HIGH-FIDELITY, ANGLE-RESOLVED SIMULATION MODEL FOR PREDICTIONS OF MULTI-CYLINDER ENGINE INSTANTANEOUS SPEED AND TORQUE

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

Summary: *A non-linear, angle-resolved, multi-cylinder SI engine computational model has been developed for predictions of instantaneous crankshaft speed and torque. The computation is firmly based on high-fidelity, comprehensive thermodynamic, steady-state, Two-Zone, Zero-Dimensional combustion model followed by a detailed analytical component model of engine friction and mechanical losses. Predictions of both engine cranking and steady-state operation conditions reveal strong dependence of variable engine moment of inertia and crankshaft instantaneous speed signal. The uncertainties and variation in the masses of the reciprocating slider mechanism components are further analysed in order to establish the impact on the instantaneous torque and speed profile. Self-tuning concept based on Levenberg-Marquardt Box-Constrained Optimisation algorithm has been introduced for model parameter identification.*
Keywords: *spark ignition engine, two-zone model, mechanical losses, instantaneous crankshaft speed, optimisation*

1. 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. The possibility to obtain these parameters from instantaneous crank shaft speed signal is extremely attractive and beneficial considering simplicity of nonintrusive, available measurement compared to pressure indicating, particularly in case of mass-production engines. The issues regarding the identification of Wiebe parametric 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 Least Squares (LSQ) set upon measured and

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simulated instantaneous crankshaft angular speed which is determined from the solution of the engine dynamics torque balance equation. The comprehensive Two-Zone SI 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. The simulation model was employed as to evaluate the engine components mass imbalances influence on combustion analysis based on instantaneous engine crankshaft angular speed.

2. SIMULATION MODEL LAYOUT

2.1 Combustion submodel structure

Zero-Dimensional (0D) Two-Zone (2Z) thermodynamic model is used as a basis for SI Engine combustion simulation. Model applied here follows the structure essentially proposed by Pischinger [1]. The set of governing equations is derived on the basis of The First Law of Thermodynamics for open systems applied to the cylinder volume, assuming in stationary process. Unburned mixture and combustion products are separated by means of thin zone of combustion reactions (flame front), heat transfer between the zones is neglected, and pressure distribution in each zone is assumed homogenous and equal, while gas temperature, composition and properties are homogenous within each zone. The combustion mixture, having in mind SI Engine operation, is assumed homogenous comprising fuel vapour, air and residual combustion products. The heat release is introduced by means of Wiebe parametric single stage model. More details on engine combustion model structure can be found in [1–4].

Thermodynamic properties of the mixture and combustion products are calculated approximately using NASA 9-coefficients polynomials [5], while species molar concentrations are calculated by means of 12 component Olikara-Borman [6] 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. Revised approach based on piston ring gap area and pressure related discharge coefficient formula proposed by Wannatong [7] is used to simulate cylinder charge leakage through crevices.

2.2 Engine dynamics and mechanical losses submodels

Assuming crank and connecting shafts rigid, Single Degrees of Freedom (1-DoF) dynamic engine model is applied to simulate crankshaft instantaneous angular speed. Engine-dynamometer dynamic system is presented in Fig. 1. 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 available from manufacturing specification. Torque balance equation for engine-dynamometer system arises from kinetic energy equation:

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

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.

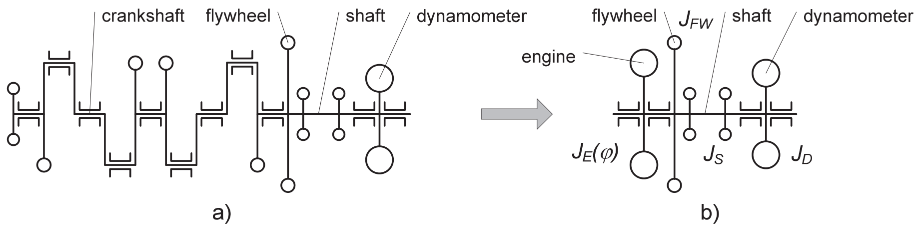

Fig. 1 *Engine dynamic model (a) and reduced, equivalent 1-DoF dynamic model (b)*

Friction losses in piston-cylinder contact are calculated by means of angle-resolved models presented by Taraza [9] relying on basic lubrication theory and Stribeck diagram. Crank and cam shaft bearings are assumed short radial bearings with Hydro-Dynamic Lubrication (HDL), for which the friction losses are modelled by equation proposed by Ocvirk. [9]. The friction losses in cam-tappet contact are modelled using approximate solution of Reynolds equations for Elasto-HydroDynamic Lubrication (EHDL) proposed by Teodorescu [10]. Friction models listed here exceed the scope of this paper, while more detailed approach can be found in selected literature.

3. SIMULATION RESULTS

Simulation model is employed at first instance as to validate the influence of uncertainties of both rotating components moment of inertia and piston group mass on instantaneous angular speed. The analysis was performed for serial-production port fuel injection petrol engine DMB M202PB13 by varying cumulative moment of inertia for crankshaft, connecting shaft and dynamometer (Fig. 2) and piston mass (Fig. 3) within the range ±10% refer to nominal values at reference operating point (n=3000 min^{-1}/ WOT). In both cases, instantaneous angular speed deviates, as expected, most noticeable in the close vicinity of the each TDC, however, the range is quite limited, staying below ±0.03%.

The second stage of numerical experiment treats the influence of uncertainties in both variables on numerical process set up to identify basic combustion model parameters: heat transfer coefficient correction factor $CF_{\alpha W}$, combustion duration $\Delta\varphi_{CD}$ and Wiebe model shape parameter m. The reference values for heat transfer coefficient are established using Woschni model, recently updated for SI engine operation (Chang, [2, 3, 8]). Wiebe model parameters for given operation point are predicted using Lindström/Bonatesta empirical models [2-4]. The influence of cumulative moment of inertia decreased for 10% on identified instantaneous angular speed is displayed in Fig. 4. In-cylinder pressure trace, MFB and RoHR (Fig. 4b) are all identified with the highest accuracy and the differences cannot be observed within the resolution of graphic presentation. The shape parameter *m* and combustion duration $\Delta\varphi_{CD}$ are identified in 17. and 18. iteration, respectively, while heat transfer coefficient correction factor $CF_{\alpha W}$, is identified with deviation of +8%. The case related to 10% increased cumulative moment of inertia of rotating components (not shown) exhibits slightly increased deviation in identified values: cca 0.6 °CA in combustion duration, Wiebe function shape parameter value lower for 1.9% and heat transfer coefficient higher for cca. 6%.

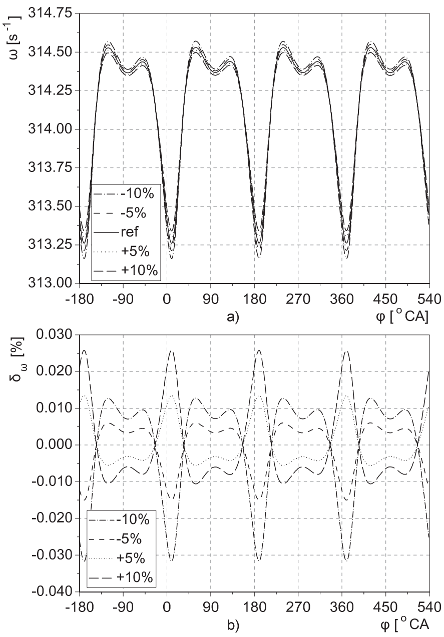

Fig. 2 *The influence of rotating mass inertia on instantaneous angular speed (a) and its deviation (b) (n=3000 min^{-1}, θ=100%)*

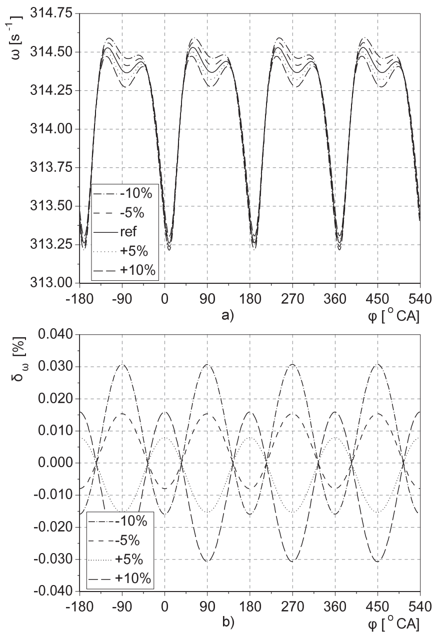

Fig. 3 *The influence of piston mass uncertainty on instantaneous angular speed (a) and its deviation (b) (n=3000 min^{-1}, θ=100%)*

Piston mass influences the IC engine variable moment of inertia. However, the influence of mass uncertainties of each piston cannot be analysed and displayed separately because all piston mass disturbances due to production tolerances are combined and comprised within the engine moment of inertia. Piston mass is at first assumed decreased for 10% which is analysed in respect to heat release model parameters identification. Fig. 5a shows the trace of instantaneous angular speed deviation during iterative optimisation process. Relative difference of cca. 0.005% is observed at the end of 20th iteration. Wiebe function shape parameter is identified with the relative error of −1.8%, combustion duration is slightly longer (cca. 0.5 °CA), and heat transfer coefficient is lower for cca. 2.1%. The opposite case treats piston mass increased for 10% (not presented). Identification process exhibits strong convergence, which is observed on all previous cases. However, identified values for three basic model parameters deviate more intensively. Wiebe function shape parameter is identified with deviation of 4.7% and heat transfer coefficient with deviation of cca. 33%. The deviation for combustion duration remains, however, lower, i.e. 1 °CA.

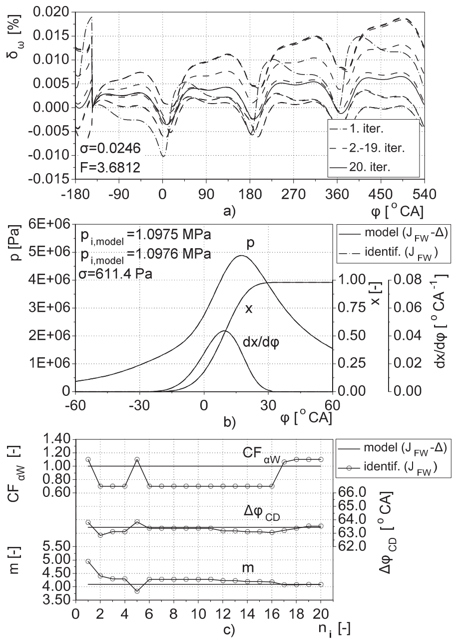

Fig. 4 *Model parameters identification: angular speed relative deviation during iterative optimisation (a); pressure, MFB and RoHR (b); heat transfer coefficient correction factor, combustion duration and form factor (c) (n=3000 min^{-1}, θ=100%)*

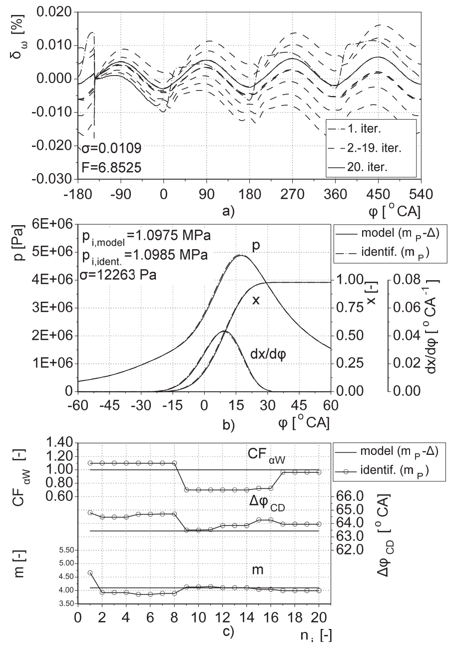

Fig. 5 *Model parameters identification: angular speed relative deviation during iterative optimisation (a); pressure, MFB and RoHR (b); heat transfer coefficient correction factor, combustion duration and form factor (c) (n=3000 min^{-1}, θ=100%)*

4. CONCLUSION

Complex, angle based, nonlinear model for combustion, engine friction and engine dynamics is presented in this paper. The set of SI engine simulation models was used to perform sensitivity analysis concerning the influences of uncertainties in IC engine slider mechanism component's masses and rotating components inertia on instantaneous crankshaft angular speed. The second stage analysis was performed using Box Constrained Optimisation based on Levenberg-Marquardt algorithm in order to establish the influence of mass and inertia uncertainties on in-cylinder pressure, MFB and RoHR tuning process. The analysis shows that uncertainties in both rotating components moment of inertia and piston mass influence the process of model parameters identification. Deviation of shape parameter, combustion duration and heat transfer coefficient exist, compared to reference values established through modelling. However, differences in pressure trace, MFB, RoHR and instantaneous angular speed provided at the end of iterative optimisation process can be hardly observed within the resolution of graphical presentation. The uncertainties analysed in this paper, have, practically, no influence in terms of identification process performance and convergence.

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