CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION OF LITERATURE SURVEY

In the first phase of this research work, available literature relevant to this work was reviewed.

2.2 HISTORICAL BACKGROUND

In the beginning, all engine experiments were designed for burning a variety of gases, including natural gas, hydrogen, and propane. There had been many investigations on hydrogen enriched combustion in internal combustion engines. Rivaz (1807) of Switzerland invented an internal combustion engine with electric ignition which used the mixture of hydrogen and oxygen as fuel. He designed a car for his engine. This was the first internal combustion powered automobile (Bruno 1996, Eckermann 2001, Dutton 2006). Later, he obtained French patent for his invention in 1807. The sketch of his engine taken from his patent is shown in Figure 2.1. Cecil (1820) described a hydrogen engine in his paper entitled "On the application of hydrogen gas to produce a moving power in machinery; with a description of an engine which is moved by pressure of the atmosphere upon a vacuum caused by explosions of hydrogen gas and atmospheric air." In this document, he explained how to use the energy of hydrogen to power an engine and how the hydrogen engine could be built. This is probably one of the most primitive inventions made in hydrogen-fueled engines.

In 1863, Lenoir made a test drive from Paris to Joinville-le-Pont with his hydrogen gas fueled one cylinder internal combustion engine Hippomobile with a top speed of 9 km in 3 hours (Energylibrary 2014). Verne (1874) in his book 'The Mysterious Island' wrote the following sentence: "Water decomposed into its primitive elements by electricity, which will then have become a powerful and manageable force. Water will one day be employed as a fuel, that hydrogen and oxygen which constitute it, used singly or together, will furnish an inexhaustible source of light and heat of an intensity of which coal is not capable. Some day the coal rooms of steamers and the tenders of locomotives will, instead of coal, be stored with these two condensed gases, which will burn in the furnaces with enormous calorific power". Now his prediction becomes true gradually. In 1920 Erren converted over 1000 S.I engines into hydrogen fueled engines (Erren & Campbell 1933). His convertion included trucks and buses. For his inventions he got patent in Great Britain in 1932 (Erren, 1932) and later in the United States in 1939 (Erren, 1939).

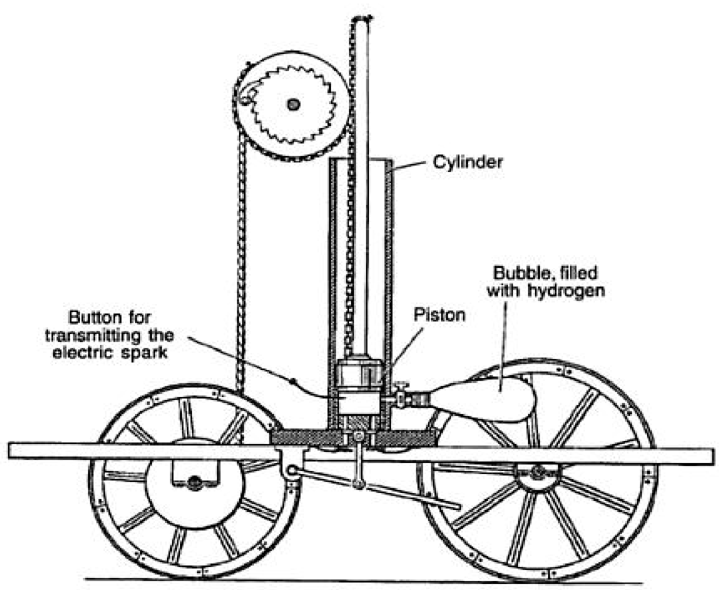

Figure 2.1 Patent drawing of Rivaz

2.3 HYDROGEN ASSISTED COMBUSTION

Hydrogen can compensate some of the demand for hydrocarbon fuel by being combusted along with gasoline, diesel, or natural gas in an internal combustion engine. This type of combustion is called dual-fuel combustion. It either uses very small amounts of hydrogen to modify combustion or uses a large amount of hydrogen as the principal source of energy in the combustion chamber. This type of operation has been investigated by numerous researchers for several types of hydrogen assisted combustions.

2.4 COMBUSTION OF HYDROGEN WITH GASOLINE

Stebar & Parks (1974) investigated about the hydrogen supplemention by means of extending lean operating limits of gasoline engines to control the NO_X emissions. They carried out their test in a single cylinder engine. Their results showed that small additions of hydrogen to the fuel resulted in very low NO_X and CO emissions for hydrogen-isooctane mixtures leaner than 0.55 equivalence ratio. They also obtained significant improvement in thermal efficiency beyond isooctane lean limit operation. However, HC emissions increased markedly at these lean conditions. They concluded that the success of hydrogen supplemented fuel approach would ultimately hinge on the development of both a means of controlling hydrocarbon emissions and a suitable hydrogen source on board the vehicle.

Houseman & Hoehn (1974) presented the first engine dynamometer test results for a modified fuel system based on hydrogen enrichment for a V-8 IC engine. The engine burnt mixtures of gasoline and hydrogen under ultra lean conditions and yielded extremely low NO_X emissions with increased engine efficiency. They produced hydrogen in a compact on-board generator from gasoline and air. They cooled hydrogen-rich product gas and mixed with the normal combustion air in a modified carburettor. The engine was then operated in the conventional manner on atomized gasoline with spark ignition, but with hydrogen-enriched air and with a high spark advance of 40°-50° BTDC. Thus the engine received two charges of fuel: a charge of gaseous fuel from the hydrogen generator, and the normal gasoline charge. The results on hydrogen enrichment were compared with the 1973 V-8 baseline stock engine with emission controls and the same engine without controls and operated at maximum efficiency under lean conditions. Relative to the stock 1973 350 CID engine, an approximate 10% reduction in brake specific fuel consumption was measured over the entire level road load speed range. For the same condition, NO_X emissions were reduced to below the equivalent 1977 EPA Standards.

Rose (1995) made researches on the method and apparatus for enhancing combustion in an ICE through electrolysis and produced hydrogen along with oxygen yielded enhanced combustion at low engine loads for all types of engines.

2.5 COMBUSTION OF HYDROGEN WITH DIESEL

Varde & Frame (1983) carried out an experimental study to investigate the possibility of reducing diesel particulates in the exhaust of the diesel engine by aspirating small quantities of gaseous hydrogen in the intake of the engine. For this study, they used a single cylinder, naturally aspirated, four stroke, DI diesel engine with compression ratio of 17.4:1. They found that hydrogen flow rate equivalent to about 10% of the total energy, substantially reduced smoke emissions at part loads. At the full rated load, reduction in smoke levels was limited. They related this to the lower amounts of excess air available in the cylinder. They found that the engine thermal efficiency was dependent on the portion of hydrogen energy, out of the total input energy supplied to the engine. In this investigation, they conducted two different types of tests. In the first set of tests, hydrogen flow rate was maintained constant while the diesel fuel flow rate was increased to increase the engine output at constant engine speed. In the second set of tests, again the engine was made to run at a constant speed of 40 rps but the hydrogen flow rate was varied. The maximum hydrogen flow rates used in this set of tests comprised about 14% of the total energy at full rated load and about 17% at 82% of full rated load. In general, the efficiency steadily increased as the portion of hydrogen energy increased at both the power levels. At the lowest hydrogen fuelling rate, the engine efficiency either decreased or remained almost constant relative to the baseline operation, i.e., when no hydrogen was supplied to engine. The premixed hydrogen fuel to air equivalence ratios at these fuelling rates were extremely low, typically 0 to 0.03, which might make the fuel to burn in a very erratic manner. They stated that the combustion of hydrogen air mixtures at such low hydrogen fuel concentrations was dependent on the local temperature around parcels of fuel mixtures. At 82% of full load, the overall temperature in the combustion chamber was lower than full rated load operation. As a result, the mixtures containing very low hydrogen content would burn better as full rated load than at part load operation. When the flow rate of hydrogen was 0.65 KJ/s, the resulting thermal efficiency was consistently lower than the baseline value. On the other hand, increasing the rate to 1.65 KJ/s resulted in higher thermal efficiency over all the load range. Peak pressure increased sharply beyond about 11% of hydrogen in the mixture at full rated load. At the same time, the time for the occurance of peak pressure got decreased from the baseline value. They also noticed that the peak cylinder pressures for mixtures containing less than 6% hydrogen energy was higher than the baseline value but it occured late. The late occurence of peak cylinder pressure at low rates of hydrogen energy supply was believed to be due to delayed burning of hydrogen in the combustion chamber. Increasing the portion of hydrogen increased the exhaust temperature due to rapid combustion and higher flame temperature. They witnessed smoke levels starting to decrease as hydrogen content was increased. They related this to increased H/C ratio of the fuel. At part loads, smoke levels got decreased by over 50%. Oxides of nitrogen increased faster than hydrocarbons as hydrogen content was increased. They attributed this to higher local temperature due to rapid combustion of hydrogen.

Roy et al (2010) investigated the engine performance and emissions of a super charged four-stroke, single cylinder, water cooled diesel engine fueled with hydrogen and ignited by a pilot amount of diesel fuel in dual-fuel mode. The engine was tested for use as a cogeneration engine. The experiments were carried out at a constant pilot injection pressure of 80 MPa and pilot quantity of 3 mg/cycle for different fuel-air equivalence ratios and at various injection timings without and with charge dilution. The intake pressure of air was kept constant at 200 kPa and the temperature was maitained at 30^{o}C throughout the study. Their experimental strategy was to optimize the injection timing to maximize the engine power at different fuel-air equivalence ratios without knocking and within the limit of the maximum cylinder pressure. They first tested the engine with hydrogen-operation condition up to the maximum possible fuel-air equivalence ratio of 0.3. A maximum IMEP of 908 kPa and a thermal efficiency of about 42% were obtained. They observed that the equivalence ratio could not be further increased due to knocking of the engine. The emission of CO was only about 5 ppm, and that of HC was about 15 ppm. However, the NO_X emissions were high, 100 to 200 ppm. Then they performed charge dilution by N_2 to obtain lower NO_X emissions. They achieved 100% reduction in NO_X. According to them, this was due to the dilution by N_2 gas which paved the way for injection of higher amount of hydrogen without knocking. Because of this charge dilution, they got 13% higher IMEP than IMEP produced without charge dilution. At an equivalence ratio of 0.20, the maximum cylinder pressure increased gradually with advancing injection timings. The maximum cylinder pressure was 9.27 MPa at an injection timing of 10^{o} BTDC, and reached its highest level of about 12 MPa at 18^{o} BTDC.

The maximum cylinder pressure at an injection timing of 5^o BTDC and an equivalence ratio of 0.25 was 8.6 MPa, and reached its highest level of 12.6 MPa at 13^o BTDC. The maximum cylinder pressure at an injection timing of 4^o BTDC and an equivalence ratio of 0.30 was 8.75 MPa, and reached its highest level of about 10 MPa at 6.5^o BTDC. The maximum cylinder pressure was very low at an equivalence ratio of 0.30 because in that case the injection timing needed to be retarded to avoid knocking. At a constant equivalence ratio, the NO_X emission increased with advanced injection timings. Advancing the injection timing increased the peak cylinder pressure, and higher peak cylinder pressures resulted in higher peak burned gas temperatures, and hence more NO_X emission. More NO_X was produced as the equivalence ratio got increased, although the injection timings were retarded. The highest NO_X emission level was about 200 ppm at an equivalence ratio of 0.25 to 0.30. HC emitted by the dual-fuel engine fueled by hydrogen varied from only 14 to 18 ppm. The CO emitted by the dual-fuel engine fueled by hydrogen varied from only 5 to 7 ppm. The level of NO_X of 200 ppm with hydrogen-operation was reduced to 0 ppm level with 60% N_2 dilution. There was about 98% and 99% reduction in NO_X for 40% and 50% N_2 dilution, respectively. However, HC increased to the level of about 80 ppm with 60% N_2 dilution. CO increased to the levels of about 80 ppm and 500 ppm with 50% and 60% N_2 dilution, respectively. They concluded that by diluting the charge with N_2, the hydrogen engine could be operated without engine knock.

Lilik et al (2010) reported about the hydrogen assisted diesel combustion on a DDC/VM Motor 2.5L, 4-cylinder, turbo-charged, common rail, direct injection light-duty diesel engine. Their main focus was on the study of exhaust emissions of the engine. They substituted hydrogen for diesel fuel on an energy basis of 0%, 2.5%, 5%, 7.5%, 10% and 15% by aspirating hydrogen into the engine's intake air. For their test, they have selected four speeds and load conditions of the engine. They observed a significant retardation in injection timing of the engine's electronic control unit (ECU) when hydrogen was aspirated. This resulted in significant reduction in NO_X emission. They also observed that the same emission reductions were possible without aspirating hydrogen by manually retarding the injection timing. To study the hydrogen assisted diesel combustion, they locked the injection timings of the pilot and the main fuel. They also used computational fluid dynamics analysis (CFD) for hydrogen assisted diesel combustion. CFD of the hydrogen assisted diesel combustion process captured the trend and reproduced the experimentally observed trends of hydrogen's effect on engine emissions. They found that the hydrogen addition caused the maximum in-cylinder pressure to increase in all modes. The effect was greater in the high load modes, where more complete combustion of the fuel occurred. At 1800 rpm and 75% maximum output with 15% hydrogen substitution, the maximum pressure got increased by 2% over base line condition and at 3600 rpm and 75% maximum output, the maximum pressure increased by 7%. Also, they observed that the maximum pressure peak occurred earlier in the high load modes. The substitution of hydrogen for diesel fuel decreased the amount of diesel fuel injected in both the pilot and main injections. When hydrogen assisted the diesel combustion, there was a slight ignition delay in the premixed combustion phase. They further stated that this was due to the fact that the diesel fuel acted as a pilot to ignite the hydrogen, since hydrogen has a lower cetane number than diesel fuel. They further stated that increasing levels of hydrogen slightly increased the apparent heat release rate of the premixed combustion phase. With the increase in hydrogen, less diesel fuel was injected. Thus, less heat was absorbed during the fuel vaporization phase between the premixed combustion phase and the mixing-controlled combustion phase of the main injection. The heat release during the mixing-controlled combustion phase was decreased with the increase of hydrogen substitution.

Welch & Wallace (1990) converted a single-cylinder Lister ST-1 direct injection diesel engine to operate on hydrogen to evaluate its performance and combustion characteristics. They admitted hydrogen gas at 10.3 MPa pressure to the engine combustion chamber through a hydraulically-actuated injection valve which controlled the timing and duration of the hydrogen injection. They provided ignition of hydrogen by a continuously operating sheathed glow plug that was used in passenger car diesel engines to assist cold starting. Their results indicated that the hydrogen-fueled diesel engine could produce higher power than an ordinary diesel engine due to the absence of smoke emissions. Another positive feature was NO_X emissions got reduced compared to the ordinary diesel engine. Indicated efficiency of the hydrogen-fueled diesel engine was about 90% of that of the original diesel at moderate loads. At very light loads, however, the efficiency of the hydrogen-fueled engine got decreased compared to that at moderate loads. They concluded that the hydrogen-fueled diesel engine with glow plug could be used to develop greater power with lower emissions than the same engine operated on diesel fuel.

Shahad & Hadi (2011) found a way to reduce the concentration of pollutants coming out from a diesel engine. They blended hydrogen gas with hydrocarbon fuels used in internal combustion engines. They carried out their experimental research in a four stroke air cooled diesel engine. Their hydrogen fueling system consisted of a hydrogen bottle, two pressure reduction valves to reduce the hydrogen pressure to 2 bars, a hydrogen flow meter and an injector. The injector was mounted on the inlet pipe at 10 cm from the engine with an angle of 45^o with the direction of injection. The hydrogen injection timing was controlled by an electronic control unit designed for this purpose. They chose three different speeds of 1000 rpm, 1250 rpm, and 1500 rpm. They also varied the load from no load to 80% of full load and the hydrogen blending ratio was varied from zero (pure diesel) to 10% (by mass) of the injected diesel fuel. Their results showed that 10% hydrogen blending reduced smoke opacity by about 65%. It increased the nitrogen oxides concentration by about 21.8% and reduced CO_2 and CO concentrations by about 27% and 32% respectively. This trend was found at all tested speeds and loads. They observed that the concentration of NO_X generally increased with hydrogen blending ratio for all loads. They related this to the improvement of combustion process caused by the presence of hydrogen in the fuel mixture which led to higher cylinder temperature. They also stated that the NO_X formation reactions were highly temperature dependent.

2.6 COMBUSTION OF HYDROGEN WITH CNG

Bysveen (2007) reported about the working characteristics of S.I engine when CNG and HCNG were used as a fuel. The engine used for his experiments was a three-cylinder, single spark plug, 2.7 litre Zetor Z4901 originally used for stationary applications. He rebuilt the engine for natural gas use by reducing the compression ratio from 17:1 to 11:1. He equipped the test engine with K-type thermo couples in the intake manifold, in the cooling water system and in the exhaust. He employed hydraulic dynamometer for loading the engine. He studied the sensitivity in spark timings for the fuels and the engine in the range of 51^o to 251^o BTDC. The CNG fuel used for this work consisted of about 99.5% vol. of CH_4, and the HCNG consisted of a mixture of 29% vol. of hydrogen. His results showed that the brake thermal efficiency was considerably higher using HCNG than using pure CNG. This effect was most pronounced for the high engine speeds. In general, he observed less production of unburned hydrocarbons when adding hydrogen to the CNG for a given excess air ratio. He reported that this was due to the fact that the lean limit for pure methane–air mixtures was much richer than the lean limit for hydrogen-enriched methane–air mixtures. With H_2 addition, a smaller quenching zone resulted; this enabled the flame to propagate closer to the walls. He further observed that the addition of hydrogen to methane–air mixtures increased the combustion speed and the combustion temperatures; it led to increased NO_X emissions compared to pure natural gas.

Mohammed et al (2011) investigated on the performance and emission of a CNG-DI and spark-ignition engine when a small amount of hydrogen was added to the CNG using in-situ mixing. They set the injection timing to 30^o BTDC, kept the air–fuel ratio at stoichiometric, and the ignition timing to maximum brake torque. They performed experiments at 2000, 3000, and 4000 rpm of engine speeds with WOT conditions. From their results, it was interpreted that the introduction of a small amount of hydrogen improved the engine performance, brake specific energy consumption, and cylinder pressures. The CO emission of the engine got decreased until the engine speed reached 3000 rpm and then started to increase with the increase in engine speed. They stated that this was mainly due to increase in completeness of combustion process and sufficiency of oxygen. At high speeds, the CO emissions tended to increase due to retardation in timing which also resulted in poor combustion. For all rates of hydrogen THC tended to decrease. They attributed this to decrease in the carbon fraction in the fuel blends and the increase in combustion temperature due to increase in H_2 fractions.

Cowan et al (2010) reported about the effects of gaseous fuel additives on a pilot-ignited, directly injected natural gas engine. The additives used in their investigation were propane, ethane, hydrogen and nitrogen. They used a single cylinder test engine equipped with a prototype fuelling system for their study. They controlled the diesel and natural gas injection processes by electronic control operated multi-fuel injector. They equipped the engine with a custom air-exchange system to ensure that the charge conditions were independent of variations in fuel composition and injection timing. They prepared the nitrogen, ethane, and propane fuel blends using bottled gas combined with commercially distributed natural gas in large volume storage tanks. They left the blends in the storage tanks for at least 48 hours to ensure that they were fully mixed before being supplied to the high-pressure gas compression system for supply to the engine. To avoid condensation of the heavy hydrocarbons, the kept all concentrations below the saturation partial pressure at all times. They selected mid-load condition for their investigation to compare the effects of the various fuels. For their study, they controlled the combustion timing by varying the timing of the start of the pilot fuel injection process. The timing of the gas start-of-injection (GSOI) was fixed at 1.0 ms after the end of the diesel injection. The 50% IHR was used as the control variable representing the combustion timing. They adjusted the start-of-injection timing for the different fuel blends to maintain the 50% IHR at the specified value. For all timings, the engine's power output was held constant by varying the mass flow rate of the gaseous fuel. They fixed pilot quantity at 5% of the total fuel on an energy basis; this amounted to approximately 6 mg diesel/cycle for all the conditions tested. The pilot diesel and gaseous fuel rail pressures were constant at 21 MPa for all the tests. Their results showed that the hydrogen addition to the fuel resulted in an increase in ignitability for the gaseous fuel, and a corresponding reduction in ignition delay. The effects of ethane and propane were similar to those of hydrogen. They observed higher NO_X emissions when ethane, propane, or hydrogen was added to the combustion process. They related this to increase in adiabatic flame temperatures as they were generated pre-dominantly through the strongly temperature-dependent thermal NO mechanism. All the fuel additives reduced hydrocarbon emissions. When compared with other additives, the hydrogen reduced more HC emissions. They related this to higher radical concentrations and a wider flammability range which resulted in more complete combustion of the fuel.

2.7 COMBUSTION OF HYDROGEN WITH LPG

Lata et al (2012) made an experimental investigation on performance and emission of a dual fuel operation of a 4 cylinder, turbocharged, inter-cooled, 62.5 kW genset diesel engine with hydrogen, liquefied petroleum gas (LPG) and mixture of LPG and hydrogen as secondary fuels. They carried out the experiments at a wide range of load conditions of the engine with different gaseous fuel substitutions. When only hydrogen was used as secondary fuel, the maximum enhancement in the brake thermal efficiency was 17% which was obtained with 30% of secondary fuel. When only LPG was used as secondary fuel, maximum enhancement in the brake thermal efficiency was 6% with 40% of secondary fuel. They observed that compared to the pure diesel operation, proportion of unburnt HC and CO got increased while emission of NO_X and smoke got reduced in both cases. On the other hand, when 40% of the mixture of LPG and hydrgen was used in the ratio of 70:30 as secondary fuel, brake thermal efficiency got enhanced by 27% and HC emission got reduced by 68%. Further, they observed that the dual fuel diesel engine showed lower thermal efficiency at lower load conditions as compared to diesel. They attributed this to the fact that at low concentration of hydrogen or LPG in the intake air, the combustion spread throughout the gas-air mixture. This caused high heat transfer losses to the adjacent walls. While, in the case of diesel engines under light load condition, the penetration of the diesel spray was such that it did not reach the cylinder walls and the combustion was confined to piston bowl and also, the surrounding coatings of air acted as insulation in between burnt gases and the walls, which reduced heat losses thereby giving better thermal efficiencies with diesel. They found that this short coming of low efficiency at lower load condition in a dual fuel operation could be removed when a mixture of hydrogen and LPG was used as the secondary fuel at higher than 10% load condition.

Rao et al (2008) performed experiments on a conventional diesel engine operating on dual-fuel mode using diesel and LPG. The experiments were done at a constant speed of 1500 rpm and under varying load conditions. They indicated that with the dual-fuel mode of operation, precious diesel could be conserved up to 80%. However, in their work, it was done only up to 45% due to severe engine vibrations. The brake power of the engine was found to be about 15% more on the dual-fuel operation, while the brake specific fuel consumption was found to be about 30% lower than diesel fuel mode of operation. They related this to better mixing of air and LPG and improved combustion efficiency.

Qi et al (2007) conducted an experimental investigation on a single cylinder direct injection diesel engine modified to operate in dual fuel mode with diesel-LPG as fuels. They used various rates of LPG–diesel blends for their experiments. They compressed LPG of 0, 10, 20, 30, and 40% by pressured nitrogen gas to mix with the diesel fuel in a liquid form. They concluded that LPG-diesel blended fuel combustion was a promising technique for controlling both NO_X and smoke emissions even on existing DI diesel engines.

2.8 COMBUSTION OF HYDROGEN WITH METHANE

Wallner et al (2007) analyzed the combustion properties of hydrogen/methane blends (5% and 20% methane by volume in hydrogen equal to 30% and 65% methane by mass in hydrogen) and compared them to those of pure hydrogen as a reference. They confirmed that only minor adjustments in spark timing and injection duration were necessary for an engine to operate on pure hydrogen and hydrogen/methane blends. They used automotive size, spark-ignited, single-cylinder, supercharged 6.0-L V-8 research engine having a compression ratio of 11.4:1 and maximum torque of 30 Nm for their investigations. They ran the engine at two different speeds of 2000 rpm and 4000 rpm. They selected three load conditions for their engine analysis as IMEP of 2 bar, 4 bar, and 6 bar. They performed a detailed analysis of the combustion behavior in order to evaluate the influence of blending of different concentrations of methane and hydrogen. They chose the spark timing as constant at 10 deg CA before top dead center (BTDC). They observed that in pure hydrogen operation, combustion took only about 25 deg CA whereas in 5% methane blend, it was 35 deg CA and in 20% methane blend, it was significantly longer to about 55 deg CA. They found extremely short combustion duration for close-to-stoichiometric pure hydrogen operation that resulted in high combustion temperatures and, thus, it increased the wall heat losses. They also noticed that the maximum rate of heat release was significantly higher for pure hydrogen operation, which also resulted in a higher combustion peak pressure of 45 bar for pure hydrogen compared to 30 bar for the 20% methane blend. These results showed that to achieve the maximum efficiency, the spark timing had to be advanced in blended operation compared to pure hydrogen operation. The NO_X emission was more at IMEP of 6 bar compared to IMEP of 2 bar and IMEP of 4 bar. They stated that at a higher engine load like 6 bar IMEP, due to higher combustion temperatures and the NO_X emission depended upon the logarithmic scale of temperatures, it got increased exponentially.

Zhou et al (2013) conducted an experimental investigation on combustion and emission characteristics of a compression ignition engine using diesel as pilot fuel and methane, hydrogen and methane/hydrogen mixture as gaseous fuels at 1800 rpm. The test engine was mounted on an eddy-current dynamometer. They measured the in-cylinder pressure by a piezo electric sensor of Kistler make and the pressure signals were amplified with a charge amplifier. A crank-angle encoder was employed for crank-angle signal acquisition at a revolution of 0.5° CA. The intake and exhaust gas temperatures were measured by K-type thermocouples. For gaseous emissions, total HC was measured with a heated flame ionization detector. NO/NO_X was measured with a heated chemiluminescent analyzer. CO and CO_2 were measured with non-dispersive infrared analyzers. O_2 was measured with a portable gas analyzer. During the investigation they observed that the ULSD-hydrogen combustion became unstable and hard to control at high loads. When hydrogen was enriched in methane, the BTE got increased at all loads. With the addition of hydrogen into methane, the peak cylinder pressure got increased relative to ULSD-Methane operation and this effect was more apparent at 90% load. At BMEP of 0.71 MPa, for ULSD-Methane dual-fuel engine, the peak heat release rate increased apparently compared with the baseline operation. The heat release rate profile for ULSD-Hydrogen revealed that the main combustion phase occurred at premixed combustion phase and the heat released during diffusion combustion phase was reduced a lot relative to other cases. They found that the CO emission increased sharply when the combustion of metane and ULSD had taken place. This was due to the incomplete combustion of methane. When ULSD-Hydrogen was combusted in dual-fuel mode, the CO emission decreased at all load sowing to the direct replacement of the carbon content from hydrogen to diesel fuel. The addition of hydrogen into the methane extended the flammability limit of methane and the incomplete combustion of methane was alleviated. When the engine was operated at 90% of the full load with hydrogen induction, CO emission got reduced by nearly 25% compared to base line operation.

It was further observed that the total HC emission was high when ULSD-methane was combusted. On the otherhand, when ULSD-Hydrogen was combusted, the total HC emission got decreased. For the BMEP 0f 0.08 MPa, 0.24 MPa and 0.41 MPa, the total HC emission was 12.01, 10.26, 9.03, 7.69, and 0.78 times than baseline for Methane, H30-M70, H50, M50, H70-M30 and Hydrogen, respectively. For ULSD-Methane and ULSD-Hydrogen dual-fuel combustion, NO_X emission got decreased slightly at lower load and increased at medium to high loads. This was due to the higher combustion temperature and faster burning rate of hydrogen than methane, ULSD-Hydrogen combustion enhanced the NO_X formation. When small quantity of hydrogen was mixed with methane, it reduced NO_X emission. But, when the quantity of hydrogen was increased, the NO_X emission got increased. At H50-M50 case, the NO_X was basically the same with ULSD-Methane operation.

2.9 COMBUSTION OF HYDROGEN WITH MISCELLANEOUS GASES

Park et al (2011) experimentally investigated the effect of addition of hydrogen on the performance and emission characteristics of a naturally aspirated S.I engine which was fueled with biogas. They ran the engine at constant engine rotational speed of 1800 rpm under a 60 kW power output condition. They blended H_2 fractions ranging from 5 to 30% to the biogas. Their engine test results indicated that the addition of hydrogen improved in-cylinder combustion characteristics, extending lean operating limit as well as reducing THC emissions while elevating NO_X generation. In terms of efficiency, however, they observed a competition between enhanced combustion stability and increased cooling energy loss with a rise in H_2 concentration. They got maximum engine efficiency at 5% to 10% of H_2 concentration. They reported that an increase of H_2 improved flame propagation speed and extended lean flammability limit while NO_X increased. As H_2% was increased, the burn duration got decreased due to the improvement in the propagation speed of the blended fuel combustion. In addition, they observed no knocking or back-fire phenomena during engine operations for all the fuel conditions. This meant that stable and efficient combustion could be achieved even in the lowest quality gas by H_2 addition while abnormal combustion was still suppressed.

Sahoo et al (2012) carried out the experiments in a Kirloskar TV1 diesel engine to evaluate its characteristics when syngas mixture of hydrogen and carbon monoxide was inducted into the combustion of diesel. The engine used for their study was a single cylinder, water cooled, direct injection, four stroke, having a bore of 87.5 mm, stroke of 110 mm, compression ratio of 17.5:1, rated power of 5.2 kW at 1500 rpm. They analyzed the flue gas compositions using a multi-component analyzer based on infrared and chemical cell technique. Their results showed that the 100% H_2 syngas mode resulted in a maximum in-cylinder pressure and combustion temperature which in-turn increased the NO_X emissions and the exhaust gas temperature compared to that of 75% and 50% H_2 syngas modes. They observed the NO_X emissions of 127 ppm, 175 ppm, and 220 ppm at peak power output for 50%, 75%, and 100% H_2 syngas modes respectively. They related this to the higher flame speed and higher energy content of the syngas at 100% H_2 syngas mode.

Mohammadi et al (2005) carried out an investigation on diesel engine used for power generation to see the effects of addition of LCG (Low Calorific Gases) and LCG with small portion of hydrogen and nitrogen on performance and emissions characteristics of the engine. These gases were originally produced in various chemical processes such as gasification of solid wastes or biomass. The test engine used by them was a four-stroke single cylinder naturally aspirated direct-injection diesel engine (Yanmar NFD-170) with a bore of 102 mm and a stroke of 105 mm, injection nozzle spray angle of 150° with four holes and with 0.29 mm hole diameter. They carried out the combustion analysis by measuring in-cylinder pressure at every 1° CA using piezoelectric pressure transducer (Kistler 6052A). They used diesel having a density of 828 kg/m^3, lower heating value of 44200 kJ/kg, and cetane number of 55 for this tests. They conducted all experiments at thermally steady state of the engine with injection timing of 12° BTDC and engine speed of 1800 rpm. They introduced nitrogen from a high pressure vessel into the intake of the engine using a gas mixer installed at downstream of surge tank. And, they introduced hydrogen gas using an orifice nozzle with diameter of 6 mm. They measured the flow rate of both gases preciously using thermal mass flow meters. In their experiment, they first adjusted the flow rate and composition of LCG and then the amount of diesel fuel injected to achieve considered output. They fixed the engine load as constant at brake mean effective pressure of 0.6 MPa. Their results showed that at rH = 0 and rLCG=25% when 25% of intake air was replaced with nitrogen, the efficiency of the engine was slightly lower than diesel fuel operation. However, when they introduced hydrogen with LCG, it lowered the consumption of diesel fuel. At rLCG=25% and rH=30%, the corresponding saving in consumption of diesel fuel was about 40%. At rH=0 when only nitrogen was added to the engine intake, it increased ignition delay with little effects on combustion process. However, at given rLCG, increasing the hydrogen concentration, promoted the premixed and diffusion combustions and it resulted in higher peak combustion pressure and temperature. Increasing rH increased the peak level and advancement in its timing.

Fang et al (2008) investigated the driving performance and emission characteristics of a 125 cc motor cycle equipped with an onboard plasma reformer for producing Hydrogen Rich Gas (HRG). To produce HRG, they inducted butane with suitable air flow rate into the plasma reformer. They ran the motorcycle under steady and transient conditions on a chassis dynamometer to assess the driving performance and exhaust emissions. Prior to run, they optimized the operation parameters of the plasma reformer in a series of tests and they concluded that the O_2/C ratio of 0.55 and a butane supply rate of 1.16 lpm was the optimum condition to produce HRG. They used gas chromatograph of Agilent 6850 GC for analyzing the gas emission and a scanning electron microscope for observing carbon deposit arising from the reforming process. For analyzing the driving tests, they used Japanese made Horiba 554JA emission analyzer; US made CAI 600 NO_X analyzer, a fuel flow meter, an oscilloscope and a temperature data recorder. From their results, it was interpreted that at O_2/C ratio of 0.55, the NO_X emission at a vehicle speed of 40 km/h got reduced from 600 ppm to 220 ppm. They attributed this to the diluting effect of HRG, as it contained CO_2 and N_2 also. They observed that when 2.95% HRG was added, the highest peak pressure was obtained. Further, in the addition of 4.11%, the pressure rise rate became slower and the peak pressure also became lower than other conditions. They concluded that the acceleration characteristics of the vehicle were similar under both fuelling systems.

Cecrle et al (2012) injected a hydrogen/carbon monoxide mixture into the inlet manifold of a biodiesel fuled dual-fuel diesel engine to evaluate its characteristics. The engine used for their testing was a Yanmar L100V single-cylinder DI diesel engine with a compression ratio of 21.2. The other operating parameters of the engine were speed of the engine mainted at 3600 rotations per minute, injection time as 15.5^0 before piston top-dead-center with a pressure of 19.6 MPa. To provide load on the engine, they employed a North-Star electric generator coupled to the crankshaft. They outfitted various sensors to measure the ambient air temperature, pressure, and relative humidity, engine air mass flow, engine intake air emperature and pressure, fuel mass flow, fuel density, engine torque, exhaust port temperature, downstream exhaust gas temperature and pressure, and generator load on the engine and test stand. For their experiments they used Reformate Assisted Gas consisting of 57% H_2 and 43% CO. They had chosen this mixture because it represented the best-case scenario for a non-catalyzed system in regard to assisted mixture energy as it was the partial oxidation of glycerin without any formation of complete products of combustion. When they added reformatted gas to the intake of the engine, the biodiesel fuel flow rate got dropped significantly. This illustrated that the reformate mixture increased the fuel economy of biodiesel under all loading conditions. CO_2 emissions also got increased for the 50% load point. The addition of reformate also reduced total HC emissions. They reasoned this to a hotter burn that would also have acted to diminish the incomplete combustion.

Plaksin et al (2008) conducted a study on reduction of NO_X in diesel engine emissions by using a hydrogen-rich synthesis gas produced by plasmatron fuel reformer. They activated 10% to 20% of the diesel fuel in an arc discharge and turned them into plasma chemical reformation fuel by using a DC arc plasmatron that was fabricated to increase the ability of gas activation. They got the yielding of diesel fuel reformation upto 80% to 100% when small quantity of diesel fuel in range of 6 ml/min was used. They supplied this synthesis gas mixture which contained hydrogen, carbon dioxide, carbon monoxide, nitrogen, and hydrocarbons into the engine together with the rest of the fuel-air mixture. They reported decrease in the NO_X content in the emissions of the engine upto 23% and simultaneously the fuel combustion efficiency got increased.

2.10 INFLUENCE OF ADDITION OF HYDROGEN AND OXYGEN MIXTURE IN COMBUSTION

Wang et al (2011) compared the effects of hydrogen and hydrogen-oxygen blends (hydroxygen) additions on the performance of a gasoline engine at 1400 rpm and with a manifold absolute pressure of 61.5 kPa. The tests were carried out on a 1.6 L, SI engine manufactured by Beijing Hyundai Motors. The rated power of the engine was 82.32 kW at 6000 rpm and a rated torque of 143.28 Nm at 4500 rpm. They applied a hybrid electronic control unit to adjust the hydrogen and hydroxygen volume fractions in the intake increasing from 0% to about 3% and keep the hydrogen-to-oxygen mole ratio at 2:1 in hydroxygen tests. For each testing condition, the gasoline flow rate was adjusted to maintain the mixture global excess air ratio of 1. First, they ran the engine with pure gasoline then with hydrogen and hydroxygen with varying volume fractions in the intake as 0% to 3% to simulate the case of hydrogen and oxygen produced by a water electrolysis process. Their test results confirmed that engine fuel energy flow rate was decreased after hydrogen addition but increased with hydroxygen blending. They found that when hydrogen or hydroxygen volume fraction in the intake was lower than 2%, the hydroxygen-blended gasoline engine produced a higher thermal efficiency than the hydrogen-blended gasoline engine. They stated that this increase in brake thermal efficiency was due to the addition of hydrogen which helped to enhance the fast and complete combustion of the fuel-air mixture. They achieved the peak value of 35.7% at the standard hydroxygen volume fraction in the intake of 0.75%. They explained that the possible reason was that the addition of hydroxygen increased the oxygen fraction in the intake; this slightly reduced the fuel-rich area in the cylinder and this in turn increased the complete combustion of the fuel-air mixtures. They observed that both engines indicated thermal efficiency and fuel energy flow rates were raised after the standard hydroxygen blending. They attributed this to the ignition energy of hydrogen which was only 1/10 of that of gasoline and the addition of hydrogen stimulated the formation of O and OH radicals. Since hydrogen has a short quenching distance, they witnessed a decrease in HC emissions caused by the crevice effect. They also related this to chemical equilibrium process. As the raised cylinder temperature after hydrogen or hydroxygen addition helped to ease the formation of HC emissions during the combustion process. They observed that the CO emission got increased with the increase of hydrogen volume fraction in the intake whereas it got decreased with the increase of the standard hydroxygen addition fraction. When they raised hydroxygen volume fraction in the intake as 0% to 2.8%, CO got reduced by 21.86% for the standard hydroxygen-blended gasoline engine. NO_X emissions were raised after hydrogen and hydroxygen additions. They attributed this to a high adiabatic flame temperature caused by the additions of hydrogen and hydroxygen.

Karagoz et al (2012) studied the effect of hydrogen-oxygen mixture on S.I engine performance and emission characteristics. They introduced the gas mixture into the inlet manifold of the engine. They selected three different supplementary fuels which contained 0% H_2, 3% H_2 + 1.5% O_2, and 6% H_2 + 3% O_2 by volume fractions of intake air. They used a mass-flow meter with a measurement uncertainty of 1%. They reduced the flow fluctuations of H_2/O_2 mixture by using a buffer tank. Their test results showed that a 6% H_2 + 3% O_2 addition increased engine brake power from 19.09 kW to 20.52 kW at 3500 rpm engine speed. The brake torque got increased from 51.28 Nm to 63.24 Nm at 2000 rpm engine speed. An increase of BMEP from 575.3 kPa to 710.4 kPa at 2000 rpm was achieved with 6% H_2 and 3% O_2 addition. Best thermal efficiencies were achieved partly by 3% and 6% hydrogen addition. An increase in brake thermal efficiency from 21.77% to 24.50% at 2000 rpm engine speed was achieved using 6% gasoline-hydrogen mixture. A BSFC got decreased from 372.4 g/kWh to 330.9 g/kWh at 2000 rpm engine speed. HC emission got reduced from 274 ppm to 84 ppm at 2000 rpm engine speed when 6% H_2 and 3% O_2 mixture was used as a supplementary fuel. NO_X emission got increased from 848 ppm to 1297 ppm at 2000 rpm due to higher in-cylinder temperature levels. They observed lower CO emissions and higher CO_2 emissions as a consequence of improved combustion.

2.11 COMBUSTION UNDER THE INFLUENCE OF HYDROGEN AND OXYGEN MIXTURE PRODUCED ESPECIALLY BY ELECTRO-CHEMICAL DISSOCIATION OF WATER

Shrestha et al (2000) conducted experiments on a Chevrolet Silverado 6.5 L turbocharged V8 diesel engine. They used three units of hydrogen generation system (HGS) each having a capacity to produce hydrogen-oxygen mixture of 690 cm^3/min by the process of water electrolysis. They tested the vehicle in three test driving cycles i.e., U.S. Federal Testing Protocol (FTP), Japanese 11 Mode Test Schedule (JAPANESE 11), and Economic Commission for Europe Schedule (ECE 1504A). They equipped the vehicle with an on-board diagnostics system, which continuously monitored the engine parameters during the test. In this test they used MD-GAS-5C gas analyser to measure CO, HC and NO_X. This analyser was interfaced with NID-7000 software for exhaust gas analysis. The real time exhaust information was collected in conjuction with vehicle load, power and torque outputs, with a sampling frequency of 10 Hz. Their result showed that the addition of hydrogen to the main fuel could be beneficial for the combustion process in internal combustion engines. Similarly, oxygen enrichment in the intake was shown to provide substantial control in particulate emissions, improved thermal efficiency, and reduced engine-out emissions in diesel engines. According to their results, Particulate matter (PM) got reduced up to 60%, CO up to 30% and NO_X up to 19% when compared with diesel combustion.

Shrestha & Karim (1999) reported that the addition of small quantity of hydrogen and oxygen produced by the electrical dissociation of water to the petrochemical fuel might contribute towards the speeding of the combustion process of internal combustion engine and bring about significant improvements in performance and emissions. For this investigation they tested an SI engine operated with methane over a range of operating conditions. One of the main features of methane fueled spark ignition engines is their relatively slow flame propagation rates in comparison to liquid fuel applications which may lead to relatively lower power output and efficiency with increased emissions and cyclic variations. This is especially pronounced at operational equivalence ratios that are much leaner than the stoichiometric value. They suggested that the addition to the methane with the products of water electrolysis generated on-board of a vehicle might produce some improvement in engine performance and also suggested that the above procedure could be effectively implemented for relatively lean mixtures and low compression ratios.

Uykur et al (2001) studied the effects of the addition of small amounts of water electrolysis products on laminar premixed methane/air flames using chemical kinetic simulation methods. They used CHEMKIN kinetic simulation package with the GRI kinetic mechanism. Pollutant concentrations, flame speeds, temperature profiles and lean flammability limits of methane/air, methane/hydrogen/air, and methane/hydrogen/oxygen/air systems were compared at different addition percentages and equivalence ratios from 1.4 to the lean flammability limit. The addition of 10% to 20% hydrogen in the fuel was found to have a small effect in improving flame speed and lean flammability limit properties. However, the addition of oxygen and hydrogen in the same ratio as found in water was shown to be beneficial. Improvements in the flame speeds of methane/air mixtures by the addition of 10% hydrogen and its associated oxygen were equivalent to the improvements obtained by the addition of 20% of hydrogen. They claimed that in near stoichiometric mixtures, the addition of oxygen substantially increased the NO_X concentrations, but for lean mixtures no increase in NO_X was predicted. CO emissions got reduced when hydrogen displaced carbon containing fuels.

Sobiesiak et al (2002) explored the impact of the addition of small amounts of molecular and atomic hydrogen/oxygen on laminar burning velocity, pollutant concentrations and adiabatic flame temperatures of premixed, laminar, freely propagating iso-octane flames using CHEMKIN kinetic simulation package and a chemical kinetic mechanism at different equivalence ratios. They concluded that hydrogen/oxygen additives increased the laminar burning velocities. Also, carbon monoxide emissions got reduced due to increase in OH concentrations in every stoichiometric ratio examined. In addition, the mixture of hydrogen and oxygen increased the adiabatic flame temperature of iso-octane/air combustion which resulted in increase in NO_X emission.

Yilmaz et al (2010) investigated the effect of hydroxy gas addition on compression ignition engine exhaust emissions and engine performance characteristics. They used a four cylinder, four stroke, compression ignition (CI) engine for their study. They fed the hydroxy gas to the intake manifold of a direct-injection CI engine by a hydroxy system and a hydroxy electronic control unit (HECU) under various loads. They produced hydroxy gas (HHO) by the electrolysis process of different electrolytes of KOH(aq), NaOH(aq), NaCl(aq) with various electrode designs in a leak proof plexi glass reactor (hydrogen generator). The experiment results showed that constant HHO flow rate at low engine speeds turned advantages of HHO system into disadvantages for engine torque, carbon monoxide (CO), hydrocarbon (HC) emissions and specific fuel consumption (SFC). Investigations demonstrated that HHO flow rate had to be diminished in relation to engine speed. In order to overcome this disadvantage they designed and manufactured a hydroxy electronic control unit (HECU). The result of their investigation showed that HHO gas addition to the engine without any modification resulted in increasing engine torque output by an average of 19.1%, reducing CO emissions by an average of 13.5%, HC emissions by an average of 5% and SFC by an average of 14%. The reason for this happening was, the increase in power was due to oxygen concentration of hydroxy gas and better mixing of hydroxy with air and fuel that yielded enhanced combustion. High laminar flame velocity of hydroxy yielded decreased ignition delay and shorter combustion period that led to ideal constant-volume combustion.

Al-Rousan (2010) conducted experiments about the effect of HHO gas supplementation on performance characteristics of a 197cc (Honda G 200) single-cylinder S.I engine with a compression ratio of 6:5:1. The result showed that the optimal surface area of an electrode was needed to generate sufficient amount of HHO. It was nearly twenty times that of the piston surface area. Also, the volume of water needed in the cell was about one and half times that of the engine capacity. The engine was subjected to a test of constant load with variable speed (from 1000 to 2500 rpm). The auxiliary equipments used for this test were tachometer for engine speed, voltmeter for cell voltage, thermometers for ambient temperature, and thermocouples for exhaust gas temperature, clamp meter for current measurements and flow meter for fuel consumption. He supplied a power of 30 amps with 0-20V of DC to the fuel cell. He concluded that by supplementing HHO gas into the combustion process of S.I engine, the brake thermal efficiency got increased on an average of 3% to 8% and fuel consumption got reduced by 20% to 30%.

Milen & Kiril (2004) carried out experiments to evaluate the influence of the addition of hydrogen-oxygen mixture obtained from electro chemically decomposed water to the inlet air of a single cylinder direct injection diesel engine having cylinder bore of 98 mm and piston stroke of 130 mm. They loaded the engine with DC dynamometer of Mezvetin MS 2218–4 make. The fuel consumption was measured by mass method. The air consumption was measured by laminar flowmeter of Cussons M79RH make. The smoke was measured by Hartridge MK3 smokemeter. The NO emissions were measured by gas analyzer Radas 1. The indicated pressure data were collected by piezo electric pressure transducer of Kistler 6509 make and crank shaft position encoder Heidenhain rod 428D 163 make. The engine speed was measured by frequency meter FM1100. This test run was conducted at constant load with variable speed of 1300 rpm to 1800 rpm. They maintained hydrogen-oxygen flowrate at 240 l/h and the injection timing at 18^{o} BTDC. The results showed that the engine power got increased with hydrogen-oxygen mixture addition. The average power improvement obtained was 15%. The peak pressure improvement at 1500 rpm was 14.8%. They concluded that when a diesel engine was running with a small amount of hydrogen addition, smaller than the present investigation, the NO_X emissions were very lower in comparison with the case without hydrogen addition. They attributed this reduction in NO_X with hydrogen addition to superior combustion characteristics of hydrogen that burns more rapidly and cleanly than hydrocarbon fuels because its amount was smaller and entered combustion reactions at higher velocity. Also, due to its lower activation energy, it incurred more molecular collisions than heavier hydrocarbon molecules. These characteristics might, not only have improved the combustion process but also enhanced the transport processes by reducing the hot spots in the combustion chamber that were one of the major contributors to NO_X emissions in IC engines.

The Canadian Hydrogen Energy Company Ltd (Canada 2005) tested the effects of injection of hydrogen in a 1992 Detroit diesel heavy-duty engine. The test was carried out on a go-power (Model DT-2000) heavy-duty dynamometer rated at 800 HP. This test was based on the widely used AVL 8-Mode heavy duty cycle for engine performance and emissions testing. In their test of direct injection diesel engines, the Hydrogen Fuel Injection (HFI) kit injected the gases during intake so that it got thoroughly mixed with intake air prior to diesel injection. The electrolysis cell of the HFI kit was constructed of seamless nickel tubing and the water chamber was made of seamless stainless steel. The power for the electrolysis was supplied from the vehicle's engine battery and hydrogen was only produced, on demand, when the vehicle engine was operating. Their experimental data showed that hydrogen burned nearly one order of magnitude faster than petroleum fuels, thus approaching ideal thermodynamic cycle. Hydrogen has a shorter flame quench distance, allowing flames to travel closer to the cold zones, thus improving combustion. These facts resulted in the reduction of fuel consumption by 4.44%, reduction of THC emissions by 6.17%, reduction of CO emissions by 0.39%, reduction of NO_X emissions by 4.34%, and reduction of PM emissions by 7%.

Birtas et al (2011) conducted an investigation on a diesel engine running with small amounts of the Hydrogen Rich Gas (HRG) provided by a water electrolyzer aspirated in the air stream inducted in the cylinder. The test engine was a naturally aspirated direct injection tractor diesel engine with 4 cylinders in line having the total capacity of 3759 cm^3, nominal power of 50 kW at 2400 rpm, maximum torque of 228 Nm at 1400 rpm, and the compression ratio of 17.5. The engine was operated at light and medium loads and at various speeds. They measured the engine operating parameters like instant torque, speed, engine air consumption, fuel consumption, fuel temperature, exhaust emissions (CO_2, CO, NO_X and THC) and the smoke opacity. Their results showed that at 2400 rpm and 40% load, the brake thermal efficiency (BTE) got decreased by 1% with 4% HRG (fuel energy) enrichment. This became 1.9% with 8% enrichment with HRG. The same trend was found at 60% load. At HRG addition of 4%, the BTE got reduced by 0.4% at 2400 rpm and by 0.8% at 1400 rpm. With 8% HRG, the relative decrease was 1.14% at 1400 rpm and 0.7% at 1000 rpm. They averaged 300 consecutives cycles for main cylinder pressure characteristics. With 4% HRG, the peak pressure got increased by 1.4% at 2400 rpm at 40% load. The rate of pressure rise got increased by 12% at 2400 rpm at 40% load. The maximum rate of heat release got increased due to the intensity of the first phase of combustion when HRG was added. This increase was 1.8% and 3% at 2400 rpm and at 40% load with HRG addition of 4% and 7.5% respectively. Oxides of nitrogen emission got decreased by 13.3% at 2400 rpm when the load on the engine was 40% with 4% of HRG addition. At 2000 rpm, the decrease was 12%. At higher HRG fractions, the decreasing trend of NO_X emission was reversed. With 15% HRG, the emission of NO_X had a relative increase of about 8%. They concluded that the higher NO_X emission with HRG enrichment was due to the extended Zeldovich kinetic mechanism. This also increased the peak pressure and the temperature when HRG was added. The emission of CO got lowered by 6.7% at 2400 rpm and at 40% load with HRG enrichment of 4%. This reduction was 8.3% at 1000 rpm and 11.5% at 1400 rpm. CO_2 emissions were continuously reduced when HRG was added. With 4% addition, the CO_2 values were less compared to the diesel operated engine by 3% at 2400 rpm and by 2.4% at 1000 rpm. Increasing the enrichment to 8%, the reductions rose to 5.3% at 1400 rpm and to 5.2% at 1000 rpm. At 40% of load, the total hydrocarbon emission (THC) got reduced by 1.7% and 7.6% at 1000 rpm by the enrichment of 4% and 8% of HRG respectively. They attributed this reduction to the lower presence of carbon and a possible more complete combustion. At 40% of load, the Filter Smoke Number (FSN) got decreased by 30% at 2400 rpm and by 8.9% at 1000 rpm when the HRG enrichment was 4%. The FSN change by HRG addition could also be attributed to a change in nature of the particulate matter produced. Without HRG induction, the particulates were black. With HRG induction, the particulates were colorless.

Niculae & Chiriac (2013) made a solution to recover the decrease of the gasoline spark ignition engines output which occurs in the case of transition to an alternative fuel like liquefied petroleum gas. For this study, they had chosen the numerical simulation by using the AVL BOOST code v. 2009. This solution involved the engine's dual fuelling with a mixture of LPG and hydrogen rich gas (HRG). This gas was resulting from water electrolysis. In their study, the spark ignition engine was fuelled successively with gasoline and with liquefied petroleum gas combined with different HRG flow rates of 5, 10, and 20 lpm corresponding to mass fractions of 3.7%, 7.9%, and 15.3% respectively. The engine was run at 2500 rpm. They also studied the effect of a possible increase of the compression ratio associated with the using of the minimum HRG flow rate on the engine's performance and efficiency. Their numerical study showed that even with changing the compression ratio of the engine to present the 9.5 to 11, the effective output of the gasoline engine was not recovered when pure LPG fuel was used with optimized start of combustion and the combustion duration. But, when HRG of 5 lpm was added with LPG at the compression ratio of 11, they successfully recovered the effective power output of the gasoline engine with optimized start of combustion and the combustion duration.

Bari & Esmaeil (2010) evaluated the performance of a conventional diesel engine when the diesel fuel was enhanced with hydrogen-oxygen mixture. They generated hydrogen-oxygen mixture on-board through electrolysis of water. They carried out experiment on a Hino, four-cylinder, direct injection, and water cooled diesel engine having a compression ratio of 17.9:1 and maximum output power of 38 kW at 1500 rpm. The engine was mounted to an electrical generator and the generator was then connected to an adjustable load cell to put load on the engine. In order to simplify the setup, they used 24V DC external power supply to generate H_2/O_2 mixture. But in reality it can be produced from the vehicle battery/alternator. They carried out experimental works under constant speed of 1500 rpm with three different power levels of 19 kW, 22 kW, and 28 kW and with varying amount of H_2/O_2 mixture. The experimental results showed that by using 4.84%, 6.06%, and 6.12% of total diesel equivalent of H_2/O_2 mixture, the brake thermal efficiency increased from 32.0% to 34.6%, 32.9% to 35.8%, and 34.7% to

36.3% at 19 kW, 22 kW, and 28 kW, respectively. These resulted in 15.07%, 15.16%, and 14.96% fuel savings. The emissions of HC, CO_2, and CO got decreased, whereas the NO_X emission got increased. At 19 kW the HC emission dropped from 187 ppm to 85 ppm with 31.75 lpm induction of H_2/O_2. At 22 kW and 28 kW the HC emission decreased from 189 ppm to 93 ppm by adding 29.84 lpm and from 192 ppm to 97 ppm by adding 30.6 lpm of H_2/O_2, respectively. NO_X emission was found to have increased from 220 ppm to 280 ppm, 232 ppm to 307 ppm, and 270 ppm to 339 ppm at19 kW, 22 kW, and 28 kW of load, respectively. The minimum amount of CO_2 was achieved at 19 kW as 2.06 ppm with 31.75 lpm of H_2/O_2 induction. The lowest CO_2 level was found as 3.17 ppm and 3.54 ppm at 22 kW and 28 kW, respectively. CO was reduced from 0.26% to 0.005% at 19 kW, from 0.24% to 0.012% at 22 kW and from 0.26% to 0.021% at 28 kW. They concluded that the flame speed of hydrogen was nine times faster than the flame speed of diesel. Therefore, the burning of diesel in the presence of hydrogen resulted in overall faster and more complete combustion. The peak pressure was also obtained closer to TDC which produced more work.

Musmar & Al-Rousan (2011) introduced HHO gas along with air into the intake manifold of a single cylinder unmodified Honda G 200, 197 cc, S.I. engine to find its impact on the combustion and emissions characterstics of the engine. The tests were conducted by varying the engine speed from 1000 rpm to 2300 rpm with constant load. A compact fuel cell had been designed for generating HHO gas for this test. The fuel cell used in this research was basically an electrolyte cell which decomposed distilled water (H_2O) into HHO. The caloric value of HHO gas was three times that of gasoline. The fuel cell was made of plates of stainless steel grade 316 L. The cell plates had an anode and cathode. Both of them were made of the same materials. According to their experience, stainless steel grade 302 and 304 could be used for the cathode. But, the anode must be made of stainless steel grade 316 L. Their results showed that a mixture of HHO, air, and gasoline caused a reduction in the concentration of emissions and an enhancement in engine efficiency. The NO_X emission got reduced to about 50%, the carbon monoxide concentration got reduced to about 20%. And, there was a reduction in fuel consumption in the range of 20% to 30%.

Dulger & Ozcelik (2000) reported that by using on-board hydrogen, the fuel consumption and the engine-out emissions could be reduced. To prove this, they carried out tests in four cars. These cars were a 1993 model Volvo 940, a 1996 model Mercedes 280, a 1992 model Fiat Kartal and a 1992 model Fiat Dogan. In their study, the hydrogen was generated by the process of electrolysis of water. In their test, the tap water was electrolysed by closed cell electrode technology. For long lasting nature, the cathode electrode of the cell was made up of compacted coal particles bonded together by a novel material. The anode electrode was made of platinum. Because of the use of carbon electrode, electrolysis efficiency got increased and heating of the electrodes got reduced. Due to the simultaneous production and consumption of hydrogen, no storage was used. The results showed that a small amount of hydrogen added to the incoming fuel-air mixture would enhance the flame velocity and permit the engine to operate with leaner mixtures. Consequently, because of the high catalytic tendency of hydrogen, the fuel burnt more completely and yielded significant reduction in exhaust emissions.

The driving tests under city traffic conditions showed that the fuel consumption for the Volvo 940 dropped to 6 l/100 km from 10.5 l/100 km, a reduction of 43%. The result for the Mercedes 280 was a drop from 11 l/100 km to 7 l/100 km, a reduction of 36%. The Fiat Kartal engine consumed 9.5 l/100 km without the system. With the system installed, the fuel consumption was 7 l/100 km which corresponded to a 26% reduction. The Fiat Dogan engine yielded 9 l/100 km without the system and 6 l/100 km with the system, a reduction of 33%. These results clearly demonstrated the fuel savings potential of the hydro gas system. Emission tests showed that exhaust emissions such as CO, CO_2, and hydrocarbons were not affected negatively by the system. Moreover, these emissions reduced up to a margin of 40 ± 50% depending on the type of the engine. Also, no performance penalty was observed. Acceleration, torque, and maximum power remained unchanged. Therefore, without altering any performance criteria, the system yielded 35 ± 40% fuel savings and reduced exhaust emissions.

Niculae et al (2013) contributed the results of an experimental research where LPG-air mixture was enriched with a Hydrogen Rich Gas (HRG) produced by the electrical dissociation of water. They carried out experiments on a four stroke SI engine. Main specifications of the engine were: 4 cylinders, 76 mm x 77 mm bore x stroke, 1397 cm^3 displaced volume, 9.5:1 compression ratio. LPG and HRG gas were individually supplied to the engine. The LPG was introduced in the original carburettor adequately modified, the HRG was introduced upstream of the carburettor. The addition of HRG was quantified by the gas flow rate of the electrolyser (5, 10 or 20 lpm). They carried out the experiments at engine light and mid load condition to measure engine torque and efficiency, exhaust emissions like NO_X, CO, CO_2, HC and cyclic variability related to combustion characteristics. They first inducted pure LPG to a certain relative air-fuel ratio λ. HRG was then gradually introduced and LPG was adequately reduced, to reach the same λ. By a similar manner, the relative air-fuel ratio was progressively increased to about λ=1.3 at light load, and λ=1.7 at mid-load. They also conducted tests with gasoline at various λ values for comparison. Their test result showed that when HRG of 10 lpm was added, the BTE got increased by 15% more than gasoline at λ=1.25. NO_X emission got increased when LPG was enriched with HRG. At 1600 rpm at λ=1.1, NO_X was higher by 30% with 5 lpm addition of HRG, by 50% with 10 lpm of HRG and by 116% with 20 lpm of HRG. The NO_X increase became much higher at λ=1.2. They stated that this was due to the extended Zeldovich kinetic mechanism. The CO emission was 20% less compared with pure LPG at λ=1.2, 43% at λ=1.4 and 64% at λ=1.65. With 20 lpm of HRG addition, HC emission was same as pure LPG, but became lower for leaner fuel-air mixtures. The difference was 40% at λ=1.2, 77 % at λ=1.4 and 90% at λ=1.65.

Chiriac et al (2006) investigated the outcome of the addition of Hydrogen Rich Gas (HRG) to the gasoline-air mixture of a four cylinder gasoline engine having a compression ratio of 9.5:1. The HRG was produced by the electrical dissociation of water. Experiments were carried out at engine light load (about 3 bar NIMEP) and partial load (about 6 bar NIMEP). The first load was thus representative by its highest sensitivity to the fuel-air ratio, while the second load was representative of typical engine operation. Experiments were carried out close to stoichiometric conditions and at lambda of 1.2. They did detailed measurements, namely, engine torque and efficiency, exhaust emissions, cyclic variability, heat release rates, and combustion duration. They concluded that all experiments with hydrogen addition had confirmed an extended range of the engine stable operation towards the lean limits. The faster combustion which approached the ideal Otto cycle and reduced engine knock tendency. The experimental results showed that the break thermal efficiency got increased by 7.4% and net indicated mean effective pressure (NIMEP) got increased by 5.6% more than gasoline combustion when 300 l/h of HRG was inducted. The overall burning duration also appeared to lower by HRG addition. CO emission concentrations were also substantially decreased by enriching the fuel with HRG. In lean mixtures, the presence of hydrogen acted as a catalyst in the CO oxidation kinetics. Also, the HC emissions got reduced due to improved combustion variability by decreasing the incomplete combustion of fuel-air mixture.

Asad & Wattoo (2003) reported about the usage of hydrogen in a CNG fueled engine. The engine used for their analysis was Suzuki Swift Sedan 1.3 L G13BA having a bore of 77 mm and stroke of 78 mm, displacement volume of 1293 cm^3, compression ratio of 7:1 and producing maximum power of 50 kW at 6000 rpm. The hydrogen used for this analysis was generated on-board by water electrolysis process. They carried out the electrolysis process by using tap water. For this, they stepped the alternator voltage to 96V and converted the same to DC power of 96V using rectifier and supplied it to on-board generator to generate hydrogen-oxygen mixture. They used stainless steel 316 material for anode electrode and carbon-graphite for cathode electrode. This system produced 17 liters per hour of gas mixture. In this study, they did not store the hydrogen-oxygen mixture; instead they generated and used the same when the engine was running. This also resulted in safe operation. To study the performance of the on-board hydrogen system, they conducted various tests to find the effects of hydrogen addition on fuel economy, engine power, efficiency, and exhaust emissions. They did not carry out any adjustments in engine ignition time and it was 6^o CA BTDC. They carried out all the experiments under urban and high-way driving conditions. From their results, it was interpreted that there was an increase in fuel economy by 5% to 7%. Also, under urban driving conditions the power got increased by about 4% to 7% when the hydrogen gas mixture was used with CNG. They attributed this to clean burning characteristics of hydrogen. They observed that at idle speed of 750 rpm, the CO emission for CNG was 0.07% by volume whereas when hydrogen mixture was added to CNG the CO emission was nearly zero. They also observed that the HC emission got reduced when hydrogen-oxygen gas mixture was added to CNG. It was 37 ppm for CNG operation and only 8 ppm for CNG with hydrogen mixture operation. They concluded that by inducting a small amount of hydrogen gas mixture, the flammability limit of fuel could be enhanced which in-turn reduced the exhaust emissions of the engine with increase in power.

Wang et al (2012a) examined the effects of H_2/O_2 addition on regulated harmful pollutants coming out from a heavy-duty diesel engine. They used a six cylinder heavy-duty diesel engine of Cummins make. The rated power of the engine was 118 kW at 2500 rpm and the maximum torque was 534 Nm at 1600 rpm. In their study, they used an oxy-hydrogen generator to generate the mixture of oxygen and hydrogen (H_2/O_2) by the process of water electrolysis. The generated gas mixture was aspirated into the diesel combustion process at various flow rates. Their results showed that brake thermal efficiency got increased from 31.1% of neat diesel combustion to 31.4% for 10 lpm and to 39.9% for 70 lpm of H_2/O_2 addition. They stated that this increase in brake thermal efficiency was due to better mixing of hydrogen with air which resulted in better combustion. The BSFC of neat diesel was 254.74 g/kWh, and was 262.06 g/kWh, 262.75 g/kWh, 260.42 g/kWh, 263.80 g/kWh, 246.51 g/kWh, 229.56 g/kWh, and 228.01 g/kWh for 10 lpm to 70 lpm of H_2/O_2 mixture addition in steps of 10 lpm, respectively. The THC concentration for neat diesel was 3.6 ppm and it got decreased to 3.51 ppm for 10 lpm of H_2/O_2 mixture, and to 3.25 ppm for 70 lpm of H_2/O_2 addition. They attributed this reduction of THC to high flame velocity of hydrogen and also to the absence of carbon atom in hydrogen fuel. The CO concentration was 26 ppm for neat diesel, and was decreased to 24 ppm for 70 lpm of H_2/O_2 addition. They related this decrease in CO emission to the operation of the hydrogen dual fuel engine at leaner equivalence ratios. They noticed the CO_2 concentration for neat diesel as 7906 ppm, and it got decreased to 7893 ppm for 10 lpm of H_2/O_2 addition and to 7523 ppm for 70 lpm of H_2/O_2 addition. They stated that the reduction of CO_2 emission was due to less carbon element in the formed mixture of fuels than that in the neat diesel. In their study, they obtained the NO_X concentration of 60.05 ppm for neat diesel, and it got increased to 60.49 ppm for 10 lpm of H_2/O_2 mixture, and to 67.22 ppm for 70 lpm of H_2/O_2 addition. They attributed this higher concentration of NO_X to both higher temperature and more available oxygen in the formed mixture. From their results, it was concluded that by using H_2/O_2 mixture in a diesel combustion process, the pollutants coming out from the diesel engine could be effectively reduced except NO_X emission.

2.12 COMBUSTION OF HYDROGEN UNDER VARIOUS INJECTION TIMINGS

Saravanan et al (2007) made an experimental investigation in a diesel engine using hydrogen as fuel with diesel as an ignition source for hydrogen. Hydrogen was injected into the intake port, while diesel was injected directly inside the cylinder. They varied the injection parameters such as injection timing and injection duration of hydrogen for a wider range at a constant injection timing of diesel as 23^o BTDC. They kept hydrogen flow rate as 10 lpm for varied load conditions. Their results showed that the brake thermal efficiency got increased from 23.6% to 29.4% for the injection timing of 5^o ATDC with injection duration of 90^o CA compared to diesel. But the maximum brake thermal efficiency obtained was 31.67% at 15^o ATDC with 60^o CA duration but at this condition they observed a knocking of the engine. The increase in brake thermal efficiency was attributed to better mixing of hydrogen with air that resulting in enhanced combustion. They found, the NO_X emission was minimum of 783 ppm for the injection timing of 5^o ATDC with the injection duration of 60^o CA compared to diesel of 1981 ppm at 75% load. The NO_X emission for diesel fuel operation at full load was 1806 ppm, whereas it got reduced to 888 ppm with injection duration of 90^o CA with injection timing of 5^o ATDC. The lowest NO_X of 705 ppm was obtained at full load with 60^o CA while the injection was made at TDC. The reduction was due to the operation of hydrogen engine at leaner equivalence ratios. At no load the HC was 3 ppm for the start of injection at 5^o AGTDC with 90^o injection duration for hydrogen operation compared to 19 ppm for diesel. At full load the hydrogen operated dual fuel engine resulting in increase in HC compared to diesel. For hydrogen operation it was 7 ppm at the injection timing of 5^o AGTDC with 90^o injection duration compared to diesel which was 42 ppm. The reduction in HC was due to the higher burning velocity of hydrogen; the absence of carbon in hydrogen fuel also reduced the HC emissions to a great extent. At 75% load condition the hydrogen operated engine at the injection timing of 5^o

ATDC and with 90^o injection duration, the smoke value was 0.4 BSN whereas for diesel it was 2 BSN. The lowest smoke value of 0.3 BSN was observed at the injection timing of TDC with 60^o injection duration. At full load for the start of injection at 5^o ATDC with 90^o injection duration, the smoke of 0.3 BSN was obtained compared to diesel of 3.8 BSN. They observed that at no load the CO levels of hydrogen operated engine at all operating conditions was less than diesel. The CO of 0.01% vol. for hydrogen operation at injection timing of 5^o ATDC and 90^o injection duration compared to that of diesel of 0.17% vol. At 75% load the peak pressure was 64.3 bar for the start of injection at 5^o ATDC with 90^o injection duration for hydrogen compared to diesel operation of 68.9 bar. At full load the hydrogen operated dual fuel engine resulted in an increase in peak pressure compared to diesel. The peak pressure for hydrogen operation was 71.7 bar at an injection timing of 5^o ATDC and 90^o injection duration compared to diesel of 73.7 bar. This was due to higher burning velocity of hydrogen, which made combustion as instantaneous. The peak heat release rate of hydrogen operated engine was 87.6 J/^{o}CA compared to diesel of 81.5 J/^{o}CA. The maximum heat addition also occured nearer to ITDC for hydrogen operation, which made the cycle efficiency to increase. They concluded that by operating the diesel engine in dual mode with hydrogen, one had to optimize the injection timing and injection duration.

Roy et al (2011) investigated the engine performance and emissions of a supercharged engine fueled by hydrogen, and three other hydrogen-containing gaseous fuels such as primary fuels, and diesel as pilot fuel in dual-fuel mode. They used energy share of primary fuels to about 90% or more, and the rest of the energy by diesel fuel. The hydrogen-containing fuels tested in this study were 13.7% H_2 content producer gas, 20% H_2 content producer gas and 56.8% H_2 content coke oven gas (COG). These fuels were simulated by mixing individual gas components in the appropriate proportions and inducted into the intake pipe with mass flow controllers. They kept intake pressure of air as constant at 200 kPa (supercharged) and the temperature as 30^{o}C throughout the study. For supercharging, they used a rotary type stationary compressor to compress the intake air to the desired pressure level, and controlled by a solenoid valve controlled regulator. This homogenous air-fuel mixture was ignited by a small quantity of diesel fuel, known as the pilot fuel that was injected towards the end of the compression stroke. Experiments were carried out at a constant pilot injection pressure and pilot quantity for different fuel-air equivalence ratios and at various injection timings. Their experimental strategy was to optimize the pilot injection timing to maximize engine power at different fuel-air equivalence ratios without knocking and within the limit of the maximum cylinder pressure. They observed better thermal efficiency with the increase in H_2 content in the fuels, and neat H_2 as a primary fuel produced the highest thermal efficiency. They decreased fuel-air equivalence ratio with the increase in H_2 content in the fuels to avoid knocking. Thus, neat H_2 operation produced less maximum power than other fuels, because of much leaner operations. In their tests they obtained two stages of combustion. This was an indicator of maximum power output conditions and a precursor of knocking combustion. The IMEP increased with advanced injection timings. The maximum IMEP of 1444 kPa occurred at an injection timing of 7^o BTDC. The maximum cylinder pressure was 12.81 MPa at an injection timing of 20^o BTDC, and reached its highest value of 14.47 MPa at 23^o BTDC. The cylinder pressure and temperature got increased when the injection timing was advanced. They also noticed that with advancement in injection time, the HC emission got decreased. They found that CO and HC emissions got reduced by 98% to 99.9% and NOx emissions by about 85% to 90% in neat H_2 operation compared to other fuels operation.

Fathi et al (2011) made a numerical analysis on a direct injection spark-ignition (DISI) hydrogen-fueled engine using 3-dimensional CFD methods and the results were compared with experimental data. For the analysis, they selected a single cylinder, 4-stroke, natural aspirated SI engine, which was converted from a direct injection diesel engine (Yanmar NFD-170), was used to evaluate engine operating characteristics such as injection timing, initial charge temperature, and initial charge pressure. In their study, they used AVL FIRE CFD code for analyzing the parameters. This code uses a 3D moving mesh for combustion chamber simulation. For their analysis of effect of initial charge temperature on the emissions and combustion characteristics, they selected three injection timings 130^o BTDC, 100^o BTDC, and 80^o BTDC. By comparison of the profiles, it was understood that by early injection of hydrogen into the cylinder, the air and fuel could mix homogeneously and combustion could occur completely. Therefore, the work done per cycle also got increased. Retarding the hydrogen injection into the combustion chamber resulted in a decrease in the work done per cycle. This was due to the time needed for air-fuel mixing, which decreased with the retarding of the hydrogen injection such that the air-fuel mixture would be inhomogeneous. Additionally, a high concentration of hydrogen around the spark plug combustion started fast and ceased rapidly. In other words, the combustion duration decreased and therefore pressure was less in the expansion stroke, which resulted in decreased work. By increasing the initial charge temperature, the maximum in-cylinder pressure was decreased. By increasing the initial charge temperature, the density of inlet mass and the amount of oxygen got decreased, which led to deficient combustion and a reduction of maximum in-cylinder pressure. By increasing the initial charge temperature, the oxygen concentration got decreased and the amount of energy needed for decomposition of N_2 and O_2 molecules got increased. Because of the increased mean and maximum in-cylinder temperature more NO_X emissions were produced. Inhomogeneous mixture formation as a result of retarding the injection timing (late injection) caused incomplete combustion and, therefore, the maximum in-cylinder temperature also got decreased. Because of decreased inlet air mass as a result of increasing the initial charge temperature, the IMEP decreased and ISFC increased. In early injection, due to complete combustion, IMEP increased and ISFC decreased. By increasing the initial pressure, the mean and maximum in-cylinder pressure got increased. This might be due to the fact that with the increasing of the initial pressure, the inlet air mass got increased. They concluded that by advancing the hydrogen injection time had a significant effect on engine performance characteristics such as work, IMEP, ISFC, and the production of NO_X emissions. By changing initial conditions such as pressure and temperature, optimum engine operating conditions could be found for DISI engines with alternative fuels.

Shioji & Mohammadi (2006) carried out an investigation on diesel engine performance and emission characteristics when LCG (Low Calorific Gases) and LCG with small portion of hydrogen was inducted into the inlet manifold of engine with varied injection time of diesel fuel. The test engine used by them was a four-stroke single cylinder naturally aspirated direct-injection diesel engine (Yanmar NFD-170) with a bore of 102 mm and a stroke of 105 mm. In their experiment, the original injection pump was replaced with a jerk type pump (Zexel Type A) and nozzle (DLL-P) with four holes and hole size of 0.24 mm to match the plunger size. They varied the injection timing of diesel fuel in the range of 7.5^o BTDC to 15^o BTDC at the engine load of 0.6 MPa. In their experiment, they fixed the amount and composition of LCG as 15% and H_2 as 30%. Their results indicated that advancing injection timing improved thermal efficiency. This trend was similar for both fuels. This advancement in injection timing also improved smoke, TC and CO emissions but the NO_X emission got worsened. They observed that at LCG addition, even at advanced injection condition the NO_X emission level was much lower than that in diesel fuel operation. From their results, it was obvious that even in advancd injection condition the LCG addition gave low NO_X emission, along with almost same thermal efficiency as diesel fuel operation.

Tomita et al (2000) investigated the effect of hydrogen injection in a diesel engine running on dual fuel mode. The engine used for their study was a single cylinder, four-stroke diesel engine with bore and stroke of 92 mm and 96 mm, respectively. The compression ratio was 17.7, and the type of combustion chamber was deep dish. They varied the the injection timing of light oil over a very wide range from 60^o BTDC to 5^o ATDC. They equipped a pressure transducer in the cylinder head to determine the cylinder pressure. They inducted gaseous hydrogen into the engine cylinder from an intake port and injected light oil in a conventional way into the cylinder. For reference, they took ordinary diesel condition that only used air in the cylinder. Their results showed that when the injection timing of the light oil was earlier, the value of the second peak of the heat release became smaller because of the increase in the rate of premixed combustion. When the injection timing was 11.7^o BTDC, the value of the first peak was the largest. When the injection timing was earlier than 30.4^o BTDC, the value of the first peak became smaller without second peak and the heat release rate showed the characteristics of premixed combustion. They reasoned this to longer ignition delay. The ignition delay got increased with advancing of injection timing. When they advanced the injection timing to near 25^o BTDC, the NO_X got increased and had the maximum value. When the value of NO_X was large, the heat release rate was also large. When the injection timing was advanced more, the value of NO_X decreased gradually. They attributed this to well-mixing of light oil with air. They witnessed a trade-off relation between hydrocarbon and the nitrogen oxide when hydrogen was inducted. They stated that this was due to more complete combustion. They further observed that when hydrogen was mixed with inlet air, emissions of HC, CO and CO_2 got decreased without smoke. However, the brake thermal efficiency got decreased slightly less than the ordinary diesel combustion. In particular, both smoke and NO_X were almost zero and HC was low when the injection timing was significantly advanced.

In another experiment, Tomita et al (2001) investigated a dual fuel engine of hydrogen and diesel oil. Hydrogen was inducted into the intake port with air and diesel oil was injected into the cylinder. They varied the SOI timing of the diesel fuel across a wide range, holding the overall equivalence ratio equal with and without the addition of hydrogen. Their test results showed a very low

NO_X emission when SOI was advanced to or beyond 40° BTDC, with NO_X emissions rising with later injection timing. This was reasoned to have occurred because of the thorough mixing of the hydrogen/air mixture and the diesel fuel before ignition. They also observed that CO_2 got decreased proportionally to the amount of hydrogen substituted for diesel fuel, due to less carbon available in the reactants. They stated that the addition of hydrogen caused an increased ignition delay and they reasoned this to the large mole fraction of hydrogen in the air which displaced oxygen. They witnessed both smoke and NO_X becoming almost zero and HC was low when the injection timing was significantly advanced although the engine operation became unstable. In overall they obtained slight decrease in efficiency of the engine with the addition of hydrogen.

2.13 COOLING LOSS IN HYDROGEN COMBUSTION ENGINES

Shudo et al (2001) analyzed the factors influencing the thermal efficiency of a homogeneous charge spark ignition (SI) engine fuelled with hydrogen, focusing on the degree of constant volume and the cooling loss. They used a four-stroke four-cylinder spark ignition gas engine which was modified from a gasoline engine for passenger cars (bore 85 mm, stroke 88 mm, compression ratio 8.5) for their tests. They supplied hydrogen or methane continuously into the intake manifold of the test engine. The fuel gas flow rate was measured with a mass flow meter (Oval, F203S). In all the experiments, they kept the engine speed as constant at 1500 rpm and the volumetric efficiency as 50 percent including fuel gas in order to avoid the flash-back phenomenon. They measured the in-cylinder pressure with a piezoelectric type pressure transducer (AVL, GM12D). They averaged the pressure data over 200 cycles and used it to calculate the indicated thermal efficiency, the rate of heat release, the degree of constant volume, etc. They measured the instantaneous temperature using a thin-film-type thermocouple (Medtherm, TCS-103E, chromel– constantan type). They analysed the exhaust gas using an exhaust gas analyser (Horiba, MEXA 9100) for calculation of the combustion efficiency. They evaluated the cooling loss from the burning gas to the cylinder walls in a homogeneous charge SI engine quantitatively by analysing the cylinder pressure diagram and the exhaust gas composition. They also obtained degree of constant volume burning and the degree of constant volume cooling by fitting the Wiebe function to the rate of heat release calculated using the cylinder pressure diagram. To evaluate the cooling loss, they compared the hydrogen combustion with methane. They revealed that the cooling loss in hydrogen combustion was higher than that of methane combustion due to a thinner quenching distance and faster burning velocity for hydrogen combustion. They also measured the cylinder pressure and instantaneous wall temperature by varying the ignition timing for both hydrogen and methane combustion. They stated that hydrogen combustion had a shorter combustion period due to its higher burning velocity than methane. They observed that the combustion chamber wall temperature tended to increase with an advance in ignition timing for both fuels. They noted that the apparent rate of heat release $dQ/d\theta$ in hydrogen combustion was greatly influenced by the cooling loss. They attributed this higher cooling loss of hydrogen combustion to a thin temperature boundary layer thickness on the combustion chamber wall due to a shorter quenching distance. They also stated that the high burning velocity of hydrogen combustion might also have caused intense convection between the burning gas and the wall, it resulted in an enhancement of the heat transfer. They further noticed that in combustion of both fuels, the cooling loss ratio tended to increase with an advance in ignition timing. This was significantly higher in hydrogen combustion than methane combustion at the same ignition timing. The cooling loss tended to decrease with the retardation of ignition timing for both hydrogen and methane combustion. They confirmed this trend by taking instantaneous heat flux measurements at a representative location in the cylinder head. They obtained lower thermal efficiency in hydrogen combustion than that of methane combustion in throttled conditions. This was due to higher cooling loss ratio and the higher degree of constant volume cooling in hydrogen combustion. They concluded that to improve the thermal efficiency of hydrogen-fuelled engines, it is essential to reduce the cooling loss.

Saravanan & Nagarajan (2009) investigated the effect of hydrogen addition on diesel engine performance and emission characteristics, by injecting hydrogen in timed manifold injection, port timed injection and carburetion technique along with reduced cooling water flow rate of 75% in order to decrease the percentage of cooling loss. They conducted their experiments in a single-cylinder, four-stroke, water-cooled, direct-injection diesel engine running at a rated power of 3.78 kW coupled to an electrical generator. Their results showed that the brake thermal efficiency for hydrogen with diesel as an ignition source in timed port injection technique was 27.3% at full load and 26.2% in manifold injection whereas in the diesel combustion it was 23%. NO_X emission in timed port injection technique was 34% higher and in carburetion technique, it was 8% higher compared to baseline diesel. With timed port injection technique, the smoke level varied from 2.1 BSN to 3.5 BSN with a reduction of 45% at full rated load compared to diesel fuel. The HC emission was lower at full load compared to the base line diesel, the maximum being 0.047 g/kWh and 0.039 g/kWh in the case of timed manifold and port injection technique respectively. In carburetion technique, the HC emission was 0.106 g/kWh at full load compared to base line diesel fuel of 0.12 g/kWh. Their study very clearly gave a direction that by changing the cooling water parameters, the performance as well as emission characteristics of the engine could be enhanced.

Shudo (2005) reported about the cooling loss of internal combustion engine when hydrogen was used in the combustion process. For his study, he chose a 4-cylinder, 4-stroke, spark-ignition engine having a bore and stroke of 85 mm and 88 mm, respectively. He measured the hydrogen with a mass-flow meter Oval F203S and he continuously supplied hydrogen to the intake pipe of the engine. He kept the engine speed as fixed at 1500 rpm, excess air ratio at 1.0 and volumetric efficiency at 35% including the fuel gas. He measured the in- cylinder pressure data with a piezoelectric type pressure transducer AVL GM12D installed in the cylinder head. For calculating the apparent rate of heat release and the in-cylinder gas temperature, he averaged the cylinder pressure data for 50 cycles. He observed larger changes in the composition and the thermo-physical properties of the in-cylinder gas. He related this to larger burning velocity of hydrogen than hydrocarbons. He calculated the heat transfer coefficient from the rate of cooling, which was estimated by the apparent rate of heat release and Wiebe function. Then he calculated representative velocity of the in-cylinder gas derived from the turbulent heat transfer equation for pipe flows by using the obtained heat transfer coefficient. He proved that by adjusting the velocity term in the widely used equation of Woschni, the equation might calculate better results for hydrogen combustion.

Shudo (2007) proved that a direct injection stratified charge is an effective technique to improve the thermal efficiency of hydrogen combustion engines. For his study he used a spark ignition engine fueled with hydrogen with a bore of 85 mm, a stroke of 88 mm, and a compression ratio of 8.5. He maintained the engine speed at 1500rpm, and charged the engine with a homogenous mixture of hydrogen and air through the intake pipe of the engine. He set the mixing ratio to stoichiometric which resulted in higher cooling losses than leaner mixture conditions. He carried out the experiments at different spark ignition timings of the engine. In order to avoid backfire from the engine particularly with stoichiometric mixture, he controlled the volumetric efficiency to 50% by using a throttle valve. He measured the combustion pressure with a piezoelectric type pressure transducer AVL GM12D installed in the cylinder head and averaged over 45 cycles. He calculated the apparent rate of heat release $dQ/d\theta$ with the pressure data. He observed that the apparent rate of heat release $dQ/d\theta$ turned negative just after the end of combustion, and this signified that a part of heat released by the combustion was transferred to the combustion chamber walls. He concluded that retardation in the spark ignition timing could decrease the cooling loss by lowering the maximum combustion temperature. However this method also decreased the degree of constant volume and did not lead to effective thermal efficiency improvements. He proved that the increase in the apparent heat release fraction effectively led to improvements in thermal efficiency, because the stratified charge reduced the cooling loss without lowering the degree of constant volume. Therefore, a method that reduces cooling loss without lowering the degree of constant volume will be effective to improve the thermal efficiency of internal combustion engines.

Sudheesh & Mallikarjuna (2010) made an experimental investigation on the acetylene-fueled engine operated in homogeneous charge compression ignition (HCCI) mode to study the effect of cooling water flow direction on intake charge temperature, heating requirements, and performance of the engine. They conducted their study in a single-cylinder, water-cooled, CI engine operated in HCCI mode with acetylene as fuel. They used an external electrical heater to heat the intake charge to achieve HCCI combustion. They kept engine speed and cooling water outlet temperature as constant. In their study, they changed the coolant water flow direction in HCCI mode from bottom-up to top-down flow. They did this in order to reduce heat transfer from cylinder liner to cooling water so that higher cylinder liner temperature would be maintained. They operated the engine also in the conventional CI mode to find the effect of HCCI mode with the conventional and reverse cooling water flow directions using different intake charge temperatures at different loads. Their results showed that the reverse cooling water flow direction showed about 14% to 50% reduction in external intake charge heating at different load conditions as compared to the conventional cooling water flow direction. The brake thermal efficiency got improved by about 5% to 10% at different load conditions. Nitric oxide (NO) and smoke emissions were very low in both cases. However, hydrocarbon (HC) and carbon monoxide (CO) emissions were higher than conventional CI mode in both the cases. In conventional flow direction, peak heat-release occured at about 364 CAD, whereas for reverse flow direction, it was at about 358 CAD. In reverse flow direction, the combustion phase occured closer to TDC. Whereas, in conventional case it occured far away after compression TDC. This reduced the net work output. They attributed this to higher cylinder liner temperatures with reverse flow direction in which heat transfered into cylinder gases. Also, peak heat release in former case was about 80 J/CAD, whereas in later case, it was about 150 J/CAD. They concluded that by reducing heat transfer from cylinder liner to cooling water, the performance and emission characteristics of the engine could be improved.

Torregrosa et al (2006) investigated about the influence of coolant and inlet charge temperature on the emissions and performances of a DI diesel engine. For their study, they used a unique test rig which could control the important operational parameters of the engine. They converted multi-cylinder engine to single cylinder operated engine. They selected three operating points to represent the European homologation cycle and tested the engine. They controled the temperature of the coolant leaving the engine block by regulating the tap water flow through the shell and tube heat exchanger. The tap water valve was regulated by a PID controller which used the temperature measurement at the exit of the engine. They used an external screw compressor with a dryer to supply the required air. They conveyed the dry air to the test bench where it passed through a heater. Their experimental results showed that the coolant temperature influenced the combustion process through the temperature of the combustion chamber walls. When they raised the coolant and inlet charge temperature, they witnessed diminishing effect in hydrocarbon emissions and increasing effect in NO_X emissions. At low load these effects were more pronounced. The NO_X emissions got increased significantly with increasing air temperature. They stated that as NO_X formation was a thermally activated process and thus when the air temperature was higher, NO_X formation was enhanced. They observed increase in

HC emissions when the temperatures of wall and the inlet air were lower. They related this to enhanced flame quenching and increase in the ignition delay period. They further observed that at low loads, more HCs were emitted when the walls were colder. They related this to incomplete combustion. They concluded that by optimizing the cooling water and the inlet air temperatures, the emissions from the engine could be greatly reduced.

2.14 COMBUSTION UNDER VARIOUS INJECTION PRESSURE OF DIESEL FUEL

Rao et al (2011) made an experimental investigation on LPG, a by-product of petroleum refining process to use it as a supplementary fuel for diesel engine. They used four-stroke, single-cylinder diesel engine, most widely used in agricultural sector for their experimentation. They varied fuel injection pressure of diesel by adjusting the spring stiffness of the fuel pump. Their experimental set up consisted of a single-cylinder, four-stroke diesel engine connected to an eddy-current dynamometer for the loading of the engine. It was provided with necessary instruments for combustion pressure and crank angle measurements. The signals were interfaced to a computer through an engine indicator to obtain pressure-crank angle diagram. They also made provision for interfacing air flow, fuel flow, temperatures and load measurement. They measured the fuel injection pressure by using a mechanical type of hand operated pressure gauge. They varied the injection pressure of diesel fuel in the range of 180 bar to 220 bar with a step of 10 bar. Their results showed that the brake thermal efficiency of the engine at low engine load of 20% was found to decrease with an increase in the LPG content. They related this to poor combustion characteristics of air-LPG mixture at low load conditions. High auto-ignition temperature of LPG might be the basis for the poor combustion characteristics of air-LPG mixture. At low engine load of 20%, at an injection pressure of 180 bar, the brake thermal efficiency was found to be 13.5% on diesel fuel mode, and reduced to 9.1% on dual-fuel mode with 50%

LPG energy. At mid engine load of 60%, the fuel injection pressure had a significant effect on the brake thermal efficiency of the engine. On diesel fuel mode of operation, it got increased from 21.1% at an injection pressure of 180 bar to 22.3% when the fuel was injected at 200 bar. At 80% engine load, on dual-fuel mode of operation with 20% LPG energy, it was found to be 25.7% when the fuel injection pressure was 180 bar, and increased to 26.8% when fuel was injected at 200 bar. They observed decrease in smoke density with an increase in the fuel injection pressure on both the modes of operation. They obtained lowest smoke when the fuel injection occurred at 220 bar. The smoke of 8.8 HSU of diesel operation got decreased to 2.9 HSU on dual-fuel mode with 50% LPG. They related this to better combustion of air-LPG mixture. They further observed that the emissions of NO got increased with an increase in the fuel injection pressure on both the modes of engine operation. They concluded that at lower engine loads, the engine performance on dual-fuel mode was found to be inferior compared to that of the diesel fuel mode. At mid and higher engine loads, the dual-fuel mode of operation was found to be superior compared to that of the diesel fuel mode of operation. At low engine loads, the engine might be operated on the diesel fuel mode, and at higher engine loads, the engine could be switched over to the dual-fuel mode.

Bakar et al (2008) reported about the effects of fuel injection pressure on engine performance of a diesel engine with four-cylinder, two-stroke, direct injection. They investigated engine performance values such as indicated pressure (IP), indicated horse power (IHP), shaft horse power (SHP), brake horse power (BHP), break mean effective pressure (BMEP) and fuel consumption both of variation engine speeds - fixed load and fixed engine speed - variation loads by changing the fuel injection pressure from 180 bar to 220 bar. They ran the engine in the range of 600 rpm to 1600 rpm with the interval of 200 rpm and the fuel injection pressure setting from 180 bar to 220 bar with the interval of 10 bar. In their second phase of experiments, the diesel engine loads were tested in 55% to 80% of the rated load with the interval of 5% and they fixed the engine speed as 1600 rpm. Their experimental result showed that the engine performance parameters like IP, IHP, SHP, BHP, and BMEP got increased when the injection pressure of the fuel was increased. They related this to excellent mixing of fuel and air in the combustion chamber. This also resulted in less un-burnt fuel than the lower injection pressure combustion operation. In the second phase of experiments on fixed engine speed and variation engine loads, they obtained highest engine performance at the injection pressure 220 bar. They observed increase in the specific fuel consumption when the engine speed was increased. They found economical specific fuel consumption at the injection pressure of 200 bar. They observed enlargement in fuel particle diameters when the fuel injection pressure was low. This also resulted in an increase in ignition delay period and an increase in cylinder pressure. On the otherhand, they witnessed a decrease in ignition delay period when the injection pressure was increased. This also resulted in an increase in the performance of the engine. They ascribed this to a decrease in the fuel particle diameters and an increase in the degree of formation of homogeneous mixture of fuel and air. They also noticed that if the fuel injection pressure was too higher, ignition delay period became shorter. So, the possibilities of homogeneous mixing got decreased and combustion efficiency got reduced. They attributed this to more energy needed to drive the injection system by reducing the leak flow and by dynamically adjusting the maximum pressure to the actual needs of the engine.

Siebers & Pickett (2002) investigated about the effects of injection pressure and orifice diameter on soot in diesel fuel jets under quiescent, direct-injection (DI) diesel engine conditions. They carried out these investigations in a constant-volume combustion vessel with complete optical access. The injector used was an electronically-controlled, common-rail injector. They considered injection pressures between 40 MPa and 190 MPa and orifice diameters between

501.1m and 180 μm. They measured the soot with a line-of-sight laser extinction technique and visualized with planar laser-induced incandescence. They determined flame lift-off lengths used in the analysis of the soot measurements with time-averaged OH chemiluminescence imaging. Their results showed that the peak soot in a fuel jet decreased with increasing injection pressure and decreasing orifice diameter. The decrease in soot with increasing injection pressure was linear with increasing injection velocity (i.e., the square-root of the pressure drop across the injector orifice). The decrease in soot with decreasing orifice diameter was such that for the smallest orifice diameter considered (50 μm), no soot was detectable for the ambient gas and injector conditions considered. These direct measurements of soot within a fuel jet, coupled with estimates of the amount of air entrained relative to the amount of fuel injected, confirmed that air entrainment and fuel-air mixing upstream of the lift-off length had an important role in determining the soot levels within a diesel fuel jet.

Ghazikhani et al (2007b) conducted an investigation on the soot emission level on an OM314 DI diesel engine. The engine used for their investigation was a four-cylinder, four-stroke, direct injection, naturally aspirated diesel engine with a peak power of 63 kW and peak torque of 235 Nm at 1830 rpm. This was a 2006 production engine Daimler OM314 with a compression ratio of 17:1, bore and stroke of 97 mm and 128 mm and displacement volume of 3.784 L. They installed and operated the engine on a 112 kW DXF Heenan & Froude hydraulic dynamometer. They used AVL 415 smart sampler to measure soot emission levels in the exhaust gas. They mounted an electrical speed meter on the engine flywheel to measure the engine speed. During the tests, they operated the engine at three different conditions, naturally aspirated, LIMP turbocharged and elevated injection pressure from 200 bars to 240 bars. In each stage, in order to investigate the soot contents, they tested the engine under ECE-R49, 13 mode standard test. They observed that the naturally aspirated engine had acceptable performance characteristics but needed urgent attention in view of the soot emission levels. LIMP turbocharged engine had reduced the soot emission level to the extent of 35%. They related this to the higher pressure and temperature of intake-air and reduction of ignition delay which improved the soot burnout process. When the injection pressure was elevated, they observed that the specific soot pollution decreased by about 26% less than the turbocharged engine. They stated that this was due to smaller fuel particles diameter and better formation of mixing fuel to air. They further observed that the soot emission level of the engine was 21% less than the Euro I stndard level but it did not match with the Euro II standard level.

Ghazikhani & Darbandi (2010) studied the effects of injection pressure on BSFC and exhaust emissions on a direct-injection turbocharged diesel engine. They measured emissions and engine BSFC values in 13 speed and load conditions based on ECE-R49 test by changing injection pressure from 200 to 300 bars. The engine used for their test was OM314 turbocharged diesel engine equipped with a waste gate supported by Borg Warner Company to convert the naturally aspirated engine to a turbo-charged diesel engine. The test rig was equipped with a 112 kW DXF Heenan & Froude hydraulic dynamometer. They measured engine emissions using a Plint-RE205 gas analyzer for unburned hydrocarbons as C6H14, CO, CO_2 and O_2. Their experimental apparatus composed of AVL-415 smoke meter, thermocouples type K by using an interface connected to a PC, volumetric fuel meter, air meter, pressure gauges and electrical engine speed meter. Their results showed that the specific HC emission got reduced when the injection pressure was increased. They related this to a better mixing of air and fuel. They also stated that if injection pressure is too high, the HC emission got increased because of impinging on the body of the cylinder and cooling loss. The oxygen emission got reduced when the injection pressure was increased due to lower ignition delay and higher combustion rate at better fuel air mixing. The CO emission got increased when the injection pressure was too high. They reasoned this to higher exhaust gas temperature which caused for the dissociation of CO_2 to CO and O_2 with higher injection pressures. The amount of specific CO_2 emission got increased upto moderate rise in injection pressure. They attributed this to better mixing of fuel and air which in-turn increased combustion efficiency. They further observed that the specific O_2 emission got decreased with rise in injection pressures. The specific NO_X emission was almost constant inspite of variation of injection pressures. They related this to not much change in peak temperature. They noticed that the smoke level got reduced when injection pressure was increased to 235 bars; after that smoke level got increased when injection pressure was increased to 300 bars. They stated that this was due to lower ignition delay and better combustion rate. But, when the injection pressure was increased to more than 235 bars, the ignition delay period became shorter because of increase in cylinder temperature. This in-turn decreased possibility of homogeneous mixing and resulted in increase in smoke level. They obtained improvement in engine specific fuel consumption with higher injection pressure due to lower ignition delay and higher combustion rate at better fuel air mixing. They concluded that variable injection pressure was a better one than the constant injection pressure.

2.15 COMBUSTION UNDER VARIOUS INLET AIR TEMPERATURES

Paykani et al (2011) made an experimental investigation on a Lister 8-1 dual fuel (diesel - natural gas) engine to examine the simultaneous effect of inlet air pre-heating and EGR on performance and emission characteristics of a dual fuel engine. As the use of EGR at high levels was unable to improve the engine performance at part loads, they combined EGR with pre-heating of inlet air for their experiments. Their results showed that there was a slight increase in thermal efficiency, CO and HC emissions got reduced by 24% and 31%, respectively. The UBHC emission also got reduced to some extent. The NO_X emissions got decreased by 21%. They related this to the lower combustion temperature due to the much inert gas brought by EGR and decreased oxygen concentration in the cylinder. They observed that when the intake temperature of air was increased, the flame was propagating successfully through the gaseous fuel air mixture. They further observed that when the inlet charge temperature was increased, it resulted in decrease in ignition delay period and eventually improved the combustion process. They witnessed a close trend of CO emission variation with intake mixture temperature. When the inlet air charge temperature was increased, the CO emission got decreased. They stated that this was due to an increase in the reaction rate of the fuel mixture and resulted in widened flammability limits.

Shahadat et al (2005) conducted the experiment in a four-stroke DI diesel engine to evaluate its characteristics when preheated air was inducted in a combustion process. The engine used for their analysis was a single cylinder, 4-stroke, water cooled diesel engine having a bore of 95 mm and stroke of 115 mm. It produced rated output of 10 kW at a rated speed of 2000 rpm and having a compression ratio 20:1. They used conventional diesel fuel and diesel-kerosene blend as fuels for their experiments. They preheated the inlet air from 32°C to 55°C and 60°C by taking heat from the exhaust gases. Their results showed that there was a significant reduction in NO_X emission when the air was preheated. They obtained the NO_X emission of 125 ppm, 360 ppm, 460 ppm, and 465 ppm when the air was not preheated in diesel fuel mode for the speeds of 900 rpm, 950 rpm, 1000 rpm, and 1150 rpm respectively. When air was pre-heated to 55ºC, the NO_X emissions got reduced to 100 ppm, 125 ppm, 175 ppm, and 210 ppm for the above mentioned speeds. They attributed this remarkable reduction to reduction in ignition delay which in-turn reduced engine emissions. For the above mentioned speeds the CO emission got reduced from 625 ppm to 390 ppm, 550 ppm to 325 ppm, 500 ppm to 275 ppm and 425 ppm to 250 ppm respectively. They reasoned this to increase in combustion rate of unburnt hydrocarbons. The brake thermal efficiency also got increased when the air was pre-heated. When the inlet air temperature was increased from 32ºC to 45ºC, 55ºC, and 60ºC, they obtained increase in brake thermal efficiency from 22% to 26%, 27%, and 28% respectively. They related this increase in brake thermal efficiency to increase in the degree of constant volume combustion. They concluded that by increasing the inlet air charge temperature, the brake thermal efficiency of the engine could be increased with reduction in emissions.

Murayama et al (1971) investigated the effect of inlet air preheating on in-cylinder combustion and exhaust emissions of a diesel engine when various fuels of low cetane number were used. The engine used for their study was single cylinder, vertical type, water cooled, four stroke diesel engine having a stroke volume of 780 cc and compression ratio of 19:1. They pre-heated the inlet air in the range of 20ºC to 100ºC by using electrical heater of capacity of 2 kW inserted into a inlet manifold of the engine. The multiple fuels used in their study were gas oil, undoped gasoline, regular gasoline and premium gasoline. Their results showed that the combustibility of engine could be improved remarkably by using pre-heated air. The heat release rate, rate of pressure rise, and peak cylinder pressure got decreased. They related this to decrease in ignition lag period. This was more pronounced when the injection time was advanced. The specific fuel consumption and the brake thermal efficiency got increased. They related this to increase in specific heat of combustion mixture and reduction of excess air factor. They observed decrease in smoke level when inlet air was pre-heated. They concluded that by pre-heating the inlet air, the low cetane number fuels could be comfortably burnt in a diesel engine.

Naber & Siebers (1998) investigated the auto-ignition and combustion of hydrogen in a constant volume combustion vessel under simulated direct-injection (DI) diesel engine conditions. The parameters varied in the investigation included: the injection pressure and temperature, the orifice diameter, and the ambient gas pressure, temperature and composition. They conducted experiments in a constant volume combustion vessel using a high pressure gaseous fuel injector. The vessel had a disk-shaped combustion chamber (114 mm diameter and 28.6 mm width) with sapphire windows at each end to permit full field line-of-sight optical access. They designed the combustion vessel for the study of an extended range of ambient conditions including combustion pressures to 35 MPa. They used a modified version of fuel injector developed by Diesel Technology Corporation for a natural gas DI compression ignition engine with an electronic control. For their experiments, they supplied high purity gaseous hydrogen from a gas cylinder and boosted to injection pressures by an oilless pump. Their results showed that the ignition delay of hydrogen under DI diesel conditions had a strong, Arrhenius dependence on temperature of inlet air condition which reflected the importance of chemical kinetics; however, the dependence on the other parameters examined was small. For gas temperatures greater than 1120 K with oxygen concentrations as low as 5% (by volume), they observed ignition delays of less than l.0 ms. They measured ignition delays for two different initial hydrogen temperatures of 450 K and 410 K. When comparing the data of the above two temperatures, they found that the ignition delay got increased with a decrease in the fuel temperature. And, they also found that in comparison to the effect of change in ambient gas temperature, the effect of a change in the fuel temperature was small. They concluded that compression ignition of hydrogen was possible in a diesel engine at reasonable TDC conditions.

Alam et al (2005) conducted experiments to study the effects of changing the inlet air temperature on the performance and emission characteristics of a Cummins 5.9L, turbocharged, water cooled, six-cylinder, 4-valves per cylinder direct injection (DI) diesel engine. The other specifications of the test engine were: rated power of 235 hp at 2700 rpm, bore and stroke of 102 mm and 120 mm, compression ratio of 16.3:1, displacement volume of 5.9 L, injection system of Bosch VP-44 rotary distributor pump, and swirl ratio of 2.45. Their experimental system consisted of an engine, dynamometer, controller, combustion analysis instrumentation, and different emissions analyzers. They fitted the engine with an ECM that monitored engine performance and controled different events automatically, especially the start of injection (SOI), injection timing advancement or retardation. As the engine was fitted with a turbocharger, they cooled the compressed hot air coming out from the turbocharger in an aftercooler to control the inlet air temperature. They manipulated water circulation to the aftercooler to control the inlet manifold air temperature. They varied the inlet manifold air temperature between 30^{o}C and 60^{o}C. They collected all the combustion, performance and emissions data in each and every temperature of inlet air by keeping the engine speed as constant at 1408 rpm with an external load of 5.12 kNm. Their experimental results showed that a small decrease of cylinder pressure was observed with an increase in inlet air temperature. However, the peak pressures were observed at the same crank angle. The intake air temperature affected the ignition delay due to its effect on charge conditions during the delay period. They observed decrease in ignition delay with an increase in inlet air temperature. They attributed this to the increase in vaporization of fuel-air mixture to make a combustible mixture. They witnessed earlier start of combustion when the inlet air temperature was increased. On the other hand, the higher the inlet air temperature, the lower the premixed and diffusion burning peaks. It was also true that the premixed combustion stage decreased with shorter ignition delay. They further observed that average in-cylinder temperature increased with an increase in inlet air temperature. When the inlet air temperature was changed from 30^{o}C to 60^{o}C, the average cylinder temperatures got increased by 150K. They witnessed higher NO_X emissions when the temperature of the inlet air was increased due to increase in average cylinder temperatures. The air flow rate decreased with increase in inlet air temperature. Similarly, specific fuel consumption got decreased with an increase in inlet air temperature which was confirmed with the higher exhaust gas temperature. CO emissions got increased and hydrocarbon emissions got decreased with an increase in inlet air temperature. They related this to the decrease in air-fuel ratio with an increase in inlet air temperature.

2.16 COMBUSTION UNDER PREHEATED FUEL

Shepherd (1982) patented an invention related to apparatus associated with a fuel injection of a diesel engine. This apparatus preheat the fuel prior to injection into engine. Thus, the general object of his invention was to increase the fuel efficiency. The system and apparatus of his invention was based on storing, preheating and delivering preheated fuel to a conventional fuel injection diesel truck engine. A conventional fuel pump was arranged to draw the fuel through a conventional fuel filter from an insulated fuel preheat tank which in turn received the fuel by gravity feed both from a pair of vented fuel tanks as well as from an auxiliary overflow fuel tank arranged to store excess fuel pumped to the engine. The preheat tank was equipped with a series of coils inside the tank through which hot engine coolant was bypassed and returned to the engine cooling system. He found that the optimum fuel temperature should be in the range of 140^{o}F to 150^{o}F. If the temperature of the preheated fuel exceeded over 150^{o}F, he controlled the temperature by the conventional thermostatic coolant control of the diesel engine employed to power the truck/tractor unit. He also provided an additional temperature control which stoped flow of the hot coolant from the engine to the fuel preheat tank whenever the temperature of the fuel being supplied to the engine exceed 150^{o}F. At any time, if the temperature of the fuel being admitted to the engine droped below 140^{o}F, the invented apparatus restored flow of hot coolant to the fuel preheat tank. By this invented apparatus, he proved that the fuel economy of the diesel truck engine could be improved without sacrificing the efficiency of the engine.

Rowley & Gonzalez (2012) patented an invention related to pre-heating fuel such as gasoline, alcohol, kerosene, diesel or ethanol before pumping the super heated fuel into a fuel rail system on mechanical or electronic fuel injected internal combustion engines. They used free heat or scavenged heat from the coolant system and/or circulating oil from the crankcase of an internal combustion engine to preheat the fuel. They connected the fuel preheater device physically to part of the engine cooling water circulating system and/or hoses connecting circulating crankcase oil. They maintained about 125°F to 200°F of temperature in a fuel preheater chamber to superheat the fuel. In this temperature range most of the liquid fuels got superheated. Then, this superheated fuel was made to travel across a metal catalyst in the form of rods and/or a thin flexible multi-layer sandwich of different types of metal ribbons/wires which further broke down long hydrocarbon chains into shorter hydrocarbon chains. This galvanic reaction took place among the different metal types coming into contact with each other. This reaction acted as a catalyst and injected metallic ions into the fuel that passed over the metal rods/ribbons/wires. Then, the superheated fuel with the metal ions was made to pass to the injector pump which forced the partially vaporized fluid to the fuel rail and then to the individual fuel injectors which mixed with the incoming air from the intake manifold. They repeatedly tested the engine with this invented apparatus and found an increase of 10% to 20% in fuel economy and reduction of 20% in greenhouse gases emission in highway driving conditions.

Crowther & Crowther (2012) patented an invention related to a device for warming diesel fuel before it was injected into the engine. The fuel warming device comprised a cylindrical housing for heating the fuel. The housing had a first end, a second end, an inner surface, and an outer surface which altogether enclosed in an inner compartment. They installed the housing in a vehicle near the engine. The housing was constructed using copper. A copper coil tube had a first end and a second end. It spanned through the length of housing in the inner compartment. The first end of the copper coil tube extended outwardly through the first end of the housing and the second end of the copper coil tube extended outwardly through the second end of the housing. They allowed fuel to enter through first end and exit through second end. The fuel exiting the copper coil tube was subsequently injected into the engine of the vehicle. A heating fluid was disposed in the inner compartment of the housing. The heating fluid filled the inner compartment of the housing and surrounded the copper coil tube. The heating fluid helped to transfer heat quickly and evenly distributed the heat in the inner compartment of the housing. The heating fluid helped to hold heat in the housing. A heating element was disposed in the inner compartment of the housing. The heating element was for increasing the temperature of the heating fluid that surrounded the copper coil tube. When the heating element was activated, the temperature of the heating fluid got increased, which warmed the copper coil tube. Fuel that ran through the copper coil tube was then warmed by the copper coil tube. They provided the heating element of a 12-volt submersible heater. The heating element was operatively/electrically connected to an electrical connection component. They concluded that by using fuel warming systems to warm fuel before the fuel was injected into the engine; the efficiency of the combustion inside the engine got increased. They also reported that their fuel warming system could be used for both gasoline and diesel fuel.

2.17 COMBUSTION UNDER OXYGEN ENRICHED AIR

Wartinbee (1971) made an engine dynamometer study to determine the effects of oxygen enriched air on exhaust emissions. His results indicated that compared to operation with lean air-fuel mixtures, hydrocarbon emissions were reduced substantially, carbon monoxide emissions were similar, and oxides of nitrogen emissions increased significantly. He also found that octane requirements and fuel consumption were higher with oxygen enrichment. He concluded that these emissions and performance characteristics were due to the higher peak combustion temperatures associated with high oxygen concentrations.

Maxwell et al (1993) evaluated the effect of oxygen enriched air on engine performance and exhaust emissions of a single-cylinder, 4-stroke, spark-ignition engine. They evaluated with both gasoline and natural gas. They varied the oxygen content of the intake air between 20.9% (ambient air) and 25%. Their test results indicated that the use of oxygen enriched air produced a significant increase in power output, improved fuel conversion efficiency, lower specific fuel consumption, higher exhaust gas temperature and a significant reduction in carbon monoxide and hydrocarbon emissions when the engine was fueled with either gasoline or natural gas.

Ng et al (1993) tested a production spark ignition engine powered vehicle (3.1-L Chevrolet Lumina, model year 1990) with oxygen- enriched intake air containing 25% and 28% oxygen by volume to determine any difficulties in running the vehicle and to evaluate emissions benefits. They tested the engine with Standard Federal Test Procedure (FTP) emissions test cycles. Their results showed that the engine ran satisfactorily without vehicle performance anomalies. The emissions of carbon monoxide and hydrocarbons got reduced significantly in all three phases of the emissions test cycle, compared with normal air (21% oxygen). Carbon monoxide emissions from the engine with the three-way catalytic converter removed were significantly reduced in the cold-phase of the test cycle. The catalytic converter also had improved carbon monoxide conversion efficiency under the oxygen-enriched air conditions. The detailed results of hydrocarbon speciation indicated large reductions in 1,3-butadiene, formaldehyde, acetaldehyde, and benzene from the engine with the oxygen-enriched air. They also observed that the catalytic converter out ozone got reduced by 60% with the 25% oxygen-content air. But, they observed significant increase in NO_X emissions on all percentages of oxygen enrichment. They concluded that adding oxygen to the intake air was a better method than adding it to the fuel.

Ghojel et al (1983) studied the effect of the partial pressure of O_2 in the intake charge of an I.D.I. diesel engine on the various operating parameters and the exhaust emissions. They varied oxygen content in the intake between 21% and 40% by volume. They evaluated the engine performance and emissions at constant engine speed and injection timing. Their research revealed that enriching the intake air with oxygen led to a large decrease in ignition delay and reduced combustion noise. The fuel economy, the power output and the exhaust temperature remained almost constant. HC and CO emissions decreased and smoke levels dropped substantially, while NO_X emissions increased in proportion with the O_2 added.

2.18 SUMMARY OF THE LITERATURE REVIEW

From the above literature review, the following preliminary conclusions have been arrived at. The technical facts based on these conclusions have been integrated in the present experimental investigation. The important conclusions are:

- Hydrogen is one of the primary sources which can fulfill the future energy requirements.
- Hydrogen is the possible energy which can cause least harm to the environment.
- The stringent future emission norms could be met easily only with use of hydrogen.
- A modification of the fuel is the best way to reduce engine-out emissions without modifying the present configurations of the engines.
- Hydrogen can be used in the engines by way of inducting in the inlet manifold, inducting in the inlet port and also by direct injection into cylinders.
- While using hydrogen, care should be taken because of its nature of back firing.
- The literature clearly shows that hydrogen can be used in a diesel engine and in the gasoline engines without any modification to the present configurations of the engine.

- As a whole, using hydrogen as a fuel / additive in an engine, due to its fast flame velocity, high diffusion rate, low activation energy, low quenching distance resulted in an increase in brake thermal efficiency, decrease in all engine-out emissions except NO_X emission. This happens because when hydrogen is used, the premixed burning phase of combustion gets increased. This in-turn increases the in-cylinder pressure and temperature and also heat release rate.

- At low load conditions of the engine, the hydrogen is not an active participant in combustion process. This is because the self ignition temperature of hydrogen is more. It needs an ignition starter to start its combustion. At low load conditions, the temperatures are always low. This resulted in high hydrogen emission in the exhaust of the engine at low load conditions.

- Limited studies were found in hydrogen / oxygen enriched hydrogen gas. Mostly, they were based on varying the flow rates of the gas without modification to the operating parameters of the engine.

From the above literature review, it is evident that more work is needed to optimize the combustion of hydrogen / oxygen enriched hydrogen in diesel engines. The research problem for the present work was selected based on the above literature review. The following chapter briefly elucidates the problem its definition, its objectives, and scope of the present work.