EFFECT OF IGNITION TIMINING ON PERFORMANCE AND EMISSION CHARACTERISTICS OF A SINGLE CYLINDER FOUR SROKE CYCLE PETROL ENGINE USING HYDROGEN AND PETROL AS ALTERNATIVE FUELS.

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ABSTRACT

This paper presents the computational investigations carried out on a single cylinder four stroke cycle petrol engine with hydrogen as an alternative fuel. The software named BOOST software from AVL Austria has been used to carry out the investigations. The engine model is created by joining the various elements available in the software in the graphical user interface. The thermodynamic properties are evaluated by applying the first law of thermodynamics to the engine as an open system when valves are open and engine as a closed system when valves are closed. In order to include the effects of gas exchange in the intake and exhaust manifolds when the valves are open, the conservation equations for mass, momentum and energy have been applied. The engine geometry fixes the design parameters. In the operating variables the speed was maintained constant while the ignition timing besides the corresponding air-fuel ratio were varied for both petrol and hydrogen engines to carry out the investigations. The mapping of the engine was done by using conventional petrol as fuel. The investigations were repeated with data for proposed hydrogen engine and its performance and emissions characteristics were studied. Results showed that in comparison with the conventional petrol fuel the power output of the engine was decreased with hydrogen. However the brake specific fuel consumption was lower with hydrogen fuel. The CO and HC emissions were reduced with hydrogen while as there was increase in NOx emissions.

It is concluded that hydrogen can successfully replace petrol as an alternative future fuel in spark ignition engines..

Keywords : Engine , Petrol , Hydrogen , Alternate fuels , Simulation , Performance , Emissions

INTRODUCTION

In case of spark ignition engines the air-fuel mixture is ignited by producing a spark in cylinder towards the end of the compression stroke. The pressure developed in the cylinder at each crank angle depends upon the spark timing with respect to top centre. The power developed by the engine depends upon the torque developed by the engine which in turn depends upon the pressue development inside the cylinder. The torque developed by the engine varies and depends upon the spark timing of the engine. The engine produces the maximum torque at a particular spark timing which is known as the maximum brake torque timing (MBT timing). If the spark timing is advanced or retarded with respect to MBT timing the torque developed by the engine decreases.[1].

Hydrogen is seen as one of the important energy vectors of the next century. Hydrogen as a renewable energy source provides the potential for a sustainable development particularly in the transportation sector. Hydrogen-driven vehicles reduce both local as well as global emissions.

Hydrogen-fueled engines are known for many advantages, among which is the very low concentration of pollutants in the exhaust gases compared to internal combustion engines using traditional or other alternative fuels. Further on, because of the wide flammability limits and the high flame propagation speed of hydrogen, a hydrogen-fueled engine is capable of very lean combustion ~up to air-to-fuel ratios of 51.[2]

A GM/Crusader V-8 engine for hydrogen use at the Laboratory of Transport Technology (Ghent University).The engine was intended for the propulsion of a midsize hydrogen city bus for public demonstration. For a complete control of the combustion process and to increase the resistance to backfire (explosion of the air–fuel mixture in the intake manifold), a sequential timed multipoint injection of hydrogen and an electronic management system is chosen. The results as a function of the engine parameters (ignition timing, injection timing and duration, injection pressure) are given.

Finally, the goals of the development of the engine were reached with power output of 90 kW, torque of 300 Nm, extremely low emission levels, and backfire-safe operation.[2].

For Hydrogen-fueled internal combustion engines reasonably fast and accurate predictive computational tools are essential for practical design, control and optimization of hydrogen engines. To serve for this broader purpose, a computational model, which has been widely used for gasoline and diesel engines, is investigated for its capability to simulate hydrogen engines.

In this study, engine simulations were employed to study the performance, combustion and emission characteristics of a hydrogen-fueled engine.

The simulations generally agreed well (typically 10% difference) with the measurements under similar engine operational conditions. In particular, the variations of peak in-cylinder pressure, heat release rate, brake power, brake thermal efficiency, exhaust temperature, and NOx emissions were predicted close to the measured values within experimental and computational uncertainties. [3]

A GM/Crusader V-8 engine was converted for hydrogen use in the Laboratory of Transport Technology (Ghent University). The engine was intended for the propulsion of a midsize hydrogen city bus for public demonstration Power regulation only by the air-to-fuel ratio (as for diesel engines) against a throttle regulation (normal gasoline or gas regulation) resulted in the development of the engine with power output of 90 Kw, torque of 300 Nm, extremely low emission levels, and backfire-safe operation.[4].

A quasi-dimensional two-zone model for the operation of S.I. engines fueled with hydrogen was investigated. In this the engine combustion chamber at any instant of time during combustion was considered to be divided into two temporally varying zones: a burned zone and an unburned zone. The model incorporates a detailed chemical kinetic model scheme of 30 reaction steps and 12 species, to simulate the oxidation reactions of hydrogen in air. A knock prediction model for S.I. engine was considered for operation on hydrogen. The effects of changes in operating conditions, including a very wide range of variations in the equivalence ratio on the onset of knock and its intensity, combustion duration, power, efficiency, and operational limits were investigated. The effects of changes in some of the key operational engine variables, such as compression ratio, intake temperature, and spark timing were computed.

The results of this predictive approach were shown to validate well against the corresponding experimental results, obtained mostly in a variable compression ratio CFR engine.[5].

The authors in their review paper on hydrogen fuelled internal combustion engines (H2ICEs) state that a number of manufacturers are now leasing demonstration vehicles to consumers using hydrogen-fueled internal combustion engines (H2ICEs).Developing countries in particular are pushing for H2ICEs (powering two- and three-wheelers as well as passenger cars and buses) to decrease local pollution at an affordable cost.

Topics that are discussed include fundamentals of the combustion of hydrogen, details on the different mixture formation strategies and their emissions characteristics, measures to convert existing vehicles, dedicated hydrogen engine features, a state of the art on increasing power output and efficiency while controlling emissions and modeling.

It was concluded that Hydrogen seems to be a viable solution for future transportation, and the hydrogen internal combustion engine could act as a bridging technology towards a widespread hydrogen infrastructure, since hydrogen combustion engine vehicles can initially be designed for bi-fuel applications. Although hydrogen is the most abundant element in the universe, it is not readily available in its molecular form and has to be produced using other energy sources. Hydrogen is therefore considered an energy carrier rather than an energy source.

Hydrogen internal combustion engine vehicles have a long history, with the earliest attempts dating back to 1807. Major contributions to the development and demonstration of hydrogen internal combustion engines have been made by Musashi Institute of Technology, BMW as well as Ford Motor Company. Modern H2ICE vehicles have shown emissions levels that are only a fraction of the most stringent standards while exceeding the fuel economy numbers of their conventional-fuel counterparts. Apart from use as a neat fuel, hydrogen is also considered as a combustion enhancer. [6]

Table 1 at the end gives physical and chemical properties of Hydrogen and Petrol which help us to investigate the feasibility of using Hydrogen as an alternative fuel to diesel for power generation.[7]

THEORETICAL BASIS[8]

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.

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

AIR REQUIREMENT AND HEATING VALUE

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:

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

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.

FRACTAL COMBUSTION MODEL

The fractal combustion model for SI engines, implemented in BOOST, predicts the rate of heat release in a homogeneous charge engine. Thereby the influence of the following parameters is considered [C8]:

- The combustion chamber shape
- The spark plug location and spark timing
- The composition of the cylinder charge (residuals, recirculated exhaust gas, air and fuel vapor)
- The macroscopic charge motion and turbulence level

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 (combustion products) and unburned mixture (fresh charge) while assuming that the temperatures of these two mixtures is different.

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

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

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

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

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

GAS EXCHANGE PROCESS , BASIC EQUATION

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

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

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

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

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).cosψ-r.cos(ψ+α)-l. $\sqrt{1-\{\frac{r}{l}.\sin(\psi+\alpha)-\frac{e}{l}\}^2}$ ----------------------------(Eq.11)

ψ = arcsin($\frac{e}{r+l}$) --(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:

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

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

c = ln{ $\frac{T_{L,TDC}}{T_{L,BDC}}$ } --(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:

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

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

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:

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

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

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RESULTS AND DISCUSSION

EFFECT OF IGNITION TIMING ON POWER

The Fig.1 below shows the effect of ignition timing on power. It is seen that there is an optimum ignition timing for maximum power generation in case of both the fuels petrol as well as hydrogen. The power decreases if the ignition timing is advanced or retarded with respect to the MBT timing or the ignition timing suitable for maximum power generation in a particular engine with particular design and operating parameters

The optimum ignition timing for maximum power generation for petrol is -30 degrees BTC while it is -10 degrees BTC for hydrogen because of the different physical and chemical properties for these two fuels.

The power produced by the engine with petrol fuel is more than the power produced by the same engine with hydrogen fuel as the volumetric efficiency of the engine with petrol fuel is more than the volumetric efficiency of the engine in hydrogen mode. Also the stoichiometric air to fuel ratio for hydrogen is 34.3 as compared to 14.6 as that of petrol. Since the engine is operated under slightly rich mode than the stoichiometric value for maximum power generation, the amount of petrol which goes to the engine of the same displacement volume for each cycle is higher than the amount of hydrogen which increases the power output in petrol mode.

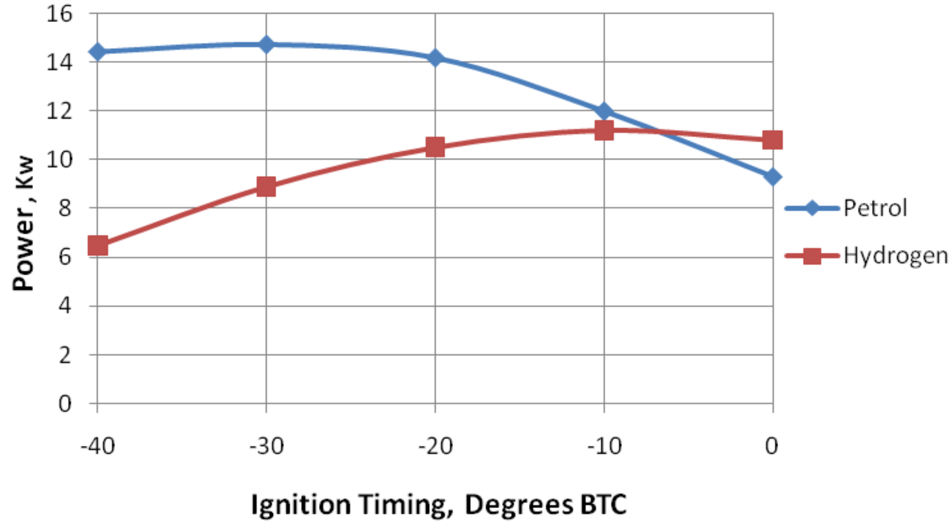

Fig.1 Effect of Ignition Timing on Power

EFFECT OF IGNITION TIMING ON BRAKE SPECIFIC FUEL CONSUMPTION.

Fig.2 below shows the effect of ignition timing on brake specific fuel consumption which is defined as fuel consumed per unit power output. There is an optimum ignition timing at which the BSFC of the engine is minimum with each fuel. The BSFC increases in either direction as the ignition timing is advanced or retarded with respect to the optimum value of ignition timing.

The engine is operated on slightly rich mixtures than the stoichiometric values of 14.6 for petrol and 34.3 for hydrogen for maximum power generation. Although the power output with petrol fuel is on higher side the BSFC is lower for the engine run by hydrogen fuel. This is because the drop in the mass of hydrogen going to cylinder in each cycle in case of hydrogen engine is much larger as compared to the decrease in corresponding power output, which brings down the value of the BSFC for hydrogen engine. The optimum ignition timing for petrol for minimum BSFC is -30 degrees BTC while it is -10 degrees for hydrogen fuel.

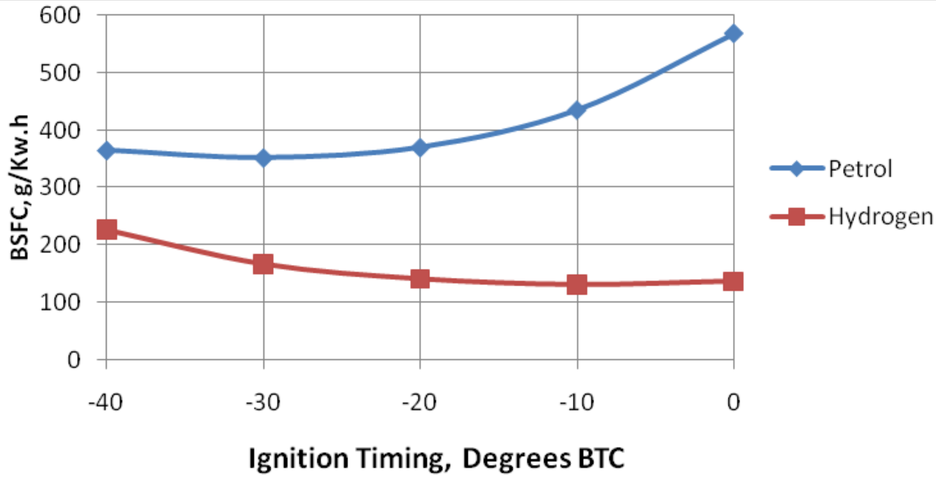

Fig.2: Effect of Ignition Timing on Brake Specific Fuel Consumption

EFFECT OF IGNITION TIMING ON TORQUE.

Fig.3 below shows the effect of ignition timing on torque. It is seen that there is an optimum ignition timing for maximum torque produced by the engine in case of both the fuels petrol as well as hydrogen. The torque decreases if the ignition timing is advanced or retarded with respect to the MBT timing (the maximum brake torque timing) suitable for a particular engine with particular design and operating parameters.

The optimum ignition timing for petrol is -30 degrees BTC while it is -10 degrees BTC for hydrogen because of the different physical and chemical properties for these two fuels.

The torque produced by the engine with petrol fuel is more than the torque produced by the same engine with hydrogen fuel due to the following reasons.

The volumetric efficiency of the engine with petrol fuel is more than the volumetric efficiency of the engine in hydrogen mode due to which more amount of petrol goes to the engine as compared to hydrogen. Also the stoichiometric air to fuel ratio for hydrogen is 34.3 as compared to 14.6 as that of petrol. Since the engine is operated under slightly rich mode than the stoichiometric value for maximum power generation, the amount of petrol which goes to the engine of the same displacement volume for each cycle is higher than the amount of hydrogen.

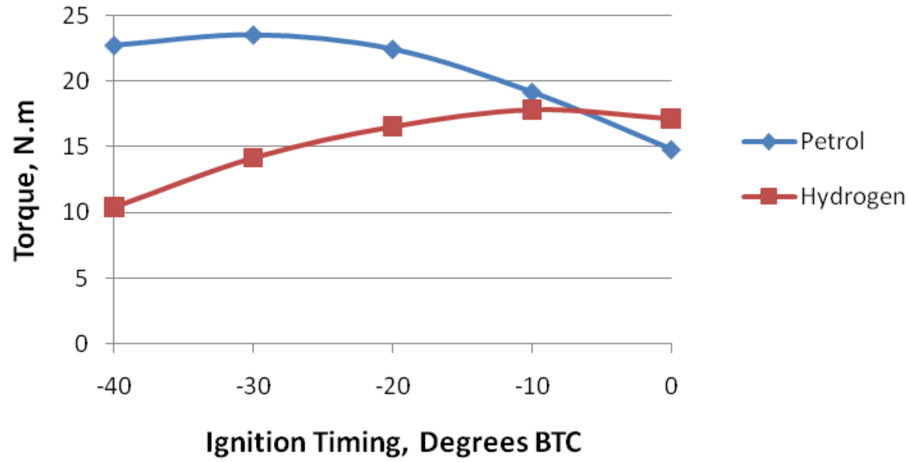

Fig.3 Effect of Ignition Timing on Torque

EFFECT OF IGNITION TIMING ON EXHAUST GAS TEMPERATURE AT EXHAUST VALVE OPENING.

Fig.4 below shows the effect of ignition timing on exhaust gas temperature.

The temperatures of the gases leaving the cylinder at exhaust valve increases by retarding the ignition timing of the mixture with respect to TC in case of petrol, because the energy released by combustion of petrol results in overall increase in the temperature of the gases in cylinder at EVO (exhaust valve opening timing) timing while transferring rest to piston in terms of indicated power and cylinder walls in terms of the heat lost through cooling system of the engine

In case of hydrogen the temperatures of the gases leaving the cylinder at exhaust valve opening first comes down as ignition timing is retarded till -20 degrees BTC and then increases by further retarding the ignition timing of the mixture with respect to TC, because the volumetric efficiency of the engine with respect to ambient first shows a marginal decrease as ignition timing is retarded until -20 degrees BTC and then increases by further retarding the ignition timing.

However this does not necessarily increase the mean effective pressure of the engine in each case as the overall area under the pressure-crank angle diagram, which ultimately represents the work done and hence the torque developed by the engine, is maximum corresponding to the optimum ignition timing or MBT timing for each type of fuel as given in results above.

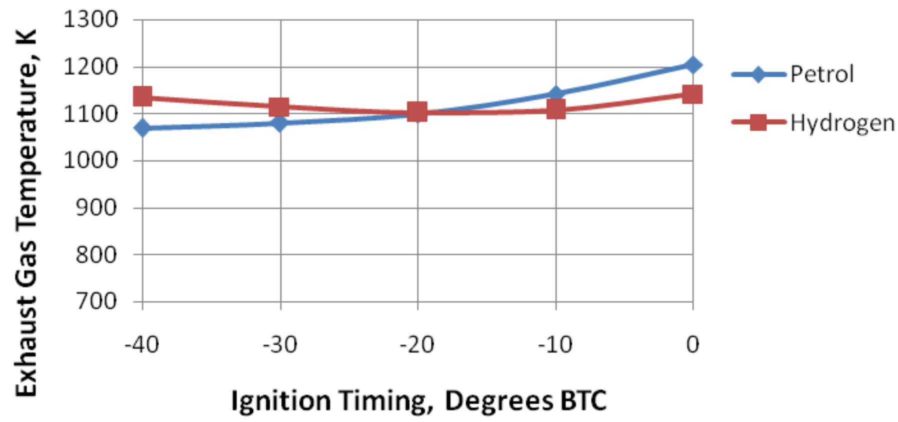

Fig.4 Effect of Ignition Timing on Exhaust Gas Temperature

EFFECT OF IGNITION TIMING ON CO EMISSIONS.

Fig.5 below shows the effect of ignition timing on carbon monoxide emissions.
CO emissions are higher in case of petrol fuel as compared to zero CO emissions in case of hydrogen fuel. Since there are no carbon atoms in hydrogen molecule so no CO emissions are produced by the engine run by hydrogen fuel. As the petrol fuel is broadly represented by octane so far as its chemical characteristics are to be explained, each petrol molecule has 8 carbon atoms in it. On incomplete combustion it results in substantial amount of CO formation.
There is slight increase in CO emissions in case of petrol engine with retard in ignition timing with respect to TC as the amount of time required for complete conversion of carbon into carbon dioxide is reduced for same amount of fuel in each cycle. The overall time for spark, flame initiation, flame propagation and flame termination is reduced.

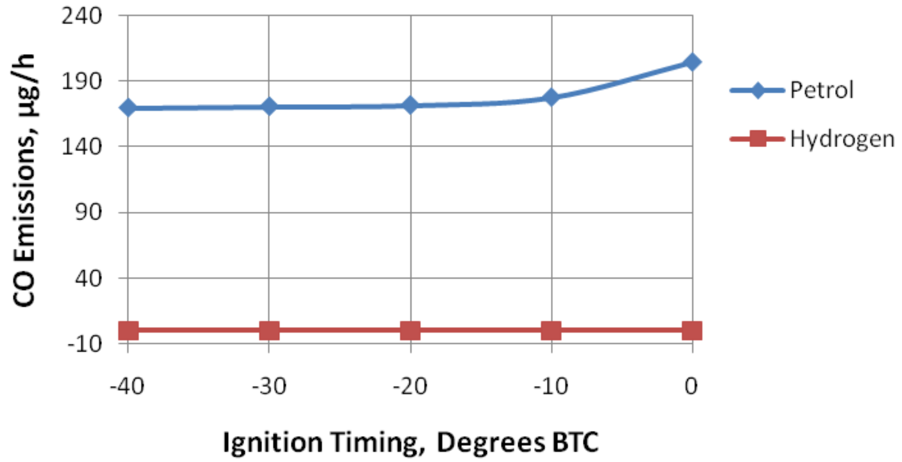

Fig.5 Effect of Ignition Timing on CO Emissions

EFFECT OF IGNITION TIMING ON NO_X EMISSIONS

Fig.6 below shows the effect of ignition timing on NOx emissions. The NOx emissions are higher in case of engine run by hydrogen fuel as compared to petrol fuel under same design conditions. All operating conditions except the air fuel ratio are same in each case. Since the hydrogen engine is run at 34.3 air-fuel ratio as compared to 14.6 to that of petrol engine under stoichiometric combustion conditions, more amount of air and therefore more amount of oxygen and nitrogen goes to hydrogen engine for each cycle which results in higher NOx formation in hydrogen mode

The NOx emissions from the engine are reduced by retarding the ignition timing in case of engine run by petrol fuel as the higher temperatures produced in the gases in the engine cylinder at EVO timing favour the catalytic reduction of NOx emissions in the catalytic convertor. The efficiency of the catalytic convertor gets increased at higher temperatures of exhaust gases close to the light off temperatures of the catalyst.

The NOx emissions leaving the exhaust manifold from the hydrogen engine fairly remain at same higher values as the comparatively lower values of the temperatures of the gases in engine cylinder at EVO timing do not favour the catalytic reduction of NOx emissions in the catalytic convertor.

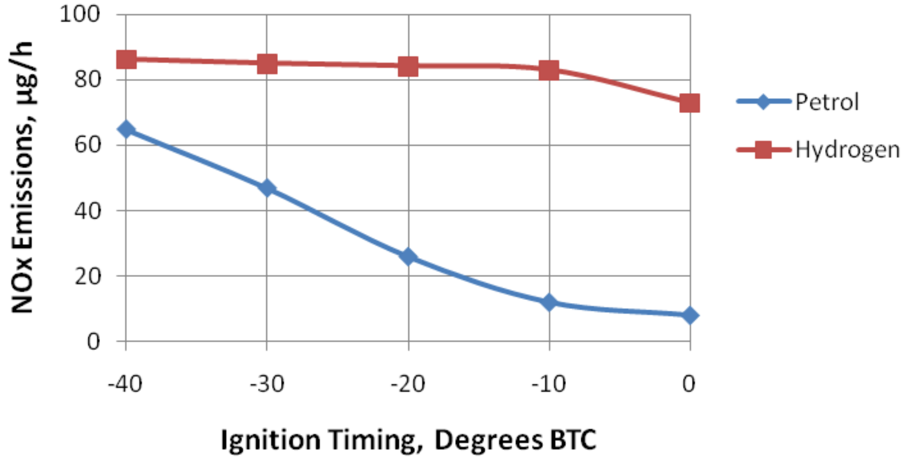

Fig.6 Effect of Ignition Timing on NOx Emissions

EFFECT OF IGNITION TIMING ON HC EMISSIONS

Fig.7 below shows the effect of ignition timing on hydrocarbon emissions from the engine.

Higher HC emissions are formed in case of petrol engine as compared to hydrogen engine under similar design and operating conditions except the air to fuel ratio of the mixture in each case. Hydrogen as such is not a hydrocarbon as it has only hydrogen atoms in its molecule, the hydrocarbon emissions in case of four stroke cycle engine run by hydrogen fuel is due to the leakage or exhaust of the lubricating oil through the exhaust valve while it is in circulation from crank case to engine piston and cylinder walls.

There is marginal increase in HC emissions in case of hydrogen engine by retarding the ignition timing. This is because as the temperatures of the gases in cylinder at EVO timing are increased by retarding the ignition timing, more amount of lubricating oil under circulation gets vapourised and carried away with engine exhaust.

The hydrocarbon emissions from the engine are reduced by retarding the ignition timing in case of petrol fuel as the higher temperatures produced in the gases in the engine cylinder at EVO timing favour the catalytic oxidation of HC emissions in the catalytic convertor. The efficiency of the catalytic convertor gets increased at higher temperatures of exhaust gases close to the light off temperatures of the catalyst

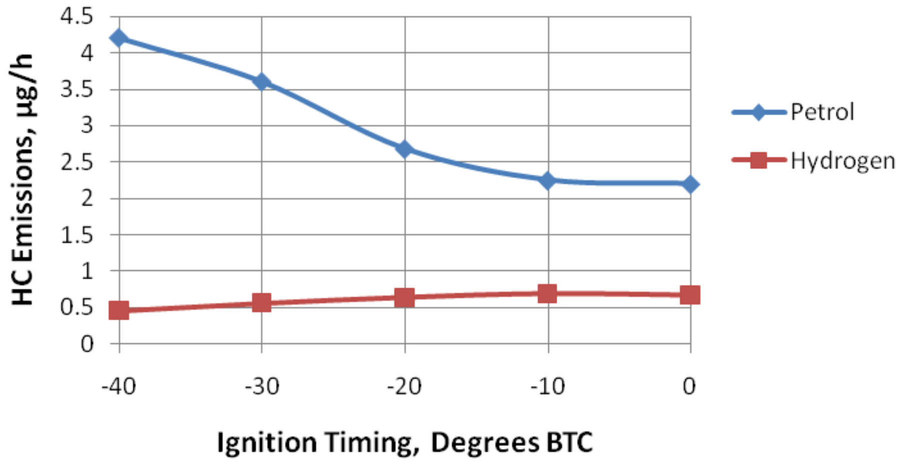

Fig.7 Effect of Ignition Timing on HC Emissions

CONCLUSIONS

1. Hydrogen can be used as an alternative fuel in spark ignition engines used as prime movers although the power produced by the engines run by hydrogen fuel is on lower side when operated in the carbureted mode.
2. The hydrogen engine runs more economically as compared to petrol engine.
3. Marginal HC emissions are produced by hydrogen engine as compared to petrol engine.
4. No CO emissions are produced by the engine run by hydrogen fuel.
5. The NOx emissions produced by the engine run by hydrogen fuel are on higher side as compared to petrol. This can be brought down by using a suitable reduction type of catalytic convertor in the exhaust manifold which shows higher reduction efficiencies at lower light off temperatures.
6. The optimum ignition timing for the engine in petrol mode is -30 degrees BTC while as it is -10 degrees BTC for the same engine in hydrogen mode

ACKNOWLEDGEMENTS

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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

APPENDIX-B

PETROL ENGINE SPECIFICATIONS	
Bore	84 mm
Stroke	90 mm
Compression Ratio	9
Number of Cylinders	1
Operating Speed	6000 rpm

APPENDIX-C

Table 1: **PHYSICAL AND CHEMICAL PROPERTIES OF PETROL AND HYDROGEN [8]**

Fuel Property	Hydrogen	Petrol
Formula	H_2	C_4 TO C_{12}
Molecular weight	2.02	100-105
Lower heating value, MJ/Kg	121	42.5
Stoichiometric air-fuel ratio, weight	34.3	14.6
Octane No.	130	80-98

REFERENCES

[1] Heywood John B., "Fundamentals of internal combustion engines" McGraw Hill International Publication, 1989.

[2] Sierens R etal.“Hydrogen-Fueled Engine”, Journal of Engineering for Gas Turbine and Power, ASME, 2001

[3] Shravan K. etal.“Computational modeling, validation, and utilization for predicting the performance, combustion and emission characteristics of hydrogen IC engines” The Elsevier, Journal of Energy, 2010.

[4] Sierens R. etal.“Experimental Study of a Hydrogen-Fueled Engine” Journal of Engineering for Gas Turbine and Power, ASME, 2001

[5] Li Hailin etal., “Hydrogen Fueled Spark-Ignition Engines Predictive and Experimental Performance” Vol. 128, Transactions of the ASME, 2006.

[6] Verhelst Sebastian etal., “Hydrogen-fueled internal combustion engines” Elsevier Journal of Progress in Energy and Combustion,2009

[7] Bechtold Richard L,” Alternative Fuels Handbook” SAE Publication

[8] AVL LIST GmbH , AVL BOOST Theory, Version 2009.1

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