STUDIES ON COMPRESSION IGNITION ENGINE TO ESTABLISH EFFECTS OF INJECTION PRESSURE, COMPRESSION RATIO AND NANO ADDITIVES – A REVIEW

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ABSTRACT

The review is focused to throw light on the effects of the fuel injection pressure, compression ratio and nano additives in the compression ignition (CI) engines on the performance, combustion and emission characteristics. However the CI engines have good reliability and performance characteristics, they emit hazardous emissions like unburned hydrocarbons, carbon dioxide, carbon monoxide, particulate matters, nitrogen oxides, etc. with the exhaust gases. Several studies have been carried out to improve the performance and combustion characteristics while reducing the level of pollutants in the exhaust gases and developed mainly three techniques. They are engine hardware modification, improvement in the fuel injection system and the exhaust gas treatment. Recently some studies were reported regarding the application of nanoparticles in the improvement of thermophysical properties of diesel fuel. The type and size of the nanoparticle added with fuel play a significant role in the engine/combustion characteristics. The review reveals that various techniques are available for improving the working characteristics, but not a single method which suits in all workingconditions.

Keywords: *CI engines, injection pressure, compression ratio, biodiesel, nanoparticles.*

INTRODUCTION

The importance of compression ignition (CI) engine is due to its higher thermal efficiency than efficiency of a diesel engine depends upon the conversion rate of the chemical energy of the fuel into heat release. The rate of heat release depends on the amount of fuel injected, ignition at an adequate time and the combustion process. The diesel engines emit hazardous emissions such as oxides of nitrogen (NO_x), particulate matter (PM), unburned hydrocarbons (UBHC), soot, smoke, carbon dioxide (CO_2spark ignition (SI) engines, effective performance, reliability and engine design. The), carbon monoxide (CO), etc. Various researches have been done to reduce the exhaust gas emissions, mainly by modifying the engine design, fuel modification, varying engine operating parameters like fuel injection pressure, injection timing, working speed, compression ratio, intake pressure etc. and exhaust gastreatment.

Fuel modification methods include adding some specific oxygenated chemical agents (ethanol, diethyl ether, dimethyl ether, 2-methoxy ethyl acetate, etc.) and nano additives (cerium oxide, aluminium, aniline nitrate, carbon nanotubes, etc.) to the diesel fuel to enhance the combustion, leading to reduced level of harmful pollutants in the exhaust gases. Technical advances since the introduction of direct injection diesel engines have steadily increased the fuel injection pressure. The main advantage of higher injection pressure is reduced particulate emission and the smoke number. In addition, it helps to achieve a better quality mixture in the combustionchamber.

EFFECT OF FUEL INJECTION PRESSURE

High injection pressure is an effective method to improve diesel engine performance and decrease particulate matter emission due to improved spray atomisation and fuel-air mixing, resulting in a more distributed vapour phase [1, 2]. Consequently it results in high local temperature, which affects NO_x emissions adversely, which can be reduced by combining increased injection pressure with exhaust gas recirculation [3]. The injected fuel particles get smaller depending on the increasing injection pressure and the NO_x formation is reduced by reducing the ignition delay [4].

Icingur and Altiparmak [5] experimentally studied the effects of injection pressure and cetane number on compression ignition engine performance and emissions. They conducted experiments in a four stroke, four cylinder DI diesel engine by varying the fuel injection pressure; from 100 – 250 bar and observed that the injected fuel particles get smaller depending on the increasing injection pressure, and NO_x formation was reduced by reducing the ignition delay. By varying the fuel injection pressure from 100 to 250 bar, the smoke level was reduced from 1.2 to 0.4 for diesel with cetane number of 54, whereas for diesel fuel with cetane number of 46, smoke level decreased from 2.05 to 0.6 respectively, which was because of the smaller drop size of the injected fuel with increase in injection pressure. Although the smoke level slightly increased with fuel cetane number, in the case of changing the injection pressure, the smoke level was affected by the pressure rather than cetane number. The smoke level was found to be very high when the injection pressure was 100 bar as at lower injection pressure, premixing in combustion process was in a bad condition because of dripping of fuelparticles.

Ismet Celikten [6] investigated the effect of the injection pressure with an electrical dynamometer assembled on 4-cylinder and 4-stroke indirect injection diesel engine. Variations for engine performance and emissions were studied at various throttle positions. In 50% throttle position, maximum CO emission obtained was 4000 ppm at 1500 rpm for 150 bar. When the injection pressure was reduced to 100 bar, the CO emission reduced to 1800 ppm. Increasing the injection pressure to 250 bar reduced CO emission to less than 100 ppm. As the engine speed reduced, the emission also decreased except at 150 bar. Over 2500 rpm, CO_2 emission for design injection pressure of 150 bar was lower than other pressures. Descending order was obtained at 100–200–250–150 bar with 8.5, 7, 6.5, 4 % respectively for 3500 rpm. With decrease in engine speed, CO_2 emission also decreased. NO_x emission was between 300 and 100 ppm at 150 bar. The emission descending order was 250–200–100–150 bar with 320. 300, 180, 40 ppm respectively at 3500 rpm. SO_2 emission was very high at 100 bar than the others for all engine rpm. SO_2 emission descending order was obtained at 100–150–200–250 bar with 300ppm for 100 bar and approximately 20-25 for other pressures at 3500rpm. Maximum SO_2 value was about 500 ppm for 100 bar at 1500 rpm. For high injection pressures such as 200–250 bar and low engine speeds, the emission was very low. In addition, the lower the engine speed, the lower was the SO_2 value. So, low injection pressures from design injection pressure were a disadvantage for SO_2. In 75% throttle position maximum CO emission obtained was 4000 ppm at 1500 rpm for 150 bar. CO emissionwasalsohighat100barwithavalueof1800ppm.Withanincreaseininjection pressure to 250 bar, the CO emission decreased to 1100 ppm. Higher pressures compared with standard injection pressure gave an advantage of reducing CO emission between 1500 and 2500 rpm. CO_2 was reduced when injection pressure was increased. The highest value was 14 % at 100 bar for 1500 rpm, which reduced to 10.8 % at 250 bar. NO_x emission was between 300 and 100 ppm at 100 bar, between 300 and 600 ppm at 250 bar. Minimum NO_x values were obtained at 100 bar. Descending order for NO_x was 250–150–200–100 bar with 600, 500, 450, 280 ppm respectively at 3500 rpm. SO_2 emission was very high at 100 bar than the others. Maximum SO_2 value of 700 ppm was obtained at 2500 rpm for 100 bar. With increase in the injection pressure, the SO_2 value decreased to less than 200 ppm. Forpressures except 100 bar, emission values were generally between 100 and 200 ppm. For 100 % throttle position, maximum CO emission was 4000 ppm obtained for 1500 rpm at 150 bar. Reducing the injection pressure to 100 bar slightly decreased the emission to 3900 ppm. Increasing the pressure to 200 bar decreased the emission to less than 100 ppm, the minimum CO value being almost nil in 3000 rpm at 250 bar. High value of CO_2 was 14% for 1500 rpm obtained at100bar.Whiletheemissionwasabout10%at250bar,thevaluewasbetween12and13 % at design injection pressure. In general, CO_2 reduced as injection pressure increased. NO_x emission was between 280 and 450 ppm at 100 bar, between 280 and 650 ppm at 250 bar. Minimum value was 280ppm for 1500 rpm obtained at 100 bar. NO_x emission reduced as injection pressure decreased. Maximum SO_2 value of 300 ppm was obtained at 100 bar and 4000–4500 rpm. SO_2 emission descending order was 150–200–250–100 bar . For pressures except 100 bar, emission values were generally between 100 and 250 ppm.

Ozer et al. [7] experimentally investigated the effect of injection pressure in a turbocharged indirect injection (IDI) diesel engine. Ethanol–Diesel emulsions (10 % ethanol-90 % diesel) were tested with 150, 200 and 250 bar fuel injection pressures. Ethanol had higher latent heats of vaporization than of diesel which caused low vaporization and mixing of fuel and air. Increasing the fuel injection pressure decreased the particle diameter and caused the diesel–ethanol fuel spray to vaporize quickly. But the liquid fuel cannot penetrate deeply into the combustion chamber and hence higher injection pressures initially generated faster combustion rates, resulting in higher temperatures. However, the initial combustion with the spray was restricted to a small region near the injector, and the flame spread around the chamber through slow propagation and caused an inefficient conversion process of heat to work. NO_x emission was about 300 ppm at 150 bar and 1500 rpm, which increased to a maximum of 700 ppm at 2500 rpm with an increase in injection pressure to 200 bar and with further increase in injection pressure to 250 bar, the emission decreased to a maximum of 500 ppm at 2500 rpm due to the effects of active combustion caused by smaller particles of atomized fuel. Thus higher NO_x emissions were observed with increase in injection pressure due to the effects of active combustion. An injection pressure of 150 bar at 1500 rpm gave a high CO value of 2600 ppm. Increasing the injection pressure to 200 and 250 bar caused fairly low CO (less than 100 ppm) in all engine speeds, especially between 1500 and 2500 rpm due to the good fuel–air mixing and easy and complete combustion of the smaller droplets.

Wang et al. [8] experimentally investigated the effects of ultra-high injection pressure and micro-hole nozzle on flame structure and soot formation of impinging diesel spray in a high-pressure, high-temperature constant volume combustion vessel at injection pressures of 100, 200 and 300 MPa. They ascertained that the highest injection pressure (300 MPa) gave better engine performance and decreased particulate matter emission due to improved spray atomization, fuel–air mixing and larger flame length. The flame length increased more rapidly and achieved quasi-steady length earlier for high injection pressure. The advance of quasi-steady stage for high injection pressure indicated an enhancement of soot oxidation. The wide dispersion of impinging spray was responsible for large flame length and flame area at high injection pressure. With microhole nozzle (0.08 mm), impinging spray flame showed lower soot formation at injection pressure of 100 MPa. At injection pressures of 200 and 300 MPa, the two-colour optical system did not detect soot images which were due to much weak soot formation under such conditions. For 0.16 mm nozzle, high injection pressure led to slightly lower soot luminosities. Even 300 MPa, an ultra-high injection pressure; the reduction of soot luminosities was not significant which might be due to the high injection rate. The soot reduction with the increase in injection pressure was mainly due to better spray atomization. Ignition delay for impinging spray flame was shortened with the increase of injection pressure because of improved spray atomization and enhanced fuel–air mixing rate. The study indicated that micro-hole nozzle and especially the combination of micro-hole nozzle and ultrahigh injection pressure provided an effective method to reduce soot formation in combustion chamber of dieselengine.

Pandian et al. [9] experimentally investigated the effect of injection system parameters on performance and emission characteristics of a twin cylinder compression ignition direct injection engine fuelled with pongamia biodiesel–diesel blend. Increasing the injection pressure from 150 bar to 225 bar increased brake thermal efficiency (BTE) from 21.82 % at 18° Before Top Dead Centre (BTDC) to 26.59 % at 27° before TDC respectively, and NO_x emissions from 316 ppm at 27° BTDC to 420 ppm at 30° BTDC respectively with reduction in CO from 0.98 % volume at 18° BTDC to 0.23% volume at 30° BTDC respectively, UBHC from 98 ppm at 18° BTDC to 52 ppm at 30° BTDC respectively and smoke opacity from92 % at 18° BTDC to 59 % at 30° BTDC respectively. With increase in injection pressure, better atomization of the fuel resulted in the smaller droplet size; faster evaporation of fuel sprays; and improved reaction between fuel and air which resulted in comparatively better combustion and contributed for higher BTE and NO_x emission with lesser CO, HC and smoke emissions at all injection timings. Beyond 225 bar of injection pressure, faster velocity of the fuel jets caused most fuel particles to hit the wall of combustion chamber where the fuel particles got cooled and not participated in the combustion process effectively which resulted in incomplete combustion. The experiment found that an injection pressure of 225 bar combined with advanced injection timing 27° before top dead centre (TDC) produced highest BTE with maximum NO_x emissions and lesser emissions of CO, HC and smoke emissions while the low injection pressure (150 bar) combined with the retarded injection timing (18°BTDC) resulted in the opposite trend to that of previousone.

Puhan et al. [10] experimentally found the effect of injection pressure on performance, emission and combustion characteristics of high linolenic linseed oil methyl ester (LOME) in a single cylinder, four stroke, constant speed, vertical, air cooled, and direct injection diesel engine with a rated output of 4.4 kW at 1500 rpm. At all loads and injection pressures; the fuel consumption was higher in the case of LOME compared to diesel which was due to higher density and lower heating value of LOME compared to diesel. The efficiency at full load was closer to diesel fuel because of improved atomization and better mixing process at higher injection pressures (Fig.1). At 240 bar injection pressure the efficiency was marginally higher than diesel due to the better combustion of LOME. CO_2 emissions for LOME (Fig.2) were comparatively higher than that of diesel fuel. As the heating value of LOME was low and had almost the same cetane number as diesel fuel and higher viscosity, the ignition delay was more for LOME which in turn produced more oxides of nitrogen (Fig.3). But injecting the fuel at higher pressures partly solved the problem by making the physical process to take place faster and hence the production of oxides of nitrogen reduced. The CO emissions (Fig.4) for LOME at all the injection pressures were lower than that of diesel fuel which was due to better combustion of LOME compared to diesel fuel. As the injection pressure was increased the fuel droplet travelled with a high velocity and that may hit the wall of the combustion chamber, which led to higher un-burnt hydrocarbon emissions(Fig.5).

The experimental investigation on the effect injection pressure in a direct injection diesel engine running on Jatropha methyl ester was done by Jindal et al. [11] in single cylinder, water cooled, four stroke, variable compression ratio (VCR) diesel engine. For 100% biodiesel, performance tests were carried out at three injection pressures (150, 200, 250 bar). The BSFC decreased with load significantly as the power output per unit fuel consumption increased at higher loads. It was found that the BSFC was always higher for B100 (100 % biodiesel) than B0 (100 % diesel) by about 25– 34% at standard rated engine parameters which was due to the fact that the esters of vegetable oils have lower heat value when compared to diesel and therefore more bio-diesel is needed to maintain the power output. During trials with three injection pressures, the effect of change in injection pressure on BSFC was found significant with lowest BSFC at 200 bar up to 50% load and 250 bar for higher loads. The decrease in BSFC can be attributed to the more efficient utilization of the fuel at higher injection pressures because of betteratomization.

EFFECT OF COMPRESSION RATIO

Air is compressed in the engine cylinder of the diesel engines and subsequently the pressurised diesel fuel is injected into the combustion chamber. Thus, mixture formation elapses in the engine‟s combustion chamber extremely quickly and ignition occurs without any external ignition source solely by transferring the heat from the compressed air to the fuel. Therefore, the diesel engine is an engine with „internal mixture formation‟ and „auto-ignition‟. The injection rate and the speed of mixture formation influence energy conversion in diesel engines. Since mixture formation is heterogeneous, the flame propagation typical in gasoline engines is absent and any danger of „knocking combustion‟ is eliminated.Therefore, high compression ratios can be produced in diesel engines, which benefit efficiency as well as an engine‟s torque characteristic. Since, as the compression ratio increases, the air density in the combustion chamber increases when ignition is applied, the temperature peaks of the fuel gas drop at the sources of ignition. Furthermore, the pressure curve and rate of heat release can be shaped more smoothly as the compression ratio increases. This reduces nitrogen oxide emission. Thus, a high compression ratio is also beneficial foremission performance. The compression work and thus the final compression temperature increase as the compression ratio increases [5].

Performance, emission and combustion characteristics of a variable compression ratio engine using methyl esters of waste cooking oil and diesel blends was experimented by Muralidharan and Vasudevan [12]. The performance and emission characteristics of variable compression ratio engine using various blends at compression ratios 18:1, 19:1, 20:1, 21:1 and 22:1 for 50% load and its comparison with the results of standard diesel fuel were done. The brake thermal efficiency of the blend B40 (40 % biodiesel + 60 % diesel) was slightly higher than that of the standard diesel at higher compression ratios, and was maximum for a compression ratio of 21, as for standard diesel and blend B40- 26.08 and 31.48 % respectively. The specific fuel consumption of B40 blend was lower than that of all other blends at compression ratio 20 and 21 which may be due to fuel density, viscosity and heating value of the fuels. B40 had higher energy content than B60 and B80, but lower than B20 and diesel. The specific fuel consumption of B40 blend at the compression ratio of 21 was 0.259 kg/kWh whereas for diesel it was 0.314 kg/kWh. At higher percentage of blends, the specific fuel consumption increased due to the decrease in calorific value for the higher blends. The blends B20, B40, B60 and B80 with standard diesel had a reduction in brake power. Brake power decreased at higher compression ratio due to the conversion from the chemical energy to mechanical energy. Due to the lower heating value of the blends and unstable combustion the brake power decreased. The maximum brake power obtained for B40 and diesel at a compression ratio 21 was 2.07 kW and 2.12 kW respectively. The mechanical efficiency of the blend B40 increased with the increase in compression ratio, when compared to that of standard diesel, the maximum obtained for compression ratio 21 being 52.53 %. The NO_x emission for diesel and other blends increased with the increase of compression ratio, the reason being higher peak temperature. For the compression ratio of 21, NO_x emission from the waste cooking oil blend B40 was higher than that of diesel, 621 ppm and 640 ppm respectively. The CO emission of the blend B40 was close to the standard diesel and it was found to be higher for compression ratio 21. The other blends B20, B60 and B80 had slightly lesser CO emission for compression ratio 21. The percentage of CO increased due to rising temperature in the combustion chamber, physical and chemical properties of the fuel, air–fuel ratio, shortage of oxygen at high speed, and lesser amount of time available for complete combustion. The blend emitted higher percentage of CO_2 than diesel at lower compression ratios and vice versa. More amount of CO_2 was an indication of complete combustion of fuel in the combustionchamber.

The experimental investigation on the effect of compression ratio and injection pressure in a direct injection diesel engine running on Jatropha methyl ester (JME) was done by Jindal et al. [11]. In the study, single cylinder, water cooled, four stroke, VCR (Variable Compression Ratio) diesel engine connected to eddy current type dynamometer for loading was run with JME (B100) at different compression ratios and injection pressures to evaluate the performance and emissions. For diesel the performance tests were conducted with 17.5 compression ratio and injection pressure of 210 bar was maintained at rated speed of 1500 rpm. For 100% biodiesel, performance tests were carried out at three injection pressures (150, 200,250bar)andforeachinjectionpressure,threedifferentcompressionratios(16,17and

18) were selected. The BSFC decreased with load significantly for both the fuels as the power output per unit fuel consumption increased at higher loads. It was found that the BSFC was always higher for B100 than B0 by about 25– 34% at standard rated engine parameters which was due to the fact that the esters of vegetable oils have lower heat value when compared to diesel and therefore more bio-diesel was needed to maintain the power output. It was observed that, as the compression ratio of the engine was increased; BSFC for B100 decreased (improved). Out of the three compression ratios, CR-18 gave lowest BSFC as, at higher compression ratio, brake power increased. During trials with different combinations of compression ratio and injection pressures, the synergic effect was seen with the combination of injecting fuel at 250 IP while maintaining the compression ratio as 18, where the BSFC was minimum for whole of the load range with an improvement of about 10 % over standard setting (17.5 CR/210 IP) of the engine at full load. At higher compression ratios, the engine efficiency improved. At a compression ratio of 18, the BTE improved by 5.5 % which can be attributed to better combustion and higher lubricity of bio-diesel. Similarly, improvement was found for higher injection pressure of 250 bar (2.3 %) because of better combustion due to finer breakup of fuel droplets providing more surface area and better mixing with air. On comparing all combinations of compression ratio and injection pressures with several sets of readings, it was found that at the combination of compression ratio 18 and injection pressure 250 bar, the engine delivered highest brake thermal efficiency (about 8.2 % higher than that of standard settings at full load). The unburned hydrocarbons, oxides of nitrogen (NO_x ppm), exhaust temperature and smoke opacity of the exhaust were found lower for B100 as compared to diesel at standard settings of the engine whereas carbon dioxide (CO_2 %) and carbon monoxide (CO %) emissions were higher. The HC emissions were lower by more than 50 %, NO_x by 25 %, exhaust temperature by 10 % and smoke opacity by 20 %. CO and CO_2 emissions are higher by 38 % and 2 % respectively. The exhaust gas temperature was also lower indicating lower temperature of combustion inside the cylinder, which caused lower levels of NO_x emissions. The reduction in HC and smoke emission was mainly due to the fact that bio-diesel had about 10–11 % oxygen contents which helped in better combustion of the fuel inside the cylinder. The higher levels of CO emission proved incomplete energy conversion which was also reflected in lower BTHE with bio-diesel. HC emissions tend to increase with increase in compression ratio and also on reduction in injection pressure. Lowest HC emissions (16 ppm) were found for compression ratio of 17 with injection pressure of 200 bar. At lower compression ratio, insufficient heat of compression delays ignition whereas at high compression ratio, dilution by residual gases hampers the combustion. The un-burnt HC emissions were higher for low injection pressure (150 bar) for all compression ratios since at lower injection pressures, atomization was poor and large droplets were formed leading to more unburned hydrocarbons in the exhaust. It was observed that with increase in compression ratio, CO emission and smoke opacity decreased whereas with increase in injection pressure for a given compression ratio, CO increased while smoke decreased. Higher levels of HC emission explain the reduction in CO with higher compression ratio. With higher injection pressure, increase in CO was due to poor diffusion flame combustion. At lower compression ratio, the temperature reached was also low and thusmoreCOwasexhaustedfromengine.TheminimumvalueofCOwasfoundas0.10%

at a compression ratio of 18 with injection pressure of 150 bar and minimum smoke opacity (20.5 %) was with IP-250 for CR-18. CO_2 emissions increase both with compression ratio and injection pressure indicating better combustion at higher compression ratio and higher injection pressures. The emission of NO_x was more sensitive to compression ratio at lower injection pressures as compared to higher compression ratios. With low injection pressure, increase in compression ratio facilitated the combustion of larger droplets because of high temperature of compression, whereas, high injection pressure converted the fuel into smaller droplets with good combustion suppressing the effect of compression pressure. The highest emissions were observed for high compression ratio and low injection pressure which was due to greater availability of fuel inside the cylinder (as reflected by increased specific fuel consumption at low injection pressure) and higher peak pressure leading to high temperatures. Minimum value of NO_x emission (165 ppm) was found for injection pressure of 250 bar with compression ratio of17.

Impact of compression ratio and injection parameters on a direct injection diesel engine fuelled with biodiesel-blended diesel fuel was experimentally studied by Sayin and Gumus [13]. The experiments were conducted on a four-stroke, naturally aspirated, single cylinder DI diesel engine. The original compression ratio (CR), injection pressure (IP) and injection timing (IT) of the engine were 18:1, 20 MPa and 20° BTDC, respectively. The increase in the biodiesel ratio in the fuel blend increased NO_x emissions by 3.23, 14.41, 30.04 and 38.46 % for B5 (5 % biodiesel + 95 % diesel), B20, B50 and B100, respectively, at reduced CR. The oxygen content of biodiesel increased the cylinder maximum gas temperature with better combustion and hence increased the NO_x concentration. The increased CR increased the NO_x emissions by 15.56 % and reduced CR decreased NO_x emissions by 12.45 % compared to results of original CR for B50. Reduced CR was to reduce the in-cylinder temperatures, and thus flame temperatures during the combustion to suppress NO_x emissions. The increase in biodiesel ratio in the fuel blend decreased the smoke opacity (OP) by 15.56 % and reduced CR increased the OP by 12.45 compared to the results of original CR for B50. As the CR increased, the maximum temperature during the combustion increased, which in turn, decreased OP. Increasing the IP decreased the OP. When the IP was increased, fuel particle diameters became smaller. Therefore, fuel air mixture became better through the combustion period, and so OP was less. The increase in biodiesel ratio in the fuel blends reduced HC emissions. At original CR, the HC decreased by 6.04, 49.81, 51.14 and 64.22 % for B5, B20, B50 and B100, respectively. HC concentration decreased with biodiesel addition, as adding oxygenated fuels can decrease HC. Oxygen enrichment was favourable to the oxidation of HCintheexpansionandexhaustprocesses.IncreasedCRreducedtheHCemissionsby4.39 % and reduced CR increased it by 35.50 % for B20 when compared with original CR. At lower CR, insufficient heat of compression delayed ignition and so HC emissions increased. The increased IP decreased the HC emissions by 0.30 % and the decreased IP raised the HC emissions by 2.73 % compared to original IP for B5. At lower IP, atomization was poor and largedropletsformedleadingtomoreHC.AtreducedIP,theCOdecreasedby11.23,13.96, 19.13 and 19.99 % for B5, B20, B50 and B100. CO concentration of the engine with the biodiesel-diesel blends decreased with the increase of the oxygen mass fraction in the blends duetothecombustionpromotionfromtheoxygenenrichment.IncreasedCRdecreasedthe

CO emissions by 37.09 % and reduced CR increased CO emissions by 9.67 % compared to results of ORG CR for B100. At lower CR, insufficient heat of compression delayed ignition and so CO emissions increased. The reason was that the increased CR increased the air temperature inside the cylinder thereby reduced the ignition lag and caused better and more complete burning of the fuel. For B20, the reduction in the emission of CO was 6.85 %, when the IP was increased from 18 to 22 MPa. The BSFC generally increased with the increase in biodiesel percentage in the fuel blend. The increments in BSFC are 1.94, 5.50, 8.27 and11.04 % for B5, B20, B50 and B100 at original IT, respectively, compared to the results of diesel fuel. It was because of the decrease in the lower heating value (LHV) of the blends by adding biodiesel, which required more fuel to be injected into the cylinder to get the same power output, leading to the increase in the BSFC. Increased CR decreased the BSFC by 1.78 %and 2.79 % for B50 compared with original and reduced CR, respectively, because with an increase in CR, the maximum cylinder pressure increased due to the fuel injected in hotter combustion chamber which led to higher effective power. Therefore, fuel consumption per output power decreased. The minimum BSFC values were obtained with the increased IPs because of improved atomization and better mixing process. The increased IP decreased BSFC values by 17.26 and 19.60 % compared to the original IP and decreased IP for B20. The BTE usually increased with the increase in biodiesel percentage in the fuel blend. Thus, the primary reason for the decrease in the BTE of biodiesels was the higher BSFC in spite of lower LHV of biodiesels. The change in BTE with respect to diesel fuel is 1.99, 3.69, 5.99 and 9.06 %, respectively, at reduced CR. The effects of the variation in CR on the BTE indicated that higher CRs improve the engine efficiency was due to better combustion and higher lubricity of biodiesel. The increased CR increased the BTE by 0.55 % and 1.33 % for B5 compared to the results of ORG and reduced CR, respectively. With respect to IP, change in the BTE was obtained at increased IP which was due to better combustion due to finer atomization offuel.

NANOTECHNOLOGY

Nanotechnology primarily deal with the synthesis, characterization and exploration of nanostructured materials. These materials are characterized by at least one dimension in the nanometer (1nm = 10^{-9} m) range [14]. Nanoparticle (powders) of ceramic materials has been produced in large scales by employing both physical and chemical methods. There has been considerable progress in the preparation of nanocrystals of metals, semiconductors, and magnetic materials by employing colloid chemical methods [15]. Since the discovery of carbon nanotubes, there has been considerable progress in the synthesis of multi- and single-walled nanotubes (MWNTs and SWNTs) and bundles of aligned nanotubes [16]. The size of nanoparticle overcomes the difficulties of settling, abrasion and clogging compared to micro-sized particles. Nano-materials possess unique mechanical, magnetic, optical, electric and thermal properties. Nanofluids can be described as a solid-liquid mixture which consists of nanoparticles and a base liquid [17] and they are materialised by suspending nanoparticles of average size of 100nm in conventional heat transfer fluids such as water, oil, ethylene gycol etc. The essential requirements of nanofluids, should possess uniform and stable suspension, negligible agglomeration of nanoparticles, passable durability, and no chemical change of nanoparticles in the base fluid. The improved heat transfer performance of nanofluids is due to the fact that the nanoparticles exhibit [18] properties like increase in surface area and heat transfer capacity of the fluid, improvisation in thermal conductivity, causing more collisions and interactions between the fluid particles and flow passages, causing more turbulence and mixing of thefuel.

EFFECT OF NANO ADDITIVES

Ambrozik and Chlopek [19] listed some basic requirements for catalytic fuel additives to the conventional fuel:

1. The catalyst added to the fuel should undergo decrement in particulates emission in the enginecylinder.
2. The chemical stability of the catalytic additive should be stable over long time with thefuel.
3. The use of catalytic additives in the fuel should not enhance the hazardousemissions.

The catalyst is characterised by its size and the reduced size of catalyst creates more active surfaces where the chemical reaction takes place and thus leads to improved reaction efficiency. Nanoparticles are the best choice as catalyst as they possess potential thermo-physical properties such as enhanced surface-area to volume ratio, enhanced radiative/mass transfer properties and enhanced ignition temperature characteristics. Yetter et al. [20] found that the nano catalyst presence in hydrocarbon fuels may lead to lower ignition temperature, shorter ignition delay and conversion of carbon into carbon dioxide. The experiment by Kulkarni et al. [21] on an inline-four cylinder diesel engine with turbocharger and liquid cooling concluded that by employing nanofluids in water coolant jacket, the engine will heat up faster and may result in the reduction of emission, since the higher concentration pollutants are emitted during the engine warm up. As the nanoparticles concentration was increased, the magnitude of specific heat was decreased, as the nanopaticles require less time to heat up during startup.

Efforts have been made to add nanoparticles to the conventional diesel fuel as well as biodiesels. The effect of achieving the desired properties for the fuels is the major application of nanofluid technology [22]. Fuels of bio-origin provide a feasible solution to the twin crisis of„fossilfueldepletion‟and„environmentaldegradation‟.Theburningoforganicallyderived fuels does not contribute any additional CO_2 into the atmosphere, as the carbon released is the same as the carbon absorbed by the plants as they grow. Using organic fuels is therefore beneficial to both the environment and the atmosphere [23]. Problems with bio-diesel are that they have lower calorific value, higher flash point, higher viscosity, poor cold flow properties, poor oxidative stability and comparatively higher emission of nitrogen oxides [24]. Sajith et al. [25] studied on the effects of cerium oxide (CeO_2) nano-particle (size: 10 – 20 nm) blended jatropha biodiesel by conducting extensive performance tests in a single cylinder compression ignition engine (naturally aspirated, four stroke, single cylinder, water-cooled compression ignition). The dosing level of the cerium oxide nanoparticle samples (by weight) in the base fuel was varied from 20 to 80 ppm. As an effect of added nano particles, thebiodieselshowedanincreasingtrendfortheflashpoint.Inaddition,theyobservedthe volatility of the fuel increases with the quantity of the fuel additive. The cerium oxide nanoparticles present in the fuel promote longer and more complete combustion, compared to the base fuel as cerium oxide acts as an oxygen buffer and thus increases the efficiency. A maximum increase of 1.5% in the brake thermal efficiency was obtained when the dosing level was varied from 20 to 80 ppm, with a maximum improvement observed at a dosing level of 80 ppm. Cerium oxide oxidizes the carbon deposits from the engine leading to efficient operation and reduced fuel consumption. Hydrocarbon emission was found to be significantly reduced on the addition of the additive; for 80ppm CeO_2, 20ppm of HC at no load to 100ppm at 5kW load and 70ppm to 180ppm for biodiesel without CeO_2. Cerium oxide supplies the oxygen for the reduction of the hydrocarbon as well as the soot and gets converted to cerous oxide (Ce_2O_3). An average reduction of 25% to 40% in the hydrocarbon emissions was obtained for additive dosing levels ranging from 40 to 80 ppm of the additive. The NO_x emission was found to be generally reduced on the addition of cerium oxide nanoparticles to bio diesel, where an average reduction of around 30% was found to occur with a dosing level of 80 ppmnanoparticles.

Sadhik and Anand [26] did experimental investigation using carbon nanotubes blended water–diesel emulsion fuel to establish the performance, emission, and combustion characteristics of diesel engine using carbon nanotubes (CNT) blended water–diesel emulsion fuels in a single-cylinder diesel engine coupled with an electrical loading device. The potential benefit of water–diesel emulsion fuel was the combined effect of „micro-explosion‟ and „secondary atomization‟ phenomena, which break the large fuel droplets into smaller ones due to the volatility differences between the diesel and the water. The microexplosions in the water–diesel emulsion fuel are the result of instantaneous vaporization of water droplets present in the diesel fuel. Once the water droplets encapsulated in the diesel fuel are exposed to high pressure and high temperature in the engine cylinder, the water present in the diesel fuel will absorb heat quickly (since the boiling point of water is lower than that of diesel) and explodes through the surrounding oil layers. This phenomenon is known as micro-explosion. Subsequently, the secondary atomization phenomenon follows immediately and provides a number of secondary fuel droplets of fine size, which evaporate very quickly. The reduced ignition delay characteristics for CNT blended water–diesel emulsion fuels could possibly be attributed to better atomization, improved ignition properties and cetane number, and enhanced surface area-to-volume ratio of CNT during the combustion in the engine cylinder. Furthermore, the addition of CNT to the water–diesel emulsion fuel enhanced the cetane number of the fuel. The fuel with the higher cetane number shortens ignition delay in a diesel engine. Hence, the addition of CNT to the water–diesel emulsion fuel has produced shortened ignition delay compared to that of water–diesel emulsion fuel at all the loads. The brake thermal efficiency of the CNT blended water–diesel emulsion fuels was improved particularly at the higher loads could be due to the better combustion characteristics of CNT along with micro-explosion and secondary atomization. Subsequently, the secondary atomization of encapsulated CNT water droplets takes place with a large number of fine droplets and reacts with diesel fuel and air effectively (owing to the enhanced surface-area-to-volume ratio of CNT). In case of D2S5W50CNT fuel, the catalytic activity may be enhanceddduetothehighdosageofCNTcomparedtothatofD2S5W25CNT.Hencethe brake thermal efficiency was higher for D2S5W50CNT compared to that of D2S5W25CNT. The CNT blended water–diesel emulsion fuels produced lower NO_x emission compared to that of water–diesel emulsion fuel due to lower combustion temperature (950ppm for D2S5W50CNT at 5.3 bmep and 1050ppm for D2S5W) which was because of reduced ignition delay and the heat absorption of the vaporization of the water during the combustion in the engine cylinder. The magnitude of CO emission for the CNT blended water–diesel emulsion fuels was marginally lower compared to that of neat diesel and water–diesel emulsion fuel (0.17, 0.18 and 0.185 % vol. respectively at 5.3 bmep) which could be due to the occurrence of secondary atomization phenomenon of the encapsulated CNT water droplets in the combustion chamber. The unburnt HC emissions were also marginally lower for the CNT blended water–diesel emulsion fuels than the water– diesel emulsion fuel (75, 82 ppm respectively at 5.3 bmep) due to catalytic activity and improved combustion characteristics of CNT, which led to improvedcombustion.

Kao et al. [27] studied about aqueous aluminum nanofluid combustion in diesel fuel by conducting experiment in a single cylinder diesel engine, with aqueous alumina blended with diesel. At greater engine speeds the combustion with diesel fuel was more efficient than with aqueous alumina blended with diesel fuel because of the too large moisture content in aqueous alumina blended with diesel fuel, which caused the decomposing energy to become insufficient. The reduced smoke concentrations at 1200 and 1800 rpm (18 and 19% respectively) indicated that the combustion with aqueous alumina blended diesel was more complete at those speeds than with diesel fuel. At 2400 rpm the smoke concentration showed no significant difference between aqueous alumina blended diesel fuel and diesel fuel. The mean smoke concentration of aqueous alumina blended diesel fuel was lower than that from diesel fuel in all loads for engine speeds less than 2200 rpm as adding aqueous aluminum nanofluid in diesel fuel promoted more complete combustion and the aluminum nanopowder additive contained moisture that produced a microexplosion on the emulsified fuel drains and hence reduced the smoke concentration. NO_x concentration for aqueous alumina blended diesel fuel was lower than that of diesel fuel at all loads at 1200 and 1800 rpm (365ppm and 410ppm respectively), because the nanopowder additive contains moisture causing micro-explosions in the emulsified fuel drains. At 1200 rpm at maximum load hydrogen burning resulted in the more complete combustion and hence the NO_x concentration caused by aqueous alumina blended diesel fuel was less than that caused by diesel fuel. The latent heat of evaporation of water and its high thermal capacity also reduced the temperatures in the combustion chamber, thus retarding nitrogen oxidebuild-up.

Sadhik and Anand [28] found the role of nanoadditive blended jatropha biodiesel emulsion (JBD) fuel on the working characteristics of a diesel engine by conducting experiment in a single cylinder, four stroke, air cooled, direct injection engine. For nanoparticle blended biodiesel emulsion fuels, the magnitude of ignition delay were reduced considerably, owing to the quick evaporation rate, improved ignition properties, and enhanced surface area/volume ratio characteristics. During the ignition delay period, the presence of nanoparticles in the fuel-air mixture improved (as a result of enhanced surface area/volume ratio) due to early combustion. The evaporation rate was reduced for the nanoparticle blended emulsion fuels than that of JBDS15W fuel at all hot-plate surface temperatures, dueto encased water droplet present in biodiesel emulsion fuel, which had enhanced the heat transfer rate during the evaporation. At all loads the brake thermal efficiency of the nanoparticle blended emulsion fuels was high compared to that of JBDS15W and JBD fuels, which could be due to the combined effect of microexplosion and secondary atomization along with the presence of potential nanoparticles. Once the alumina nanoparticles encased water droplet biodiesel emulsion fuel subjected to high pressure and high temperature environment in the engine cylinder, the water droplets will absorb the heat quickly, which led to the explosion of water droplets through the surrounding oil layers called microexplosion. Subsequently, a number of secondary fuel droplets of very fine size were produced, which also evaporates quickly. So, the formation of secondary droplets in the combustion chamber enhanced the fuel-air mixing in the presence of potential nanoparticles. The nanoparticles, possess enhanced surface area/volume ratio and reactive surfaces which led to higher chemical reactivity to act as a catalyst. Hence the brake thermal efficiency of nanoparticle blended biodiesel emulsion fuels was improved, owing to the catalytic activity of alumina nanoparticles. The brake thermal efficiency of JBDS15W fuel is low due to the effect of water addition, longer ignition delay, and reduced engine temperature. The brake specific fuel consumption was improved for the nanoparticle blended biodiesel emulsion fuels compared to that of JBDS15W and JBD which could be due to the existence of alumina nanoparticles in the encapsulated water droplets present in the biodiesel emulsion fuel. Maximum efficiencies were25,26,28,28.5and29%forJBD,JBDS15W,JBDS15W25A,JBDS15W50A,and JBDS15W100A respectively. Owing to the enhanced surface area/volume ratio and shorten ignition delay characteristics of nanoparticles, sufficient fuel could have accrued in the combustion chamber to undergo a possible catalytic effect in the unit volume of the fuel during the combustion. That effect could have led to improvement in the specific fuel consumption for the nanoparticle blended biodiesel emulsion fuels compared to that of biodiesel emulsion fuel. The formation of NO_x emission in diesel engines was due to the high temperature environment and oxygen availability during the combustion. There was a reduction of NO_x emissions at all brake mean effective pressure (BMEP) for the nanoparticle blended biodiesel emulsion fuels compared to that of the JBDS15W fuel due to reduced ignition delay, early combustion, and ample fuel accumulation in the engine cylinder; maximum reduction being from 1200 to 880 ppm at 0.53 MPa of BMEP for 100 ppm amount of nanoparticle. There was a marginal decrement in CO emissions (Fig.6) for the nanoparticle biodiesel emulsion fuels compared to that of JBDS15W which could be due to the enhanced surface area/volume ratio and high catalytic activity of alumina nanoparticles present in the biodiesel emulsion fuel, which led to shortened ignition delay. Due to that, there was an enhancement in the fuel-air mixing in the combustion chamber, leading to complete combustion. For nanoparticle blended biodiesel emulsion fuels, the magnitude of unburnt HC emissions (Fig.6) was reduced due to hydrocarbon oxidation and secondary atomization effects. Also, the degree of fuel-air mixing in the presence of nanoparticles could have improved, on the account of secondary atomization and improved catalytic effect associated with the nanoparticle blended biodiesel emulsion fuels. The addition of nanoparticles to the biodiesel emulsion fuel imparted shorter ignition delay, enhanced evaporation, and improved ignitioncharacteristics.Duetotheaboveeffectsassociatedwiththenanoparticleblended biodiesel emulsion fuels, an ample fuel could have been induced in the combustion chamber prior to the ignition, leading to better combustion, air-fuel mixing, resulting in reduced smoke emissions.

SUMMARY

Nanoparticles are added to the diesel fuel as they possess potential thermo-physical properties such as enhanced surface-area to volume ratio, enhanced radiative/mass transfer properties and enhanced ignition temperature characteristics. These properties help in better combustion of diesel fuel and thus improvement in performance as well as in emission characteristics. Research on the optimum usage of nanoparticles in diesel/biodiesel fuels are still going on and are mostly in the developing stage. Also, the adding nanoparticles should not cause a new mode of air pollution. Varying the injection pressure and/or compression ratio has also led to improvement in efficiency and emission characteristics of diesel engines. The right combination of injection pressure and compression ratio varies with the fuel properties. Now-a-days, the importance of biodiesel is increasing day by day and hence addition of proper nanoparticle (as catalyst) and adjustment of the compression ratio and injection pressure will give the best results in case of performance and emissioncharacteristics.

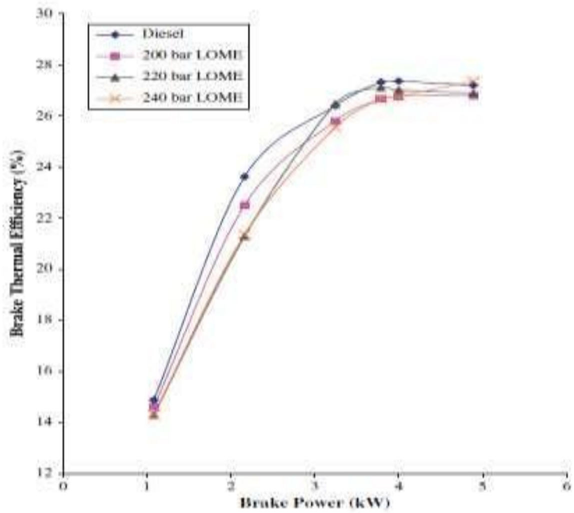

Figure 1: Variation of brake thermal efficiency with brake power (ref. [10]).

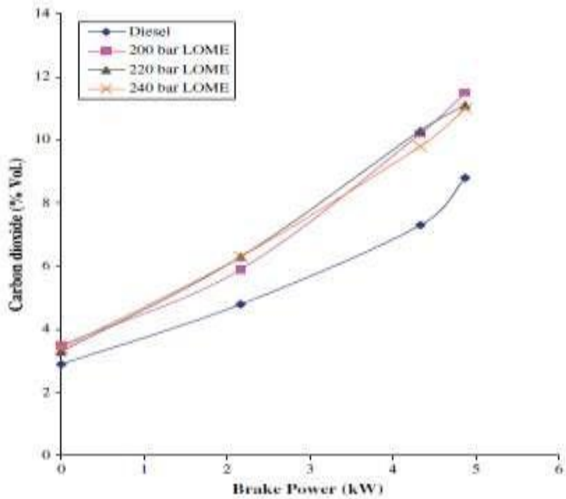

Figure 2: Variation of carbon dioxide with brake power (ref. [10]).

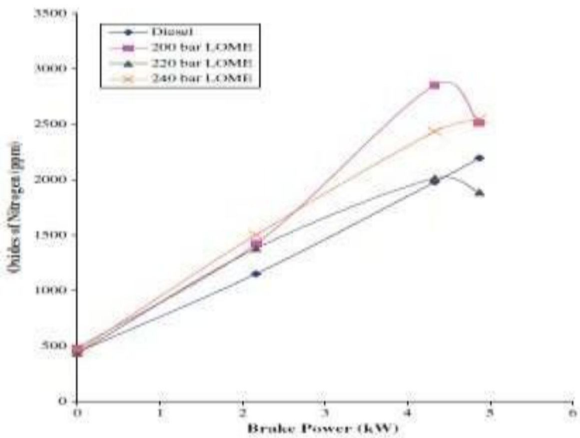

Figure 3: Variation of oxides of nitrogen with brake power (ref. [10]).

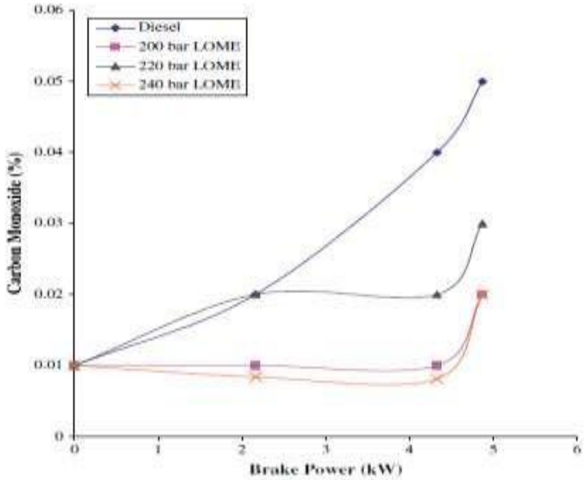

Figure 4: Variation of carbon monoxide with brake power. (ref. [10]).

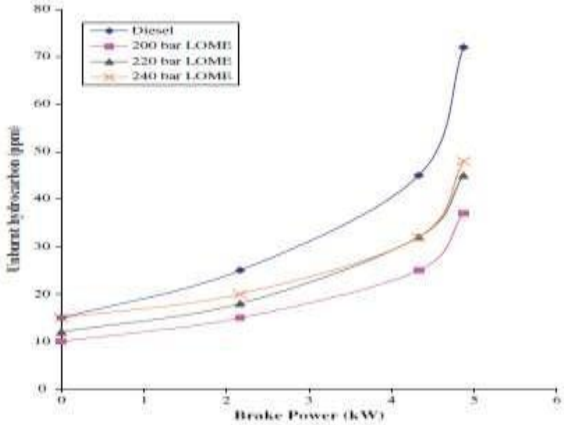

Figure 5: Variation of UBHC with brake power. (ref. [10]).

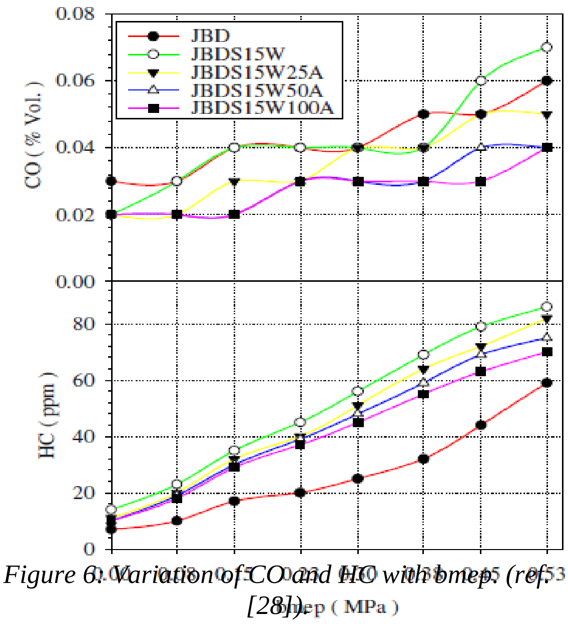

Figure 6: Variation of CO and HC with bmep. (ref. [28]).

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