IN-CYLINDER PRESSURE RECONSTRUCTION FOR MULTICYLINDER SI-ENGINE BY COMBINED PROCESSING OF ENGINE SPEED AND ONE CYLINDER PRESSURE

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ABSTRACT

The paper presents an approach to reconstruct cylinder-individual pressure of each combustion cycle by processing the instantaneous fluctuations of the engine speed and the in-cylinder pressure of one cylinder. A new pressure model with a feasible number of parameters is combined with a model-based torque estimation in order to reconstruct the pressure traces. The performance of the proposed algorithm is demonstrated with measurement data acquired from a vehicle with a four cylinder spark ignition (SI) engine.

1. INTRODUCTION

Advanced engine control systems require feedback about the combustion process in order to reduce exhaust gas emissions and improve fuel efficiency. The in-cylinder pressure provides adequate information about the engine performance. It is directly related to heat release, pollutant formation, gas exchange, and other important items [1]. At the current stage, however, when considering series production, measuring the pressure of each cylinder is still prohibitive by a number of conflicts such as production costs and engine dimensions. Thus, alternative methods to obtain cylinder pressure are of interest.

The reconstruction of the cylinder pressure has been investigated before. The proposed approaches are primarily based on the analysis of the instantaneous fluctuations in the engine angular velocity [2, 3]. Larsson et al. [4] investigated pressure estimation using torque sensors. Structure-borne sound, measured with accelerometers mounted on the surface of the engine block, can also be used in this context [5]. Though all these approaches are successful, most of them are limited with respect to accuracy and regions of engine speed. In addition they often require a large number of coefficients to be correctly adjusted. However, it is obviously difficult to extract the pressure trace or properties only using indirect signals.

Therefore we try to reconstruct the feedback information from the combustion chamber of each cylinder with desired accuracy by using a reduced number of pressure sensors, at best only one, and the already available engine speed signal. In [6, 7] the estimation of mean indicated pressure (IMEP) of each cylinder based on the proposed concept was presented. This contribution addresses the problem of reconstructing the cylinder-individual pressure traces by combined processing of the instantaneous fluctuations of the engine speed and the pressure of only one cylinder, the so-called key cylinder. In contrast to the methods mentioned before, the key cylinder approach has significant advantages. The available pressure signal provides new possibilities with respect to the signal modelling as well as its calibration.

The paper is organized as follows. In section 2, the approximation of the crankshaft torque is presented. A method for differentiation based on polynomial fitting is used in this context. The reconstruction of the in-cylinder pressure represents the problem of estimating multiple signals from a single source, the engine torque. Considering the superposition of the pressure traces and the relation between torque and pressure, direct inversion of the system is not possible. In order to solve this problem, a new parametric pressure model with a feasible number of parameters is introduced in section 3. In the subsequent section, an algorithm for pressure decomposition and offset compensation is described. Following that, torque estimation is combined with the proposed signal model to fit the parameters of the unknown pressure traces. In section 6 the algorithm is applied to measurement data before the paper concludes with section 7.

2. TORQUE ESTIMATION

Assuming a stiff crankshaft, the resulting torque of the crankshaft τ can be described according to [3] by

$$\tau = \tau_{ind} - \tau_{fric} - \tau_{load} = \theta(\varphi)\,\frac{d\dot{\varphi}}{d\varphi}\,\dot{\varphi} + \frac{1}{2}\,\frac{d\theta(\varphi)}{d\varphi}\,\dot{\varphi}^2, \qquad (1)$$

where τ_{ind} is the indicated torque, τ_{fric} the friction torque, τ_{load} the load torque, $\theta(\varphi)$ the crank angle dependent inertia, φ the crank angle, and $\dot{\varphi} = \frac{d\varphi}{dt}$. τ_{ind} is caused by the in-cylinder pressure p

$$\tau_{ind}(\varphi) = \sum_{l=0}^{z-1}(p_l(\varphi) - p_0)\,h(\varphi - l\,\frac{4\pi}{z}) \qquad (2)$$

with $h(\varphi) = A\,r\,\left(\sin\varphi + \frac{\lambda\,\sin\varphi\,\cos\varphi - \mu\,\cos\varphi}{\sqrt{1-\lambda^2\,\sin^2\varphi + 2\,\lambda\,\mu\,\sin\varphi - \mu^2}}\right)$, where p_0 is the ambient pressure, z the number of cylinders, l the cylinder index according to the firing order, A the piston area, r the crank radius, λ the connecting rod ratio, and μ the axial offset ratio.

We generally define the alternating component $[\tau]_\sim$ of τ as

$$[\tau(\varphi)]_\sim = \tau(\varphi) - \overline{\tau}. \qquad (3)$$

$\overline{\tau}$ presents the mean value of τ for the current engine cycle. Considering the steady state engine, the sum of the mean components of all acting torques is balanced.

$$\overline{\tau}_{ind} - \overline{\tau}_{fric} - \overline{\tau}_{load} = 0 \qquad (4)$$

The load torque during one cycle is nearly constant. Thus, the alternating component of the indicated torque can be expressed as

$$[\tau_{ind}(\varphi)]_\sim = \theta(\varphi)\,\frac{d\dot{\varphi}}{d\varphi}\,\dot{\varphi} + \frac{1}{2}\,\frac{d\theta(\varphi)}{d\varphi}\,\dot{\varphi}^2 + [\tau_{fric}(\varphi)]_\sim. \quad (5)$$

Methods for the compensation of deterministic disturbances, such as friction and incremental errors of the toothed wheel, as well as an approach for approximating $\overline{\tau}_{ind}$, have been presented in [6, 7]. Hence, only the angular velocity $\dot{\varphi}$ and its differential remain to be estimated in order to obtain the engine torque.

Engine speed is measured indirectly using a flywheel gear with 60 teeth. It provides trigger pulses for sampling the clock signal. Consequently the angle equidistant term $t(\varphi_i)$ is measured with angular resolution of 6°. The time resolution depends on the clock frequency. Considering that

$$\dot{\varphi} = \frac{1}{\frac{dt}{d\varphi}}, \quad (6)$$

we will estimate $\frac{dt}{d\varphi}$ first and then calculate $\dot{\varphi}$ according to (6).

The differentiation is implemented performing a linear least-squares fit of a polynomial of degree N, within a moving data window with an odd number of samples F. The method provides an accurate estimation using a minimal number of data samples. This is an important criterion with respect to the decoupling of single combustions, especially in case of misfire. In the following, the equivalent FIR-presentation of the procedure will be derived, in order to obtain the frequency response and use it to determine adequate values for N and F.

The polynomial is defined as a function of window indices x. Note that x does not depend on the window position and always has the same values: $-L, ..., 0, ...L$, with $L = \frac{F-1}{2}$. Considering the derivative at the sample n, we define the vectors
$\underline{t}_n = (t_{n-L}, t_{n-L+1}, ..., t_{n+L-1}, t_{n+L})'$, $\underline{a} = (a_0, ..., a_N)'$,
$\underline{w} = (w_{n-L}, w_{n-L+1}, ..., w_{n+L-1}, w_{n+L})'$, and

$$\mathbf{X} = \begin{pmatrix} 1 & (-L)^1 & \cdots & (-L)^{N-1} & (-L)^N \\ 1 & (-L+1)^1 & \cdots & (-L+1)^{N-1} & (-L+1)^N \\ \vdots & \vdots & \cdots & \vdots & \vdots \\ 1 & (L-1)^1 & \cdots & (L-1)^{N-1} & (L-1)^N \\ 1 & (L)^1 & \cdots & (L)^{N-1} & (L)^N \end{pmatrix}$$

so that

$$\underline{t}_n = \mathbf{X}\,\underline{a} + \underline{w} \quad (7)$$

where $\underline{w}$ is the vector of measurement noise. The least-squares fit of the polynomial coefficients yields:

$$\hat{\underline{a}} = \mathbf{D}\,\underline{t}_n \quad (8)$$

with $\mathbf{D} = (\mathbf{X}'\,\mathbf{X})^{-1}\,\mathbf{X}'$. Since the m-th derivative of a polynomial $y(\underline{a}, x)$, defined as

$$y(\underline{a}, x) = a_0 + a_1\,x + a_2\,x^2 + a_3\,x^3 + \,...\, + a_N\,x^N, \quad (9)$$

for $x = 0$ is

$$\frac{d^m\,y(\underline{a}, 0)}{dx^m} = m!\,a_m, \quad (10)$$

only the coefficient a_1 has to be computed for the first derivative. Hence, according to (8) only the second row of the matrix $\mathbf{D}$, in the following denoted as vector $\underline{D}_2$, is required. Applying the first row of $\mathbf{D}$ smoothes the signal t and is also known as the Savitzky-Golay method [8]. Thus, the differential at the sample n can be estimated as

$$\left.\frac{dt}{d\varphi}\right|_{\varphi=\varphi_n} = \underline{D}_2\,\underline{t}_n = \sum_{i=-L}^{L} D_{2,i+L+1}\,t_{n+i}. \quad (11)$$

Note that $\underline{\mathbf{D}}$ does not depend on the measured data and can consequently be precomputed for given N and F. Expression (11) amounts to the FIR-filtering of the data $\underline{t}_n$ with filter coefficients $\underline{D}_2$. The advantage of this representation is that the frequency response of the procedure can be calculated (see examples in Fig. 1). Additionally, using constant coefficients significantly reduces the computational costs for an on-line algorithm.

Choosing $N = 5$, $F = 11$ for engine speeds lower than 4000 rpm, and $N = 3$, $F = 13$ for higher speeds, $t(\varphi_i)$ is smoothed first and then differentiated with sufficient accuracy. The differentiation of $\dot{\varphi}$ in (5) is approximated with the same approach, thus the torque trace is completely estimated.

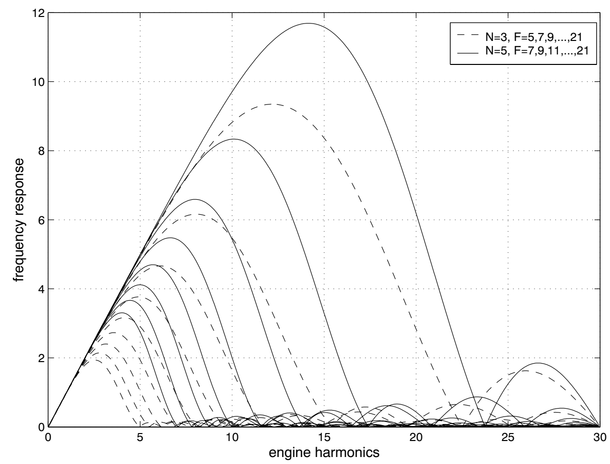

Fig. 1. Frequency response of the procedure including smoothing and differentiation with the polynomial approach. Engine harmonics present the frequency normalized to the engine speed.

3. PRESSURE SIGNAL MODEL

When analyzing a large number of measured pressure traces at the same operating point, some common characteristics can be observed. In the angle region before ignition, normal combustion as well as misfires show identical traces. The difference between the pressure of a misfire and normal combustion usually shows the same shape varying in the amplitude and angular position. These stochastic variations are caused by differing charge compositions and starting points of the combustion. The pressure rise for the misfire case is caused by the compression.

Considering this observation, the pressure trace is decomposed into two parts: one due to the compression, $g(\varphi)$ and another one due to the combustion, $f(\varphi)$, as shown in figure 2. Hence the following pressure signal model for a single combustion is proposed

$$p(\underline{\vartheta}, \varphi) = g(\varphi) + \alpha\,f(\varphi - \delta) \quad (12)$$

where $\underline{\vartheta} = (\alpha, \delta)$. The component $g(\varphi)$ represents the pressure trace that would occur without ignition. It depends primarily on the manifold pressure and the operating conditions. Since these are widely constant during one engine cycle, $g(\varphi)$ can be assumed to be identical for all cylinders in this period.

The shape of $f(\varphi)$ depends on the operating point. An additional advantage of the key cylinder approach in this context is that a suitable trace can be determined adaptively using the available pressure signal.

Thus the basic idea is to obtain the current estimation of $g(\varphi)$ and $f(\varphi)$, decomposing the key cylinder pressure of each combustion. The estimation of $\underline{\vartheta}$ will be carried out subsequently in the torque domain.

4. PRESSURE DECOMPOSITION AND OFFSET COMPENSATION

In the angular region before the start of combustion the compression curve $g(\varphi)$ is identical to the complete pressure $p(\varphi)$. Thus only the remainder of the trace has to be approximated. Neglecting the wall heat losses, the compression can be assumed as an adiabatic process:

$$g(\varphi)\, V(\varphi)^{\kappa} = C, \tag{13}$$

where κ is the adiabatic exponent, C a constant and $V(\varphi)$ the stroke volume calculated from engine dimensions. Using this assumption, the missing part of the curve can be extrapolated. Therefore the unknown parameters κ and $ln\, C$ have to be estimated. Additionally, we have to consider that the piezoelectric sensors, usually used for measuring the pressure, register only the fluctuations of the signal. Since the absolute value is not correct, the offset error of the measured pressure $p_M(\varphi)$ has to be compensated.

$$p(\varphi) = p_M(\varphi) + \Delta p \tag{14}$$

By means of (14) the model (13) for the region before ignition can be expressed as

$$(p_M(\varphi) + \Delta p)\, V(\varphi)^{\kappa} = p_M(\varphi)\,(1 + \Delta p/p_M(\varphi))\, V(\varphi)^{\kappa} = C \tag{15}$$

or equivalently

$$\ln p_M(\varphi) + \ln(1 + \Delta p/p_M(\varphi)) + \kappa\, \ln V(\varphi) = \ln C. \tag{16}$$

Assuming the cylinder pressure at -180 to $-175°$ to be equal to the available manifold pressure, the offset error is limited to ± 0.5 bar. Thus the data closely to the ignition fulfills $|\Delta p/p_M(\varphi)| \ll 1$. Using the Taylor-approximation,

$$\ln(1 + \Delta p/p_M(\varphi)) \approx \Delta p/p_M(\varphi), \tag{17}$$

we obtain

$$\ln C - \Delta p\, p_M(\varphi)^{-1} - \kappa\; \ln V(\varphi) \approx \ln p_M(\varphi). \tag{18}$$

The parameters κ, $\ln C$ and Δp can be estimated using the least squares method. Considering the measured pressure for n crank angles $\varphi_1 < \varphi_2 ... < \varphi_n < 0°$ before ignition, the model (18) yields

$$\ln C - \Delta p\, p_M(\varphi_i)^{-1} - \kappa\, \ln V(\varphi_i) \approx \ln p_M(\varphi_i), \tag{19}$$

with the measurement noise w_i. We define the vectors $\underline{\gamma} = (\ln C, \Delta p, \kappa)'$, $\underline{y} = (\ln p_M(\varphi_1), ..., \ln p_M(\varphi_n))'$, and

$$\mathbf{Z} = \begin{pmatrix} 1 & -p_M(\varphi_1)^{-1} & -\ln V(\varphi_1) \\ \vdots & \vdots & \vdots \\ 1 & -p_M(\varphi_n)^{-1} & -\ln V(\varphi_n) \end{pmatrix} \tag{20}$$

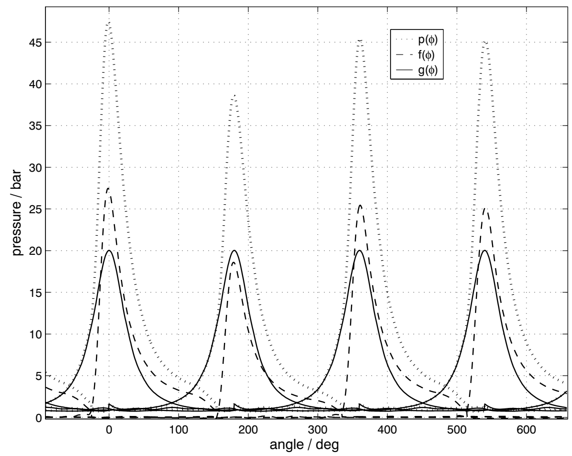

Fig. 2. Pressure components: complete pressure $p(\varphi)$, compression component $g(\varphi)$, combustion component$f(\varphi)$

so that the parameter vector $\underline{\gamma}$ can be obtained as follows:

$$\hat{\underline{\gamma}} = (\mathbf{Z}'\,\mathbf{Z})^{-1}\,\mathbf{Z}'\,\underline{y}. \tag{21}$$

Thus the pressure decomposition is completed applying the estimated parameters to (13, 14).

5. RECONSTRUCTION

In order to reconstruct the unknown pressures according to (12), the terms $g(\varphi)$ and $f(\varphi)$ for each engine cycle are obtained by decomposing the key cylinder pressure.

Consider now the task of estimating the unknown parameters $\underline{\vartheta}$. To simplify the notation, a single engine cycle is regarded. Assuming $g(\varphi)$ as identical for all cylinders, the torque part due to the compression τ_{comb} can be calculated as

$$\tau_{comb}(\varphi) = \tau_{ind}(\varphi) - \sum_{l=0}^{z-1} (g_{kc}(\varphi - l\,\frac{4\pi}{z}) - p_0)\, h(\varphi - l\,\frac{4\pi}{z}), \tag{22}$$

where $g_{kc}(\varphi)$ is the compression pressure of the key cylinder. During one cycle of the experimental four cylinder engine, three sets of parameters have to be determined. Since the components $f(\varphi)$ of each cylinder are widely decoupled (see Fig.2), the trace of $\tau_{comb}(\varphi)$ can be divided into four windows, each containing a signature of only one combustion. We define

$$p_{comb}(\underline{\vartheta}, \varphi) = \alpha\, f(\varphi - \delta). \tag{23}$$

Thus the parameter vectors can be estimated separately solving the following nonlinear least-squares problem for each window

$$\min_{\underline{\vartheta}} \sum_{i=k_1}^{k_2} |\tau_{comb}(\varphi_i) - p_{comb}(\underline{\vartheta}, \varphi_i)\, h(\varphi_i)|^2\,, \tag{24}$$

where k_1 and k_2 are the indices limiting the considered data window.

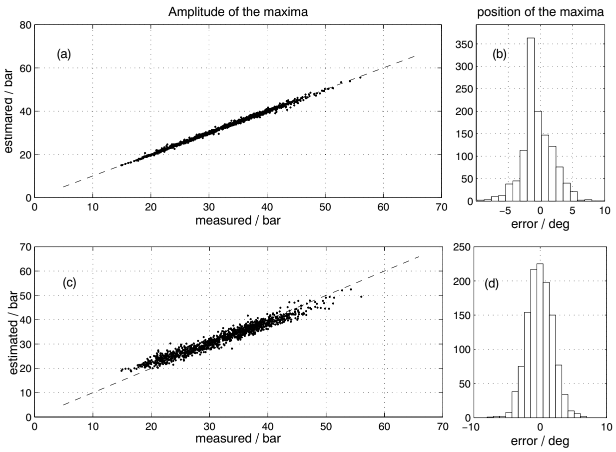

Fig. 3. Reconstruction results using the torque computed from measured pressure (a and b), and using engine speed (c and d), 4000 rpm and varying load, 1200 combustions.

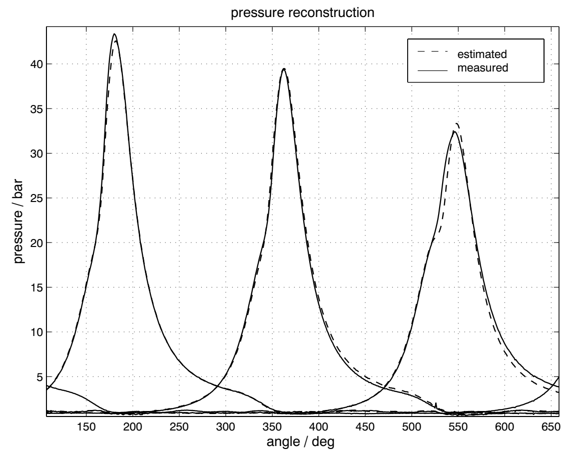

Fig. 4. Pressure reconstruction, an example of traces reconstructed using engine speed.

6. EXPERIMENTAL RESULTS

The algorithm was applied to measurements obtained from a vehicle with a 1.4 l four cylinder, direct injection SI-engine, at different operating points. All cylinders were equipped with a pressure sensor. One of them was assumed to be the key cylinder, and the remaining ones provided the reference. The pressure signal was sampled with a resolution of 1°. Engine speed was measured using a toothed gear with increments of 6°. The estimated torque trace was interpolated to the same resolution as the pressure. The performance of the proposed approach is demonstrated exemplarily for one measurement. Figure 3 presents the results for 1200 combustions at a relatively high engine speed of 4000 rpm and varying load.

Firstly, the pressure is reconstructed using the torque computed from the measured pressure (a,b). This corresponds to the reconstruction assuming that the torque trace is ideally estimated. Results show that the proposed signal model performs well regarding the considered system inversion.

Secondly, the reconstruction is performed using the torque estimated from engine velocity (c,d). The estimated positions of the maxima are comparable to the first case. With respect to the amplitude of maxima, the reconstruction is slightly worse due to the errors of torque estimation, but still very accurate. An example of the traces is shown in figure 4. The accuracy of the method is obviously limited by the torque estimation. For extremely high engine speeds, over 5000 rpm, the signal-to-noise ratio of velocity measurements becomes poor, and corrupts the estimation significantly.

7. CONCLUSIONS

A parametric pressure model, based on processing the pressure of one cylinder, is combined with torque estimation. Accurate feedback from the combustion chamber of each cylinder over a wide region of operating points is obtained using only one pressure sensor and the engine speed signal. Thus the proposed key cylinder approach provides new opportunities for cost-efficient cylinder-individual engine control and diagnostics. A promising possibility to further improve the accuracy is to implement a more complex engine model.

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