

An Experimental Study of Turbocharged Hydrogen Fuelled Internal Combustion Engine

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ABSTRACT

Hydrogen is considered as one of the potential alternate fuel and when compared to other alternate fuels like CNG, LPG, Ethanol etc., it has unique properties due to absence of carbon. In the current work, Hydrogen engine of 2.5 L, four cylinder, spark ignited Turbocharged-Intercooled engine is developed for Mini Bus application. Multi-point fuel injection system is used for injecting the hydrogen in the intake manifold. Initially, boost simulation is performed to select the optimum compression ratio and turbocharger. The literature review has shown that in-order to get the minimum NO_x emissions Hydrogen engines must be operated between equivalence ratios ranging from 0.5 to 0.6. In the present study, full throttle performance is conducted mainly with the above equivalence ratio range with minimum advance for Maximum Brake Torque (MBT) ignition timing. At each operating point, the performance, emissions and combustion parameters are recorded and analyzed in detail. It has been observed that a power of 66.7 kW @3200 rpm and a torque of 180 Nm @2000 rpm are achieved by using neat Hydrogen on Turbo charged engine. The base performance data is compared with CNG naturally aspirated engine (running using stoichiometric operation) and it has been observed that Hydrogen engine is having similar performance above 2000 rpm, however there is torque drop due to operation on Hydrogen which has limitation of the turbocharger to boost at lower engine speeds (running using equivalence ratio ranging from 0.5 to 0.6). Such lean operation also helps to avoid backfire which the hydrogen engine is quite vulnerable. It has also been observed that the combustion stability of the Hydrogen engine is very good at all Indicated Mean Effective Pressure (IMEP) values irrespective of the engine speed with a Coefficient of Variance (COV) around 1-2%.

Keywords: Hydrogen Engine, Combustion, Emissions, Performance, COV

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INTRODUCTION

The conventional petroleum base fuels such as Gasoline and Diesel are being used as a major source of fuels in transport vehicles. This sector is facing a potential threat due to uncertainty and interdependence of petroleum fuel availability and environmental issues like air pollution, global warming etc. These issues can be addressed to certain extent by use of alternate fuels. But most of the alternate fuels are hydrocarbon based that emit CO₂, which is a Greenhouse gas (GHG), may be at a lower rate than conventional fuels. Apart from CO₂ emissions, other pollutants such as CO, HC & NO_x are emitted from the automobiles contribute to ambient air pollution. In order to reduce the CO₂ emission and ambient emissions, various fuels have been considered as substitutes to the conventional hydrocarbon based fuels.

The potential alternate fuels to replace them are mainly alcohols, Liquefied Petroleum Gas (LPG), Compressed Natural Gas (CNG), and Hydrogen (H₂), vegetable oils, bio gas, producer gas and Liquefied Natural Gas (LNG). Among all alternate fuels, Hydrogen (H₂) can be considered as a unique alternate fuel as it is a carbon less fuel and renewable fuel with capability to produce ultra-low emissions. Hydrogen gas, the simplest element in the universe, is one of the cleanest burning alternative fuel available and has the potential to eliminate all carbon emissions. Hydrogen can be adapted to both Spark Ignition (SI) engine and Compression Ignition (CI) engine. In SI engines, hydrogen can be used as a sole fuel, but in the case of CI engine dual fuelling technique can be adopted [2].

Since Hydrogen is a carbon free fuel, on burning hydrogen it will produces only water as its product. Hence the formation of HC, CO & CO₂ during the combustion can be completely avoided.

trace amount of these compounds may be formed due to the partial burning of lubricating oil in the combustion chamber. NO_x is one of the major pollutants in hydrogen operated SI and CI engines [2]. NO_x formed by the thermal dissociation and oxidation of nitrogen (N₂) in atmospheric air during combustion, is the only undesirable engine out emission from a neat hydrogen propelled engine [3]. But coupling this with unique characteristic of hydrogen, low lean-flammability limit, which allows stable combustion even at highly diluted condition [4], achieve near zero engine out emission.

Another characteristic of hydrogen is its tendency of pre-ignition and backfiring [5]. It is observed that these issues can be tackled if the proper selection of design parameters such as equivalence ratio, compression ratio is determined for stable engine operation. Hence in the current work, we chose to arrive at a best possible equivalence ratio and compression ratio to develop a practical & stable hydrogen engine which is targeted to be used on mini bus vehicle for transportation. The study emphasis on the process of development and on the testing of the hydrogen engine to understand the performance, emission and combustion characteristics.

METHODOLOGY

The following methodology is used in the present work.

We have checked in the literature the best possible region of equivalence ratio with which we can operate engine to get the minimum NO_x emissions. Initial simulation was done with AVL boost simulation package to determine an optimum compression ratio and turbocharger configuration.

Hydrogen engine operating strategy is to determine the optimum equivalence ratio for minimal NO_x emissions. Based on the optimum simulation data, a four cylinder Hydrogen fuelled engine is redesigned for Hydrogen from a base CNG engine.

Further, in this paper, emphasis is given on the hydrogen engine testing to understand the performance, emission and combustion characteristics.

HYDROGEN ENGINE OPERATING STRATEGY

The main parameters which need to be considered while operating a Hydrogen engine are performance and emissions. The Hydrogen engine should give best possible performance with lowest NO_x emissions. Considering the learning from the previous work, if we reduce the combustion chamber flame temperature or in-cylinder thermal loading, NO_x emissions from the Hydrogen engine reduces [6]. At the same time, it also reduces the heat losses from the combustion chamber walls, which will lead to increase in thermal efficiency. Due to this, risk of backfiring and pre-ignition also reduces as the reduction in combustion chamber temperature indirectly leads to reduction in hotspots inside the combustion chamber. The flame temperature can be reduced by using a very lean mixture.

To achieve lowest NO_x emissions, we need to operate the engine in the equivalence ratio ranging from 0.5 to 0.6 which can be observed from Figure 1. At this equivalence ratio range, flame temperature gets reduced and it will lead to reduction in NO_x emissions [7]. Hence we have decided to operate the engine in the above equivalence ratio range which is a conventional qualitative method of governing. This equivalence ratio range also reduces the risk of backfiring and pre-ignition.

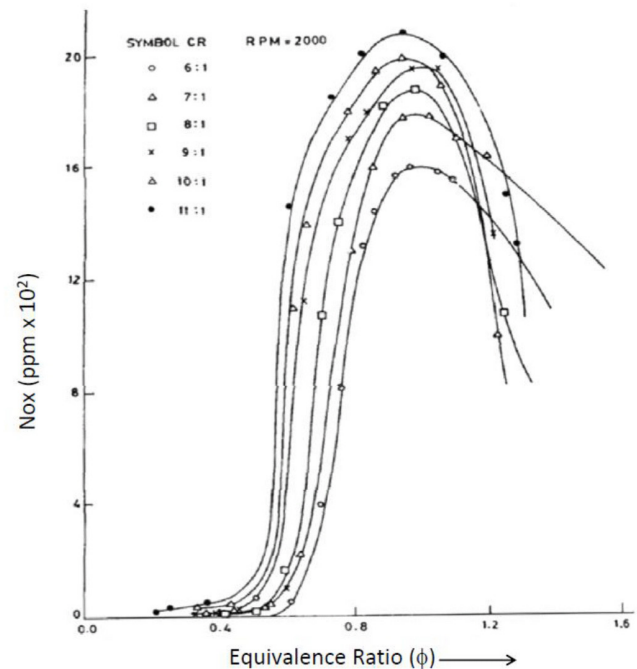


Figure 1. NO_x Emission as a function of equivalence ratio for different Compression ratio [7]

AVL BOOST MODEL PREPARATION FOR HYDROGEN ENGINE

AVL Boost is one-dimensional gas exchange engine combustion simulation software, which can predict performance at steady state operating conditions. It can predict specific fuel consumption, power, torque, and mean effective pressures at full throttle and steady state operating conditions [8]. In the present study, AVL Boost is used to simulate the performance of Hydrogen engine.

The Hydrogen engine specifications are mentioned in the Table 1.

Table 1. Hydrogen Engine Specification

Base Specifications	
Number of Cylinder	4
Stroke	88.9 mm
Bore	100.3 mm
Number of Valves per cylinder	2
Fuel	Hydrogen
Aspiration	Turbocharged, Intercooled
Power	66.7 kW @ 3600 rpm
Torque	180 Nm @ 2000 rpm
Compression Ratio	12

The first step of simulation task is to create an accurate engine model which should consist of all the elements most similar to the existing physical engine. Figure 2 represents the engine model used for the current simulation.

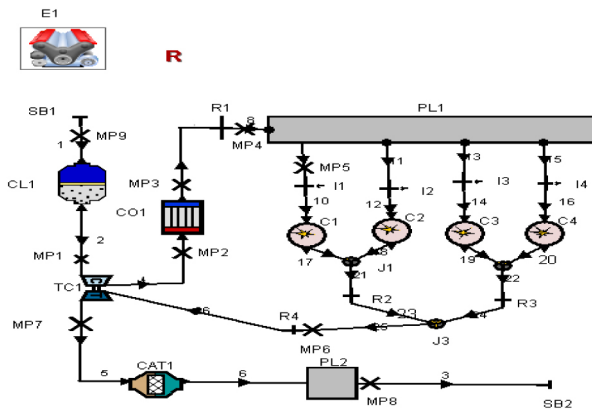


Figure 2. Boost Simulation Model

AVL Boost enables a detailed simulation of entire engine including the complete intake and exhaust system. Figure 2 shows the AVL Boost simulation model of the 2.5 L Hydrogen engine.

Description of the Model

The air is aspirated from system boundary SB1 through the intake snorkel pipe 1 to the air cleaner CL1. The cleaned air is sent to the turbocharger compressor via the pipe 2. TC1 is the turbocharger, from TC1 compressed air is sent to the intercooler CO1 via pipe 4. The intercooled air is passed through the restrictor R1 (which is modeled similar to the Throttle Body) to the intake manifold.

The intake manifold is split into two volumes. PL1 represents the plenum volume and the pipes 9, 11, 13 and 15 represent the runners for the cylinders C1 to C4 respectively. Pipes 10, 12, 14, 16 represent the intake ports for the cylinders C1 to C4 respectively. Injectors I1 to I4 are placed before the intake ports of the cylinders C1 to C4 respectively.

The exhaust ports are modeled in a similar way by the pipes 17, 18, 19 and 20. Before going into the exhaust manifold, the exhaust ports 17, 18 are connected to a common port 21 via junction J1 and the ports 19, 20 are branched to a common port 22 via junction J2 respectively. The exhaust ports 21 and 22 are connected at the flanges represented by restriction R2, R3 to the exhaust runner pipes 23 and 24 respectively. The exhaust runner pipes 23 and 24 are connected to junction J3 and from there they are sent to the exhaust outlet flange represented by restriction R4 through pipe 25. Exhaust gas enters into the turbocharger through pipe 26.

Exhaust gas expands in the turbocharger TC1 and goes to the catalytic converter CAT1 through pipe 5. From catalytic converter CAT1, the exhaust goes to the plenum PL2 (which is modeled similar to the silencer) through pipe 6. The exhaust goes finally let into atmosphere from plenum PL2 to ambient - system boundary SB2 through tail pipe 3.

Details of the Combustion Model

For thermodynamic calculations, rate of heat release is needed to be provided as input. The vibe function is often used to approximate the actual heat release characteristic of the engine [8].

$$\frac{dx}{d\alpha} = \left(\frac{a}{\Delta\alpha_c} \right) \cdot (m+1) \cdot y^m \cdot e^{-a \cdot y^{(m+1)}} \quad (1)$$

$$dx = \frac{dQ}{Q} \quad (2)$$

$$y = \frac{\alpha - \alpha_0}{\Delta\alpha_c} \quad (3)$$

where,

Q is the Total Fuel Input

α is the crank angle

α_0 is the start of combustion

$\Delta\alpha_c$ is the Combustion duration

m is the shape parameter

a is the vibe parameter

The integral of the Vibe function gives the fraction of fuel mass which was burned since the start of combustion [7].

$$x = \int \frac{dx}{d\alpha} \cdot d\alpha = 1 - e^{-a \cdot y^{(m+1)}} \quad (4)$$

where,

x is the mass fraction burned

In the current work, Vibe 2 zone combustion model is used. The inputs for the Vibe 2 Zone combustion model are same as the Vibe function. However instead of one mass averaged temperature, two temperatures (burned and unburned zones) are calculated [9]. One more advantage of using this model is that this model also predicts the knocking characteristics of the engine, provided the actual heat release is described properly by the vibe function specified [9].

Vibe parameter (a) is specified as 6.9 as the combustion is assumed as complete combustion.

During our single cylinder hydrogen engine testing, pressure crank angle histories are captured. Using the same experimental pressure crank angle histories of our single cylinder hydrogen engine test data and the other parameters of the Vibe i.e., duration of combustion, shape parameter are calculated. Typical vibe parameters which are from that analysis are indicated below for reference [6].

During our previous work, a good correlation between the simulation work and experimental data is shown and hence, in the current work simulation is used to determine the compression ratio and turbocharger configuration. As the validation of Boost simulation data with respect to experimental data is already shown in our previous work [6], hence in this paper more emphasis is given on the experimental test data.

Table 2. Typical Vibe Parameters [6]

Speed (rpm)	Shape Parameter	Combustion Duration (°)
1600	2.3	20
2000	2.2	23.1
2400	2.24	28

Details of the Engine Friction Model

Engine friction adversely affects the maximum work output and fuel economy characteristics of an engine and directly accounts for much of the difference in fuel consumption between cold and full-warm engine operation [8]. For calculation of engine friction, the Shayler, Leong, Murphy - Model is used which is available in the Boost program. Based on the set of engine type and engine geometry related input data, Boost program calculates the frictional mean effective pressure (FMEP) of the engine and is provided as input to the simulation.

Details of the Knock Model

The knock model available in the Boost calculates the minimum octane number required for the engine operation free of knock [8].

The threshold for the onset of knock is exceeded if the integral [8].

$$\int_0^t \frac{1}{\tau_{id}(t)} dt \quad (5)$$

where,

T_{id} is the ignition delay at the unburned zone's condition is larger than one before the end of combustion is reached.

The ignition delay for the knock model depends on the octane number of the fuel and the gas condition. The equation for ignition delay is given below [8].

$$\tau_{id} = A \cdot \left(\frac{ON}{100}\right)^a \cdot p^{-n} \cdot e^{\frac{B}{T}} \quad (6)$$

where,

T_{id} is the ignition delay in ms

ON octane number of the fuel

p is the pressure in atm

T is the temperature in K

A, a, n, B are the model constants

$A = 17.68$ ms

$a = 3.402$

$n = 1.7$

$B = 3800$ K

The approach for determining the model constants are by experimentally running the engine at different knocking conditions and obtaining the pressure crank angle histories at different conditions for knocking combustion. Fine tuning the above parameters in Boost case explorer in such a way that the predicted pressure curves are matching with the experimental pressure curves.

Since there are no model constants are available for hydrogen fuel and also no experimental cylinder pressure data with knocking conditions, the model constants which are specified in the Boost manual for the default condition are taken as the same for our simulation work and are as mentioned above.

Details of the Heat Transfer Model

The heat transfer model used for the simulation is Woschini model which is available in the Boost software and this model calculates the heat transfer rates to the piston, cylinder head, and liner [8]. Wall temperatures are provided as the input to the heat transfer model and the wall temperatures of piston, cylinder head, liner at TDC and liner at BDC are assumed as 585 K, 585 K, 585 K and 408 K respectively which are taken from our single cylinder hydrogen work as they provided a good correlation with the experimental work [6].

Details of the Species Model & Simulation Convergence Criteria

Classic species transport model is used to provide the thermodynamic properties.

The cycle calculation is done at a time step chosen automatically by the Boost software based on the CFL (Courant, Friedrichs and Lewy) stability criteria [8]. The cycle simulation is repeated till convergence to IMEP with specified error value is achieved.

Details of the Turbocharger Model

Boost provides two methods to model a Turbocharger.

1. Simplified Model
2. Full Model

Simplified Model is meant for steady state performance. This model provides further three modes for turbocharger simulation [8].