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Computer Simulation Studies on a Single Cylinder Four Stroke Cycle Spark Ignition Engine Using Gasoline and Blend of Ethanol and Gasoline as Alternative Fuels

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ABSTRACT

This paper describes the results of computational studies on a single cylinder four stroke cycle spark ignition engine using ethanol-gasoline blends as an alternative fuel to petrol. The simulation is done in the professional engine simulation software from AVL Austria named as BOOST(AVL, 2009). The modeling methodology involves the use of first law of thermodynamics and Navier -Stokes equations. The software gave successful results in both the cases. It was observed that the power output was reduced by about 35% with 85% ethanol blended gasoline as an alternative fuel due to its lower calorific value, there was increase in brake specific fuel consumption due to lesser power under similar conditions and due to lower calorific value of ethanol blended gasoline. The CO emissions were reduced with ethanol blends as fuel because of less carbon atoms, NOx was reduced with ethanol blended gasoline due to lesser exhaust gas temperatures as compared to pure gasoline. The exhaust gas temperatures were higher by about 12% with pure gasoline mode , and HC emissions were also reduced with ethanol blends due to better combustion as the inherent oxygen in ethanol assists in combustion. it is proposed that 85% ethanol +15% gasoline blend can be successfully used in petrol engine as an alternative fuel.

Keywords : Engine , petrol , ethanol , alternate fuels , simulation , performance , emissions

INTRODUCTION

Computer simulation studies helps to predict the behavior of the engine in the petrol and ethanol-petrol blend modes. We prepare the models for petrol and ethanol-petrol blend fuel modes for the engine systems and feed the actual data corresponding to the design and operating conditions of the system. It helps to simulate the results without actually performing experiments. Thus a lot of money and time is saved. Moreover we can simulate and compute those results which are very difficult to be measured experimentally. Favorable computed results pave the way for further experimental investigations.

The objectives are to investigate the feasibility of Ethanol as alternative fuel in petrol engines. Table 1 gives physico-chemical properties of ethanol and petrol which help us to investigate the feasibility of using Ethanol as an alternative fuel to petrol (Richard, 2005). Muharrem Eyidogan *et al*. (2010) did experimental investigations on engine fitted with chassis dynamometer usinf E5 and E10. They found that the brake specific consumption of fuel increased. Yung-Chen Yao *et al*. (2012) used E15 on motorcycle engine fitted with carburetor and fuel injection system. They found that CO emissions were reduced with E15. Hakan (2005) did experimental and theoretical investigation using gasolineethanol blends in spark-ignition engines. He found that performance improved and emissions were reduced. Serder *et al*. (2007) did comparative study of mathematical and experimental analysis of spark ignition engine performance used ethanolgasoline blend fuel. He found that brake torque improved but BSFC also increased. Schifter *et al*. (2011) investigated the combustion and emissions behavior for ethanolgasoline blends in a single cylinder engine. They found that performance was satisfactory. Al-Hasan (2003) investigated the effect of ethanolunleaded gasoline blends on engine performance and exhaust emission. He found that power, torque and efficiency improved. Also BSFC decreased. Fikret *et al.* (2004) investigated the use of ethanolgasoline blend as a fuel in an SI engine using E60 fuel. They found that torque improved, CO and HC emissions were reduced. fuel. They found that torque improved, CO and HC emissions were reduced.

THEORETICAL BASIS

The theoretical background including the basic equations for all elements used in the present model is summarized below to give a better understanding of the program.

The Cylinder, High Pressure Cycle, Basic Equation.

The calculation of the high pressure cycle of an internal combustion engine is based on the first law of thermodynamics:

$$\frac{d(m_c.u)}{d\alpha} = -\frac{p_c.dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_w}{d\alpha} - \frac{h_{BB}.dm_{BB}}{d\alpha}$$

(Eq.1)

where

$\frac{d(m_c.u)}{d\alpha}$ = change of the internal energy in the cylinder.

$-\frac{p_c.dV}{d\alpha}$ = piston work.

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

$\sum \frac{dQ_w}{d\alpha}$ = wall heat losses

$\frac{h_{BB}.dm_{BB}}{d\alpha}$ = enthalpy flow due to blow-by

$\frac{dm_{BB}}{d\alpha}$ = blow-by mass flow

The first law of thermodynamics for high pressure cycle states that the change of internal energy in the cylinder is equal to the sum of piston work, fuel heat input, wall heat losses and the enthalpy flow due to blow-by.

Eq.1 is valid for engines with internal and external mixture preparation. However the terms which take into account the change of gas composition due to combustion, are treated differently for internal and external mixture preparation.

For internal mixture preparation it is assumed that

- The fuel added to the cylinder charge is immediately burnt.
- The combustion products mix instantaneously with the rest of cylinder charge and thus form a uniform mixture.
- As a consequence, the Air-Fuel ratio of the charge diminishes continuously from a high value at the start of combustion to the final value at the end of combustion.

In order to solve this equation, models for the combustion process and the wall heat transfer, as well as the gas properties as a function of pressure, temperature, and gas composition are required.

Together with the gas equation

$$p_c = \frac{1}{V} . m_c . R_o . T_c \text{ ----- (Eq.2)}$$

Establishing the relation between pressure, temperature and density, Eq. 2 for in-cylinder temperature can be solved using a Runge-Kutta method. Once the cylinder gas temperature is known, the cylinder gas pressure can be obtained from the gas equation.

Combustion model

The following equation for the stoichiometric air requirement specifies how much air is required for a complete combustion of 1 kg fuel:

$$L_{st} = 137.85 \cdot \left(\frac{c}{12.01} + \frac{h}{4.032} + \frac{s}{32.06} - \frac{o}{32.0}\right)$$

[kg Air/kg Fuel] ------------(Eq.3)

For lean combustion, the total heat supplied during the cycle can be calculated from the amount of fuel in the cylinder and the lower heating value of the fuel. The lower heating value is a fuel property and can be calculated from the following formula:

$$H_u = 34835 \cdot c + 93870 \cdot h + 6280 \cdot n + 10465 \cdot s - 10800 \cdot o - 2440 \cdot w \text{ [kj/kg]} \text{ -------(Eq.4)}$$

In rich air fuel mixture combustion, the total heat supplied during the cycle is limited by the amount of air in the cylinder. The fuel is totally converted to combustion products even if the amount of air available is less than the amount of stoichiometric air.

Heat Release Approach.

The vibe function is used to approximate the actual heat release characteristics of an engine:

$$\frac{dx}{d\alpha} = \frac{a}{\Delta\alpha_c} \cdot (m+1) \cdot y^m \cdot e^{-a \cdot y^{(m+1)}} \quad \text{(Eq.5)}$$

$$dx = \frac{dQ}{Q} \quad \text{(Eq.6)}$$

$$y = \alpha - \frac{\alpha_0}{\Delta\alpha_c} \quad \text{(Eq.7)}$$

The integral of the vibe function gives the fraction of the fuel mass which was burned since the start of combustion:

$$x = \int \left(\frac{dx}{d\alpha}.d\alpha\right) = 1\text{-}e^{-a.y(m+1)} \qquad \text{(Eq.8)}$$

Gas Exchange Process , Basic Equation

The equation for the simulation of the gas exchange process is also the first law of thermodynamics:

$$\frac{d(m_c.u)}{d\alpha} = -p_c.\frac{dV}{d\alpha} - \sum \frac{dQ_w}{d\alpha} + \sum \frac{dm_i}{d\alpha}.h_i - \sum \frac{dm_e}{d\alpha}.h_e \qquad \text{(Eq.9)}$$

The variation of the mass in the cylinder can be calculated from the sum of the in-flowing and out-flowing masses:

$$\frac{dm_c}{d\alpha} = \sum \frac{dm_i}{d\alpha} - \sum \frac{dm_e}{d\alpha} \qquad \text{(Eq.10)}$$

Piston Motion

Piston motion applies to both the high pressure cycle and the gas exchange process.

For a standard crank train the piston motion as a function of the crank angle α can be written as:

$$s = (r+l).\cos\psi - r.\cos(\psi+\alpha) - l.\sqrt{1-\left\{\frac{r}{l}.\sin(\psi+\alpha)-\frac{e}{l}\right\}^2} \qquad \text{(Eq.11)}$$

$$\psi = \arcsin\left(\frac{e}{r+l}\right) \qquad \text{(Eq.12)}$$

Heat Transfer

The heat transfer to the walls of the combustion chamber, i.e. the cylinder head, the piston, and the cylinder liner, is calculated from:

$$Q_{wi} = Ai\ .\alpha_{w}.\ (T_c\text{-}T_{wi}) \qquad \text{(Eq.13)}$$

In the case of the liner wall temperature, the axial temperature variation between the piston TDC and BDC position is taken into account:

$$T_L = T_{L,TDC} \cdot \frac{1 - e^{-cx}}{x.c} \qquad \text{(Eq.14)}$$

$$c = \ln\{\frac{T_{L,TDC}}{T_{L,BDC}}\} \qquad \text{(Eq.15)}$$

For the calculation of the heat transfer coefficient, the Woschni 1978 heat transfer model is used.

Woschni Model

The woschni model published in 1978 for the high pressure cycle is summarized as follows:

$$\alpha_w = 130.D^{-0.2}.p_c{}^{0.8}.T_c{}^{-0.53}.\left[C_1.c_m + C_2.\frac{V_D.T_{c,1}}{p_{c,1}.V_{c,1}}.(p_c - p_{c,o})\right]^{0.8} \qquad \text{(Eq.16)}$$

C1 = 2.28+0.308.cu/cm

C2 = 0.00324 for DI engines

C2 = 0.00622 for IDI engines

For the gas exchange process, the heat transfer coefficient is given by following equation:

$$\alpha_w = 130.D^{-0.2}.p_c{}^{0.8}.T_c{}^{-0.53}.(C_3.c_m)^{0.8} \qquad \text{(Eq.17)}$$

$C_3 = 6.18+0.417.c_u/c_m$

Pipe Flow

The one dimensional gas dynamics in a pipe are described by the continuity equation

$$\frac{\partial \rho}{\partial t} = -\frac{\partial(\rho.u)}{\partial x} - \rho.u.\frac{1}{A}.\frac{dA}{dx}, \qquad \text{(Eq.18)}$$

the equation for the conservation of momentum

$$\frac{\partial(\rho.u)}{\partial t} = -\frac{\partial(\rho.u^2 + p)}{\partial x} - \rho.u^2.\frac{1}{A}.\frac{\partial A}{\partial x} - \frac{F_R}{V}, \quad \text{(Eq.19)}$$

and by the energy equation

$$\frac{\partial E}{\partial t} = -\frac{\partial[u.(E+p)]}{\partial x} - u.(E+p).\frac{1}{A}.\frac{dA}{dx} + \frac{q_w}{V}. \qquad \text{(Eq.20)}$$

The wall friction force can be determined from the wall friction factor λf :

$$\frac{FR}{V} = \frac{\lambda f}{2.D}.\rho.u.|u| \qquad \text{(Eq.21)}$$

Using the Reynold's analogy, the wall heat flow in the pipe can be calculated from the friction force and the difference between wall temperature and gas temperature:

$$\frac{q_w}{V} = \frac{\lambda_f}{2.D}.\rho.|u|.c_p.(Tw - T) \qquad \text{(Eq.22)}$$

During the course of numerical integration of the conservation laws defined in the Eq.20, Eq.21 and Eq.22, special attention should be focused on the control of the time step. In order to achieve a stable solution, the CFL criterion (stability criterion defined by Courant, Friedrichs and Lewy) must be met:

$$\Delta t \leq \frac{\Delta x}{u + a} \qquad \text{(Eq.23)}$$

This means that a certain relation between the time step and the lengths of the cells must be met. The time step to cell size relation is determined at the beginning of the calculation on the basis of the specified initial conditions in the pipes. However, the CFL criterion is checked every time step during the calculation. If the criterion is not met because of significantly changed flow conditions in the pipes, the time step is reduced automatically.

An ENO scheme is used for the solution of the set of non-linear differential equations discussed above. The ENO scheme is based on a finite volume approach. This means that the solution at the end of the time step is obtained from the value at the beginning of the time step and from the fluxes over the cell borders

RESULTS AND DISCUSSION

Effect of Speed on Power

The Fig.1 below shows the effect of speed on power. It is seen as the speed increases the power also increases due to more number of power cycles per unit time. Further it is seen that the power developed by petrol as fuel is higher than ethanol-petrol blend due to higher calorific value of neat petrol.

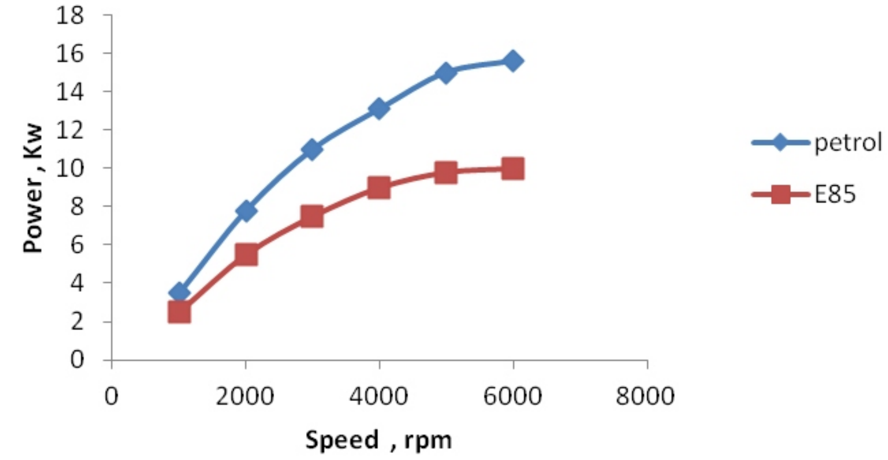

Fig.1 Effect of Speed on Power

Effect of Speed on Brake Specific fuel Consumption. (bsfc)

Fig.2 below shows the effect of speed on brake specific fuel consumption (fuel consumed per unit power output) . It is seen that the operation of petrol engine is more economical as the bsfc depends on power developed under similar conditions which is higher for petrol.

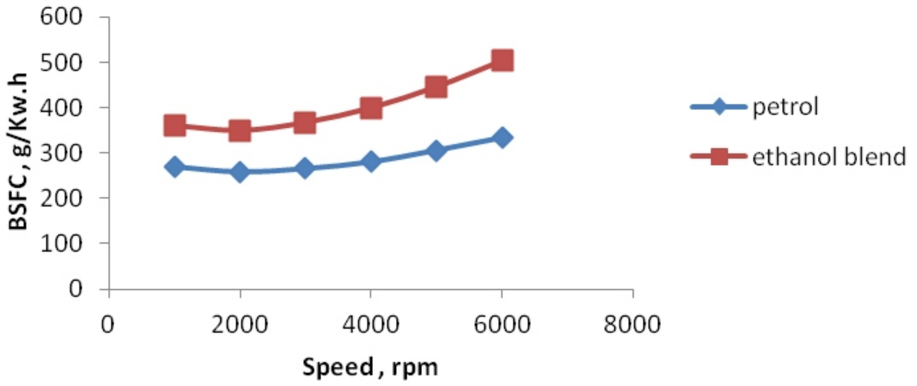

Fig.2 Effect of Speed on brake specific fuel consumption.

Effect of Speed on Exhaust Gas Temperature.

Fig.3 below shows the effect of speed on exhaust gas temperature. It is seen from the graph that the exhaust gas temperatures in case of petrol is higher. This is also a clear indication that petrol produces higher temperatures due to overall better combustion characteristics and higher heating value.

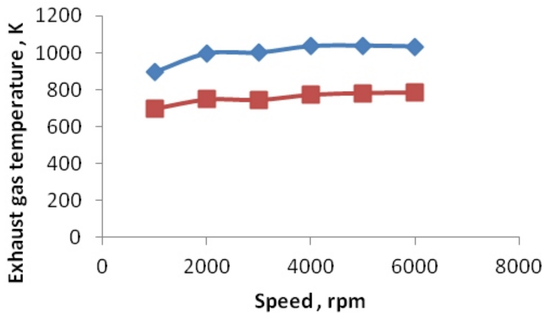

Fig.3 Effect of Speed on exhaust gas temperature.

Effect of Speed on Hydrocarbon (HC) Emissions

Fig.4 below shows the effect of speed on hydrocarbon emissions. The HC emissions are much less with ethanol-petrol blend fuel. The stoichiometric air fuel ratio of ethanol is much lesser than that of petrol. Also the ethanol has inherent oxygen in its molecular structure. These two reasons help in better combustion of ethanol-petrol blend which reduces the HC emissions.

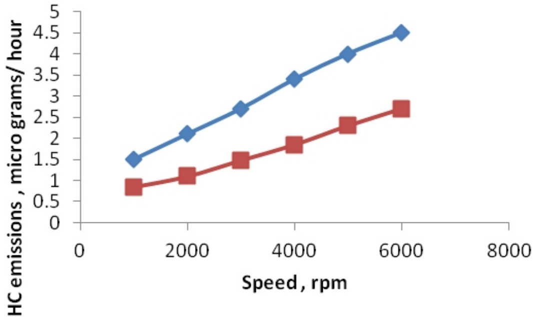

Fig.4 Effect of Speed on Hydrocarbon emissions.

Effect of Speed on Carbon monoxide (CO) Emissions

Fig.5 below shows the effect of speed on carbon monoxide emissions. CO emissions with petrol are significant due to many carbon atoms in petrol. Much lesser CO emissions are produced with ethanol blend fuel as there are less number of carbon atom in ethanol-petrol blend. The CO emissions with petrol are higher at higher speeds as there are more number of power cycles at higher speed.

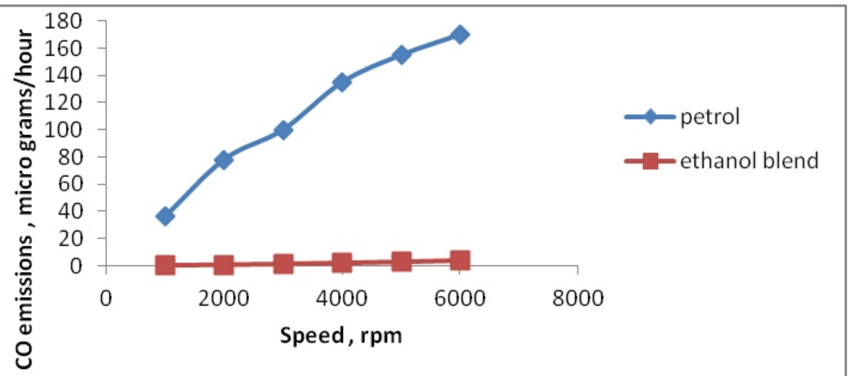

Fig.5 Effect of Speed on Carbon monoxide emissions.

Effect of Speed on NOX Emissions

Fig.6 below shows the effect of speed on NOx emissions. The NOx emissions with petrol fuel are higher as higher temperatures are produced with petrol as compared to ethanol-petrol blend. For the formation of NOx emissions presence of oxygen and nitrogen from air at higher temperatures for long time is desirable and favourable condition.

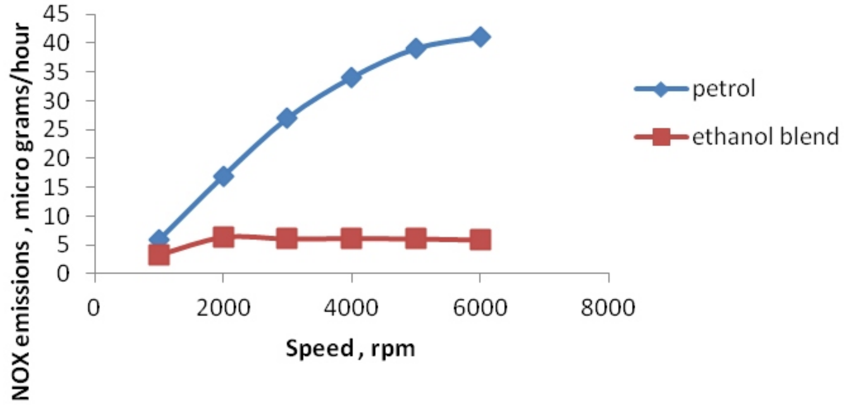

Fig.6 Effect of Speed on Nitric oxides emissions

CONCLUSIONS

1. Ethanol can safely be used as an 85% blend with petrol in petrol engines as an alternative.

2. Pollution formation from engines using Ethanol-petrol blends is much less.

3. Lesser power is produced by Ethanol-petrol blend as compared to petrol.

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APPENDIX-A

Nomenclature

a = speed of sound
A = pipe cross-section
A_{eff} = effective flow area
A_i = surface area (cylinder head, piston, liner)
AF_{CP} = air fuel ratio of combustion products
A_{geo} = geometrical flow area
c = mass fraction of carbon in the fuel
c_V = specific heat at constant volume
c_p = specific heat at constant pressure
C1 = 2.28+0.308.cu/cm
C2 = 0.00324 for DI engines
C2 = 0.00622 for IDI engines
Cm = mean piston speed
Cu = circumferential velocity
c_u = circumferential velocity
D = cylinder bore
D = pipe diameter
dm_i = mass element flowing into the cylinder
dm_e = mass element flowing out of the cylinder
d_{vi} = inner valve seat diameter (reference diameter)
= blow-by mass flow
e = piston pin offset
E = energy content of the gas (=ρ. +
f = fraction of evaporation heat from the cylinder charge
FR = wall friction force
h = mass fraction of hydrogen in the fuel
hBB = enthalpy of blow-by
hi = enthalpy of in-flowing mass
he = enthalpy of the mass leaving the cylinder
Hu = lower heating value
k = ratio of specific heats
l = con-rod length
m = shape factor
= mass flow rate
mc = mass in the cylinder
mev = evaporating fuel
mpl = mass in the plenum
n = mass fraction of nitrogen in the fuel
o = mass fraction of oxygen in the fuel
p = static pressure
P01 = upstream stagnation pressure
Pc,o = cylinder pressure of the motored engine[bar]
Pc,1 = pressure in the cylinder at IVC[bar]
ppl = pressure in the plenum
pc = cylinder pressure
p2 = downstream static pressure
qev = evaporation heat of the fuel
qw = wall heat flow

Q = total fuel heat input
QF = fuel energy
Qwi = wall heat flow (cylinder head, piston, liner)
r = crank radius
R0 = gas constant
s = piston distance from TDC
t = time
T = temperature
Tc,1 = temperature in the cylinder at intake valve closing (IVC)
Tc = gas temperature in the cylinder
Twi = wall temperature (cylinder head, piston, liner)
TL = liner temperature
T L,TDC = liner temperature at TDC position
T L,BDC = liner temperature at BDC position
Tw = pipe wall temperature
T01 = upstream stagnation temperature
u = specific internal energy
u = flow velocity
V = cylinder volume
V = cell volume (A.dx)
VD = displacement per cylinder
w = mass fraction of water in the fuel
x = relative stroke (actual piston position related to full stroke)
x = coordinate along the pipe axis
α = crank angle
αo = start of combustion
Δαc = combustion duration
αw = heat transfer coefficient
ρ = density
μσ = flow coefficient of the port
ψ = crank angle between vertical crank position and piston TDC position
= wall friction coefficient
Δt = time step
Δx = cell length

APPENDIX-B

Petrol Engine Specifications	
Bore	84 mm
Stroke	90 mm
Compression Ratio	9
Number of Cylinders	1

APPENDIX-C

Table 1: Physico-Chemical Properties of Petrol and Ethanol [2]

Fuel Property	Ethanol	Petrol
Formula	C_2H_6O	C4 to C12
Molecular weight	46.07	100-105
Lower heating value, MJ/Kg	26.8	42.5
Stoichiometric air - fuel ratio, weight	9	14.7
Octane number	113	80-98

ETHNO-BOTANY