Particle Formation and Emissions from Dual Fueled CNG DI and Gasoline PFI SI Research Engine

F. Catapano, S. Di Iorio, P. Sementa, B. M. Vaglieco
Istituto Motori CNR
b.m.vaglieco@im.cnr.it

Abstract
Compressed natural gas (CNG) is considered as cleaner fuel compared to gasoline and has emerged as an alternative transport fuel in view of its abundant availability. CNG has been implemented in transport sector globally. PM and NO_x emissions were reduced, nevertheless, recent studies highlighted the presence of ultrafine particle emissions at the exhaust. The present study deals of the effect of CNG on particle formation and emissions when it was direct injected and when it was dual fueled with gasoline. In this latter case, the CNG was direct injected and the gasoline port fuel injected. Measurements were performed in cylinder and at the exhaust of a single cylinder SI research engine similar to modern commercial engine for passenger cars. The in-cylinder formation process of particles was evaluated by high spatial and temporal resolution optical measurements based on 2D chemiluminescence measurements from UV to visible. Two “ad hoc” filters, at 310 and 431 nm, were used to detect two representative radical species that influence the in-cylinder pollutants formation and oxidation. In particular, OH* and CH* spatial distribution were measurement to evaluate the local air fuel ratio in the cylinder too. Simultaneously, the exhaust emissions were characterized and the particle size distribution function was measured by an Engine Exhaust Particle Sizer in the range from 5.6 to 560 nm. The local air fuel ratio spatial distribution was strongly correlated to oxidation and particle emissions.

Introduction
Currently, the possible fuel depletion as well as the growing concerns on environmental issues prompt to the use of more environmentally friendly fuel for urban vehicles. The use of gaseous fuels can be considered a right choice also as transition for next future of internal combustion engines and the retrofitting of old cars too as an alternative to reduce air pollutant emissions in congested towns. The use of methane was encouraged over time especially in countries where there was lack fossil resource or where there was abundance natural gas resources. Recently, several studies on greenhouse gas intensity of natural gas, whose main component is methane, have shown that natural gas reduces GHG emissions from passenger cars on a Well-to-Wheel (WtW) basis by 23% compared with petrol and by 7% compared with diesel. Moreover, on the heavy-duty application, benefits compared to diesel are of 16% for compressed natural gas (CNG) and up to 15% for liquid natural gas (LNG). Also in the maritime sector, overall WtW benefits are up to

21% compared to conventional Heavy-Fuel-Oil (HFO) fuels [1]. The CNG properties make it suitable for the use in the spark ignition engines. It has a high-octane number, and hence high auto-ignition temperature and anti-knocking property. Then, spark ignition engines fueled with CNG can run at higher compression ratios, thus producing higher thermal efficiencies. It mixes uniformly with air, resulting in efficient combustion and a substantial reduction of the emissions at the exhaust [2-4]. On the other hand, the slow burning velocity, the poor lean-burn capability and the air displacement, lead to large cycle-by-cycle variations, lower engine power output and large fuel consumption [5].

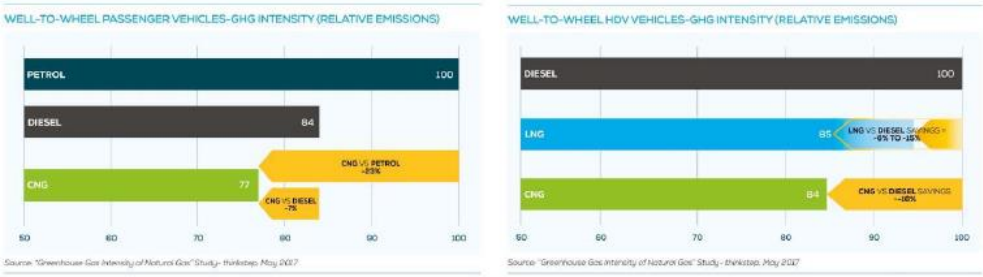

Figure 1-WTW Passenger vehicle relative emission using gaseous and fossil fuels [1]

The combustion of gaseous fuels is cleaner than liquid fuels. It can reduce the NO_x and PM, several studies indicated that particle concentration levels emitted from CNG fueled engines are lower than gasoline the port fuel injection (PFI) SI engines [5, 6], as well as contributing to the reduction of CO_2 emissions, due to the low carbon-to-hydrogen ratio. Nevertheless, recent studies highlighted the presence of ultrafine particle at the exhaust of CNG engines. To overcome these phenomena, different injection strategies based on simultaneous use of CNG direct injected (DI) and gasoline in the duct were proposed and evaluated. Moreover, the phenomena that induced their formation were evaluated by simultaneous in-cylinder optical measurements and conventional one.

Experimental Apparatus and Procedures

An optically accessible four-valve single cylinder research engine (Figure 2) was used. The engine was equipped with the prototype GDI cylinder head of a 250 cc engine widely used in Europe, it is provided with a hole between the two intake valves for the direct injection system. The head is characterized by pent-roof chamber engine mounted on an elongated piston. The engine reached a maximum speed of 5000 rpm. It was modified to run fueled with CNG and gasoline both simultaneously (Dual Fuel configuration) and not. The CNG was supplied by a pressurized bottle using a pressure regulators typically set to 8 bar directly in to the combustion chamber. A Synerject strata injector in house modified was used for the gaseous fuel. For the liquid fuel injection in the intake port was used a commercial injector with 3 holes. It is a commercial injector for the real reference

engine. For the DI was used a Magneti Marelli prototypal 6-holes injector. Further details on the experimental procedure can be retrieved in [5,6].
The head of engine had a centrally located spark plug and a quartz pressure transducer flush-installed in the combustion chamber to measure the in-cylinder pressure. A crankshaft encoder with a resolution of 0.1 crank angle degree (CAD) was used to trigger the pressure measurements. The in-cylinder pressure, the rate of chemical energy release and the related parameters were evaluated on an individual cycle basis and/or averaged on 400 cycles [5, 6].
The design of this engine utilizes a classic extended piston with piston-crown window similar to that originally introduced by Bowditch in 1961 [7]. This is of relatively high importance for this study, given that the design did not allow any lubricant droplets to enter the combustion chamber (i.e. the optically accessible part of the engine features self-lubricating Teflon-bronze piston rings); therefore, no contribution that could have originated from the oil interfere with the formation of particulate.

Optical Apparatus

Natural flame emission passed through the sapphire window fitted in the piston crown and was reflected toward the optical detection assembly by an UV-enhanced mirror inclined at 45°. The combustion process was followed by means of the 2D UV-visible digital imaging techniques. A CCD camera with sensitivity only in the visible range was used. It was equipped with a 50 mm focal length, f/3.8 Nikon lens. Moreover, CH* and OH* measurements were performed by using a dichroic filter, two band pass filters and two ICCD cameras equipped with a F/3.8 UV Nikon objectives with 105 mm focal length. The dichroic filter is a pass-through of the UV light from 230 to 360 nm the luminosity was forwarded in the direction of the first ICCD by means a 310 nm band pass filter to detect the OH* luminosity; at the same time the light above 360 nm is reflected towards the second ICCD camera. In this case a band pass filter centred at 431 nm was used to detect the CH* luminosity. The ICCD has an array size of 1024x1024 pixels with a pixel size of 19x19 μm^2 and 16-bit dynamic range digitization at 100 kHz. More details are reported in ref. [5.6].

Procedures

Two injection systems were used both simultaneously and separately to perform the PFI, the DI and the dual fuel (DF). In PFI configuration, the engine was fueled with gasoline (GPFI). In DI configuration, the engine was fueled with gasoline (GDI) and CNG (CNGDI). For DF configuration, the gasoline and CNG were separately injected. The gasoline was port fuel injected and the CNG direct injected (DF). The DF ratio was defined on energy basis. The 20% of the total energy was given by the combustion of port injected gasoline. The remaining 80% of the energy supply was due to the combustion of the CNG. More details are reported in ref. [5.6].The engine was equipped with a programmable system for the injection

and ignition management. In particular, the duration of injection (DOI) was properly set to obtain stoichiometric equivalence ratio and lambda 1.3. The start of spark (SOS) and the end of injection (EOI) was chosen to optimize the combustion considering the coefficient of variation of the indicated mean effective pressure (IMEP) and the exhaust emissions. Moreover, for DI the EOI was properly chosen to reproduce a stratified combustion. The gasoline injection pressure was fixed at 100 bar for DI and 3.5 bar for PFI, the injection pressure was 8 bar for the CNG.
Engine was operated at 2000 rpm-full load, chosen as representative of the urban driving condition in the New European Driving Cycle (NEDC). All the tests were carried out at stoichiometric and lean conditions. The air fuel ratio was measured by a linear lambda sensor at exhaust.

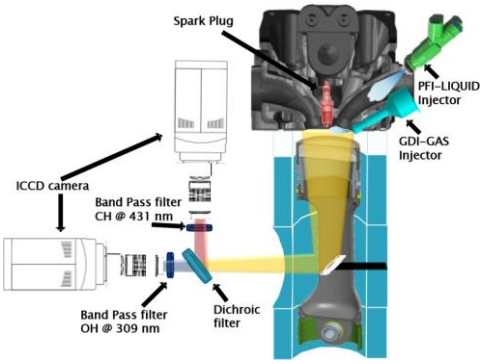

Figure 2. Experimental apparatus for optical investigations and a detail of head view with the injectors used

Exhaust Measurements

An opacimeter was used for evaluating the exhaust gas opacity. Particle number and size distributions were measured with an Engine Exhaust Particle Sizer 3090 by electrical mobility methods. It measures particle sizes from 5.6 to 560 nm with a sizing resolution of 32 channels. Before entering the EEPS, the sample of exhaust gas was taken by a 1.5 m long line heated at 150°C and it was diluted with the Dekati® Engine Exhaust Diluter, a Particle Measurement Program compliant conditioning system. Steady-state measurements of CO, CO_2, UHC and NO_x were detected. Moreover, Methane HC emissions were measured with a HP 5890 gas chromatography.

Experimental Results

The effect of dual fueling on engine combustion and performance were evaluated by the indicated data and the IMEP at two different air-fuel ratios. Figure 3 shows the in cylinder pressure and rate of heat release (ROHR) for all the injection strategies and fuels in stoichiometric condition.

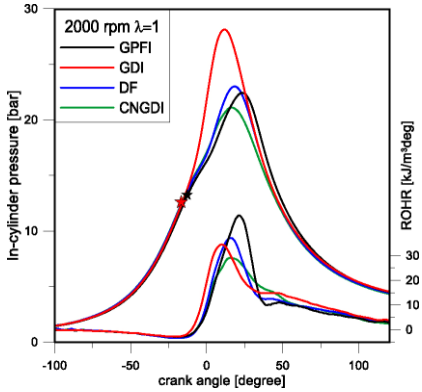

Figure 3. In-cylinder pressure and rate of heat release for all injection strategies at 2000 rpm in stoichiometric condition

The GDI configuration has the highest in-cylinder pressure than the other configurations. This can be ascribed to the stratification of the charge that results in a faster combustion that evolves closer to the TDC lowering the heat transfer through the cylinder wall so increasing the in-cylinder pressure. The positive effect of the charge stratification is less evident for CNGDI, for which the pressure curve was lower and the combustion is slower than GPFI as well. In this case, the beneficial effect of the stratification is counteracted by the CNG properties such as the slow flame propagation and the higher diffusivity. When CNG was dual fueled, the pressure was higher and more advanced than CNGDI and GPFI, indicating that the gasoline improves the CNG combustion better exploiting all the advantages of the charge stratification. Nonetheless, for CNGDI, despite the same SOS the combustion evolution is closer to that of GPFI, suggesting that the better charge stratification of GDI plays an important role, resulting in an improved kernel formation and flame propagation as observed from optical measurements too [6]. From the analysis of the ROHR can be distinguished a fast main combustion phase subsequently the spark and a much slower combustion phase, occurring late in the expansion stroke. Moreover, the flame front sweeps the combustion chamber, burning the air/fuel mixture in a quite premixed combustion [6]. At the end of the main combustion phase, the ROHR shows a diffusive combustion phase. The charge stratification of the GDI is also evident looking at the sharp increase of the ROHR during the first combustion phase. At the same time, the more evident diffusive phase, ascribable to the combustion of the liquid fuel, evidenced a bad evaporation and the fuel impingement that are typical issues of the charge stratification. CNGDI is characterized by a slow combustion. When dual fueled, the combustion is accelerated, as evidenced by the sharper ROHR, even if it is slower than the GDI. Nevertheless, the diffusive phase is quite lower with respect to GDI configuration because the direct injected fuel is the CNG leading to the reduction of the typical drawbacks of the charge stratification evidenced in the GDI. Moreover, despite the same ignition timing of GPFI, the DF combustion is

slightly advanced suggesting a more favorable kernel formation. This can be due to the combined effect of the combustion of gasoline as well as the ignition of gaseous fuel in the immediate vicinity of the combustion center of the gasoline combustion as observed in a previous paper [5, 6].

Figure 4 depicts the particle size distribution functions (PSDF) for all the engine injection configurations. For GDI the PSDF shows a strong accumulation mode centered around 80 nm in both stoichiometric and lean conditions. GPFI shows a bimodal size distribution where the weight of the nucleation is stronger in stoichiometric condition and comparable to the accumulation mode in lean condition. The accumulation mode is lower of about 1 order of magnitude with respect to the GDI and it is shifted towards smaller diameter in both stoichiometric and lean conditions. For CNGDI, the PSDF has a strong nucleation mode peaked around 10nm in both the operating conditions. A weak accumulation mode is observed. It is interesting noting that in lean condition, the emissions of particle smaller than 10 nm are larger than that in GDI. For DF, the PSDF is quite similar to that of GPFI. A strong nucleation mode can be observed both in stoichiometric and lean conditions. Nevertheless, for both the stoichiometric and lean conditions, the DF has a lower particle number concentration than GPFI and the curve is shifted toward smaller diameter. Moreover, a slightly larger emission of particles smaller than 10 nm with respect to GPFI is observed in lean condition.

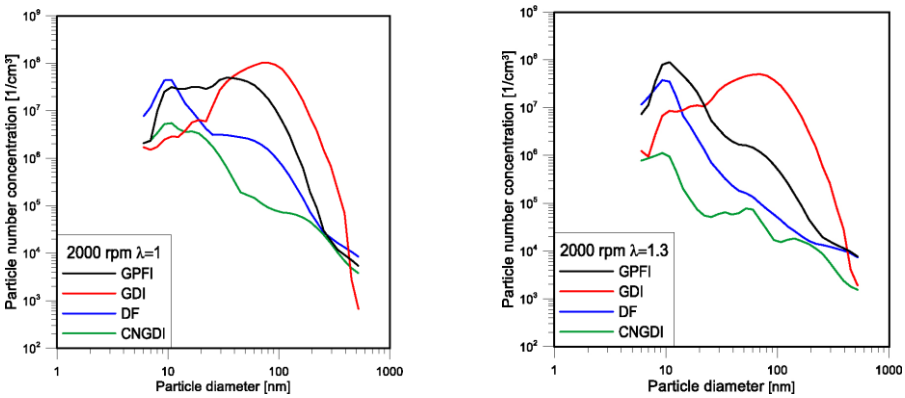

Figure 4. Particle size distribution functions for all the injection configurations in stoichiometric (left) and lean (right) condition.

For a comprehensive analysis of the effect of CNG and dual fuel on particle emissions, the combustion process was analysed by in cylinder optical techniques for detailing the formation and oxidation processes occurring in the combustion chamber with high spatial and temporal resolution. The optical analysis allows valuable information not only on combustion evolution but also on pollutant formation.

To better analyse the combustion process, natural flame chemiluminescence was

detected by 2D-digital imaging measurements performed in the visible range. Measurements in the UV range were carried out as well. In particular, narrow band pass filters at wavelengths typical of OH* (λ=310 nm) and CH* (λ=431 nm), respectively, were used. As known by literature, OH* is the main oxidant of soot and CH* is a flame front marker because it exists only in a narrow layer of the reaction zone taking part in the decomposition process of the fuel molecules in hydrocarbon flames. Furthermore, OH* and CH* can provide information also on the equivalence ratio. The local AFR can be evaluated from OH* and CH* chemiluminescence emissions from the following empirical equation for liquid and gaseous fuels [5]:

$$AFR=0.894-0.26 \ln (I_{OH}/I_{CH} -0.597) \quad (1)$$

$$AFR=0.599-0.314 \ln (I_{OH}/I_{CH} -0.344) \quad (2)$$

where I=Emission Intensity of OH* and CH* measured at ambient pressure.
Figure 5 depicts the images of the emissions of the natural flame, of the OH* and CH* radicals detected during the combustion process in stoichiometric condition, and the correspondent lambda spatial distribution for the GPFI, CNGDI and DF configurations.

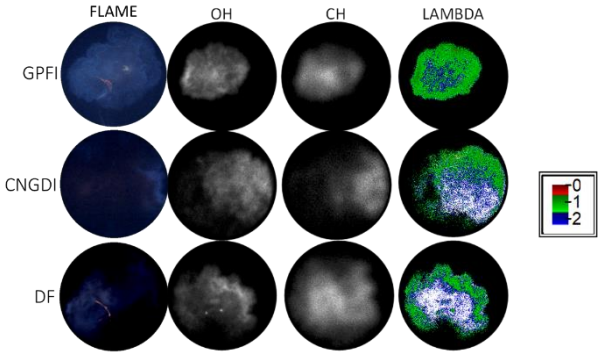

Figure 5. Flame, OH* and CH* emissions and Lambda spatial distribution at typical crank angles of the combustion process in stoichiometric condition

In GPFI the presence of a diffusive flame, characterized by a yellow-orange flame mainly due to thermal radiation of hot soot particles [5-7], in correspondence of the intake valve is well recognizable by the natural flame chemiluminescence. In PFISI engines, in fact, despite the high temperature of the valves, liquid fuel can deposit on the valves surface during the fuel injection forming a liquid film that burns in a diffusive way. In DF, a weak flame is recognizable close to the intake valve. The lower luminosity is due to the lower amount of gasoline and indicates a lower soot production. For GNGDI, weak luminous and not localized flames, where nuclei soot particle is formed, are observed. These can be due to the presence of rich zone likely due to the presence of lube oil passing through the valve stem seals. In real engines, it is still not entirely clear which mechanism is the main contributor to oil consumption due to the complexity of the phenomenon. It is controlled by engine

architecture and by the design of four salient escape routes: the piston rings, the turbocharger seals, the valve stem seals, and the positive crankcase ventilation system. In the optical engine, unlubricated Teflon-Bronze rings are used then lube oil may reach the combustion chamber only by the valve stem seals. It is important noting that, looking at the particle emissions, a small leak through the exhaust valve generates more particulate matter emissions than a far larger leak through the inlet valve, simply because, in the former case the oil is oxidized less effectively [6].

The chemiluminescence emission @ 431 nm is lower with respect to the signal at 310 nm. Nevertheless, they show a similar behaviour.

The analysis of the local lambda values shows for CNGDI and DF a more heterogeneous mixture. The central region of the flame, in fact, rapidly becomes lean indicating a fast decomposition of the fuel molecules. In DF combustion, the turbulence due to the gas direct injection coupled with the lower carbon content of the gas enhance the gasoline vaporization reducing the flame rich region so reducing the formation of soot. At the same time, the larger presence of lean region results in an improved oxidation process.

Conclusions

The CNG-gasoline dual fuel combustion was characterized in a SI 4-stroke small engine. The thermodynamic analysis suggests that under dual fuel operating mode the quality of the gaseous fuel combustion was improved by the liquid fuel supplementary addition. Gasoline favors the propagation of the flame front, resulting in an improvement of the combustion.

Moreover, the beneficial effect of CNG in terms of particle emissions reduction is ascribable to the low particle formation due to the gaseous properties, such as no C-C bound, probably most of the particles measured for CNGDI condition are due to leaking of the lubricating oil through the valves. When the CNG is dual fuelled with gasoline the particle emissions are quite lower than GPFI because of the lower contribution of gasoline injected in the intake duct. DF weak flame is recognizable close to the intake valve. The lower luminosity can be ascribed to the lower amount of gasoline and indicates a lower soot production.

The combined analysis of optical and particle exhaust measurements highlights that the use of direct injection of CNG both in the conventional and the dual fuel configuration, in which gasoline was injected in the intake duct, leads to soot formation and particle emissions. Nevertheless, the mean diameter detected at exhaust is smaller than 20 nm. The use of “ad-hoc” after treatment should be carried out to better evaluate the dual-fuel engine efficiency.

References

1. http://ngvemissionsstudy.eu/
2. Korakianitis, T., Namasivayam, A. M. and Crookes, R. J. (2011). Natural-gas fueled spark-ignition (SI) and compression-ignition (CI) engine performance

and emissions. Prog. Energ. Combust., 37, 89-112.

3. Aslam, M.U., Masjuki, H.H., Kalam, M.A., Abdesselam, H., Mahlia, T.M.I., Amalina, M.A.,(2006). An experimental investigation of CNG as an alternative fuel for a retrofitted gasoline vehicle. Fuel 85, 717–724.

4. Huang, X., Wang, Y. Xing, Z., Du, K. (2016) Emission factors of air pollutants from CNG-gasoline bi-fuel vehicles: Part II. CO, HC and NOx Science of the Total Environment, 565 698–705.

5. Di Iorio, S., Sementa, P., and Vaglieco, B., M., (2013) Experimental Investigation of a Methane-Gasoline Dual-Fuel Combustion in a Small Displacement Optical Engine, SAE Technical Paper 2013-24-0046, 2013, doi:10.4271/2013-24-0046.

6. Catapano, F., Di Iorio, S., Sementa, P., and Vaglieco, B., M., (2017) "Particle Formation and Emissions in an Optical Small Displacement SI Engine Dual Fueled with CNG DI and Gasoline PFI," SAE Technical Paper 2017-24-0092, 2017, doi:10.4271/2017-24-0092.

7. Ladommatos, N. and Zhao, H., “Engine Combustion Instrumentation and Diagnostics,” (Warrendale, SAE International, 2001), doi:10.4271/R-264, 2001.