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The effect of alcohol fumigation on diesel engine performance and emissions

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Abstract

The effects of ethanol fumigation (i.e. the addition of ethanol to the intake air manifold) and ethanol–diesel fuel blends on the performance and emissions of a single cylinder diesel engine have been investigated experimentally and compared. An attempt was made to determine the optimum percentage of ethanol that gives lower emissions and better performance at the same time. This was done by using a simple fumigation technique. The results show that both the fumigation and blends methods have the same behavior in affecting performance and emissions, but the improvement in using the fumigation method was better than when using blends. The optimum percentage for ethanol fumigation is 20%. This percentage produces an increase of 7.5% in brake thermal efficiency, 55% in CO emissions, 36% in HC emissions and reduction of 51% in soot mass concentration. The optimum percentage for ethanol–diesel fuel blends is 15%. This produces an increase of 3.6% in brake thermal efficiency, 43.3% in CO emissions, 34% in HC and a reduction of 32% in soot mass concentration. © 1999 Elsevier Science Ltd. All rights reserved.

Keywords: Fumigation; Diesel emission; Blends; Spraying nozzle; Sampling probe

1. Introduction

It is apparent from the increasing popularity of light-duty diesel engines that alternative fuels, such as alcohols, must be applicable to diesel combustion if they are to contribute significantly as substitutes for petroleum-based fuels. However, in the past, little attention has

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been given to the utilization of alcohol fuels in compression ignition engines [1]. This is due to the difficulties encountered while attempting to use alcohols in diesel engines. The main difficulties are:

1. More alcohol fuel than diesel fuel is required by mass and volume [2].
2. Large percentages of alcohol do not mix with diesel fuel, hence use of diesel–alcohol blends is not feasible [3]. Also, the blends were not stable and separate in the presence of trace amounts of water [4].
3. Alcohols have extremely low cetane numbers, whereas the diesel engine is known to prefer high cetane number fuels (45–55) which auto-ignite easily and give small ignition delay [5].
4. Diesel fuels serve as lubricants for diesel engine. Alcohol fuels do not have the same lubricating qualities [2,4].
5. The poor auto-ignition capability of alcohols is responsible for severe knock due to rapid burning of vaporized alcohol [1,4] and combustion quenching caused by high latent heat of vaporization and subsequent charge cooling [4].

Although replacing diesel fuel entirely by alcohols is very difficult, an increased interest has emerged for the use of alcohols, and particularly lower alcohols (methanol and ethanol) with different amounts and different techniques in diesel engines as a dual fuel operation during recent years.

There are several techniques involving alcohol–diesel dual fuel operation. The ignition of alcohol in dual fuel operation is ensured by the high self-ignition diesel fuel. The most common methods for achieving dual fuel operation are [6]:

1. Alcohol fumigation — the addition of alcohols to the intake air charge, displacing up to 50% of diesel fuel demand.
2. Dual injection — separate injection systems for each fuel, displacing up to 90% of diesel fuel demand.
3. Alcohol–diesel fuel blend — mixture of the fuels just prior to injection, displacing up to 25% of diesel fuel demand.
4. Alcohol–diesel fuel emulsion — using an emulsifier to mix the fuels to prevent separation, displacing up to 25% diesel fuel demand.

The techniques we are concerned with in this study (the simplest) are alcohol fumigation and alcohol–diesel blends.

Fumigation is a method by which alcohol is introduced into the engine by carburetting, vaporizing or injecting the alcohol into the intake air stream. This requires the addition of a carburettor, vaporizer or injector, along with a separate fuel tank, lines and controls.

Fumigation has some following advantages:

1. It requires a minimum of modification to the engine, since alcohol injector is placed at the take air manifold. Also, flow control of the fuel can be managed by a simplified device and fuel supply system.
2. The alcohol fuel system is separate from the diesel system. This flexibility enables diesel engines, equipped with the fumigation system, to be operated with diesel fuel only. The engine can switch from dual fuel to diesel fuel operation and vice-versa by disconnection and connection of the alcohol source to the injector.

3. If an engine is limited in power output due to smoke emissions, fumigated ethanol could increase the power output because alcohol tends to reduce smoke. This is because of good mixing of the injected charge with alcohol.
4. Fumigation can substitute alcohol for diesel fuel. Up to 50% of the fuel energy can be derived from alcohol by fumigation [3].

The easiest method by which alcohols can be used in diesel engines is in the form of blends. For lower alcohols, this approach is limited to ethanol because methanol is not soluble or has very limited solubility in the diesel fuel [3].

An advantage of ethanol–diesel fuel solutions is that few major component changes are required for their use. Small adjustments to the injection timing and fuel delivery may be necessary to restore full power. The adjustments depend on the ethanol concentration and the combustion effects of ethanol [3,7]. In this study, no modification on the engine was made for blends, since the amounts used were within the permitted range.

Weidmann and Menard [8] used a standard Volkswagen 4-cylinder, swirl-chamber diesel engine to test the performance of alcohol–diesel fuel blends. The alcohols involved were ethanol and methanol. Their object was to report on the development of an engine/fuel concept designed for alcohol–diesel fuel blends. They reported that HC and CO emissions were increased and NO_x emissions decreased compared to diesel fuel. Also, alcohol–diesel fuel blends emit more aldehydes and less polycyclic aromatic hydrocarbons (PAH).

Czerwinski [9] tested a 4-cylinder, heavy duty, direct injection diesel engine in which 30% ethanol and 15% rape oil mixtures were used. He found that the addition of 30% ethanol to the diesel fuel causes longer ignition delay. The combustion temperatures were lower. At full load, all emissions were lower. At lower loads and speeds, CO and HC emissions were increased. It was possible to obtain emissions similar to diesel fuel, but with reduced power output up to 12.5%.

Concerned with alcohol fumigation, Broukhiyan and Lestz [4] applied ethanol fumigation to a 5.7 I, V-8, light duty, indirect injection diesel engine by using a pressurized nitrogen cylinder with secondary air supply in amounts up to 50% of the total fuel energy. Their research was undertaken to study the effect of ethanol fumigation on the performance (efficiency), combustion knock characteristics and exhaust emissions. For all conditions except the 1/4 rack setting (light load) condition, modest thermal efficiency gains were observed upon fumigation. However, engine roughness or the occurrence of severe knock limited the maximum amount of ethanol that could be fumigated. Brake specific NO_x concentrations were found to decrease for all conditions tested. While decreasing the mass of particulate emitted, ethanol fumigation enhanced the biological activity of that particulate.

Hayes et. al. [6] tested 100, 125, 150, 175 and 200 proof ethanol as fumigants in a 6-cylinder turbocharger diesel engine at 2400 rpm. Ethanol was injected directly into the intake ports. Six electronic fuel injectors were used, one for each cylinder. They found that the lower proofs reduced the maximum rate of pressure rise. Any proof at lower load reduced NO levels. HC emissions increased greatly as the ethanol substitution was increased without depending on the ethanol proof.

It can be seen from the literature survey reviewed earlier that there is a lack of information about comparison of alcohols fumigation with alcohols as blends with diesel fuel, and there is no detailed information about the effects of alcohols on diesel engine smoke or particulate matter emissions. Since the comparison between these two methods needs a base to start from our base was using ethanol percentages as a fraction of the diesel energy input at full rack setting. These percentages are to be supplied to the engine while keeping the total fuel energy constant as obtained from diesel fuel operation at full rack setting. This could be achieved through constructing an effective simple fumigation technique.

2. Experimental apparatus and test procedure

The engine used for this study was a single cylinder, four stroke, direct injection, variable compression ratio, diesel engine with a swept volume of 582 cm^3. The engine is naturally aspirated and water cooled. The engine was coupled to an electrical generator through which load was applied by increasing the field voltage. A fixed 20° injection timing and 18 compression ratio were used throughout the experiments. Indicators on the test bed show the following quantities which are measured electrically: engine speed, brake power and various temperatures

Fig. 1 shows a schematic of the ethanol fumigation system. Ethanol was fumigated into the intake air charge and introduced in the engine as a vapor or mist, dependent on the degree of vaporization which occurred.

A simple fumigation system was used, consisting of a single hole, direct opening configuration spraying nozzle. It was selected to achieve ethanol delivery at relatively low pressure. The nozzle has a diameter of about 0.25 mm. Since the obtained nozzle flow rate was relatively high, the produced ethanol jet was allowed to hit a partition in order to get ethanol

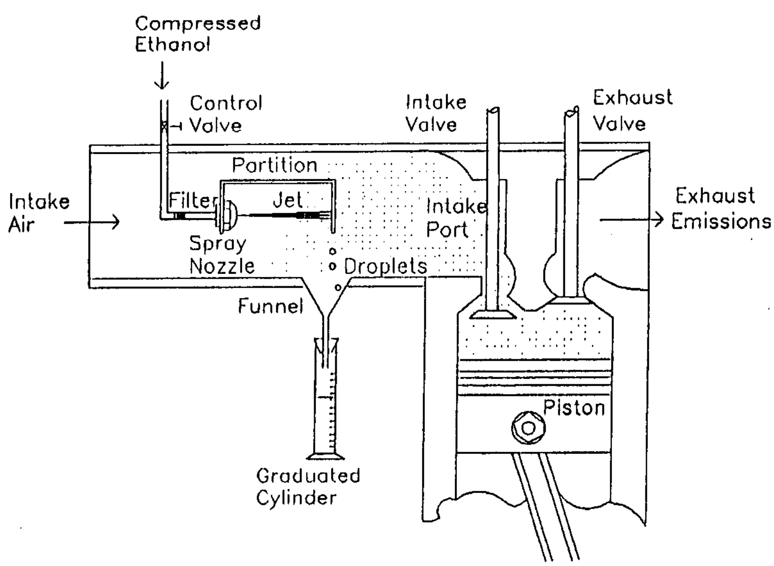

Fig. 1. Schematic diagram of ethanol fumigation system.

mist which is directly mixed with air before entering the engine. An electrically driven air compressor was used to supply ethanol to the nozzle.

The nozzle was positioned approximately 50 cm ahead of the inlet manifold. This allowed the ethanol to be mixed with the intake air for a sufficient period, providing uniform mixing. The intake manifold was provided with a transparent window for optical inspection of the ethanol–air mixture.

Particulate exhaust emissions were drawn using a special sampling probe. Particulates were collected on Teflon-coated glass fiber filters. The filters, manufactured by Pallflex Products Corporation (Type TX40 HI20-WW), measured 25 mm in diameter and were held in a 25 mm filter holder which is connected to the diffuser end by a copper elbow fitting.

An electrically driven air compressor was used to make a vacuum such that exhaust particulates can be extracted continuously from the main exhaust flow and collected on the filter. The flow through the filter was maintained nearly constant by using an orifice at the outlet of the sampler. This flow was measured by using an air flow meter.

The exhaust gas was sampled and analyzed using a 'Sun Gas Analyzer' (SGA1000). The gaseous pollutants treated in this study where CO, CO_2, and HC. The concentration of each gas is measured relative to the sample taken continuously and digitally.

Smoke samples were analyzed using a 'Sun Electric Advanced Smoke Analyzer' (ASA200), which is a compact unit for measuring the smoke density from the exhaust system of diesel powered engines. The Sun Electric ASA200 Smoke Meter uses a modulated light emitting diode (LED) as a light source and a solid state photodiode light receiver. The sophisticated electronics measure, the light level received through a portion of the exhaust smoke passing through the meter. The density of the smoke is then calculated, and a result is given in the form of the Smoke Absorption Coefficient (K).

In all tests the engine was allowed to warm up for 15 min to reach equilibrium. This was determined by monitoring the exhaust and coolant temperatures. The engine was tested at different rack settings (depending on the percent of substitution of the same energy input obtained from diesel fuel at full rack setting) and at different speeds. The rack settings used were always between 3/4 rack and full rack settings.

Since it was decided to determine the effect of substituting ethanol for diesel fuel while keeping the total fuel energy constant, as obtained from diesel fuel at full rack setting it was first necessary to determine the total fuel energy supplied for each condition within the test matrix used. This was done by running the engine at each test condition on the baseline fuel, Fractional ethanol energy flow rates of 5, 10 and 20% were chosen.

The filters used for particulate matter collection were prepared and weighed in the following manner: unloaded (clean) filters were backed in an oven for 1 h at 230°C, then they were weighed using a sensitive digital balance of ± 0.1 mg accuracy, which was calibrated before weighing. The samples were drawn through the filter for 1 min. The difference in mass between these two stages of weighing is the total mass of particulate matter.

In order to find the soot particles mass, these filters were backed again in the oven at 230°C for 1 h to remove the moisture and the volatile matters, then they were re-weighed. The difference in mass between these filters and the clean ones is the mass of soot particles.

3. Results and discussion

Fig. 2 shows the variation of brake thermal efficiency with introducing ethanol in both methods. The figure shows a slight improvement in the efficiency with increasing ethanol substitution. This improvement was approximately 7.5 and 5.4% over the entire speed range for ethanol fumigation and ethanol–diesel fuel blends, respectively. It can be seen from this figure that ethanol has the same effect at all speeds. The maximum increase in the efficiency was at 1500 rpm. The improvement in the thermal efficiency is attributed to the changes occurring in the combustion process. The physical and chemical differences in fuel structure of ethanol and diesel fuel lead to a combination of changes in the combustion process.

In general, the slight gains in thermal efficiency with increased ethanol substitution may be attributed to the increase in the ignition delay, so a rapid rate of energy is released which reduces the heat loss from the engine because there is not enough time for this heat to leave the cylinder through heat transfer to the coolant.

The blended fuels compared least favorably with fumigation at all speeds and approach the diesel fuel in some cases. This could be due to the following reasons:

- The physical properties of diesel fuel are changed when ethanol is added in solution (blend). The addition of ethanol causes the viscosity of diesel fuel to decrease. Also, the addition of ethanol in solutions with diesel fuel causes the cetane rating to drop and the heating values to be lower.
- Evaporation of ethanol in the intake air (fumigation case) lowers the intake mixture

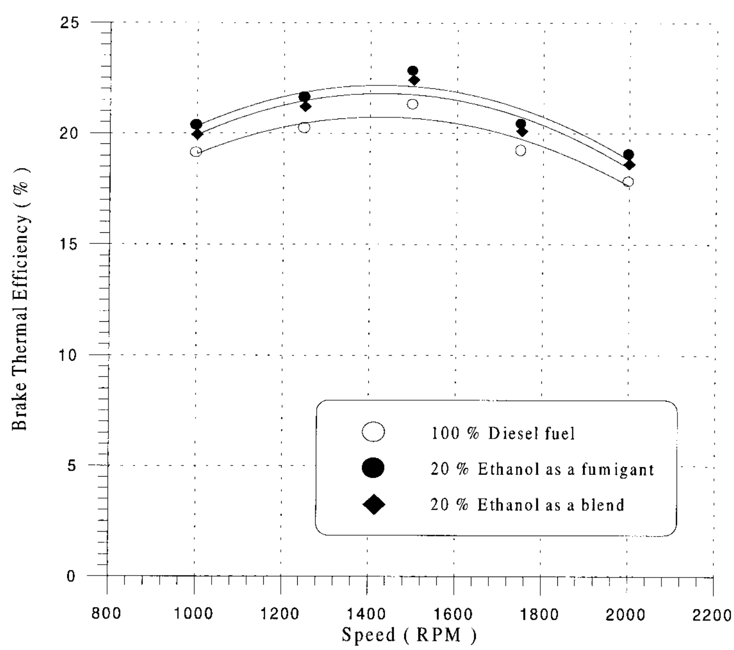

Fig. 2. Brake thermal efficiency versus speed for ethanol fumigation and for fuel blends.

temperature and increases it's density. Thus, as more air is made available in the cylinder, greater amounts of power can be generated if the right proportion of fuel is added.

Figs. 3 and 4 show the effect of ethanol substitution on CO and HC production, respectively. The maximum increase in CO and HC emissions was at 20% ethanol for both the fumigation and blends methods. Also, the CO and HC emissions were always higher when using the blended fuels than when the engine operated with fumigation.

For 20% ethanol fumigation, the increase in CO emissions was in the range of 21–55% at the speed range used, and for 20% ethanol as a blend with diesel fuel, the increase in CO emissions was in the range of 28–71.5% at the same speed range used.

The increase in the CO levels with increasing ethanol substitution is a result of incomplete combustion of the ethanol–air mixture. Factors causing combustion deterioration (such as high latent heats of vaporization) could be responsible for the increased CO production. Combustion temperatures may have had a significant effect. A thickened quench layer created by the cooling effect of vaporizing alcohol could have played a major role in the increased CO production.

Another reason for the increasing CO production is the increase in ignition delay. This could lead to a lower temperatures throughout the cycle. This results in combustion of a proportion of the fuel in the expansion stroke, which lowers temperatures and reduces the CO oxidation reaction rate.

The produced emissions of CO from fumigation were less than for blends. Combustion temperatures for fumigation may be higher than for blends, better air utilization due to the presence of a homogeneous ethanol charge and lower effect of previous reasons when applying

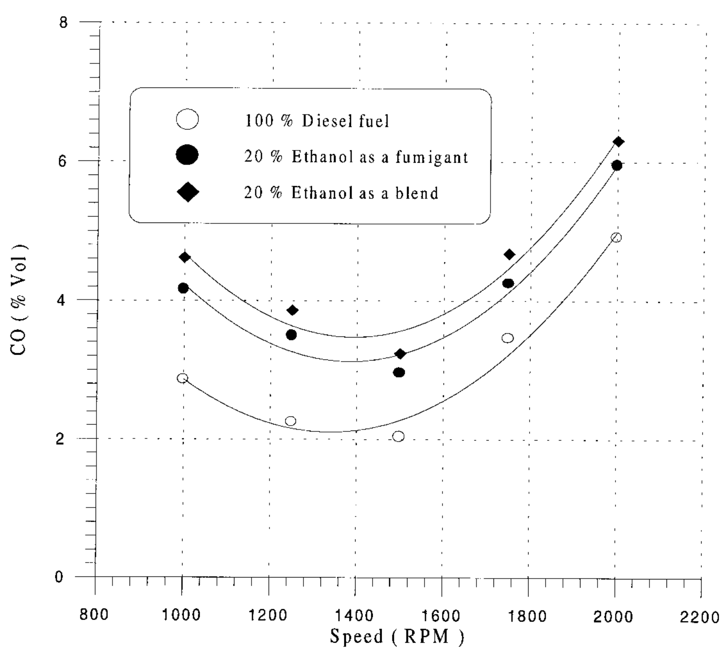

Fig. 3. CO emissions versus speed for ethanol fumigation and for fuel blends.

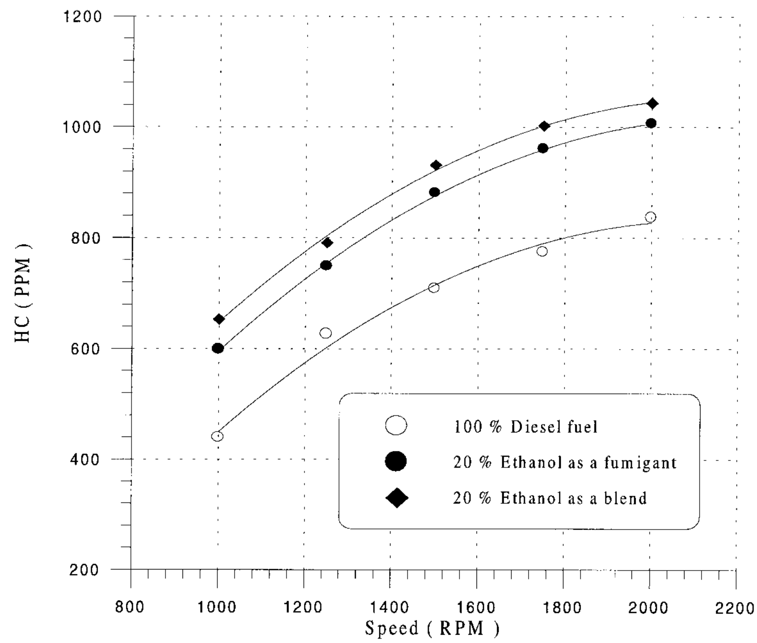

Fig. 4. HC emissions versus speed for ethanol fumigation and for fuels blends.

ethanol fumigation may have lowered the CO emissions for fumigation compared with blends. Also, the minimum increase in CO emissions was at the higher speeds because of more turbulence in the cylinder (effective mixing) and relatively high combustion temperatures compared with the lower speeds.

Fig. 4 shows the effect of ethanol substitution on HC emissions. For 20% ethanol fumigation, the increases in HC emissions were between 20 and 36%, and for 20% ethanol as a blend with diesel fuel, the increases were between 25 and 49% over the entire speed range. IT is noticed that there is a resemblance in the results concerning CO and HC emissions production.

The HC emissions tend to increase because of the quench layer of unburned fumigated ethanol present during fumigation. There is no quench layer with diesel fuel injection alone because the combustion is droplet-diffusion-controlled and completely surrounded by air. Also, the high latent heat of vaporization can produce slow vaporization and mixing of fuel and air. These factors result in high HC levels.

Fig. 5 shows the effect of ethanol substitution on engine smoke. The smoke measurements were plotted as a smoke absorption coefficient. This is a number which gives an indication about the exhaust emissions density.

There is a decrease in smoke coefficient of 30–48% for 20% ethanol as a fumigant, decreases of 11–25% for 20% ethanol as a blend with diesel fuel and a decrease of 18.5–33.3% for 15% ethanol as blend. This decrease was the maximum over the entire speed range used. It appears that the optimum percentage of ethanol for smoke reduction are 20 and 15% for the fumigation and blends methods, respectively.

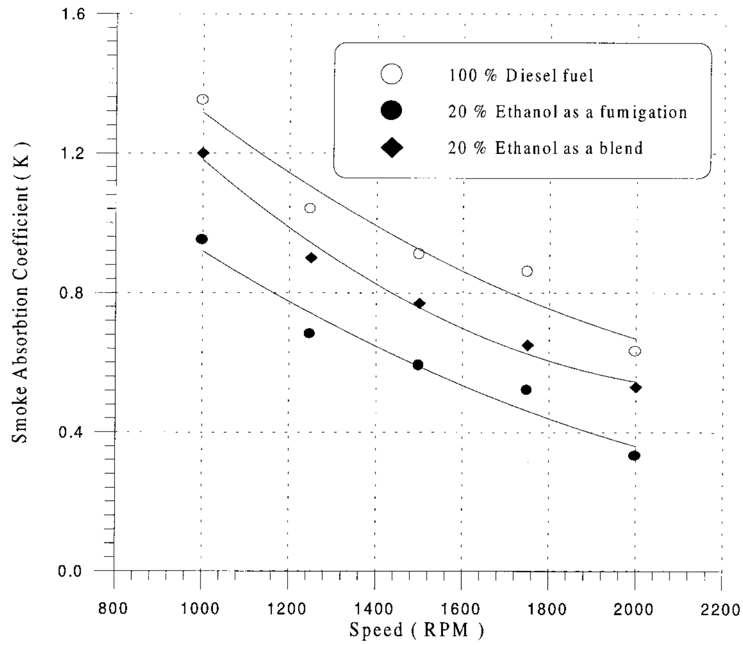

Fig. 5. Engine smoke versus speed for ethanol fumigation and for fuels blends.

The recognized drastic reduction in smoke coefficient, as more amount of ethanol were used, is attributed to several reasons. Here, the charge cooling increases ignition delay and, thus, enhances the mixing of diesel fuel with the ethanol–air mixture which, in turn, makes for better air utilization and less smoke. Also, diesel fuel has a high tendency to soot formation due to its low H/C ratio and the nature of its combustion process. Using ethanol, either as a blend or as a fumigant in a diesel engine, increases the hydrogen content in the mixture and eventually reduces the engine smoke and leads to a soot free combustion of ethanol under normal diesel engine operating conditions.

The soot mass concentrations for ethanol fumigation, diesel fuel blends and diesel fuel operation are shown in Fig. 6. From this figure, it can be seen that there is a matching between smoke and soot measurements, and both methods confirm each other. Soot concentration represents the mass fraction of soot in the exhaust. It is given in milligrams of soot per kilogram of exhaust.

From Fig. 6, the maximum decrease (over the entire speed range) in soot concentrations was 33–51% for 20% ethanol fumigation, 10–22% for 20% ethanol blends and 15–32.5% for 15% ethanol–diesel fuel blends. Soot formation, when applying ethanol in both methods, shows a strong dependence on the amounts of ethanol used. The decrease in soot formation rate could be attributed to the same reasons responsible for the smoke decrease.

From the results discussed previously, it is apparent that using ethanol in diesel engines causes increased HC and CO emissions and reduces particulate or smoke. The emissions of CO and HC can be a strong limiting factor in the proportion of ethanol that can be used. Though these emissions are increased, it should be kept in mind that in many cases, they are quite low

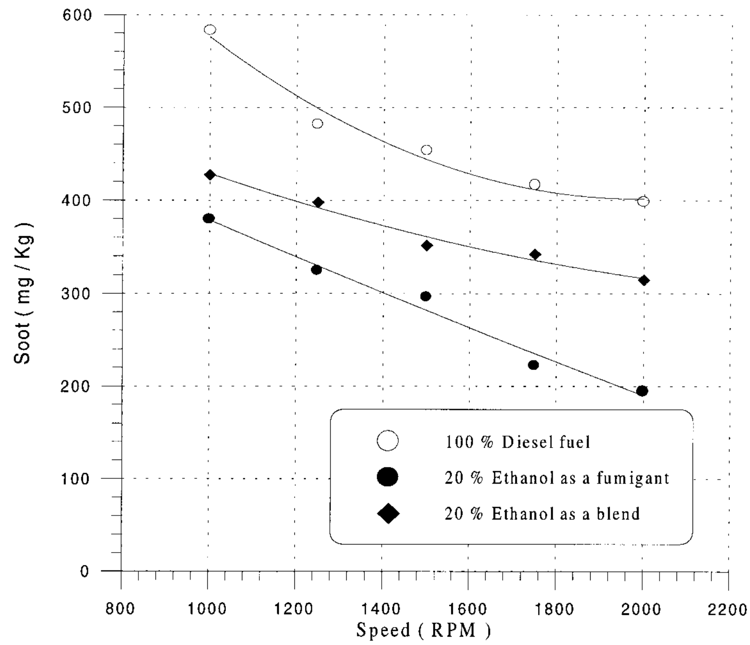

Fig. 6. Soot emissions versus speed for ethanol fumigation and for fuels blends.

to start with and small increases may be entirely acceptable. Also, ethanol fumigation of diesel engines holds the promise of reducing smoke from the current generation of diesel engines to which this simple method is being applied.

4. Conclusions

This work was undertaken to study and compare the effects of ethanol fumigation and ethanol–diesel fuel blends on the performance (efficiency) and exhaust emissions of a diesel engine. This was achieved by using a simple fumigation technique and a new method of introducing ethanol percentages. The conclusions which may be drawn from this study are as follows:

- In all cases, the use of ethanol as a blend with diesel fuel compared least favorably with fumigation, and the use of such fumigation technique is effective and gives reasonable results.
- Based on the above results, the optimum percentage of ethanol appears to be 20 and 15% for ethanol fumigation and ethanol–diesel fuel blends operations, respectively. The use of 20% ethanol as a fumigant can produce an increase of 7.5% in the brake thermal efficiency, 55% in CO emissions levels and 36% in HC emissions levels. Also, this fumigation percentage produces a decrease of 48% in engine smoke and 51% in soot mass concentration.

- The use of 15% ethanol as a blend with diesel fuel can produce an increase of 3.6% in brake thermal efficiency, 43.4% in CO emissions and 34.2% in HC emissions. It can also produce a reduction of 33.3% in engine smoke and 32.5% in the soot mass concentration.

The maximum increases and decreases mentioned in the above results are over the entire speed range chosen.

References

[1] Heisey JB, Lestz SS. Aqueous alcohol fumigation of a single-cylinder DI diesel engine, SAE Paper No. 811208, 1981.
[2] Doann H-A. Alcohol fuels. Boulder, CO: Westview Press, 1982.
[3] Eugene EE, Bechtold RL, Timbario TJ, McCallum PW. State-of-the-art report on the use of alcohols in diesel engines, SAE Paper No. 840118, 1984.
[4] Broukhiyan EMH, Lestz SS. Ethanol fumigation of a light duty automotive diesel engine, SAE Paper No. 811209, 1981.
[5] Baranescu RA. Fumigation of alcohols in multi-cylinder diesel engine-evaluation of potential, SAE Paper No. 860308, 1986.
[6] Hayes TK, Savage LD, White RA, Sorenson SC. The effect of fumigation of different ethanol proofs on a turbo-charged diesel engine, SAE Paper No. 880497, 1988.
[7] Jiang J, Ottikkutti P, Gerpen JV, Meter DV. The effect of alcohol fumigation on diesel flame temperature and emissions, SAE Paper No. 900386, 1990.
[8] Weidmann K, Menard H. Fleet test, performance and emissions of diesel engine using different alcohol fuel blends, SAE Paper No. 841331, 1984.
[9] Czerwinski J. Performance of HD-DI-diesel engine with addition of ethanol and rapeseed oil, SAE Paper No. 940545, 1994.