EFFECT OF START OF COMBUSTION TIMING ON THE OCTANE REQUIREMENT, PERFORMANCE AND EMISSION CHARACTERISTICS OF A V6 TCI BI-TURBO SPARK IGNITION ENGINE USING PETROL AND METHANE AS ALTERNATIVE FUELS.

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ABSTRACT

This paper presents the variable start of combustion timing based computational research investigations on a V6 TCI BI-TURBOspark ignition engineusing petrol and methane as alternative fuels. The research investigations are done to explore the octane number requirements of petrol and methane for successfully turbo-charging the spark ignition engines needed for power generation with above mentioned alternative fuels.
The investigations were done by using the professional thermodynamic simulation software named as AVL BOOST. This software uses thermodynamic models for combustion, heat transfer, pollutants formation, frictional power and octane number requirement for fuels. The software also uses the laws of conservation of mass and energy for computing results in the engine cylinders. For thermodynamic calculations in both intake manifold and exhaust manifold the above two laws are supplemented by the law of conservation of momentum. The partial differential equations involved in the simulations are solved by using numerical methods.
First the model for a V6 TCI BI-TURBO spark ignition engine was selected for the above mentioned research investigations.
The model was run in the petrol mode under variable start of combustion timings and results were computed for octane number requirement of the petrol over the entire range of the start of combustion timing. The performance and emissions characteristics of the engine were also studied under the same conditions. The speed and air-fuel ratio were considered corresponding to the condition for maximum power generation
The computational investigations were repeated with the methane version of the same engine. Again the octane ratings needed for methane were computed under variable start of combustion timings. The performance and emission characteristics of the engine were also investigated in the methane mode. The speed and air-fuel ratio were selected corresponding to condition for maximum power generation in the methane mode.
It was observed that for successful commercialization of the multi-cylinder turbo-charged spark ignition engines the octane rating of the petrol needs to be upgraded. This will further help in choosing the start of combustion timing and therefore optimum spark timing of these engines which will help in boosting the power and torque of these engines using the concept of turbo-charging..

Keywords: Multi-cylinder Engine,Spark Ignition, Petrol, Methane,Turbo-charging,Octane Number Requirement,Performance, Emission and Octane on Demand.

INTRODUCTION

In order to boost the power of multi-cylinder spark ignition engines further using the concept of turbo-charging with inter-cooling and commercialize these successfully we need to design these engines accordingly. Further the octane ratings of the commercial spark ignition engine fuels like petrol and methane have to be upgraded and maintained as needed for achieving the normal combustion characteristics in all cylinders during the entire combustion period.The fuel supply to these engines under the concept of “octane on demand” will eliminate all possible chances of knock based abnormal combustion in any of the engine cylinders.

The full cycle results for these engines need to be simulated by investigating the possibilities of improving the performance and emissions characteristics of these engines with an open mind. The design and operating parameters of the engine will help in tracing the thermodynamic properties of the working fluid in the engine components. These thermodynamic properties of the air-fuel mixture inside the engine cylinders during the combustion period will ultimately help in selecting a suitable fuel with the octane ratings as needed for achieving the needed performance over the entire range of speed and load with normal combustion characteristics. This will ultimately remove any possibility of knock based abnormal combustion in any cylinder during any combustion period..

John B. Heywood in the various chapters of his text book writes thatthe presence or absence of knock in an engine depends primarily on the anti-knock quality of the fuel, which is defined by the fuel’s octane number. It determines whether or not a fuel will knock in a given engine under given operating conditions: the higher the octane number, the higher the resistance to knock. The octane number requirement of an engine is defined as the minimum fuel octane number that will resist knock throughout its speed and load range.

The following factors affect an engine’s octane requirement: composition of the fuel, combustion chamber geometry, charge motion, spark-advance curve, inlet-air, intake manifold and water jacket temperatures, carburetor or fuel injector air/fuel ratio calibration, the ambient conditions-pressure, temperature and relative humidity-during the requirement determination.

It has been seen experimentally that the time available for combustion reduces at high speeds if the spark timing is maintained constant. This ultimately reduces the torque developed by the engine at high speeds.It is seen that the drastic torque loss of upto 15% could be avoided by correspondingly varying the spark advance say from 10 degrees of crank angle before TC to 50 degrees of crank angle before TC. If the spark advance is maintained at high values of crank angle even at lower speeds this actually increases the torque developed by the engine and therefore pressure and temperature developed in the engine cylinder during combustion. This increases the tendency to knock with the same conventional fuels with low octane number decided with decreasing the spark advance at low speeds rather than allowing the torque amplification. However for power boosting purposes without knock in naturally aspirated engines and more seriously in turbocharged spark ignition engines, this needs higher or variable octane rated fuel, known as the octane on demand concept, to be used for power generation.

The 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. 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]

Bourhisetal 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]

Ameretalconducted simulation tests on a a singlecylinder 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]

Mohananetal 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]

Barattaetal 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:

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

where

$\frac{d(m_c.u)}{d\alpha}$ = change of the internal energy in the cylinder.

$-\frac{p_c.dV}{d\alpha}$ = piston work.

$\frac{dQ_F}{d\alpha}$ = fuel heat input.

$\sum \frac{dQ_w}{d\alpha}$ = wall heat losses

$\frac{h_{BB}.dm_{BB}}{d\alpha}$ = enthalpy flow due to blow-by

$\frac{dm_{BB}}{d\alpha}$ = blow-by mass flow

The first law of thermodynamics for high pressure cycle states that the change of internal energy in the cylinder is equal to the sum of piston work, fuel heat input, wall heat losses and the enthalpy flow due to blow-by.
In order to solve this equation, models for the combustion process and the wall heat transfer, as well as the gas properties as a function of pressure, temperature, and gas composition are required.
Together with the gas equation

$$p_c = \frac{1}{V} .m_c.R_o.T_c \text{ --(Eq.2)}$$

Establishing the relation between pressure, temperature and density, Eq. 2 for in-cylinder temperature can be solved using a Runge-Kutta method. Once the cylinder gas temperature is known, the cylinder gas pressure can be obtained from the gas equation.

COMBUSTION MODEL

AIR REQUIREMENT AND HEATING VALUE

STOICHIOMETRIC AIR-FUEL MIXTURE

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

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

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

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

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

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

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

HEAT TRANSFER

The heat transfer to the walls of the combustion chamber, i.e. the cylinder head, the piston, and the cylinder liner, is calculated from:

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

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

For the calculation of the heat transfer coefficient, the Woschni1978 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:

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

$C_3 = 6.18+0.417.c_u/c_m$

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:

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

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

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 START OF COMBUSTION TIMING ON OCTANE NUMBER REQUIREMENT

The Fig.1 below showsthe effect of start of combustion timing on the octane demand of the engine for petrol and methane fuels. It is clear that when the start of combustion is varied from -10 degrees of crank angle before TC to -40 degrees of crank angle before TC the range of the octane number requirement for petrol fuel varies between 97 and 110 and the maximum octane demand is at -30 degrees of crank angle before TC.

Also the range of the octane number required for methane between the same range of variation of the start of combustion timing is between 101 to 112 and the maximum octane demand for methane is at -30 degrees of crank angle before TC.

With the help of Fig.2 it is clear that for the value of the start of combustion timing needed for maximum power generation the octane number required for petrol fuel is 108 and the octane number requirement for methane fuel is 111.

The variation of the octane number requirement of the fuels with respect to the variable start of combustion timing for both petrol and methane can be explained as follows. The 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 combustion chamber and also the time for which these high pressures and temperatures prevail in both burned and unburned zones, the octane demand of the engine 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.

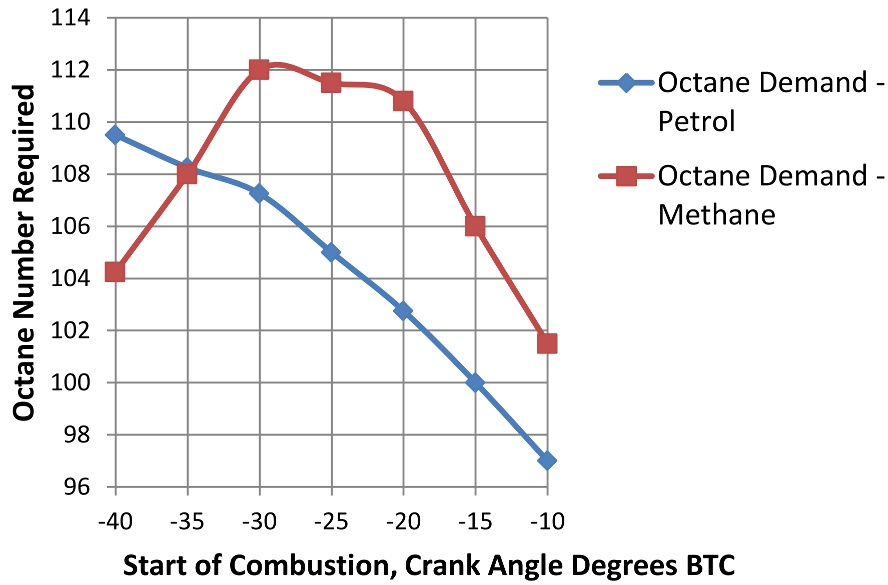

Fig.1 Effect of Start of Combustion on Octane Number Required

EFFECT OF START OF COMBUSTION TIMING ON ENGINE POWER.

The Fig.2 below showsthe effect of start of combustion timing on the power developed by the engine in both petrol and methane modes.

For both petrol and methane fuels the power developed by the engine varies when the start of combustion timing is varied from -10 to -40 degrees of crank angle before TC.

If the octane number required for petrol fuel can be adjusted at the refinery level as demanded by the results in figure 1 the start of combustion timing should be designed at -30 degrees of crank angle before TC for the petrol engine for the condition of maximum power generation.

Similarly if the octane number required for methane fuel can be adjusted at the refinery level as demanded by the results in figure 1 the start of combustion timing should be designed at -20 degrees of crank angle before TC for the engine in the methane mode if power developed by the engine has to be maximized.

More power is developed by the engine in the petrol mode as compared to the methane mode because the volumetric efficiency of the engine in the petrol mode is higher than the volumetric efficiency of the engine in the methane mode.

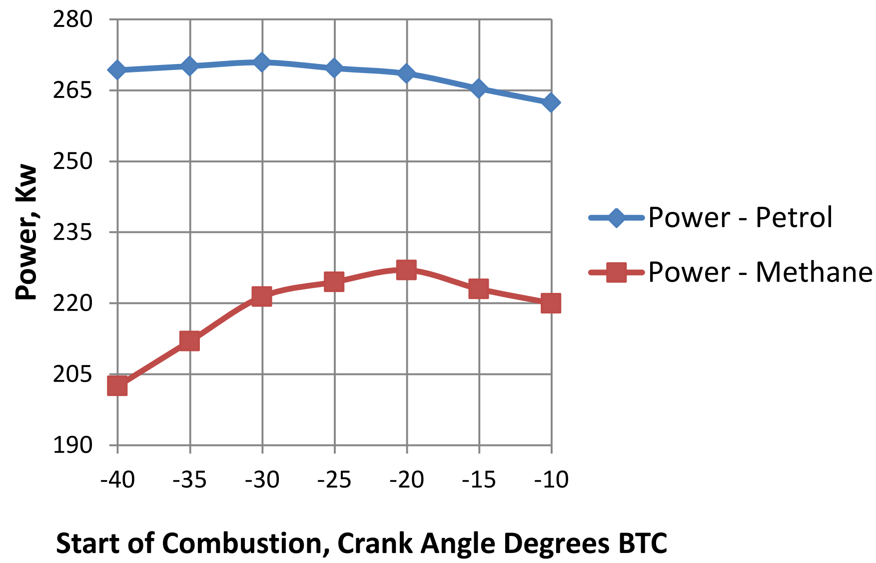

Fig.2 Effect of Start of Combustion on Power

EFFECT OF START OF COMBUSTION TIMING ON ENGINE TORQUE.

The Fig.3 below shows the effect of start of combustion timing on the torque developed by the engine in petrol and methane modes.

For both petrol and methane fuels the torque developed by the engine varies when the start of combustion timing is varied from -10 to -40 degrees of crank angle before TC.

If the octane number required for petrol fuel can be adjusted at the refinery level as demanded by the results in figure 1 the start of combustion timing should be designed at -30 degrees of crank angle before TC for the petrol engine for maximum torque to be developed.

Similarly if the octane number required for methane fuel can be adjusted at the refinery level as demanded by the results in figure 1 the start of combustion timing should be designed at -20 degrees of crank angle before TC for the engine in the methane mode under the condition for maximum torque generation.

The petrol engine develops more torque as compared to the torque developed by the same engine when operated in the methane mode because the volumetric efficiency of the engine in the petrol mode is higher than the volumetric efficiency of the engine in the methane mode.

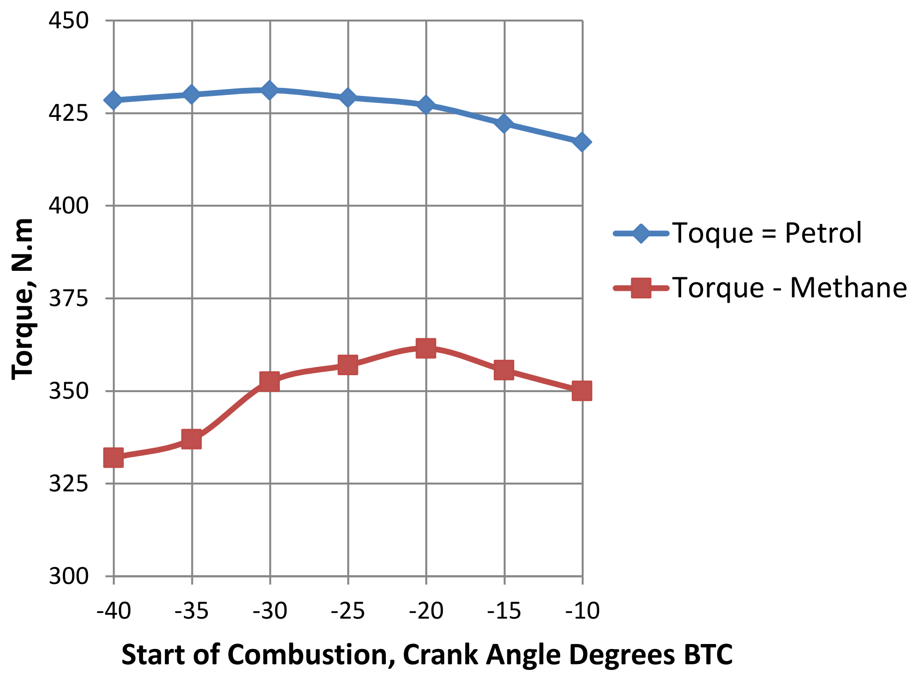

Fig.3 Effect of Start of Combustion on Torque

EFFECT OF START OF COMBUSTION TIMING ON BRAKE SPECIFIC FUEL CONSUMPTION OF THE ENGINE.

The Fig.4 below shows the effect of start of combustion timing on the brake specific fuel consumption of the engine in petrol and methane modes.

The brake specific fuel consumption of the engine in both petrol and methane modes varies when the start of combustion timing is varied from -10 to -40 degrees of crank angle before TC.

The brake specific fuel consumption of the engine in the petrol mode is minimum corresponding to -30 degrees of crank angle before TC of the start of combustion timing.

Also the brake specific fuel consumption of the engine in the methane mode is minimum corresponding to -20 degrees of crank angle before TC of the start of combustion timing.

The fuel consumption per unit of energy developed for the engine under consideration in the petrol mode is lower than the fuel consumption per unit of energy developed by the same engine in the methane mode because the power developed by the engine in the petrol mode is higher than the power developed by the same engine when operated in the methane mode and the difference in power generation dominates the terms involved in the definition of the brake specific fuel consumption of an engine.

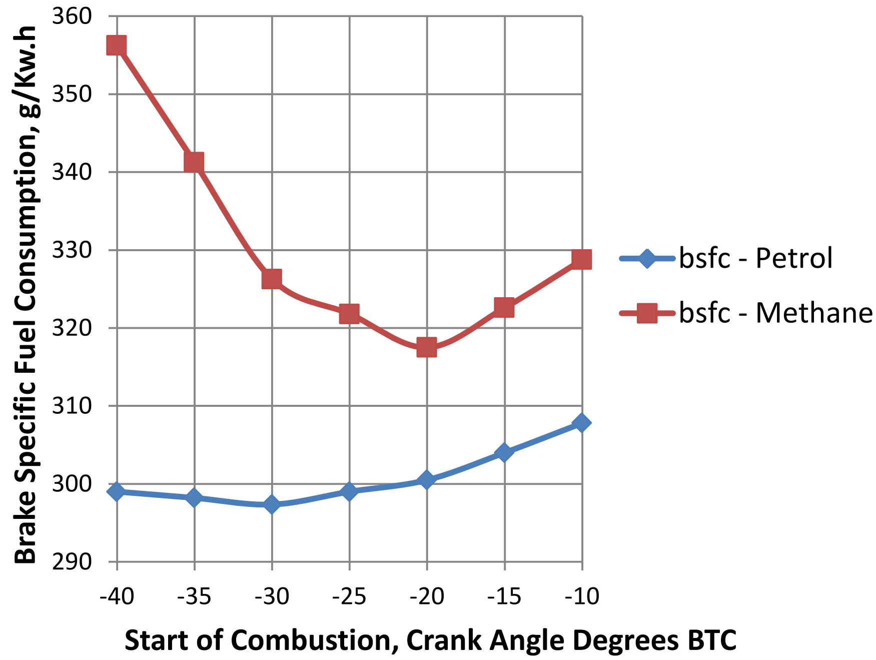

Fig.4 Effect of Start of Combustion on Brake Specific Fuel Consumption

EFFECT OF START OF COMBUSTION TIMING ON CO EMISSIONS

The Fig.5 below shows the effect of start of combustion on CO emissions produced by the engine.In case of both the fuels petrol and methane the CO emissions vary when start of combustion timing is varied between -10 degrees of crank angle before TC to -40 degrees of crank angle before TC.

The engine in the petrol version produces minimum CO emissions when the start of combustion timing is set at -30 degrees of crank angle before TC. As seen from the foregoing results the petrol fuel based engine also develops maximum power and torque with the same start of combustion timing. This is because of better combustion characteristics or higher percentage conversion of carbon in the petrol fuel into carbon-dioxide releasing more heat energy in the exothermic reactions in the cylinder which produce higher temperatures and pressures to further boost the engine performance characteristics.

The methane version of the same engine produces minimum CO emissions when the start of combustion timing is set at -20 degrees of crank angle before TC. As seen from the foregoing results the methane fuel based engine also develops maximum power and torque with the same start of combustion timing. This is because of better combustion characteristics or higher percentage conversion of carbon in the methane fuel into carbon-dioxide which further boosts power and torque because of higher temperatures and pressures developed during the combustion duration.

The engine produces more CO emissions in the methane mode as compared to petrol mode per unit of energy basis. This is because the engine in the petrol mode fundamentally generates more power as compared to the power produced in methane mode which brings down the CO emissions from the engine in petrol version per unit energy basis.

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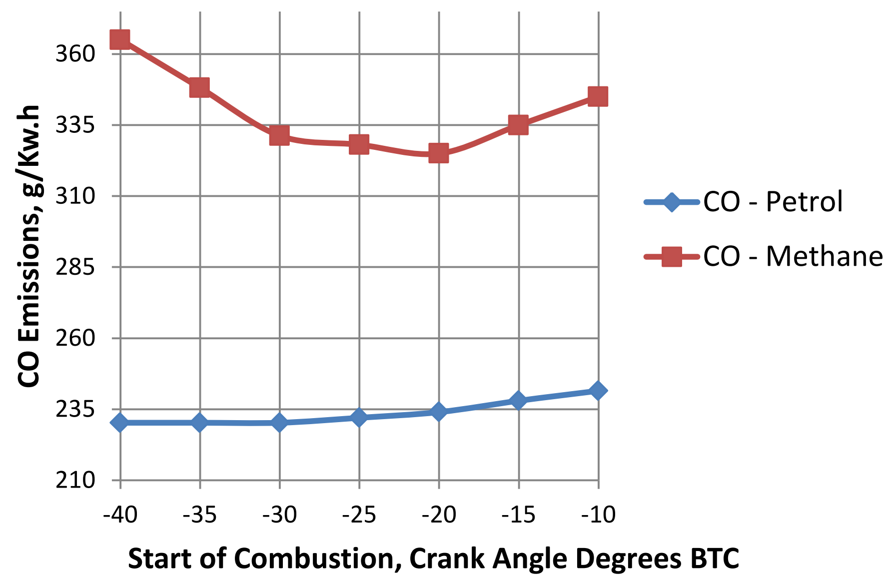

Fig.5 Effect of Start of Combustion on CO Emissions

EFFECT OF START OF COMBUSTIONTIMING ON HC EMISSIONS.

The Fig.6 below shows the effect of start of combustion timing on HC emissions produced by the engine. The HC emissions produced by the engine increases in both petrol and methane modes as the start of combustion timing is further advanced from -10 degrees of crank angle before TC to -40 degrees of crank angle before TC.At the optimum start of combustion timing of -30 degrees of crank angle before TC for petrol fuel and -20 degrees of crank angle before TC for methane fuel the HC emissions produced by the engine are 1.32g/Kw.h and 1.42g/Kw.h respectively. If the start of combustion timing is further advanced the HC emissions produced increase with both fuels which can be because of several complex factors like more time available for the hydrocarbons to get absorbed by the lubricating oil on the cylinder walls and thereby getting released and exhausted from the cylinder towards the end of combustion period. As seen from the graph the HC emissions decrease with both petrol and methane by further retarding the start of combustion timing with respect to TC as less amount of hydrocarbon could now get absorbed by the lubricating oil on the combustion chamber walls in shorter period of time. The other reason is that with the further advance in the start of combustion timing, the higher amount of the hydrocarbon fuels, petrol and methane, get filled into the crevice volumes in the gap between piston and cylinder above the piston rings. This is possible because with the piston moving upwards towards TC from a greater distance from TC with increased advance in start of combustion timing piston there is sufficient time to force more amount of hydrocarbons fuel into the crevice volumes by the rapidly increasing pressure in the cylinder.This fraction of the hydrocarbon fuel escape combustion and is exhausted from the cylinder as the exhaust valve is opened.

The engine produces lesser HC emissions per unit of energy developed in the petrol mode as compared to methane mode since the fuel consumption per unit of energy produced in the petrol mode is lower than the fuel consumption per unit of energy developed in the methane mode.

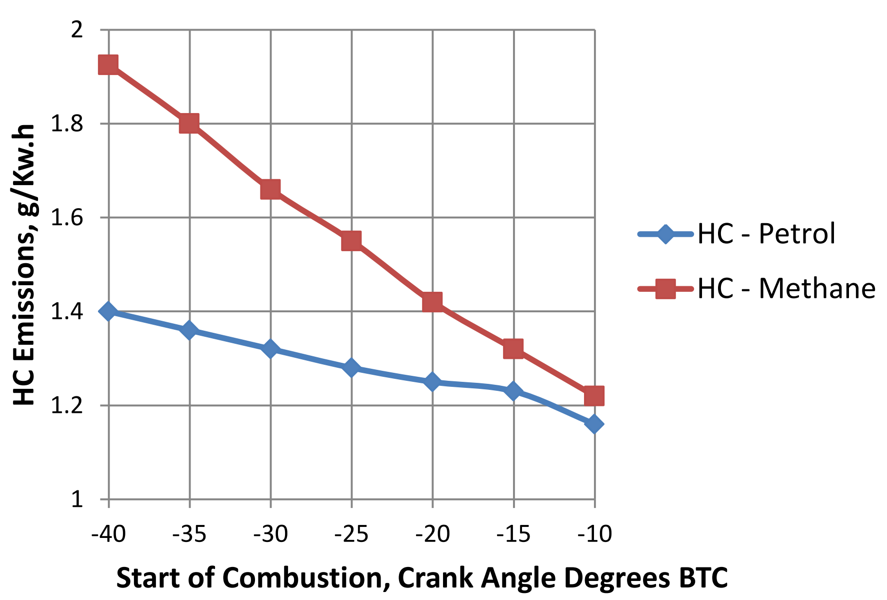

Fig.6 Effect of Start of Combustion on HC Emissions

EFFECT OF START OF COMBUSTION TIMING ON NOX EMISSIONS.

The Fig.7 below showsthe effect of start of combustion timing on the NOx emissions produced by the engine with both petrol and methane as fuels.

The NOx emissions produced by the engine increasewith both the fuels by further advancing the start of combustion timing from -10 degrees of crank angle before TC to -40 degrees of crank angle before TC. This is because the time available for the conversion of oxygen and nitrogen of the air in the cylinder into various oxides of nitrogen increases by advancing the start of combustion timing.

The engine produces more NOx emissions in the petrol mode as compared to the methane mode

Since the temperatures developed in the cylinder in the petrol mode are higher than the temperatures produced in the cylinder in the methane mode.

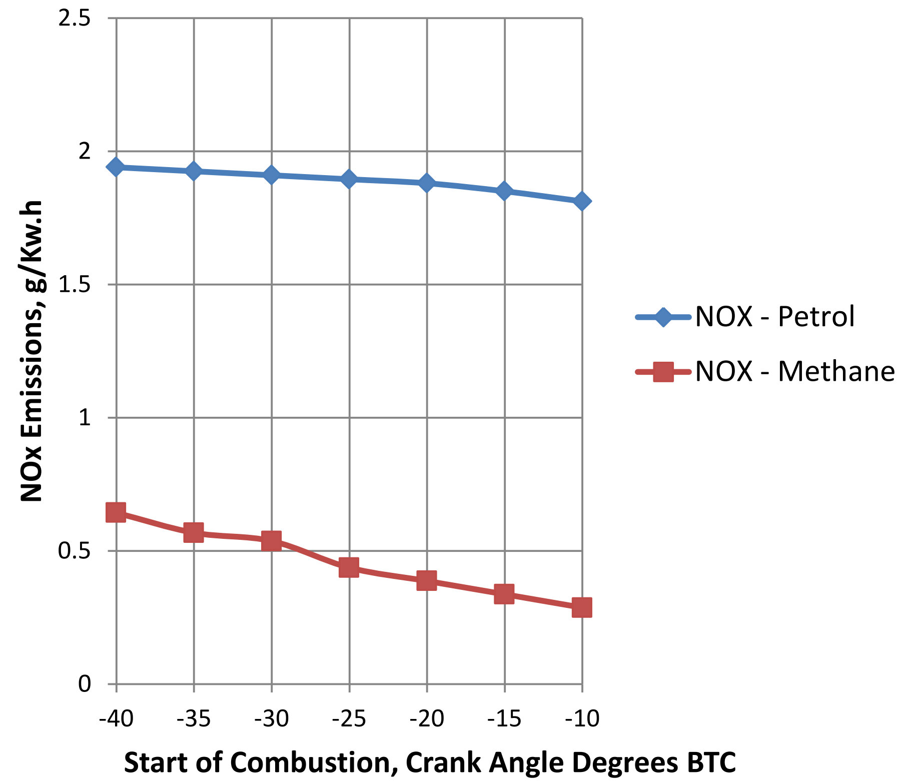

Fig.7 Effect of Start of Combustion on NOx Emissions

EFFECT OF START OF COMBUSTIONTIMING ON EXHAUST GAS TEMPERSATURE.

The Fig.8 below shows the effect of start of combustion timing on the exhaust gas temperature of the engine in both petrol and methane modes.

With petrol as fuel, the exhaust gas temperature increases by retarding the start of combustion timing. This is because by retarding the start of combustion timing towards TC the HC emissions produced by the engine decrease. This increase in the exothermic conversion of HC in the fuels into corresponding compounds releases more amount of energy which increases the temperature of the exhaust gas.

With methane as fuel, the exhaust gas temperature increases by retarding the start of combustion timing from -30 degrees of crank angle before TC. This is because by retarding the start of combustion timing towards TC from -30 degrees of crank angle before TC the HC emissions produced by the engine decrease. This increase in the conversion of HC in the fuel into corresponding compounds releases more amount of energy which increases the temperature of the exhaust gas. By further advancing the start of combustion timing from -30 degrees of crank angle before TC to -40 degrees of crank angle before TC the exhaust gas temperature increases because the fuel consumption per unit of energy produced increases in this direction for methane fuel.

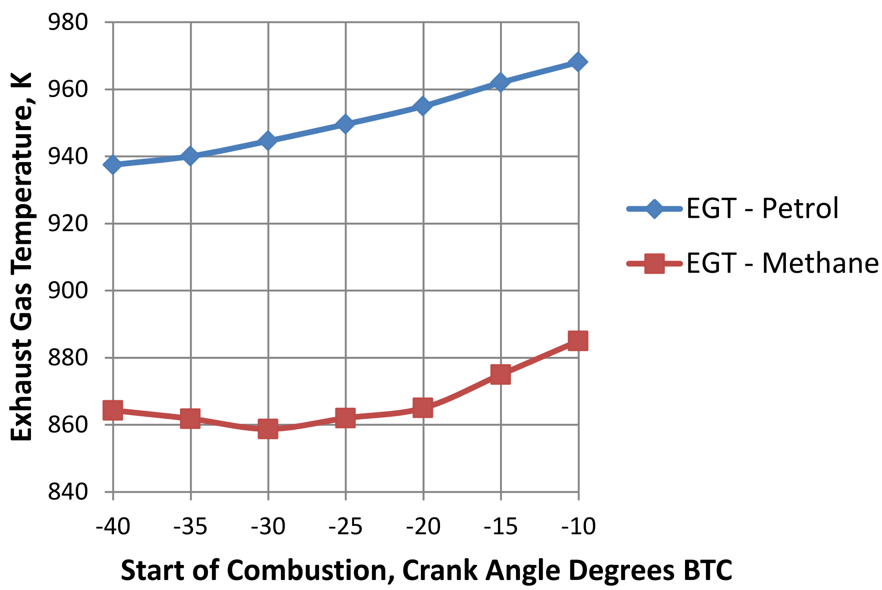

Fig.8 Effect of Start of Combustion on Exhaust Gas Temperature

CONCLUSIONS

1. The engines develops maximum power and torque with start of combustion timing at 30 degrees of crank angle before TC for petrol and 20 degrees of crank angle before TC for methane.
2. The engine has the minimum brake specific fuel consumption of petrol with the start of combustion timing at30 degrees of crank angle before TC for petrol and 20 degrees of crank angle before TC for methane.
3. The engine produces minimum CO emissions with the start of combustion timing at 30 degrees of crank angle before TC for petrol and 20 degrees of crank angle before TC for methane.
4. In order to design and commercialize the petrol based turbocharged spark ignition engine with the above mentioned best possible performance and emission characteristics the octane number requirement of the commercial petrol is 108.
5. For commercializing the methane version of the same turbocharged spark ignition engine successfully with normal combustion characteristics the octane number requirement for methane has to be 111.

6. The octane rating of 108 for petrol and 111 for methane for the engine under consideration will eliminate all possible chances of knock during combustion.

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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
μσ = 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[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
Operating Value of Air-Fuel Ratio.	13	15

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