COMPUTATIONAL PERFORMANCE EVALUATION OF A FOUR STROKE CYCLE SPARK IGNITION ENGINE USING PETROL AND PROPANE 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 Propane as an alternative fuel to petrol. The simulation is done in the professional engine simulation software from AVL Austria named as BOOST. The modeling methodology involves the use of first law of thermodynamics for engine as an open system when valves are open and engine as a closed system when valves are closed. To include the effect of gas exchange in the intake and exhaust manifolds when the valves are open , the modeling is done using Navier -Stokes equations for manifolds. The design parameters are fixed by engine geometry. A matrix was prepared for the operating variables to carry out the simulation. First the data was used as per petrol engine needs, and results for engine performance were generated. The operation was revised with data for proposed propane fuelled engine and its performance were studied. The software gave successful results in both the cases. It was observed that the power output was increased with propane as an alternative fuel due to its higher calorific value, there was reduction in brake specific fuel consumption due to higher calorific value of propane. The emissions will also reduced with propane as fuel because of less carbon atoms in propane as compared to petrol fuel. It is proposed that propane can be successfully used in petrol engine as an alternative fuel.

Keywords *: Engine , Petrol , Propane , Alternate fuels , Simulation , Performance , Emissions*

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Introduction

Computer simulation studies helps to predict the behavior of the engine in the petrol and Propane fuel modes. We prepare the models for Petrol and Propane 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 Propane as alternative fuel in petrol engines. It is also intended to choose Propane as fuel because it has less number of carbon atoms and as compared to petrol and therefore will produce minimal pollution. Table 1 at the end gives physico-chemical properties of propane and petrol which help us to investigate the feasibility of using propane as an alternative fuel to petrol.

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:

$$\frac{d(m_c.u)}{d\alpha} = -\frac{p_c.dV}{d\alpha} + \frac{dQ_F}{d\alpha} - \sum \frac{dQ_w}{d\alpha} - \frac{h_{BB}.dm_{BB}}{d\alpha} \quad \text{----------(Eq.1)}$$

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:

$$L_{st} = 137.85 .(\frac{c}{12.01} + \frac{h}{4.032} + \frac{s}{32.06} - \frac{o}{32.0}) \text{ [kg Air/kg Fuel] ----------(Eq.3)}$$

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 . c +93870 . h +6280 . n +10465 . s -10800 . o -2440 . w [kj/kg] -------(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:

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

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

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

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:

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

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

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:

$$Q_{wi} = Ai \,.\alpha_w \,.\, (T_c - T_{wi}) \quad \text{----------(Eq.13)}$$

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

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

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

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

C2 = 0.00622 for IDI engines

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

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

$C_3 = 6.18+0.417.c_u/c_m$

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. In addition, the amount of fuel specified is fed into the air flow.

In the case of carburetor model, the fuel flow is set to a specified percentage of the instantaneous mass flow.

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.

The fuel is added in gaseous form to the pipe flow. No evaporation is considered.

Pipe Flow

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

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

the equation for the conservation of momentum

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

and by the energy equation

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

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

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

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:

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

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:

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

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 slightly higher than propane due to higher volumetric efficiency with petrol.

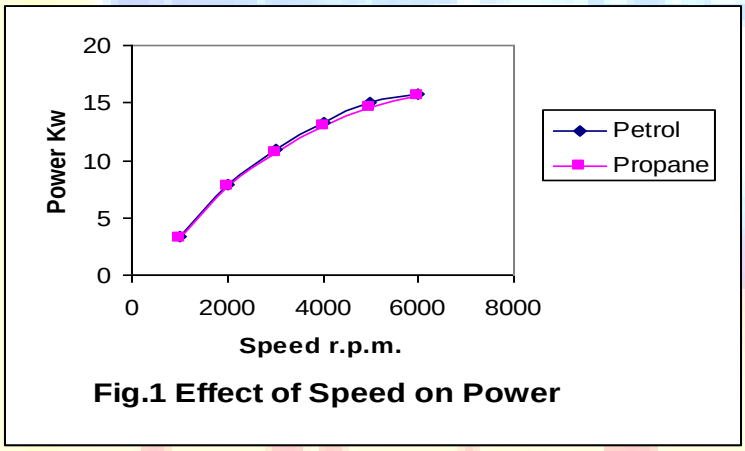

Fig.1 Effect of Speed on Power

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 propane engine is approximately as economical as the petrol fuel. From the physical and chemical properties of petrol and propane it is seen that propane has higher heating value but at same time its volumetric efficiency is lesser as it converts more easily into gaseous phase on release of pressure. Since the power with each fuel is comparable so the bsfc also remains comparable.

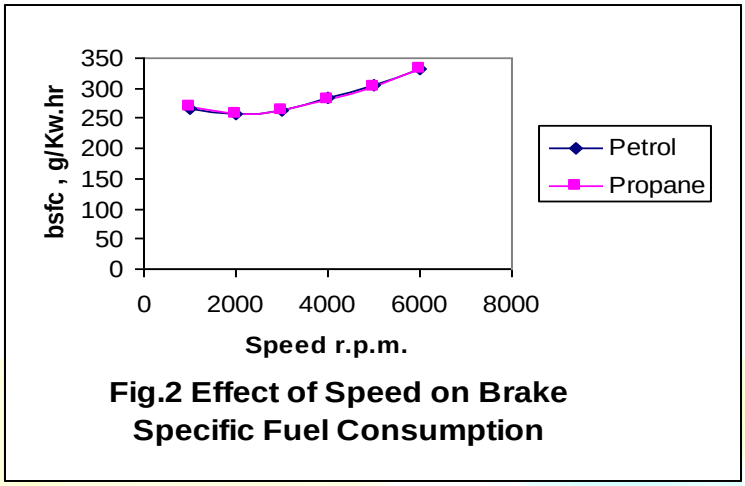

Fig.2 Effect of Speed on Brake Specific Fuel Consumption

Effect of Speed on Torque.

Fig.3 below shows the effect of speed on torque. It is seen that in each case the maximum torque is produced at 2000 r.p.m.. This is because the combustion characteristics and in-cylinder pressure development is best at this speed. Petrol produces slightly better torque due to its higher volumetric efficiency as compared to propane.

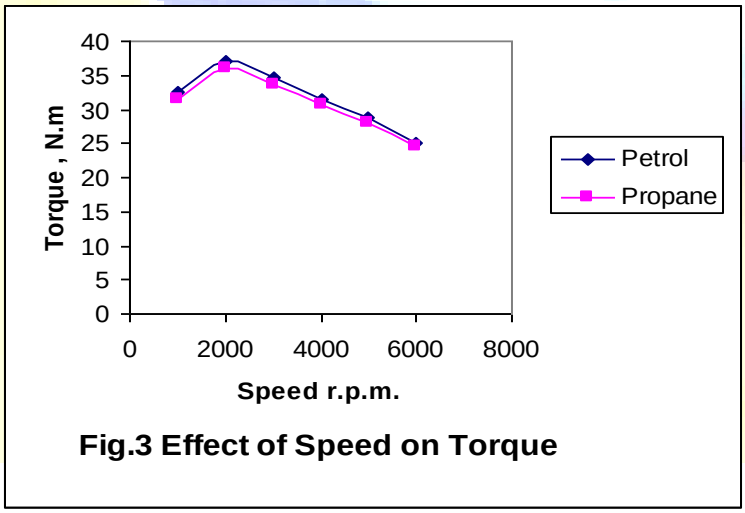

Fig.3 Effect of Speed on Torque

Effect of Speed on Exhaust Gas Temperature.

Fig.4 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 and propane are comparable. This is also a clear indication that propane also produces higher temperatures and produces comparable power as its heating value is higher than petrol. But the volumetric efficiency of propane is lower.

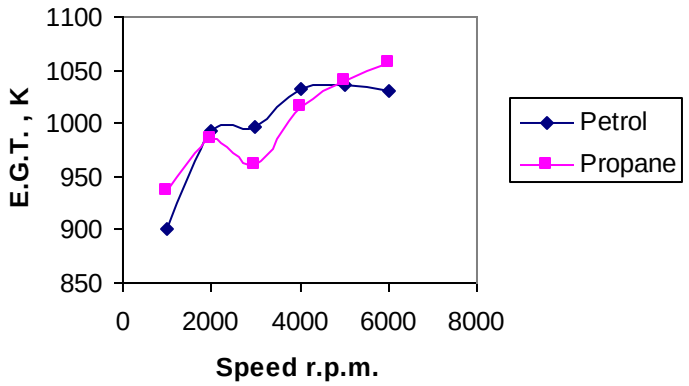

Fig.4 Effect of Speed on Exhaust Gas Temperature

Conclusions

1. Propane can safely be used in petrol engines as an alternative fuel to conventional petrol.
2. Pollution formation from engines using Propane as fuel will be less due to less carbon atoms in propane fuel.
3. Comparable power is produced by propane.
4. The Propane engine operation is as economical as petrol approximately.
5. Since the Octane number of propane is higher than that of petrol so there are lesser chances of knocking. It will result in better combustion and also gives way to increase the compression ratio of engine slightly which can boost power futher.

Acknowledgements

Author is thankful to AVL company of Austria and its unit AVL India Ltd Gurgaon in general and Mr. Abishek Agarwal of AVL India Ltd Gurgaon in particular for providing BOOST engine simulation software with free license for academic purposes.

References

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[2] Richard L. Bechtold , Alternative Fuels Handbook , SAE Publication.

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)
$\frac{dm_{BB}}{d\alpha}$	=	blow-by mass flow
e	=	piston pin offset
E	=	energy content of the gas ($=\rho.\bar{c}_V.T + \frac{1}{2}.\rho.u^2$)
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

$P_{c,o}$	=	cylinder pressure of the motored engine[bar]
$P_{c,1}$	=	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
$T_{c,1}$	=	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

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 Propane

Fuel Property	Propane	Petrol
Formula	C3H8	C4 to C12
Molecular weight	44.1	100-105
Lower heating value, MJ/Kg	46.44	42.5
Stoichiometric air-fuel ratio, weight	15.67	14.7
Octane number	110	80-98