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Effect of Compression Ratio on a Single Cylinder Spark Ignition Engine Using Ethanol-Gasoline Blends for power Generation

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ABSTRACT

A computational study on performance and emission characteristics of single cylinder, four stroke SI engine operating on ethanol-gasoline blends was carried out on AVL BOOST at different compression ratios. In this paper Vibe two zone model was selected for the study of combustion analysis and simulation were carried out for different blends of ethanol and gasoline i.e. from 0% to 30% of ethanol by volume with the increment of 10. The computational results show that gasoline fuel produces more power and torque than E10, E20 and E30. Whereas there is a considerable decrease in the emission of HC and CO but NO_X increases with increase in ethanol content.

1. Introduction

In this century, the demand of crude oil and petroleum products is increasing day by day due to enormous increase in a number of vehicles. Recently, gasoline and diesel are the most commonly used fuels and in future they become scare and most costly. With the increase in the demand of gasoline and diesel and the depletion of these fuels, the alternative fuel technology will become most common in the coming decades. There is relatively very small number of IC engines that uses alternative fuel as a working fluid. Some developing countries use alternative fuels for their vehicles because of the high cost of petroleum products. Another reason for the use of alternative fuel is the reduction in emissions over the gasoline and diesel emission problems. The alternative fuel engines are modified engine that were originally designed for gasoline fuelling. A lot of researches have been done to determine the various performance and emission characteristics of an engine fuelled with dual fuel. Even there are some diesel engines available in the market using methanol or natural gas and a small amount of diesel fuel that is injected at the proper time to ignite both fuels. At present most of the alternative fuels are costly since the quantity used is less. The cost of the alternative fuel will be less if they were used at the same magnitude as the gasoline because the cost of manufacturing, distribution and marketing would be less. The major problem is the distribution of alternative fuel where the alternative fuel is available to the public. It is profitable if there are well service stations for the distribution of alternative fuels and there are enough automobiles that will use alternative fuels. Alcohols are an attractive alternative fuels mainly methanol and ethanol. These can be obtained from number of sources i.e. natural as well as manufactured. These fuels have high octane number which improves its anti-knock characteristics [1].

2. Literature survey

Mali Anup D. [2] studied the effect of compression ratio on performance of 4-stroke spark ignition engine. He investigated various performance parameters at various compression ratios. He changed the clearance volume by changing the cavity volume of cylinder head and also by piston height. By this he will be able to get different values of clearance volume. The result shows that with the increase in compression ratio there will be increase in fuel efficiency and power output. A Y F Bokhary [3] presented the investigations on the Utilization of Ethanol-Unleaded Gasoline Blends on SI Engine Performance and Exhaust Gas Emission. He studied the performance and emission characteristics in a spark ignition four-stroke, single cylinder German CT 300 VCR SI engine using ethanol-unleaded gasoline blend. The different range of ethanol-gasoline were used for the test at full throttle opening position and variable engine speed ranging from 1100 - 2000 rpm with an increment of 300 rpm. The maximum percentage of ethanol substitution was 15%. The results showed that various parameters such as the brake torque, brake power, brake mean effective pressure, volumetric and brake thermal efficiencies increases by using ethanol-gasoline blends whereas brake specific fuel consumption reduces. Also the results showed that ethanol-gasoline blends reduce carbon monoxide (CO) and carbon dioxide (CO_2) emission, while the nitric oxide (NO) and oxygen (O_2) concentration increases. J-J Zheng [4] presented the effect of the compression ratio on the performance and combustion of a natural-gas direct-injection engine. He studied the combustion and emission characteristics of a natural-gas direct injection spark ignition engine under different compression ratios. Due to the increase in compression ratio the penetration distance of the natural-gas jet is decreased and relatively strong mixture stratification is formed which results in fast burning rate and a high thermal

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efficiency, especially at low and medium engine loads. The result shows that there is an increase in thermal efficiency with a compression ratio up to a limit of 12 at high engine loads. There is also an increase in cylinder gas pressure with the increase in compression ratio which results in decrease in flame development duration and this behavior becomes more obvious with increase in the compression ratio at low loads or for lean mixture combustion. Also result shows that with the increase in compression ratio, the exhaust hydrocarbon (HC) and carbon monoxide emissions decreased while the exhaust nitrogen oxide emission is increased. Experiments showed that a compression ratio of 12 is a reasonable value for a compressed-natural-gas direct-injection engine to obtain a better thermal efficiency without a large penalty of emissions. Achinta Sarkar [5] presented the performance and emission characteristics of SI engine running on different ethanol-gasoline Blends. He presented a review of the performance and emission characteristics of SI engine using ethanol as a fuel. The review is based on the works of different researchers and scientist available in the literature. Ethanol can be obtained from sugarcane, crop residues, cellulose, agricultural biomass, municipal waste etc. The experiment conducted by different researchers and their experimental results shows that brake specific fuel consumption, brake torque, indicated power, thermal efficiency increases or decreases depending upon the operating condition of the engine and ethanol percentage in the ethanol-gasoline blends. However, the compression ratio always increases due to enhancement of the octane number of the blend. On the other hand volumetric efficiency increases with the increase in ethanol percentage in the blends. Also result shows the reduction in the emission of unburned hydrocarbon and carbon monoxide with ethanol and ethanol gasoline blend. But the CO_2 emission is more with ethanol and NOx emission increases or decreases depending upon the engine operating conditions. N. Ravi Kumar [6] presented the effects of compression ratio and EGR on performance, combustion and emissions of Di injection diesel engine. He has studied the effects of cooled exhaust gas recirculation on performance, combustion and emissions of a Diesel engine at various parameters like compression ratios with different loads (no load to full load) and for different EGR rates (0-10%). The result shows that with the increase in compression ratio the brake thermal efficiency increases but specific fuel consumption and combustion duration and smoke opacity decreases. While with raise in % of EGR the percentage increase in brake thermal was up to 13.5%. Result also shows that with raise in % EGR the NOx emissions was gradually decreases by 11% to 85% at different compression ratios.

3. Present Work

In this paper the performance and emission characteristics of single cylinder spark ignition engine using blends of ethanol to gasoline with ratios from 0% to 30% (i.e. E0, E10, E20 and E30) by volume were analyzed in the AVL BOOST simulation at different compression ratios.

Table 1. Test Engine Specifications [7]

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

The Engine Specifications is given in Table 1. Also the Physical-Chemical characteristics of gasoline and ethanol are shown in Table 2.

Table 2. Physical-Chemical Characteristics of Gasoline and Ethanol [8]

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

3.1 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 [9]. In order to design the model, all the elements required are selected and then placed in the working area. Then the elements are arranged in a particular order and are connected by means of pipes. The designed 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). The figure 1 displays the created model:

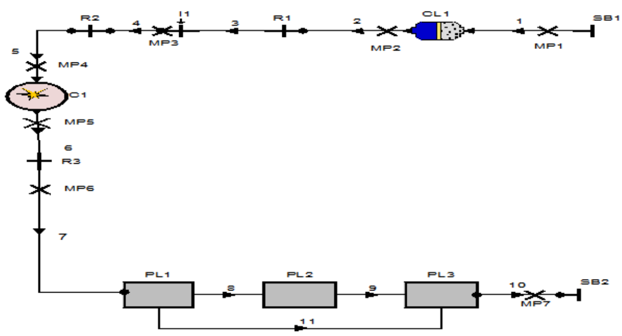

Figure 1. Model of the Engine used for Parametric Investigations.

3.2. Combustion Model

In this paper Vibe two zone model was selected for the study of combustion analysis in single cylinder spark ignition engine. In Vibe two zone model, the combustion chamber is divided into two regions i.e. unburned and burned gas region [10]. The first law of thermodynamics is applied to each zone to predict the rate of fuel consumed with respect to crank angle.

The following equations (1, 2) govern the Vibe two zone model [11]

$$\frac{dm_b u_b}{d\alpha}=-p_c\frac{dV_b}{d\alpha}-\sum\frac{dQ_F}{d\alpha}+h_u\frac{dm_b}{d\alpha}-h_{BB,b}\frac{dm_{BB,b}}{d\alpha} \quad (1)$$

Table 3. Six reactions based on Zeldovich mechanism.

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

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

Where

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

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

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

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

$h_u \frac{dm_b}{d\alpha}$ = 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 [12]. The following 6 reactions (based on the well known Zeldovich mechanism) are taken into account(Table 3):

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

$$\alpha = \frac{c_{NO,act}}{c_{NO,equ}} \cdot \frac{1}{C_{PostprocMult}} \qquad AK_2 = \frac{r_1}{r_2 + r_3} \qquad AK_4 = \frac{r_4}{r_5 + r_6}$$

Where

$C_{PostProcMult}$ = Denotes Post Processing Multiplier

$C_{KineticMult}$ = Denotes Kinetic Multiplier

c = Denotes molar concentration in equilibrium

r i = Denotes reactions rates of Zeldovich mechanism

The CO formation model implemented in **BOOST** is based on Onorati et al. [13]. The final rate of CO production/destruction in [mole/cm3s] is calculated as:

$$r_{CO} = C_{Const} \cdot (r_1 + r_2) \cdot (1-\alpha) \quad (4)$$

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

Where

c = Denotes molar concentration in equilibrium

r i = 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 [14]. The mass in the crevices at any time period is given by equation (5):

$$m_{crevice} = \frac{p \cdot V_{crevice} \cdot M}{R T_{piston}} \quad (5)$$

mcrevice = Denotes mass of unburned charge in the crevices [kg]

p = Denotes cylinder pressure [Pa]

Vcrevice = Denotes total crevice volume [m3]

M = Denotes unburned molecular weight [kg/kmol]

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

Tpiston = Denotes piston temperature [K]

4. Results and discussion

The present study concentrated on the emission and performance characteristics of the ethanol-gasoline blends in single cylinder spark ignition engine. Different concentration of the blends (from E10% to E30% of ethanol by volume) was analyzed using AVL BOOST for variable compression ratio under constant engine speed. The results are divided into different subsections based on the parameter analyzed.

4.1. Effect of compression ratio on Power and Torque

Figure 2 and figure 3 shows the effect of compression ratio on power and torque. The power increases with the increase in compression ratio because more mechanical energy is extracted from a given mass of air-fuel mixture due to higher thermal efficiency. The higher compression ratio allows the same combustion temperature to be reached with less fuel due to longer expansion cycle which in turn creates more mechanical power output and also decreases the exhaust temperature. As we know that power is directly proportional to torque, so torque also increases with the increase in compression ratio. Further, more it is seen that pure gasoline shows higher power and torque than the ethanol-gasoline blends (E10, E20 AND E30) because the calorific of ethanol-gasoline blends is half than the pure gasoline. Thus pure gasoline shows higher power than the ethanol-gasoline blends.

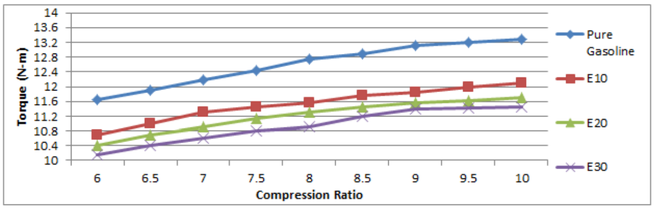

Figure 2. Effect of Compression-Ratio on Power

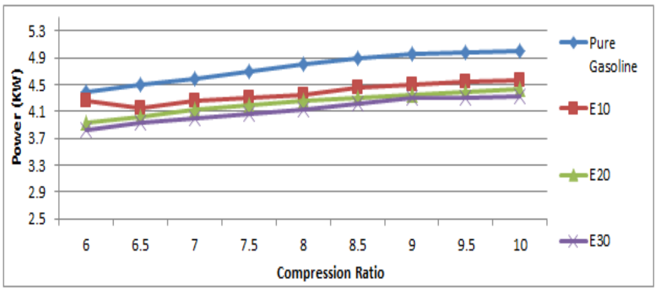

Figure 3. Effect of Compression-Ratio on Torque

4.2. Effect of Compression Ratio on Brake Specific Fuel Consumption (BSFC)

Figure 4 shows the effect of compression ratio on BSFC. It is clear from figure 4 that BSFC decreases with the increase in compression ratio may be due to the ignition delay and more combustion duration. But the BSFC remains higher for higher blend as almost twice as much ethanol as gasoline must be burned to give the same energy input to the engine. As the ethanol content increases more amount of fuel is required to obtain the energy input. Thus E30 shows higher BSFC than E10, E20 and pure gasoline.

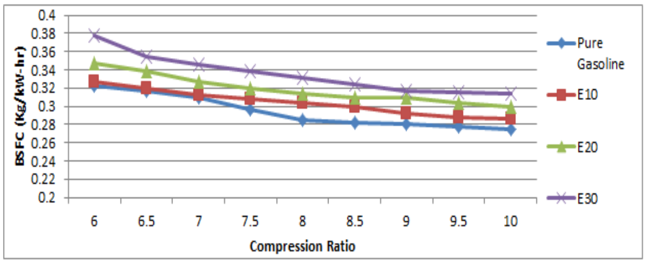

Figure 4. Effect of Compression-Ratio on BSFC

4.3. Effect of Compression Ratio on Exhaust Gas Temperature (EGT)

Figure 5 shows the effect of compression ratio on EGT. Exhaust gas temperature gives the effectiveness of utilization of heat energy produced during combustion of fuel. The exhaust gas temperature decreases with the increase in compression ratio, the reason may be that with increase in compression ratio, more amount of heat is utilized to increase the power. Further it is seen that E30 shows lower exhaust gas temperature because ethanol is oxygenated fuel due to which complete combustion takes place and in turn lower the exhaust gas temperature. Thus E30 shows lower exhaust gas temperature because it contains more content of ethanol than E10 and E20.

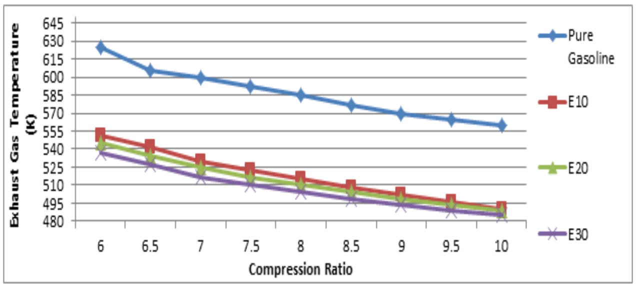

Figure 5. Effect of Compression-Ratio on Exhaust gas temperature

4.4. Effect of Compression Ratio on Carbon Monoxide (CO) Emissions

Figure 6 shows the effect of compression ratio on carbon monoxide emission. The carbon monoxide (CO) emission decreases with the increase in compression ratio because the air-fuel temperature inside the cylinder increases with the increase in compression ratio which consequently reduces the delay period due to better and complete burning of fuel. Further it is seen that with the addition of ethanol the CO emission decreases because of additional oxygen present in the ethanol which converts all carbon to carbon dioxide (CO_2). Thus E30 shows lower CO emissions.

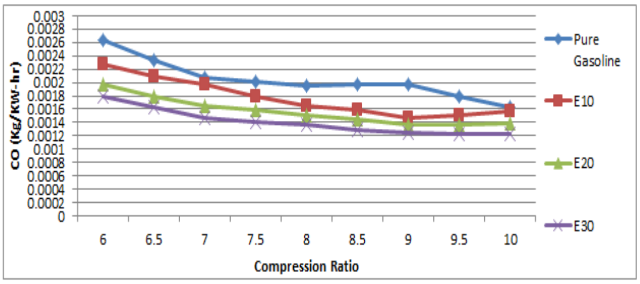

Figure 6. Effect of Compression-Ratio on CO emissions

4.5. Effect of Compression Ratio on Hydrocarbon (HC) Emissions

Figure 7 shows the effect of compression ratio on HC emissions. It is clear from figure that HC emissions increases with the increase in compression ratio, the reason may be due to unburned mixture remained in crevices during compression and remained unburned after flame passage because flame cannot propagate into the crevices. Further, it is seen that hydrocarbon emission reduces with increase in ethanol content because alcohols posses higher laminar flame speed compared to gasoline which lead to more complete combustion. Thus E30 shows lower HC emission.

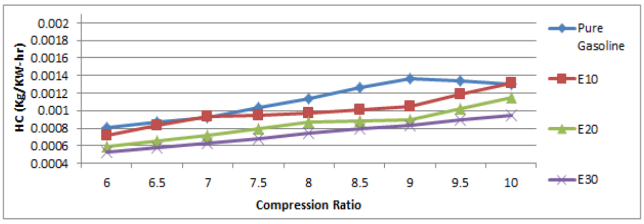

Figure 7. Effect of Compression-Ratio on HC emissions

4.6. Effect of load on NO_X Emissions

Figure 8 shows the effect of compression ratio on NO_X emission. The NO_X emission slightly decreases with increase in compression ratio because when the engine produced the maximum brake power, NOx concentrations reduced with increased compression ratio (CR), since due to interference of new parameter. It was the optimum spark ignition timing, As the CR increased the mixture temperature increased inside combustion chamber due to optimum spark ignition timing which improving the burning and increasing the flame propagation velocity, causing the optimum spark timing to retard, to insure knock preventing. This operation reduces the NOx formation required time; this phenomenon will appear in Spark timing study clearly. Further, it is seen that NO_X emission increases with increase in ethanol content. The reason may be due to addition oxygen present in the ethanol and peak temperature. Thus, E30 shows higher NO_X emission than E10, E20 and gasoline.

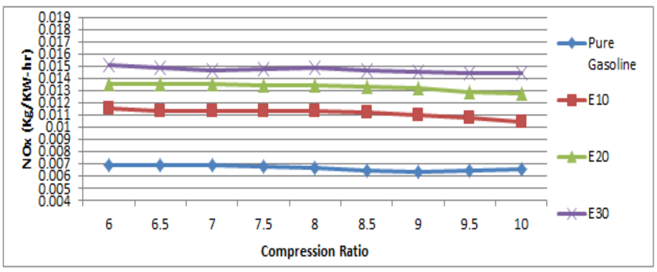

Figure 8. Effect of Compression-Ratio on NO_X emissions

5. Conclusion

The following conclusion can be made:

1) It is possible to use ethanol-gasoline blends as an alternative fuel.

2) It is possible to do blending up to 30% of ethanol by volume.

3) The blends show reduced CO and HC emissions as compared to gasoline. However there is a considerable increase in NO_X emission. Thus there is a scope for future studies how to reduce the NO_X emissions.

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