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EFFECT OF GASOLINE - ETHANOL BLENDS ON PERFORMANCE AND EMISSION CHARACTERISTICS OF A SINGLE CYLINDER AIR COOLED MOTOR BIKE SI ENGINE

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Abstract

This paper investigates the effect of using gasoline-ethanol (GE) blends on performance and exhaust emission of a four stroke 150 cc single cylinder air cooled spark ignition (SI) engine, without any modifications. Experiments were conducted at part load and different engine speeds ranging from 3000 to 5000 rpm, without and with catalytic converter. Ethanol content was varied from 5 percentage to 20 percentage by volume and four different blends (E5, E10, E15 and E20) were tested. Fuel consumption, engine speed, air fuel ratio, exhaust gas temperature and exhaust emissions were measured during each experiment. Brake thermal efficiency ($\eta_{b,th}$), volumetric efficiency (η_{vol}), brake specific fuel consumption (BSFC) and excess air factor were calculated for each test run. Brake specific fuel consumption, volumetric efficiency and excess air factor increased with ethanol percentage in the blend. Carbon monoxide (CO), hydrocarbon (HC) and oxides of nitrogen (NO_x) emissions decreased with blends.

Keywords: Gasoline–Ethanol Blends, Air cooled SI engine, Performance, Emissions.

1. Introduction

Fuel additives are very important, since many of these additives when added to fuel improve engine's performance and emissions. One of the most important additive to improve combustion efficiency is oxygenates (oxygen containing organic compounds) which contribute oxygen for combustion of air-fuel mixture. Several oxygenates have been used as fuel additives, such as methanol, ethanol, tertiary butyl alcohol and methyl tertiary butyl ether. As a fuel for spark-ignition engines, alcohol has some advantages over gasoline, such as better anti-knock characteristics (higher octane number) and reduced CO and HC emissions. The

Nomenclatures

$(AFR)_{act}$	Actual air-fuel ratio of fuel blend
$(AFR)_{st,b}$	Stoichiometric air-fuel ratio of fuel blend
$(AFR)_{st,i}$	Molar stoichiometric air-fuel ratio of fuel blend
$(LHV)_b$	Lower heating value of fuel blend, kJ kg^{-1}
$(LHV)_i$	Lower heating value of given component in fuel blend, kJ kg^{-1}
$\dot{m}_a$	Air mass flow rate, kg h^{-1}
$\dot{m}_f$	Mass flow rate of fuel, kg h^{-1}
N	Engine speed, rpm
P	Atmospheric pressure, kPa
Q	Volume flow of fuel, cm^3
R	Air constant, kJ kg^{-1} K^{-1}
T	Engine torque, N m
T_a	Ambient temperature, K
t	Time required to consume 10 cm^3 of fuel, s
V_s	Swept volume of engine, m^3
v_i	Volume fraction of given component in fuel blend, vol.%

Greek Symbols

$\eta_{b,th}$	Brake thermal efficiency
η_{vol}	Volumetric efficiency, %
ρ_i	Density of given component in fuel blend, kg m^{-3}
ρ_b	Density of fuel blend, kg m^{-3}
λ	Excess air factor

Abbreviations

BP	Brake power, kW
BSFC	Brake specific fuel consumption, kg kW^{-1} h^{-1}
BSEC	Brake specific energy consumption, kJ kW^{-1} h^{-1}

latent heat of evaporation of alcohol is higher than that of gasoline, which makes the temperature of the intake manifold lower and increases the volumetric efficiency. But the same property may present cold-start problems. The stoichiometric air–fuel ratio of alcohol is lower than that of gasoline and hence the required amount of air for complete combustion is lower for alcohol [1].

Although alcohol has these advantages, it suffers from some drawbacks. It may cause the blended fuel to contain water [2], and further result in corrosion problems on the mechanical components, especially on the components made of copper, brass or aluminium. To minimize this problem on fuel delivery system, such materials mentioned above must be avoided [3]. Alcohol can react with rubber parts and cause a jam in the fuel pipe. Therefore, it is advised to use fluorocarbon as a replacement for rubber [4]. The heating value of alcohol is also lower than that of gasoline which calls for more alcohol fuel to achieve the same power output.

Presently, ethanol is regarded as most suitable substitute fuel for petroleum based fuels for use in automobiles. The main reason for advocating ethanol is that it can be manufactured from natural products or waste materials, whereas gasoline is produced from non-renewable natural resources. In addition, ethanol shows good anti-knock characteristics as evident from its octane number, shown in Table 1. However, economic reasons still limit its usage on a large scale. Currently, instead of pure ethanol, a blend of ethanol and gasoline is considered as more attractive fuel with good anti-knock characteristics.

Table 1. Fuel properties.

Property	Gasoline	Ethanol
Formula (liquid)	C_8H_{18}	C_2H_6O
Molecular weight (kg / kmol)	114.15	46.07
Density (kg/m^3)	765	785
Heat of vaporization (kJ/kg)	305	840
Specific heat (kJ/kgK) Liquid	2.4	1.7
Specific heat (kJ/kgK) Vapour	2.5	1.93
Lower Heating Value (kJ/kg)	44,000	26,900
Stoichiometric air–fuel ratio by mass	14.6	9.00
Research Octane Number	92	108.6
Motor Octane Number	85	89.7
Enthalpy of formation (MJ/kmol) Liquid	259.28	224.10
Enthalpy of formation (MJ/kmol) Gas	277.0	234.6

Engine tests with various blend ratios of ethanol in gasoline have shown that E10 increases the engine power by 5 percent and the octane number is increased by 5 percent for each 10 percent ethanol added. Under various compression ratios of the engine, the optimum blend was found to be E10 [5-7]. Another set of experiments with E5, E10 and E15 blends has shown that the best performance was achieved with E5 by increasing thermal efficiency by 4 percent under low speed conditions and 20 percent at the high speed conditions [8].

The effect of gasoline-ethanol blends on exhaust emissions has been experimentally investigated by different researchers. It is reported that the concentration of CO was reduced by about 40–50 percent at the lean side near stoichiometry. Also, the concentration of CO decreased as the percent of ethanol increased in the blend. It has been observed that oxygenates significantly decrease the CO, NO_x and HC emissions at the stoichiometric air–fuel ratio [9, 10]. From another set of experiments with three-fourth throttle opening position and variable engine speed ranging from 1000 to 4000 rpm, it is shown that gasoline-ethanol blends increase the brake power, torque, volumetric and brake thermal efficiencies and decrease brake specific fuel consumption and equivalence air–fuel ratio. CO and HC emissions concentrations in the engine exhaust have decreased, but CO_2

concentration has increased. Among various blends, E20 has been found to give the best results for all measured parameters at all engine speeds [11].

Emission characteristics of a four-stroke motorcycle engine with E10 have been investigated at different driving modes on the chassis dynamometers. The results indicate that CO and HC emissions in the engine exhaust are lower with E10 as compared to unleaded gasoline, whereas NO_X emissions are not affected significantly. It is also found that the E10 fuelled motorcycle engine produces more ethylene, acetaldehyde and ethanol emissions compared to unleaded gasoline engine [12]. Cold start behaviour of single cylinder air cooled motor bike engine fuelled with different gasoline ethanol blends with intake air heating has been studied, and E10 has been reported as optimum blend [13] .

The literature on use of gasoline-ethanol blends in multi cylinder or water cooled SI engines is available in plenty. E5 or E10 or E15 is reported to be optimum blend based on different sets of experiments with variables such as ignition timing, compression ratio and engine speed. But, very few references are found on gasoline-ethanol blends used in single cylinder air cooled motorcycle engine. Moreover, the comparison on behaviour of present day catalytic converter with gasoline-ethanol blends is missing in the literature.

Hence, the current work aims at presenting the results on the effect of different blends on i) performance characteristics in an air cooled motor bike engine without modifying fuel system, ignition timing and compression ratio and ii) emission characteristics with and without catalytic converter.

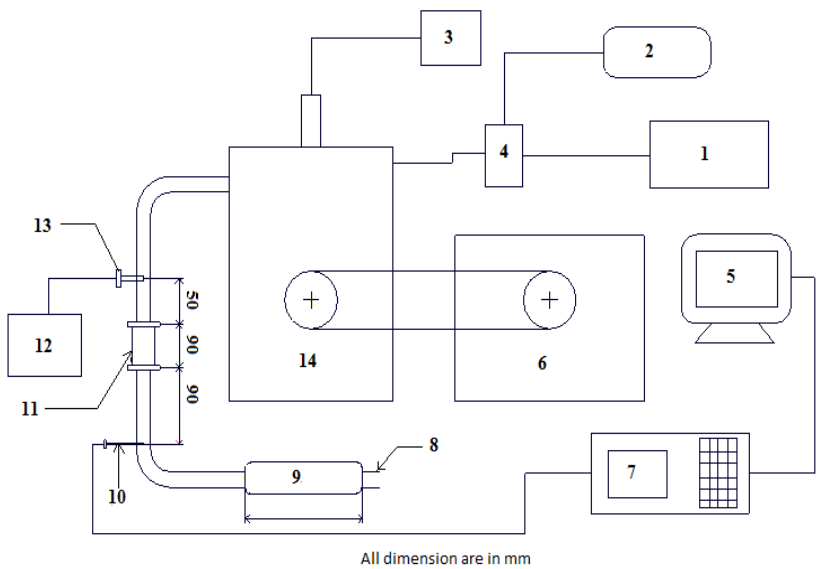
2. Experimental Set up and Procedure

Experiments were conducted on a single cylinder, four stroke, air cooled, DTSi (Digital Twin Spark ignition) engine with different gasoline-ethanol blends (E0, E5, E10, E15 and E20) and the detailed specifications are given in Table 2. The fuel blends were prepared just before starting the experiment to ensure that the fuel mixture is homogenous and no water is formed by reaction of ethanol with water vapour in the atmosphere. The test engine – clutch – gear box assembly was coupled through sprocket-chain to an eddy current dynamometer equipped with load control switches and the complete set-up is shown in Fig. 1.

After starting, the engine was allowed to run at idling for a period of 10 min. Tests were performed at engine speeds of 3000, 3500, 4000, 4500 and 5000 rpm at constant load. The load and speed range were chosen considering the normal on-road driving conditions. The desired engine speed was maintained by adjusting the throttle. Fuel consumption rate for 10 cc was measured using a calibrated burette and a stopwatch. The concentrations of the exhaust emissions (HC, CO, CO_2, O_2 and NO_x) and air–fuel ratio were recorded for every two seconds for duration of 30 seconds at each engine speed and the average value is presented in this paper. “Crypton Gas Analyser” and K-type thermocouple were used to measure exhaust emissions and temperature respectively. The specifications of exhaust gas analyser and catalytic converter are given in Tables 3 and 4 respectively. In the composition of the catalytic converter, 0:37:3 represent the amount of Pt, Pd and Rh in grams per cubic feet of catalytic converter volume.

Table 2. Test engine specifications.

Engine Type	Four-stroke, Air cooled, single cylinder, SI Engine
Bore x Stroke	58 mm x 56.4 mm
Maximum power	11 kW @ 8500 rpm
Maximum Torque	12.76 Nm @ 6500 rpm
Compression ratio	9.5:1
Fuel supply system	Constant Vacuum carburettor
Fuel	92 ON unleaded gasoline – anhydrous ethanol blends
Idling speed	1400 ± 50 rpm
Valve timing	Intake opens 12.1° CA before TDC Intake closes 55.5° CA after BDC Exhaust opens 36.5° CA before BDC Exhaust closes 14.1° CA after TDC
Spark timing	9.1° CA before TDC
Speed reduction ratio	3.19 in fifth gear
Dynamometer details	Maximum torque: 1.4 kg-m at 3000 rpm Torque arm length: 90 mm Coupling : Gear box shaft and dynamometer shaft Transmission efficiency: 0.85



1. Air filter
2. Fuel Tank
3. RPM meter
4. Carburetor
5. Computer
6. Eddy current dynamometer
7. Exhaust gas analyser
8. Tail pipe
9. Muffler
10. Exhaust gas probe
11. Cataytic converter
12. Temperature display unit
13. Thermocouple
14. Engine set up

Fig. 1. Experimental set-up.

Table 3. Exhaust gas analyzer specifications.

Gas	Range (by volume)	Accuracy	Resolution
CO	0 to 10%	± 0.06 %	0.01 %
HC	0 to 10000 ppm	± 12 ppm	1ppm
CO_2	0 to 20%	± 0.5 %	0.1 %
O_2	0 to 25%	± 0.1 %	0.01 %
NO_x	0 to 2000 ppm	± 5 ppm	1 ppm

Table 4. Catalytic converter specifications.

Composition	JM 665/40/0:37:3
Catalyst loading	1.412 kg/m^3
Platinum	0 kg/m^3
Palladium	1.3066 kg/m^3
Rhodium	0.106 kg/m^3
Size (diameter x length)	33 x 60 mm
Cells per square inch (cpsi)	100

3. Results and Discussion

3.1. Performance characteristics

In order to calculate engine performance parameters, the following basic expressions were used. The nomenclature for the symbols used is given at the end of this paper.

$$\dot{m}_f = \frac{0.0036 Q \rho_b}{t} \quad \text{kg/h}$$

$$BP = \frac{2\pi N T}{60000} \quad \text{kW}$$

$$BSFC = \frac{\dot{m}_f}{BP} \quad \text{kg/kWh}$$

$$BSEC = BSFC * (LHV)_b \quad \text{kJ/kWh}$$

$$\eta_{b,th} = \frac{3600}{BSEC} * 100\ \%$$

$$\eta_v = \frac{\dot{m}_a\ R\ T_a}{30\ V_s N\ P} * 100\ \%$$

$$\rho_b = \sum \rho_i\ v_i \quad \text{kg/m}^3$$

$$LHV_b = \sum \left(\frac{\rho_i\ v_i}{\rho_b}\right) (LHV)_i \quad \text{kJ/kg}$$

$$(AFR)_{act} = \frac{\dot{m}_a}{\dot{m}_f}$$

$$\lambda = \frac{(AFR)_{act}}{(AFR)_{st,b}}$$

$$(AFR)_{st,b} = \sum \left(\frac{\rho_i \; v_i}{\rho_b}\right) (AFR)_{st,i}$$

The effect of the ethanol blends on the fuel consumption is shown in Fig. 2. Mass flow rate of fuel increases as the ethanol percentage increases for all engine speeds. This behavior is attributed to the LHV per unit mass of the blends, which is lower than that of neat gasoline. Therefore, the amount of fuel introduced into the engine cylinder for a given desired fuel energy input has to be greater with blends. Engine speed is increased by opening the throttle valve wider. This increases inlet air velocity increasing the pressure drop. This causes more fuel to be drawn from float chamber of the carburetor. The engine running was not stable with E20 blend at 4500 and 5000 rpm. This could be due to higher mass flow rate of fuel which in turn results in greater cooling in combustion chamber. This lead to flame quenching and misfiring at these higher engine speeds with E20. Hence, the results at 3000, 3500 and 4000 rpm only are given for E20 blend.

Though the results of gasoline-ethanol blends up to E20 are presented in this paper, attempts were made to test E25 blend too. But starting of the engine was very difficult and it required multiple attempts. Moreover the engine produced very high amounts of HC and CO emissions during its unsteady operation. Misfiring and flame quenching are the reasons for this engine behaviour and hence, the ethanol content in the blend was limited to 20 percent.

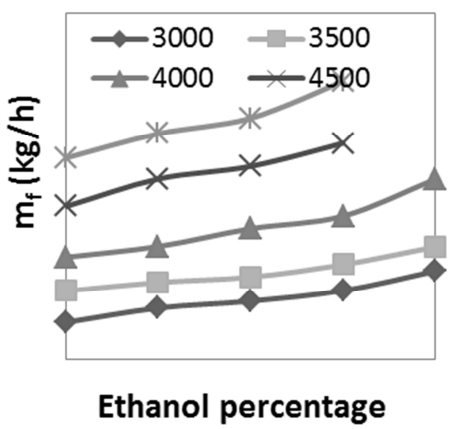


Fig. 2. Effect of Ethanol percentage on mass flow rate of fuel.

The variation of BSFC for all fuel blends at different engine speeds is shown in Fig. 3. BSFC is the mass flow rate of fuel consumed to develop unit power output. As ethanol percentage increases, BSFC increases, as heating value of the blends is lower than that of neat gasoline. Though, fuel consumption on kg/h basis increased with speed, BSFC decreased at 3500 and

4000 rpm for almost all fuels. This is attributed to higher volumetric efficiency at these engine speeds than at other engine speeds. This is discussed with reference to Fig. 7.

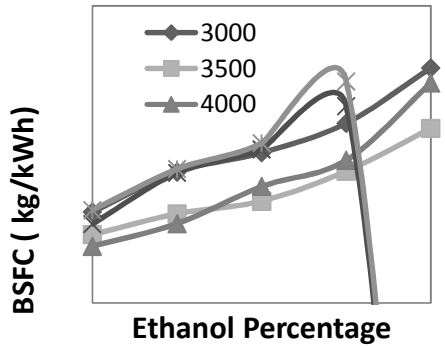


Fig. 3. Effect of Ethanol percentage on brake specific fuel consumption.

The effect of blends on brake thermal efficiency is shown in Fig. 4. It is inversely proportional BSFC. Without any modification in the compression ratio, intake system, fuel system and ignition timing, it was shown that BSFC increased as ethanol content in the blend increased. Therefore, $\eta_{b,th}$ decreases as ethanol percentage increases. On comparison with neat gasoline, E15 and E20 showed significant reduction than E5 and E10. And for any particular fuel blend, $\eta_{b,th}$ was higher at intermediate speeds than at 3000/4500/5000 rpm.

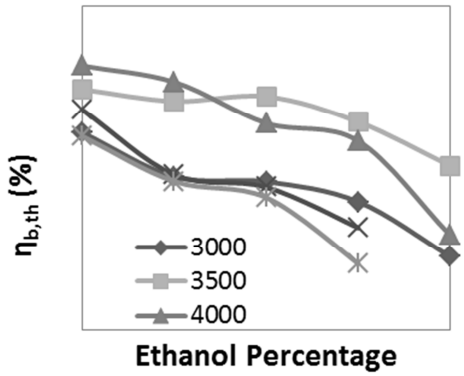


Fig. 4. Effect of Ethanol percentage on brake thermal efficiency.

Exhaust temperatures for different blends are shown in Fig. 5. Generally, for a particular blend, exhaust temperature increased with engine speed. This is because of higher heat release of more quantity of air fuel mixture at higher speeds. At particular engine speed, exhaust temperature decreased with ethanol percentage due to cooler engine components. The drop in temperature of the

engine components is attributed to higher heat of vapourization of ethanol blends compared to neat gasoline.

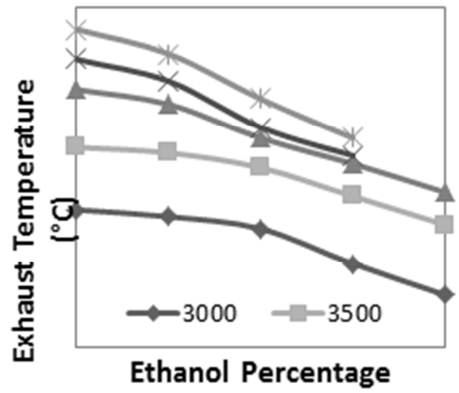


Fig. 5. Effect of Ethanol percentage on exhaust gas temperature.

The effect of the gasoline-ethanol blends on excess air factor is shown in Fig. 6. It increases as the ethanol percentage increases. This effect is attributed to two factors. First is the decrease in the stoichiometric air-fuel ratio of the blends, since the stoichiometric air-fuel ratio of neat ethanol is lower than that of gasoline. Secondly, increase of actual air-fuel ratio with the blends due to the oxygen content in ethanol. As the engine speed increases to 3500 and 4000 rpm, λ increases slightly. This is because of higher amount of air inducted which in turn increases air- fuel ratio. This is evident from the volumetric efficiency curves shown in Fig. 7. With further increase in the engine speed beyond 4000 rpm, λ decreases, since the amount of intake air decreases due to relatively higher engine temperatures as shown in Fig. 5.

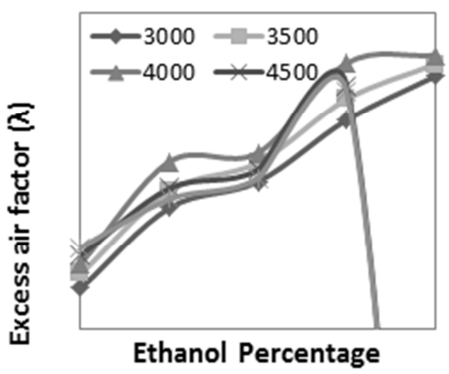


Fig. 6. Effect of Ethanol percentage on excess air factor.

An increase in volumetric efficiency was observed with increase in ethanol percentage. This is attributed to the lower suction temperature with ethanol blends which results in higher density of the charge and higher mass flow rate of air-fuel mixture. For a particular blend, volumetric efficiency was higher at 3500 and

4500 rpm than at other speeds. This is due to the higher temperature of the engine components at higher speeds which decreases density. However, increase in volumetric efficiency at 3500/4000 rpm compared to 3000 rpm is due to the valve timing which causes higher mass flow rate of charge at the intermediate speed range of the engine.

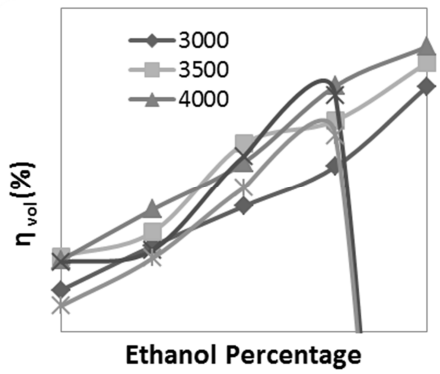


Fig. 7. Effect of Ethanol percentage on volumetric efficiency.

3.2. Emission characteristics

The effect of gasoline-ethanol blends on CO, HC and NO_x emissions, without and with catalytic converter are discussed here.

Figures 8(a) and 8(b) show the effect of blends on CO emissions without and with catalytic converter, respectively. As speed increased, CO decreased due to higher engine temperature and as ethanol percentage increased, CO decreased as the mixture became leaner. This is evident from Fig. 6. At least 50% CO was removed by catalytic converter with E0 and E5 and more than 75% CO was reduced with other blends. This is attributed to more oxygen available in the exhaust with E10, E15 and E20 compared to E0 and E5.

The effect of blends on HC emissions without and with catalytic converter is shown in Figs. 8(c) and 8(d) respectively. As speed increased, HC decreased due to higher engine temperature and as ethanol percentage increased, HC decreased as the mixture became leaner. But this is effect holds good only up to E15. E20 showed an increase in HC emissions at higher engine speeds. The reason for this could be misfiring or flame quenching with more mass of cold air-fuel mixtures. However, with catalytic converter, reduction was observed. This is due to oxidation of these HC emissions with more oxygen and higher temperatures at higher engine speeds.

Figures 8(e) and 8(f) show the effect of blends on NO_x emissions without and with catalytic converter, respectively. As speed increased, NO_x increased due to higher engine temperature. As ethanol percentage increased, HC decreased by small amount. This is attributed to lower combustion temperature with blends. At the same time, oxygen available for combustion also increases with ethanol percentage. From this result, it is clear that the more pronouncing factor for formation of NO_x emissions is combustion temperature than availability of

oxygen. With catalytic converter, NO_x were less by about 50% for all blends. But, it showed increasing trend with ethanol percentage. This might be because of formation of small amounts of NO_x in catalytic converter in the presence of oxygen and nitrogen in the exhaust at elevated temperatures.

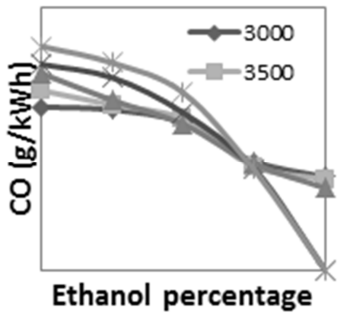


Fig. 8(a). CO – without converter.

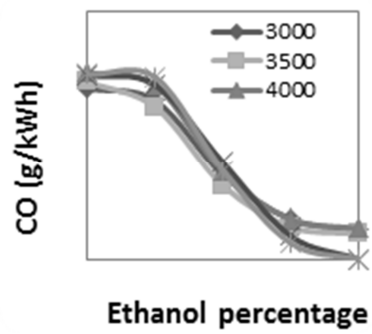


Fig. 8(b). CO - with converter.

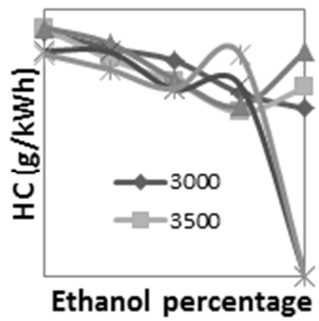


Fig. 8(c). HC - without converter.

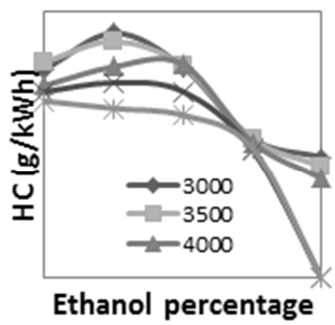


Fig. 8(d). – with converter.

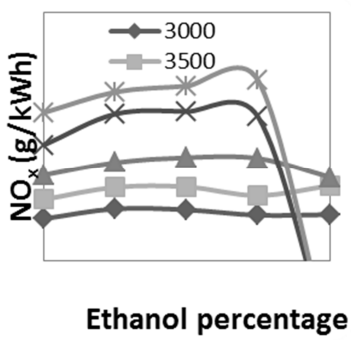


Fig. 8(e). NO_x – without converter.

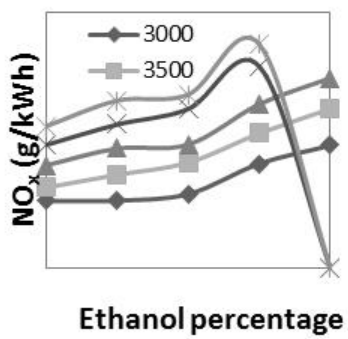


Fig. 8(f). NO_x – with converter.

Fig. 8. Variations of CO, HC and NO_x emissions with ethanol percentage at different engine speeds without and with catalytic converter.

4. Conclusions

Gasoline-ethanol blends were experimented at part-throttle operation of motorcycle engine, without modifying compression ratio, intake system, fuel system and ignition timing, without and with catalytic converter. The following conclusions are derived from the results.

- Blends up to E15 showed smooth and satisfactory engine operation and E20 blend resulted in choking and stalling at higher engine speeds.
- Brake specific fuel consumption was higher for blends due to lower heating value of the blends compared to neat gasoline and correspondingly there was reduction in brake thermal efficiency. Among the blends, E5 and E10 were closer to neat gasoline.
- Volumetric efficiency and excess air factor increased with ethanol percentage in the blend due to higher heat of vapourization and oxygen in ethanol.
- Exhaust temperature decreased with blends. This is because of lower engine temperature and higher excess air in the air-fuel mixture.
- CO, HC and NO_x emissions decreased with ethanol blends due to leaner air-fuel mixtures. Catalytic converter showed reduction in all three emissions. The reduction percentage increased with ethanol percentage in the blend. A small increase in NO_x emissions was noticed with catalytic converter.
- As E5 and E10 blends show equivalent performance characteristics and superior emission behaviour compared to neat gasoline in part-throttle conditions, they can be used as substitute fuels for neat gasoline in motorcycle engine with no modifications.

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