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Investigations on effects of ethanol blending on performance and combustion characteristics of gasoline fuelled lean burn SI engine

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Abstract. This study discusses the effects of addition of small amount of ethanol on performance and emission behaviour of a gasoline fuelled lean burn SI engine. Experiments were carried out in a single cylinder diesel engine which is made to operate as SI engine on lean condition with gasoline as a fuel. The engine was operated at wide open throttle at a compression ratio of 10.5:1 and 1500 rpm at diverse equivalent ratios by injecting the fuel into manifold. The test outcomes stipulated that use of ethanol-gasoline blend (10% by volume) was better compared with pure gasoline. The test results also revealed that there was an accountable increase in brake power output, brake thermal efficiency and an apparent extension in the lean limit of operation with ethanol-gasoline blend. In emission front, reduction in hydrocarbon and carbon monoxide emissions were observed. There was an increase in carbon dioxide and NO emissions due to better combustion and high in-cylinder temperature. On the whole it is concluded that small ethanol addition to gasoline improves performance and reduces emission for lean operating SI engine.

1. Introduction

In challenges to concerns of limited fossil-fuel resources, increasing global air pollution and energy security compels the researchers go for more and more alternative cleaner energy sources. It also emphasizes the need to study the technological alternatives to provide fresh look towards energy supply and environmental protection [1] and [2]. For this purpose, using alternate fuels, like alcohols in gasoline engines have a great possibility to decrease the reliance on fossil fuels, exhaust emissions [3]. Among the alcohols, ethanol has been recognized as having the ability to increase the quality of air when used as an additive on gasoline engines because of its good knock resistance characteristics, high latent heat of vaporization, high oxygen content, high flame propagation speed and reduce the HC and CO emissions. The properties of gasoline and alcohol shown in Table 1. Presently low concentrations of Ethanol can be used in SI engines without any modifications. SI engines can be fuelled by clean ethanol (99.9%) but it necessities some changes to the engine [4], [5] and [6].

Quite a lot of works have been conducted to compare the performance characteristics of alcohol blends (ethanol) with gasoline for stoichiometric combustion mode. Gravalos. I. et al. [7] worked on non- road engine with five different kinds of lower and higher alcohols–gasoline mixture test fuels ranging from 10-30%. The alcohol blend consists of five different alcohols. The test outcomes show that exhaust emissions of un-burnt hydrocarbon and carbon monoxide from the lower and higher molecular mass alcohol-gasoline blend are lesser than emission from the gasoline and drop being

higher with the higher fraction of ethanol in the mixture. Al.Hasan [8] investigates performance and emission characteristics of 10 test unleaded gasoline and ethanol blends starting from 0 to 25% with an addition of 2.5%.The test outcomes indicated that 20% ethanol mixture gave good results in terms of performance and emissions. Alok kumar et al. [9] studied the possible problems with higher ethanol/gasoline blends on a single cylinder SI engine with 10%, 30% and 70% ethanol and gasoline blends. The experimental results shows that performances decreased by 10-14% with 70% ethanol blend at all operating conditions compared to neat gasoline while at optimized operating conditions the performance was same as neat gasoline and reduced emissions. Venugopal Thangavel et al. [10] performed series of engine tests by developing the dual fuel injection system real time controller to allow the ethanol and gasoline into intake port of a single cylinder SI engine. The results showed that efficiency and torque are better for 50% blend due to the faster combustion as a result of better mixing of a fuel as compared to conventional blending. John M. Storey et al. [11] using the turbo charged direct injection spark-ignition (DISI) gasoline engine investigated the performance of ethanol gasoline mixtures. The highest percentage of ethanol handled was 20%.The results shows that as ethanol blend increases Particulate matter emissions decreases in both FTP and US06 cycles. Hu¨seyin Serdar Yu¨cesu et al. [12] investigated the effect of gasoline-ethanol blend and compression ratio at stoichiometric air fuel ratio under different speeds on performance of single cylinder four stroke SI engine. The results indicated that brake specific fuel consumption was improved for the blend E60 and torque output increased with compression ratio along with 11% and 10.8% average reduction of CO and HC emissions for the blends Ethanol 40% and 60% blends at a speed of 2000 rpm. Tayler et al. [13] examined experimentally the outcomes of the alcohol and gasoline blend on emissions. The alcohols included were ethanol, methanol, i-propanol and n-propanol by different mass ratio of oxygen. While using alcohol and gasoline blend, the reduction of CO and HC emissions were observed by 40% and 75% respectively. Wu et al. [14] conducted the experiments to know the effects of air fuel ratio on performance and emission characteristics of SI engine using gasoline-ethanol blends. The experimental result shows that torque output of engine increases without appreciable increase in the BSHC. With the increment in ethanol content CO and HC emissions were reduced. The smallest amount of CO_2 was obtained with blend E30; finally they concluded that 30% ethanol blend can reduce emissions effectively. Palmer [15] conducted a series of experiments using different blend rates of gasoline-ethanol fuel in engine testing. The results stipulated that 10% Ethanol blend to gasoline improves 5% power output and 5% improvement in octane number for every 10% ethanol blend addition. Taljaard et al. [16] conducted a study on effect of oxygenates in gasoline on the performance and emission characteristics of SI engine at stoichiometric condition. Finally they accomplished the results that oxygenates reduce CO, HC and NOx emissions significantly. El-kassaby [17] conducted a chain of tests by using ethanol-gasoline blends up to 40% at different compression ratios to know the SI engine performance. The results indicated that, engine indicated power improves with ethanol blend and maximum was observed for 10% gasoline and ethanol blends.

From the literature, several researchers have performed experiments related to the consequences of alcohol-gasoline blend performance and emission characteristics of SI engine at stoichiometric and rich conditions. The prime intension of current research is to analyze the aftermath of manifold injection technique and 10% ethanol blend in gasoline on SI engine performance and emission behaviour under lean conditions. Moreover in the present study, tests were executed using a dedicated gasoline injection controller unit developed by authors.

2. Materials and method
In this work, the alcohol-gasoline mixture was provided on volume basis. Ethanol with concentration of 10% was blend with 90% neat gasoline; this blend was called as E10. The fuel blend was prepared just before the test to make sure that the alcohol-gasoline mixture is uniform and no water is created by the reaction of alcohol with water vapour in the ambiance. Experiments were performed on the 4-stroke, air cooled, single cylinder SI engine with displacement volume 661cc with a compression ratio 10.5 and power of 4.4kW at 1500 rpm. The detailed specifications of engine are given in table 2.

Table 1. Properties of the fuels [9] [10] [11] [12].

Properties	Gasoline	Ethanol
Chemical structure	C_4 to C_{12} (C_7H_{17})	CH_3–CH_2–OH
Physical state	liquid	Liquid
Molecular weight	95-120	46
Density (kg.m^3 at 15^{0}C)	737	794
Oxygen w/%	0	34.8
Stoichiometric air fuel ratio (kg . kg^{-1})	14.7:1	9:1
Lower heat value (kJ . kg^{-1})	43,440	26,770
Auto-ignition temperature (^{o}C)	371	465
Latent heat (kJ . kg^{-1})	310	904
Research Octane number	86-94	108
Flammability limits Vol %	1.4-7.6	4.3-19
Flame velocity (cm.s^{-1})	33	39
Boiling point (^{0}C)	30-200	78.5

Table 2. Engine specifications.

Type	Four stroke, air cooled, single cylinder, overhead valve, compression ignition engine modified to run in the SI mode
Make	Kirloskar TAF1
Fuel	LPG, Gasoline, Ethanol
Number of cylinders	One
Bore x stroke	87.5 x 110 mm
Displacement volume	661.5 cc
Compression ratio	10.5:1
Connecting rod length	231 mm
Rated power	4.4 kW @ 1500 rpm
Valve timing	Inlet valve opening : 4.5° before Top Dead Centre
	Inlet valve closing :35.5° after Bottom Dead Centre
	Exhaust valve opening : 35.5° before Bottom Dead Centre
	Exhaust valve closing : 4.5° after Top Dead Centre

Basically the engine was altered from the Kirloskar TAF1 diesel engine by modifying the compression ratio, incorporating the ignition system and spark plug and replacing the fuel injection pump and the injector with carburetor system. The overall system made favourable for gasoline operation in SI mode. The simplified diagram of investigational set up is shown in figure 1.

The spark timing could be varied sharply by 1° crank angle from 280° to 360° (0° to 80° before TDC) through an electronic control unit based circuit with an input from proximity sensor. The compression ratio was changed by modifying the clearance region dimensions and it was fixed at 10.5:1. Shim plates of standard thickness were placed between crankcase and cylinder head to compensate the required clearance.

The intake manifold specially designed to accommodate an electrically operated gasoline injector in such a position that contact of spray with wall is minimal. A dedicated gasoline injection circuit consisting of microcontroller ATMEGA 8A was developed to generate the pulses for the correct duration to energize the injector by taking input from the crank angle encoder. The pulse width can be controlled within 1ms. The injector works on a pressure range 0-3 bar, voltage of 10-12 V and it was fed by fuel feed pump.
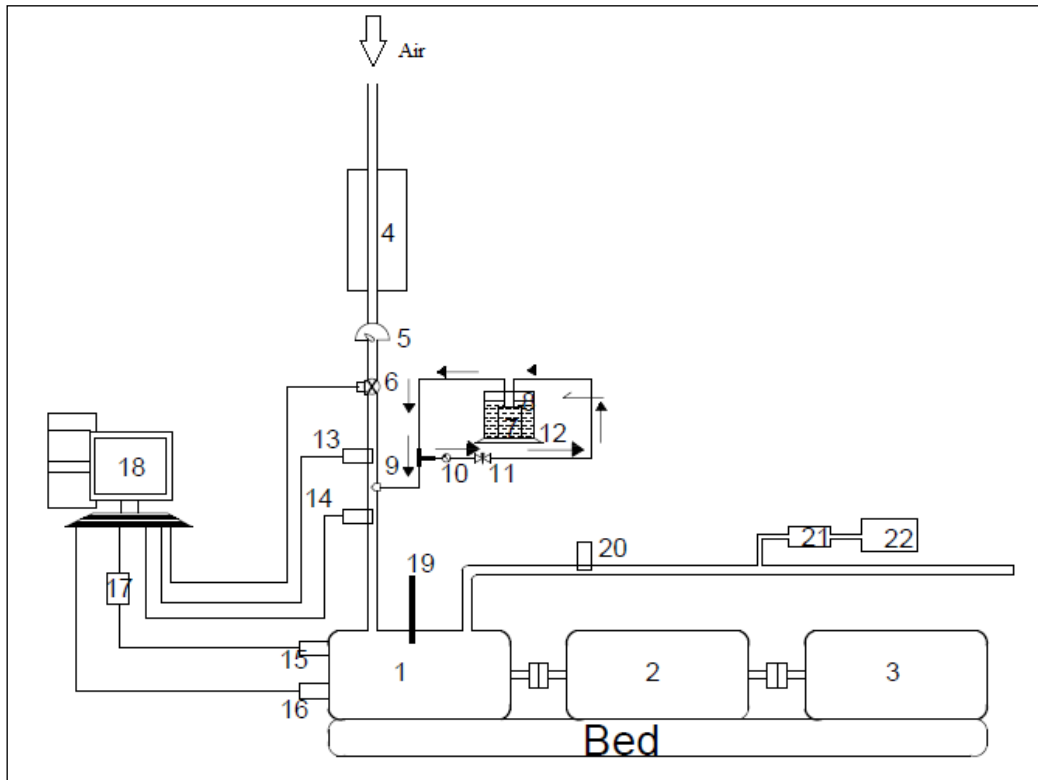
Figure 1. Layout of the experimental setup.
1. Engine 2. Eddy current dynamometer 3. Motoring unit 4.Surge tank for air 5.Air flow meter 6.Throttle valve 7.Fuel pump 8.Fuel tank 9.Fuel injector 10.pressure gauge 11.fuel control valve 12.weightpan 13.Intake air temperature sensor 14.Intake manifold pressure sensor 15.Incylinder pressure sensor 16.Crank angle encoder sensor 17.Charge amplifier 18.Computer 19.Spark plug 20.Exhaust gas temperature sensor 21.Ice path 22. 5 – Gas analyzer

The engine was loaded by water cooled eddy current dynamometer system which was conditioned to operate constantly at 1500 rpm. The throttle position sensor was connected to a digital display in order to vary the exact throttle opening position exactly. The exhaust gas was analyzed with five gas analyzer.

2.1. Experimental conditions
Experiments were conducted at constant throttle opening condition at 100% i.e. Wide Throttle Opening (WOT) based on the ECU controller module connected to a throttle position sensor fixed in the intake manifold. The gasoline fuel pump was immersed in the fuel tank and it is maintained at a pressure of 3 bars with the ball valve present in the return line as shown in figure 1. A positive displacement type air flow meter was used for measuring the flow rate of air. A surge tank was coupled on to the intake side to minimize the airflow variations. The flow rate of the fuel was measured with the help of the weight pan on mass basis. The air and fuel flow rates were used in calculation of equivalence ratio.

The air-fuel ratio was adjusted from leanest to rich by adjusting the amount of fuel fed to the injector by varying pulse width module through microcontroller, based on the input from the crank angle encoder. The leanest mixture (lowest equivalence ratio) limit was limited by the unstable operation of the engine. At each load MBT (Maximum Brake Torque) spark timing was maintained. Measurements were for all time taken after the engine attains a steadiness of operation. Torque, engine speed, MBT timing, airflow rate, fuel flow rate, pulse width modulation, intake and exhaust temperature, HC, CO, CO_2 and NO exhaust emission levels were recorded at each equivalence ratio.

3. Results and discussion

All the results were obtained at steady speed of 1500 rpm and at throttle opening position of 100%. The load was varied based on the fuel inlet from rich to leanest possible limit. The performance and emission characteristics were compared between pure gasoline and E10 blend.

3.1. Performance parameters

The variations of brake power with equivalence ratio are shown in figure 2 for E0 and E10 blend. At an equivalence ratio of 0.55, the brake power output was 2.1 kW for E10 which was 76% higher than that of E0 which was about 1.3 kW. However, at higher brake thermal efficiency region of 0.71, the brake power output for E10 was 4.3 kW which was higher than E0 which was 3.9 kW signifying an increase of 9%.The increment in brake power with blend E10 was due to the 'cooling effect' of ethanol. As observed in table 1, the latent heat of ethanol is significantly more compared with gasoline.
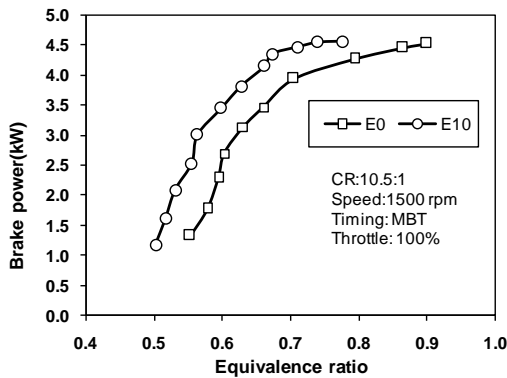
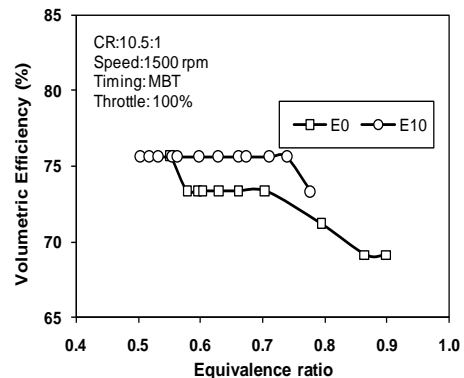**Figure 2.** Variation of brake power.

Figure 3. Variation of volumetric efficiency

The addition of the ethanol to gasoline causes an increment in latent heat for the blend compared with gasoline, which results in a cooler charge and hence denser charge entering into engine. Moreover, high flame speed of ethanol increases the combustion rate and enhances the power output significantly. The 'cooling effect' of ethanol clearly states that with E10 blend engine should have enhanced breathing capacity (volumetric efficiency) when compared to that of engine with E0 as shown in figure 3.The volumetric efficiency was improved at a range of 3 to 5% with E10 blend compare to E0 at all equivalence ratios. Figure 4 shows the variation of brake thermal efficiency with equivalence ratio for E0 and E10 blend.
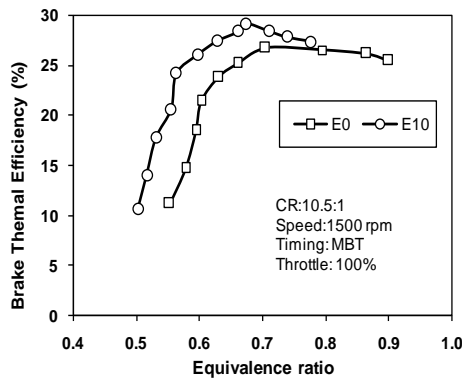
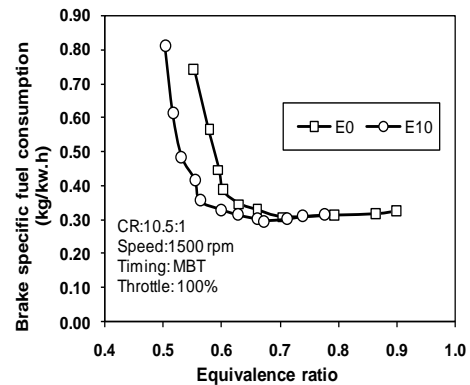**Figure 4.** Variation of brake thermal efficiency.

Figure 5. Variation of brake specific fuel consumption.

The peak brake thermal efficiency for E10 blend was found to be 28.5% at an equivalence ratio 0.71 and was about 26.8% for E0 blend. This shows an improvement of 7% high brake thermal efficiency for E10 blend. The improvement in brake thermal efficiency with E10 blend is due to its high latent heat vaporization compared with the E0.The large value of latent heat of vaporization increases the quantity of fuel vapour in the working charge during compression stroke which leads to homogeneity of the fuel – air mixture and causes the combustion efficiently and reduces the heat loss to cylinder walls due to incomplete combustion. It can be also observed that the blend E10 extends the lean burn limit from an equivalence ratio 0.55 to 0.50 signifying slight improvement in fuel consumption with E10 blend as shown in figure 5.

3.2. Emission Parameters
Figure 6 shows the variation of HC emissions with equivalence ratio for E0 and E10 blend. The HC emissions, within the stable limit of combustion on the leaner side of 0.71 equivalence ratios for the E10 blend were less compared to E0. The rise in HC emission was abrupt at lower equivalence ratios due to misfiring. The rise in HC emissions for E10 slightly higher compared with E0.It is due to addition of ethanol with base gasoline, which increases the oxygen concentration in the blend leading to cause more misfiring for E10 blend compared with E0 at lower lean limits between 0.50 and 0.71.
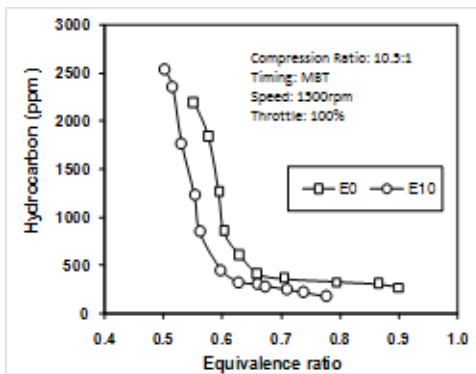

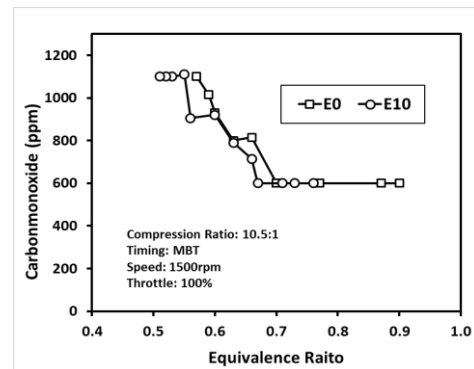**Figure 6.** Variation of hydrocarbon emission.

Figure 7. Variation of carbon monoxide emission.

The decreasing trend of HC emissions towards rich limit is primarily due to better combustion. Figure 7 shows that at an equivalence ratio 0.71 the CO emissions were 0.06% and 0.08% for E10 and E0 respectively. This can be also observed from Figure 8 which shows the trend of CO_2 emissions of blend E10 which was higher because of complete combustion.
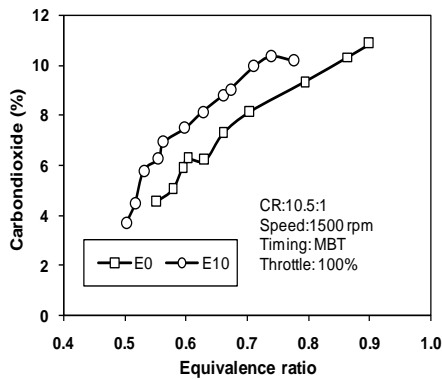

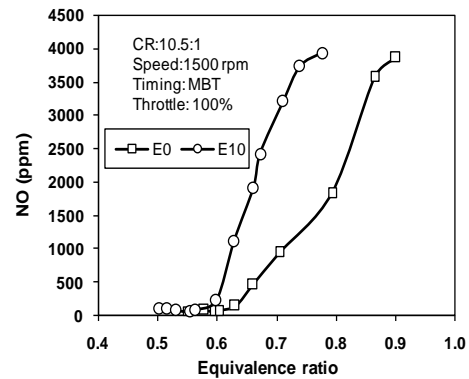**Figure 8.** Variation of carbon dioxide emission.

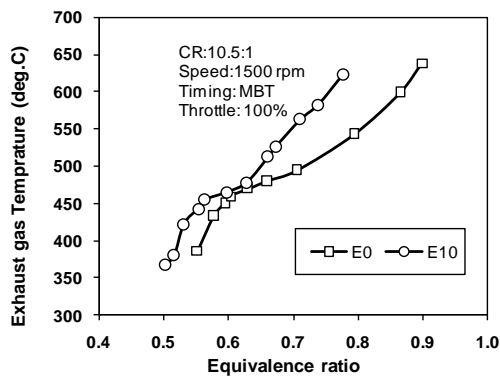

Figure 9. Variation of nitric oxide emission.

Figure 10. Variation of exhaust gas temperature.

The NO emission comparison from figure 9 clearly indicates that there was an increase of 59% at an equivalence ratio 0.71 because of higher combustion rate, this can be also be inferred from Figure 10 that higher exhaust gas temperature occurs for E10.Hence, it can be stated that E10 blend reduce CO and HC emissions and increases the NO emissions significantly.

4. Conclusion

An electronically controlled manifold injection system was successfully developed and tested; comparison of performance and emission characteristics of ethanol-gasoline blend E10 with pure gasoline E0 was also done. Based on the experimentation the following outcomes can be arrived.

- There was a significant improvement in brake power output and brake thermal efficiency with E10 as compared to the E0.There was a improvement of 10% brake power output and 7% brake thermal efficiency and 5% volumetric efficiency at higher loads. The maximum brake thermal efficiency under lean operating range reduces slightly brake specific fuel consumption as well.
- Lean extension limit was possible from an equivalence ratio 0.55 using E0 to an equivalence ratio of 0.50, about 5% increase by using E10 blend.
- The CO emission was less for E10 blend as compared to E0 and also the decrease in HC emissions was observed.
- The NO emissions were 59% high for E10 blend as compared to E0, due to high cylinder temperatures and enhanced combustion.

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