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Energy Procedia 81 (2015) 950 – 959

Energy Procedia

69th Conference of the Italian Thermal Engineering Association, ATI 2014

# Accelerometer Based Methodology for Combustion Parameters Estimation

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**Abstract**

Due to increasingly stringent emission regulations and the need of more efficient powertrains, obtaining information about combustion process becomes a key factor. Low-cost in-cylinder pressure sensors are being developed, but they still present long-term reliability issues, and represent a considerable part of the engine management system cost. Research is being conducted in order to develop methodologies for extracting relevant combustion information using standard sensors already installed on-board. The present work introduces a methodology for combustion parameters estimation, through a control-oriented analysis of structure-borne sound. The paper also shows experimental results obtained applying the estimation methodology to different passenger car engines.


Peer-review under responsibility of the Scientific Committee of ATI 2014

*Keywords:* gasoline engine, diesel engine, control, accelerometer signal, in-cylinder pressure signal, combustion parameters, closed-loop control, spark advance, start of injection.

## 1. Introduction

In order to comply with most recent emission regulations, it has been already widely demonstrated by many researchers that a significant reduction of pollutant emissions can be achieved by using closed-loop combustion control, based on real time in-cylinder pressure analysis [1,2,3].

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Peer-review under responsibility of the Scientific Committee of ATI 2014
doi:10.1016/j.egypro.2015.12.152

Cost and long-term reliability of in-cylinder pressure sensors still represent the main issues towards the on-board implementation of closed-loop combustion control algorithms. Even if the poor sensors' reliability is overcome in the future, they will still represent an important rate of the overall control system's cost. Due to these reasons, many methodologies to estimate fundamental combustion parameters, based on transducers already mounted on-board, are available in scientific literature. For example: in-cylinder pressure reconstruction from torque sensor [4], analysis of crankshaft speed fluctuations [5,6,7] or structure-borne sound using vibration transducers [8].

The paper presents a methodology to extract useful information for combustion phasing closed-loop control, based on vibration signal analysis. This approach has been applied on both gasoline and diesel engines; the use of on-board vibration signal information is a widespread solution adopted for knock onset detection in gasoline engines [9], whereas in the diesel application is not as common, but its introduction would not result in an unacceptable cost increase.

The development of a control system able to detect the combustion phase within the engine cycle, would enable the engine efficiency (or the engine torque) to be maximized. Indeed, it is well known that the torque output of an engine is strongly correlated to the combustion phase. Many parameters can be defined to represent the combustion phase: the most commonly used are the Pressure Peak Position (PPP) and the Center of Combustion (CA50), which is defined as the crankshaft angle where the 50% of fuel mass is burned. The combustion phase and the previously mentioned efficiency-related parameters, are influenced by spark advance (for spark ignition engines), and by the start of main injection (for compressed ignition engines). Therefore, knowing these quantities would enable the implementation of a closed-loop combustion control system. Figure 1.a and 1.b show some experimental results, for the gasoline engine used in this work, in which the normalized Indicated Mean Effective Pressure (IMEP) is represented versus PPP and CA50.

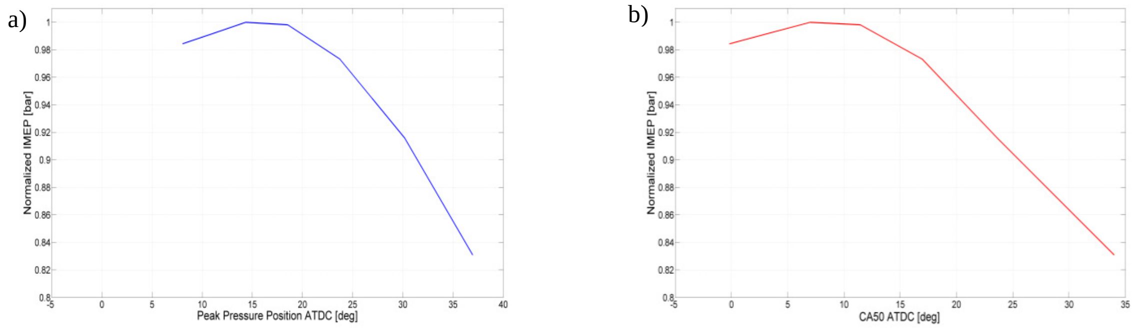


Figure 1. IMEP versus PPP (a) and IMEP versus CA50 (b). Spark ignition engine.

The two combustion phase parameters taken into account have interesting properties; the values they assume when maximizing the IMEP can be considered to be constrained in a narrow range, similar for every engine and essentially constant through the overall operating range. These properties are demonstrated from experimental experiences and can be used to identify a target value. The typical crank angle in which the peak pressure must be placed to achieve the maximum torque output is about 16 deg. after TDC (Top Dead Center), as shown by Hubbard et. al. [10], while for the CA50, the optimal position is about 10 deg. after TDC, as shown by Heywood [11].

Typically, in commercial ECUs, the combustion phase parameters are not calculated, since this implies the need to directly measure the pressure inside the combustion chamber. The development of alternative ways to estimate these quantities using non-intrusive methodologies or sensors already installed on-board, would avoid increasing the production costs, the design complexity and the long-term reliability issues related to the use of in-cylinder pressure sensors. This paper represents a first attempt to explore the possibility of estimating combustion phase parameters such as PPP, by means of real-time processing the vibration signal.

## 2. Experimental setup

The gasoline engine chosen for this study is a 4-cylinder 1.4L super- and turbo-charged direct-injection system, for passenger car applications. Table 1 shows the main technical data of the engine.

Table 1. Technical data of the gasoline engine

| **Displacement** | 1390 $cm^3$ |
|---|---|
| **Stroke** | 75.6 mm |
| **Bore** | 76.5 mm |
| **Compression Ratio** | 10.0 |
| **Max Torque** | 220 Nm @ 1500-4000rpm |
| **Max Power** | 103 kW @ 6000 rpm |
| **Number of valves** | 4 per cylinder |
| **Injection system** | Direct Injection |

The transducer used to measure the engine vibration is the one already installed on-board between cylinders 2 and 3, dedicated to knock detection.

During the experiments carried out on a test bench, in-cylinder pressure and vibration signals have been recorded at high frequency (simultaneously sampled at 200 kHz), whereas the most important ECU variables at 50 Hz.

The operating points tested under steady-state conditions are shown in Figure 2.

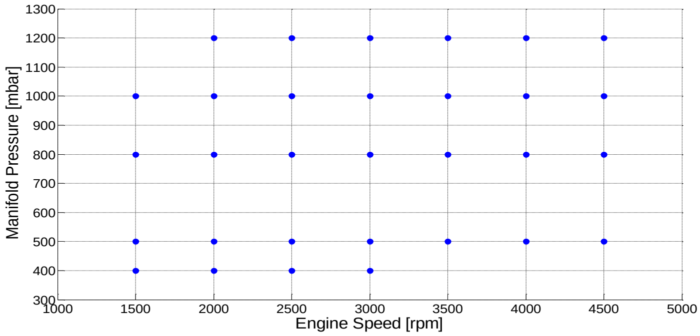


Figure 2. Operating points tested on the gasoline engine.

Regarding the diesel engine, the experiments have been carried out on a 4-cylinder 1.3L turbo-charged common-rail system, for passenger car applications. The main technical data are reported in Table 2.

Table 2. Technical data of the diesel engine.

| **Displacement** | 1248 $cm^3$ |
|---|---|
| **Stroke** | 82.0 mm |
| **Bore** | 69.6 mm |
| **Compression Ratio** | 16.8 |
| **Max Torque** | 200 Nm @ 1500 rpm |
| **Max Power** | 70 kW @ 3800 rpm |
| **Number of valves** | 4 per cylinder |
| **Injection system** | Common Rail Multi-Jet |

As the vibration measurement was not available, a Bruel&Kjaer 4393 accelerometer has been installed on the engine block, between cylinders 2 and 3. The in-cylinder pressure and vibration signals have been recorded at high frequency (simultaneously sampled at 100 kHz), whereas the most important ECU variables at 200 Hz.

The engine speed range analyzed goes from 1500 to 3000 rpm with steps of 500 rpm, while the load range has been discretized from 0 to 12 bar with steps of 2 bar of Brake Mean Effective Pressure (BMEP). The sensors' setup, similar for both engines, is shown in Figure 3.

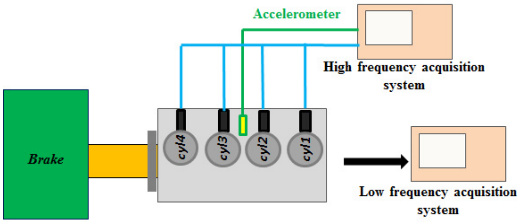


Figure 3. Sensors' configuration for the gasoline and diesel engines.

## 3. Preliminary analysis of the vibration signal

In order to extract information about the combustion process, in-cylinder pressure signal can be used to calculate the most important combustion parameters, such as Indicated Mean Effective Pressure (IMEP), Ignition Delay (ID), Start of Combustion (SoC) and Center of Combustion (CA50).

To compare in-cylinder pressure and engine vibration signal, the two quantities need to show a satisfying coherence in the frequency domain associated to the combustion process. Figure 4 and Figure 5 show the coherence function between in-cylinder pressure and vibration trace for both engines considered in this study, calculated using the equation (1).

$$C_{xy}(f) = \frac{|P_{xy}(f)|^2}{P_{xx}(f)P_{yy}(f)} \tag{1}$$

$C_{xy}(f)$ represents the magnitude squared coherence estimate. It is a function of frequency with values between 0 and 1 that indicates how well $x$ corresponds to $y$ at each frequency. The magnitude squared coherence is a function of the power spectral densities ($P_{xx}(f)$ and $P_{yy}(f)$) of $x$ and $y$ and the cross power spectral density ($P_{xy}(f)$).

Observing Figure 4 and Figure 5 it can be noticed that the engine firing frequency and its multiples have a coherence value close to 1 in both cases taken into consideration. This means that the firing frequency and its harmonics spread through the engine block and they are captured in the same way by both pressure and vibration transducers.

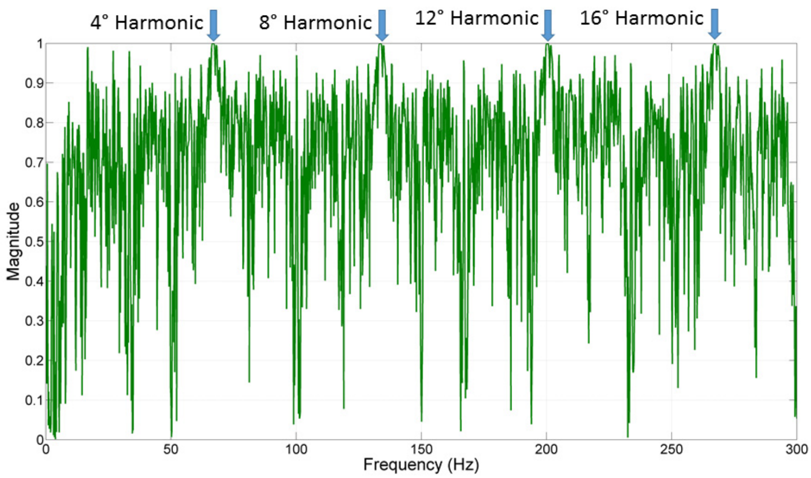


Figure 4. Coherence function between in-cylinder pressure and vibration trace for the gasoline engine, at 2000 rpm.

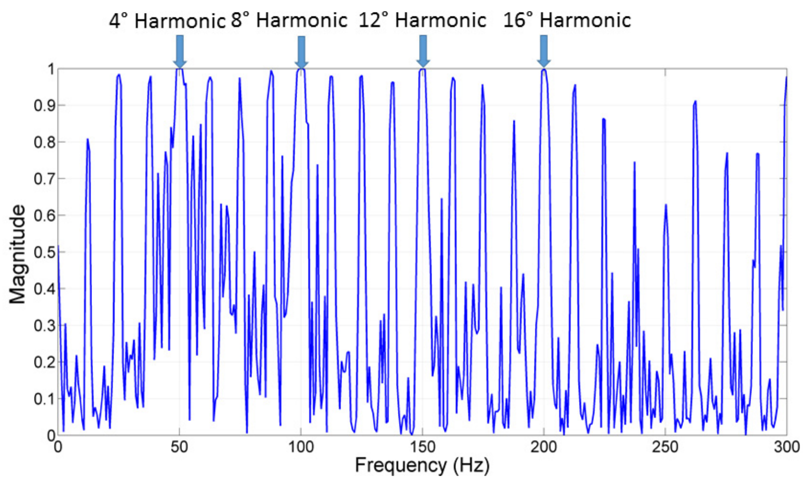


Figure 5. Coherence function between in-cylinder pressure and vibration trace for the diesel engine, at 1500 rpm.

Due to the good coherence between accelerometer and in-cylinder pressure, it is reasonable to assume that the information contained in the pressure trace about the combustion phenomenon, can be extracted from the vibration signal as well. This means that relevant combustion parameters, such as PPP and SoC, can in principle be extracted from the accelerometer signal, if properly processed.

## 4. Peak pressure position detection for the gasoline engine

The torque output of a spark ignition engine is influenced by the combustion phase, which can be optimized by adjusting the Spark Advance (SA). The spark timing is controlled by the ECU in open loop, usually using the values stored in a map spanned by engine speed and load. The SA values collected in the map are determined by performing specific tests on the test bench for each operating condition. Due to the numerous experiments needed to derive a proper SA map, this approach is time demanding; moreover, the SA values in the map are usually conservative, in order to ensure safe running of the engine. The implementation of a system able to estimate in real-time the combustion phase would enable to update the SA map during the engine entire life, allowing continuous maximization of its efficiency.

Figure 6.a shows the comparison between cylinder pressure and vibration trace. The accelerometer signal has been low pass filtered, in order to isolate only the harmonic components useful for the identification of the combustion parameter. The correlation between the two waveforms is hard to tell, but if the pressure signal first derivative is used instead of the pressure trace itself (Figure 6.b), a clearer information can be extracted. The similarity between the acceleration signal and the pressure first derivative could be explained by assuming that a vibration that propagates through the engine block can be caused by the force impulses generated by the pressure gradient variation. Hence, it may be concluded that the accelerometer signal is particularly sensitive to pressure gradient variations.

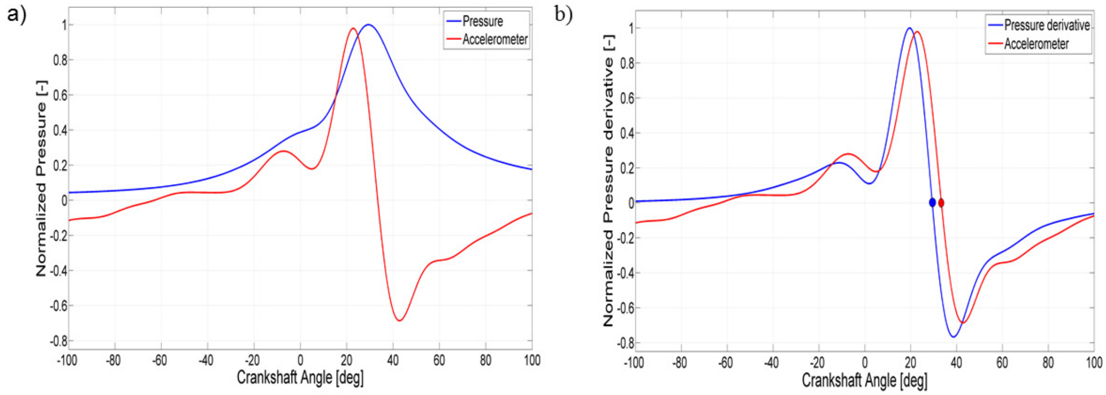


Figure 6. (a) Comparison between in-cylinder pressure and accelerometer signal; (b) Comparison between in-cylinder pressure first derivative and accelerometer signal.

Observing Figure 6.b, the zero-crossing angle of the pressure derivative signal is in fact significantly correlated to the acceleration zero-crossing position. It can be noted a time delay between the zero-crossing of the two signals, due to the propagation time through the engine block. In order to estimate the PPP properly, such time delay can easily be compensated.

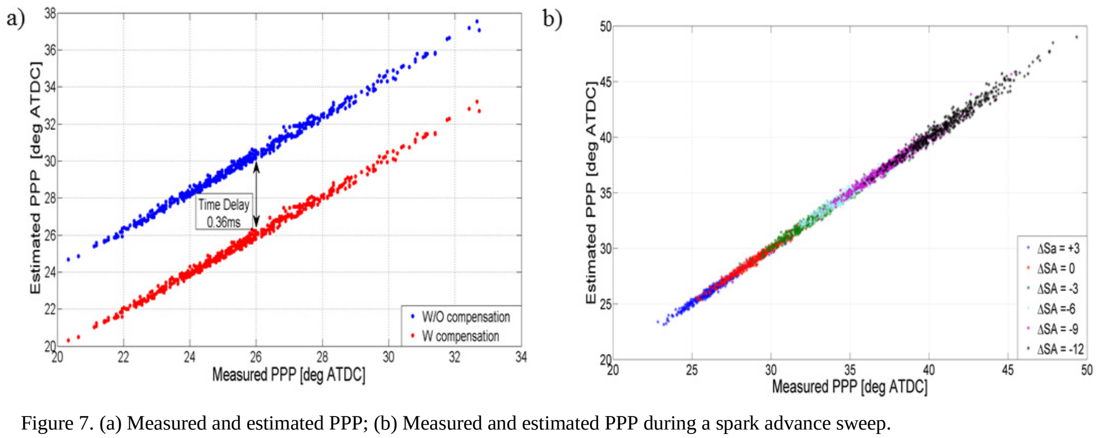


Figure 7. (a) Measured and estimated PPP; (b) Measured and estimated PPP during a spark advance sweep.

In Figure 7.a, the correlation between PPP calculated with the in-cylinder pressure and PPP detected by accelerometer's zero-crossing is shown, for a test at 2000 rpm and 1200 mbar. The constant time delay identified is 0.36 ms. The Root Mean Square Error (RMSE) is equal to 0.12 CA deg. Furthermore, Figure 7.b shows the same result, but extended to a spark advance sweep. In this case, the time delay has been already compensated and the overall RMSE is equal to 0.22 CA deg.

The pressure peak position estimation methodology has been applied on each operating point listed in Figure 2 and the experimental results, in terms of linear correlation, for the cylinders 2 and 3, are shown in Table 3 and Table 4.

Table 3. Correlation factor between PPP calculated via in-cylinder pressure signal and PPP estimated via acceleration signal, for cylinder 2.

| | 400 [mbar] | 500 [mbar] | 800 [mbar] | 1000 [mbar] | 1200 [mbar] |
|---|---|---|---|---|---|
| 1500 [rpm] | 3.81% | 87.81% | 98.39% | | |
| 2000 [rpm] | 6.21% | 44.64% | 93.15% | 96.73% | |
| 2500 [rpm] | | 34.19% | 91.45% | 94.93% | 96.92% |
| 3000 [rpm] | | 98.70% | 99.53% | 96.15% | 96.78% |
| 3500 [rpm] | | 83.52% | 84.88% | 87.94% | 94.62% |
| 4000 [rpm] | | 90.89% | 95.56% | 94.99% | 97.05% |
| 4500 [rpm] | | 69.74% | 90.07% | 91.53% | |

Table 4. Correlation factor between PPP calculated via in-cylinder pressure and PPP estimated via acceleration signal, for cylinder 3.

| | 400 [mbar] | 500 [mbar] | 800 [mbar] | 1000 [mbar] | 1200 [mbar] |
|---|---|---|---|---|---|
| 1500 [rpm] | 7.80% | 90.09% | 98.67% | | |
| 2000 [rpm] | 28.32% | 58.99% | 93.89% | 95.71% | |
| 2500 [rpm] | | 59.96% | 91.09% | 94.58% | 96.73% |
| 3000 [rpm] | | 46.26% | 99.18% | 95.67% | 96.27% |
| 3500 [rpm] | | 80.10% | 87.06% | 91.36% | 93.48% |
| 4000 [rpm] | | 89.17% | 95.08% | 91.72% | 90.35% |
| 4500 [rpm] | | 73.85% | 88.34% | 90.87% | |

The correlation coefficients have been calculated using Pearson's linear correlation, expressed by Equation (2):

$$r(x,y)=\frac{\sum_{i=1}^{N}(x_i-\bar{x})(y_i-\bar{y})}{\sqrt{\sum_{i=1}^{N}(x_i-\bar{x})^2\sum_{i=1}^{N}(y_i-\bar{y})^2}} \tag{2}$$

Observing the above tables, it turns out that the correlation coefficients generally assume very high values, except for low speed and low load regions. At low load, the variations of the pressure gradient within the cylinder are not sufficiently intense to generate a pulse that can be propagated through the engine block, and reach the acceleration sensor.

## 5. Diesel combustion indexes estimation

This section of the paper shows how the vibration signal has been used to estimate the PPP for the diesel engine. In order to achieve the proposed goal, a tri-axial accelerometer (Bruel&Kjaer type 4393) has been mounted between cylinders 2 and 3. According to Arnone et al. [12], in order to isolate only the harmonic components of interest, a band-pass filter has been applied to the vibration signal, rather than a low-pass filter as the one used for the gasoline engine (presented above).

The same methodology developed and tested for a gasoline engine application has been applied, namely the acceleration signal zero-crossing position has been used to estimate the pressure peak position. Figure 8 shows the comparison between in-cylinder pressure signal and the band pass filtered acceleration signal, for a test run at 1500 rpm and 4 bar BMEP.

Figure 9 shows the comparison between measured and estimated pressure peak, during the same test (1500 rpm and 4 bar BMEP). The time delay between in-cylinder pressure and vibration signals may be easily compensated also in this case. The RMSE for PPP detection is 0.21 deg.

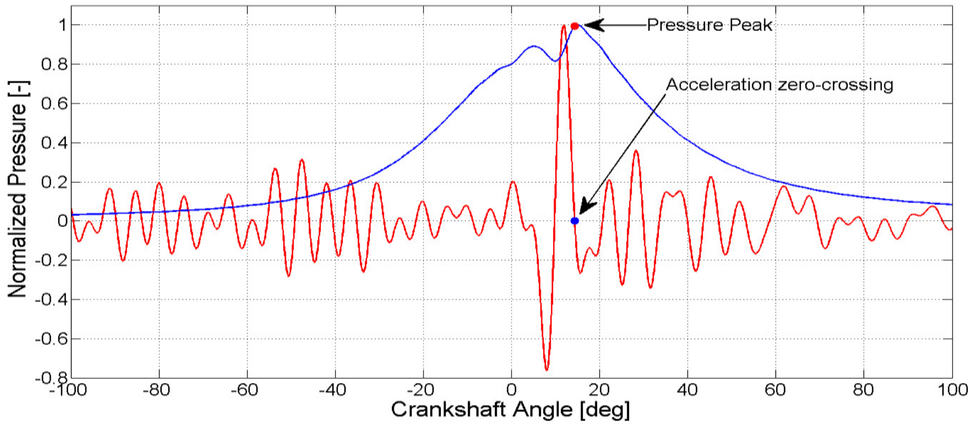


Figure 8. Comparison between in-cylinder pressure and accelerometer signals.

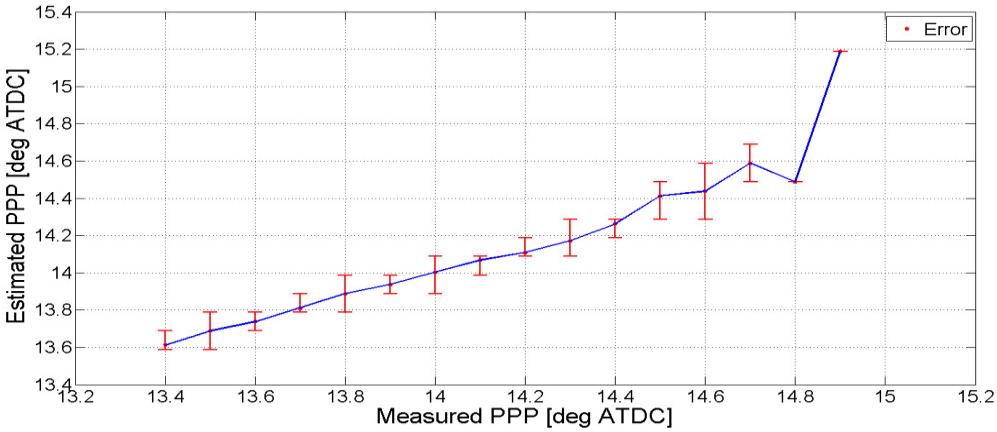


Figure 9. Measured PPP from in-cylinder pressure signal, and estimated from acceleration signal, for a test run at 1500 rpm and 4 bar BMEP.

By deeply analyzing the in-cylinder pressure signal, accurate information about the combustion process can be extracted. For example: it is possible to identify the Start of Combustion of the Pre and Main injection, by analyzing the Rate of Heat Release (RoHR) calculated using the equation (3).

$$RoHR = \frac{1}{\gamma - 1} \cdot V \cdot \frac{dp}{d\theta} + \frac{\gamma}{\gamma - 1} \cdot p \cdot \frac{dV}{d\theta} \tag{3}$$

In the light of the high coherence levels of the frequencies related to the combustion process, as above shown, it is reasonable to think that relevant information about the combustion process can be extracted also from the acceleration signal. In order to prove this, the RoHR has been compared with the structure-borne sound (Figure 10). Observing Figure 10, it is still hard to identify the correlation between the two waveforms; the acceleration signal has then been compared to the Rate of Heat Release First Derivative (RoHRD), as shown in Figure 11, resulting in a much greater similarity between the two signals.

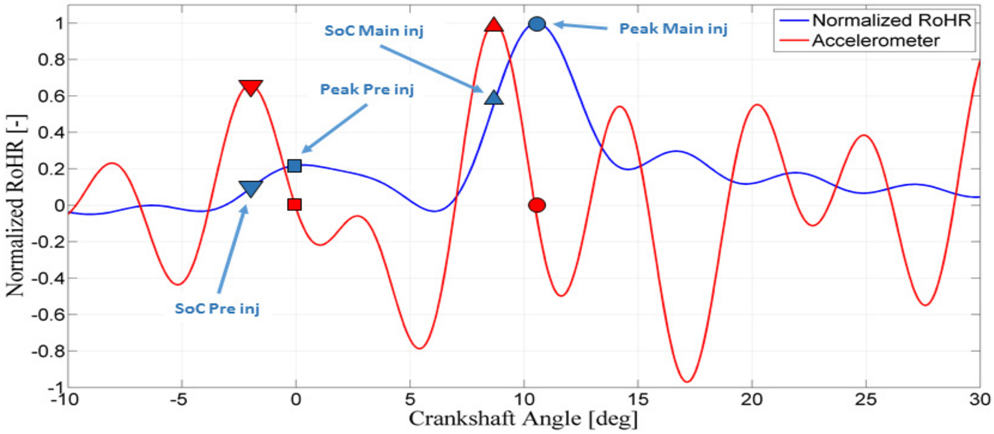


Figure 10. Comparison between RoHR and structure-borne sound, after time delay compensation.

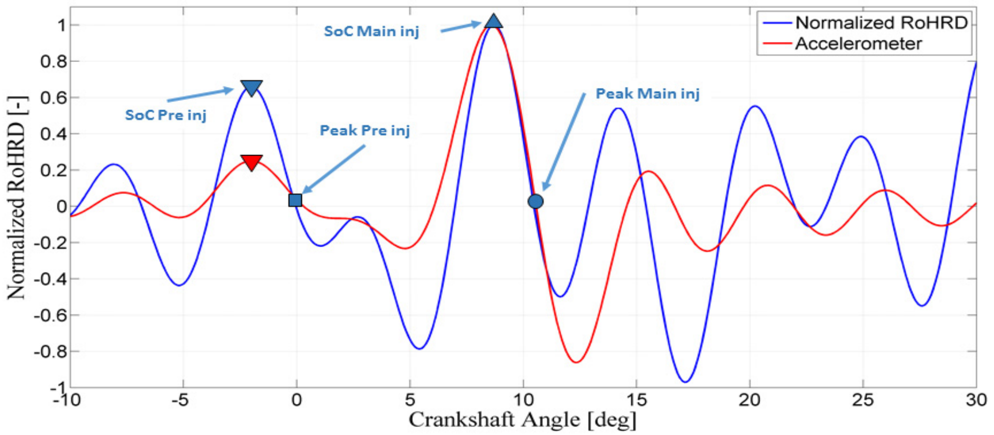


Figure 11. Comparison between RoHR first derivative and acceleration signal, after time delay compensation.

Looking at Figure 11, the similar information content of the two waveforms can be clearly noticed. The peaks of the vibration signal can be correlated to the Start of Combustion (SoC) of Pre and Main injections, and the acceleration and the RoHRD zero-crossings seem to correspond, providing further insight into the combustion process, demonstrating that the proposed approach appears to be promising also for diesel engine applications.

## 6. Conclusion

The paper presents the development of a methodology useful to extract important combustion-related parameters from the engine block vibration signal. The methodology has been successfully applied on both gasoline and diesel engines, for light duty applications. The results obtained during the tests can be considered sufficiently accurate to enable closed-loop combustion control, for both the types of engines taken into consideration, but the performance of the detection system is still not satisfying for the cylinders located farther from the accelerometer transducer. Several tests are being conducted to increase the signal to noise ratio, by varying the accelerometer position and considering also the installation of more than one accelerometer.

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