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Detailed combustion analysis of a supercharged double-fueled spark ignition engine

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Detailed combustion analysis of a supercharged double-fueled spark ignition engine

Abstract: The main goal of researches in the field of automotive engineering is to obtain a large scale implementation of low or zero-emissions vehicles in order to substantially reduce air pollution in urban areas. A fundamental step toward this green transition is represented by the improvement of current internal combustion engines in terms of fuel economy and pollutant emissions. The spark ignition (SI) engines of modern light duty vehicle are supercharged, down-sized and equipped with direct injection. The gaseous fuels, such as liquefied petroleum gas (LPG) or natural gas (NG), proved to be a valid alternative to gasoline in order to reduce pollutant emissions and increase fuel economy. In previous works the authors investigated the simultaneous combustion, in a SI engine, of gasoline and a gaseous fuel (referred to as Double-Fuel operation, DF) both in the naturally aspirated and supercharged version; a significant increment of engine efficiency and a great reduction of pollutant emissions were obtained with respect to pure gasoline operation, with almost unchanged performance.

This paper is a development of the previous work and shows the results of a detailed heat release analysis, performed on the DF supercharged engine fueled with mixtures of gasoline and natural gas, in order to highlight the effects of engine speed, charging pressure and fuel mixture composition (the proportion between gasoline and NG) on the combustion speed.

It was found that both gasoline content in the DF mixture and supercharging pressure contribute to increase the combustion speed, which, in some cases, produced engine indicated efficiency increments up to 5%. The wide set of experimental data presented in this paper allows to better understand the combustion behavior of gasoline-NG fuel mixtures and can be also used to calibrate combustion sub-models integrated in engine numerical simulations.

Key-Words: spark ignition engine, combustion, fuel mixtures, natural gas.

Introduction

The anthropic global warming is one of the most discussed topic in the last decades and almost all countries have a plan to reduce greenhouse gas emissions in the next years. A parallel, and not less important, topic is the reduction of harmful pollutant emissions in urban areas where most people live. One of the main sources of pollutant emissions in urban areas is the vehicular traffic and for this reason the environmental regulations became increasingly stringent. To reduce pollutant emissions and improve fuel economy the research in the field of internal combustion (IC) engines produced, in the last decades, many innovations such as gasoline direct injection [1] and engine downsizing [2] coupled with supercharging [3] [4] [5] [6].

The pollutant emissions of an IC engine are strongly related to fuel properties and also to the combustion process. The main pollutant emissions of Diesel engines are particulate matter (PM) and nitrous oxides (NOx). Gasoline fueled spark ignition (SI) engines, endowed of port fuel injection (PFI) or direct injection system, mainly emit unburned hydrocarbons (HC), carbon monoxide (CO) and nitrous oxides; the gasoline direct injection (GDI) engines also emit significant levels of PM, although lower than Diesel engines. When running at high-full load, gasoline SI engines are usually operated with rich air/fuel (A/F) mixtures with the aim to cool inlet air (exploiting the heat of vaporization of gasoline) and prevent dangerous knocking phenomena; such rich mixtures (even +25% with respect to stoichiometric) cause very high levels of CO and HC in the engine out emissions, and very low oxidizing efficiency of the three-way catalyst: as a result, in there operating conditions, the most part of the CO and HC produced by the engine is also emitted by the tailpipe. Such mixture enrichments are not adopted when the spark ignition engine is fueled by gaseous fuels (Compressed Natural Gas or LPG), for two main reasons: the first is that gaseous fuels are characterized by a considerably higher knock resistance with respect to gasoline (gasoline MON is 85, LPG MON is 93 [7], while CNG MON is 122[8]) which makes gaseous fuels less critical; the second is that gaseous fuels have already lost their heat of vaporization (being already gaseous), and hence a mixture enrichment would not produce the cooling effect obtained with gasoline. In the full load condition, gaseous fueled SI engines may be however operated with slightly rich mixtures (3%-5%) with the aim to maximize flame propagation speed and hence engine power, but the resulting CO and HC emissions are very low with respect to a gasoline operation. Moreover, the better mixing properties of gaseous fuels with respect to liquid injected fuels also allows to obtain very good and homogeneous air-fuel mixtures, with the result of a more complete combustion and hence lower HC and negligible PM emissions. For the already mentioned higher knock resistance, gas fueled supercharged engines adopt higher compression ratios, compared to gasoline engines, and benefit of higher efficiency and power density. Gaseous fuels also have the advantage of a low production cost and uniform geopolitical availability while their low density, compared to gasoline, reduces the volumetric efficiency of port injected engines: the direct injection [9] and/or supercharging however can eliminate this drawback.

In the past decades gaseous fuels have been studied [10] [11] [12] and gas fueled vehicles have been presented in the market as a valid alternative to gasoline and diesel ones, frequently in the bi-fuel version, i.e. endowed of two separate and complete injection systems, which allow the vehicle to run either on gasoline or on the gaseous fuel; many researches have been also carried out regarding methane or hydrogen fueled engines [13] as well as natural gas (NG) [14].

A SI bi-fuel engine can be also fueled in a third operating mode, called Double-Fuel (DF) mode [15] [16] [17], which consists in injecting both fuels during the same engine cycle. This operating mode can be easily implemented in bi-fuel engines exploiting the two separate injection systems already available on board, and only requires a software editing of the Electronic Control Unit (ECU). It is worth highlighting that the Double-Fuel combustion is quite different from the well-known Dual Fuel combustion (operated in Compression Ignition engines), in which a small quantity of Diesel fuel is directly injected into the premixed air-gaseous fuel mixture, and its auto-ignition acts as igniter to start the flame propagation combustion of the air-gaseous fuel charge; in the Double-Fuel operation, instead, the two fuels are homogeneously mixed with air and simultaneously burn through the same flame front.

Previous works from the authors of this paper showed that, in a naturally aspirated spark ignition engine, the simultaneous combustion of gaseous fuel (NG or LPG) and gasoline (i.e. the DF operation) allows to reach, in comparison to the pure gasoline operation, a remarkable increase (+26%) of the brake thermal efficiency (BTE) and an extreme reduction (-90%) of HC and CO pollutant emissions, whilst maintaining almost the same (-4%) power output [15] [16]. These results were made possible on account of the high knock resistance of gaseous fuels which permitted to adopt stoichiometric A/F mixtures and optimal spark advance up to full load operation. Dealing with the DF operation with natural gas and gasoline, it was also observed that the maximum brake thermal efficiency was reached employing fuels mixture with 50% in mass of both fuels, while the peak performances were obtained with a fuels mixture with a 30% in mass of natural gas.

The same authors also experimented [17] the supercharged DF engine in order to further investigate the efficiency, pollutant emissions and performance improvements obtainable. A lot of experimental tests were carried out: the engine was fueled with gasoline-NG mixtures in different proportions and imposing different supercharging pressures; in the best case scenario engine performance increased up to 20% with respect to pure NG operation, while, compared to pure gasoline operation, indicated efficiency increased up to 32% and the pollutant emissions were significantly reduced (-75% CO and -66% HC).

Many authors, in the last years, operated a turbocharged SI engine [18] [19] [20] in DF mode experiencing various benefits with respect to both "only gasoline" and "only NG" modes. The authors of these papers however could not control the supercharging pressure which was a direct consequence of the interaction between the turbocharging system and the engine: hence the number of the test conditions analyzed was limited. In [17] instead the authors of the present paper performed an analytical study of the supercharged DF engine performance under many different boost pressure conditions.

Other researchers [21] investigated the performance and emissions improvement of the Double-Fuel injection (also frequently referred to as dual-injection) SI engine (NG direct injection plus gasoline port injection), under lean-burn conditions, with respect to sole gasoline mode; they experimented a decrease of both Brake Specific Fuel Consumption (BSFC) and pollutant emissions (HC, CO and PM). Many researchers investigated the simultaneous combustion of gasoline and hydrogen in a SI engine: in [22] the authors experienced a decrease of BSFC and NOx emissions, with respect to sole gasoline operation, by optimizing water injection and Start of Combustion (SOC) while in [23] the authors performed a heat and exergy balance. In [24] the authors performed a comparative analysis on a NG fueled engine by adding either gasoline or methanol; they found a BTE increment, with respect to sole NG operation, when adding methanol while a decrement when adding gasoline and also the HC and CO were lowered by adding methanol but raised by adding gasoline. Finally some researchers [25] [26] investigated the lean burn operation of methanol/hydrogen dual-injection SI engines; they found BTE increments and HC and CO reductions with respect to sole methanol operation together with a faster combustion and a smoother engine operation.

The most important process influencing engine performance, efficiency and emissions, is indeed the combustion; in particular, a faster combustion brings higher pressure and temperature and in turn higher thermodynamic efficiency. The combustion heat release process is studied by using the Mass Fraction Burned (MFB) curve evaluated by analyzing the indicated pressure. Numerical simulations are a fundamental tool in the engine design and optimization procedure and since combustion is the most important phase of engine cycle a proper combustion sub-model calibration is a key aspect of the whole design process. The combustion sub model calibration requires a lot of experimental data in the form of MFB curves obtained in many different engine operating conditions (manifold absolute pressure (MAP) and engine speed).

Considering the absence of literature about the combustion analysis of gasoline-NG mixtures in SI engines, the authors further developed the study of the supercharged DF engine [17], by evaluating the effects of fuel mixture composition, supercharging pressure and engine speed on combustion speed. Hence the results of this study are fundamental to better understand the combustion behavior of gasoline-NG fuel mixtures as well as to obtain a wide experimental database for the calibration of combustion simulation models.

At equal operating parameters (engine speed, engine load, mixture strength and spark advance) natural gas exhibits a slower flame propagation than gasoline [27] [28] thus giving a longer combustion and, in turn a lower thermodynamic efficiency. When gasoline and natural gas are mixed together a brand new fuel is obtained whose combustion speed depends on mixture composition (the mass percentage of the two fuels).

To summarize, the combustion speed is influenced by engine speed that promotes turbulence inside the combustion chamber, by supercharging pressure that influences pressure and temperature during combustion, and finally by fuel mixture composition; all these aspects have been studied through the experimental MFB curve evaluated for each engine operating condition tested in the previous work [17]; each MFB curve has been studied both as a whole and subdivided into its main three parts: the flame front development phase, the fast flame propagation phase and flame extinction phase; in this way the influence of the various operating conditions has been evaluated with great detail and a big amount of experimental data have been collected and will be used, in a future development of this research, to properly calibrate a combustion model. Moreover, with the aim to better understand the role played by each single parameter, the study has been carried out both in the time domain and in the crank angle domain: for this reason, as example, the extension of the whole combustion, as well as of each single part, has been evaluated both as time interval and as crank angle interval: the term "duration" has been hence employed to indicate the time interval (expressed in milliseconds), while, the related angle interval has been indicated as "arc" (measured in Crank Angle Degrees, CAD); it is obvious that the combustion arc is proportional to the combustion duration through the engine speed, and both have an inverse proportionality with the speed of combustion: faster combustion will give hence lower combustion duration or combustion arc.

The comparison between the combustion speed of pure fuels (gasoline and NG) and that of fuel mixtures allows to identify and separate the effects of the engine and the effects of the fuel, hence the results of this work can be used to calibrate combustion models that will be used in the design process of different engines fueled with the same fuel mixtures.

Experimental setup

In this paragraph the authors report a short description of the experimental setup used to obtain the MFB curves analyzed in the paper; a more detailed description of the setup is however available in previous work [17] dealing with the performance and efficiency obtainable by DF operation on a supercharged spark ignition engine.

As reported in Figure 1, where the layout of the employed engine test bed is shown, a bi-fuel PFI SI engine from FIAT (whose specifications are reported in Table 1) was coupled to an eddy current dynamometer (*Schenck W130*), and supercharged by means of a Roots compressor *(Finder BLW 80-2)* powered by a *Control Techniques* brushless motor. The speed of the supercharger was feedback controlled by means of a simple PID controller with the aim to maintain every desired level of MAP for every engine speed, while an intercooler was implemented to maintain almost constant the inlet air temperature.

The DF operation was implemented exploiting the two separate port fuel injection systems (one for each fuel) already available on the original bi-fuel engine: each desired mixture of NG and gasoline was hence obtained by modulating the injection time of each fuel, whose specifications are reported in Table 2 and Table 3.

It is worth to point out that the power absorbed by the supercharging system was not employed anyway to obtain the effective power produced by the engine due to the following reasons: firstly, the compressor rotational speed was varied by the PID controller to obtain constant MAP values, apart from the engine speed and fuel mixture, thus causing a continuous variation of the compressor-engine “speed-ratio” resulting in compressor efficiencies far from best values and out of line with conventional mechanical supercharging devices; secondly, the measurable electric power absorbed by the supercharging system included the efficiency of the brushless motor, whose contribution could not be hence separated. As a consequence, only indicated parameters could be evaluated.

Table 1 - Engine specs

Number of cylinders	4
Total displacement [cm^3]	1242
Bore [mm]	70.80
Stroke [mm]	78.86
Compression ratio	9.8
Rod to crank ratio	3.27
Intake valve/cylinder	1
Exhaust valve/cylinder	1
Gasoline injection system	PFI, Bosch EV6
NG injection system	PFI, Bosch EV1

Table 2 - Gasoline properties

Liquid phase density at 15 °C [kg/m^3]	740
Equivalent H/C ratio [29]	1.85
Stoichiometric air/fuel mass ratio	14.7
Lower heating value [MJ/kg] [30]	43.4
Motor Octane Number	85
Laminar burning velocity (1 bar, 358 K, λ=0.9) [cm/s] [27]	49.5

Table 3 – Natural Gas properties

Methane - CH_4	[%vol]	85.79
Ethane - C_2H_6	[%vol]	7.86
Propane - C_3H_8	[%vol]	1.61
N-butane - C_4H_{10}	[%vol]	0.19
Isobutane - C_4H_{10}	[%vol]	0.28
Butylene - C_4H_8	[%vol]	0.05
Isopentane - C_5H_{12}	[%vol]	0.06
N-pentane - C_5H_{12}	[%vol]	0.06
Carbon dioxide - CO_2	[%vol]	1.04
Nitrogen - N_2	[%vol]	2.96
Helium – He	[%vol]	0.09
Hydrogen/Carbon Ratio		3.76
Stoichiometric air/fuel ratio		16.9
Lower heating value [MJ/kg] [31]		46.67
Measured MON [8]		122.1
Laminar burning velocity (1 bar, 358 K, λ=1.0) [cm/s] [28]		37.5

The composition of the fuel mixture was defined by its NG mass fraction (defined as the ratio between the NG mass and the whole injected fuel mass), which was varied between 0% (i.e. pure gasoline), 40%, 60%, 80% and 100% (only natural gas), as reported in Table 4, where the operating conditions tested are resumed. Table 4 also reports the Lower Heating Value (LHV) of the different fuel mixtures used in the tests; the NG mass fraction has been used to identify the fuel mixture instead of LHV because the mass flows have been measured with great precision (the error is 1% of read value) while the LHV of both gasoline and NG are difficult to evaluate with such precision.

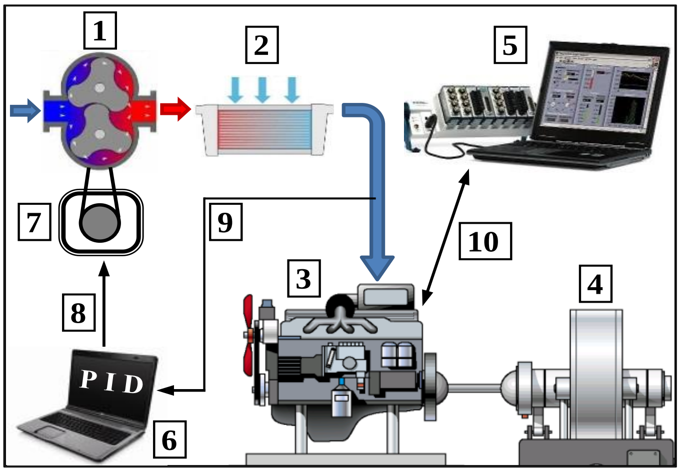


Figure 1 – schematic layout of the test bench: 1) supercharger; 2) intercooler; 3) SI bi-fuel engine; 4) eddy current dynamometer; 5) data acquisition system and engine control module; 6) brushless motor PID control; 7) brushless motor; 8) brushless speed control signal; 9) supercharging pressure sensor signal; 10) engine sensors output signals and input controls.

Table 4 - Operating conditions of the test performed

Engine speed [rpm]	from 1500 to 5000 with steps of 500
MAP [bar]	1.0, 1.2, 1.4, 1.6
Inlet temperature [°C]	28±10
NG mass fraction [%]	0, 40, 60, 80, 100
Overall air/fuel ratio	Stoichiometric
Spark advance (max. allowed for best efficiency) [CAD BTDC]	gasoline: from 8.5 to 22
	DF mixtures: from 11 to 31
LHV of the DF mixtures [MJ/kg] [30] [31]	40% NG: 44.71 60% NG: 45.36 80% NG: 46.02

The engine, operated with pure gasoline, could not be supercharged because of knocking occurrence and the manifold absolute pressure was hence maintained at 1.0 bar.

For each boosting pressure and fuel mixture adopted, the rotation speed of the engine was incremented with steps of 500 rpm starting from 1500 up to 5000 rpm, as also reported in Table 4, were the complete list of the operating conditions tested is reported. A stoichiometric overall A/F ratio was maintained both in the DF operation and in the pure natural gas test, with the aim to obtain the best compromise between engine efficiency and pollutant emissions. With pure gasoline, instead, the A/F was set to the factory settings (reported in Figure 2) to avoid knocking phenomena. As can be observed in Table 4, a maximum deviation of 10°C from mean value was achieved in the engine inlet temperature: this deviation however refers to the total number of experimental tests, i.e. all engine speeds, mixtures composition and boosting pressures tested. Higher inlet temperatures were obtained at higher boosting pressures, hence the test performed at the same MAP are perfectly comparable.

The mass flows of both fuels employed were measured by means of two Coriolis effect mass flow meters (*Endress+Hauser PROMASS* and *Bronkhorst mini CORI-FLOW*), while the engine inlet air mass flow was measured using an *Endress+Hauser Prowirl* vortex flow meter.

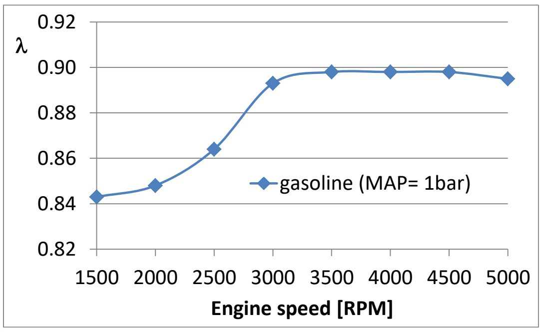


Figure 2 - Relative A/F ratio (λ) values adopted in the pure gasoline mode

Table 5 – Accuracy of the instrumentation used in the test.

Sensor	Accuracy
MAP sensor	±1% FSO (2.38 bar)
Gasoline mass flow meter	±1% of reading
NG mass flow meter	±1% of reading
Air mass flow meter	±1% of reading
Dynamometer	±2% of reading
Combustion chamber pressure sensor	linearity error < ±0.3% FSO
Combustion chamber pressure sensor	thermal sensitivity shift ≤±0.5% at temperature between 200 and 300°C

A piezoelectric pressure sensor *AVL GU13X* flush mounted in the combustion chamber was employed to measure in-cylinder pressure. Table 5 reports the accuracy of the instrumentation used in the test.

100 consecutive pressure cycles were sampled on every operating condition, individually compensated by means of the manifold absolute pressure [32] [33] and employed to obtain a single average pressure cycle, adopted to evaluate the MFB curve through the Rassweiler and Withrow method [34]. A capacitive sensor (*Kistler 2629B*), whose precision is 0.1 CAD, was previously employed to determine the correct Top Dead Centre (TDC) position, which, as known, is a critical aspect when performing indicating analysis.

In-cylinder pressure signal together with the other engine parameters were sampled using high-speed National Instruments data acquisition boards and counter boards, using the output pulses from a 360 ppr incremental optical encoder, installed on the engine crankshaft, as sampling clock and trigger, thus obtaining a sampling resolution of 1 CAD.

The spark advance was controlled to set the Location of the Peak Pressure (LPP) as close as possible to the value usually adopted for the maximum brake torque condition [35] [36], i.e. 15 CAD After Top Dead Centre (ATDC). With the aim to avoid dangerous knocking phenomena, the output signal from a piezoelectric accelerometer (*Brüel & Kjær Cubic DeltaTron*) fixed on the engine block, was monitored on a 100MHz Agilent oscilloscope to detect any possible knock occurrence. As a result, it was observed that pure natural-gas always allowed a knock free operation with the best combustion phasing (LPP at 15 CAD ATDC), while in DF mode, the probability of knocking occurrence increased with gasoline content in the fuel mixtures, often forcing to adopt retarded combustions.

Results and discussion

The aim of the present work is to show the results of an extensive series of experimental test performed on a supercharged spark ignition engine fueled with mixtures of natural gas and gasoline in different proportions, highlighting the effects of engine speed, supercharging pressure and fuel mixture composition on combustion speed.

The A and B diagrams in Figure 3 report the MFB curves, obtained at different engine speed, for the fuel mixture with 80% NG and MAP=1.6 bar; in A the MFB is plotted as function of time while in B as function of crank angle. In the diagrams *C* and *D* of Figure 3, instead, the time derivatives and the crank angle derivatives are shown respectively. As clear from the A diagram in Figure 3, the combustion duration (measured in milliseconds) decreases with increasing engine speed, due to the higher turbulence induced by the intake process, giving as result an almost constant combustion arc measured in CAD, as shown in the B graph of Figure 3; this means that, in the engine tested, the flame front propagation speed exhibits a proportional variation with engine speed. A similar behavior was found for all the operating conditions tested. The observed proportionality between combustion speed and engine speed, depicted in Figure 3 A and C, falls within theoretical expectations as far as the turbulence intensity in the cylinder near TDC position is proportional to mean piston speed [37]; the tested engine is endowed of a single inlet duct and has both flat cylinder and piston heads so the turbulence near TDC is not particularly amplified by swirl or tumble motions.

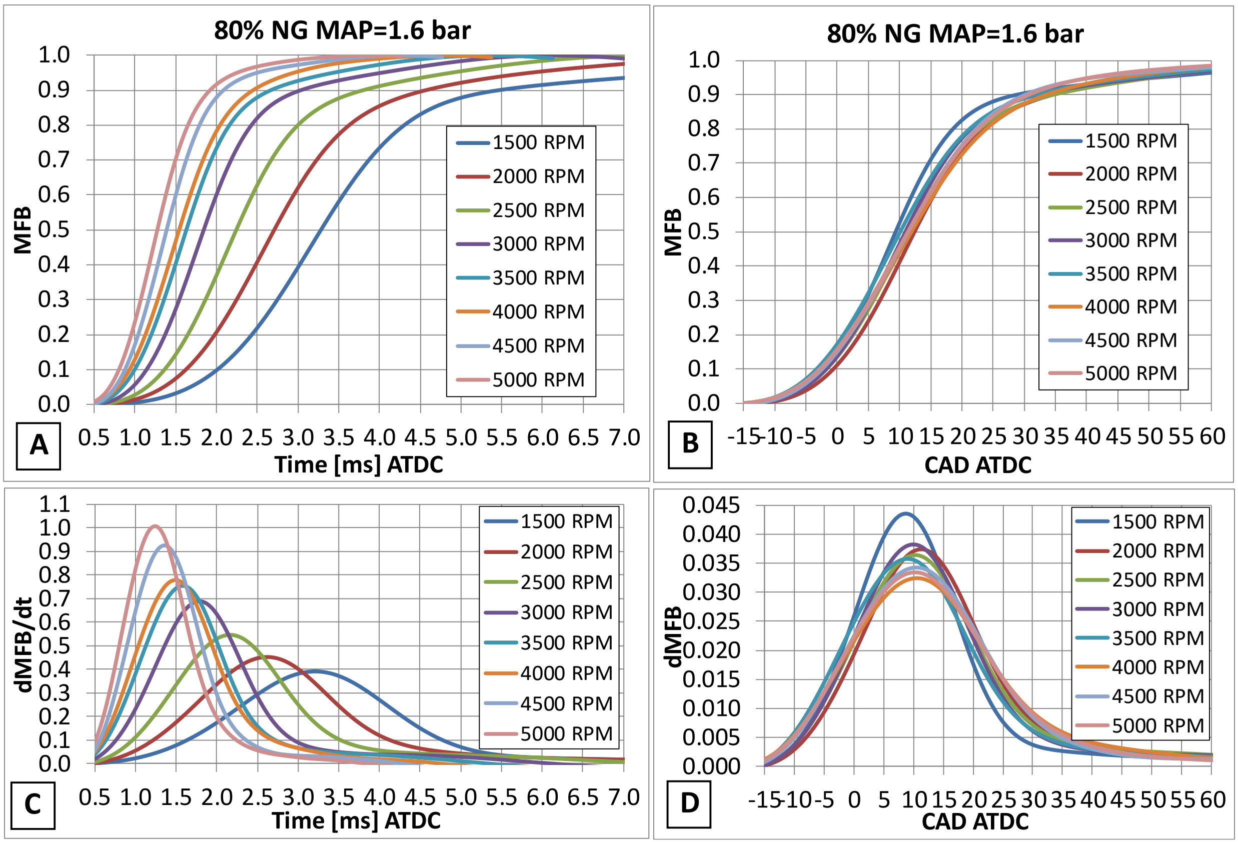
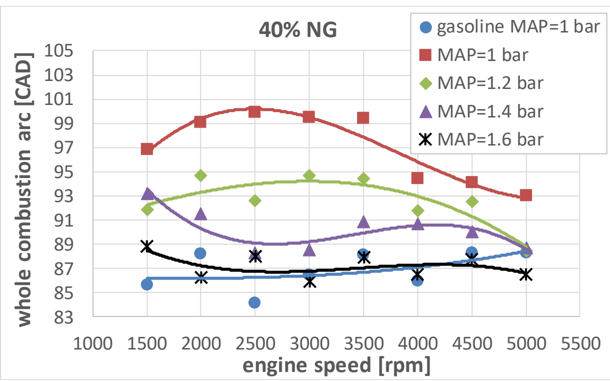


Figure 3 - MFB curves (A, B) and derivatives (C, D), at different engine speed, plotted as function of time (A, C) and as function of Crank Angle Degrees (B, D)

1. Whole combustion

The thermodynamic efficiency is maximum when the combustion is isochoric (constant volume) so the most important parameter to control is the combustion arc, which starts with the spark and is supposed to finish when MFB= 99%, being the last 1% not relevant and prone to cause great uncertainties on the actual end of combustion determination. Figure 4 shows, for the 40% NG fuel mixture, the combustion arc as function of engine speed evaluated at different supercharging pressures: as already mentioned the combustion arc is almost constant as function of engine speed. Figure 5 instead shows the combustion arc as function of %NG, for the constant engine speed of 3500 rpm and for different boosting pressures: it can be observed that at MAP=1 bar and 1.2 bar, the NG concentration in the fuel mixture does not influence the combustion arc, while at higher MAP levels (1.4 bar and 1.6 bar) the mixtures with higher gasoline content produce, as expected, smaller combustion arc (i.e. a faster combustion). A similar trend has been observed at different engine speed. The dashed line in Figure 5 shows the combustion arc length obtained with gasoline at MAP=1 bar and it is reported, as reference, also in the following diagrams.



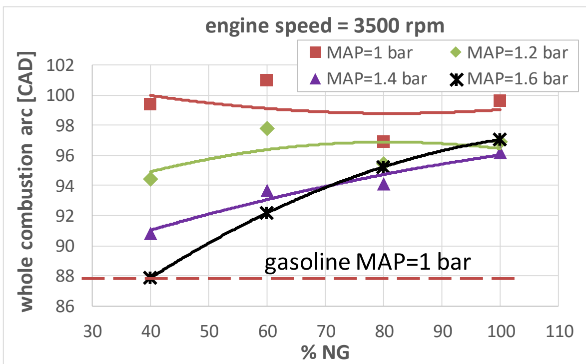


Figure 4 - combustion arc as function of engine speed, 40% NG

Figure 5 - combustion arc as function of %NG at 3500 rpm

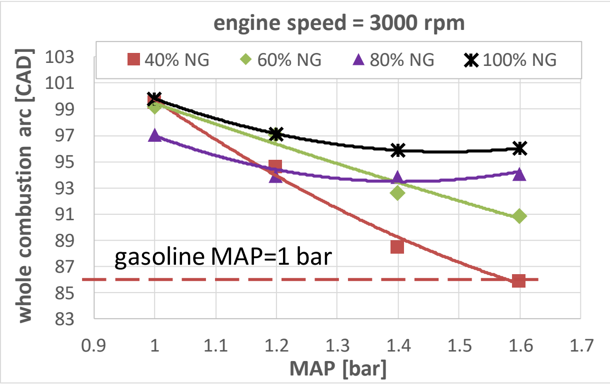


Figure 6 - combustion arc as function of MAP at 3000 rpm

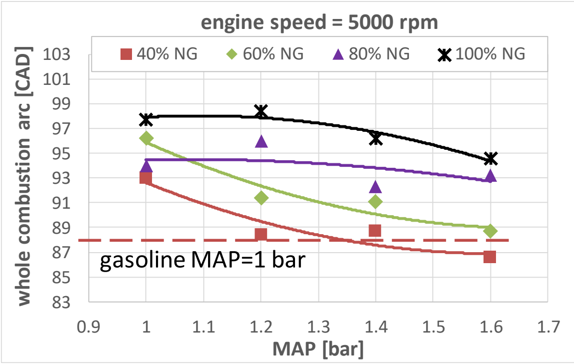


Figure 7 - combustion arc as function of MAP at 5000 rpm

In Figure 6 the combustion arc is plotted as function of MAP at the fixed engine speed of 3000 rpm and for different fuel mixture compositions: it can be observed that the boost pressure has a very limited influence on combustion arc when the fuel mixture has a high NG content (80% and 100% NG) while increasing the gasoline content in the fuel mixture produces a marked decrease of the combustion arc with MAP (8 CAD for the 60% NG and 13 CAD for the 40% NG mixture). A similar trend is also observed at higher engine speed, as shown in Figure 7 which refers to 5000 rpm, with a smaller reduction of the combustion arc (7 CAD for the 60% NG and 6 CAD for the 40% NG mixture). It can be concluded that, for each fuels mixture tested, a higher MAP level causes a faster combustion, and this effect is stronger with higher gasoline content in the fuel mixture. At MAP=1.6 bar the mixture with 40% NG shows an overall combustion arc reduction of about 10 CAD with respect to MAP=1 bar, meaning a combustion speed similar to the pure gasoline operation (Figure 4 and Figure 6).

Aiming to a deep investigation of the effects of engine speed, fuel mixture composition and MAP on combustion speed, the authors decided to divide the combustion arc into three zones: the flame front development, going from Spark timing to MFB=10%, that is mainly characterized by a laminar flame front propagation; the fast flame propagation, where MFB goes from 10% to 80% with a nearly linear trend, that is characterized by a turbulent flame propagation and produces the most relevant thermodynamic effects, and finally the flame extinction, where MFB goes from 80% to 99%, that is characterized by a progressive decrease of the flame propagation speed due to the depletion of burning material and to the flame front cooling, caused by the approach to the combustion chamber walls.

2. Flame front development

Figure 8 shows the crank angle covered by flame front development (FFD) as function of engine speed, evaluated at different supercharging pressures and for the 80% NG fuel mixture (a similar trend has been observed also for the other fuel mixtures tested): as can be noted, the FFD arc increases with engine speed for all the fuel mixtures and supercharging pressures tested; this means that the early stage of the combustion is not dominated by turbulence, otherwise a constant FFD arc would be observed (as therefore observed with regards to the entire combustion, see Figure 3 and Figure 4, whose duration proportionally decreases with engine speed resulting in an almost constant combustion arc). This is consistent with the widely diffused theory of the turbulent flame propagation: when the flame front dimension is in the order of the turbulence scale (i.e. during its early stage), the wrinkling effect of turbulence is less important [37]. It must be, however, pointed out that a slight effect of turbulence is present in the results obtained: the flame front development arc revealed an average overall increment of about 10 CAD, which means that, an engine speed increase of more than three times (from 1500 to 5000 rpm), caused the FFD arc to increase less than two times (while a complete null turbulence effect would give the triplication of the FFD arc). It can be hence concluded that, in its early stage, the flame front propagation is less influenced by turbulence and mainly depends on pressure and temperature.

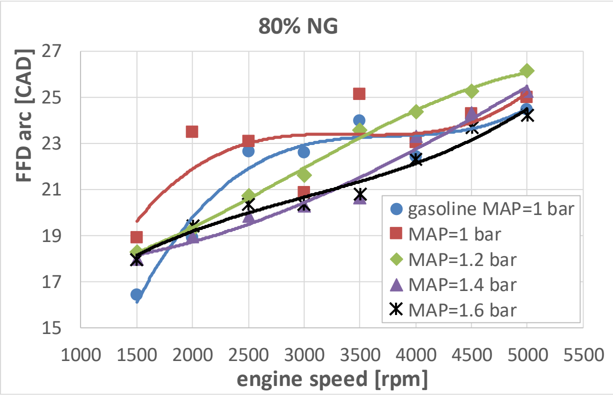


Figure 8 - FFD arc as function of engine speed, 80% NG

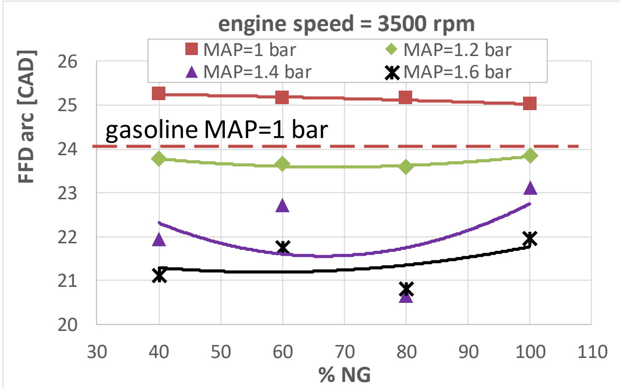


Figure 9 - FFD arc as function of %NG at 3500 rpm

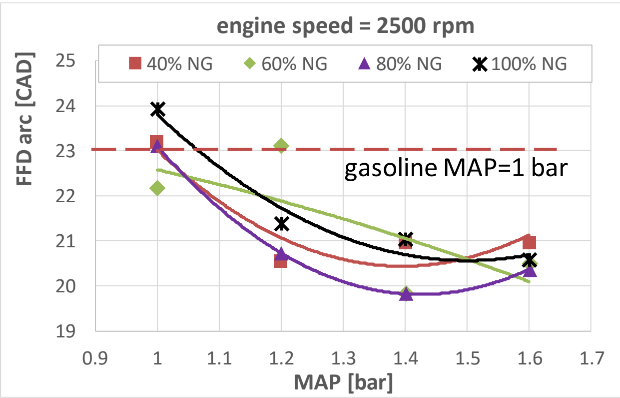


Figure 10 - FFD arc as function of MAP at 2500 rpm

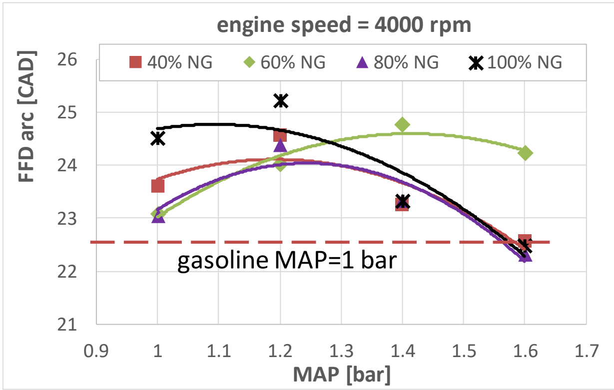


Figure 11 - FFD arc as function of MAP at 4000 rpm

Increasing the supercharging pressure slightly reduces the flame front development phase at medium-low engine speed, as can be seen in Figure 10, while reveals a limited effect for medium high engine speed (Figure 11). These trends can be explained considering the effects of pressure and temperature on the laminar burning velocity (LBV) [27] [28] that governs the early stage of combustion. LBV reduces with increasing pressure and increases with increasing temperature; supercharging the engine produces, at the SOC, higher pressures but also higher temperatures due to the reduced effect of wall heat transfer; the spark timing must be also taken in to account: when MAP increases the allowed spark advance tends to decrease to avoid knocking and this produces a higher temperature at the SOC due to the compression phase; from the analysis of Figure 10 and Figure 11 it can be concluded that the increased temperature inside the combustion chamber has a greater effect than the increased pressure because there is an overall reduction of FFD arc with MAP.

As regards the mixture composition, it was found a negligible effect on FFD arc, as can be seen in Figure 9; similar trends were observed for other engine speed; at MAP=1 bar all the DF mixtures exhibit a higher FFD arc compared to gasoline (Figure 9) and this is coherent with the lower LBV of NG compared to gasoline (as reported in Table 2 and Table 3); when MAP increases, the higher temperatures produce a proportional decrease of the FFD for all the DF mixtures.

3. Fast flame propagation

Figure 12, Figure 13, Figure 14 and Figure 15 show the effects of engine speed, mixture composition and supercharging pressure on the fast flame propagation (FFP) phase; during this phase 70% of the whole combustion heat is released hence small variations of this arc could have significant effects on the thermodynamic engine cycle.

The naturally aspirated engine (MAP=1 bar) shows an oscillating trend (with almost constant mean value) of the FFP arc versus engine speed for all the fuel mixtures tested: as example, Figure 12 shows the results for the 60% NG fuel mixture; when MAP is increased, all the fuel mixtures exhibit an FFP arc decrease at low rpm and an increase at high rpm with respect to MAP=1 bar (Figure 14, Figure 15), as a result the FFP arc of the supercharged engine shows an increasing trend as function of engine speed with a maximum variation of about 10 CAD. At MAP=1 bar the increase of engine speed, promoting turbulence, produces a great increment of combustion speed, a great reduction of the FFP duration and, as a consequence, an almost constant FFP arc; at higher MAP levels, instead, engine speed has a lower effect on combustion speed, giving a smaller reduction of the FFP duration and, as a consequence, an increase of the FFP arc, as shown in Figure 12. Increasing MAP produces an increase of both pressure and temperature in the combustion chamber together with an increment of the combustion speed, as proved by the FFP arc reduction of roughly 5 CAD shown in Figure 14; this is due to the positive effect of temperature on LBV, already discussed for FFD arc, considering that the turbulent burning velocity is strictly related to LBV [37]. At higher engine speed (Figure 15) the opposite effects of pressure and temperature compensate each other and the FFP arc results almost unchanged as function of MAP; probably, at higher engine speed, wall heat transfer are greatly reduced and this produces a sort of saturation effect on both gas temperature and LBV.

NG concentration in the fuel mixture (Figure 13) produces small increments of FFP arc when MAP is 1 and 1.2 bar and no increment or a small decrement when MAP is 1.4 and 1.6 bar respectively; this is probably due to opposite effects: the presence of NG tends to reduce the combustion speed because of its lower LBV compared to gasoline (as reported in Table 2 and Table 3) [27] [28] but the increasing spark advance, due to the higher knock resistance of NG, tends to produce higher temperatures during combustion and in turn higher burning speed.

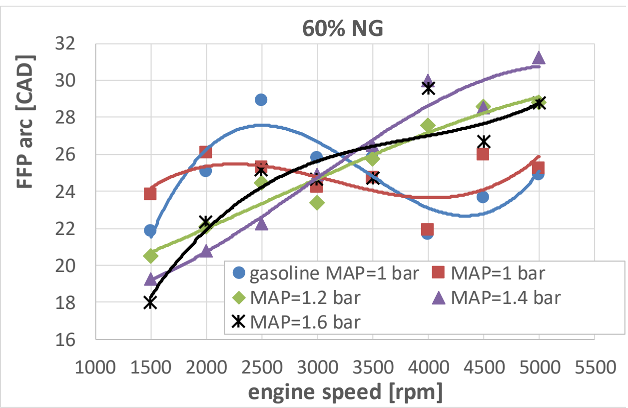


Figure 12 - FFP arc as function of engine speed, 60% NG

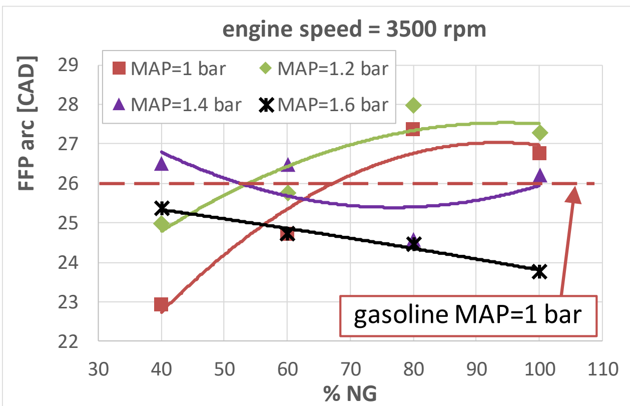


Figure 13 - FFP arc as function of %NG at 3500 rpm

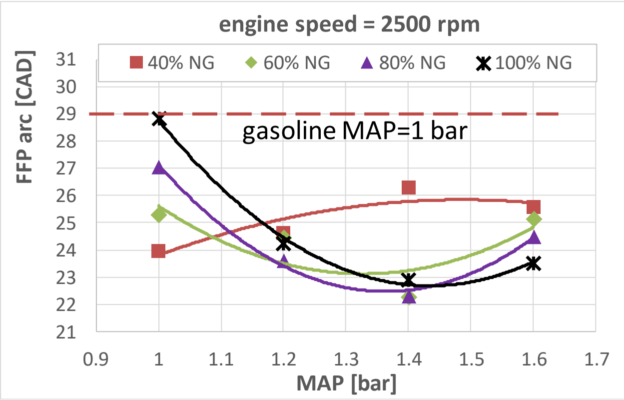


Figure 14 - FFP arc as function of MAP at 2500 rpm

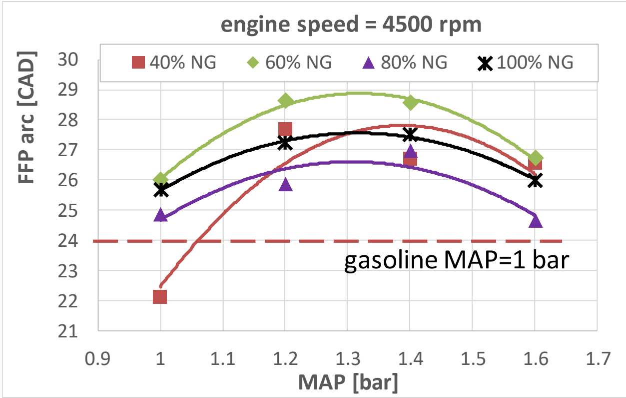


Figure 15 - FFP arc as function of MAP at 4500 rpm

4. Flame extinction

Figure 16 shows the duration of the flame extinction phase as function of engine speed evaluated at different MAP levels for the 40% NG fuel mixture, a similar trend has been observed for the other fuel mixtures. The results show a decreasing flame extinction arc with engine speed, with a maximum variation of about -20 CAD that compensates the increasing trends shown on previous Figure 8 and Figure 12: the overall result is the almost constant trend of the whole combustion arc as function of engine speed shown in Figure 4.

The decreasing trend of flame extinction arc shown in Figure 16 means that the turbulence induced by engine speed accelerates so much the flame extinction duration that also the related arc decreases with increasing engine speed.

The effect of boosting pressure on the flame extinction arc is instead reported in Figure 18 and Figure 19: it can be observed that with 80% and 100% of NG in the fuel mixtures, the flame extinction arc remains almost constant or slightly increases as function of MAP, while the trend is decreasing with 40% and 60% of NG in the fuel mixtures; in particular for the 40% NG mixture at MAP=1.6 bar the flame extinction arc decreases of about 10 CAD and almost equals that of gasoline at MAP=1 bar (Figure 18 and Figure 16); at higher engine speed the flame extinction arc becomes, in some cases, even lower than that of gasoline (Figure 19 and Figure 16); these trends can be explained, once again, with the two conflicting effects of pressure and temperature on LBV and in turn on the turbulent burning velocity; moreover, during the extinction phase the flame front reduces its temperature because of both the cooling effect of the approaching combustion chamber walls and the expansion phase.

The NG content in the fuel mixture has a small effect on the flame extinction both for MAP=1 bar and 1.2 bar while, for higher MAP levels, increasing the gasoline content in the fuel mixture significantly reduces the flame extinction arc (Figure 17); This strong effect of fuel mixture composition on combustion extinction arc, almost absent in the previous parts of the combustion (FFD and FFP), motivated a further analysis exposed in the following section.

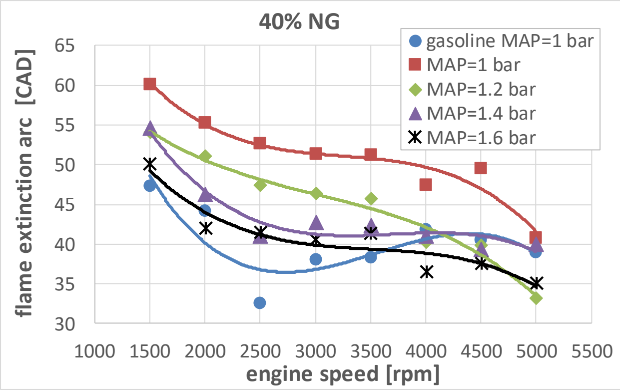


Figure 16 - flame extinction arc as function of engine speed, 40% NG

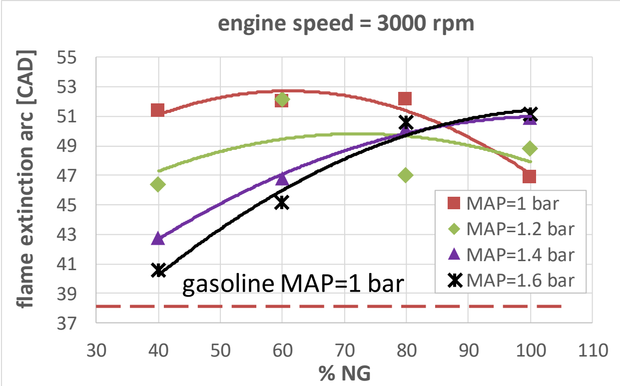


Figure 17 – flame extinction arc as function of %NG at 3000 rpm

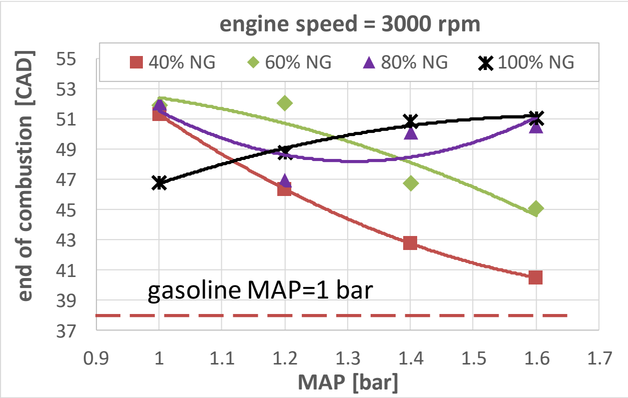


Figure 18 – flame extinction arc as function of MAP at 3000 rpm

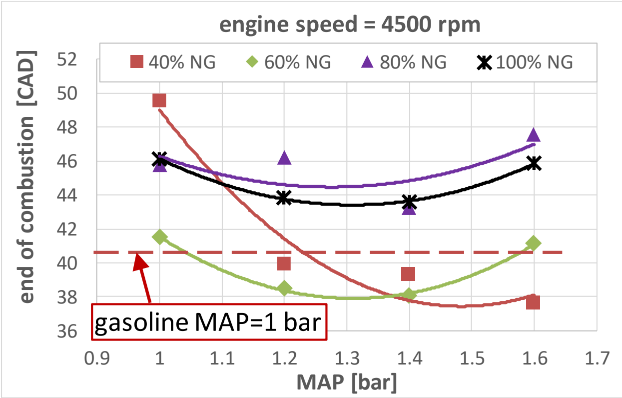


Figure 19 – flame extinction arc as function of MAP at 4500 rpm

From this detailed analysis two important conclusions can be drawn: in the engine tested, about one fourth of the whole combustion arc is needed for the flame front development, one fourth for the fast flame propagation and the remaining half is instead needed for the flame extinction phase; this means that in the engine tested the last 20% of MFB involves the same combustion arc of the first 80%; the other important result is that the 10 CAD shortening of combustion arc due to MAP increase for the 40% NG mixture (Figure 4) regards only the flame extinction phase, since neither the flame front development nor the fast flame propagation phase are remarkably shortened by MAP increase. The shortening effect of MAP is higher when gasoline content increases in the fuel mixture and it is concentrated in the last 20% of MFB; to better highlight this effect Figure 20 reports the MFB and combustion speed diagrams obtained with the 40% NG mixture at different MAP levels and at 3000 rpm; the curves are plotted only in the final part of the combustion where their trend becomes different.

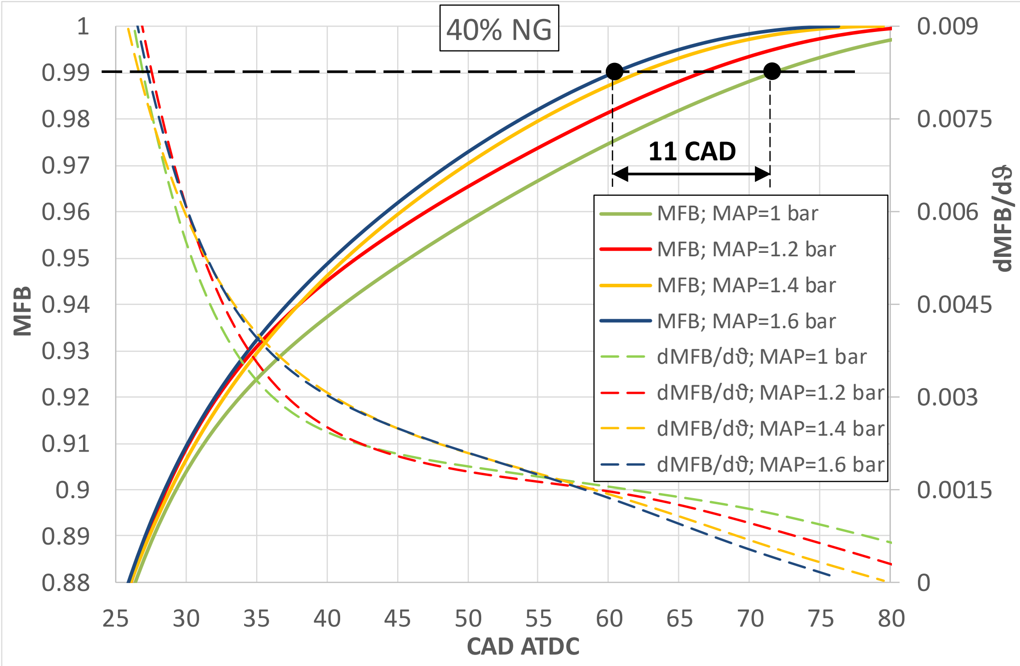


Figure 20 – MFB and combustion speed in the combustion extinction zone (40% NG, different MAP)

In the range between 30 CAD and 55 CAD ATDC it is evident the combustion speed increase with MAP (dashed lines in Figure 20), which, in turn, causes a faster combustion extinction. A plausible hypothesis to justify this behavior, in addition to the higher temperatures, could be a sort of controlled knocking of the gasoline-NG mixture; when MAP increases, the higher pressure and temperature of the end gas probably lead gasoline to auto-ignition, but the high octane rating of the NG in the fuel mixture [8] mitigates this phenomenon avoiding macroscopic knocking: the final effect is that the local auto ignition of gasoline accelerates the combustion extinction. This hypothesis could explain why the MAP increase produces greater effects in fuel mixtures with higher gasoline content and mainly in the final part of the combustion.

5. Effects on engine efficiency

In a SI engine, for a given operating condition of speed and load, the Indicated Thermal Efficiency (ITE) is influenced by different factors: the combustion phasing, its duration and the A/F ratio. If the combustion duration is fixed, the ITE is greatly influenced by combustion phasing that depends on spark advance. As mentioned before, knock occurrence prevented from optimal combustion phasing when low NG concentration and high boost pressure were adopted in DF operation. As a result, even with a faster flame propagation, the indicated efficiency revealed lower because of the retarded combustion. Figure 21 shows both LPP and ITE of the 40% NG mixture at different MAP levels: as can be observed, increasing boost pressure forced to retard the combustion (LPP increased) and consequently, despite the reduction of combustion arc (Figure 4), the ITE decreased. When the flame propagation speed increases at the end of combustion, i.e. in the last 20% of the MFB curve, there are no remarkable effects on engine ITE; on the contrary, when the flame propagation speed increases during the rapid combustion phase, i.e. during the 10-80% of MFB curve, the positive effect on engine ITE is greater.

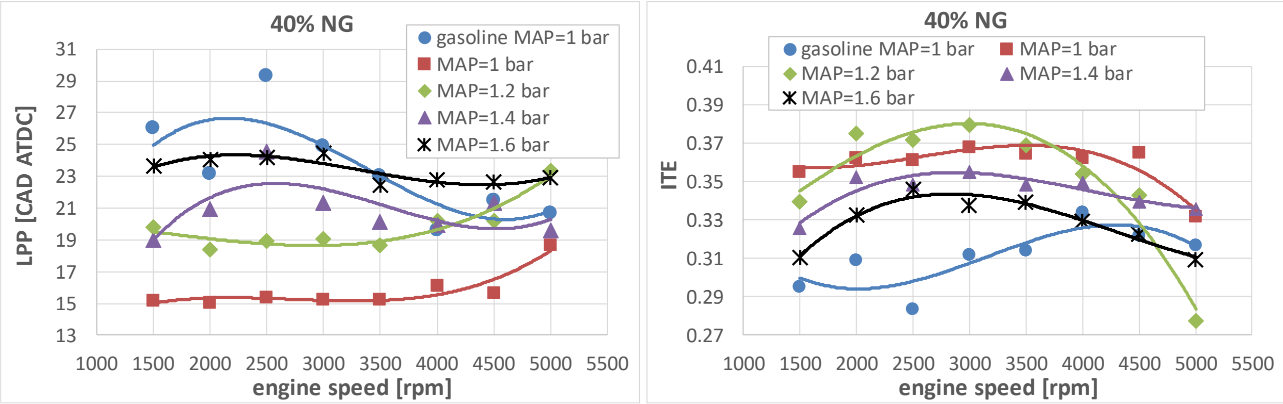


Figure 21 – LPP and ITE for the 40% NG mixture at different MAP

Figure 22 shows, on the top left, the engine ITE obtained at MAP=1 for the different fuel mixtures tested; as already mentioned the DF mixtures and 100% NG produced higher engine ITE, compared to gasoline, due to the stoichiometric air-fuel ratio and to a better combustion phasing; it can also be noted that the 60% NG and 40% NG allowed to obtain higher engine efficiencies than 100% NG and this can be explained only with a faster combustion. The bottom section of Figure 22 shows that both LPP and whole combustion arc of the various DF mixtures are quite similar but observing the top right of Figure 22 it is evident that fast flame propagation arc is reduced with increasing gasoline content in the DF mixture with variations up to 5 CAD, which represents a 20% of the average FFP arc (about 25 CAD). It can be concluded, hence, that the 20% reduction of the fast flame propagation phase obtained increasing the gasoline content in the fuel mixture, with the same air-fuel ratio and combustion phase, produced a remarkable engine ITE increment, about 5% in the analyzed cases, while a 20% reduction of the flame extinction, obtained by supercharging, produced negligible positive effects on engine ITE, widely counterbalanced by the negative effects of a retarded combustion phase.

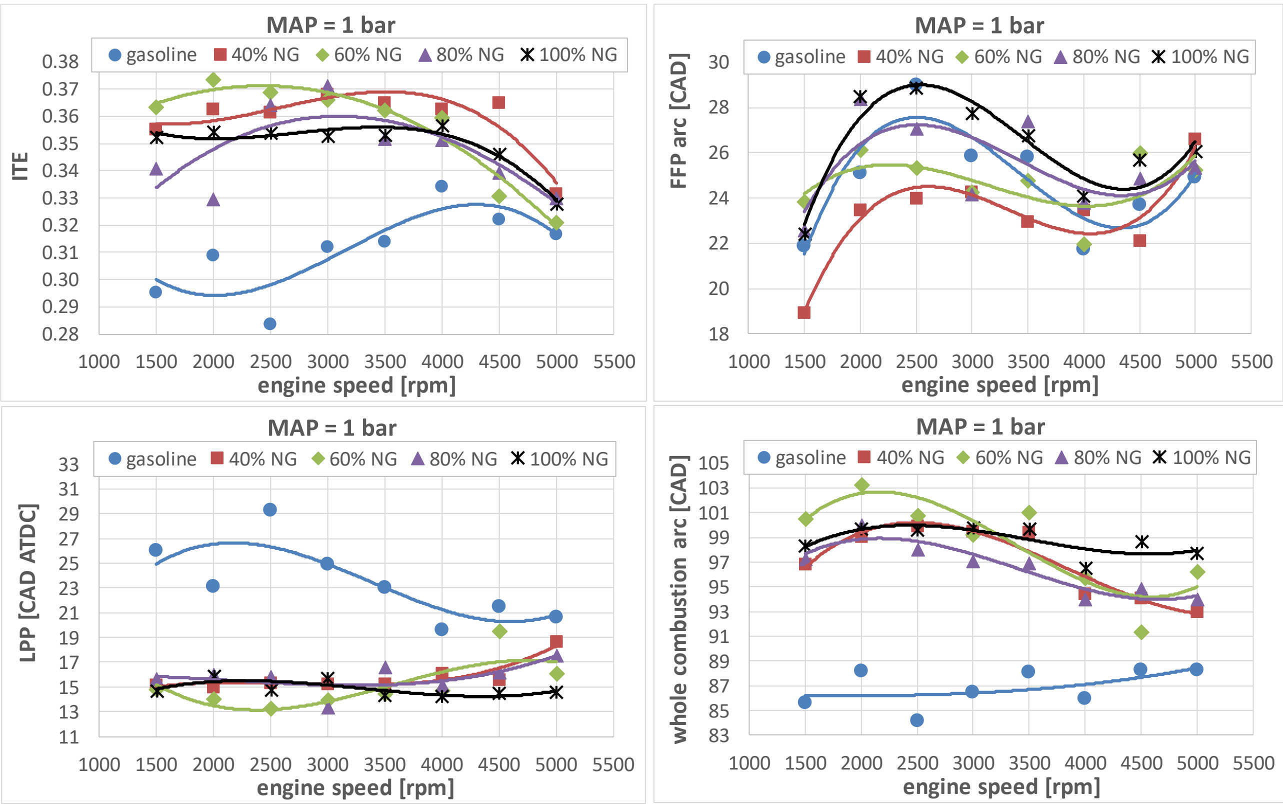


Figure 22 – ITE, LPP, FFP and whole combustion arc for the different fuel mixtures at MAP=1 bar

Conclusions

This work shows the results of an extensive series of experimental test carried out on a spark ignition engine, supercharged by means of a Roots compressor, and fueled with several different mixtures of natural gas and gasoline, with the aim to study the effects of engine speed, supercharging pressure and fuel mixture composition on the flame front propagation speed. The combustion arc has been divided in three parts to deeply investigate the combustion evolution. A MAP increase produced, for each mixture tested, an increase of the combustion speed and this effect revealed stronger with higher gasoline content in the fuel mixture; in particular, the reduction was found to be concentrated in the last part of combustion (80%<MFB<99%), the effect on engine indicated efficiency revealed limited and widely counterbalanced by the retarded combustion phase. On the contrary, the aspirated engine (MAP=1 bar) showed a reduction of the fast flame propagation arc (10%<MFB<80%) when increasing gasoline content in the fuel mixtures and this produced up to 5% indicated engine efficiency increments. Regarding the engine speed of rotation, its strong effect on the flame propagation speed produced an almost constant combustion arc; more in detail, increasing engine speed produced an increase of both flame front development arc and fast flame propagation arc, which however resulted counterbalanced by a decrease of the flame extinction arc.

These are indeed very interesting results that bring new knowledge on the combustion behavior of gasoline - natural gas fuel mixtures and represent the basis of a future research work in which the experimental results presented in this paper will be used to properly calibrate a dedicated combustion model.

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SYMBOLS AND ABBREVIATIONS

ATDC = After Top Dead Centre
A/F = Air/Fuel ratio
(A/F)ST = Stoichiometric Air/Fuel ratio
BMEP = Brake Mean Effective Pressure
BSFC = Brake Specific Fuel Consumption
BTDC = Before Top Dead Centre
BTE = Brake Thermal Efficiency
CAD = Crank Angle Degrees
CNG = Compressed Natural Gas
CO = Carbon Monoxide
DAQ = Data Acquisition
DF = Double-Fuel
ECU = Electronic Control Unit
FFD = Flame Front Development
FFP = Fast Flame Propagation
GDI= Gasoline Direct Injection
HC = Hydrocarbon
IC= Internal Combustion

IGBT = Insulated Gate Bipolar Transistor
IMEP = Indicated Mean Effective Pressure
IMEPm = Measured IMEP
ITE = Indicated Thermal Efficiency
LBV= Laminar Burning Velocity
LHV= Lower Heating Value of the fuel
LPG= Liquefied Petroleum Gas
LPP = Location of Pressure Peak
MAP = Manifold Absolute Pressure
MFB= Mass Fraction Burned
MON = Motor Octane Number
NG= Natural Gas
NO_X = Nitrogen Oxide
PID = Proportional Integral Derivative
PM= Particulate Matter
ppr = pulse per revolution
RON = Research Octane Number
SOC = Start of Combustion
SI = Spark Ignition
TDC= Top Dead Centre
THC = Total Hydrocarbon
UEGO = Universal Exhaust Gas Oxigen

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37 John B. Heywood, “Internal Combustion Engine Fundamentals, Second Edition”, 2018, McGraw-Hill Education, ISBN: 9781260116106.

The Reviewers' comments have been numbered and are reported BLACK while the authors' rebuttals are reported in RED. All the changes in the revised manuscript are written in RED.

Reviewers' comments: **Reviewer #1**

The authors have performed a Detailed combustion analysis of a supercharged double-fuel spark ignition engine. The subject is interesting, and the archival nature of the work is clear. However, this is not translated to the manuscript where several issues are verified which need to be fully amended prior submission.

The **discussion section** is a main concern on the paper since it lacks referencing, explanations and, in several cases, when an explanation is given, it is contradictory. Please, find a list of some issues that were verified in the manuscript that can be useful to improve your work. Given the extent of the issues that were verified, I do not recommend the paper for publication.

Introduction

1) What does it mean combustion arc? It does not seem a usual term from the engine field.

The expression "combustion arc" has been employed by the authors to indicate the angular interval of the combustion (which is hence expressed in Crank Angle Degrees, CAD), different from "combustion duration" which instead indicates the time interval (i.e. expressed in seconds) of the same combustion; with the aim to avoid any confusion between the two concepts, the authors preferred to adopt different terms to denote the "combustion time interval" and the "combustion angular interval", above all in consideration of the very different behavior that the two parameters may exhibit: as example, the graphs in Figure 3 show that the time interval of the combustion (i.e. the combustion duration) changes substantially with the engine speed, while, passing to the crank angle domain, the combustion arc (i.e. the angular duration of the combustion) is almost constant with the engine speed.

A clarification has been properly introduced in the revised version of the paper.

2) Please, when do you refer to an engine parameter, specify if it is brake or the type of indicated (gross or net). It is helpful to understand the comparison basis that is considered. For example, when you refer to efficiency in the abstract.

In the revised version of the paper the authors paid more attention clarifying whether the indicated or the brake thermal efficiency is concerned.

It has been also clarified that, being useless the power drained by the supercharging system, only indicated efficiency could be evaluated in the supercharged Double Fueled engine.

As regards the abstract, the efficiency increments obtained in the previous works refer both to the naturally aspirated and to the supercharged Double Fueled spark ignition engine: in the first case, brake thermal efficiency is concerned, while in the second (as explained) the indicated efficiency is dealt with. For this reason, in the abstract, the authors could not specify which kind of efficiency is mentioned when referring to previous works.

3) Avoid the use of too many references in the same line.

The authors apologize for the references, but when the referenced papers deals with same topic the citations must be placed in the same line and the results commented together.

4) "The pollutant emissions of an IC engine are strongly related to fuel properties and also to the combustion process. Gasoline fueled engines mainly emit unburned hydrocarbons (HC) and carbon monoxide (CO), in particular when a rich air/fuel (AF) mixture is used in order to prevent knocking. Diesel engines emit high levels of particulate matter (PM) and nitrous oxides (NOx)." This is an oversimplification. Gasoline engines also produces significant concentrations of NOx emissions. It may be easier to deal with them using TWC, nonetheless, their levels are still significant. Modern GDI also struggles with high particulate emissions. Please, comment.

Although in some operating conditions (high loads) Spark Ignition (SI) engines produce high levels of NOx emissions, this kind of pollutant is easily cut down by the three-way catalyst: as a matter of fact, the SI has never suffered for relevant problems related to NOx emissions reduction. Diesel engines instead emit high levels of NOx also in the lower load region, and their control requires more sophisticated (and expensive) devices (i.e. SCR catalyst); the reasons are mainly related to the operation under overall lean and non-homogeneous A/F mixture, which causes wide availability of both oxygen and nitrogen, and to the maximum local temperatures during combustion, which are difficult to control and activate the dissociation phenomenon: as a result, NOx reduction has always been a great concern for car manufacturer, which, in the last five years, reduced the production of Diesel passenger car, above all in the small displacement section. The use of non-homogeneous mixture is also the cause for the strong production of PM which affects Diesel engines combustion. Port Fuel Injected (PFI) SI engines are not characterized by relevant PM emissions (which are not regulated for such engines) since operated always with homogeneous mixtures, while GDI engines emit relevant PM only when operated in the stratified charge mode (i.e. with non-homogeneous mixtures); in any case, Diesel engines emit abundantly higher levels of PM compared to PFI and GDI SI engines, which hence required the adoption of proper devices (e.g. DPF). In the introduction section, the authors briefly resumed the above-mentioned considerations. A more thorough comparison between the emissions of Diesel and SI engines has been added to the text.

5) "Gas fueled engines (NG or LPG), compared to the above mentioned, exhibit lower CO and HC emissions and negligible levels of PM; this result is obtained thanks to the high knock resistance of gaseous fuel that allows a stoichiometric AF ratio in all engine operative conditions." I would expect higher CO and HC giving the same compression ratio, since its is harder to ignite and also NG has a lower flame speed. Please, comment.

As is known, when running at high-full load condition, the gasoline spark ignition engine is fuelled with very rich mixtures with the aim to cool the air charge (exploiting the heat of vaporization of gasoline) and avoid dangerous knocking phenomena: as a result, mixtures enrichment in the order of 20%-25% are operated at full load; this is the reason why gasoline spark ignition engines exhibit at the full load condition, a higher fuel consumption with respect to medium load; such rich mixtures cause very high levels of carbon monoxide (CO) and hydrocarbons (HC) in the engine out emissions, and very low oxidizing efficiency of the three-way catalyst: as a result, the most part of the CO and HC produced by the engine is also emitted by the tailpipe.

Such mixture enrichments are not operated when the spark ignition engine is fueled by gaseous fuels, for two main reasons: firstly, gaseous fuels are characterized by a considerably higher knock resistance with respect to gasoline (gasoline MON=85, LPG MON=93, CNG MON=122) which makes their combustion less critical; secondly, gaseous fuels have already lost their heat of vaporization (being already gaseous), and hence a mixture enrichment would not produce the cooling effect obtained with gasoline. In the full load condition, gaseous fueled spark ignition may be however operated with slightly rich mixtures (3%-5%) with the aim to maximize flame propagation speed and hence engine power, but the resulting CO and HC emissions are very low with respect to a gasoline operation. Moreover, the better mixing properties of gaseous fuels with respect to liquid injected fuels also allows to obtain very good and homogeneous air-fuel mixtures, with the result of a more complete combustion, resulting in lower HC emissions and negligible PM formation.

Such clarification has been introduced in the revised version of the paper.

6) I would rather use Dual Fuel than Double fuel for DF since it is more conventional in the internal combustion engine community.

In the Double Fuel operation, the two fuels are homogeneously mixed with air and burn simultaneously through the same flame front; this is quite different from the well-known Dual Fuel combustion (operated in Compression Ignition engines), in which a small quantity of diesel fuel is directly injected into the premixed air-gaseous fuel mixture, and its auto-ignition acts as igniter to start the flame propagation combustion of the air-gaseous fuel charge.

Such explanation has also been introduced in the revised version of the paper

7) "This operating mode can be easily implemented in bi-fuel engines, thanks to the double injection system available on board, and only requires a software editing of the Electronic Control Unit (ECU)." That is partially true. If you are injecting both fuels separately, other barriers appear such as the introduction of an additional fuel tank, the consumer preference, etc.

Maybe a misunderstanding occurred, since, as reported in the text cited by the reviewer, bi-fuel engines are concerned in the Double Fuel operation: such engines are already endowed of two separate and complete injection systems, including two separate fuel tanks, hence no barriers appear.

The consumer is not involved, since the DF operation described by the authors should be implemented by the Electronic Control Unit (ECU) which, in turn, should have been properly programmed to this purpose. The DF operation would allow a better exploitation of the energy resource on account of the lower specific consumption obtained by the stoichiometric overall mixture.

With the aim to avoid other possible misunderstandings, the authors further clarified these concept in the revised version of the paper.

8) The introduction section needs further refinements:
1. It must be clearer.
2. The authors have used mainly their own references during the introduction to discuss about the subject. Please, add references from other research groups that reinforces your claim.
3. The English grammar need to be improved.

The introduction has been reviewed and improved and more references from other research groups have been added.

Experimental Setup:
9) Are the authors accounting the energy required by the supercharger to reach the desired pressure in the efficiency calculations? Please, add comments about this in the manuscript.

The power drained by the supercharging system was not employed in the calculations (the reasons are explained in the revised version of the paper), hence in the supercharged DF operation the authors could evaluated only indicated efficiency.

10) 10°C is a significant variation for a given operating condition. Please, discuss the effect of having this variation in the results.

The authors apologize for the misunderstanding, the deviation of 10°C refers to the total number of experimental tests, i.e. all engine speeds, mixtures composition and boosting pressures tested. Higher inlet temperatures were obtained at higher boosting pressures, hence the test performed at the same MAP are perfectly comparable.

A clarification has been introduced in the revised version of the paper.

11) Please, add a table with the accuracy of your experimental setup.

A table (Table 5) with the accuracy of the experimental setup has been added in the text.

Results
12) Figure 3 A and C are not needed in the discussion.

Although it may appear that Figure 3 A and C are not essential in the discussion, the authors believe their presence is fundamental to better understand the influence of engine speed, and in turn of induced turbulence, on combustion evolution; besides the mentioned figures have been used to clarify the difference between the time duration and the angular duration of the combustion.

13) Please, don't use the term combustion arc.

As already explained, the expression “combustion arc” is employed by the authors to indicate the angular interval of the combustion, which differs from the time interval of the combustion, usually indicated with “combustion duration”. Since, for the sake of clarity, in this paper it is important to maintain a separation between the two parameters, the authors were obliged to adopt different expressions. “Combustion arc” was considered the shortest and most effective choice. In alternative the expression “angular interval of the combustion” could be employed, but this would have a very bad impact on the readability and clarity of the paper.

14) Combustion duration is generally defined by using the values from CA90-CA10, where the numbers stands for the MF values. If you observe your graphs from Figure 3, you can state that your combustion lasts for 30-40 CAD. This is reasonable and agrees with the literature. The values presented in Figure 6 and Figure 7 are not representative and infers an excessive uncertainty in their reporting. Please, modify this.

Although it is true that the combustion phase going from 10% to 90% of MFB brings the greatest thermodynamic effects, in this paper the authors presented a very thorough description of combustion analyzing all the phases involved (flame development, fast propagation and extinction), in particular in Figures 12, 13, 14 and 15 the fast flame propagation phase has been analyzed that involves percentages of MFB between 10% and 80% and represents the almost linear part of the MFB diagram.

15) General comment. Please, review all graphs to include the units of the parameters. Consider including them in the legend too. For example, MAP= 1.4 bar.

All the graphs have been reviewed and now include the units of the parameters.

16) The presentation of the paper needs to be improved. In general, it has low graph quality, the labels are not in capital letters, etc.

The graph quality has been improved.

17) Which kind of mass fraction burned is presented. Is it heat transfer accounted?

In this paper the authors always referred to the mass fraction burned, which is proportional, through the fuel lower calorific value, to the heat released by the combustion, frequently indicated as Gross Heat Released. When the heat transfer to the wall is accounted for, hence the heat received by the gas is concerned, which usually denoted as Net Heat Released.

18) It is not possible to see the differences between some of the engine speeds.

The graphs have been improved to better highlight all the quantities.

19) "In Figure 6 the combustion arc is plotted as function of MAP at the fixed engine speed of 3000 rpm". Figure 6 labels indicates NG as variable.

In Figure 6 each curve represents the combustion arc (reported in the ordinate axis) as function of MAP (in the abscissa). The several curves differ from each other for the NG concentration: hence the NG% is the curve parameter, which is constant for each curve.

20) "the flame front development, going from Spark timing to MFB=10%". Up to 10%, correct?

Yes, correct

21) Please, reference your assumptions regarding the flame development phases and the respective MFB for them.

It is widely accepted that the 10% MFB corresponds to the end of the flame development phase as far as from that point the MFB trend becomes almost linear up to 80% MFB; also the reviewer #1 at point 14 says: “Combustion duration is generally defined by using the values from CA90-CA10, where the numbers stands for the MF values” confirming that CA10 is generally known as the end of combustion development phase and the start of rapid combustion phase.

22) "Figure 8 shows the crank angle covered by flame front development (FFD) as function of engine speed, evaluated at different supercharging pressures and for the 80% NG fuel mixture (a similar trend has been observed also for the other fuel mixtures tested): as can be noted, the FFD arc increases with engine speed for all the fuel mixtures and supercharging pressures tested; this happens because, in its early stage, the flame front propagation is less influenced by turbulence and mainly depends on pressure and temperature. For this reason, increasing engine speed, the flame front development arc revealed an average overall increment of about 10 CAD from the 1500 rpm to the 5000 rpm condition." If in the early phases, the combustion is influenced by the pressure and temperature and not by the engine speed, why you see an opposite trend in the graphs? In several cases the authors are crossing concepts to justify the results.

For a better understanding, it is useful to remember what has been observed in terms of whole combustion arc (i.e. the angular interval related to the whole combustion) and whole combustion duration (i.e. the time interval related to the whole combustion) in the graphs of Figure 3 at the beginning of the paper: when the MFB (or its derivative) is represented in the time domain, a great variation is noted with the engine speed, while when the representation is in the crank angle domain, it can be observed an almost null variation of the combustion arc. This was explained considering the strong effect played by the turbulence on the speed of combustion, whose duration (seconds) revealed hence reduced almost proportionally with respect to the engine speed, thus giving as final result an almost constant combustion arc. When the effect of the turbulence is strong, the combustion is accelerated and hence its duration reduces.

The same reasoning applies to the FFD: the stronger is the effect of turbulence (which increases with engine speed), the lower will be the variation of the angular interval related to the FFD. Since the graph of Figure 8 denotes a substantial variation of the FFD arc with respect to the engine speed, it can be concluded that the effect of turbulence was not so strong to maintain an almost constant FFD arc. And this is consistent with the well-known theory of the turbulent flame propagation: when, at the beginning, the flame front is very small, it does not take advantage of the turbulence level.

With the aim to avoid misunderstanding, the author introduced a clarification in the revised version of the paper.

23) "oscillating trend (with constant mean value)". How is this mean value calculated?

With "oscillating trend (with constant mean value)" the authors meant that the trend is neither decreasing nor increasing but roughly constant; the mentioned “mean value” has not been mathematically evaluated (because it would be useless) but only roughly estimated.

24) The result section must be improved. There are several misconceptions and explanations that are not fully correct. The text description does not follow what is observed in the graphs. In some

cases, the authors refers to the impact of a variable as negligible and the other as important. However, analyzing the graph, the opposite is verified. Please, correct these discussions.

It would be better to exactly know which graphs the reviewer is referring to. If the reviewer refers to the graphs where the combustion arc is plotted against engine speed, the explanation of the misunderstanding is fairly simple: when the authors say that the engine speed has a great influence on combustion time duration this is not in contrast with the graphs of Figure 4 where the combustion angular duration (the combustion arc) is almost constant with increasing engine speed and the reason lies in the obvious difference between the time duration of a phenomenon and its angular duration. In other words, if the engine speed would not affect at all the combustion time duration, this should remain constant among the different engine speeds, and the values representing the combustion arc in figure 4 would increase with a direct proportionality to the engine speed (i.e. doubling the engine speed should imply a double combustion arc if the combustion duration is constant); the constant trend observed in the combustion arc of figure 4, instead, means exactly the contrary: increasing the engine speed produced such an increase of the combustion speed (and hence such a reduction of the combustion duration) to give the almost constant trend of the combustion arc shown in figure 4. The other graphs, obtained at constant engine speed, have been employed by the authors to eliminate the effects of engine speed on combustion duration and to highlight the effects of other variables such as MAP and fuel mixture composition.

Reviewer #2

General Comments

This paper presented a series of experimental tests performed on a supercharged, spark-ignition engine, using different mixtures of natural gas and gasoline. The goal was to study the effects of engine speed, supercharging pressure and fuel mixture composition on the combustion speed. By dividing the combustion temporal arc in three portions, the authors were able to investigate the evolution of the combustion and eventually its effects on indicated thermal efficiency. This work is definitely of interest to the community especially due to the limited literature regarding Double Fuel engine operation.

1) The level of the English language used in this manuscript is overall acceptable with some minor flaws scattered throughout the paper. Some examples are provided here below in the "General Comments" section of this review, while some others are listed in the "Specific Comments" section. This reviewer advises for a thorough, final proofreading of the manuscript to ensure the grammar and orthography are appropriate.

The manuscript has been revised and the English language has been improved

2) The authors often use the expression "Thanks to", which is somehow informal. One example of a more appropriate form for such situations would be "due to". Please consider replacing all instances of "Thanks to".

The authors kindly appreciated the reviewer recommendation and modified the paper accordingly.

3) Throughout the manuscript, there is an over-abundance of commas. One example is in the following sentence where the comma is not needed before the two instances of the word "that": "To resume, the combustion speed is influenced by engine speed, that promotes turbulence inside the combustion chamber, by supercharging pressure, that influences pressure and temperature during combustion, and finally by fuel mixture composition." Similar minor mistakes can be found elsewhere in the manuscript.

   The minor mistakes have been corrected throughout the manuscript.

4) Aside from reporting that the injection systems are of the PFI type for both gasoline and natural gas fuels (Table 1), the author do not explicitly mention it anywhere in the text. The reviewer's opinion is that this is a very important detail that should be made clear in the manuscript.

   The authors modified the manuscript pointing out in the “Experimental setup” section that the engine involved is a bi-fuel PFI spark ignition engine.

   In general, the manuscript should be accepted for publication after providing all the minor revisions needed to address the comments in this review.

   Specific Comments

Introduction

5) * In the sentence "… this result is obtained thanks to the high knock resistance of gaseous fuel that allows a stoichiometric AF ratio in all engine operative conditions.", the correct form is "… engine operating conditions". This minor mistake is found in other instances in the paper. Please correct each one of them.

The authors gratefully acknowledge the reviewer for its recommendation and modified the paper accordingly.

6) * The authors mention their previous work (refs. [13] and [14]) to provide an example of the benefits of using Double Fuel (gasoline + natural gas) operations. It would be helpful if the authors also reported a brief note about the relative mass split between the two fuels. This would help to provide an idea of how much natural gas allowed for the benefits they reported.

As indicated in the revised version of the paper, in the gasoline-natural gas DF operation, the best efficiency was obtained with 50% in mass of natural gas, while for the peak performance a 30% in mass of natural gas was necessary.

7) * The authors should spell out the THC acronym at its first use. In addition, this acronym is used only once throughout the manuscript, which makes its use not needed.

The authors corrected the manuscript substituting THC with HC.

8) * A suggestion to improve the readability of this sentence: "In medium/high load conditions, the supercharged gasoline engines, to avoid knocking, operate with rich A/F ratio and retarded combustion thus producing high fuel consumption and pollutant emissions."
The authors should consider rephrasing the sentence above as follows: "In medium/high load conditions, the supercharged gasoline engines operate with rich A/F ratio and retarded combustion to avoid knocking, thus producing high fuel consumption and pollutant emissions."

The whole sentence has been re-organized following the reviewer suggestion.

9) * In the sentence: "When gasoline and natural gas are mixed together a brand new fuel is obtained which combustion speed depends on mixture composition", please replace "which" with "whose".

The manuscript has been corrected.

10) * In the sentence: "To resume, the combustion speed…" there is a chance that "resume" is not the correct verb. Is it possible that the right verb is "To summarize…"? If yes, please edit this sentence.

The authors gratefully acknowledge the reviewer for its recommendation and modified the paper accordingly.

Experimental Setup

11) * In the sentence: "…a more detailed description of the setup is however available on a previous work [15] dealing with…", please replace with "…a more detailed description of the setup is however available in previous work [15] which deals with…".

The authors modified the paper following the reviewer recommendation.

12) * In Table 1, please change the units of displacement from [cc] to [cm3]. Also, please indicate that 1242 cm3 is the total displacement of all four cylinders combined.

The unit has been corrected and total displacement indicated in the table.

13) * In Table 2, please correct the square bracket for the fuel density which is currently formatted as a superscript.

The density unit has been corrected.

14) * In Table 2, the authors report an 85% volumetric concentration of Methane for the Natural Gas fuel. Is there any chance they can report the overall composition by mentioning any other hydrocarbon species and their concentrations? This would be very useful for the replicability of the current study as well as for future computational studies.

As suggested by the reviewer, the full composition of the natural gas employed in the test has been introduced in Table 3 (formerly Table 2) – Natural Gas properties.

15) * In Table 3, can the author add a note with the range of spark advance? While it is clear why they maximized it, a numeric reference in Table 3 would allow the reader to put the range into perspective.

In Table 4 (formerly Table 3) two rows have been added to show the range of spark advances used in the tests, the first refers to pure gasoline operation while the second refers to the DF mixtures (including 100% NG).

16) * In Figure 1, the DAQ acronym has been used, but its formal introduction is only reported one page later. The authors should spell out the DAQ acronym in the caption of Figure 1 for clarity.

The Figure caption has been modified according to the reviewer comment.

Results and discussion

17) * In the analysis of Figure 8, the authors express the duration of the flame front development in crank angle degrees. It is possible to notice that at 5000 rpm the FFD duration increases to less than

double the value at 1500 rpm. However, the engine speed at 5000 rpm is more than three times as large, which implies that the actual FFD duration is decreasing in terms of absolute time. The authors' discussion should highlight this phenomenon.

A detailed discussion has been added to the text to address the mentioned phenomenon.

18) * All the analyses and plots refer to the average cycle and therefore do not report any information on cyclic variability of the investigated quantities. Have the authors though about showing the observed variability by means of bars or shaded areas. It would be very interesting to understand if variability depends on any of the parameters, such as MAP or engine speed and would add to the value of the manuscript.

When the authors investigated the indicated parameters (indicated mean effective pressure (IMEP) and indicated thermal efficiency (ITE)) the effect of cyclic variability was explored by evaluating both IMEP and ITE from each single pressure curve sampled: then the average and the dispersion could be easily calculated, and the results are already published in their previous paper on the supercharged DF spark ignition engine. In this paper instead, for each operating condition, the MFB has been calculated on the average pressure cycle, which means that the different MFB values needed to compute the three parts of the combustion arc (i.e. 10% MFB, 80% MFB and 99% MFB) have been evaluated only once for each operating condition. To arrange a cyclic variability investigation would require, instead, repeating the whole analysis on each single MFB curve evaluated for each operating condition, and the number of numeric values to analyze and display would be difficult to manage; moreover, in the authors opinion, adding dispersion bars to the current graphs would also detract from the clarity the presentation. On the basis of these observations, and considering that the amount of information contained in the graphs already presented in the paper is very large, the authors excluded to perform also the cyclic variability analysis in this paper. Nonetheless this is indeed a very interesting hint for a further development of this research work, which will be seriously taken into consideration by the authors.

19) * In Figure 20, please add a legend that clearly identifies what the continues and dashed lines represent and which axis they should be read on.

In figure 20 a legend has been added to clarify what the continuous and dashed lines represent and now it is clear also which axis they refer to.

Reviewer #3

This paper presents a detailed analysis of engine speed, natural gas fraction, and boost pressure effects on the burn duration of a double-fueled natural gas / gasoline SI engine. The authors do a thorough job on presenting the data and explaining some of the main trends and conclusions. This reviewer recommends the work be published but has some recommendations that could help improve the clarity and impact of the manuscript.

1) Is "double-fueled" the correct terminology? I have seen "dual-fueled" in prior literature, but perhaps this is reserved for compression ignition applications only. Is there a distinction in terminology between SI/CI for dual-fuel applications or are the authors simply keeping consistent with prior work they presented? The term "bi-fuel" is also used in the manuscript. For the sake of clarity, they authors may want to list all of these synonyms together and note that they can be used interchangeably.

The authors first experimented the simultaneous combustion of gasoline and CNG several years ago (2008), when no literature references were available on this topic. The terminology Double Fuel was hence selected to identify the flame propagation combustion in a homogeneous charge composed by air and by the two fuels. The Dual Fuel combustion instead is the combustion realized in compression ignition engines where a small quantity of the most reactive fuel (e.g. Diesel fuel) is directly injected within a homogeneous mixture of air and less reactive fuel (e.g. methane): the auto-ignition (due to compression) of the most reactive fuel acts as igniter to start the flame propagation combustion of the less reactive fuel homogeneously mixed with air. The Double Fuel concept is hence quite different from the Dual Fuel, and this is the reason why the authors maintain a different terminology.

The bi-fuel engines are instead engines (usually spark ignition engines) which are endowed of two separate fueling and injection systems, e.g. gasoline and natural gas or gasoline and LPG; since the two injection systems are already available, the Double Fuel combustion can be easily implemented in such engines by means of a simple ECU programming.

Such explanations/clarifications have been introduced in the revised version of the paper.

2) In general, there are numerous grammatical inconsistencies throughout the work. Although English is likely a second language for the authors, the overall flow and presentation of information should be improved, perhaps through a third party editing service and at a minimum, during the last stage of journal editing.

The manuscript has been revised and the English language has been improved

3) The introduction section is fairly well written and provides a good, concise summary of prior work in the field. The following comments are therefore intended to clarify and strengthen the impact of the work. There are claims about substantially reducing (presumably engine-out) CO and THC emissions, but these would normally be managed by the the three-way-catalyst (TWC) under stoichiometric operation. Therefore, the authors should make clearer what the advantage is in reducing these emissions for DF NG-gasoline engines. Does it help with conditions when the engine is operated outside stoichiometry and the TWC is not as efficient? In my opinion, the real benefits of this technology would focus on any gains in fuel economy or reductions in engine-out

PM. For example, a very low PM emitting SI engine might avoid needing to use a gasoline particulate filter to meet future regulations. Have there been measurements for particulates in these DF engines before or in other work? If so, what are the quantified benefits? If the benefits are large, these should be emphasized even more during the introduction to motivate the research area.

The main advantage of Double Fuel combustion relies in the possibility to operate the engine with overall stoichiometric mixture even at full load; gasoline fueled engines, instead, when running at high-full load condition, employ very rich mixtures with the aim to cool the air charge (exploiting the heat of vaporization of gasoline) and avoid dangerous knocking phenomena: as a result, mixtures enrichment in the order of 20%-25% are operated at full load; this is the reason why gasoline spark ignition engines exhibit at the full load condition, a higher fuel consumption with respect to medium load; such rich mixtures cause very high levels of carbon monoxide (CO) and hydrocarbons (HC) in the engine out emissions, and very low oxidizing efficiency of the three-way catalyst: as a result, the most part of the CO and HC produced by the engine is also emitted by the tailpipe. Such mixture enrichments are not operated when the spark ignition engine is fueled by gaseous fuels, for two main reasons: firstly, gaseous fuels are characterized by a considerably higher knock resistance with respect to gasoline (gasoline MON=85, LPG MON=93, CNG MON=122) which makes their combustion less critical; secondly, gaseous fuels have already lost their heat of vaporization (being already gaseous), and hence a mixture enrichment would not produce the cooling effect obtained with gasoline. The possibility to adopt overall stoichiometric mixture with the Double Fuel combustion hence allows to strongly reduce CO and HC emissions as well as the fuel consumption.

As regards the PM, the authors agree with the reviewer: running always with overall stoichiometric mixture with air certainly produces a noticeably reduction in the particulate formation; unfortunately, the authors could not measure both PM and NOx, hence there are not data available to validate this. The engine test bed is well equipped and would be ready to perform these measurements, but the lack of a proper exhaust gas analyzer is the obstacle.

4) The results and discussion section is one long section without any logical break up to help the reader easily digest the content. It is recommended that the authors create a few subsections to aid in reading through the manuscript. One logical approach to this would be sections focusing on (i) whole combustion arc, (ii) flame front development arc, (iii) flame front propagation arc, etc.

The result and discussion section has been divided as suggested by the reviewer.

5) Figure 3 - Is the result of proportional flame front propagation speed with engine speed surprising or unique? Given that the turbulence intensity in the cylinder near TDC for non-swirling charge motions is proportional to mean piston speed, this falls nicely within theoretical expectations (see Heywood, pg. 341, Equation 8.23). Perhaps a note on this should be added to the manuscript indicating that this falls within expectations. Also, the MFB and dMFB at 1503 RPM seems to fall slightly outside the scatter of other curves. Is this still within the expected range or is there an explanation for this outlier?

The proportionality between flame front propagation speed and engine speed falls within theoretical expectations (a reference to the suggested formula by Heywood has been added to the text), what is not so obvious is the so perfect inverse proportionality between combustion time duration and engine speed that produces an almost constant combustion arc (CAD) as function of engine speed for the tested engine. The curves of MFB and dMFB at 1500 rpm fall slightly outside the scatter of other curves but this falls within the expected range and there is not a specific reason; a similar trend has been observed in some other cases with random curves falling outside the common scatter, in other cases again all the curves are packed together.

6) Figure 4 to 6 - What does the "gasoline" line pertain to in the figures? Is this the gasoline combustion arc corresponding to MAP = 1 only? This should be clarified in this and all subsequent figures.

The gasoline dashed line refers to the combustion arc obtained with gasoline at MAP=1 bar as is already stated in the text; all the figures have been modified to better clarify this point.

7) In general, the results are very thorough, but there seems to be a lack of physical explanation or interpretation of all the results which would help the reader better understand the overall characteristics of DF NG-gasoline engines. The authors could greatly improve the quality and impact of the manuscript with better descriptions of why they are seeing certain individual trends in the data and compare these with simplified theoretical expectations (or predictions, if possible). Some examples of these points will be provided below.

Considering the flame front development arc analysis (FFD in Figures 8 to 11):

9) It is not clear (to me) why the FFD arc, which is most closely related to laminar flame speed, is so strongly influenced by engine speed (Figure 8), can this be explained with simple laminar flame speed theory or engine effects on the main parameters governing laminar flame speed?

The explanation is quite simple: if the engine speed had no influence at all on laminar flame speed the FFD time duration would be perfectly constant as function of engine speed and, as a consequence, in the diagram of figure 8 the graphs should increase with a direct proportionality as function of engine speed because they report the trend of FFD angular duration and not the FFD time duration. Observing figure 8 it is clear that: although engine speed increases more than three times (from 1500 to 5000 rpm), the FFD arc increases less than two times and this means that the increasing engine speed slightly reduces the FFD time duration. This clarification has been added in the text.

10) Why does FFD arc decrease with an increase in MAP (Figure 9)? Theoretically, increasing the pressure will decrease laminar flame speed and increase the FFD arc. There must be compensating effects of temperature due to the higher pressure, changing spark timing, etc. This should try to be explained rather than simply stating what was observed. Perhaps including the governing equation for laminar flame speed could help to discuss these impacts better.

The increasing pressure has the effect of reducing laminar flame velocity (LFV) and this should increase the FFD arc, as correctly noted by the reviewer; a possible explanation is the contrasting effect of increasing temperature due to both the reduced thermal exchanges and the changed spark timing. This explanation has been added to the text.

11) What about the interesting trends in FFD vs. MAP at different engine speeds? Why do NG-containing fuels have a larger FFD arc than gasoline at higher engine speed (i.e., Figure 11, 4000 RPM)? Methane has a higher laminar burning velocity than gasoline and should have shorter FFD arc? Is a simple explanation complicated due to different spark timings and thermodynamic conditions at the start of combustion? If so, this should be noted.

Methane has a lower laminar burning velocity than gasoline, as can be found in literature (and shown in the revised version of the paper), so there is no surprise in finding higher FFD arc than gasoline; nonetheless the thermodynamic conditions plays a fundamental role in the determination of the actual FFD arc; a detailed discussion about this topic has been added to the text.

12) What causes the local min/max in Figures 10 & 11? Are there competing laminar flame speed effects (i.e., reduction in FFD due to higher temperatures because of increased MAP but then overtaken by the detrimental effect of pressure on flame speed)?

The local min/max are determined indeed by contrasting effects of pressure and temperature that are influenced in turn by engine speed, that modifies heat exchanges and turbulence, spark timing and MAP. A wide explanation has been added to the text.

13) Would it be possible to do simple flame speed calculations to explain these trends better?

It is not clear if the reviewer refers to the calculation of the “real” flame speed propagation, which is turbulent, or if the reviewer means the calculation of the laminar flame speed in the thermodynamic conditions of the gas inside the cylinder. In the first case, a two-zone combustion model should be adopted to obtain the turbulent flame propagation from the experimental MFB curve, but this obviously goes far beyond the scope of this paper. In the second case, the estimation of the probable laminar flame speed would require the evaluation of the unburned gas temperature as function of the crank angle and for each operating conditions: this large amount of calculations however would not add relevant information with respect to the simple proportion (already traced in the revised version of the paper) between the laminar flame speed of gasoline and natural gas: considering an average relative A/F ratio (lambda) of 0.9 for gasoline operation, the laminar flame speed at ambient conditions is 49.5 cm/s, while for stoichiometric operation with natural gas at the same thermodynamic conditions the laminar flame speed is 37.5 cm/s. This simple evaluation show that gasoline is characterized by roughly a 30% higher flame speed.

14) Since burn duration is related to laminar burning velocity, perhaps a typical values for gasoline and methane should be included in Table 2.

According to the reviewer suggestion, typical values of the laminar burning velocity for both gasoline and natural gas have been introduced in Table 2 and Table 3 (formerly Table 2).

15) In general, the other analysis areas (i.e., FFP and flame extinction arcs) could benefit from deeper/more coherent explanations of the trends as described above.

A deeper explanation of the trends observed in the graphs has been added to all the analysis areas (FFP and flame extinction arcs)

16) Please also check your abbreviation list as not all of the ones indicated in the manuscript appear.

The abbreviation list has been checked.