POSSIBILITIES TO IDENTIFY ENGINE COMBUSTION MODEL PARAMETERS BY ANALYSIS OF THE INSTANTANEOUS CRANKSHAFT ANGULAR SPEED

by

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In this paper, novel method for obtaining information about combustion process in individual cylinders of a multi-cylinder spark ignition engine based on instantaneous crankshaft angular velocity is presented. The method is based on robust box constrained Levenberg-Marquardt minimization of non-linear least squares given for measured and simulated instantaneous crankshaft angular speed which is determined from the solution of the engine dynamics torque balance equation. Combination of in-house developed comprehensive zero-dimensional two-zone spark ignition engine combustion model and analytical friction loss model in angular domain have been applied to provide sensitivity and error analysis regarding Wiebe combustion model parameters, heat transfer coefficient, and compression ratio. The analysis is employed to evaluate the basic starting assumption and possibility to provide reliable combustion analysis based on instantaneous engine crankshaft angular speed.

Key words: *spark ignition engine, two-zone model, mechanical losses, instantaneous crankshaft speed, optimization*

Introduction

Combustion analysis based upon in-cylinder pressure measurement represents crucial application in engine development and design. In order to quantify combustion parameters in internal combustion (IC) engines, burn rate analysis is used mainly to determine burn angles, normalized variable of mass fraction burned (MFB), and in differential form, the rate of heat release (RoHR). The analysis is based upon the First Law of Thermodynamics and, depending on accuracy and details required, includes the sub-models for the effects of heat transfer, residual fraction, gas leakage through crevices, and gas properties. The method is usually referred to as direct method due to the fact that solution of the equation for the first law of thermodynamics provides the value of RoHR directly. The major drawback arises from the deterministic approach, *i. e.*, combustion analysis is largely affected by number of uncertainties whose can not be identified explicitly. This mainly stands for heat transfer and gas leakage models which are hard to calibrate independently and determination of compression ratio and errors introduced by measurement such as pressure sensor offset and pressure trace-angle phase shift. Analysis and application of deterministic method has been demonstrated in the authors' previous work [1] regarding both spark ignition (SI) and compression ignition (CI) engines and more information can be also found in selected literature [2, 3].

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The most frequently used parametric model to describe combustion in IC engines is based on original Wiebe function. The MFB and RoHR are given in simple exponential form relying on few input parameters which is an obvious advantage when it comes to process modelling. However, in order to obtain two essential model parameters from pressure related combustion analysis based on direct method – form factor and combustion duration angle, additional computational effort is required. Stochastic methods or indirect method, as referenced in this work, incorporating minimization of difference between simulated and measured in-cylinder pressure solve this problem intrinsically. Additionally, this concept enables one to identify simultaneously the heat transfer model constants, while real compression ratio value, gas leakage and measurement biases can be identified from pressure trace without combustion, which is beneficial when it comes to high accuracy. This concept has been demonstrated successfully in the past in terms of combustion analysis in CI engines [4] utilizing box constrained Levenberg-Marquardt minimization of non-linear least squares (LSQ) given for measured and simulated in-cylinder pressure. Regarding SI engines, which is the subject of this work, the same concept was demonstrated utilizing Newton-Gauss minimization algorithm [5, 6].

The possibility to obtain information on combustion process in individual cylinders by analysing signal from crankshaft sensors such as instantaneous torque or crank shaft speed is extremely attractive. The reason for that comes from relative simplicity of measuring crankshaft torque or speed compared to individual in-cylinder pressure measurement which is considered impractical in case of mass-production engines. Anderson *et al*. [7] used a Torque Ratio concept based on direct torque measurement proposed by Shagerberg and Mc Kelvey [8] to evaluate heat release process in individual cylinders of a SI engine. Relationship to pressure ratio concept (cylinder pressure to corresponding motored pressure) proposed by Matekunas was obtained using single-zone model proposed by Heywood and Gatowski followed by Wiebe's parametric heat release model. LSQ minimization was in turn used to provide tuning of combustion model parameters.

Hamedović *et al.* [9] proposed in-cylinder pressure signal decomposition to combustion and compression terms to extract the combustion contribution in individual cylinders of a SI engine from crankshaft angular speed. Mean components of gas, load and friction torques were assumed balanced, while alternating component of disturbance torque (term used to summarise friction and torsion effects) was determined by comparing gas torque obtained from pressure measurement in reference cylinder and instantaneous crankshaft speed [10]. Engine speed disturbances due to incremental disc geometric faults were corrected using improved crankshaft energy concept [11-13]. Similar approach related to combustion in CI engine was also presented by Weißenborn *et al*. [14].

This paper addresses the application of the indirect concept applied to the identification of Wiebe combustion model parameters and heat transfer coefficient correction based on crankshaft instantaneous angular speed. The method is based on robust box constrained Levenberg-Marquardt minimization of nonlinear LSQ given for measured and simulated instantaneous crankshaft angular speed determined from the solution of the engine dynamics torque balance equation. Combination of comprehensive engine process model and analytical friction loss model in angular domain has been applied to provide sensitivity and error analysis regarding crucial model input parameters. This analysis will be used to evaluate the basic starting assumption and possibility to provide reliable combustion analysis based on instantaneous engine crankshaft angular speed.

Engine model

Governing equations

The simulation of SI engine combustion was developed on basis of zero-dimensional (0-D) two-zone (2-Z) thermodynamic model. Engine combustion chamber as open thermodynamic system is displayed schematically in fig. 1. In this work, approach proposed by Pischinger [15] is used.

The 0-D2-Z model has been built upon following assumptions: the process is unsteady; unburned mixture and combustion products are separated by means of thin zone of combustion reactions (flame front); heat transfer between zones is neglected; pressure distribution is homogenous and equal in each zone; temperature and gas composition and properties are homogenous within each zone; the combustion mixture is homogenous and consists of fuel vapour, air and residual combustion products. The basic equation is derived from the conservation of energy law (the First Law of Thermodynamics for open systems) applied to the cylinder volume. Transforming to differential form in angular domain φ and introducing index i to denote zones of unburned mixture (u) and burned combustion products (b), equation can be expressed in general form for both zones:

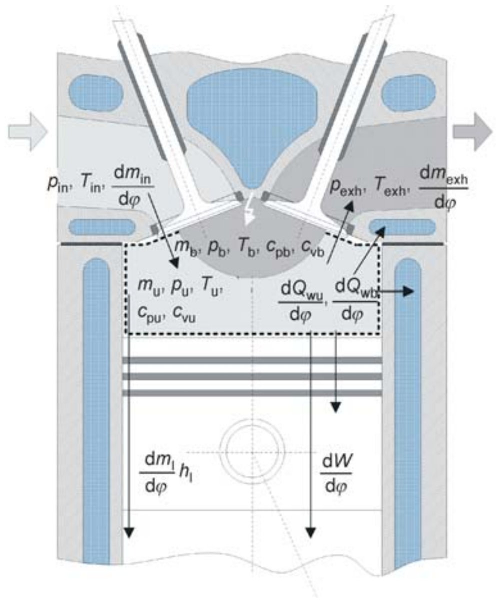

Figure 1. SI engine cylinder as a control volume in combustion model

$$\frac{dU_i}{d\varphi} = \frac{d(m_i u_i)}{d\varphi} = \frac{dm_i}{d\varphi} u_i + m_i \frac{du_i}{d\varphi} = \frac{dQ_i}{d\varphi} - p\frac{dV_i}{d\varphi} \tag{1}$$

Mass balance equation is applied for both zones in order to incorporate necessary rates of change coming from combustion ($dm_c/d\varphi$), intake ($dm_{in,i}/d\varphi$), exhaust flow ($dm_{exh,i}/d\varphi$), and gas leakage ($dm_l/d\varphi$). Mass change rate for burned gas and unburned mixture is given in general form (2):

$$\frac{dm_i}{d\varphi} = k_{c,i}\frac{dm_c}{d\varphi} - k_l \frac{dm_{l,i}}{d\varphi} + k_{in,i}\sum_{i_{iv}=1}^{n_{iv}}\left(\frac{dm_{in,i}}{d\varphi}\right)_{i_{iv}} + k_{exh,i}\sum_{i_{ev}=1}^{n_{ev}}\left(\frac{dm_{exh,i}}{d\varphi}\right)_{i_{ev}} \tag{2}$$

Equation supports multi-valve and asymmetric valve timing configuration for each intake (index i_{iv}) and exhaust valve (index i_{ev}), while intermittent and repeatable nature of the whole is taken into account by means of simple programme switch indicators (k_c, k_l, k_{in}, and k_{exh}) that change their values in discrete domain (0 or 1 depending of the process phase). The rate of energy exchange in angular domain for both zones is defined using the same general approach:

$$\frac{dQ_i}{d\varphi} = -\frac{dQ_{w,i}}{d\varphi} + k_{c,i}\frac{dm_c}{d\varphi}h_u - k_l\frac{dm_{l,i}}{d\varphi}h_i + \\ + k_{in,i}\sum_{i_{iv}=1}^{n_{iv}}\left(\frac{dm_{in,i}}{d\varphi}\right)_{i_{iv}} h_{o,in,i} + k_{exh,i}\sum_{i_{ev}=1}^{n_{ev}}\left(\frac{dm_{exh,i}}{d\varphi}\right)_{i_{ev}} h_{o,exh,i} \tag{3}$$

The right side of the equation represents the sum of heat loss through the cylinder walls ($\mathrm{d}Q_\mathrm{w}/\mathrm{d}\varphi$), the change of gas enthalpy due to combustion ($h_\mathrm{u}\mathrm{d}m_\mathrm{c}/\mathrm{d}\varphi$), gas leakage ($h_\mathrm{i}\mathrm{d}m_\mathrm{l,i}/\mathrm{d}\varphi$) and flow through intake ($h_\mathrm{o,in,i}\mathrm{d}m_\mathrm{in,i}/\mathrm{d}\varphi$) and exhaust valve ports ($h_\mathrm{o,exh,i}\mathrm{d}m_\mathrm{exh,i}$). Differential equation for unburned mixture and combustion products temperatures is obtained by introducing derivative of gas volume for each zone in logarithmic form and partial derivatives of specific internal energy u and gas constant R in respect of pressure and temperature into eq. (1):

$$\frac{\mathrm{d}T_i}{\mathrm{d}\varphi}=\frac{\dfrac{\mathrm{d}Q_i}{\mathrm{d}\varphi}-\dfrac{\mathrm{d}m_i}{\mathrm{d}\varphi}h_i}{m_i\left(\dfrac{\partial u_i}{\partial T_i}+T_i\dfrac{\partial R_i}{\partial T_i}+R_i\right)}+\frac{\dfrac{\mathrm{d}p}{\mathrm{d}\varphi}\left(\dfrac{R_iT_i}{p}-T_i\dfrac{\partial R_i}{\partial T_i}-\dfrac{\partial u_i}{\partial p}\right)}{\dfrac{\partial u_i}{\partial T_i}+T_i\dfrac{\partial R_i}{\partial T_i}+R_i} \tag{4}$$

Mass balance eq. (5) is used to obtain derivative of specific volume for cylinder charge (6):

$$m=m_\mathrm{b}+m_\mathrm{u}=xm+(1-x)m \tag{5}$$

$$\frac{\mathrm{d}}{\mathrm{d}\varphi}\left(\frac{V}{m}\right)=\frac{\mathrm{d}V}{\mathrm{d}\varphi}\frac{1}{m}-\frac{V}{m^2}\frac{\mathrm{d}m}{\mathrm{d}\varphi}=\frac{\mathrm{d}x}{\mathrm{d}\varphi}v_b+x\frac{\mathrm{d}v_\mathrm{b}}{\mathrm{d}\varphi}-\frac{\mathrm{d}x}{\mathrm{d}\varphi}v_\mathrm{u}+(1-x)\frac{\mathrm{d}v_\mathrm{u}}{\mathrm{d}\varphi} \tag{6}$$

where ratio $x = m_\mathrm{b}/m$ represents the mass fraction of burned mixture. Since the pressure is presumed homogenous and equal in both zones, differential equation for cylinder charge pressure can be derived from eq. (6) introducing eq. (4) and partial derivatives of specific volume with respect to pressure and temperature:

$$\frac{\mathrm{d}p}{\mathrm{d}\varphi}=\frac{\dfrac{\mathrm{d}V}{\mathrm{d}\varphi}\dfrac{1}{m}-\dfrac{V}{m^2}\dfrac{\mathrm{d}m}{\mathrm{d}\varphi}-\dfrac{\mathrm{d}x}{\mathrm{d}\varphi}\left(\dfrac{R_\mathrm{b}T_\mathrm{b}-R_\mathrm{u}T_\mathrm{u}}{p}\right)-f_\mathrm{p1}}{f_\mathrm{p2}} \tag{7}$$

$$f_\mathrm{p1}=\frac{\partial v_\mathrm{b}}{\mathrm{d}T_\mathrm{b}}\left[\frac{\dfrac{\mathrm{d}Q_\mathrm{b}}{\mathrm{d}\varphi}-\dfrac{\mathrm{d}m_\mathrm{b}}{\mathrm{d}\varphi}h_\mathrm{b}}{m\left(\dfrac{\partial u_\mathrm{b}}{\partial T_\mathrm{b}}+T_b\dfrac{\partial R_\mathrm{b}}{\partial T_\mathrm{b}}+R_b\right)}\right]+\frac{\partial v_\mathrm{u}}{\mathrm{d}T_\mathrm{u}}\left[\frac{\dfrac{\mathrm{d}Q_\mathrm{u}}{\mathrm{d}\varphi}-\dfrac{\mathrm{d}m_\mathrm{u}}{\mathrm{d}\varphi}h_\mathrm{u}}{m\left(\dfrac{\partial u_\mathrm{u}}{\partial T_\mathrm{u}}+T_\mathrm{u}\dfrac{\partial R_\mathrm{u}}{\partial T_\mathrm{u}}+R_\mathrm{u}\right)}\right] \tag{8}$$

$$f_\mathrm{p2}=x\frac{\partial v_\mathrm{b}}{\mathrm{d}T_\mathrm{b}}\frac{\dfrac{R_\mathrm{b}T_\mathrm{b}}{p}-T_\mathrm{b}\dfrac{\partial R_\mathrm{b}}{\partial p}-\dfrac{\partial u_\mathrm{b}}{\partial p}}{\dfrac{\partial u_\mathrm{b}}{\partial T_\mathrm{b}}+T_\mathrm{b}\dfrac{\partial R_\mathrm{b}}{\partial T_\mathrm{b}}+R_\mathrm{b}}+(1-x)\frac{\partial v_\mathrm{u}}{\mathrm{d}T_\mathrm{u}}\frac{\dfrac{R_\mathrm{u}T_\mathrm{u}}{p}-T_\mathrm{u}\dfrac{\partial R_\mathrm{u}}{\partial p}-\dfrac{\partial u_\mathrm{u}}{\partial p}}{\dfrac{\partial u_\mathrm{u}}{\partial T_\mathrm{u}}+T_\mathrm{u}\dfrac{\partial R_\mathrm{u}}{\partial T_\mathrm{u}}+R_\mathrm{u}} \tag{9}$$

Thermodynamic properties of the mixture and components of combustion products are approximated by means of NASA 9-coefficients polynomials provided by Gordon and McBride [16]. Molar concentrations of the species and respective partial derivatives of specific internal energy u, specific volume v, and gas constants R, have been obtained on assumption of equilibrium composition of combustion products and calculated by means of Olikara-Borman [17] model. The instantaneous gas mass flow through valves was modelled assuming one-dimensional, quasi-steady, compressible isentropic flow, while discharge coefficients for both intake and exhaust valves were determined experimentally, using flow test bench. The same assumptions were employed regarding instantaneous mass flow through crevices and revised approach based on piston ring gap area and pressure related discharge coefficient formula proposed by Wannatong [18] was used.

Heat release model

If the start of combustion (φ_{SOC}) is known, which can be for SI engines, in most general case, regarded as ignition advance (φ_{IGN}), only two parameters in Wiebe model must be tuned in order to reproduce heat release during combustion correctly – m, defined as form factor and $\Delta\varphi_{\mathrm{CD}}$ defined as combustion duration angle. The fraction of burned mixture x is given in exponential form:

$$x(\varphi) = 1 - \exp\left[-a\left(\frac{\varphi - \varphi_{\mathrm{SOC}}}{\Delta\varphi_{\mathrm{CD}}}\right)^{m+1}\right] \tag{10}$$

In order to establish initial guess and box constraints required for optimization procedure and so narrow the search field, method for approximate parameter determination is required. Provided the set of model parameters for the reference operating point, limited predictions can be obtained using correlations proposed by Csallner, Witt or Lindstrom [5]. To provide direct prediction of form factor m for stoichiometric operation, as a function of mean piston speed (c_{m}), ignition advance (φ_{IGN}), and residual mass fraction (x_{RG}) proposed by Bonatesta *et al*. [19]:

$$m = 3.46 c_m^{-0.225} (1 + \sqrt{\varphi_{\mathrm{IGN}}})^{0.35} (1 - 1.28 x_{\mathrm{RG}}) - 1 \tag{11}$$

Simple approximate correction for both lean and rich range is made by introducing the function proposed by Lindstrom [5], based on model for laminar flame speed summarised in [20]. Provided the engine speed n for given operating point and form factor approximated by eq. (11), combustion duration $\Delta\varphi_{\mathrm{CD}}$ is calculated using simple linear correlation proposed by Lindstrom [5]:

$$\Delta\varphi_{\mathrm{CD}} = 23.7529(3.51\cdot 10^{-4} n + 5.71 - m) \tag{12}$$

Heat transfer model

Heat transfer from the gas charge to the cylinder walls is modelled by means of Newton's law of convective heat transfer. Introducing index i to denote corresponding zones (u and b) and index j to denote surfaces of the piston crown, cylinder liner and cylinder head, respectively, Newton's equation becomes:

$$\frac{\mathrm{d}Q_{\mathrm{w},i}}{\mathrm{d}t} = \alpha_{\mathrm{w},i} \sum A_{\mathrm{w},i,j} (T_i - T_{\mathrm{w},j}) \tag{13}$$

Extensive comparative analysis for most popular relations regarding instantaneous heat transfer coefficient w can be found in literature [15, 23-25]. In terms of qualitative accuracy, most frequently used models proposed by Woschni and Fieger [21], Hohenberg [22] or Annand as presented in [20] display comparable and satisfactory results. Model proposed by Woschni provides the most detailed approach in terms of process phase distinctions and their influences on heat transfer phenomena, but applied without calibration, often fails to predict instantaneous heat coefficient with sufficient accuracy. Compared to other two most often used global models, Woschni model appears to be the least accurate in the case of SI engine, giving slightly under-estimated heat flux values during compression and over-estimated values during combustion phase [23-25]. Deploying single correction factor in order to calibrate heat transfer coefficient, which is usual practice, would be insufficient, and more accurate equation is required. Revised version of Woschni and Fieger model proposed by Chang *et al.* [23] is used. It is primarily intended for HCCI operation, however, Cho *et al.* [24] referenced this model for direct injection SI engines (DISIE), and Wang *et al.* [25] validated it for DISIE homogenous charge operation which makes this model interesting for conventional Port Fuel Injection SI engines, as well. Revised model introduces instantaneous combustion chamber height H_{cc} instead of cylinder diameter D as characteristic length in original equation, which is motivated by widely accepted chamber height as characteristic length in in-cylinder global flow and turbulence simulations. In that way, under-predicted heat flux during the late stage of compression can be compensated, because prior to ignition, the chamber height is usually less than cylinder diameter. Gas temperature influence has been decreased by optimizing exponent (–0.73 instead of original value –0.53). Chang's equation is given as follows:

$$\alpha_{\mathrm{w}} = 0.0130 H_{\mathrm{cc}}^{-0.2} p^{0.8} T^{-0.73} \left[C_{\mathrm{w}1} c_m + \frac{C_{\mathrm{w}2}}{6} \frac{V_{\mathrm{h}} T_1}{p_1 V_1} (p - p_m) \right]^{0.8} \tag{14}$$

The changes were validated and related to high-pressure phase exclusively, and for low-pressure gas exchange process, original model set-up proposed by Woschni must be used. Surface temperatures $T_{\mathrm{w,j}}$, are computed and refined through iterative numeric process assuming spatially averaged values.

Engine dynamic and friction model

If the crankshaft is assumed as a rigid body, angular speed, and acceleration would be influenced exclusively by gas pressure, friction losses, inertia, and external load torque. However, dynamic response of the shafting caused by variation in excitation torques is superimposed changing the character of angular speed variations. Detailed models of elastic engine shafting for both speed and gas pressure torque prediction are widely used [8, 26-28], but issues related to insufficiently accurate prediction was reported as well, and addressed to some level of imbalance in friction modelling and neglecting auxiliaries powertrain [8, 26]. The wealth of literature, however, references simplified Single Degree of Freedom (1-DoF) [9-14, 29-31] or 2-DoF rigid body dynamic models [32]. In this work, considering model application to the steady-state operation, simplified 1-DoF dynamic engine model is applied along with detailed angle resolved friction model. Dynamic system engine-dynamometer is presented in fig. 2.

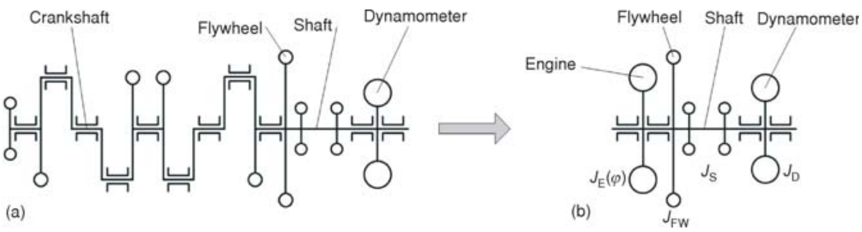

Figure 2. Engine dynamic model (a) and reduced, equivalent 1-DoF dynamic model (b)

Engine mass moment of inertia J_{E} is a function of both mass and position of slider mechanism components, and therefore, variable in angle domain. Engine inertia and its first derivative in respect of crank angle are calculated by means of dynamically equivalent model (Hafner and Maass [33]), while inertia of flywheel J_{FW}, connecting shaft J_{S} and dynamometer J_{D}

are known values obtained through 3-D modelling or from experiment. Assuming crank and connecting shafts as rigid bodies [30, 31], torque balance equation for engine-dynamometer system can be derived from kinetic energy equation applying Newton's principle:

$$[J_{\mathrm{E}}(\varphi) + J_{\mathrm{FW}} + J_{\mathrm{S}} + J_{\mathrm{J}}]\ddot{\varphi}(\varphi) + \frac{1}{2}\frac{\mathrm{d}J_{\mathrm{E}}(\varphi)}{\mathrm{d}\varphi}\dot{\varphi}(\varphi)^2 = T_{\mathrm{G}}(\varphi) - T_{\mathrm{F}}(\varphi) - T_{\mathrm{L}} \tag{15}$$

Superimposed, variable gas-pressure torque contributions from individual cylinders are denoted by T_{G}, and T_{L} is the load torque which is measured and assumed constant in time/angle domain. The term T_{F} denotes the sum of torques from mechanical losses in tribologic systems and auxiliaries. Provided the focus on statistical markers, such as maximum in-cylinder pressure and its angular position with respect to TDC, neglecting engine friction losses and auxiliaries power consumption can be justified because the influence of engine friction torque is small around the TDC. Angle or time averaged models of engine mechanical losses (*e. g.* model Chen-Flynn as applied by Chen and Moskwa [30] or model Milington-Hartles applied by Filipi and Assanis [31]) can be deployed in first instance to improve predictivity. If it comes to identification of combustion model parameters, detailed high sensitivity angle-resolved friction losses model must be applied. Analytical models based on Reynolds equation are available [34], although cumbersome and demanding, provide universal approach and minimize the influence of empirical constants. Friction losses in piston-cylinder contact were calculated by means of angle-resolved models presented by Taraza [35] relying on basic lubrication theory and Stribeck diagram. Assuming crank and cam shaft bearings short radial bearings with hydro-dynamic lubrication (HDL) the friction losses are modelled by equation proposed by Ocvirk. Theoretical background is given by Stachowiak [36], and detailed application in terms of IC engine bearings is presented by Taraza [35]. The friction in cam-tappet contact, assuming elasto-hydrodynamic lubrication (EHDL), was modelled by means of approximate solution of Reynolds equation presented by Teodorescu [37].

Model sensitivity and error analysis

Combustion model parameters identification based on instantaneous angular speed is conducted by minimization of least squares objective function (LSQOF) constructed upon the set of modelled and measured data:

$$F(X) = \sum_{i=1}^{N} w_{\mathrm{f},i}[\omega_{e,i}(\varphi_i) - \omega_{m,i}(\varphi_i, X)]^2 = \sum_{i=1}^{N} w_{\mathrm{f},i} f_i^2(\varphi_i, X) \tag{16}$$

The term X is a colon-matrix comprising model parameters (in this case $X = [m\Delta\varphi_{\mathrm{CD}}CF_{\alpha\mathrm{W}}]^{\mathrm{T}}$), φ – the independent variable, represented here as crank angle, ω_{e} and ω_{m} represent vectors of angular speed values provided through experiment and modelling, respectively, both of same size N. The term w_{f} represents weight factor which is set up as to increase or decrease the importance and influence of each individual point. In this work, the LSQOF $F(X)$ is minimized by means of box constrained Levenberg-Marquardt optimization algorithm which was proved more reliable in case of complex non-linear systems than Newton-Gauss method, often regarded as basic in minimization of LSQ [5].

Preliminary analysis was performed in order to gain first experience and validate starting assumptions weather combustion analysis on basis of instantaneous angular speed is viable or not. Analysis is performed as to provide information on model sensitivity and robustness, to explore the level of angular speed sensitivity to individual model parameters and to investigate the nature and magnitude of errors in identification if uncertainties in model parameters exist.

Table 1. Reference values and limits of engine model parameters used in sensitivity analysis

Model parameter	Reference value	Parameter limits
Wiebe comb. model form factor	m_{ref} = 4.03 (model Bonatesta [16])	±5%, ±10%
Combustion duration	$\varphi_{\mathrm{CD,ref}}$ = 63.4° CA (model Lindstrom [5])	±5%, ±10%
Compression ratio	$\varepsilon_{\mathrm{ref}}$ = 9.2 (engine spec., tab. 3)	±2%, ±4%
Heat transfer coefficient	$\alpha_{\mathrm{W,ref}}$ (model Chang [20])	±10%, ±20%

The first numerical experiment was performed by varying individually the values of form factor, combustion duration, compression ratio and heat transfer coefficient in order to analyse angular speed model sensitivity. Reference values for form factor, combustion duration and heat transfer coefficient were established through models presented in previous sections (tab. 1).

The boundaries are defined according to observed uncertainties of each relevant model. In order to encounter both high load and high inertia forces, analysis was performed at reference operating point n = 3000 min^{-1}/WOT.

The level of sensitivity differs for parameters considered in this analysis. The character of the relative error in respect to reference condition is similar in cases of Wiebe model parameters having in mind that both variables affect heat release dynamics in the same way. The influence of form factor on modelled in-cylinder pressure trace and instantaneous angular speed is presented in fig. 3. By increasing the value of form factor for given combustion duration, heat release is delayed, therefore, losses in the late phase of compression are reduced and contribution to the angular speed is observed during expansion. The change in angular speed of ±0.012% can be expected if form factor is varied within ±10%. The influence of combustion duration on instantaneous angular speed is presented in fig. 4. The increase of combustion duration for given value of form factor, causes delayed heat release.

Instantaneous angular speed appears to be more sensitive to changes in combustion duration, and changes between –0.040 and 0.053% can be observed for combustion duration varied within 10%. The influence of heat transfer coefficient on instantaneous angular speed is presented in fig. 5. Having in mind that the

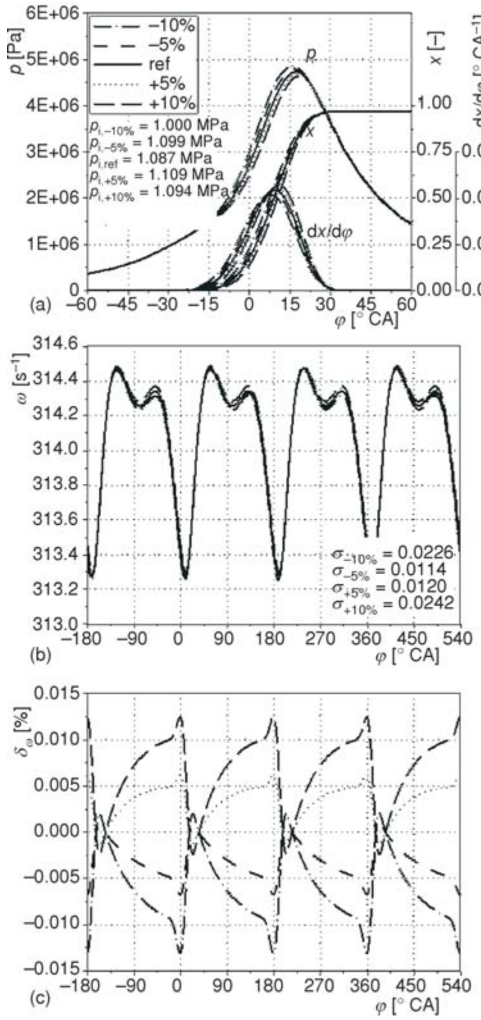

Figure 3. The influence of Wiebe form factor on in-cylinder pressure (a), angular speed (b), and angular speed deviation (c) in respect to reference values (n = 3000 min^{-1}, WOT)

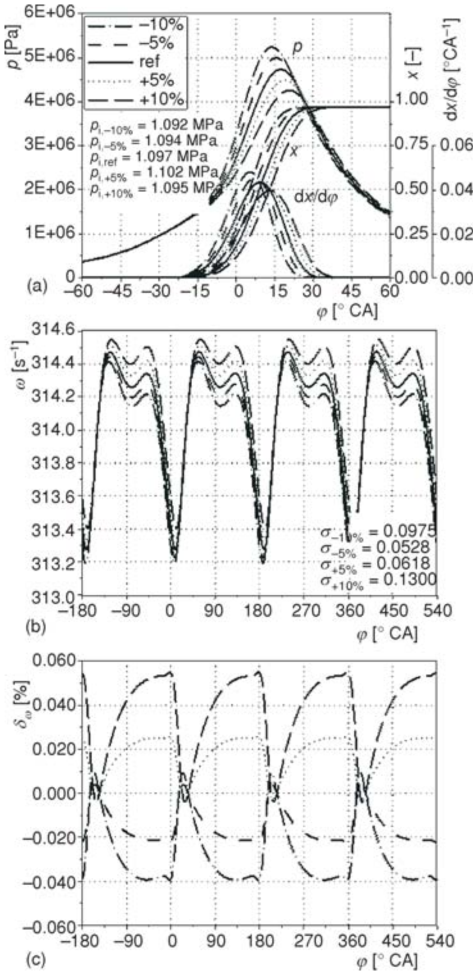

Figure 4. The influence of combustion duration on in-cylinder pressure (a), angular speed (b), and angular speed deviation (c) in respect to ref. values (n = 3000 min^{-1}, WOT)

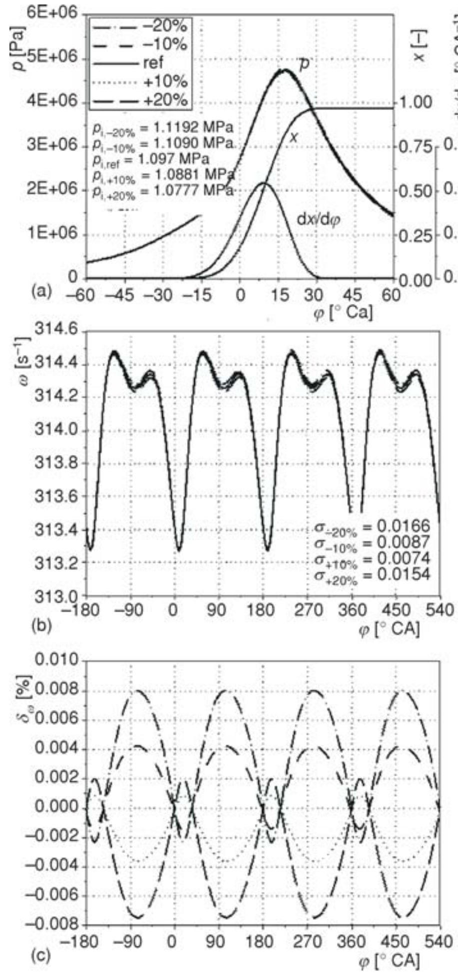

Figure 5. The influence of heat transfer coefficient on in-cylinder pressure (a), angula speed (b), and angular speed deviation (c) in respect to ref. values (n = 3000 min^{-1}, WOT)

heat transfer coefficient has been varied within extended limits (±20%) compared to other three model variables, deviation of angular speed is rather small, approximately ±0.008%. Decreasing heat transfer coefficient produces higher values of IMEP, and so, influences increase in instantaneous angular speed.

The influence of compression ratio on instantaneous angular speed is presented in fig. 6. By increasing its value, instantaneous angular speed decreases locally, around each TDC, which is caused by losses during the late compression. For discrete change of +4% in compression ratio, decrease in angular speed in proximity of TDC is ~0.02%. Between successive TDC, increase of 0.006% is observed, which is due to positive influence of increased compression ratio on process efficiency.

Analysis, designed as to vary one model parameter independently of others affecting the combustion process, does not reflect real conditions because their interrelationships are ne-

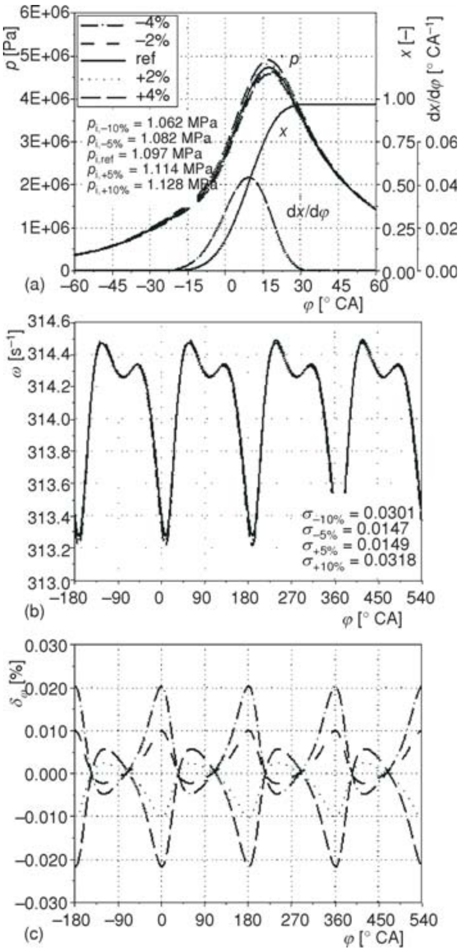

Figure 6. The influence of compression ratio on in-cylinder pressure (a), angular speed (b), and angular speed deviation (c) in respect to reference values (n = 3000 min^{-1}, WOT

glected. Despite the fact that changes of angular speed observed through numerical experiment are extremely small – 10^{-4} to 10^{-5}, this analysis provides valuable conclusions about instantaneous angular speed sensitivity. Sensitivity analysis also shows that instantaneous angular speed responds to model parameter variations and implies that identification procedure based on numerical optimisation is viable. However, compression ratio appears to be influential and must be treated as systematic error.

Systematic errors

The uncertainties in identification process which could appear due to systematic errors in compression ratio are validated using virtual experiment. In order to minimise disturbances, instead of measurement, model, given by eqs. (4) and (7)-(15) is used to obtain in-cylinder pressure and instantaneous angular speed assuming compression ratio 2% lower in respect to declared value (tab. 1), which is within production tolerances and also regarded to be real in respect to thermal dilatation. In the second step, production tolerances and usual compression ratio decrease due to thermal dilatation were neglected and Wiebe model parameters and heat transfer coefficient correction factor ($CF_{\alpha W}$) were identified upon declared compression ratio using minimisation of LSQF given in eq. (16). This is done upon the difference between engine speed obtained by modelling through virtual experiment (further referred as modelled) and that obtained through identification procedure (referred as identified).

In order to ensure convergence of box constrained optimisation applied to LSQF, boundaries are set in respect to reference values given in tab. 1 as: for heat transfer coefficient was optimised within ±40% (presented through correction factor $CF_{\alpha W}$), form factor m within ±40% and combustion duration within ±10%. Modelled and identified instantaneous angular speed and its relative error during the iterative optimisation procedure are displayed in fig. 7. After 20 iterations, which is an arbitrary defined limit, relative error of instantaneous angular speed approaches value of 0.003%, while LSQOF reaches 0.936. The difference between modelled and identified angular speed obviously exists, however, its value remains below the rεsolution of graphical presentation and can not be observed. The results of combustion model parameters identification (fig. 8), are not as impressive as in the case of angular speed, although the difference in IMEP is low (~1 kPa). Form factor m was identified with very high relative error of +10% compared to reference value. The heat transfer coefficient was identified with relative error of +9% in respect to reference value obtained through Chen model. The

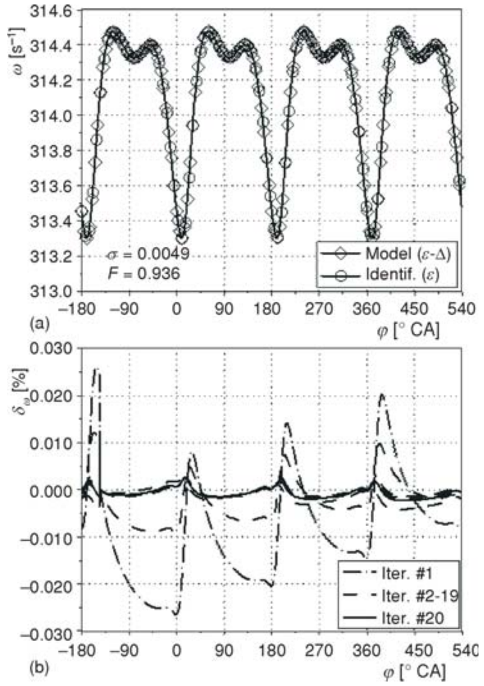

Figure 7. Angular speed identification: modeled and identified angular speed (a) and relative deviation during iterative optimisation (b) (n = 3000 min^{-1}, WOT)

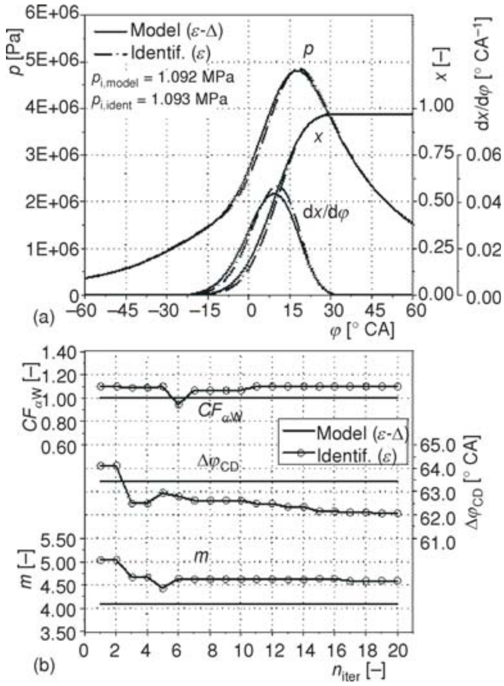

Figure 8. Model parameters identification: pressure, MFB and RoHR (a), correction factor of the heat transfer coefficient, combustion duration and form factor (n = 3000 min^{-1}, WOT)

result of the identification of combustion duration appears better than in previous cases reaching relative error of ~2.3%. The analysis shows that accurate value of compression ratio is required prior to combustion model identification and must be identified independently. When the combustion is switched off and engine is fully warmed up, compression and expansion processes can be simulated with high accuracy, therefore, engine cranking can be used to identify compression ratio. This case is investigated in separate numerical experiment considering cranking speed of 250 min^{-1} and WOT operation (fig. 9). The angular speed trace shows symmetry around each TDC, although this symmetry is not perfect (thermodynamic losses and gas leakage), which is, actually true for the in-cylinder pressure, as well. Deviation between the angular speed trace obtained for the reference compression ratio and those obtained for values varied within the range of ±4% are higher than those observed for normal engine operation mainly due to lower inertia torque to gas pressure torque ratio. The highest values of app. ±1.5% are observed around each TDC, which is sufficient to perform reliable identification of compression ratio during engine cranking.

Experimental results

Models and methods presented in previous sections are used to validate assumption on combustion analysis based on real measurement of instantaneous angular speed. The experiment was conducted on serial-production port fuel injection petrol engine DMB M202PB13. Technical data are given in tab. 2. In-cylinder pressure and instantaneous angular speed were measured simultaneously in steady state operating regime using different angle domains. In-cylinder pressure was

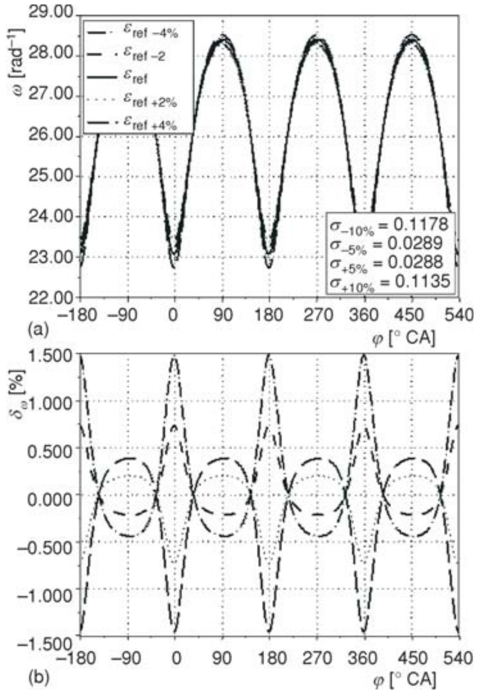

Figure 9. The influence of compression ratio and angular speed (a) and angular speed relative deviation in respect to reference value (b) (engine cranking n = 250 min^{-1}, WOT)

Table 2. Main engine specification

Description	Value
Engine manufacturer	DMB
Engine type	SI, MPI, M202PB13
Bore/stroke	80.5/67.4 mm
No. of cylinders	4
Compression ratio	9.2 (+0.2/ 0.1)
Max. power output	52 kW @ 5800 min^{-1}
Cooling system	Liquid
Fuel system	Port injection

measured in 2nd cylinder (quartz sensor AVL 8QP505. in angular domain using optical incremental encoder with angular resolution of 1° CA positioned at the front end of the crankshaft.

TDC phase shift and pressure sensor offset were identified simultaneously from pressure recordings for cycles without combustion using LSQ minimisation. Pressure signal for cycles with combustion was corrected for phase shift, and pressure sensor offset was identified simultaneously with combustion model parameters, minimizing the LSQOF for measured and modelled in-cylinder pressure.

Angular speed was measured using Hall sensor at the flywheel with angular resolution of 3° CA. The angular speed signal was corrected for flywheel run-out using the method proposed by Kiencke [29]. The run-out and its angular position with respect to reference point were identified simultaneously minimizing the OF for modelled and measured angular speed during cranking. The tooth pitch was corrected using method proposed by Ferenbach *et al.* [11-13]. The signal was post processed by interpolating to the domain of pressure measurement (resolution of 1° CA), averaging on a sequence of cycles and then smoothed using cubic spline approximation in order to retain differentiability up to 2nd order.

The compression ratio was identified using angular speed measurement during the cranking of the fully warmed engine. The identified value of 9.18 was used then to provide complete combustion analysis based on in-cylinder pressure and instantaneous angular speed measurement. The results presented here as an example are obtained for the same steady-state operating point (n = 2990 min^{-1}, WOT). Angular speed deviation (fig. 10) still exists (~–0.002%) however, its character differs and descending trend in identified angular speed can be observed. Identified angular deceleration, obviously, arises from torque imbalance caused by higher friction losses obtained from model used here.

The results of Wiebe model parameters identification based on in-cylinder pressure and angular speed measurement are presented in fig. 11. The identification based on pressure measurement gives combustion duration of 63.9° CA (*e. g.* combined Bonatesta-Lindstrom model gives 63.4° CA), and form factor of 4.09 (*e. g.* model Bonatesta gives value 4.03). The

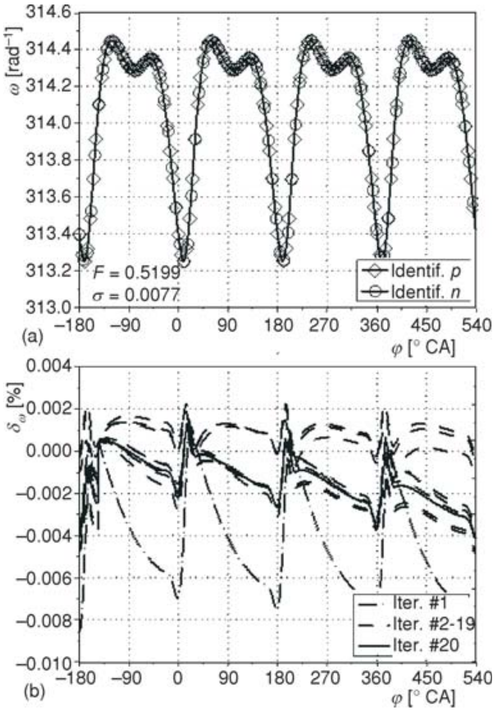

Figure 10. Angular speed identification: measured and identified angular speed (a) and relative deviation during iterative optimisation (b) (n = 2990 min $^{-1}$, WOT)

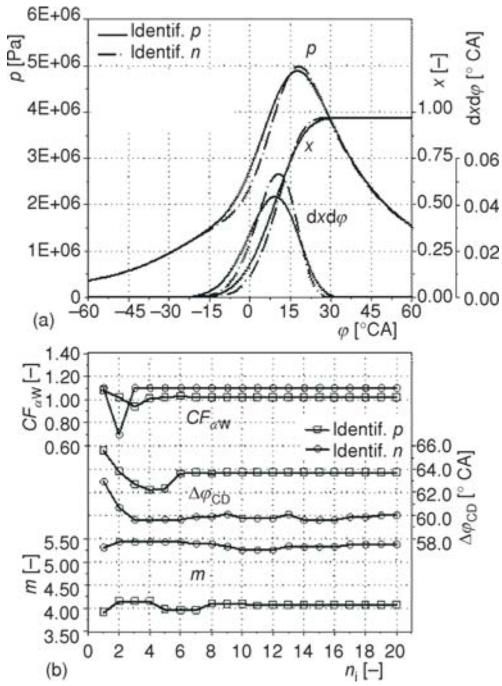

Figure 11. Model parameers identification: pressure, MFB, and RoHR (a), correction factor of the heat transfer coefficient, combustion duration and form factor (n = 2990 min^{-1}, WOT)

same procedure, based on angular speed measurement gives shorter combustion angle of 60.1° CA (error 5.27%) and form factor of 5.4 (error +32.03%).The heat transfer coefficient is identified with deviation of app. +1.9% based on pressure measurement, while using angular speed measurement, deviation rises up to +11%.

In order to compensate higher values of friction losses obtained through combined Taraza-Teodorescu model and continuously descending character of instantaneous angular speed, LSQ minimisation forces combustion model parameters identification in such manner as to minimize the influence of gas pressure. The gas pressure contributes predominantly the friction forces in piston ring-cylinder contact during the very start of expansion stroke but friction torque is only slightly affected because its tangential component is small. In the case of piston skirt-cylinder contact, gas pressure contributes largely the friction force during expansion mid stroke through normal component, but in this case tangential component of the friction force approaches highest values and so affects friction torque significantly. Having in mind these aspects, the optimisation routine acts expectedly, as to minimize gas pressure influence during the expansion mid stroke by shifting the heat release during combustion towards TDC through shortening combustion duration and by increasing heat loss through higher value of heat transfer coefficient. The analysis shows that small uncertainties in friction losses model affects largely the combustion process identification and even the application of highly sophisticated, detailed angle resolved friction losses model can not provide sufficient accuracy required for indirect combustion identification based on instantaneous angular speed. Friction losses superimposed to inertia effects and crankshaft torsional deformation appear as a transfer function between cylinder process and crankshaft speed and therefore, largely affect the optimisation process and results of indirect combustion analysis. These phenomena must be encountered in identification process as well, in order to provide reliable results comparable to direct combustion analysis method used as reference. However, the approach and results presented in this work shows potentials for fast and cost effective combustion analysis based on nonintrusive, already available measurement of instantaneous crankshaft angular speed.

Conclusions

Comprehensive, non-linear, angle-resolved model for combustion, engine friction and engine dynamics is presented. The set of models is employed to perform detailed parametric sensitivity analysis concerning the influences of Wiebe model parameters, heat transfer coefficient, and compression ratio on instantaneous crankshaft angular speed. The combustion duration appears as the most influential model parameter, giving maximal deviations of ±0.04% for varied combustion angles within the range of ±10%. The form factor, varied within ±10%, produces deviations of as high as ±0.01%. The heat transfer coefficient is less influential having in mind that variation was performed within ±20% in respect to the reference value. The compression ratio produces highest deviations around each TDC, and so affects combustion and instantaneous angular speed. Levenberg-Marquardt BCO for LSQOF was applied to investigate numerically the influence of uncertainties in compression ratio values on model parameters identification accuracy. In spite of the compression ratio error of 2%, the results of the instantaneous angular speed identification are positive, approaching error of 0.003%. Large deviations in combustion model parameters were observed: +10% for Wiebe form factor, +2.3% for combustion duration angle and +9% for heat transfer coefficient. Simulation shows that angular speed during engine cranking can be used to identify the compression ratio. Experimental validation shows that regardless detailed modelling, high accuracy in combustion model parameters identification still can't be guaranteed without systematic approach. Uncertainties in applied friction models appear influential, and therefore must be subjected to identification, as well. However, procedure presented in this work, proved sufficiently sensitive and robust, and with minor improvements could be used as alternative method for combustion analysis.

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Nomenclature

A – area, surface area, [m^2]
A_w – surface area of the wall, [m^2]
a – combustion efficiency parameter, [–]
c_m – piston mean velocity, [ms^{-1}]
J_E – engine mass moment of inertia, [kgm^2]
J_D – dynamometer mass moment of inertia, [kgm^2]
J_{FW} – flywheel mass moment of inertia, [kgm^2]
J_S – shaft mass moment of inertia, [kgm^2]
m – mass, [kg], Wiebe form factor, [–]
m_b – mass of burned products, [kg]
m_l – mass lost through crevices, [kg]
m_u – mass of unburned mixture, [kg]
n – engine speed, [min^{-1}]
p – pressure, [Pa]
Q – heat, [J]
Q_c – heat released by combustion, [J]
Q_w – heat transferred to the walls, [J]
Q_l – heat lost through crevices [J]
R – special gas constant, [Jkg^{-1}K^{-1}]
T – temperature, [K]; torque [Nm]
T_b – temperature of burned gas, [K]

T_F – friction torque, [Nm]
T_G – gas pressure torque, [Nm]
T_L – load torque, [Nm]
T_u – temperature of unburned mixture, [K]
T_w – wall temperature, [K]
U – internal energy, [J]
u – specific internal energy, [Jkg^{-1}]
V – volume, [m^3]
v – specific volume, [m^3kg^{-1}]
x – normalized heat released, [–]
x_{RG} – mass fraction of residual gases, [–]

Greek symbols

α_W – heat transfer coefficient, [$Jm^{-2}K^{-1}$]
ε – compression ratio, [–]
φ – crank angle, [deg. CA]
φ_{IGN} – ignition advance, [deg. CA]
φ_{SOC} – start of combustion, [deg. CA]
$\Delta\varphi_{CD}$ – combustion duration, [deg. CA]
ω – angular speed, [rad^{-1}]

Acronyms

CI – compression ignition
MFB – mass fraction burned
IC – internal combustion
IMEP – indicated mean effective pressure
RoHR – rate of heat release
SI – spark ignition
TDC – top dead centre
LSQ – least squares
LSQF – least squares objective function
WOT – wide open throtle

References

[1] Tomić, M., *et al*., A Quick, Simplified Approach to the Evaluation of Combustion Rate from an Internal Combustion Engine Indicator Diagram, *Thermal Science, 12* (2008), 1, pp. 85-102
[2] Hohenberg, G., Killman, I., Basic Findings Obtained from Measurement of the Combustion Process, 19th Fisita Congress, Melbourne, Australia, 1982, paper 8216
[3] Brunt, M. F. J., Pond, C. R., Evaluation of Techniques for Absolute Cylinder Pressure Correction, SAE International Congress & Exposition, Detroit, Mich., USA, SAE paper 970036, 1997
[4] Tomić, M., The Identification of Parameters of a Diesel Engine Thermodynamic and Flow Processes Mathematical Model (in Serbian), Ph. D. thesis, Faculty of Mechanical Engineering, University of Belgrade, Belgrade, 1987
[5] Lindstrom, F., Empirical Combustion Modeling in SI Engines, Ph. D. thesis, Machine Design, Royal Institute of Technology (KTH), Stockholm, 2005
[6] Tunestal, P., Self-Tuning Gross Heat Release Computation for Internal Combustion Engines, *Control Engineering Practice, 17* (2009), 4, pp. 518-524
[7] Andersson, I., *et al*., The Torque Ratio Concept for Combustion Monitoring of Internal Combustion Engines, *Control Engineering Practice, 20* (2012), 6, pp. 561-568
[8] Schagerberg, S., McKelvey, T., Instantaneous Crankshaft Torque Measurements – Modeling and Validation, SAE 2003 World Congress & Exhibition, Detroit, Mich., USA, 2000, Session: Electronic Engine Controls SAE technical paper 2003-01-0713
[9] Hamedović, H, *et al*., In-Cylinder Pressure Reconstruction for Multi-Cylinder SI-Engine by Combined Processing of Speed and One Cylinder Pressure, *Proceedings,* International Conference on Acoustics, Speech, and Signal Processing, 2005, IEEE, 2005
[10] Raichle, F., *et al*., Method and Device for Detecting the Angular Position Signal in an Internal Combustion Engine (in German), U. S. patent WO 2005/083252 A1, 2005
[11] Fehrenbach, H., *et al*., Determination of Engine Torque from the Angular Speed Signal (in German) *Motortechnische Zeitschrift – MTZ, 63* (2002), 12, pp. 1020-1027
[12] Fehrenbach, H., *et al*., Torque Determination of Internal Combustion Engines by Evaluating the Crankshaft Angular Velocity (in German), *Motortechnische Zeitschrift – MTZ. 59* (1998), 5, pp. 314-319
[13] Fehrenbach, H., *et al*., Compensation of the Encoder Wheel Error when Driving (in German), *Motortechnische Zeitschrift – MTZ. 63* (2002), 7-8, pp. 588-591
[14] Weißenborn, E., *et al*., Adaptation of a Zero-Dimensional Cylinder Pressure Model for Diesel Engines Using the Crankshaft Rotational Speed, *Mechanical Systems and Signal Processing, 25* (2011), 6, pp. 1887-1910
[15] Pischinger, R., *et al*., Thermodynamik der Verbrenungsmaschinen (Thermodynamics of Internal Combustion Engines – in German), Springer-Verlag, Wien, 2002
[16] Gordon, S., McBride, B. J. Computer Program for Calculation of Complex Chemical Equilibrium Compositions and Applications: Part I – Analysis, NASA reference publication, Report No. RP-1311-P1

[17] Olikara, C., Borman, G. L., A Computer Program for Calculating Properties of Equilibrium Combustion Products with Some Applications to IC Engines, SAE paper 750468, 1975
[18] Wannatong, K., Simulation Algorithm for Piston Ring Dynamics, *Simulation Modelling Practice and Theory, 16* (2008), 1, pp. 127-146
[19] Bonatesta, F., *et al*., Burn Angles and Form Factors for Wiebe Function Fits to Mass Fraction Burned Curves of a Spark Ignition Engine with Variable Valve Timing, *International Journal of Engine Research, 11* (2010), 2, pp. 177-186
[20] Heywood, J. B., *Internal Combustion Engine Fundamentals*, McGraw-Hill Inc., New York, USA, 1988
[21] Woschni, G., Fieger, J., Experimental Determination of the Average Local Heat Transfer Coefficients in a Gasoline Engine (in German), *Motortechnische Zeitschrift - MTZ, 42* (1981), 6, pp. 229-234
[22] Hohenberg, G., Calculation of the Gas Heat Transfer in Diesel Engines (in German), *Motortechnische Zeitschrift – MTZ, 41* (1980), 7/8, pp. 321-326
[23] Chang, J., *et al*., New Heat Transfer Correlation for an HCCI Engine Derived from Measurements of Instantaneous Surface Heat Flux, Powertrain & Fluid Systems Conference & Exhibition, Tampa, Fla., USA, 2004, Session: Homogeneous Charge Compression Ignition, SAE paper 2004-01-2996
[24] Cho, K.-W., *et al*., Experimental Investigation of Combustion and Heat Transfer in a Direct-Injection Spark Ignition Engine via Instantaneous Combustion Chamber Surface Temperature Measurements, *Proceedings*, Institution of Mechanical Engineers, *Part D: Journal of Automobile Engineering, 222* (2008), 11, pp. 2219-2233
[25] Wang, X., *et al.*, Heat Release and Heat Flux in a Spray-Guided Direct-Injection Gasoline Engine, *Proceedings*, Institution of Mechanical Engineers, *Part D: Journal of Automobile Engineering, 221* (2007), 11, pp. 1441-1452
[26] Taraza, D., Possibilities to Reconstruct Indicator Diagrams by Analysis of the Angular Motion of the Crankshaft, International Off-Highway & Powerplant Congress & Exposition, Milwaukee, Wis., USA, 1993, Session: International Off-Highway & Powerplant Congress & Exposition, SAE technical papers, SAE paper 932414
[27] Kallenberger, C., Hamedović, H., Evaluation of Torque Estimation Using Gray-Box and Physical Crankshaft Modeling, *Proceedings*, 15th European Signal Processing Conference (EUSIPCO 2007), Poznan, Poland, 2007, pp. 1529-1532
[28] Kallenberger, C., Hamedović, H., Comparison of the Extended Kalman Filter and the Unscented Kalman Filter for Parameter Estimation in Combustion Engines, *Proceedings*, 15th European Signal Processing Conference (EUSIPCO 2007), Poznan, Poland, 2007, pp. 2424-2428
[29] Kiencke, U., Engine Misfire Detection, *Control Engineering Practice, 7* (1999), 2, pp. 203-208
[30] Chen, S. X., Moskwa, J. J., Application of Nonlinear Sliding-Mode Observers for Cylinder Pressure Reconstruction, *Control Engineering Practice, 5* (1997), 8, pp. 1115-1121
[31] Filipi, Z. S., Assanis, D. N., A Nonlinear, Transient, Single-Cylinder Diesel Engine Simulation for Predictions of Instantaneous Engine Speed and Torque, *Journal of Engineering for Gas Turbines and Power, 123* (2001), 4, pp. 951-959
[32] Connolly, F. T., Yagle, A. E., Modeling and Identification of the Combustion Pressure Process in Internal Combustion Engines?: *II Experimental Results. Journal of Engineering for Gas Turbines and Power, 115* (1993), 10, pp. 801-809
[33] Hafner, K. E., *Theory of Engine Vibration of Internal Combustion Engine*, Springer Verlag, Wien, 1984
[34] Taylor, C. M., *Engine Tribology: Tribology Series*, Elsevier Science Publishers B.V., Amsterdam, Netherlands, 1993
[35] Taraza, D., *et al*., Friction Losses in Multi-Cylinder Diesel Engines, SAE World Congress, Detroit, Mich., USA, 2000, Session: CI & SI Power Cylinder Systems, SAE technical paper 2000-01-0921, 2000
[36] Stachowiak, G. W., Batchelor, A. W., *Engineering Tribology*, Butterworth-Heinemann, Elsevier Inc. , 2006
[37] Teodorescu, M., *et al*., Simplified Elasto-Hydrodynamic Friction Model of the Cam-Tappet Contact, SAE World Congress & Exhibition, Detroit, Mich., USA, SAE technical paper 2003-01-0985, 2003

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