Robust cylinder pressure estimation in heavy-duty diesel engines

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Robust Cylinder Pressure Estimation in Heavy-Duty Diesel Engines

Serkan Kulah¹, Alexandru Forrai¹, Frank Rentmeester¹, Tijs Donkers² and Frank Willems¹,²

Abstract
The robustness of a new single cylinder pressure sensor concept is experimentally demonstrated on a six cylinder heavy-duty Diesel engine. Using a single cylinder pressure sensor and a crank angle sensor, this single cylinder pressure sensor concept estimates the in-cylinder pressure traces in the remaining cylinders by applying a real-time, flexible crankshaft model combined with an adaptation algorithm. The single cylinder pressure sensor concept is implemented on CPU/FPGA based hardware. For steady-state engine operating conditions, the added value of the adaptation algorithm is demonstrated for cases in which a fuel quantity change or start of injection change is applied in a single, non-instrumented cylinder. It is shown that for steady state and transient engine conditions, the cylinder pressure traces and corresponding combustion parameters, IMEP, PMAX, and CA50, can be estimated with 1.2 [bar], 6.0 [bar], and 1.1 [deg CA] inaccuracy, respectively.

Keywords
Cylinder pressure, virtual sensing, combustion parameters, combustion control, real-time

Introduction
Cylinder pressure-based combustion control is widely introduced for passenger cars. Benefits include: smaller variations in emission levels, robustness to fuel quality variation, reduced fuel consumption due to more accurate (multi-pulse) fuel injection, and minimized after treatment size. In addition, it enables the introduction of advanced, high-efficient combustion concepts. The application in truck engines is foreseen, but challenges related to system costs, durability, and impact on the cylinder head design need to be overcome.

In¹, a new single cylinder pressure sensor (CPS) concept for heavy-duty Diesel engines was introduced. Compared to other papers, e.g.,² - ⁶, the work in¹ focuses on heavy-duty Diesel powertrains, which are characterized by a relatively flexible crankshaft in contrast to the existing passenger car applications. Furthermore, in the work of⁷ direct torque measurements are used for combustion phasing estimation and control in a Diesel engine.

In this work, a model-based approach is followed, which originates from optimal control theory for signal tracking. Using measurements from a single cylinder pressure sensor and crank angle sensor, the individual in-cylinder pressure signals are reconstructed on-line with a real-time, 10 body crankshaft model and an adaptation mechanism. The potential of the single CPS is experimentally demonstrated on a modern six cylinder Euro-VI Diesel engine.

Contrary to our work in¹, detailed model identification results are presented and the concept is implemented and tested on the engine. Also, in order to illustrate the robustness of the concept, the fuel quantity and start of injection change in a single cylinder are changed. For these test cases, the estimated cylinder pressure traces and relevant combustion control parameters are presented and compared with measured values. These start of injection changes and the earlier mentioned model identification results are also new compared to¹.

Cylinder pressure estimation concept
Using a single in-cylinder pressure sensor and a crank angle encoder, the pressure in the five remaining cylinders are estimated. As the system is a multi-input single-output (MISO) system, the solution to this problem is not unique and the system model is not invertible. To resolve this issue, the proposed single CPS follows four steps to estimate the cylinder pressures:

- Initial estimate of all the induced torques on the crankshaft is obtained based on the in-cylinder pressure of the cylinder that is measured directly;
- Unknown external load on the crankshaft is estimated from crank angle measurement;
- Initial induced torque estimates are corrected based on the difference between measured and estimated crankshaft angular velocity using an optimal signal tracking algorithm;
- In-cylinder pressures are reconstructed from the induced torque using the crankshaft kinematics (i.e., an inverse piston model).

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Induced torque and external load estimation

The concept of cylinder pressure estimation is illustrated in Fig. 1. First, the available information from the measured cylinder pressure of one cylinder is used to determine an initial estimate of the induced torque denoted by $\tau^{est}$. This estimate is also applied to the other pistons. This induced torque consists of a part that is induced by reciprocating inertia of the piston assembly (determined from geometrical data), denoted by $\tau^{osc}$, and a part that is related to combustion, denoted by $\tau^{comb}$, i.e.,

$$\tau^{est} = \tau^{osc} + \tau^{comb}$$  \hspace{1cm} (1)

In a real tank, various drive-train configurations can be found and the external torque $\tau^{load}$ is not measured. Therefore, the external torque action on the engine at the flywheel is estimated from the dynamic crank angle signal. The crankshaft acceleration signal is used with the total rotating inertia information to calculate the net load torque on the shaft, which is later used to compute the external torque. Note that also the effect of auxiliaries is lumped in this torque estimation.

By applying the estimated torque $\tau^{est}$ and the estimated external torque $\tau^{load}$ to the flexible crankshaft model, a real-time estimate of the crank angle denoted by $\hat{\theta}$ can be found. However, this step requires a mathematical model of the flexible crankshaft, which has to be implemented in real-time. Based on its inputs $\tau^{est}$ and $\tau^{load}$, it estimates the $\theta$.

The crankshaft model

The model is assumed to be a multi-body system containing 10 bodies, each having a lumped inertia and are connected by a stiffness and a damping, the configuration is shown in Fig. 2. The flywheel is connected to the engine dynamometer (electrical drive) via a torsional damper and a drive shaft. Because the torsional damper does decouple the crank shaft from the drive regarding high frequent oscillations, it is assumed that it does not play a role in the dynamic behaviour.

The equations that relates position of all the bodies to each other can be given by the general differential equation:

$$J\ddot{\theta} + B\dot{\theta} + K\theta = \tau$$  \hspace{1cm} (2)

where $\theta = [\theta_1, \theta_2, ..., \theta_{10}]^T$ is the angular position vector containing the position of each body, $J$ is the inertia matrix, $B$ is the structural damping matrix, $K$ is the stiffness matrix, and $\tau$ is the torque vector acting on the bodies.

The equation above can be rewritten in a state-space form with the usual notations as:

$$\begin{align*}
\dot{x} &= Ax + Bu \\
y &= Cx
\end{align*}$$  \hspace{1cm} (3)

where the subscript $c$ stands for continuous-time state-space representation. The state vector is defined as: $x = [\theta_1, \theta_2, ..., \theta_{10}]^T$ and the input vector is $u = [\tau^{load}, \tau^{est}, ..., \tau_{T}^{est}, 0, 0, 0]^T$. Before the single CPS can be validated experimentally, the parameters of the flexible crankshaft model have to be identified. How the parameters of the crankshaft model are identified and validated experimentally is described in the experimental results section of this article.

On a heavy-duty Diesel engine, the crank angle encoder sensor can be placed at either the flywheel or the auxiliaries pulley of the crank shaft. Considering ease of access, sensor type and practical issues, for experimental studies, one encoder sensor located at the auxiliaries side of the engine is used with single CPS. Since the single CPS is using a single in-cylinder pressure sensor, the question arises: on which cylinder the pressure sensor shall be placed? The performed theoretical study shows that the controllability of the system is not influenced by the location of the pressure sensor. Taking into account practical reasons (easy sensor access), the pressure sensor is mounted on cylinder 1, which is the closest to the crank angle encoder.

Next, assuming that the model parameters are identified, having the estimated input vector $u$, a real-time estimation of the output $y$ (angular velocity of the $i^{th}$ body) can be found. More specifically, according to Fig. 2, the crank angle encoder is mounted on the torsion damper on the $10^{th}$ body. Therefore, we are interested to estimate the velocity of the $10^{th}$ body (denoted by $\hat{\theta}_{10}$) and compare it with the measured one by the encoder.

As was already mentioned, the initial induced torque estimates $\tau^{est}$ will be corrected by $\tau^{\Delta}$ based on the difference between measured and estimated crankshaft velocity using an optimal signal tracking algorithm. This is discussed in more detail in the next subsection.

<table>
<thead>
<tr>
<th>Engine Speed</th>
<th>External Torque</th>
<th>Adaptation Algorithm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Estimated $P_{10}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>Inverse Piston Model</td>
</tr>
</tbody>
</table>

Figure 1. Single cylinder pressure sensor - block diagram.

Virtual Cylinder Pressure Sensor

![Diagram of a virtual cylinder pressure sensor](image-url)
The adaptation algorithm

The single CPS concept proposed in this paper is using one pressure sensor to measure the in-cylinder pressure, the other in-cylinder pressures shall be estimated. Among the individual cylinders, the in-cylinder pressures can be different, i.e. due to fuel offsets or differences in start of injection in case of individual cylinder control. Therefore, the induced torque estimates, which are based on the measured in-cylinder pressure need to be corrected using an adaptation algorithm.

First, the continuous-time state-space representation is written in discrete-time form, at time moment \( t = kh, k \in \mathbb{N} \) and \( h > 0 \) the sampling time:

\[
\begin{align*}
    x(k+1) &= A_dx(k) + B_d\left(u(k) + u^\Delta(k)\right) \\
    y(k) &= C_d x(k)
\end{align*}
\]

where the subscript \( d \) stands for the discrete-time state-space representation. Furthermore, \( u(k) = [\tau_{\text{load}}(k), \tau_{\Delta}^2(k), ..., \tau_{\Delta}^6(k), 0, 0, 0]^T \) and \( u^\Delta(k) = \tau^\Delta(k) \) is the correction term, given by the adaptation algorithm.

The adaptation algorithm determines the required change \( \tau^\Delta(k) \) of the estimated combustion torque per cylinder, such that the estimated crank angle rotation matches the measured engine rotation signal. This is formulated as a signal tracking problem. More precisely, the estimated combustion torque is the solution of the following linear quadratic optimal control problem:

\[
\min \frac{1}{2} \sum_{k=0}^{N-1} \left[ y(k) - y^\text{meas}(k) \right]^2 + \frac{1}{\gamma} \left( \tau^\Delta(k) \right)^2
\]

for some \( \gamma > 0 \), where \( \hat{k} \) is the sample number.

In the equation above, \( y(k) \) and \( y^\text{meas}(k) \) are the estimated and measured angular velocity of the \( 10^{th} \) body. Since this position is sampled with 1 crank angle resolution and one engine cycle corresponds to 720 CAD (crank angle degree), \( N \) is chosen as: \( N = 720 \).

With the weighting factor \( \gamma \), the relative importance of a small tracking error compared to small torque corrections is tuned. From an initial experimental parameter study, it is found that the optimal weighting factor that gives a minimal estimation error in the signal tracking problem equals \( \gamma = 10^6 \). For more details and background on the single CPS, the interested reader is referred to [1].

Torque to pressure calculation

In the final step the cylinder pressures are converted to the corrected induced torques \( (\tau^\text{est} + \tau^\Delta) \) for the individual cylinders by applying an inverse piston model, i.e., a kinematic crank-slider model. However, as is briefly presented below, a singularity exists at the top dead center (TDC), which makes it difficult to calculate back the pressure from a given torque. It can be shown that the combustion torque \( \tau_i \) developed by \( i^{th} \) cylinder is:

\[
\tau_i = k_E \cdot p_i \cdot \sin(\theta_i + \phi_i)
\]

where the subscript \( i \) relates to the \( i^{th} \) cylinder, \( k_E \) is an engine dependent constant, \( p_i \) is the pressure, \( \theta_i \) is the crank angle and \( \phi_i \) is rod angle measured from top dead center.

From the equation above, we can observe that when \( \sin(\theta_i + \phi_i) \) is zero the torque becomes zero. If the torque is given (i.e. the torque is estimated) due to the singularity, the pressure cannot be calculated by solving (6), since the torque at TDC is zero but the pressure is different from zero. Furthermore, in real-time this singularity exist not only at the angle, where \( \sin(\theta_i + \phi_i) \) is zero, but also the computations become inaccurate when \( \sin(\theta_i + \phi_i) \) is close to zero.

In order to avoid the singularity, a solution is presented in [7], where a nonlinear mapping is defined between the measured torque and main combustion parameters. In this case, the pressure is not estimated directly but the main combustion parameters (which are pressure dependent) are estimated. Instead of using the technique proposed by, a different method is proposed in this article for estimating the \( i^{th} \) cylinder pressure. First, the torque \( \tau^\text{est} \) is estimated, then the torque is corrected by \( \tau^\Delta \) and, finally, the estimated pressure is calculated as:

\[
p_i^\text{est} = p_1 \left( 1 + \frac{\tau_i}{\tau_i^\text{max}} \right)
\]

where \( p_i^\text{est} \) is the estimated pressure for cylinders \( i = 2, ..., 6 \), \( p_1 \) is the measured pressure, and \( \tau_1^\text{max} \) is the maximum value of the torque, generated by pressure \( p_1 \). Note that in the equation above the pressure and the torque signal of cylinder
The experimental set-up

The work is performed on a state-of-the-art Euro-VI heavy-duty 13 liter Diesel engine. For the combustion analysis, a Kistler 6125 pressure transducer is installed in each cylinder and an AVL 365 crank angle encoder is mounted on the torsion damper, as illustrated in Fig. 2. Data is acquired with an AVL Indimodul, which samples signals with a resolution of 1 crank angle degree [deg].

The external load is applied by coupling the Diesel engine with an electric drive (engine dynamometer). In the test set-up, the standard engine ECU is bypassed, such that the fueling can be controlled for each cylinder individually.

The hardware - from Speedgoat - consists of an Intel Core i3 dual core CPU clocked at 3.3 GHz and a programmable Xilinx FPGA I/O board for code deployment. This combined with powerful Simulink Real-Time technology to automatically distribute performance optimized models to multiple cores allows application execution times to be significantly reduced, especially for complex models. For the software environment Matlab/Simulink is used, which allows rapid prototyping and automatic code generation and deployment. The cylinder pressure estimation algorithm runs on the CPU, having a sampling frequency of 10 [kHz]. Similar to\textsuperscript{9}, real-time heat release and IMEP calculations are implemented on the FPGA. A picture of the experimental set-up, showing the hardware setup with the Diesel engine placed in the test room, is shown in Fig. 3.

Experimental results

First, the identification and the validation of the crankshaft model is described. Next, experiments and results for validating the single CPS are presented and discussed.

Crankshaft model identification and validation

Based on measurements with 100% fueling in all cylinders, a parameter set is identified, which minimizes the error between the measured and predicted angular velocity in a least squares sense. During the identification, the model inputs are the measured load torque, induced torques, which are determined based on six measured cylinder pressure signals, and the angular velocity, which is derived from the measured angular position.

The model is identified and validated for different steady-state operating points, which are shown in Fig. 4. The operating points, defined by engine speed and engine torque, are presented on a relative (or often called per-unit) scale. The operating points are normalized according to the maximum engine speed and maximum engine torque.

As a remark, for practical implementation a single model (i.e., single parameter set), which describes the full operating range is preferred. This minimizes the required floorspace for implementation and avoids the need of model switching. However, local models have been investigated too, but resulted only in slight improvements in estimation accuracy.

Therefore, in this work a single model is used that is identified for test point 1 in Fig. 4. The prediction accuracy of this model is validated for the other 7 test points. As shown in in Fig. 5, the crank angle-resolved angular velocity is predicted with good accuracy (less than 3% absolute relative error) over the studied operating range. Note that the dynamics can differ significantly for the various operating points, but these are well-captured in amplitude and phase.

Single cylinder pressure sensing validation

In this section, the experimental validation of the single cylinder pressure sensor concept is presented. For validation of its robustness, experiments with fuel quantity and start of injection (SOI) offsets have been performed. During these experiments, the pressure sensor information of cylinder 1 is used in the single CPS. Cylinder pressure measurements and corresponding combustion parameters in the other cylinders are used for reference in the validation process.

Fuel offset: In the nominal case, the fueling is equal in all cylinders and corresponds to the specified operating point with 100% fueling. For cylinder 6, a fuel offset is applied to create in-cylinder conditions that are different from the instrumented cylinder 1. This mimics possible injector fouling or aging.

For a significant fuel reduction of 30%, experimental results of the single cylinder pressure sensor are presented.
in Fig. 6, which illustrates that the estimated peak cylinder pressure in cylinder 6 is reduced. Note that if the adaptation is not active, the cylinder pressure levels remain similar to pressure in cylinder 1.

The results - averaged over 100 cycles - show that the estimated cylinder pressure for the case with fuel reduction is in good agreement with the measured one, even for the most distant cylinder 6 - less than 5% absolute relative error.

**SOI offset:** To further demonstrate robustness of the single CPS, variations in the start of injection are applied to cylinder 6. These possible variations are related to production tolerances or fueling inaccuracies.

In Fig. 7, the measured and estimated cylinder pressure are shown for a SOI change of 3 [deg CA]. Also in this case, the results - averaged over 100 cycles - show that the estimated pressure in cylinder 6 shifts accordingly. Good agreement with the measurements is found in this studied case - less than 5% absolute relative error.

**Estimation of combustion parameters:** The single CPS is planned to be applied in closed-loop combustion control strategies. These strategies typically focus on controlling combustion parameters that are related to in-cylinder pressure, such as Indicated Mean Effective Pressure (IMEP), crank angle at 50% heat release (CA50) and peak cylinder pressure (PMAX). Therefore, apart from the impact on pressure estimation, the impact on estimation accuracy of the relevant combustion parameters is also studied for fueling and SOI offset in cylinder 6. The results are shown in Fig. 8 and Fig. 9, respectively.
The top figure shows the estimated pressure in cylinder 6. In the lower figures the momentary gross IMEP, CA50, and PMAX are plotted as a function of time. Similar to Fig. 8, these parameters are calculated in real-time and are available at the end of the combustion cycle (indicated by circle). In all these cases the adaptation algorithm presented in the previous section was active.

As illustrated by these figures, the specific offset is applied around t=11.7 [s]. In all figures the results corresponding to actual cylinder pressure measurement are shown for reference.

For the 30% fuel reduction case (see Fig. 8), it is observed that the single CPS is able to estimate the main combustion parameters with relatively good accuracy (less than 5% absolute relative error). Despite the introduction of the offset, the gross IMEP and PMAX are closely and immediately tracked.

This is mainly due to the adaptation mechanism; if the adaptation mechanism is not active, the studied combustion parameters remain unchanged. These errors and potential limitations on control performance are suppressed by the application of the adaptation mechanism. Similar trends are found for a SOI change of -3 [deg CA], see Fig. 9.

However, CA50 is more difficult to track. This will be analyzed in more detail in the sensitivity study section below.
In order to further test the robustness of the single CPS, a more detailed sensitivity analysis is performed. More precisely, around the nominal operating conditions for test point 4, SOI and fueling sweeps are applied to cylinder 6. The estimated combustion parameters’ (CA50 and IMEP) changes are compared with the values corresponding to measured pressure traces. As a remark, the combustion parameters (CA50 and IMEP) are averaged over 100 engine cycles.

**Fuel quantity sweep:** Fig. 10 shows the CA50 and IMEP variations corresponding to estimated and measured cylinder pressures for ±30% fueling variation in cylinder 6. This figure clearly illustrates that the adaptation mechanism is capable to capture the impact of both positive and negative fuel variations on the studied combustion parameters. This is in line with the observations in Fig. 8. The maximum CA50 and IMEP estimation error is 0.6 [deg] and 0.4 [bar], respectively.

**SOI sweep:** In a similar manner, injection timing are performed within ±3 [deg CA]. The results are shown in Fig. 11. Both estimated and measured CA50 and IMEP values show linear trends, down to injection timing advance of -2 [deg CA]. The IMEP variation is well captured - maximum error of 0.5 [bar]. For increasing SOI, the estimated CA50 is retarding as in measurements. However, the adaptation mechanism shows different sensitivity than the measurements. This results in a maximum estimation error of 2 [deg CA].

**Transient test cycle analysis**

The single CPS performance during transient engine conditions are presented in this section. For this experiment, a fast transient engine test cycle is created connecting all the operating points presented in Fig. 4. The prepared test cycle consists of 2000 engine cycles, with an average of 250 engine cycles for each test point and 50 engine cycles for fast transient speed and/or torque ramps. For the transient experiment, the pressure sensor information from cylinder 1 is used in the single CPS. During the transient test an external disturbance of −300 [ms] fuelling bias is applied on the nominal injection duration of the cylinder 6 using the fuelling control system. This disturbance is created in order to obtain different in-cylinder conditions for cylinder 6 compared to the cylinder 1.

The single CPS performance in the presence of an external disturbance is evaluated by looking at the combustion parameters which are determined for each individual cycle. Additionally, to visualize the single CPS performance, three data sets are plotted on the Figures 12, 13 and 14. Estimation Off (Est Off) is the case in which the cylinder 1 in-cylinder pressure (reference cylinder) signal is used to calculate combustion parameters. Estimation On (Est On) is the case where the single CPS output with adaptation for cylinder 6 pressure is used to calculate combustion parameters and measured signal (Meas.) is the case where the combustion parameters are calculated by using direct in-cylinder measurements of cylinder 6. On the figures, the test points (TP) are highlighted with the corresponding numbers. TPs 1, 2, 4, 5 and 7 considers relatively steady state regions whereas TPs 3, 6 and 8 include fast torque/speed ramps. The vertical lines show the boundaries considered for dividing TPs for analysis.

In Fig. 12 the CA50 results for Cylinder 6 are plotted. In the figure, each test point is highlighted and both steady state and transient sections are included in the data analysis. In Table 1 the obtained mean CA50 error values for both estimation on and off cases are presented. Single CPS reduced the CA50 error - between the reference cylinder
Table 1. Error analysis of combustion parameters during transient test cycle for the estimation on and off conditions

<table>
<thead>
<tr>
<th>Parameter</th>
<th>TP1</th>
<th>TP2</th>
<th>TP3</th>
<th>TP4</th>
<th>TP5</th>
<th>TP6</th>
<th>TP7</th>
<th>TP8</th>
<th>Avg</th>
</tr>
</thead>
<tbody>
<tr>
<td>CA50 mean error in [degCA]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Estimation Off</td>
<td>2.5</td>
<td>2.4</td>
<td>2.0</td>
<td>2.3</td>
<td>3.2</td>
<td>2.2</td>
<td>1.8</td>
<td>2.0</td>
<td>2.3</td>
</tr>
<tr>
<td>Estimation On</td>
<td>1.0</td>
<td>0.9</td>
<td>0.5</td>
<td>1.8</td>
<td>2.3</td>
<td>1.1</td>
<td>0.9</td>
<td>0.6</td>
<td>1.1</td>
</tr>
<tr>
<td>IMEP mean error in [bar]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Estimation Off</td>
<td>2.2</td>
<td>2.3</td>
<td>2.4</td>
<td>2.6</td>
<td>2.7</td>
<td>2.6</td>
<td>2.2</td>
<td>2.8</td>
<td>2.5</td>
</tr>
<tr>
<td>Estimation On</td>
<td>0.9</td>
<td>1.0</td>
<td>2.0</td>
<td>1.0</td>
<td>0.8</td>
<td>0.9</td>
<td>1.7</td>
<td>1.1</td>
<td>1.2</td>
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<tr>
<td>PMAX mean error in [bar]</td>
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<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Estimation Off</td>
<td>3.4</td>
<td>3.5</td>
<td>2.2</td>
<td>6.7</td>
<td>6.9</td>
<td>2.8</td>
<td>1.3</td>
<td>3.3</td>
<td>3.8</td>
</tr>
<tr>
<td>Estimation On</td>
<td>4.2</td>
<td>3.4</td>
<td>9.0</td>
<td>5.0</td>
<td>6.9</td>
<td>3.6</td>
<td>5.0</td>
<td>10.0</td>
<td>6.0</td>
</tr>
</tbody>
</table>

Figure 12. CA50 calculated with Initial estimated and measured pressure for cylinder 6 for the transient test cycle

Figure 13. IMEP calculated with Initial estimated and measured pressure for cylinder 6 for the transient test cycle

Figure 14. PMAX calculated with Initial estimated and measured pressure for cylinder 6 for the transient test cycle

and the disturbed cylinder - from 2.3 [deg CA] to 1.1 [deg CA]. For the low load points (TP4 and TP5) estimation accuracy of CA50 is relatively lower compared to high load points (TP1, TP2 and TP3). At low load points the CA50 of cylinder 6 gets closer to the 0 [deg CA] of cylinder 6 (TDC), where observability of pressure signal from the crank shaft oscillations becomes lower and results in performance reduction in single CPS for CA50 determination.

Similarly in Fig. 13 the IMEP results for cylinder 6 are presented. As presented in Table 1, single CPS reduced the IMEP error from 2.5 [bar] to 1.2 [bar] which is caused by the external disturbance. Single CPS performance for IMEP correction is obtained lower for the low speed high load points (TP3 and TP7) compared to other test points. These operation points will require further investigations.

In Fig. 14 cylinder 6 peak cylinder pressure (PMAX) results are presented for the aforementioned cases. For PMAX detection single CPS increased the initial error with respect to the reference cylinder. The average PMAX error (see Table 1) is increased from 3.8 [bar] to 6 [bar] during the transient test cycle. Especially high error values are obtained for the test points (TP3 and TP8) where the fast transients (speed/torque ramps) are included in the TP windows. Nonetheless, the maximum PMAX error is still obtained within the 5% error band relative to the measured PMAX values (TP3).
Conclusions

A new single cylinder pressure sensor concept is successfully implemented and its robustness is demonstrated on a six cylinder heavy-duty Diesel engine. The potential of this concept is evaluated at different steady state and transient engine operating conditions by varying fuel quantity and start of injection in a single non-instrumented cylinder.

Based on the studied cases it is concluded that:

- the estimated angular velocity based on the identified real-time 10-body crankshaft model shows good agreement with the measured angular velocity: the maximum absolute error is 3% over wide engine operating range. Efficient implementation is feasible since a single model parameter set is sufficient;
- the added value of the pressure adaptation algorithm is shown for fuel quantity and start of injection offset. The main combustion parameters are predicted with relatively good accuracy: considering both steady state and transient operating conditions with external disturbances (SOI and FV variations), IMEP, PMAX and CA50 can be approximated with ±1.1 [bar], ±6.0 [bar] and ±1.1 [deg CA] inaccuracy.

CA50 and PMAX estimation with high accuracy still remains a challenge in case of start of injection change and transient engine conditions. It has been identified that the lower observability of pressure signal at and around TDC via crank shaft oscillations (due to engine kinematics) is the main reason for the less accurate estimation of these combustion parameters. Further improvement of the single CPS for these parameters over the entire engine operating envelope including speed and load step changes and demonstrating the concept with multi-pulse fueling strategies are subject of future work. Ultimate goal is integration of this concept into a virtual NOx sensor, as discussed in 9.

References