PERFORMANCE ANALYSIS OF 1,4 DIOXANE–ETHANOL–DIESEL BLENDS ON DIESEL ENGINES WITH AND WITHOUT THERMAL BARRIER COATING

by

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1,4 dioxane, a new additive 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 10% 1,4 dioxane additive, and then to generate performance, combustion and emissions data for evaluation of different 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. Drastic reduction in smoke density is found with the blends as compared to neat diesel and the reduction is still better for coated engine. NO_x *emissions were found to be high for coated engines than the normal engine for the blends. The oxygen enriched fuel increases the peak pressure and rate of pressure rise with increase in ethanol ratio and is still superior for coated engine. Heat release pattern shows higher premixed combustion rate with the blends. Longer ignition delay and shorter combustion duration are found with all blends than neat diesel fuel.*

Key words: *blended fuel, additives, thermal barrier coating, performance, emission characteristics*

Introduction

Reducing the emissions and fuel consumption are no longer future goals; instead they are the demands of the day. Indiscriminate extraction and increased consumption of fossil fuels have led to the reduction in carbon based resources. Alternative fuels promise to harmonize sustainable development, energy conservation management efficiency, and environmental preservation. Due to the shortage of petroleum product and its increasing cost, efforts are on to develop alternative fuels especially to diesel oil for full or partial replacement.

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Diesel engines have the advantages of high thermal efficiency, lower emission of CO and HC. However, they have the disadvantage of producing smoke, particulate matter and NO_x and it is difficult to reduce both NO_x, and smoke density simultaneously in diesel engine due to trade off between NO_x and smoke. It follows, therefore, that substantial amount of effort has been directed at providing solutions to these problems. Among various developments to reduce emissions, the application of oxygenated fuels to diesel engines is an effective way to reduce smoke emissions. The potentiality of oxygenated fuels to suppress soot precursor formation is dominated by molecular structure as well as fuel oxygen contents [1]. When oxygen contents in the fuel reach approximately 30% by mass, smokeless combustion in diesel engines could be realized [2]. Since ethanol is a widely available oxygenate with a long history of use in gasoline blends it has also been considered as a potential oxygenate for diesel fuel blending. Researchers have investigated the use of ethanol in diesel engines over the past several decades. The limited miscibility at lower temperature, less heating value, poor lubricating properties and the required minor variations in fuel delivery systems restrict the use of ethanol in diesel fuel [3]. Also the addition of ethanol to diesel fuel decreases the blends' viscosity and causes cetane number of the blends linear reduction at ambient temperature [4]. Usually, when ethanol contents in the blends reach 20-40%, high concentrations of additives are needed to ensure the mixture homogeneity in the presence of high water contents, and to attain the required cetane number for suitable ignition [5, 6]. The first phase of this study has focused on using ethanol as an oxygenated blend component in diesel fuel with appropriate additives to solve the above problem without any engine modifications.

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 [7, 8]. Kamo [9] 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*. [10] 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 alcohols, 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*., [11-13]. 1,4 dioxane (1,4-dioxacy- clohexane) miscible in ethanol and in diesel is investigated in this study. The hydrocarbon moieties of these molecules constitute the hydrophobic portion of the structure due to their strong affinity over diesel fuel while the two oxygen molecules forms very strong hydrogen bond with ethanol as shown in fig. 1, therefore it is non-ionic and form a stable, homogenous emulsion

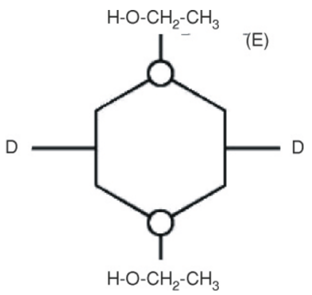

Figure 1. Representation of a micelle between the dioxane (Dy), diesel (D), and ethanol (E)

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 tab. 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 employed to note the exhaust gas temperature.

Table 1. Engine specification

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 rpm
Dynamometer	Eddy current
Injection timing	23° bTDC
Injection pressure	220 kbar, direct injection

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 0.025 m^{-1}. The equipment has a microprocessor controlled program sequence to check the measurement process and to store such values as pressure, temperature, opacity, 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 is given in tab. 2.

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	0~22 vol.%	0.01 vol.%	< 2 vol % 0.1 vol.
NO_x	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 °C	1 °C	4 °C

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 piezoelectric transducer. The transducer was mounted flush on the cylinder head surface for avoiding passage effects. A piezoelectric transducer produces a charge 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 fig. 2.

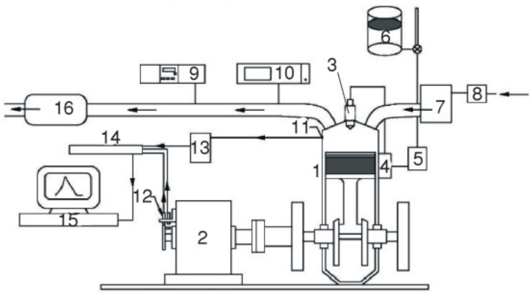

Figure 2. Experimental set-up
(1) – Kirloskar TV1 engine, (2) – eddy current dynamometer, (3) – injector, (4) – fuel pump, (5) – fuel filter, (6) – fuel tank, (7) – air stabilizing tank, (8) – air filter, (9) – AVL somke meter, (10) – AVL Di-gas analyser, (11) – pressure transducer, (12) – TDC encoder, (13) – charge amplifier, (14) – indimeter, (15) – monitor, (16) – exhaust silencer

Base data was generated with standard diesel fuel. Subsequently three fuel blends, namely 70D:20E, 65D:25E, and 60D:30E along with 10% dioxane which was found as optimum percentage by volume were prepared and tested. The mixing protocol consisted of first blending the emulsifier (dioxane) into the ethanol and then blending this mixture into the diesel fuel. The properties of diesel, ethanol, and dioxane are presented in tab. 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 NO_x 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 ZrO_2-Al_2O_3 coated piston, cylinder liner, head and bottom of the valves and the results were compared.

Table 3. Properties of test fuels

	Molecular formula	Molecular weight	Density at 20 °C (·10³ kg/m²)	Boiling point [°C]	Flashpoint [°C]	Viscosity [mPa·s]	% of oxygen by weight	Cetane number
Diesel	C_xH_y	190-220	0.829	180-360	65-88	3.35	0	45-50
Dioxane	$C_4H_8O_2$	88	1.034	101	12	1.20	36	50
Ethanol	C_2H_5OH	46	0.79	78.4	13	1.20	34.7	8

Source: Table on Gasoline and Gasohol from Alcohols and Ethers, API Publication 4261, Second Edition (July1988), United States Environmental Protection Agency

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 (Sulzer Metco NiCrAlY-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 discussion

The stoichiometric air requirement for the combustion of ethanol is lower, since they already contain oxygen in its structure. Also dioxane is having two atoms of oxygen in its structure, hence larger amount of fuel can be burnt in a given amount of air and hence the brake specific fuel consumption (BSFC) decreases for the blends compared with baseline fuel.

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 3 shows the specific fuel consumption for different ethanol additions with and without thermal insulation. Among the blends 70D:20E:10Dy 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 [14].

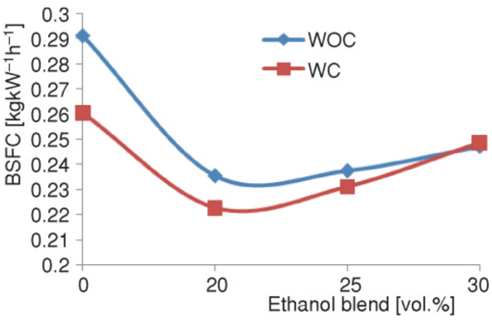

Figure 3. Brake specific fuel consumption for different ethanol blends at full load

TBC engine improved the brake thermal efficiency (BTE) of sole fuel by 2% 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.* [10].The presence of oxygen due to ethanol and dioxane in the oxygenated fuel, improve the combustion, especially diffusion combustion and hence increase the BTE. Figure 4 compares the effect of oxygenated fuel blend on the BTE

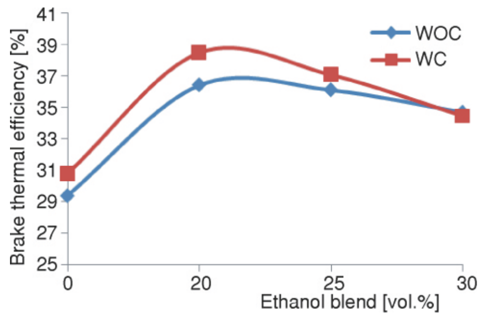

Figure 4. Brake thermal efficiency for different ethanol blends at full load

for the standard and TBC engine. The maximum BTE occur for 70D:20E:10Dy blend ratio. The BTE of a normal engine for 70D:20E:10Dy blends are nearly 6% over sole fuel at peak load and 3% 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.

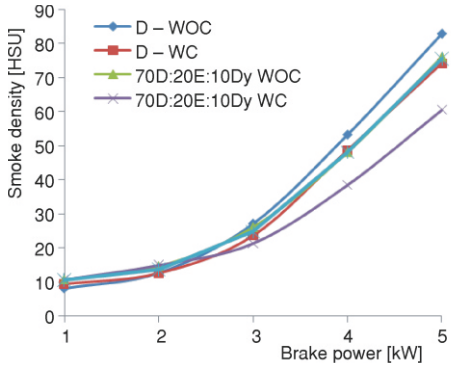

Figure 5. Variation of smoke density with brake power for 70D: 20E: 10Dy blend

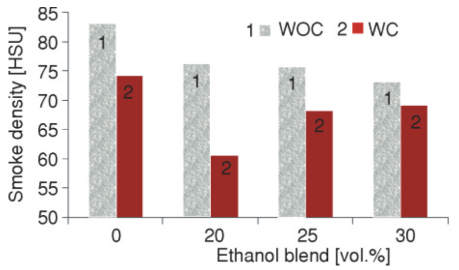

Figure 6. Smoke density for different ethanol blends at full load

The addition of ethanol and dioxane, decreasing the smoke density especially between part load to maximum load as shown in fig. 5 due to increased heat release rate and more complete combustion of the oxygenated fuel. The variation of smoke density for different ethanol blends at peak load is shown in fig. 6. Reduction of 10% in smoke density for 70D:20E:10Dy 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 15 HSU reductions was observed for 70D:20E:10Dy 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.

NO_x emissions are predominantly temperature phenomena [5]. 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*. [15]. Ramu *et al*. [10] 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 fig. 7. It can be seen that NO_x emissions of lower blends increase more rapidly than those of higher ethanol proportion at peak load.

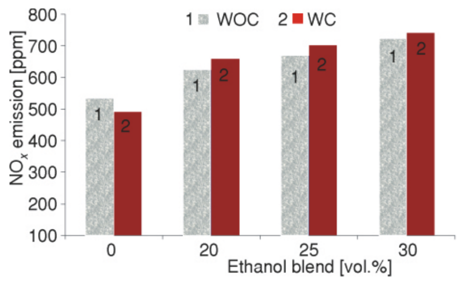

Figure 7. NO_x emission for different ethanol blends at peak load

The maximum increase in NO_x emissions occur at 50~100% full load conditions because of higher combustion temperature and longer com-

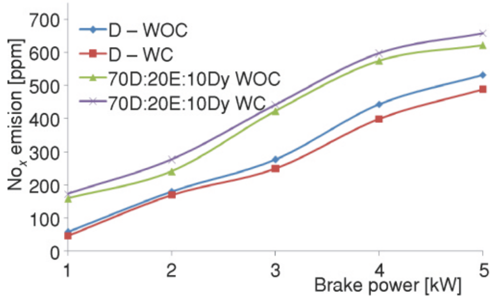

Figure 8. NO_x emissions against brake power

bustion duration due to rich oxygen circumstance from ethanol in the mixture as illustrated in fig. 8.

Figure 9 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 10 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 flame is 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 due to the presence of dioxane and ethanol increases the combustion pressure and temperature for oxygenated fuels and hence reduces the HC emissions at insulated conditions of the engine.

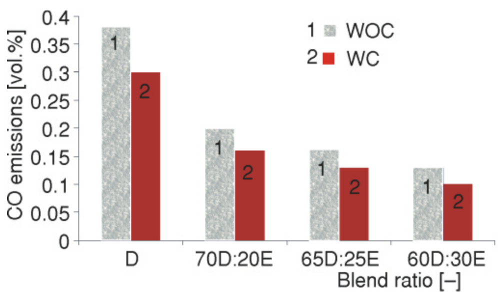

Figure 9. CO emission for different ethanol blends at peak load

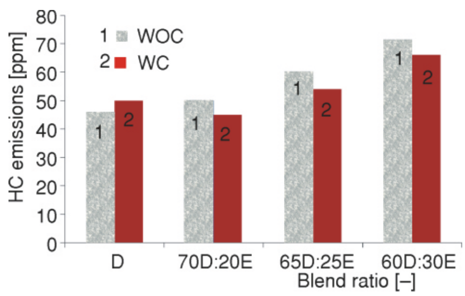

Figure 10. HC emission for different ethanol blends at peak load

Dioxane and ethanol contain oxygen molecule that increase the spray optimization and evaporation. Hence it improves the combustion process of the engine. Figure 11 illustrate 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.* [16] obtained the same result for 15% ethanol but with no appreciable difference in the maximum pressure due to the lower cetane number of ethanol. In this case, the increase in pressure is due to the presence of the dioxane 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 75 bar for the sole fuel and is 77 bar for 70D:20E:10Dy blends for a normal engine. Whereas, the peak pressure for sole fuel is 70 bar and is increased to 82 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 fig. 12, 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*. [16] 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 70D:20E:10Dy shows higher heat release 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.

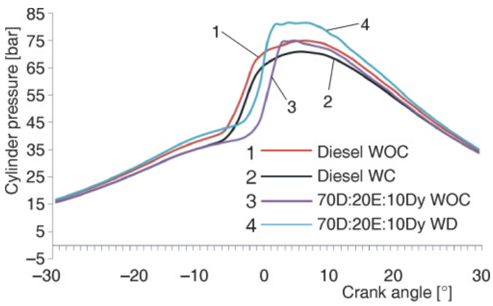

Figure 11. Cylinder pressure against crank angle

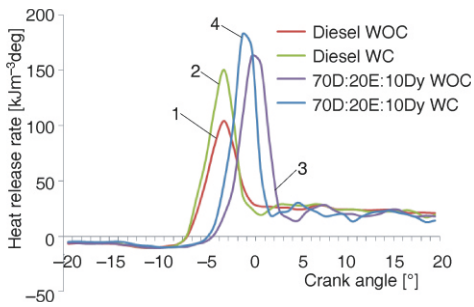

Figure 12. Heat release rate against crank angle

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. 70D:20E:10Dy shows lower specific fuel consumption, and is further decreased for coated engines.

6% improvement of brake thermal efficiency for 70D:20E:10Dy blend when compared to sole fuel, where as the increase is 9% for TBC engines.
Smoke reduction is 9 HSU for 70D:20E:10Dy at peak load for the normal engine and is increased to 15 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 70D:20E:10Dy 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 70D:20E:10Dy blends 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 – brake thermal efficiency
D – diesel
Dy – 1,4 dioxane
E – ethanol
HSU – Hartridge smoke unit
TBC – thermal barrier coating
WC – with coating
WOC – without coating

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