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Air Pollutants and Performance Characteristics of Ethanol-Diesel Blends in CI Engines

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Abstract

Owing to the energy crisis and pollution problems of today, investigations have concentrated on decreasing fuel consumption and on lowering the concentration of toxic components in combustion products by using non-petroleum, renewable, sustainable and non-polluting fuels. While conventional energy sources such as natural gas, oil and coal are non-renewable, alcohol can be coupled to renewable and sustainable energy sources.

In this study, the combustion characteristics and emissions of diesel fuel and ethanol blends were compared. The tests were performed at steady state conditions in a four-cylinder DI diesel engine at full load at 1500-rpm engine speed. The experimental results showed that diesel ethanol blends provided significant reductions in CO, unburned HC, and NOx. Ethanol blends had a 13.8% increase in brake-specific fuel consumption due to its lower heating value. The results indicated that ethanol may be blended with diesel fuel to be used without any modification on the engine.

Keywords: Diesel, ethanol, cetane, solution, oxygenate, performance, emissions, noise.

ملوثات الهواء ومواصفات أداء خلائط ديزل-ايثانول في محركات الاشتعال بالانضغاط

الخلاصة

أهتمت الأبحاث العلمية الحديثة بتقليل استهلاك الوقود وتقليل الملوثات السامة الناتجة من الاحتراق، بسبب الأزمة العالمية الخاصة بالطاقة ومشاكل التلوث، وذلك باستخدام أنواع من الوقود المتجدد، المتوفر وغير مسبب للتلوث، ويمكن اعتبار الكحول مصدر طاقة متجددة ومتوفرة، بينما أنواع الوقود التقليدي مثل الغاز الطبيعي، النفط والفحم غير متجددة.

في هذة الدراسة، تمت مقارنة مواصفات أداء وملوثات وقود ديزل وخلائط من الديزل-الايثانول، وأجريت التجارب باستخدام محرك رباعي الاسطوانات ذي حقن مباشروعند حمل كلي وسرعة 1500 rpm، وبينت النتائج أن خلائط الديزل-الأيثانول تسبب نقصان بملوثات CO، الهيدروكاربونات غير المحترقة HC، وأكاسيد النيتروجين NOx. كما يسبب خليط الايثانول ارتفاع في استهلاك الوقود النوعي المكبحي بحدود 13.8 %، بسبب انخفاض القيمة الحرارية له، وبينت النتائج أن الايثانول يمكن استخدامه باضافتة الى وقود الديزل بدون عمل أي تحويرات في المحرك.

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1. Introduction:

The world is presently confronted with the twin crises of fossil fuel depletion and environmental degradation. Indiscriminate extraction and lavish consumption of fossil fuels have led to reduction in underground-based carbon resources. The search for alternative fuels, which promise a harmonious correlation with sustainable development, energy conservation, efficiency and environmental preservation, has become highly pronounced in the present context (Energy Sources, 2006).

Growing concern regarding the potential global warming impact of fossil fuel combustion and the ambition to diversify energy resources have lead to the identification of targets for the introduction of alternative, climate neutral transportation fuels. Currently available non-fossil fuels are almost exclusively bio-ethanol and bio-diesel (fatty acid methyl ester). On a somewhat longer term, biomass gasification followed by a synthesis process is expected to become a major route towards the production of non-fossil renewable fuel (Demirbas, 2009 & Frijters, 2006).

Alcohol is an alternative transportation fuel since it has properties, which would allow its use in existing engines with minor hardware modifications. Consequently there has been renewed interest in the ethanol–diesel blends with particular emphasis on emissions reductions. An additional factor that makes ethanol attractive as a fuel extender or substitute is that it is a renewable resource (Agarwal, 2007).

Ethanol, which is the same chemical as the alcohol in alcoholic beverages, can reach 96% purity by volume by distillation, and is as clear as water. This is enough for straight-ethanol combustion. Most blends of diesohol are typically made with 10–15% alcohol, 85–90% automotive diesel and a blending agent. Diesel and alcohol do not mix easily, so formulating diesohol requires the use of additives to create stable blends (Boehman, 2005).

The addition of ethanol to diesel fuel affects certain key properties with particular reference to blend stability, viscosity and lubricity, energy content and cetane number (Shivanand, 2009). Materials compatibility and corrosiveness are also important factors that need to be considered (Hansen, 2005). Diesohol, while used in compression ignition engines as an alternative to diesel, behaves quite differently from diesel. The alcohol component in the fuel:

- Changes the combustion characteristics of the fuel.
- Alters tailpipe emissions and engine operability.

- Changes fuel storage and handling requirements (Xing-cai, 2004).

There are many techniques by which ethanol can be used as a fuel in compression ignition engines (Ahmed, 2005). The easiest method by which ethanol could be used is in the form of solutions, but ethanol has limited solubility in diesel; hence ethanol/diesel solutions are restricted to small percentages (typically 20%) (Colban, 2007). This problem of limited solubility has been overcome by emulsions which have the capability of accommodating larger displacement of diesel up to 40% by volume. But the major drawbacks of emulsions are the cost of emulsifiers and poor low temperature physical properties (Ajav, 1997). However, there are many obstacles of using ethanol in CI engines, which are listed as follows:

1. Ethanol has limited solubility in diesel fuel. Phase separation and water tolerance in ethanol–diesel blend fuel are crucial problem (Brusster, 2002).
2. Ethanol fuel has an extremely low cetane number, whereas the diesel engine is prefer to high cetane number fuels which makes auto-ignition easily and gives small ignition delay (Chaichan, 2009).
3. The dynamic viscosity of ethanol is much lower than that of the diesel fuel, so that the lubricity is a potential concern of ethanol–diesel blend fuel (Nagaraju, 2008).

Reference (Chaichan, 2010 & Ciniviz, 2010) studies the effect of adding ethanol to diesel fuel in single cylinder engine. The studies conclutions were: The brake specific fuel consumption for dual fuel, in general, is higher than that for the diesel engine for low and medium loads; however, it becomes less than the diesel engine for high loads. The brake thermal efficiency for diesel engine is higher than that for dual fuel engine, for low and medium loads, while it becomes lower for high loads. In general, the addition of ethanol in the mixture improves the efficiency.

The experimental studies of (Nabin, 2006 & Merritt, 2005) were carried out in a multi-cylinder diesel engine. The results showed an appreciable reduction of emissions such as particulate matter, oxides of nitrogen, smoke density, unregulated emission-benzo (a) pyrene and marginal increase in the performance when compared with normal diesel engine. The presence of ethanol generates different physicochemical modifications of the fossil fuel, notably reductions of the cetane number, lower heat content, viscosity, flash point, and pour point. These modifications changed the spray evaporation properties, combustion performance, and engine-out emissions.

The present paper considers only ethanol-in-diesel solutions since these alone have neither a significant incremental vehicular cost nor high emulsifier fuel-additive cost (often rivaling the cost of the ethanol in the fuel). The objective of this study is to investigate the effect of different ethanol blends on the combustion characteristics of diesel fuel. Therefore, to accomplish this objective, the experiments were carried out. Firstly, four blends of ethanol (10, 15, 20 and 25%) were selected based on the there ability to dissolve in diesel fuel without separation with help of cetane number improver and emulsifier.

2. Experimental setup

2. 1. Materials

The commercial diesel fuel and ethanol (99.7% purity) were used in this work. The ethanol–diesel blended fuels were formed by blending together the following components, diesel fuel, ethanol, solublizer and cetane number improver. The blending preparation procedure was first to blend the cetane number improver and solublizer (1.5%v/v for all ethanol–diesel blended fuels) into the ethanol, and then blend this mixture into the diesel fuel. Therefore, some basic properties of 15% ethanol–diesel blended fuel (without cetane number improver) were measured, and the lower heat values, carbon content, and oxygen content of blended fuels were calculated. The fuel density was measured by weighting a known volume of oil; the oil viscosity was measured by using a dynamic viscometer, the measurement principle consisted of measuring the time needed for a known volume of oil to drop from a viscometer; the flash point was measured by using a close-cup method; the surface tension of fuels were measured by a surface tensiometer. The oil density, surface tension and liquid viscosity were measured at 20^{o}C, all the measurements were repeated at least three times, and the average results are shown in Table 2.

2.2. Equipments

Experimental apparatus of engine under study is DI, water cooled four cylinders, in-line, natural aspirated FIAT diesel engine whose major specifications are shown in Table 1. The engine was coupled to a hydraulic dynamometer through which load was applied by increasing the torque. Fig. 1 gives a photo of the tested engine.

The Multigas mode 4880 emissions analyzer was used to measure the concentration of nitrogen oxide (NOx), unburned total hydrocarbon (HC), CO_2 and CO. The anoalyser detects the CO, CO_2, HC and O_2 content. The gases are picked up from the engine exhaust pipe by meons of the probe. They are separated from water they contain through the condensate separating filter, and then they are conveyed in lhe measuring cell. A ray of infrared light, which is generated by the transmitter, is send through the optical filters on to the measured elements. The gases which are contained in the measuring cell absorb the ray of light of different wave lengths; according to their concentration. The H_2, N_2 and O_2 gases due to their moleculor composition (they have the same number of atoms), do not absorb the emitted rays. This prevents from measuring the concentration through the infrared system. The CO, CO_2 and HC gases, because of their molecular composition, absorb the infrared rays at specific wavelengths (absorption spectrum). However the analyser is equipped with a chemical kind sensor through which the oxygen percentage is measured.

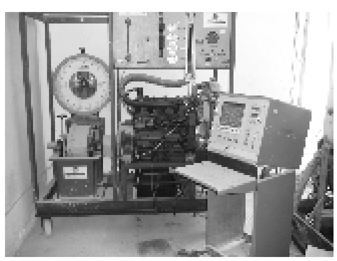

Figure (1) A photo of the used engine

Overall sound pressure was measured by precision sound level meter supplied with microphone type 4615 Italy made, as appears in fig. 2; the devise was calibrated by standered calibrator type pisto phone 4220. It measures overall sound pressure in desiple units (dP).

Figure (2) Overall sound pressure used in the tests

2.3. Analysis

The following equations were used in calculating engine performance parameters:

1- Brake power (*kW*):

$$bp = \frac{2\pi * N * T}{60 * 1000}$$

2- Brake mean effective pressure (*kN/m*^{2}):

$$bmep = bp \times \frac{2 * 60}{V_{sn} * N}$$

3- Fuel mass flow rate (*kg/sec*):

$$\dot{m}_f = \frac{v_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time}$$

4- Air mass flow rate (*kg/sec*):

$$\dot{m}_{a,act.} = \frac{12\sqrt{h_o} * 0.85}{3600} \times \rho_{air} \ kg/sec$$

$$\dot{m}_{u_{theo.}} = V_{s.u} \times \frac{N}{60 * 2} \times \rho_{air} \;\; kg/sec$$

5- Brake specific fuel consumpotion (*kg/kW.hr*):

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600$$

6-Total fuel heat (*kW*):

$$Q_t = \dot{m}_f \times LCV$$

7- Brake thermal efficiency (*%*):

$$\eta_{bth.} = \frac{bp}{Q_t} \times 100$$

3. Results and Discussion

Brake specific fuel consumption (bsfc) variation with load for the tested fuels is shown in Fig. 3. BSFC of diesel-ethanol blends are higher than that of the diesel fuel at low and medium loads. More fuel is necessary to compensate low net calorific value of blend fuel to achieve combustion process, or to produce the same amount of heat compared to the higher one. Therefore, 25% ethanol blends with the lowest net calorific value among the test fuels shows the highest bsfc.

Blends bsfc improved for high loads about 2.5% at 10% blend, and reaches 10% with 25% blend at subjected load equals 8.46 bar. Alcohol presence in more quantities inside combustion chamber with its high oxygen content, improved combustion characteristics and reduced the unburned lost fuel with exhaust.

Fig. 4 confirms the above mentioned; increasing ethanol percentage in the mixture at certain load improves bsfc, because of better utilization of combustion by increasing oxygen percentage inside combustion chamber. For lower ethanol blends the fuel consumption became higher due to the differences in heating values between the two fuels.

Diesel engine gives better brake thermal efficiency for low and medium loads, while diesel-ethanol blends engine record an improvement at high loads, because of better combustion with more oxygen, as shown in fig. 5. Brake thermal efficiency deteriorates at low loads because of mixture temperature reduction. Also, ethanol takes its high evaporation energy from combustion chamber, reducing its temperature and pressure. Hence, increasing delay period, and this reduced the output power. Incomplete combustion can be considered another reason.

Fig. 6 clarifies total power utilization and alcohols share rates in energy inside the fuel mixture. Increasing ethanol percentage in the blend increase the total power needed to carry out the desired output power in accelerating rate at the beginning. To equalize total mixture heating value reduction and alcohols cooling effect. After that, it started to improve due to better advantages of combustion air, and reaching approximately complete combustion. It can be seen clearly that improvement rate became better for high engine loads.

Misfire limits were studied as fig 7 shown. For tested loads range, it was restricted between values (49%, 39%, 34.3%, 31.8% and 29.04%) ethanol for loads values (no load, 2.04, 3.4, 6.7 and 8.4) bar respectively. Misfire happens for long delay periods and puts limits for the replaced diesel fuel with alcohols. Extensive pressure rise rate engaged with fast and violent combustion of alcohol/air mixture refer to self ignition. The charge after primary reactions increases the beforehand cylinder pressure and temperature, and this increased the diesel delay period. Alcohol replacement is limited by cooling effect and delay period problems at low and medium loads. While for high loads replacements it is limited by fast combustion that causing knock.

Fig. 8 shows engine operation with 10% ethanol and neat diesel, at constant load (7.44 bar) and speed (1500 rpm) conditions and variable injection timing. BSFC of diesel fuel was higher in general, except at optimum injection timing where diesel's bsfc improved and became less than 10% ethanol. The reduction was about 3% at optimum injection timing, and when injection timing retarded, diesel's bsfc increased to approach 6% at injection timing 23.5^{o}BTDC, which is considered late for this engine.

Retarding injection timing away from optimum injection timing (38.5°BTDC for this engine) increased bsfc gradually at 35°BTDC. The increament value for 10% ethanol reached 13% compared with neat diesel. BSFC increased sharply to reach 40% compared to its value at optimum injection timing. Late injection timing causes high pressure rates due to fuel injection close to top dead centre. Combustion will take place after piston falls at expansion stroke without allowing for appropriate mixture preparing time. This reduction in maximum cylinder pressure will require more fuel to be injected into combustion chamber to compensate exhausted unburned fuel. The cooling effect of evaporated ethanol will increase delay period also, slowing down pressure rise inside cylinder.

Fig. 9 shows that brake thermal efficiency of diesel engine at optimum injection timing was better from that for 10% ethanol blend. Compared to diesel-ethanol blends at retarded timings, it was lower than that resulted from 10% ethanol. At optimum timing the blends efficiency became higher than diesel fuel, and can be improved more if the injection timing can be advanced. However, this solution has side effects represented in producing high pressure rates, combustion noise aggravation and possibility of knock taking place.

Exhaust gas temperatures reduced with increasing ethanol fraction in the mixture, as fig. 10 shows, due to reduction in mixture total heating values.

Engine speed effect was studied when the engine operated with several diesel-ethanol blends, as fig. 11 represents. Brake power reduced with increasing ethanol percentage in the mixture. The maximum reduction was with 25% ethanol addition, due to its low heating value. Brake power increased for medium speeds and reduced for low and high speeds. Based on the results of bp developed by the engine on different diesel-ethanol blends, it can be said that they have nearly similar power producing capability as diesel fuel.

Brake specific fuel consumption increased with ethanol ratio increase in the mixture, as fig. 12 indicates. When engine speed increased bsfc for all fuel were increased. Ethanol blends influence was clear due to the mixture heating values reduction. This reduction in the total produced energy reflected on exhausted gas temperatures, fig.13, which was reduced with ethanol percentage increment.

A compareson of recent results was made with researches study similar issue. Meiring et al., 1983 reported a 5% drop in maximum fuel delivery when evaluating a 30% ethanol–diesel blend in a tractor engine fitted with a rotary distributor pump. Hansen et al., 2000 measured a 7– 10% decrease in power at rated speed with a 15% dry ethanol, 2.35% PEC additive and 82.65% diesel fuel blend run in a Cummins 5.9 L engine. Kass et al., 2001 checked the torque output from the same model engine with two blends containing 10% and 15% dry ethanol, respectively, and 2% GE Betz additive, and reported an approximate 8% reduction for both fuel blends.

Xing-cai, 2004 studied the effects of cetane number (CN) improver on performance, emissions, combustion characteristics and heat release rate of a 4-cylinder high speed DI compression ignition engine fuelled with ethanol–diesel blend fuels containing various proportions of CN improver. The main result was that bsfc increased, but the diesel equivalent BSFC decreased, and the thermal efficiency improved remarkably when diesel engine fueled with blends. The CN improver has a positive effect on engine thermal efficiency and fuel consumption. It can be observed results approach despite the diversity of tested engines and conditions.

The NOx emissions from the tested fuels are shown in Fig. 14. NOx formation in combustion process is mainly controlled by the combustion temperature. On the other hand, the combustion temperature is depended on the injection timing of the fuel, ignition delay time and combustion pattern. From the experimental results, there are no differences in NOx emissions of diesel-ethanol blends and the diesel fuel at lower load level. But at 75- 100% load level, about 5-10% reduction can be seen in NOx concentrations. The reduction of NOx emission from diesel-ethanol blends compared to that of diesel fuel, are probably due to the lower combustion temperature related with the shorter ignition delay and lower heat release rate of blends.

The HC emissions of the tested fuels are shown in Fig. 15. From the experimental results, the HC emissions from all diesel-ethanol blends fuels are lower than that of the diesel fuel. The lower HC emission from blended fuels was probably due to the oxygen content in ethanol chemical structure. The present of this oxygen allows the fuel to burn completely, so fewer unburned fuel emission result. Therefore, more ethanol in diesel-ethanol blends shows more reductions in HC emission.

The CO emissions from tested fuels are shown in fig. 16. Generally, CO is generated when there is not enough oxygen to convert all carbon to CO_2, some fuel does not get burned and some carbon ends up as CO. The other factors of CO emission are poor fuel air mixing; local fuel rich region and incomplete combustion will create some CO. From the experimental results, the CO emission from diesel-ethanol (from10 to 20%) blends is lower than that of the diesel fuel at all load levels. While for 25% blend has higher CO emission compared to the diesel fuel at maximum loads, due to high combustion chamber temperatures which increase dissociation from CO_2 to CO.

From figures (14, 15 and 16) it can be seen that, at 100% load level, the HC emissions from 25% blend reduced about 51%. NOx concentrations reduced about 22.5%, than that of the diesel fuel. The CO emission from these blends is higher with about 15%. While at no load operation, HC emissions reduced 81%, NOx reduced about 65% and CO concentrations reduction was about 70%.

Figure 17 represents exhaust CO_2 concentrations which are higher in diesel-ethanol blends case for most loads than diesel fuel. This an evidence of improvements in combustion process, so all carbons find its desired oxygen particles to react with, reducing HC and CO as shown in figures 15 and 16.

Sound or noise increased with increasing load (fig. 18).Sound values of diesel-ethanol blends were found lower (8.5-10%) than the sound values of diesel fuel throughout all loads. The minimum reduction was (4.2%) observed at the low loads, and the maximum decrease (16.05%) at the maximum loads. Combustion improvements due to ethanol blends reduced noise, although it still higher than accepted limits, and the rig must be isolated with proper design and materials to reach acceptable levels.

NOx emission from diesel engine is a primary concern. The NOx emissions from test fuels are illustrated in Fig. 19. The NOx emission from blends is slight lower and diesel fuel, and (10 and 15% ethanol) are higher than the other blended fuels. But the NOx emission of 25% blends was lower. As described previously, the NOx formation in combustion process is very complicated, generally, the earlier combustion timing can result higher combustion temperature and higher NOx emission. Therefore, the NOx emissions from tested fuels reduced with engine speed increase, and may be influenced with the time available for formation.

HC concentrations reduced with engine speed increase (fig. 20) for diesel fuel, and for 10, 15 and 20 % ethanol blends, and increased for 25% ethanol fraction at medium and high speeds. At low speeds HC reduced about 44% with 25% ethanol fraction, and at high speeds it was reduced about 9% at 20% ethanol blend, and increased for 25% ethanol with about 4% at 1900 rpm. In addition to incomplete combustion due to mixture turbulence was increased, the trapping of fuel in crevices and boundary layers, can be considered as a main cause resulted in low oxidation at low temperatures accompanied with increasing ethanol fraction in the mixture.

In general, CO emissions reduced with increasing ethanol in the mixture, the CO emission from 25% ethanol blend is the lowest and that for diesel fuel is the highest. This is shown in Fig. 21. The CO emissions reduced about 20% compared to diesel fuel at low speeds, and reduced with levels are nearly the same at medium speeds; these concentrations reduced about 16.67% at high speeds. Combustion improved with ethanol addition at low and medium speed, due to increment in oxygen particles inside the chamber reducing exhausted CO, this improvement affected with increasing speed and reduction in heating values which diminished the differences.

A review for former results was made by comparing it with researches study similar issue. The results of (Spreen, 1999 & Kass et al., 2001) showed a consistent reduction in NOx varied about 4–5%. Both reported decreases in CO emissions occurred, while HC increased substantially, but both were still well below the regulated emissions limit. Schaus et al., 2000 reported decreases in NOx concentrations being dependent on speed and load of the engine.

Xing-cai, 2004 studied the effects of CN improver on performance and emissions of 4-cylinders diesel engine. The study concluded that NOx emissions decreased when diesel engine fueled with blends; and the NOx emissions further reduced when CN improver was added to blends. CO emission increased remarkably at lower and medium loads, while the increasing degree of CO emission decreased with increase in the CN improver addition. HC emission is very lower at overall engine operating conditions.

Although the former papers varied away from the recent work results in emissions reduction rates. They agreed that in general adding ethanol to diesel fuel reduces emitted emission. The reduction ratio depends on engine fuel metering technology, fuel additives, and exhaust control technology, age of the vehicle, maintenance history, test procedure, and test conditions.

4. Conclusions

Ethanol diesel blends are at an earlier stage of development compared to other alternatives as CNG or LPG. These blends can also reduce GHG emissions although with commercially available sources of ethanol the reduction is not as great as desired to prevent using additional equipments to cut emissions. On an energy basis, ethanol provides about the little reductions in produced bp and increased of bsfc. Ethanol lowers the flash point of the fuel below the legal limit for diesel fuel. This means that ethanol diesel blends will have to be treated as gasoline rather than as a diesel fuel. Ethanol has a low cetane rating and ethanol diesel blends will require the addition of a cetane improver to provide adequate performance of the fuel. This additive as well as the emulsifier required to keep the ethanol and diesel in suspension add to the cost of the fuel.

The addition of ethanol to diesel fuel does reduce the HC, carbon monoxide, NOx emissions and noise. The main results can be obtained as follows:

1. The bsfc increased, but the diesel equivalent bsfc decreased, and the thermal efficiency improved remarkably when diesel engine fueled with blends. The ethanol addition has a positive effect on engine thermal efficiency, and on fuel consumption for high loads.
2. NOx emissions decreased when diesel engine fueled with blends; and it further reduced with speed increase.
3. CO emission reduced remarkably at lower and medium loads, and at low and medium speeds, while the

increasing degree of CO emission decreased with increase in the speed. HC emission is very lower at overall engine operating conditions.

4. The ignition delay prolonged, and the total combustion duration shortened for ethanol–diesel blend fuel when compared to diesel fuel. It is desired to modify injection timing with the used bled.

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Nomenclature

TDC	top dead centre
BMEP	brake mean effective pressure
BSFC	brake specific fuel consumption
BTE	brake thermal efficiency
CA	crank angle
OIT	optimum injection timing
HC	unburned hydrocarbons
NOx	nitrogen oxides

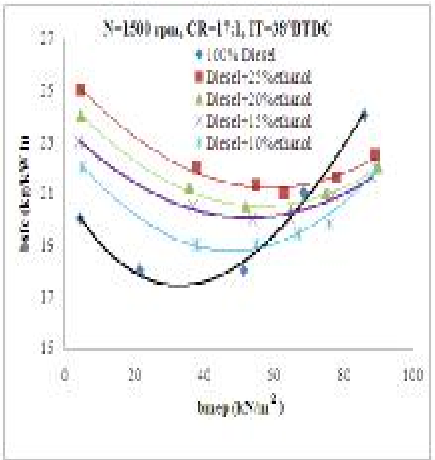
Table (1) Tested engine specifications

Engine type	**4cyl., 4-stroke**
Engine model	**TD 313 Diesel engine reg**
Combustion type	**DI, water cooled, natural aspirated**
Displacement	**3.666 L**
Valve per cylinder	**two**
Bore	**100 mm**
Stroke	**110 mm**
Compression ratio	**17**
Fuel injection pump	**Unit pump** **26 mm diameter plunger**
Fuel injection nozzle	**Hole nozzle** **10 nozzle holes** **Nozzle hole dia. (0.48mm)** **Spray angle= 160^o** **Nozzle opening pressure=40 Mpa**

Table (2) Fuel properties for diesel and ethanol.

specification	Diesel	Ethanol	15% ethanol-diesel
Chemical formula	$C_{10.8}H_{18.7}$	C_2H_5OH	-
Mole weight (g)	148.3	46.1	-
Density (g/cm^3 at 20^{o}C)	0.84	0.789	0.828
Boiling point (^{o}C)	180-330	78	-
Heat of evaporation (kJ/kg)	280	856	-
Lower heat value (MJ/kg)	42.5	27.0	39.5
Liquid viscosity (cP at 20^{o}C)	3.03	1.2	2.87
Surface tension (mN/m at 20^{o}C)	34.1	28.9	27.5
Flash point (^{o}C)	78	13.5	14
Stoichiometric air fuel ratio	14.4	9	-
Cetane number	45	5-8	-
Auto-ignition (^{o}C)	235	423	

Carbon content (wt%)	87.4	52.2	80.3
Oxygen content (wt%)	0	34.3	5.1



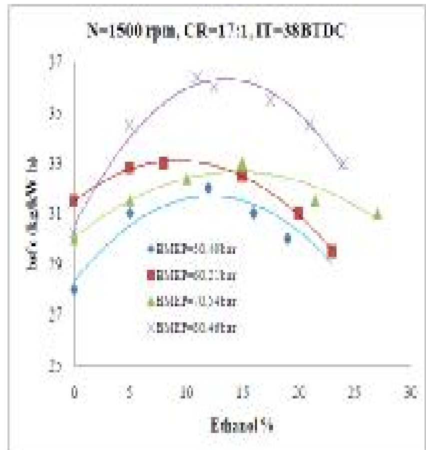
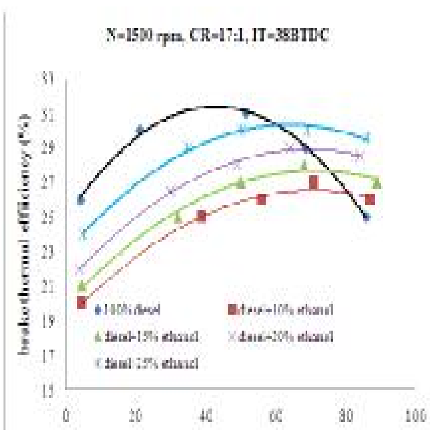
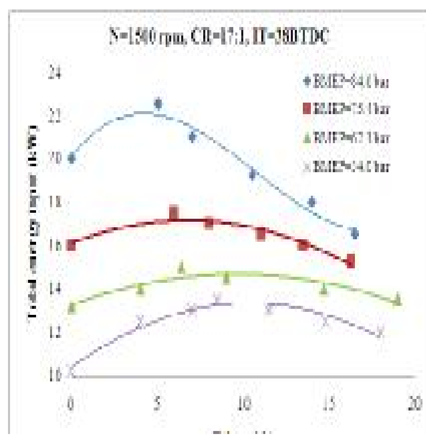
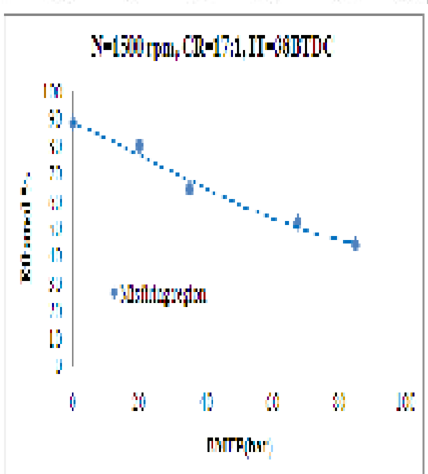


Figure (3) Brake specific fuel consumption of the tested fuels

Figure (4) Brake specific fuel consumption with ethanol percentage in the fuels







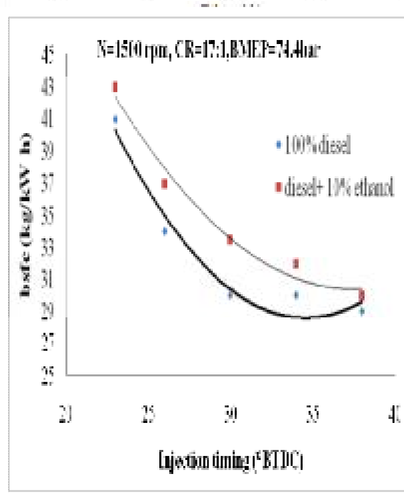


Figure (7) Misfiring region with

Figure (8) Brake specific fuel

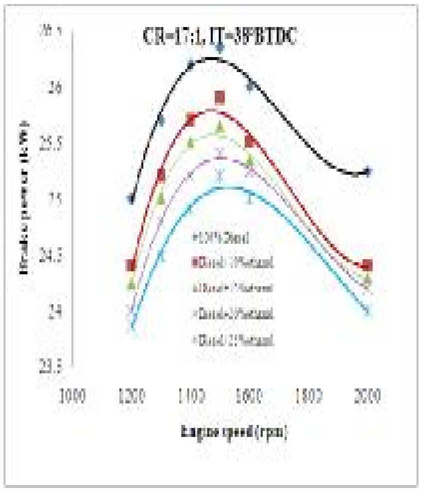


Figure (11) Brake power with engine speed relationship

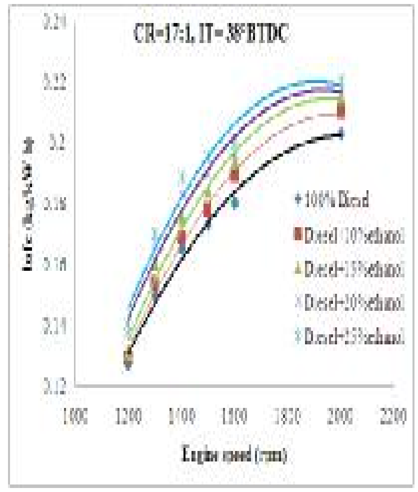


Figure (12) Brake specific fuel consumption with engine speed relationship

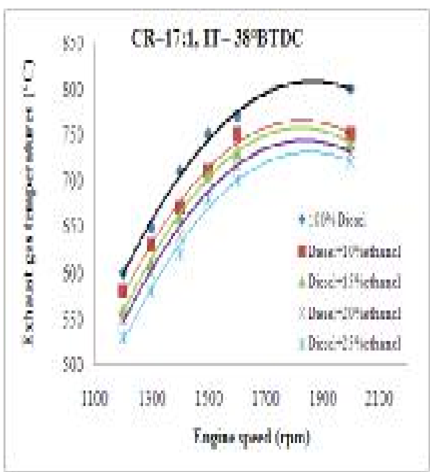


Figure (13) Exhaust gas temperatures with engine speed relationship

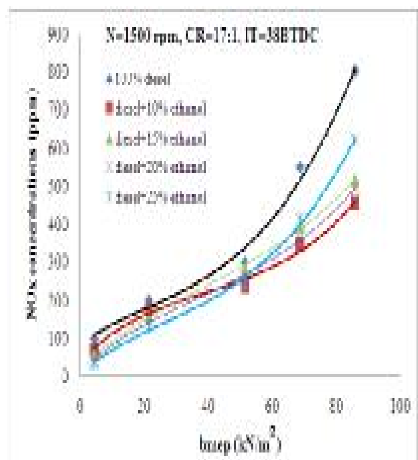


Figure (14) NOx concentrations variation with bmep relationship

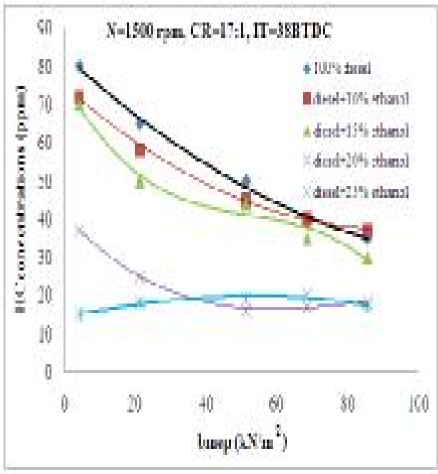


Figure (15) HC concentrations with bmep relationship

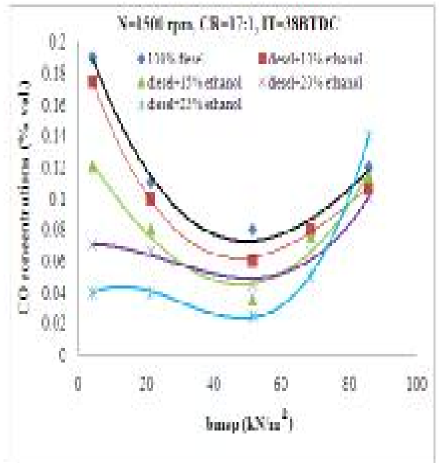


Figure (16) CO concentrations with bmep relationship

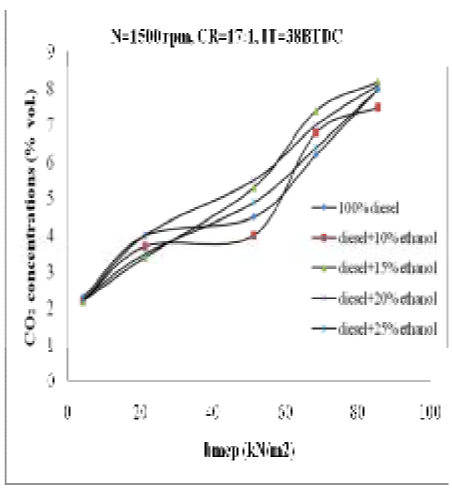


Figure (17) CO_2 concentrations with bmep relationship

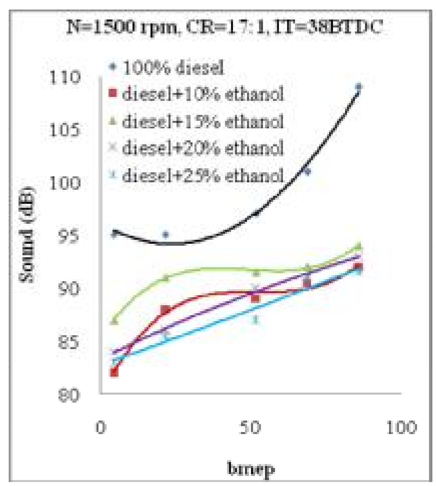


Figure (18) Sound levels with bmep relationship

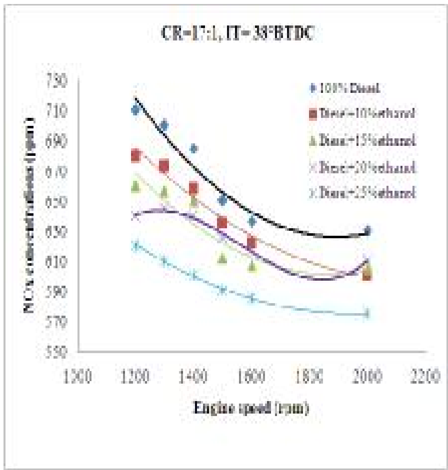


Figure (19) NOx concentrations with engine speed relationship

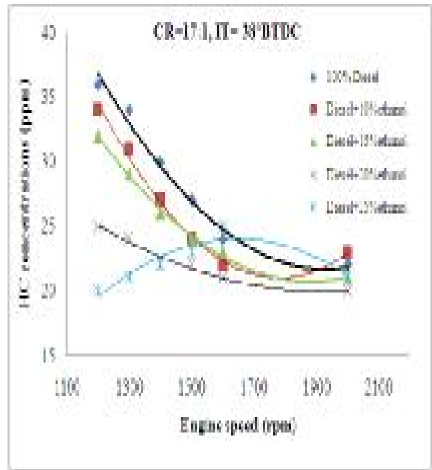


Figure (20) HC concentrations with engine speed relationship

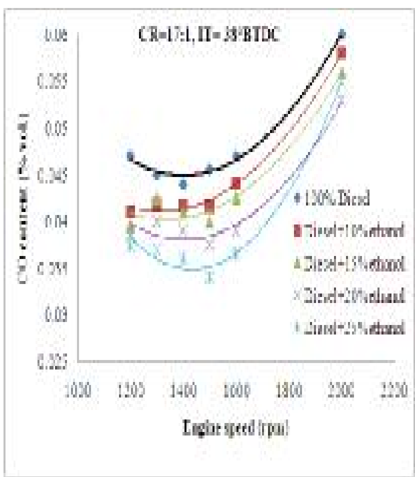


Figure (21) CO concentrations with engine speed relationship