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Optimization Of Start Of Combustion Timing For Methanol And Petrol As Two Alternative Fuels For A Single Cylinder Spark Ignition Engine

M. Marouf Wani
Mechanical Engineering Department,
National Institute of Technology, J&K, Srinagar, India. PIN 190006

ABSTRACT

This paper presents the comparative results for optimization of the start of combustion timings for methanol and petrol as two alternative fuels for a single cylinder spark ignition engine. The results were generated by the computational thermodynamic investigations in the professional internal combustion simulation software AVL BOOST. The results were computed at the maximum engine speed of 6000rpm under rich mixture conditions for both methanol and petrol corresponding to the possible development of maximum power. The software uses the basic conservation laws of mass, energy and momentum for computing the values of thermodynamic and gas dynamic variables in all the components of the modeled engine. The values of the thermodynamic and gas dynamic variables are computed throughout the four stroke cycle on crank angle basis. The software uses the numerical finite volume method for solving the conservation equations for mass and energy and momentum in the manifolds for computing the properties of air and exhaust gas. The thermodynamic simulations in the engine cylinder are based on solving the first law of thermodynamics for open and closed cycles.

First the modeled engine was run in petrol mode and the results were simulated for engine performance and emissions characteristics. The result was also computed for the octane demand of the engine under variable start of combustion timing operation. The computational investigations were repeated with methanol as an alternative fuel to petrol.

The current spark ignition engine design is based on the octane ratings of the commercial spark ignition engine fuels. The results of the computational investigations on the basis of start of combustion timing, which follows the spark timing of the engines, show how the performance and emission characteristics of the spark ignition engines can be changed or optimized further.

Keywords: Start of Combustion Timing, Spark Ignition, Internal Combustion Engine, Methanol, Petrol, Octane Requirement, Performance and Emissions

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Introduction

The spark ignition engines use petrol , LPG, CNG and ethanol as its alternative fuels for power generation for A.C generators and automotive applications. The design of these spark ignition engines is governed by the principles of normal combustion without knock. So the design parameters like compression ratio, spark timing and the cooling system design is done accordingly. However the spark timings of these engines is decided on the basis of the octane numbers of

the commercial fuels like petrol, LPG, CNG and ethanol etc for the reasons mentioned above. There is a possibility to improve the engine performance on the basis of optimum spark timings provided the fuels available commercially are improved to meet the corresponding octane demand of the engine. Methanol as a liquid fuel has physical and chemical properties suitable for combustion in spark ignition engines and has been tested and verified by many researchers as a suitable fuel for spark ignition engines.

Fleming R. et al., conducted experimental investigations with methanol on a single-cylinder research engine, a 4-cylinder 122-CID (2000 cc) engine, and a 8-cylinder 350-CID engine.

The results showed that the single cylinder engine could operate leaner than the multi-cylinder engine since non-uniform distribution of air-fuel mixture in multi-cylinder engine for each cycle was observed. The methanol based engines were 5% more economical than their corresponding gasoline versions. Also the methanol based engines produced substantially lower nitrogen oxides emissions.[1]

Harrington J., et al. conducted experimental investigations on a single cylinder engine with methanol and indolene as two fuels over wide range of speed and load.

The results showed that methanol fuel exhibits faster burning rate, (shorter ignition delay period and combustion interval). Also with same engine air flows and equivalence ratios, methanol produced more power than indolene. Methanol based engine consumed more fuel but energy consumption rate was lower with methanol. The methanol engine produced lower NO emissions but CO and HC emissions were higher in any one of the two versions depending upon the operating value of the equivalence ratio. [2]

Sumio Ito et al., conducted fundamental research on a conventional spark ignition engine using methanol as its fuel. The results showed that the thermal efficiency was greater with methanol. Also methanol showed pre-ignition problems by increasing the compression ratio of the engine. The unburned fuel and aldehyde emissions were greater with the methanol fuel but the evaporative emissions were lower with methanol. The methanol showed cold start problems which was overcome in the dual fuel mode. [3]

Gardiner D et al., conducted experimental investigations on a small spark ignition engine under sub-zero temperature conditions to study the comparative cold start problems with methanol, indolene and commercial gasoline fuels. The results showed that the methanol based engine failed to start at below about zero degree Celsius temperature whereas the indolene fuelled engine could start easily up to - 45 degree Celsius temperatures. [4]

Adelman H conducted experimental investigations with methanol in both spark ignition combustion and compression ignition combustion types of engine designs. The compression ignition combustion engine design was modified by incorporating an additional spark ignition system in it to be used for both diesel and methanol operation. The results showed that, in case of the compression ignition combustion based engine design, the methanol fuel eliminated the particulates and also produced lesser NOx emissions. However the fuel consumption was higher with methanol as compared to diesel as its fuel. Again with methanol as fuel, the modified compression ignition combustion type of engine design gave higher thermal efficiency than the conventional spark ignition type of engine design. The NOx and CO emissions were comparable with both the types of designs mentioned above. Also the unburned fuel emissions were higher with

the modified compression ignition type of engine design.[5]

Pannone G et al., conducted experimental investigations with methanol on a single cylinder spark ignition engine under naturally aspirated and turbocharged conditions under varying air-fuel ratios. The results were that lean operation of turbocharged engine gave higher thermal efficiency as compared to stoichiometric operation of naturally aspirated version. Also the carbon monoxide and nitrogen emissions were reduced in case of the turbocharged engine. However the unburned fuel and the aldehyde emissions were increased with the turbocharged version.[6]

Niwa K et al., conducted controlled fleet tests with M85 fuel on Otto-type vehicles in Japan. The results confirmed it as a fuel for Otto-type vehicles in Japan with improved and durable fuel injector technology, improved cold start ability design and reduced formaldehyde emissions technology.[7]

Ramadan B et al., conducted numerical investigations with methanol on a direct injection 4 stroke cycle spark ignition engine using two types of bowl-in-pistons, under swirl and no-swirl conditions and under variable air-fuel ratios. The investigations were done for the closed period of the cycle only. The results showed that the fuel-air mixing, combustion and flame propagation were significantly improved when swirl was turned on. This further resulted higher peak pressures in the cylinder as well as heat loss across the cylinder walls. The investigations further showed incomplete combustion under stoichiometric operation of the engine.[8]

Brusstar M et al., conducted experimental investigations on a medium duty turbocharged spark ignition engine, using ethanol and methanol blends with petrol with port fuel injection system and high compression ratios. The alcohols used were derived from the renewable biomass source. The results indicated that the bio-derived fuels were cost effective and resulted in efficient operation of the engine.[9]

Zhang Fan et al., conducted computational and experimental investigations on a SI engine using M10, M20 and M30 as methanol-gasoline blended fuels for estimating aldehyde emissions formation. The computation was done using AVL BOOST software and the experimental investigations were conducted using Fourier transform infrared spectrometer. Both the simulated as well as experimental results show that the formaldehyde emissions increased with increased percentage of methanol In the methanol-gasoline blends.[10]

Methanol can be produced from natural gas, coal and wood. Some favorable physical and chemical properties of a mixture of 85 percent methanol and 15 percent gasoline known as M85 make it possible to use it as an alternative liquid fuel for spark ignition engine based vehicles. However difficulties with methanol arise from its low energy, a non-visible flame with M100 (neat methanol), cold starting difficulties, lubricant contamination, increased engine wear, increased formaldehyde emissions, materials incompatibility.[11]

Theoretical Basis[12]

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.

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

$p_c = \frac{1}{V}.m_c.R_o.T_c$ ----------(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

Air Requirement And Heating Value modeling is given below.

STOICHIOMETRIC AIR-FUEL MIXTURE

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

$$L_{st}=137.85\left(\frac{c}{12.01}+\frac{h}{4.032}+\frac{s}{32.06}-\frac{o}{32.0}\right)\ [\text{kgAir/kgFuel}] \quad \text{----------(Eq.3)}$$

LEAN MIXTURE

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.

RICH MIXTURE

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.

HEATING VALUE

The lower heating value is a fuel property and can be calculated from the following formula:

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

HEAT RELEASE APPROACH.

Vibe Two Zone

The rate of heat release and mass fraction burned is specified by the Vibe function given by equation No.5 below.

The first law of thermodynamics is applied separately to the burned and unburned mixture while assuming that the temperatures of these two mixtures is different.

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

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

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

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

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

GAS EXCHANGE PROCESS

Basic Equation

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

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

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

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

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:

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

$$\psi = \arcsin(\frac{e}{r+l}) \quad \text{----------(Eq.12)}$$

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:

$Q_{wi} = Ai .\alpha_w . (T_c - T_{wi})$ ----------(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:

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

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

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:

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

C1 = 2.28+0.308.cu/cm

C2 = 0.00324 for DI engines

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

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

$C_3 = 6.18+0.417.c_u/c_m$

FUEL INJECTOR

The fuel injector model is based on the calculation algorithm of the flow restriction. This means that the air flow rate in the fuel injector depends on the pressure difference across the injector and is calculated using the specified flow coefficients.

For the injector model, a measuring point must be specified at the location of the air flow meter. In this case the mean air flow at the air flow meter location during the last complete cycle is used to determine the amount of fuel. As is the case for continuous fuel injection, the fuelling rate is constant over crank angle.

PIPE FLOW

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

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

the equation for the conservation of momentum

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

and by the energy equation

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

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

$$\frac{FR}{V} = \frac{\lambda f}{2.D}.\rho.u.|u|$$ ----------(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:

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

$$\Delta t \le \frac{\Delta x}{u + a}$$ ----------(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

KNOCK MODEL

Ignition Delay And Octane Number Requirement

AVL Boost uses the following equation based model proposed by Hires etal. for the calculation of ignition delay in combustion.

$\tau_{iD} = A \left(\frac{ON}{100}\right)^{a} p^{-n} e^{B/T}$ --=---------- (Eq.24)

where

$\tau_{iD} = A \left(\frac{ON}{100}\right)^{a} p^{-n} e^{B/T}$

τ_{iD} = ignition delay

ON = Octane Number Requirement

A = 17.68 ms

B = 3800 K

a = 3.402

n = 1.7

Methodology Used in Present Investigations

The engine was first modeled in the graphical user interface of the software using the basic elements available in it.

The boundary conditions for the model were set and all other input data was given to the elements in the model.

The first law of thermodynamics was solved for computing the properties of gas in the engine cylinder as an open system if any of the valves was open and as a closed system if both the valves were closed.

In order to generate data for gas properties in the manifold pipes, where the properties like pressure, density, velocity etc propagate like waves, the use of conservation equations for mass and momentum was also done with the help of

numerical finite volume method.

The results computed on crank angle incremental basis were integrated for full four stroke cycle to get the torque and power developed by the engine.

Various models available in the software for emission formation mechanism were used to compute the values of emissions produced by the engine.

Results And Discussions

EFFECT OF START OF COMBUSTION TIMING ON OCTANE NUMBER REQUIREMENT.

The Fig.1 below shows the effect of the start of combustion timing on the octane demand of the engine for petrol and methanol fuels.

It is seen from the figure that the octane demand of the engine is higher with petrol fuel as compared to methanol. This is due to the development of higher temperatures in the engine cylinder with petrol fuel as compared to methanol. Further it is clear that the octane demand of the engine varies with respect to the start of combustion timing for both petrol and methanol fuels. This is because the pressure and temperature developed in the engine cylinder vary with the start of combustion timing. The pressure and temperature are the dominant factors control-ling the knock model under consideration. The maximum octane demand of the engine with petrol and methanol fuels are 72.75 and 70.25 respectively which are much below the octane numbers of
commercialized petrol and methanol.

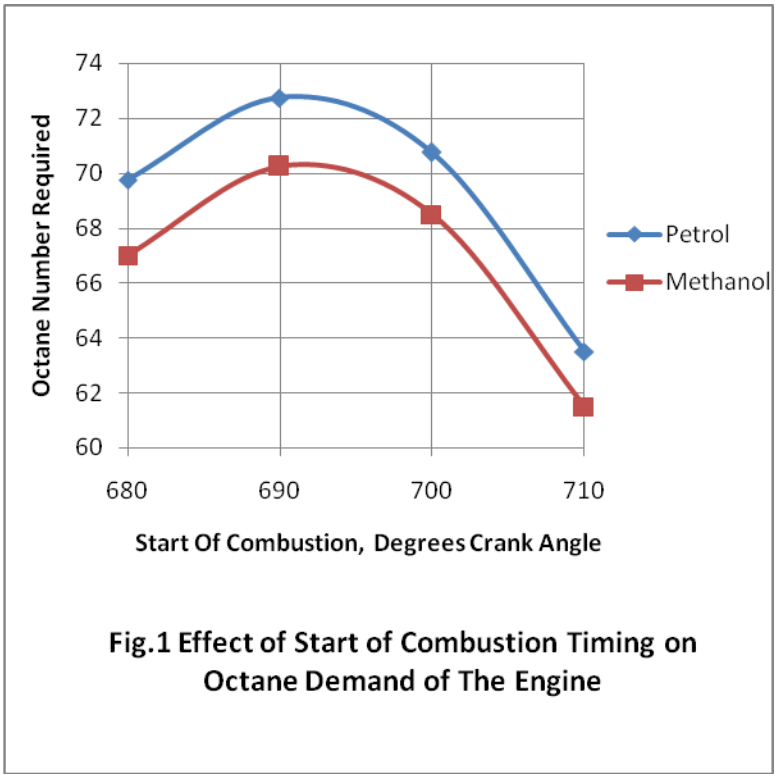

Fig.1 Effect of Start of Combustion Timing on Octane Demand of The Engine

EFFECT OF START OF COMBUSTION TIMING ON ENGINE POWER

The Fig.2 below shows the effect of start of combustion timing on the power developed by the engine in the petrol and methanol modes.

It is seen that the engine develops more power with petrol as compared to methanol. This is

because the heating value for petrol is higher than the heating value of methanol. In spite of the higher mass of methanol consumed by the engine per cycle as compared to petrol more amount of thermal energy is released with petrol fuel. This results in the development of higher pressures and higher temperatures with petrol fuel inside the engine cylinder as compared to methanol fuel. The higher positive pressure differentials across the piston with petrol fuel results in the development of higher powers in petrol mode.

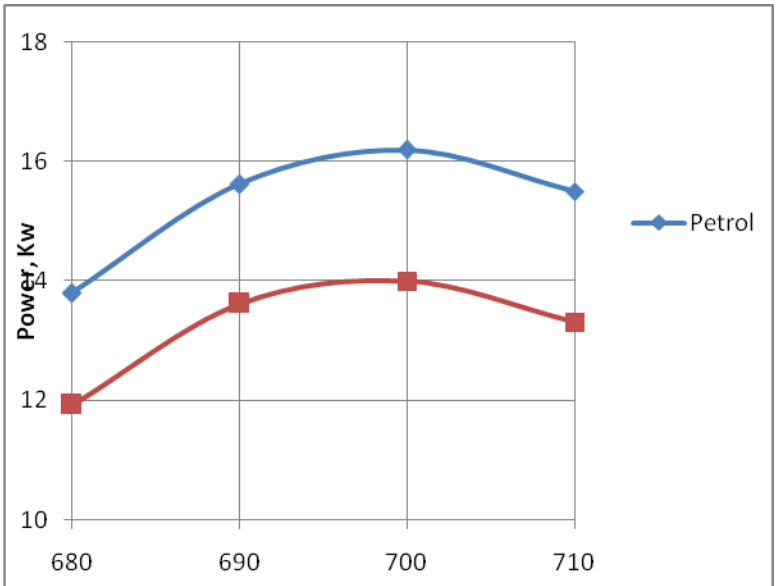

Fig.2 Effect of Start of Combustion Timing on Power Developed by The Engine

EFFECT OF START OF COMBUSTION TIMING ON ENGINE TORQUE.

The Fig.3 below the effect of start of combustion timing on the torque developed by the engine with petrol and methanol as two alternative fuels.

It is seen that the engine develops higher torque with petrol as compared to methanol as its fuel. This is because more amount of heat is released with petrol fuel as compared to methanol due to the higher heating value of petrol. The higher pressures acting on the piston inside the engine cylinder during the entire cycle develops more torque at the engine crank shaft due to transfer of higher forces along the connecting rod.

Further it is seen that torque varies with respect to the start of combustion timing. The torque developed with both petrol and methanol fuels is maximum when the start of combustion timing is 700 crank angle degrees. This is because the combustion efficiency of the engine is best with the above mentioned start of combustion timing. This actually corresponds to the maximum brake torque spark timing for this engine.

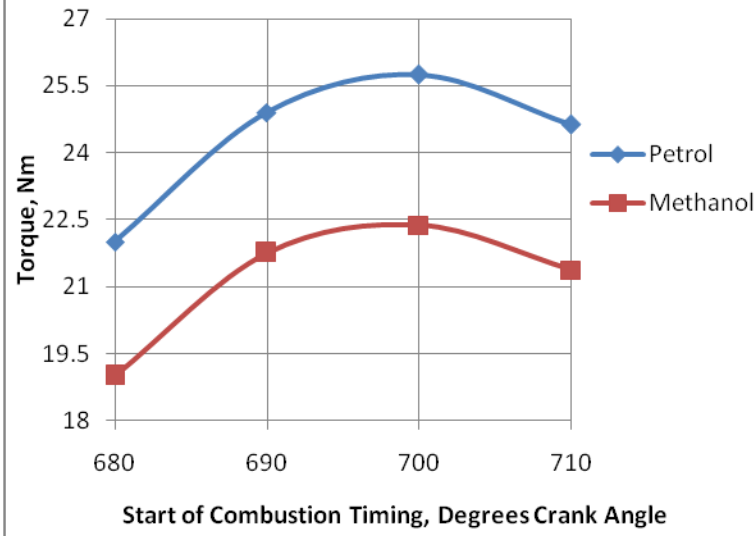

Fig.3 Effect of Start of Combustion Timing on Torque Developed by The Engine

EFFECT OF START OF COMBUSTION TIMING ON BRAKE SPECIFIC FUEL CONSUMPTION.

The Fig.4 below shows the effect of start of combustion timing on the brake specific fuel consumption of the engine using methanol and petrol as its fuels.

It is seen from the figure that the fuel consumed per unit energy output is higher with methanol fuel as compared to petrol. This is because both the fuel flow rate and the power developed by the engine are more favorable with petrol fuel which result in the economical operation of the engine in petrol mode as compared to methanol version of same engine.

It is further seen that the brake specific fuel consumption of the engine varies as the start of combustion timing is varied. The fuel consumption of the engine with either of the two fuels is minimum when the start of combustion timing is 700 crank angle degrees. This is because the power developed by the engine with both the fuels is maximum with this start of combustion timing.

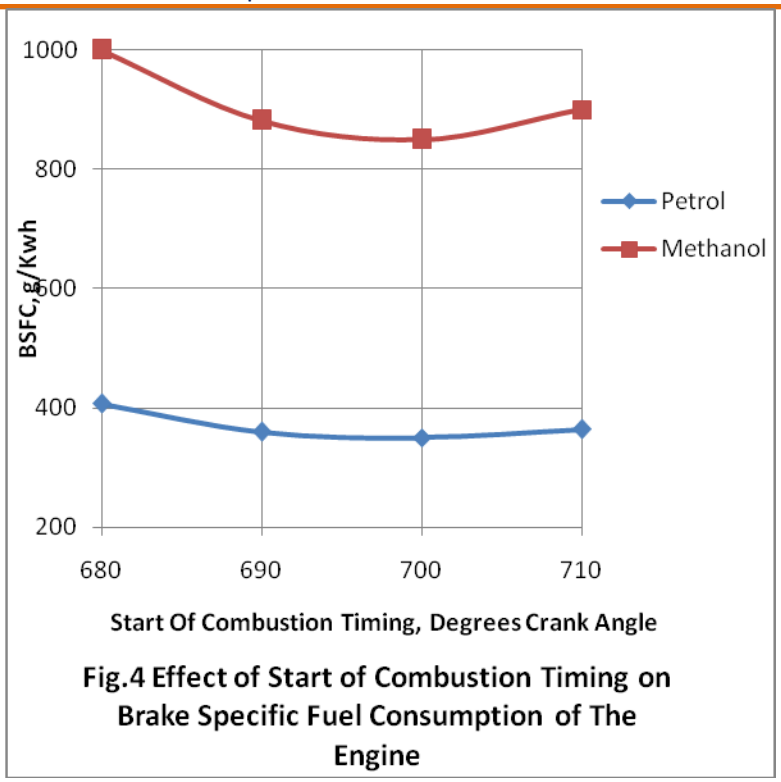

Fig.4 Effect of Start of Combustion Timing on Brake Specific Fuel Consumption of The Engine

EFFECT OF START OF COMBUSTION TIMING ON CO EMISSIONS

The Fig.5 below shows the effect of start of combustion timing on the formation of CO emissions with petrol and methanol as two alternative fuels for the engine under consideration. It is seen from the figure that the engine produces more CO emissions in methanol mode as compared to petrol mode. This is because the methanol consumption per cycle is higher than the petrol consumption per cycle for the engine under same design and similar operating conditions.

Further the engine produces minimum CO emissions when the start of combustion timing is set at 700 crank angle degrees with either of the two fuels considered. This is because the combustion characteristics inside the engine cylinder with either of the two fuels is the best with the above mentioned start of combustion timing.

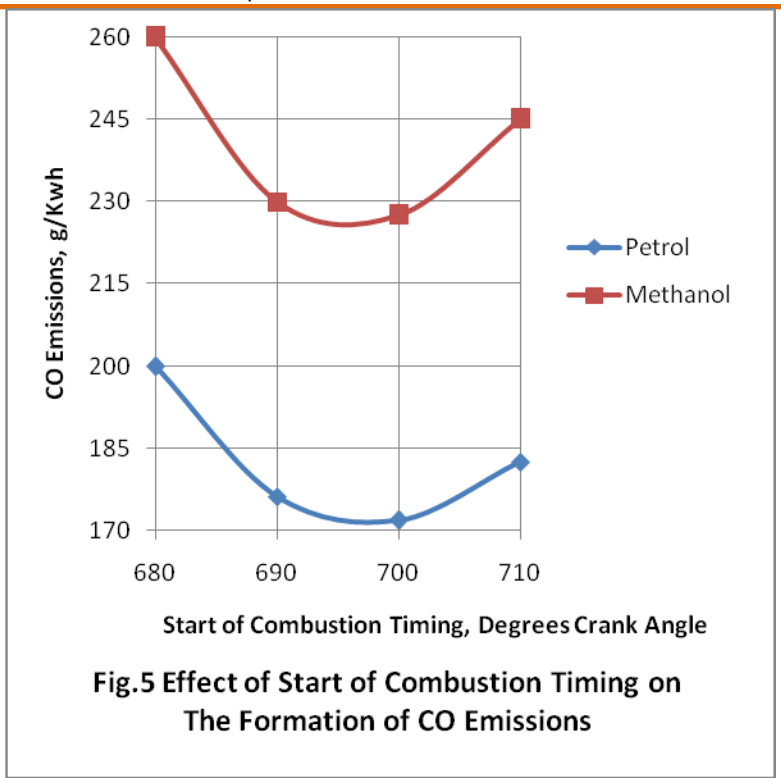

Fig.5 Effect of Start of Combustion Timing on The Formation of CO Emissions

EFFECT OF START OF COMBUSTION TIMING ON HC EMISSIONS.

The Fig.6 below shows the effect of start of combustion timing on the formation of HC emissions from the engine in the petrol and methanol modes.

It is clear that the HC emissions produced by the engine per unit of energy output are higher with methanol as compared to petrol. This is due to higher fuel consumption per unit energy output with methanol fuel as compared to petrol.

Further it is seen that the HC emissions produced by the engine vary with the start of combustion timing. The HC emissions produced by the engine, with either of the two fuels under consideration, decrease as the start of combustion timing is retarded. This is due to the combined effects of the variation of fuel consumption per unit of energy output and the variation of the power

developed by the engine with respect to start of combustion timing.

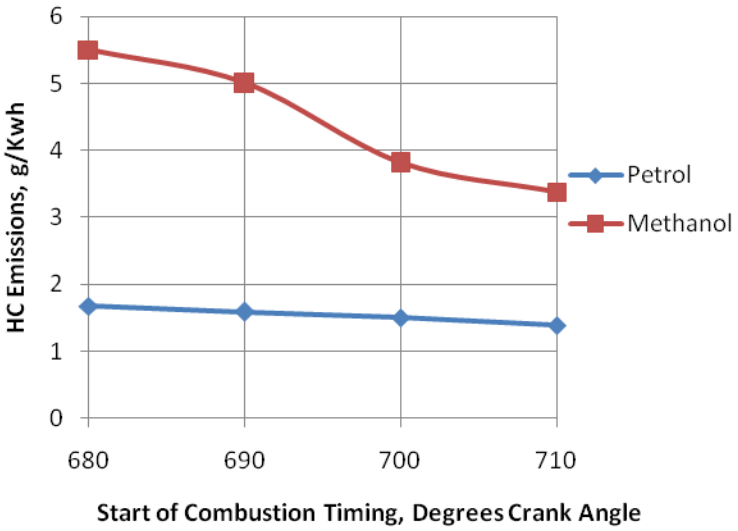

Fig.6 Effect of Start of Combustion Timing on Formation of HC Emissions

EFFECT OF START OF COMBUSTION TIMING ON NOX EMISSIONS.

The Fig.7 below shows the effect of start of combustion timing on the NOx emissions produced by the engine with petrol and methanol as its fuels.

It is seen that the engine produces more NOx emissions in its petrol version as compared to its methanol version. This is due to the development of higher temperatures in the engine cylinder with petrol as compared to methanol.

Further it is also seen that the NOx emissions produced by the engine with either methanol or petrol decrease as the start of combustion timing is retarded. This is due to the combined effect of the variation of the brake specific fuel consumption with start of combustion timing and the variation of the exhaust gas temperature with start of combustion timing for both the fuels under consideration.

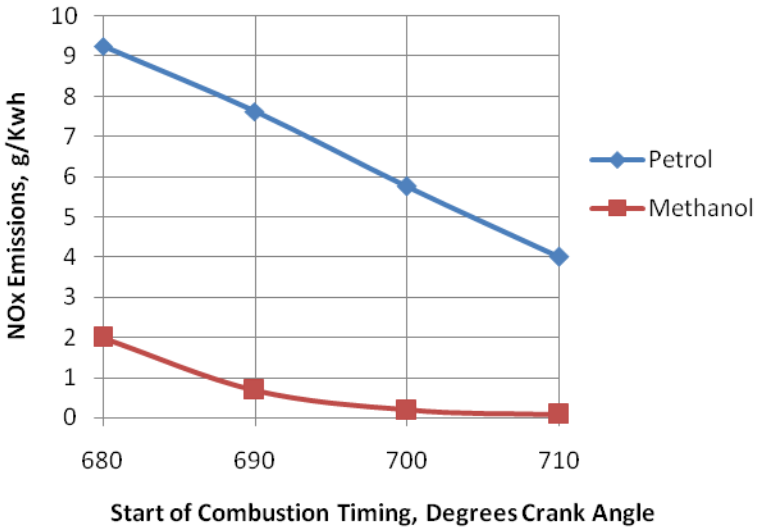

Fig.7 Effect of Start of Combustion Timing on Formation of NOx Emissions

EFFECT OF START OF COMBUSTION TIMING ON EXHAUST GAS TEMPERATURE.

The Fig.8 below shows the effect of start of combustion timing on the exhaust gas temperature of the engine in methanol and petrol modes.

It is seen from the figure that the engine develops higher exhaust gas temperatures with petrol fuel as compared to methanol. This is because of higher heating value of petrol as compared to methanol.

Further it is seen that the exhaust gas temperature varies with respect to the start of combustion timing of the engine. this is because the fuel consumption per unit of energy output also varies with the start of combustion timing accordingly

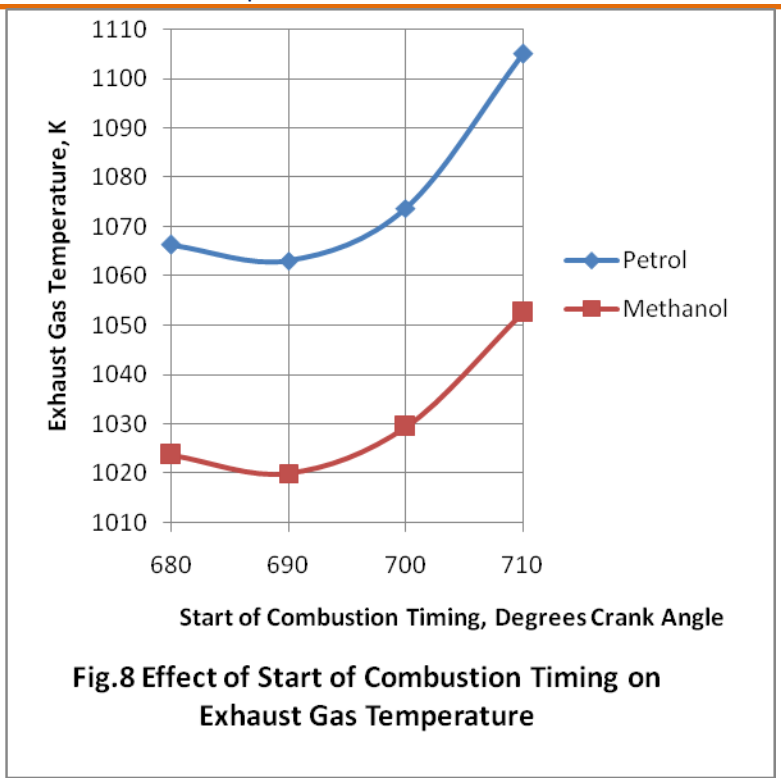

Fig.8 Effect of Start of Combustion Timing on Exhaust Gas Temperature

Conclusions

1. Both petrol and methanol can be used as alternative fuels in spark ignition engines since the octane numbers of the commercial methanol and petrol fuels are much higher than the simulated maximum octane demand of the engine with either of these two fuels.
2. The performance of the engine in terms of power, torque and the fuel consumption for either of these two fuels is best with the start of combustion timing as 700 degrees of crank angle.
3. The engine will produce minimum CO, HC and NOx emissions per unit of energy output when the start of combustion timing is designed for 700 crank angle degrees.

Acknowledgement

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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
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)
$\frac{dm_{BB}}{d\alpha}$ = blow-by mass flow
e = piston pin offset
E = energy content of the gas (=ρ. $\bar{c}_V.T + \frac{1}{2}.\rho.u^2$)
f = fraction of evaporation heat from the cylinder charge
F_R = wall friction force
h = mass fraction of hydrogen in the fuel
h_{BB} = enthalpy of blow-by
h_i = enthalpy of in-flowing mass
h_e = enthalpy of the mass leaving the cylinder
H_u = lower heating value
k = ratio of specific heats
l = con-rod length
m = shape factor
$\dot{m}$ = mass flow rate
m_c = mass in the cylinder
m_{ev} = evaporating fuel
m_{pl} = mass in the plenum

n = mass fraction of nitrogen in the fuel
o = mass fraction of oxygen in the fuel
p = static pressure
P_{01} = upstream stagnation pressure
Pc,o = cylinder pressure of the motored engine[bar]
Pc,1 = pressure in the cylinder at IVC[bar]
p_{pl} = pressure in the plenum
p_c = cylinder pressure
p_2 = downstream static pressure
q_{ev} = evaporation heat of the fuel
q_w = wall heat flow
Q = total fuel heat input
Q_F = fuel energy
Q_{wi}= wall heat flow (cylinder head, piston, liner)
r = crank radius
R_0 = gas constant
s = piston distance from TDC
t = time
T = temperature
Tc,1= temperature in the cylinder at intake valve closing (IVC)
T_c = gas temperature in the cylinder
T_{wi} = wall temperature (cylinder head, piston, liner)
T_L = liner temperature
$T_{L,TDC}$ = liner temperature at TDC position
$T_{L,BDC}$ = liner temperature at BDC position
T_w = pipe wall temperature
T_{01}= 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
α_o= start of combustion
$\Delta\alpha_c$= combustion duration
α_w = heat transfer coefficient
ρ = density
$\mu\sigma$ = flow coefficient of the port
ψ = crank angle between vertical crank position and piston TDC position

λf = wall friction coefficient
Δt = time step
Δx= cell length

Table-1

Engine Specifications	
Engine Type	Four Stroke
Method Of Ignition	Spark Ignition
Displacement	500 cc
Compression Ratio	9
Number Of Cylinders	1
Rated Speed	6000 rpm

Table-2

Physical And Chemical Properties Of Petrol And Methanol[13]

Fuel Property	Petrol	Methanol
Formula	C4 TO C12	CH3OH
Density, Kg/m3	750	791
Lower heating value, MJ/Kg	42.5 MJ/KG	19.5
Stoichiometric air-fuel ratio, weight	14.5	6.4
Octane No. MON RON	 84 92	 92 110

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