

A Model-Based Injection-Timing Strategy for Combustion-Timing Control

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

ABSTRACT

The combustion timing in internal combustion engines affects the fuel consumption, in-cylinder peak pressure, engine noise and emission levels. The combination of an in-cylinder pressure sensor together with a direct injection fuel system lends itself well for cycle-to-cycle control of the combustion timing.

This paper presents a method of controlling the combustion timing by the use of a cycle-to-cycle injection-timing algorithm. At each cycle the currently estimated heat-release rate is used to predict the in-cylinder pressure change due to a combustion-timing shift.

The prediction is then used to obtain a cycle-to-cycle model that relates combustion timing to gross indicated mean effective pressure, max pressure and max pressure derivative. Then the injection timing that controls the combustion timing is decided by solving an optimization problem involving the model obtained.

The controller was experimentally tested on a Scania D13 heavy-duty diesel engine in the lower load range with a fuel mixture of 80 volume % gasoline and 20 volume % n-heptane.

Controller-performance test results are presented for different controller and model parameter settings at three different operating points. The results show that the controller was consistent in converging to a combustion timing point that was dependent on the magnitude of the modeled heat transfer rate but not necessarily the same as the experimentally found most efficient point. The convergence rate of the controller could be set by varying a design parameter β , which is the controller resistance to changing the injection timing when moving towards a more efficient operating point. Choosing β was shown to be a trade-off between controller convergence speed and cycle-to-cycle variation. The controller was also able to fulfill predefined constraints on max pressure and max pressure derivative.

CITATION: Ingesson, G., Yin, L., Johansson, R., and Tunestal, P., "A Model-Based Injection-Timing Strategy for Combustion-Timing Control," *SAE Int. J. Engines* 8(3):2015, doi:10.4271/2015-01-0870.

INTRODUCTION

During the compression and expansion strokes the in-cylinder pressure changes with crank angle as a result of volume change, combustion and heat transfer. The phasing of combustion relative to top-dead center (TDC) and the shape of the combustion rate affects the in-cylinder pressure [1].

Ideally, the combustion phasing should be set to minimize the specific fuel consumption such that constraints on cylinder pressure, noise and emissions are fulfilled. Assuming that such an ideal operating point exists, it will vary with operating conditions, and if the combustion timing is controlled with injection timing, the injection timing for the ideal combustion timing will also vary with operating conditions according to [2] [3].

Traditionally, the way to decide injection timing is to experimentally calibrate the most favorable injection timing for each operating point and then use this setting in open-loop operation. This might however

be sensitive to drift, especially in more sensitive combustion modes which was shown by Olsson et al. [4]. The approach also demands extensive experimental work.

Combustion-timing closed-loop control strategies could be divided into two subgroups, those who aim to control the combustion timing to follow a predefined combustion timing reference crank angle. Or those that aim to optimize engine performance subject to the combustion timing.

Examples of the former were investigated by Bengtsson et al. [5], where quantities such as crank angle for 50 % heat released, θ_{50} , crank angle for maximum pressure and crank angle for maximum heat release rate among others were used as feedback signals that indicate combustion timing. Closed-loop control of θ_{50} was obtained using physically-based auto-ignition models by Widd et al. [6], Shaver et al. [7] and Willems et al. [8], to mention a few. Nakayama et al [9] designed a model-based feed-forward auto-

ignition controller. However, the problem of deciding the optimal combustion timing reference still remains unsolved by using these controllers alone.

Karlsson et al. [10] showed that dynamical black-box models between injection timing, indicated mean effective pressure, max pressure derivative and NO_x level could be identified and used in Model Predictive Control in order to minimize specific fuel consumption subject to emission and pressure constraints. Lewander, Widd et al. designed an extremum-seeking controller to find the combustion timing for lowest specific fuel consumption with the use of θ_{50} feedback control in a CI engine [11], similar controllers were investigated by Hellström et al [12], and Killingsworth et al. [13] in SI and HCCI engines respectively.

This study investigates a single-zone model-based cycle-to-cycle combustion-timing controller that aims to maximize the gross indicated mean effective pressure subject to cylinder pressure constraints. The controller estimates the current cycle heat release rate, shifts this heat release rate relative to TDC and then predicts the cylinder pressure change due to this shift by simulation. The predicted pressure change is then used to formulate an optimization problem whose solution gives the wanted combustion timing and thus the wanted injection timing for the next cycle. This scheme is then repeated every cycle.

THE COMBUSTION TIMING CONTROLLER

The controller computation of the injection timing for the preceding cycle after each cycle could be divided in three steps:

- I. Estimation of the heat release rate of the current cycle.
- II. Prediction of the cylinder pressure due to a crank angle shift of the heat release rate.
- III. Computation of a shift in injection timing by solving an optimization problem by using gradient information obtained in step II.

Heat Release Rate Estimation

The heat release rate $dQ_c/d\theta$ is 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

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

where γ is the ratio of specific heats c_p/c_v , p is the measured in-cylinder pressure, V is the cylinder volume and θ the crank angle. $dQ_{ht}/d\theta$ is the convective heat transfer rate from the combustion chamber gases to the combustion chamber wall, and is modeled with the relation

$$\frac{dQ_{ht}}{d\theta} = h_c A (T - T_w), \quad (2)$$

where A is the combustion chamber surface area, T is the mean gas temperature and T_w the mean wall temperature respectively. The convection coefficient h_c was in this work modeled as a Woschni expression [14]

$$h_c = \alpha C_0 B^{0.2} p^{0.8} T^{-0.55} \omega^{0.8}, \quad (3)$$

where α is an empirical scaling constant, $C_0 = 1.3 \cdot 10^{-2}$, B is the cylinder bore. ω is the characteristic in-cylinder gas velocity and is modeled as

$$\omega = C_1 S_p + C_2 \frac{VT_{ivc}}{p_{ivc} V_{ivc}} (p - p_m), \quad (4)$$

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 . C_1 and C_2 are calibration constants and the suffix *ivc* denotes inlet valve closing timing. Since T is not measured inside the cylinder, the temperature in the cylinder at inlet valve closing time T_{ivc} is assumed to be equal to the inlet manifold temperature which is measured close to the cylinder inlet valve. T is then computed for all other θ using the ideal gas relation

$$T = \frac{pVT_{ivc}}{p_{ivc}V_{ivc}} = \frac{pV}{nR} \quad (5)$$

where n is the number of moles and R the ideal gas constant. γ depends on the cylinder gas composition and temperature, in this work it was estimated assuming that p and V satisfied the adiabatic relation (6) during the compression stroke with a constant C

$$pV^\gamma = C \quad (6)$$

γ and C could then be estimated from p and V with (6) using linear regression methods [15].

Pressure Prediction

After estimating $dQ_c/d\theta$ using (1) the objective is to predict the pressure change due to a change in the combustion timing. Now it is assumed that the pressure change due to a change in combustion timing in the next cycle is equivalent to the effect of a static crank angle shift of the heat release rate in the current cycle $dQ_c^0/d\theta$. This assumption relies on weak cycle-to-cycle dynamics of the combustion event and that the heat release rate shape is not changing too much with smaller changes in combustion timing. So first $dQ_c^0/d\theta$ is shifted back and forth with crank angle $\Delta\theta$

$$\frac{dQ_c^+}{d\theta} = \frac{dQ_c^0(\theta + \Delta\theta)}{d\theta}, \quad \frac{dQ_c^-}{d\theta} = \frac{dQ_c^0(\theta - \Delta\theta)}{d\theta} \quad (7)$$

Pressure changes p^+ and p^- corresponding to $dQ_c^+/d\theta$ and $dQ_c^-/d\theta$ are then simulated by first linearizing (23) w.r.t. p around the current cycle pressure p^0 , heat release rate $dQ_c^0/d\theta$ and T_w .

$$\frac{dp}{d\theta} = -\frac{\gamma}{V} \frac{dV}{d\theta} p + \frac{\gamma-1}{V} \left(\frac{dQ_c}{d\theta} - h_c A \left(\frac{pV}{nR} - T_w \right) \right) \quad (8)$$

Denoting the nonlinear term in the heat transfer model

$$\mu(p, \theta) \equiv \frac{(\gamma-1)h_c A}{nR} p, \quad (9)$$

the linearized system is given by (10) where Δp is the first order deviation from p^0

$$\frac{d\Delta p}{d\theta} = -\left(\frac{\gamma}{V} \frac{dV}{d\theta} + \frac{d\mu(p^0, \theta)}{dp} \right) \Delta p + \frac{\gamma-1}{V} \frac{d\Delta Q_c}{d\theta}, \quad (10)$$

p^+ and p^- are then given by $\Delta p + p^0$ where Δp is simulated with (10) where $d\Delta Q_c/d\theta = dQ_c^+/d\theta - dQ_c^0/d\theta$ and $d\Delta Q_c/d\theta = dQ_c^-/d\theta - dQ_c^0/d\theta$.

The reason for linearizing the system is that Δp is given by the expression

$$\Delta p(\theta) = \int_{\theta_{ivc}}^{\theta} \Phi(\theta, \vartheta) \Gamma(\vartheta) \frac{d\Delta Q_c(\vartheta)}{d\vartheta} d\vartheta, \quad (11)$$

where

$$\Phi(\theta, \vartheta) = e^{\int_{\vartheta}^{\theta} \left(-\frac{d\mu(p^0, \tau)}{dp} \right) d\tau} \left(\frac{V(\vartheta)}{V(\theta)} \right)^{\gamma} \quad (12)$$

and

$$\Gamma(\vartheta) = \frac{(\gamma-1)}{V(\vartheta)} \quad (13)$$

which makes Δp computationally cheap to simulate.

Injection-Timing Shift Optimization

With the predicted pressures p^+ and p^- computed, interesting quantities such as indicated gross mean effective pressure p_{IMEP_g} (from now on denoted p_{IMEP} , for ease of notation), max pressure p_{max} and max pressure derivative dp_{max} can also be predicted

$$p_{IMEP}^+ = \frac{1}{V_d} \int_{V_{ivc}}^{V_{evo}} p^+ dV, \quad p_{IMEP}^- = \frac{1}{V_d} \int_{V_{ivc}}^{V_{evo}} p^- dV \quad (14)$$

$$p_{max}^+ = \max_{\theta} p^+, \quad p_{max}^- = \max_{\theta} p^-, \quad (15)$$

$$dp_{max}^+ = \max_{\theta} dp^+, \quad dp_{max}^- = \max_{\theta} dp^-. \quad (16)$$

Moreover, the estimated derivatives of these quantities with respect to the combustion timing crank angle shift $\Delta\theta$ can be computed

$$\frac{dp_{IMEP}}{d\Delta\theta} = \frac{p_{IMEP}^+ - p_{IMEP}^-}{2\Delta\theta}, \quad (17)$$

$$\frac{dp_{max}}{d\Delta\theta} = \frac{p_{max}^+ - p_{max}^-}{2\Delta\theta}, \quad (18)$$

$$\frac{d(dp_{max})}{d\Delta\theta} = \frac{(dp^+)_{max} - (dp^-)_{max}}{2\Delta\theta}. \quad (19)$$

With the gradients (17), (18), (19) a simple prediction model for p_{IMEP} , p_{max} and dp_{max} in the next cycle with respect to $\Delta\theta$ is given by

$$\begin{bmatrix} p_{IMEP} \\ p_{max} \\ dp_{max} \end{bmatrix} = \begin{bmatrix} p_{IMEP}^0 \\ p_{max}^0 \\ dp_{max}^0 \end{bmatrix} + \begin{bmatrix} \frac{dp_{IMEP}}{d\Delta\theta} \\ \frac{dp_{max}}{d\Delta\theta} \\ \frac{d(dp_{max})}{d\Delta\theta} \end{bmatrix} \Delta\theta \quad (20)$$

With this model, an optimization problem for finding the wanted $\Delta\theta$ can be formulated

$$\underset{\Delta\theta}{\text{minimize}} \quad -p_{IMEP} + \beta(\Delta\theta)^2 \quad (21)$$

subject to:

$$\begin{bmatrix} p_{IMEP} \\ p_{max} \\ dp_{max} \end{bmatrix} = \begin{bmatrix} p_{IMEP}^0 \\ p_{max}^0 \\ dp_{max}^0 \end{bmatrix} + \begin{bmatrix} \frac{dp_{IMEP}}{d\Delta\theta} \\ \frac{dp_{max}}{d\Delta\theta} \\ \frac{d(dp_{max})}{d\Delta\theta} \end{bmatrix} \Delta\theta,$$

$$\begin{bmatrix} p_{max} \\ dp_{max} \\ |\Delta\theta| \end{bmatrix} \leq \begin{bmatrix} l_{p_{max}} \\ l_{dp_{max}} \\ l_{\Delta\theta} \end{bmatrix},$$

where β is a cost weight for changing the combustion timing and $l_{p_{max}}$, $l_{dp_{max}}$ and $l_{\Delta\theta}$ are upper limits on p_{max} , dp_{max} and the absolute value of $\Delta\theta$. Now by solving (21) an optimal choice for a combustion timing shift $\Delta\theta$ can be chosen for the next cycle. The optimization problem can be rephrased by interchanging $\Delta\theta$ and $\Delta\theta_{opt}$ if the combustion