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# Combustion Parameters Estimation and Control Using Vibration Signal : Application to the Diesel HCCI Engine

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*Abstract*— **This paper proposes a method to estimate and to control the combustion timing of a Diesel Homogeneous Charge Compression Ignition (HCCI) engine without cylinder pressure transducers. The principle of the proposed controller is to correct the start of injection (SoI) in order to compensate the effects of the air path errors or injection system perturbation. The controller takes as inputs the middle of combustion estimated from the vibration trace recorded with a knock sensor fitted at the surface of the engine block. The estimated combustion parameters are computed in real-time and sent to the engine control system in a cycle to cycle manner. The contribution of this paper is combustion state estimator and controller that allows to maintain the desired combustion timings. Tests are performed on a test bench to compare the estimated and measured combustion timing. The closed-loop controller is validated in steady engine conditions and the obtained results demonstrate the potential of this approach.**

## I. INTRODUCTION

Increasing demands on emissions reduction and efficiency lead to a progressive introduction of cleaner combustion concepts. For the Diesel engine, applications of low temperature combustion (LTC) concept have recently become of major interest. This kind of combustion can be achieved by preparing a highly diluted mixture of air, fuel and burned gases achieving the auto-ignition of this mixture simultaneously in the whole space of the cylinder. The NOx production is linked with flame temperature, thus, due to high EGR rate, the flame temperature tends to decrease and the NOx emissions accordingly [1]. Very low level of NOx emission can be achieved thanks to the HCCI combustion [2] such that no dedicated post-treatment systems are required. However, the potential benefits of HCCI combustion in terms of emissions are counterbalanced by its high sensitivity to in-cylinder thermodynamic conditions since auto-ignition delay and heat release rate are mainly controlled by chemical kinetics. The first order variables are : the intake pressure a the intake valve closing (IVC), the burn gas rate (BGR) at IVC, the intake manifold temperature at IVC. The fuel path controller must be adapted to takes into account the possible errors between the desired operating point and the real operating condition. For instance, during engine transients, the intake manifold gas state (pressure, temperature and composition) cannot be achieved instantaneously. Thus, the engine fuel path controller must takes into account of possible errors in order to avoid non-optimal combustions. Cylinder to cylinder BGR discrepancies are also possible due to the intake system design and aerodynamic phenomena. This leads to cylinder to cylinder torque production unbalance. Moreover, too large BGR overshoots may cause misfires and engine instability. The in-cylinder temperature is also strongly affected by the engine bloc and piston temperature history. Finally, the fuel properties such as the cetane index tend to change the combustion auto-ignition delay.

The combustion in HCCI engine can be perturbed by many sources, its control is inherently more difficult than in standard internal combustion engines and needs more information about the combustion process. This can be achieved by using cylinder pressure sensor [3], [4]. The cylinder pressure signal provides detailed information about the combustion process extracted through the rate of heat release (ROHR) analysis. Direct monitoring of pressure within the cylinder allows an accurate diagnosis of the combustion process. On one side, the cylinder pressure signal provide relevant information for closed loop control. On the other side, the cylinder pressure sensors are expensive and have a short lifetime. For these reasons, commercial car implementations are limited. Indirect measurements or state estimation are preferable in most of the cases. One emergent approach relies on the analysis of the engine vibration signal.

This paper presents a combustion state observer and a closed loop combustion timing controller without cylinder pressure sensors. The combustion parameters estimation from the engine block vibration is demonstrated in section II. The tuning of the combustion observer and the methodology to perform the combustion mapping are presented in section III-A. A SISO controller is implemented on the engine management system to correct the SoI with the estimated combustion timing as a main feedback variable (section III-B). The experimental facilities including the HCCI engine, the combustion analysis platform and the rapid prototyping systems are presented in the section III-C. The experimental results are reported in section III-D. The paper closes with conclusion and future directions of this work.

## II. COMBUSTION STATE ESTIMATION FROM VIBRATION TRACE

The pressure signal reconstruction from the engine block vibration has been recognized as a difficult issue due to highly nonlinear relation between cylinder pressure evolution and vibration signal. For this purpose, signal deconvolution or inverse filtering provides quite good results [5]. Antoni *et*

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*al.* showed that better pressure reconstruction is possible by taking into account the non-stationary nature of the vibration signals [6], [7]. The structural vibration signal simultaneously collects information on several cylinders. In this framework, Urlaub and Böhme proposed an algorithm to estimate pressures in each cylinder from the structure-borne sound recorded on the surface of the engine block [8]. Instead of a whole combustion history reconstruction, we choose to estimate only two global variables that can characterize the quality of the combustion over one engine cycle : the combustion timing and the combustion noise. The main issue is to separate the combustion from all the other noise signatures on the accelerometric signal. In the first part of this work, we performed an off-line analysis to understand the relation between the two signals (pressure and knock). From this off-line analysis we could also determine the best settings for the on-line combustion parameter extraction algorithm.

### A. *Cylinder pressure and vibration signals off-line analysis*

This section presents the off-line analysis of the cylinder and knock signals (Fig. 1). The main goal is to recover the information related to the combustion phenomena from the engine block vibration.

Firstly, a crank-angle windowing is applied separate the combustion from the other cylinders noise sources. The window position is defined relatively to the top dead center where the combustion occurs. The window must be large enough to include the vibration trace in any cases over the whole engine mapping.

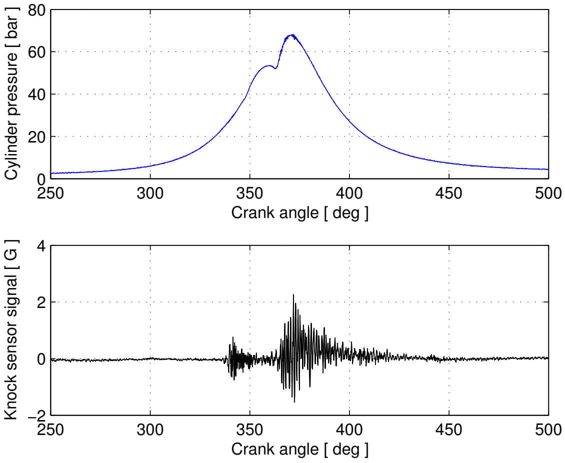

Fig. 1. Example of cylinder pressure (top) and knock sensor (bottom) acquisitions at 1500 rpm, 7 bar of IMEP. The angular resolution is 0.1 CAD (Crank Angle Degree).

Secondly, an off-line time-frequency analysis of the cylinder pressure and the resulting knock signal determines the frequency range where the information about the combustion can be extracted. Examples of time-frequency maps on the same operating point are represented in figures 2.

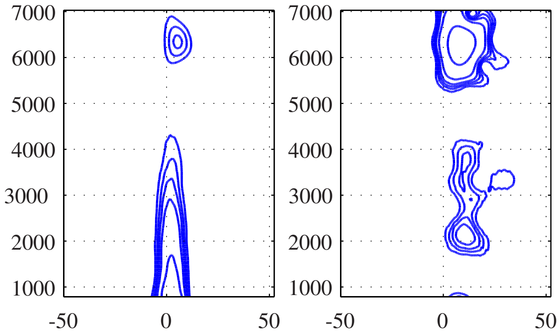

Fig. 2. Time-frequency analysis of pressure and knock signals : Wigner-Ville spectrum on pressure gradient (left), Wigner-Ville spectrum on knock signal (right). X and y axis units are crank angle degree and frequency respectively. This time-frequency analysis has been performed at 1500 rpm, 3 bar of IMEP in HCCI mode with a single injection using the Wigner-Ville transform [9].

One can observe in the first angular-frequency representation that the pressure signal exhibits different components within the same crank angle window where the combustion process occurs. The low frequency component is related to the auto ignition and appears few crank angle degrees before the TDC. This part of the Wigner-Ville spectrum has the higher amplitude and is related to the rapid increase of the cylinder pressure after the auto ignition. The other component is characterized by narrower frequency content (at 7 kHz). It corresponds to resonance due to the excitation of the cylinder cavity. A careful study of the angular-frequency map of the vibration signal demonstrates the relevance of the low frequency band width of the knock signal. However, this components shows some differences due to the propagation path that corresponds to a convolution of this signal with engine block transfer function. The low frequency component is strongly distorted but contains the key information about the source of interest. Energy around 3 and 4 kHz can be linked to common rail injection signature on accelerometric signature it was shown in the studies made in [10], [11]. This phenomena is not a relevant part of the signal.

In order to select the frequency band-width of interest, we combine the time frequency analysis with the coherence function analysis. The value of the coherence function between the knock signal $k$ and the cylinder pressure $p_{cyl}$ gives frequencies where the information of combustion can be extracted from the vibration trace. The coherence function $\rho(f)$ can be estimated by [12] :

$$\rho(f) = \frac{\left| \sum_{i=1}^{n} p_{cyl,i}(f) k_i^*(f) \right|^2}{\sum_{i=1}^{n} |p_{cyl,i}(f)|^2 \sum_{i=1}^{n} |k_i(f)|^2} \qquad (1)$$

where $p_{cyl}(f)$ and $k(f)$ are respectively the Fourier transforms of the cylinder pressure and the knock signals. $\rho(f) \in [0-1]$ and higher is $\rho$, higher is the correlation between knock and cylinder pressure. Figure 3 displays the coherence according to the frequency for the HCCI engine.

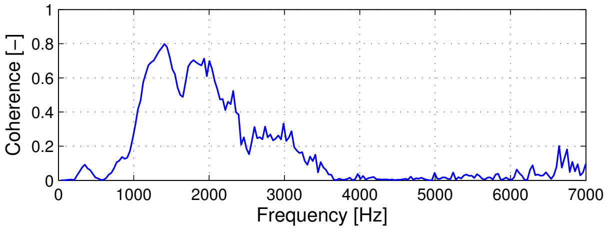

Fig. 3. Coherence between in-cylinder pressure and accelerometric signal

With all these off-line considerations, we define a band-pass filter to separate the combustion from the other noise sources on the vibration signal. The cut-off frequencies are 1000 Hz and 2500 Hz due to high coherence values ($\rho > 0.3$). The center frequency is chosen as the higher intensity peak on the spectrum analysis (Fig. 2). Here, the selected frequency is 2 kHz. it is an important point to set our extraction algorithm parameters (see II-B).

### *B. Combustion state observer design*

On a fixed operating point, the accelerometer signal is periodic with respect to the angular domain. Moreover, this signal is directly linked to the combustion. Indeed, it is filtered version of the combustion vibration. The filtering is due to the propagation of the signal in the mechanical structure. The signal we are interesting in is this "source" signal. To estimate this signal, we consider that the mechanical structure is a finite dimensional linear filter where the input is the vibration we want to estimate, and the output is the accelerometer measurement.

Consider the linear system driven by an unknown periodic input signal $w(t)$

$$\left\{ \begin{array}{l} \dot{x} = Ax + A_0 w(t) \\ y = Cx \end{array} \right. \tag{2}$$

where $x(t) \in \mathbb{R}^n$ is the state and $A$, $A_0$, $C$ are matrices in $\mathcal{M}_{n,n}(\mathbb{R})$, $\mathcal{M}_{n,1}(\mathbb{R})$ and $\mathcal{M}_{1,n}(\mathbb{R})$ respectively. The goal of our study is the estimation of the $T_0$-periodic continuous input signal $w(t) \in \mathbb{R}$, with $1 = dim(w) = dim(y) \leq n = dim(x)$, through its Fourier decomposition over the $h$ harmonics

$$w(t) \triangleq \sum_{k \in \mathcal{I}_h} c_k e^{\imath k \omega_0 t} \text{ with } \omega_0 \triangleq \frac{2\pi}{T_0} \tag{3}$$

where $\mathcal{I}_h = \{-h, \ldots, h\}$ is the set of $h$ modes. The Fourier decomposition is constant, i.e. $\dot{c}_k = 0 \; \forall k \in \mathcal{I}_h$.

Corresponding to state-space model, we define a time-varying Luenberger type observer

$$\left\{ \begin{array}{rcl} \dot{\hat{x}} & = & A\hat{x} + A_0 (\sum_{k \in \mathcal{I}_h} \hat{c}_k e^{\imath k \omega_0 t}) - L(C\hat{x} - y) \\ \dot{\hat{c}}_k & = & -e^{-\imath k \omega_0 t} L_k (C\hat{x} - y), \qquad \forall k \in \mathcal{I}_h, k \geq 0 \end{array} \right. \tag{4}$$

The design of $L$ and $\{L_k\}_{k \in \mathcal{I}_h}$ are defined, along with convergence proof, in [13].

Thus, we have a real time estimation of the source signal and its Fourier decomposition. Based on these information, in the next section, we estimate combustion parameters.

### *C. From the source signal to combustion parameters estimation*

We defined the energy released by the combustion as the instantaneous energy of the estimated source vibration. This correspond to the squared sum of the $k$ Fourier coefficients of the harmonic decomposition (3) :

$$E(\theta) \triangleq \sum_{k \in \mathcal{I}_h} \|\hat{c}\|_k^2 \tag{5}$$

Only the harmonics included in the bandwidth of interest are kept. The combustion energy estimated from the vibration signal is more sensitive than the one computed from the cylinder pressure signal. In order to get robust feedback variable, the estimated combustion variables are not directly computed from the raw energy signal $E(\theta)$. The energy provided by the combustion observer is fitted with a Gaussian function $f(\theta)$ that respects the shape of the most relevant part of this signal :

$$f(\theta) = a_1 \exp\left( -\frac{(\theta - a_2)^2}{a_3^2} \right) \tag{6}$$

The parameters, $a_i$, are identified online by using nonlinear recursive fitting method. The initial parameters for learning are updated for each engine cycle and chosen to be in the angular range where the combustion may occur. This is usually set a couple of degree after the start of injection taking into account of the auto-ignition delay and the propagation time of the shock wave. The two combustion variables of interest are :

- The Middle of Combustion ($MoC$) is linked to the maximum of the rate of heat released during the combustion stroke. It corresponds to the angle where the instantaneous energy is maximum. The estimated occurrence of the middle of combustion is defined as the center of the Gaussian function : $MoC = a_2$.
- The Energy of Combustion ($EoC$) is related to the total energy released by the combustion. The $EoC$ is defined as $EoC = a_1 a_3$. This parameter is an image of the combustion intensity filtered through the engine block. This corresponds to the cylinder combustion noise.

## III. CLOSED LOOP COMBUSTION CONTROL

### *A. Online combustion state algorithm*

The on-line combustion estimation algorithm is embedded into a dedicated platform (see section III-C). The estimation algorithm works as follow :

**Step 1 : Crank angle windowing.** The knock signal is recorded between 20 CAD before TDC and 50 CAD after TDC. One data frame is recorded at each engine cycle for each cylinders. This acquisition window is wide enough to include the relevant part of the knock signal in any engine operating conditions. The most relevant part of the vibration

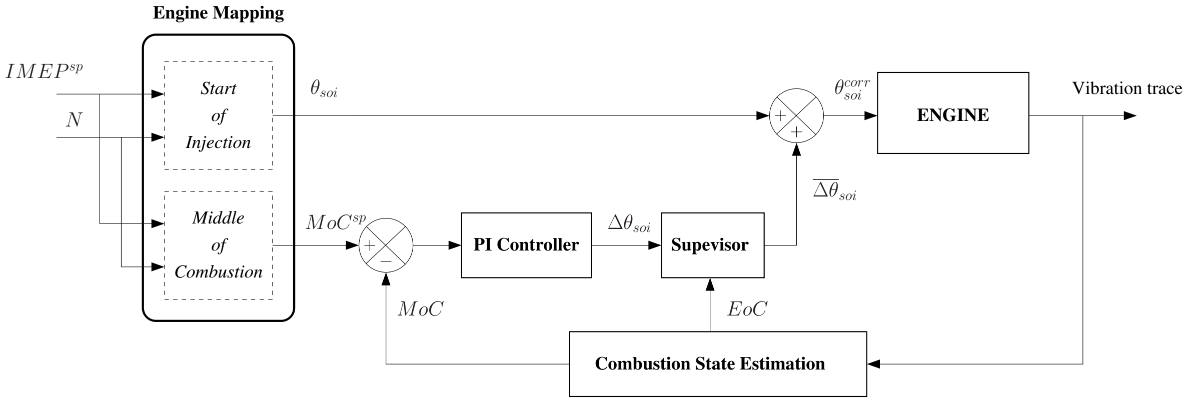

Fig. 4. Start of injection control scheme based on open loop and closed loop correction term.

signal is the one that contains the impact of the combustion stroke.

**Step 2 : Frequency windowing.** The vibration data series are then filtered with a fourth order Butterworth band-pass filter. The lower and upper cut-off frequencies are 1000 Hz and 2500 Hz respectively. The cut-off frequency values depend on the engine architecture and are chosen according to the time-frequency analysis (Fig. 2) and the coherence function (Fig. 3).

**Step 3 : Combustion energy estimation using the state observer.** The periodic observer has been set to estimates only one coefficient of the Fourier decomposition. We take a single frequency of 2000 Hz according to the spectrogram (Fig. 2) as suggested in section II-A.

**Step 4 : Combustion variables extraction from the Fourier energy.** The estimated energy returned by the observer is fitted with the Gaussian function (equation (6)) and the estimated combustion variables are calculated according to the definitions given in section II-C.

All these tasks are performed in real time in order to provide the $MoC$ and $EoC$ at the end of each engine cycle for a given cylinder. The algorithm has been tested for a set of operation points. For each engine setpoint, we study the variance between the middle of combustion ($MoC$) and the measured combustion timing, $\theta_{50}$, computed from the cylinder pressure with a standard ROHR analysis [14], [15]. The results obtained in steady state and transient tests show that the combustion timing can be used as a closed loop variable (Tab. I and Fig. 5).

### B. Start of injection controller

The main task is to design an algorithm that controls the combustion timing ($\theta_{50}$) using the proposed combustion indicators as feedback variables. The goal is to maintain the combustion timing as close as possible to its optimal value. The controller architecture is shown in figure 4. The principle of the proposed controller is to add an correction value $\Delta\theta_{soi}$ to the start of injection mapping $\theta_{soi}$. The SoI is the most influent control input to compensate the various perturbation affecting the HCCI combustion : temperature, air path errors or fuel properties. The SoI values are stored in maps as a function of the engine speed and engine load setpoint. Its values are optimized at the steady test bench. In parallel, knock sensor data series are recorded using the high-frequency rapid prototyping platform (see section III-C) so that for each calibrated point we can determine off-line the $MoC$ setpoint. For each engine operating point, the $MoC$ and $EoC$ setpoints are calculated off-line from the recorded knock data averaged over 100 engine cycles. Here, we implement a single-input single-output proportional-integral controller in order to control the $MoC$ :

$$\Delta\theta_{soi} = k_p(MoC^{sp} - MoC) + k_i \int (MoC^{sp} - MoC)dt \tag{7}$$

| IMEP | 1000 rpm | 1500 rpm |
|---|---|---|
| 3 bar | 1.0 | 1.3 |
| 4 bar | 0.9 | 1.6 |
| 5 bar | 0.5 | 1.1 |
| 6 bar | 1.2 | 1.1 |
| 7 bar | 1.3 | 1.8 |

TABLE I
VARIANCE ON PARTIAL ENGINE MAP $std(\text{MoC} - \theta_{50})$ COMPUTED FROM 100 ENGINE CYCLES.

The $MoC$ variance (Fig. 5) is not an issue since the combustion controller gains are small and the input signal $MoC$ is filtered. Here, the purpose is not to perform a cycle to cycle combustion control. The control law (7) cannot be applied directly to the system in all the cases for sake of stability. For very low values of the $EoC$, the energy released by the combustion is strongly reduced compare to other vibration sources. In these conditions, the $MoC$ does not represent the real combustion timing since the energy released by the combustion is not transmitted to the knock sensor. The $MoC$ cannot be used as a feedback variable for closed loop control of the SoI. Thus, a supervisor is required to saturate the SoI correction. The supervisor limits the amplitude of the SoI correction when the $EoC$ is too far from its setpoint such

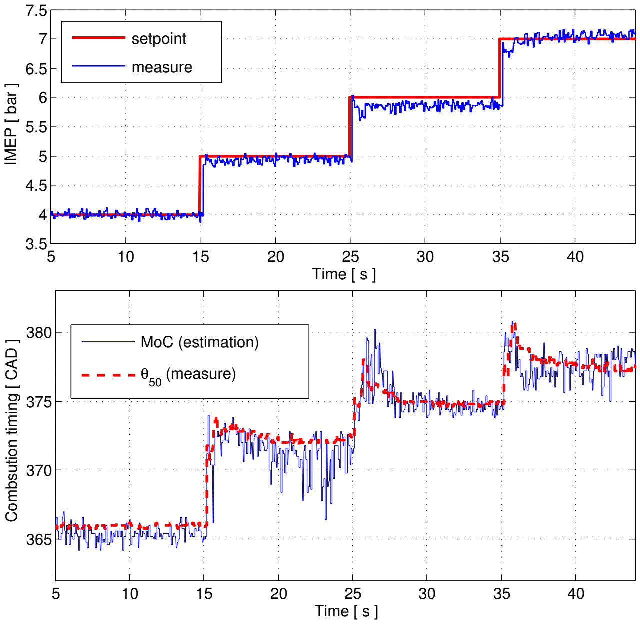

Fig. 5. Comparison between measured ($\theta_{50}$) and estimated ($MoC$) middle of combustion during engine torque transients at constant engine speed (1500 rpm). The top figure represents the IMEP setpoint and measure, the bottom figure displays the combustion timing.

that

$$\overline{\Delta\theta}_{soi} = \begin{cases} -\widehat{\Delta\theta}_{soi} & \text{if } EoC < EoC^{sp} - 3 \\ \Delta\theta_{soi} & \text{if } EoC^{sp} - 3 \leq EoC \leq EoC^{sp} + 4 \\ +\widehat{\Delta\theta}_{soi} & \text{if } EoC > EoC^{sp} + 4 \end{cases} \tag{8}$$

The amplitude of the saturation also depends on the stability of the combustion. Then, the value of the saturation depends on the $EoC$ standard deviation. Here, we use a centered Gaussian function :

$$\widehat{\Delta\theta}_{soi} = \Delta_{max} \, \exp\left(-\frac{{\sigma_{EoC}}^2}{2\alpha^2}\right) \tag{9}$$

where $\Delta_{max}$ is the maximum amplitude of the SoI correction, $\alpha$ is the width of the Gaussian function and $\sigma_{EoC}$ corresponds to the $EoC$ standard deviation computed over a sliding window. The maximum amplitude of the start of injection correction is limited by the standard deviation of the estimated engine noise $\sigma_{EoC}$. Late combustion timings usually decreases the combustion stability and higher cycle to cycle variability is observed. In that case, the $EoC$ standard deviation increases and the SoI correction is reduced by the supervisor. This simple rule avoids to apply a wrong correction to the SoI when the combustion timing estimation ($MoC$) is not robust. Finally, the corrected start of injection applied to the system is $\theta_{soi}^{corr} = \theta_{soi} + \overline{\Delta\theta}_{soi}$. Note that an anti-windup is added to the controller.

### C. Experimental setup

IFP has designed a dual mode combustion compression ignition engine able to reach low NOx emission while maintaining good performances. The dual mode engine application called NADI (Narrow Angle Direct Injection) applies HCCI at part load and switches to conventional Diesel combustion to reach full load requirements. The figure 6 displays the experimental engine architecture and actuators. The engine is a four cylinders direct injection Diesel engine working in both conventional combustion mode and HCCI mode. The engine is fitted with high pressure injection system. The variable geometry turbocharger (VGT), exhaust gas recirculation (EGR) valve and intake throttle are in charge to control the air and EGR flow rates. Cylinder pressure signals are recorded using piezoelectric sensors mounted on the glow plug location. The charge yielded by the piezoelectric sensor is converted into a proportional voltage signal by the charge amplifier. This signal is sampled at 0.1 CAD intervals. An encoder mounted on the engine shaft providing 7200 pulses per revolution synchronizes the pressure acquisitions. The engine has also been instrumented with a knock sensor. Developed engine control exploits combustion parameters computed with algorithms applied in real time on knock sensors signals. In order to achieve this, an innovative prototyping setup is implemented, composed of two prototyping platforms :

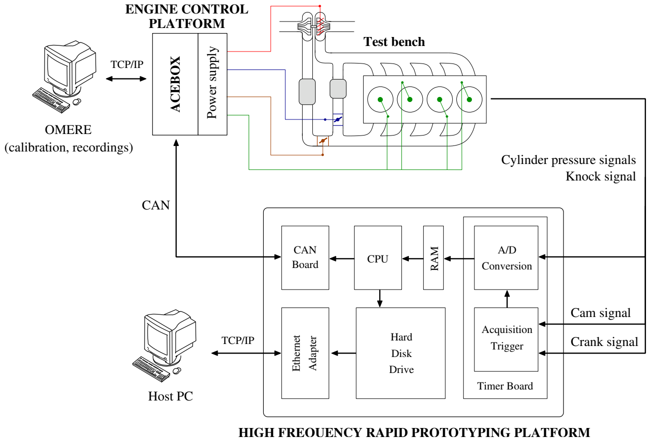

Fig. 6. Experimental setup.

1) *The engine control platform ACEBox (Automotive Control Embedded Based on xPCtarget)* : it consists in an industrial PC housing a dedicated timer board developed at IFP. This board and its specific acquisitions and commands I/O allow to update real-time control algorithms engine synchronously (each TDC or 6 CAD for example). This platform supports the engine air path and fuel path controllers.

2) *The high-frequency rapid prototyping platform* : It is a PC based real-time environment platform dedicated to high frequency acquisitions and rapid prototyping of engine synchronous processing algorithms. The frame based acquisition board developed at IFP provides ability to acquire data on 8 independent channels at specific angular windows, with a resolution of 0.1 CAD. More technical details about this platform are given in [16]. This platform is used for knock and

pressure signals acquisition, recording and real-time processing. In recording mode, the acquired frames are stored on hard disk drive for post-processing purpose. In real-time processing mode, the combustion analysis and combustion observer results are send to the ACEBox each TDC, cylinder to cylinder and cycle to cycle.

Communication and data exchange means between platforms is based on CAN protocol. Both platforms work with xPC Target real-time kernel provided by The MathWorks. By this prototyping means, control and signal processing algorithms are brought rapidly from executable specifications implemented in SIMULINK to test bench or vehicle applications.

### D. Control results

The combustion observer-controller structure (Fig.4) is tested over two cases where perturbations are added to the engine the system on the air path and on the fuel path.

*Case 1 : Perturbation on the intake manifold gas composition (Fig. 7).*

The aspirated gas composition is a key variable for the HCCI combustion. For the first test, the intake manifold gas composition is modified manually in order to create an offset with the calibrated value. As far as the intake manifold burnt gas rate decreases, the combustion timing is advanced. This effect is highlighted on the measured combustion timing of the cylinder 2. This latter has no correction on its SoI. The SoI correction is only applied to the cylinder 1. For this cylinder, the start of injection correction follows the evolution of the BGR perturbation such that the combustion timing is kept constant. The measured and estimated combustion timing are very close in spite of a small offset.

*Case 2 : Perturbation on the combustion timing (Fig. 8).*

The purpose of the second test is to valid the observer controller strategy in the case of a fuel perturbation. A modification of the start of combustion is simulated by adding a bias on the injection timing. This perturbation can be representative of a fuel properties modification. The test is performed on a steady engine setpoint. The offset applied to the start of injection has the effect to shift the combustion timing later than its optimal value. This effect is clearly detected on the $MoC$. The engine noise is reduced since the combustion timing is very late. The $EoC$ behavior is close to the one of the measured combustion noise in spite of a higher standard deviation. In order to compensate the combustion phasing offset the closed loop controller is enabled a time $= 15s$. From that time, the start of injection is corrected in order to recover the $MoC$ setpoint. In that case, the fuel system perturbation is perfectly rejected.

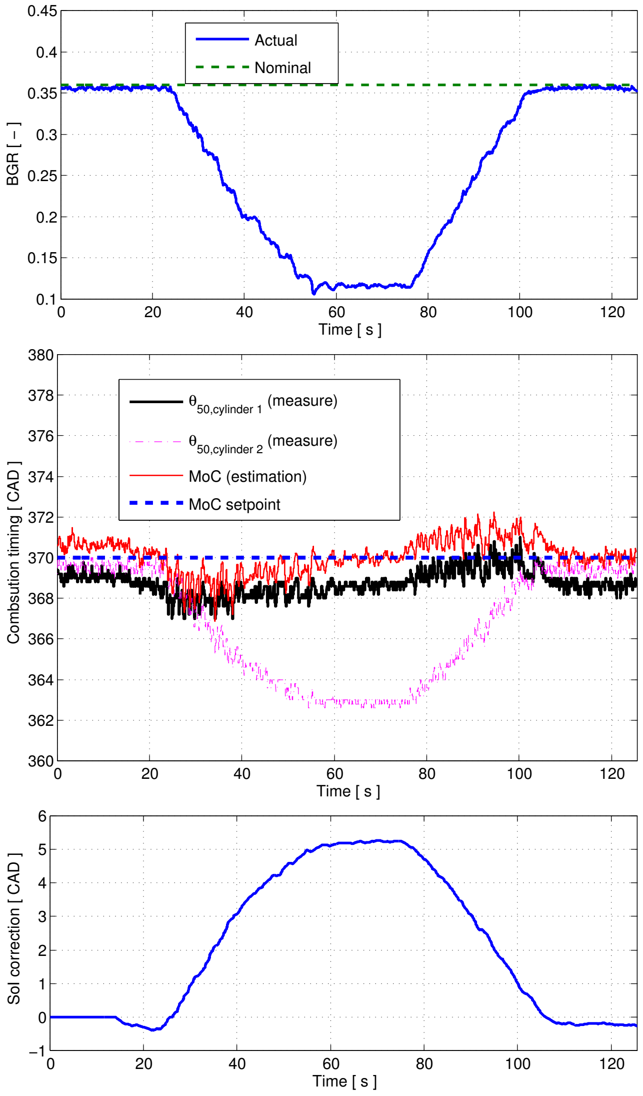

Fig. 7. Closed loop $MoC$ control on a steady setpoint (N=1000 rpm, IMEP = 4 bar). Case 1 : perturbation on the intake manifold gas composition. From top to bottom : BGR, combustion timing and SoI correction. The controller is enabled at t=15 s.

## IV. CONCLUSION & FUTURE WORK

This paper has presented a method to estimate and to control the combustion timing of a Diesel HCCI engine without cylinder pressure sensor. The combustion state estimator relies on a high frequency periodic observer that take as input the vibration signal recorded on the engine bloc with a knock sensor. It gives the estimated values of the noise and of the combustion timing. The estimated combustion timing has been used as a main feedback variable in a SISO controller. The estimated noise help to supervise and to saturate the start of injection correction. Two cases of combustion control are presented. The proposed method is efficient to compensate the effect of perturbations coming from the intake manifold gas composition or from the injection system. This low cost solution is a possible path to control the new combustion concepts such as HCCI or CAI. The next phases of this

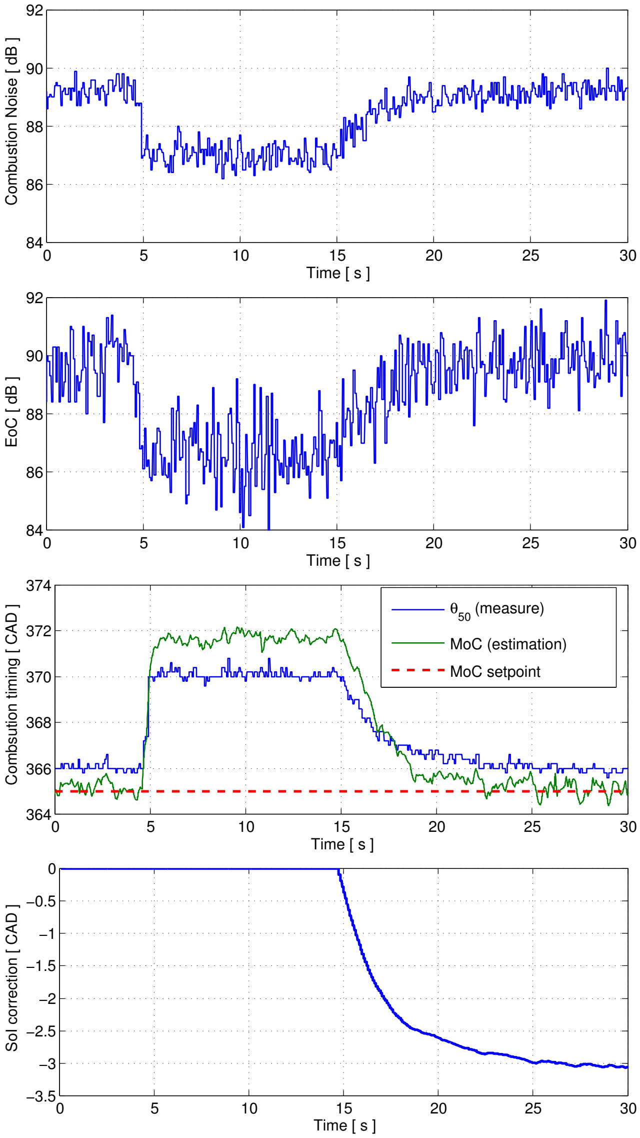

Fig. 8. Closed loop $MoC$ control on a steady setpoint (N=1000 rpm, IMEP = 4 bar). Case 2 : perturbation on the combustion timing $\theta_{50}$. An offset of 4 CAD is applied on the injection setting in order to modify the combustion timing (at t=5 s). From top to bottom : measured combustion noise, estimated combustion noise, combustion timing and SoI correction. The controller is enabled at t=15 s.

work are to investigate the potential of the method to control the four cylinders simultaneously with a single knock sensor during transient operating conditions. The actual control structure is probably not suitable for this purpose and we believe that a multivariable controller based on the two main combustion indicators ($MoC$ and $EoC$) is a task of interest for further study. Another demanding application concerns the cylinder to cylinder noise balancing for Diesel engine cold start.

## V. LIST OF ABBREVIATIONS

| | |
|---|---|
| BGR | Burn Gas Rate |
| CAD | Crank Angle Degree |
| EGR | Exhaust Gas Recirculation |
| $EoC$ | Energy of Combustion |
| HCCI | Homogeneous Charge Compression Ignition |
| IMEP | Indicated Mean Effective Pressure |
| IVC | Intake Valve Closing |
| LTC | Low Temperature Combustion |
| $\theta_x$ | $x$% of Mass Fuel Burnt |
| $\theta_{soi}$ | Injection timing |
| $MoC$ | Middle of Combustion |
| NOx | Nitrogen Oxides |
| ROHR | Rate Of Heat Release |
| SISO | Single Input Single Output |
| SoI | Start of Injection |
| TDC | Top Dead Center |

## REFERENCES

[1] J. B. Heywood, *Internal Combustion Engine Fundamentals*. McGraw-Hill, 1988.

[2] B. Walter and B. Gatellier, “Development of the high power NADI concept using dual-mode diesel combustion to achieve zero NOx and particulate emissions,” *SAE Paper*, no. 2002-01-1744, 2002.

[3] D. Schiefer, R. Maennel, and W. Nardoni, “Advantages of Diesel engine control using in-cylinder pressure information for closed loop control,” *SAE Paper*, no. 2003-01-0364, 2003.

[4] S. Leonhardt, N. Müller, and R. Isermann, “Methods for engine supervision and control based on cylinder pressure information,” *IEEE/ASME Transactions on Mechatronics*, vol. 4, no. 3, pp. 235–245, September 1999.

[5] Y. Gao and R. B. Randall, “Reconstruction of Diesel engine cylinder pressure using a time domain smoothing technique,” *Mechanical Systems and Signal Processing*, vol. 13, no. 5, pp. 709–722, 1999.

[6] J. Antoni, J. Danière, and F. Guillet, “Effective vibration analysis of IC engines using cyclostationarity. Part I : A methodology for condition monitoring,” *Journal of Sound and Vibration*, vol. 257, no. 5, pp. 815–837, 2002.

[7] J. Antoni, J. Danire, F. Guillet, and R. Randall, “Effective vibration analysis of IC engines using cyclostationarity. Part II : New results on the reconstruction of the cylinder pressure,” *Journal of Sound and Vibration*, vol. 257, no. 5, pp. 839–856, 2002.

[8] M. Urlaub and J. F. Böhme, “Reconstruction of pressure signals on structure-borne sound for knock investigation,” *SAE Paper*, no. 2004-01-0522, 2004.

[9] P. Flandrin, *Temps-Fréquence*, ser. Traité des Nouvelles Technologies, Traitement du Signal. Paris: Hermès, 1993.

[10] B. A. Badawi, M. Kholosy, A. A. Omer, and M. A. Shahin, “Identification of diesel engine events from acoustic signals using independent component analysis and time-frequency analysis,” in *SAE Paper*, no. 2007-01-2278, 2007.

[11] Q. Leclère, L. Polac, B. Laulagnet, and J. L. Guyader, “Vibro-acoustique des moteurs d’automobile,” *Techniques de l’ingénieur*, pp. BM 2773–1 to BM 2773–19, 2005.

[12] J. S. Bendat and A. G. Piersol, *Random data : Analysis and measurement procedures*. John Wiley and Sons Ltd, 2000.

[13] J. Chauvin, G. Corde, N. Petit, and P. Rouchon, “Periodic input estimation for linear periodic systems: Automotive engine applications,” *Automatica*, vol. 43, pp. 971–980, 2007.

[14] R. B. Krieger and G. L. Borman, “The computation of apparent heat release for internal combustion engines,” *Proceedings of Diesel Gaz Power, ASME*, no. 66-WA/DGP-4, 1966.

[15] J. A. Gatowski, E. N. Balles, K. M. Chun, F. E. Nelson, J. A. Ekchian, and J. B. Heywood, “Heat release analysis of engine pressure data,” in *SAE Paper*, no. 841359, 1984.

[16] O. Grondin, L. Duval, F. Guillemin, S. Ker, G. Corde, and C. Vigild, “Rapid-prototyping multi-sensors processing platform for real time engine control and diagnosis,” in *Proceedings of the Fifth IFAC Symposium on Advances in Automotive Control*, California, USA, August 2007.