

Numerical and Experimental Investigation of Fuel Effects on Knock Occurrence and Combustion Noise in a 2-Stroke Engine

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ABSTRACT

Knock occurrence is a widely recognized phenomenon to be controlled during the development and optimization of S.I. engines, since it bounds both compression ratio and spark advance, hence reducing the potential in gaining a lower fuel consumption. As a consequence, a clear understanding of the engine parameters affecting the onset of auto-ignition is mandatory for the engine setup. In view of the complexity of the phenomena, the use of combined experimental and numerical investigations is very promising. The paper reports such a combined activity, targeted at characterizing the combustion behavior of a small unit displacement two-stroke SI engine operated with either Gasoline or Natural Gas (CNG).

In the paper, detailed multi-cycle 3D-CFD analyses, starting for preliminary 1D computed boundary conditions, are performed to accurately characterize the engine behavior in terms of scavenging efficiency and combustion. In order to assess the accuracy of the adopted numerical approach, comparisons between numerical forecasts and experimental measurements of instantaneous in-cylinder pressure histories are carried out for both gasoline- and CNG-fueled engine operations. 3D analyses are also used to investigate the knock sensitivity of the engine to variations of spark timings in a limited set of operating conditions.

The activity is simultaneously developed within a 1D modeling framework, where a detailed quasi-dimensional combustion and knock model is applied to perform a wider investigation of engine performance and knock occurrence for both Gasoline and Natural Gas fuelling. Results from 3D simulations are here used to improve the 1D simulations through a better description of scavenging and combustion processes.

Once validated, 1D analyses are in particular finalized to find the knock-limited spark advance by changing both compression ratio and spark timing in order to reduce the fuel consumption. In this phase, a dedicated routine is also developed to have information on combustion related noise, which may limit fuel consumption improvements. Further confirmations on the validity of the 1D approach to the modeling of the knock onset are derived from full-3D knocking analyses over a limited set of engine operating conditions.

Advantages and limitations of CNG operations of the engine are briefly pointed out at the end of the paper.

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INTRODUCTION

Alternative-fueled engines are undergoing a constant growth of interest in the global transportation market as a feasible response to the increase of both pollutant emission limitations and conventional fuel price. In fact, most governments are adopting increasingly limiting regulations on ICE pollutants and fuel consumption as a response to the global warming concern and the connection between hydrocarbon consumption and greenhouse gas emissions. Different “alternative-fuels”, ranging from bio-fuels to low-carbon fuels up to hydrogen, are extensively investigated in order to find possible solutions to the above mentioned problems. Among the available options, Compressed Natural Gas (CNG) is one of the most promising possibilities, since it combines low emission levels (mainly CO, CO₂ and highly reactive hydrocarbons) to reduced running costs.

The growing interest towards CNG fueled engines is particularly evident in recently exploded mass transportation markets such as India and south-east Asia, where the environmental issue and the need for low-cost mobility is of crucial importance. In such countries, two-wheel and three-wheel vehicle production has been expanding rapidly over the past several years especially in the urbanized areas. In fact, motorcycles can play an important role in fulfilling both personal and commercial transportation needs in view of the smaller physical size of vehicles, allowing for easier navigation in heavily congested areas. Furthermore, purchase and maintenance are substantially lower than the corresponding costs for even small automobiles, making the types of low-cost motorcycles the most economically efficient transportation option [1].

A major drawback of low-cost motorcycles is that they emit substantial quantities of hydrocarbons (HCs), carbon monoxide (CO), and particulate matter (PM). These pollutants are widely recognized to have significant adverse health effects and deteriorate environmental quality. To address the serious pollution problems posed by two-wheel vehicles, a growing number of countries worldwide have implemented, or are in the process of implementing, motor vehicle pollution control programs aimed at substantially reducing harmful emissions from spark-ignited two-wheel vehicles, as demonstrated by the research efforts paid in this field by important manufacturers [2,3,4].

While waiting for mid-to-long-term fashionable solutions such as electric or hybrid electric motorcycles, CNG appears today to be one of the most promising short-term / low-cost technologies to reduce emission levels while limiting production and running costs.

This is confirmed by the high population of three-wheeled, CNG powered motorcycles in certain Indian cities and some south-east Asia countries as a response to air quality-driven clean-fuel requirements; further diffusion of CNG fueled vehicles on the market is primarily limited by the presence of a capillary network of fueling stations.

Many outstanding contributions dealing either with the development of specifically CNG-fueled engines or with the conversion to CNG of existing units for automotive applications can be found in literature, while only a few of them deal with two-stroke engines for motorcycles or mopeds [5, 6]. The utilization of the NG within such engines can lead to relevant improvements in NMHC, NO_x and CO emissions, despite some challenges can be found in evaporated fuel and unburned HCs. Nevertheless, properly optimized CNG vehicles show potential benefits in both emissions and fuel consumption with reduced cost increase and design modification with respect to, for example, the adoption of more expensive technologies such as direct-injection and/or supercharging.

The development of strategies and technologies for the specific engine application without increasing the R&D costs can profit from robust and accurate computational simulations allowing the researchers to understand and optimize the fundamental processes governing the engine operation. A wide set of both 1D and 3D numerical procedures can be found in literature aiming at carefully modeling the scavenging and combustion processes, as reported for example in [7, 8].

On one hand, 1D tools can be extensively used to assess the influence of both operating and design parameters on engine performance from a global perspective. 1D simulation reduced computational costs allow the researchers to quickly address the global effects of a wide range of solutions and strategies on the overall engine performance; furthermore, 1D tools can provide time-history of pressure and temperature to be used as boundary conditions by the more complex 3D analyses.

On the other hand, full-3D analyses are mandatory whenever point wise flow patterns and geometrical details are needed to accurately understand and optimize the engine behavior. 3D simulation proves to be useful also both in providing 1D simulations input parameters which are difficult to measure at the engine bench and in exploring engine optimization paths which could lead to engine damage at the engine testbed.

Despite coupled 1D-3D simulations are nowadays growing in popularity in the research community, the high computational costs typical of the 3D simulations and the numerical issues arising at the 1D/3D interfaces limit their extensive application to the late development stage.

The paper reports results from a combined experimental and numerical investigation of a two-stroke engine for motorcycle applications. Measurements are used to tune and validate an in-house developed combustion model included in a commercial 1D code, and a commercial CFD tool for the full-3D multicycle analysis of scavenging and combustion.

A preliminary investigation of some engine operations for the CNG-fueled engine are reported in [9], where the advantages in terms of knock resistance of the CNG vs. gasoline were highlighted. The analysis is now extended and improved:

- 1D and 3D data are better integrated and iteratively exchanged in order to improve both the simulation realms;
- a wider set of engine operations, now including gasoline-fueled ones at different engine speeds, is modeled in order to better investigate the effects of the fuel on both combustion evolution and engine performance;
- knock-limit of the engine is further addressed for gasoline fuel; a knock-free operation of the tested engine is confirmed for CNG fuelling;
- a procedure to characterize the combustion related noise of the engine is developed in order to better address the operative limits of the engine.

The numerical models (1D, 3D and combustion noise) are finally used to investigate the engine operative limits under both gasoline and CNG operations. In particular, in order to improve CNG fuelled engine performance, the higher knock-resistance of CNG fuel is exploited through spark advance variations and compression ratio increase. Potential benefits of the adoption of CNG are also verified in terms of combustion related noise.

EXPERIMENTAL ANALYSES

The Engine

As reported in [9], experimental investigations are carried out on a crankcase-scavenged, carbureted, spark-ignited two stroke single-cylinder engine. The engine displacement is 50 cm³. The engine is shown in Figure 1 below, while Table 1 reports some technical specifications.

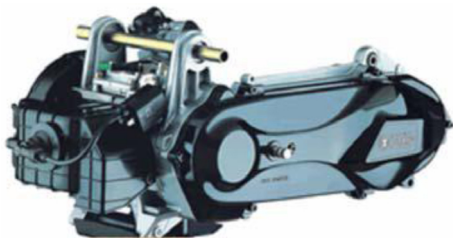


Figure 1. The Morini Engine

Table 1. Engine Specifications

Engine model	Ported Two-Stroke SI
Scavenging type	Schnurle
Displacement	50 cm ³
Compression ratio	11.7:1
Bore, Stroke	41, 37.4 mm
Maximum Power	4.1 kW @8000 rpm
Maximum Torque	4.7 Nm @7500 rpm
Intake Port Open/Number/Width	56° BBDC / 5 / 11 mm
Exhaust Port Open/Number/Width	81° BBDC / 1 / 26 mm

Experimental apparatus

The engine is installed on a DC test bench; both instantaneous and overall sensors are used. Location of the

sensors and the relative legend are shown in Figure 2 and Table 2 respectively.

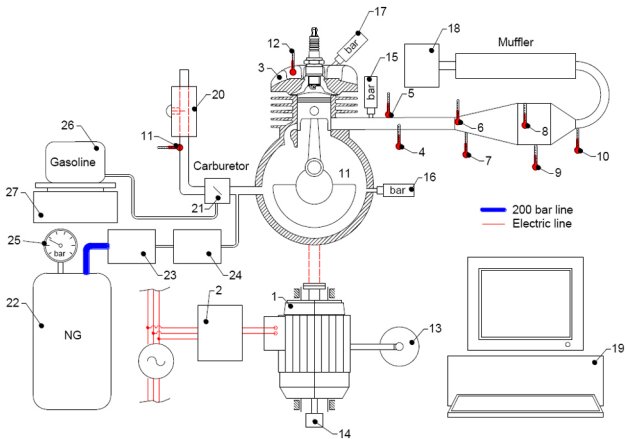


Figure 2. Experimental testbed scheme.

Instantaneous pressures inside the crankcase, the cylinder and the exhaust pipe are measured with piezoquartz-transducers at engine speeds ranging from 4000 to 7000 rpm at WOT conditions. To ensure the right functioning of the exhaust piezo-transducer, it is also cooled through a custom-made water circuit. Additional experimental data are collected like: air mass flow rate, torque, speed, fuel flow rate, gas and wall temperatures in a number of sections along the exhaust system. All temperatures are acquired by K type thermocouples.

Table 2. Sensors Legend.

1	DC Electric motor
2	Central Command electric motor
3	Engine
4-12	Thermocouples
13	Force transducer
14	Encoder
15-16	Fast piezoresistive pressure transducer
17	Miniature dynamic pressure transducers
18	Exhaust Analyzer
19	PC + DAQ Card
20	Hot wire anemometer
21	Angular position sensor
22	Gas Tank
23	Natural Gas pressure reducer
24	Natural gas mass flow meter
25	Pressure Gauge
26	Petrol tank
27	Balance

The air mass flow rate measurement is carried out by a customized hot film anemometer. The sensor and the electronic circuit of a commercial anemometer are placed into a new suitable custom-made structure in order to ensure accurate measurement despite the low air flow rate. The device is calibrated with stationary tests in comparison with a more accurate mass flow rate measurement.

At each operation point, the engine is fuelled with both gasoline and CNG. The latter is injected in the intake pipe

upstream the crankcase, at a moderate over-pressure above the atmospheric one. NG flow rate is adjusted through an electronic control valve, incorporated in the NG mass flow rate meter (Bronkhorst F202AC).

NG tests are carried out approximately at the same air index ($\lambda = \alpha/\alpha_{st}$) provided by the standard carburetor in the gasoline tests. To reduce variations in operating conditions, the NG fuelling test is performed immediately after the gasoline one. During the test the air index is firstly checked by the hot film anemometer for the air mass flow rate; this value is then adjusted by post-processing data from exhaust gas emissions measurements.

The injection of a gaseous fuel is expected to reduce the air flow rate and power output. The latter is in particular due to the combined effects of the lower fuel flow rate and slower NG laminar flame speed. A partial compensation is due to the slightly higher lower heating value of the CNG (45471 kJ/kg) with respect to the gasoline fuel (43700 kJ/kg). The slower NG combustion also indicates the need to adapt the spark timing to the CNG fuelling. However, due to limitations of the experimental equipment, CNG tests are carried out at the same spark timing of Gasoline one.

The collected data are used to validate both 1D and 3D models. Then, in order to improve CNG operation, the validated models are employed to analyze variations in the spark timing and compression ratio. In the following, experimental data and numerical results always refer to WOT operation and same air index for gasoline and CNG.

COMPUTATIONAL METHODS

1D model - whole engine analysis

Combustion Model

1D analyses are carried out with GT-Power® commercial product [10]. A in-house developed combustion model is however utilized, exploiting the “user model” features of the GT-Power code. Differently from the standard combustion model, relying on a two-equation entrainment model [11], the developed approach applies to the “wrinkled flamelets” combustion regime. The latter is widely recognized to be the one really occurring inside internal combustion engines [12,13]. In the wrinkled flamelets regime the flame front is corrugated by the convective action of turbulent eddies which determine an increase in the flame front area and burning rate. The model, extensively described in [14,17,18,19,20] is based on a fractal schematization of the flame front surface and covers the different burning phases occurring during the overall reaction process, namely: flame initiation, laminar burning, turbulent burning, and wall combustion.

In the model, mass burning rate, dm_b/dt , is computed as a function of the flame front increase area A_T/A_L :

$$\frac{dm_b}{dt} = \rho_u A_T S_L = \rho_u \left(\frac{A_T}{A_L} \right) A_L S_L \quad (1)$$

Following the fractal geometry concepts, the above area ratio can be expressed in terms of the ratio of maximum and minimum wrinkling scales, $L_{\max}$ - $L_{\min}$, and fractal dimension D_3 :

$$\left(\frac{A_T}{A_L} \right)_{fr} = \left(\frac{L_{\max}}{L_{\min}} \right)^{D_3-2} \quad (2)$$

In eq. (2), the instantaneous flame radius and the Kolmogorov length scales are chosen as the maximum and minimum wrinkling scales, respectively. The D_3 dimension indeed depends on the ratio between the turbulence intensity u' and the laminar flame speed S_L [15]. Proper laminar flame speed correlation, holding for both Gasoline [16] and Methane [21] fuels, are included in the model.

Eq. (2) is the classical formulation of a fractal combustion model [12,13,16], while in the present paper, two main modification are introduced, both important in two-stroke engine operation. In a small-size two-stroke engine, in fact, the rotational speed may be as high as 10.000 rpm. At high engine speeds, however, the flame wrinkling determined by the turbulent flow field can be so intense to stretch and break the flame, producing a multiple connected flame front, with “islands” of unburned mixture trapped within the burned gas zone. In this case, the combustion regime enters the “distributed reaction zone”. This occurs when the “Karlovitz number”, Ka , exceeds the unity. On this basis, it is introduced a correction term in the fractal burning rate expression (2). For $Ka > 1$, the laminar flame area is slightly enhanced to account for the additional reaction zone due to the presence of the unburned mixture pockets:

$$\frac{A_T}{A_L} = \left(\frac{A_T}{A_L} \right)_{fr} \cdot \Lambda_1 \quad \Lambda_1 = (1 + w_{turb} k_{AL,1} Ka) \quad (3)$$

The following definitions of Karlovitz Number, Ka , Damkholer Number, Da , and flame thickness, δ_f , are utilized in the model:

$$Ka = \frac{(u'/S_L)^3}{(L_i/\delta_f)^2} = \frac{(u'/S_L)}{(Da)^2} \quad Da = \frac{L_i/\delta_f}{(u'/S_L)} \quad \delta_f = \frac{\nu_u}{S_L} \quad (4)$$

L_i being the integral length scale, δ_f the flame thickness and ν_u the kinematic viscosity of unburned gas zone. The correction coefficient $k_{AL,1}$ is usually in the range (0.0 - 0.1), while the weight w_{turb} in eq.(3) allows to gradually apply the correction during laminar-turbulent flame transition (w_{turb} going from 0 to 1). The above correction is activated only for $Ka > 1$, which may occur at high speeds (high u') and/or low loads (low S_L). A high residual gas fraction, typical of two-

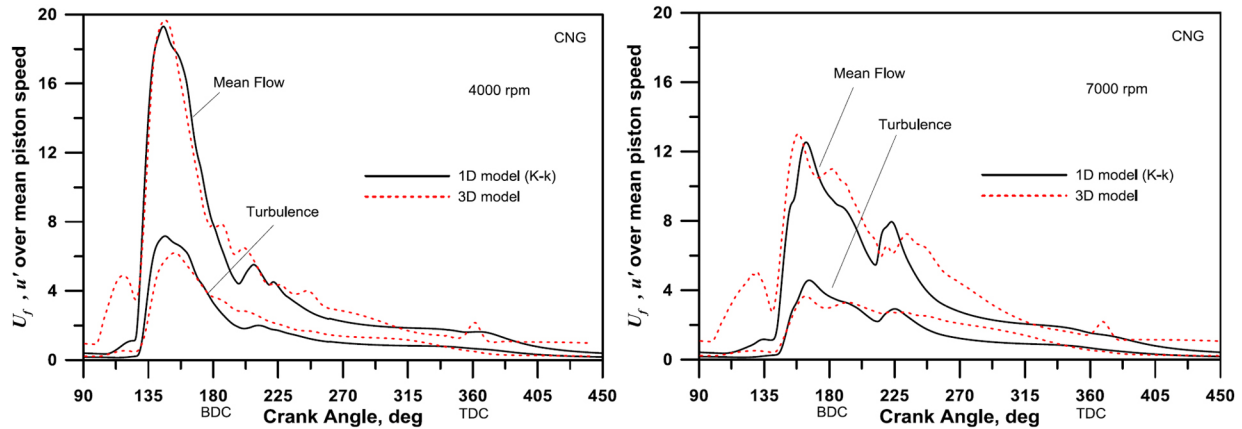


Figure 3. In-cylinder average gas velocity and turbulence intensity - 4000 and 7000 rpm, CNG fuel.

stroke engines, is also responsible for a low S_L and increased Ka .

A further modification is required in engines where a relevant flame distortion is expected. This once again occurs in a ported two-stroke engine, where the very intense mean flow motion produced during scavenging process is conserved in a greater extent at spark time. Flame distortion is expected to be higher for higher mean flow motions (U_f) and lower turbulent flame speeds (S_T). For this reason, a reasonable correction is proposed, as shown:

$$\frac{A_T}{A_L} = \left(\frac{A_T}{A_L} \right)_{fr} \cdot \Lambda_1 \cdot \Lambda_2 \quad \Lambda_2 = \left(1 + w_{turb} k_{AL,2} \frac{U_f}{S_T} \right) \quad (5)$$

$$S_T = S_L \left(\frac{A_T}{A_L} \right)_{fr} \quad (6)$$

Mean flow velocity U_f is derived by the integration of the K equation, presented in next paragraph. The tuning coefficient, $k_{AL,2}$, is of order 0.0 - 0.2.

Turbulence Model

Wrinkling scales in eq. (2) are computed as a function of the characteristics of the turbulent flow field inside the cylinder, utilizing a two-equation $K-k$ approach [22,23].

$$\begin{aligned} \frac{dK}{dt} &= c_{in} \frac{1}{2} \dot{m}_{in} u_{in}^2 + c_{ex} \frac{1}{2} \dot{m}_{ex} u_{ex}^2 - P - K \frac{\dot{m}_{ex}}{m} + c_{imbl} K \frac{\dot{P}_{cyl}}{\rho_u} \\ \frac{dk}{dt} &= P - D - k \frac{\dot{m}_{ex}}{m} + c_{imbl} k \frac{\dot{P}_{cyl}}{\rho_{cyl}} \\ P &= c_P \frac{K}{t'} \quad D = c_D \frac{k}{t'} \\ \left(K = \frac{1}{2} m U_f^2 \right) \quad \left(k = \frac{3}{2} m u'^2 \right) \quad \left(t' = \frac{L_I}{u'} \right) \quad \left(L_I = c_i \sqrt[3]{V} \right) \end{aligned} \quad (7)$$

In the above balance equations, K is the kinetic energy of the mean flow field (U_f) - whose production is mainly related

to the intake and exhaust flow rates ($\dot{m}_{in}$ and $\dot{m}_{ex}$) - k is the kinetic energy of the turbulent flow field (assumed homogeneous and isotropic), while D is its dissipation rate. P represents a turbulent production term, which characterizes the energy transfer between the mean and the turbulent flow field (*energy-cascade* mechanism). Density variation terms, in the unburned and whole cylinder zones, account for turbulence enhancement/decay due to tumble vortex compression/expansion, weighted by a tuning constant, c_{imbl} . Turbulence production and dissipation rates are defined through simple expressions, including two additional tuning constants c_P and c_D . The turbulence intensity is derived from the k definition. The integral length scale L_I is evaluated as a characteristic dimension of the instantaneous cylinder volume, and, consequently, the Kolmogorov length scale can be estimated (for isotropic turbulence), as:

$$l_k = \frac{L_I}{Re_t^{3/4}} \quad Re_t = \frac{u' L_I}{\nu_u} \quad (8)$$

To select proper values of tuning constants in eqs. (7), preliminary 3D analyses are carried out - with boundary conditions provided by the 1D code - to characterize in-cylinder mean flow velocity and turbulence levels at different engine speeds. As an example, extracted data for the 4000 and 7000 rpm engine speeds, CNG fuel, are reported in Figure 3. As visible in the plots, in-cylinder average mean gas velocity and turbulent fluctuations - non-dimensionalized respect to the mean piston speed - show a similar trend (dashed profiles), reaching a peak level toward the end of the induction stroke, and rapidly decaying during the compression. Turbulence decay is faster than mean flow energy decay, reaching a very low value a few degrees BTDC.

Figure 3 in particular show that a unique set of tuning constant for eqs. (7) is able to correctly reproduce the 3D turbulence results with a good accuracy at different engine

speeds. Similar results were obtained for different engine speeds and for gasoline fuel, as well.

Scavenging Model

As known, a classical limitation of 0D models is the possibility to correctly describe the scavenging process: the actual mixing between the incoming mixture and burned gases, and the short-circuiting at the exhaust in fact hardly depend on the in-cylinder 3D flow distribution, port geometry and operating conditions [24,25]. To handle these complex phenomena, GT-Power code includes a simplified approach, where a “Scavenging Law” is assigned in input. This profile quantifies the amount of burned products exhausted at the port as a function of the instantaneous in-cylinder residual fraction. Once again, to overcome the 0D limitations, preliminary 3D analyses are utilized and processed to derive a 3D computed scavenging law, as shown in Figure 4. The latter puts into evidence that, initially, the scavenging behaves similarly to a ‘perfect-displacement’ of burned gases, while later during the process the contemporary presence of mixing and short-circuiting phenomena determine a sudden decrease of the exhaust residual ratio. Even if the shown profile may vary with fuel type and operating conditions, it was verified that differences are negligible and a single scavenging law was actually utilized in 1D analyses at each speed and for both fuels.

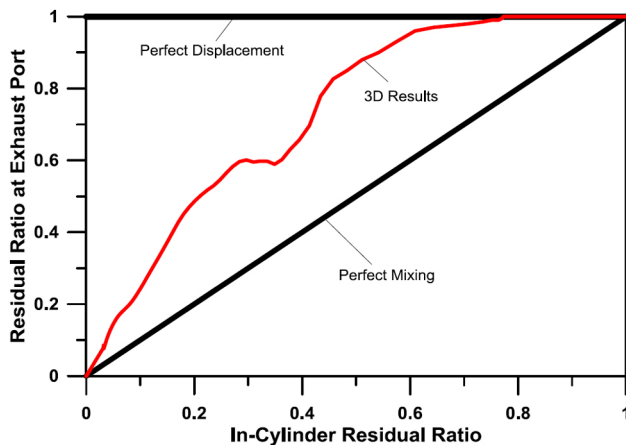


Figure 4. 3D computed scavenging law utilized in 1D analyses

1D-3D interaction

As discussed, 1D model accuracy is greatly enhanced in terms of both turbulence characteristics and scavenging profile, thanks to 3D modeling. This allows to accurately predict overall engine performance and intake and exhaust pressures to be used as boundary conditions for the 3D tool. An iterative exchange of information is hence required between the two modeling approaches. Of course, the loop between 1D and 3D simulation tools is performed until differences in terms of key-parameter predictions are negligible.

1D knock model

A further feature of the 1D model is the presence of a procedure accounting for knocking onset. It is based on the solution of a kinetic scheme in the unburned gas zone, whose thermodynamic conditions are defined by the previously described combustion model. For Gasoline operation, a reduced kinetic mechanism for the oxidation of iso-octane and n-heptane mixtures is employed. It was developed by Tanaka and Keck [26] and includes 5 elements, 32 species and 55 reactions. The mechanism handles both low and high temperature reactions and is particularly tuned to reproduce the end-gas conditions in a SI engine. For NG fuelling, the GRI-Mech 3.0 mechanism [27] is indeed utilized (5 elements, 53 species and 325 reactions).

The solution of the chemical kinetic scheme allows to compute variations on unburned species concentration (dx_i/dt) and the related heat released, dQ_{chem}/dt [9,23]. Unfortunately, the 0D treatment of the in-cylinder process doesn't allow to reproduce high-frequency knock-induced pressure oscillation. For this reason, to quantify knock occurrence and intensity, it is not possible to resort to classical knock indices (*MAPO*, *IMPO* or *DKI* - which will be indeed considered in the 3D section). Nevertheless, it is possible to define a parameter, x_{Qub} , estimating the percentage of fuel's chemical heat released in the unburned zone during autoignition, as:

$$\frac{dQ_{chem}}{dt} = -m_u \sum_{i=1}^{N_{spec}} e_i \frac{dx_i}{dt} \quad x_{Qub} = \frac{\int \frac{dQ_{chem}}{dt} dt}{m_f LHV} * 100 \quad (9)$$

Previous investigations [23] showed that a reliable threshold value for knocking onset is about 2%. In the following analyses, the spark timing is advanced until the above value is overcome, so defining the ‘knock-limited’ spark advance.

Combustion noise model

Besides the knocking onset, another important phenomenon which may limit the maximum spark timing is the combustion radiated noise. Spark advancing in fact determines an increase in pressure rise rate, directly related to the external radiated noise [28]. For this reason a procedure is developed to quantify the effects of the combustion characteristics on overall noise, for the engine under test operated with either Gasoline and Natural Gas (CNG).

Radiated engine noise, as known, is due to engine structural vibration induced by the in-cylinder pressure exciting the engine structure, and directly causing it to vibrate and radiate noise (direct combustion noise) [28,29]. Actually, the overall noise also includes a mechanical contribution (primary and secondary mechanical noise), produced by moving parts and associated with small impacts due to the crankshaft mechanism, including pistons, connecting rods, camshaft, etc. The analysis of the mechanical noise is beyond

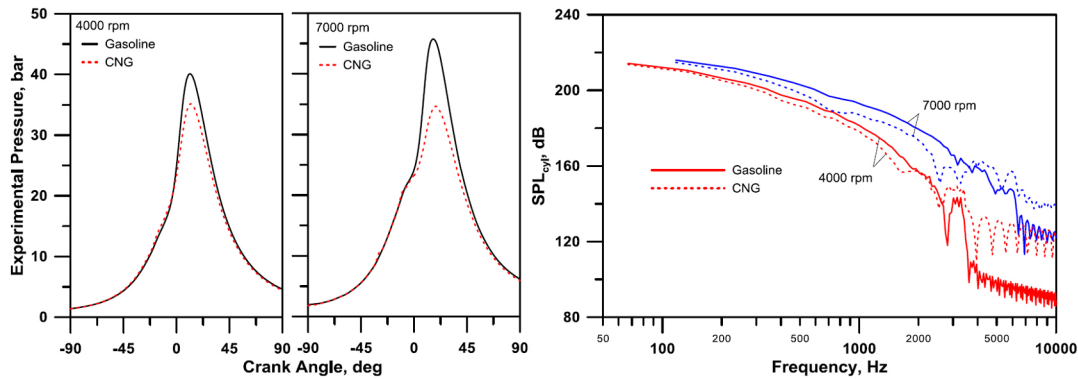


Figure 5. Experimental Pressure and SPL at 4000 to 7000 rpm, WOT operation, for Gasoline and CNG

the scope of the present analysis and it is assumed to be a constant in model.

As said, direct combustion noise is due to the rapidly fluctuating cylinder pressure forces, but it is also depending on the attenuation characteristics of the engine block. The combustion excitation is related to the shape, magnitude and repetition of the cylinder pressure rise and, consequently, any change to the combustion process, due as an example to a different injected fuel, affects the combustion noise, too. The proposed approach is based on the 'attenuation curve', in which a relationship between the in-cylinder Sound Pressure Level (*SPL*) and radiated noise level is established.

The first step of the analysis is the computation of the frequency contents of the experimentally measured pressures, through the employment of a Fast Fourier Transform (FFT). The *SPL* is then defined as:

$$SPL_{cyl} = 20 \log_{10} \left(\frac{\tilde{p}}{p_0} \right) \quad (10)$$

where p_0 is the standard reference level of $2 \cdot 10^{-5}$ Pa, and $\tilde{p}$ denotes the effective value of the pressure signal:

$$\tilde{p} = \sqrt{\frac{1}{T} \int_0^T p(t)^2 dt} \quad (11)$$

As an example, Figure 5 shows the experimental pressure traces at 4000 and 7000 rpm, at WOT condition, for gasoline and CNG, and the related *SPL* up to 10000 Hz.

As expected, at each engine speed, the CNG fuel exhibits a slower burning rate and a lower pressure peak than Gasoline. Consequently, the in-cylinder *SPL* at both speed is reduced. The slightly higher high frequency content (above 4000 Hz) does not contribute too much at the overall radiated noise.

The latter, as already said, depends on structural attenuation of the engine block, assumed to behave as a linear system. Under the additional hypothesis of free field conditions, the external sound pressure level (SPL_{ext}), at 1m from the engine, can be written as:

$$SPL_{ext}(dBA) = SPL_{cyl}(dB) + SA(dB) + AW(dB) \quad (12)$$

In the above equation, *SA* term represents the engine Structure Attenuation, shown in Figure 6. Although, not exactly referring to the tested powertrain, the considered profile accounts for strong attenuation at low and high frequency range, as verified in most of the actual internal combustion engines [28]. In addition, since the same profile is utilized for both gasoline and CNG, the relative difference of radiated noise between the two fuels is expected to be well captured.

In order to obtain an output in dBA, an additional filtering is applied using the A-Weighting (*AW*) filter reported in the same Figure 6, which takes into account the human ear perception.

Repeating the above described procedure for each experimental pressure cycle, the external radiated noise resulting from eq. (12) can be easily predicted, as shown in Figure 7a. Due to uncertainties in the specification of the *SA* profile in Figure 6, the maximum in-cylinder pressure rise rate ($dp/d\theta_{max}$, bar/deg) is also reported as a function of engine speed and injected fuel (Figure 7b). Both Figures clearly confirm that CNG operation is quieter at each operating condition. In particular, the effects of slower CNG combustion increase with the engine speed (see Figure 5) determining a reduced noise emission at high speeds.

Figure 7a also shows that maximum radiated noise, for gasoline combustion, approaches 103 dBA. In further analyses related to engine optimization in CNG operation, the latter value will be hence considered as the maximum admissible noise level.

3D model - Detailed in-cylinder analysis

Computational Grid

3D-CFD in-cylinder simulations are carried out using Star-CD licensed by CD-Adapco; the mesh is generated through the es-ice tool and is made up of mainly hexahedral cells which are progressively distorted, activated and deactivated to account for the piston motion. The