

Simultaneous Control of Combustion Timing and Ignition Delay in Multi-Cylinder Partially Premixed Combustion

Gabriel Ingesson, Lianhao Yin, Rolf Johansson, and Per Tunestal
Lund University

ABSTRACT

In low-temperature combustion concepts such as partially premixed combustion, the ignition delay should be large enough in order to ensure sufficient fuel and air mixing before the start of combustion. It is also necessary that the combustion timing is sufficiently well phased for high thermal efficiency.

Since the ignition delay and combustion timing are intimately coupled, the decoupling of these two quantities gives rise to an interesting multiple input, multiple output control problem where the control of the air system and the fuel injection system have to be combined. In a multi-cylinder engine this problem becomes underdetermined or uncontrollable with more outputs than inputs.

This article investigates model-based cycle-to-cycle cylinder-individual closed-loop control of the ignition delay and the combustion phasing in a multi-cylinder heavy-duty DI engine running on a gasoline fuel mixture. The controller design of choice was model predictive control (MPC) which is a suitable design for multiple input/output systems with actuator constraints. Ignition delay and combustion phasing were extracted from cooled in-cylinder pressure sensors and controlled by manipulating injection timings, the gas mixture temperature and exhaust-gas recirculation (EGR) ratio using a dual EGR-path system and a fast thermal-management (FTM) system.

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INTRODUCTION

In direct-injection (DI) engines, low temperature combustion is an often used name for combustion concepts where the ignition delay is prolonged in order to enhance fuel and air premixing. Since an increased ignition delay gives the fuel more time to penetrate the gas mixture before the combustion starts, locally leaner mixtures are provided during combustion which both reduces the formation of particulate matter and NO_x due to lower combustion temperatures [1].

The ignition delay could be prolonged by manipulating the cylinder-gas state when the fuel is injected into the cylinder. This can be done by varying the injection timing in relation to the piston position, since the ignition delay depends on the cylinder gas state at the time of injection [2]. The ignition delay could also be prolonged by diluting the inducted gas with cooled exhaust gas recirculation (EGR) which decreases the oxygen concentration and increases the heat capacity of the inducted cylinder gases, this decreases the in-cylinder peak temperatures and slows down the early chemical reactions and thus increases the ignition delay [3]. EGR dilution also effectively reduces NO_x emissions due to the lower temperatures and oxygen levels. The temperature and density of the mixture at injection could also be altered by cooling the charge prior to the intake manifold. This has previously been used to control the combustion phasing in HCCI [4].

During the past decade a concept called partially premixed combustion (PPC) has been under investigation at Lund University. By the simultaneous use of high EGR ratios, early injection timings and gasoline-like fuels, it has been found that a prolonged ignition delay can be obtained together with low NO_x and soot emission levels together with high efficiency [5].

In order to achieve high thermal efficiency, the combustion timing has to be phased so that the highest possible pressure during the expansion stroke is obtained without excessive heat transfer losses and high pressures during the compression stroke. The combustion phasing depends both on the ignition delay and the injection timing unless the injection timing is very advanced. In order to maintain efficient PPC, the ignition delay and the combustion phasing which are tightly coupled have to be controlled simultaneously. This work presents a centralized controller design that solves this problem by executing coordinated control action in the fuel-injection system and the gas-flow system.

Previous research on the subject was carried out by Lewander et al. [6] who controlled the ignition delay in PPC while keeping the combustion phasing within an acceptable range using the fuel injection timings and a model predictive control (MPC) design. Karlsson et al. [7] controlled the ignition delay and combustion phasing using a linear-quadratic regulator (LQR) design in order to

minimize emissions using the back-pressure valve and injection timings as system inputs. Other previous work on optimal control of the fuel and air path system in conventional diesel engines was done by Alfieri et al [8] who used injection timings and EGR actuation to follow optimal trajectories for λ and NO_x for low fuel consumption and emission levels. Atkinson et al [9] designed a neural network model calibrated to experimental data in order to set multiple fuel injection timings and the EGR flow in order to obtain optimal transient performance.

This article investigates model predictive control (MPC) for modelbased cycle-to-cycle cylinder-individual closed-loop control of the ignition delay and combustion phasing in a multi-cylinder heavy-duty DI engine running on a gasoline-fuel mixture. The control of the output variables is done by altering the injection timings, the valve positions of a dual-path EGR system and the valve positions of an inlet-manifold fast thermal-management (FTM) system. The controller is evaluated during load and speed changes and during setpoint changes at stationary conditions. The primary aim of this work was to find a controller design that could handle this multiple input/output control problem given by this particular engine setup, due to hard constraints on valve positions, MPC was considered to be the best available design option.

EXPERIMENTAL SETUP

Engine Setup

The experimental engine was a Scania D13 6-cylinder heavy-duty diesel engine with specifications displayed in Table 1. The engine speed was controlled with a 355 kW AC motor that worked both as an engine motor and brake. The engine was boosted with a fixed geometry turbine compressor combination and the inlet-manifold gas temperature was controlled by varying the valve positions θ_{Hot} and θ_{Cool} in the FTM system prior to the inlet manifold. The intake oxygen concentration was controlled by varying valve positions in the short and long-route EGR paths θ_{SR} , θ_{LR} . The gas-system layout is displayed in Figure 1.

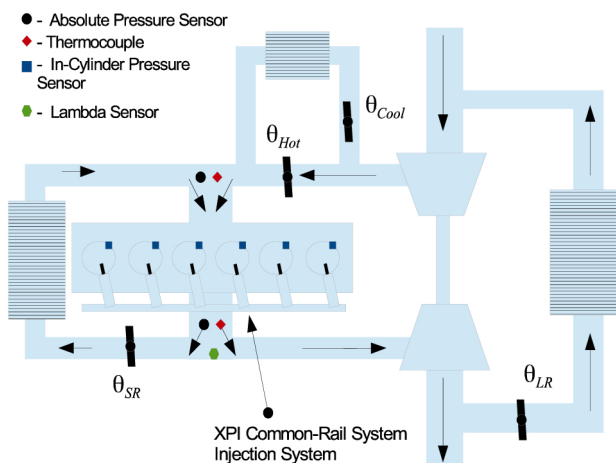


Figure 1. The Scania D13 gas-system setup, the valve angles θ_{Cool} and θ_{Hot} were used to control the inlet manifold gas temperature while the valve angles θ_{SR} and θ_{LR} were used to dilute the intake air with EGR.

Table 1. Engine Specifications.

Total Displaced Volume	12.74 dm ³
Number of Cylinders	6
Stroke	160 mm
Bore	130 mm
Connecting Rod	255 mm
Compression Ratio	18:1
Number of Valves	4

Engine Control and Measurement System

The entire engine control system was programmed with LabVIEW™ which is a graphical programming environment developed by National Instruments. The real-time system consisted of a NI PXIe-8135 embedded controller (2.3 GHz quad-core processor), NI PXI-7854/7854 R (Multifunction reconfigurable I/O (RIO) with Virtex 5-LX110/LX30 FPGA). The FPGA was considered as a configurable hardware that worked as a flexible AO / DIO, it was also used for AD acquisition, triggered by the crank-angle encoder. The ADC sampled the analog signals with a 16-bit resolution. The user interface was run on a separate host PC with a Windows 7 operating system which communicated with the real-time system over TCP/IP.

The in-cylinder pressure was measured with water-cooled Kistler 7061B pressure sensors and was sampled with the Leine-Linde crank-angle encoder pulse every 0.2 crank-angle degree. Inlet manifold and exhaust pressures were measured with Keller PAA-23S absolute pressure sensors. Inlet manifold and exhaust temperatures were measured with K-type thermocouples. No emissions were measured during the experiments presented in this paper.

The fuel injection system was a production Xtra high Pressure Injection (XPI) common-rail injection system for the Scania D13 engines. The common-rail pressure, injection timings, durations and valve positions were controlled from the real-time system using Drivven drivers.

The fuel used in the experiments was a mix of 80 volume % gasoline and 20 volume % N-heptane.

The heat-release analysis and controller computations were run in MathScript RT Module nodes inside a timed-loop block on the real-time target. The timed loop was triggered by the FPGA once the in-cylinder pressure was sampled. All computations were done using floating point arithmetic.

SIGNAL PROCESSING

In order to extract combustion information from the measured in-cylinder pressure p , model-based signal processing, also called heatrelease analysis, was performed every cycle for each cylinder. In order to account for the pressure sensor offset, the pressure sensor signal was pegged w.r.t. to the measured inlet manifold pressure at inlet valve closing (IVC), high-frequency noise in p was attenuated using a zero-phase digital filter.

The heat release rate $dQ_c/d\theta$ was estimated by treating the combustion chamber content as a single-zone open system. Applying the first law of thermodynamics and the ideal gas law yields [12]:

$$\frac{dQ_c}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta}, \quad (1)$$

where V is the cylinder volume and θ the crank angle. The ratio of specific heats $\gamma = c_p/c_v$ depends on the cylinder gas composition and temperature, in this work γ was estimated assuming that p and V satisfy the adiabatic relation (2) during the compression stroke with a constant C

$$pV^\gamma = C \quad (2)$$

γ and C were estimated from p and V with (2) using linear regression methods [10]. The convective heat-transfer rate from the combustion chamber gases to the combustion chamber wall $dQ_{ht}/d\theta$ was modeled with the relation

$$\frac{dQ_{ht}}{d\theta} = \frac{h_c A}{6N_{Speed}} (T - T_w), \quad (3)$$

where A is the combustion chamber surface area, T is the mean gas temperature, T_w the mean wall temperature and N_{Speed} is the engine speed in rpm. The convection coefficient h_c was in this work modeled as a Woschni expression [11]

$$h_c = C_0 B^{0.2} p^{0.8} T^{-0.55} \omega^{0.8}, \quad (4)$$

where $C_0 = 1.3 \cdot 10^{-2}$, B is the cylinder bore, ω is the characteristic in-cylinder gas velocity, modeled as

$$\omega = C_1 S_p + C_2 \frac{V T_{IVC}}{p_{IVC} V_{IVC}} (p - p_m) \quad (5)$$

where the first term describes the general gas movement which depends on the mean piston speed S_p , the second term models the gas movement due to combustion and depends on the difference between the measured pressure p and the motored pressure p_m . The calibration constants C_1 and C_2 were manually tuned in this work. The in-cylinder temperature T was estimated using the measured intake manifold temperature T_{IM} at IVC ($T_{IM} = T_{IVC}$) which gives

$$T = T_{IVC} \frac{pV}{p_{IVC} V_{IVC}}. \quad (6)$$

With the accumulated heat release Q obtained from Equation (1), combustion timing θ_{50} and the combustion start θ_{10} were computed using

$$\theta_x = \arg\max_{\theta} \left(Q(\theta) - \frac{x}{100} \max_{\theta} Q(\theta) \right), \quad x = \{10, 50\} \quad (7)$$

The ignition delay τ in milliseconds is then defined as

$$\tau = \frac{\theta_{10} - \theta_{SOI}}{0.006 N_{Speed}} \quad (8)$$

where the start of injection θ_{SOI} is the crank angle of the injector current impulse.

MODELING

A physics based correlation model was used to model the ignition-delay behavior whilst a calibrated static model was used to model the gains from the gas-system valves to the intake manifold composition and temperature. This section covers the description of these models and how they were used for controller design.

Ignition-Delay Model

The following mean-value model was used to model τ

$$\tau = A \exp\left(\frac{E_a}{RT}\right) [\overline{O_2}]^\alpha \quad (9)$$

where E_a is an apparent activation energy, R is the universal gas constant and A , α , are fuel dependent empirical parameters, $\bar{T}$ is the mean-cylinder temperature between θ_{SOI} and θ_{10} , $[\overline{O_2}]$ is the mean-cylinder oxygen concentration in the cylinder during the same time period. Similar models are presented by Heywood [12] and were used by Kempinski and Rife [13] and also by Donald and Eyzat [14] to parameterize the ignition delay for different fuels and engines. The oxygen concentration was computed as follows

$$[O_2] = \frac{[O_2]_{IVC}}{V} V_d \quad (10)$$

Where $[O_2]_{IVC}$ is the in-cylinder gas concentration at inlet-valve closing, consisting of oxygen coming from the air mass flow, the EGR mass flow and the residual gases.

This gives the expression for τ , assuming constant engine speed

$$\tau = A \exp\left(-\frac{E_a}{R \frac{\int_{\theta_{SOI}}^{\theta_{SOI}+\tau} T_{IVC} \left(\frac{V_{IVC}}{V(\theta)}\right)^{\gamma-1} d\theta}{\int_{\theta_{SOI}}^{\theta_{SOI}+\tau} d\theta}}\right) \left(\frac{\int_{\theta_{SOI}}^{\theta_{SOI}+\tau} \frac{[O_2]_{IVC}}{V(\theta)} V_d d\theta}{\int_{\theta_{SOI}}^{\theta_{SOI}+\tau} d\theta} \right)^\alpha \quad (11)$$

Model parameters for (9) were identified at 5, 10 bar gross indicated mean effective pressure p_{IMEP_g} and 1200, 1500 rpm by minimizing the model squared output error with respect to an identification data set. The ignition delay was varied by changing the injection timing and the valve positions in the EGR and IM system. In Figure 2, model output