

A Comparative Low Speed Pre-Ignition (LSPI) Study in Downsized SI Gasoline and CI Diesel-Methane Dual Fuel Engines

Jean-Marc Zaccardi and David Serrano
IFP Energies Nouvelles

ABSTRACT

Low speed pre-ignition (LSPI) in downsized spark-ignition engines has been studied for more than a decade but no definitive explanation has been found regarding the exact sources of auto-ignition. No single mechanism can explain all the occurrences of LSPI and that each engine should be considered as a particular case supporting different conditions for auto-ignition.

In a different context, dual fuel Diesel-Methane engines have been more recently studied in large to medium bore compression ignition engines. However, if Dual Fuel combustion is less knock sensitive, LSPI remains one of the main limitations of low-end torque also for dual fuel engines. Indeed, in some cases, premature ignition of CNG can be observed before the Diesel pilot injection as LSPI can classically be observed before the spark in gasoline engines.

This article aims at highlighting the similarities and discrepancies between LSPI phenomena in SI gasoline and dual fuel engines. The comparison of both cases bring a novel perspective on some proposed LSPI mechanisms in literature and help in understanding possible conflicting theories.

The first essential characteristic of dual fuel engines lies in the use of a gaseous fuel (port-injected in the studied case). The mixture homogeneity within the combustion chamber is thus expected to be much better than with a liquid fuel. Compared to classical direct injection gasoline engines, fuel impacts on the walls don't exist and fuel / lubricant interactions must be reassessed as if to explain LSPI.

The high octane number and auto-ignition delay times of methane must also be considered but counterbalanced by the high compression ratios used. Critical conditions for auto-ignitive LSPI can thus be compared for SI and dual fuel engines.

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INTRODUCTION

Nowadays, Diesel Compression Ignited (CI) engines are commonly used for transportation in Europe because they offer higher fuel efficiency and lower CO₂ emissions than Spark Ignited (SI) engines at equal rated power. However, stringent emission regulations on NO_x and PM are making after-treatment systems for Diesel CI engines expensive and more and more complex. That is why alternative paths are explored like the dual fuel concept.

The general principle consists in initiating the combustion of a main fuel with a pilot injection of a secondary fuel characterized by a high cetane number like Diesel fuel. The main fuel should have therefore a high octane number (like gasoline, ethanol or methane) to maintain the high compression ratio of CI engines,

synonym of high thermal efficiency. Such concepts have already been well investigated by IFPEN with the combinations of Diesel / gasoline [1] and Diesel / methane fuels [2][3].

The concept of dual fuel Diesel-methane is well known for example in heavy duty applications as the high octane number of methane allows converting easily existing CI engines without varying compression ratio, thus reducing adaptation costs. However, only limited Diesel substitution ratios between 30 to 70 % [4] are used on this type of heavy duty applications while recent studies have shown that dual fuel combustion is possible over a large operating range with high Diesel fuel substitution ratios, up to 99 % [2][3][5] allowing high CO₂ savings. In this perspective, IFPEN has applied the dual fuel concept to a passenger car multi-cylinder engine [3]. The first objective of this approach was to lower consumption compared to bi-fuel gasoline/methane SI engines by approaching CI

engine efficiency. The second aim was to reduce tailpipe emissions compared to Diesel CI engines by using simple after-treatment (like three-way catalyst). In this purpose, from mid to high loads, dual fuel combustion is operated at stoichiometry with huge CO₂ benefits beyond theoretical Diesel full substitution by methane. Indeed, CO₂ has been reduced by up to ~28% and fuel consumption by up to 7% at full load in dual fuel conditions compared to pure Diesel operation [3]. From low to mid loads, dual fuel combustion is then operated at lean burn with EGR.

Moreover, among the different concepts for alternative efficient engines, implementing a dual fuel approach seems easier in an engine than complex kinematic like for instance multi-link engine. It requires “only” an additional injection system (injector, rail, tank and security valves) for the gaseous fuel. However, dual fuel combustion suffers from major combustion issues that need to be addressed. At low loads, significant unburned methane emissions can result from both combustion chamber geometry and combustion conditions. Indeed, dual fuel engines are generally derived from Diesel engines using pistons with high crevices that can trap port-injected methane (especially high top land heights that are required because of steel inserts and cooling oil galleries [6]). Emissions and efficiency also depend on flow motion inside the combustion chamber: Carlucci et al. [7] have compared swirl and tumble motion and they found that the charge bulk flow induced by a swirl inlet improve the charge mixing of the Diesel spray and the partial propagation of the turbulent flame fronts on the air-methane mixture. Thus, the oxidation of the air-methane regions more distant from the pilot ignition nuclei is enhanced : HC emissions are then drastically reduced and combustion duration is shortened. Besides, if the methane / air mixture is too lean, meaning out of flammability limits, combustion misfiring and flame quenching both on walls and bulk can occur, causing high unburned HC emissions. Ishiyama et al. [8] and Konigsson et al. [5][6][9] gave lean limits values for equivalence ratio around 0.45 for avoiding misfires and around 0.55 for reasonable unburned HC emissions. It seems that clean dual fuel combustion is a challenging task. Many scientists have tried to list the effects of all engine parameters like the Diesel pilot Start Of Injection (SOI), Diesel rail pressure or Exhaust Gas Recirculation (EGR) rate [4][5][8].

Other indirect issues exist: some recent studies have investigated the Diesel injector fouling issue caused by high temperatures reached by the injector nozzle [9]. Indeed, the reduction in Diesel flow through the nozzle decreases heat dissipation, thus inducing high temperatures and causing the formation of deposits inside the nozzle. This item remains a key issue for future industrialization of dual fuel solutions. At high loads, maximal performance can be reduced by the occurrence of knock [10] but also of Low-Speed Pre-Ignition (LSPI) [3][11][12], thus limiting the compression ratios that can be used compared to the Diesel baseline engines.

LSPI has been widely observed on highly charged SI gasoline engines over the last 10 years. It has sometimes been referred to as Super-Knock [13] or Mega-Knock [14] but more often as LSPI [15]. In fact, “mega-knock” pressure oscillations appear because of the premature and uncontrolled ignition. The first occurrences of this abnormal combustion dates back to the very beginning of years 2000 in Europe but surprisingly, there is still no consensus on the exact causes of the irregular auto-ignition phenomenon. Considering the great variety of engines, fuels and operating conditions used for the reported analysis of LSPI, it is however not surprising that different hypotheses emerged. In fact, the authors are deeply convinced that there has never been only one cause to LSPI and that each engine should rather be considered as a particular case of the more general theory of auto-ignition [16]. The increase in temperature and pressure levels with load explains indeed the highly unstable state of the mixture at the end of the compression stroke. In these conditions, spatial, cycle-to-cycle and cylinder-to-cylinder fluctuations can easily unbalance the system and lead to uncontrolled auto-ignitions.

The experimental study of this last form of irregular combustion is complicated due to its variable occurrence mode. It is generally a rare and occasional event but it can also appear on a regular basis and even sometimes alternatively with normal combustions [17]. Defining relevant test and quantification methodologies and performing reliable analyses are challenging as LSPI events are usually very violent and cannot be controlled as simply as knocking combustion. Moreover, due to its intensity in terms of maximal cylinder pressure, LSPI is now widely seen as a crucial issue and a strong limitation, for the downsizing of SI engines [18].

This article aims at highlighting the similarities and discrepancies between LSPI phenomena in SI gasoline and dual fuel Diesel-methane engines. Indeed, basic considerations on hardware configurations, mixture formation processes and ignition in both types of engines can first enhance our understanding of uncontrolled auto-ignition at high load. Then, the detailed comparison of events recorded in both cases can bring a novel perspective on some proposed LSPI mechanisms and help in understanding possible conflicting theories.

To this end, experimental results recorded on single and multi-cylinder dual fuel engines will be compared with previously obtained results on SI engines. Engines and fuels used for this analysis are first introduced in experimental setup section. Then, the next part gives an overview of the combustion processes followed during normal operation and pre-ignition in Diesel-methane fuelled engines. The following section is dedicated to a comparative analysis of LSPI events recorded on SI and dual fuel engines during parametric and hardware variations. This comparative analysis will also be completed with a literature review discussing the main LSPI mechanisms. Last section finally points out the most probable LSPI mechanism associated with dual fuel engines.

EXPERIMENTAL SETUP

Engines

Results presented in this article have been obtained on three different engines: a multi-cylinder SI engine and two Diesel engines used for dual fuel operation (see [Table 1](#)).

Table 1. Dual fuel and SI engines specifications

	PSA DV6d 4-cylinder	IFPEN Single cylinder	PSA EP6Dt 4-cylinder
Combustion	CI & Dual fuel	CI & Dual fuel	SI
Displacement	1560 cm ³	499 cm ³	1598 cm ³
Bore x Stroke	75 x 88.3 mm	85 x 88 mm	77 x 85.8 mm
CR	16:1	17:1	10.5:1
Connecting rod length	136.80 mm	145.00 mm	138.54 mm
Valves / cyl	2	4	4
IVO @ 1mm lift	17 CAD ATDC	33 CAD ATDC	VVT
IVC @ 1mm lift	10 CAD ABDC	30 CAD ABDC	VVT
EVO @ 1 mm lift	16 CAD BBDC	23 CAD BBDC	22.5 CAD BBDC
EVC @ 1 mm lift	10 CAD BTDC	17 CAD BTDC	7.5 CAD BTDC
Diesel FIS (DI)	Bosch CRI2.2 (400bar)	Bosch CRI3.1 (400bar)	
Gas FIS (PFI)	Bosch NGI (9bar abs.)	Landi Renzo (3 bar abs.)	
Gasoline FIS (DI)			Bosch HDEV5 (<200bar)
EGR	Cooled HP	Cooled HP	Cooled LP

As for dual fuel engines, the tests have been carried out on a single cylinder engine (displacement of 499 cm³) and on 4-cylinder PSA 1560 cm³ DV6d Diesel engine (80 kW Euro 6 variant using a variable nozzle turbine), both being originally designed for Diesel fuel operation. Methane could be injected directly but in our case port fuel injection was preferred so as to keep possible full Diesel operation. That way, only minor changes were necessary on the baseline engines to allow dual fuel operation. Gas injection systems were implemented without modifying the global engines packaging. In both cases, the gas is injected into the intake duct of each cylinder with Start Of Injection (SOI) set at 360 CAD before combustion Top Dead Center (TDC), just upstream of the inlet valves. Thus, methane is assumed to be homogeneously distributed in each cylinder.

For all the dual fuel tests, the Diesel injection strategy was simplified by keeping only one sole injection. The injection duration was kept as low as possible in order to maintain the lowest and stable Diesel pilot quantity. The injection pressure was set to 400 bar for all the tests. The Diesel mass fraction was then rather low (around 1 to 6 % except for very low loads). The equivalence ratio ϕ was regulated via the measurement of a wideband lambda sensor and by adjusting only the methane injection duration to maintain the selected equivalence ratio. As a remark, each time ϕ is mentioned in this article, it refers to the in-cylinder global equivalence ratio including the Diesel and methane fuels. The ratio of each fuel is detailed through the mention of Diesel and methane mass fractions.

Several configurations were also tested on each of these engines (in terms of compression ratios or swirl motions). Regarding SI engines, results presented in this article have been recorded on a serial production turbocharged engine (displacement 1598 cm³, see [Table 1](#)) being run most of the time with direct injection in lateral position. Different configurations have been tested in terms of injection strategy, charge motion (variable tumble level) and dilution (cooled low pressure EGR). For all of these configurations, optimal injection and ignition settings were determined to minimize exhaust emissions and to optimize the combustion timing.

No after-treatment system was used on these engines. Therefore, to be representative of full production exhaust lines, the exhaust back-pressure was adjusted with a dedicated throttle (the same approach was followed for the single cylinder engine fed by an external sonic nozzle system).

Table 2. Limits of engine operation for multi-cylinder engines (exhaust temperature limited at 750 °C for the single cylinder engine)

	Dual Fuel engines	SI engine
Average max. cylinder pressure	≤ 160 bar	≤ 120 bar
CoV of IMEP	≤ 3 %	≤ 3 %
Upstream turbine temperature	≤ 810°C	≤ 950°C
Downstream turbine temperature	≤ 700°C	≤ 850°C

Safety and stability limits were also set to protect the engine integrity during the tests (at least during normal operation at full load, see [Table 2](#)). Regarding dual fuel engines, these constraints are not different from Diesel operation, as no major modification of engine hardware were made.

The temperature limits were not reached with the multi-cylinder SI engine at low engine speeds, even for loads greater than the baseline serial production engine. However, regarding dual fuel engines, the Diesel SOI is a critical parameter and must be adjusted to avoid exceeding both the maximal pressure limit (reached for early injections) and the temperature limit (reached for late injections).

Fuels

The multi-cylinder SI engine was run with a commercially available E5 fuel (RON 97) and dual fuel engines were run with an association of pure methane and a standard EN590 European Diesel fuel (B5 and cetane number > 51).

As shown by [Figures 1](#) and [2](#), auto-ignition delay times τ_i of methane are more than ten times greater than those of gasoline (represented here by a ternary reference fuel TRF consisting of n-heptane, iso-octane and toluene). The higher auto-ignition delay and octane number of methane (~130) allow maintaining knock-free operation over a broader range of engine speeds and loads than with gasoline (Fig. 1).

compression ratios can also be used while reaching higher maximal brake torque and keeping optimal combustion timings at high loads.

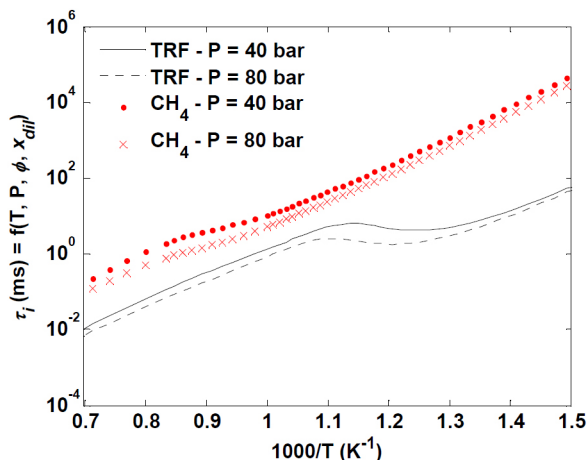


Figure 1. Calculated auto-ignition delays times τ_i for $\Phi = 0.7$ and $x_{dil} = 0$ (Chemkin calculations)

It can also be noted that methane presents great advantages compared to other high octane number fuels. Methane is notably the lowest carbon content fossil fuel, and reduces significantly CO_2 emissions compared to Diesel fuel (~26 % theoretically and more if it comes from a renewable source like biogas).

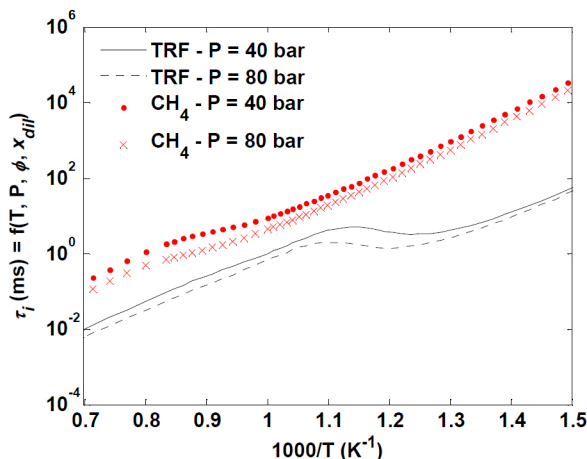


Figure 2. Calculated auto-ignition delays times τ_i for $\Phi = 1.0$ and $x_{dil} = 0$ (Chemkin calculations)

Measurements and Control

Conventional measurements of temperature and pressure were made on the coolant, oil, fuel, intake and exhaust paths. Fast pressure transducers were also used to record the instantaneous intake and exhaust pressures. Cylinder pressure has been recorded with different angular resolutions: from 0.1 CAD for analyzing knocking combustion, up to 1 CAD when statistical analysis over several minutes was required.

The engines were controlled with IFPEN in-house management system devices. These fully by-path systems allowed flexible engine control of all engine devices and notably considering that all of the engines have to use two injection systems (methane and Diesel for dual fuel engines, PFI and DI for SI engines).

GENERAL OVERVIEW OF THE COMBUSTION PROCESS

Regular Dual Fuel Combustion

The dual fuel concept has been applied by minimizing the Diesel fuel mass fraction (pilot injection size) and thus maximizing methane proportion for the whole engine operating range. In these conditions, the dual fuel combustion process can be described by four different steps:

1. The homogeneous carbureted mixture of air and methane (and sometimes with EGR too) is compressed but should not auto-ignite because of its high auto-ignition delay.
2. The combustion is then initiated by the direct injection of a Diesel fuel micro pilot quantity which auto-ignites during the end of the compression stroke. Diesel fuel is then only used as a high energy chemical ignition source and Diesel SOI is the trigger of the dual fuel combustion (excluding Diesel fuel auto-ignition).
3. Diesel combustion occurs (which can be composed by premixed and diffusion phases) with part of the mixed methane due to the mixture entrainment into the Diesel spray.
4. Methane combustion around the Diesel fuel spray [2] occurs. This last step involves complex mechanisms (including multi-propagation flames) that are currently under investigations at IFPEN [19] with optical engine.

LSPI in Dual Fuel Engines

Critical operating range

Similarly to SI gasoline engines, LSPI generally appears on dual fuel engines between 1500 and 2500 rpm and for IMEP above 20 bar. Lower engine speeds may also be concerned if high low-end torques are targeted.

Figure 3 gives an illustrative example of full load curve obtained on a specific configuration of the 4-cylinder PSA DV6d engine running on dual fuel operation. Black-filled symbols indicate the operating points on which LSPI could be recorded, and occurrence frequencies are also mentioned.

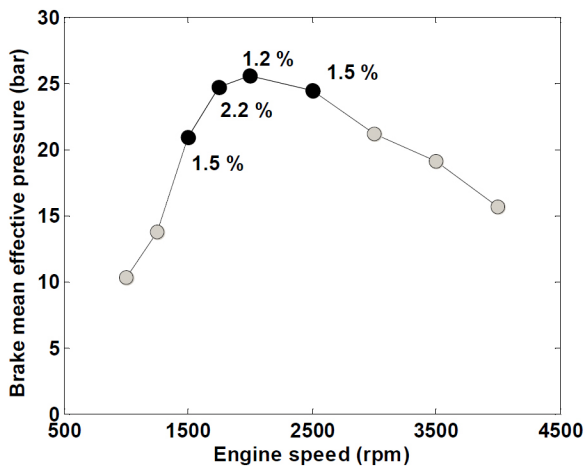


Figure 3. Exemplary full load curve ($\Phi = 1$, 0 % EGR, Diesel fuel mass fraction between 2.5 and 4.5 %)

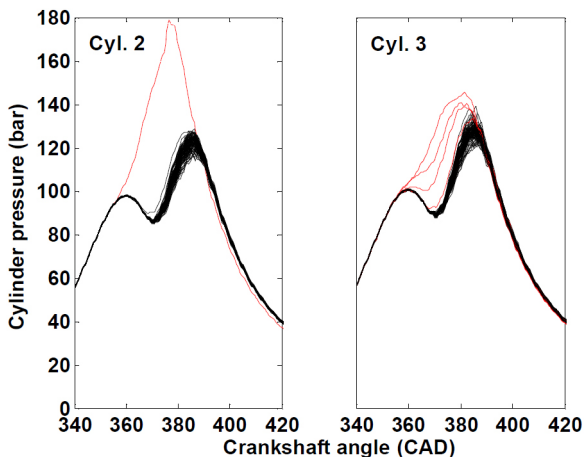


Figure 4. LSPI events recorded on cylinders n° 2 & 3 (1750 rpm, 25.4 bar IMEP, $\Phi = 1$, Diesel SOI = 362 CAD, 97.5 % methane mass fraction, 2.5 % Diesel fuel)

As already observed in SI engines, no specific pattern of occurrence can be identified in dual fuel engines. Indeed, LSPI can occur within sequences alternating with regular combustions, or as isolated events. It can also be mentioned that, until now, no cycle-to-cycle determinism has been identified in the occurrence of LSPI which remains thus a pure stochastic event. As it will be shown in next sections, causal relations can be found between mean operating conditions and LSPI occurrence frequency both in SI and dual fuel engines, but no direct determinism between successive cycles can explain the mixture premature auto-ignition [20].

LSPI Combustion Process

The LSPI combustion process in dual fuel engines is quite similar to the one that can be observed in SI engines. Indeed, a three steps combustion process can be identified as illustrated by the example in Figures 5 and 6.

1. First, the beginning of combustion can be observed before the Diesel pilot injection (as it starts before the spark in SI engines). It means that some heterogeneities, or hot

spots, within the mixture have sufficient energy to trigger an auto-ignition. This first step represents the critical characteristic of LSPI since, as for SI engines, several mechanisms can be assumed to explain how the mixture's reactivity can be enhanced to the point of being able to trigger an auto-ignition.

2. After the first auto-ignition of fresh gases, a second phase similar to a classical flame front propagation can be observed. The heat release rate associated with this second step is indeed comparable to the one obtained during normal combustion.
3. Finally, a third phase can be observed in the form of a bulk auto-ignition. The fresh gases remaining in the combustion chamber are indeed highly compressed and heated up and, therefore, a second knocking-like auto-ignition occurs. Depending on the combustion timing and on the load, this auto-ignition can lead to severe engine damages. As a remark, it can also be mentioned that these knock oscillation frequencies do not seem to be related to the engine bore as for SI engines but more to the piston bowl diameter.

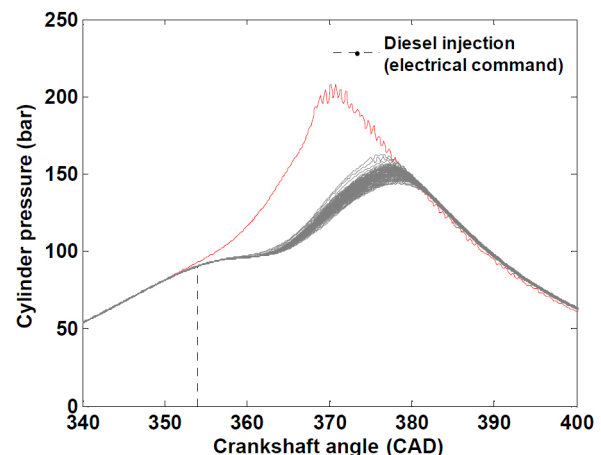


Figure 5. Typical cylinder pressure recorded on the single cylinder dual fuel engine (2000 rpm, 21 bar IMEP, Diesel SOI 354 CAD, 98 % methane, 2 % Diesel fuel, $\Phi = 0.85$)

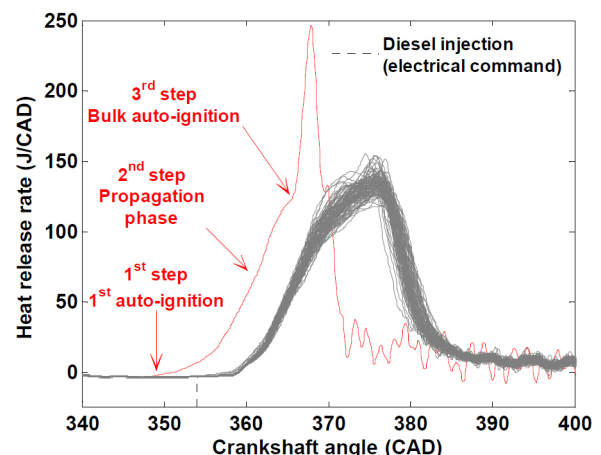


Figure 6. Heat release rates calculated for cylinder pressure signals illustrated in Figure 5