

Most Recent Discussion of the Performance and Emissions Produced by a Stratified Torch Ignition Prototype Engine Fuelled by a Blend of Ethanol and Gasoline

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ABSTRACT

In a torch ignition engine system the combustion starts in a prechamber, where the pressure increase pushes the combustion jet flames through calibrated nozzles to be precisely targeted into the main combustion chamber.

The paper presents the layout of the prototype engine and the developed fuel injection system. It continues with a detailed description of the performance of the torch ignition engine running on a gasoline/ethanol blend for different mixture stratification levels as well as engine speeds and loads. Also detailed analyses of specific fuel consumption, thermal and combustion efficiency, specific emissions of CO₂ and the main combustion parameters are carried out. A supplementary decrease in NO_x emissions was obtained by use of Brazilian pure hydrated fuel.

The paper concludes presenting the main results obtained in this work, which show significant increase of the torch ignition engine performance in comparison with the commercial baseline engine.

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INTRODUCTION

Currently, more than 1 billion vehicles operate throughout the world; this estimate is expected to increase to 1.3 billion vehicles in 2030 and will exceed 2 billion vehicles by mid-2050 [1, 2, 3]. These vehicles make approximately 70% of the global carbon monoxide (CO) and 19% of carbon dioxide (CO₂) emissions [2,4]. The increasing number of vehicles around the world increases the demand for fuel and affects the global climate, the stability of eco, -systems, the health of the population as well as the global economy [5, 6, 7, 8, 9]. These issues require the development of new internal combustion engine technologies to reduce fuel consumption and the emission of pollutant gases in particular CO₂, which emitted into the atmosphere, increases the greenhouse effect.

To reduce CO₂ emissions and to prevent an increase in the pollution levels, many countries have introduced government promotions for the use of bio-fuels [1]. In 2003 vehicles that were designed for use of blends of gasoline and ethanol (flex-fuel) technology were introduced in the Brazilian market; these vehicles are capable of operating on gasohol (gasoline blended with 18-27% volume anhydrous ethanol) or 100% hydrous ethanol (4.0-4.9% volume of water) or any blend of these fuels [10]. As a result of governmental

promotions in Brazil, the volume of flex-fuel vehicles was rapidly increased to a market share of more than 88% in 2014 [11, 12]. Based on the information published by the Brazilian National Metrology, Quality and Technology Institute (Instituto Nacional de Metrologia, Qualidade e Tecnologia - INMETRO), the estimated increase in fuel consumption due to the widespread use of flex-fuel technology in Brazil is on average approximately 6 % [11].

Experience gained with mass-produced vehicles by Honda in the late 1970's showed that an implemented torch ignition system with pre-combustion chamber could generate many flame fronts containing high thermal and kinetic energy inside the main combustion chamber. Such a system could therefore initiate a faster and more complete combustion and thereby reduce the risk of knock phenomena, which enables the use of higher volumetric compression ratios. Furthermore, the high ignition potential generated by the system enables the use of lean mixture preparation in the main chamber, which reduces the CO₂ level as well as the specific fuel consumption and NO_x emission. The very complex and costly control systems retarded further development until electronic control possibilities appeared in the 1980's. From the 80's further developments of Torch ignition systems were made [12, 13].

Therefore, it is believed that the torch ignition system can be employed to improve the engine's thermal efficiency, specific fuel consumption and emission levels.

Objective of the Study

The purpose of the study presented in the paper is to characterize and analyze the fields of specific fuel consumption, thermal efficiency and CO₂ emissions of the performance of a stratified charge Spark Ignited (SI) engine equipped with a Torch Ignition (TI) system. From the analysis of the combustion parameters will be understood the reasons that lead to performance gains or losses when a TI-system is applied.

Baseline Engine Layout

A commercial 4-cylinder, 1.6-liter, 16-valve SI-engine was used as a basis for the prototype engine layout. The mixture preparation system implemented on the engine was a direct injection high-pressure system.

The engine main data are presented in [table 1](#).

Displacement Volume	1596 cm ³
Number of Cylinders	4
Combustion chamber arrangement	Pent roof-shaped
Top piston geometry	Dish top piston crown
Camshaft	2 – DOHC without VVT
Connecting-rod length	137 mm
Bore x stroke	79.0 x 81.4 mm
Geometric compression ratio	11.0:1
Intake valve opening (B-TDC)	10° (1 mm ref.)
Intake valve closure (A-BDC)	20° (1 mm ref.)
Intake valve diameter x lift	30 x 7.95 mm
Exhaust valve closure (B-BDC)	30° (1 mm ref.)
Exhaust valve closure (A-TDC)	0° (1 mm ref.)
Exhaust valve diameter x lift	24 x 7.00 mm
Firing order	1-3-4-2

Details of TI-Pre-Chamber Layout

The entire cylinder head of the baseline engine was replaced by a specially designed new head, which enabled the implementation of the TI-system specific enhancements. [Figure 1](#) shows the layout of the new cylinder head. [Figure 2](#) shows the details of the pre-chamber adaptor. In [figure 1](#) one can identify the locations of the spark plug (2) and the direct fuel injector (3).

The pre-chamber (1) in [figure 1](#) is the location in which the primary fuel-air mixture is performed and subsequently ignited. By the pressure rise in the pre-chamber the burning charge is expelled through the transfer bore (4) in [figure 2](#) towards the main combustion chamber. At the end of the transfer bore a multi-hole diffuser was inserted, which spreads the burning gases to provide a large surface arrival in the main chamber. The pre-chamber main data is presented in [table 2](#).

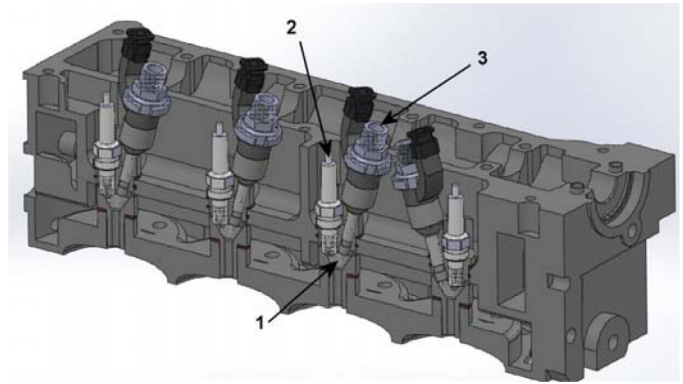


Figure 1. New cylinder head layout

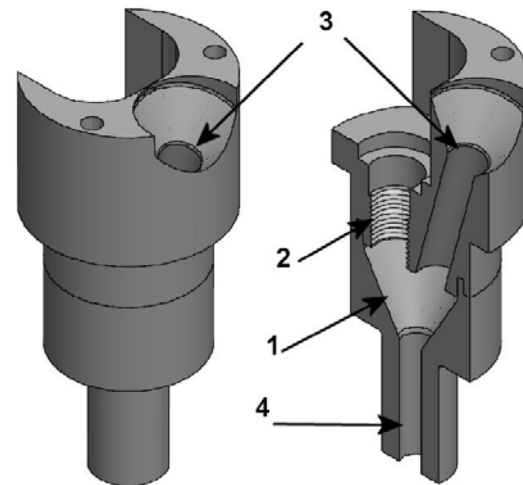


Figure 2. Pre-chamber adaptor

Table 2. Pre-chamber main data.

Volume	2.97 cc
Geometry	Conical
Diameter of transfer bore	6.0 mm
Number of holes in diffuser	1 to 5

The performance of a TI-system is conditioned by the volume of the pre-chamber and its internal geometry, the location of the spark plug and the fuel injector layout. Furthermore, the system performance depends on the dimensions, the number and configuration of the diffuser plate holes, which are located on the bottom side of the transfer tube (4) in [figure 2](#). In the present study the number of diffuser plate holes was varied from 1 to 5. [Figure 3](#) shows the different manufactured diffuser plate layouts used to determine experimentally the optimal mixture and flame distribution in the main chamber.

Numerical Simulation Tool

One of the main objectives of this study was to identify fundamental system parameters to optimize the physical phenomenon of mixture preparation in the combustion chamber. To gather the necessary

understanding it was mandatory to apply a series of complementary analysis tools in parallel with the fired engine test rig and vehicle experiments. The main supplementary tool for the present study was the numerical 3D-simulation tool NCF-3D. To determine numerically the optimal mixture and flame distribution in the main chamber is used a numerical simulation 3-D CFD tool. This thermodynamic and chemical tool can be applied to the entire engine structure contained within the boundaries of the inlet throttle body and the exhaust pipe. Such a model is called a Virtual Engine Model (VEM). A few comments will be made about the usage and the reliability of the used VEM approach. Whenever a totally predictive research work is made, such as the analysis of new combustion chamber geometry, it is very important that the user/researcher knows the validity and limits of all the assumptions related to the models and sub-models involved in the computations. As this entire knowledge is only available in very few commercial computation codes, the VEM software basis used for the present development is a non-commercial company code. This code, the NCM-3D, has been continuously co-developed and refined for more than 25 years by the authors in collaboration with Tor Vergata University in Rome (Italy). During time the code was successfully tested and used for system development on more than 30 different real engines, representing a very wide range in displacement and power output (from 5.9-liter 4-cylinder aircraft, 10-cylinder F1, middle to small displacement SI and CI and down to 50 c.c. two-stroke engines [14, 15, 16, 17]). The code was originally developed as a high performance simulation tool for optimization of the design of GDI-based combustion chamber layouts and mixture preparation components such as fuel injectors for a lean stratified combustion approach. The original version, in which the resolution of the governing 3D Navier-Stokes (N-S) equations, is based on the well-known KIVA-3V approach. The KIVA approach was originally developed by the Los Alamos Laboratory [18, 19]. The approach employs a finite volume approximation in a Cartesian or cylindrical reference system in which a multi-block structured grid is generated. The NCM-3D uses a modified κ - ϵ turbulence model, accounting for compressibility effects, during the arbitrary Lagrangean-Eulerian (ALE) integration of the averaged N-S equations. The code was successively modified by including high-level flexible dynamic grid control (one single grid is used for the entire engine cycle) and in the later rewritten versions by adding compatibility with Pentium high-speed processors and complete multi-node parallel processing capability. The specific physical models implemented in the code enable full 3D computation of both gas and fuel spray dynamics, including atomization, vaporization of multi-components fuel, internal droplet ballistics and fuel/wall interaction. In particular, the wall/liquid interaction models take into account the complete phenomenon of wall-film deposit, droplet propagation and bouncing as well as film undercutting phenomena and local surface vaporization. To avoid any type of parameter-tuned model approach a lot of care was taken with each implemented model to make its layout depend on physical relationships only. For the TI-DI approach a primary numerical simulation analysis was made to adapt a high pressure multi-hole injector for which the spray pattern was made compatible with a Spray Guided layout.



Figure 3. Different diffuser layouts

For all engines tested a stationary engine test rig was used. Initially a commercial 4-cylinder, 1.6-liter, 16-valve SI-engine was tested as a baseline, and a first prototype engine fitted with the modified cylinder head shown in figure 1. To achieve an adapted mixture preparation aspect in the pre-chamber a specially developed Magneti Marelli high-pressure multi-hole liquid fuel injector was used. The injector characteristics were chosen to produce a very low dynamic flow and spray pattern with a small cone angle. The nominal static flow of the injector was set to 1.7 g/s $\pm$ 4% at a rail-pressure of 10 MPa. In order to minimize wall wetting inside the pre-chamber the injector nozzle was fitted with a single-hole atomizer plate, which produces a very small spray cone angle (7°). Figure 4 shows the high-pressure electromagnetic direct fuel injector (a) and the dynamic flow evolution versus injection duration (b). Furthermore, the figure shows a frontal and lateral view (c) of the narrow spray corresponding to a time frame of 0.8 ms after the start of injection.

PRELIMINARY ENGINE SCREENING

In a preliminary phase the differences in performances between the stratified charge TI-engine and a baseline engine equipped with a conventional ignition system both running at homogeneous, stoichiometric mixture conditions were analyzed. The TI-engine was fitted with single-hole adaptor pre-camera.

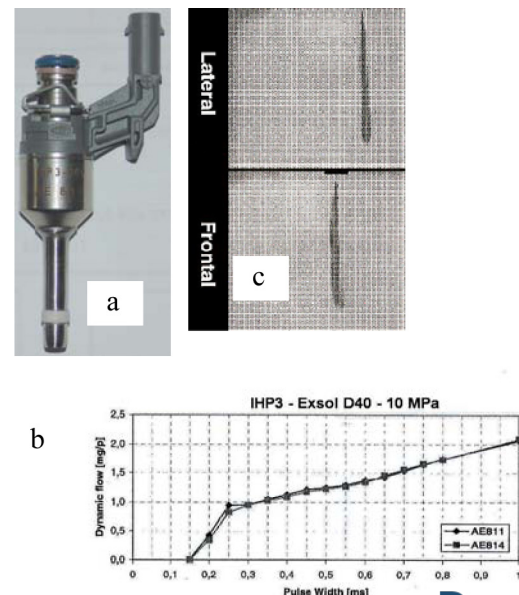


Figure 4. Fuel injector details



Nine operational test points were selected. In these load points on the engine test rig the main performance parameters were measured. To quantify the engine performance the torque, the engine speed, the inlet air mass flow rate, the fuel mass flow rate and the lambda factor were measured. Moreover, the gas temperatures in the intake and the exhaust manifolds as well as the oil and cooling water temperature levels were monitored. Also, the instantaneous in-cylinder pressure was monitored by a fast-acting pressure sensor communicating with the combustion main chamber by a bore drilled in the cylinder head. Figure 5 shows the load point matrix used.

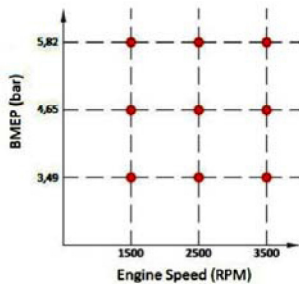


Figure 5. Load point matrix for preliminary tests

The above-mentioned main combustion parameters were compared and analyzed with the base line engine running (fuelled by E25) under homogeneous, stoichiometric mixture conditions and the TI-prototype engine (also fuelled by E25) in stratified charge conditions. The engines were coupled to a Foucault current Horiba W230 dynamometer. The DOE was related to the operational points displayed in figure 5. The Stars Automation software platform was used for control and data acquisition. To quantify the engine performance were measured the torque (Nm), the engine speed (rpm), the inlet mass flow (kg/h), the fuel flow mass rate (kg/h), the lambda factor and the exhaust and intake air temperatures as well as the oil and cooling liquid temperatures (°C). The in-cylinder instantaneous pressure was monitored by a fast acting AVL G13P sensor drilled into combustion chamber wall synchronized with a crankshaft AVL 364 encoder (resolution 0.5° crank angle). A Motec M800 EMS based on a 32-bit microprocessor controls data acquisition and allows the real time adjustment of the injection timing and duration of all tested injectors (PFI and DI). Also the injection pressure and the ignition timing are controlled by the Motec software. 200 engine cycles were acquired at each test point. The average results of acquisitions were obtained over a period of 10 s at a 10 Hz rate. Figures 6 to 8 present the results for the specific fuel consumption versus engine speed for both the baseline engine and the TI-engine obtained at three different BMEP conditions (see figure 5). In the figures, results from the baseline engine are marked by the letters “BL”, whereas the TI-prototype is identified by the letter “S” followed by the lambda factor of the test point.

Another important parameter, which shows directly the combustion quality obtained is the co-variance of the IMEP. Figure 9 shows this co-variance for both the baseline engine and the TI-engine. From the figure an important increase in the co-variance for lambda values greater than 1 for the baseline engine can be detected. As can be seen in the figure, the TI-engine behaves differently. In the TI-engine the co-variance remains fairly constant up to a lambda value of 1.20, and only a small increase can be observed when the lambda value is

increased to 1.3. At this point, it should be noted that co-variance data greater than 3% is unacceptable for production engines. The figure 9 shows the results in the center point of the test matrix at 2500 rpm.

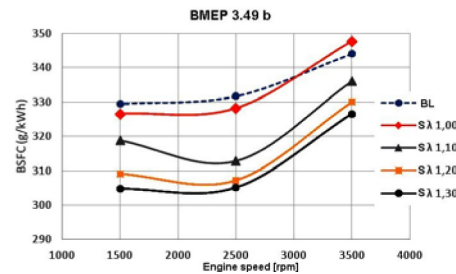


Figure 6. Specific fuel consumption versus engine Speed for 3.49 b BMEP.

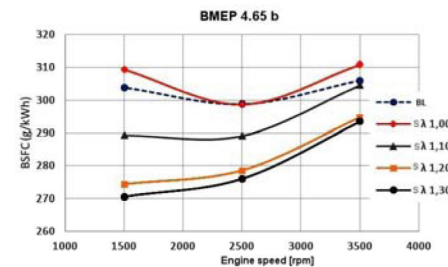


Figure 7. Specific fuel consumption versus engine Speed for 4.65 b BMEP.

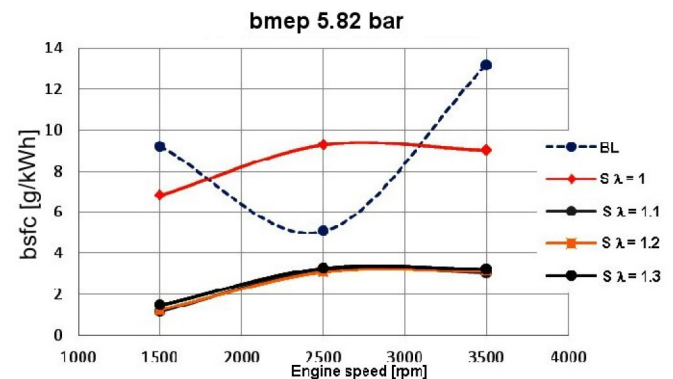


Figure 8. Specific fuel consumption versus engine Speed for 5.82 b BMEP.

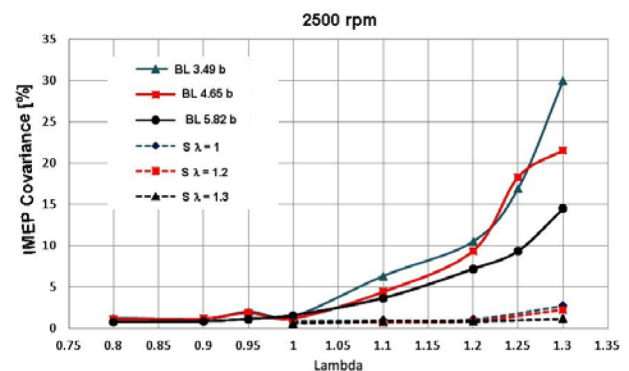


Figure 9. IMEP Covariance versus lambda

The excellent behavior of the TI-engine at increased lambda values can be explained in the following way. By the TI-system a high ignition energy is created in the pre-chamber. The system creates a stratified charge in the main combustion chamber and induces a high turbulence level by the flame propagation in the main chamber