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Thermodynamic Research Investigations On The Potential Of Green House Gas Emission Reduction By Using Methane As An Alternative Fuel In Conventional Spark Ignition Engines

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ABSTRACT

This paper presents the results of the thermodynamic research investigations on the potential of the green house gas reduction by using methane as an alternative fuel to petrol in spark ignition engine. The results were generated in the professional thermodynamic internal combustion engines simulation software AVL BOOST. The software basically uses the laws of conservation of mass, momentum and energy for generating the thermodynamic results for full cycle simulation of internal combustion engines. The software has the provision of selecting suitable combustion, heat transfer and frictional power models for different types of possible engine designs. First the engine was modelled for a single cylinder spark ignition engine. The engine was run in the petrol mode and the results were generated for the green house gas emissions produced by the engine under variable speed operation. The computational investigations were repeated by using low carbon fuel methane as an alternative fuel for the spark ignition engine. The engine was run with rich air-fuel mixtures with both the fuels under consideration to represent the high load conditions. The engine performance was satisfactory in both the fuel modes. It was seen that the green house gas emissions, from the spark ignition engine under consideration, were drastically reduced with methane fuel as compared to petrol fuel over the entire range of the speed of the engine.

Keywords: .GHG, Carbon dioxide, Global Warming, Petrol, Methane, Spark Ignition, Engine

INTRODUCTION

The following two chemical equations representing the combustion stoichiometry of the two spark ignition engine fuels methane and petrol.

$CH_4 + 2(O_2 + 3.773N_2) = CO_2 + 2H_2O + N_2$

$C_8H_{18} + 12.5(O_2 + 3.773N_2) = 8CO_2 + 9H_2O + 47.1625N_2$

It is seen from these stoichiometric combustion related equations that one molecule of iso-octane, which on an average basis replaces the conventional petrol, produces 8 times the number of GHG mole cules as produced by one molecule of methane.[1]

In Complete combustion, the hydrocarbons in the fuel's chemical bonds are transformed to carbon dioxide (CO_2). Its proportion is also dependent on the operating point. Here again, the proportion depends on the engine operating conditions. The amount of converted carbon dioxide in the exhaust gas is directly proportional to fuel consumption. Thus the only way to reduce carbon-dioxide emissions when using standard fuels is to reduce the fuel consumption. Carbon dioxide is a natural component of atmospheric air, and the CO_2 contained in automotive exhaust gases is not classified as a pollutant. However, it is one of the substances responsible for the greenhouse effect and the global climate change that this causes. In the period since 1920, atmospheric CO2 has risen continuously, from roughly 300 ppm to approx. 450 ppm in the year 2001. This renders the effort to reduce carbon-dioxide emissions and fuel consumption more important than ever.[2]

F. Mundroff et. al., conducted research investigations on direct injection type of internal combustion engines for both gasoline and diesel fuels. It was concluded that direct injection technology reduces the CO_2 emissions by reducing the fuel consumption of the engine.[3]

Adlercreutz, L., et. al., conducted theoretical research investigations on optimizing the combustion concept for CNG combustion on a single cylinder research engine. The overall goal was to reduce the CO2 emissions in g/Kwh by 50% compared to modern gasoline engine, while trying to maintain the performance and characteristics of the engine. It was concluded that downsizing the engine and increasing its specific performance reduced the CO_2 emissions by 45% at full load and 25-34% on part load.[4]

Johnson, T., wrote a review on CO_2 emissions and fuel consumption reductions and the technologies involved from vehicles in the road transportation sector. Regarding the CO2 emissions reduction he writes that a plan was proposed by the United Nations for upwards of 80% CO_2 reductions from the 1990 levels by 2050. Engine technology trends are indicating nominally 15% reductions using advancements in currently utilized technologies. Many of the reductions will come from the use of direct injection technology in gasoline engines and downsizing diesel and gasoline engines for more specific power.[5]

Okmoto, K., et. al., conducted experimental investigations on spark ignition engines used in vehicles for reducing the CO2 emissions as per Japanese 10-15 mode test. A new hybrid fuel technique was adopted in which a fuel additive was used with gasoline to act as a friction modifier. It was found that the fuel economy was improved by 2.4% with a 2.4% reduction in carbon dioxide emissions from vehicles. Further the improvements in engine power and vehicle acceleration were also observed.[6]

Bach, C., estimated that the 20% of all CO2 emissions originate from road traffic. He further conducted studies on the natural gas hybrid passenger cars in order to reduce the CO2 emissions. It was concluded that the use of Hybrid NGVs reduces the CO2 emissions partly because of lower carbon content of the natural gas.[7]

Mueller, W., et. al., conducted experimental investigations with a new family of three way catalyst (TWC) in order to reduce the CO2, HC, CO and NOx emissions from gasoline engines. It was concluded that the modifications in the catalyst design and its tuning with the engine performance was able to reduce the CO2 emissions from the experimental engine.[8]

Warey, A., et. al., conducted analytical investigations on the possible reduction of CO2 emissions by replacing the state of the art four stroke diesel engines by the two stroke engines for light duty vehicle applications. They developed 1-D models for a loop scavenged two stroke diesel engine and an opposed piston two stroke diesel engine. based on these models and the in-house vehicle models, projections were made for the CO2 emissions for a representative light duty vehicle over the New European Driving Cycle and the Worldwide Harmonized Light Vehicles Test Procedure. It was concluded that the loop scavenged two stroke diesel engine had about 5-6% lower CO2 emissions, while the opposed piston diesel engine had about 13-15% potential benefit.[9]

Chiara Guido, et. al., conducted experimental investigations on a 2.0 L Euro 5 compliant diesel engine in NG/Diesel based dual fuel mode. The effect of engine control parameters like CNG substitution ratio, diesel pilot injection strategies, EGR, boost pressure was investigated for possible reduction of GHG emission along with the reduction of other pollutants. It was concluded that there was more than 10% reduction of CO2 emissions besides a significant reduction in THC (Total Hydrocarbon) emissions.[10]

Mufaddel Dahodwala, et.al., conducted computational investigations on advanced after treatment system and two advanced engine models to in order to develop technical knowhow to meet the phase 2 GHG and ultra-low NOx emission standards for heavy duty diesel engines. The advanced after treatment system included DOC, an SDPF, a standalone SCR and an ammonia slip catalyst calibrated against the experimental data. The engine models comprised the advanced technologies like downsizing, down-speeding, variable compression ratio, cylinder deactivation and turbo-compounding. The results indicated that with appropriate selection of engine and after treatment technology packages the 2027 Phase 2 GHG emission standards and the proposed 2024 ultra-low NOx emission standards could be achieved simultaneously.[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($\frac{c}{12.01} + \frac{h}{4.032} + \frac{s}{32.06} - \frac{o}{32.0}$ [kgAir/kgFuel] -------------------------------(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.

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

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

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

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

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

GAS EXCHANGE PROCESS

BASIC EQUATION

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

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

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

$$\frac{dm_c}{d\alpha} = \sum\frac{dm_i}{d\alpha} - \sum\frac{dm_e}{d\alpha}$$ ---(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:

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

$$\psi = \arcsin\left(\frac{e}{r+l}\right)$$ --(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:

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

$c = \ln\{\dfrac{T_{L,TDC}}{T_{L,BDC}}\}$ --(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.

RESULTS AND DISCUSSIONS

EFFECT OF SPEED ON THE POWER DEVELOPED BY THE ENGINE

The fig.1 below shows the effect of speed on the power developed by the engine with petrol and methane fuels. It is clear that the power developed varies with respect to speed as the number of power cycles executed by the engine per unit time increase with the increase in speed. The number of power cycles executed per unit time remains a dominant factor for the power developed by the engine. Again it is seen that the engine develops higher power with petrol as compared to methane. This is because the combined effect of stoichiometric air-fuel ratio, volumetric efficiency and the heating value of fuels is in favour of the petrol as compared to methane fuel.

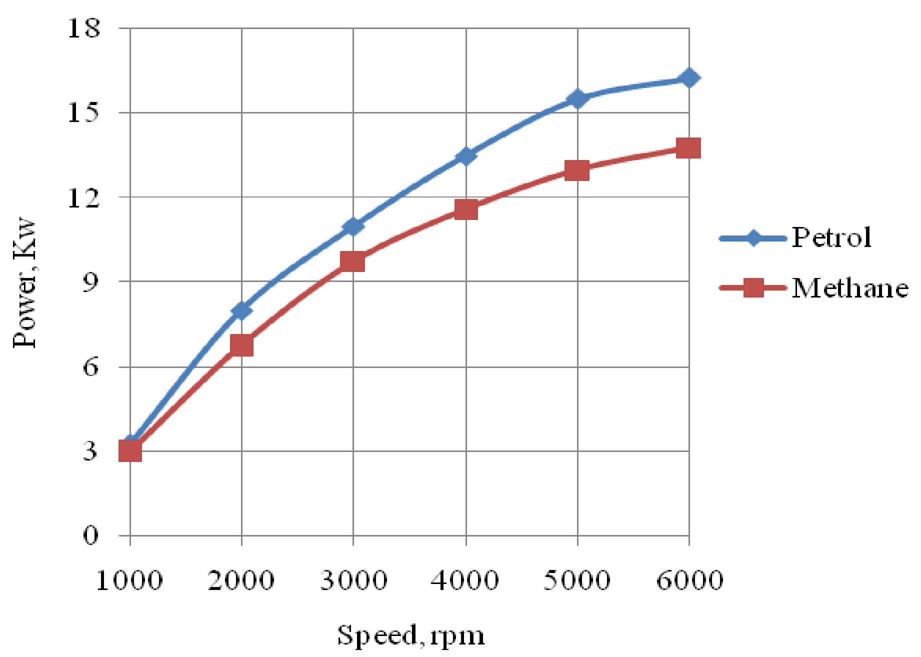

Fig.1 Effect of Speed on The Power Developed by The Engine

EFFECT OF SPEED ON THE TORQUE DEVELOPED BY THE ENGINE

The fig.2 below shows the effect of speed on the torque developed by the engine with petrol and methane as two alternative fuels. It is seen from the graph that for both the fuels under consideration the torque first increases with speed and then comes down with further increase in the engine speed. The reason is that the combustion efficiency of the engine is maximum at the speed of 2000rpm and comes down on the either side of this engine speed. The best possible combustion efficiency results in highest pressure development inside the engine cylinder which ultimately develops maximum torque at the engine crank shaft. Further the engine develops higher torque with petrol fuel as compared to methane as its fuel. This is because the heating value and the stoichiometric air-fuel ratios of the two fuels result in favour of petrol fuel as compared to methane fuel.

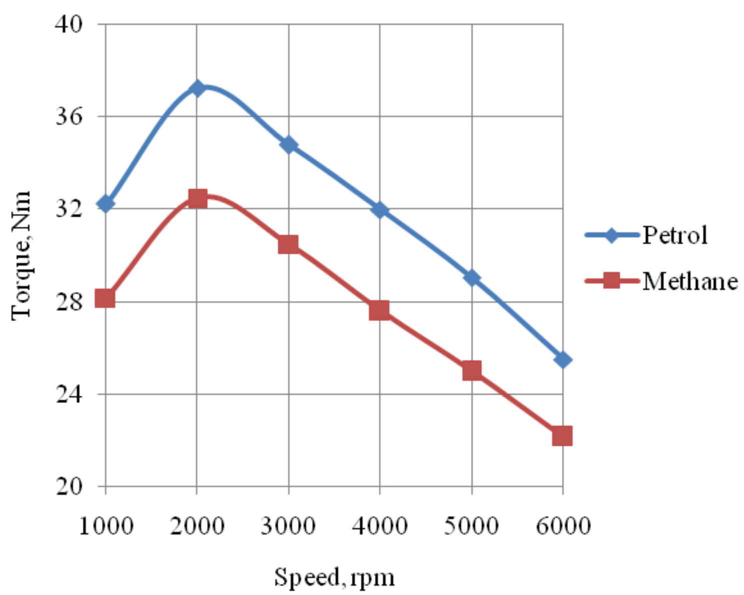

Fig.2 Effect of Speed on The Torque Developed by The Engine

EFFECT OF SPEED ON THE BRAKE SPECIFIC FUEL CONSUMPTION (BSFC) OF THE ENGINE

The fig.3 below shows the effect of speed on the brake specific fuel consumption of the engine with petrol and methane modes. It is seen that the fuel consumption per unit energy output of the engine with either of the two fuels varies with respect to engine speed. The BSFC of the engine first decreases up to the speed of 2000rpm and then increases as the speed of the engine is increased further. This is because the power developed by the engine remains a dominant numerical factor till 2000rpm. Beyond this speed the numerical value of the mass of fuel consumed per unit time remains a dominant factor. Again it is clear that the fuel consumption per unit energy output of the engine is lower with methane fuel as compared to petrol fuel. This is because the combined effect of the heating value of fuel and the stoichiometric air-fuel ratio for the same displacement volume of the engine is in favour of the methane as compared to petrol.

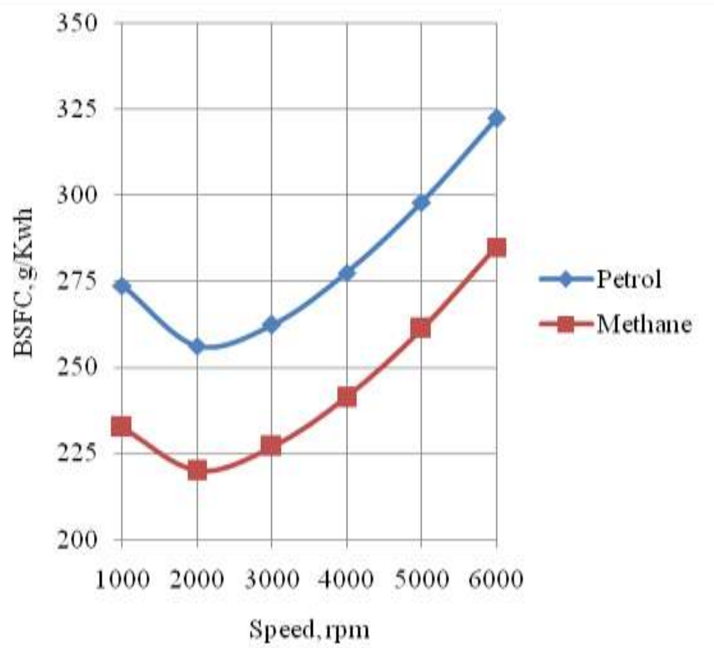

Fig.3 Effect of Speed on The Brake Specific Fuel Consumption of The Engine

EFFECT OF SPEED ON GHG (CO2) EMISSIONS PRODUCED BY THE ENGINE

The fig.4 below shows the effect of speed on the GHG emissions produced by the engine with methane and petrol fuels. It is seen from the graph that the GHG emissions produced by the engine increase with increase in speed with both methane and petrol fuels. This is because of more number of power cycles executed by the engine with higher crank shaft rotational speeds.

Further it is clear that the engine produces lower GHG emissions with methane fuel as compare to the petrol fuel. This is because the methane fuel has only one carbon atom in its molecular structure as compare to the average of eight carbon atoms in the iso-octane representing the petrol fuel.

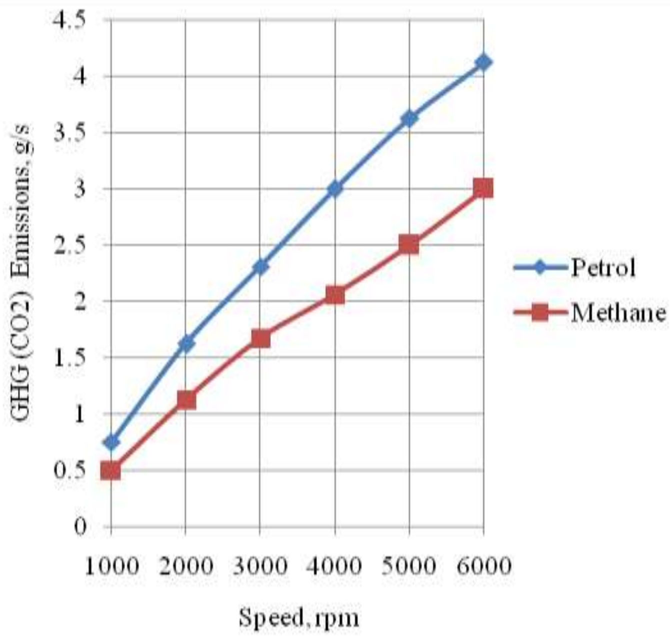

Fig.4 Effect of Speed on The GHG (CO2) Emissions Produced by The Engine

EFFECT OF SPEED ON THE MASS FRACTION OF GHG (CO_2) IN THE EXHAUST GAS OF THE ENGINE

The Fig.5 below shows the effect of speed on the percentage fraction of the GHG in the exhaust gas of the engine. It is clear from the graph that there is marginal increase in the percentage of the GHG emissions in the exhaust gas of the engine, with either of the two fuels, by increasing the engine speed. It is further seen from the graph that on percentage basis the GHG emissions are drastically reduced with methane fuel as compared to petrol fuel.

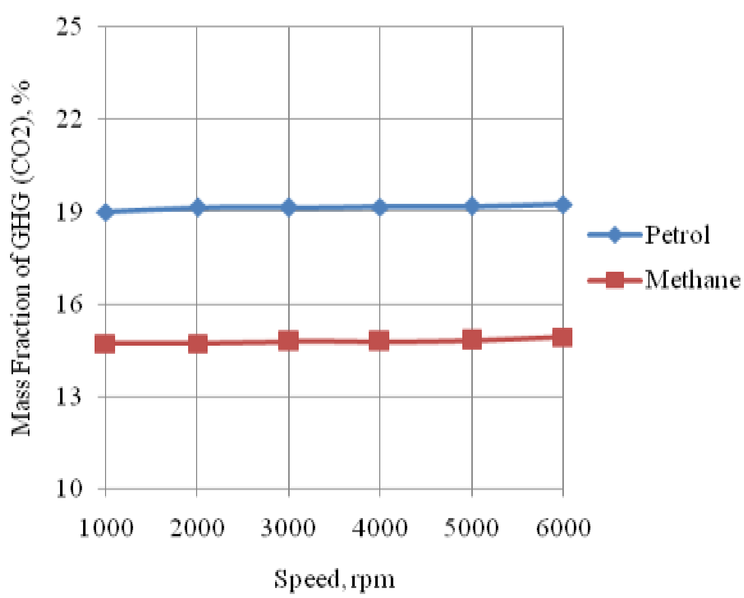

Fig.5 Effect of Speed on The Mass Fraction of GHG (CO2) Emissions

CONCLUSIONS

1. The engine performance in general is satisfactory with methane as its fuel. As compared to conventional petrol the drop in power with methane fuel varies from 7.69% at the speed of 1000rpm to 15.07% at the maximum rated speed of 6000rpm.
2. The engine develops higher torque with petrol than that developed with methane fuel. The drop in the torque developed at engine crank shaft ranges between 12.68% at its lowest speed of 1000rpm to 13.09% at its highest rated speed of 6000rpm.

3. The fuel consumption per unit energy output with methane fuel is lower than that for petrol fuel. The reduction in fuel consumption with methane fuel as compared to petrol fuel varies between 15.06% at the engine speed of 1000rpm and 11.62% at the maximum rated engine speed of 6000rpm.
4. The GHG emissions are reduced by using methane as an alternative fuel to conventional petrol in spark ignition engines. The GHG emissions reduction with methane fuel as compared to petrol fuel vary between 33.33% at the lowest engine speed of 1000rpm to 27.27% at the highest rated speed of 6000rpm.
5. On the basis of mass fraction of the exhaust gas produced by the engine, the GHG emissions are reduced by 5% with methane fuel as compared to the petrol fuel.
6. Replacing the petrol fuel in the spark ignition engines by the low carbon atom methane fuel will help in reducing the alarms of global warming.

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AVL BOOST engine simulation software with license for academic research purposes.

APPENDIX-A

NOMENCLATURE

a = speed of sound, m/sec
A = pipe cross-section, m^2
A_{eff} = effective flow area, m^2
A_i = surface area (cylinder head, piston, liner), m2
AF_{CP} = air fuel ratio of combustion products
A_{geo} = geometrical flow area, m^2
c = mass fraction of carbon in the fuel
c_V = specific heat at constant volume, J/Kg.K
c_p = specific heat at constant pressure, J/Kg.K
Cm = mean piston speed, m/sec
Cu = circumferential velocity, m/sec
c_u = circumferential velocity, m/sec
D = cylinder bore, m
dm_i = mass element flowing into the cylinder, kg
dm_e = mass element flowing out of the cylinder, kg
d_{vi} = inner valve seat diameter (reference diameter), m

$\frac{dm_{BB}}{d\alpha}$ = blow-by mass flow, kg/degree of crank angle
e = piston pin offset, m
E = energy content of the gas (=ρ. $\bar{c}_V.T + \frac{1}{2}.\rho.u^2$) J
f = fraction of evaporation heat from the cylinder charge
F_R = wall friction force, N
h = mass fraction of hydrogen in the fuel
h_{BB} = enthalpy of blow-by, J/Kg
h_i = enthalpy of in-flowing mass, J/Kg
h_e = enthalpy of the mass leaving the cylinder
H_u = lower heating value, J/Kg
k = ratio of specific heats
l = con-rod length, m
m = shape factor
$\dot{m}$ = mass flow rate, kg/sec
m_c = mass in the cylinder, kg
m_{ev} = evaporating fuel, kg
m_{pl} = mass in the plenum, kg
n = mass fraction of nitrogen in the fuel
o = mass fraction of oxygen in the fuel
p = static pressure, bar
P_{01} = upstream stagnation pressure, bar
Pc,o = cylinder pressure of the motored engine ,bar
Pc,1 = pressure in the cylinder at IVC, bar
p_{pl} = pressure in the plenum, bar
p_c = cylinder pressure, bar
p_2 = downstream static pressure, bar
q_{ev} = evaporation heat of the fuel, J/kg
q_w = wall heat flow, J
Q = total fuel heat input, J
Q_F = fuel energy, J
Q_{wi}= wall heat flow (cylinder head, piston, liner), J
r = crank radius, m
R_0 = gas constant, J/mol.K
s = piston distance from TDC, m
t = time , sec
T = temperature, K
Tc,1= temperature in the cylinder at intake valve closing (IVC), K
T_c = gas temperature in the cylinder, K
T_{wi} = wall temperature (cylinder head, piston, liner), K
T_L = liner temperature, K
$T_{L,TDC}$ = liner temperature at TDC position, K

$T_{L,BDC}$ = liner temperature at BDC position, K
T_w = pipe wall temperature, K
T_{01}= upstream stagnation temperature, K
u = specific internal energy, J/Kg
u = flow velocity, m/sec
V = cylinder volume, m^3
V = cell volume (A.dx), m^3
VD = displacement per cylinder, m^3
w = mass fraction of water in the fuel
x = relative stroke (actual piston position related to full stroke)
α = crank angle, degrees
α_o= start of combustion, crank angle degrees
$\Delta\alpha_c$= combustion duration, crank angle degrees
α_w = heat transfer coefficient, $J/m^2.K$
ρ = density, kg/m^3
$\mu\sigma$ = flow coefficient of the port
ψ = crank angle between vertical crank position and piston TDC position, degrees
λf = wall friction coefficient
Δt = time step, sec
Δx= cell length, m

APPENDIX-B

Engine Specifications	
Engine Type	Four Stroke
Method Of Ignition	Spark Ignition
Displacement, cc	500
Compression Ratio	9
Number Of Cylinders	4
Rated Speed, rpm	6000
A/F Ratio Methane Petrol	 17 14

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APPENDIX-C

PHYSICAL AND CHEMICAL PROPERTIES OF PETROL AND METHANE[1]

Fuel Property	Petrol	Methane
Formula	C4 TO C12	CH4
Density, Kg/m3	0.750	0.725
Lower heating value, MJ/Kg	42.5	45
Stoichiometric air-fuel ratio, weight	14.6	17.24
Octane No.	80-98	120
Auto-ignition Temperature, C	280	650

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