

# Development of Multi Cylinder Turbocharged Natural Gas Engine for Heavy Duty Application

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## ABSTRACT

CNG has recently seen increased penetration within the automotive industry. Due to recent sanctions on diesel fuelled vehicles, manufactures have again shifted their attention to natural gas as a suitable alternative. Turbocharging of SI engines has seen widespread application due to its benefit in terms of engine downsizing and increasing engine performance [1].

This paper discusses the methodology involved in development of a multi cylinder turbocharged natural gas engine from an existing diesel engine. Various parameters such as valve timing, intake volume, runner length, etc. were studied using 1D simulation tool GT power and based on their results an optimized configuration was selected and a proto engine was built. Electronic throttle body was used to give better transient performance and emission control. Turbocharger selection and its location plays a critical role. Turbocharger Wastegate actuator trials were conducted to select optimum actuator to restrict boost enough to meet power target. Various engine related MAP like main lambda, volumetric efficiency, MBT timing, Primary ignition timing, etc. were calibrated and saved in EMS. The full load curve was calibrated using EMS to meet performance target of 85 kW rated power and 285 Nm max torque. Further, through precise calibration of various ECU tables the engine managed to meet BS IV emission norms.

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## INTRODUCTION

Ever-increasing cost & limited resources of petroleum based fuels and the stringent emission norm has put lot of pressure on the automobiles manufacturer and researchers to improve the fuel economy and to reduce the exhaust gas emission of their vehicles [2]. This has lead manufacturers and scientist to experiment with newer technologies. Downsizing of engines have seen tremendous scope when it comes to SI fuelled engines. Use of alternative fuels along with downsizing of engines has helped engineer's meets stringent emission norms while maintaining vehicle performance and also improving fuel efficiency [3, 4].

CNG due to its low C/H ratio is known for emitting lower amounts of CO<sub>2</sub> than gasoline [3]. The improvement of fuel efficiency will directly translate into lesser CO<sub>2</sub> emission which will give manufacturers an edge once CO<sub>2</sub> is included as a regulated pollutant in the forthcoming BSVI emission norms that is to be implemented by year 2020 in India.

Turbocharging an engine has long been considered the best way to improve engine performance. Its application in diesel engines is seasoned but has rarely been used in SI engines. The presence of a throttle body in the intake stream of an SI engine can cause functional problems for a turbocharger which were investigated and solutions to avoid the same were implemented. The constant opening and closing of the throttle causes continuous zones of low and very high bottled up pressure before and after the compressor. Not venting the high pressure after the compressor when the throttle body is closed can lead to damage of the compressor bearings. CNG being a gaseous fuel of higher calorific value sees higher exhaust temperatures. Diesel engine turbocharger are designed to withstand exhaust temperatures of up to 705°C. Turbocharging a CNG engine can see exhaust temperatures exceed 800°C.

The base engine was a multi cylinder turbocharged diesel engine used on heavy duty above 3.5T applications. The base engine was converted to a turbocharged dedicated CNG engine for the same application. This paper discusses the different phases, calibration activities and experimentation that was involved in the development.

## ENGINE SPECIFICATION

Base diesel and proposed CNG engine specifications are mentioned in the table 1. It can be seen that while converting the base diesel engine to CNG application the rated power target was increased by around 20% and there was change in the required torque target too. To achieve the desired target the base BMEP level was increased from 9.9 bar to 11 bar. Hence it was indeed required to check the suitability of existing base diesel engine's components for CNG operation.

Table 1. Engine Specification

|                   | Base Diesel Engine | Proposed CNG Engine        |
|-------------------|--------------------|----------------------------|
| No of Cylinder    | 5                  | 5                          |
| Cubic Capacity    | 3.25 Ltr           | 3.25 Ltr                   |
| Power (kW @ RPM)  | 70 @ 3000          | 85 @ 3000 (3400 initially) |
| Torque (Nm @ RPM) | 255 @ 1400 - 2000  | 284 @ 1400 - 2000          |
| Injection System  | VE Mechanical      | Multi Point Fuel Injection |

## EMISSION NORMS

The new engine is developed to meet BSIV engine emission norms. And those are as per Table-2. Applicable cycle is ETC when checked for normal engine testing with no DTC/fault in the system or component.

Table 2. Emission norms

| BSIV Emission Norms    |         |         |         |         |                |
|------------------------|---------|---------|---------|---------|----------------|
| Monitored Component    | CO      | NMHC    | CH4     | NOx     | PM             |
|                        | (g/kWh) | (g/kWh) | (g/kWh) | (g/kWh) | (g/kWh)        |
| Legislation Limit (LL) | 4       | 0.55    | 1.1     | 3.5     | Not Applicable |
| Engineering Target     | ≤ 3.2   | ≤ 0.44  | ≤ 0.88  | ≤ 2.8   | Not Applicable |

## REVIEW OF EXISTING COMPONENTS AND PROPOSAL FOR NEW COMPONENT DEVELOPMENT

### Intake Manifold and Cylinder Head

Initially the existing cylinder head and intake manifold was tested for swirl. This Intake manifold has small volume and cylinder head port was designed to create swirl which is very critical for diesel engines. CNG engines requires intake manifold with larger volume for smooth operation and greater torque response [1, 5]. Below figure shows that the mean swirl number increases with the use of short runner intake manifold i.e. diesel engine intake manifold. This higher swirl is not desirable for CNG application.

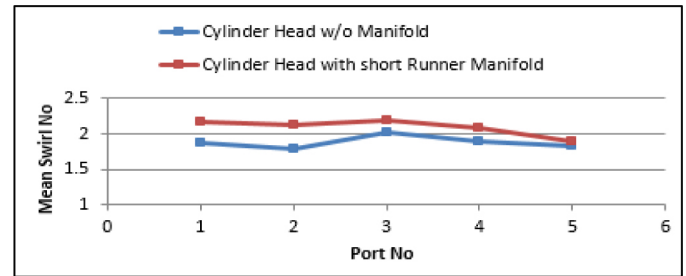


Figure 1. Mean Swirl Number for Existing Intake Manifold

So to overcome this issue, the new intake manifold was proposed on the basis of 1D and 3D simulation which has bigger plenum volume and runner length compared to diesel intake manifold which will help in storing higher charge/ air as a reservoir. Also the runner length and its orientation is finalized in such a way that it kills the swirl and improve the overall volumetric efficiency. To accommodate spark in place of diesel injector a small protrusion is created in intake port area which in turn helps to reduce the swirl. Figure 2 shows the mean swirl number comparison with new and existing intake manifold.

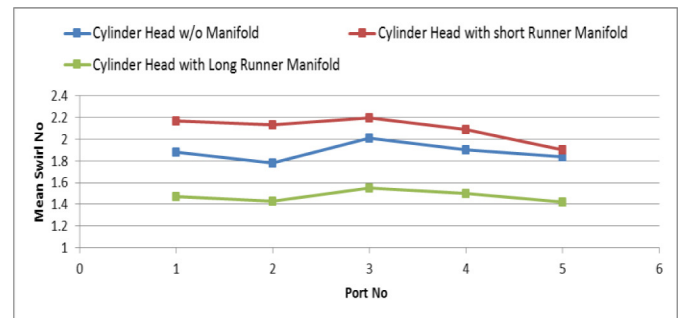


Figure 2. Mean Swirl Number for Diesel and CNG Intake Manifold

### Cooling and Lubrication System

The existing water pump was designed to meet the cooling requirement for the 70kW engine. The cooling requirement for the proposed engine was calculated and it was found that water flow requirement is 211 LPM for rated Power & 115 LPM for Max torque. So the existing water pump was not suitable for the CNG application. So accordingly new water pump is designed and used further.

The same case was with the lubrication system. The existing oil pump was also not capable for providing required oil flow with the desired pressure. So accordingly new requirement was given and new oil pump was developed. The existing oil was also reviewed. Oil with low ash additives is suggested which will give protection against carbon deposit. This oil caters low sulphur to protect after-treatment device. Also this offer outstanding resistance to pre-ignition and protection against carbon deposit on valves and spark plugs.

### Piston & Ring Pack

Piston and ring packs are one of the major contributing components to achieve desired performance and emission. Piston & ring pack are reviewed for the following point of views:

- minimize dead volumes
- minimize friction losses
- minimize oil consumption
- improve durability

Considering the increased BMEP of developed CNG engine it is recommended to use symmetric / asymmetric barrel type compression ring. The key stone ring used in diesel engine is not recommended considering its scraping operation due to low clearance & high temperature generated in CNG engines.

### Compression Ratio

For existing diesel engine CR is 18:1. For converting the same engine to run for CNG, Compression Ratio needs to be reduced. For proposed CNG engine two different CR ratios (10.5 & 11) were initially proposed. However to achieve the more power in CNG engine, boosting in the form of turbocharger is considered. Accordingly air requirement is calculated & it is observed that the inlet air temperature is the key factor to decide the CR to avoid auto ignition of CNG fuel [5]. While considering the worst case situation, atmospheric intake air temperature is ranging from 45°C to 50°C & boost pressure of around 1.5 bars, there are chances of significant increase in compression temperature. So in the present engine configuration Compression Ratio (CR) 10.5: 1 was used for further optimization. And if required CR of 11:1 will be considered.

### Combustion Chamber

The existing shape of the combustion chamber (Piston Bowl) in the diesel engine is re-entrant type, which is not suitable for CNG application. Combustion chamber for CNG engine is designed based on parameters like turbulence, low swirl and squish [5, 6]. For natural gas due to high activation energy, the laminar flame speed is low which results in longer combustion duration. Thus the total combustion period becomes prolonged as compared to diesel and petrol. This leads to loss in efficiency. Longer the total combustion time, higher chances of the remaining unburned mixture to undergo pre-flame reactions and self-ignite.

Fuel burning before top dead center increases the work required for compression, while that of burning late in the cycle performs less work on the piston during expansion. This problem of low flame speed can be solved by two ways. The first is that the ignition timing could be advanced. The other option is the development of combustion chamber specifically for natural gas operation by Increasing turbulence & squish in the combustion chamber to increase flame speed. A squish generated charge motion combustion chamber had its effects on the burning rates. High levels of turbulence generated from the squish affected the faster burning rates, which resulted in improvement in thermal efficiencies. Figure 3 shows the typical combustion bowl shape for diesel and CNG engine.

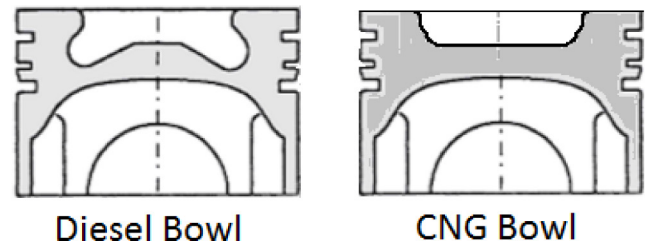


Figure 3. Typical Diesel and CNG combustion bowl

### Piston Pin

Due to change in the peak firing pressure of the engine, it is essential to check the strength of the piston pin. FE analysis is done for checking structural integrity of the piston pin.

The analysis is based on the simple assumption that; pin will act as a simply supported beam when assembled in the piston. Thus, pin is held on its bottom side in patch of 120 degrees. (Assuming that to be the contact patch.) Gas load is applied on the pin, at each node. This force is also applied on contact patch of 120 degrees. Appropriate material properties are assigned to the material for determining the stress and the displacements. Analysis is done on three different pressure conditions (90bar, 100bar, 110bar). Figure 4. Shows the case set up for the piston pin analysis.

The existing diesel piston pin is having 5.5 mm wall thickness and 74mm length. The max. Stress observed at 110 bar combustion pressure is 454 MPa which is greater than maximum fatigue strength of the material selected i.e. 420 MPa and hence existing piston pin was not suitable for the CNG engine application.

So accordingly the suggestions were made and the Pin thickness was increased to 7mm and length of the pin was increased by 3mm and again the simulation was done. It was found that max. Stress observed at 110 bar combustion pressure is 340 MPa, which was very much below the acceptable limit and hence it was accepted.

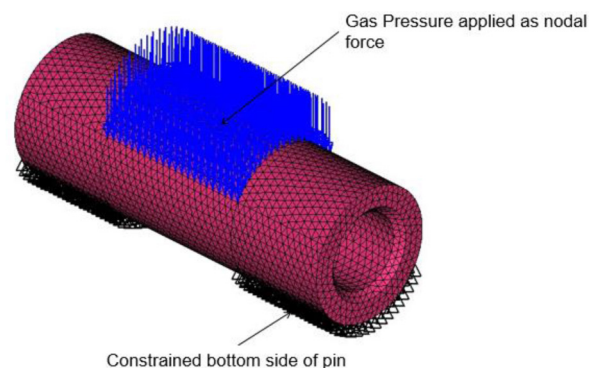


Figure 4. FEA case set up for piston pin analysis

Table 3. Shows the difference between the maximum stress level and deflection for existing and modified piston pin.

Table 3. Piston Pin Simulation Result Summary

| Case | Analysis Pressure (Bar) | Deformation at the loading boundary (mm) |                | Max. Stress (MPa) |                | Stress Due to Boundaries (MPa) |                |
|------|-------------------------|------------------------------------------|----------------|-------------------|----------------|--------------------------------|----------------|
|      |                         | (Existing Pin)                           | (Modified Pin) | (Existing Pin)    | (Modified Pin) | (Existing Pin)                 | (Modified Pin) |
| 1    | 90                      | 0.0211                                   | 0.0134         | 362               | 278            | 792                            | 712            |
| 2    | 100                     | 0.0223                                   | 0.0149         | 402               | 309.1          | 874                            | 791            |
| 3    | 110                     | 0.0251                                   | 0.0164         | 454               | 340            | 900                            | 870            |

Stress due to boundaries is not the realistic stress and hence can be neglected

### Catalytic Convertor

The base engine was not having any after treatment system as the applicable norms was BSIII. For this proposed BSIV CNG engine the three way catalyst was recommended. The light off temperature suggested by the catalytic converter manufacturer was around 350°C. The loading of the new catalytic converter was 60 gm/ft<sup>3</sup> and the relative concentration for Pt/Pd/Rh was 0/90/10 while keeping the cell density constant at 400 cpsi.

### Turbocharger

As CNG engine is having more power target than the base diesel engine, turbocharger is the prime need to fulfill the power target. The proposed CNG engine is running on stoichiometric principle so for calculating the required air flow rate initially the SFC target was set for each engine operating speed. Then on the basis of fuel consumption the required air flow was determined. For this engine intercooler was also considered as after turbocharging the temperature of the intake air will rise drastically. This rise in temperature may further reduce the air flow and may cause increase in the compression temperature which may then lead to knocking. So the air temperature after intercooler was fixed and further calculations for turbocharger was done.

Initially two different calculations one without EGR and other with EGR were done. The actual test results of the selected turbocharger are discussed in the subsequent part of this paper.

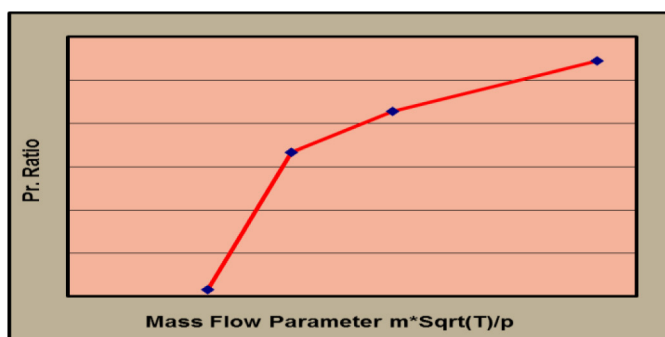


Figure 5. With EGR Compressor Details

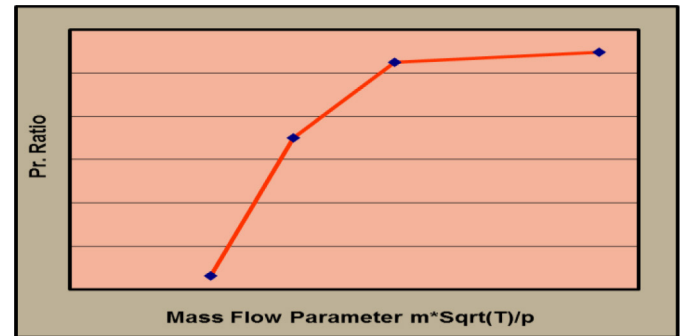


Figure 6. Without EGR Compressor Details

## 1D SIMULATION

1D simulation model is built in the GT power software. This model is built to predict the effect of different valve timing, plenum volume and runner length of Intake Manifold on engine performance [7]. A schematic engine model is shown in Figure.7 Actual engine components are schematically represented by symbols. Geometrical and operating boundary conditions are applied as an input to the model. Simulation runs are taken at full throttle condition for maximum power engine speed, maximum torque engine speeds and idle speed.

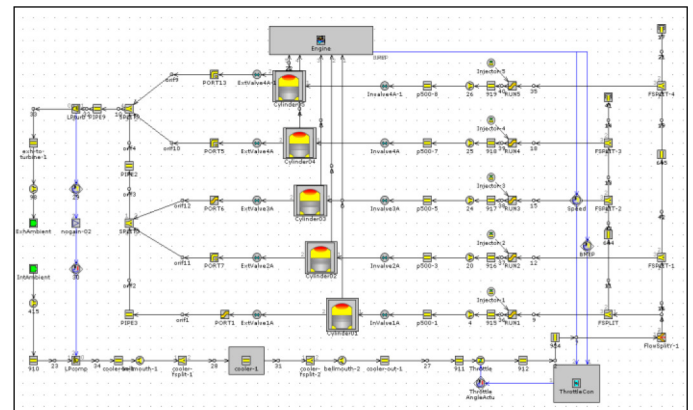


Figure 7. Schematic 1D simulation model

### Valve Timing Study

Existing valve timing was taken as a base for further investigation. To have the minimum changes in cam shaft profile, only cam lobe shifting was considered & accordingly intake & exhaust valve timing was retarded & advanced by 10°. So there are total 9 cases was studied with different IVO, IVC, EVO, EVC events. The effect of various valve timings on the volumetric efficiency, BSFC, Residual fraction, Air flow rate, and intake pressure is shown in figure no 8 to 13 respectively.