International Congress Motor Vehicles & Motors 2012

Kragujevac, October 3rd-5th, 2012

MVM2012-032

Slobodan Popović[1]
Miroljub Tomić[2], Nenad Miljić[3], Marko Kitanović[4], Predrag Mrđa[5],

THE INFLUENCE OF DYNAMIC ENGINE MODEL PARAMETERS ON CRANKSHAFT INSTANTANEOUS ANGULAR SPEED – SENSITIVITY AND ERROR ANALYSIS

ABSTRACT: Most frequently used approach to describe combustion process in IC engines is based on Wiebe function. Its parameters are strongly influenced by geometric and in-cylinder gas- and thermodynamic processes and can't be predicted easily and with sufficient accuracy using available empirical relations. Along with deterministic methods lacking in systematic approach, indirect analysis methods based on minimization of error between simulated and measured in-cylinder pressure have been successfully demonstrated. In this paper the same concept is introduced to identify combustion model parameters of a multi-cylinder SI engine based on box constrained Levenberg-Marquardt minimization of nonlinear Least Squares (LSQ) given for measured and simulated instantaneous crankshaft angular speed determined from the solution of the engine dynamics torque balance equation. For predictions of gas pressure and engine torque, a nonlinear, detailed, angle resolved two-zone, zero-dimensional multi-cylinder SI engine combustion model has been developed as well as a detailed analytical component model of engine friction and mechanical losses. The models are used to analyse the influences of engine combustion and dynamics model parameters on instantaneous crankshaft speed and to investigate biases in measurement in order to recognize and establish the constraints in process of combustion model parameters identification.

KEYWORDS: Spark Ignition Engine, two-zone model, mechanical losses, instantaneous crankshaft speed, optimization

INTRODUCTION

Combustion analysis, comprising most commonly the determination of burn angles, normalized variable of Mass Fraction Burned (MFB), and the Rate of Heat Release (RoHR) is crucial when it comes to engine development and design. In order to quantify combustion parameters in IC engines, in-cylinder pressure measurement and analysis based upon First Law of Thermodynamics are required. Depending on accuracy level and details required, the analysis includes the additional sub-models for the effects of heat transfer, residual fraction, gas leakage through crevices and gas properties. This method is referred to as direct method because of the fact that solution of the equation for the First Law of Thermodynamics applied to the open thermodynamic system represented by engine

[1] *Slobodan Popović, Research and Teaching Assistant, Internal Combustion Engines Department, Faculty of Mechanical Engineering, University of Belgrade, Kraljice Marije 16, 11120 Belgrade 35, Serbia, spopovic@mas.bg.ac.rs*
[2] *Miroljub Tomić, Head of Department, Internal Combustion Engines Department, Faculty of Mechanical Engineering, University of Belgrade, Kraljice Marije 16, 11120 Belgrade 35, Serbia, mtomic@mas.bg.ac.rs*
[3] *Nenad Miljić, Research and Teaching Assistant, Internal Combustion Engines Department, Faculty of Mechanical Engineering, University of Belgrade, Kraljice Marije 16, 11120 Belgrade 35, Serbia, nmiljic@mas.bg.ac.rs*
[4] *Marko Kitanović, Research Assistant, Internal Combustion Engines Department, Faculty of Mechanical Engineering, University of Belgrade, Kraljice Marije 16, 11120 Belgrade 35, Serbia, mkitanovic@mas.bg.ac.rs*
[5] *Predrag Mrdja, Research Assistant, Internal Combustion Engines Department, Faculty of Mechanical Engineering, University of Belgrade, Kraljice Marije 16, 11120 Belgrade 35, Serbia, pmrdja@mas.bg.ac.rs*

combustion chamber provides the value of RoHR directly. The authors have previously demonstrated simplified deterministic approach [1] regarding both SI and CI engines and more information can be also found in literature [2,3]. The crucial drawback comes from the fact that the procedure is heavily affected by number of uncertainties whose can't be identified explicitly. This mainly applies to the determination of compression ratio and errors introduced by measurement (pressure sensor offset and pressure trace-angle phase shift) and heat transfer and gas leakage models calibration.

When it comes to process simulation, parametric model capable to describe heat release during combustion is advantageous. The most frequently used model is that proposed by Wiebe which employs simple exponential form to provide values of MFB and RoHR. Two basic model parameters – form factor and combustion duration angle, can be retrieved from combustion analysis but additional computational effort is required. Using stochastic or indirect method, based on minimization of difference between simulated and measured in-cylinder pressure solve this problem intrinsically. Additionally, method provides simultaneous identification of heat transfer model constants. When it comes to high accuracy, real compression ratio value, gas leakage and measurement biases are beneficial and can be identified from pressure trace without combustion. Indirect method has been demonstrated successfully in terms of combustion analysis in CI engines [4] through box constrained Levenberg-Marquardt minimization of nonlinear Least Squares (LSQ) given for measured and simulated in-cylinder pressure. The same principle was demonstrated in respect to Si engines, utilizing Newton-Gauss minimization algorithm [5,6].

The possibility to obtain combustion analysis from instantaneous crank shaft speed signal is extremely attractive and beneficial considering relative simplicity of nonintrusive, already available measurement compared to impractical pressure indicating, particularly in case of mass-production engines. The issues regarding the identification of Wiebe combustion model parameters based on crankshaft instantaneous angular speed is presented in this work. The indirect method presented here employs box constrained Levenberg-Marquardt minimization of nonlinear LSQ constructed upon measured and simulated instantaneous crankshaft angular speed which is determined from the solution of the engine dynamics torque balance equation. The comprehensive two-zone engine combustion model and detailed analytical friction loss model in angular domain has been applied to provide sensitivity and error analysis regarding basic 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.

SIMULATION MODEL

Governing equations

The simulation of SI engine combustion was developed on basis of Zero-Dimensional (0D) Two-Zone (2Z) thermodynamic model which is essentially proposed by Pischinger [7].The model is designed upon following assumptions: the process is instationary; 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 The First Law of Thermodynamics for open systems applied to the cylinder volume. Introducing index i to denote zones of unburned mixture (u) and burned combustion products (b), equation in angular domain φ is given in general differential form for both zones:

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

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

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

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 in eq. (3):

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

The right side represents the sum of heat loss through the cylinder walls ($dQ_w/d\varphi$), the change of gas enthalpy due to combustion ($h_u\cdot dm_c/d\varphi$), gas leakage ($h_i\cdot dm_{l,i}/d\varphi$) and flow through intake ($h_{o,in,i}\cdot dm_{in,i}/d\varphi\cdot$) and exhaust valve ports ($h_{o,exh,i}\cdot dm_{exh,i}$). Gas temperature differential equation for both zones 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 to pressure and temperature into eq. (3):

$$\frac{dT_i}{d\varphi}=\frac{\frac{dQ_i}{d\varphi}-\frac{dm_i}{d\varphi}h_i}{m_i\left(\frac{\partial u_i}{\partial T_i}+T_i\frac{\partial R_i}{\partial T_i}+R_i\right)}+\frac{\frac{dp}{d\varphi}\left(\frac{R_iT_i}{p}-T_i\frac{\partial R_i}{\partial p}-\frac{\partial u_i}{\partial p}\right)}{\left(\frac{\partial u_i}{\partial T_i}+T_i\frac{\partial R_i}{\partial T_i}+R_i\right)} \tag{4}$$

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

$$m=m_b+m_u=xm+(1-x)m \tag{5}$$

$$\frac{d}{d\varphi}\left(\frac{V}{m}\right)=\frac{dV}{d\varphi}\cdot\frac{1}{m}-\frac{V}{m^2}\cdot\frac{dm}{d\varphi}=\frac{dx}{d\varphi}\cdot v_b+x\cdot\frac{dv_b}{d\varphi}-\frac{dx}{d\varphi}\cdot v_u+(1-x)\cdot\frac{dv_u}{d\varphi} \tag{6}$$

where ratio $x=m_b/m$ represents the mass fraction of burned mixture. Assuming homogenous and equal pressure in both zones, differential equation for cylinder charge pressure can be obtained from eq. (6) introducing eq. (4) and partial derivatives of specific volume with respect to pressure and temperature:

$$\frac{dp}{d\varphi}=\frac{\frac{dV}{d\varphi}\frac{1}{m}-\frac{V}{m^2}\frac{dm}{d\varphi}-\frac{dx}{d\varphi}\left(\frac{R_bT_b-R_uT_u}{p}\right)-\frac{\partial v_b}{\partial T_b}\left[\frac{\frac{dQ_b}{d\varphi}-\frac{dm_b}{d\varphi}h_b}{m\left(\frac{\partial u_b}{\partial T_b}+T_b\frac{\partial R_b}{\partial T_b}+R_b\right)}\right]+\frac{\partial v_u}{\partial T_u}\left[\frac{\frac{dQ_u}{d\varphi}-\frac{dm_u}{d\varphi}h_u}{m\left(\frac{\partial u_u}{\partial T_u}+T_u\frac{\partial R_u}{\partial T_u}+R_u\right)}\right]}{x\frac{\partial v_b}{\partial T_b}\frac{\left(\frac{R_bT_b}{p}-T_b\frac{\partial R_b}{\partial p}-\frac{\partial u_b}{\partial p}\right)}{\left(\frac{\partial u_b}{\partial T_b}+T_b\frac{\partial R_b}{\partial T_b}+R_b\right)}+(1-x)\frac{\partial v_u}{\partial T_u}\frac{\left(\frac{R_uT_u}{p}-T_u\frac{\partial R_u}{\partial p}-\frac{\partial u_u}{\partial p}\right)}{\left(\frac{\partial u_u}{\partial T_u}+T_u\frac{\partial R_u}{\partial T_u}+R_u\right)}} \tag{7}$$

Thermodynamic properties of the mixture and combustion products are calculated approximately using NASA 9-coefficients polynomials [8], while species molar concentrations are calculated by means of 12 component Olikara-Borman [9] model assuming equilibrium composition. One-dimensional, quasi-steady, compressible isentropic flow was assumed to model instantaneous gas mass flow through valves and crevices. Discharge coefficients for both intake and exhaust valves were determined experimentally, using flow test bench. Regarding gas flow through crevices revised approach based on piston ring gap area and pressure related discharge coefficient formula proposed by Wannatong [10] is used.

Heat release model

Wiebe heat release model provides the fraction of burned mixture x in simple exponential form:

$$x(\varphi)=1-exp\left[-a\cdot\left(\frac{\varphi-\varphi_{SOC}}{\Delta\varphi_{CD}}\right)^{m+1}\right] \tag{8}$$

Considering SI engines, the start of combustion (φ_{SOC}) can be regarded as ignition advance angle (φ_{IGN}) which is generally known variable. In order to obtain heat release during combustion, only form factor m and combustion duration $\Delta\varphi_{CD}$ must be identified. Initial guess and box constraints required to initialize optimization procedure can be obtained using Csallner, Witt or Lindström [5] correlations provided previously identified model parameters for reference operating point. More practical approximate approach by Bonatesta *et al.* [11] is used here 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}):

$$m=3.46\cdot c_m^{-0.225}\cdot\left(1+\sqrt{\varphi_{IGN}}\right)^{0.35}\cdot(1-1.28\cdot x_{RG})-1 \tag{9}$$

Combustion duration $\Delta\varphi_{CD}$ can be approximated using simple linear correlation proposed by Lindström [5]:

$$\Delta\varphi_{CD}=23.7529\cdot\left(3.51\cdot10^{-4}\cdot n+5.71\text{-}m\right) \tag{10}$$

In order to encounter for reach or lean mixture operation, form factor correction proposed by Lindström [5] which is based on Blizard and Keck [12] model for laminar flame speed is used.

Heat transfer model

Heat transfer from the cylinder charge to the walls is modelled generally by means of Newton's law of convective heat transfer. Regarding 0D2Z model, Newton's equation is given as follows

$$\frac{dQ_{w,i}}{dt}=\alpha_{w,i}\cdot\sum_{j=1}^{n}A_{w,i,j}\cdot\left(T_i\text{-}T_{w,j}\right) \tag{11}$$

where index i denotes corresponding zones (u and b) and index j denotes surfaces of the piston crown, cylinder liner and cylinder head, respectively. Instantaneous heat transfer coefficient α_w introduces a great number of issues being a complex function of instantaneous pressure, temperature, engine speed and geometric and flow characteristics of combustion chamber and so affects global accuracy of gas pressure and temperature prediction. A wealth of empirical relations for instantaneous heat transfer coefficient have been proposed in the past relying generally on dimensional analysis for turbulent flow correlating Nusselt, Prandtl and Reynolds numbers, however only few proved to be sufficiently universal and reliable (*e.g.* Annand [12], Woschni [13] or Hohenberg [14]). Here, basic Woschni model [13], recently revised by Chang *et al.* [15] is used. Although adapted for HCCI combustion system, this model was applied for Direct Injection SI engines (Cho *et al.* [16]) and validated for homogenous charge operation (Wang *et al.* [17]) as well. Instead of cylinder diameter D as characteristic length in original Woschni equation, instantaneous combustion chamber height $H_{cc}(\varphi)$ is used to compensate under-predicted heat flux during the late stage of compression previously reported in literature [15–17]. Gas temperature influence has been decreased by optimizing exponent (–0,73 instead of –0,53). The influence of engine load on charge flow velocity were observed too strong in original model and therefore, corrected through term C_{w2} by scaling factor of 1/6. Chang's equation is given as follows:

$$\alpha_w=0.0130H_{CC}(\varphi)^{-0.2}p^{0.8}T^{-0.73}\left[C_{w1}c_m+\frac{C_{w2}}{6}\frac{V_hT_1}{p_1V_1}(p-p_m)\right]^{0.8} \tag{12}$$

The revised expression corresponds to high-pressure phase while for gas exchange process, original model setup proposed by Woschni [13] must be used. Surface temperatures $T_{w,j}$, are initiated by means of Atkins-French approximate model and further refined through iterative numeric process assuming spatially averaged values.

Engine dynamics and friction losses model

In the most ideal case which implies the crankshaft as a rigid body, angular speed and acceleration would be influenced by gas pressure, friction losses, inertia and external load torque. The engine shafting, however, responds dynamically as the result of variation in excitation torques changing the character of angular speed variations. Elastic engine shafting models are widely used to simulate angular speed, but issues related to insufficiently accurate prediction were reported and addressed to simplified or inconsistent friction modelling and neglected auxiliaries power consumption [18,19]. The number of sources, however, reference simplified Single Degrees of Freedom (1-DoF) [20–22] or 2-DoF rigid body dynamic models [23].

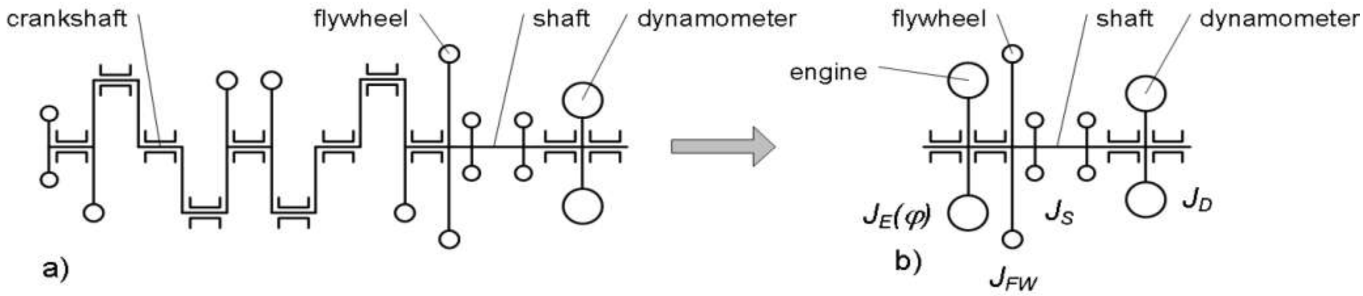

Figure 1 Engine dynamic model (a) and reduced, equivalent 1-DoF dynamic model (b)

In this work, simplified 1-DoF dynamic engine model is applied to simulate crankshaft instantaneous angular speed. Engine-dynamometer dynamic system is presented in Figure 2. Engine mass moment of inertia J_E is assumed as a function of both mass and position of slider mechanism components in respect to shaft angle position. Engine inertia and its first derivative in respect of crank angle are calculated by means of dynamically equivalent model, while inertia of flywheel J_{FW}, connecting shaft J_S and dynamometer J_D are known constants obtained from manufacturing specification. Assuming crank and connecting shafts rigid [21,22], torque balance equation for engine-dynamometer system arises from kinetic energy equation (Newton's principle):

$$[J_E(\varphi)+J_{FW}+J_S+J_D]\cdot\ddot{\varphi}(\varphi)+\frac{1}{2}\cdot\frac{dJ_E(\varphi)}{d\varphi}\cdot\dot{\varphi}(\varphi)^2=T_G(\varphi)-T_F(\varphi)-T_L \quad (13)$$

Gas-pressure torque contributions from individual cylinders are denoted by term T_G while T_L is the measured load torque. The term T_F denotes the sum of torques from mechanical losses in tribological systems and auxiliaries.
The influence of engine friction torque is small in the vicinity of TDC, and neglecting engine friction losses and auxiliaries power consumption seems acceptable if the analysis objective is concentrated mainly on combustion statistical markers (*e.g.* peak cylinder pressure and its position). Angle or time averaged models of mechanical losses (*e.g.* Chen-Flynn [21] or Milington–Hartles [22],) can improve the accuracy, but considering delicate process of combustion model parameters identification, detailed, high sensitivity angle-resolved model of friction losses must be applied. Analytical models based on Reynolds equation are available (*e.g.* Dawson [24]), although cumbersome and demanding, provide universal approach and minimize the influence of empirical constants. In this work, friction losses in piston-cylinder contact are calculated by means of angle-resolved models presented by Taraza [25] 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 for this method is presented by Stachowiak [26], and detailed application in terms of IC engine bearings is presented by Taraza [25]. Cam-tappet contact is assumed to obey the Elasto-Hydrodynamic Lubrication Theory (EHDL) and the friction losses are modelled by means of approximate solution of Reynolds equation presented by Teodorescu [27]. Friction models listed here exceed the scope of this paper and more details can be found in selected literature.

SENSITIVITY ANALYSIS

Series of numerical experiments was conducted in order to gain necessary experience and validate the assumption weather combustion analysis based on instantaneous angular speed signal was possible or not. Preliminary analysis was 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. The analysis was conducted for serial-production port fuel injection petrol engine DMB M202PB13 (technical data given in Table 1)

Table 1 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	52 kW @ 5800 min^{-1}
Cooling system	liquid
Fuel system	Port Injection

Table 2 Reference values and limits of engine model parameters used in sensitivity analysis

Model parameter	**Reference value**	**Limits**
Wiebe comb. model form factor	m_{ref} =4,03 (model Bonatesta [11])	± 5%, ± 10%
Combustion duration	$\varphi_{CD,ref}$ =63,4°CA (model Lindström [5])	± 5%, ± 10%,
Compression ratio	ε_{ref} =9,2 (engine spec., tab. 3)	± 2%, ± 4%
Heat transfer coefficient	$\alpha_{W,ref}$ (model Chen [15])	±10%, ±20%
Gas leakage through crevices	$A_{l,ref}$ (model Wannatong [10])	0.5; 2.0; 5,0; 10,0

In the first instance, numerical experiment was performed by varying individually the values of form factor, combustion duration, compression ratio, heat transfer coefficient and gas leakage in order to analyse engine simulation model sensitivity. Reference values for form factor, combustion duration and heat transfer coefficient were established through models presented in previous sections (Table. 2). 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 character of the relative error in respect to reference condition is similar in cases of Wiebe parameters having in mind that both variables affect combustion process in the same way. The influence of form factor on gas pressure and instantaneous angular speed is presented in Figure 2. For given combustion duration, increasing the form factor causes delayed heat release and reduced losses in the late phase of compression. Therefore, 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 Figure 3. Instantaneous angular speed is observed to be more sensitive to changes in combustion duration, and differences between –0,040 and 0,053% can be observed for combustion duration varied within the range of ±10%.

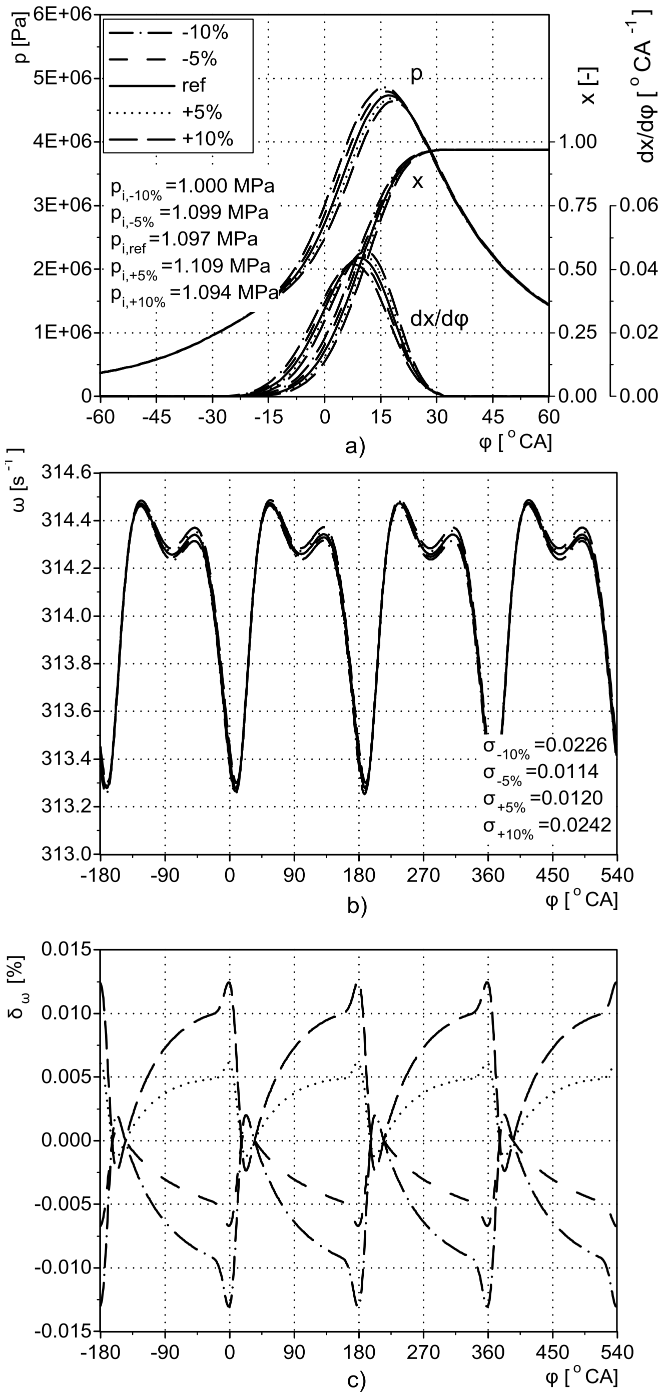

Figure 2 The influence of Wiebe form factor on in-cylinder pressure (a), angular speed (b) and angular speed deviation (c) in respect to reference values

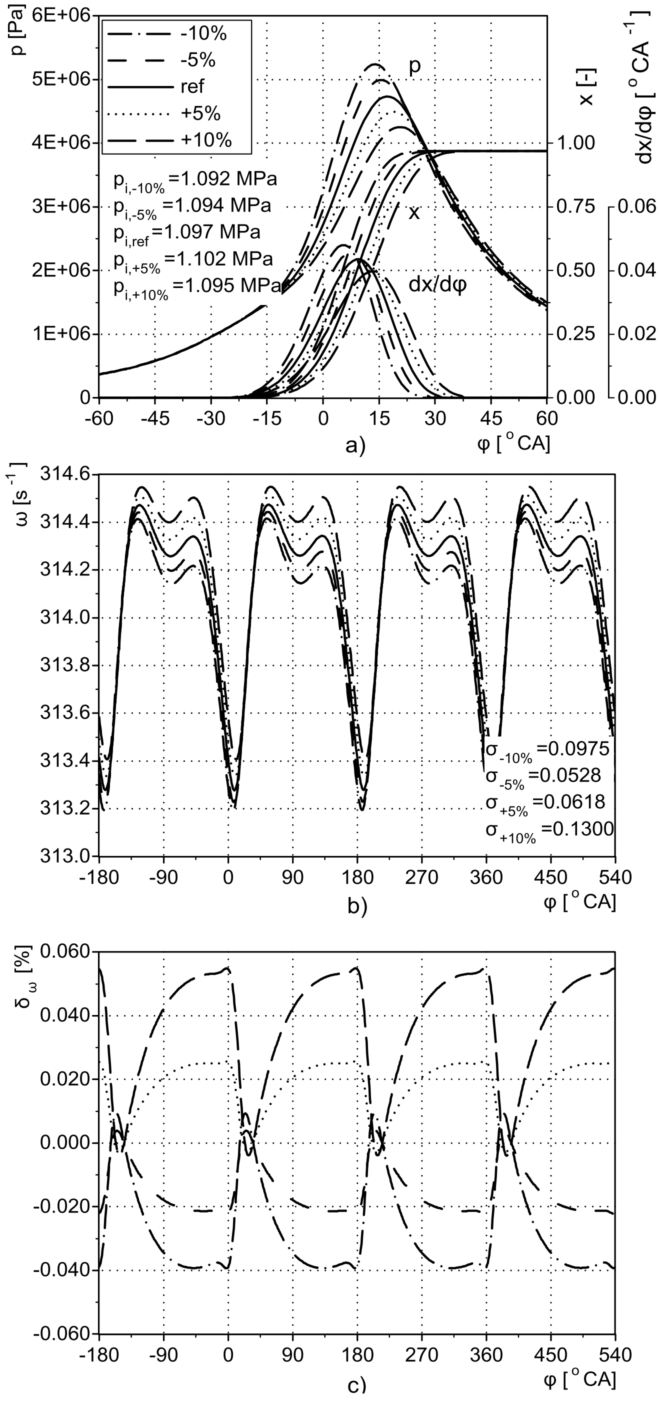

Figure 3 The influence of combustion duration on in-cylinder pressure (a), angular speed (b) and angular speed deviation (c) in respect to reference values

The influence of heat transfer coefficient on instantaneous angular speed is presented in Figure 4 Having in mind 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 Figure 5. 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 app. 0,02%. Between successive TDCs, increase of 0,006% is observed, which is due to positive influence of increased compression ratio on process efficiency.

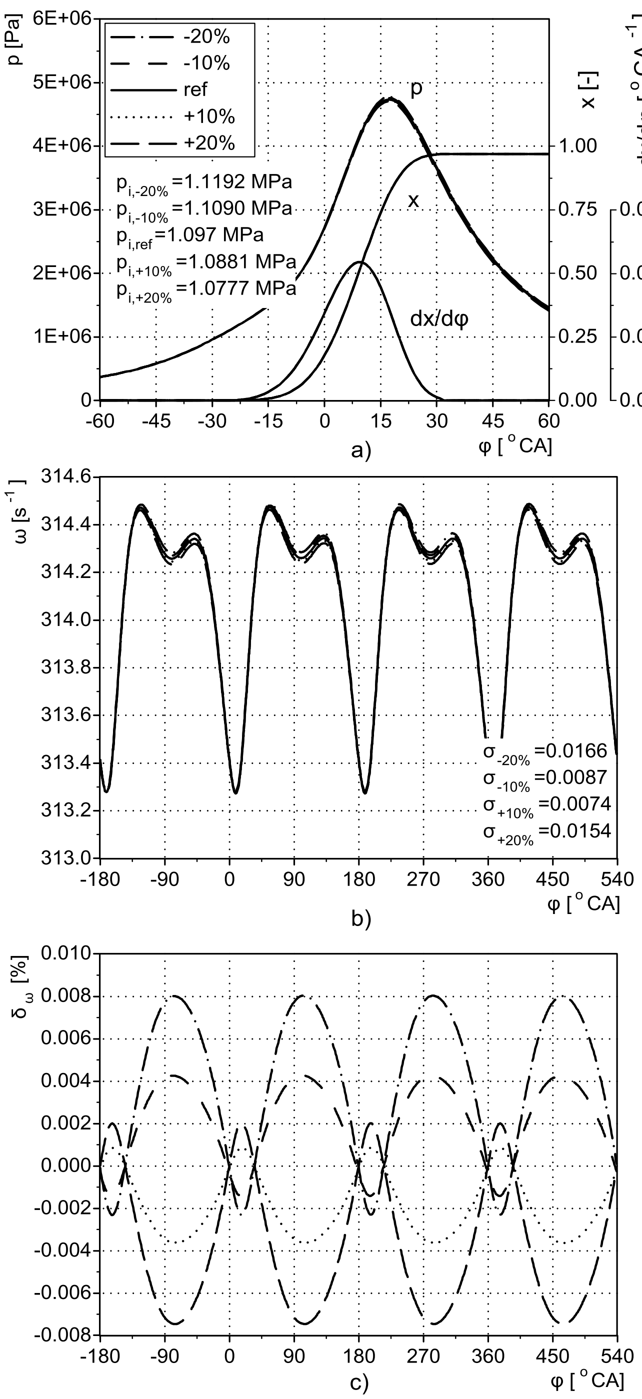

Figure 4 The influence of heat transfer coefficient on in-cylinder pressure (a), angular speed (b) and angular speed deviation (c) in respect to reference values

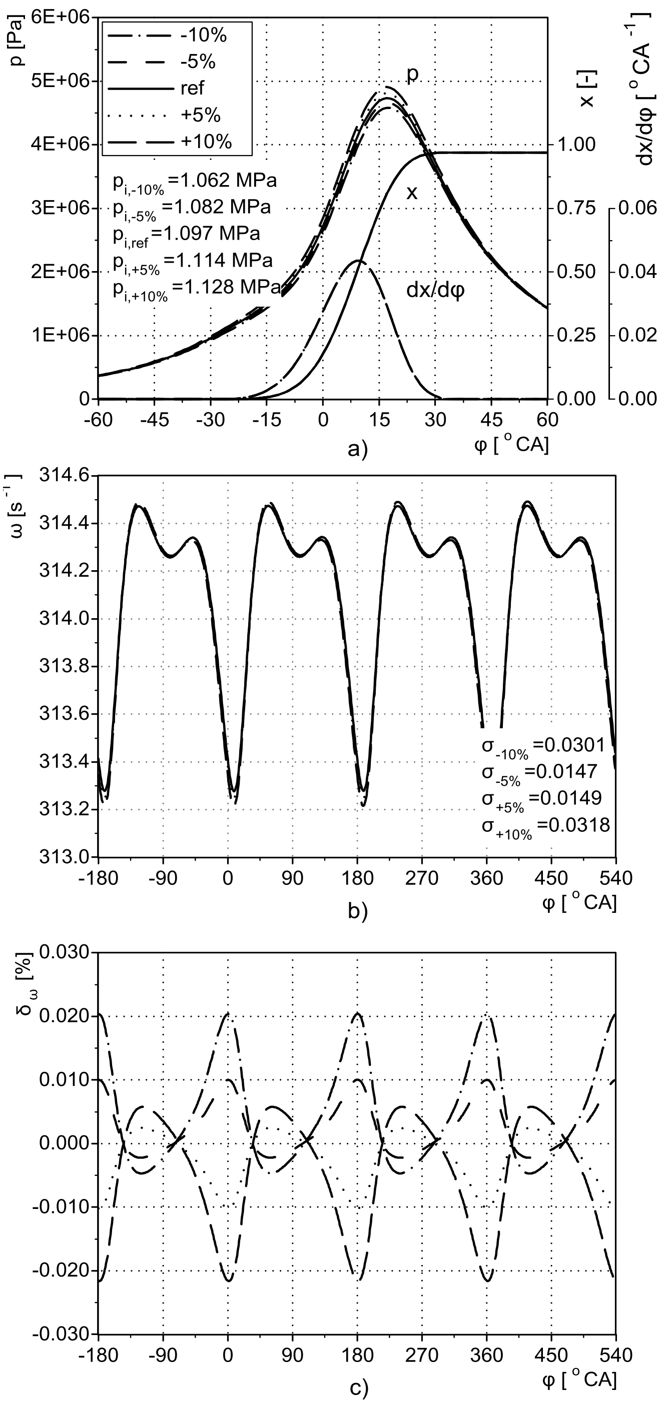

Figure 5 The influence of compression ratio on in-cylinder pressure (a), angular speed (b) and angular speed deviation (c) in respect to reference values

The influence of gas leakage on instantaneous angular speed was considered by varying the crevices effective flow area. The difference in gas pressure is hardly visible (Figure 6a), however increased IMEP by 0,6% is observed for effective flow area decreased for 50%. Increased crevices produce lower values of IMEP (*e.g.* -1.6% for 10 times larger flow area, which is less probable case). Angular speed diagram is purposely omitted because the change in angular speed is hard to observe within the resolution of graphic presentation. However, angular speed deviation exists (Figure 6b), and can be observed around each TDC. This corresponds well with the highest leakage rates obtained in each cylinder during combustion. The gas leakage is, however, 2-10 times less influential than other four model parameters considered in this analysis.

Analysis, designed as to vary one model parameter independently of others affecting the combustion process, does not reflect real conditions because their interrelationships are neglected. The changes of angular speed observed in numerical experiment, are extremely small – 10^{-4} to 10^{-5}, however, the analysis provides valuable conclusions about instantaneous angular speed sensitivity. Sensitivity analysis shows that instantaneous angular speed responds to variations of model parameter considered here and implies that identification procedure based on numerical optimisation is possible. However, compression ratio appears to be influential and must be treated as systematic error.

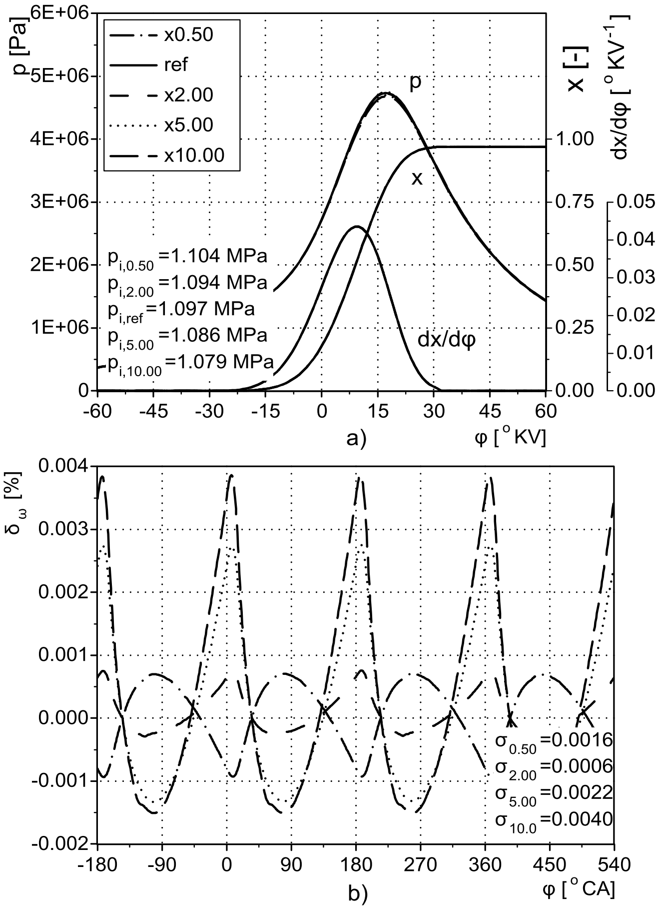

Figure 6 The influence of gas leakage on in-cylinder pressure (a), and angular speed deviation (b) in respect to reference values

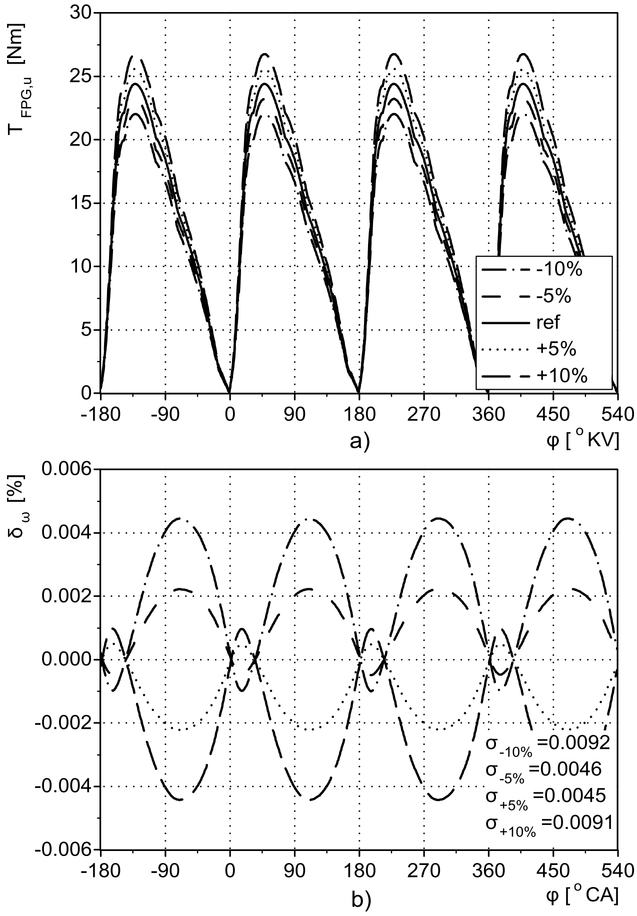

Figure 7 The piston friction losses torque (a) and its influence on angular speed deviation (b) in respect to reference values

The second numerical experiment refers to the influence of friction losses in main tribolgical systems. The analysis was performed in simplified form by varying friction coefficients obtained from previously referenced models within the range of ±10%. The friction torque in piston-cylinder contact and its influence on angular speed are presented in Figure 7. Angular speed is less affected by piston friction in vicinity of TDC (combustion phase) where friction torque diminishes because of crank slider mechanism kinematics. However, the influence is increased during compression and expansion phase where normal component of gas pressure force dominate in friction torque.

The influence of friction losses in crankshaft bearings is displayed in Figure 8. The character of friction torque (Figure 8a) incorporates a small deviation because of superimposed torques from uneven number of bearings which is also observed in the case of angular speed deviation (Figure 8b). The friction in bearings increases during combustion and affects angular speed around each TDC within the range of ±0.0001%.

The friction torque in cam-tappet contact and its influence on crankshaft angular speed is presented in Figure 9. The friction torque resembles sinusoidal form reaching repeatedly highest values in the middle of each stroke (low tangential speed followed by high load in contact). Local peaks observed before and after each TDC correspond to high impact loads at valve opening and closing angles. The influence of cam-tappet friction on angular speed is rather small and falls within the accuracy of integration.

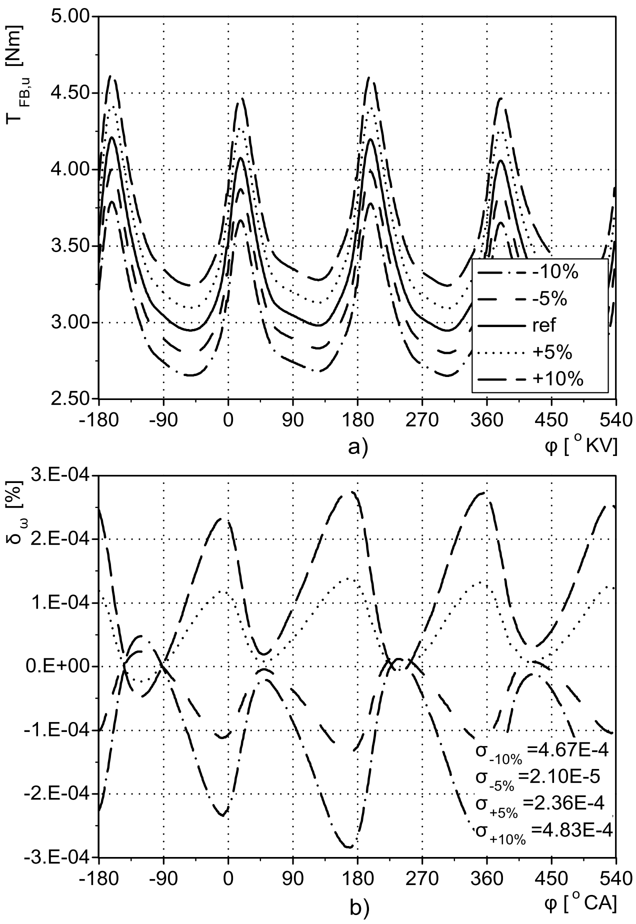

Figure 8 The friction losses torque in crankshaft bearings (a) and its influence on angular speed deviation (b) in respect to reference values

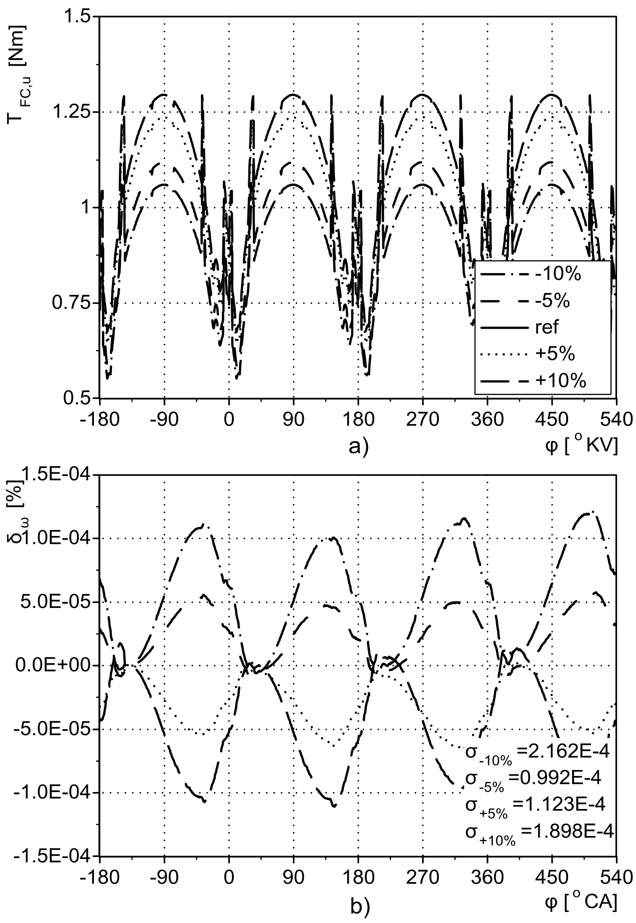

Figure 9 The friction losses torque in cam-tappet contact (a) and its influence on angular speed deviation (b) in respect to reference values

Model parameter identification 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:

$$F(X)=\sum_{i=1}^{N} w_{f,i}\cdot\left[\omega_{e,i}(\varphi_i)-\omega_{m,i}(\varphi_i,X)\right]^2=\sum_{i=1}^{N} w_{f,i}\cdot f_i^2(\varphi_i,X) \tag{14}$$

Colon-matrix X comprises model parameters (in this case $X=[m\ \Delta\varphi_{CD}\ CF_{\alpha W}]^T$), φ is crank angle considered as independent variable, while 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 used to accommodate the importance and/or influence of each individual point acquired by measurement. 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 nonlinear systems than Newton-Gauss method, often regarded as basic in minimization of LSQ [5].

The next numerical experiment refers to uncertainties in identification process expected to appear due to compression ratio systematic error. Cylinder charge pressure and instantaneous angular speed were modelled for assumed real compression ratio deliberately decreased for 2% in respect to reference value in order to encounter usually neglected production tolerances and thermal dilatation of combustion chamber. The boundaries for box constrained optimisation are set in respect to reference values given in Table 1. Heat transfer coefficient was optimised within ±40% (presented through correction factor $CF_{\alpha W}$), form factor m within ±40% and combustion duration within ±10%.

Simulated and identified instantaneous angular speed and its relative error during the iterative optimisation procedure are displayed in Figure 10. 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 absolute difference between simulated and identified angular speed exists, however, it can’t be observed within the resolution of graphical presentation (Figure 10a). The results of combustion model parameters identification (Figure 11) are less satisfactory compared to angular speed, although the difference in IMEP is fairly low (approximately 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 result of the identification of combustion duration appears better than in previous cases reaching relative error of app. 2,3%.

The analysis implies that correction must be applied for the value of compression ratio Therefore, its independent identification must be conducted prior to heat release model identification. Engine cranking can be used to identify compression ratio because compression and expansion processes for fully warmed cylinder can be simulated with high accuracy.

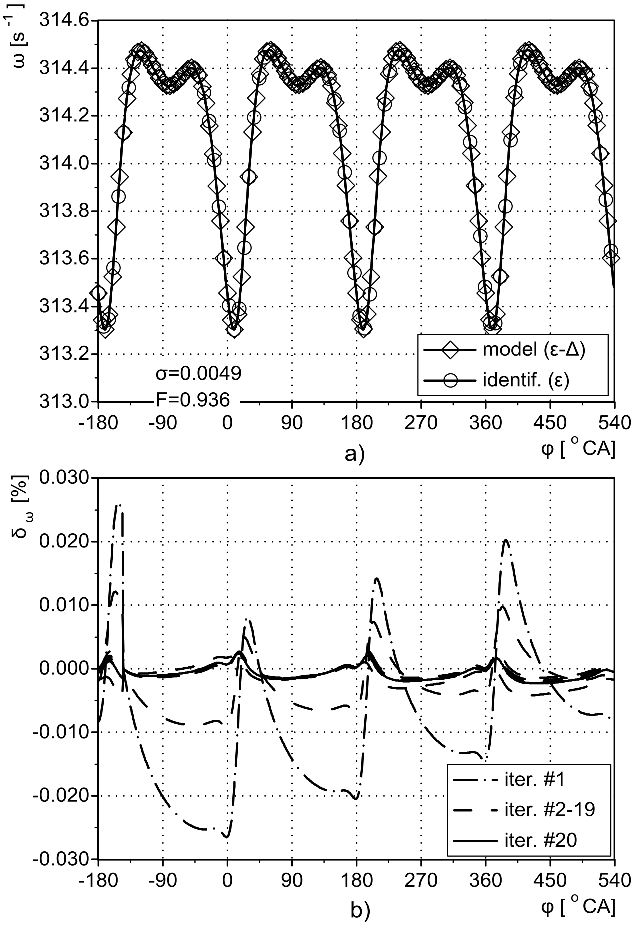

Figure 10 Angular speed identification: modelled and identified angular speed (a) and relative deviation during iterative optimisation (b)

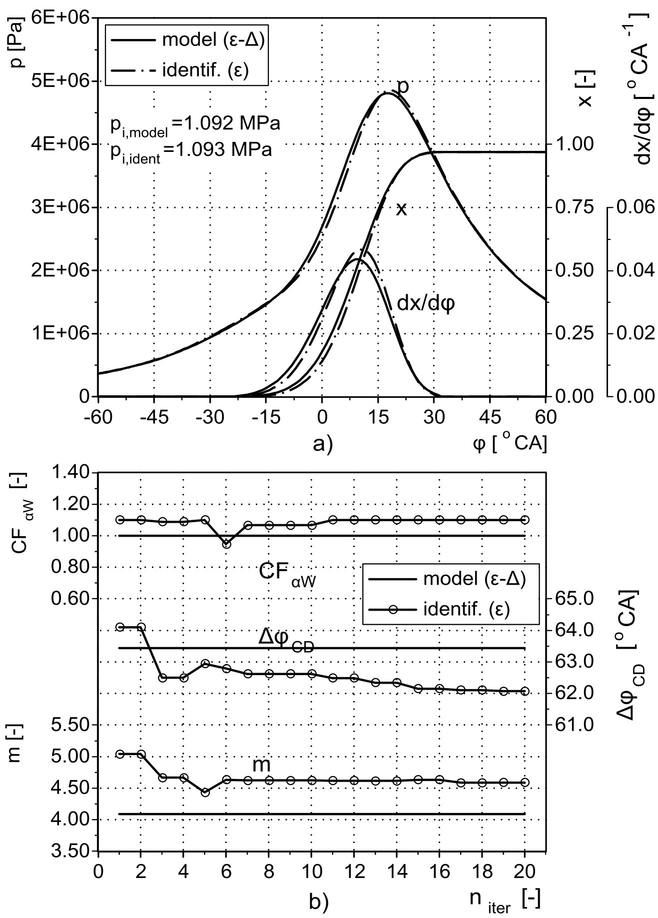

Figure 11 Model parameters identification: pressure, MFB and RoHR (a); correction factor of the heat transfer coefficient, combustion duration and form factor (b)

The influence of mechanical losses is frequently neglected when one comes to modelling of instantaneous crankshaft speed. The numerical approach presented in previous section, however, implies that friction losses, particularly those occurring in piston-cylinder contact s friction (Figure 7), affects crankshaft speed with intensity comparable to that of compression ratio. Therefore it is worthwhile to investigate its influence on combustion model parameters identification. The cylinder pressure was modelled using Chen heat transfer formula and nominal compression ratio value. To encounter the influence of friction in piston-cylinder contact, the crankshaft speed was simulated using hypothetical values of friction coefficient in piston-cylinder contact deliberately scaled down by 10% in respect to the nominal values obtained by Taraza model, while friction losses in bearings and cam-tappet contact remained unchanged. Combustion model parameters were identified assuming nominal values of friction coefficients.

Simulated and identified instantaneous angular speed and its relative error during the iterative optimisation procedure are displayed in Figure 12. Regarding angular speed, identification yields satisfactory results (Figure 12a), however, positive relative error (Figure 12b) shows that friction torque imbalance exists. The results of combustion model parameters identification based on simulated angular speed are presented in analysed in Figure 13. The deviation for all parameters apears significant and beyond real and expected values for the operating point considered in this case. Combustion duration was identified with 68,1 °CA compared to 63,4 °CA obtained through combined Bonatesta-Lindström (+7,5%). Form factor (2.9 compared to modelled 4.03) and heat transfer coefficient

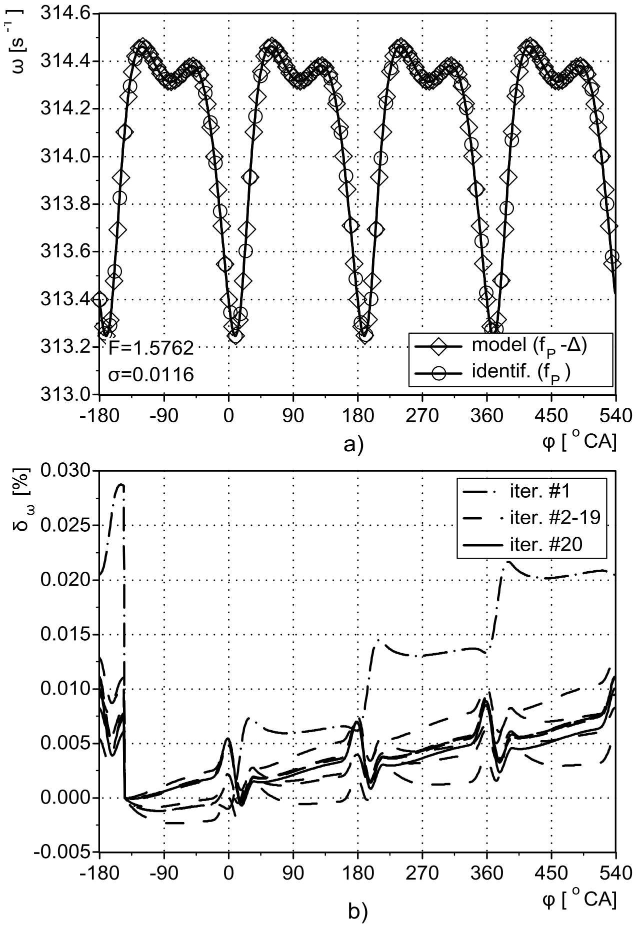

Figure 12 Angular speed identification: modelled and identified angular speed (a) and relative deviation during iterative optimisation (b)

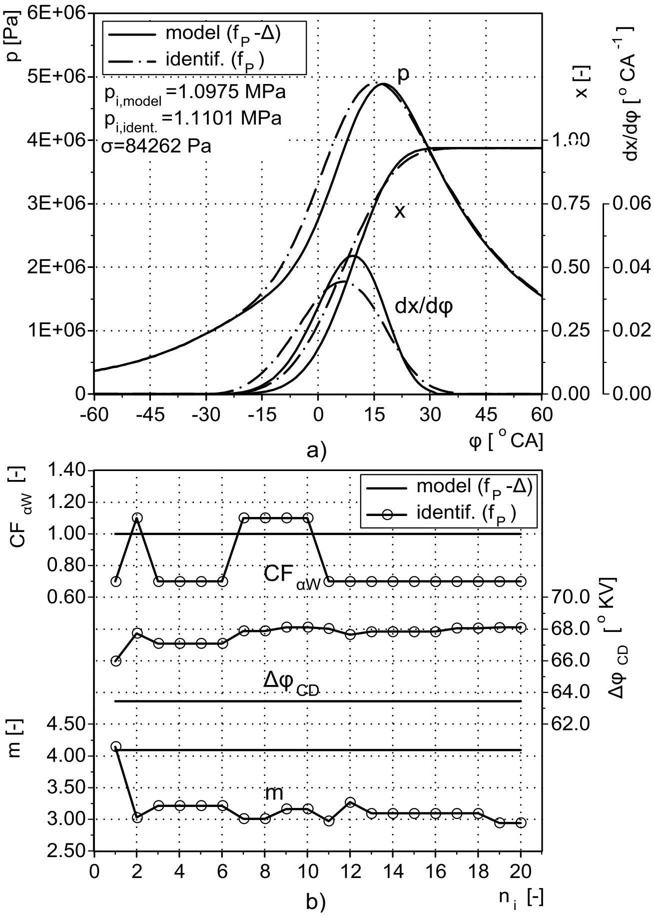

Figure 13 Model parameters identification: pressure, MFB and RoHR (a); heat transfer coefficient correction factor, combustion duration and form factor

were identified with even higher negative deviations, –25% and –31% respectively. In order to compensate higher original values of friction losses obtained through combined Taraza-Teodorescu model, numerical optimisation acts as to obtain higher heat release rates at the very start of combustion. The result seems contradictory, because higher IMEP values accomplished through faster heat release at the beginning would lead to even more significant crankshaft acceleration. This however, does not occur, because friction losses in piston-cylinder contact depend on directly on gas pressure acting on the back surface of each piston ring.

Uncertainties in friction losses model, obviously, affects largely the combustion process identification and even the application of detailed angle resolved friction losses model can't 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 combustion process and crankshaft speed and therefore, largely affect the optimisation process and results of indirect combustion analysis. These phenomena must be carefully analysed and incorporated in identification process as well, in order to provide results comparable to direct combustion analysis method used as reference. However, the approach and presented here shows potentials for fast and cost effective combustion analysis based on measurement of instantaneous crankshaft angular speed.

CONCLUSIONS

Complex angle based, nonlinear model for combustion, engine friction and engine dynamics is presented in this paper. The set of models was used to perform 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.

Box-constrained Levenberg-Marquardt optimisation routine for LSQOF was applied to investigate the influence of compression ratio uncertainties 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.

Uncertainties in friction models, however, appear influential and must be subjected to identification. Procedure presented in this work, proved sufficiently sensitive and robust, and provided improved in terms of friction coefficient identification could be used as alternative method for combustion analysis.

ACKNOWLEDGEMENTS

The results presented in this paper have been obtained through the research projects NPEE-290025 and TR-14074 realized under the financial support by the Serbian Ministry of Science and Education within the National Energy Efficiency Program.

REFERENCES

1. Tomic, 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. XIX Fisita Congress, Melbourne, Australia, 1982, paper 8216
3. Brunt, M. F. J., Pond, C. R., Evaluation of techniques for absolute cylinder pressure correction. SAE 1997 International Congress & Exposition, Detroit, MI, USA, 1997, SAE Technical Papers, SAE 970036.
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. Lindström, F., Empirical Combustion Modeling in SI Engines, Ph.D. thesis. Machine Design. Royal Institute of Technology (KTH), Stockholm, 2005
6. Tunestål, P.. Self-tuning gross heat release computation for internal combustion engines. *Control Engineering Practice*, *17* (2009), 4, pp. 518–524.
7. Pischinger, R., Klell, M., Sams, T., Thermodynamics of internal combustion engines (in German), Springer-Verlag. Wien, 2002
8. Gordon, S., McBride, B. J. (1994). Computer Program for Calculation of Complex Chemical Equilibrium Compositions and Applications: Part I - Analysis. *NASA reference Publication*, Report No. RP–1311–P1.
9. Olikara, C., Borman, G. L., A Computer Program for Calculating Properties of Equilibrium Combustion Products with Some Applications to IC Engines. SAE Technical Papers, 1975, SAE 750468.
10. Wannatong, K., Simulation algorithm for piston ring dynamics, *Simulation Modelling Practice and Theory*, 16 (2008), 1, pp. 127–146.
11. Bonatesta, F., Waters, B., Shayler, P. J., 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.
12. Heywood, J. B., Internal Combustion Engine Fundamentals. McGraw-Hill Inc., New York, 1988
13. 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.
14. Hohenberg, G. (1980). Calculation of the gas heat transfer in diesel engines (in German), *Motortechnische Zeitschrift - MTZ*, *41 (1980), 7/8*, pp. 321–326.
15. Chang, J., *et. al.*, New Heat Transfer Correlation for an HCCI Engine Derived from Measurements of Instantaneous Surface Heat Flux, 2004 Powertrain & Fluid Systems Conference & Exhibition, Tampa, FL, USA, 2004, Session: Homogeneous Charge Compression Ignition SAE Technical Papers, 2004, SAE 2004–01–2996.
16. Cho, K.-W., *et. al.*, (2008). Experimental investigation of combustion and heat transfer in a direct-injection spark ignition engine via instantaneous combustion chamber surface temperature measurements. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *222 (2008), 11*, pp. 2219–2233.
17. Wang, X., *et. al.*, Heat release and heat flux in a spray-guided direct-injection gasoline engine. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *221 (2007), 11*, pp. 1441–1452.
18. Taraza, D., Possibilities to Reconstruct Indicator Diagrams by Analysis of the Angular Motion of the Crankshaft, International Off-Highway & Powerplant Congress & Exposition, Milwaukee, WI, USA, 1993, Session: International Off-Highway & Powerplant Congress & Exposition, SAE Technical Papers, SAE 932414.

19. Schagerberg, S., McKelvey, T., Instantaneous Crankshaft Torque Measurements – Modeling and Validation, SAE 2003 World Congress & Exhibition, Detroit, MI, USA, 2000, Session: Electronic Engine Controls SAE Technical Papers, SAE 2003–01–0713.
20. Fehrenbach, H., *et. al*., Determination of engine torque from the angular speed signal (in German) *Motortechnische Zeitschrift - MTZ*, 63 (2002), 12, pp. 1020-1027
21. 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.
22. 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
23. 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.
24. Taylor, C. M., Engine Tribology: Tribology Series, Elsevier Science Publishers B.V., Amsterdam, Netherlands, 1993
25. Taraza, D., Henein, N., Bryzik, W., Friction Losses in Multi-Cylinder Diesel Engines, SAE 2000 World Congress, Detroit, MI, USA, 2000, Session: CI & SI Power Cylinder Systems, SAE Technical Papers, 2000, SAE 2000–01–0921
26. Stachowiak, G. W., Batchelor, A. W., Engineering Tribology, Butterworth-Heinemann, Elsevier Inc. , 2006
27. Teodorescu, M., *et. al.*, Simplified Elasto-Hydrodynamic Friction Model of the Cam-Tappet Contact, SAE 2003 World Congress & Exhibition, Detroit, MI, USA, 2003, SAE Technical Papers, SAE 2003–01–0985.