COMPUTATIONAL INVESTIGATIONS FOR ASSESSING THE FEASIBILITY OF TURBOCHARGING A MULTICYLINDER SPARK IGNITION ENGINE BY USING PETROL AND METHANE AS ALTERNATIVE FUELS.

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ABSTRACT

This paper presents the results of computational research investigations for turbo-charging a multi-cylinder petrol engine, using Indian commercial petrol and methane as alternative fuels, while trying to improve its performance and emission characteristics.
The investigations were done in the professional thermodynamic simulation software named as AVL BOOST. The various laws of thermodynamics along with many concepts from gas dynamics and mechanics are used in simulating the results using numerical methods for solving the equations involved.
The model was created for proposed multi-cylinder turbocharged spark ignition engine using graphical user interface available in the software.
The model was run in the petrol mode and the octane number requirement was investigated over its entire operating range of speed. Also the corresponding performance and emissions characteristics were also computed.
The investigations were repeated by using methane as an alternative fuel in the designed multi-cylinder turbocharged spark ignition engine mode.
It was seen that both the fuels petrol and methane fulfill the octane number requirement as per the indian commercial petrol and methane at and above the operating speed of 3000 rpm. At lower operating speeds, say from the idle operating speed of the engine up to 3000 rpm, we need to improve the octane ratings of the Indian commercial petrol and methane if used for turbocharged spark ignition engines.
The performance and emissions are otherwise satisfactory with both the fuels used in the turbocharged spark ignition engine. The petrol version gives a better performance as compared to the methane version for the same engine under similar conditions. The HC and NOX emissions are lower with methane as fuel.
Keywords: Engine, Multi-cylinder, Petrol, Methane, Turbo-charging, Octane Requirement, Performance, Emissions.

INTRODUCTION

The concept of turbo-charging for power boosting of engines is well established so far as compression ignition engines are concerned.

The literature available shows that the concept of power boosting of spark ignition engines by the various methods of turbo-charging is limited by the knock that can occur during combustion.

This can be taken care of by testing the use of high octane fuel like ethanol or CNG. However it will not produce highest possible thermal efficiency over entire range of speed and load as the octane requirement is not the same for the entire range of operation of the engine.

In order to achieve a maximum boost in engine power and at the same time to have the highest possible thermal efficiency for each cycle of engine corresponding to entire range of speed and load. we can try to investigate the use of two grades of the same hydrocarbon fuel, say petrol, one having a lower and the other a higher octane rating, in combination under different operating conditions.

The other possibilities are the use of two different hydrocarbons, say by using petrol as a base fuel and adding a required percentage of a high octane fuel like ethanol or methane to meet the overall octane requirement of the fuel in the dual fuel mode over the entire range of speed and load.

Indra F in his paper discusses the progress in the Development of Turbo-Charged Spark Ignition Engines for Passenger Cars. He observed that good results can be obtained with forced-induction engines incorporating four valves per cylinder. The most suitable applications are primarily in the performance car category, or for fast touring saloons.

He further observed that the higher output levels necessitate the use of inter-cooling, sophisticated ignition and injection systems, heat resistant materials and more efficient cooling, all of which make turbo-charged engines more expensive.[1]

Mohanan etal developed the computer simulation methodology for conducting the studies on a methanol fueled turbocharged multi-cylinder automotive spark ignition engine using the concepts of thermodynamics for evaluating its performance and emission characteristics. The computational studies were done with gasoline and methanol as two alternative fuels for two cases involving the original manifolds and the modified manifolds fitted with the turbocharger. The matching of the turbocharger with the engine was also studied.

The results showed an increase in power output, lower nitrogen oxide and carbon monoxide emissions and improved brake specific energy consumption for the methanol fueled engine as compared to the gasoline version for both the original and modified engine designs operating on naturally aspirated and turbocharged conditions.

The available experimental results validated the accuracy of the engine modeling based computational thermodynamic methodology.[2]

Bromberg etal conducted experimental and computer simulation based research investigations for the octane requirement of a turbocharged spark ignition engine in various driving cycles under a wide range of speed and load.

The methodology for the experiments involved the use of high octane PRF fuels and gasoline-ethanol blends after carefully defining the octane limits under different operating conditions.

The above results were used for engine-in-vehicle simulations for calculation of the octane requirements of the models for a passenger car and a medium duty truck under various driving cycles.

The authors also conducted the parametric studies for analyzing the effects of spark retard, engine downsizing at fixed vehicle performance, and vehicle types, on engine efficiency, fuel economy, and ethanol consumption.

It was concluded that the high octane fuel (e.g., E85) effectively suppresses knock, but the octane ratings of such fuels are much above what is required under normal driving conditions.
In view of above the authors further optimized the octane requirement of the engine itself over its full range of operation under each practical driving cycle for a turbocharged engine.
The authors concluded that the average octane ratings of fuel needed in real-world driving were in the 60-80 RON range. The maximum RON required was 90-100. Downsizing and vehicle loading in trucks increased octane requirement substantially. Simulating the results for engine's under required octane based fuels produced by varying the amount of ethanol in the mixture of a dual fuel system, it was observed that it can significantly increase the average engine brake efficiency (about 30% increase) and fuel economy (about 26%) depending on driving details. The above increased ethanol substitution could be brought down by retarding the spark timing by 5 crank angle degrees without compromising with the efficiency.[3]
Baratta etal conducted experimental investigations on a 4 cylinder turbocharged spark ignition production engine to assess its performance and emissions with three fuels namely CNG and CNG/Hydrogen Blends (15% and 25% in volume of H_2).
The methodology involved the modification of the engine into a dedicated CNG engine.
The first part of the experimental investigations were carried out at MBT timing under stoichiometric conditions. The experiments were conducted at constant speed under variable load operation first followed by constant speed operation at variable load.
Investigations were also conducted by varying the spark timings and relative air-fuel ratios and at the same time maintaining the speed and load constant.
The cyclic and cylinder to cylinder variations were also investigated by recording pressure traces.
It was concluded that the addition of hydrogen demands retarding the spark timing for developing maximum brake torque under stoichiometric conditions.
The addition of hydrogen resulted in lowering the brake specific fuel consumption due to higher heating value of hydrogen. This ultimately increased the fuel conversion efficiency of the engine. There was reduction in total hydrocarbon emissions and CO emissions with increased substitution of hydrogen under duel fuel operation. However the NO_x emissions were increased.
The addition of hydrogen resulted in an increase in the lean operation limit of the engine with respect to CNG operation.
There was significant cylinder-to-cylinder variations with all the fuels under consideration due to non uniform spatial distribution of fuels among various cylinders which in turn affects the effectiveness of the cooling system and the uniform air/fuel mixture distribution.[4]

Bourhis etal conducted research investigations on a tubocharged spark ignition engine in a dual fuel mode to optimize the octane number requirement of the fuel by varying the fuel properties and the engine fuel injection parameters.
The methodology involved the utilization of the concepts about knock in spark ignition engines that the efficiency of spark ignition engines is limited towards high loads by the occurrence of knock, which is linked to the octane number of the fuel. Further running the engine at its optimal efficiency requires a high octane number at high load, whereas a low octane number can be used at low load.
The occurrence of knock can be avoided by adjusting the octane number (RON) of the fuel injected in the combustion chamber by developing the concept of "Octane on Demand" (OOD).
The logic explained above will help to operate the engine which will practically have best possible thermal efficiency for each cycle at the corresponding operating load for that particular cycle.

The above concept was implemented by a dual fuel injection strategy involving a low-RON base fuel and a high-RON octane booster. The ratio of fuel quantity on each injector is adapted to fit the RON requirement function of engine operating conditions.
As per the authors this OOD concept entails a good characterization of the octane requirement needed to run the engine at its optimal efficiency over the entire map.
To achieve the best fuel combination concept the authors performed the engine testing on direct and indirect injection multi-cylinder engine by using the fuel combinations involving three different octane boosters namely ethanol, reformate and a blend of butanol isomers (SuperButolTM)) with two base fuels namely a very low-RON naphtha-based fuel (RON 71) and a non-oxygenated gasoline (RON 91)).
The above dual fuel injection based OOD concept was tested by using both GDI and PFI mixture preparation techniques.

Results reveal that the injection configuration has quite a low effect on the octane booster demand (to keep the engine at its optimal combustion phasing). Ethanol is confirmed to be a remarkable octane booster, for its high latent heat of vaporization and its favorable sensitivity (defined as RON – MON). It is also shown that the base fuel, compared to the high octane fuel, has a weaker effect on the octane booster requirement, promoting the use of a less processed naphtha-based low-RON fuel (71) as a base fuel.
0D-simulation results, based on experimental data, reveal that a consumption around 25 %v/v of octane booster on the WLTP cycle is sufficient to keep the engine running on its optimal efficiency with a 71 RON base fuel. This leads to about 4% CO_2 savings over the WLTP cycle (the more loaded the driving cycle, the larger the CO_2 gains). [5]

THEORETICAL BASIS.[6]

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:

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

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:

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

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.

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.

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

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

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

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

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

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:

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

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

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:

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

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

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

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

RESULTS AND DISCUSSIONS.

EFFECT OF SPEED ON OCTANE NUMBER REQUIREMENT.

The Fig.1 below shows that higher octane number rated petrol and methane are needed at low speeds as compared to high speeds.

Comparatively higher pressures are developed in each cycle at lower speeds as compared to that at higher speeds. So as per the knock model considered the ignition delay is lower at lower speeds which necessitates the use of a high octane number fuel. Also since lower pressures are developed at higher speeds on cycle basis so as per the equation for knock model the ignition delay increases. This requires a lower octane number fuel at high speeds since there are no chances of knock due to self ignition of the entire mixture or only a part of end gas mixture.

This is basically because the combustion characteristics and therefore thermal efficiency is better at lower speeds than at higher speeds. At lower speeds there is sufficient time for proper combustion as well as proper flame propagation for better or complete combustion resulting in increasing trend of pressure development up to the speed of maximum torque condition. The pressure developed beyond this speed are still higher than that developed at rated speed and other intermediate speeds.

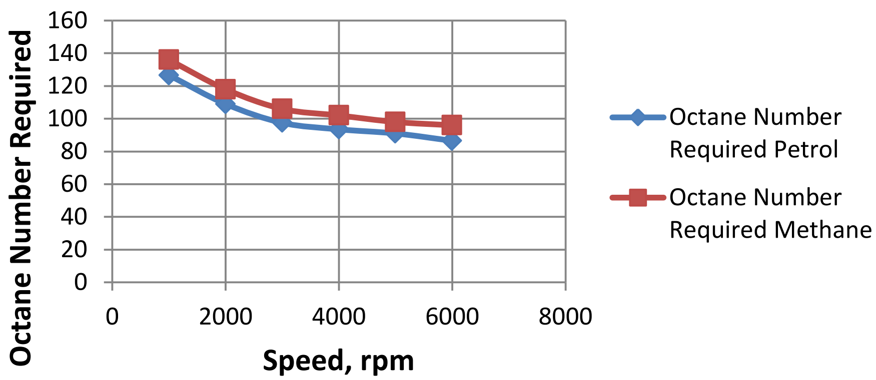

Fig.1 Effect of Speed on O.N. Requirement

EFFECT OF SPEED ON ENGINE POWER

The Fig.2 below show that as the speed is increased the power also increases with both fuels petrol and methane due to more number of power cycles per unit time at higher speeds. The power produced in the petrol mode is higher than the power produced with methane as fuel due to lower volumetric efficiency with gaseous fuel methane.

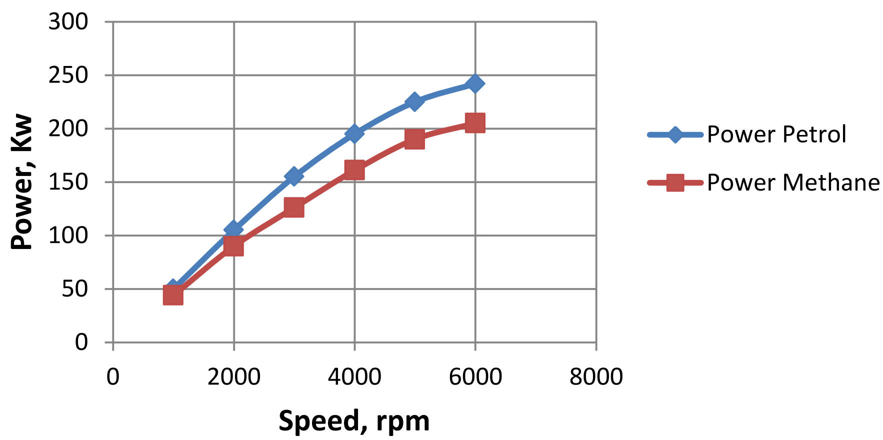

Fig.2 Effect of Speed on Power

EFFECT OF SPEED ON ENGINE TORQUE.

The Fig.3 below shows that with both fuels petrol and methane the torque first increases and then decreases with respect to speed, this is because the overall combustion characteristics are best at the maximum torque condition which gives rise to higher peak pressures at the spee for maximum torque.

Also the torque produced by petrol as fuel is on the higher side than that produced with methane as fuel. This is because of overall higher pressures generated during entire cycle in the petrol mode as compared to the methane mode due to higher volumetric efficiency achieved with petrol as fuel as compared to methane as fuel. Higher volumetric efficiency with petrol sends additional amount mixture of petrol and air when compared with methane for the same displacement volume with the corresponding values of operating air to fuel ratios being very close.

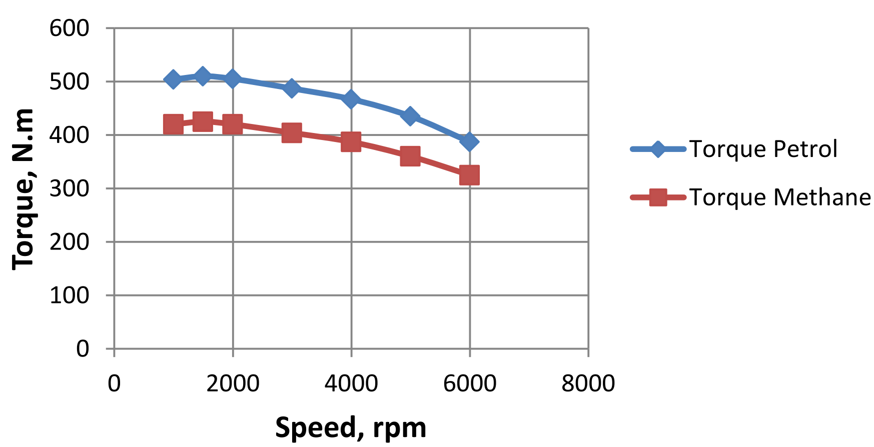

Fig.3 Effect of Speed on Torque

EFFECT OF SPEED ON BRAKE SPECIFIC FUEL CONSUMPTION.

The Fig.4 below shows that with both fuels the values of the brake specific fuel consumption first decreases with speed and then increases as the speed is further increased. The value of the brake specific fuel consumption is minimum at the speed of 2000 rpm. This speed is also the condition for maximum torque where the combustion efficiency is best for any cycle at that speed.

Also the values of BSFC are lower with petrol as fuel as compared to methane.

This is because of the higher power development in the petrol mode which predominates other operating parameters like rate of fuel consumption.

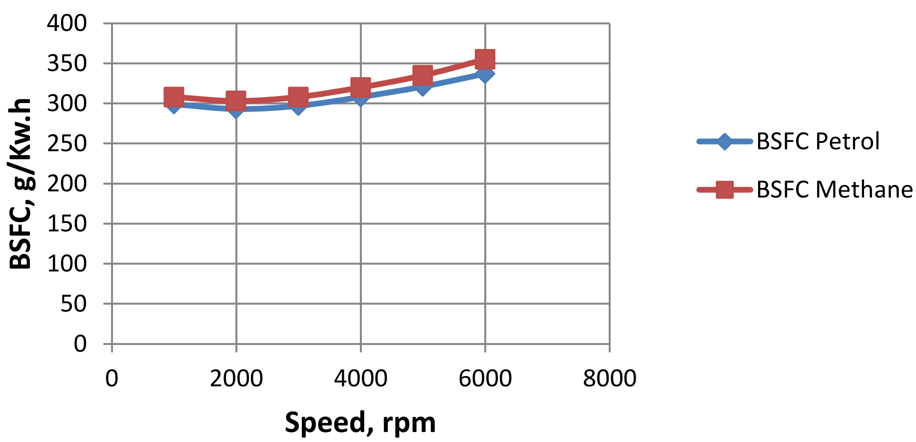

Fig.4 Effect of Speed on BSFC

EFFECT OF SPEED ON UN-BURNT HC EMISSIONS.

The Fig.5 below shows that the HC emissions increase as the speed increases with both petrol and methane as fuel. This is due to more number of power cycles per unit time at higher speeds.

The HC emissions are on the higher side with petrol mode as more amount of petrol goes to engine for each cycle due to higher volumetric efficiency with petrol as fuel.

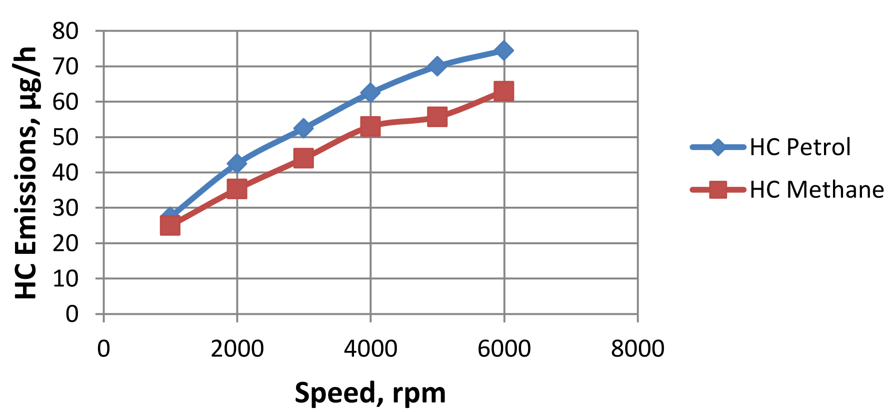

Fig.5 Effect of Speed on HC Emissions

EFFECT OF SPEED ON NOX EMISSIONS.

The Fig.6 below shows that with both fuels petrol and methane the NOx emissions increase with speed due to more number of power cycles at higher speeds.

Also the NOx emissions produced with petrol as fuel are much higher than that produced with methane as fuel. Since the NOx emissions are produced from the oxygen and nitrogen of air only. Due to higher volumetric efficiency in the petrol mode, more amount of air goes to engine for each cycle in the petrol mode, which contributes higher NOx emissions in the petrol mode as compared to the methane mode. Also higher peak temperatures are produced in the cylinder in the petrol mode as compared to methane mode which also is essential condition for the higher rate of NOx formation in engines.

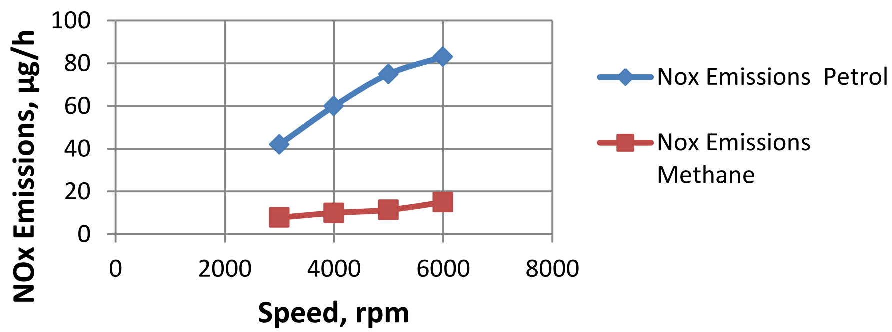

Fig.6 Effect of Speed on NOx Emissions

EFFECT OF SPEED ON EXHAUST GAS TEMPERSATURE.

The Fig.7 below shows that with both petrol and methane as fuels the exhaust gas temperature increases with speed due to more number of power cycles per unit time at higher speeds.

Also higher exhaust gas temperatures are produced with petrol as fuel as compared to methane. This is due to more amount of petrol being consumed for each cycle due to the higher volumetric efficiency in the petrol mode.

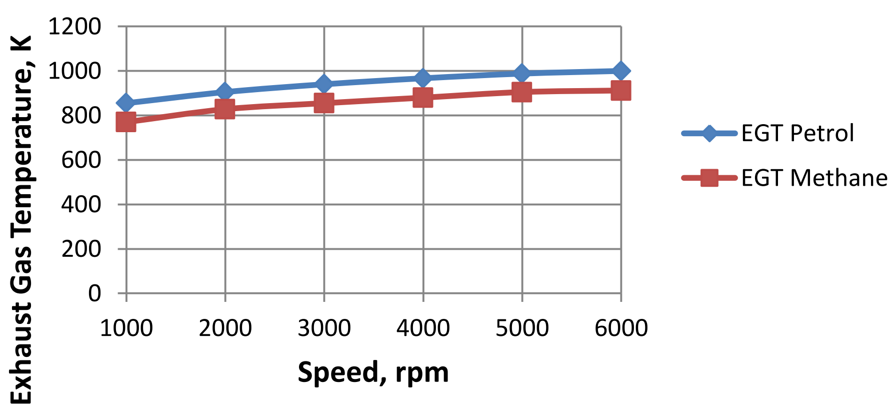

Fig.7 Effect of Speed on Exhaust Gas Temperature.

CONCLUSIONS

1. It is seen that both the fuels petrol and methane fulfill the octane number requirement beyond the operating speed of 3000 rpm onwards as per the octane ratings of Indian commercial petrol and methane
2. It is proposed that the Indian petro-chemical industries may produce petrol and methane with higher octane ratings as desired by the computational results above to suit the requirements of turbocharged spark ignition engines at low speeds up to idle rpm of the engine.
3. It is further proposed that a micro-controller based fuel injection system should be designed to supply those grades of petrol and methane with the required octane numbers corresponding to variable speed operation.
4. It is further concluded that for constant speed operation of turbocharged spark ignition engines as may be used in power generation either the operating speed of the generator set could be designed at 3000 rpm or else for a lower operating speed petrol or methane with the corresponding required octane number should be produced in petroleum refineries.
5. The initial cost of turbocharged diesel engines is high. The naturally aspirated version of this class of engines is not acceptable in market because of inferior comparative performance on the basis of total displacement volume.

   The cheaper alternative is the naturally aspirated spark ignition engine which needs comparative displacement volume as that of the turbocharged diesel engine version for the same power.

   Turbo-charging a spark ignition engine will improve its performance further but will also result in increasing its initial cost. This cost rise may not be acceptable to the low investors in both domestic and commercial applications.

   Hence it is proposed that a micro-controller based dual fuel turbocharged park ignition engine can be designed with high octane methane as fuel at lower speeds up to 3000 rpm and either methane or lower octane fuel petrol as fuel at higher speeds up to the rated speed of the engine.

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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

ENGINE SPECIFICATIONS	
Engine Type	V6 TCI Bi-Turbo
Method of Ignition	Spark Ignition
Bore	82 mm
Stroke	84.6 mm
Compression Ratio	8.5
Number of Cylinders	6
Rated Speed	6000 rpm

APPENDIX-C

PHYSICAL AND CHEMICAL PROPERTIES OF PETROL AND METHANE[7]

Fuel Property	Petrol	Methane
Formula	C4 TO C12	CH4
Density, Kg/m3	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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