

Virtual Cylinder Pressure Sensor for Transient Operation in Heavy-Duty Engines

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ABSTRACT

Cylinder pressure-based combustion control is widely introduced for passenger cars. Benefits include enhanced emission 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 need to be overcome related to durability, increased system costs, and impact on the cylinder head.

In this paper, a new single cylinder pressure sensor concept for heavy-duty Diesel engines is presented. Compared to previous studies, this work focuses on heavy-duty Diesel powertrains, which are characterized by a relatively flexible crank shaft in contrast to the existing passenger car applications. By combining model predictions of the external load and of the crank angle variations, individual cylinder pressure variations are determined following a signal tracking approach.

The potential of this virtual cylinder pressure sensor concept is demonstrated for a six cylinder Diesel engine that operates under transient load conditions. A comparison with experimental data shows that relevant combustion control parameters, such as, CA50 and IMEP can be approximated with ± 1.5 [°CA] and ± 0.5 [bar] accuracy, respectively, during transients.

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INTRODUCTION

With the availability of production-type cylinder pressure sensors, engine control entered a new era for passenger cars. Using this sensor information, closed-loop combustion control has become feasible. Namely, this in-cylinder pressure information can be used for controlling combustion phasing and indicated power output for each cylinder on a cycle-to-cycle basis. This is a crucial enabler for advanced combustion concepts, such as Homogeneous Charge Compression Ignition (HCCI), Pre-mixed Charge Compression Ignition (PCCI), and Reactivity Controlled Compression Ignition (RCCI). These concepts rely on controlled auto ignition and often lack the direct control of combustion phasing and rate of heat release. Besides improved combustion stability of advanced concepts, closed-loop combustion control is also beneficial for conventional Diesel combustion because of the following reasons [1]:

- Enhancing robustness as it minimizes the effect of fuel variations, production tolerances, ageing on emissions;

- Reducing fuel consumption by increased (multi-pulse) fuelling accuracy;
- Reducing calibration time: with closed-loop combustion control engine calibration can be automated.

As a result of these benefits, closed-loop combustion control has gained interest over the past few years. For light-duty applications, several engines with closed-loop combustion control systems are currently available in the market [2], [3]. Closed-loop combustion control is based on real-time, in-cylinder pressure information, typically using direct measurements from pressure sensors. For heavy duty (HD) applications, the required durability for these sensors is very demanding and installation of a pressure sensor in each cylinder is expensive. Furthermore, considerable changes are needed in the cylinder head design to properly position these sensors.

Instead of measuring the in-cylinder pressure directly, several studies propose to use high frequent (0.1 - 0.5 [°CA sampling]) crankshaft measurements with crankshaft models to estimate IMEP and CA50

[4]. However, these high-frequent crank-angle measurements are not available in commercial HD engines. In order to mitigate this issue, [5] considers a rigid crankshaft dynamic model to reconstruct the dynamic torque profiles, which can be used to estimate the in-cylinder pressures. The crankshaft speed measurements are used with a dynamic crankshaft model to reconstruct the total torque on the crankshaft. Subsequently, the torque traces are used to calculate the in-cylinder pressures via crank-slider kinematics. In [6], a combustion model with rigid crankshaft assumption is considered to calculate combustion phases, which are used to calculate in-cylinder pressures. Unfortunately, torsional vibrations in HD engines are significant due to the large dimensions of the engine. As such, the crankshaft of a HD engine cannot be modelled as a rigid body.

Instead of using only crank angle measurements, [7] proposes to use a torque sensor positioned on the crankshaft to obtain the net torque acting on the crankshaft. Using a MISO flexible crankshaft model and model inversion techniques, in the frequency domain, in-cylinder pressures are obtained from the measured torque signal. However, this approach is not suitable, since the exact inversion of the model is not possible when crank-angle position is measured instead of torque. Moreover, in order to estimate the induced torque on one cylinder, all other torques (friction, mass, load) are assumed to be known in [7]. This is possible under the assumption that the torque induced by the oscillating pistons is negligible. For HD engines, the torque induced by oscillating pistons does have a significant effect and, therefore, cannot be neglected. In [8], heat release models (using Wiebe functions) are applied in the calculation of the in-cylinder pressures. The calculated pressures are used to estimate the crankshaft rotation by utilizing a rigid-body crankshaft model, while a sliding-mode observer corrects the estimated pressures. Tuning the heat-release parameters and the time-varying observer gains becomes cumbersome in case this method is to be extended towards a flexible crankshaft model.

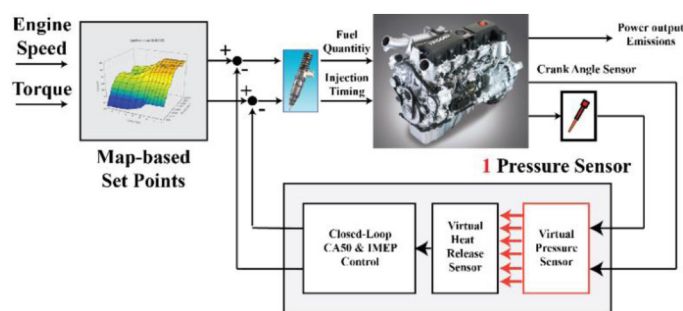


Figure 1. Closed-loop combustion control concept.

In order to overcome the aforementioned technical issues, this paper presents a novel model-based method that solves the in-cylinder pressure estimation problem for HD Diesel engines by using optimal control theory. Based on a flexible crankshaft model and an adaptation mechanism, the individual in-cylinder pressure signals are estimated using a single cylinder pressure sensor. The estimated in-cylinder pressures can be used for closed-loop combustion control,

as also indicated in Figure 1. The potential of this single-cylinder pressure sensor concept is demonstrated for a six-cylinder Diesel engine by comparing the output of the virtual sensor with measurements. The main focus of this paper is on transient conditions in conventional Diesel combustion mode, but initial results for Diesel-CNG RCCI combustion are also presented.

EXPERIMENTAL SETUP

A modern, six cylinder HD Diesel engine has been used to demonstrate the potential of the proposed virtual cylinder pressure sensor concept. The engine under consideration is equipped with exhaust-gas recirculation (EGR), variable turbine geometry (VTG) and common-rail fuel injection equipment.

Measurement System

For the development of the proposed concept, high-frequent measurements are taken on an engine test bench. An overview of the measurement setup is presented in Figure 2. As shown in the figure, the measurement setup uses a crank angle encoder and six in-cylinder pressure sensors, which are measured with an AVL Indimodul and are sampled with a resolution of 0.1 [°CA]. The crank angle variations are measured using an AVL 365 crank angle encoder. All cylinders are equipped with Kistler 6056A type piezoelectric pressure transducers and will be used later to validate pressure estimation results. The position encoder was mounted to the torsional damper housing at the front of the crankshaft, while the load was applied to the flywheel at the rear of the crankshaft via a coupling. External load is applied to the engine using an external electric drive connected to the flywheel.

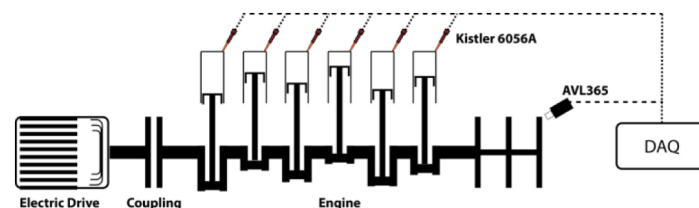


Figure 2. Schematic overview of the measurement setup overview with electric drive, engine and sensors.

FLEXIBLE CRANKSHAFT MODEL

A flexible crankshaft model is used in the virtual pressure sensor to describe the effect of cylinder pressures on the crank angle and angular velocity. The lumped-mass representation of the flexible crankshaft model with corresponding inertias is shown in Figure 3. The actual crankshaft drives the oil-pump and camshaft by means of a gear transmission at the flywheel side. On the auxiliaries' side, a torsional damper is located, which rotates accessories by means of a belt drive. A water pump, air condition compressor, and alternator are some of the auxiliaries connected to the belt drive. The external load is applied onto the crankshaft from the flywheel side and dynamic effects of the clutch are not considered.

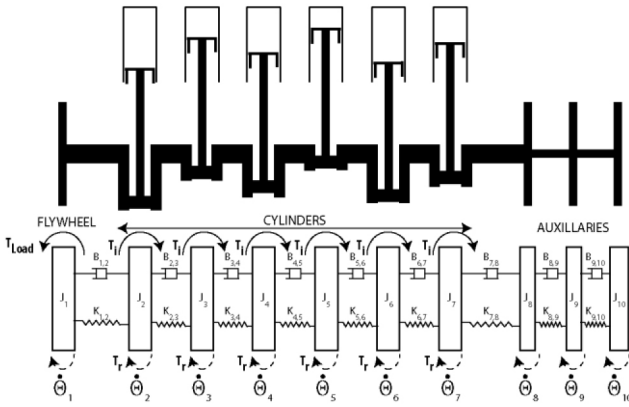


Figure 3. The lumped-mass model of the crankshaft.

In the model, the torque applied to the pistons is decomposed into two components: a torque τ^{comb} induced by the combustion and a torque τ^{osc} induced by the reciprocating inertia of the piston assembly. In a HD engine, the cylinder assemblies have a considerable amount of mass, which oscillates when the engine operates. Therefore, the oscillations in the cylinder assembly results in a torque effect on the crankshaft. This torque induced by the reciprocating inertia is considerable and has to be included in the model. All lumped inertias are interconnected by using spring and damper elements. The following assumptions have been made to generate the model:

- Structural damping of the crankshaft is assumed to be linear and assumed between lumped mass elements in order to obtain a linear model with coefficients $b_{1,2}, b_{2,3}, \dots, b_{9,10}$;
- Stiffness of the crankshaft is assumed to be linear and to be between each lumped mass element with coefficients $k_{1,2}, k_{2,3}, \dots, k_{9,10}$;
- Crankshaft inertias $J_1, J_2, \dots, J_{10}$ corresponding to the different elements on the shaft; flywheel, individual cylinders, auxiliaries are assumed to be concentrated at discrete locations;
- Friction-induced torque effects, ageing effects and small variation on engine parameters are neglected.

The engine model utilized in this research is obtained from the equations of motion of the system. By defining the state variables:

$$x = [\theta_1, \dot{\theta}_1, \dots, \theta_{10}, \dot{\theta}_{10}]^T, \quad (1)$$

a linear state-space model is obtained of the form:

$$\begin{cases} \dot{x} = Ax + B u(x, p_i) \\ y = Cx \end{cases} \quad (2)$$

with:

$$u := [\tau^{load}, \tau_2^{comb}(\theta_2, p_2) + \tau_2^{osc}(\theta_2, \dot{\theta}_2), \dots, \tau_7^{comb}(\theta_7, p_7) + \tau_7^{osc}(\theta_7, \dot{\theta}_7)]^T \quad (3)$$

in which τ^{load} is the external load and $y = \theta_1 \in \mathbb{R}$ is the measured output. More details on the applied model can be found in [9].

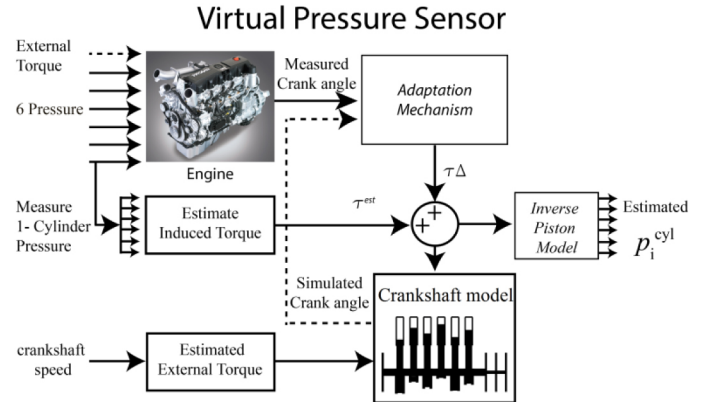


Figure 4. Schematic of the virtual cylinder pressure sensor concept.

VIRTUAL CYLINDER PRESSURE SENSOR

Figure 4 illustrates the proposed virtual cylinder pressure sensor (CPS) concept. Using a single in-cylinder pressure sensor and a crank angle encoder, the pressure in the five remaining cylinders is 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 virtual cylinder pressure sensor (repeatedly) follows four steps to estimate the cylinder pressures:

1. In-cylinder pressure of one cylinder is measured directly to obtain an initial estimate of all the induced torques on the crankshaft;
2. External load on the crankshaft is unknown in a vehicle and has to be estimated from crank angle measurement;
3. Initial induced torque estimates are corrected based on the difference between measured and estimated crank angle position using an optimal signal tracking algorithm;
4. In-cylinder pressures are reconstructed from the induced torque using the crank-slider kinematics, i.e., an inverse piston model.

The various steps are discussed in more detail below.

Step 1 - Initial Estimate of Induced Torque

In the first step, the measured pressure trace is used for an initial estimate τ^{est} of the induced torques on the crankshaft, using the (static) relation between pressure and induced torque (which only depends on crank angle position). This step is based on three underlying assumptions:

- All six cylinders in the engine are geometrically identical and also the intake manifold pressures are (approximately) the same;
- Crank angle and angular velocity of each cylinder deviate only slightly from the calculated relative positions using the slider-crank relations;
- The same fueling strategy is applied to all cylinders in terms of quantity and timing.

Based on the first assumption, the pressure traces for all cylinders are approximately shifted versions of each other (owing to the firing order of the cylinders) until the fuel is injected into the cylinder. After a pressure trace of cylinder 1 is measured, an estimate of the induced torque of all cylinders τ_i^{est} can be calculated using piston kinematic relations. More precisely, τ_i^{est} is a combination of a shifted version of τ_2^{comb} (initial estimate of τ_i^{comb}) and τ_i^{osc} .

Step 2 - Estimation of External Load Torque

The virtual cylinder pressure sensor is used to estimate the in-cylinder pressure traces during transient engine conditions. Transient conditions are characterized by engine speed and external load torque variations. For given cylinder pressures and external load torque input, the engine model simulates the crankshaft rotation. In order to estimate the cylinder pressure from the crankshaft rotation, an estimate of the load torque has to be available.

To estimate the load torque τ^{load} , the engine is assumed to be decoupled from the rest of the powertrain at the flywheel side. This allows the influence of the drivetrain on the crank shaft to be modelled as an external torque effect acting on the flywheel. Moreover, the auxiliary torque τ^{aux} is assumed to be lumped with the load torque. The total net torque acting on the crankshaft is given by:

$$\tau^{net} = \sum_{i=2}^7 \tau_i^{est} + \tau^{load}, \quad (4)$$

where i refers to the elements in Figure 3. Using crank-angle encoder information, the torque τ^{net} can be estimated using a rigid body approximation of the crankshaft model, which allows τ^{load} to be computed.

Step 3 - Improving the Estimated Torque using Crank Angle Measurements

In the third step, the initial estimates of the induced torques are improved in real time using the described crankshaft model and an adaptation mechanism. The calculated induced torques are applied to the flexible crankshaft model, which is described in the previous section.

Using the crankshaft model and the external torque estimation, the initial estimate of the induced torques are used to simulate the crankshaft rotation. As a result of small torque differences between the actual and the assumed induced torques, an error occurs between the actual and the calculated crankshaft angle. At this stage, the problem is casted into a signal tracking problem. In particular, the virtual CPS is the solution to a linear quadratic optimal control problem. The objective is to find model inputs, i.e., corrections τ^Δ on the initially estimated torque τ_i^{est} of the individual cylinders, such that the measured crank angle matches the predictions made by the model.

For the signal tracking algorithm, we take a finite-time perspective with a finite number of samples. This can be achieved by considering one complete combustion cycle at a time, and by handling the problem in the discrete-time domain. The initial condition of the

model (2) is taken by assuming $\theta_i = 0$ and $\dot{\theta}_i = \dot{\theta}_1$ for each body of the model, since the angular velocity $\dot{\theta}_1$ can be measured directly. The model used in the signal tracker algorithm is an exact discretization of the linear state-space model (2) and is given by:

$$\begin{cases} x[k+1] = A_d x[k] + B_d (\tau^{nom}[k] + \Gamma[k] \tau^\Delta[k]) \\ y[k] = C_d x[k] \end{cases} \quad (5)$$

where $\tau^{nom} = [\tau^{load}, \tau_2^{est}, \dots, \tau_7^{est}]^T$. Now, given a measurement of the crank angle, denoted by y^{meas} , we determine τ^Δ , by solving:

$$\min_{\tau^\Delta} \sum_{k=0}^{N-1} \frac{1}{2} (y[k] - y^{meas}[k])^2 + \frac{1}{2\gamma} (\tau^\Delta[k])^2 \quad (6)$$

for some (large) $\gamma > 0$ subject to the discrete-time system equation (5). Here, we use crank angle measurement with 1 [°CA] resolution, so $N=720$. The optimization problem (6) is a standard linear quadratic tracking problem, which can be tackled by solving a Ricatti difference equation and two recursive equations.

Step-4: In-Cylinder Pressure Reconstruction

The final step in the virtual cylinder pressure sensing process is to reconstruct the in-cylinder pressure traces using the assumed τ^{est} and the calculated torque adjustment τ^Δ per cylinder. The calculated torques are used to reconstruct the pressure traces for all cylinders using the inverse slider-crank equations.

CONCEPT VALIDATION

For validation and demonstration of the virtual CPS concept, a simulation study is performed using real measurements. This is done by comparing the measured and the predicted in-cylinder pressures and the crankshaft angular velocity signals. Experiments are performed at different combinations of engine speed, external load and combustion types, as shown in Table 1.

Method

For the steady-state tests, data of 200 combustion cycles is collected at constant engine speed and load torque of the engine test bench. For the estimation, one random cycle out of 200 cycles is used instead of cycle averaged data. This method is selected to create more realistic estimation conditions. In the results presented below, the pressure in cylinder 1 is measured directly and the other cylinder pressures are estimated. In order to assess the potential of this approach, the predicted in-cylinder pressures as well as the predicted angular velocity are compared with measurements. The pressure traces are plotted corresponding to the cylinder firing order: 1-5-3-6-2-4. The TDC position of the measured cylinder is taken as a reference. The predicted crankshaft angular velocity is the simulated response $\dot{\theta}_1$ (at flywheel side) in the crankshaft model. This response is compared with the measured signal from the crank angle encoder. For the studied cycle, the corresponding Peak Pressure Rise (PPR), Indicated Mean Effective Pressure (IMEP) and CA50 are also examined.