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Influence of fuel bio-alcohol content on the performance of a turbo-charged, PFI, spark-ignition engine

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ABSTRACT

In this work, the behavior of an engine running with binary and ternary mixtures of n-butanol, ethanol and gasoline has been investigated.

Analyses have been performed at different engine speed both at low load and at high load. For each blend, the air to fuel ratio has been kept stoichiometric, while the spark time has been tuned in order to maximize the engine brake torque and hence the engine fuel conversion efficiency. The performance, the combustion characteristics and the pollutant emissions of the engine fueled by biofuel mixtures have been compared to those characterizing the engine running with neat gasoline.

In an attempt to provide a guideline for the development of engines running with every mix of gasoline and alcohol, measurements are presented as a function of the oxygen content of the fuel. When the fuel oxygen content increases, results show that the optimal spark time must be retarded at part load while must be advanced at high load. At this operation, the maximum obtainable efficiency increases, the CO_2 specific emissions decrease almost linearly together with NO_x and HC specific emissions, the engine torque remains practically the same, while the brake specific fuel consumption increases.

1. Introduction

Due to the increase of the economic activities and the wider use of vehicles especially in developing countries, the demand for transport fuels is increasing. On the other hand, stricter rules for the control of both pollutant and climate-altering emissions induce to improve the conventional fuel performance and to use alternative fuels in conventional internal combustion engines. From a technological point of view, downsizing, gasoline direct injection, variable valve timing and actuation, exhaust gas recirculation, variable compression ratio are some of the approaches aimed at improving the thermal efficiency of spark ignition engines. Some of these technologies, coupled to the usage of bio-fuels, could lead to significant reduction in CO_2 emissions.

Biofuels used in lieu of gasoline also reduce global CO_2 emissions. Compared to pure gasoline, the massive use of bio-alcohol as a transportation fuel would save significant amounts of fossil fuels, while reducing greenhouse gas emissions by almost half. On the other hand, conventional gasoline engines are 20–30% less efficient than Diesel engines. Therefore, the need to improve the fuel economy of gasoline engines is a major challenge to meet future CO_2 emission targets for passenger cars.

For spark ignition engines, there is a wide range of biofuels alternative to gasoline. Today, ethanol and butanol are among the most discussed [1].

Bioethanol already contributes from 20 to 30% of the fuel market in the United States and Brazil due to its accessibility, low cost and compatibility with modern engines without modifications [2]. Biobutanol in its various isomeric structures can be used as a fuel for conventional engines. Today, it is considered to be of great interest since its physical properties are very similar to those of gasoline [3].

Ethanol and butanol can be obtained from various types of crops, food crops (first generation biofuels) and non-food crops (second generation biofuels). They can derive both from thermo-chemical processes [4] and fermentation [5]. In recent years, several technologies that convert cellulose into butanol and ethanol are in the R & D phase [6]. As an example, the ABE process allows to produce acetone, n-butanol and ethanol typically in a proportion of 3:6:1 by mass [7,8].

Many studies are available in literature on the use of alcohol-gasoline blends in spark ignition engines. In general, they focus

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on the use of binary blends of an alcohol with gasoline. The optimization of the engine operating parameters according to the fuel characteristics is not always considered. Issues related to emissions are particularly investigated. As an example, Rice [9] compared the exhaust gas emissions using butanol, ethanol and methanol. In particular, the emissions of CO, NO_x and HC have been detected by a 4-cylinder Chrysler engine in different conditions with 20% by volume of alcohol/gasoline. It has been observed that with a mixture of alcohol/gasoline a CO reduction was obtained, caused by the lower stoichiometric air-fuel ratios of the alcohol fuels due to their partially oxidized nature.

In Ref. [10], the influence of butanol addition on the emissions of a gasoline spark-ignition engine has been investigated. By running the engine at stoichiometric air/fuel mixtures, it has been found that the specific emissions change very little with the alcohol content of the blend. However, a 40% butanol/60% gasoline blend minimizes the HC emissions, while a greater alcohol content increases them. It is shown that the NOx emissions strongly depend on the operative air to fuel ratio. Stoichiometric air to fuel mixtures with an alcohol content equal to 80% produce less NOx than gasoline. It has also been found that the addition of butanol improves the combustion stability and reduces the ignition delay.

Numerical analyses by Scala et al., show the thermal efficiency of a spark ignition engine burning stoichiometric mixtures tends to grow when ethanol or butanol is added to gasoline [11]. This is mainly due to the greater laminar flame speed of alcohols compared to that of gasoline. As a consequence, the spark timing of an engine running with gasoline-alcohol mixtures has to be retarded to achieve its best thermal efficiency. Many of the results reported in the present paper confirm this trend, enlarging the analysis to numerous engine operating points and different fuel alcohol contents.

Szwaja and Naber stated n-butanol burns faster than gasoline at the same conditions approaching the combustion to an ideal constant volume process [12]. This circumstance has been deepened in the present paper analyzing the influence of the engine load on the combustion duration with different kinds of fueling.

Further experiments and theoretical calculations showed that ethanol added fuels show reduction in carbon monoxide (CO), carbon dioxide (CO_2) and nitrogen oxide (NO_x) emissions without significant loss of power compared to gasoline. But it was measured that the reduction of the temperature inside the cylinder increases the hydrocarbon (HC) emission [13]. Ethanol blends in lower proportions showed an increment in the range of 2.31–4.16% for the engine torque and 0.29–4.77% for the brake power. Brake specific fuel consumption (BSFC) increased for higher volume of ethanol content in the range 5.17–56%. The brake thermal efficiency (BTE) increased in the range 2.5–6% when ethanol–gasoline blends were used [14].

Fewer studies have been carried out regarding the use of ternary mixtures. Many of these concern engines firing with gasoline-ethanol-methanol mixtures. In Ref. [15], it is shown that ternary blends (6% ethanol, 6% methanol and 88% gasoline) and binary blend (12% ethanol, 88% gasoline and 12% methanol, 88% gasoline) allow marginally improving both engine thermal efficiency and engine power compared to pure gasoline. Similar results have been found in Ref. [16] where lower rate of blends (3–10 vol% of bio-ethanol and methanol in gasoline) have been considered. Results show also that pollutant emissions decrease when these ternary blends are used instead of pure gasoline.

Also gasoline-nbutanol-methanol mixtures have been investigated [17]. Results show that ternary blends improve the engine performance compared to neat gasoline. Considering that Acetone-Butanol-Ethanol mixture (ABE) is an intermediate product in the ABE process, blends of pure gasoline and ABE (0%–80% vol. ABE) have been also considered [18]. Tests have been done at a given engine operating point, using different air to fuel ratio but keeping constant the spark timing. Results show the emissions of unburned hydrocarbons increase with respect to pure gasoline while CO emissions decrease. The efficiency changes with respect to gasoline according to the ABE content in the fuel mixture.

The interest in different alcohol blends comes from the circumstance that current biofuel production technologies lead to mixtures in which different alcohols can be present in variable quantities. Furthermore, the diffusion of biofuels will lead to the presence on the market of alcoholic mixtures produced in various ways and in different compositions. Such mixtures can be used as such or, more presumably, will be mixed with gasoline in order to reduce the dependence on fossil fuels. As said above, ethanol-gasoline blends are widely used as a transportation fuel in numerous countries (as an example in Brazil, USA, North Europe, India, etc.). Naturally, the alcohol content in the mixtures varies according to the local market. Furthermore, research activities on bio-butanol, bringing into light the good potential of butanol as a gasoline mixing fuel, have increased the interest in this kind of biofuel. Thus, both the bio-alcohol content present in the fuel mixture and its chemical composition could change among different areas of use.

In this scenario and in order to assess the potential of different kinds of bio-alcohol mixed with gasoline and different bio-alcohol content in the mixtures, in this paper, the behavior of a spark ignition engine, running with some binary n-butanol-gasoline mixtures, ethanol-gasoline mixtures and ternary n-butanol-ethanol-gasoline mixtures, has been investigated.

In most of the papers here cited, engine performances are evaluated at single engine operating points and optimizations have not often been proposed. In this paper, a systematic analysis of the engine fired with alcohol-gasoline blends, together with the optimization of the spark advance is presented. Investigations have been carried out varying the engine speed both at low load and at high load. According to the fuel blend burnt, the engine control parameters have been tuned in order to obtain the best thermal conversion efficiency with stoichiometric air-to-fuel ratios. In an attempt to generalize the results, the main results are presented as a function of the oxygen content of the fuel mixture. The idea is to provide a guideline for the development of flexible engines able to fire different gasoline-alcohol blends according to the available market in different countries. As it has been reported in the paper, many fuel properties that influence the engine behavior depend almost linearly on the oxygen content of the fuel mixture.

2. Experimental setup

2.1. Setup

Tests have been performed on a turbocharged, port fuel injected, spark-ignition engine (Table 1). The experimental apparatus (Fig. 1) has been widely described in Ref. [19] and in Ref. [20]. Briefly, the

Table 1
–Main engines specifications.

Model	4 Cylinders, 16 Valves Turbocharged SI Engine
Turbocharger group	IHI RHF3
Displacement	1368 cm^3
Bore/Stroke/Con. Rod	72/84/129 mm
Compression Ratio	9.8
Max Power (ISO Conditions)	110 kW @ 5500 rpm
Max Torque (ISO Conditions)	206 Nm @ 2250 rpm

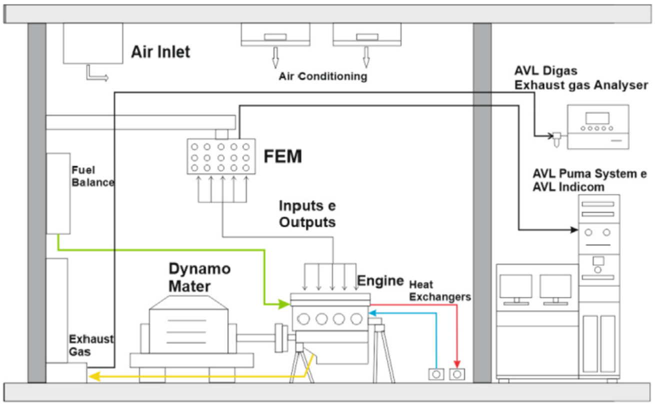

Fig. 1. Test bench sketch.

engine is coupled to an eddy current dynamometer (FE 260 Borghi&Saveri) that allows controlling the engine speed by measuring the delivered engine torque. The main pressures and temperatures characterizing the engine operation have been monitored during the tests. By means of several Front End Module (FEM), an AVL Puma 5.3 System controls the tests providing also to the data acquisition. The fuel consumption has been measured by an Assing fuel balance (inaccuracy less than 1%), while a Horiba UEGO sensor has been used to measure the air excess ratio (inaccuracy less than 4%).

The in-cylinder pressure curves relative to the first cylinder of the engine have been acquired by means of a quartz pressure transducer (AVL GM14D, sensitivity 19 pC/bar) mounted flush to the combustion chamber. For each engine operating point, 500 cycles have been acquired by means of an AVL Indicom system. Data have been sampled every one crank angle degree when the engine operated at part load, every 0.2 crank angle degrees when the engine ran at full load. Engine emissions have been measured by an AVL Digas device; it allows to measure HC (resolutions < 100 ppm), NOx (<10 ppm) CO (<0.01%) and CO2 concentrations (<0.1%).

An on-line programmable engine control unit allowed tuning both the injection timing and the ignition timing during the tests.

2.2. Analyzed fuel mixtures

Several kinds of alcohol-gasoline blends have been considered (Table 2). Fuels have been mixed by mass content.

In some engines, ethanol can give corrosion problems when it is present in high concentrations. Thus, for ethanol, the maximum concentration used is 10%.

The fuel oxygen content increases when the alcohol content increases. The G50B40E10 fuel, characterized by a bio-alcohol content of 50%, has an oxygen content of about 12%.

Based on pure fuel's data, some properties of the blend can be easily calculated according to the mixture composition (Table 2). In particular, the lower heating value and the latent heat of vaporization are weighted mass average values. The stoichiometric air to fuel ratio is calculated according to Ref. [21]. Density has been calculated assuming that the total volume of the mixture is the sum of the volumes of individual components. The heat released per unit of stoichiometric air-fuel mixture has been calculated assuming the complete oxidation of the fuel blend. The CO_2 amount produced per unit of heat released has been calculated in the same way.

The stoichiometric air-to-fuel ratio linearly decreases with the oxygen content. The lower heating value also decreases when the oxygen content increases, while both density and latent heat of vaporization increase. Regardless of fuel composition, these trends are almost linear.

2.3. Test procedure

The engine behavior has been evaluated both at medium load and at high load considering medium-low engine speeds (Table 3).

Since the engine output can be handled by means of both the throttle valve and the boost pressure, the engine load has been

Table 2
Alcohol – Gasoline Blends. For gasoline a mean chemical composition has been considered.

		G100B0E0	G85B15E0	G90B0E10	G78B15E7	G60B40E0	G50B40E10
Gasoline-butanol-ethanol mass content	%	100-0-0	85-15-0	90-0-10	78-15-7	60-40-0	50-40-10
H/C atomic ratio		1.87	1.940	1.938	1.994	2.077	2.167
O/C atomic ratio		0.0	0.029	0.031	0.053	0.083	0.122
Carbon Content	kg/kg	0.865	0.832	0.830	0.808	0.778	0.744
Hydrogen content	kg/kg	0.135	0.136	0.135	0.135	0.136	0.135
Oxygen content	kg/kg	0.0	0.032	0.035	0.057	0.086	0.121
Stoichiometric Air to Fuel ratio	kg/kg	14.6	14.1	14.0	13.7	13.2	12.7
Lower heating value	MJ/Kg	43.4	41.85	41.83	40.75	39.28	37.7
Latent heat of vaporization	kJ/kg	350.0	382.9	407.4	423.1	437.8	495.1
Density	kg/m^3	750	758	755	762	773	778
Heat released per unit of stoichiometric air-fuel mixture At ISO condition	MJ/m^3	3.459	3.450	3.456	3.448	3.434	3.429
CO_2 per heat released	g/MJ	73.0	72.8	72.6	72.5	72.5	72.1

Table 3
Test cases.

Engine operating point	Engine speed	Manifold Absolute Pressure	Air excess
	[rpm]	[bar]	[–]
N = 1500/Map = 0.66	1500	0.66	1
N = 2000/Map = 0.66	2000	0.66	1
N = 3000/Map = 0.66	3000	0.66	1
N = 1500/Map = 1.12	1500	1.12	1
N = 2500/Map = 1.22	2500	1.22	1
N = 3000/Map = 1.08	3000	1.08	1

referred to the absolute pressure in the intake manifold (Map). Actually, at standard operating conditions and at a given engine speed, this parameter determines the air mass flow entering the cylinder, i.e. the engine torque output. Moreover, many engine control units use lookup tables indexed by both engine speed and Map.

At medium load, the engine has been throttled so that the pressure of the intake manifold is equal to 0.66 bar. At this operating point, the engine running with straight gasoline delivers a torque of about 60 Nm (about the 30% of the maximum torque). At high load, the throttle is fully open and the engine is supercharged.

Both at low and high load, the fuel injection time has been adjusted so to feed the engine with a stoichiometric mixture. In this way, the catalyst can cut down the pollutants released by the engine independently of the mixture used.

During the tests, the ignition angle was changed in order to determine the point of maximum brake torque (hence of maximum engine brake thermal efficiency). Measurements were made once the stationary operating conditions were reached.

For each mixture, three test campaigns were carried out. Results shown in the following are the averaged values derived from these test campaigns.

Each fuel mixture was prepared in a fuel tank as usual for onboard vehicle applications. The mixtures have been well mixed at the test beginning. However, stratification effects due to the different density of the fuels could lead to a certain dispersion of the engine performances.

3. Results

As the alcohol content changes in the fuel mixture, the behavior of the engine has been investigated considering its optimal operating conditions, i.e. running the engine with the spark advance that maximizes the brake thermal efficiency.

Considering the partial load operation, Fig. 2 shows the measured brake thermal efficiency as a function of the spark advance. By varying the fuel mixtures, the engine response is not univocal. In general, the engine efficiency tends to increase as the bio-alcohol content increases in the fuel mixture. G50B40E10 blend and G60B40E0 blend reach the higher efficiencies. In two cases out of three, the G85B15E0 mixture shows an efficiency curve lower than that of pure gasoline. This can be explained considering that alcohols speed up the combustion process, so the optimal spark advance tends to decrease when the fuel alcohol content increases. Furthermore, the alcohol greater latent heat of vaporization tends to decrease the charge temperature and therefore the heat losses through the cylinder walls.

At high load, engine optimization is conditioned by the knock onset and by the maximum acceptable temperature of exhaust gases. Knock onset limits the maximum spark advance allowable, while the exhaust gas temperature determines the maximum air to fuel ratio.

As an example, Fig. 3 shows the engine brake thermal efficiency, the knock intensity and the gas temperature measured when the spark advance is changed for the N = 2500/Map = 1.22 case.

Knock intensity has been evaluated according to Ref. [21]. Knock events excite the combustion chamber to resonate at its natural frequency. The subsequent oscillations in the acquired in-cylinder pressure curve can be used to measure the knock intensity of an individual cycle. In particular, for each engine cycle, the cylinder pressure signal has been filtered. The knock intensity of an individual cycle is calculated by comparing the maximum amplitude of pressure oscillations (i.e. the so called *mapo*) with a threshold value ($mapo_{th}$) calculated by means of a statistical approach. To take into account the cycle-to-cycle variation, a knock index relative to an engine operating point is calculated by the following relationship:

$$KI = \frac{\sum_{j=1}^{N_{cyc}} (mapo - mapo_{th})}{Mapo_{th} \bullet N_{cyc}} \bullet 100$$

where N_{cyc} is the number of acquired cycles. This relationship considers both the percentage of individual knocking cycles and the extent of knocking events relative to the given engine operating point. Previously analyses [22] have shown that a knock index equal to 5 is acceptable for this engine.

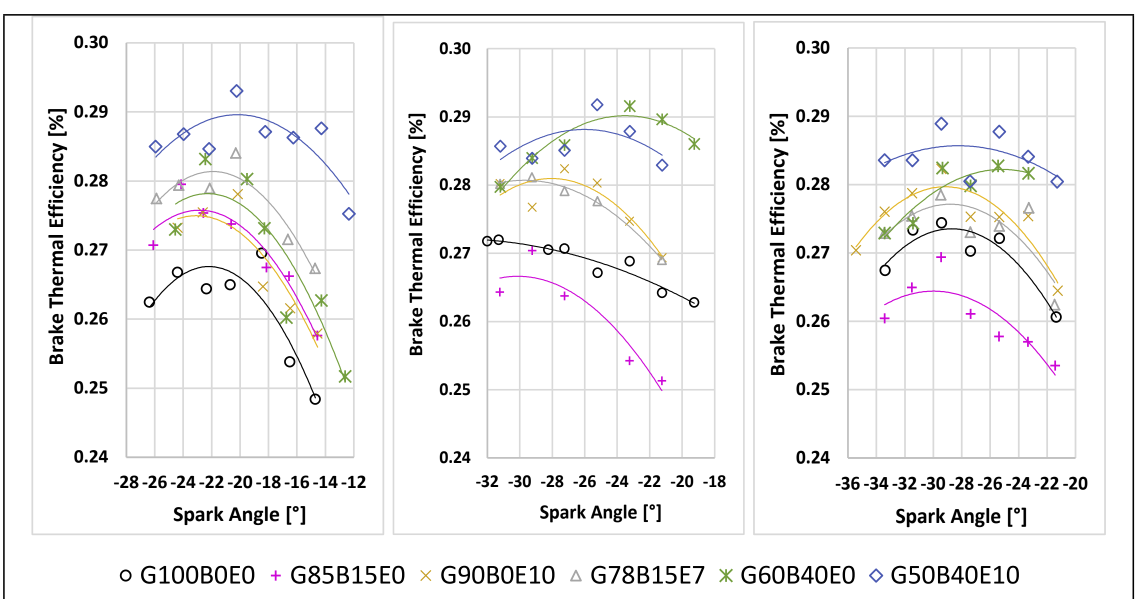

Fig. 2. Brake thermal efficiency as a function of the spark angle for different fuel blends. Partial load: N = 1500/Map = 0.66 (left), N = 2000/Map = 0.66 (middle), N = 3000/Map = 0.66 (right).

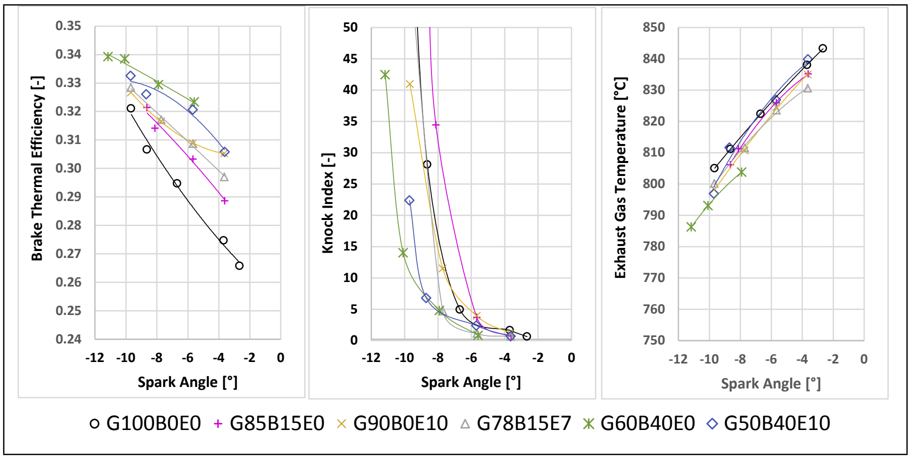

Fig. 3. Brake thermal efficiency, knock intensity and mean exhaust gas temperature for different fuel blends. Test case: N = 2500/Map = 1.22.

The optimum ignition angle cannot be reached due to knock events. The spark advance that maximizes the efficiency is knock limited; it depends on the alcohol content of the mixture. The exhaust gases temperature is always below 900 °C (i.e the maximum temperature allowable for the turbine), thus it is not necessary to run the engine with rich mixtures.

For all the cases, Fig. 4 shows the spark advance that maximizes the brake thermal efficiency as a function of the fuel oxygen content.

At low load, the optimal advance tends to decrease almost linearly with the oxygen content. This result agrees with what was found by means of numerical analyses in Ref. [11]. However, the differences are small. Even when the engine runs with high-alcohol mixtures, it could operate with the base spark advance set for gasoline without evident efficiency penalizations.

At high load, the spark advance maximizing the efficiency tends to grow almost linearly with the fuel oxygen content. As previously mentioned, at high load the spark advance is limited by the knock onset. Alcohols have greater resistance to detonation than gasoline, therefore the knock limited spark angle is more anticipated when the alcohol content in the fuel mixture increases.

For the cases with a low boost pressure (N = 1500/Map = 1.12 and N = 3000/Map = 1.08), the optimal spark advance varies little as already seen at partial loads. For the N = 2500/Map = 1.22 case, the allowed spark advance grows more significantly when the fuel oxygen content increases.

As already shown in Figs. 2 and 3, the maximum obtainable brake thermal efficiency marginally increases with the fuel oxygen content (Fig. 5) both at low load and at high load.

At low load, the linear regression curves are almost overlapping. This highlights a low dependence of the efficiency on engine speed. The G50B40E10 blend achieves a maximum efficiency about two percentage points higher than that obtained with pure gasoline. At high load, the efficiency gain is even more pronounced.

On average, fuels characterized by high fractions of biofuel (G60B40E0 and G50B40E10), increase the maximum brake thermal efficiency with respect to pure gasoline by about 10% at high load and about 7% at low load.

Fig. 6 shows the torque values measured at the optimal spark timing. Despite the differences in the lower heating values (Table 2), the engine torque is practically unaffected by the presence of bio-alcohol in the fuel. In fact, the engine burns a volume of air-fuel mixture approximately equal to the engine displacement. Table 2 shows that the heat released by the same volume of air-gas-alcohol mixture is practically constant for the examined cases.

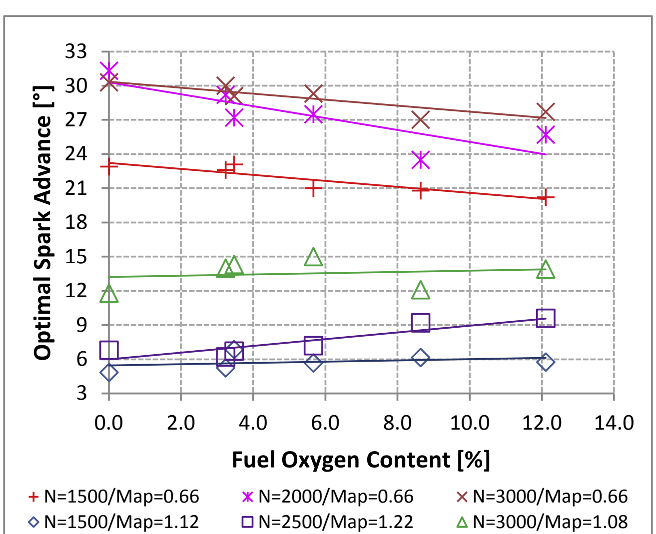

Fig. 4. Optimal spark advance versus the fuel oxygen content.

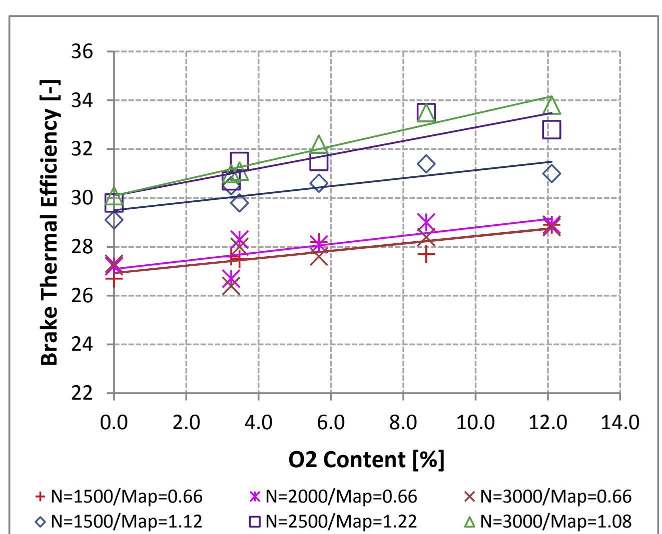

Fig. 5. Maximum brake thermal efficiency at low and at high loads.

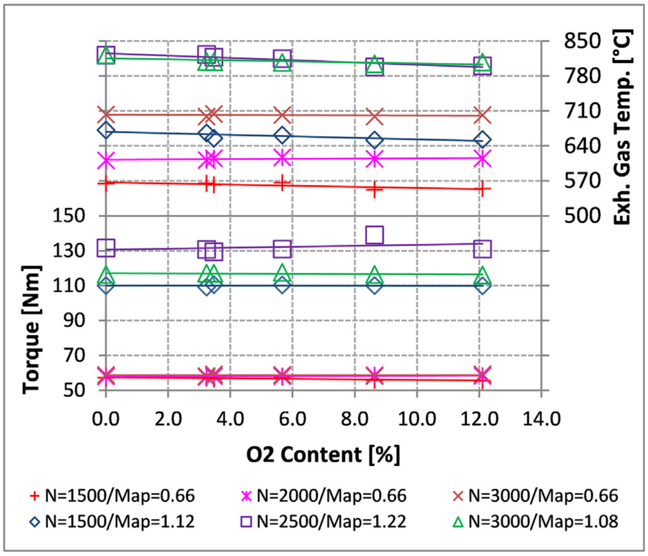

Fig. 6. Torque and exhaust gas temperature at optimal spark advance.

The measured exhaust gas temperature is also reported in Fig. 6. The exhaust gas temperature depends on both the maximum temperature reached during the combustion phase and the spark angle. Advancing the start of combustion, it ends earlier, increasing the effective gas expansion inside the cylinder. As a consequence, the gas temperature is lower at the exhaust valve opening.

At part load, when the alcohol content in the fuel increases, the greater latent heat of vaporization tends to lower the maximum temperature reached into the combustion chamber. On the other hand, spark angles are more retarded. As a result of conflicting effects, the exhaust gas temperature does not vary with the fuel burned. At high load, exhaust temperatures decrease almost linearly with the fuel oxygen content. This result can be justified considering both the greater latent heat of vaporization of alcohols compared to that of pure gasoline and the greater spark advances characterizing the engine fueled with alcohol-gasoline blends.

Fig. 7 show the combustion durations characterizing the engine at the points of maximum brake thermal efficiency. Conventionally, the combustion development is calculated as the duration necessary to burn the first ten percent of the fuel mass, while the turbulent combustion duration refers to the fuel mass burning in the range 10–90%. As usual, the burnt fuel mass curves have been derived from the in-cylinder pressure curves by means of the so-called heat release analysis.

The laminar flame speed value affects the combustion development. At the same conditions, alcohols have a laminar flame speed some higher than gasoline [11]. Furthermore, at part load, due to the lower spark advances, the charge temperature in the ignition zone is higher when the engine burns alcohol-gasoline blends. As a consequence, the duration of the first combustion phase marginally decreases with increasing the fuel oxygen content.

Vice versa, at high loads the engine running with gasoline-alcohol blends operates with more advanced spark timing. Due to also the greater latent heat of vaporization of alcohols, at the ignition time, the charge temperature decreases according to the fuel alcohol content. Therefore, the laminar flame speed decreases in the ignition area and the development combustion phase tends to marginally grow.

The turbulent flame propagation phase is mainly controlled by the turbulence intensity at the flame front. Then, the duration of the second combustion phase depends on the regime and on the load, but not on the chemical composition of the charge.

The measured specific fuel consumption is shown in Fig. 8. This parameter can be useful in order to evaluate the operating cost of the engine. Both the lower heating value and the stoichiometric air to fuel ratio decrease when the fuel alcohol content increases. As a consequence the specific fuel consumption increases with the oxygen content of the fuel blend. As in previous cases, the trend is roughly linear.

Fig. 9 shows the CO_2 specific emissions measured when the engine runs at the optimal spark advance with different alcohol-gasoline blends.

These specific emissions have been derived from the composition of the exhaust gases measured at the engine outlet. According to the data shown in Table 2, CO_2 decreases almost linearly with the oxygen content of the fuel. For blends with higher alcohol

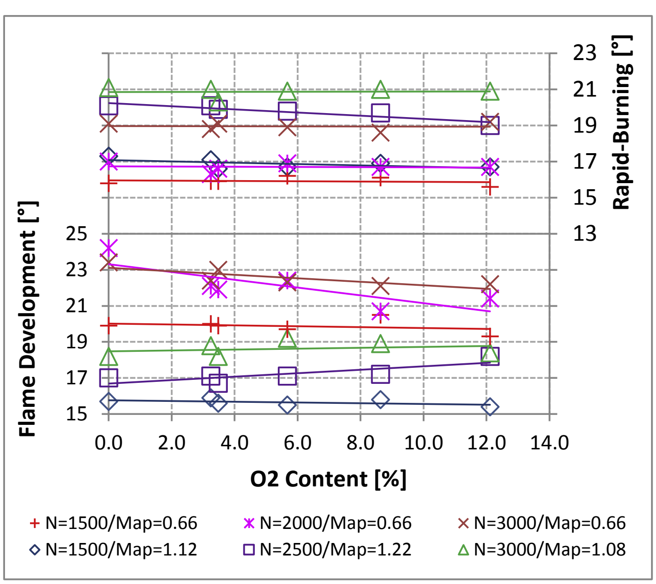

Fig. 7. Combustion durations (flame development angle and rapid-burning angle) versus the oxygen content of the fuel mixture at optimal spark advance.

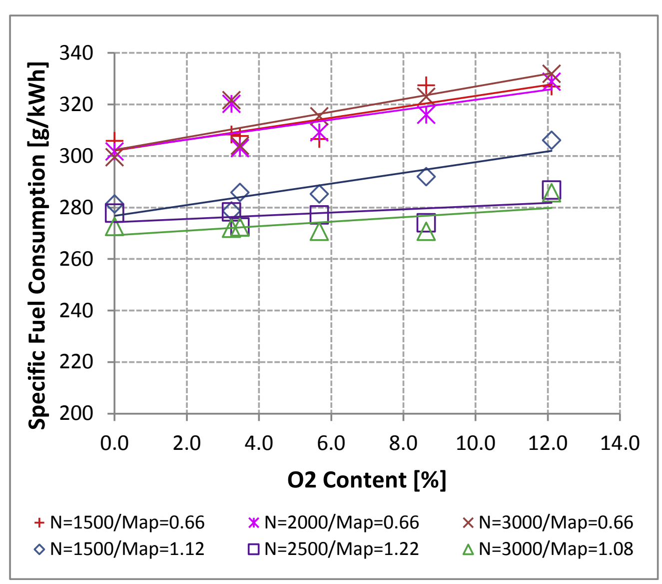

Fig. 8. Specific fuel consumption versus the oxygen content of the fuel blend at optimal spark advance.

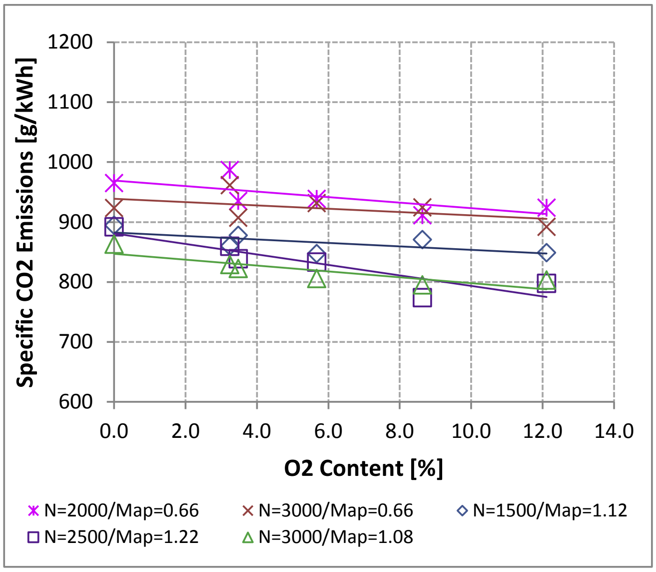

Fig. 9. Specific CO2 emissions versus the oxygen content of the fuel blend at optimal spark advance.

concentrations, at part load, an average decrease of 3.6% has been observed in comparison to pure gasoline fueling. At high loads, according to the trend of the fuel conversion efficiency which increases with the fuel oxygen content, the average CO_2 reduction is around 8%.

The specific emissions of NO_x and HC change very little when gasoline is mixed with biofuels. They tend to marginally decrease when the O_2 content in the fuel mixture increases (Fig. 10). The decrease in NO_x can be explained as due to lower maximum temperatures inside the combustion chamber. Similar results have been found both for butanol/gasoline blends in Refs. [10,23] and ethanol/gasoline in Refs. [14,24].

The small HC reduction can be explained with a more complete combustion of the fuel mixture as a consequence of the alcohol chemical structure which is simpler than that of gasoline.

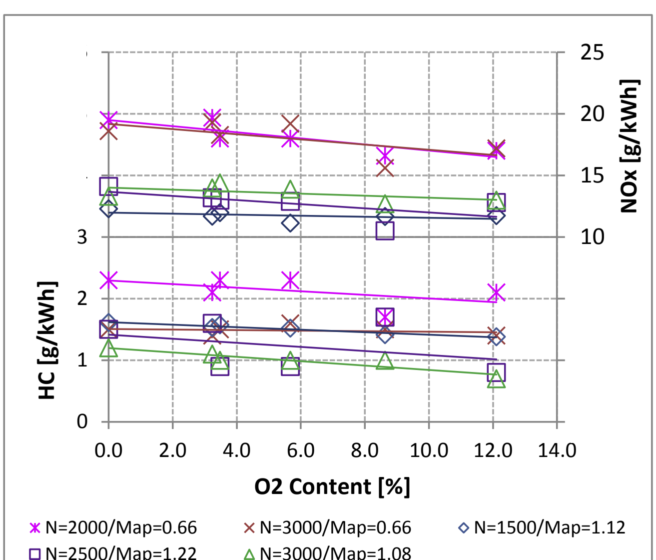

Fig. 10. Specific HC and NOx emissions versus the oxygen content of the fuel blend at optimal spark advance.

4. Conclusions

The behavior of different alcohol-gasoline blends, firing in a downsized spark-ignition engine, has been analyzed in numerous engine operating points. In particular, n-butanol and ethanol, both deriving from bio-masses, have been considered. Binary mixtures (butanol-gasoline or ethanol-gasoline) and ternary mixtures (butanol-ethanol-gasoline) have been analyzed at low and high engine load and at different engine speeds.

In order to have a synthetic parameter, able to represent the alcohol content in the fuel mixture, most results have been presented as a function of the fuel oxygen content.

The first obtained results show the engine fuel conversion efficiency is positively affected by the alcohol content of the fuel burnt. In the most operating points here analyzed, the efficiency increases with the fuel oxygen content. In particular, at high load operation, the maximum efficiency could increase up to three percentage points when alcohol-gasoline mixtures are adopted instead of pure gasoline. This circumstance allows concluding the increase in brake fuel specific consumption, growing with the oxygen content of fuel, is substantially due to the smaller heating values of bio-alcohols.

At low load, the optimum spark advance is not more sensitive to the fuel composition. Increasing the alcohol percentage in the mixture, the spark advance can be reduced due to the higher alcohol flame speed compared to that of pure gasoline. At high load, the spark advance is knock limited. Due to the alcohol higher knock resistance, with respect to gasoline, the spark time can be advanced. So the fuel conversion efficiency may benefit from this new engine setting.

As an example, G50B40E10 mixture (40% butanol, 10% ethanol and 50% gasoline) has allowed a spark advance angle about up to five crank angle degrees greater than that of pure gasoline. This mixture achieved the highest engine efficiency at different levels of load (manifold absolute pressure) and rotational speed. Compared to pure gasoline, a maximum efficiency gain, ranging from 4% to 13% for the different analyzed points, has been obtained. Also the combustion duration has been measured according to the different engine fueling. As it has been explained in the paper, at low engine load a small decrease in the combustion duration (about 10%), with the increase of alcohol content, has been observed. At high load, the different fuel mixtures have shown a combustion duration quite similar to that of pure gasoline fueling. So, since this parameter undergoes very small modifications (particularly at high load operation), it seems just weakly influencing the engine efficiency. According to the trend of the fuel conversion efficiency which increases with the fuel oxygen content, at high load levels, the average CO_2 reduction is around 8%.

At high load, compared to gasoline fueling, a reduction of about 50° in peak temperature values during the alcohol-gasoline mixture combustion has been observed. This has produced a decrease of about 10% in NO_x formation. Similarly, a small reduction in HC specific emissions has been obtained.

At the end, the experimental analyses here presented confirm that alcohols are interesting transportation fuels. Main engine performances remain similar to those obtainable with gasoline fueling. As said above, in some operating points, even some improvements are observable. So, producing the alcohol fuels starting from biomasses could represent an effective way for the reduction of CO_2 emissions.

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