VOL. 8, NO. 12, DECEMBER 2013 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

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EXPERIMENTAL INVESTIGATIONS OF A DI-DIESEL ENGINE WITH AND WITHOUT THERMAL BARRIER COATING DRIVEN BY TETRA METHYL AMMONIUM BROMIDE-ETHANOL-DIESEL EMULSION

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ABSTRACT

An experimental study is conducted to evaluate the effects of using blends of ethanol with conventional diesel fuel with 10%, 20%, 25% and 30% (by vol) ethanol, on the combustion and emissions of a standard, fully in the method four stroke, air cooled, direct injection (DI) with and without thermal barrier coating. New additive Tetra Methyl Ammonium Bromide allows the splash blending of ethanol in diesel in a clear solution. The objective of this investigation is to first create a stable ethanol-diesel blended fuel with 2% additive and then to generate performance combustion and emissions data for evaluation of differential ethanol content on a single cylinder diesel engine with and without thermal barrier coating. Results show improved performance with blends compared to neat fuel for all conditions of the engine. The effect of ethanol blended diesel fuels on brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), smoke and NOx emission has been investigated. The results indicate that with the increase of ethanol in the blends, smoke reduces significantly, BTE improves slightly and combustion duration decreases and the reduction is still better for coated engine NOx emissions were found to be low for coated engine than the normal engine for the blends.

Keywords: DI-diesel engine, thermal barrier coating, blended fuel, additives, performance, emission characteristics.

INTRODUCTION

Engine manufactures all over the world have achieved to develop diesel engines with high thermal efficiency and specific power output, always trying to keep inside the limits of the imposed emissions regulation, which every day become more stringent. Significant achievements for the development of cleaner diesel engines have been made, over the last years, by following various engine-related techniques, such as for example the use of common-rail systems, fuel injection control strategies, exhaust gas recirculation, exhaust gas after-treatment, etc. [1, 2]. Furthermore, especially for the reduction of pollutant emissions, researchers focused their interest on the domain of fuel- related techniques, such as for example the use of alternative fuels often in fumigated form [3, 4], or gaseous fuels of renewable nature that are friendly to the environment [5-7] or oxygenated fuels that show the ability to reduce particulate emissions (8, 9) usually with an escorting increase of the emitted nitrogen oxides. Sometimes, in order to obtain the desirable results, resort is being made to the simultaneous use of engine-and fuel-related techniques. Dwindling crude oil reserves and their constantly increasing prices have placed increasingly sensitive loads on the trade balances of the non-oil producing countries and, meanwhile, have come to represent a threat to the existence of the developing and industrialized countries. Then, considerable attention has been paid on the development of alternative fuel sources in various countries, with particular emphasis on the bio-fuels that posses the added advantage of being renewable fuels that can be replenished through the growth of plants or production of livestock, showing an ad hoc advantage in reducing the emitted carbon dioxide [10, 11]. There is a commitment by the USA government to increase bio-energy three-fold in 10 years, which has added impetus to the search for viable bio-fuels [12]. Approximately one third of the heat released by the combustion of the fuel in a diesel engine is dissipated to the cooling medium. Ceramic coatings have application as thermal barriers to improve the efficiency of the engines, by reducing energy loss and cooling requirements [13, 14]. Kamo [15] indicated thin ceramic coating at a thickness of about 0.004" on the piston and cylinder head surface were more effective in reducing heat rejection. Ramu *et al*., [16] also indicated the same for ZrO_2-Al_2O_3 and SiC coatings. Hence the second phase of the study concentrates on the influence of thin zirconia and alumina coated piston crown, cylinder head on performance and emissions characteristics. Literary survey revealed that several oxygenated organic compounds (ether, amino alcohol surfactants, etc.) may be used as additives and when the ethanol concentration increases beyond 20% high concentrations of additives needed to stable the mixture. Choosing unsuitable organic additive meets with several difficulties: immiscible fuel-alcohol blends, difficult to handle, very expensive, etc., [17-19]. Tetra Methyl Ammonium Bromide miscible in ethanol and in diesel is investigated in this study.

Parameter tested and experimental procedure

Experiments were conducted on a single-cylinder, air-cooled, direct injection diesel engine developing a power output of 5.2 kW at 1500 rpm connected with a water cooled eddy current dynamometer. The engine was operated at a constant speed of 1500 rpm and standard injection pressure of 220 bar. The specification of the engine is given in Table-1. The fuel flow rate was measured on volume basis using a burette and a stop watch. K-type thermocouple and a digital display were

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employed to note the exhaust gas temperature. Smoke level was measured using a standard AVL437C smoke meter. The gas to be measured is fed into a chamber with non-reflective inner surfaces. Light produced by an incandescent bulb scatter on the photo cell from reflections or diffused light inside the chamber. The system converts the current delivered from the photocell in to a linear function of the received light within the operating temperature range. The absorption coefficient is calculated in accordance with ECE-R24 ISO 3173 with an accuracy of $0.025m^{-1}$. The equipment has a micro processor controlled program sequence to check the measurement process and to store such values as pressure, temperature, capacity, and absorption.

Exhaust emissions of unburned HC, CO, CO_2, O_2, and NO_x were measured on the dry basis. A non-dispersive infrared (NDIR - AVL-444 digas) analyzer was used. The exhaust sample to be evaluated was passed through a cold trap (moisture separator) and filter element to prevent water vapour and particulates from entering into the analyzer.

The analyzer was periodically calibrated according to the instructions of the manufacturer. HC and NO_x were measured in ppm hexane equivalents and CO, CO_2, and O_2 emissions were measured in terms of volume percentage. The accuracy and the measuring range of the analyzer are given in Table-2.

Table-1. Specification of the engine.

Type	**Vertical, water cooled, four stroke**
Number of cylinder	One
Bore	87.5 mm
Stroke	110 mm
Compression ratio	17.5:1
Maximum power	5.2 kW
Speed	1500 rev/min
Dynamometer	Eddy current
Injection timing	23° before TDC
Injection pressure	220 bar

Table-2. Accuracy and measuring range of AVL-444 digas analyzer.

Measured parameter	**Measuring range**	**Resolution**	**Accuracy**
CO	0~ 10 vol. %	0.01 vol. %	< 0.6 vol. % ± 0.03 vol.
CO_2	0~ 20 vol. %	0.1 vol. %	< 10 vol. % ± 0.5 vol.
HC	0~ 20000 ppm vol.	≤ 2000:1 ppm vol.	< 200 ppm vol. % ± 10 ppm vol.
O_2	0~ 22vol. %	0.01 vol. %	< 2 vol % ± 0.1 vol.
NOx	0~ 5000 ppm vol.	1 ppm vol.	< 500 ppm vol. % ± 50 ppm vol.
Engine speed	400~ 6000 rpm	1 rpm	± 1% of ind. value
Oil temperature	-30~ 125^0C	1^0C	± 4^0C

AVL combustion analyzer with 619 indimeter Hardware and Indwin software version 2.2 is used to measure in cylinder pressure, heat release rate, indicated mean effective pressure, etc. It consists of inbuilt analog to digital convertor, charge amplifier with PC interface. In cylinder was measured with a water-cooled piezoeletric transducer. The transducer was mounted flush on the cylinder head surface for avoiding passage effects. A piezoelectric transducer produces a charges output, which is proportional to the in cylinder pressure. The charge output was supplied to the inbuilt charge amplifier of the AVL combustion analyzer where it was amplified for an equivalent voltage. A 12-bit analog to digital (A/D) converter was used to convert analog signals to digital form. The A/D converter had external and internal trigging facility with sixteen single ended channels. Data from 100 consecutive cycles can be recorded. Recorded signals were processed with specially developed software to obtain combustion parameters like peak pressure, maximum rate of pressure rise, heat release rate, etc. The schematic experimental set-up is shown in Figure-1.

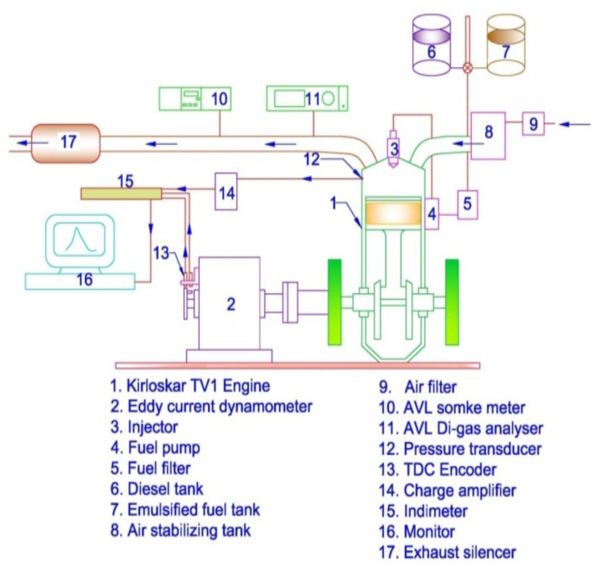

Figure-1. Experimental set-up.

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Base data was generated with standard diesel fuel. Subsequently four fuel blends, namely 90D: 10E, 0D: 20E, 75D: 25E and 70D: 30E along with 2% TMAB which was found as optimum percentage by volume were prepared and tested. The mixing protocol consisted of first blending the emulsifier into the ethanol and then blending this mixture into the diesel fuel. The properties of diesel, ethanol, and TMAB are presented in Table-3. Readings were taken, when the engine was operated at a constant speed of 1500 rpm for all loads. Parameter like engine speed, fuel flow, and the emission characteristic like NOx and smoke were recorded. The performance of the engine was evaluated in terms of brake thermal efficiency, brake power, and brake specific fuel consumption from the parameters. The combustion characteristics like cylinder pressure and heat release rate were noted for different blends. The experiments were repeated for the same fuels after thermally insulated the engine with a thin layer ZrO2-Al2O3 coated piston, cylinder liner, head and bottom of the valves and the results were compared.

Table-3. Properties of test fuels.

Fuel	Molecular formula	Molecular weight	Density at 20^0C (10^3 kg/m^2)	Boiling point (^{0}C)	Flash point (^{0}C)	Viscosity (m Pas)	% of oxygen by weight	Cetane number
Diesel	C_xH_y	190-220	0.829	180-360	65-88	3.35	0	45-50
Ethanol	C_2H_5OH	46	0.79	78.4	13	1.20	34.7	8

Preparation of coatings

Commercially available ZrO_2 and Al_2O_3 ceramic headstock powders (Sulzer Metco) with particle sizes ranging from 38.5 to 63 µm and Ni-20Cr-6Al-Y metal powder (SulzerMetcoNiCrAlY-9) with particle size ranging from 10 to 100 µm were used. The surfaces were grit blasted using 400 mesh Al_2O_3 powder. The substrates were grit blasted until a surface roughness of alumina (Ra-4) was achieved. The grit blasted substrates were ultrasonically cleaned using anhydrous ethylene alcohol and dried in cold air prior to coating deposition. A NiCrAlY bond coat of about 150 µm was air plasma sprayed on to the substrate. ZrO_2 of 150 µm was deposited over the bond coat and Al_2O_3 was sprayed over ZrO_2 coat. The thickness of Al_2O_3 was also150 µm. Air plasma spray system (Ion Arc 40 kW) was used to deposit the coating. No air cooling on the back side of the substrates was applied during the spraying process.

RESULTS AND DISCUSSIONS

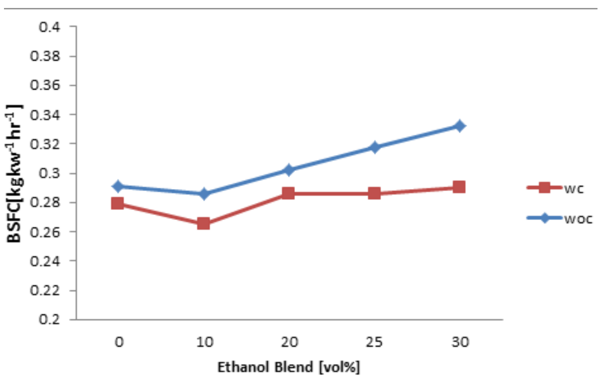

Figure-2. Brake specific fuel consumption for different ethanol blends at full load.

Ethanol has lower heat value than diesel fuel. As the amount of ethanol in the blends increases, heat value of the blends decreases. In order to maintain the same power, more fuels are consumed. As a result, BSFC will increase as the blended fuels with high ethanol concentration are used. Figure-2 shows the specific fuel consumption for different ethanol additions with and without thermal insulation. Among the blends 90D: 10E ratio shows minimum specific fuel consumption to other blends and sole fuel. Decrease in BSFC is observed for thermal barrier coating (TBC) engines due to substantial reduction in combustion chamber heat transfer and reduced friction due to increased wall temperature as indicated by Thring [20].

TBC engine improved the brake thermal efficiency (BTE) of sole fuel by 8% when compared to the standard engine due to the in cylinder heat transfer reduction and increase in combustion duration as indicated by Ramu *et al.*, [16]. The presence of oxygen due to ethanol (oxygenated fuel), improve the combustion, especially diffusion combustion and hence increase the BTE. Figure-3 compares the effect of oxygenated fuel blend on the BTE for the standard and TBC engine. The maximum BTE occur for 90D: 10E blend ratio. The BTE of a normal engine for 90D: 10E blend is almost same when compare to sole fuel at peak load and 8% addition was observed for TBC engine. The BTE decreases with increase in ethanol quantity as it reduces total heat value of the mixture, but still superior to sole fuel.

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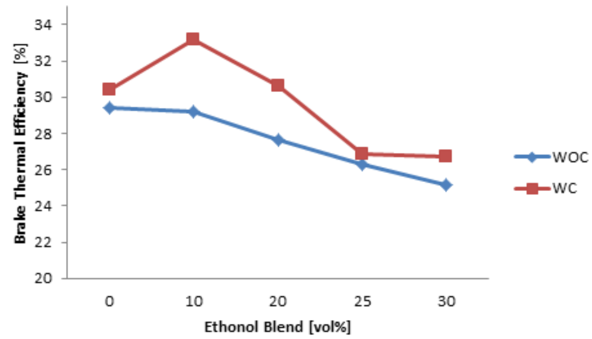

Figure-3. Brake thermal efficiency for different ethanol blends at full load.

The addition of ethanol and TMAB, decreasing the smoke density especially between part load to maximum load as shown in Figure-4 due to increased heat release rate and more complete combustion of the oxygenated fuel.

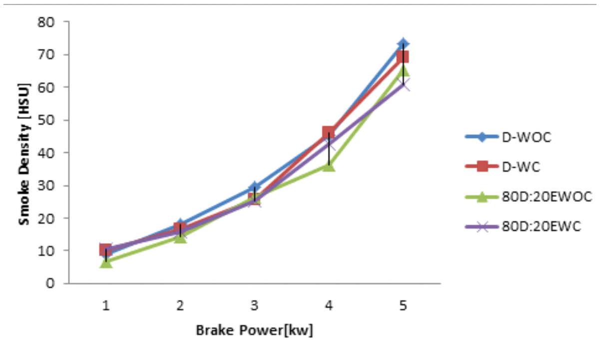

Figure-4. Variation of smoke density with brake power for 80D: 20E blend.

The variation of smoke density for different ethanol blends at peak load is shown in Figure-5.

Reduction of 12% in smoke density for 80D: 20E blend ratios were observed at peak load and further reduction was observed for the engine with thermal barrier coating because of the decreased quenching distance and the increased lean flammability limit. The higher temperatures both in the gases and at the combustion chamber walls of the TBC engine assist in permitting the oxidation reactions to proceed close to completion. A maximum of 8 HSU reductions was observed for 80D: 20E blend ratio on TBC engine against sole fuel at normal conditions. The results reveal that the tendency to generate soot from the fuel-rich regions inside diesel diffusion flame is decreased by ethanol in the blends.

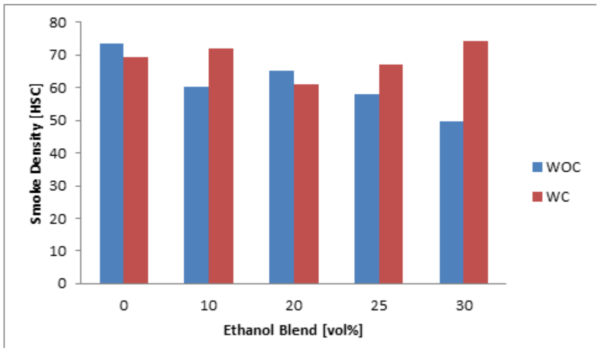

Figure-5. Smoke density for different ethanol blends at full load.

NO_x emissions are predominantly temperature phenomena [21]. Late combustion due to change in the delay period lower the peak pressure of the sole fuel in TBC engines. Since the peak pressure rise is lower, for the same value of mass, the peak gas temperature may also be lower, resulting reduced NO_x emissions for sole fuel. The same trend is observed by Assanis *et al*., [22]. Ramu *et al*., [16] also found lower NO_x for zirconia alumina coated engine for diesel fuel. However, the presence of oxygen increase the heat release rate and maximum pressure rise for the oxygenated fuel and hence the NO_x emission will be high for TBC engine than the standard engine. The anticipated increase in NO_x emissions as a function of increasing ethanol concentration is apparent in Figure-6. It can be seen that NO_x emissions of lower blends increase more rapidly than those of higher ethanol proportion at peak load.

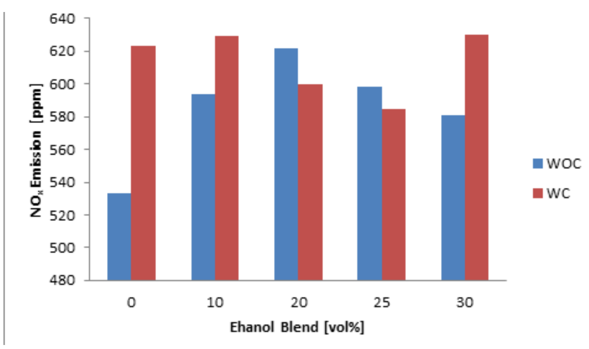

Figure-6. NO_x emission for different ethanol blends at peak load.

The maximum increase in NO_x emissions occur at 50~100% full load conditions because of higher combustion temperature and longer combustion duration due to rich oxygen circumstance from ethanol in the mixture as illustrated in Figure-7.

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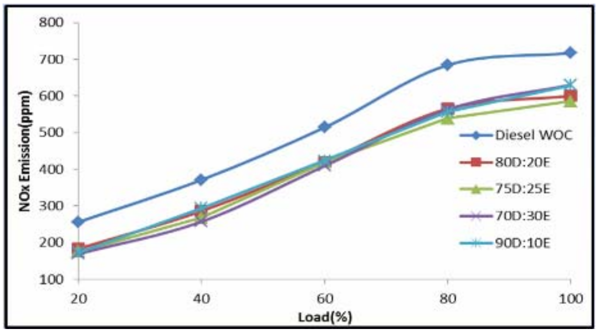

Figure-7. NO_x emissions against brake power.

Figure-8 shows in a bar chart diagram, the CO exhaust emissions for the neat diesel fuel and the various percentages of the ethanol in its blends with diesel fuel, at the peak load. One can observe that the CO emitted by the ethanol-diesel fuel blends is lower than that for the corresponding neat diesel fuel case, with the reduction being higher the higher the percentage of ethanol in the blend. This may be attributed to the engine running overall "leaner", with the combustion being now assisted by the presence of the fuel-bound oxygen of the ethanol even in locally rich zones. TBC reduces the CO emissions due to the in cylinder heat transfer reduction and increase in combustion duration.

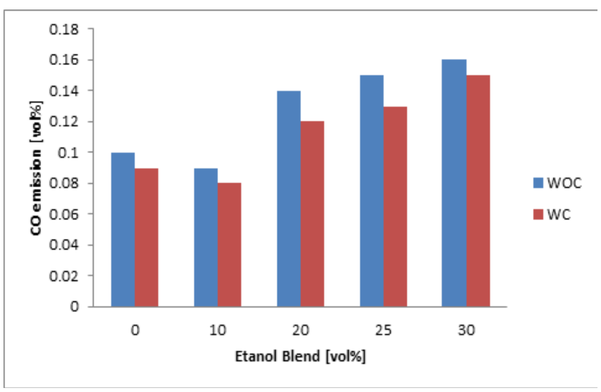

Figure-8. Coemission for different ethanol blends at peak load.

Figure-9 shows in a bar chart diagram, the total unburned HC exhaust emissions for the neat diesel fuel and the various percentages of the ethanol in its blends with diesel fuel at peak load. One can observe that the HC emitted by the ethanol-diesel fuel blends are higher than those for the corresponding neat diesel fuel case, with the increase being higher the higher the percentage of ethanol in the blend. The increase of HC with the addition of ethanol is due to the higher heat of evaporation of the ethanol blends causing slower evaporation and so slower and poorer fuel-air mixing, to the increased spray penetration causing unwanted fuel impingement on the chamber walls (and so flame quenching) and cushioning in the ring and areas, and to the increase with ethanol of the so-called "lean outer flame zone" where flameis unable to exist. Late combustion due to change in the delay period lower the peak pressure of the sole fuel in TBC engines and hence increases the HC emissions. However, the presence of excess oxygen in ethanol increases the combustion pressure and temperature for oxygenated fuels and hence reduces the HC emissions at insulated conditions of the engine.

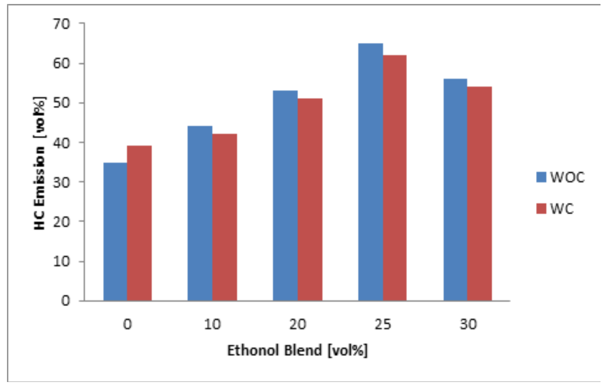

Figure-9. H Cemission for different ethanol blends at peak load.

Ethanol contain oxygen molecule that increase the spray optimization and evaporation. Hence it improves the combustion process of the engine. Figure-10 illustrates cylinder pressure traces of ethanol blended diesel fuels. It is found that at the same engine speed and maximum load, the ignition delay for the oxygenated blend is higher (the pressure rise due to combustion starts later) than the corresponding one for the neat diesel fuel case, while there is a slight increase in the maximum pressure. Rakopoulos *et al*., [23] obtained the same result for 15% ethanol but with no appreciable difference in the maximum pressure due to the lower cetane number of ethanol.

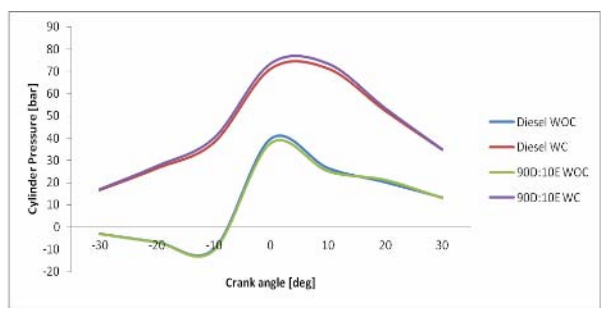

Figure-10. Cylinder pressure against crank angle.

In this case, the increase in pressure is due to the presence of the TMAB which improves the cetane number of the mixture. TBC decreases the ignition delay for the oxygenated fuel due to the increasing gas temperature and hence the cylinder pressure. The peak pressure of sole fuel is 69.9 bar for the sole fuel and is 71.6 bar for 90D: 10E blends for a normal engine. Whereas, the peak pressure for sole fuel is 68 bar and is increased to 74 bar for TBC engine. It can also be seen that for the oxygenated fuel on TBC engine higher pressure region change sharply as with diesel engine, but the durations of the higher pressure period is shorter than that of diesel engine.

One can again observe, from Figure-11, that the ignition delay for the oxygenated blend is higher than the corresponding one for the neat diesel fuel case, while its premixed combustion peak is much higher and sharper. It is the lower cetane number of ethanol that causes the increase of ignition delay and so the increased amount of "prepared" fuel (to this end may also assist the easier evaporation of ethanol) for combustion after the start of ignition and is reflected in cylinder pressure. But Rakopoulos *et al*., [23] not experienced any increase in cylinder pressure for 15% ethanol (without any cetane improver) probably because of the counteracting effect of later combustion in a lower temperature environment.

It can be seen that for the engine without thermal insulation heat release rate curves of the oxygenated fuel blends and sole fuel shows similar curve pattern although the rate of heat release for the 90D:10E shows higher heat release rate than sole fuel. The reason is the rate of diffusion combustion of the oxygenated fuel increasing the heat release rate consequently oxygenated fuel has controlled rate of pre-mixed combustion. The heat release rate is further increased for TBC engines due to increased pre-mixed combustion.

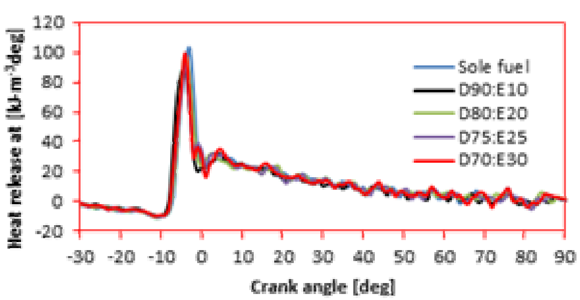

Figure-11. Heat release rate against crank angle.

CONCLUSIONS

The main conclusions of this study are:

- The brake specific fuel consumption increase with increase in ethanol blend in diesel fuel but less than sole fuel. 90D: 10E shows lower specific fuel consumption, and is further decreased for coated engines.
- The brake thermal efficiency for 90D: 10E blend is almost same when compared to sole fuel, where as the increase is 8% for TBC engines.
- Smoke reduction is 8 HSU for 80D: 20E at peak load for the normal engine and is decreased to 8 HSU for the coated engines.
- All blends shows increase in NO_x emission when compared to sole fuel at all engine conditions. Cylinder pressure is higher for 90D: 10E blends than other blends with and without thermal barrier coating.
- The CO emissions were reduced with the use of the ethanol-diesel fuel blends with respect to that of the neat diesel fuel, with this reduction being higher the higher the percentage of ethanol in the blend. Further reduction was observed for TBC engine.
- The unburned HC emissions were increased with the use of the ethanol-diesel fuel blends with respect to that of the neat diesel fuel, with this increase being higher the higher the percentage of ethanol in the blend. TBC increased the HC emissions for sole fuel; on the other hand it decreased the HC emissions for the oxygenated fuels.
- The peak pressure and heat release rate for blends are higher than sole fuel and is maximum for coated engines.

On the whole it is concluded that 90D:10E with 2% TMAB as surfactant can be used as fuel in a compression ignition engine with improved performance and significant reduction in exhaust emissions except NO_x as compared to neat diesel and that can be controlled by other techniques like turbo charging, exhaust gas recirculation, etc. The ethanol ratio can further be improved in thermally insulated conditions.

Acronyms

BSFC	-	Brake Specific Fuel Consumption
BTE TMAB	- -	Brake Thermal Efficiency Tetra Methyl Ammonium Bromide
D E	- -	Diesel Ethanol
HSU TBC	- -	Hartridge Smoke Unit Thermal Barrier Coating
WC WOC	- -	With Coating Without Coating

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