# PERFORMANCE OPTIMIZATION OF A MULTI-FUEL SPARK IGNITION ENGINE

**José Guilherme Coelho Baêta**
Fiat Isvor do Brasil
Universidade Federal de Minas Gerais – UFMG
E-mail: baeta@isvorfiat.com.br

**Rogério Jorge Amorim**
Universidade Federal de Minas Gerais – UFMG
E-mail: rogeriojamorim@yahoo.com.br

**Ramón Molina Valle**
Universidade Federal de Minas Gerais – UFMG
E-mail: ramon@demec.ufmg.br

**José Eduardo Mautone Barros**
Centro Federal de Educação Tecnológica de Minas Gerais - CEFET-MG
E-mail: emaultone@zaz.com.br

**Remo Dias Bahia de Carvalho**
Fundação Centro Tecnológico de Minas Gerais – CETEC MG
E-mail: remo@cetec.br

***Abstract.*** *The new trends of the automotive market require the application of new technologies to a concept of engines, which allows for the use of different types of fuel. The multi-fuel engines available in the market display only one compression ratio, therefore being subject to optimization, as to obtain maximum efficiency the engine must work with a variable compression ratio. Although technically possible, this procedure is not considered feasible for a low-cost product. This work proposes a system, which allows for each type of fuel to attain peak efficiency through a variance in the engine intake pressure and without changing its compression ratio, a feature that can be added to a low-cost product. The gains obtained with this project will be shown in each stage of the experiment. The methodology developed to adjust and calibrate the EMS for liquids and gaseous fuel is shown. With this work we intend to create an alternative optimizing concept for the 1.3 8v FIRE FLEX MULTI-FUEL engine to allow it to attain peak efficiency when one of the following types of fuel is used: Gasoline E25, E94 ethyl hydrate, any level of mixed Gasoline E25 and Alcohol E94, and CNG.*

**Keywordws:** multi-fuel engines, compression ratio, optimization, efficiency, intake pressure.

**1. Introduction.** The growing demand for multi-fuel cars results from the recurrent changes of prices and availability of fuel in Brazil. This is due to a number of external factors, such as the price of the dollar, the demand for alcohol production instead of sugar, and the introduction in the market of methane gas used as fuel. To respond to this demand the car industry has adapted their engines for the continuous use of various types of fuel. The most common type of fuel used in cars in Brazil is gasoline. However, there has been a number of proposals aimed at finding alternative and economically viable sources of energy, and at facing both the fluctuation of price and the availability of oil. During the oil crisis of the 1970´s Brazil implemented the PROÁLCOOL (Alcohol-based Car Fuel National Project) aimed at finding an alternative fuel for the Otto-cycle engine [1]. In the 1980s Brazil implemented the PLANGÁS (Methane-based Car Fuel National Project) aimed at finding a substitute for the Diesel oil used in cargo and passenger vehicles, and at reducing air pollution in the metropolitan areas [2]. At present, the country has achieved some experience in the development of a concept of alcohol engines with a satisfactory performance. There has been a growing demand for the use of methane due to its low price and low level of exhaust emission. However, the gasoline and alcohol engines transformed into CNG have shown a significant reduction in efficiency as a result of their low compression ratio for CNG. The proposal here is to add the use of methane gas to the dual fuel engine by installing an innovative system. This system uses also a 5th generation methane gas multipoint fuel injection from BRC. The FIRE FLEX engine has only one compression ratio, which allows for its optimization since it should work with a variable compression ratio. However, although technically possible to achieve, the cost of production of this feature is impracticable for this category of engines. This work aims at a careful examination and improvement of the performance of the FIRE FLEX MULTI-FUEL engine so much as to attain its peak efficiency when any of the three types of fuel is used. This optimization is achieved with the installation of an adequate system, which produces a variance in the intake pressure without changing the engine compression ratio.

### 2.1 Objectives

The main objective of this project is to create an optimizing system for the 1.3 8v FIRE FLEX MULTI-FUEL engine in such a way that it attains its peak efficiency when using one of the following types of fuel: Gasoline E25, Alcohol E94, any level of mixed Gasoline E25 and Alcohol E94, and CNG.

### 2.2 Methodology

This work develops an experimental methodology through which it is attained the highest performance level of the 1.3 8v FIRE FLEX engine when using one of the following types of fuel: Gasoline E25, E94 ethyl hydrate, any level of mixed Gasoline E25 and Alcohol E94, and CNG. For the purposes of this work, some partners companies have provided an important contribution supplying a great part of the systems and needed components like: engines, linear lambda sensors, sets of pistons for the compression ratio of 12.5:1 and 15:1. Other systems were bought like a CNG multipoint injection system and the MoTeC electronic control unit that were needed to the whole tuning of the engine control system. The 1.3 8v FIRE FLEX engine in this standard configuration is coupled to the Magneti Marelli IAW 4AF.FF ECU so that the use of Gasoline E25, Alcohol E94 and any percentage of mixture between both can be controlled. Before going to the dynamometer, some preliminary stages were necessary. These stages have already been described previously by "Baêta, (2004)". Therefore after discerning analysis of all available engine control units, the MoTeC M4 was chosen like the best option to be used in this work because it allows setting the system to operate with any kind of internal combustion engine. The entire procedures of programming and adjusting can be done easily with this type of ECU, being a powerful tool for scientific combustion engine researches. A large application of this ECU can be found in racing teams; some categories of car racing use this system due to a large number of options in the program setup, because of the help information quality and due to the great technical support. The main MoTeC M4 ECU specifications could be found at "MoTeC Internet site (2004)". To optimize this Multifuel engine at first it has been necessary to know the engine performance behavior operating with different compression ratios using all types of previously mentioned fuel. Beyond of performance tests, a global volumetric efficiency test was accomplished for each compression ratio following the "FIAT standard 7-A3511, (2004)". This test is very important to check if the volumetric efficiency has been influenced by the used compression ratios, justifying or not the need of changing the diagram valves through the camshaft. The equation (1) was used to calculate the global volumetric efficiency.

$$\eta = \frac{1000 \cdot F \cdot \left(A/F\right)_{STOICHIOMETRIC} \cdot \lambda}{V/2 \cdot \left[1{,}2928\left(P/1{,}0133\right) \cdot \left(273/\left(273{,}t\right)\right)\right] \cdot n \cdot 60} \qquad (1)$$

Where F is the fuel consumption (kg/h), (A/F) is the stoichiometric air-fuel ratio, λ is the lambda value (kg air actual/ kg air stoich), V is the volumetric displacement (dm3), P is the atmospheric pressure (bar), t is the manifold air temperature (°C), 1.2928 is the air density at 0°C and 1.0133 bars and n is the engine speed (rpm)

For global volumetric efficiency tests onto bench dynamometer it is usual to use liquid fuels and the repeatability for this test is well known. In this work the global volumetric efficiency tests have been performed using E94 and CNG. The CNG tests have shown the same repeatability for all used compression ratios. It is important to note that inside-of-the-cylinder CNG occupies a bigger volume compared to liquid fuels due to its bigger specific volume, justifying the least global volumetric efficiency result The equation (2) was used to calculate the fuel volumetric flow into the cylinder.

$$\dot{m}_t * \upsilon_t = \dot{m}_{air} * \upsilon_{air} + \dot{m}_{fuel} * \upsilon_{fuel} \qquad (2)$$

The figure 1 shows the difference in the Global volumetric efficiency tests for E94 and CNG using 12.5:1 compression ratio. The average difference is about 8.72%. These results presented here are basically the same for the others compression ratios with an insignificant variation. So in this case there is no need to modify the valve diagram.

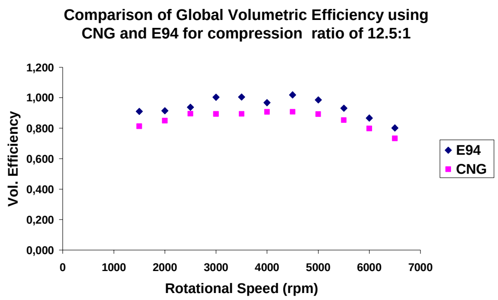


Figure 1 – Comparison of Vol. Eff. for CNG and E94

Figure 2 – The Spark plug transducer, charge amplifier oscillographic data logger

During all tests the cylinder pressure was monitored for each configuration aiming to know the combustion behavior inside the combustion chamber using an AVL pressure transducer GU 13Z-24 incorporated inside the AVL spark plug F5DPRT. The pressure transducer specifications are: sensitivity of 15.60 pC/bar, linearity < ± 0.3%, measure range 200 bars, temperature range 400 °C and natural frequency 130 kHz.

The signal acquired from the transducer has been amplified by the charge amplifier Kistler Type 5011 and sent to the Yokogawa OR1400 oscillographic recorder. The figure 2 shows the Pressure transducer, amplifier and the data logger used. The pressure and rotational speed signals in Volts acquired by Yokogawa data logger for E94 with 12.5:1 compression ratio, 1500 rpm and 100% of load is shown in Fig. 3.

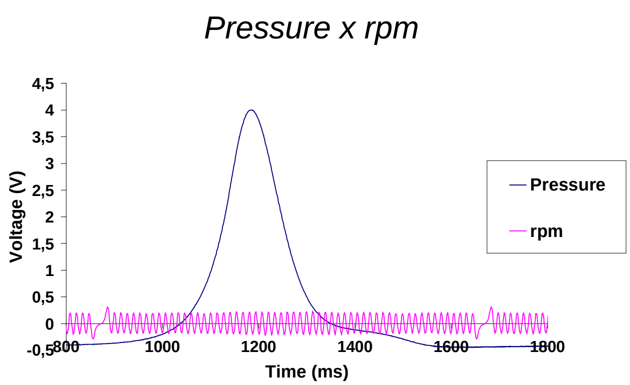


Figure 3 – The pressure and rotational speed in Volts

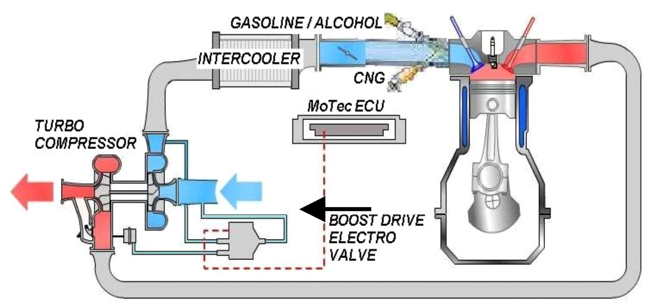


Figure 4 – Scheme of the multi-fuel optimization.

Note that to convert the pressure in Volts to bars the specifications are: transducer sensitivity is 15.60 pC/bar; amplifier factor is 100 bar/V and data logger configuration is 2.56 pC/bar. So the conversion factor is:

$$\frac{2.56\, {}^{pC}\!/_{bar}}{15.6\, {}^{pC}\!/_{bar}} * 100\, {}^{bar}\!/_{V} = 16.4\, {}^{bar}\!/_{V} \tag{3}$$

Lots of data can be extracted from the pressure data by crankshaft angle like the combustion velocity and the unburned fraction mass. Based on all acquired data the optimization process can be done using a turbo compressor where the boost pressure is controlled electronically. The figure 4 shows the Scheme of the multi-fuel optimization. The entire stages of this work can be summarized as follow:

- Aspirated engine with compression ratio of 11:1 ( Original EMS using E25, E94 and 50% of both fuels); REFERENCE PERFORMANCE;
- Aspirated engine with compression ratio of 11:1 ( MoTeC M4 EMS using E25, E94, 50% of both fuels and CNG);
- Aspirated engine with compression ratio of 12,5:1 (MoTeC M4 EMS using E25, E94, 50% of both fuels and CNG);
- Aspirated engine with compression ratio of 15:1 (MoTeC M4 System using E94 and CNG);
- Turbo engine with compression ratio of 11:1 (MoTeC M4 System using E25, E94, 50% of both fuels and CNG).

Before performing the tests with MoTeC ECU it was necessary to obtain the performance curves from the original engine with IAW 4AF.FF ECU. This original configuration was in conformance with the emissions “NBR ISO 6601 standard, (2001)”. The acquisition process of the whole performance curves in this work was in conformance with “NBR ISO 1585 standard, (1996)”. So the engine has been installed onto a bench dynamometer with original system and components and soon afterwards the break in process of the engine, the performance curves have been acquired for E25, E94 and the 50% of mixture between both fuels. Then the MoTeC ECU and its harness were installed substituting the original electronic control system. In addition the CNG multipoint injection system was also installed. The engine was fuelled with gasoline E25 and the ECU tuning process applied to the EMS to find the MBT (Maximum Break Torque) for each site has been done. However, in some points the LKL (Lower Knock Limit) is achieved before the MBT. This phenomenon basically depends on the internal combustion chamber temperature, the fuel octane rating in use, the actual air/fuel ratio and the mixture atomization. In those cases, the ignition advance has been maintained 0.5 degrees before reaching the LKL. The same procedure were used with the alcohol E94, the mixed E25 and E94 (50-50%), and the CNG. At this step is very important to understand the entire process applied to calibrate and adjust the MoTeC EMS. So for liquid fuels, the used sequence was:

1- Set the software main setup according to the engine, sensors and components;
2- Choose the calculation method for load and efficiency;
3- Calibrate all sensor tables;
4- Calibrate the Crank Index Position;
5- Tune one row of the fuel and ignition tables for the 100% of load (in case of running with turbo, tune the pressure boost electro-valve map);
6- Tune the injector dead time compensation table changing the battery voltage aiming to keep the same lambda factor using 100% of load;
7- Adjust and optimize the dwell table changing the battery voltage, based on the coil primary current using 100% of load;
8- Adjust the injection timing angle aiming the best SFC using 100% of load;
9- Go back to the fuel and ignition main tables and tune all points aiming best performance according to the maximum exhaust temperature, desired lambda and MBT or knocking (in case of running with turbo, tune the pressure boost electro-valve map);
10- Insert a lambda closed loop according to the allowed maximum exhaust temperature and keeping the open loop in high loads in agreement with performance request.

For CNG the differences in the tuning process applied were the steps 8 and 10 described as follow:
8- The “injection timing table” calibration was no longer made by calculating the specific fuel consumption point by point for each engine speed, but observing the variations in lambda value. According to this method, the richer the lambda value is, the better is fuel delivery.
10- The entire fuel map was set to reach lambda value equals 1.00 independently the engine speed and load. Since the CNG is super heated and there is no effective exchange heat transfers between the fuel and the combustion chamber, the exhaust temperature keeps invariant for richer mixture. Other facts are: at first, there is no significant increase on torque for richer mixture in a CNG-fuelled engine. Figure 5 displays the graph of Torque X □ for E94 and CNG. Second, due to CNG pressure in the cylinders vary from 200MPa to 10MPa; it causes a change in the CNG specific volume, justifying little deviations in lambda values for the same injection pulse width. Moreover, the up-to-date EMS for gaseous fuel applications measures the fuel pressure and temperature in order to compensate the specific volume variation. Because all these points it has been used a lambda full closed loop map.

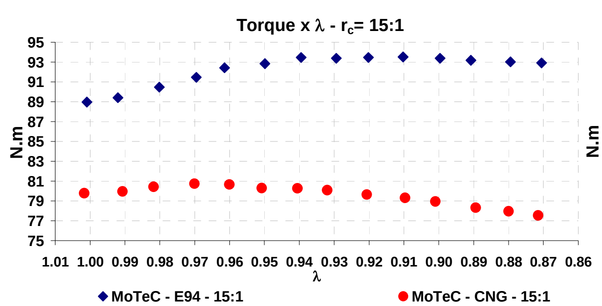


Figure 5 – Torque x l graph for E94 and CNG with compression ratio of 15:1 at WOT and 2500 rpm

In order to measure the methane consumption, after the pressure regulator a flow meter was coupled to the whole system. The consumption calculation was made through an approximation of the CNG with the ideal gas model. CNG composition that was necessary to calculate the relative molar mass, with the percentage of its components, was provided by GASMIG.

All the needed CNG for the project and the set of 6 cylinders were supplied by IGÁS. The sequence described below shows the entire process used to perform each stage of the work:

- Calibration and adjustment of the EMS;
- Data collection concerning the 1.3 8v engine performance, volumetric efficiency and cylinder pressure;
- Replacement of the pistons (if necessary);

The table (1) presents the configurations for each compression ratio and fuels tested during the aspirated engine development.

Table 1 – Description of the tests performed

| | Compression ratio | | |
|---|---|---|---|
| **Fuels** | **11:1** | **12.5:1** | **15:1** |
| **E25** | X | X | |
| **50% MIX** | X | X | |
| **E94** | X | X | X |
| **CNG** | X | X | X |

Figure 6 – Pistons for different compression ratios

The figure 6 shows the pistons for the compression ratio of 15:1 and 12.5:1 respectively that were used as reference to evaluate the proposed solution that was presented in this work. For the development of the proposed turbo stage, the same sequence described for aspirated will be performed. So the replacement of the four pistons with the original pistons was performed, keeping the compression ratio of 11:1. During this stage, the turbo compressor system will be coupled to the engine. Due to the low boost pressure used, there is no reason to use an intercooler in the intake system. At this stage before continuing the tests, a simulation program developed by "Barros, (2003)" will be utilized to reproduce the same data acquired for all aspirated configurations. So beyond of the validation process, the engine data can be acquired from the software for several compression ratios making possible to analyze the data for small intervals. After this, the better-acquired data for each fuel can be used to optimize the performance during the turbo engine adjustments. Afterwards, the ECU is tuned to attain maximum peak efficiency using the maximum turbo boost pressure feasible for each kind of fuel seeking maximum performance. For this new configuration an intercooler installed to the intake system must be necessary.

### 2.3 Results

The first result data that were obtained in this work were the global volumetric efficiency tests. Figure 7 presents the global volumetric efficiency for CNG for all compression ratios used. The obtained results revealed a very good repeatability for gaseous fuel.

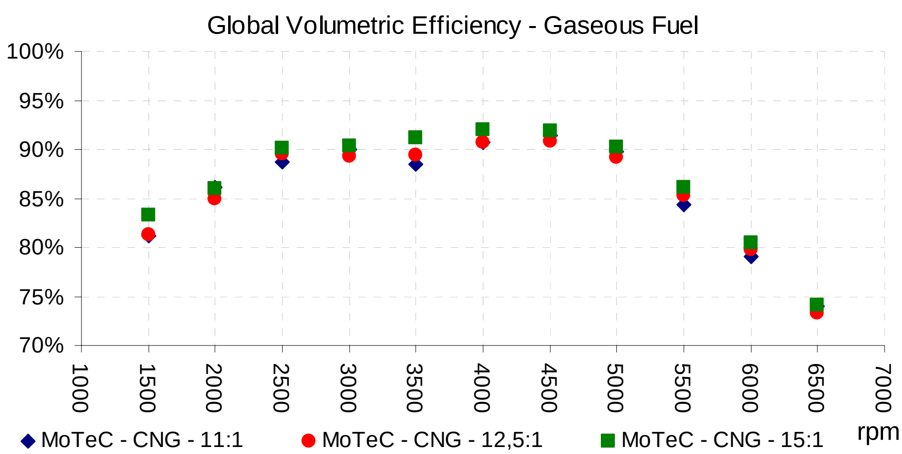


Figure 7 – Global volumetric efficiency for CNG

Figures 8, 9, 10 and 11 present the corrected torque, corrected power, SFC and lambda curves obtained from the tests for E25 for compression ratios of 11:1 and 12.5:1. Therefore the E25 fuelled engine could not work properly with the compression ratio of 15:1 due to high exhaust temperatures and knocking phenomenon at high loads and high speeds.

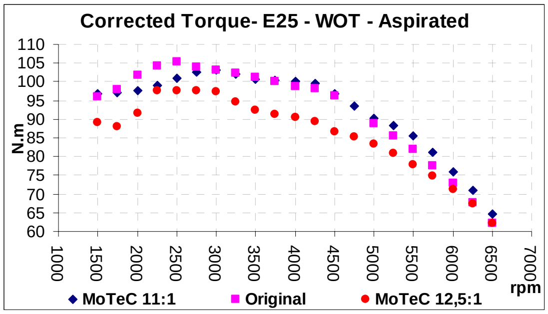


Figure 8 – Corrected torque for E25 fuelled engine

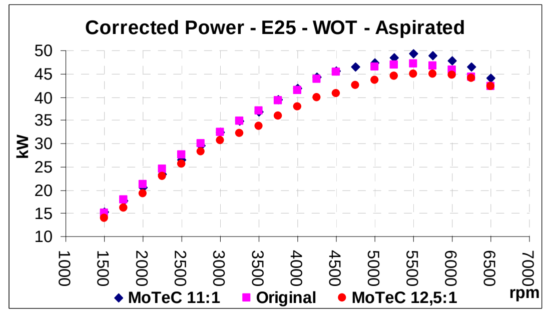


Figure 9 – Corrected power for E25 fuelled engine

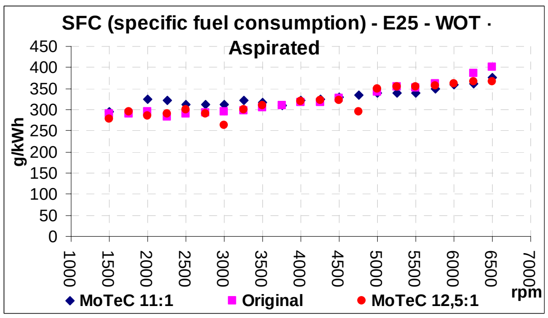


Figure 10 – SFC for E25 fuelled engine

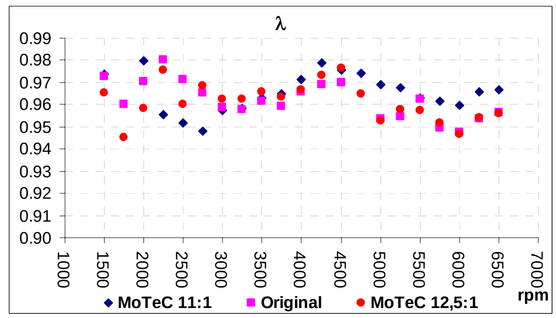


Figure 11 – Lambda factor for E25 fuelled engine

Figures 12, 13, 14 and 15 present the corrected torque, corrected power, SFC and lambda curves obtained from the tests for E94 for compression ratios of 11:1, 12.5:1 and 15:1.

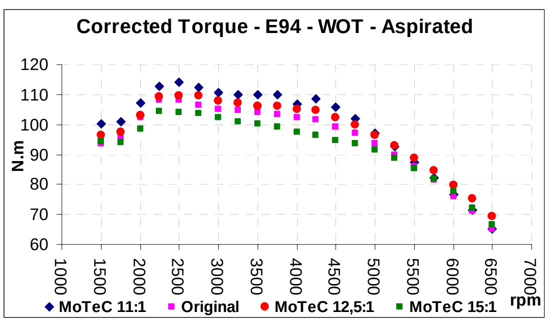


Figure 12 – Corrected torque for E94 fuelled engine

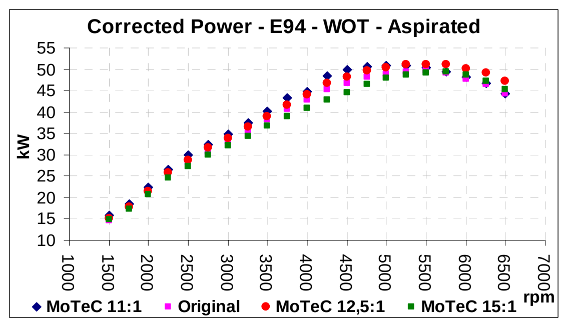


Figure 13 – Corrected power for E94 fuelled engine

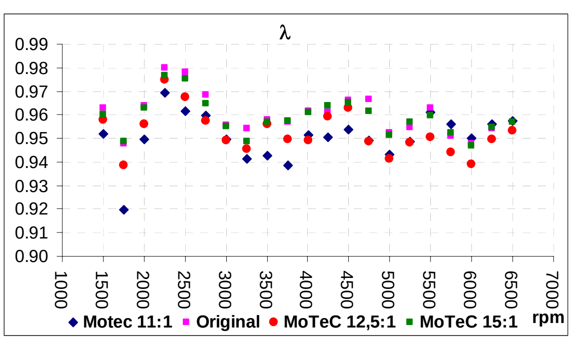


Figure 14 – SFC for E94 fuelled engine

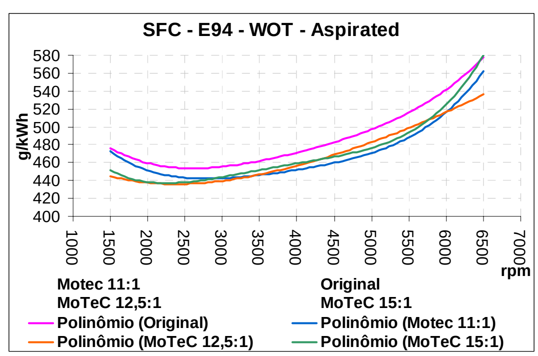


Figure 15 – Lambda factor for E94 fuelled engine

Figures 16, 17, 18 and 19 present the corrected torque, corrected power, SFC and lambda curves obtained from the tests for the mixture of 50% of E94 and 50% of E25 in volume for compression ratios of 11:1 and 12.5:1. Because of the previous explanation the compression ratio of 15:1 has not been used for this configuration.

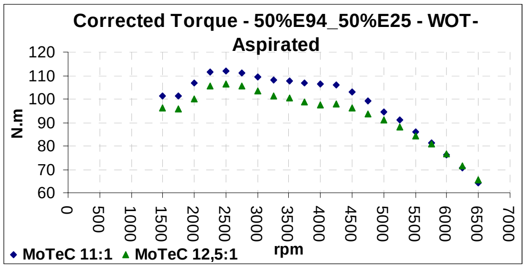


Figure 16 – Corrected torque mixture 50% fuelled engine

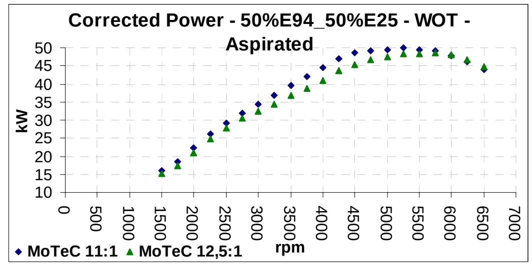


Figure 17 – Corrected Power mixture 50% fuelled engine

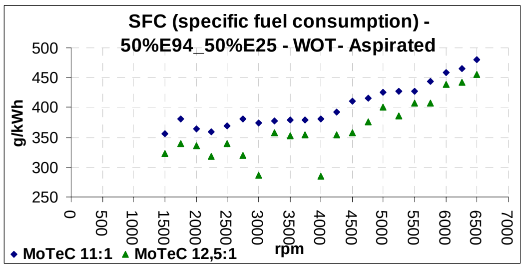


Figure 18 – SFC for mixture 50% fuelled engine

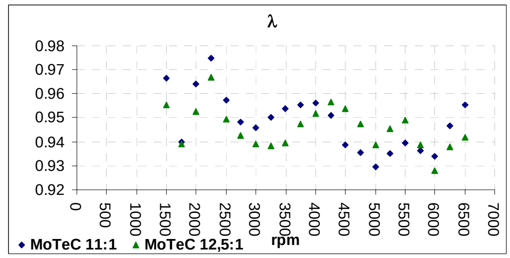


Figure 19 – Lambda factor for mixture 50% fuelled engine

Figures 20, 21, 22 and 23 present the corrected torque, corrected power, SFC and lambda curves obtained from the tests for CNG using compression ratios of 11:1, 12.5:1 and 15:1 all of them compared with E25 performance for 11:1.

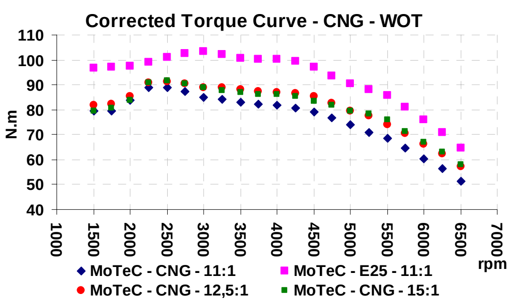


Figure 20 – Corrected torque for CNG fuelled engine

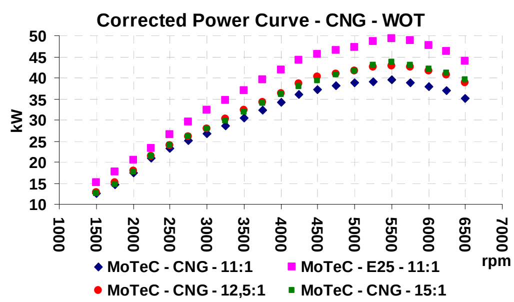


Figure 21 – Corrected Power for CNG fuelled engine

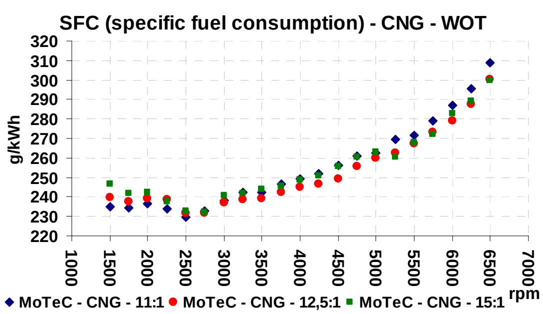


Figure 22 – SFC for CNG fuelled engine

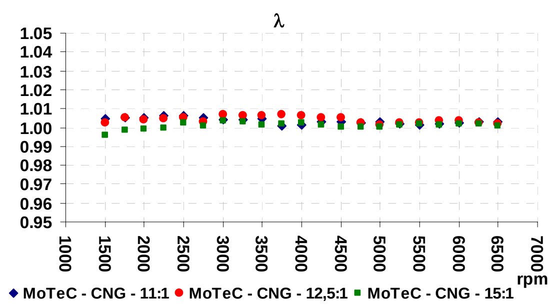


Figure 23 – Lambda factor for CNG fuelled engine

The stage of the optimization through the turbo compression are not completed up to now, thereafter the results for the optimization stage will be revealed in further works.

### 2.4 Conclusions

The first conclusion is that for compression ratio of 15:1 using E94 the torque increases for richer mixtures up to $\lambda$=0.94. This magnitude of torque keeps almost the same up to $\lambda$= 0.90, for richer mixtures the thermodynamic efficiency decreases due to the cooling of the combustion chamber by the fuel. It is explained due to the great quantity of heat absorbed by the fuel to vaporize itself (latent heat of vaporization). For CNG the magnitude of the torque does not increase significantly up to λ = 0.97 and from this value, richer mixtures cause a drastic fall in the torque due to the big volume in the cylinder filled by the gaseous fuel. The CNG generates accurate data to evaluate Global volumetric efficiency tests. The EMS calibrations performed for all configurations tested have demonstrated the higher compression ratio for each fuel, the more probability to find LKL before MBT, analogous, the higher the fuel octane rating or methane number (gaseous) the more probability to find MBT before LKL.

According to discerning analyses the correct power and torque for E25 for the compression ratio of 12.5:1 decreases when compared with 11:1, from 3500 to 5000 rpm. It can be easily explained by the tendency of knocking for this fuel with this compression ratio and the need of the increasing in the advance ignition in this band. This increase was not possible due to high combustion chamber temperature reached. The attempt to keep the same values of lambda has intensified the decrease of the ignition advance and consequently the torque and power. Despite the contribution of the higher compression ratio the SFC does not show significant alteration for the compression ratio of 12.5:1 due to lower ignition advance too.

Analyzing the E94 graphs for compression ratio of 15:1 it shows the lowest performance data compared to other compression ratios, being inappropriate to run E94 in this engine hardware. Regarding the differences found in λ, Consumption, SFC, Corrected power and torque the results display no significant difference between 11:1 and 12.5:1, mainly because of the same SFC. An emission analyzes for both cases based on the effects of changing in the EMS controlled variables like "ignition advance" can decide the better configuration to be used.

According to the graphs for mixture of 50% of E94 and 50% of E25 in volume the results display almost the same performance index, but the compression ratio of 12.5:1 is a little better, mainly because of the lower SFC. So an emission analyzes for both cases can decide the better configuration to be adopted.

The compression ratio of 11:1 is the lowest performance data compared to other compression ratios, being a low compression ratio to extract the maximum performance from this gaseous fuel. Then comparing the results for the compression ratios of 12.5:1 and 15:1, according to the results obtained, there is no significant difference between both. However the expected results should be demonstrated that the compression ratio of 15:1 was better for CNG. The modification in the combustion chamber geometry by the piston geometry used influences badly the combustion behavior, mainly the combustion velocity. It would be explained in details in further works. Another problem was the high temperature reached on the top of the pistons due to the great quantity of mass on this region, obligating the use of lower ignition advance angle causing losses in performance data for the compression ratio of 15:1.

## 3. Acknowledgements

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