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The Experimental Validation of a New Thermodynamic Method for TDC Determination

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ABSTRACT

In-cylinder pressure analysis is becoming more and more important both for research and development purpose and for control and diagnosis of internal combustion engines; directly measured by means of a combustion chamber pressure transducers or evaluated by analysing instantaneous engine speed [1,2,3,4], in-cylinder pressure allows the evaluation of indicated mean effective pressure (IMEP), combustion heat release, combustion phase, friction pressure, etc...It is well known to internal combustion engine researchers that for a right evaluation of these quantities the exact determination of Top Dead Centre (TDC) is of vital importance: a 1° error on TDC determination can lead to evaluation errors of about 10% on the IMEP and 25% on the heat released by the combustion. In this paper the authors present the experimental validation of an original thermodynamic method for the correct evaluation of the "loss angle", i.e. the angular phase shift between the TDC location and the pressure peak location. The validation has been carried out on a spark ignition engine comparing the results of the thermodynamic method, whose input is the in-cylinder pressure acquired in a "motored" cylinder (i.e. without combustion), with those obtained from a commercial available TDC sensor. The comparative tests aimed to characterize the precision of the proposed method.

INTRODUCTION

In-cylinder pressure analysis is nowadays an indispensable tool in internal combustion engine research & development. It allows the measure of some important performance related parameters, such as indicated mean effective pressure, mean friction pressure, indicated fuel consumption, heat release rate, mass fraction burned, etc. Moreover, future automotive engine will probably be equipped with in-cylinder pressure sensors for continuous combustion monitoring and control, in order to fulfil the more and more severe emission limits. For these reasons, in-cylinder pressure analysis must be carried out with maximum accuracy, in order to minimize the effects of its characteristic measurement errors. The exact determination of the top dead centre (TDC) position is of vital importance, since a 1° crank angle (CA) error can cause up to a 10% evaluation error on IMEP and 25% error on the heat released by the combustion: the position of the crankshaft (and hence the volume inside the cylinder) should be known with the accuracy of at least 0.1 crank angle degrees (CAD), which is not an easy task, even if the engine dimensions are well known. A good TDC determination can be pursued by means of a dedicated capacitive TDC sensor, which allows a dynamic measurement (i.e. while the cylinder is motored) with the required 0.1 degrees accuracy. Such a sensor has a substantial cost and its use is not really fast, since it must be fitted in the spark plug hole of the cylinder. A different approach can be followed using a thermodynamic method whose input is the in-cylinder pressure, sampled in a "motored" cylinder during the compression and expansion strokes: some of these methods can be found in literature [1, 2, 3, 4, 5, 6]. This paper will discuss a new thermodynamic method for the TDC position evaluation together with the experimental validation.

THE THERMODYNAMIC METHOD: BASE THEORY

The compression and expansion processes in a motored engine can be described observing the energy transformations regarding the unitary mass which remains in the cylinder. The first law of thermodynamics states that:

$$\delta q - p\delta v = \delta u \qquad (1)$$

where δq represents the specific heat received by the gas from the cylinder walls during the crank rotation $\delta\vartheta$, p and v represent the gas pressure and specific volume, while δu is the specific internal energy variation.

The gas involved in the process can be assumed to be a perfect gas, thus the following equations are also valid:

$$p\ v = R'\,T \Rightarrow \frac{\mathrm{d}p}{p} + \frac{\mathrm{d}v}{v} = \frac{\mathrm{d}T}{T}$$

$$\delta u = c_V\ \delta T \tag{2}$$

being T the gas temperature, R' the gas constant, c_P and c_V the constant pressure and constant volume specific heat.

The compression-expansion process in a motored engine can be assumed to be frictionless, hence the second law of thermodynamics states that the specific entropy variation δS of the in-cylinder gas in the crank rotation $\delta\vartheta$ is:

$$\delta\, S = \frac{\delta q}{T} \tag{3}$$

thus, from equation (1) and (2) the specific entropy variation results:

$$\delta\, S = \frac{\delta q}{T} = \ c_P \frac{\delta v}{v} + c_V \frac{\delta p}{p} \tag{4}$$

Due to mass leakage through piston rings and valve seats, the available volume V for the in-cylinder gas increases, hence its specific volume changes:

$$V = v \cdot m\ \Rightarrow\ \frac{\mathrm{d}v}{v} = \frac{\mathrm{d}V}{V} - \frac{\mathrm{d}m}{m} \tag{5}$$

where m represent the in-cylinder mass.

Hence, considering the finite increment "δ" due to a crank rotation $\delta\vartheta$, the specific entropy variation in equation (4) will now result:

$$\delta S = c_P \frac{\delta V}{V} + c_V \frac{\delta p}{p} - c_P \frac{\delta m}{m} \tag{6}$$

being δm the mass escaping from the cylinder (hence $\delta m \leq 0$) during the crank rotation $\delta\vartheta$; hence the in-cylinder pressure changes is:

$$\delta p = \frac{1}{V}\left[\delta Q(k-1) - kp\,\delta V\right] + k\,p\,\frac{\delta m}{m} \tag{7}$$

here $\delta Q = m\ \delta q$ represents the heat received by the gas and k is the isentropic coefficient $=c_P/c_V$. Equation (7) shows that, in an ideal adiabatic ($\delta Q = \delta m = 0$) motored engine, pressure would reach its maximum ($\delta p = 0$) when the volume is minimum ($\delta V = 0$), hence the Location of Peak Pressure (LPP) would coincide with the TDC position: in the temperature-entropy diagram the in-cylinder pressure evolutions would be represented by two coincident segments (AB and BA in figure 1), being $\delta S = 0$. In a real motored engine the pressure variation is caused both by volume changes and by two phenomena related to the real machine, i.e. the heat transferred δQ (which is negative when the gas temperature is higher than wall temperature, i.e. $\delta Q \propto (T_W - T_{GAS})$) and the mass leakage δm (which is negative whenever in-cylinder pressure is higher than outer pressure): hence, from equation (4), $\delta S \neq 0$. Equation (7) clearly shows that both heat transfer and mass leakage cause pressure rise to be zero when the volume changes are still negative (i.e. during compression), hence the pressure curve becomes asymmetric, with respect to the TDC, shifting the LPP in advance with respect to the TDC position (see the diabatic evolution in figure 1 or the real pressure curve in figure 2). The angular distance between LPP and TDC location (L_{TDC}) is the so called "loss angle" (ϑ_{loss} in figure 2), which can assume values between 0.4 and 1 crank angle degrees (CAD) depending on the entity of the heat transfer and mass leakage.

$$\vartheta_{loss} = L_{PP} - L_{TDC} \tag{8}$$

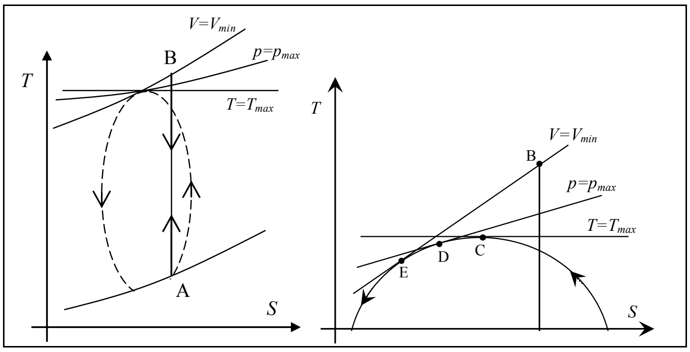

figure 1 Temperature-Entropy diagram of the compression-expansion process in a motored cylinder: ideal engine (segments AB and BA) and diabatic engine (dashed curve); the peak pressure (point D) occurs before the TDC (point E).

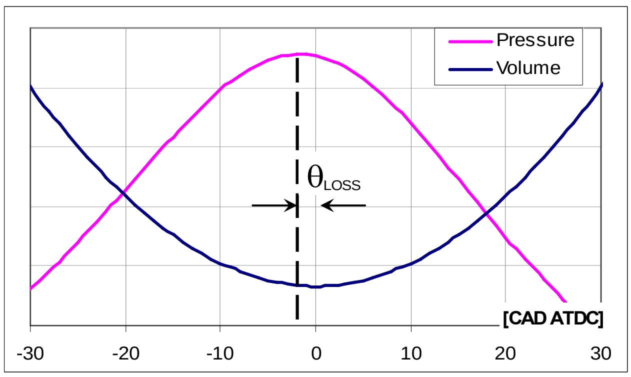

figure 2 Qualitative progress of in-cylinder pressure and volume near TDC

The loss angle represents the error committed when the (motored) pressure curve is phased with respect to volume by setting LPP=0. Since it can be significantly greater than the allowable TDC position error of 0.1 degrees, it is safer to be able to evaluate it. Equation (6) shows that two easily measurable quantities, the in-cylinder pressure and volume, allow the evaluation of the entropy variation (i.e. heat transfer) together with the mass leakage by means of the functions $\delta V/V$ and $\delta p/p$, which are plotted as example in figure 3; defining the "loss function" F so that:

$$\delta F = c_P \frac{\delta V}{V} + c_V \frac{\delta p}{p} \tag{9}$$

it will result:

$$\delta F = \delta\, S + c_P \frac{\delta m}{m} \tag{10}$$

The entity of the variation of the "loss function" δF, which gathers the sum of the two "*losses*", is then determined by the capability of the cylinder walls to exchange heat with the gas and by the amount of gas escaping from the cylinder. The qualitative progress of the "*loss function"* variation in a real cylinder during a compression-expansion process, together with its two constitutive terms δS and $c_P\ \delta m/m$, is shown for example in figure 4: as can be seen entropy variation (which depends on the difference T_{gas}-T_{wall}) and mass leakage (related to the difference between in-cylinder pressure and outer pressure) have a similar trend, reaching a minimum near the TDC: it follows that, in this position, the δF value is the sum of the two loss angle causes. Following this concept the authors tried to draw information on the loss angle entity directly from the loss function variation.

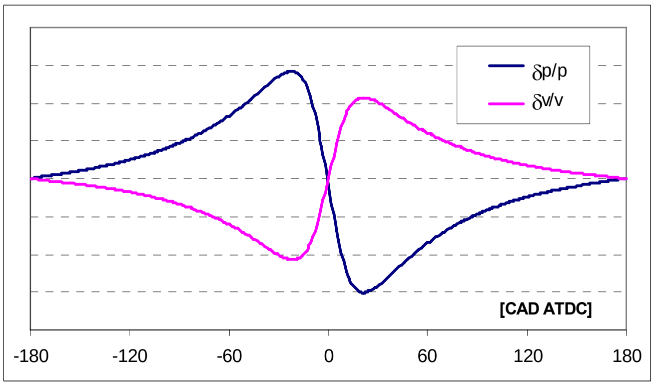

figure 3 Qualitative progress of δV/V and δp/p (δϑ=1 crank angle degree)

When the gas pressure reaches the peak value (i.e. at the LPP), the ratio $\delta p/p$ is zero, and equation (9) becomes:

$$\delta F_{LPP} = \left[c_p \frac{\delta V}{V} \right]_{LPP} \tag{11}$$

Equation (11) shows that at the peak pressure position the knowledge of the function δF allows to determine the value of $\delta V/V$ (see figure 3) which, depending only on engine geometry, is a known function of the crankshaft position, i.e. of the loss angle.

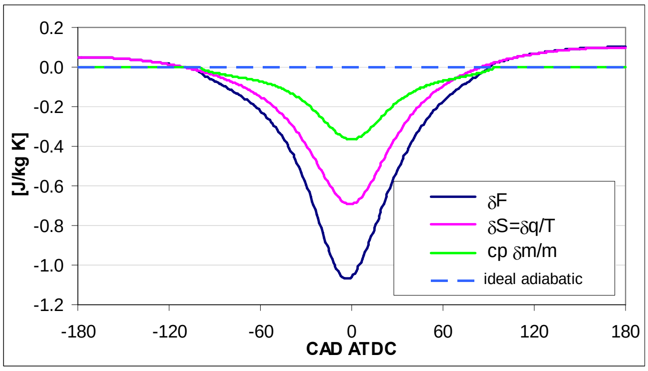

figure 4 Loss function variation δF and its constitutive terms vs. crank position (δϑ=1 crank angle degree).

The function $\delta V/V$ can be expressed as:

$$\frac{\delta V}{V} = \frac{\sin(\vartheta)\left(1 + \frac{\cos(\vartheta)}{\sqrt{\mu^2 - \sin^2(\vartheta)}}\right)\delta\vartheta}{\frac{2}{\rho - 1} + \mu + 1 - \cos(\vartheta) - \sqrt{\mu^2 - \sin^2(\vartheta)}} \tag{12}$$

where ρ is the compression ratio and μ is the rod to crank ratio (i.e. the ratio between connecting rod length and crank radius). Since the loss angle is normally around 1 degree (=0.017radians), further approximations can be made:

$$\sin(\vartheta_{loss}) \approx \vartheta_{loss} \quad \text{and} \quad \cos(\vartheta_{loss}) \approx 1$$

Thus at the LPP equation (12) becomes:

$$\left[\frac{1}{V}\frac{\delta V}{\delta\vartheta}\right]_{LPP} = \frac{\vartheta_{loss}\left(1 + \frac{1}{\sqrt{\mu^2 - \vartheta_{loss}^{\ 2}}}\right)}{\frac{2}{\rho - 1} + \mu - \sqrt{\mu^2 - \vartheta_{loss}^{\ 2}}} \tag{13}$$

Hence, being $\vartheta^2_{loss} << \mu^2$, equations (13) and (11) yield:

$$\vartheta_{loss} = \frac{2}{\rho - 1} \cdot \frac{\mu}{\mu + 1} \cdot \left[\frac{1}{c_p}\frac{\delta F}{\delta\vartheta}\right]_{LPP} \tag{14}$$

This demonstrates that the loss angle can be easily correlated to the loss function increment δF evaluated at the peak pressure position. Unfortunately the δF undergoes great distortions even with small phase errors between $\delta p/p$ and $\delta V/V$. Figure 5 shows some δF curves calculated assuming different phase errors (expressed as a fraction of the loss angle). As can be seen, even phasing the pressure diagrams with the loss angle error (setting LPP=0) a considerable evaluation error on δF_{LPP} would be introduced. This fact, without a reliable way to evaluate the δF at peak pressure position, would make equation (14) useless.

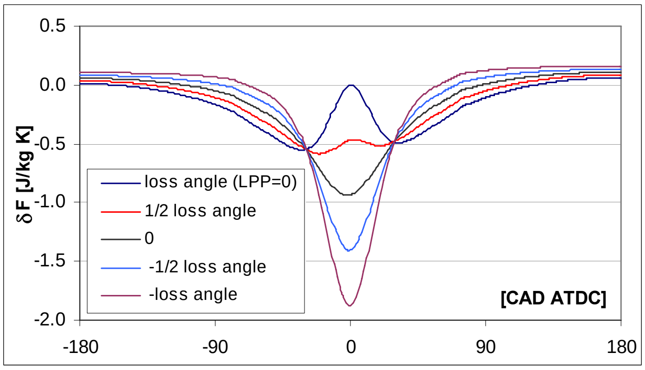

figure 5 Loss function increment δF for different phase errors ($\delta\vartheta$=1 crank angle degree)

The figure 5 however shows the existence of two zones in which the various curves assume the same values: here (about ±30 CAD ATDC in figure 3) the two functions $\delta p/p$ and $\delta V/V$ reach their extreme values and hence are poorly influenced by a little phase error (such as the loss angle); for this reason, according to equation (9), in these two crank positions the loss function increment remains almost unchanged. This means that assuming a TDC position error equal to the loss angle (easily achievable setting LPP=0), the values assumed by the loss function variation δF_1 and δF_2 in the two positions relative to the minimum and maximum $\delta V/V$, will be nearly correct. Hence, in order to determine the loss angle from equation (14), a correlation between δF_1, δF_2 and δF_{LPP} has been evaluated [7]: it has been found that, for a given engine, a proportionality relation exists between δF_{LPP} and δF_m (the mean value between δF_1 and δF_2):

$$\delta F_{LPP} = E_P \cdot \delta F_m \tag{15}$$

$$\delta F_m = \left(\frac{\delta F_1 + \delta F_2}{2} \right) = \left(\frac{\delta F_{\min\ dv/v} + \delta F_{\max\ dv/v}}{2} \right)$$

The constant E_p in equation (15) can be evaluated once the engine geometry and the heat transfer law are known and can be assumed equal to 1.95 as first approximation; this value has been determined in [7].

Summarizing, once the pressure cycle has been sampled, the procedure for the TDC estimation consists of 5 simple steps, here resumed:

1. the phase of the pressure cycle is adjusted setting LPP=0 (in this way the position error is exactly equal to the loss angle).
2. evaluation of the angular position ϑ_1 and ϑ_2 of the minimum and maximum $\delta V/V$, for example using the following equation [7]
$$\vartheta_{1,2} = \mp 76.307 \cdot \mu^{0.123} \cdot \rho^{-0.466} \tag{16}$$
where ρ is the engine compression ratio while μ is the rod to crank ratio (i.e. the ratio between the rod length and the crank radius).
3. calculus of the loss function increments δF_1 and δF_2 at the angular position ϑ_1 and ϑ_2 by means of equation (9) and hence calculus of their mean value δF_m=1/2 (δF_1+δF_2).
4. Evaluation of the loss function increment at the peak pressure position δF_{LPP} by means of the mean value δF_m through the proportionality constant E_p (equation (15)) which can be assumed to be 1.95 [7].
5. The loss angle ϑ_{loss}, and hence the TDC location, can be evaluated by means of equation (14).

EXPERIMENTAL TESTS

In order to verify the capability of the proposed method to determine the TDC position, some experimental tests have been carried out using a FIAT four cylinders spark ignition engine, whose characteristics are reported in table 1, connected to an eddy-current dynamometer.

Number of cylinders	4
Displacement	1242 [cc]
Bore	70.8 [mm]
Stroke	78.9 [mm]
Compression ratio	9.8
Rod/Crank ratio	3.27

Table 1 engine characteristics

The in-cylinder pressure has been measured in a motored cylinder on different conditions of manifold absolute pressure (MAP) and engine speed using an AVL GU13X piezoelectric sensor (installed by means of its ZC32 spark plug adaptor), while the MAP has been measured by means of a DRUCK piezoresistive pressure sensor.
The TDC location has been measured under the same condition of MAP and engine speed by means of a capacitive Kistler TDC sensor system 2629B placed in the spark plug hole of the motored cylinder; the measurement accuracy of this kind of sensor is 0.1° CA.
The data acquisition has been performed by the use of a National Instruments DAQ card 6062, using a 360 ppr encoder to trigger and clock the acquisition, thus sampling both pressure and TDC sensor signal with 1 crank angle degree resolution.

TDC LOCATION MEASUREMENT

A typical TDC sensor signal is visible in figure 6; it has been acquired using an Agilent 100MHz oscilloscope with 0.005 ms sample period that corresponds, at 1500 rpm engine speed, to a crank rotation of 0.045°. The same signal has been downsampled with 1 CAD resolution (figure 7) and interpolated, in a range of 7 CAD around the maximum, by means of a 4^{th} degree polynomial, which was then used to compute the location of the maximum value, obtaining the same TDC position of the curve sampled with a 0.045 CAD resolution. The same procedure has been followed using many different TDC sensor curves acquired by means of the oscilloscope at different engine speed, obtaining always a precise matching of the TDC positions. This demonstrates that a 1 CAD sampling resolution allows the estimation of the TDC position with the required accuracy of 0.1 degrees. This procedure has also been applied to in-cylinder pressure curves using a 3^{rd} order polynomial, confirming the capability to appraise the location of peak pressure with the same accuracy of 0.1 degrees.

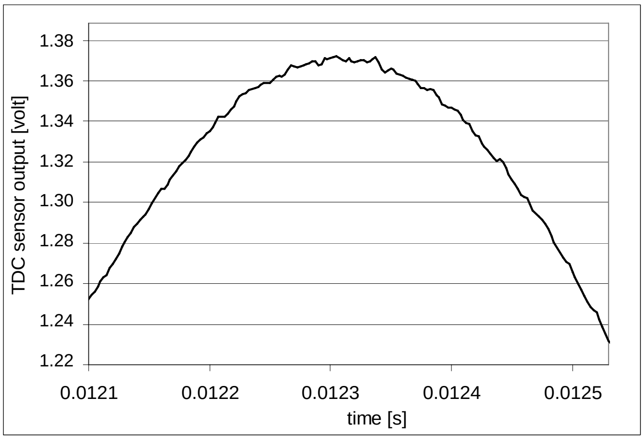

figure 6 A typical TDC sensor signal acquired with 0.005 ms resolution (1500 rpm)

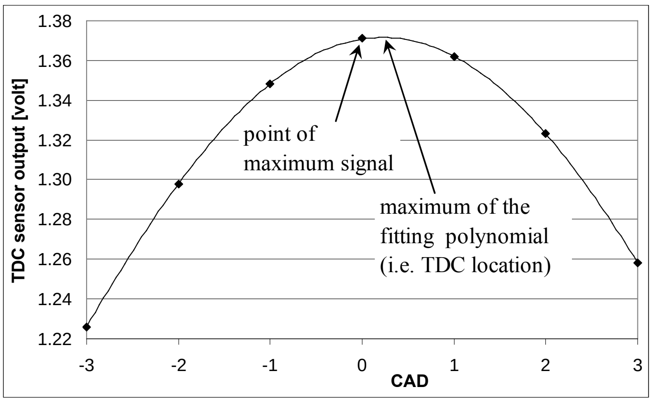

figure 7 An example of TDC sensor signal, acquired with 1 CAD resolution, fitted by a 4^{th} degree polynomial

For each test condition 100 consecutive TDC sensor curves have been acquired, one for each engine cycle (720° CA). For each curve the 4^{th} degrees polynomial fitting procedure has been applied thus obtaining the TDC location with the required accuracy of 0.1 degrees. In table 2 the results of all tests are reported in terms of mean TDC location and scattering. As can be seen the measured TDC location was 0.28 CAD, with a scattering (i.e. the difference between the maximum and the minimum value), over the different test conditions, inside the range of the sensor measurement accuracy (±0.06 CAD), while the mean scattering over 100 consecutive curves was ±0.21° CA (see the last column of table 2).

engine speed [rpm]	MAP [bar]	mean TDC location [°CA]	scattering [°CA]
1500	0.4	0.29	±0.22
1500	0.5	0.29	±0.23
1500	0.6	0.33	±0.21
2000	0.4	0.21	±0.20
2000	0.5	0.34	±0.20
2000	0.6	0.34	±0.24
2500	0.4	0.24	±0.21
2500	0.5	0.24	±0.19
2500	0.6	0.29	±0.25
3000	0.4	0.28	±0.14
3000	0.5	0.27	±0.17
3000	0.6	0.24	±0.20
	mean	**0.28 ±0.06**	**±0.21**

Table 2 TDC location

TDC DETERMINATION BY MEANS OF THE THERMODYNAMIC METHOD

The method proposed for the TDC position evaluation requires the measure of in-cylinder pressure under motored condition. In order to apply the method proposed, each pressure curve has to be phased setting LPP=0 CAD: in this way a positioning error equal to the loss angle is committed.

The in-cylinder pressure has been interpolated by means of a 3^{rd} degree polynomial in a range of 7 CAD around the maximum pressure value for the LPP determination; a 3^{rd} degree polynomial was also used to fit the $\delta p/p$ values in a range of 40 CAD around the two locations corresponding to the minimum and maximum $\delta V/V$: this procedure corresponds to a low-pass filtering that eliminates the high frequency noise, allowing a reliable evaluation of the loss function increment δF.

On the base of the fitting polynomial, the $\delta p/p$ and hence the δF values have been calculated in the two locations of minimum and maximum $\delta V/V$ (i.e. around ±30 CAD ATDC for the engine used in the test). The loss angle has been then calculated using equations (15) and (14), in which has been assumed c_p=1060 J/kg K and c_v=773 J/kg K considering the mean gas temperature between 30° CA BTDC and LPP. From equation (8) it was then possible to evaluate the TDC location.

Since the used pressure transducer is a piezoelectric sensor the pressure curves had to be compensated in order to obtain the absolute values. Three different techniques have been taken into consideration to compensate the pressure curves [8, 9]:

- MAP method: the in-cylinder pressure value at bottom dead centre (BDC) is changed to match the MAP value (measured by the apposite sensor) at the same CA.
- Mean MAP method: the in-cylinder pressure value at bottom dead centre (BDC) is changed to match the mean MAP value measured along the whole engine cycle (720° CA).
- Polytropic exponent method: the pressure curve is shifted imposing the exponent γ of the polytropic transformation during the compression phase of the gas.

In order to evaluate the influence of pressure curves compensation on the results, both the MAP method and Polytropic exponent method where used. The latter was employed considering five different polytropic exponent values (1.30, 1.32 … 1.38).
Thus, for each test condition, the 100 acquired pressure curves were firstly compensated and then employed for the thermodynamic evaluation of the TDC position.
The values obtained for the loss angle (ϑ_{loss}) and TDC position, for every test condition, are reported in table 4 (polytropic exponent compensation with γ=1.34); each result represents the mean of 100 values, whose scattering is also reported in the last column. Table 4 also shows the values of measured loss angle ($L_{TDCmeasured}$ - L_{PP}). The loss angle values (ϑ_{loss}) and TDC position calculated for each compensation method are reported in table 5: here only the mean values over the different test conditions are reported (i.e. the last row in table 4). It is clear that the thermodynamic method results are quite unaffected by the compensation method used.

engine speed [rpm]	MAP [bar]	measured ϑ_{loss} [°CA]	evaluated ϑ_{loss} [°CA]	scattering ϑ_{loss} [°CA]	TDC [°CA]	scattering [° CA]
1500	0.4	-0.82	-0.72	1.52	0.19	±1.52
1500	0.5	-0.79	-0.78	1.09	0.28	±1.04
1500	0.6	-0.74	-0.75	0.94	0.34	±0.92
2000	0.4	-0.73	-0.76	1.48	0.24	±1.45
2000	0.5	-0.73	-0.49	1.32	0.11	±1.24
2000	0.6	-0.70	-0.76	1.02	0.40	±0.97
2500	0.4	-0.75	-0.63	1.31	0.12	±1.40
2500	0.5	-0.73	-0.64	1.00	0.15	±0.99
2500	0.6	-0.75	-0.63	0.83	0.18	±0.83
3000	0.4	-0.68	-0.64	1.14	0.25	±1.15
3000	0.5	-0.74	-0.56	0.88	0.10	±0.93
3000	0.6	-0.67	-0.46	0.72	0.02	±0.72
mean		**-0.73** ±0.08	**-0.65** ±0.16	**±1.10**	**0.20** ±0.19	**±1.10**

Table 4 Results of the thermodynamic method (pressure curves compensated by means of the polytropic index γ=1.34)

The mean value of the TDC found by means of thermodynamic method (table 4, column 6), is quite coherent with the mean measure obtained by the use of the TDC sensor: 0.20 and 0.28 CAD respectively. Moreover the scattering of the mean values, over the different test conditions, is ±0.19 CAD that is three times the scattering obtained by the use of the TDC sensor (±0.06 CAD). If the scattering over the 100 values of each test condition is considered (see last column in table 2 and 4) the thermodynamic method gives values of ±1.10 CAD against the ±0.21 CAD of the sensor. The authors consider this a quite satisfactory result, since the method is based on the pressure curves values which are affected both by measurement uncertainties and by cycle-by-cycle variations even under motored conditions.

Compensation method	Mean ϑ_{loss} [°CA]	Mean scattering ϑ_{loss} [°CA]	Mean TDC [°CA]	Mean scattering TDC [°CA]
MAP	-0.65 ±0.18	±1.09	0.19 ±0.18	±1.08
Polytropic exponent γ=1.30	-0.65 ±0.16	±1.09	0.19 ±0.19	±1.08
Polytropic exponent γ=1.32	-0.65 ±0.16	±1.10	0.19 ±0.19	±1.09
Polytropic exponent γ=1.34	-0.65 ±0.16	±1.10	0.20 ±0.19	±1.10
Polytropic exponent γ=1.36	-0.66 ±0.16	±1.11	0.20 ±0.19	±1.10
Polytropic exponent γ=1.38	-0.66 ±0.16	±1.12	0.21 ±0.19	±1.11

Table 5 Influence of pressure curves compensation

CONCLUSIONS

In this paper a new thermodynamic method for the TDC determination based on the in-cylinder pressure analysis has been validated by means of experimental tests. The TDC location, obtained by the use of the thermodynamic method, has been compared with the measure carried out by means of a commercial TDC sensor: the results obtained show a quite good matching. The scattering of TDC position mean values, over the different test conditions, are also quite similar, and this represents a good result too, considering that the thermodynamic method is based on the analysis of pressure curves. The thermodynamic method shows a good prediction capability poorly by changes in operative conditions (namely crankshaft speed and MAP) and by pressure measurement uncertainties.

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DEFINITIONS, ACRONYMS, ABBREVIATIONS

TDC: top dead centre

ATDC: after top dead centre

BTDC: before top dead centre

CA: crank angle

CAD: crank angle degrees

IMEP: indicated mean effective pressure

LPP: location of peak pressure [CAD]

MAP: manifold absolute pressure [bar]

$\mathbf{L_{TDC\ measured}}$: TDC location measured by the TDC sensor [CAD]

δ**Y**: increment of the generic function Y for a crank rotation $\delta\vartheta$

q: specific heat received by the gas from the cylinder walls [J/kg]

p: gas pressure [Pa]

v: gas specific volume [m^3/kg]

V: in-cylinder volume [m^3]

$\mathbf{T_W}$: cylinder walls mean temperature [K]

$\mathbf{T_{GAS}}$: gas mean temperature [K]

u: specific internal energy of the gas [J/kg]

R': gas constant [J/kg K]

$\mathbf{c_p}$, $\mathbf{c_v}$: constant pressure and constant volume specific heat of the gas [J/kg K]

S: gas specific entropy [J/kg K]

m: mass of the in-cylinder gas [kg]

Q: heat received by the mass *m* of the gas from the cylinder walls [J]

δ**F**: loss function variation for a given $\delta\vartheta$ crank rotation[J/kg K]

Greek

$\delta\vartheta$: finite crank rotation

ϑ: crank angle [CAD]

ϑ_{loss}: loss angle [CAD]

ρ: engine compression ratio

μ: rod to crank ratio

γ: polytropic exponent