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Investigation of Performance and Emission Characteristics using Low Reactivity Fuel and Biodiesel Blended Nanoparticles in Diesel Engine

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***Abstract*— Alternate fuels and introduction of new combustion technology has been the major topic in these days due to its importance in environmental issue and cost. This work investigates the effect of low reactivity and high reactivity fuels in diesel engine to reduce the emission and to improve the efficiency. The experiments are performed on single cylinder research diesel engine by adapting dual fuel operation using gasoline as a low reactivity fuel which is injected near the port and various diesel blends injected directly into the cylinder. Here low reactivity fuel is kept same for all investigations but high reactivity fuel blends are changed. All tests were carried out at 1500rpm 23 CAD BTDC (DI) on various loading conditions. The port fuel injection takes place at 20 CAD ATDC (during intake stroke) which is controlled by ECU. Bio diesel used for direct injection (blended with diesel) is Annona squamosa (custard apple) seed oil. Various blending ratios used for DI: B20-80, B20A25, B20A50. Results suggest that in-cylinder fuel gradients strongly affect the engine efficiency. In addition NOx and smoke emission are reduced drastically while using proper blends with slight increase in HC and CO but using nanoparticles it get reduced further.**

Keywords*— Diesel, Gasoline, Aluminium oxide nanoparticles, Annona squamosa oil, Combustion, Performance and Emission.

1. INTRODUCTION

The increasing concern about the environment has been one of the major forces behind ongoing research to obtain cleaner sources of energy and to optimize the use of existing technologies. Internal combustion engines (ICE) have been around for more than one century and have been identified as one major source of air pollution. The stringent regulations introduced around the world to limit the pollutant emissions of internal combustion engines (ICE) present a major challenge for the engine research community. In spite of its efficiency, conventional mixing-controlled diesel combustion in CI diesel engines requires complex and costly exhaust after treatment systems to reach the NOx and soot limitation values proposed in the current regulations, such as EURO VI. Additionally to the complexity of the after treatment systems, the use of DPF (to reduce soot emissions) and LNT or SCR (to minimize NOx emissions) requires a periodically regeneration (opting rich) or the introduction of a reducing agent, which penalizes the fuel consumption. Thus, in order to reduce after treatment costs and fuel consumption it is necessary to avoid the generation of these pollutants in the focus of the emission, i.e. during the combustion development. D. Ganesh, G. Nagarajan [1] in their study a vaporized diesel fuel was mixed with air to form a homogeneous mixture and inducted into the cylinder during the intake stroke. To control the early ignition of diesel vapour-air mixture, cooled (30 C) Exhaust Gas Recirculation (EGR) technique was adopted. Experiments were conducted with diesel vapour induction without EGR and diesel vapour induction with 10%, 20% and 30% EGR and results are compared with conventional diesel fuel operation (DI @ 23 before Top Dead Centre (bTDC) and 200 bar injection pressure). The investigation of diesel HCCI showed the potential of this type of combustion. HCCI is a promising concept for achieving low emissions at part load operations. It offers a solution for what is considered as a major drawback of diesel engines. This technique can be successfully applied to traditional direct injection diesel engines with low extra costs and no modification to the DI system by performing the mixture formation in the intake manifold. However, high HC and CO emissions are a major disadvantage. Even though it may be possible to reduce these emissions in the exhaust system with an oxidation catalyst, they also reduce the fuel efficiency considerably. The investigation has shown that a diesel engine can run on a homogeneous fuel/air mixture that is generated externally in a fuel vaporizer. The engine combustion with a homogenized mixture via fuel vaporizer is demonstrated. The aim of low NOx and smoke emission was achieved with fuel consumption about 12% higher compared to conventional diesel operation when operating engine with 30% EGR. The main reasons for the fuel consumption penalty are the unburned fuel and the vaporizer loss. In order to delay and control the combustion, EGR was used in this investigation up to 30%. With EGR, combustion was controlled and delayed. The combustion occurs close to TDC rather than well before TDC due to EGR on combustion pressure and temperature. A further increase in the EGR for retarding the combustion would lead to higher HC/CO emission and to combustion instability. Thus the simultaneous reduction of NOx and Smoke emission with fuel vaporizer system was achieved in the limited engine operation area. Xingcai Lu et al [2] this work investigated the effects of an in-cylinder active thermo-atmosphere environment (ATAE) on the diesel engine combustion and emissions in a single-cylinder diesel engine. Port-fuel injection of n-heptane was used to prepare the lean fuel/air mixture, and active radicals and heat release were found during the low-temperature reaction (LTR) and high-temperature reaction (HTR). The premixed ratio of n-heptane was used to denote the ATAE intensity. The effects of ATAE intensity and fuel delivery angle of directly injected diesel fuel on the combustion and emissions were evaluated. The experimental results reveal that, as the premixed ratio of n-heptane increases, both the maximum values of the heat release rate (HRR) in the LTR and HTR increase, which means that the ATAE intensity increases. With the increase of the ATAE intensity, the peak value of the diffusion combustion decreases and NOx emissions and smoke opacity reduce substantially at first and attain to the lowest levels at a certain point. Once the ATAE intensity exceeds this critical value, NOx emissions begin to increase monotonously but the smoke opacity increases to a peak point and then begins to further decrease. Overall, under those operating conditions, the optimized ATAE intensity is about 20−30%. Mustafa et al [3] As an alternative combustion mode, the HCCI combustion has some benefits compared to conventional SI and CI engines, such as low NOx emission and high thermal efficiency. However, this combustion mode can produce higher UHC and CO emissions than those of conventional engines. In the naturally aspirated HCCI engines, the low engine output power limits its use in the current engine technologies. The increase in CO emissions associated with retarded SOI timing which caused fuel stratification and less homogeneity of the mixture. Despite the increase in CO emission with retarded SOI timing the overall combustion efficiency was improved as a result of the reductions in UHC emissions. As the CO emissions increased with the engine speed for all three boost pressure as a function of SOI timing, the NOx emissions did not show similar trend. M. M. Rahman et al [4] incorporates the port injection fuel delivery system (PFI) injects hydrogen directly into the intake manifold at each intake port rather than drawing fuel in at a central point. Typically, Hydrogen is injected into the manifold after the beginning of the intake stroke. Hydrogen can be introduced in the intake manifold either by continuous or timed injection. The former method produces undesirable combustion problems, less flexible and controllable. But the latter method, timed port fuel injection (PFI) is a strong candidate and extensive studies indicated the ability of its adoption. The calling sounds for adopting this technique are supported by a considerable set of advantages. It can be easily installed only with simple modification and its cost is low .The flow rate of hydrogen supplied can also be controlled conveniently. External mixture formation by means of port fuel injection also has been demonstrated to result in higher engine efficiencies, extended lean operation, lower cyclic variation and lower NOx production. This is the consequence of the higher mixture homogeneity due to longer mixing times for PFI. Hanson et al [5] A study of partially premixed combustion (PPC) with non-oxygenated 91 pump octane number (PON) commercially available gasoline was performed using a heavy-duty (HD) compression-ignition (CI) 2.44-l Caterpillar 3401E single-cylinder oil test engine (SCOTE). The experimental conditions selected were a net-indicated mean effective pressure (IMEP) of 11.5 bar, an engine speed of 1300 rev/min, an intake temperature of 40°C with intake and exhaust pressures of 200 and 207 kPa, respectively. The baseline case for all studies presented had 0% exhaust gas recirculation (EGR), used a dual-injection strategy a -137 deg ATDC pilot SOI and a -6 deg ATDC main start-of-injection (SOI) timing with a 30/70% pilot/main fuel split for a total of 5.3 kg/h fueling (equating to approximately 50% load). Combustion and emissions characteristics were explored relative to the baseline case by sweeping main and pilot SOI timings, injection split fuel percentage, intake pressure, load and EGR levels. The results from these tests produced low engine-out NOx and PM emissions. Interestingly, with EGR rates over 20%, both NOx and PM were simultaneously reduced while maintaining or even lowering indicated specific fuel consumption (ISFC). The results presented in this paper demonstrate promising in-cylinder emissions reductions from the use of gasoline in HD CI engines. Splitter et al [6], demonstrated as a promising method to achieve high efficiency – clean combustion. Engine experiments were performed in a heavy-duty test engine over a range of loads. Detailed computational fluid dynamics modeling was then used to explain the experimentally observed trends. Specifically, it was found that RCCI combustion is capable of operating over a wide range of engine loads with near zero levels of NO*x* and soot, acceptable pressure rise rate and ringing intensity, and very high indicated efficiency. The comparison between RCCI and conventional diesel showed a reduction in NO*x* by three orders of magnitude, a reduction in soot by a factor of six, and an increase in gross indicated efficiency of 16.4 per cent (i.e. 7.9 per cent more of the fuel energy was converted to useful work). The simulation results showed that the improvement in fuel conversion efficiency was due both to reductions in heat transfer losses and improved control over the start- and end-of-combustion. Shuaiying et al [7], investigates the effects of diesel injection strategies on combustion, emissions, fuel economy and the operation range with high efficiency and low emissions fueled with gasoline/diesel dual fuel on a modified single-cylinder diesel engine. This gasoline/diesel dual- fuel combustion mode proposes port fuel injection of gasoline and direct injection of diesel fuel with rapid in-cylinder fuel blending. Single and double injection strategies were employed in the engine experiments at 1500 rev/min and 50 mg/cycle total equivalent diesel fuelling rate. The experimental results showed that this combustion mode had the capability of achieving high efficiency with near zero NOx and soot emissions by using an early injection timing of single (E-single) strategy with high gasoline ratio. Parameters were optimized in double injection strategy included the first and second injection timing, injected diesel mass split between the two injections and the premixed gasoline ratio. Compared to other three injection strategies, the early second injection timing (E-SOI2) strategy achieved the lowest indicated specific fuel consumption (ISFC) of 173 g/kW h, the NOx and soot emissions were below 0.2, 0.003 g/kW h respectively, but the maximum pressure rise rate (MPRR) was penalized, while the late second injection timing (L-SOI2) strategy was most favorable at reducing MPRR because of prolonged combustion duration. E-single and E-SOI2 strategies were difficult at high loads due to the high MPRR. Therefore, L-SOI2 strategy is an effective approach to expand the operation range to higher load. However, the upper load was limited by high soot emissions, which demanded increasing the diesel injection pressure. Under the operating conditions with optimized parameters, the maximum indicated mean effective pressure (IMEP) can be expanded up to 1.391 MPa by using L-SOI2 strategy with increased fuel mass while still maintaining good emissions and MPRR within a given criteria. Jesus Benajes et al [8], investigated the effect of low reactivity fuel characteristics and blending ratio on low load RCCI, performance and emission using four different low reactivity fuels : E10-95, E10-98, E20-95 and E85 (Port Fuel Injected) while keeping constant the same high reactivity fuel: diesel B7 (Direct Injected). The result of their experiment showed that in-cylinder fuel reactivity strongly affects the engine efficiency at low load. Specifically a reduced reactivity gradient allows an improvement of 4.5% in terms of gross indicated efficiency when the proper blending ratio is used. In addition, EURO VI NOx and soot emission levels are fulfilled with a strong reduction in CO and HC compared with the case of high reactivity gradient among the low and high reactivity fuel. Jesus Benajes et al [9], investigates the effect of the direct injection timing and blending ratio on RCCI performance and engine emissions out at different engine loads using four low reactivity fuels : E10-95, E10-98, E20-95 and E85 (Port Fuel Injected) while keeping constant the same high reactivity fuel: diesel B7 (Direct Injected). Results suggest that notable higher diesel amount is required to achieve a stable combustion using E85. This fact leads to higher NOx levels and unacceptable ringing intensity. By contrast, EURO VI NOx and soot levels are fulfilled with: E10-95, E10-98, and E20-95. Finally, the higher reactivity of E10-95 results in a significant reduction in CO and HC emissions, mainly at low load. Ming Zheng et al [10], investigated the butanol-fuelled HCCI combustion on a high efficiency diesel engine .

This work, n-butanol that has a low reactivity and high volatility is studied for HCCI combustion on single cylinder high CR (18.2:1) diesel engine without any modification in air path system. The results indicate that n-butanol HCCI combustion offers the benefit of ultra-low NOx and smoke emission with minimal requirements for intake dilution through EGR. Also the low reactivity helps in realizing an optimal combustion phasing, and thermal efficiency levels comparable to that of conventional diesel combustion are consistently achieved.

2. MATERIALS AND METHODS

2.1. Preparation of biodiesel

Transesterification is basically a chronological reaction where triglycerides are first reduced to diglycerides and to monoglycerides. The monoglycerides are finally reduced to fatty acid esters. Equipments used for transesterification reaction are magnetic stirrer, thermometer, and beaker. Raw materials are custard apple seed oil (Annona squamosa), methanol, and potassium hydroxide. Custard apple seed oil was measured to a capacity of 1000 ml and filled into the first beaker and stirred at 1000 rpm. The oil was warmed up to 60°C and 5g of potassium hydroxide was dissolved in 250ml of methanol followed by forceful stirring. This catalyst/alcohol mixture was added to the custard apple seed oil and stirred vigorously at 1000 rpm for 1 hour at 60°C. Crude glycerine, the heavier liquid, was separated at the bottom and methyl ester on the top. After completion, water at 80°C was added to double volume of methyl ester, and then stirred for 15 min. The glycerine was allowed to settle again. The process was repeated until the ester layer becomes clear.

2.2. Preparation of fuel blend

For the blending of aluminium oxide nanoparticles in Annona squamosa oil, a sample (25mg/l and 50mg/l) of nanoparticle is added and uniform dispersion done using sonicator.

2.2.1. Properties of biodiesel blend samples

The properties of biodiesel were determined by standard methods like bomb calorimeter, redwood viscometer, open cup apparatus and shown in Table 1.

Table 1 Properties of fuel

Properties	Diesel	B20	B20A25	B20A50
Flash point (°C)	42	54	52	50
Fire point (°C)	50	59	58	55
Kinematic viscosity @40 °C (x10^{-6} m^2/s)	0.116	0.193	0.241	0.376
Calorific value (kJ/kg)	43500	41402	41791	42031
Density (kg/m^3)	830	840	840	841

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3. EXPERIMENTAL SETUP AND TEST PROCEDURE

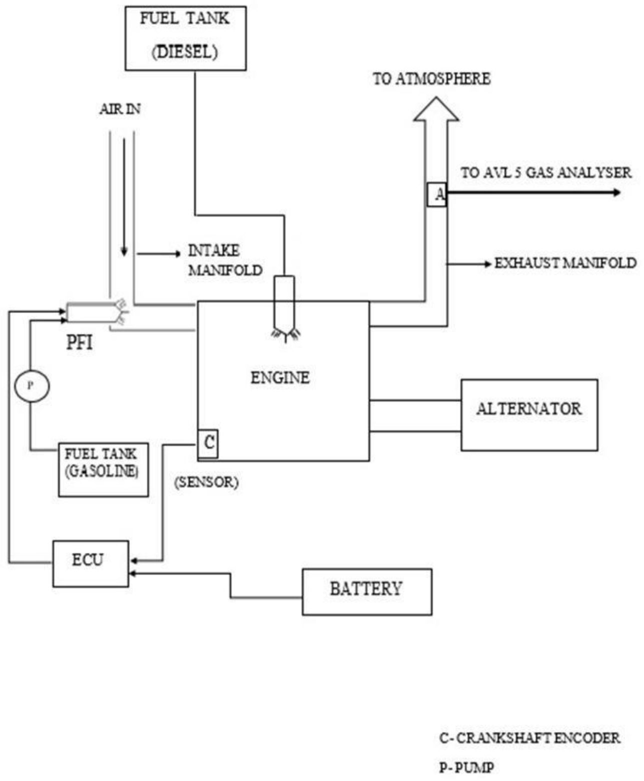

Figure 1 Experimental setup

Table 2 Engine Specification	
Type	Vertical, air cooled, four stroke single cylinder diesel engine
Make	Kirloskar
Number of cylinders	One
Bore	87.5 mm
Stroke	110 mm
Compression ratio	17.5:1
Maximum power	4.4 Kw
Dynamometer	Electrical
Speed	1500 rpm
Injection timing	23 deg CA(before TDC)
Injection pressure	200 bar

Table 3 Port fuel injector specification	
Make	Bosch
Max pr	5 Bar
Distance from inlet valve	160 mm
Controlled by	ECU
Injection timing	20^0 ATDC
Injection amount	20%

The experiment was performed firstly with diesel and then with gasoline (PFI) and biodiesel blends (i.e. Diesel, B20, B20+gasoline, B20A25+gasoline, B20A50+gasoline). Diesel fuel was filled in burette. Then the diesel fuel is supplied to engine by accessing the valves provided on fuel supply line. Electrical power supply is provided to control panel, 5 gas analyzer and smoke meter. The engine is started under no load condition by hand cranking using decompression lever. Then engine is run under no load condition for a few minute so that the speed of engine, 5 gas analyzer and smoke meter is get warmed up and stabilized respectively. After that the engine was made to run on desired load with the help of an electrical alternator. As the load on engine increased from no load to desired load, engine rpm decreases. The desired constant rpm is maintained using screw arrangement. After that run the engine for three minutes so that it can stabilize. Then supply valve of diesel was closed and the valve of burette was opened so that fuel filled up again. The time taken for 10cc of fuel consumption is noted. Temperature of exhaust gases from the engine is displayed on the digital control panel. Unburned hydrocarbon, carbon monoxide, nitrogen oxides and smoke opacity are noted down with the help of 5 gas analyzer and smoke meter respectively. These procedure is repeated for loads of 0%, 25%, 50%, 75%, 100% keeping the rpm constant [1500rpm].After the records of experiment is noted, engine was brought to no load condition and stopped. Before starting the next experiment engine is allowed to cool till exhaust gas temperature reaches the room temperature. The experiment is repeated by replacing the conventional intake manifold with newly designed manifold which holds port fuel injector. Port Fuel injector carries gasoline fuel which is premixed with air before entering into cylinder is controlled using ECU. Consequently, the gasoline injection timing was fixed 20 CAD bTDC to allow the fuel flow. The experiment is also carried out with the same setup for different blending ratios of biodiesel with high reactivity fuel (diesel) i.e. with B20, B20A25 and B20A50.

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4. RESULT AND DISCUSSIONS

In the present study, the performance, emission, and combustion characteristics of the engine fuelled with diesel, biodiesel (B20), gasoline & biodiesel (G-B20), gasoline & biodiesel blended aluminium oxide nanoparticles of 25ppm (G-B20A25), gasoline & biodiesel blended aluminium oxide nanoparticles of 50ppm (G-B20A50) blends were compared and discussed.

4.1. Engine performance parameters

4.1.1. Brake specific fuel consumption

Brake Specific Fuel Consumption (or BSFC) is the ratio between the engine's fuel mass consumption and the crankshaft power it is producing. The fuel consumption of the dual fuel combustion shows the reduced consumption of fuel when compared to the diesel fuel. Fig 2 shows the constant reduction in the consumption of dual fuels throughout the engine loads. Brake Specific Fuel Consumption is decreased at higher loads due to reduction in pumping losses. The brake specific fuel consumption slightly increases with the increasing amount of biodiesel in the fuel blend.

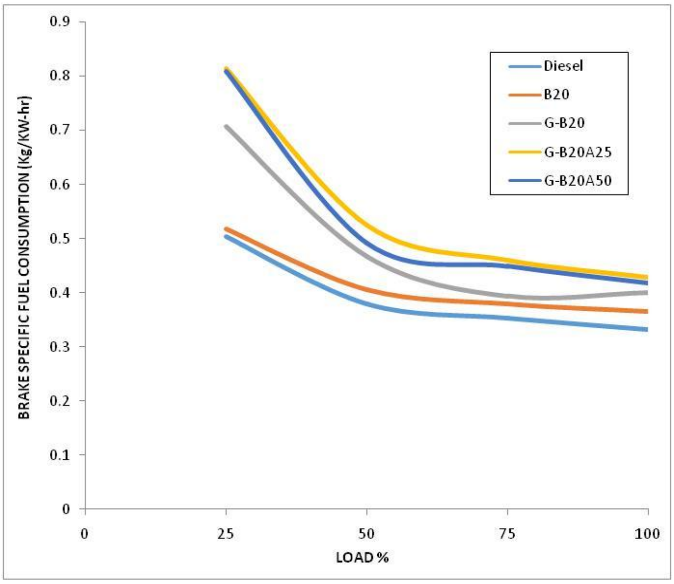

Figure 2 Brake specific fuel consumption Vs Load

4.1.2. Brake thermal efficiency

The Brake thermal efficiency is defined as the ratio of work output at the engine shaft to the energy supplied by fuel. It is a measure of the engine’s ability to make efficient use of fuel. Figure 3 shows the variation of the brake thermal efficiency with the load. The results show that the brake thermal efficiency of the dual fueled engine is reduced when compared with single fueled diesel engine in all loads because of lower combustion chamber temperature. But at higher loads brake thermal efficiency is slightly improved while using nanoparticle blend when compared to G-B20.Use of low reactivity fuel (Gasoline) in PFI and high reactivity fuels in DI slightly decreases the η_{bt} when compared to conventional combustion.

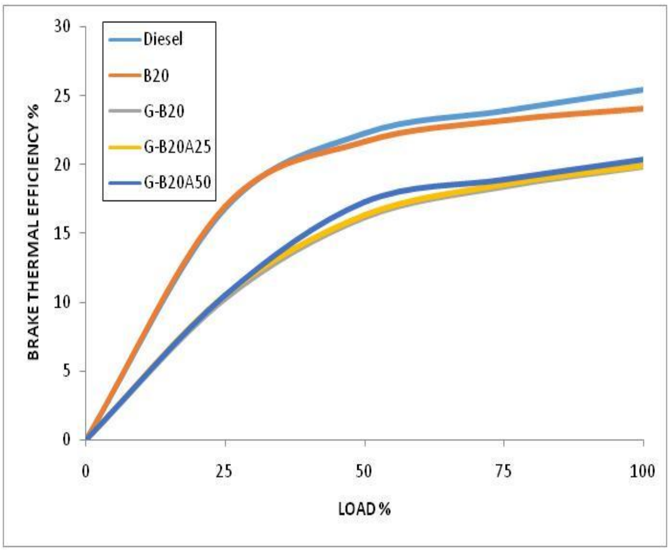

Figure 3 Brake thermal efficiency Vs Load

4.2. Emission parameters

4.2.1. Hydrocarbon

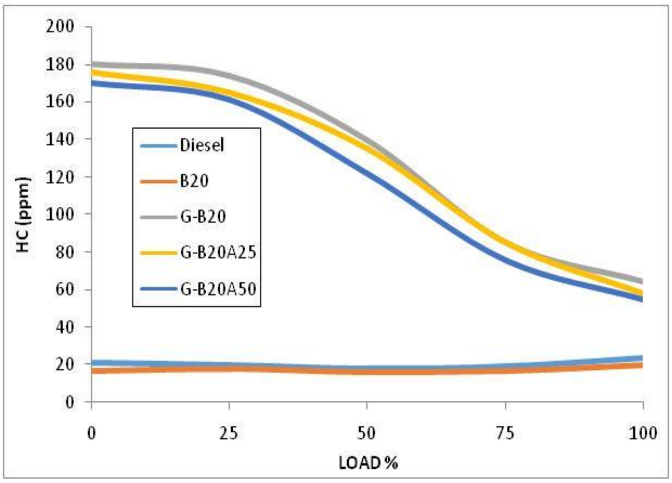

Figure 4 Hydrocarbon Vs Load

Since CI engine operates with an overall fuel-lean equivalence ratio, it has only 1/5th of HC emissions than SI engines. But fig 4 shows that use of gasoline in port (in all blends) results in more HC emissions when compared to conventional combustion mode. This is due to non-availability of oxygen to burn all the fuel and use of gasoline fuel. Some of the other reasons are wall deposition, oil film absorption, crevice volume etc. But at higher loads HC emission for dual fuel mode is decreased also use of nanoparticles reduces HC emissions further since aluminium oxide nanoparticles are oxidation catalyst.

4.2.2. Carbon monoxide

Fig 5 shows the variation of carbon monoxide emission among various blends. Poor mixing, local rich regions, insufficient oxygen may also be the source for CO emissions. Even though blend are present (which has some amount of

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oxygen), Presence of gasoline may be the reason for more CO emission than conventional diesel fuel. But influence of nanoparticles reduces CO emission further due to oxygen content present in Al_2O_3.Thus nanoparticles enhances the CO oxidation.

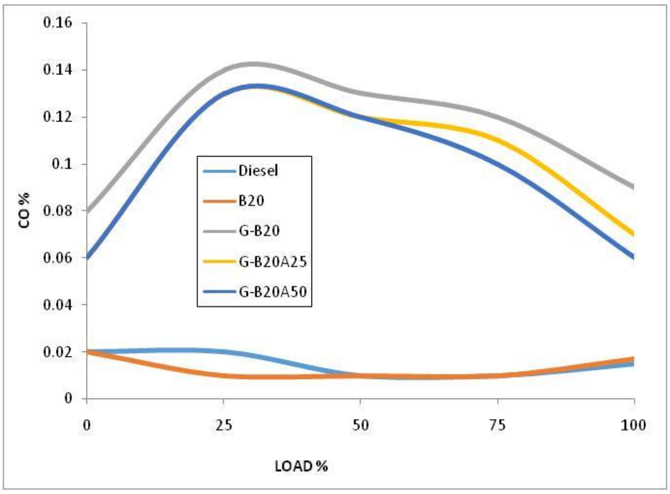

Figure 5 Carbon monoxides Vs Load

4.2.3 Oxides of nitrogen

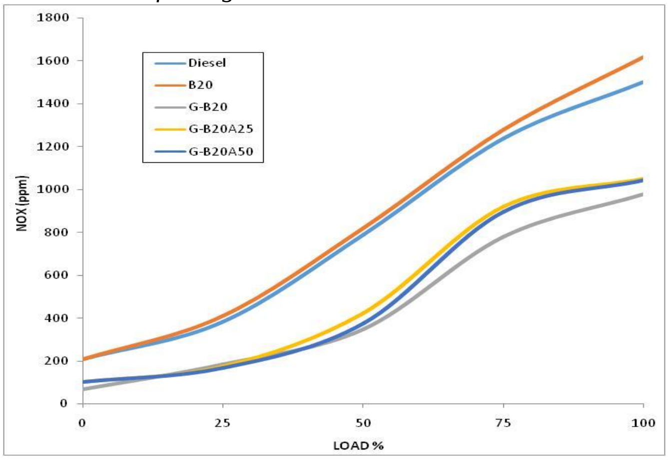

Figure 6 Oxides of nitrogen Vs Load

Fig 6 shows the variation of NO_x emission with respect to load. NO_X emission while using dual fuel injection considerably decreases because of lower combustion chamber temperature (reactivity controlled). Also it reduces 30-35% of NOx emission at high load condition for pure gasoline (PFI) + B20 (DI). It is slightly higher for G-B20A25 and G-B20A50 blends at higher loads but lesser than diesel fuel.

4.2.4 Smoke emission

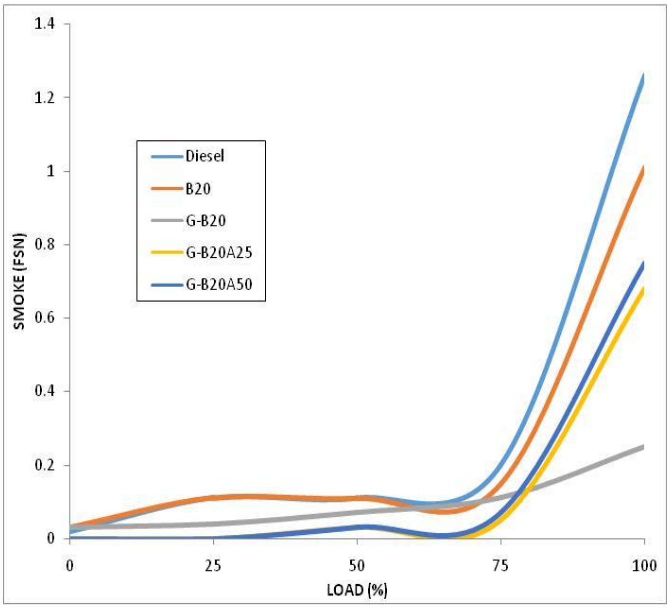

Figure 7 Smoke emissions Vs Load

Smokeless emission obtained using reactivity controlled compression ignition is shown in Fig 7 at low and part loads but it is slightly higher at high loads because of reduced availability of oxygen. When compared to diesel only 50% of smoke is emitted at higher loads by nanoparticle fuel blends.

4.3. Combustion parameters

4.3.1. Cylinder pressure

The pressure rise rate is then plotted against crank angle for the same time period. A typical pressure rise rate against crank angle is shown in Fig 8 it can be seen from this figure that the slope of the pressure- angle curve increases during compression and the combustion period until it reaches the highest value at a certain crank angle then the slope starts to decrease. The pressure rise of G-B20A50 and neat diesel was slightly increased when compared to other blends as load is increased. This indicates that ignition delay of dual fuel blends are more when compared with diesel.

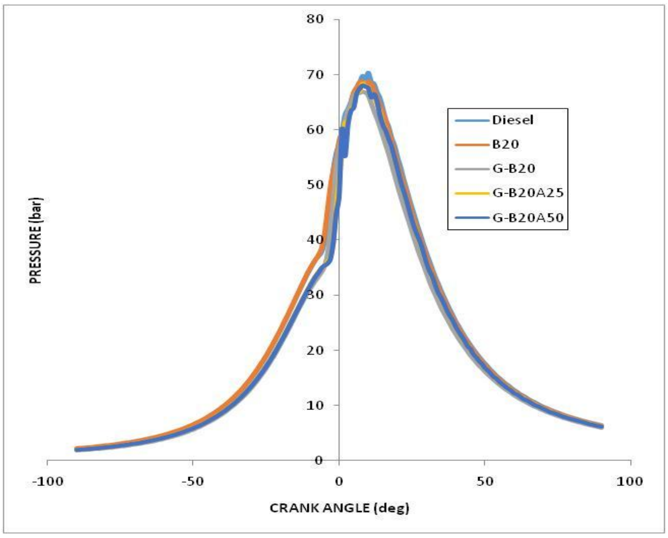

Figure 8 Cylinder pressure Vs Crank angle

4.3.2. Heat release rate

Heat release rate of various fuel blends is shown in Fig 9. Here neat diesel fuel and G-B20A50 have higher heat release rate, this is due to higher calorific value. Meanwhile, HRR of other blends are low due to lower CV. HRR of blended fuels (G-B20 and G-B20A25)reduced due to in homogeneity or presence of oxygen content also higher cetane number may be the reason. Ignition delay of dual fuel mode is increased hence combustion phasing can be controlled. But in case of G-B20A50 HRR is more due to influence of nanoparticles also knocking occurs. Knocking can be reduced in case of G-B20 and G-B20A25.

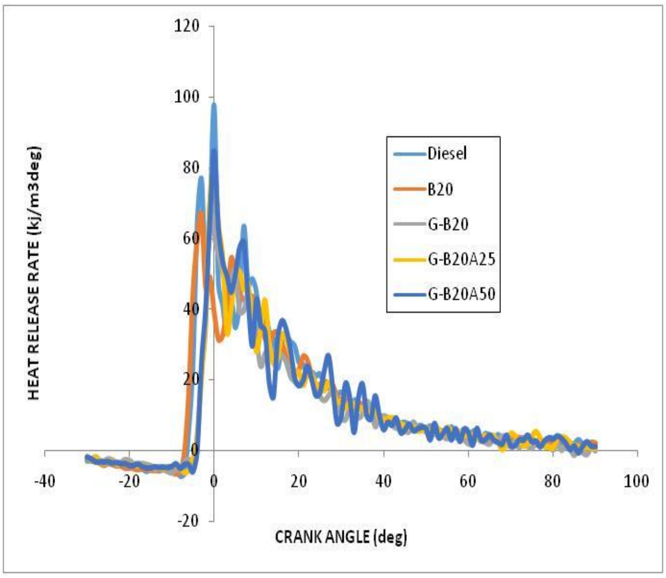

Figure 9 Heat release rate Vs Crank angle

6. CONCLUSION

The experimental study on performance and emission characteristics of low reactivity fuels and biodiesel nanoparticle blends were performed and compared with the conventional diesel fuel. Reactivity controlled compression ignition engine reduces the NO_x emissions to about 30-35% in case of dual fueled injection on an engine. May be use of EGR or prolonging ignition delay may further reduce the NOx at higher loads and about 100% reduction in smoke emissions at low and part loads, during higher loads it is slightly noticed but only 20% of conventional mode, but use of nanoparticles increases smoke emission only 50% of conventional mode. However, HC and CO emissions in both the cases were comparatively high at low and part loads when compared to neat diesel due to low combustion chamber temperature, but use of nanoparticles in blend reduces the HC and CO emissions. No improvement was observed in brake thermal efficiency for dual fuel injection (low reactivity fuel (PFI) and high reactivity fuel (DI) compared to conventional single DI diesel engine. However, use of nanoparticles may slightly improve brake thermal efficiency when compared to G-B20. The brake specific energy consumption is increased for all dual injection strategies than single diesel DI engine at various loads. The heat release rate decreases with all dual fuel blends. But use of nanoparticles more than 25mg increases HRR hence knocking may occur. The premixed combustion phase of conventional diesel fuel is the main reason for higher heat release rate in neat diesel. In-cylinder pressure and temperature is decreased in all dual fuel strategies, but slightly higher in case of G-B20A50.

NOMENCLATURE	
ECU	Electronic Control Unit
CO	Carbon monoxide
CO_2	Carbon dioxide
HC	Hydrocarbon
NOx	Oxides of nitrogen
B20	Diesel 80% + Annona squamosa oil 20%
G-B20A25	Diesel 80% + Annona squamosa oil 20%+Aluminium oxide nanoparticles 25ppm
G-B20A50	Diesel 80% + Annona squamosa oil 20%+Aluminium oxide nanoparticles 50ppm
GD	Gasoline+ Diesel
PFI	Port Fuel Injection
RCCI	Reactivity Controlled Compression Ignition
FSN	Filter smoke number

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