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# A STUDY OF INJECTION TIMING FOR A DIESEL ENGINE OPERATING WITH GASOIL AND HRG GAS

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*Această lucrare are ca scop evaluarea tendințelor de modificare a performanțelor şi emisiilor unui motor diesel cu variația avansului la injecție când motorul este alimentat în mod dual, folosind programul de simulare AVL Boost. În acest sens modelul creat simulează funcționarea unui motor diesel prevăzut cu un sistem suplimentar de alimentare. Sistemul normal de injecție pentru alimentarea cu motorină este cuplat cu un sistem de injecție extern care asigură îmbogătirea cu gaz HRG (Gaz Bogat în Hidrogen) a aerului admis. Acest gaz este obținut prin electroliza apei şi este furnizat motorului, prin introducere în sistemul de admisie, la diferite debite corespunzătoare procentelor energetice de $H_2$ utilizate în amestecul combustibil (0%, 1,46%, 3,38% si 5,85%).*

*This work addresses the possibility to asses the changing of performance and exhaust emissions with the injection timing variation when a diesel engine operates in dual fuel mode using the simulation program AVL Boost. In this sense the created model simulates the operation of a diesel engine which is provided with a supplementary fuelling system. The normal diesel fuel injection system is associated with an external injection system which ensures the enrichment of the input air with HRG - Hydrogen Rich Gas. This gas is obtained by water electrolysis and is supplied at different flow rates to the engine intake manifold corresponding to different energetic $H_2$ fractions (0%, 1.46%, 3.38% and 5.85%) for the overall mixed fuel composition.*

**Key words:** diesel engine, dual fueling, hydrogen rich gas, injection timing, performance, emissions

## 1. Introduction

Assessments on the new engines performance and emissions obtainable are actually performed in the research and development stages using dedicated simulation tools which offer the possibility to envisage what are the best paths to be followed. Using computational codes for fundamental processes simulation, in conjunction with experiments at the test bench becomes nowadays a common

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practice for the engines development activities, to obtain fast and reliable results with low costs. The existing codes simulate the processes within cylinder (gas flow, air-fuel mixing, combustion and heat transfer) intake and exhaust systems (gas flow phenomena) fuel-injection systems (fuel flow and injection phenomena) etc., in specific conditions imposed by the details of the engine component design and operation conditions [1].

Large differences in the complexity of the mathematical models have resulted in the existing variety of simulation codes. According to the degree of complexity, they are based on a variable amount of input data, defined the engine designed details, physical properties and particular coefficients (as discharge coefficients for flows, heat transfer, fuel-air mixing and combustion). The efficient use of the different computational codes, for reliable analysis, parametric studies and prediction of engine performance needs a preliminary calibration stage, based on experiments carried out at the test bed.

The AVL Boost program, v2009.1 is based on the calculation models associated to each engine component, which are assembled in the data pre-processor package.

According to a graphical programming method and with an interface containing pre-defined constituent elements of the engine, one can specify the relevant geometrical and technical features of every main part of the engine. A symbolic model of the engine is thus designed, which is physically similar to the investigated engine.

## 2. AVL Boost Model

A symbolic model of a tractor diesel engine used for experimental research on a test bed was thus created.

The main elements of the engine symbolic model are pipes (1, 2, 3, 4, 5, 6, 7, 8, 9, 10, 11, 12 si 13) and manifolds (PL1, PL2, PL3 si PL4), junctions, air filter (CL1), cylinders (C1, C2, C3 si C4), injector, measuring points (MP1, MP2, MP3 si MP4) and system boundaries (SB1 si SB2). All these components need design and operational data corresponding to the operation condition which is investigated (Fig. 1) [2].

The fluid flows through pipes and manifolds are simulated by one-dimensional model, with flow coefficients which have to be defined as initial data. The flow is defined by continuity, momentum, and energy conservation equations; the friction coefficient and the heat transfer to the walls are variable along the pipes [3].

The cylinder is obviously the most complex part of the engine model and it contains the most complicated associated submodels, implying thus significantly more input data than all the other components. Connections of the cylinders with the intake and exhaust manifolds are symbolized by pipes and they represent the ports in the cylinder head. In the case of the multicylinder engines the manifolds are considered plenums and sometimes the connection between ports and manifolds are symbolized by junctions. [4]

In the AVL BOOST code the engine is considered as a block of identical cylinders, exchanging successively mass and energy with the surroundings environment trough the valves, according to the given succession of the cylinders operation. The engine geometric dimensions, the piston movement, the firing order, the fuel amount, the combustion characteristic and some corresponding estimates for the thermodynamic parameters of the cylinder charge when the exhaust valve opening and for the heat exchange inside the cylinder must be given as input data.

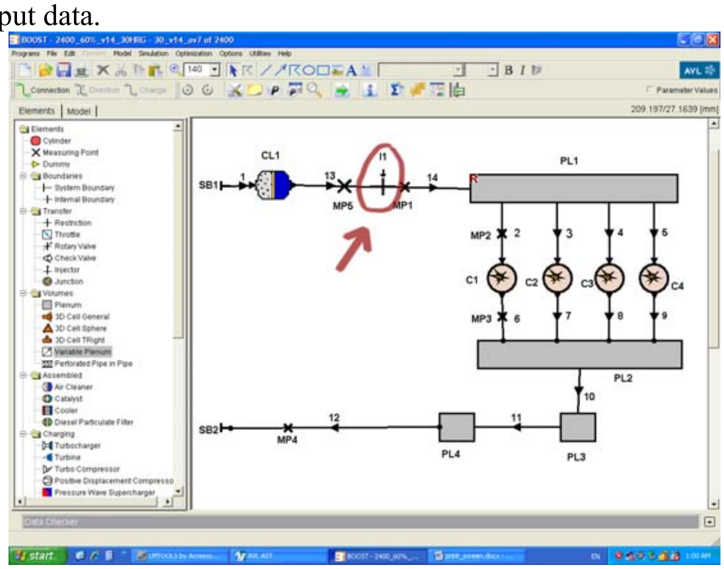

Fig. 1. The engine symbolic model created for the simulation of the HRG gas injection in the intake manifold

Combustion process modeling for diesel engines can be obtained with a real rate of heat release, with a formal type characteristic (single or double Vibe) or with other type of sub-models such as: Woschni/Anisits, Hiroyasu, AVL MCC (Fig. 2). [3]

The AVL MCC model was chosen in this study (Mixing Controlled Combustion) for the prediction of the combustion characteristics. This model considers the effects of the premixed (PMC) and diffusion (MCC) controlled combustion processes according to equation (1) [5]:

$$\frac{dQ_{total}}{d\alpha} = \frac{dQ_{MCC}}{d\alpha} + \frac{dQ_{PMC}}{d\alpha} \tag{1}$$

where: $Q_{total}$ total heat release [kJ];
$Q_{MCC}$ cumulative heat release for the mixture controlled combustion [kJ];
$Q_{PMC}$ cumulative heat release for the premixed mixture combustion [kJ] .

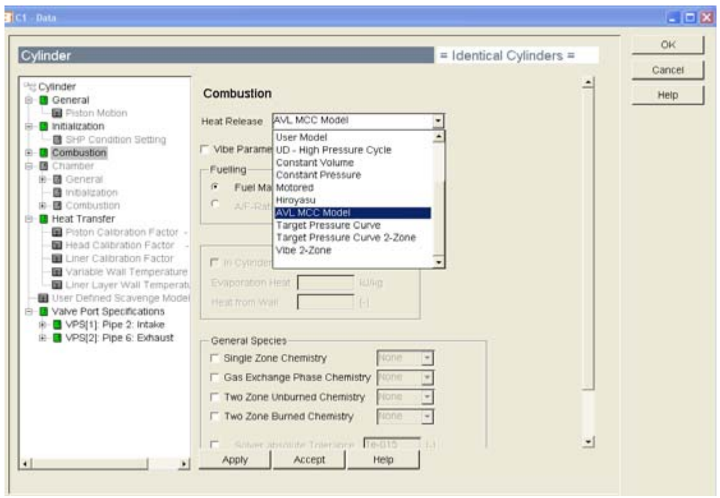

Fig. 2. Combustion models characterized by the type of heat release

The using of the AVL MCC combustion model, requires input data which characterize the injection process (number and injector holes diameter, injection pressure in fuel line, flow coefficients) and the combustion process (autoignition delay factor, combustion parameters, for the in cylinder turbulent flow, for the dissipated kinetic energy, for the EGR influence and for the premixed combustion) (Fig. 3).

These sophisticated submodels are more related to the injection system characteristics and also offer the opportunity to evaluate the formation of NOx emissions, CO and soot. Nitric oxide emission resides in two parameters: NOx Kinetic Multiplier-parameter to adjust the NOx producing kinetics and NOx Postprocessing Multiplier- parameter for setting up the results of NOx generating calculation. The carbon monoxide emission is assesed by CO Kinetic Multiplier-parameter to adjust to CO producing kinetic. The fraction of the smoke particles is ended by two constants Soot Production Constant - to characterize the speed of soot formation and Soot Constant Consumption - to characterize the oxidation rate of soot particles.

The heat transfer to the combustion chamber walls can be evaluated using various computational submodels (e.g. Woschni 1978, Woschni 1990, Hohenberg, Lorenz, Model AVL 2000). In this simulation study Woschni's 1990 correlation was selected with the following calculation relations (equation 2, 3) [3]:

$$Q_{wi} = A_i \cdot \alpha_w \cdot (T_c - T_{wi}) \tag{2}$$

where: $Q_{wi}$ is the convective heat flux to combustion chamber surroundings (cylinder head, piston, liner); Ai the corresponding surface areas, $\alpha_w$ convective heat transfer coefficient; $T_c$ average cylinder charge temperature; $T_{wi}$ wall temperature (cylinder head, piston, liner). [5]

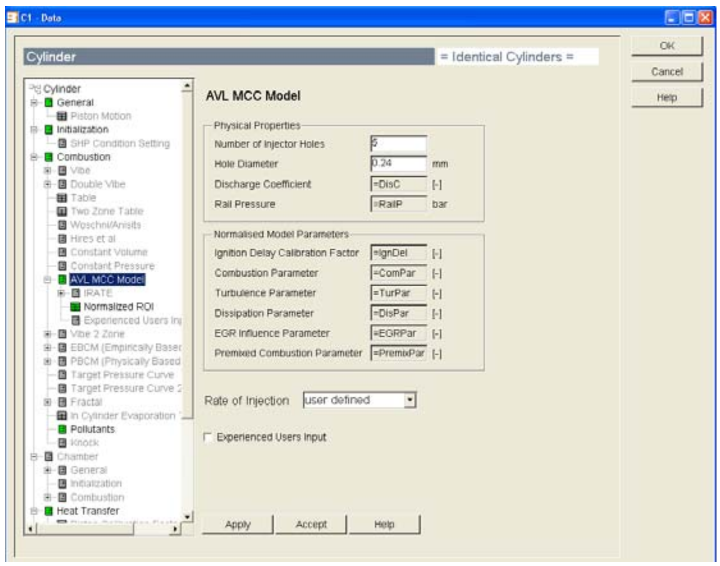

Fig. 3. Input data for AVL MCC combustion model

This correlation aims to ensuring at a more accurate prediction of the heat transfer by connecting it with the engine operation condition including therefore the indicated mean effective pressure IMEP (equation 3):

$$\alpha_w = 130 \cdot D^{-0.2} \cdot p_c^{0.8} \cdot T_c^{-0.53} \cdot \left\{ c_1 \cdot c_m \cdot \left[ 1 + 2\left( \frac{V_{TDC}}{V} \right)^2 \cdot IMEP^{-0.2} \right] \right\}^{0.8} \tag{3}$$

where: $D$ is the cylinder bore; $p_c$ actual cylinder pressure; $c_1 = 2.28 + 0.308 * c_u/c_m$; $c_m$ mean piston speed; cu circumferential velocity; VTDC cylinder volume at TDC, V actual cylinder volume [5].

The normal diesel fuel injection system is associated with an external injection system which ensures the enrichment of the input air with HRG-Hydrogen Rich Gas. This gas is obtained by water electrolysis and is supplied at different flow rates to the engine intake manifold corresponding to different energetic $H_2$ fractions (0%, 1.46%, 3.38% and 5.85%) for the overall mixed fuel composition.

In the last years, industrial hydrogen was produced not only by methane steam reforming but also by water electrolysis technologies which have been continuously improved in order to reduce the energy consumption.

New type of electrolyzers based on particular technologies which do not separate the resulting $H_2$ and $O_2$, gas products ( known as hydroxi gas, HHO, Brown gas, Rhode gas) produce hydrogen-oxygen mixtures that contain hydrogen 64...67% (vol), oxygen 31...33% and 0...5% other compounds of hydrogen with oxygen. In our case this gas named HRG gas, was considered as a stoichiometric hydrogen-oxygen mixture. Therefore for the gas fuel supply on the symbolic model of the engine, HRG input was delivered by the gas injector (I1 in figure 1) and it was considered that the mass fractions of the two gases are 88.88% for oxygen and 11.12% for hydrogen. The corresponding gas flow was continuously injected in the intake manifold and adjusted in constant flow rates by an electronic flow meter Alicat Scientific provided with pressure regulator [2].

Concerning the combustion process, the program needed also as input data the mass percent of diesel fuel and hydrogen from the total amount which participate to the combustion. These fractions are changed as function of the HRG gas flow rate introduced in the intake manifold and as function of the diesel fuel consumption settled experimentally on the test bed. The diesel fuel-HRG mixture lower heating value was automatically calculated based on these fractions and on the lower heating values of the each fuel (diesel and hydrogen), taken from the AVL Boost program data base.

The simulations were based on the hypothesis of internal mixture formation due to the in-cylinder direct injection of the diesel fuel. Thus it was assumed that:

- fuel added to the cylinder charge is immediately combusted;
- combustion products mix instantaneously with the rest of the cylinder charge and form a uniform and homogeneous mixture;
- as a consequence, the cylinder charge A/F ratio diminishes continuously from an initial value at the start of combustion to a final value at the end of combustion.

## 3. Engine test bed

The experimental investigation, for this simulation study was performed at the University Politehnica of Bucharest on a tractor diesel engine, made by Uzina Tractorul Brasov. The engine was internationally certified in Czech Republic in 2004. The engine technical specifications are listed below:

Engine model: 2404.050
Number of cylinders: 4 line
Engine capacity: 3758 $cm^3$
Number of valves: 8
Power: 50 kW at 2400 rpm
Naturally aspirated
Maximum torque: 228 Nm at 1100 rpm
Direct Injection
Relative air/fuel ratio: 1.4
Compression ratio: 17.5
Bore: 102 mm
Stroke: 115 mm
Connecting rod length: 182 mm

The engine was mounted on a test bench specially provided with dedicated equipment for accurate measurement and control of the operating condition parameters. Engine loaded was performed by an eddy current dynamometer type AVL ALPHA 160. Fuel consumption was gravimetrically measured by an AVL Dynamic Fuel Meter 733 S coupled with an AVL Fuel Temperature Controller 753 C. Exhaust gas emissions were determined using a gas analyzer HORIBA MEXA 7170 D. Cylinder pressure traces have been registered by means of an AVL Indiset 620 supported by the AVL Indicom 1.6 software.

The testing matrix was fitted-out by the automatic regulating and control system AVL Puma Open. This test bench management system facilitated the accomplishment of all stages of the testing matrix, with a fast response time and allowed in the same time the primary data processing.

## 4. Model calibration on experimental data

Before starting intensive parametric studies to perform engine optimization, a careful calibration procedure of the engine model is always necessary. Almost every time, it is necessary to perform more than one simple calibration check point in order to increase the model prediction accuracy.

The model calibration is normally based on the cylinder pressure trace and on as much possible measured parameters defining the engine operation regime. These parameters are experimentally obtained by test bed investigation and they may be: power, torque, air and fuel consumptions, volumetric efficiency, air-fuel ratio, residual burned gases fraction, local pressures and temperatures measured in different points of interest along the intake and the exhaust system. System boundary conditions are also extremely important because they imposed the relation between the engine performance and emissions parameters and the ambient [6].

Therefore, local temperatures and pressures measured on the testing bed are compared with the AVL Boost program results for different locations:

- intake manifold - Measuring Point 1 – MP1(in Boost) and T_admis and P_admis (on the testing matrix)
- exhaust manifold - Measuring Point 3 – MP3 (in Boost) and T_EXH_1 and P_EXH_1 (on the testing matrix)
- emissions gas analysis sample probe position– Measuring Point 4 – MP4 (in Boost) and T_EXH and P_EXH (on the testing matrix).

Fig. 4 and table 1 show that pressures and temperatures calculated with AVL Boost, and presented with AVL IMPRESS Chart are comparable with the measured and registered averages values by AVL Puma Open on test bed. These data figure the measured and calculated values for the engine operation condition at part load 60%, speed 2400 rpm, and for 5.85% $H_2$ energetic fraction of HRG in fuel mixture [2].

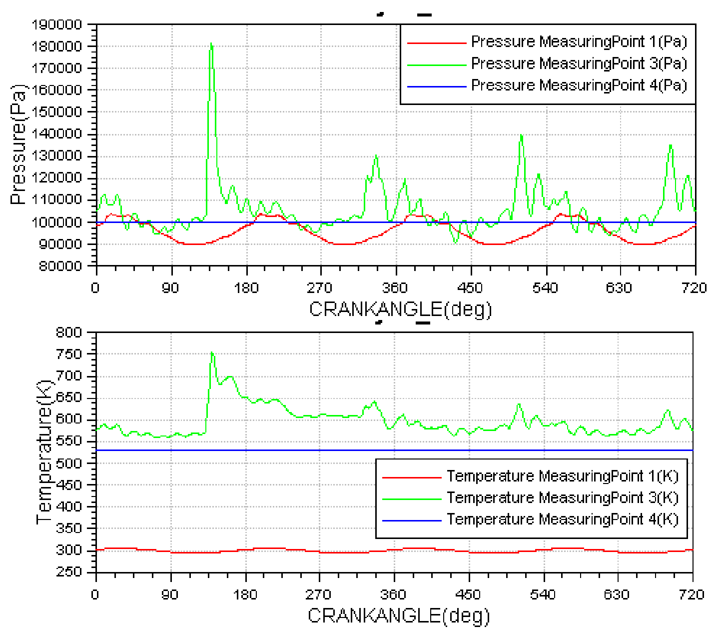


Fig. 4. Local temperatures and pressures calculated with AVL Boost in different locations for engine operating condition 60% load, 2400 rpm, and 5.85% $H_2$

Comparison of the experimental pressure trace (as an average diagram for 300 consecutive cycles registered) with the resulted curve obtained from AVL

Boost simulation is shown in Fig. 5. This comparison shows a good agreement between experiment and simulation and an accurate model calibration.

*Table 1*

**Local temperatures and pressures measured on test bed in the same locations and for the same engine operating condition**

| P_admis | P_EXH | P_EXH_1 | T_admis | T_EXH | T_EXH_1 |
|---|---|---|---|---|---|
| Pa | Pa | Pa | K | K | K |
| -103000 | 100250 | 122000 | 298 | 564 | 651 |

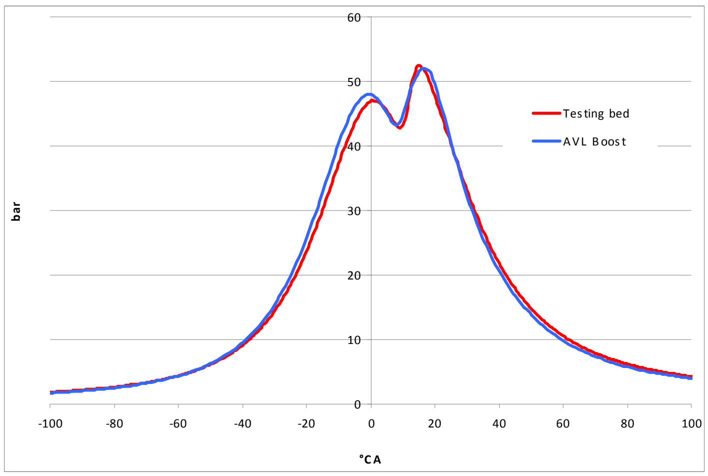


Fig. 5. Pressure traces comparison for model calibration at 60% load, 2400 rpm, and 0% $H_2$

The study of the effect of the injection timing is accomplished for the run at 2400 rpm and 60% load. For each of the energetic additive proportions with HRG pursued ( 0%, 1.46%, 3.38% and 5.85% ) calibration of the model pursuing the experimental data was undertaken.

It can be noticed that the relative deviations of the simulation results face to the reference experimental data are less than 0.25% for effective power, 2.9% for maximum pressure and 5.15% for brake thermal efficiency BTE (table 2 and 3).

After the calibration procedure is finalized, the model can be used for engine the optimization activity based on parametric studies. Many design and operational parameters can be modified, as for example valves timing and valves lift curves, combustion chamber geometry, injection timing and injection system characteristics, air-fuel ratio, heat release characteristics etc.

In this study, the influence of diesel fuel injection timing variation on the engine performance and pollutant emissions formation was considered. The injection characteristic was modified by changing the injection timing, the total injection duration and the normalized injection rate being unchanged as it can be seen in Fig. 6.

*Table 2*

**Model calibration results for 0% and 1.46% HRG**

| | Measured | | Simulated | | Relative deviation % | |
|---|---|---|---|---|---|---|
| $H_2$ energetic fraction [%] | 0% | 1.46% | 0% | 1.46% | 0% | 1.46% |
| Effective power [kW] | 26.84 | 26.84 | 26.84 | 26.79 | 0.00 | 0.19 |
| Diesel fuel consumption [kg/h] | 7.66 | 7.54 | 7.66 | 7.54 | 0.00 | 0.00 |
| Lambda | 2.39 | 2.43 | 2.39 | 2.43 | 0.00 | 0.00 |
| Maximum cylinder pressure [bar] | 53.05 | 53.80 | 52.64 | 52.25 | 0.77 | 2.88 |
| NOx emission [ppm] | 345 | 370 | 345 | 370 | 0.00 | 0.00 |
| CO emission[ppm] | 336 | 302 | 336 | 302 | 0.00 | 0.00 |
| Specific smoke value [g/kWh] | 0.1957 | 0.1870 | 0.1953 | 0.1860 | 0.21 | 0.52 |
| BTE [%] | 29.75 | 29.66 | 28.22 | 28.19 | 5.12 | 4.96 |
| Indicated efficiency [%] | 38.02 | 37.91 | 38.50 | 38.49 | -1.26 | -1.52 |

*Table 3*

**Model calibration results for 3.38% and 5.85% HRG**

| | Measured | | Simulated | | Relative deviation % | |
|---|---|---|---|---|---|---|
| $H_2$ energetic fraction [%] | 3.38% | 5.85% | 3.38% | 5.85% | 3% | 5.85% |
| Effective power [kW] | 26.84 | 26.84 | 26.78 | 26.78 | 0.22 | 0.22 |
| Diesel fuel consumption [kg/h] | 7.41 | 7.23 | 7.41 | 7.23 | 0.00 | 0.00 |
| Lambda | 2.45 | 2.49 | 2.45 | 2.49 | 0.00 | 0.00 |
| Maximum cylinder pressure [bar] | 53.84 | 54.06 | 52.31 | 53.28 | 2.84 | 1.44 |
| NOx emission [ppm] | 386 | 406 | 386 | 406 | 0.00 | 0.00 |
| CO emission[ppm] | 288 | 264 | 288 | 264 | 0.00 | 0.00 |
| Specific smoke value [g/kWh] | 0.1505 | 0.1423 | 0.1507 | 0.1424 | -0.15 | -0.05 |
| BTE [%] | 29.59 | 29.57 | 28.12 | 28.07 | 4.99 | 5.05 |
| Indicated efficiency [%] | 38.08 | 37.88 | 38.38 | 38.32 | -0.78 | -1.16 |

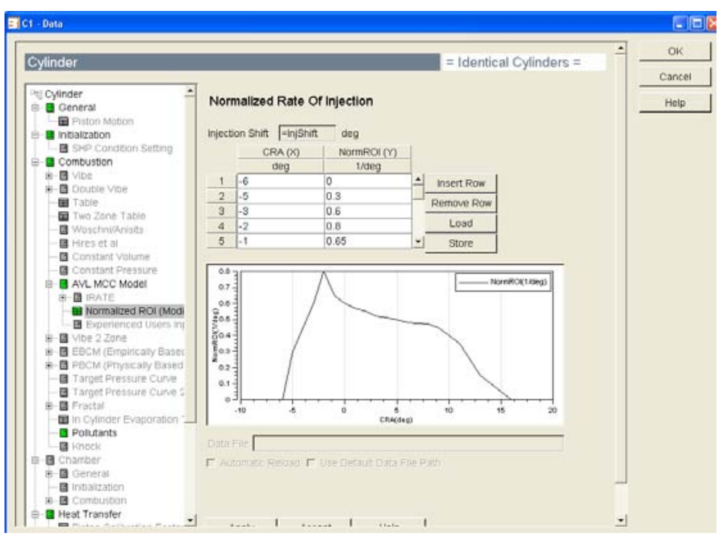

Fig. 6. Normalized injection rate as an unchanged input parameter

## 5. Injection timing effects

The AVL Boost program adapted for engine operating conditions with air enriched HRG in the intake system was used to study the effects of diesel fuel injection variation on performance and emissions. The study tried to evaluate the possibility to compensate the decrease in engine efficiency determined due to HRG addition, which was experimentally established by means of injection timing optimization.

The parametric study of the influence of injection timing variation was accomplished for different values ranging from 4°CA to 22°CA. The main results of the simulation are figured below in Figs. 7, 8 and 9.

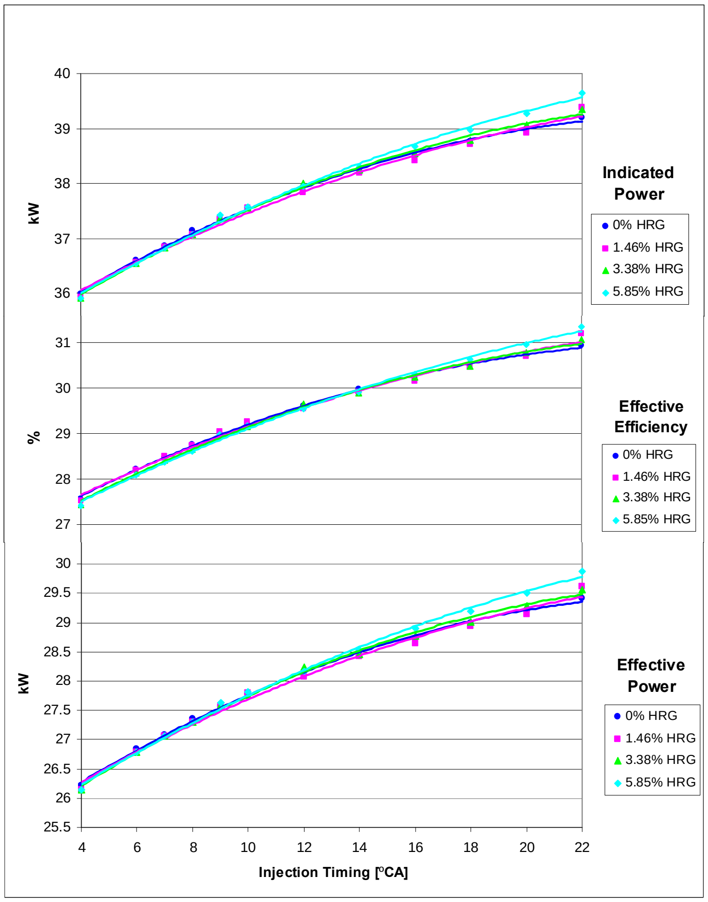


Fig. 7. The influence of injection timing on indicated power, effective efficiency and effective power at 60% load and 2400 rpm

The following aspects can be noticed:

The increase of the injection timing from 6 °CA to 8 °CA, for a constant HRG flow rate corresponding to 3.38% $H_2$, leads to an increase of the effective power by 3.8%, of the maximum pressure by 11.3 % and of the brake thermal efficiency by 1.92 %; these effects can be attributed to the combustion improvement due to its shifting towards top dead center.

The operation with 3.38% $H_2$ and injection timing 7 °CA relative to the reference condition (0% $H_2$, and injection timing 6 °CA), is associated with a decrease of the smoke number by 25%, of the CO emission by 17%, and with an increase of the NOx emission by 24%. The maximum cylinder pressure records a moderate increase from 52.64 bar to 55.27 bar.

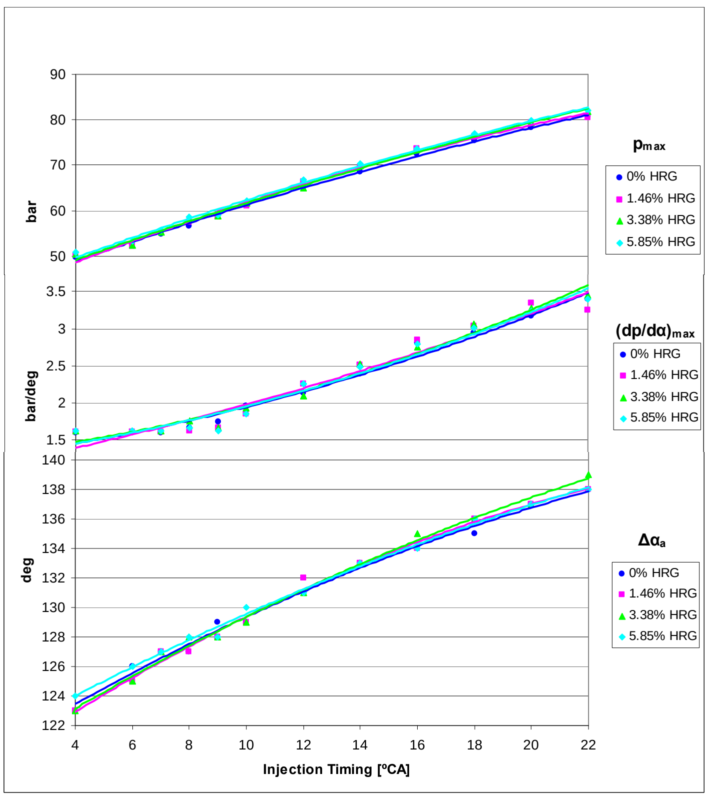


Fig. 8. The influence of injection timing on maximum pressure, maximum pressure rise and on total combustion duration at 60% load and 2400rpm

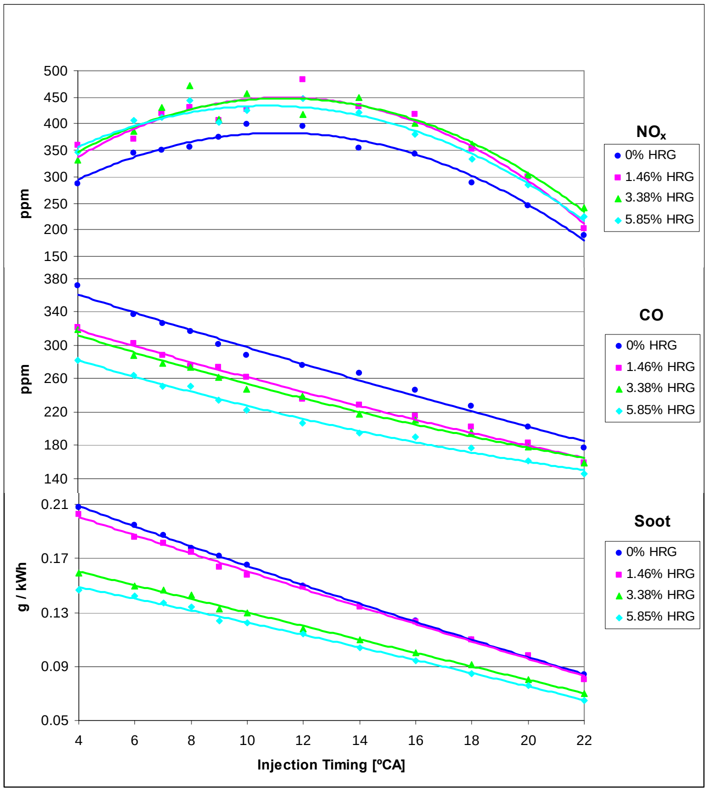


Fig. 9. The influence of injection timing on NOx, CO and smoke, at 60% load and 2400rpm

## 6. Conclusions

In conclusion, the simulations performed with the AVL Boost program have showed that for reduced HRG flow rates and optimized diesel fuel injection timing, itcan be obtained:

1. Slight increase in the engine brake thermal efficiency and moderate increase of the maximum cylinder pressure.
2. Significant smoke and CO emission reductions, but important NOx emission increase, which however can be reduced by promoting the cooled EGR solution.
3. These aspects seem to be particularly interesting from the perspective of modifying the injection characteristic of the Delphi rotary type injection pump, in direction of advancing the static injection timing.

**Acknowledgments:**

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