OCTANE REQUIREMENT, PERFORMANCE AND EMISSION CHARACTERISTICS OF A V6 TCI BI-TURBO SPARK IGNITION ENGINE UNDER VARIABLE SPEED OPERATION USING PETROL AND ETHANOL AS ALTERNATIVE FUELS.

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ABSTRACT
This paper presents the results of computer simulation based thermodynamic analysis on a turbocharged V6 TCI BI-Turbo spark ignition engine under variable speed operation using petrol and ethanol as alternative fuels. The results have been simulated in the professional thermodynamic simulation software named AVL BOOST.
First the modeled engine was run in the petrol mode and the results were computed for the octane requirement for petrol for the turbocharged configuration of the spark ignition engine under consideration. The results were also computed for its performance and emission characteristics.
The computational investigations were repeated on the same engine using ethanol as a second fuel.
The engine demands gasoline as well as ethanol having varied octane ratings over its entire range of the speed. The engine also gives satisfactory performance with both the fuels petrol and ethanol.
The engine produces higher power and torque output in the petrol mode as compared to the ethanol mode. The brake specific fuel consumption of the engine in the petrol mode is lower than that in the ethanol mode.
However the ethanol based engine produces less CO and HC emissions. The engine produces higher NOx emissions with petrol fuel than that produced with ethanol as a fuel.

Keywords: *Octane Requirement, Turbocharged, Multi-cylinder, Spark Ignition Engine, Performance, Emissions, Petrol and Ethanol.*

INTRODUCTION
The design of the naturally aspirated spark ignition engines is done on the basis of the petrol or gasoline available from the petroleum refineries. Accordingly its air flow modeling is done. The fuel supply system as well as the optimum spark timings are designed and tuned accordingly.
Since the physical and chemical properties of gasoline, which includes its octane number, put some constraints for further power boosting and also downsizing the commercialized gasoline engine. The power boosting and also the downsizing of the diesel based compression ignition engines has already been done by incorporated the use of various configurations of turbochargers. This technology till to-date has not been adopted and exploited for the downsizing and power boosting of petrol engines due to knock based limitations governed by the octane number of the commercialized gasoline.
Also the octane requirement of the engine for petrol fuel varies over the full range of speed and load on the engine. This necessitates the need for the design of a “Octane On Demand” type of fuel supply system for the turbocharged spark ignition engines. This basically will involve the supply of a combination of two types of fuels, one having a high octane number and the other having a lower octane number. The fuel supply system will supply a certain mixture of these two fuels as governed by the octane demand of the engine at a particular load and speed. One of the fuels can be the conventional gasoline or petrol and the second fuel can be ethanol having a higher octane rating as compared to petrol. The combination of these two fuels will allow the use of turbochargers for further downsizing and power boosting of the existing spark ignition engines run by petrol.

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]

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.[4]

Francesco Catapano etal. conducted experimental investigations to study the effects of air fuel mixing and combustion behavior of gasoline ethanol blends in a GDI wall guided turbocharged multi-cylinder optical engine. They observed that the addition of ethanol in gasoline allowed an improvement of engine performance in terms of IMEP, COV, IMEP and emissions.[5]

Thiago etal conducted computer simulation studies on a turbocharged spark ignition engine using ethanol. The engine gave the satisfactory results. [6]

Alberto etal conducted simulation studies on a directly injection turbocharged spark ignition ethanol engine. The results showed that the ethanol has higher knock resistance than petrol. Further it was concluded that the direct injection and turbo-charging are the key features of high efficiency and high power density ethanol engines [7].

Young etal conducted experimental investigations for octane requirement of a turbocharged spark ignition engine under various driving cycles

It was found that downsizing or turbocharging as well as vehicle loading in trucks increased octane requirement substantially.[8]

Jose etal conducted experimental investigations for exploring the limits of a downsized ethanol direct injection spark ignited engine in different configurations in order to replace high displacement gasoline engines.

It was concluded that 28% of fuel consumption reduction was achieved by means of an extreme downsizing. 53% of downsizing was reached by using cutting edge technologies like using a twin stage compressor for achieving a larger pressure ratio. A significant decrease in the engine emissions was also achieved. [9]

Hulser etal conducted investigations on the origin of pre-ignition in a highly boosted SI engine using biofuels namely tetrahydro2methylfuran (2MTHF) and 2methylfuran (2MF), in addition to the conventional ethanol and petrol fuels. The primary objective was to investigate the influence of molecular biofuel structure on the locations of preignition in a realistic, highly charged SI engine at low speed by state-of-the-art optical measurements.

It was concluded that the pre-ignition tendency of these fuels decreases with increasing RON. Further as compared to other bio-fuels ethanol reveals a relatively high pre-ignition tendency, although its octane numbers (RON and MON) are particularly high. [10]

Thewes etal conducted experimental investigations for analyzing the Effect of BioFuels on the Combustion in a Downsized DI SI Engine.In this study the fuel influence of several biofuel candidates on homogeneous engine combustion systems with direct injection is investigated.

The results reveal Ethanol and 2Butanol are the two most knock resistant fuels. Hence these two fuels enable the highest efficiency improvements versus RON95 fuel ranging from 3.6% 12.7% for Ethanol as a result of a compression ratio increase of 5 units. Tetrahydro2methylfuran has a worse knock resistance and a decreased thermal efficiency due to the required reduction in compression ratio by 1.5 units.

In general, 1Butanol and 2Butanol emit higher amounts of HC emissions in all operation points combined with significantly increased particle emissions at high loads indicating a worse mixture formation. [11]

Guilliam etal conducted experimental investigations on the use of ethanol's double octane boosting effect with low RON naphtha based fuel for an Octane on Demand concept for SI Engine

The results showed that the fuel combination [naphtha; ethanol] offers the most promising boosting effect. The dedicated tests on an up to date gasoline direct injection multi cylinder engine revealed that naphtha based fuel of RON 71 can be used over a significant area of the engine map. Around two thirds of the engine map can be run using a moderate ethanol rate within the range 0% to 40%, making this OOD concept compatible within the E10 E20 context.[12]

Zhang etal conducted experiments on the study of the lifecycle based optimized use of ethanol gasoline blends for turbocharged engines. The study involved a lifecycle (well to wheel) analysis to determine the CO2 emissions associated with ethanol blended gasoline in optimized turbocharged engines. The study involves a more accurate assessment on the best achievable CO2 emission of ethanol blended gasoline mixtures in future engines.

The results showed that the engine downsizing technology can yield a CO2 reduction of up to 25.5% in a two stage downsized turbocharged engine burning the optimum sugarcane based fuel blend. [13]

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:

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

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

$p_c = \frac{1}{V}.m_c.R_o.T_c$ ---(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 .$\left(\frac{c}{12.01} + \frac{h}{4.032} + \frac{s}{32.06} - \frac{o}{32.0}\right)$ [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.

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

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

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

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

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

GAS EXCHANGE PROCESS , BASIC EQUATION

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

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

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

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

PISTON MOTION

Piston motion applies to both the high pressure cycle and the gas exchange process.
For a standard crank train the piston motion as a function of the crank angle α can be written as:

s= (r+l).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 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

$$\frac{\partial(\rho.u)}{\partial t} = -\frac{\partial(\rho.u^2 + p)}{\partial x} - \rho.u^2.\frac{1}{A}.\frac{\partial A}{\partial x} - \frac{F_R}{V}, \text{------------------------------(Eq.19)}$$

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 SPEED ON OCTANE NUMBER REQUIREMENT.

The Fig.1 below shows the effect of speed on the octane number requirement of the engine for petrol and ethanol fuels. The figure shows that higher octane number rated petrol and ethanol are needed at low speeds as compared to high speeds. This is because 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.

Also the octane requirement for petrol is higher than that for ethanol as higher pressures and temperatures are developed with petrol fuel.

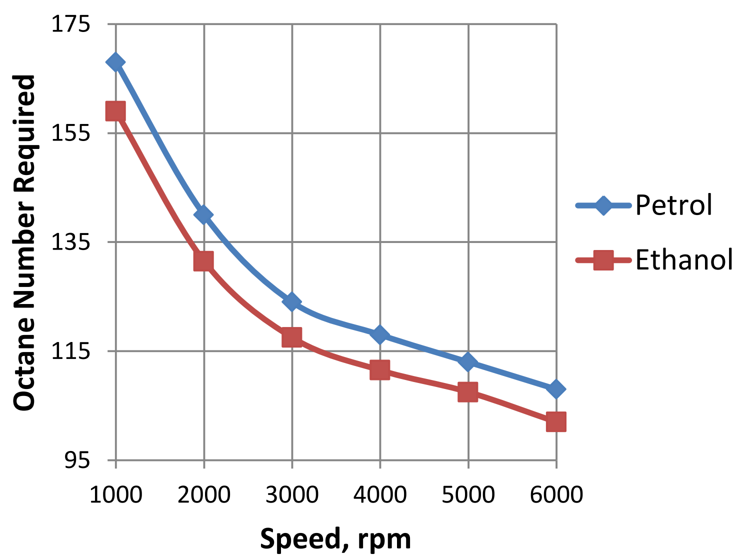

Fig.1 Effect of Speed on Octane Number Requirement

EFFECT OF SPEED ON ENGINE POWER

The Fig.2 Below shows the effect of speed on power developed by the engine with petrol and ethanol as fuels.

It is seen from the figure that the power increases with speed for both the fuels petrol and etanol. This is because the number of power cycles per unit time increase by increasing the engine speed.

The power produced by the engine in the petrol mode is higher than that produced with ethanol as fuel. This is because the heating value of petrol is higher than that of ethanol. Although the stoichiometric air-fuel ratio of ethanol is lower than that of petrol which allows additional amount of ethanol fuel to be supplied per cycle for the same displacement volume, but the higher heating value of petrol still dominates.

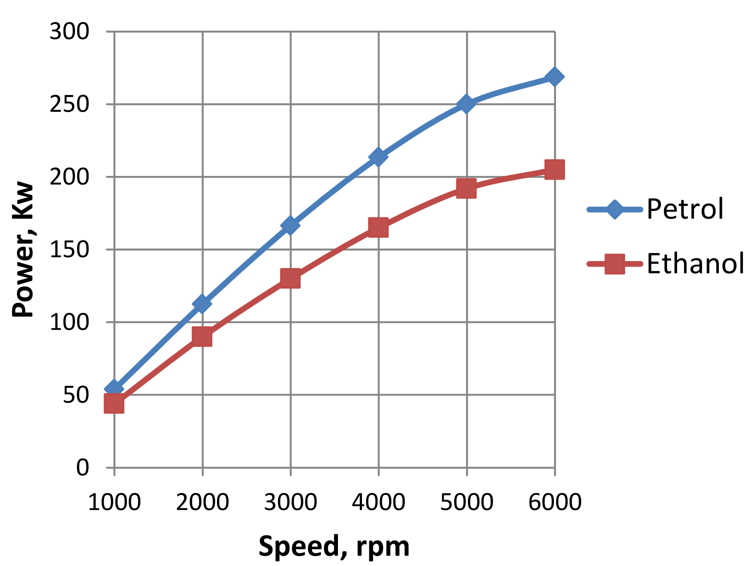

Fig.2 Effect of Speed on Power

EFFECT OF SPEED ON ENGINE TORQUE.

The Fig 3 below shows the effect of speed on engine torque for both petrol and ethanol as fuels. It is seen that the torque produced by the engine is maximum at the speed of 2000 rpm for both the fuels petrol and ethanol. The torque developed is lower on both the sides of above speed of 2000 rpm. This is because the combustion characteristics is best at the speed of 2000 rpm.

Further it is seen that the engine produces higher torque with petrol than ethanol due to the higher heating value of petrol than ethanol.

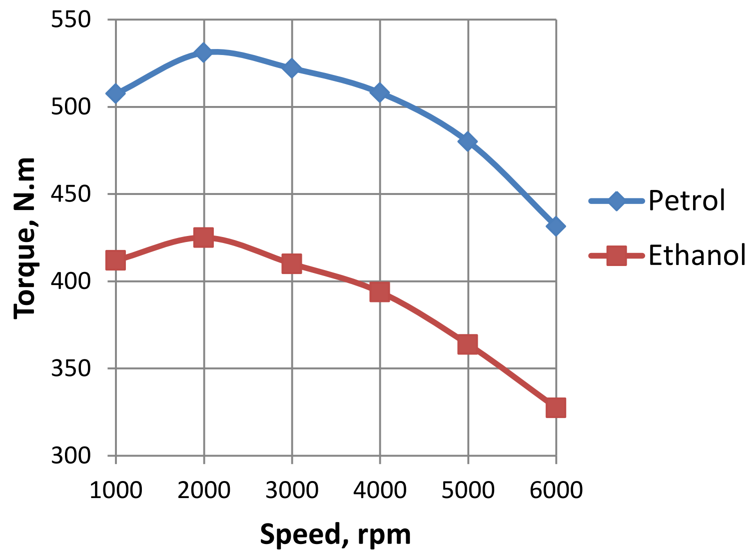

Fig.3 Effect of Speed on Torque

EFFECT OF SPEED ON BRAKE SPECIFIC FUEL CONSUMPTION.

The Fig,4 below shows the effect of speed on the fuel economy of the engine with petrol and ethanol fuels. It is seen that the brake specific fuel consumption first decreases with speed and then again increases. The BSFC is minimum at the speed of 2000 rpm for both the fuels under consideration. The value of BSFC depends upon the ratio of mass consumption rate of fuel and power developed by the engine. The power developed by the engine increases with speed with the corresponding increase in the mass flow rate of fuel. However the power increases at a faster rate up to 2000 rpm which decreases the BSFC for the engine. as the speed is increased beyond 2000 rpm the fuel consumption rate becomes more dominant which increases the brake specific fuel consumption of the engine. the BSFC for petrol is lower than that for the simulated ethanol engine as the power developed by the petrol engine is higher than the power developed by the engine with ethanol as a fuel.

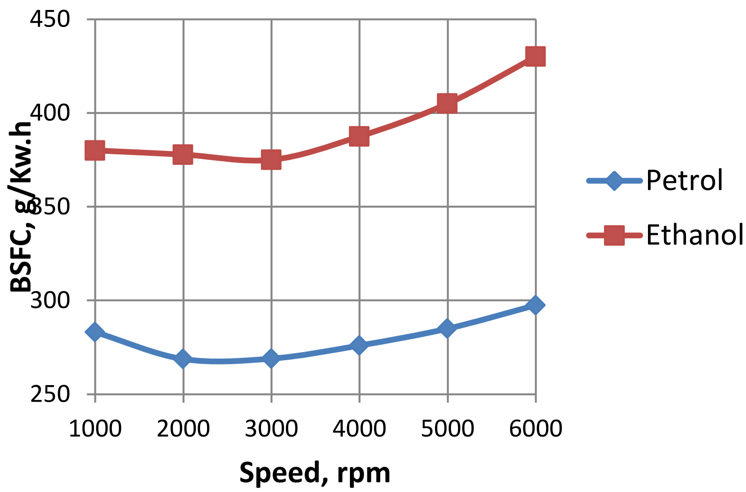

Fig.4 Effect of Speed on Brake Specific Fuel Consumption.

EFFECT OF SPEED ON CO EMISSIONS

The Fig.5 below shows the effect of speed on CO emissions with petrol and ethanol as fuels. It is seen that the CO emissions per unit of energy developed increase as the speed increases with both petrol and ethanol as fuels. This is because by increasing the speed the engine executes more number of power cycles per unit time and therefore the fuel consumption rate increases which increase the CO emissions produced as the speed is increased.

Further the engine produces less CO emissions with ethanol fuel as compared to petrol for the engine under consideration while maintaining the identical design and operating conditions. This is because the ethanol fuel has inherent oxygen in its molecular structure which aids in the further conversion of CO into CO_2 thus reducing its concentration in the exhaust gas.

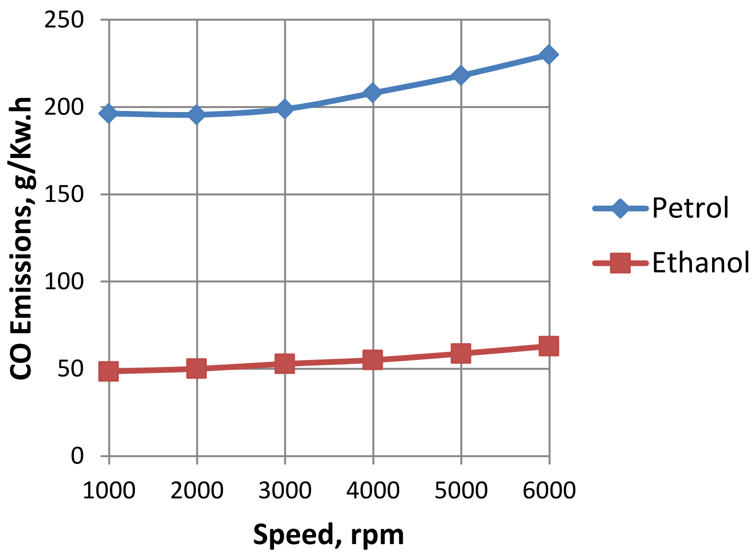

Fig.5 Effect of Speed on CO Emissions

EFFECT OF SPEED ON HC EMISSIONS.

The Fig.6 below shows the effect of speed on HC formation with petrol and ethanol fuels for the engine under consideration.

The HC emission formation from the engine is higher with petrol fuel as compared to ethanol as a fuel. This is because the combustion is better in the ethanol mode due to the presence of oxygen in the ethanol molecule which improves the combustion efficiency and thus reduces the HC formation in the ethanol mode.

In case of both the fuels, petrol and ethanol, the mass flow rate of fuel as well the power developed by the engine increases with speed at different rates.

In case of engine operated with petrol, the power increases at a higher rate as compared to the fuel consumption rate upto 4000 rpm so that the HC emissions formed are decreased upto this speed. Beyond this speed the fuel flow rate becomes dominant as compared to power development which increases the HC emissions.

The HC emissions produced by the engine with ethanol fuel increase as the speed increases. This is because the increase in the fuel consumption rate dominates as compared to increase in power with speed.

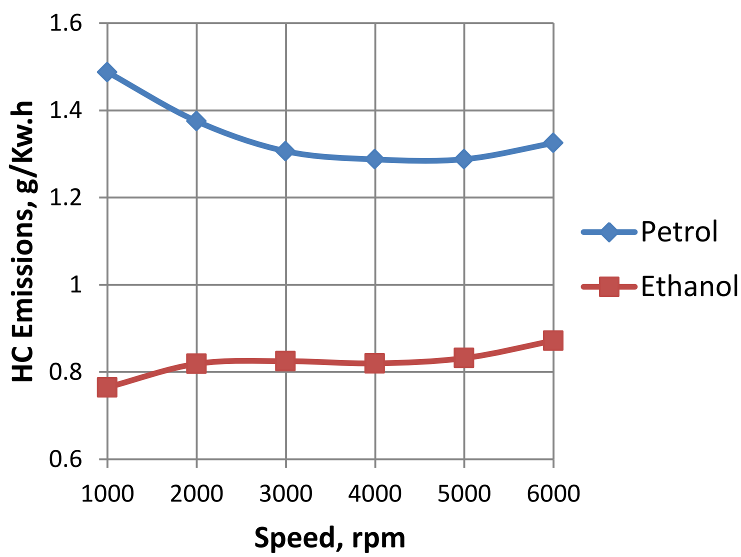

Fig.6 Effect of Speed on HC Emissions

EFFECT OF SPEED ON NOX EMISSIONS.

The Fig.7 below shows the effect of speed on NOx emissions produced by the simulated engine with petrol and ethanol as fuels.

The NOx emissions produced by the engine depends on the high temperatures available in the engine cylinder as well as the substantial time available for allowing the oxygen and nitrogen of the air to react and form NOx emissions. These two conditions are different for the fuels under consideration when used in the same engine.

In case of petrol the NOx emissions increase as the speed increases. This is because the temperatures developed by the engine in the petrol mode are on higher side and increase as the speed increases which favors the rise in the NOx formation rate with respect to speed.

In the case of of ethanol fuel the magnitude of the temperatures developed in the engine cylinder is on the lower side. Further with the increase in speed as the time available for the oxygen and nitrogen to react decreases, the NOx formation rate with ethanol fuel decreases with the increase in speed. Also the NOx emissions produced by the engine with ethanol at the speed of 1000 rpm shows a much high value due to more time available at this lower engine speed which favours NOx formation.

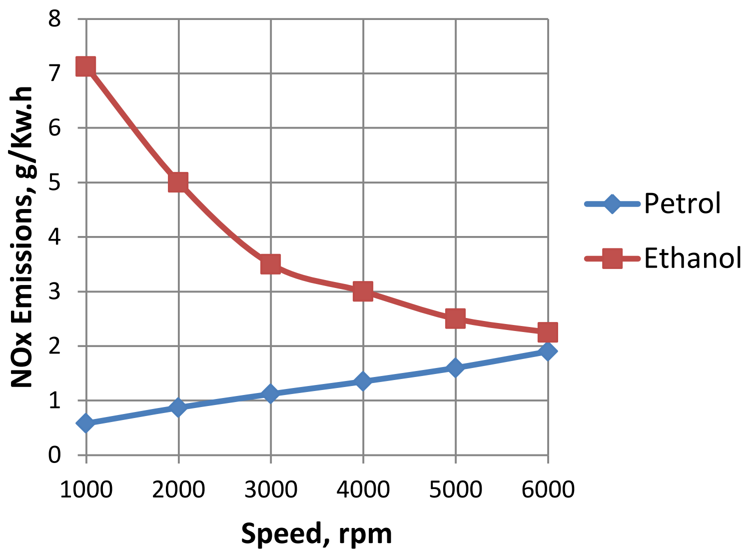

Fig.7 Effect of Speed on NOx Emissions

EFFECT OF SPEED ON EXHAUST GAS TEMPERSATURE.

The Fig.8 below shows the effect of speed on the exhaust gas temperature produced by the engine. it is seen that the exhaust gas temperature increases with speed with both petrol and ethanol fuels. This is because as the speed increases the number of power cycles executed by the engine increase which increases the exhaust gas temperature since the corresponding time available for heat transfer to the coolant decreases. In other words the percentage of energy loss per cycle due to heat transfer across the system boundary or walls decrease with the increase in engine speed.

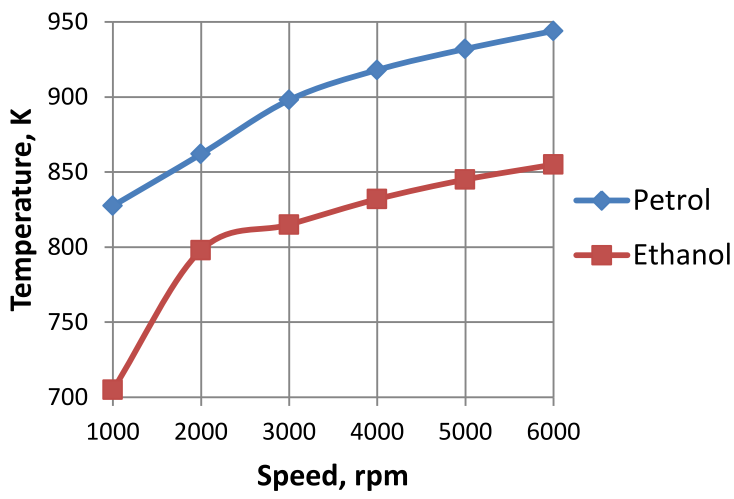

Fig.8 Effect of Speed on Exhaust Gas Temperature

CONCLUSIONS

1. In general turbocharged spark ignition engines demand high octane rated fuels.
2. It is possible to boost the power of a spark ignition engine using a turbocharger provided fuels petrol and ethanol have high octane ratings.
3. Downsizing of a spark ignition engine with the help of a turbocharger can be made possible with high octane rated petrol and ethanol.
4. The octane demand of the turbocharged spark ignition engine for both petrol and ethanol varies with respect to speed and is higher is lower speeds.
5. The turbocharged engine demands higher octane rating for petrol than ethanol.
6. The engine under consideration produces higher power and torque in petrol mode than with ethanol mode.
7. The brake specific fuel consumption of the engine is lower in petrol mode as compared to that with ethanol fuel.
8. The ethanol based engine produces less CO and HC emissions as compared to petrol.
9. The ethanol based engine produces higher NOx emissions than the petrol based 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 (=$\rho.\bar{c_V}.T + \frac{1}{2}.\rho.u^2$)
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

APPENDIX-C
PHYSICAL AND CHEMICAL PROPERTIES OF PETROL AND ETHANOL[15]

Fuel Property	Petrol	Ethanol
Formula	C4 TO C12	C_2H_5OH
Density, Kg/m3	750	800
Lower heating value, MJ/Kg	42.5	26.8
Stoichiometric air-fuel ratio, weight	14.6	9.0
Octane No.	80-98	108

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