EFFECT OF AIR-FUEL RATIO ON THE OCTANE NUMBER REQUIREMENT, PERFORMANCE AND EMISSION CHARACTERISTICS OF A CONSTANT SPEED V6 TCI BI-TURBO SPARK IGNITION ENGINE USING PETROL AND METHANE AS ALTERNATIVE FUELS.

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ABSTRACT

This paper presents the results of computational research investigations on a V6 TCI BI-TURBO spark ignition engine, using petrol and methane as alternative fuels under variable load and constant speed operation. The aim of these investigations is to explore the successful possibilities of turbo-charging the spark ignition engines needed for power generation and other constant speed operations.
Emission characteristics. This will help in the compact design of such a category of engines.
The investigations were done in the professional thermodynamic simulation software named as AVL BOOST. This software uses the concepts and laws of thermodynamics and gas dynamics for solving engine related problems. Models for combustion analysis, frictional power calculations, heat transfer analysis etc are used for assisting in solving the problems while trying to arrive at required conclusions. The numerical methods are used for solving the equations involved.
The model for a V6 TCI BI-TURBO spark ignition engine was selected for the above mentioned research investigations.
The model was first run in the petrol mode under constant speed operation. The air-fuel ratio was varied during these simulations which actually represents the variable load condition for the same. The results were generated for the octane number requirement as well as performance and emissions characteristics.
The investigations were repeated by using methane as an alternative fuel to petrol for the above engine under similar running conditions.
The computational investigations on the engine shows successful performance with both petrol and methane as fuels while demanding much higher octane ratings for both of these fuels than available commercially.
The petrol version of the engine produces more power than the methane version whereas the brake specific fuel consumption of the engine in the petrol mode is lower than that in the methane mode.
The petrol version of the engine produces higher CO emissions than methane version whereas the methane version of the engine produces lesser HC and NOx emissions than the petrol version.
Keywords: Multi-cylinder Engine, Spark Ignition, Petrol, Methane, Turbo-charging, Constant Speed, Variable Load ,Octane Requirement, Performance and Emissions.

INTRODUCTION

The concept of turbo-charging a spark ignition engine is aimed to downsize the engine for the same power or torque to be developed as compared to a naturally aspirated spark ignition engine. In other words turbo-charging an engine having a certain displacement volume will increase its power and torque and at the same time reduce its brake specific fuel consumption. Both the above designs will find their own particular applications depending whether weight and size reduction for a particular power is required or performance improvement for the same size is required.

Turbo-charging the spark ignition engines faces knock based combustion problems both under variable load and constant speed operation as well as in the variable speed mode with commercial petrol as available in the market. In order to commercialize the turbo-charged spark ignition engines successfully, modifications in the engine design and its operating conditions are needed to be done. Further high octane rated fuels need to be produced and commercialized which will prevent the occurrence of knock, throughout the operating range of these engines, at the thermodynamic properties of the charge in the cylinder during combustion process.

John B. Heywood in the various chapters of his text book writes that most modern turbocharged engines use port fuel injection. This provides easier electronic control of fuel flow and improves the dynamic response of the system by reducing fuel transport delays. The bmep of most production spark-ignition engines at wide-open throttle is knock-limited over part of the engine speed range. The compression ratio (8-12) is usually set at a sufficiently high value so that some spark retard from MBT timing is needed to avoid knock for the expected range of available fuel octane rating and sensitivity. If the end of combustion process is progressively delayed by retarding the spark timing, the peak cylinder pressure occurs later in the expansion stroke and is reduced in magnitude. Attempts to boost the output of a given size spark ignition engine by an inlet air compression device that increases air pressure and temperature will aggravate the knock problem, since the end gas pressure and temperature will increase. However the higher output for a given displacement volume will decrease engine specific weight and volume. Also for the same maximum power, the smaller turbocharged engine should offer better fuel economy at part load. At a given part-load torque requirement, the mechanical efficiency (lower frictional loss) of the smaller turbocharged engine is higher. The variables that are adjusted to control knock in turbocharged SI engine are :compression ratio, spark retard from optimum, charge air temperature (inter-cooling), and fuel-air equivalence ratio. Most turbocharged SI engines now use a knock sensor and ignition-timing control system so that timing can be adjusted continuously to avoid knock without unnecessary retard. With a knock sensor, sensing above-normal vibration levels on the cylinder head, ignition timing can be automatically adjusted in response to changes in fuel octane rating and sensitivity, and ambient conditions. The occurrence of knock at high speeds corresponding to WOT for such engines is avoided by reducing the exhaust flow through the turbine as speed increases by bypassing a substantial fraction of the exhaust around the turbine through the wastegate or flow control valve. Turbo-charging the naturally aspirated 2.3-dm3 engine results in a 36 percent increase in maximum engine torque under certain conditions. In a vehicle context, the low-speed part-load advantage of the smaller size but equal power turbocharged engine should result in an average fuel economy benefit relative to the larger naturally aspirated engine. Knock is a phenomenon that is governed by both engine and fuel factors; its presence or absence in an engine depends on the antiknock quality of the fuel. Individual hydrocarbon compounds vary enormously in their ability to resist knock, depending on their molecular size and structure. Practical fuels are blends of a large number of individual hydrocarbon compounds. The octane number requirement of an engine or vehicle or vehicle-engine combination is defined as the maximum fuel octane number that will resist knock throughout the engine's operating speed and load range. [1]

Bourhis etal in his paper describes the research investigations on the effect of properties of the fuels used and engine injection configuration effects on the octane on demand concept for a dual fuel turbocharged spark ignition engine .

The methodology used involved using a dual fuel injection strategy, involving a low-RON base fuel and a high-RON octane booster. Both GDI and PFI concepts were tried on the above mentioned multi-cylinder engine for these investigations.

The percentage by volume of two fuels on each injector was regulated to fit the RON requirement function of engine operating conditions.

To find the best fuel combination a very low-RON naphtha-based fuel (RON 71) and a non-oxygenated gasoline (RON 91) were used as base fuels.

The three different octane boosters tried were ethanol, reformate and butanol isomer.

The results indicate that the injection configuration GDI or PFI has quite a low effect on the octane booster demand needed to keep the engine at its optimal combustion phasing.

0D-simulation results, based on experimental data, revealed that the substitution of about 25 % by volume of octane booster on the WLTP cycle is sufficient to keep the engine running on its optimal efficiency with a 71 RON base fuel

The results showed about 4% savings on CO2 emissions over the WLTP cycle. The savings on CO2 emissions increase further with the increase in load on the, WLTP, driving cycle considered in the investigations.[2]

Amer etal conducted simulation tests on a a single cylinder direct injection spark ignition (DISI) engine to investigate the effect of fuel properties, like octane number, on its knock characteristics under turbocharged conditions. In addition fuel effects on particulate emissions at part-throttle were measured.

The methodology used involved the use of different fuels having RON in the range of 95 and 105. Different configurations of Turbochargers for raising the inlet air pressure up to 3.4 bar absolute were also used in simulating the results.

The authors concluded that with the boosting levels tried in the investigations there is potential for downsizing a 3.2-liter engine to 1.5 liters using a 2-stage turbo-charger.

The authors further conducted vehicle based simulations after incorporating the above engine based achievements and observed that it reduces the fuel consumption of the vehicle by 16% with the base fuel of 95 RON which increases up to 19% with a fuel of RON of 99.6.[3]

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.[4]

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.[5]

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.[6]

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.[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:

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

STOICHIOMETRIC AIR-FUEL MIXTURE

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

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

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.

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

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

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

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:

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

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

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:

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

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

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

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

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:

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

C1 = 2.28+0.308.cu/cm

C2 = 0.00324 for DI engines

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

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

$C_3 = 6.18+0.417.c_u/c_m$

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.

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:

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

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

KNOCK MODEL

IGNITION DELAY AND OCTANE NUMBER REQUIREMENT.

AVL Boost uses the following equation based model 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 AIR-FUEL RATIO ON OCTANE NUMBER REQUIREMENT.

The Fig.1 below shows that octane number requirement with both the fuels first increases with air-fuel ratio and then decreases under constant speed operation.

In case of petrol the octane requirement ranges between 120.9 and 120.7. The highest octane requirement is at an air-fuel ratio of 13 corresponding to maximum torque generation.

In case of methane as a fuel the octane requirement ranges between 124.5 and 129.75. The highest octane requirement with methane fuel is at an air-fuel ratio of range of 15 and 16 corresponding to the condition of maximum torque generation.

The maximum pressure and temperature are developed in the engine cylinder at the same value and range of air-fuel ratio considered above for both the fuels under consideration. The pressure and temperature developed in the engine cylinder decrease on either side of the peak based value of air-fuel ratio mentioned above.

Since chances of knock for any particular fuel during combustion increases with the increase in pressure and temperature developed in both burned and unburned zones in the case of two zone combustion modeling for an engine cylinder, so octane demand also increases accordingly. The unburned zone pressure and temperature are more important as this can lead to faster development of pre-flame reactions in the air-fuel mixture of the unburned zone. The higher values can also lead to sudden and earlier auto-ignition of the end gas mixture in the unburned zone.

The octane ratings of methane needed for the engine under consideration are in the range of 126.5 and 129.75. Also the octane ratings of petrol needed in the petrol version of the engine are in the range of 120.73 and 120.96.

The commercially available petrol and methane have octane ratings of 80-98 and 120 respectively. The octane ratings of both of these two fuels are to be upgraded if used for power generation with turbocharged spark ignition engine under consideration.

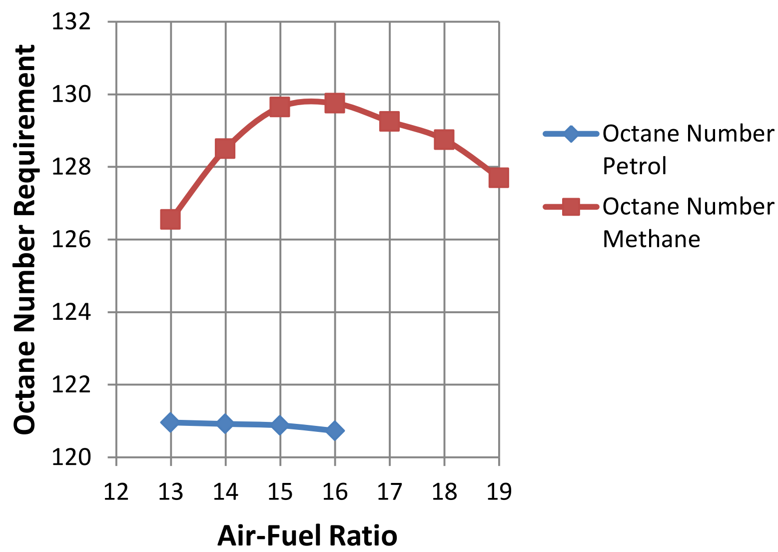

Fig.1 Effect of Air-Fuel Ratio on Octane Number Requirement

EFFECT OF AIR-FUEL RATIO ON ENGINE POWER

The Fig.2 below shows the effect of air-fuel ratio on engine power with petrol and methane as fuels for the engine under consideration.

In case of petrol the power first increases up to the air-fuel ratio 13 and then again decreases. The higher power is developed with this value of air-fuel ratio for rich mixture since the combustion of petrol in cylinders tries to consume entire amount of air available leaving no further scope of improvement.

In case of methane the power first increases up to the air-fuel ratio 15 and then again decreases. The higher power is developed with this value of air-fuel ratio for rich mixture since the combustion of methane in cylinders tries to consume entire amount of air available leaving no further scope for improvement under design and operating conditions considered.

More power is developed with petrol as fuel as compared to methane. Under the operating considered for this engine the volumetric efficiency achieved with petrol varies from 193.92 to 194.14 as compared to the range of 170.9 to 172.3 achieved with methane as a fuel. This drop in volumetric efficiency together with higher stoichiometric value of air-fuel ratio for methane lowers the power that can be developed by the same engine.

Although the lower heating value of methane is higher than petrol but the less power is developed with methane as a fuel. This is due to large drop in volumetric efficiency with this fuel which means less amount of charge goes to engine cylinders which dominates the drop in power.

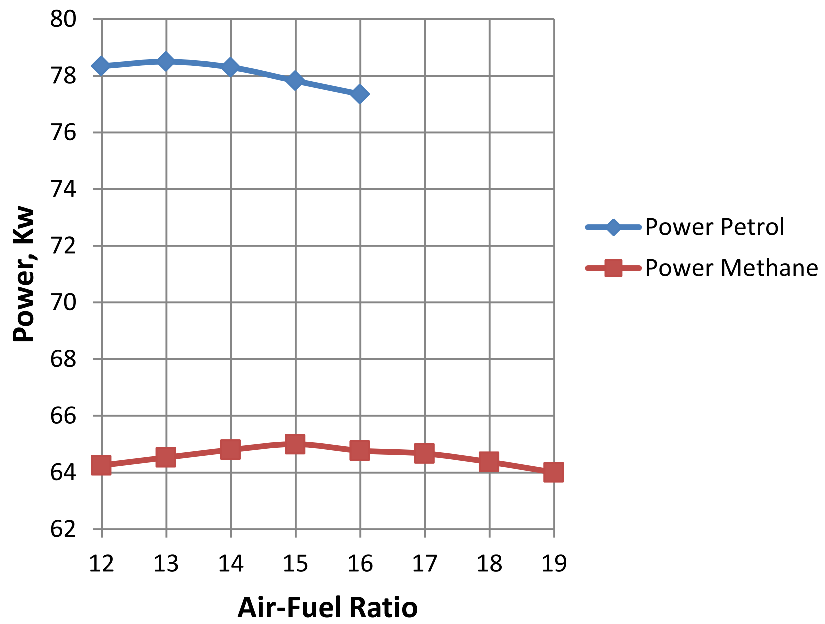

Fig.2 Effect of Air-Fuel Ratio on Power

EFFECT OF AIR-FUEL RATIO ON ENGINE TORQUE.

The Fig.3 below shows the effect of air-fuel ratio on engine torque with both petrol and methane as fuels.

In case of petrol version of the engine under consideration the torque increases up to air-fuel ratio of 13 and then decreases. This is because the overall conditions for combustion are best at this value of air-fuel ratio which leads to development of higher pressures responsible for maximizing the forces acting at crank shaft for torque development.

In case of methane version of the engine under consideration the torque increases up to air-fuel ratio of 15 and then decreases. This is because the overall conditions for combustion are best at this value of air-fuel ratio which leads to development of higher pressures responsible for maximizing the forces acting at crank shaft for torque development.

The torque developed in case of petrol version is higher than the torque developed in case of methane version. The reason for this that more amount of charge goes to petrol version of engine due to higher volumetric efficiency in this mode as compared to methane version.

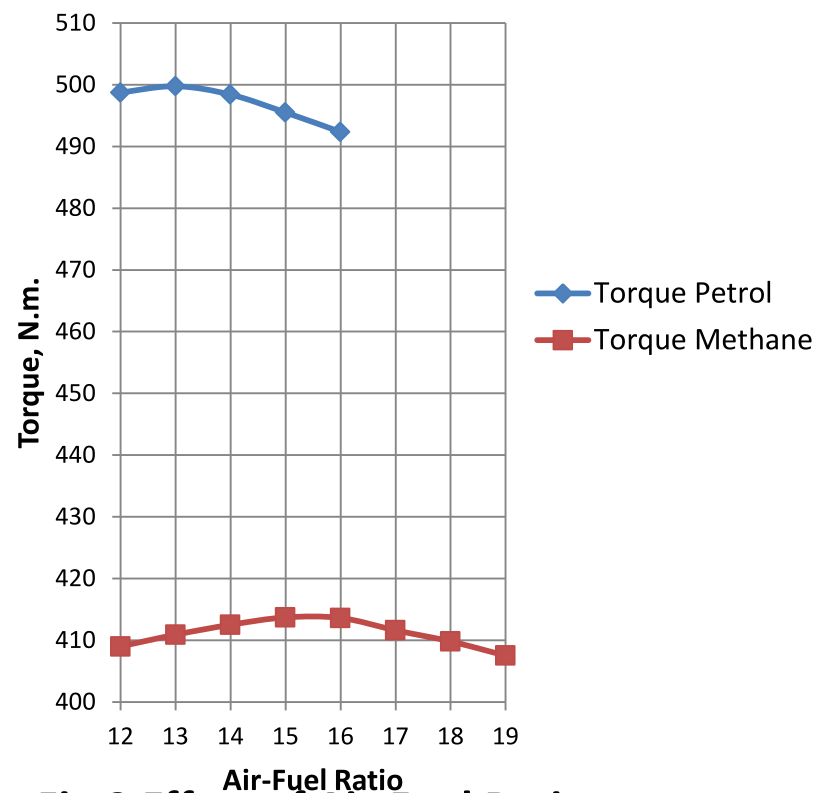

Fig.3 Effect of Air-Fuel Ratio on Torque

EFFECT OF AIR-FUEL RATIO ON BRAKE SPECIFIC FUEL CONSUMPTION.

The Fig.4 below shows the effect of air-fuel ratio on the brake specific consumption of the engine using petrol and methane as fuels.

The brake specific consumption of the engine in petrol version decreases up to air-fuel ratio of 13 since the power developed increases and the rate of petrol aspirated decreases. The brake specific fuel consumption of the engine in petrol version decreases beyond the air-fuel of 13 because the rate of decrease of petrol supplied to the engine is higher than the rate of decrease of power developed by the engine.

The brake specific consumption of the engine in methane version decreases up to air-fuel ratio of 15 since the power developed increases and the rate of methane aspirated decreases. The brake specific fuel consumption of the engine in methane version decreases beyond the air-fuel of 15 because the rate of decrease of methane aspirated by the engine is higher than the rate of decrease of power developed by the engine up to air-fuel ratio of 18. Beyond air-fuel ratio 18 the BSFC shows an increasing trend due to the higher rate at which power drops to the rate at which the mass flow rate of fuel consumed by the engine drops.

The brake specific fuel consumption is higher for methane than petrol as fuel for the engine under consideration. This is because much higher power is developed in petrol mode due to higher volumetric efficiency in petrol mode resulting in increase in power to be developed. Also the mass of fuel aspirated by the engine per cycle in the methane mode is higher than rate of petrol consumption per cycle for the same engine which increases the value of brake specific fuel consumption for methane version of the engine under consideration.

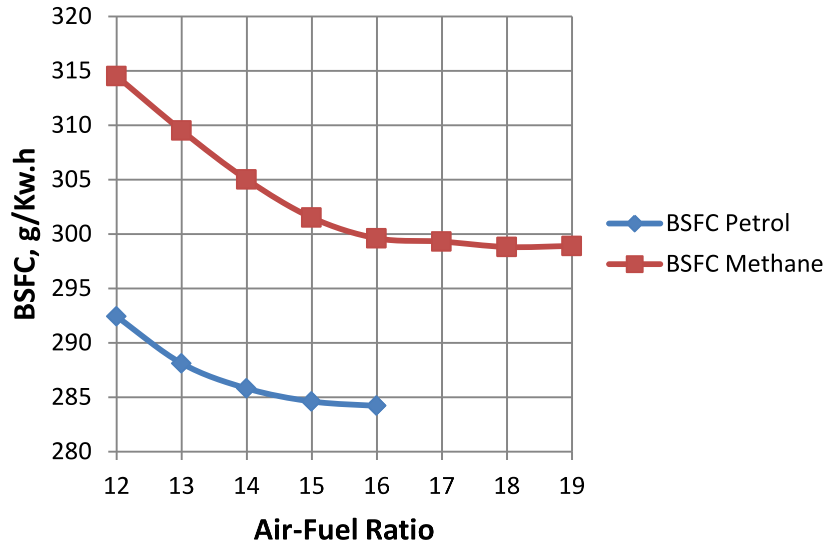

Fig.4 Effect of Air-Fuel Ratio on Brake Specific Fuel Consumption

EFFECT OF AIR-FUEL RATIO ON CO EMISSIONS

The Fig.5 below shows the effect of air-fuel ratio on CO formation from the engine in both petrol and methane mode.

For engine in petrol mode up to the value of air-fuel ratio of 15 and for the engine in the methane mode up to the value of air-fuel ratio 18 the CO emissions decrease towards leaner side. This is because more amount of air is available for combustion in the leaner mode. Beyond the values of air-fuel ratio of 15 in the petrol mode and 18 in the methane mode the CO formation shows an increasing trend. This is because of comparatively less favourable combustion conditions like lesser flame speed in the leaner mode under consideration.

The CO emissions are higher in the methane mode as compared to petrol mode. This is due to lower volumetric efficiency with methane as fuel resulting in availability of less amount of air for complete conversion of carbon to carbon dioxide..

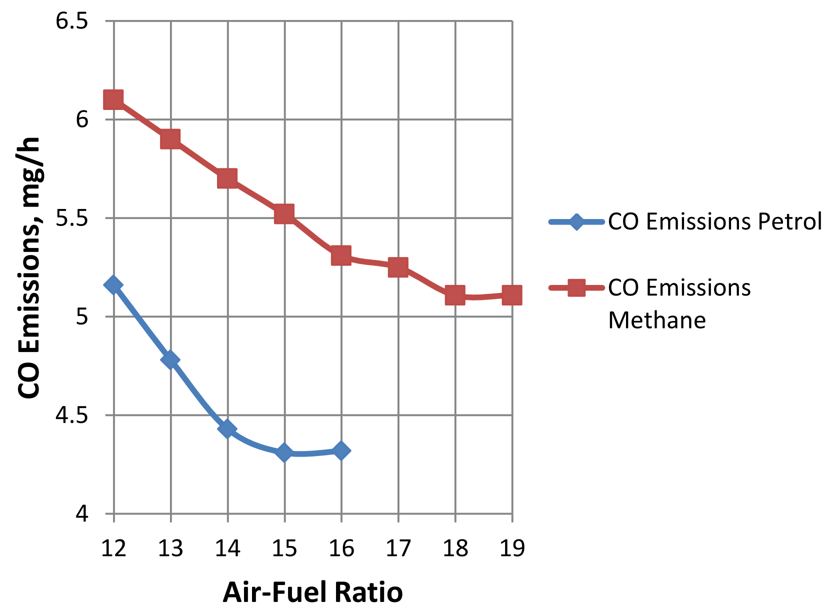

Fig.5 Effect of Air-Fuel Ratio on CO Emissions.

EFFECT OF AIR-FUEL RATIO ON HC EMISSIONS.

The Fig.6 below shows the effect of air-fuel ratio on the formation of HC emissions from the engine in petrol and methane modes.

The HC emissions decrease with air-fuel ratio in both modes as more amount of air is available for combustion in leaner modes.

The petrol version of the engine produces more HC emissions as compared to methane version.

This is because of higher specific volume of methane as a gas as compared to petrol. So comparatively much lesser amount methane is aspirated into the engine as compared to petrol. This results in a higher rate of the formation of HC emissions in the petrol mode as compared to methane mode.

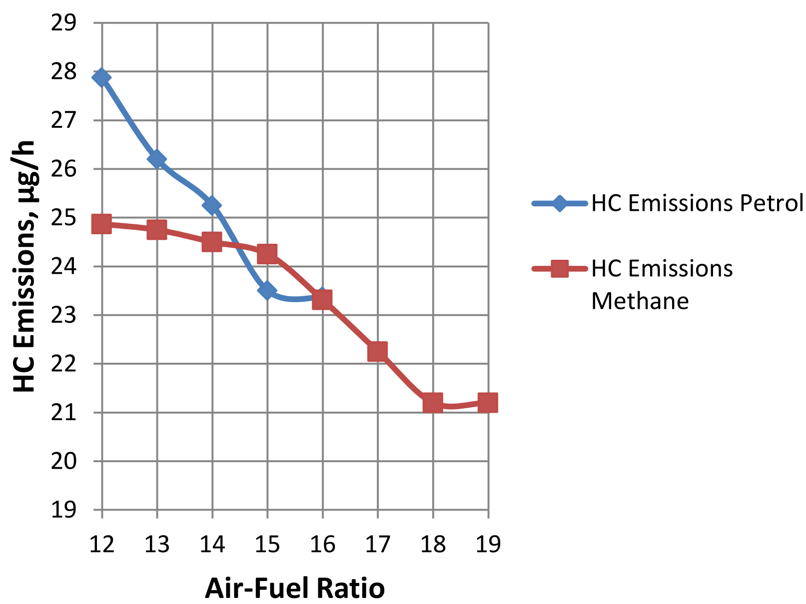

Fig.6 Effect of Air-Fuel Ratio on HC Emissions

EFFECT OF AIR-FUEL RATIO ON NOX EMISSIONS.

The Fig.7 below shows the effect of air-fuel ratio on the formation of NOx emissions from the engine in the petrol and methane mode.

The rate of NOx formation with petrol as fuel increases with air-fuel ratio as more amount of air goes to cylinders in the leaner modes. This together with the high temperatures developed in the engine cylinders throughout the range of air-fuel ratio results in increasing trend of NOx emissions with air-fuel ratio.

The rate of NOx formation with methane as fuel increases upto air-fuel ratio of 18 as more amount of air goes to cylinders in the leaner modes. This together with favourable temperatures developed in the engine cylinders up to the same value of air-fuel ratio results in increasing trend of NOx emissions with air-fuel ratio. Beyond the air-fuel ratio of 18 the NOx emissions decrease as the temperatures developed in the engine cylinders decrease with air-fuel ratio for the methane version of this engine throughout the range of air-fuel ratio.

The petrol version of the engine produces more NOx emissions as compared to the methane version of the same engine. This is because higher amount of air is aspirated by the engine in the petrol mode as compared to the methane mode. Also higher temperatures are developed in the engine cylinders in the petrol mode than methane mode. These two factors are favourable for higher rates of NOx formation in the engine in petrol mode as compared to methane mode as per the reasons behind the mechanism of NOx formation in I C engines.

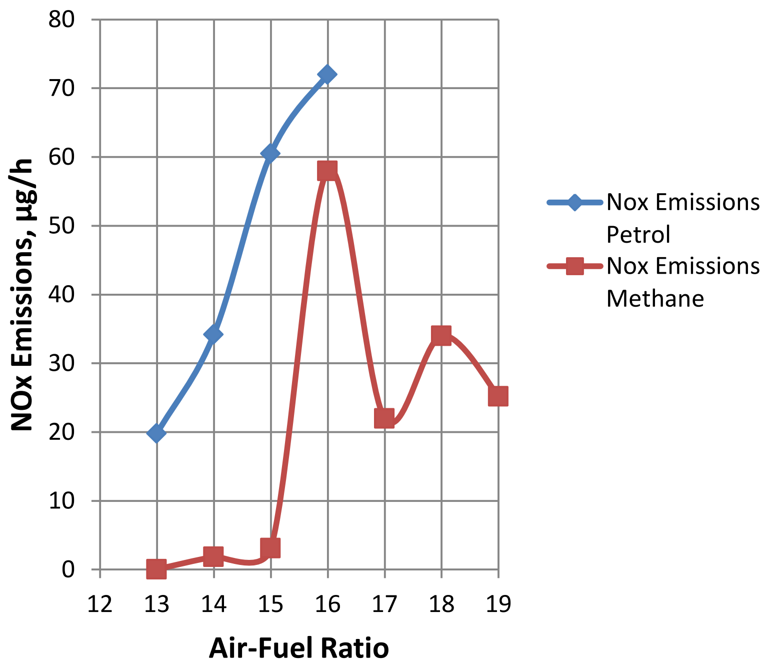

Fig.7 Effect of Air-Fuel Ratio on NOx Emissions.

EFFECT OF AIR-FUEL RATIO ON EXHAUST GAS TEMPERSATURE.

The Fig.8 below shows the effect of air-fuel ratio on exhaust gas temperature of the engine in petrol and methane modes.

The exhaust gas temperature is almost constant with entire range of air-fuel ratio for both petrol and methane modes.

The exhaust gas temperature is higher with petrol as compared to methane as fuel in the same engine. The reason is that more amount of petrol is consumed by the engine in petrol mode as compared to methane mode. The heat liberated during combustion with higher mass of petrol is more than the heat liberated with methane aspirated although the lower heating value of methane is higher than that of petrol. The increase in the heat energy liberated due to higher mass of petrol consumed by the engine per cycle dominates the corresponding heat liberated with lower mass of methane having a higher heating value. The lower value of the stoichiometric air-fuel ratio for petrol as compared to methane further favours the formation of higher temperatures with petrol as a fuel.

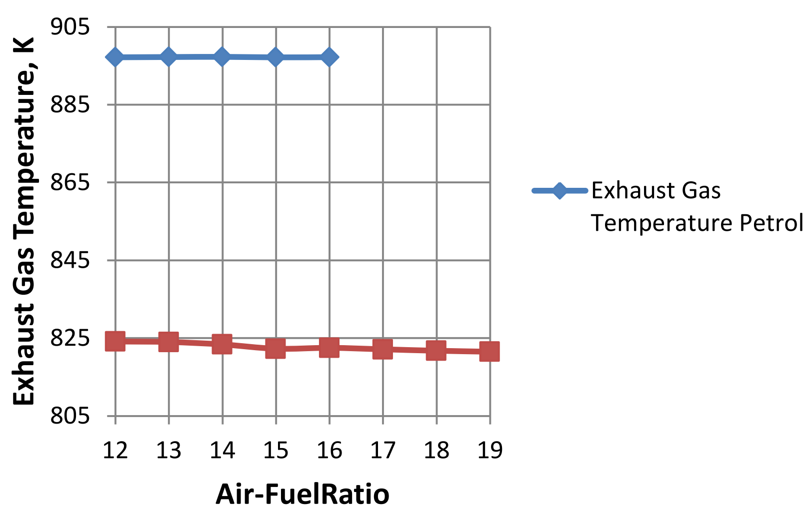

Fig.8 Effect of Air-Fuel Ratio on Exhaust Gas Temperature.

CONCLUSIONS

1. The commercial petrol and methane available in the market need to be upgraded to octane numbers of more than 121 and 130 respectively if the engine under consideration is used for power development using these fuels with normal combustion characteristics over the entire range of air-fuel ratio or load under constant speed operation.
2. The engine shows successful performance characteristics with both petrol and methane as fuels.
3. The petrol version of the engine produces more power than the methane version.
4. The brake specific fuel consumption of the engine in the petrol mode is lower than that in the methane mode.
5. The petrol version of the engine produces higher CO emissions than methane version.
6. The methane version of the engine produces lesser HC and NOx emissions than the petrol version.
7. Since turbo-charging of the spark ignition engines is limited by the knock characteristics of these engines with fuels under consideration when operated at variable air-fuel ratio or load under constant speed, the production of high octane rated fuels like petrol, methane and ethanol etc in the refineries could help in commercializing these engines. This could further open the door for further development of these engines towards the turbo-compounded versions as has been done in case of the compression ignition engines.

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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	1500 rpm

APPENDIX-C

PHYSICAL AND CHEMICAL PROPERTIES OF PETROL AND METHANE[9]

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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