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# Water Injection Effects In A Single-Cylinder CFR Engine

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## ABSTRACT

Though analysed by a few researches, the practice of water injection in Spark Ignition Engines (SI-ICE) does not yield homogeneous results, owing to various typologies of engines used for experiments.

In this paper the effects of water injection in the intake pipe are investigated from both a theoretical and experimental viewpoint. Pressure vs. time diagrams were recorded on a single-cylinder CFR engine at AGIP PETROLI, Priolo (CT).

Tests were performed according to Research and Motor Method (ASTM). Water was supplied by a continuous injection system inclusive of comparatively high pressure pump. The engine was fed with low O.N. base gasoline (cheap products, intermediate of refinery processes). The water to fuel mass flow rate ratio was varied in the range 0 to 1.5.

The NOx emissions measurements confirm the tremendous effectiveness of water injection in reducing the engine environmental impact.

Test data have been used to implement a detonation model that allows to predict water injection effects.

Results have shown that water injection really represents a *new way* to avoid detonation, to reduce compression work and to control NOx formation in SI engines.

## INTRODUCTION

The use of water as supplement to fuel-air mixture has been considered repeatedly since the early development of internal combustion engines (ICEs). Water has been added as an internal coolant, as a knock suppressant and a means to control emissions. Observations generally regard water effects on engine performance, knock and emissions.

Clearly the practise of water injection in ICES is not new [1], but nowadays it is considered carefully for the benefits it allows. The most important are the current requirement for raising the octane number (O.N.) of fuel (for the replacement of the primary anti-knock additive in petroleum refining technology), and the reduction of NOx emission in the exhaust gas.

Early designs of high compression ratio aircraft piston engines employed a water vapour induction system to increase the fuel anti-knock rating and engine output. With 10% injection of water[1], during intake stroke fuel octane rating is boosted about 2 to 4 units with simultaneous increases in the engine output of 30 - 50 % [2, 3].

The use of water-fuel emulsion for control NOx emission was described by Nicholls [4]. It was found that 10% water in gasoline caused 10-20% reductions in nitrogen oxides. Bignardi et al. [5], while proposing a kinetic model, computed the composition, temperature, internal energy and nitric oxides content of the exhaust gas, under influence of water added to fuel.

Russian researches, Satpov and Lusho [6], have reported the effects of water presence in fuel on SI engines. They indicated that the NOx content in exhaust gas dropped by 1.3 % for 1% addition of water to emulsified fuel.

All observations are attributed to a cooling effect due to the high latent heat of vaporisation of water and to a water induced slow-down of the combustion process. Direct test evidence on latent heat effects and on water induced changes in combustion rates does not yield homogeneous results, owing to various typologies of the engines used for experiments.

In this paper the effects of water injection in the intake pipe of a single cylinder CFR engine was investigated. Tests were performed at AGIP Petroli Fuels Control Laboratories, Priolo (CT), according to Research and Motor Method (ASTM).

Water was supplied by a continuous injection system inclusive of a comparatively high pressure pump. The water nozzle (De Lavan - 45° spray, 0.2 gal/h under the pressure of 1 MPa), was located in the intake manifold downstream from carburettor, shortly before the inlet valve. Engine test data with base fuel and gasoline-water mixture were compared to identify the water injection effects.

Actually, the instability of the emulsified fuel[2] was the main disadvantage encountered in the utilisation of

1. Water per unit fuel (volume)

water-fuel emulsions. Moreover it is necessary to create the emulsion shortly before its introduction in the combustion chamber since no emulsion promoter was employed[3].

## EXPERIMENTAL APPARATUS

A standard engine[4] arrangement and instrumentation was utilised except for:

- a piezoelectric pressure transducer located in the knock plug hole;
- a positive displacement water pump;
- a piezoelectric pressure transducer mounted at the inner surface of the intake pipe;
- a charge amplifier;
- a digital data recorder system;
- a modified intake pipe provided with a special water nozzle.

Part of instrumentation set-up and some parts of the modified engine are shown in fig.1. Table 1 shows a list of the main technical properties of experimental devices.

Pressure data were recorded, as well as crank-angle by a storage oscilloscope data system. Motoring pressure trace was also monitored immediately following the fired cycle by shorting instantaneously the primary circuit of engine ignition. Fuel consumption was measured using a 100 ml burettes and a digital electronic counter. Air flow into the engine intake pipe was measured by a calibrated Venturi flow meter. The intake pipe was provided with a special water nozzle expressly built in order to allow controlling water injection rate.

Fuel was introduced at the carburettor “Venturi” (9/16”) via standard jet. Adjustment of fuel and water flow rates, to the engine, was obtained by varying water supply pressure (in the range of 0.5 to 2 MPa), fuel head, and by controlling water back-flow to the pump suction, by means of a fine needle valve.

A fitting was introduced amid the standard intake pipe so that the water nozzle was properly located. Pipe walls wetting was minimised and interference was avoided with gasoline spray. A scheme of the modified intake pipe with water injector is shown in figure 2.

Exhaust emissions of CO and $CO_2$ were monitored with a non-dispersive infrared analyser (NDIR), unburned hydrocarbons by a flame ionisation detector (FID), and oxides of nitrogen were measured by chemiluminescence method after dilution with zero air.

## FUELS

Several blended gasolines produced at AGIP Petroli, coming from "CR30" plant, were tested at AGIP Experimental Fuel Control Station, both Research and Motor methods. However better results have been found out over a base product called “BL2 CR30”[5] rated at 64 O.N.M.M (68 O.N.R.M.).

Table 2 shows a list of some of the physical and chemical properties of this base gasoline.

It is well known that fuel octane quality limits performance and efficiency on SI-ICE. Gains can be achieved by both design provision and refinery processes.

The reduction of the amount of anti-knock additive substances consequent the use of water injection as anti-knock system would have a remarkable implications from at least two points of view:

- keeping the real cost of industrial product low;
- minimising the engine environmental impact.

These two concepts form together one of the aim of this paper.

## LABORATORY ENGINE TESTS

It is widely accepted that the audible knock in SI engines is caused by pressure waves due to very fast heat release. Such fast combustion cannot occur from the mechanisms that govern normal turbulent flame propagation but rather from chemical reactions proceeding in the unburned gas ahead of the flame front which make this *end gas* auto-ignite almost instantaneously.

At the end of the suction stroke, finely atomised, liquid water gasoline and gas are compressed. The vaporisation of the liquid will cool the gas and consequently less work will be required for the engine compression stroke. At the same time, the liquid whose vaporisation is delayed until the piston is in the neighbourhood of TDC, requires a negligible compression work, while, later in vapour phase, it is able to supply the piston with significant expansion work. Water, with its high latent heat, is particularly suited for this purpose.

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2. The practical implementation of this concept involves the solution of many technical problems because the production of a high quality emulsion is required whilst the add-on kits used to retrofit the existing engines should result as compact, reliable, and cheap as possible.
3. Several techniques are under investigations for emulsion product, based on mechanical (ranging from high pressure homogenization to cavitation and ultrasonic phenomena) or chemical concepts.
4. Waukesha Engine Division – Dresser Industries , Wisconsin, USA.
5. This is a Virgin Naphtha coming directly from topping plant .To increase the original O.N. M.M. to 85, typical of commercial fuel, major total cost is about 45-50 $/ton.

Comparative tests show that a noticeable change in compression pressure does not occur until approximately 20 deg BTDC as it is shown in pressure – crank angle diagrams of figure 3.

The decrease in compression pressure corresponds to less than 20% vaporisation of the injected water. To obtain this scope, the intake temperature was raised to[6] 416.2 K while the engine speed was kept 900 r/min. With lower intake temperatures, vaporisation was negligible since no appreciable pressures reductions were detected.

Experimental test data show that water injection into an unheated manifold, when the engine speeds is reasonably high, will not give water a sufficient time to vaporise during the compression stroke. Of course, in normal operation of a naturally aspirated engine the water does not vaporise until after combustion is well under way. This effect well explains why injection of water into very highly compressed or supercharged engines has been successful.

During combustion development the maximum peak pressure location, as it is shown in figures 3, 4 indicates how well the engine can transform its gas pressure into output work due to the beneficial effects of the water injection. The results covered up to this point demonstrate the influence of water injection on the efficiency at constant value of the compression ratio. The substantial advantage obtained by water injection regards the regular combustion development that should be impossible, otherwise. The comparison with detonation mode leads to the consideration that the engine can operate at the maximum non knocking compression ratio, then its efficiency can be increased.

It is well evident that water injection slows the combustion process down in SI engine. Thus to maintain standard MBT spark setting the timing should be advanced during test, when using water injection gasoline combinations. Unfortunately such a procedure has not been performed, because the available test rig was not provided with dynamometer.

It is worth to know that keeping the spark advance constant while increasing the water rate leads to a loss of potential piston work gain.

In the following figs. 4,5 are shown p–V diagrams that demonstrate the influence of the water injection/fuel mass ratio "*s*"[7] , in the work cycle, at constant value of the volumetric compression ratio (**r**) and the maximum non knocking **r** value, at constant "*s*" for a fixed fuel.

6. In standard CFR engine the temperature control set is provided in °F

7. In the following context it is meant by water injection rate, the ratio of water to fuel mass flow rate.

Based on this result the efficiency gain can be obtained by taking advantage of the octane quality of water / gasoline combinations only.

Several octane tests on different base gasoline according to R.M. were carried out and experimental results are shown in Table 3.

As mentioned in an earlier section, all experimental findings were normalised with respect to the data obtained using base gasoline at standard spark advance. The influence of water injection was experimentally measured and best results were carried out using base gasoline called "BL2 CR30".

Effects of water injection over octane tests in accordance with both Research and Motor methods, are shown for a fixed fuel in fig. 7.

As it is shown in fig. 7, using base gasoline with water injection, the R.M. octane number rose from 70 ÷ 93, and M.M. octane number rose from 64 ÷ 90, while "*s*" was varied in the range 0 ÷1.5.

The better beneficial effect of water injection appears in the field of 1.25 "*s*" value.

The exhaust temperature did not significantly change for rich mixture conditions when water was injected in the range of 0.5 water injection rate. However for water injection rate of 1 ÷ 1.50, exhaust gas temperatures decreased somewhat. Shown on the fig. 8 are the exhaust temperature changes which were measured during engine tests. Water injection increased the total mass of charge in the cylinder since the air fuel flow rates were held constant for each test. Thus most of the observed temperature drop can be accounted for by the increased dilution of the charge.

Also $NO_x$, HC, and $CO/CO_2$ concentrations in the exhaust gas, were detected under standard knocking intensity with and without water injection.

$NO_x$ emissions were significantly reduced with water injection and this reduction was present in any case. The $NO_x$ level was reduced more then 50 %, while water / fuel ratio was about 1.50. The $NO_x$ levels for the fixed fuel, are plotted versus water to fuel mass ratio (0 ≤ "s" ≤ 1.5) and diagram is shown in the fig. 9.

The effect of water injection caused a small drop in carbon monoxide and hydrocarbons emission increased, probably due to change in quench layer thickness and gas temperatures. However this hydrocarbons increase is related to the fuel components and chemical participants.

## DETONATION INDEX

It is well known that auto-ignition of perfectly homogeneous mixture is controlled by factors such as temperature, density, turbulence and chemical composition of fuel. Starting from the consideration that the preflame reactions of a real mixture in a real engine begin early during the compression stroke and continue at an accelerating rate as the compression pressure and temperature increase, compression and combustion events are hereinafter considered.

Water injection into air-fuel mixture affects parameters such as instantaneous gas temperature and heat transfer rate. Since these variables are extremely difficult to measure experimentally, pressure data were processed.

An engine cycle study that models the compression and combustion processes is implemented and proposed in the following.

Thermodynamic effects of water injection spark ignition engine are considered. To better demonstrate these changes in work cycles, pressure-volume (p-V) diagrams are considered in order to develop theoretical calculations.

As far as a mixture of only gasoline and air is induced into cylinder, standard methods can be employed in order to calculate the instantaneous gas temperature from cylinder pressure data.

Suitable assumptions are usually introduced, about the fuel evaporation process, that are reasonably in accordance with experimental evidence (i.e. all fuel in vapour phase at intake valve closure).

Unfortunately no such assumption can be made for injected water, whose lower vapour pressure and higher latent heat cause vaporisation to be delayed well after the beginning of the compression stroke.

In order to estimate the instantaneous gas temperature value into combustion chamber a mathematical model has been developed starting from experimental data carried out in CFR engine tests.

A gas mixture composed of air, fuel, ad water have been considered. This assumption of “pseudo-fluid”[8] allows to determine the correct value of the mixture constants $R_u$ and $R_b$ for unburned and burned gas, by comparing instantaneous values of experimental gas pressure and maximum value of gas temperature carried out during motored engine tests.

In order to develop a model that allows to define a detonation index to predict this phenomenon, a thermodynamics calculation for compression stroke and combustion process is implemented.

The combustion chamber was divided into two regions: unburned and burnt. The charge was assumed to be composed of ideal gases (frozen in the unburned gas region and in chemical equilibrium in the burnt gas region), and the first law of thermodynamic, equation of state, mass and volume conservation were applied to the burnt and unburned gases.

The pressure was assumed to be uniform throughout the cylinder and *r* first order ordinary differential equations can be obtained for the pressure, mass, volume, composition, and temperature of the burnt and unburnt gases.

In this model the two zones are separated by a spherical surface of discontinuity (the flame front). The model was based on the assumptions that the flame front was animated by radial advancement and leakage flows into crevices were neglected.

The equations governing the burnt and unburnt gas zones can be written as:

$$m = m_u + m_b$$

$$\frac{dm}{d\theta} = \frac{dm_u}{d\theta} + \frac{dm_b}{d\theta}$$

$$V = V_u + V_b$$

$$p \cdot V_u = m_u \cdot R_u \cdot T_u$$

$$p \cdot V_b = m_b \cdot R_b \cdot T_b$$

$$v = \frac{V}{m} = x_b \cdot v_b + (1 - x_b) \cdot v_u$$

$$v_b = \frac{V_b}{m_b} \qquad v_u = \frac{V_u}{m_u}$$

$$\frac{d(m_b \cdot e_b)}{d\theta} = -p\frac{dV_b}{d\theta} - \frac{dQ_b}{d\theta} + h_u \frac{dm_{b,R}}{d\theta}$$

$$\frac{d(m_u \cdot e_u)}{d\theta} = -p\frac{dV_u}{d\theta} - \frac{dQ_u}{d\theta} + h_u \frac{dm_{u,R}}{d\theta}$$

$$e_u = e_u^0 + \int_{T_o}^{T_u} c_{v,u}\, dT$$

$$e_b = e_b^0 + \int_{T_o}^{T_b} c_{v,b}\, dT$$

$$h_u = e_u + p\frac{V_u}{m_u} \qquad h_b = e_b + p\frac{V_b}{m_b}$$

where the subscription u indicates the unburnt gases and the subscription b indicates the burnt gases.

Note that:

$$\frac{dm_{u,R}}{d\theta} = -\frac{dm_{b,R}}{d\theta} = -\frac{dm_c}{d\theta}$$

where $dm_c/d\theta$ is the combustion or mass burning rate.

In absence of leakage flows into crevices:

$$\frac{dm_{u,R}}{d\theta} = \frac{dm_u}{d\theta} \qquad \frac{dm_{b,R}}{d\theta} = \frac{dm_b}{d\theta}$$

8. The remaining vaporization rate will be developed during combustion process.

During combustion phase the mass burnt fraction was defined as follows, according to Wiebe model.

$$x_b = \frac{m_b}{m} = 1 - e^{\ln(0,001)\cdot\left(\frac{\theta-\theta_s}{\Delta\vartheta_b}\right)^n}$$

This set of equations is not closed (i.e. there are more unknowns than equations). Closure can be achieved by specifying the mass burning rate $dm_c/d\theta$ and the geometry of the flame front.

Thanks to this water mass burning rate it is possible to take in account the "vaporization degree" of water during compression stroke. Calculated values of gas temperature and pressure during compression stroke are shown in figs 10 and 11 for different water mass flow rate.

Details of this model are reported in [7].

Detonation phenomena depend on the story of pressure and temperature of the end gas in a variable time range.

Pre-reaction, during this time, within the unburned fraction may be completed with the consequence of auto-ignition presence into the gas fraction. The upper limit and lower limit of the reaction time are variable. In ICEs the lower limit is identified with the time of the ignition (starting of the combustion $t_0$). According to the theory of Livengood and Wu [14], the portion of the pre-reaction that occurs in the generic time t subsequent the starting time $t_0$, can be represented in the following form:

$$\int_{t_0}^{t} \frac{1}{\tau}\, dt$$

where $\tau$ is the instantaneous time delay, as a function of pressure and temperature.

This equation allows to determine the time $t_{det.}$ when the detonation starts (if it is present), because the equation can be written in the following form that describes the trend of the pre-reaction in the all field of the time delay:

$$\int_{t_0}^{t_{det.}} \frac{1}{\tau}\, dt = 1$$

Using this equation, it is possible to predict the onset of the detonation, when the value of $t_{det.}$ is fixed, assuming $t_{det.}$ as the instant in which the maximum pressure in combustion chamber is achieved. Thus the following identity can be written as:

$$t_{det.} = t_{p\,max}$$

To evaluate a detonation index, it is necessary to specify the expression of the time delay as function of pressure, temperature and mixture strength.

According to several authors [7, 13], this expression can be written as:

$$\tau = A\cdot\frac{e^{\frac{B}{T}}}{p^C}$$

where A, B, C are constants depending on the mixture features. In this case the values of constants are:

$$A = 17.68\cdot\left(\frac{O.N.}{100}\right)^{3.402}$$

$$B = 3800$$

$$C = 1.7$$

where O.N. is the octane number of the fuel.

Then, not knocking conditions can be represented by :

$$\int_{t_0}^{t_{p\max}} \frac{p^C}{A\cdot e^{\frac{B}{T}}}\, dt \le 1$$

$$\frac{30}{\pi\cdot n}\int_{\varphi_0}^{\varphi_{p\max}} \frac{p^C}{A\cdot e^{\frac{B}{T}}}\, d\varphi \le 1$$

where n is the engine speed and $\varphi_0$, $\varphi_{p\,max}$ are respectively the crank angle when the combustion begins, and the crank angle at combustion peak pressure value on the normal[9] cycle.

Thus it is possible to define an index that allows to know whether the detonation occurs or not. The expression for this index is:

$$I_{det.} = \frac{30}{\pi\cdot n}\int_{\varphi_0}^{\varphi_{p\max}} \frac{p^C}{A\cdot e^{\frac{B}{T}}}\, d\varphi$$

For the not knocking condition it is found:

$$I_{det.} = \frac{1}{n}\int_{\varphi_0}^{\varphi_{p\max}} \frac{p^C}{e^{\frac{B}{T}}}\, d\varphi \le \frac{\pi}{30}\cdot A$$

The term at the right hand of this equation can be interpreted such as a "detonation index required" [10] from the fuel and the left hand such as a "detonation index available" for the engine. The first index depends on fuel characteristics exclusively, while the second one depends on the engine features only.

Detonation index required and available versus water mass flow rate are shown in figs 12 and 13. It is possible to underline that detonation margin grows up when fuel O.N.M.M. runs from 65 to 90, according to diagrams

9. Normal cycle is the cycle optimised for the maximum mean pressure.

10. Note the analogy with pumps.

shown in figs 4, 5 and 6. The detonation index required (blue line in fig. 12) decreases while the water to fuel mass ratio grows up from 0 to 1.5 according to diagrams shown in figs 4, 5 and 6. The detonation index available (red line in fig. 12) increases while the water to fuel mass ratio grows up according to diagram shown in figure 7.

## TABLES AND FIGURES

Table 1.

| Description of Device | Model |
|---|---|
| Data Acquisition Board | AX5412 High Speed |
| Piezoelectric Transducer located in the knock plug hole | Kistler 7063 |
| Piezoelectric Transducer mounted in the inner surface of intake pipe | Kistler 601A |
| Charge Amplifier | Kistler type 5007 |
| Position Transducer | Elettroprogetti s.a.s. |
| Thermocouple | Type K (NiCr-NiAl)<br>[ 0°C ÷ 1260°C ] |
| Water Nozzle | Delavan 45° spray<br>0.2 gal / h |
| Positive Displacement<br>Water Pump | Bosch<br>9 / 100 |
| Rated delivery pressure<br>Rated flow rate<br>Pump speed<br>Max Power | 10 [ MPa ]<br>9 [ $dm^3$/min ]<br>2800 [ r/min ]<br>2 [ kW ] |

Table 2.

| Chemical and Physical properties | Base Gasoline BL2 CR30 |
|---|---|
| | |
| Motor Octane Number (N.O.M.M.) | 64 |
| | |
| Density [ $kg/dm^3$ ] | 0.6776 |
| Lower Heating Value [ MJ/kg ] | 44.83 |
| Reid Vapour Pressure [ kPa ] | 70.33 |
| | |
| Sulphur [ ppm ] by vol. | 4 |
| Olefins [ % ] by mass | 1.2 |
| Naphthene [ % ] by mass | 10.0 |
| ISO-Paraffin [ % ] by mass | 40.2 |
| NOR-Paraffin [ % ] by mass | 44.3 |
| Aromatic Tot. [ % ] by mass | 4.3 |
| Aromatic Superior [ % ] by mass | 0.7 |
| Aromatic Index | 20.0 |
| Benzene [ % ] by mass | 1.8 |
| Toluene [ % ] by mass | 1.0 |
| Xilene [ % ] by mass | 0.8 |

Table 3.

| O.N. RESEARCH METOD | | | | | |
|---|---|---|---|---|---|
| | s=0 | s=0.5 | s=1 | s=1.25 | s=1.5 |
| CR30 "BL1" | 34 | 42 | 50 | - | - |
| CR30 "BM1" | 52 | 61 | 69 | - | - |
| CR30 "BL2" | 70 | 77 | 87 | 91 | 95 |
| CR30 "BL3" | 73 | 81 | 90 | - | - |

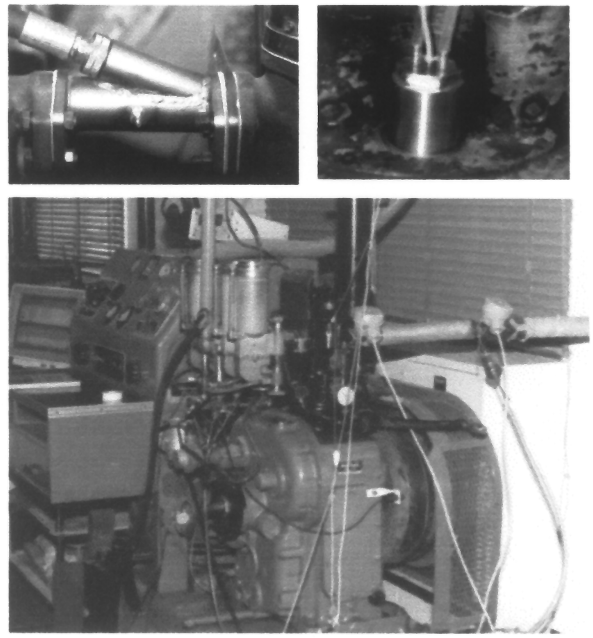


**Fig. 1**

Engine set-up and some parts of the modified engine

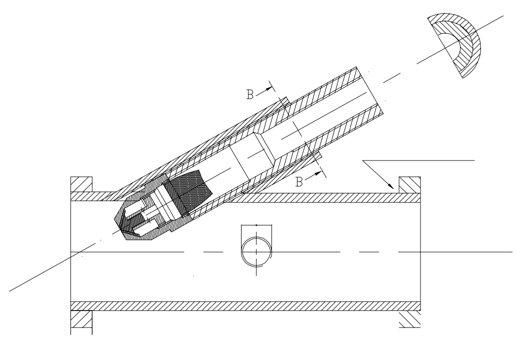
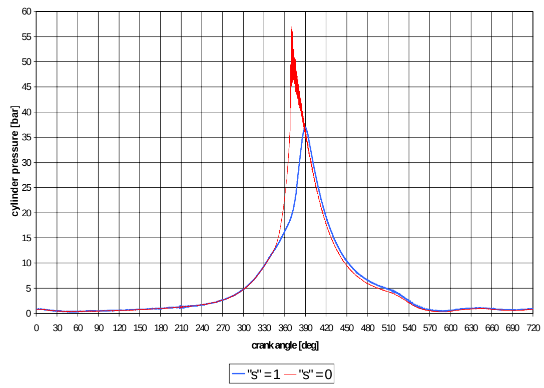



**Fig. 2**

Scheme of the modified intake pipe with water injector



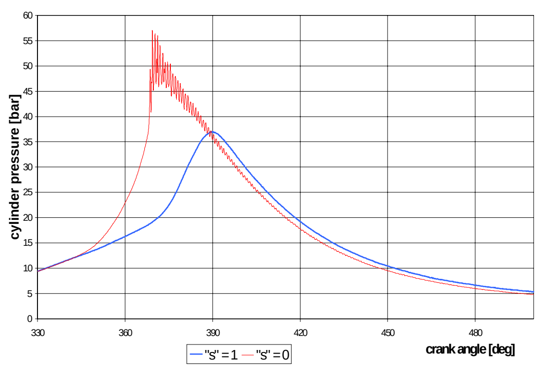




**Fig. 3**

Effect of water injection over pressure traces of CFR engine

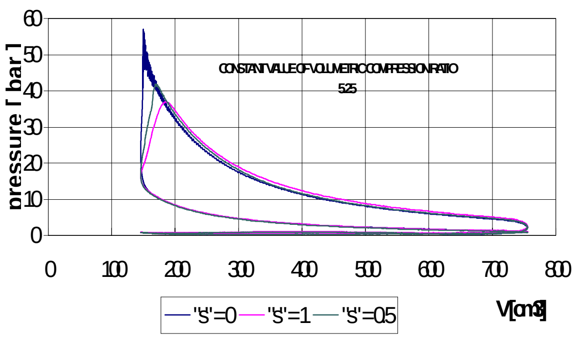



**Fig. 4**

Effect of water injection on the indicated pressure-volume diagram

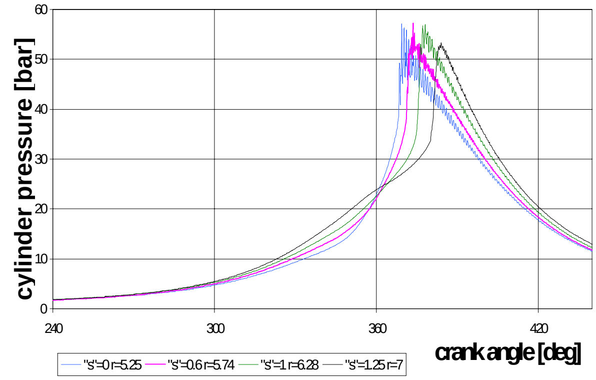



**Fig. 5**

Effect of the water injection on pressure – crank angle diagram (fixed fuel ρ variable)

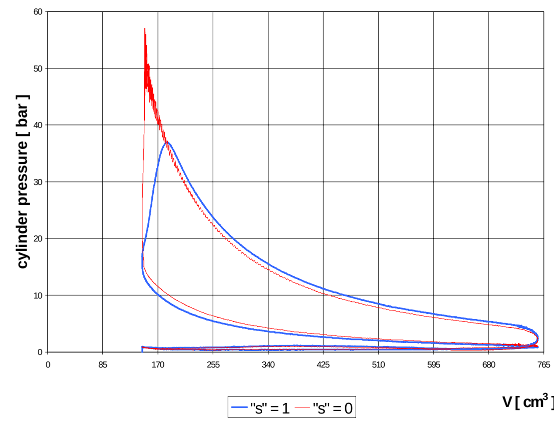



**Fig. 6**

Effect of the water injection on the diagram area

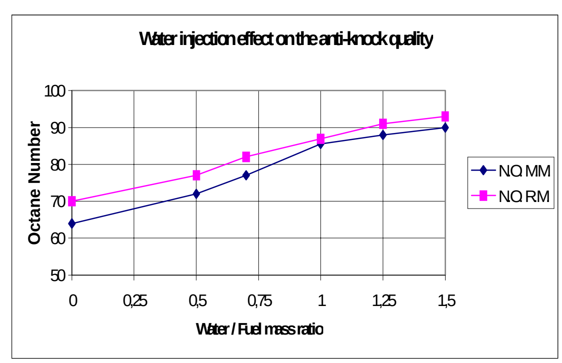



**Fig. 7**

Effect of water content on the anti-knock quality of water-gasoline combination

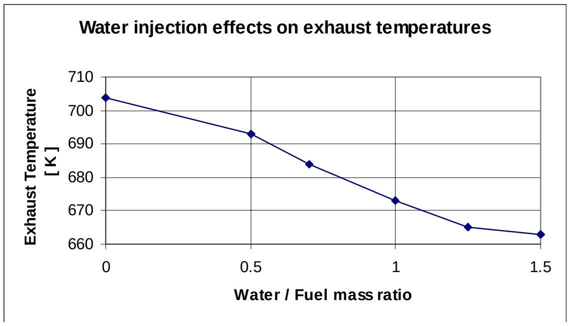



**Fig. 8**

Effect of water content on the exhaust temperatures of water-gasoline combination

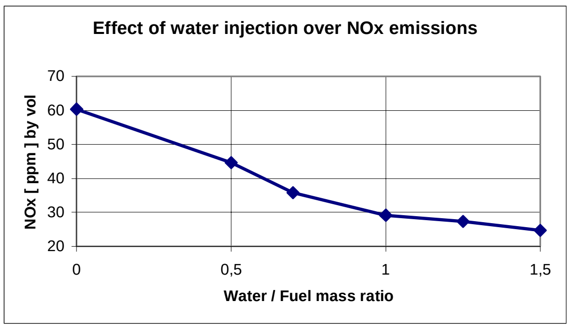



**Fig. 9**

Effect of water content on the $NO_x$ emissions of water-gasoline combination

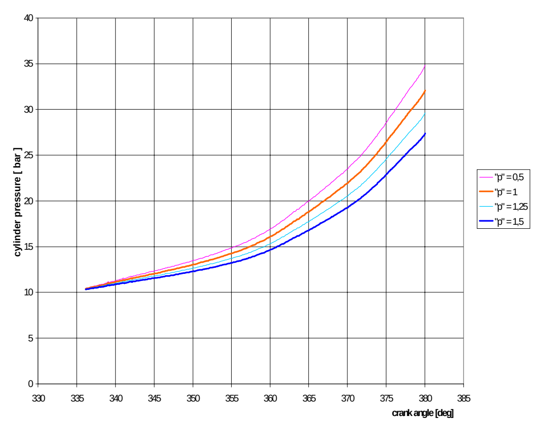



**Fig. 10**

Calculated pressures versus water / fuel mass ratio

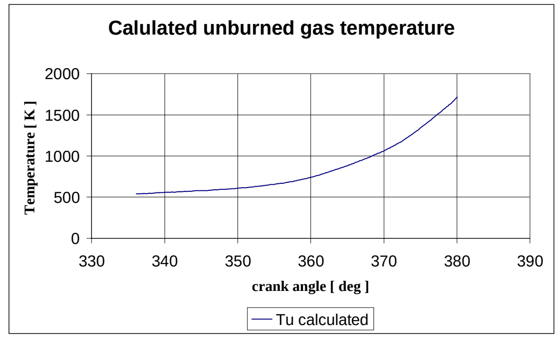



**Fig. 11**

Temperature of the unburned mass fraction during combustion development (“s” = 1.25)

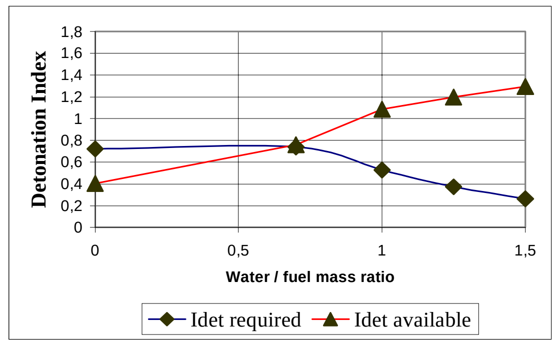



**Fig. 12**

Detonation Index “required” and “available” versus water / fuel mass ratio

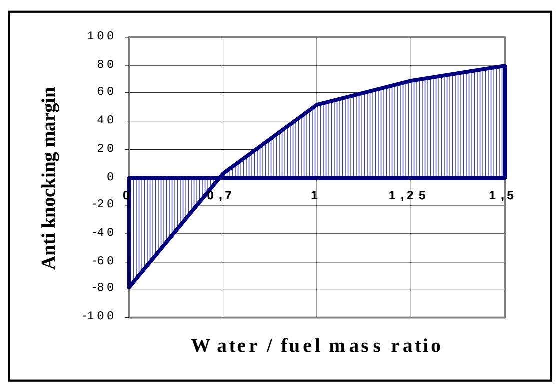



**Fig. 13**

Anti knocking margin versus water / fuel mass ratio

## CONCLUSIONS

The development of SI- ICE has been accompanied by the desire to raise the compression ratio for increased engine efficiency and better fuel economy. One obstacle to this gain in the economy, at times, has been the octane quality of the available gasoline. To overcome this limitation, water was proposed by several authors as an antiknock to suppress detonation. The information generated during these studies, under a variety of engine typologies and running conditions, does not yield homogeneous results.

In this paper water injection effects on a single cylinder CFR-SI engine have been investigated from both theoretical and experimental viewpoint. Tests were performed at AGIP Petroli Fuels Control Laboratories, Priolo (CT).

The primary effects of the examined water injection system were concerned with the utilisation of very low O.N. base gasoline (cheap products, intermediate of refinery processes) in SI-ICE, antiknock characteristics, detonation index, and exhaust emission.

Based on the experimental findings in engine fed with base gasoline mixture and gasoline-water-air combinations, it is possible to conclude that:

1. Water injection system was tested on CFR engine according to Research and Motor methods (ASTM) in order to increase the antiknock rating of a virgin naphtha rated at 64 O.N.M.M (68 O.N.R.M.).
2. With regard to the tested base gasoline, by simultaneously increase of volumetric compression ratio and water injected mass flow rate, measured Research Octane number increases to 93 from 70 and Motor Octane number to 90 from 64.

   At present the best results were obtained with water injection/fuel mass flow rate in the field over one ratio (1 < “s” ≤ 1.25) at standard engine spark advance.
3. Experimental data showed nitric oxide reductions of over 50% with water injection/fuel ratio in the range from 1 to 1.25 (1< “s ”< 1.25). Water injection caused a drop in carbon monoxide emission concentrations; this effect was more noticeable with rich mixture running conditions where CO levels were relatively high. Conversely, water injection in amounts equal to the fuel flow caused about 20 % increase of hydrocarbon emissions in the exhaust gas.
4. Using both experimental p-V data and calculations results, a model that allows to determine detonation index to predict water injection effects for a fixed fuel, while “s” was varied from 1.5 to 0, has been implemented. The comparison between required and available detonation index leads to the definition of a suitable anti knocking margin that allows to extend investigations to several low O.N. petrochemical products.

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## NOMENCLATURE

**s:** water to fuel mass ratio
**m:** total gas mass
**$m_u$:** unburned gas mass
**$m_b$:** burned gas mass
**θ:** crank angle
**V:** total gas volume
**$V_u$:** unburned gas volume
**$V_b$:** burned gas volume
**$R_u$:** unburned gas constant
**$R_b$:** burned gas constant
**v:** gas specific volume
**$v_u$:** unburned gas specific volume
**$v_b$:** burnedgas specific volume
**$x_b$:** gas mass burnt fraction
**$h_u$:** unburned gas specific enthalpy
**$h_b$:** burned gas specific enthalpy
**$e_u$:** unburned specific internal energy
**$e_b$:** burned specific internal energy
**$e_u^0$:** unburned specific internal energy of formation at the reference temperature
**$e_b^0$:** burned specific internal energy of formation at the reference temperature
**p:** cylinder pressure
**T:** absolute temperature
**t:** time
**$\tau$ :** time delay
**n:** engine speed
**$I_{det}$:** detonation index

subscripts

**u:** unburned
**b:** burned
**R:** reaction
**CR:** crevices