

Towards an Innovative Combination of Natural Gas and Liquid Fuel Injection in Spark Ignition Engines

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ABSTRACT

In order to address the CO₂ emissions issue, and to diversify the energy for transportation, CNG (Compressed Natural Gas) is considered as one of the most promising alternative fuels given its high octane number. However, gaseous injection decreases volumetric efficiency, impacting directly the maximal torque through a reduction of the cylinder fill-up.

To overcome this drawback, both independent natural gas and gasoline indirect injection systems with dedicated engine control were fitted on a RENAULT 2.0L turbocharged SI (Spark Ignition) engine and were adapted for simultaneous operation. The main objective of this innovative combination of gas and liquid fuel injections is to increase the volumetric efficiency without losing the high knocking resistance of methane.

This paper describes how the CIGAL™ (Concomitant Injection of Gas And Liquid fuels) concept can generate a synergy effect between both fuels, which allows combustion efficiency to be increased significantly on turbocharged SI engines. Indeed, the gas and gasoline resultant mixture can be injected in stoichiometric conditions even at full load with spark timing set close to MBT (Maximum Brake Torque), thus leading to excellent performances, compliance with stringent regulations, good specific consumptions and reduced CO₂ emissions.

This injection strategy can also be extended to other liquid fuels, such as ethanol, and gaseous fuels like LPG (Liquid Petroleum Gas): the purpose remains having the best distribution between gas and liquid injections in order to

maximize their advantages. Like for methane and gasoline, a strong synergy was observed with LPG and gasoline concomitant injection, leading to lower specific consumptions compared to those obtained in single fuel injection.

This study demonstrates that the concept of concomitant injection offers direct benefits on the autonomy management of gas and gasoline powered vehicles. It also opens very interesting perspectives for meeting future emission regulations using only a 3-way catalyst since stoichiometry is maintained over the whole engine operating range.

INTRODUCTION

The use of alternative fuels such as CNG in spark ignited engines is considered as a very interesting way to reduce CO₂ emissions in short term and to limit the dependency to crude oil [1].

The combustion of CNG, which is primarily composed of methane, leads to many benefits linked to its specific thermodynamic and chemical properties. For instance, due to its H/C (Hydrogen over Carbon ratio) molecular ratio close to 4 (vs. 1.8 for gasoline), the theoretical CO₂ emissions of CNG for the same energy introduced in the combustion chamber are 23% lower than with gasoline in stoichiometric conditions [2].

Moreover, compared to gasoline, CNG offers an improved knock resistance as a consequence of its high equivalent octane number. Indeed the RON of methane is about 130 compared to 95-98 for premium European unleaded gasoline fuels. Engines can thus be designed with a higher

compression ratio than for conventional gasoline engines and MBT spark advances can be applied in continuous stoichiometric conditions whatever the load and engine speed.

These key properties of CNG increase thermal efficiency and offer CO₂ savings which have been highlighted by several Well-to-Wheel analyses [3, 4].

Nevertheless, some drawbacks associated to the use of gaseous fuels should be mentioned. Unless direct injection is adopted, a port fuel injection CNG engine will suffer power output losses over 10% compared to the same size gasoline engine due to the displacement of the air by the gaseous fuel [5]. This issue has been partially addressed by using turbocharged engines which represent the latest CNG commercial vehicle developments. However, unlike liquid fuels, gaseous fuels lack latent heat of vaporization which cools the fuel/air mixture. This reduces volumetric efficiency of CNG engines compared to gasoline ones [6]. Furthermore, because of the absence of a liquid fuel spray acting as a cooling agent and a lubricant, intake valves are exposed to higher temperatures, and higher valve and seat wear rates are observed on CNG engines [6, 7]. Additionally, the intake valves of these engines are also subjected to increasing deposit formation due to the lack of gasoline spray with detergent additive as a cleaning agent.

In order to get the most of both fuel's advantages, this paper presents an innovative injection concept consisting in injecting simultaneously a gaseous and a liquid fuel during the same engine cycle: the CIGAL™ concept (Concomitant Injection of Gas And Liquid fuels). This idea has been reported in two patent applications [8, 9], the main principle being the injection of a fuel having a low energy density and a high octane number (e.g. CNG) coupled with the injection of a fuel having a high energy density and a low octane number (e.g. gasoline).

Stein et al. [10] have performed recently direct injection of E85 coupled with port-injection of gasoline on a turbocharged engine with increased compression ratio. Thank to its high octane number, the use of approximately 40% in mass of E85 allowed higher BMEPs (Brake Mean Effective Pressure) than what could be achieved using 100% gasoline, due to the fact that the stoichiometric conditions could only be reached in bi-fuel mode. Consequently reduced fuel consumptions were obtained, hence a better usage of gasoline.

In this paper, the potential of the concomitant injection of gas (methane, LPG) and gasoline is explored on a turbocharged, port-injected SI engine with increased compression ratio. The objective of this work is to improve the performances reached with gas by adding a liquid fuel, while keeping a high knock

resistance of the fuel mixture. In other words, the main idea is to exploit the high octane number of the gaseous fuel (e.g. methane), and the density and latent heat of vaporization of the liquid fuel (e.g. gasoline) in order to generate a synergy between them. Depending on expectations in terms of engine settings, the methane/gasoline ratio can be adapted depending on the operating point. This solution offers promising perspectives for the after-treatment of exhaust gases and for on-board energy management in a bi-fuel vehicle.

EXPERIMENTAL

EXPERIMENTAL SET-UP

ENGINE TECHNICAL CONFIGURATION

The tests were carried out on the RENAULT F4Rt engine. This is a 2.0L four-cylinders, spark ignited, turbocharged, variable intake valve timing, indirect injection gasoline engine.

This engine was specially adapted for methane operation in order to maximize its performances and efficiency in methane mode, while allowing operation with gasoline without any performance penalty compared to the stock engine. This optimization work is described in details in a dedicated paper [11].

Different changes to the stock configuration were made. An indirect gas injection system was specifically designed and implemented without modifying the global engine packaging. Four low pressure gas injectors were connected to a feed ramp and the gas was injected in the intake manifold, just before the valve ([Fig. 1](#)). The gas injection pressure was controlled at 2 bar above the intake manifold air pressure to ensure injection flow control. The gasoline injection system remained unchanged.

(See [Figure 1](#) after last section of paper.)

Since higher thermal and mechanical constraints in the combustion chamber were expected, different valves were used to avoid valve seat wear (material and geometry of valves and seats). Reinforced cylinder head, connecting rods and connecting rod bearings were used for a safe operation of the engine at higher firing pressures.

In order to reach the performance targets, the compression ratio was increased from 9.5:1 to 11.5:1. Piston shape and combustion chamber (adaptation of tumble level) were designed using IFP C3D-simulation tool. Also, the turbocharging loop was adapted and specific valve timing settings associated to a Variable Valve Timing device were selected.

The [Table 1](#) shows the technical characteristics of the engine used in this study.

(See [Table 1](#) after last section of paper.)

Cylinder pressure was monitored in all cylinders with four flush mounted AVL QC34D pressure transducers. Manifold air temperature, coolant temperature, oil temperature and engine back-pressure were controlled. Gasoline consumption was measured by an AVL 733S gravimetric fuel balance, and gas consumption was measured by a MicroMotion CMF010 Coriolis type mass flow meter.

The engine was controlled with an IFP in-house management system device, called ACEbox© (Automotive Control Embedded solution based on xPCTarget). This full pass system allowed highly flexible engine control, especially for the concomitant injection mode.

FUELS

Three different fuels were tested in this study. A conventional gasoline containing 5% ethanol, representative of a European premium commercial gasoline; pure methane (CH₄) which will be referred to as methane in this study, and commercial LPG. Their properties are given in [Table 2](#).

(See [Table 2](#) after last section of paper.)

The complexity of these tests and the different types of fuels required therefore highly flexible infrastructures.

EXPERIMENTAL PROCEDURE

Mechanical, thermal, emission, stability and safety limits were identified and constrained the testing during this study. These limits are summarized in [Table 3](#).

Table 3. Limits of engine operation

Turbocharger speed	≤ 160 000 rpm
Average cylinder pressure + 2 σ	≤ 135 bar
Pre-turbine temperature	≤ 950°C
Raw HC emissions at full load	≤ 10 000 ppmC
CoV IMEP (Coefficient of variation Indicated Mean effective Pressure)	≤ 5%
Engine knock	absence of knock

The knock limit was determined both audibly and by on-line visualization of the four cylinder pressure signals via an oscilloscope.

BI-FUEL MODE EXPERIMENTAL PROCEDURE

A specific procedure, was used for operating the engine with concomitant injection of gas and liquid fuels (called bi-fuel mode). The target operating point was first tuned in gas (e.g. methane) mode only. Then, gasoline was added with an

imposed injection time while the equivalence ratio was controlled by the gas injection time via the lambda sensor. An increase of the gasoline injection time resulted therefore in an increase of the fuel mass distribution while maintaining the equivalence ratio unchanged.

The independent measurements of both the gasoline mass flow and the gaseous fuel mass flow allowed the fuel mass distribution to be calculated (i.e. ratio of the mass flow of gasoline over the total mass flow of gasoline and gas). This fuel mass distribution was then regulated by varying the injection time of gasoline.

Regarding engine tunings in bi-fuel mode, the injection timings of each fuel were kept identical to those used in single fuel mode. Spark advance was adjusted at MBT or at KLSA (Knock Limit Spark Advance). VVT settings were adjusted to the best trade-off between MBT and THC emissions.

A corrected LHV (Lower Heating Value) of the global fuel introduced into the combustion chamber (mixture of gasoline and gas) was calculated linearly from the mass flow measurements of each single fuel. This corrected LHV in bi-fuel mode is referred to as cLHV in this paper.

For example, in bi-fuel mode, $LHV_{\text{test fuel}} = cLHV$, and in methane mode, $LHV_{\text{test fuel}} = LHV_{\text{methane}}$.

CORRECTED BSFC

Whatever the test fuel (single fuel or bi-fuel), the BSFC was corrected in order to be able to compare the different specific fuel consumptions independently of the fuel's LHV. Therefore, the BSFC was corrected at the gasoline LHV according to the following formula:

$$cBFSC = BFSC_{\text{test fuel}} \times \frac{LHV_{\text{test fuel}}}{LHV_{\text{gasoline}}}$$

RESULTS AND DISCUSSION

SINGLE FUEL MODE REFERENCE TESTS

Reference full load curves ([Fig. 2](#)) were acquired with methane and gasoline in single fuel injection during the optimization phase of the engine [11].

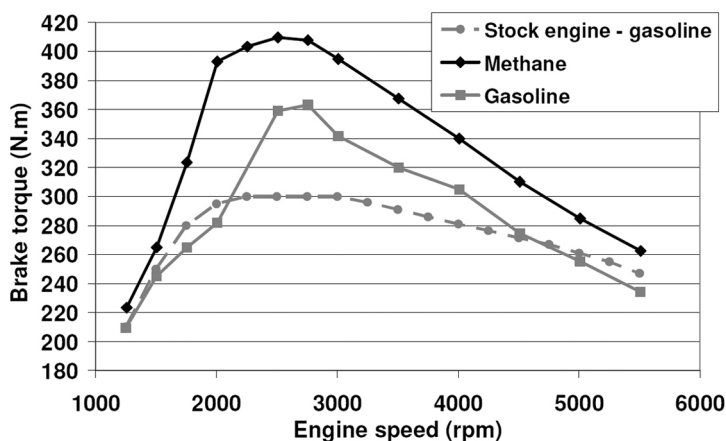


Figure 2. Full load torque in gasoline and methane modes

With gasoline, only a 5% decrease in performance below 2000 rpm was observed compared to the stock engine, given the knock sensitivity of this configuration. From 2500 rpm to maximum speed, the torque was higher than with the stock gasoline engine, with a 363 N.m MBT at 2750 rpm, highlighting the capability of the combustion chamber design to promote fast combustion rate, with heat releases close to $0.05\text{ }^{\circ}\text{CA}^{-1}$.

With methane, despite a lower volumetric efficiency, torque was significantly increased: 410 N.m torque (25.8 bar BMEP) was reached at 2500 rpm, with a maximum power output close to 150 kW from 4500 rpm to 5500 rpm.

These differences in performances between methane and gasoline are explained by the high equivalent octane number of methane that totally suppressed engine knock in this configuration. Therefore, MBT spark advances could be applied and significant boost pressures were used. It drastically improved the volumetric efficiency until the maximum turbocharger speed was reached (at 2250 rpm). Indeed, cylinder fill-up is reduced when methane is used in naturally aspirated engines. As a consequence, a stoichiometric air-fuel ratio could be maintained thus leading to a very good cBSFC (Fig. 3, Fig. 4).

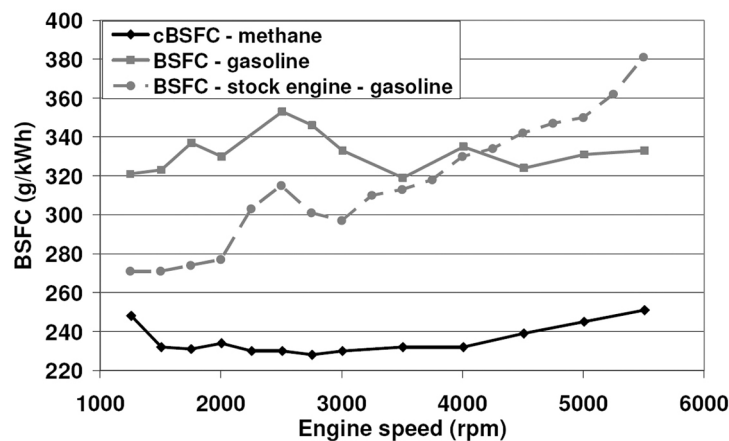


Figure 3. BSFC at full load in gasoline and methane modes

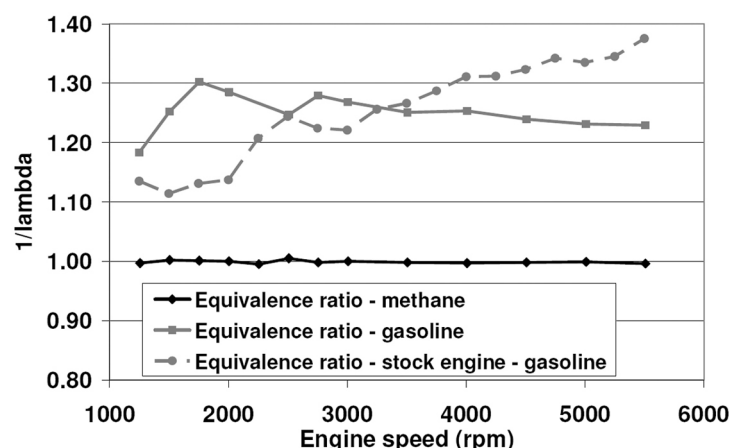


Figure 4. Equivalence ratio at full load in gasoline and methane modes

With gasoline, spark retard and enrichment had to be applied to mitigate engine knock and stability at low speed and to control the exhaust gas temperature at high speed (Fig. 5). As a consequence, the BSFC was approximately 30% higher than with methane.

It is important to underline that the differences between the stock engine and the prototype one with gasoline are explained by the differences between the technical specifications, principally the compression ratio.

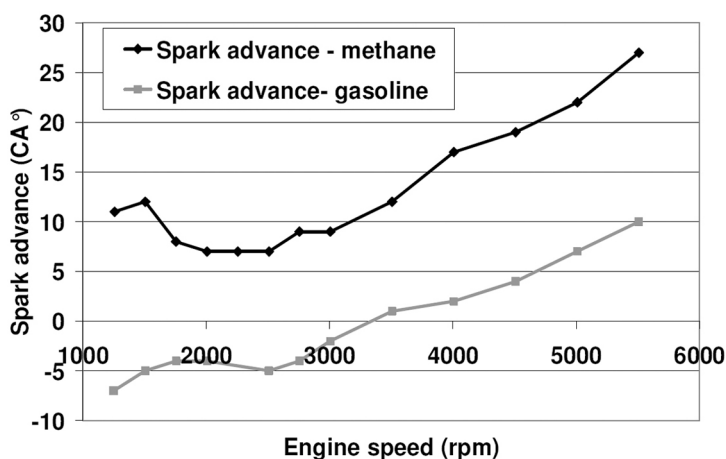


Figure 5. Spark timing at full load in gasoline and methane modes

These full load tests were taken as reference for bi-fuel mode optimization.

CONCOMITANT INJECTION OF METHANE AND GASOLINE

Different tests with concomitant injection of methane and gasoline (called “bi-fuel mode” in this paper) were carried out and compared to the above single fuel reference tests.

A first step consisted in understanding the influence of the liquid and gas fuel distribution. Then, different full load curves were acquired with different strategies. The first strategy consisted in operating the engine in continuous stoichiometric conditions and determining the gasoline/methane mass ratio which would lead to the maximum brake torque. The second strategy consisted in operating the engine with the minimum amount of methane needed to mitigate engine knock in order to achieve specific performance targets.

INFLUENCE OF FUEL DISTRIBUTION

In order to evaluate the potential of the concomitant injection concept, a fuel ratio variation was carried out at full load at 1750 rpm. The waste-gate of the turbocharger was totally closed, and spark advance was set to MBT. The VVT was used to have an IVC of 8° ABDC in order to maximize the scavenging effect and the volumetric efficiency. This IVC value was kept constant during these tests.

The test was started with 100% methane at $1/\lambda=1$, and gasoline was added little by little. The torque increased with the addition of gasoline and reached a maximum value for 50% gasoline. The torque increase was close to 50 N.m (+15%) on this point (Fig. 6).

It is important to underline that the addition of gasoline led to a progressive appearance of engine knock from 30% gasoline. It was possible to maintain spark timing at MBT up to this fuel ratio (Fig. 7). Nevertheless, for a higher gasoline content, the engine knock forced to retard the spark timing but it didn't have a significant impact on torque decrease. In fact, the volumetric efficiency (Fig. 8) was enhanced with gasoline addition as a consequence of two combined effects: the higher density of the liquid fuel whose injection allowed a part of natural gas to be replaced by air, and the cooling of the air/fuel mixture due to the latent heat of vaporization of gasoline.

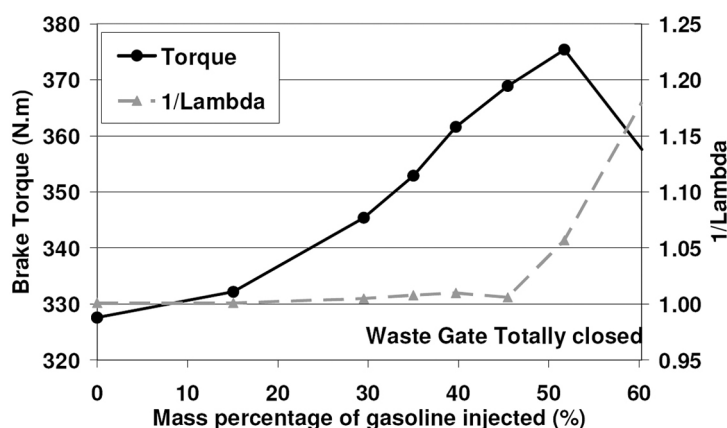


Figure 6. Fuel ratio variation at 1750 rpm at full load - impact on torque and 1/Lambda

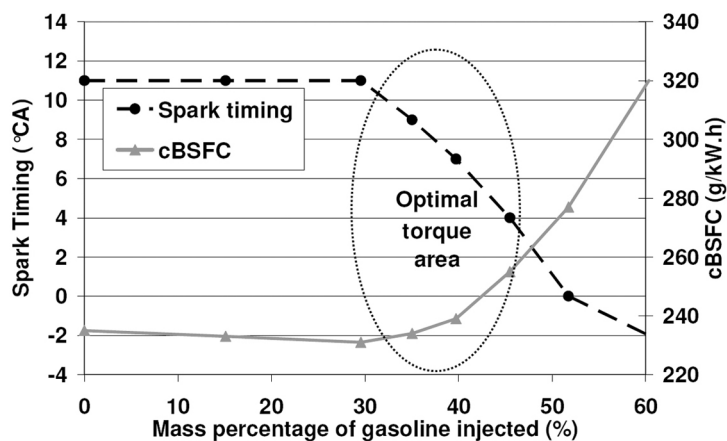


Figure 7. Fuel ratio variation at 1750 rpm at full load - spark timing, and cBSFC