VIRTUAL ENGINE MODELING (VEM) – A NEW MANDATORY DEVELOPMENT AND OPTIMIZATION TOOL FOR MASS PRODUCED THERMAL ENGINES AND THEIR MIXTURE PREPARATION SUB-SYSTEMS.

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Abstract. *For a long time researchers have been occupied by the development of numerical simulation tools to assist amongst others the automotive industry engineers to maximize engine efficiency and to decrease the engine specific fuel consumption. However, the requirements of the industry are that such simulations tools are fast, reliable (algorithms must as much as possible depend only on physics, thermodynamics and chemistry) and capable to connect to computer aided design (CAD) tools. The paper presents a complete example of a successful collaboration between the research development team and a major Brazilian mixture preparation systems manufacturer with the objective to optimize within a short time frame (3 months) the fuel injector spray characteristics for integral flex-fuel applications. The paper demonstrates the necessity to introduce a series of coupled new high-performance analysis tools for further improvement of the mixture preparation system and in particular the fuel injector performance. The paper continues with a discussion of the basic structure of the interactive Virtual Engine Model approach applied to fuel injector atomizer optimization. Test results obtained by application of the new tools on a series produced flex engine are presented. The impact of the improved spray formation capability of the optimized fuel injector atomizers is explained and experimental vehicle FTP-cycle data are reported and discussed.*

Keywords*: Virtual Engine Modeling, Physical based modeling, Industrial Application, Decreased development costs*

1. INTRODUCTION

For more than two decades researchers have worked to develop appropriate numerical simulation tools to assist amongst others the automotive industry development engineers in their task to maximize engine efficiency and to decrease the engine specific fuel consumption. However, to satisfy the requirements of the industry it is mandatory that such simulations tools are fast (less than 10 CPU-hours for a complete engine cycle), reliable (algorithms must as much as possible depend only on physics, thermodynamics and chemistry and not on experimentally tuned parameter constants) and capable to connect to Computer Aided Design (CAD) tools.

To better understand the history of development for such a simulation tool the present paper will use a non-commercial simulation package, the NCM-3-D, developed for a period of more than 20 years by Numidis SARL (Colombes - France) in collaboration with the University Tor Vergata/Numidia (Rome – Italy). The software package NCM-3-D was originally developed for specific analysis and development of Gasoline Direct Injection (GDI) and diesel DI systems. The notion of specificity to an application area is very important regarding the high-level complexity of embedded models and a feature, which often is less well adapted in most of commercial all-purpose software packages. In the present paper an example is given to illustrate the extremely important notion of specific adaptation of all sub-models based on physical relationships to an application area. This specific approach requires of course a very interrelated teamwork between the software developing/adapting management team and the potential end user of the produced results. The example chosen is in the automotive application field and presents a complete example of a successful collaboration between the software development team and a major Brazilian mixture preparation systems manufacturer (Magneti Marelli Sistemas Automotivos) with the objective to optimize within a short time frame (3 months) the fuel injector spray characteristics for integral flex-fuel application. The specific application frame of the automotive oriented simulation software is called the Virtual Engine Modeling (VEM).

2. COMMON CORE FEATURES FOR 3-D NON-STEADY SMULATION APPROACHES

During the early days of the simulation tool development it was rapidly learned that if the precision and CPU-time requirements of in particular the automotive industry were to be met, the all-purpose aspect of the software package should be abandoned and therefore the VEM approach, being the initial development, was dedicated to the Internal Combustion (IC) engine physics. Furthermore it was rapidly learned that to fulfill these requirements the SW-package must absolutely be compatible with parallel processing techniques operating in a cluster configuration. At a later stage these mandatory features were confirmed in a new development of another application area, the Wildfire approach.

As a common core is used a finite volume approximation of the governing 3-D Navier-Stokes (N-S) equations in a Cartesian or cylindrical reference system in which a multi-block grid structure is generated. This approach is based on

the well-known KIVA-3V code originally developed by the Los Alamos Laboratory [2, 3]. Unstructured grid approach, which is a feature that facilitates automatic grid generation, has not been retained neither for the VEM nor for the Wildfire applications, as a long experience in the automotive field (more than 35 different commercial engines in a displacement range from 50 cc to 5.2 l as well as Formula 1 engines) shows the absolute necessity to respect even the smallest geometrical particularities of a given combustion chamber layout. This imposes a high proportion of manual grid generation as most automatic grid generators implicitly smoothen grid properties in complicated border areas.

The original version was modified by including high-level flexible dynamic grid control and by adding compatibility with modern Pentium high-speed processors and complete multi-node parallel processing capability. In the common core a modified κ-ε turbulence model, accounting for compressibility effects, is used during the arbitrary Lagrangean Eulerian (ALE) integration of the averaged N-S equations [15].

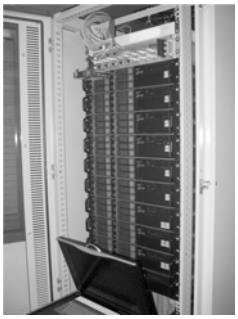


Figure 1 – Example of 8-nodes cluster

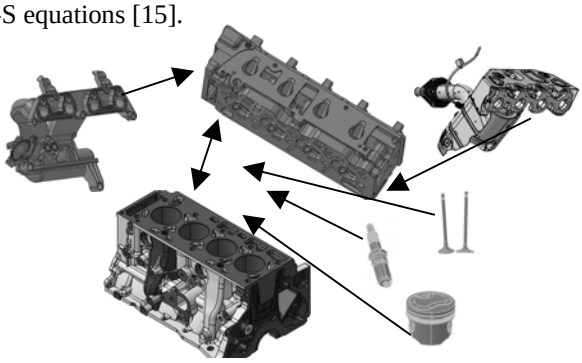


Figure 2 – Example of complete CAD model

Figure 1 shows an example of an 8-nodes parallel processing cluster, which is currently used in most VEM applications. Figure 2 shows an example of a CAD-model of a small commercial flex fuel engine, which is used to create the complete computation mesh for an equivalent VEM.

3. AUTOMOTIVE SPECIFIC MODEL IMPLEMENTATION

The key functions related to the automotive application field are the complex simulation of mixture preparation and combustion of liquid hydrocarbons. For this purpose numerous specific physical models and sub-models are developed, tested and introduced into the code. The implemented models enables full 3-D computation of both gas and fuel spray dynamics, including atomization, vaporization of multi-components fuels, internal droplet ballistics and fuel/wall interaction, as well as the complete combustion phase including auto-ignition and computation of post-combustion products (CO, CO_2, NO_X, particulates ….). The implemented combustion model for SI-combustion is of the characteristic time scale type whereas the auto-ignition model for both gasoline and diesel applications is based on reduced chemistry reactions. A few comments on these choices are necessary. For quite a period two different approaches in SI combustion approaches were tested in parallel, the Continuous Flame Model proposed by Colin et al. (CFM) [1, 4] and the Characteristic Time Scale model (CTS) suggested by Reitz et al. [26, 27].

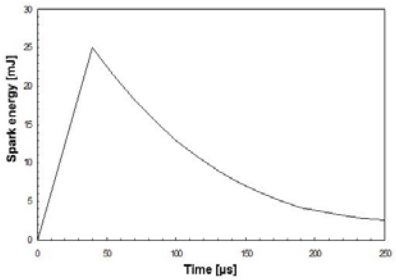


Figure 3 – Example of energy release in plug gap

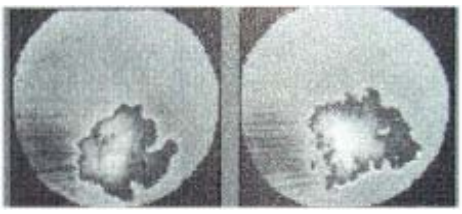


Figure 4 – Example wrinkled SI-flamelet propagation in transparent engine

For the SI-engine approach is imposed a premixed turbulent flame behavior, which is initiated as a laminar growing structure taking place immediately after the energy discharge in the spark plug gap that produces a rate of change of temperature in the plasma following Fourier's law. The energy release in the spark plug gap is for both models governed by a the source-term S, the parameters E_{ele} and E_{chem} are the electrical and chemical energy discharged in the plasma present in the spark gap.

$$S = \frac{E_{ele} + E_{chem}}{4\pi\rho c_p d_i^3 / 3}$$

The elements ρ, cp, and α are respectively the density, the specific heat and the thermal diffusion of the plasma. As can be deduced from this expression the initial kernel growth is conditioned by the energy discharged in the gap (peak and rate) and by the initial gas temperature in the chamber prior to ignition. The generic expression for the discharged energy, which varies from one engine to another, has the form:

$$E_{ele} = f_{rise}(coil, gap), f_{decay}(k_1 * \exp^{-K_2 t})$$

Figure 3 shows an example of the shape of the generic energy release function. As can be expected from the physical shape of the flame border in a real engine the assumption of continuity may have its limits. Figure 4 shows two consecutive recordings, separated by 1.5° Crank Angle (CA), of the very wrinkled flame border of a premixed turbulent flame, which is quite distant from a continuous behavior. This is taken into account by the CTS-model. In this combustion model the species considered in the computations are: C_XH_Y, O_2, H_20, CO and H_2. The main strategy of this model is to consider a spherical laminar kernel growth until a dimension equal to the characteristic Komolgorov turbulence length scale $\boldsymbol{l_K}$ is reached. Hereafter the characteristic time for equilibrium of conversion from one chemical species to another $\boldsymbol{\tau_C}$ is taken to be related to the turbulent conversion time $\boldsymbol{\tau_T}$ only. Parallel usage of the two models showed no significant difference in combustion pressure generation and heat release when near to stoichiometric mixtures are burned, but at very reach or very lean mixture conditions (GDI lean stratified) the results a very different and the CTS-model approach matches very well experimental results. Therefore this approach is the model retained for the VEM SI-combustion.

An optimized auto-ignition approach is still a matter of much discussion. – Some models introduce up to [27] approximately 2446 reactions between 544 species. However, such detailed chemical models increases the computational load within a VEM structure to a level that is far beyond the capability of even modern Pentium high-speed processors linked in a cluster network. This is the reason for the interest in simplification of combustion kinetics. As good results were obtained in both diesel and knock-research applications by the use of an adapted shell-model [20, 21], this model was used for the EURO95 gasoline CAI-ignition approach on a 1.6 l GDI-engine [20]. This model uses a highly simplified multi-step reaction mechanism to predict the spontaneous auto-ignition of hydrocarbon fuels. In the model the auto-ignition chemistry is reduced to an 8 step branching reaction scheme divided in 4 processes, initiation (step 1), propagation (step 2), branching (formation and degeneration – step 3 to step 6) and termination (step 7 and 8). A detailed reaction scheme can be found in [20]. By real engine test-rig experiments it was clearly demonstrated that for a well-defined European high-octane fuel (E95 or E98 close to an iso-octane structure) the predictions of the applied VEM for start of combustion (ignition dwell) and combustion duration were reliable. But for less controlled low-octane fuels composed of a high number of long-chained molecules (typical for the US-gasoline quality) low-temperature oxidation setoff becomes far more important and the shell model is insufficient to precisely predict the auto-ignition dwell and duration. The choice was therefore made to implement a model of higher complexity, but still manageable with respect to required computation load. The choice was a chemical kinetic model composed of 55 reactions including 32 species as first proposed by Keck et al. [28]. As combustion of long-chained heavy HC-molecules is a two-stage process involving a low-temperature cycle followed by a high-temperature event it was necessary to choose a model formulation with a complexity level, which covers both mechanisms but remains acceptable with respect to CPU-usage. The general oxidation scheme contains an initial low/intermediate temperature cycle in which large molecules break down into smaller hydrocarbon and radical molecules. The high-temperature cycle involves mainly the smaller molecules and leads to the exponentially increasing explosive burning. By this two-stage approach both the ignition delay and the specific burning rate for a given fuel can be expressed by the proper choice of fuel-dependant parameters in the chemical reaction constants. The ignition delay is defined as the time from start of compression to a pressure value of 20% of the combustion peak pressure. The burning rate is defined as the ratio between a pressure rise from 20 % to 80% of the peak pressure and the time necessary to accomplish this rise. All reaction constants are of the Arrhenius type:

$$K_i = A_i T^n \exp\left(-E_i / RT\right)$$

3.1. Coupling between simulation structures

An important feature of 3-D simulation package layout is direct links to 1-D and 0-D codes. This is the case for the presented VEM. Here are embedded in the NCM-3D frame both a 1-D and a 0-D link and they can directly communicate with the 3D structure. This is extremely useful when no physical engine is available to produce up- and downstream boundary conditions for the 3-D computational domain (1-D) or engine inertia data must be known (0-D). It is possible to generate the boundary conditions in the different load points by the 1-D code, if a classic one-dimensional computation scheme is established (tube lengths, volumes and concentrated head losses). Figure 5 illustrates the general layout of the VEM in 3-D with its 1-D and 0-D links.

The 0-D link is an engine torque equilibrium model, which computes the instantaneous forces acting on the piston at each time-step. By the knowledge of the net force acting on the piston it is possible to compute the instantaneous piston

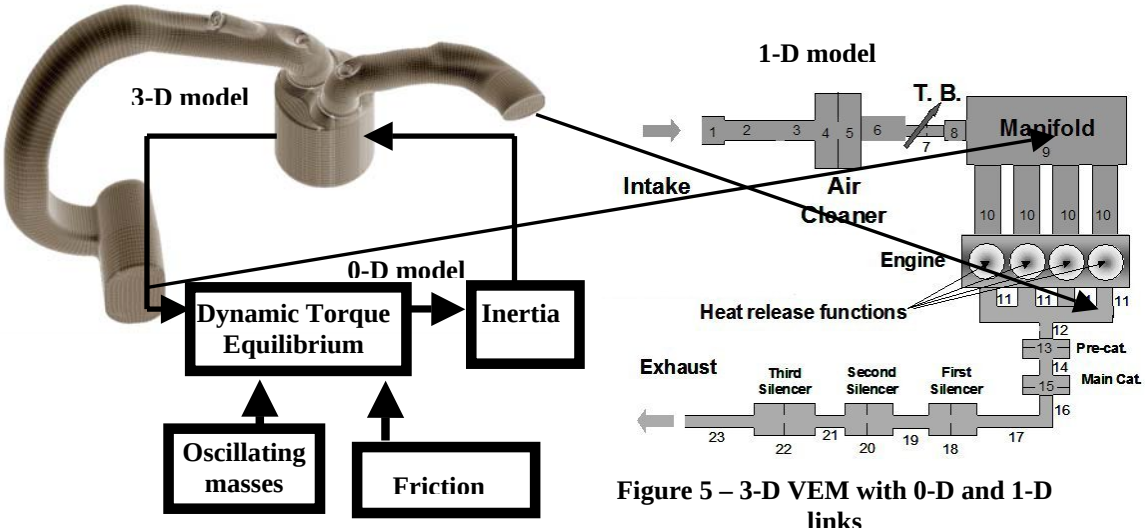


Figure 5 – 3-D VEM with 0-D and 1-D links

acceleration and velocity. The net force on the piston and hereby its time-dependant position is computed from the difference between the gas pressure acting on the piston top and the counter forces imposed by the internal engine losses (friction, inertia, oscillating masses, auxiliaries and loaded as well as unloaded bearings).

It is equally important to understand that the physical/mathematical layout of each of the embedded models whether it is for combustion, spray behavior, vaporization or fuel/wall interaction must be experimentally validated before they are used within the VEM-frame. For validation of models related to droplet dynamics, vaporization and interaction with solid walls for both Port Fuel (PFI) and Gasoline Direct (GDI) injectors are used high-level optical visualization and measurement techniques. As common base for such investigations are used specific backpressure flow vessels in which can be obtained dynamic and steady state visualizations and measurements of different spray patterns for the different spray nozzle atomizers at constant test conditions. Typically such a vessel, in which the injector under study is mounted, should allow the introduction of pre-heated ambient gas conditions with a possible backpressure range between -0.07 MPa < P_C < 0.8 MPa for gasoline fuel injectors (PFI or GDI) and up to 8 MPa for diesel injectors with respect to atmospheric conditions. Particularly, such a vessel must be equipped with the necessary software and hardware to control dynamically the injector pulse width and the duty-cycle. The different optical tools used to visualize and monitor the spray formation process are based on either continuous or phase controlled flashed illumination techniques for the spray in 3D or 2D (laser sheet) conditions. Continuous illumination is mostly used for high-speed photographic recording, whereas phase illuminated techniques apply to CCD camera recording. Measurements of velocity vector distributions in the sprays are made either by Particle Image Velocimetry (PIV – pulsed YAG-laser sheet recording) or by twin-component Phase Doppler Anemometry and droplet diameter analysis.

Figure 6 shows an example of combustion behavior correlation (flame border) based on recordings in a single cylinder transparent engine.

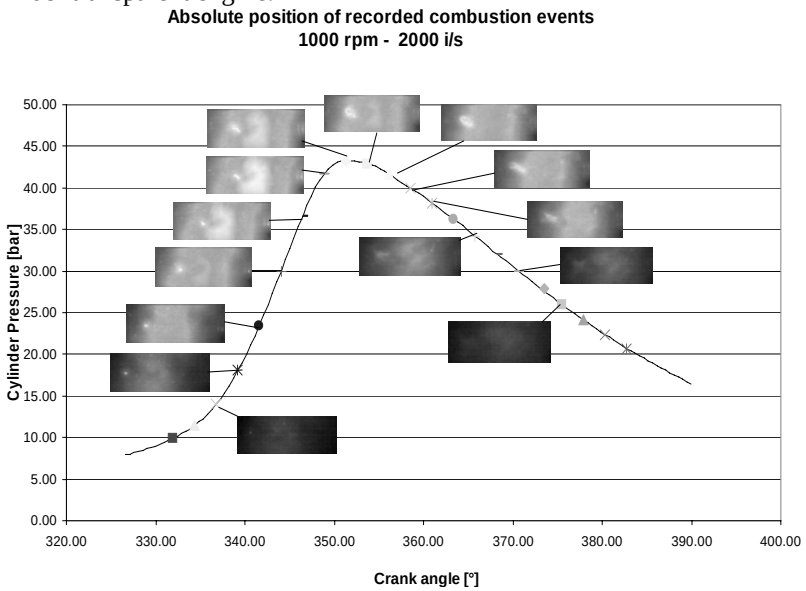


Figure 6 – Combustion model correlation on single cylinder engine

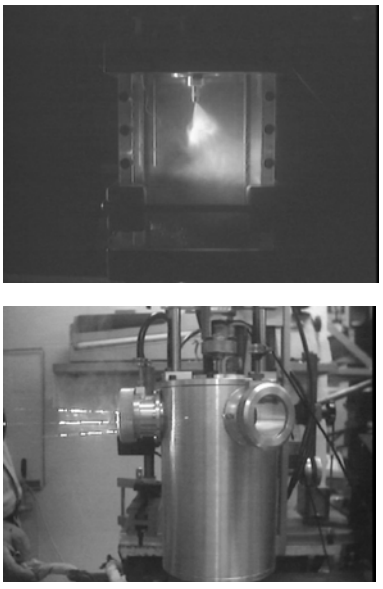


Figure 7 – Example of pressure vessels

Figure 7 shows two spray vessels for gasoline operation, one is a cubic pressure vessel with relatively large rectangular prismatic windows the other a cylindrical vessel with smaller air-shrouded windows, which can be aligned on a non-rectangular optical axis crossing. The first is mainly used for time flashed recordings and PIV, whereas the second is particularly designed for Phase Doppler measurements.

The advantage of using these high-level optical devices and in particular the PDA measurements made near to the injector/atomizer tip is that measured values of droplet and velocity distributions can be used as initial values for the spray and thereby can be avoided the time-consuming simulation of the fuel injector itself.

4. REAL-TIME AUTOMOTIVE EXAMPLE

The increased interest in combustion improvements, which can lead to a decrease of the accumulated Green House Gas (GHG) fraction in the atmosphere, has lately produced a considerable momentum in research and development efforts both in Europe, the USA and South America. The recent UN-report and the Copenhagen conference on possible climate changes have reinforced this trend particularly for SI-engines.

The conversion of conventional IC-engines to alternative fuels is one of the most cost effective means to decrease the crude oil dependence and to limit the air CO_2 pollution. Alternative fuels include Compressed Natural Gas (CNG), methanol, ethanol and bio-diesel, blended or not with conventional oil-based fuels. Following the release of the United Nation's Climate report, which demonstrated the necessity not to delay further the requisite counter measures to decrease the amount of CO_2 added to the atmosphere, political consciousness of the problem has grown. Decision makers in the European Union have in parallel with the voluntary ACEA-agreement (Association des Contructeurs Européen d'Automobile) of a CO_2 fleet limit of 120 g/km by 2012 passed a legislation, which imposes a mandatory blend of at least 10% bio-fuel into all motor fuels by 2020. These are strong indicators for the automotive industry to promote short term, immediately applicable solutions such as the integral or blended bio-fuel approach [7, 8, 11].
The oxidation process within a combustion chamber of a Carbon-based fuel inevitably produces amongst others, CO_2 in quantities proportional to the amount of fuel burned. The CO_2 is released in the atmosphere where it contributes to the reinforcement of the Greenhouse effect. If the fuel is fossil Hydrocarbon-based (crude-oil) the amount of CO_2 is purely added to the atmosphere and the result is an ever-increasing amount in the atmosphere. The speed of increase depends on the total amount of burned HC as well as on the vegetal photosynthesis, which decompose CO_2 and releases O_2. The amount of released CO_2 in modern societies is largely greater than the available vegetal photosynthesis capability. On the other hand, if a fuel, such as ethanol, is extracted from a vegetal basis the amount of CO_2 released equals the amount, which will be absorbed by replacement of the used plants. Therefore the ethanol-cycle can be considered CO_2-neutral, if the necessary energy for distillation is obtained from vegetal raw material.

4.1. Mixture preparation systems background

The research and development of Flex-fuel vehicle layouts has been ongoing for more than two decades [6, 9, 10, 18, 19, 22, 23, 24, 25], and in this matter particularly the Brazilian auto- and subsystem makers promoted by the PROÁLCOOL program, were forerunners. As a result by 2003 on the Brazilian market several series-produced small displacement passenger car engines with the capability to adapt to random fuel mixture conditions were introduced (gasoline/ethanol). At the end of year 2008, more than 90% of all new vehicles on the Brazilian market in the engine segment from 1 to 1.8 l were flex-vehicles. An important factor for this commercial success in Brazil was the early development of production and distribution facilities for ethanol or gasoline/ethanol blends [18, 19, 22]. The experience gathered during the development of the first generations of Flex-vehicles proved once again that the fuel injector is a key element in the mixture preparation process. Therefore it has become necessary to address the fuel atomizer optimization issue to accomplish the further refinement of the atomizer layout, which is mandatory to generate the optimal overall spray momentum and shape when using fuels with very different physical characteristics as well as to meet the new Brazilian PL6 legislation requirements.

The initial development objectives for a flex-fuel mixture preparation system are to find compromises for hard- and software components related to the mixture preparation process, which optimize the components performance with respect to a blended gasoline fuel with a random mix (from 0 to 100 %) of ethanol [18, 19, 22, 23]. The physical background for this optimization problem can be reduced to the following:

To optimize the engine performance it is necessary to control the combustion process. To do this an understanding of the physics and chemistry of combustion processes is fundamental. Only this understanding makes it possible to direct in a given direction one or more of the interacting parameters. The fluid- and thermodynamic conditions as well as chemical reaction rates in the combustion chamber are strongly conditioned by its thermal exchange with the environment, the presence of residual gasses from the previous cycle and the turbulent charge motion in the combustion chamber prior to ignition. Therefore the automotive engineer, who optimizes the combustion process for a PFI-mixture preparation system with a given fuel type, must control the momentum of both the aspirated gas and the fuel sprays as well as their introduction rates into the combustion chamber for all load and speed as well as thermal boundary

conditions. Only by this means it is possible to direct the development of both passive (intake ducts, combustion chamber shape) and active (throttle body and injectors) engine structures as well as the control strategies (ECU) to converge towards an optimal behavior [12, 13, 14, 16, 17].

To optimize the mixture preparation system within the frame of such complicated boundary conditions, it became clear already during initial development of a flex systems that it was necessary to introduce a series of new processes and tools such as the VEM to perform the optimization of the intake system and in particular the fuel injectors. This flex system optimization is illustrated by examples related to a small displacement (1-liter) 8 valve passenger car engine, which today is in series production for integral flex-vehicle application (figure 5 and 8).

Figure 8 – 1.0 l Flex engine hardware

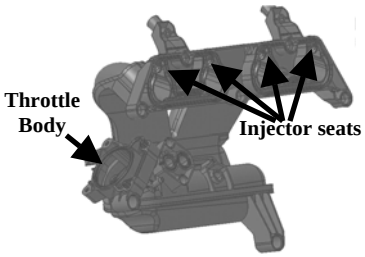
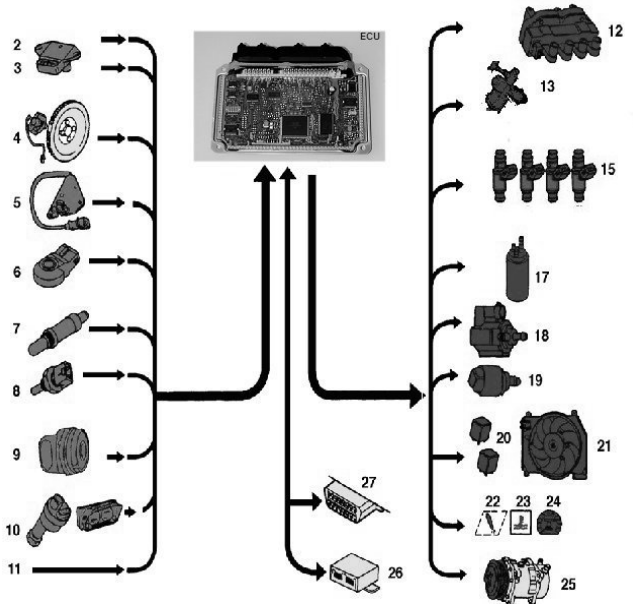


Figure 9 – 1.0 l Flex engine intake system

The engine base-layout is a 4-cylinder engine, which was originally designed and optimized to be fuelled by pure gasoline only. The combustion chamber layout is two valves per cylinder with a basically flat piston design. The compression ratio is adapted to be compatible with the use of gasoline/ethanol mixtures. The intake system is composed of a plastic manifold (figure 9) fitted with an off-center side-mounted Throttle Body (TB) and an air cleaner volume immediately above the TB-inlet.

The engine is fuelled by 4 low-pressure (rail-pressure 0.35 MPa) injectors. The injector seats are located in the intake manifold on the upstream side of the head/manifold interface. A speed density approach (engine speed versus manifold absolute pressure) is used for the mixture preparation management. The management is performed by an engine control unit (ECU) in which a Software Flex-fuel Sensor (**SFS®**) processor is implemented for detection of the composition of the fuel blend used. The percentage of ethanol in the fuel is determined by an analysis of a selected number of engine parameters. A comparison between the λ-sensor signal and the computed air/fuel ratio, based on the speed/density conditions and the metered fuel injected into the engine, makes it possible to extract the instantaneous fuel composition. The injection control and the spark advance can then be corrected to satisfy the change in average fuel characteristics conditioned by a given percentage X ($0 < X < 100$ %) of ethanol in the base gasoline. Figure 10 shows the basic structure of a production integral flex-system.



Legend:

2. MAP sensor
3. Air temperature sensor
4. Speed sensor
5. Phase sensor
6. Knock sensor
7. λ sensor
8. Water temp. sensor
12. Ignition coils
13. Pressure regulator
15. Fuel injectors
17. Fuel pump
18. Canister valve
19. Idle control valve

Figure 10 – Basic Flex Control system

In any mixture preparation system the fuel injector is a key element. The injector parameters, which are related to the fuel spray generation and dynamic behavior (geometry, penetration and momentum) must be carefully optimized to

obtain the best performance over the entire fuel blend range (0 to 100 % ethanol). For electromagnetically operated fuel injectors the main parameters are the dynamic opening/closure response time characteristics (related to the magnetic circuit design) and the dynamic formation of the spray injected into turbulent gas core in the intake manifold (related to the atomizer design). The latter atomizer design parameters must be optimized to produce the optimal ratio between the spray and the gas momentums for the mixture of gasoline and ethanol [18]. The latest third generation injector was subject to further optimization of the electromechanical performance to increase the precision and speed of the duty cycle as well as to increase the linear flow range. These activities were common for both pure gasoline and flex applications and were reported in [24]. On the other hand the optimization of the atomizer plate is more complex for a flex application compared to a pure high-quality gasoline application (example Euro 95). Therefore the usage of the specialized VEM process is necessary to perform a successful optimization.

The main objective of this project was to create virtual injector atomizer layouts, which would produce an optimized performance in fuel atomization (spray momentum adaptation to gas core momentum) over the entire blending range. As the optimized atomizer layout was to be applied to existing engines fitted with un-optimized baseline injectors to improve their cold start performances a supplementary constraint was that the new optimized atomizer could be used without any significant change in the already existing hot engine calibration based on the un-optimized atomizers. Once the optimal choice has been identified by the VEM, sets of prototype injectors can be manufactured and experimentally tested.

The baseline injector was a second generation PICO fuel injector equipped with a flat seat atomizer plate and non-oriented multiple sprays. The layout of this atomizer plate was done at a time before the VEM approach was available at an engineering level. Therefore no real spray/gas momentum ratio considerations were taken into account at that time. Figure 11 shows the baseline layout. At the engine, injectors were installed with the three holes non-oriented atomizer plates producing three independent lobes with an angular separation of 120° and each spray inclined at an angle of 9 ° with respect to the injector axis. The cone angle of each spray is 8°.

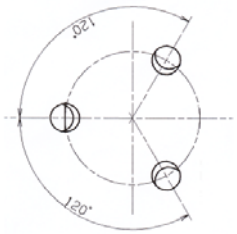


Figure 11 – Baseline atomizer layout

Engine Speed Rpm	BMEP b
Idle	-
2000	2
2500	3.5
3000	5

Table 1 – Engine test points

The notion of atomizer plate orientation is linked to the angular position of the atomizer discharge holes in the plate with respect to the valve stem. If the atomizer plate produces a single conical spray along the injector axis (no bent spray) the orientation of the plate is without importance. If the spray has an off-set (bent spray) or is multi-lobes the angular targeting of the spray can be defined to minimize the amount of fuel, which hits the valve stem above the valve head. This is generally necessary for low temperature boundary conditions. Otherwise the HC-emission can be dramatically increased from cylinders fuelled by an intake port in which a high amount of non-vaporized fuel hits the valve stem.

Another important issue for the design of an optimized atomizer plate is the number of discharge holes machined in the plate. In a first order approach, the ratio between the number of discharge holes and their diameter is determined by the required static and dynamic average flows (engine parameter), the spray momentum produced (optimized by VEM approach) and the spray targeting as well as the injector pulse-width linearity. Nevertheless, experience has shown that other indirect parameters should be kept in mind. Particularly the ratio of the effective flow area to the wetted discharge hole perimeter ς, is also a very important parameter. If this ratio is small there is an increased risk of the appearance of several significant problems. They are all related to the physical fact that during injector closure the head loss across the discharge holes is increasing rapidly and the energy necessary for atomization disappears. The result is the production of very large randomly floating droplets/clusters, which finalize their trajectory at injector closure on the downstream side of the injector atomizer plate. They thus form a more or less thick liquid film on the plate. The intensity of this phenomenon is inversely proportional to ς. A large amount of liquid film on the downstream side of the atomizer plate is very inconvenient for several reasons:

- In general the opening phase of the succeeding injection cycle will initiate by exploding the liquid diaphragm covering the discharge holes. Thereby a series of liquid clusters traveling at low velocity are produced, which will contribute to an increased wall film formation in the injector seat area.
- The amount of a liquid film present on the atomizer plate and its chemical composition (fuel composition variability from one country/state to another) is one of the three fundamental parameters, which favor injector clogging. The two other parameters are the electro-chemical potential of the plate surface (given with the

material chosen for the plate) and the temperature to which the plate can be heated in extreme conditions (example hot soak). If one of the three parameters has not reached its critical value, clogging will not be an issue.

For the above listed reasons it is therefore a rule of thumb that an atomizer plate design should be made by use of the lowest possible number of discharge holes, which is compatible with the momentum, targeting and flow criteria imposed by the engine layout.

4.2. Design of VEM experiment

For the present study engine test rig data were available for the baseline injector configurations. Earlier experiences have shown that for engines in the displacement area between 1 and 2 liters the detailed mixture preparation and transport analysis can be based on a limited number of strategically chosen load points [18, 19, 22]. The engine data at the chosen strategic test points, indicated by tab. 1, were recorded for steady state operation at both 25 °C and 90°C engine coolant temperatures at low load as well as at 90° coolant temperature at 2000, 2500 and 3000 rpm full load. At the test points the power and torque values were recorded, the throttle plate position, the injection pulse and phase, the fuel rail-pressure and temperature, the inlet and exhaust gas temperature and pressure, the engine volumetric efficiency and the λ-value also. The fuelling for the recorded experimental test points was set to 100 % ethanol. Furthermore the instantaneous cylinder combustion pressure was recorded at each test point.

The VEM analysis was conducted according to the following criteria:

- Experimental results are extracted from the virtual engine computations only after 6 complete steady state engine cycles.
- The transport ratio of liquid fuel and vapor towards the intake valve zone and into the cylinder was retained as a quality parameter with assessment of the following 10 key parameters:

1. The total fuel mass (liquid + vapor) trapped in the cylinder after intake valve closure.
2. The liquid mass impacting on the manifold runner wall.
3. The remaining liquid wall film in the valve seat area of the intake port during an engine cycle.
4. The remaining liquid film in the intake runner in both up- and downstream areas with respect to the injector seat location during an engine cycle.
5. The total fuel mass (liquid + vapor) trapped in the injector seat during an engine cycle.
6. Liquid deposit on the valve stem and head during an engine cycle.
7. Total fuel mass (liquid + vapor) present in the manifold after intake valve closure.
8. Crank Angle (CA) resolved A/F in the spark plug gap (A/F_{SP}).
9. CA-resolved A/F at the cylinder liner wall (A/F_{CW}).
10. The ratio between A/F_{SP} and the average cylinder A/F (A/F_{CYL}).

Before the start of the optimization process all the test points were simulated with the baseline injector and the computed cylinder combustion pressures compared to the experimentally recorded. A difference of less than +/- 10 % between recorded and computed pressure data guaranties that predictive data computed with other atomizer layouts will be representative of the changes in real engine performance once the new layout is manufactured and installed on the real engine.

To illustrate the type of immediate detailed information, which can be obtained by the use of the virtual engine a few examples for the 2000 rpm, 2.8 b bmep hot engine load point are given below (3 holes atomizer). The results are related to a 100% ethanol fuelling by the baseline injector. As mentioned above it is very important to gather information about the instantaneous momentum of the gas flow into which the fuel spray penetrates.

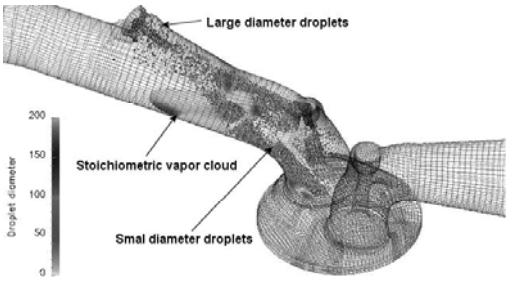


Figure 12 – General status of fuel mixture distribution at injector closure BTDC

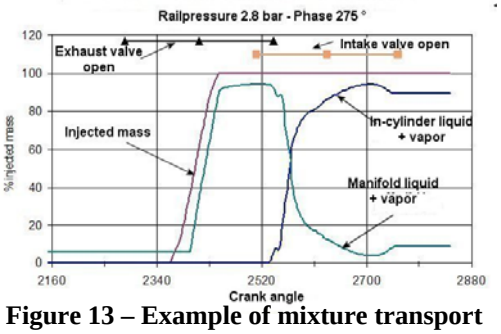


Figure 13 – Example of mixture transport within the intake system.

Figure 12 shows the general instantaneous mixture distribution in he intake system at injector closure at 40° CA Before Top Dead Centre (BTDC) of the intake stroke. Both small and medium size fuel droplets can be identified in the lower part of the intake port, whereas large droplets/clusters are present near to the injector atomizer. Furthermore the instantaneous droplet vaporization can be identified by the vapor cloud representing the presence of a stoichiometric vapor in the intake port.

Figure 13 shows an example of the mixture transport functions within the intake system and the cylinder. The graphic indicates the instantaneous amount (resolution 1° CA) of the total fuel mass (liquid + vapor) as a percentage of the injected mass in the manifold and in the cylinder.

Figure 14 shows an example of the instantaneous $(A/F)/(A/F)_{STOICHIOMETRIC}$ ratio in the spark plug area. The local A/F is integrated in spherical concentric volumes with diameters of 2, 4, 6 and 8 mm. The spheres are centered in the middle of the spark plug gap. For this particular load point the local A/F variations indicate a strong local turbulence in the spark plug gap can be identified from 155 ° CA to 80 ° CA BTDC, where after the turbulence decreases at further compression and remains low just before ignition.

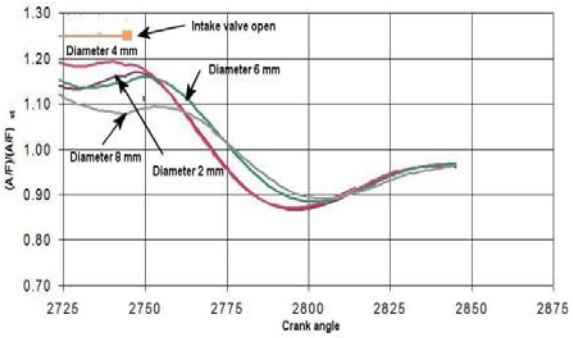


Figure 14 – Example of mixture distribution in concentric spheres centered in the spark plug gap.

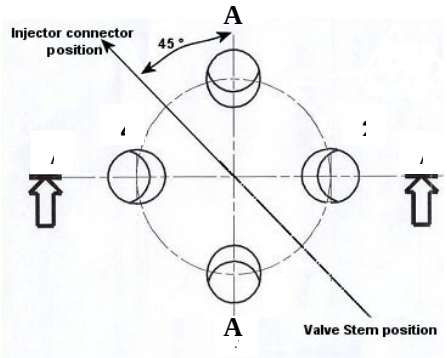


Figure 15 – Layout of optimized four discharge holes oriented atomizer layout.

5. VEM OPTIMIZATION RESULTS

New optimized atomizer layouts were found after four complete iterative loops for the engine at which the imposed boundary conditions were applied to a minimization of the wall film formation in the port area, but without the need of a change in the hot engine closed loop operated ECU calibration. For the engine the new layout of a design of four oriented discharge holes was chosen. Figure 15 shows the general structure of the optimized design. The four discharge holes are symmetrically positioned with an angular distance of 90°. The orientation of the atomizer plate is determined to place the discharge holes group 1 and 2 on one side of the axis linking the injector connector and the valve stem and the discharge holes group 3 and 4 on the other side. For engine the discharge hole diameter was adapted to a static mean flow of 1.83 g/s of ethanol at a rail-pressure of 0.27 MPa. The separation angle between sprays on axis A-A and perpendicular to this axis is 18°. The cone angle of each spray is 8°. Figure 16 gives an example for the engine at cold conditions of the important change of deposit rate in the intake port by a shift from the baseline atomizer to the new optimized atomizer.

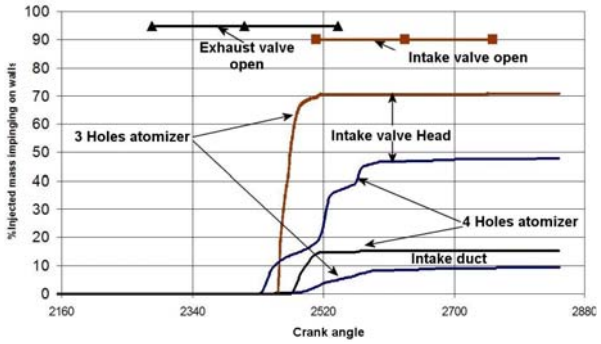


Figure 16 – Change in film deposit intensity by change in atomizer layout at cold engine conditions.

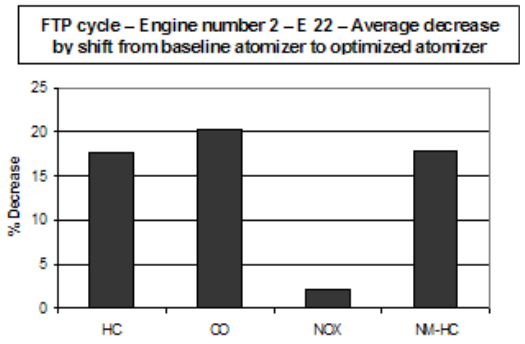


Figure 17 – FTP cycle results.

At the valve head a 30 % decrease is produced. The entire VEM optimization process was concluded in a time of less than 3 months.

200 prototype injectors were manufactured and mounted on a fleet of vehicles, which were tested at hot engine conditions and at FTP cycles. It was confirmed that the new injector layouts were compatible with existing hot engine

ECU closed loop calibration. Figure 17 shows an example of the engine at FTP vehicle test. An average decrease of 18 % in HC, 20 % in CO, 3 % in NO_X and 17 % in non-methane HC emissions was obtained.

6. CONCLUSIONS

The experience gathered during several years of intensive research and development related to flex-fuel mixture preparation systems for series application has shown once again that the fuel injector is a key element in the mixture preparation process. However a further optimization of this element requires the application of a series of new high-performance tools to accomplish this supplementary refinement of the atomizer layout. The objective of this optimization process is to generate the optimal overall spray momentum and shape when using blends of fuels with very different physical characteristics.

The new alternative optimization process described in the paper involves beyond classic engine test rig and vehicle dynamometer experiments the usage of high-level optical visualization and measurement techniques as well as numerical virtual engine modeling. A predictive VEM requires a very precise layout of the computation mesh, which is normally derived directly from the CAD-files of the engine parts and components to be analyzed within the 3-D computation domain.

The VEM based optimization process was applied to a 1.0 l series production flex-fuel engine to improve the fuel injector performance in the first phase of the FTP cycle without imposing a change in the existing hot engine calibration. The VEM is the only tool, which enables in a relatively short time interval a full dynamic analysis (both in time and space) of fuel spray behavior from different real or virtual atomizer layouts.

The VEM analysis was performed in a limited number of strategic load points at which the dynamic spray behavior was analyzed by the assessment of specific tracking of fuel mass (vapor + liquid) transport, deposit and mixing in the cylinder. The analysis was performed both at hot and cold engine conditions. A reference was created for the engine by analyzing the existing baseline injector behavior.

Several iteration loops were made with different virtual atomizer layouts. To satisfy both the hot engine calibration compatibility and decrease in exhaust gas pollutant emission the analysis converged towards a four discharge holes oriented atomizer layout for the engine.

Only the optimized atomizer layout was manufactured and used for a series of prototype injectors, which were tested in several vehicles. The vehicle tests confirmed the compatibility with existing hot engine ECU closed loop calibration. A reduction for the engine in pollutant exhaust emission during the first phase of the FTP cycle was obtained. As an average for the engine the reductions are 19 % in HC, 19% in CO, 5 % in NO_X and 18 % in non-methane HC.

The obtained reductions can be considered sufficient to enable the correct engineering margin for the series produced engine to fulfill the 2009 Brazilian emission legislation.

Currently the mixture system optimization is proceeding amongst others in the direction of introducing new cold start devices also optimized by help of a VEM-approach to make it possible to fulfill the PL6-norm and other new 2012 Brazilian emission requirements.

The described project is an excellent example of an approach, which by the combined use of a VEM and experimental work the desired objective can be reached within a time frame much shorter than what an entirely experimental approach could have produced. However, this result is due to the fact that the VEM as a simulation tool was developed not as an all-purpose tool but as a specific internal combustion engine tool with high-level parallel cluster computation capability.

7. REFERENCES

1. AVL List GMbH, “Fire Combustion Module.”, Fire User’s Manual AVL, Advanced Simulation Technologies, Graz 2006.
2. Amsden, A.A., O'Rourke, P.J, and Butler C.D., "KIVA II A Computer Program for Chemical Reactive Flows with Sprays," Los Alamos National Labs. LA -11560-MS-1989.
3. Amsden, A.A., O'Rourke, P.J, and Butler C.D., "KIVA III: A KIVA Program with Block Structured Mesh for Complex Geometries" Los Alamos National Labs. LA-12503-MS.
4. Colin O., Ducros F., Veynante D. and Poinsot T., “A thickened flame model for large eddy simulations of turbulent pre-mixed combustion.”, Phys. Fluids Vol. 12, Issue 7, 2000.
5. The European Requirement.", Proceedings from Engine and Environment, Graz, September 1997.
6. Baêta J. G., ”Experimental Analysis of Flexible Fuel Systems in Spark Ignition Engine.”, SAE Paper 2005-01-2183.
7. Brustar M. J., Stuhldreher M., Swain D. and Pidgeon W. M., “High Efficiensy and Low Emissions from a Port-injected Engine with Neat Alcohol Fuels.”, SAE Paper 2002-01-2743.
8. G. Cipolla, “Alternative Antriebe für Automobile”, Haus der Technik, Fachbuch Band 88 – 2007, pp 32-49

9. Giroldo M. B., Makant W., Werninghaus E. and Coelho E. P. D., “Development of 1.6L Flex Fuel Engine for Brazilian Market.”, SAE Paper 2005-01-4130.
10. Iwai N., “A study of Cold Startability and Mixture Formation of High-Percentage Methanol Blends.”, SAE Paper 880044.
11. Nakata K. and Utsumi S., ”The effect of Ethanol Fuel on a Spark Ignition Engine.”, SAE Paper 2006-01-2641.
12. Pontoppidan M., Bella G. and Rocco V. "Mixture formation in SI-Engine intake Systems - A simplified theoretical Model with Experimental Verification." - ASME 1996 Spring Congress
13. Pontoppidan M, Gaviani G., Bella G. and Rocco V. "Direct Fuel Injection - A Study of Injector Requirements for Different Mixture Preparation Concepts" SAE Paper 970628
14. Pontoppidan M, Gaviani G., Bella G. and Rocco V. "Improvements of GDI-Injector Optimization Tools for Enhanced SI-Engine Combustion Chamber Layout.’’, SAE Paper 980494
15. Pontoppidan M, Demaio A., Bella G. and Rocco V. “ Enhanced use of 3-D CFD and Combustion Simulation to Optimize the Development of ECU-Control Algorithms for GDI Spark Ignited Engines.” ICE 99 – Proceedings
16. Pontoppidan M. “Combined use of 3D-CFD Simulation and direct Visualization Techniques to decrease the Optimization Procedure Complexity of Combustion chamber Layout and Engine Control Pre-mapping operation necessary for Online Engine Test rig Optimization.”, Internationales Symposium für Verbrennungsdiagnostik, 2002, Baden-Baden, pp.137- 147.
17. Pontoppidan M., Gaviani G., Bella G. and Demaio A. ”Optimization by CFD simulation of Spray formation Parameters to adapt Direct Injection high-pressure Fuel injectors to high-speed SI-engines.”, SAE paper 2004-01-0539
18. Pontoppidan M., Bonfiglioli S., Damasceno F. and Montanari G., “Mixture Preparation Optimization by CFD of a Flex-Vehicle (gasoline/ethanol) Intake System Layout”, SAE-Brazil paper 2004 - No do Resumo 118
19. Pontoppidan M., Bonfiglioli S., Damasceno F. and Montanari G., “Detailed Study of Ethanol In-cylinder Combustion Behavior by means of a Numerical Virtual Engine Model Approach”, SAE-Brazil paper 2005 - No do Resumo 034
20. Pontoppidan M., Demaio A. and Rotondi R., “Parametric Study of Physical Requirements for Successful Use of a Homogenous Charge Compression Ignition (HCCI) Approach in a Direct Injected Gasoline Engine.”, SAE paper 2006-01-0632.
21. Pontoppidan M., Bella G., Demaio A; and Rotondi R., “Parametric Study of Physical Requirements for Optimization of the EGR-rate and the Spray Formation for Minimum Emissions Production over a Broad Range of Load/Speed Conditions.”, SAE paper 2006-01-1120.
22. Pontoppidan M., Bonfiglioli S., Montanari G., Bucci A. and Ewald P, “Description of preliminary Study for Technology Transfer of an Ethanol Mixture Preparation System from Automotive Application to a 4-Cylinder 5.9 liter Aircraft Engine.”, SAE-Brazil paper 2006-01-2878.
23. Pontoppidan M., Bonfiglioli S., Montanari and Ewald P, “Description of Technology Development Methods used to Implement Ethanol Mixture Preparation Systems on both Automotive and Aviation SI-engines.”, Proceedings of XVI ISAF-convention on Alcohol Fuels, Rio de Janeiro, November 2006, Section: Infrastructure, Consumption and Environment.
24. Pontoppidan M., Bonfiglioli S. and Montanari G., “Presentation of the new third Generation “Green” injector Family, PICO-ECOlogical, developed for further Improvement of Flex-fuel Engine Performance.”, SAE-Brazil paper 2007-01-2749.
25. Turner J. W., Pearson R. J., Hollabd B. and Peck R., “Alcohol-Based Fuels in High Performance Engines.”, SAE Paper 2007-01-0056.
26. Reitz R.D. and Han Z., “Turbulence Modeling of Internal Combustion Engine Using RNG κ–ε models” Comust. Sci. and Tech., vol. 106 pp 267-295, 1995.
27. Reitz R. D., Fan L. and Li Z. “Modeling Fuel Preparation and Stratified Combustion in a Gasoline Direct Injection Engine” SAE 1999-01-0175, 1999.
28. Tanaka S., Ayala F.and Keck J., “A reduced chemical kinetic model for HCCI combustion of primary reference fuels in a rapid compression machine.”, Combustion and flame 133, p 467-481, 2003.

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