PERFORMANCES AND EMISSIONS IMPROVEMENT OF AN S.I. ENGINE FUELLED BY LPG/GASOLINE MIXTURES

Emiliano Pipitone, Stefano Beccari
Department of Mechanics, University of Palermo

DOI: 10.4271/2010-01-0615

ABSTRACT

As is known gaseous fuels, such as Liquefied Petroleum Gas (LPG) and Natural Gas (NG), thanks to their good mixing capabilities, allow complete and cleaner combustion than normal gasoline, resulting in lower pollutant emissions and particulate matter. Some of the automobile producers already put on the market “bi-fuel” engines, which may be fed either with standard gasoline or with LPG. These engines, endowed of two separate injection systems, are originally designed for gasoline operation; hence they do not fully exploit the good qualities of LPG, such as its better knocking resistance, which would allow higher compression ratios. Moreover, when running with gasoline at medium high loads, the engine is often operated with rich mixture and low spark advance (with respect to the maximum brake torque value) in order to prevent from dangerous knocking phenomena: this produces both high hydrocarbon and carbon monoxide emissions and high fuel consumption. Starting from these observations, the authors experimentally investigated on the simultaneous combustion of LPG- gasoline mixtures in stoichiometric proportion with air (with different LPG/gasoline mass ratios), so as to exploit the good qualities of both fuels to obtain cleaner and more efficient combustions: the addition of LPG to the gasoline-air mixture in fact raises knocking resistance, allowing thus to run the engine with both “overall stoichiometric” mixture and more efficient spark advance even at full load, while the stoichiometric A/F ratio allows to minimize pollutant emissions. In this paper the authors present the results of an extensive experimental study in terms of engine efficiency increments and reduction of pollutant emissions with respect to the pure gasoline operation.

INTRODUCTION

Automobile market is nowadays characterized by a great diffusion of bi-fuels vehicles, i.e. vehicles endowed of Spark Ignition (SI) engines which can be run either on standard gasoline or on gaseous fuels, such as Natural Gas (NG) or LPG (Liquefied Petroleum Gas). With respect to conventional fuels, besides a lower price, these gaseous fuels also exhibit lower pollutant emissions, and this in turn makes bi-fuels vehicles suited for the urban centre transportation. However, current bi-fuel vehicles are equipped with SI engines developed for the use with gasoline, and are not optimized for LPG or natural gas. Moreover, when running with standard gasoline, these engines are usually operated with rich mixtures and poor spark advance in order to prevent dangerous knocking phenomena: these cause both high fuel consumption and high Carbon Monoxide (CO) and Hydrocarbons (HC) emissions, as shown in Figure 1, which reports with continuous lines, the emissions levels measured at full load by a series production bi-fuel engine, while, for a comparison, the dashed lines represent the probable pollutant concentrations which would be measured running the engine with stoichiometric mixtures.

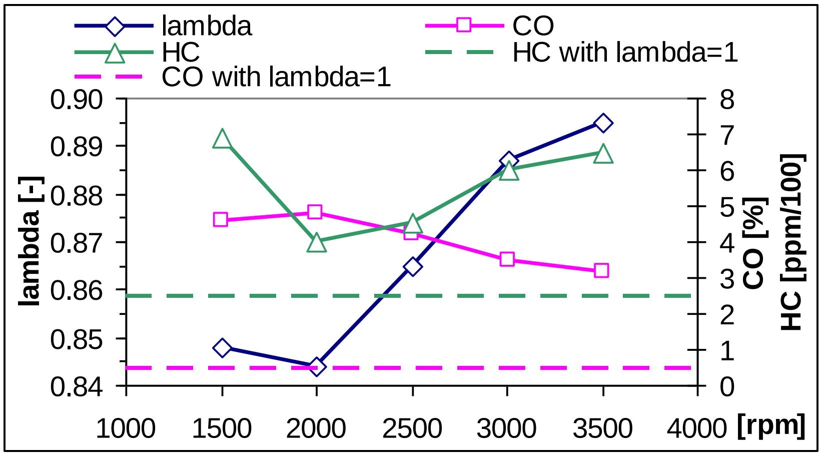

Figure 1 Pollutant raw emissions and mixture strength when running with gasoline at full load

In a previous work [1] the authors tested the simultaneous combustion of natural gas and gasoline in a SI engine under different proportions between the two fuels: the high knock resistance of natural gas allowed both to run the engine with "overall stoichiometric" mixtures even at full load and to optimize the combustion phase, thus reaching, with respect to the pure gasoline operation, efficiency increments up to 27% with negligible power losses. Moreover, fuelling the engine with "overall stoichiometric" mixture maximizes the catalyst conversion efficiency, further reducing the environmental impact of the engine. As a conclusion of this previous work, the double-fuel operation revealed a valid alternative to both pure gasoline and pure natural gas operation.

As is known, LPG also has good knocking resistance properties, being a mixture of Propane (R.O.N.=111) and Butane (R.O.N.=103). This means that also the simultaneous combustion of LPG and gasoline could allow, with respect to pure gasoline operation, significant increase in engine efficiency and decrease in pollutant emissions. The aim of the present work was hence to experimentally evaluate the advantage attainable by the combustion of gasoline-LPG mixtures, under different proportions between the two fuels.

Number of cylinders	4
Displacement [cc]	1242
Bore [mm]	70.80
Stroke [mm]	78.86
Compression ratio ρ	9.8
Rod to crank ratio μ	3.27
Intake valve/cylinder	1
Exhaust valve/cylinder	1
Gasoline Injection system	Multi Point, Port Injection
Gas Injection system	

Table 1 Main characteristics of the FIAT bi-fuel spark ignition engine used in the test

MAIN SECTION

This experimental study has been carried out at the engine test bed of the Department employing a four cylinders 8V 1242cc bi-fuel (Gasoline–Natural Gas) spark ignition engine from FIAT (whose main characteristics are presented in Table 1) connected to a Schenck eddy current dynamometer W130; the LPG injection has been performed by means of the series natural gas injectors, connecting the fuel rail (which normally contains natural gas) to an LPG tank filled with liquefied gas and whose properties are given in Table 2; a pressure regulator has been used to control the LPG injection pressure and the gas injectors have been operated by means of IGBT transistors activated by digital pulses sent using a National Instruments PCI-6602 Counter/Timer board programmed and controlled under LabVIEW. A Walbro-TDD ECU connected to a personal computer was used to control both gasoline injection and spark timing, and an Endress+Hauser Coriolis effect PROMASS 80A was employed to measure the gasoline flow, while LPG flow was deduced on the base of the real injection time using the injector flow chart, previously determined employing the Coriolis effect mass flow meter.

Butane - C_4H_{10} [%Vol.]	20
Propane – C_3H_8 [%Vol.]	80
Lower Heat Value [MJ/kg]	46.2

Table 2 Composition and properties of the LPG used in the test

In order to maintain the overall stoichiometric mixture with air, an ECM AFRecorder 2400 connected to a UEGO sensor placed in the exhaust duct was used. The in-cylinder pressure was measured using an AVL GU13X piezoelectric pressure sensor (installed by means of its ZC32 spark plug adaptor). A fundamental aspect in indicating analysis is the precise determination of the TDC position [2]: as is known, in fact, a 1 degree error (which can be introduced setting the TDC at the peak pressure position of a motored pressure cycle) can cause up to a 10% error in the IMEP estimation. In the experimental test carried out the TDC position was determined by means of the Kistler capacitive sensor 2629B, whose precision is of 0.1 Crank Angle Degrees (CAD). All the quantities were sampled by means of a high speed National Instruments DAQ Board PCI-6133 using as trigger and scan clock the pulses generated by a 360ppr incremental encoder connected to the engine crankshaft. As summarized in Table 3, the double-fuel combustion was tested under different conditions of engine speed, load (expressed by means of the Manifold Absolute Pressure, MAP) and LPG mass fraction (i.e. the ratio between the injected LPG mass and the total fuel mass). For each operative condition tested, the spark advance was chosen as the minimum between the MBT value and the knock onset spark advance. Knock occurrence was monitored by means of a piezoelectric accelerometer fastened on the engine block and connected to an oscilloscope. A Motorscan 8020 exhaust gas analyzer was also employed to measure both carbon monoxide and hydrocarbons concentration in the raw emissions.

Engine speed [rpm]	1500, 2000, 2500, 3000
Manifold Pressure [kPa]	100 (i.e. WOT), 90 and 80
LPG mass fraction [%]	20, 40, 60, 80
Overall A/F ratio	Stoichiometric
Spark Advance	knock limited or best torque

Table 3 Operative conditions tested in Double-fuel mode

EXPERIMENTAL RESULTS

The test performed confirmed that the addition of LPG to gasoline effectively raises the knock resistance of the engine, allowing to run with an “overall stoichiometric” mixture at full load and low engine speed even for the lowest amount of LPG added. The advantage is immediate in terms of engine efficiency, as shown in Figure 2, which reports the typical results of a constant speed test: here indicated and brake thermal efficiencies are plotted as function of the LPG mass fraction, which varies from 0%, i.e. pure gasoline, to 100%, which instead refers to the pure LPG mode. In the same graph, the mixture strength is also reported, in terms of air excess index “Lambda”. As shown, at 2500 rpm the double-fuel combustion allowed efficiency improvements, with respect to the pure gasoline operation, as high as 8 percentage points, that is a 26% increment. The main cause of this strong improvement is related to the mixture strength, which, thanks to the raised knock resistance of the engine, was set to stoichiometric even for a 20% LPG mass fraction.

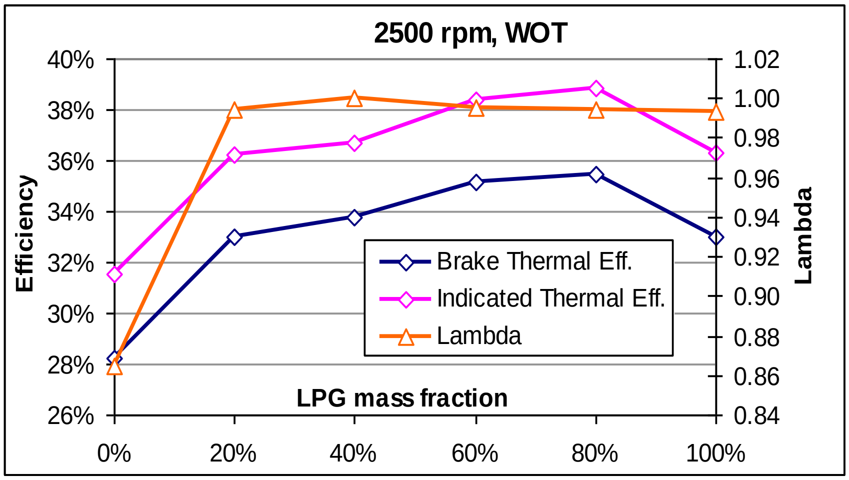

Figure 2 Indicated and brake thermal efficiencies as function of LPG mass fraction (0%=gasoline, 100%=LPG) at 2500 rpm and WOT

The second reason must be instead searched in the thermodynamic efficiency of the cycle which was improved by advancing the combustion phase toward the optimum; Figure 3, which reports the Knock Limited Spark Advance (KLSA) as a function of the LPG mass fraction, determined at full load and for each of the four engine speeds tested, clearly shows how the LPG addition to gasoline increased the knock resistance of the engine; its effect, in terms of combustion phase, can be followed in Figure 4, which reports the Location of Pressure Peak (LPP) measured in the same conditions: as evident, the addition of LPG to gasoline allowed to correct the typical bad combustion phase of gasoline at full load, characterized for the engine on the test bed by LPP values as high as 30 CAD ATDC.

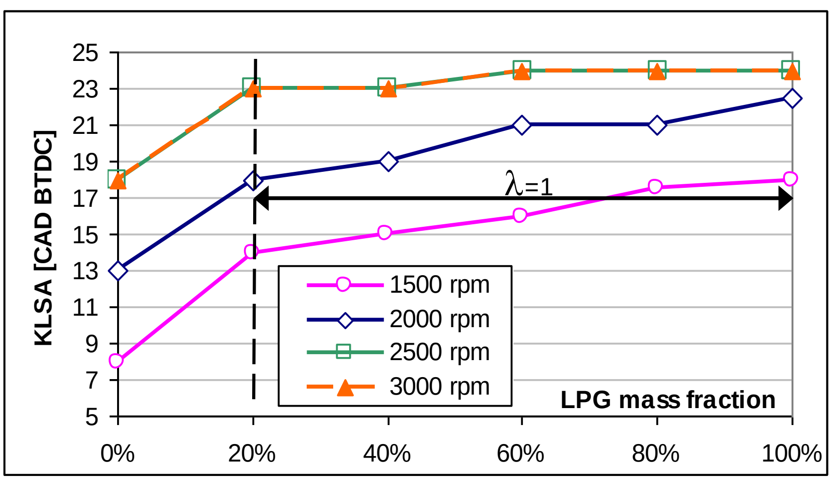

Figure 3 KLSA as function of the LPG mass fraction at WOT

Figure 2 also shows the maximum efficiency to be reached for an LPG mass fraction of 80%. Almost the same result has been obtained for the other engine speeds tested, as pointed out in Figure 5: the best efficiency has been found for LPG mass fraction between 60% and 80%. This behavior can be explained by observing Figure 6, which reports the duration of the rapid combustion (i.e. from 10% to 90% of the mass fraction burned) as function of the LPG mass fraction: the graph shows that the for an LPG mass fraction of 80%, the main part of the combustion is faster and hence gives rise to a higher engine efficiency.

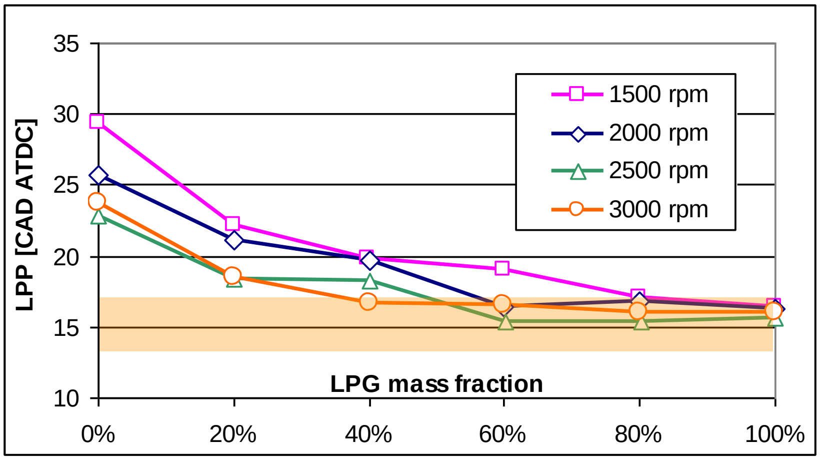

Figure 4 LPP values at full load as function of the LPG mass fraction (the orange band indicates the optimal values)

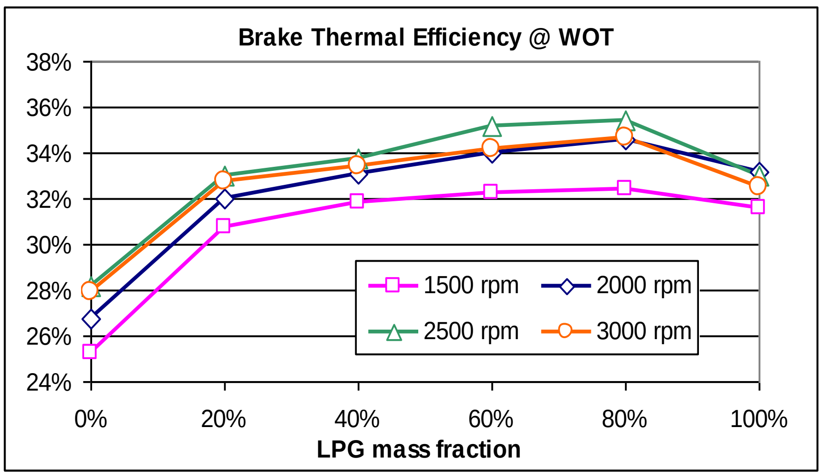

Figure 5 Brake thermal efficiencies as function of LPG mass fraction at full load and for different engine speed

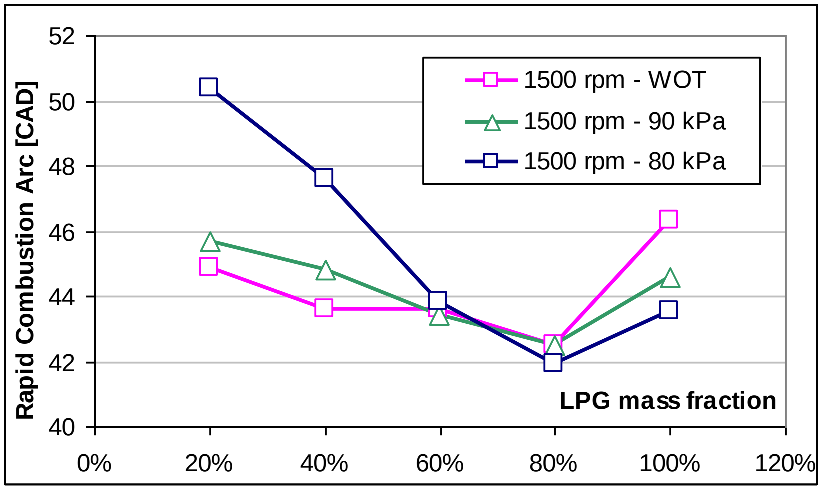

Figure 6 Rapid combustion arc (from 10% to 90% of mass fraction burned) measured at 1500 rpm as function of LPG mass fraction

The authors explained this considering that in a mixture of two fuels with significantly different knock resistance (LPG has RON≈110 while for standard gasoline RON=95), it is always the less resistant to cause knocking to occur; when running with high LPG mass fraction, the high knock resistance of LPG allows to increase spark advance (see as example the KLSA reported in Figure 3) without audible (and hence dangerous) knocking phenomena to occur, and this in turn causes the gasoline-air mixture to be subjected to high temperature and pressure, well above its knock limit: this means that, even if audible knocking does not occur, a portion of the gasoline-air mixture may undergo autoignition, thus burning almost instantaneously; this, in turn, speeds up the heat release process causing an engine efficiency increment. This phenomenon obviously does not occur in the pure LPG mode (i.e. for LPG mass fraction of 100%), because there is no gasoline which may autoignite. Figure 7 instead reports the brake thermal efficiency increments with respect to the pure gasoline operation measured at full load: as shown, the simultaneous combustion of gasoline and LPG allowed efficiency increments up to 29%.

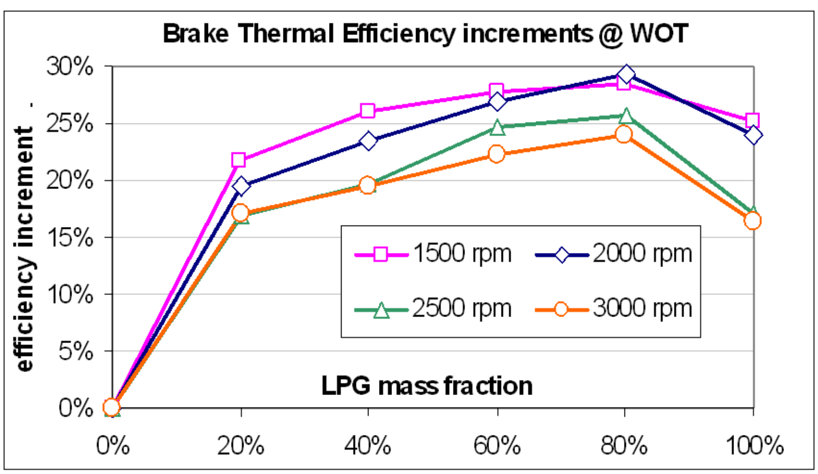

Figure 7 Brake thermal efficiency increments with respect to the pure gasoline mode measured at full load

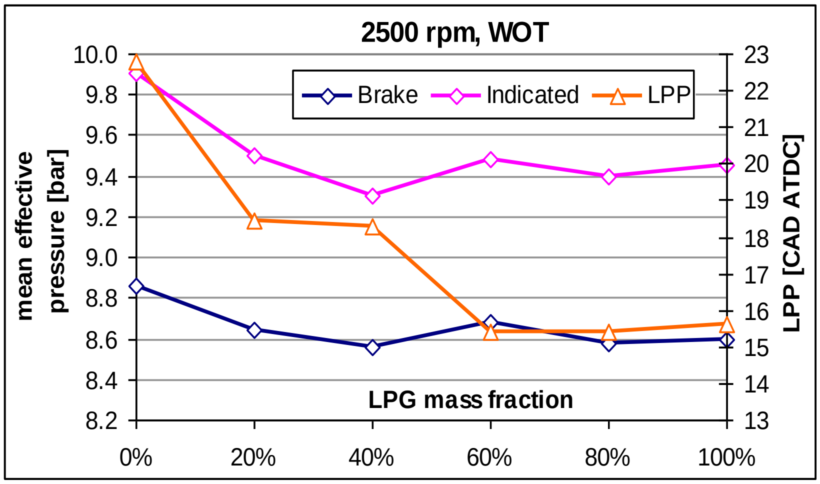

Figure 8 Brake and Indicated Mean Effective Pressure together with the LPP measured at 2500 rpm and WOT

As regards power output, in the double-fuel operation no remarkable losses have been found with respect to the pure gasoline: as can be observed, as example, in the graph in Figure 8, the power loss connected to the use of a gaseous fuel, which decreases the volumetric engine efficiency, is partially counterbalanced by the improved thermodynamic cycle efficiency due to the better combustion phase (see the LPP values reported in the same graph) that can be adopted exploiting the increased knock resistance. The highest power loss was found at 2500 rpm with a 40% LPG mass fraction and resulted to be -3.4%. In the same operative condition, the brake thermal efficiency increment resulted to be +19.6% (see the graph in Figure 7).

One of the principal benefit connected to the adoption of the double-fuel combustion in place of the pure gasoline operation is related to the lower environmental impact that can be pursued: as already pointed out in fact, in order to avoid dangerous knocking phenomena; most of the gasoline fuelled engines are operated with very rich mixtures at full load, and this causes high emissions level, above all in terms of Carbon monoxide (CO) and Hydrocarbons (HC).

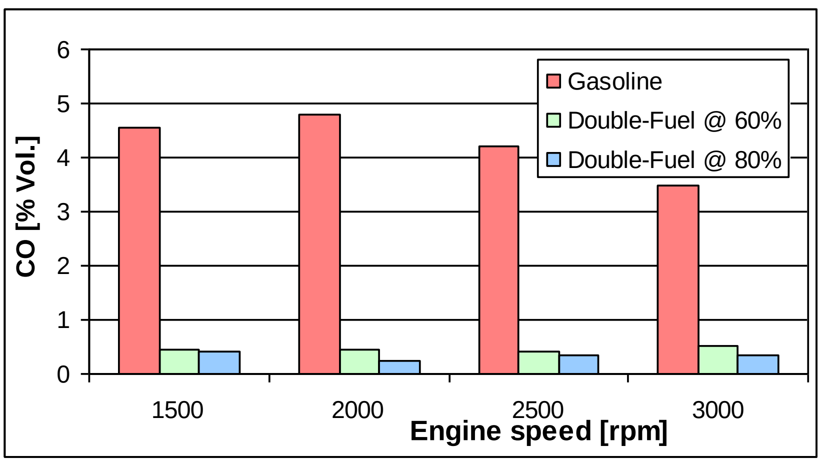

Figure 9 Raw CO emission measured at WOT for pure gasoline (λ reported in Figure 1) and Double-Fuel mode (λ =1, LPG mass fraction 60% and 80%)

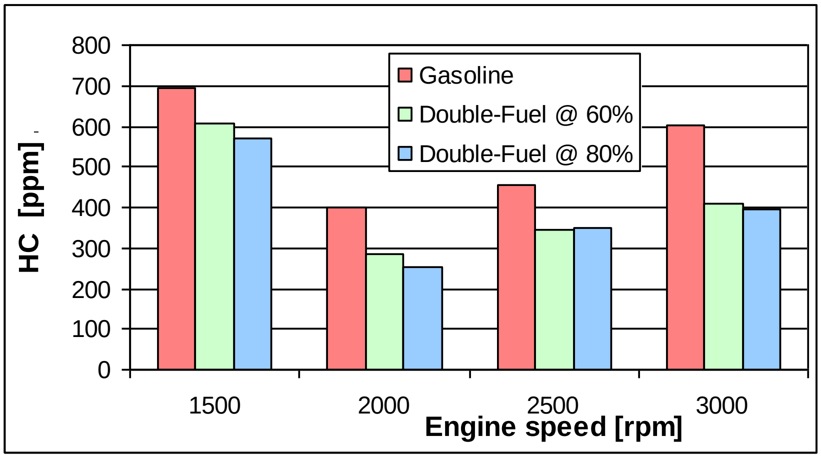

Figure 10 Raw HC emission measured at WOT for pure gasoline (λ reported in Figure 1) and Double-Fuel mode (λ=1, LPG mass fraction 60% and 80%)

Fuelling the engine with a mixture of LPG and gasoline in stoichiometric proportion with air instead produced exhaust gases with lower pollutant concentration, as demonstrated by the graphs in Figure 9 and Figure 10: here the raw emissions of both CO and HC measured at full load with pure gasoline (under the rich mixtures plotted in Figure 1) are compared to those obtained in the same operative condition in the double-fuel mode with LPG mass fraction of 60% and 80%. As concerns the carbon monoxide raw emissions, a difference of about one order of magnitude was observed (see Figure 9), while as regards hydrocarbons, the double-fuel combustion allowed a maximum concentration decrease of 32% (see Figure 10).

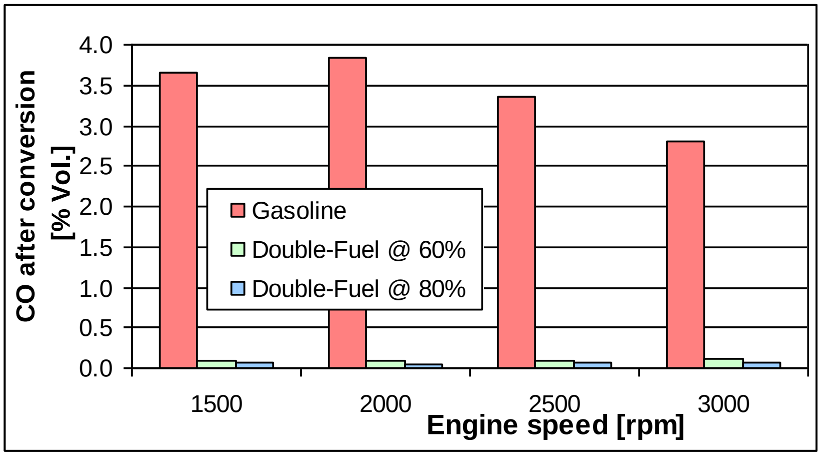

Figure 11 After treatment CO emission estimated at WOT for pure gasoline (λ reported in Figure 1) and Double-Fuel mode (λ =1, LPG mass fraction 60% and 80%)

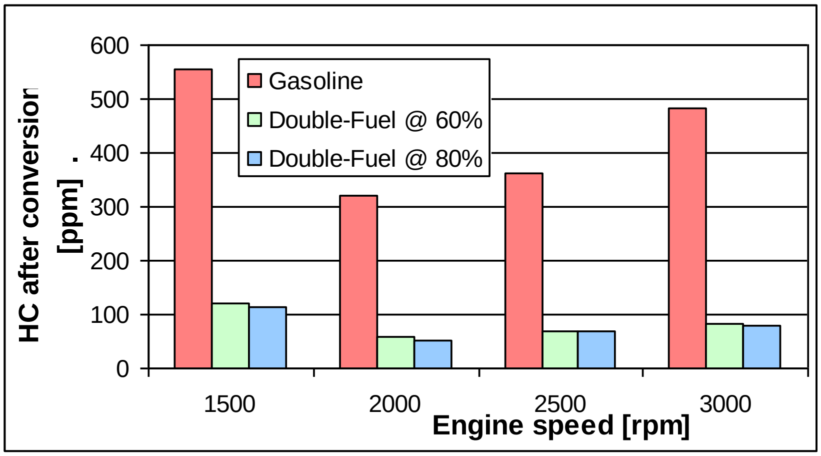

Figure 12 After treatment HC emission estimated at WOT for pure gasoline (λ reported in Figure 1) and Double-Fuel mode (λ =1, LPG mass fraction 60% and 80%)

It must be considered, however, that the great potential of the double-fuel combustion in terms of pollutant emissions relies also in the capability to keep the catalyst to its best conversion efficiency, since the engine is fuelled with an overall stoichiometric mixture. Hence, in order to evaluate the probable emissions level downstream the 3-way catalyst, the authors assumed [3, 4] a conversion efficiency of 80% for the double-fuel mode, while, considering the strong mixture enrichment operated by the series production ECU at full load, it was supposed to be 20% for the pure gasoline mode. On the base of these assumption, the out emissions level were estimated and are presented in Figure 11 and Figure 12: as can be observed, due to the catalyst conversion, the double-fuel mode could allow a hydrocarbons emissions reduction up to 84%. The results presented and discussed until this point regard the full load conditions. As shown in Table 3, however, the test have been performed also for two lower loads, namely for Manifold Absolute Pressure (MAP) of 90 and 80kPa. The results, in terms of efficiency increment with respect to the pure gasoline operation, are presented in Figure 13 and Figure 14 respectively.

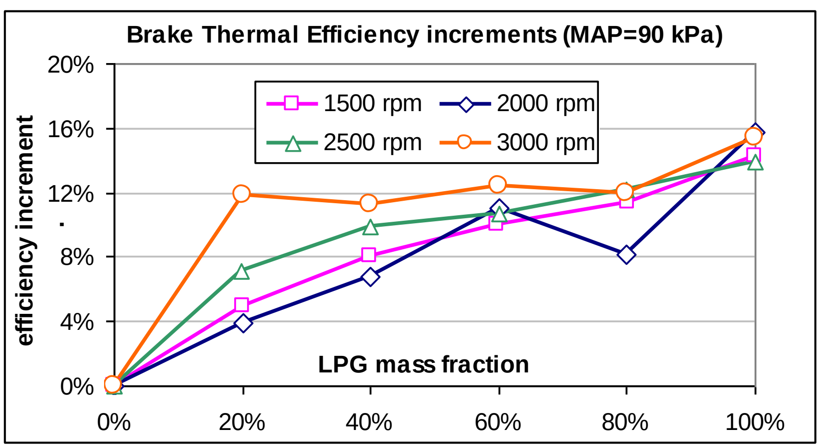

Figure 13 Brake thermal efficiency increments with respect to the pure gasoline mode measured at MAP=90kPa

As expected, the advantage of the double-fuel combustion decreases with decreasing load, since knocking becomes less dangerous and probable, and this allows to feed the engine with stoichiometric air-gasoline mixture: this tends to reduce the main advantage derived from the simultaneous combustion of gasoline and LPG. However, the engine still benefits for the best combustion phase which can be reached thanks to the improved knocking resistance, and this leads to efficiency improvements up to 16% at MAP=90kPa, and 14% at MAP=80kPa.

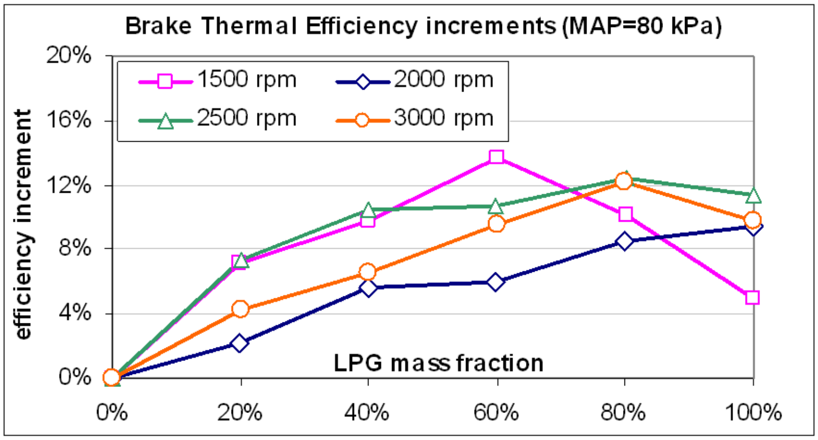

Figure 14 Brake thermal efficiency increments with respect to the pure gasoline mode measured at MAP=80kPa

CONCLUSIONS

On the base of a previous work carried out on the combustion of gasoline-natural gas mixtures in a spark ignition engine, the authors aimed to assess in this work the advantage attainable by the simultaneous combustion of LPG and gasoline. The results of an experimental study are here presented and discussed: they show that the addition of LPG to the gasoline effectively increases the knocking resistance of the engine, allowing thus to run with "overall stoichiometric" mixtures even at full load, which in turn allows to lower both fuel consumption and pollutant emissions (above all HC and CO). Moreover, the increased resistance to knock also allows to improve the thermodynamic cycle by advancing the combustion up to its best phase, which in turn implies a further improvement in terms of power output. An extensive experimental study was carried out on the engine test bed equipped with a bi-fuel spark ignition engine, under different operative conditions of speed and load and employing LPG-gasoline mixtures in many different proportions. The performance obtained by means of the double-fuel combustion, in terms of engine efficiency and power output, are here presented and compared to those achieved running the engine on pure gasoline. The results of this comparison show engine efficiency increments up to 29% at full load and "overall stoichiometric" mixture: the best results were observed for an LPG mass fraction (i.e. the ratio between the injected natural gas mass and the total fuel mass) of 80%. As regards the engine power, the increment with respect to the pure gasoline mode depends on two contrasting factors: in fact, if on one hand the addition of LPG to gasoline raises the fuel knocking resistance, thus allowing to advance the combustion to a better phase, on the other hand, being a gaseous fuel, it lowers the volumetric efficiency of the engine, thus causing a power loss. The results of the experimental tests showed that the two factors compensate each other, making the power increment to remain around zero: in the worst case, the power loss with respect to the pure gasoline mode was found to be 3.4%. As regards pollutant emissions, in the double-fuel mode the measured concentrations of both carbon monoxide and hydrocarbons were quite lower than those measured in the pure gasoline operation; the authors also estimated the emission level downstream the 3-way catalyst, and the results obtained lead to definitively consider the double-fuel combustion as a valid alternative to normal gasoline operation in bi-fuel engines.

REFERENCES

1. E. Pipitone, S. Beccari, “Performances improvement of a S.I. CNG bi-fuel engine by means of double-fuel injection”, SAE Technical Paper 2009-24-0058, DOI: 10.4271/2009-24-0058

2. Pipitone E., Beccari A., "Determination of TDC in internal combustion engines by a newly developed thermodynamic approach”, Applied Thermal Engineering, Vol. 30, Issues 14-15, October 2010, Pages 1914-1926, DOI: 10.1016/j.applthermaleng.2010.04.012

3. J. B. Heywood, "Internal Combustion Engines Fundamentals", McGraw-Hill automotive technology series, 1988, ISBN 0-07-100499-8

4. R. Stone, J. K. Ball, "Automotive engineering fundamentals", SAE International , 2004

5. E. Pipitone, A. Beccari, "A Study on the Use of Combustion Phase Indicators for MBT Spark Timing on a Bi-Fuel Engine", SAE Technical Paper 2007-24-0051, DOI: 10.4271/2007-24-0051

CONTACT INFORMATION

Ing. Emiliano Pipitone, Dipartimento di Meccanica, University of Palermo. emiliano.pipitone@unipa.it

ACKNOWLEDGMENTS

The authors would like to express their acknowledgments to Prof. Emilio Catania from Politecnico di Torino and to Ing. Andrea Gerini from CRF who made this research possible. The authors also gratefully acknowledge Mr. Beniamino Drago for his invaluable technical support.

DEFINITIONS/ABBREVIATIONS

A/F: Air to Fuel Ratio

ATDC: After Top Dead Centre

BMEP: Brake Mean Effective Pressure

BTDC: Before Top Dead Centre

CAD: Crank Angle Degrees

CNG: Compressed Natural gas

CO: Carbon Monoxide

CO_2: Carbon Dioxide

ECU: Electronic Control Unit

HC: Hydrocarbon

IMEP: Indicated Mean Effective Pressure

KLSA: Knock Limited Spark Advance

Lambda = λ = Air Excess Index = $(A/F)/(A/F)_{stoichiometric}$

LPG: Liquefied Petroleum Gas

LPP: Location of Pressure Peak

MAP: Manifold Absolute Pressure

MBT: Maximum Brake Torque

MFB: Mass Fraction Burned

MFB50: Location of 50% of Mass Fraction Burnt

NG: Natural gas

TDC: Top Dead Centre

UEGO: Universal Exhaust Gas Oxygen

WOT: Wide open throttle (full load)