

HD Diesel Engines Development for Alternative Fuel Use

Pedro M. Barroso, Xavier Ribas, Mario Pita Sr, Judith Dominguez, and Edgard De Seia
Applus IDIADA

ABSTRACT

This work is based on the development of heavy-duty diesel engines for alternative fuel use. Three diesel engines for commercial vehicle applications were studied: a 13L diesel engine was converted to a dedicated lean-burn NG engine and two diesel engines (14 and 4.25L) were converted to a dual-fuel operation with diesel-NG and diesel-LPG respectively. The dedicated NG engine conversion was achieved by means of some relevant modifications such as the reduction of the compression ratio, design of a gas injection system, design of a spark plug adapter, and implementation of a complete EMS. In relation to dual-fuel cases, some minor modifications were made to the diesel baseline engines such as the installation of the gas train components and the implementation of a gas ECU for the management of the diesel and gas injection using some CAN bus J1939 signals. The results show the feasibility of this kind of engine conversion in terms of its effect on performance and pollutants emissions, especially PM reduction, compared to the baseline diesel engine certification limits.

CITATION: Barroso, P., Ribas, X., Pita Sr, M., Dominguez, J. et al., "HD Diesel Engines Development for Alternative Fuel Use," *SAE Int. J. Engines* 8(1):2015, doi:10.4271/2015-26-0056.

INTRODUCTION

Due to its high performance, high reliability and especially for its low fuel consumption, diesel engines have been the preferred power source for heavy-duty vehicles for decades in on-road and off-road applications. Nevertheless, the reduction of emissions from diesel combustion particularly NO_x and PM, without deterioration of the aforementioned advantages is one of the most important goals that designers are facing due to the exhaust emission legislation becoming more and more stringent. The emission issues is not the only limitation, the availability of liquid fuel and the risk for the future stability of supply resulting from a significant price increase are continued to be the challenges of this kind of technology.

One of the most promising technical measures to reduce the amount of liquid fuel energy consumed by a HD diesel engine is its conversion for alternative fuels use which have potential benefits in terms of fuel cost reduction, improving local air quality, reducing noise and PM emissions. The mentioned diesel conversion for alternative fuel use can be total or partially performed. In the case of total conversion, the combustion system is changed in order to work from Diesel to spark-ignited engine; this conversion can include several modifications such as introducing spark plugs or changing the combustion chamber in order to obtain a new compression ratio [1, 2, 3].

In the case of partial engine conversion dual-fuel concept is used. A dual-fuel engine is conceived to simultaneously operate with diesel and a gaseous fuel, both of them being metered separately (Figure 1).

Two types of technology are currently considered according to the way of gas injection: high pressure direct injection and indirect injection which can be done in two ways: single point injection (before or after turbocharger) or multi-point gas injection in the intake manifold. The emissions, thermodynamics, performance of the dual-fuel engine must be similar or better than the diesel original engine by using any of two technologies [4, 5, 6, 7, 8, 9, 10].

The major advantages of the gas direct injection are that its THC and CO emissions are low, there is no limitation by knocking at high loads, and it can replace more than 90% of the diesel fuel (by energy). The main disadvantages for this technology are the introduction of the new injection system which can lead to engine cylinder head modifications for retrofitted engines and also a direct communication with OEM controller software could be necessary. In the case of high pressure direct injection of NG this can be done with CNG or LNG, being the last more suitable for long-distance transport. This technology is outside the scope of this work.

In the case of gas indirect injection the most important advantages are the simplified engine control with no direct communication with OEM, the non-intrusive technical modifications in the engine structure, and the low cost for its implementation compared to gas direct injection. The main disadvantages of gas indirect injection are that the upper load range is typically limited by knocking, higher THC and CO emissions are presented especially at part loads which

lead to the installation of a two-way catalyst in the exhaust line and substitution rates of 50% to 75% can be achieved depending on duty cycle and the level of integration with the diesel baseline engine.

There are several alternative fuels technologies available, including natural gas, LPG, bio fuels, methanol, ethanol, synthetic fuels. In this work two alternative fuels were studied: NG for dedicated and dual-fuel engine conversion and LPG for dual-fuel engine conversion hence the main objective of this paper was to describe the main activities performed in the development of a HD diesel engine used in commercial vehicles applications for alternative fuel use.

The paper is organized as follows. In the next section, the three engines tested, the test bench installation, the main instrumentation and measurement devices used are described. The tools and methods used to reach the objective of this work are explained in the methodology section and a special section with the main activities performed in the HD engine development process is included before the description of the main experimental results. Finally, the main conclusions drawn in this study are exposed at the end of the document.

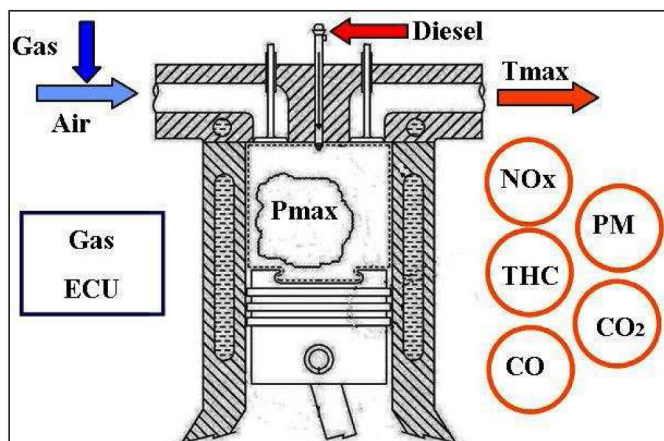


Figure 1. Dual-fuel engine concept (gas indirect injection)

THE EXPERIMENTAL SET-UP

The experimental tests were carried out on three turbocharged intercooled heavy-duty DI diesel engines for commercial vehicle applications. Table 1 summarizes the main engines specifications. The engines were fully instrumented with fast response thermocouples and mean pressure sensors, which were placed in convenient zones. The temperature of all the fluids: inlet air, EGR, cooling water, lube oil and fuels were measured by PT100 and K-type thermocouples. For pressure measurement different types of sensors were used according to pressure requirements (absolute, relative and differential). In addition to that, in order to obtain the maximum turbine inlet temperature and the maximum in-cylinder pressure (T_{max} and P_{max} in Figure 1) the exhaust manifold was fully instrumented and AVL 621 indicating system was used respectively.

The installation used in this study is a certified ISO-17025 state-of-the-art engine test cell equipped with modern measurement equipment and devices for performing activities such as engine development, engine certification and homologation as well as thermal shock, conformity of production and durability tests. The engine test bench was equipped with an active dynamometer with correspondingly maximum values of: power (450 kW), torque (2869 Nm) and speed (4500 rpm). More specifically, the dynamometer is an AC electric machine manufactured by ABB equipped with a digital high-precision torque transducer HBM T-type. These components are linked with a PC via AVL PUMA OPEN 1.5.3 - EMCON 400 management system to program and run steady-state and transient tests as well as the worldwide emission tests automatically. A general schematic of the experimental set-up used in this work is shown in Figure 2.

Table 1. Main characteristics of the three baseline diesel engines.

13L diesel engine (dedicated NG conversion)	
Bore x Stroke	135 x 150 mm
Compression ratio	16.5:1
Number of valves	2 (intake) – 2 (exhaust)
Intake configuration	VGT + CAC
Power level	352 kW @ 1900 rpm
Torque level	1900 Nm @ 1200 rpm
Injection system	Electronic Unit Injectors
Emissions level	Euro IV
14L diesel engine (diesel-NG conversion)	
Bore x Stroke	134 x 168 mm
Compression ratio	16.75:1
Number of valves	2 (intake) – 2 (exhaust)
Intake configuration	VGT + CAC + high pressure EGR (cooled)
Power level	>380 kW @ 1800 rpm
Torque level	> 2100 Nm @ 1200 - 1500 rpm
Injection system	Electronic Unit Injectors
Emissions level	EPA 2005
4.25L diesel engine (diesel-LPG conversion)	
Bore x Stroke	102 x 130 mm
Compression ratio	17.4:1
Number of valves	2 (intake) – 1 (exhaust)
Intake configuration	110 kW @ 2300 rpm
Power level	Turbocharger (mech. wastegate) + CAC
Torque level	520 Nm @ 1200 - 1500 rpm
Injection system	Unit injection pumps
Emissions level	Euro II

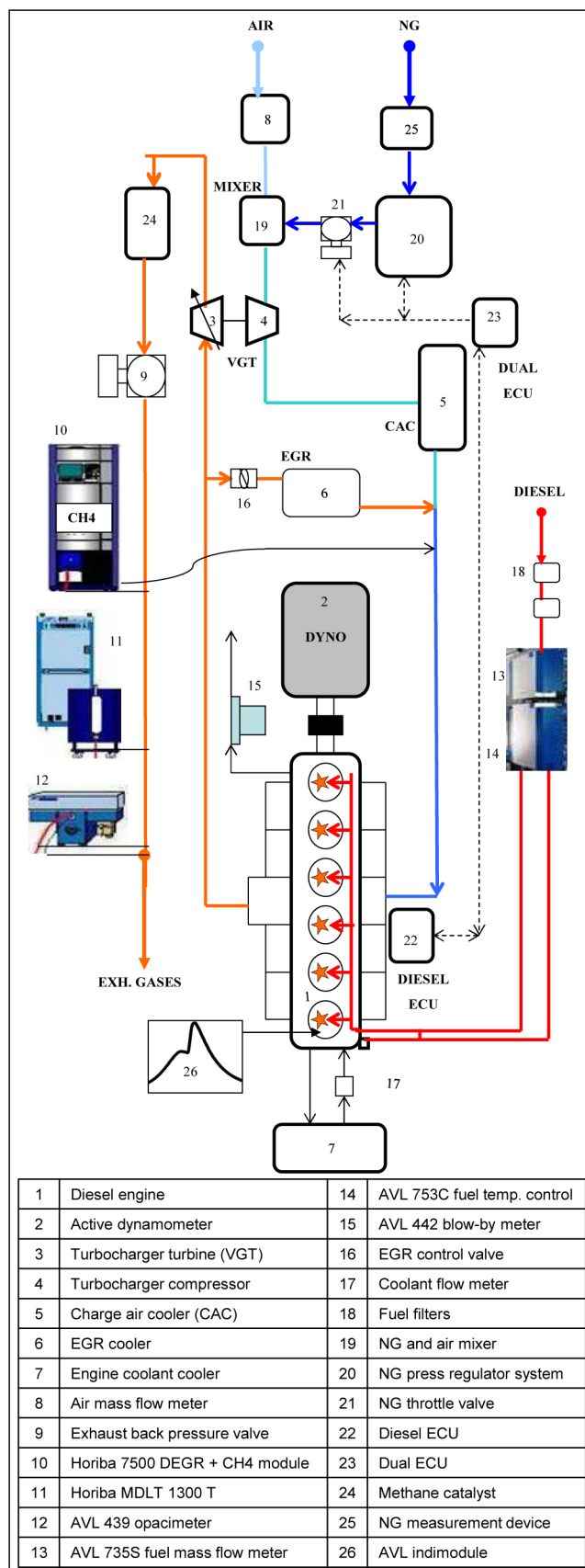


Figure 2. General schema of the engine installation at the test bench (14L dual-fuel engine conversion)

The general data flow acquisition of the installation can be seen in Figure 3. Several engine variables such as engine speed, engine torque and the most relevant temperatures and pressure not only for the engine but also from the test cell (ambient conditions) were acquired. Other equipment such as exhaust gas analyser, PM measurement device, indicating system for in-cylinder pressure and flow mass air and fuel measurement were also used. The information about diesel and gas ECU can also be acquired by PUMA OPEN management system which can receive all this information for online analysis or data post processing.

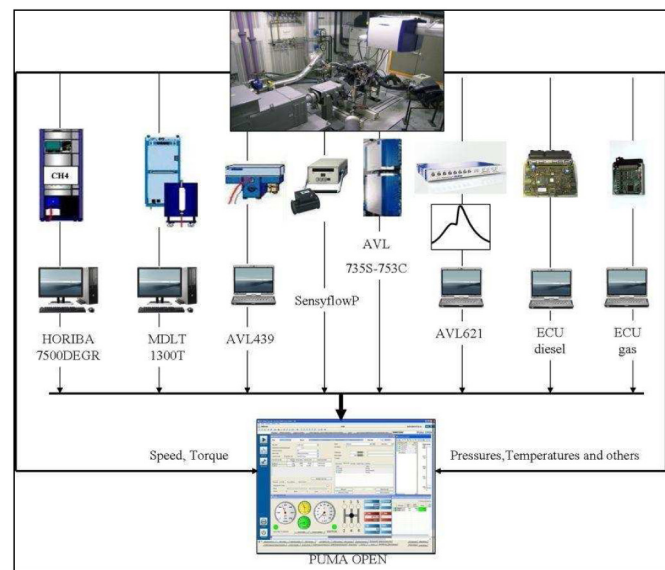


Figure 3. Data acquisition flow schema.

The test cell can work with different types of fuels such as gasoline RON 95/98, diesel, natural gas, LPG and reference fuels. Major characteristics of the three fuels used in this work are shown in the appendix section at the end of this document. The consumption and temperature control of diesel fuel was done by the combination of AVL 735S (Coriolis principle) and AVL 753C equipment. This configuration is a highly precise and continuous fuel consumption measurement and conditioning system with a maximum consumption of 125 kg/h and accuracy of 0.12% for the whole system under real test bed conditions. In the case of gas fuel consumption it was measured by a BRONKHORST gas mass flow meter with a measurement range of 1-150 kg/h.

The intake air consumption was measured by a thermal mass flow meter Sensyflow P-type with a measurement range of 0-4000 kg/h. The air flow meter was suitably installed in the test bench to prevent wakes, eddies, circulating flows, or flow pulsations which could affect the accuracy or repeatability of the measurements. The test cell is also equipped with a humidity and intake air temperature control system integrated in PUMA OPEN.

For the measurement of the typical gaseous emissions [CO₂, CO, NO_x and THC] exhaust gas analyser HORIBA 7500DEGR was used which was extended with a CH₄ module for its measurement during dual-fuel (diesel-NG) mode. A detailed explanation of the measurement principles of all these gases can be found in reference [11]. In the case of 14L dual-fuel (diesel-NG) engine, CO₂ and CO

intake manifold was also measured to calculate EGR rate according to [equation \(1\)](#) where the ambient $[\text{CO}_2]_{\text{atm}}$ concentration remains constant at 0.04%. The measured EGR was an input in the 1D engine model of this engine.

$$\text{EGR} = \frac{[\text{CO}_2]_{\text{intake}} - [\text{CO}_2]_{\text{atm}}}{[\text{CO}_2]_{\text{exhaust}} - [\text{CO}_2]_{\text{atm}}} \quad (1)$$

The PM specific emission during the worldwide emission tests was measured by micro dilution tunnel HORIBA MDLT-1300T. This equipment takes a small portion of total exhaust flow and dilutes it with air to create constant flow rate, and passes the sample through filters to measure particulate matter. After the PM is measured the filters are stored in the weighing room and conditioned for at least one hour and then weighed in a calibrated and accurate scale. The gross weight of the filters is recorded and the tare weight subtracted in the PM sample mass. If background correction is to be applied, the dilution air mass through the filters and the particulate mass shall be recorded and taken into account in the final PM result.

METHODOLOGY

The method used in the development of a HD diesel engine for alternatives fuel use could be explained by the standard “V” method shown in [Figure 4](#). The activities in the detailed design stage (left side of the “V”) are: the preliminary calculations, the identification of restrictions (Pmax, Tmax, noise level, knock, thermal stability, etc.), the computational tools and implemented models (1D, 3D and CAD), the selection of suppliers and components technically and economically feasible. The lower horizontal part of the [Figure 4](#) is the implementation stage where prototype construction activity is performed. Finally, the right side of the “V” is the integration and validation activities of the developed product. The engine test bench calibration, the verification of thermal stability and durability of the engine and components, the performance and emissions tests, the vehicle calibration and driveability are examples of such activities. It is noteworthy that the whole set of activities within the “V” should be performed within the time planned for the project represented in the upper horizontal part of [Figure 4](#).

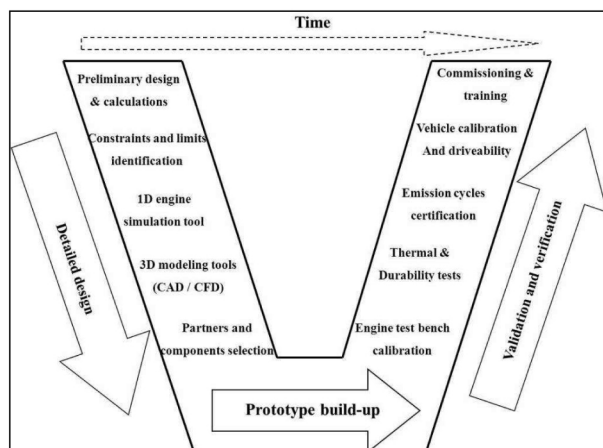


Figure 4. “V” method schema for engine development.

To reach the objective of the present work several development activities were completed (engine simulation, mechanical modifications, electronic implementations, engine test bench calibration) and also specific performance and emission tests were performed in the diesel baseline engines and in the NG dedicated and dual-fuel converted engines as shown in the experimental matrix from [Table 2](#). The performance tests were done based on D80/1269ECE and the emission tests were carried out mainly based on the worldwide emission tests according engine emission level (see [Table 1](#)). The main development activities implemented in the three baseline diesel engines for alternative fuel used will be explained in the next section of the document.

Table 2. Experimental matrix.

Engine Conversion	Development activities	Tests performed	Calibration Target
13L engine (NG fuelled)	1D and 3D simulations	Performance	Performance & behaviour
	Reduction of the compression ratio	ESC	
	Design of an spark plug adapter	ETC	
	Design of a gas injection system		
	Engine test bench calibration		
14L engine (diesel-NG)	1D simulation	Performance	Performance & emissions
	Installation of the gas train components	EPA 2005	
4.25L engine (diesel-LPG)	Installation of the EMS for dual-fuel engine operation	Performance	Performance & emissions
	Engine test bench calibration	R49-02	

The specific gaseous and PM emissions in [g/kWh] were calculated by means of in-house software developed by the authors ([Figure 5](#)) which is a GUI developed in Matlab environment. This tool has been extensively proved with certificated engines and it incorporates worldwide emissions standards libraries not only for HD and off-road vehicles, but also for light-duty and passenger cars. This GUI also performs others functions e.g. emission test validation, carbon flow check, emission deterioration factor calculation and dynamometer regression verification.

The specific exhaust emission calculations in diesel mode and NG converted engine were done according to the corresponding type-approval engine model. In the case of dual-fuel engines conversion, specific exhaust emission calculations were done using the procedure, definitions and recommendations pointed out in reference [12]. The mentioned document states that emission testing of a dual-fuel engine is complicated by the fact that the fuel used by the engine can vary between pure diesel fuel and a combination of mainly gaseous fuel with only a small amount of diesel fuel as an ignition source. The ratio between the fuels used by a dual-fuel engine can also change dynamically depending on the operating condition of the engine. As a result special precautions and restrictions are necessary to enable emission testing of these engines.

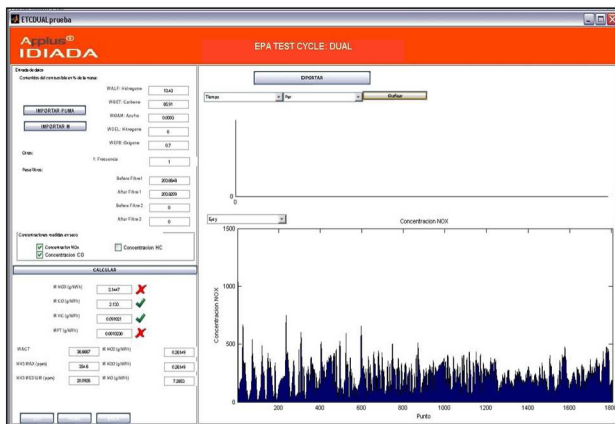


Figure 5. GUI emissions calculation software.

In the case of dual-fuel engine conversion, chemical energy supply and Gas Energy Ratio (GER) were calculated by equations (1) and (2) respectively.

$$E_{\text{Dual}} = m_{\text{Gas}} \cdot \text{LHV}_{\text{Gas}} + m_{\text{D}} \cdot \text{LHV}_{\text{D}} \quad (2)$$

$$\text{GER} = 100 \cdot \left(\frac{m_{\text{Gas}} \cdot \text{LHV}_{\text{Gas}}}{E_{\text{Dual}}} \right) \quad (3)$$

$$\eta_{\text{Dual}} = \frac{W_{\text{Effective}}}{E_{\text{Dual}}} \quad (4)$$

Where: m_{D} and m_{Gas} are the masses of diesel and gas injected [kg], LHV_{D} and LHV_{Gas} are the Lower Heating Value of diesel and gas [kJ/kg], E_{Dual} is chemical energy supply by dual-fuel [kJ], GER is the Gas Energy Ratio [%] and η_{Dual} is the brake fuel conversion efficiency [-].

DEVELOPMENT ACTIVITIES IMPLEMENTED IN THE THREE DIESEL ENGINES FOR ALTERNATIVE FUEL USE

13L Spark-Ignited NG Fuelled Engine

As mentioned before, one of the key activities in the detailed design stage of the development process is the 1D engine simulation which for several decades has been an established method in the development of internal combustion engines [13]. The 1D model was constructed (Figure 6) using a commercial simulation software which takes account of the wave dynamic manifold phenomena via robust solution of the Navier-Stokes equations. The model comprises several sub models such as combustion, heat transfer, and friction losses for instance and it was extensively validated with more than 100 tests from engine test bench measurements (Figure 6b). Once the model is validated in diesel mode some simulations and estimations were done in dedicated NG operation.

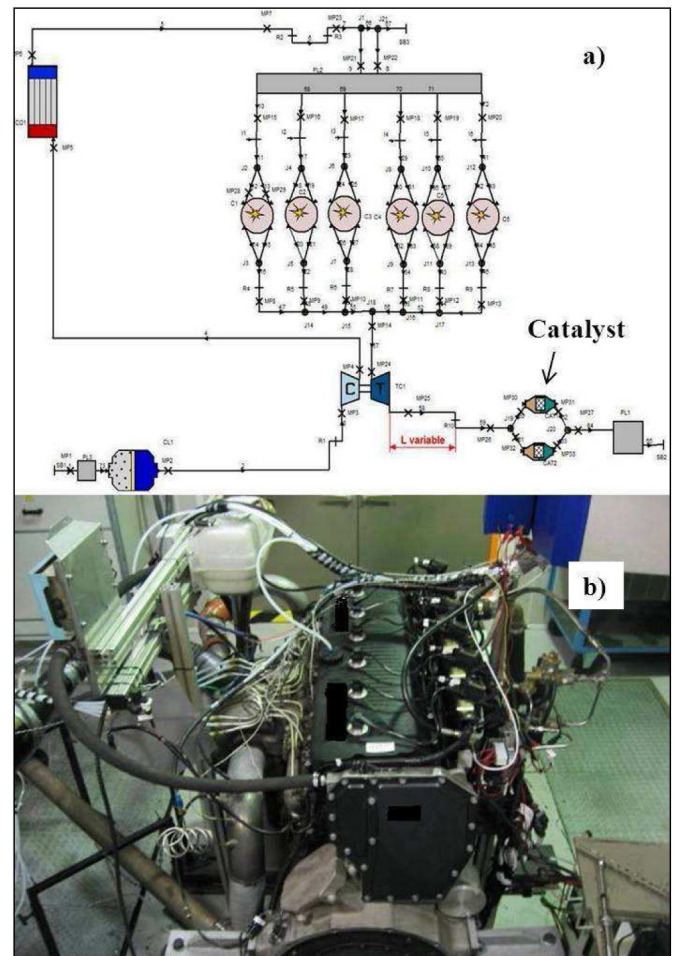


Figure 6. a) 1D engine model (NG final version), b) 13L NG engine installed at the test bench.

Despite the high octane number of NG fuel, the original diesel engine compression ratio of 16.5:1 (see Table 1) was reduced to until admissible levels for NG operation. One thermodynamic approach to estimate a suitable new compression ratio when the lack of the data is presented at the beginning of the development process is to perform several simulations varying the compression ratio with some input data technically assumed or estimated.

The simulation was done at full load condition in NG lean-burn operation at the same rated speed of the diesel engine and with some assumed and expected AFR, spark-timing and dimensionless RoHR chosen from the author's testing database as the main inputs. A value around 10.5:1 for a new compression ratio was selected taking into account several factors (weighted) such as: admissible turbine inlet temperature, less heat transferred, less expected NO_x formation in the combustion chamber (by Zeldovich mechanism), suitable fuel economy and avoiding premature wear and though operation for spark-timing engines [14].

After the new compression ratio was selected, three possibilities to modify the compression ratio without modifying the connecting rod length (This was a constraint in this project) were presented: increasing the volume of the combustion chamber in the cylinder head, increasing the cylinder head gasket thickness, and increasing the volume of the