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A Study on the Use of Combustion Phase Indicators for MBT Spark Timing on a Bi-Fuel Engine

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ABSTRACT

The performance of a spark ignition engine strongly depends on the phase of the combustion process with respect to piston motion, and hence on the spark advance; this fundamental parameter is actually controlled in open-loop by means of maps drawn up on the test bench and stored in the Electronic Control Unit (ECU). Bi-fuel engines (e.g. running either on gasoline or on natural gas) require a double mapping process in order to obtain a spark timing map for each of the fuels. This map based open-loop control however does not assure to run the engine always with the best spark timing, which can be influenced by many factors, like ambient condition of pressure, temperature and humidity, fuel properties, engine wear. A feedback control instead can maintain the spark advance at its optimal value apart from operative and boundary conditions, so as to gain the best performance (or minimum fuel consumption). Such a control can be realized using as pilot variable a combustion phase indicator, i.e. a parameter which depends exclusively on the phase of the heat release process and assumes a fixed value for optimal spark timing. The purpose of the present work is to compare the behaviour of the most used combustion phase indicators using two different fuels one after the other (common gasoline and Compressed Natural Gas, CNG) on the same engine, in order to assess the influence of different heat release progress and to verify the possibility to feedback control the spark timing apart from the fuel used. The comparison has been carried on by means of experimental test on the engine test bench, analysing in-cylinder pressure acquired with varying spark advance for different operative conditions of engine speed, load and air-to-fuel ratio.

INTRODUCTION

The phase of the heat release process with respect to piston motion is of vital importance for the achievement of the best performances from an internal combustion engine. On spark ignition engines this phase is controlled by means of maps stored in the Electronic Control Unit (ECU) which report the spark advance values to adopt for different operative conditions of engine load, represented by means of the throttle position or by the Manifold Absolute Pressure (MAP), and angular speed. The spark timing map is normally obtained during a time consuming calibration process on the engine test bench, and does not assure to run the engine with the Maximum Brake Torque (MBT) spark timing under all operative conditions, since this optimal spark timing depends also on ambient conditions of pressure, temperature and humidity, fuel qualities and engine wear; normal differences from the engine used in the mapping process [1, 2] may also influence the optimal spark timing. Thus the engine may run with a wrong spark advance, decreasing its efficiency. Fig. 1 shows, as example, the relative torque loss (ΔTorque/Torque$_{,MAX}$) caused by a bad spark timing control. The diagram refers to the tested engine when fuelled with natural gas; however almost the same results were obtained using normal gasoline. As can be seen a 2% torque loss can be caused by a 5 degrees deviation from optimal spark advance, while a 10 degrees error can decrease engine torque by 7%.

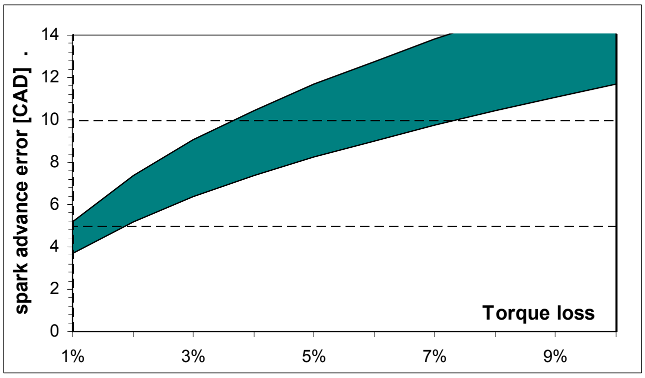

Fig. 1 Torque loss due to bad spark timing control

A closed-loop control on the spark timing instead could allow to run the engine always at its best efficiency spark advance. Such kind of control is achievable for example using as pilot variable of the control system a combustion phase indicator, that is a parameter derived from in-cylinder pressure analysis which mainly depends on combustion phase [3, 4] and assumes a reference value under optimal spark timing condition. The control system hence act on the spark advance so as to maintain the pilot variable (the combustion phase indicator) at its optimal value. Moreover, on a bi-fuel engine (e.g. running either on standard gasoline or on Compressed Natural Gas, CNG) the calibration process time is doubled since two different maps are necessary to manage the spark timing with both fuels. In this case, a closed-loop control on the spark advance may be useful both for speed-up the mapping process and to run the engine with optimal spark timing apart from operative conditions of load and speed, and, above all, from fuel kind and qualities.

MAIN SECTION

The aim of this work is to characterize the combustion phase indicators measured in a bi-fuel engine fed either with standard gasoline or with CNG, and test the possibility to feedback control the spark advance using a combustion phase indicator as pilot variable regardless of fuel kind and qualities. As concern combustion phase indicators, the most encountered in literature are:

1. Location of Pressure Peak (LPP)
2. Location of Maximum Pressure Rise (LMPR)
3. Location of 50% of Mass Fraction Burned (MFB50)
4. Relative Pressure Ratio 10 CAD ATDC (PRM10)
5. Location of Maximum Heat release Rate (LMHR)

All of them comes from in-cylinder pressure analysis, which can be directly measured with a combustion chamber pressure transducer or evaluated by means of engine speed analysis. In the following section a brief description of the listed combustion phase indicators is given.

THE COMBUSTION PHASE INDICATORS

Location of peak pressure (LPP)

According to this criterion, spark advance is set to its best value when the pressure peak is found to be 14÷16 crank angle degrees (CAD) after top dead centre (ATDC), apart from engine speed and load, and from other variables. This is one of the most encountered combustion phase indicator in literature [1, 5, 6, 7], and requires pressure sampling at least for 30 degrees ATDC. As for all the other indicators, it has been determined empirically and has not yet any theoretical explanation.

Location of Maximum Pressure Rise (LMPR)

H. A. Cook et al. in 1947 [8] showed that under optimal spark advance, the maximum pressure rise occurs about 3 degrees ATDC. Rarely encountered in literature among the combustion phase indicators, its evaluation requires pressure sampling in the interval ±20 degrees around top dead centre (TDC). A previous experimental investigation [4] revealed poor performances attainable by the use of this indicator; it revealed lower accuracy with respect to the other indicators, hence it has been excluded in the actual work.

Location of 50% of Mass Fraction Burned (MFB50)

It is well known to internal combustion engine researchers that in-cylinder pressure allows the evaluation of the fuel mass fraction burned: this can be accomplished following the procedure proposed by Rassweiler and Withrow [9], simple yet reliable, or by means of thermodynamic analysis [10, 11, 12, 13], which instead requires to know wall heat transfer law. According to this criterion, spark timing is set to the best value when MFB reaches 50% about 9 degrees ATDC [3, 4, 14, 15]. Compared to LPP and LMPR, this indicator requires a greater amount of calculus and data to sample: in-cylinder pressure in fact must be acquired during almost the whole compression and expansion strokes. Moreover, unlike LPP and LMPR, absolute pressure values are needed for a correct MFB50 evaluation: this imply a non negligible sensitivity to pressure referencing error [3, 4]. In the present work the mass fraction burned has been evaluated by means of the Rassweiler & Withrow method.

Relative Pressure Ratio 10 crank angle degree ATDC (PRM10)

Also this indicator was proposed [16] as an alternative to the MFB50; its author in fact defined it as

$$\mathrm{PRM10} = \frac{\mathrm{PR}(10) - 1}{\mathrm{PR}(55) - 1} \tag{1}$$

being PR(ϑ) the Pressure Ratio between the measured fired pressure and the evaluated motored pressure ϑ degrees ATDC (generally PR values stay between 1 and 4). Under MBT spark timing the relative pressure ratio PRM10 should assume the value 0.55, quite similar to MFB which should reach its 50% around 9 degrees ATDC. The ratio between measured and motored pressure (the latter calculated using a polytropic law) in effect follows the concept already proposed by Rassweiler & Withrow, i.e. the heat released by combustion is closely related to the pressure rise besides the compression effect. The advantage in the use of the PRM10 instead of MFB50 theoretically relies on the easier calculation and fewer data to sample: four points in fact should be enough, two taken during compression stroke for polytropic index evaluation and the other two taken 10 and 55 CAD ATDC. As a matter of fact, measurement noise and pressure referencing (absolute pressure values are needed) may require sample the transducer output for a great part of the compression stroke in order to correctly evaluate the polytropic coefficient [17, 18]. Moreover, since the expansion polytropic index not necessarily equals the compression one, both of them should be evaluated: this requires a complete pressure sampling during expansion stroke, and makes the use of the relative pressure ratio almost equivalent to the use of the mass fraction burned. For this reason it has been excluded in the comparison reported in this work.

Location of Maximum Heat release Rate (LMHR)

The authors proposed on previous works [3, 4] the Location of the Maximum Heat release Rate (LMHR) as an alternative to the 50% of the Mass Fraction Burned, since it is less influenced by in-cylinder pressure referencing or measurement errors and its evaluation does not require any extra calculus effort with respect to MFB50: in fact the mass fraction burned is normally obtained by integration. Moreover, if the mass fraction progress is adequately described by the Wiebe function, it can be demonstrated [see for example in 4] that LMHR is almost coincident with MFB50: therefore it has a set point value almost equal to MFB50, i.e. 8÷10 degrees ATDC with optimal spark timing.

THE EXPERIMENTAL TEST

In order to compare the behaviour of the combustion phase indicators measured on the same engine using two different fuels (one after the other) a wide experimental campaign has been carried out on the engine test bed of the Department of Mechanics at University of Palermo. Each of the combustion phase indicators here taken into consideration (LPP, MFB50 and LMHR) has been evaluated at the optimal spark advance found for different operative conditions of loads, speed and air-to-fuel ratios (listed in Table 1), fuelling the engine either with standard gasoline or compressed natural gas (whose composition is resumed in Table 2).

Fuel	MAP [kPa]	Speed [rpm]	λ [-]
CNG	60, 80, 100	1500, 2000, 2500, 3000, 3500	0.85, 1.0
Gasoline	48, 65	1500, 2000, 2500, 3000, 3500	0.85, 1.0

Table 1 Operative conditions tested

The engine test bed was equipped with a FIAT bi-fuel four cylinder in-line engine (whose characteristics are resumed in Table 3) connected to an eddy current dynamometer, set on constant speed braking characteristics. Since a complete spark timing test requires to test the engine with spark advance value higher than the optimal one, knocking occurrence limited the engine load in the gasoline test: in this case in fact the MAP was varied from 48 to 65 kPa, corresponding to 2.5 and 5 bar BMEP with stoichiometric air-to-fuel ratio. The use of CNG instead allowed to set the MBT spark advance even at full load. This is due to the typical higher knocking resistance of methane (octane number≈130), which is the major constituent of the CNG (as can be seen in Table 2) and to the relatively low compression ratio of the engine tested. For each of the operative conditions tested, the spark timing was varied in a range of 10 degrees across the MBT value using a Walbro TDD ECU controlled from a personal computer.

Methane – CH_4	85.484
Ethane – C_2H_6	7.615
Propane – C_3H_8	1.681
Carbon dioxide – CO_2	0.505
Nitrogen – N_2	3.947
Other	0.768

Table 2 Composition of the Natural Gas used in the test [% VOL.]

Displacement [cc]	1242
Bore [mm]	70.80
Stroke [mm]	78.86
Compression ratio	9.8
Rod length [mm]	128.95
Inlet valves/cylinder	1
Outlet valves/cylinder	1

Table 3 Characteristics of the four cylinder engine used in the tests

For each spark advance, a matrix of 50 in-cylinder pressure cycles has been recorded, together with the corresponding values of engine torque, speed, A/F, MAP and spark advance. The 50 cycles matrix was used to compute the mean pressure cycle, thus overcoming the cycle-by-cycle variation which strongly affect indicators evaluation [6]. Hence, for each operative conditions in , each indicator has been evaluated on the base of the mean pressure cycle. The data acquisition was performed by means of two National Instruments DAQ Card 6062, using the outputs of an optical 360 ppr encoder to clock and trigger the acquisition. In order to remove unwanted noise, each pressure cycle has been filtered using a 2nd order lowpass Butterwoth filter with phase shift compensation (which is a crucial feature in this kind of study) and whose cutting frequency has been determined by means of observation made on the amplitude spectrum of some sampled pressure signals: this analysis revealed that unwanted noise has frequencies higher than 45 times the engine cycle frequency. Moreover, since the mass fraction burned evaluation requires absolute pressure values and the pressure transducer employed is an un-cooled piezoelectric, (i.e. it outputs relative pressure values) each pressure cycle has been compensated with by means of the MAP technique [17, 18], which assumes that mean in-cylinder pressure around the inlet stroke BDC is equal to manifold absolute pressure: this requires the use of a MAP sensor, which is commonly integrated in modern spark ignition engine management system. A Kistler 2629B TDC system was used for the correct top dead centre determination, while A/F ratio, engine speed and spark advance were acquired by means of an ECM AFRecorder 2400 connected to a UEGO sensor placed in the exhaust duct. Fig. 2 shows the typical result of a complete spark timing test: engine torque and combustion phase indicators are plotted as functions of spark advance, with fixed engine speed, throttle position and air-to-fuel ratio.

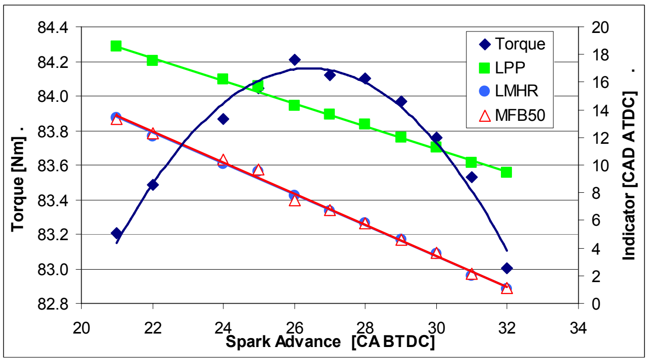

Fig. 2 Engine torque and combustion phase indicators as functions of spark advance (2500 rpm, MAP=100 kPa, λ=1.0, Fuel: Natural Gas)

The dots represents the experimental data, while the lines represents the fitting curves. As can be seen, the combustion phase indicators can be interpolated by straight lines, while engine torque is well fitted by a 2nd order polynomial (regression coefficients typically higher than 0.96 were found). The MBT spark timing was then determined as the abscissa corresponding to the maximum value of the torque fitting polynomial, and was used to establish the indicators optimal value on its regression line. It was thus possible to determine the best spark advance and the relative indicator's value for each operative conditions of Table 1 both in the gasoline and in the natural gas test. The results thus obtained allowed to estimate the mean value assumed by each indicator under optimal spark timing condition. The torque fitting curves obtained in all the operative conditions tested were also used to evaluate the relative torque losses caused by a wrong spark timing choice.

Table 4 and Table 5 resume the mean indicators optimal values together with the spreads (i.e. the amplitude of the indicators variation ranges) relatively to the stoichiometric mixture tests. As can be seen, each of the three indicators taken into consideration revealed suitable to pilot a feedback spark timing control with both the fuels used: each indicator in fact assumes a reference value when the spark advance is the best, with a limited spread over the whole tested conditions, which could cause a spark advance deviation from optimal value not higher than 2 degrees. This would allow to perform a reliable optimal spark timing control with low spark oscillations around the best value. It was also found a good agreement between the indicators reference values obtained using normal gasoline or natural gas: this means that these combustion phase indicators are almost insensitive to the different heat release laws which characterize the combustion of the two fuels used; it follows that the same indicator set point value could be used to pilot the optimal spark timing control for both the fuels.

	LPP	MFB50	LMHR
optimal value	14.2	8.1	7.8
spread	± 0.9	± 1.3	± 1.0

Table 4 Indicators mean optimal values and spread (stoichiometric mixture, Natural Gas)

	LPP	MFB50	LMHR
optimal value	13.9	7.1	6.9
spread	± 1.0	± 1.0	± 1.3

Table 5 Indicators mean optimal values and spread (stoichiometric mixture, Gasoline)

Since a spark ignition engine may be fed with rich mixture in order to prevent abnormal combustion phenomena, the authors repeated the tests fuelling the engine with a rich mixture (λ=0.85), so as to check for any dependence of the indicators optimal values from mixture strength.

	LPP	MFB50	LMHR
optimal value	14.3	8.0	7.8
spread	± 1.3	± 1.6	± 1.5

Table 6 Indicators mean optimal values and spread (λ=0.85, Natural Gas)

	LPP	MFB50	LMHR
optimal value	14.4	7.1	7.6
spread	± 1.1	± 1.4	± 1.5

Table 7 Indicators mean optimal values and spread (λ=0.85, Gasoline)

The results obtained, reported in Table 6 and Table 7, point out that enriching the mixture by a 15% has a negligible effect on the indicators reference values, which remained almost identical to those found in the stoichiometric mixture tests (see Table 4 and Table 5). Hence it can be drawn the conclusion that a feedback spark timing control can be pursued independently from fuel kind and air-to-fuel ratio using the overall mean optimal values, i.e. the mean values from Table 4, Table 5, Table 6 and Table 7, which are reported in Table 8.

	LPP	MFB50	LMHR
Mean value	14.2	7.6	7.5

Table 8 Indicators mean optimal values

In order to assess the real performance attainable by this optimal spark timing control, the authors evaluated, for each of the operative conditions tested, the spark advance errors induced by the use of the indicators mean optimal values, i.e. the difference between the real MBT spark advance (corresponding to the maximum value of the torque fitting polynomial) and the spark advance related to the use of the indicator mean set point value (obtained by means of the indicator regression line). As shown in Fig. 3, the data collected revealed that, employing the indicators mean set point, the highest spark advance deviation from optimal value was 3 degrees (with a mean error of about 1 degree) using gasoline, while feeding the engine with NG the maximum spark advance error resulted to be about 1.5 degrees, with a mean value of 0.5 degrees.

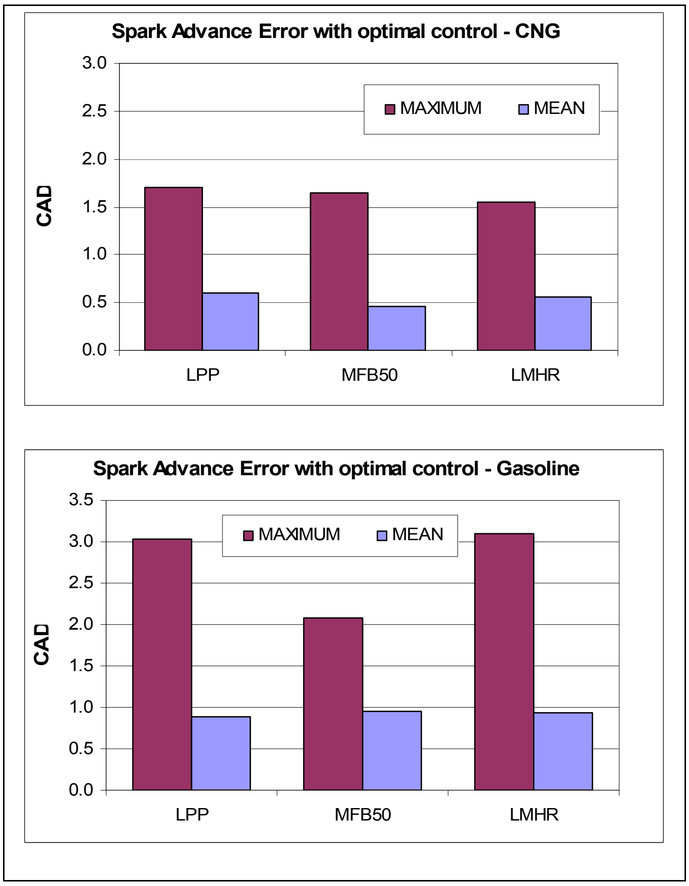

Fig. 3 Spark advance errors related to the indicators mean set point (Top: CNG, Bottom: Gasoline)

The lower limit curve in Fig. 1 reveals that, in the worst case, these maximum spark advance deviations correspond to engine torque losses of about 0.7% and 0.2% respectively. In order to effectively check for the real torque losses connected to the use of the mean indicators optimal values, the authors, by means of the fitting curve, evaluated, for each operative condition tested, the engine torque loss, i.e. the difference between the maximum torque and the torque corresponding to the indicators set point; as shown in Fig. 4, it was found that using natural gas, the highest deviation from the maximum torque was lower than 0.2%, with a mean loss not higher than 0.03%; when feeding the engine with gasoline, maximum and mean torque losses became 0.3% and 0.05%.

These results point out that the use of the indicators overall mean values can maintain the engine efficiency at its best, regardless of the fuel used and of the operative condition of engine speed, load and mixture strength.

CONCLUSION

An experimental investigation has been carried out on a bi-fuel spark ignition engine fed either with normal gasoline or natural gas with the aim to assess the possibility to use combustion phase indicators to perform a spark advance optimal control regardless of the fuel used and the operative condition. Three different combustion phase indicators have been put to the test, the Location of Peak Pressure (LPP), the Location of the 50% Mass Fraction Burnt (MFB50) and the Location of the Maximum Heat release Rate (LMHR).

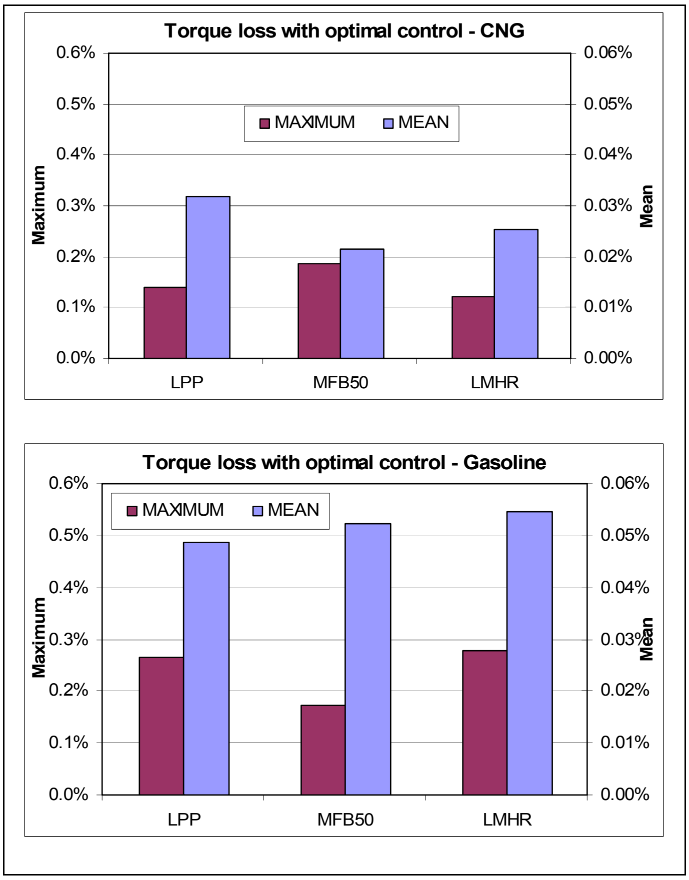

Fig. 4 Torque losses related to the indicators mean set point (Top: CNG, Bottom: Gasoline)

It has been found that each of the indicators not only allows to actuate a feedback spark timing control, but its reference value has no relevant variations with varying fuel: this means that the combustion phase indicators tested are insensitive to the different heat release laws which characterize the two used fuels. Hence a single overall mean optimal value has been determined for each indicator, apart from the fuel used. The authors also verified the performance attainable by a feedback control driven by the use of these indicators: the results obtained clearly show that spark advance can be kept next to its optimal value apart from the operative conditions, thus running the engine always at its best efficiency.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

A/F: Air to Fuel Ratio

ATDC: After Top Dead Centre

BDC: Bottom Dead Centre

BTDC: Before Top Dead Centre

BMEP: Brake Mean Effective Pressure

CAD: Crank Angle Degree

CNG: Compressed Natural gas

ECU: Electronic Control Unit

IMEP: Indicated Mean Effective Pressure

LMHR: Location of Maximum Heat release Rate

LMPR: Location of Maximum Pressure Rise

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

PRM10: Pressure Ratio Management value 10 crank angle degrees ATDC

TDC: Top Dead Centre

UEGO: Universal Exhaust Oxygen Sensor

WOT: Wide open throttle

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