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Research Paper

COMPUTATIONAL PARAMETRIC INVESTIGATIONS ON A SINGLE CYLINDER SPARK IGNITION ENGINE USING ETHANOL-GASOLINE BLENDS FOR POWER GENERATION

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This study investigates the effect of blending of ethanol and gasoline in a single cylinder four stroke cycle spark ignition OHV engine fitted to a generator. The simulation is done in the professional engine simulation software from AVL Austria named as BOOST. AVL BOOST is used as a computational Thermodynamic simulation tool to analyze the performance and emission characteristics for different blends of ethanol and gasoline (10%, 20%, and 30% of ethanol by volume). The study is carried out for 40%, 50%, 60%, 70%, 80%, 90%, 100% load conditions for constant engine speed. Results were compared with the pure gasoline. It showed that as the ethanol content increases power and torque decreases. Fuel consumption increases with increase in ethanol percentage. CO emission decreases with increase in ethanol percentage whereas HC emissions decrease at higher percentage loads. NOX emissions increase with increase in ethanol percentage.

Keywords: Engine, Gasoline, Alternative fuel, Simulation, Performance, Emission

INTRODUCTION

Ethanol is an attractive alternative fuel because of its low cost and they can be obtained from both natural and manufactured sources. The two kinds of alcohols that seem most promising are Methanol (methyl alcohol) and ethanol (ethyl alcohol) (Ganesan, 2013). Ethanol has higher antiknock characteristics compare to gasoline. As such with an alcohol fuel, engine compression ratios between 11:1 and 13:1 are usual. Today's gasoline engines use a compression ratio of around 7:1 or 9:1, much too low for pure alcohol.

Alcohol reduces harmful exhaust emissions in a properly designed engine and fuel system. Alcohol contains about half the heat energy of gasoline per litre. The stoichiometric air fuel ratio is lesser for alcohol than for gasoline. The

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Stoichiometric air fuel ratio for ethanol is 9. To provide a proper fuel air mixture, a carburetor or fuel injector fuel passage should be doubled in area to allow extra fuel flow.

Ethanol does not vapourize as easily as gasoline. Its latent heat of vapourization is much greater. This affects cold weather starting. If the alcohol liquefies in the engine then it will not burn properly. Thus, the engine may be difficult or even impossible to start in extremely cold climate. To overcome this, gasoline is introduced in the engine until the engine starts and warms up. Once the engine warms, alcohol when introduced will vapourize quickly and completely and burn normally. Even during normal operation, additional heat may be supplied to completely vapourize alcohol. Alcohol burns at about the speed of gasoline. As such, ignition timing must be changed, so that more spark advance is provided. This will give the slow burning alcohol more time to develop the pressure and power in the cylinder. Moreover, corrosion resistant materials are required for fuel system since alcohols are corrosive (Ganesan, 2013). Today, the reserves of petroleum based fuels are being rapidly depleted. It is well known that the future availability of energy resources, as well as the need for reducing CO_2 emissions from the fuels used has increased the need for the utilization of regenerative fuels (Gao *et al.*, 2007).

LITERATURE SURVEY

Piotr Bielaczyc *et al.* (2013) presented an examination of the effect of ethanol–gasoline blends' physicochemical properties on emissions from a light-duty spark ignition engine. They have conducted a series of on an unmodified European passenger car on a chassis dynamometer over the New European Driving Cycle, using a constant volume sampler and analyzers for quantification of both regulated and unregulated exhaust gas compounds, using range of ethanol-gasoline blends from E5 (5% ethanol with 90% gasoline) to E50 (50% ethanol with 50% gasoline). The result showed that certain parameters varied linearly with the addition of ethanol content and some parameters remains unchanged. Venugopal and Ramesh (2014) presented the experimental studies on the effect of injection timing in a SI engine using dual injection of n-butanol and gasoline in the intake port. In this work, they have mounted two injectors in the intake port of SI engine in order to inject gasoline and n-butanol separately so that the fuels hit the back of the intake valve. They analyzed the performance, emissions and combustion parameters of an engine using gasoline and n-butanol. Initially the n- butanol and gasoline were injected simultaneously by using two injectors but with different injection timings at 25% and 60% throttle position with 3000rpm. The results were than compared with pure gasoline and pure butanol i.e. 100% by using a single injector. The result shows that there is nearly 26% reduction in hydrocarbons emission when n-butanol and gasoline were injected simultaneously at 25% and 60% throttle position. The result also shows that at 60% throttle before the gasoline were injected; n-butanol is responsible for the reduction in hydrocarbons (HC) and carbon monoxide (CO) emissions. Ananda Srinivasan and Saravanan (2010) presented the emission reduction in SI engine using ethanol-gasoline blends on thermal barrier coated piston. In this work, the ethanol-gasoline blends along with Isoheptanol were used in multi-cylinder SI engine in order to analyze the effect of blends on various parameters, i.e., performance, emission and combustion characteristics. The blended fuel used for the test were prepared by mixing 99.9% pure ethanol and unleaded gasoline with Isoheptanol blend, in the ratio of E60 + 2.0 Isoheptanol and E50 + 1.0 Isoheptanol. The tests were conducted on general multi-Cylinder SI engine and the next tests were conducted on the same engine but the piston is coated with Alumina Titania. The result shows that there is increase in brake thermal efficiency and CO, CO_2 and NOx is slightly decreased whereas HC increased. But when the same tests were conducted on engine with coated piston, the results shows an improvement. The result shows the increase in brake thermal efficiency and reduction in CO, CO_2, HC and NOx emissions. Schifter *et al.* (2011) presented the combustion and emission behavior for ethanol-gasoline blends in a single cylinder engine. In this work, various performance and exhaust emissionson a single cylinder SI engine were analyzed at 2000 rpm. The tested fuel range varies from E0 to E20. The result shows that E10 have marginal effect in combustion rates than pure gasoline but E20 slow down combustion process and increases cyclic dispersion.

PRESENT WORK

In the current investigation AVL BOOST has been used to analyze the effect of blending of ethanol with gasoline at different concentrations. A blend of ethanol to gasoline varies from 0% to 30% by volume were used in the simulation for various load conditions. The Engine Specifications is given in Table 1. Properties of ethanol and gasoline fuels are shown in Table 2.

Table 1: Test Engine Specifications

S. No.	Type	
1.	Bore (mm)	65.09
2.	Stroke (mm)	61.91
3.	Connecting rod length (mm)	123.82
4.	Compression ratio	9
5.	Maximum power (KW)	4.8 (at 3600 rpm)
6.	Maximum torque (N-m)	4.71 (at 3600 rpm)
7.	Engine displacement volume (cm^3)	206

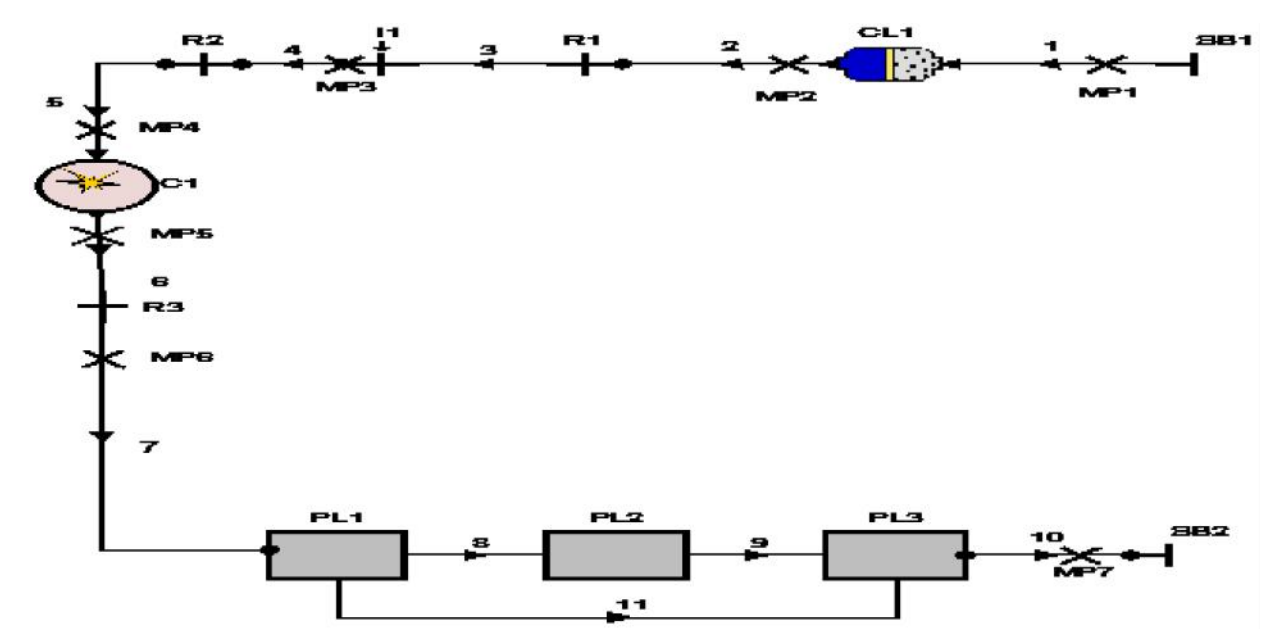
Source: Whispower AG 25000E

Table 2: Properties of Ethanol and Gasoline

Fuel Property	Ethanol	Gasoline
Formula	C_2H_5OH	C_8H_{18}
Composition weight % Carbon	52.2	85-88
Composition weight % Hydrogen	13.1	12-15
Composition weight % Oxygen	34.7	0-4
Molecular weight	46.07	100-105
Density kg/l	0.79	0.69-0.79
Specific gravity (relative density)	106-110	91
Freezing point	–114	–40
Boiling point	78	27-225
Vapor pressure, KPa at 38 °C	15.9	48-103
Specific heat, KJ/KgK	2.4	2.0
Viscosity, mPa at 20 °C	1.19	0.37-0.44
Lower heating value, MJ/Kg	26.8	30-33
Flash point, °C	13	–43
Auto-ignition temperature, °C	423	256
Stochiometric air-fuel ratio, weight	9.0	14.7
Octane number Research	108.6	88-100
Octane number Motor	89.7	80-90
Latent heat of vaporization	923	346

Source: Suat Saridemir (2012)

Figure 1: Model of the Engine Used for Parametric Investigations



SIMULATION MODELLING

The BOOST program package consists of an interactive pre-processor which assists with the preparation of the input data for the main calculation program. Results analysis is supported by an interactive post-processor. The model can be designed by placing the elements in the working area first and the connecting them with the pipes. Alternatively elements can be placed in the required order (AVL List Gmbh, AVL Boost – User Guide, 2009). The Figure 1 displays the created model:

The model consists of the following elements:1 Cylinder (C), 1 Air Cleaner (CL), 1 injector (I), 2 System Boundaries (SB), 3 Plenums (PL), 3 Restrictions (R), 7 Measuring points and 11 Pipes (Numbers).

COMBUSTION MODEL

In this work Vibe two zone model was selected for the combustion analysis. Vibe two zone model divides the combustion chamber into unburned and burned gas region (Heywood, 1988). For each zone, the first law of thermodynamics is applied to predict the rate of fuel consumed with respect to crank angle.

The following Equations (1 and 2) govern the Vibe two zone model (AVL List Gmbh, AVL Boost – Theory, 2009)

$$\frac{dm_u u_u}{d\sim} = -p_c \frac{dV_u}{d\sim} + -\sum \frac{dQ_{Wu}}{d\sim} + h_u \frac{dm_B}{d\sim} - h_{BB,u} \frac{dm_{BB,u}}{d\sim} \quad ...(2)$$

where

$\frac{dm_b u_b}{d\sim}$ = Denotes change of the internal energy of burned gas in the cylinder

$-p_c \frac{dV_b}{d\sim}$ = Denotes piston work

$\frac{dQ_F}{d\sim}$ = Denotes fuel heat input

$h_{BB,b} \frac{dm_{BB,b}}{d\sim}$ = Denotes enthalpy due to blow by

$h_u \frac{dm_b}{d\sim}$ = Denotes enthalpy flow from the unburned to the burned zone

u and *b* in the subscripts denote unburned and burned gas

The NOx formation model implemented in BOOST is based on Pattas and Häfner (Pattas and Häfner, 1973). The following 6 reactions (based on the well known Zeldovich mechanism) are taken into account (Table 3):

$$r_{NO} = C_{PostProcMult} \,.\, C_{KineticMult} \,.\, 2.0 \,.\, (1 - \sim^2) \frac{r_1}{1+\sim \cdot AK_2} \frac{r_4}{1+AK_4} \quad \text{...(3)}$$

$$\sim = \frac{c_{NO,act}}{c_{NO,equ}} \cdot \frac{1}{C_{PostprocMult}} \quad AK_2 = \frac{r_1}{r_2 + r_2} \quad AK_4 = \frac{r_4}{r_5 + r_6}$$

where

$C_{PostProcMult}$ = Denotes Post Processing Multiplier

$C_{KineticMult}$ = Denotes Kinetic Multiplier

c = Denotes molar concentration in equilibrium

ri = Denotes reactions rates of Zeldovich mechanism

The CO formation model implemented in BOOST is based on Onorati *et al.* (2001). The final rate of CO production/destruction in [mole/cm^{3}s] is calculated as:

$$r_{CO} = C_{Const} \,.\, (r_1 + r_2) \,.\, (1 - \sim) \quad \text{...(4)}$$

$$\sim = \frac{c_{CO,act}}{c_{CO,equ}}$$

where

c = Denotes molar concentration in equilibrium

ri = Denotes reactions rates based on the model

The process of formation of unburned hydrocarbons in the crevices is described by assuming that, the pressure in the cylinder and in the crevices is the same and that the temperature of the mass in the crevice volumes is equal to the piston temperature (D'Errico *et al.*, 2002). The mass in the crevices at any time period is given by Equation (5):

$$m_{crevice} = \frac{p \cdot V_{crevice} \cdot M}{RT_{piston}} \quad \text{...(5)}$$

$m_{crevice}$ = Denotes mass of unburned charge in the crevices [kg]

p = Denotes cylinder pressure [Pa]

$V_{crevice}$ = Denotes total crevice volume [m^3]

M = Denotes unburned molecular weight [kg/kmol]

R = Denotes gas constant [J/(kmol K)]

T_{piston} = Denotes piston temperature [K]

Table 3: Six Reactions Based on Zeldovich Mechanism

	Stoichiometry	**Rate $k_i = k_{0,i} .T^a . e^{(-TAi/T)}$**	**K_0 [cm^3, mol, s]**	**a [–]**	**T_A [K]**
R1	$N_2 + O = NO + N$	$r_1 = k_1 . c_{N2} . c_O$	4.93E13	0.0472	38048.01
R2	$O_2 + N = NO + O$	$r_2 = k_2 .c_{O2} . c_N$	1.48E08	1.5	2859.01
R3	$N + OH = NO + H$	$r_3 = k_3 . c_{OH} . c_N$	4.22E13	0.0	0.0
R4	$N_2O + O = NO + NO$	$r_4 = k_4 . c_{N2O} . c_O$	4.58E13	0.0	12130.6
R5	$O_2 + N_2 = N_2O + O$	$r_5 = k_5 . c_{O2} . c_{N2}$	2.25E10	0.825	50569.7
R6	$OH + N_2 = N_2O + H$	$r_6 = k_2 . c_{OH} . c_{N2}$	9.14E07	1.148	36190.66

RESULTS AND DISCUSSION

The present study concentrated on the emission and performance characteristics of the ethanol-gasoline blends. Different concentrations of the blends (E10 to E30 by volume) were analyzed using AVL BOOST for 40% to 100% load conditions under constant engine speed. The results are divided into different subsections based on the parameter analyzed.

Effect of Load on Power and Torque

The Figures 2 and 3 shows the effect of load on torque and power respectively. The increase in torque and power with the increase in percentage load is due to presence of more oxygen which results in complete combustion and increases power. Further it is seen that pure gasoline shows higher power and torque than ethanol-gasoline blends because of higher calorific value of pure gasoline.

Effect of Load on Brake Specific Fuel Consumption (BSFC)

The effect of using ethanol-gasoline blends on Brake Specific Fuel Consumption (BSFC) is shown in Figure 4. It is clear from Figure 4 that with the increase of the ethanol content the BSFC increases because of the low heat

Figure 2: Effect of Load on **Torque**

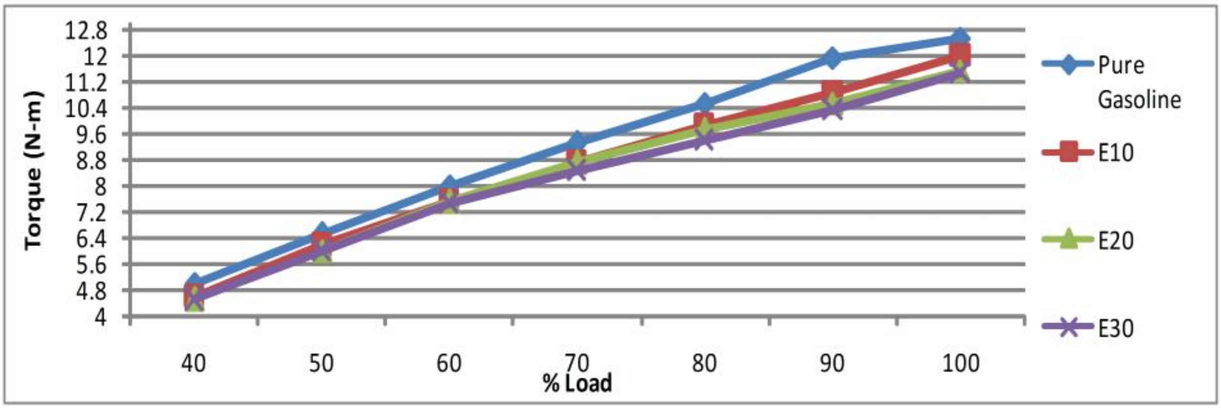


Figure 3: Effect of Load on **Power**

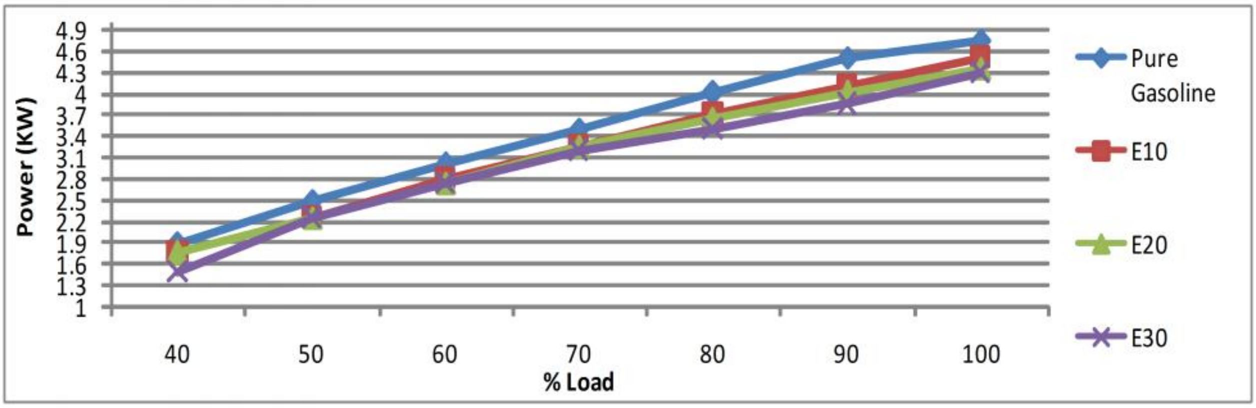


Figure 4: Effect of Load on BSFC

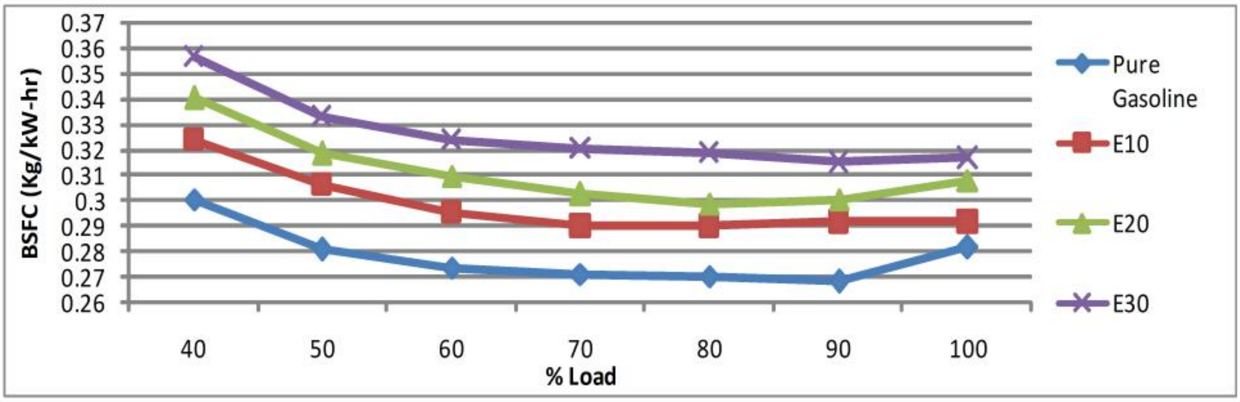
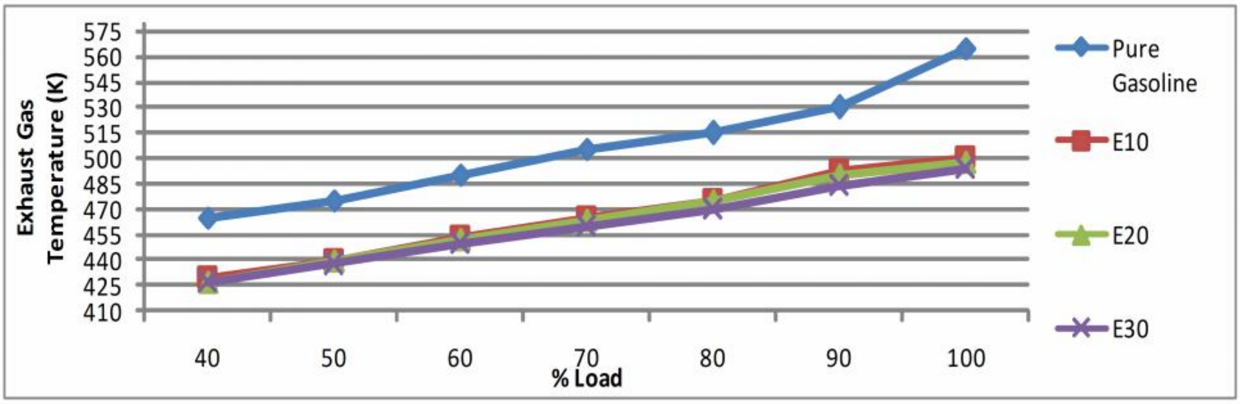


Figure 5: Effect of Load on Exhaust Gas Temperature



content per unit mass of ethanol fuel than the pure gasoline. Thus, for given desired fuel energy input more amount of fuel is introduced in the engine, thus BSFC increases with increase in ethanol content that is why E30 shows higher BSFC. Thus gasoline is more economical than ethanol-gasoline blends.

Effect of Load on Exhaust Gas Temperature

Figure 5 presents the effect of ethanol-gasoline blends on exhaust gas temperature. It is clear from Figure 5 that exhausts temperature decreases as ethanol content increases in the mixture at various engine loads because ethanol is oxygenated fuel which results in complete combustion and thus lower exhaust temperature. The latent heat of vaporization of ethanol is 2.64 times greater than gasoline. Ethanol absorbs more heat from the cylinder during vaporization (Saridemir and Ergin, 2012). So the adiabatic flame temperature of ethanol is lower than gasoline. Thus E30 shows lower exhaust gas temperature because it absorbs more heat during vaporization.

Effect of Load on Carbon Monoxide (CO) Emissions

Figure 6 shows the effect of various fuels on

Figure 6: Effect of Load on CO Emissions

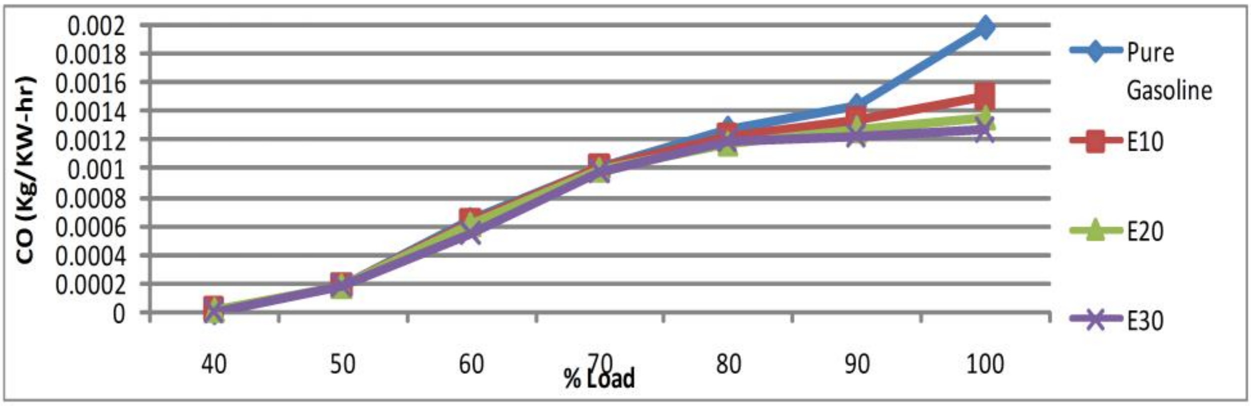
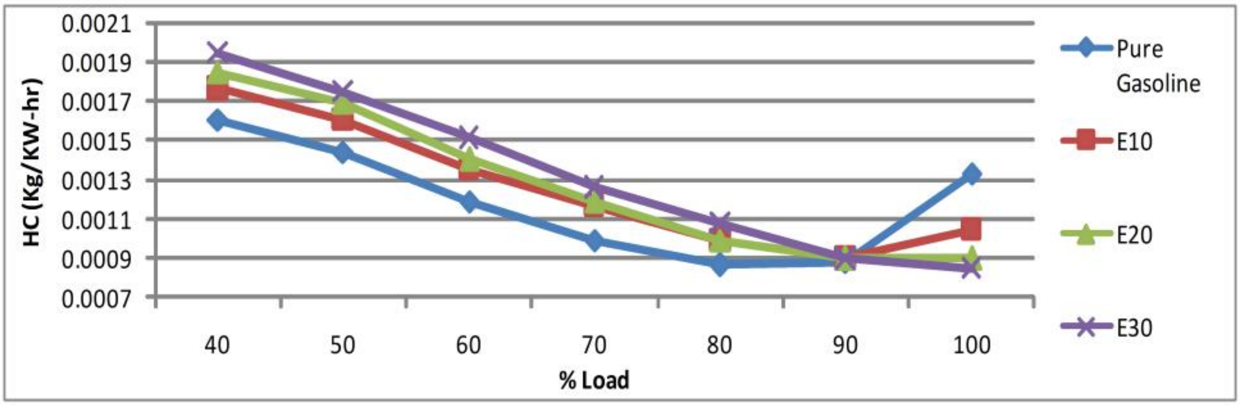


Figure 7: Effect of Load on HC Emissions



the CO emissions at different loads. The CO is formed due to incomplete combustion. With the increase of ethanol content, CO decreases because ethanol is an oxygenated fuel which results in better combustion. Thus E30 shows lower CO emission than the pure gasoline.

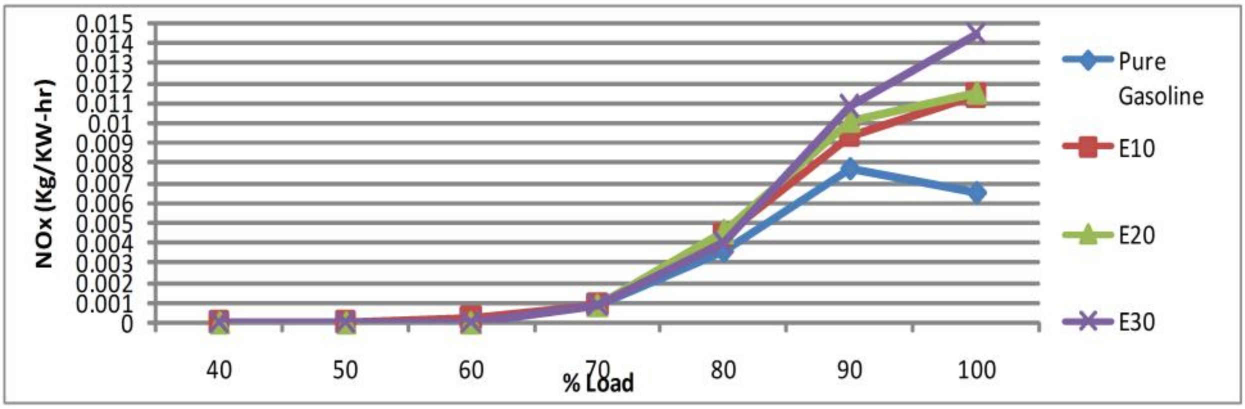
Effect of Load on Hydrocarbon (HC) Emissions

Figure 7 shows the effect of load on HC emissions. It is clear from Figure 7 that HC emission decreases with increase in percentage load because fuel rich mixtures contains enough oxygen to react with all the carbon, thus results in high HC emission in exhaust products. Thus with the increase in percentage load the fuel becomes leaner and results in decrease in HC emissions. Further, more initially ethanol-gasoline blends shows higher HC emission than gasoline because of rich mixture but at full percentage load HC emission is higher in pure gasoline than ethanol-gasoline blends. The decrease in HC emission at full load than gasoline is due to more amount of oxygen and also ethanol is an oxygenated fuel which reacts with all carbon and hydrogen.

Effect of load on NOX Emissions

Figure 8 below shows the effect of load on NOx

Figure 8: Effect of Load on NOx Emissions



emissions. The NOx emission increases with increasing load for all percentage of ethanol because with the increase in percentage load more amount of air will enter the engine which results in increase in NOx emission. With the increase of ethanol content in gasoline the NOx emission increases due to the oxygen content of the ethanol, as ethanol supplies addition oxygen for NOx formation. Also the latent heat of vapourization is higher for ethanol which results in higher pressure and temperature as compared to pure gasoline. High pressure and temperature inside the cylinder may be another reason that explains the increase in NOx formation.

CONCLUSION

1. Ethanol can be used as an alternative fuel in petrol engine.
2. Torque and power were decreased with increasing ethanol percentage.
3. The CO emissions decrease with increasing ethanol percentage whereas HC emission decreases at higher percentage load.
4. NOx emission for ethanol-gasoline blends is higher than gasoline.

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