TEKA Kom. Mot. i Energ. Roln. – OL PAN, 2011, 11c, 38–52

MODELLING AND VERIFICATION FAILURES OF A COMBUSTION ENGINE INJECTION SYSTEM

Mieczysław Dziubiński, Jacek Czarnigowski

Lublin University of Technology, Faculty of Mechanical Engineering
20-618 Lublin, Nadbystrzycka 36 Str.
E-mail: m.dziubinski@pollub.pl, j.czarnigowski@pollub.pl

Summary. Growing pressure to improve the performance of IC engines has resulted in development of improved control systems. These requirements, combined with the introduction of control algorithms that could only be simulated experimentally were the foundations of control-oriented modelling techniques. The paper presents the assumptions and requirements toward control oriented models and a model of an idling SI engine. Some results of identification and verification tests of the model are presented for the 1.5 GLI engine. On the basis of the engine model, the simulation of sensor failures had been carried out.

Key words: internal combustion engine, idle speed, control, simulation of sensors failures.

1. INTRODUCTION

During recent years, the main advances in internal combustion engine technology have been the reduction of toxic gas emissions and fuel consumption, in addition to improvement of engine work stability and reliability in steady and dynamic state.

These targets have been dictated by environment protection regulations. Neither engine construction development with optimisation of particular components, nor introduction of new materials and technologies were enough to reach these targets. Therefore, new control systems allowing for variable working conditions and dynamic changes of engine characteristics have had to be introduced and developed.

Simple control algorithms (e.g. PID) whose synthesis is based on the system transmittance analysis proved insufficient in these circumstances. This is due to their inability to account for the characteristics of the object of control. Thus, algorithms based on simple linear engine models were implemented such as LQR and H-infinity algorithms, where the selection of settings is based on mathematical processing of engine models. These algorithms have also their limitations: firstly, the model is linear and therefore only a rough approximation, and secondly, it is also stationary i.e. identified for a certain engine in a stated moment of its life cycle.

At the same time some complex algorithms emerged that, due to their nonlinearity could not be synthesized by means of mathematical transformations. Examples of these are algorithms based on adaptation theory. The settings of the regulator need to be determined empirically. A proper synthesis requires repeatable conditions of the experiments which is highly difficult or even impossible to achieve either in the test-bed or during normal operation. A solution to this problem may be simulation tests, whose results can be verified empirically on the test bed.

This method is less time-consuming and allows reduced costs of control system algorithm synthesis. This is why modelling gained on importance and control-oriented models are being developed.

2. MODEL OF THE ENGINE

Our intention was to create a mathematic model by means of which parameters of control algorithms could be determined and tests of individual algorithms of the simulated SI engine idle speed control could be conducted. In order to meet the assumptions of the object oriented model, a mean value model allowing for inertia of the engine and its control system was developed. The object of modelling was a 1.5 GLI car engine – a four-stroke four-cylinder spark ignition single point injection engine, cubic capacity of 1496 cm^3. The model was calculated twice for each rotation of the crankshaft (frequency of ignition control). The model was programmed using a C++ object programming technique. It was divided into 3 modules:

a) Fuel film,
b) Manifold,
c) Cylinder and crankshaft.

The nonstationarity and variability of the engine was expressed by changes of indicated torque determined in the last module.

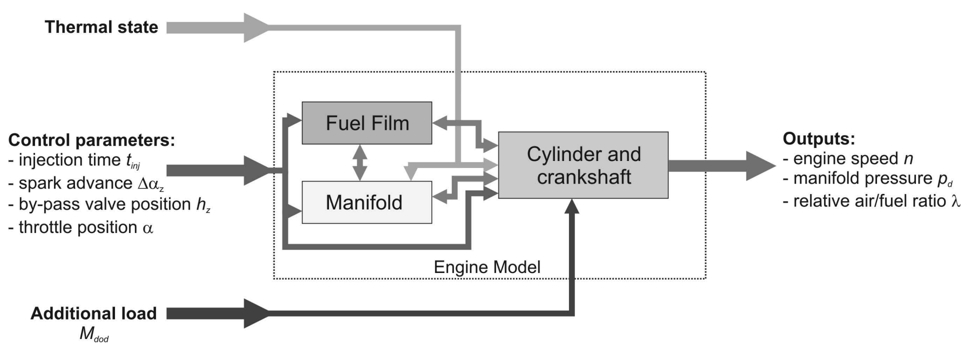

Fig. 1. Block diagram of the engine model

2.1. Fuel film module

Fuel film results from the deposit of condensed fuel on the manifold walls. It is particularly distinct in single point injection engines. The fuel film evaporates during the engine work adding to the injected dose, thus increasing the amount of fuel reaching the cylinder. In particular conditions the rate of vaporisation (and the amount of vaporised fuel) depends on the mass of fuel film.

The input to this module is the injection time t_{inj} recalculated into the mass of fuel supplied by a single injection m_{inj}. This injected fuel is divided into two parts: m_{ip} – mass of fuel evaporised directly, and m_{if} – mass of fuel deposited in the manifold as fuel film:

$$m_{ip} = (1 - X) \cdot m_{inj}, \tag{1}$$

$$m_{if} = X \cdot m_{inj}, \tag{2}$$

m_{if} increases the total mass of fuel film m_{film}. At the same time, some fuel vaporises from the fuel film (m_{fp}) adding to the fuel supplying the cylinder. Thus, the mass of fuel reaching the cylinder may be described by the following equation:

$$m_{fuel} = m_{ip} + m_{fp}. \tag{3}$$

The dynamic of fuel film is formulated by the equation describing change of fuel mass evaporating from the film:

$$\dot{m}_{fp} = \frac{1}{\tau}(-m_{fp} + X \cdot m_{inj}). \tag{4}$$

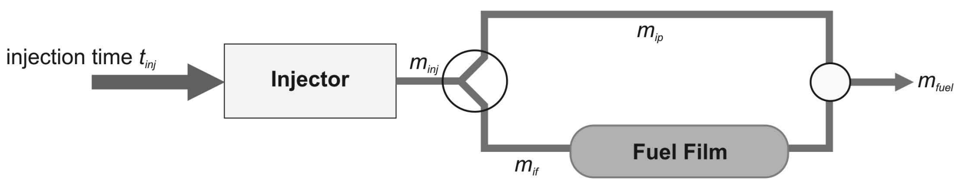

Fig. 2. Calculations in the fuel film module

2.2. Manifold module

This module is based on the mass balance in the manifold:

$$\frac{dm_{mf}}{dt} = \dot{m}_{tr} + \dot{m}_{bp} + \dot{m}_{fuel} - \dot{m}_{cyl}. \tag{5}$$

The mass flow rates entering the manifold through the throttle and the by-pass valve are described by equations of subcritical or critical flow through a Venturi throat:

$$\dot{m} = \frac{\mu \cdot F_{flow} \cdot p_0}{\sqrt{R \cdot T_0}} \cdot \psi, \tag{6}$$

where:

$$\psi = \begin{cases} \sqrt{\dfrac{2 \cdot \kappa}{\kappa - 1}\left[\left(\dfrac{p_d}{p_o}\right)^{\frac{2}{\kappa}} - \left(\dfrac{p_d}{p_o}\right)^{\frac{\kappa+1}{\kappa}}\right]} & p_d > p_o \cdot \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}} \\ \sqrt{\kappa \cdot \left(\dfrac{2}{\kappa - 1}\right)^{\frac{\kappa+1}{\kappa-1}}} & p_d \le p_o \cdot \left(\dfrac{2}{\kappa+1}\right)^{\frac{\kappa}{\kappa-1}} \end{cases}. \tag{7}$$

The mass flow rate of air-fuel mixture leaving the manifold is defined by the sum of masses reaching particular cylinders and calculated according to the following equation:

$$\dot{m}_{cyl} = \frac{1}{\Delta t} \frac{p_0 \cdot V_{cyl}}{R_{mix} \cdot T_0} \cdot \eta_v. \tag{8}$$

The output from the manifold module is the pressure in the manifold calculated from the equation of state of gas in the manifold as follows:

$$p_d = \frac{m_{mf} \cdot R \cdot T_{mf}}{V_{mf}}. \tag{9}$$

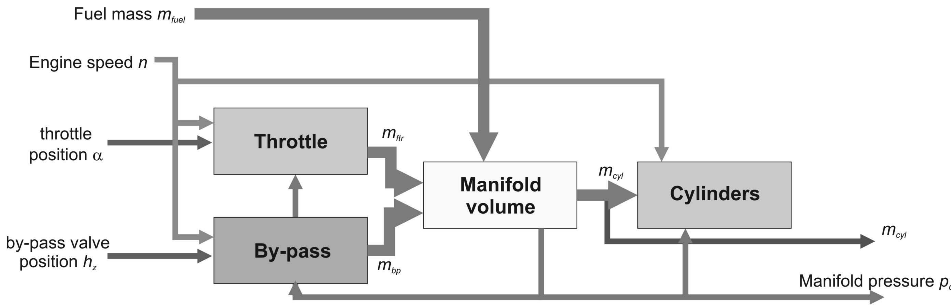

Fig. 3. Calculation of the manifold module: the algorithm

2.3. Cylinder and crankshaft module

The third module of the model is responsible for calculation of effects of engine operation, i.e. torque and combustion gases' composition. The inputs to this module are: spark advance $\Delta\alpha_z$ mass of air reaching the cylinder m_{air}, mass of fuel reaching the cylinder m_{fuel}, engine speed n, throttle position α and additional load torque M_{dod}. The parameters calculated by this module are engine speed and lambda sensor signal.

The module comprises four components:

a) indicated torque (M_i) calculation submodule,
b) engine friction torque M_b calculation submodule,
c) new engine speed calculation submodule,
d) lambda sensor submodule.

2.4. Indicated torque (Mi) calculation submodule

The value of indicated torque is calculated on the basis of:

- engine speed n,
- pressure in the manifold p_d,
- spark advance $\Delta\alpha_z$.

The torque is calculated by means of artificial neural network MLP (3-5-5-1) BP. The result of calculation is corrected by a factor dependent on the air/fuel ratio coefficient λ, according to the following formula:

$$M_i = \beta(\lambda) \cdot M_{istch}. \tag{10}$$

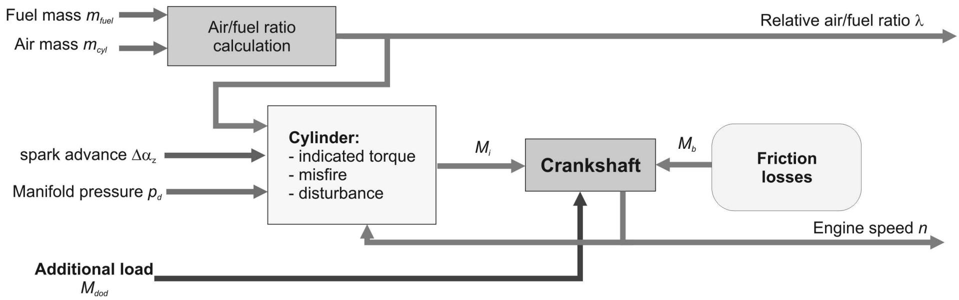

Fig. 4. Structure of the cylinder and crankshaft module

The author allowed for nonstationarity engine operation by modelling the following effects:
a) misfire,
b) uneven work of cylinders,
c) stochastic nonstationarity of indicated torque.

The nonstationarity in this module is defined according [9]. To obtain a random value of normal distribution, of stated parameters of variability:
a) mean value ,
b) standard deviation σ_{Mi},

it may be calculated from the following equation:

$$M_{i\sigma} = \bar{M}_i + \sigma_{Mi} \cdot \cos(2 \cdot \pi \cdot RND) \cdot \sqrt{-2 \cdot \log(RND)}. \tag{11}$$

In order to bind the next value of M_i with the previous one, a back propagation of the value by means of functional binding values of $M_i(k\text{-}1)$ and $M_i(k)$ may be introduced:

$$M_i(k) = (1-\xi^2) \cdot M_i(k-1) + z \cdot \xi \cdot M_{i\sigma}, \tag{12}$$

where:

$$z = 1 + 0.55 \cdot (1-\xi)^{0.75}. \tag{13}$$

2.5. Engine friction torque calculation submodule

The value of engine friction torque is calculated from a dependency on engine speed n:

$$M_b = c_0 + c_1 \cdot n + c_2 \cdot n^2. \tag{14}$$

2.6. Engine speed calculation submodule

The engine speed of the engine is calculated based on the second principle of dynamics:

$$\frac{dn}{dt} \cdot I_b = M_i - M_b - M_{dod}. \tag{15}$$

Additional load torque M_{dod} is the effect of engaging additional equipment during the engine work [1, 8]. On this basis, engine acceleration in a particular cycle and new engine speed are calculated.

2.7. Lambda sensor submodule

The last part of the cylinder and crankshaft module is the function calculating the lambda sensor signal (with signal time-lag). The function returns the voltage of the sensor on the basis of engine work parameters (actual air/fuel ratio in the cylinder). On the basis of the method presented in [Hawryluk 2001], the lag of signal was assumed to be 20 calculation cycles. The lambda sensor signal is modelled by a function described in [Pukrushpan 2004].

3. IDENTIFICATION AND VERIFICATION OF THE MODEL

The identification of the model covered idle speed as the model was designed for synthesizing and verification of idle speed control algorithms. The scope of verification was thus confined to:
- engine speed range from 600 to 1300 RPM,
- manifold pressure range from 30 to 60 kPa,
- spark advance range from 0 to 30 degrees before TDC,
- relative air/fuel ratio range from 0.8 to 1.2.

Subject to identification were parameters of the modules. Tests were conducted on the test bed equipped with a direct current brake that enabled both reception of the energy from the engine and to driving it. Fuel consumption, combustion gas composition, engine thermal state and in-cylinder pressure were measured. The cooling system allowed the engine lto be kept in a steady thermal state. The conditions during the tests were steady in terms of thermal state and loads. A choice of results is presented below.

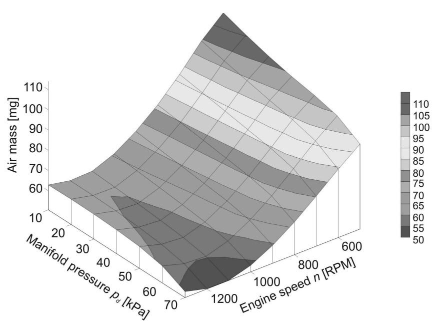

Fig. 5.Mass of air entering the manifold by closed throttle

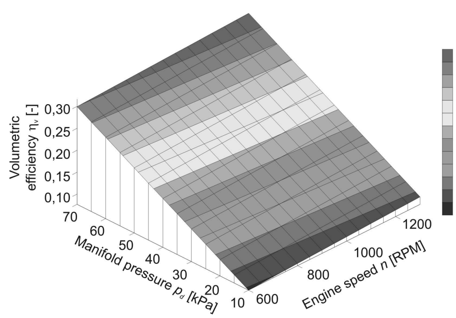

Fig. 6. Volumetric efficiency characteristics

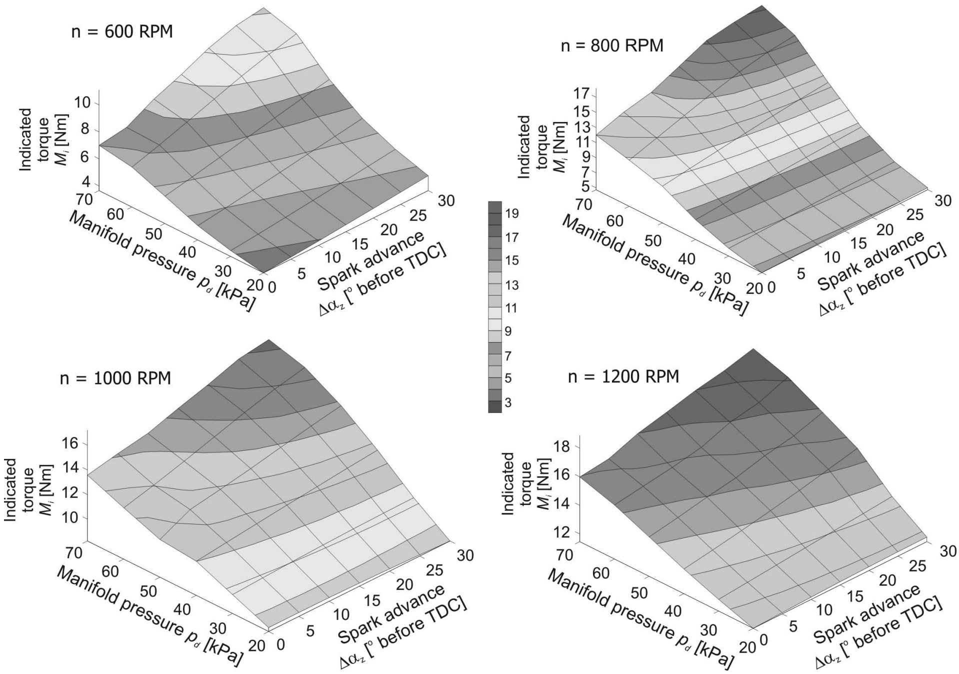

Fig. 7. Dependence of indicated torque from spark advance and engine speed by constant pressure in the manifold

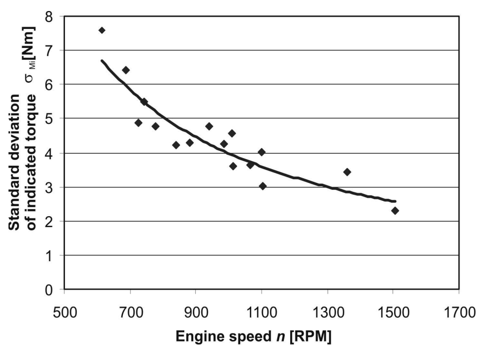

Fig. 8. Standard deviation of indicated torque

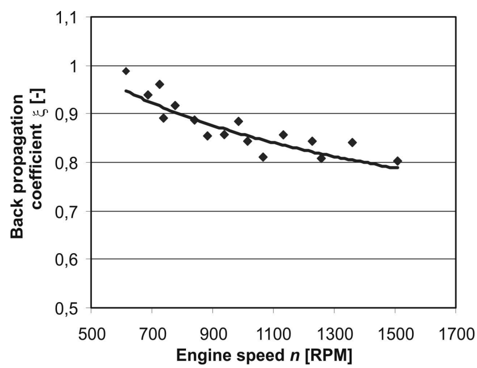

Fig. 9. Coefficient ξ of indicated torque

The last stage of the research was the verification of the model with regard to the steady state and sudden change of additional load specific to idle speed. As the main parameter of engine work was engine speed, the model adequacy analysis consisted in checking if the model correctly predicts the value of engine speed. In steady state, correlation of verification test results and model calculations was 0.95 (see Figure 12). This similarity of behaviour of the engine and its model was also clear in the case of sudden change of load without control (Figure 10).

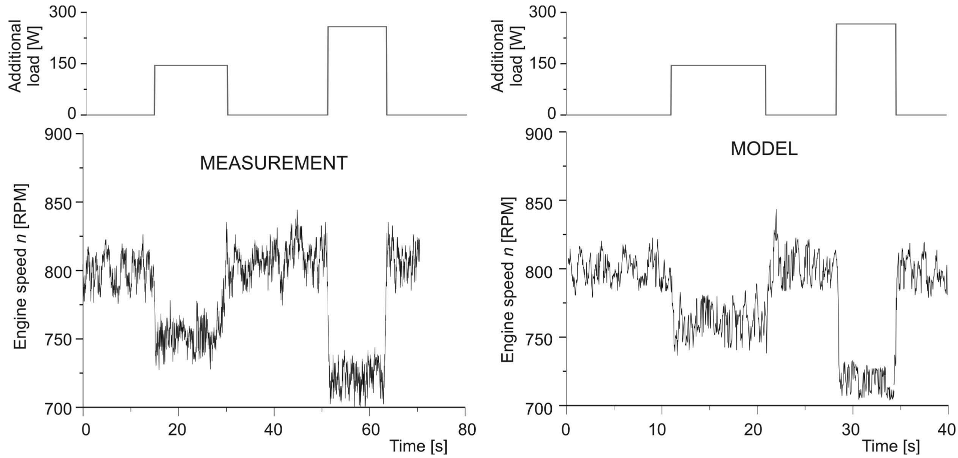

Fig. 10. Graph of rotational speed under additional load without control
– test bed measurements and results of the simulation

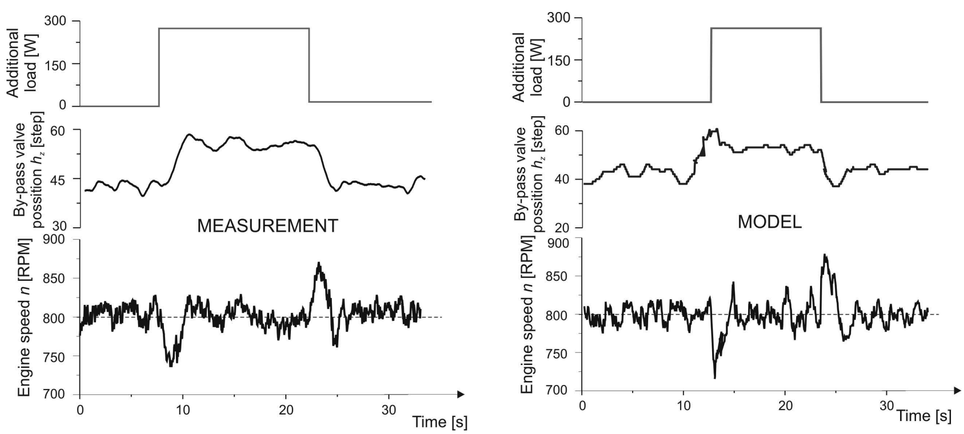

Fig. 11. Graph of rotational speed under additional load with adaptive control
– test bed measurements and results of the simulation

The model was applied to simulation tests of adaptive and PID control algorithms [Smith 1995, Stefanopoulou 1998]. Exemplary results of these tests, confirming the adequacy of the model, can be seen in Figure 11.

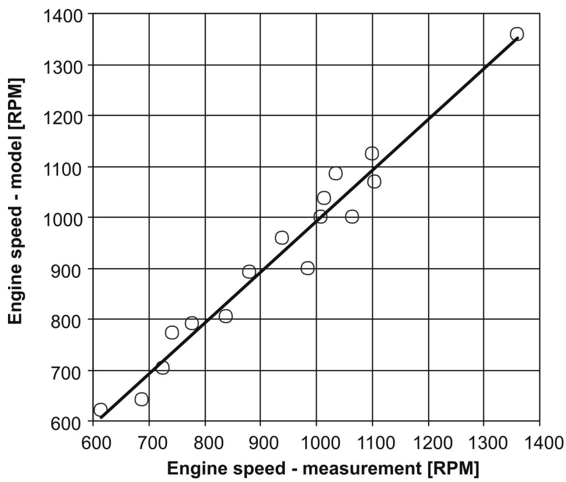

Fig. 12. Correlation of verification test results and model's calculations

LINE-BY-LINE EDITING CEASED HERE. ONLY RANDOM EDITING AFTER THIS POINT.

4. MODELLING OF SENSOR FAILURES IN ENGINE INJECTION

The simulation model for sensor failures is presented in Fig. 13.

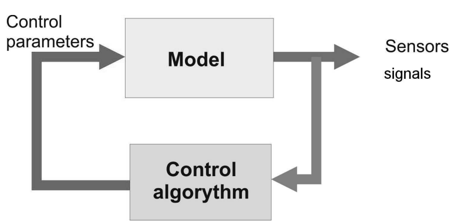

Fig. 13. Model of sensor failures simulation

4.1.Models of sensors

Sensors were modeled as separate elements (modules) and described by the characteristics presented below:

a) pressure sensor MAP,
   $U_{MAP} = 0.05294 \cdot MAP - 0.659$,
   where: U_{MAP} - sensor voltage signal [V],
   MAP – pressure in suction manifold [kPa].
b) temperature sensors:
   $U_{Tem} = 0.000002 \cdot T^3 - 0.0003 \cdot T^2 - 0.0305 \cdot T + 4.2796$,
   where: U_{Tem} - sensor voltage signal [V],
   T – temperature [K],

The modules of cooling agent temperature and the temperature of air in the suction manifold were used in the model.

c) throttling valve position sensor – potentiometric sensor:
$U_{TP} = 0.05294 \cdot TP - 0.659$,
where: U_{TP} - sensor voltage signal [V],
TP – throttling valve position [%],

d) mass air flowmeter MAF:
$U_{MAF} = 0.6525 \cdot MAF^{0.3143}$,
where: U_{MAF} - sensor voltage signal [V],
MAF – mass air flow [$g \cdot s^{-1}$].

During simulation tests, it was possible to introduce failures of any sensor. In each case, it was possible to simulate the following failures:

– fault to frame:
$U_{sen} = U_{gnd}$,
where: U_{sen} - sensor voltage signal [V],
U_{gnd} - mass voltage level [V],
– fault to admission:
$U_{sen} = U_{power}$,
where: U_{sen} - sensor voltage signal [V],
U_{power} - admission voltage level [V],
– shift of the characteristic (e.g. resulting from the shift of the sensor):
$U_{sen} = U_{sen\,(m)} + \Delta U$,
where: U_{sen} - sensor voltage signal [V],
$U_{sen\,(m)}$ - sensor voltage signal coming from the model without failures [V],
ΔU – level of sensor characteristic shift [V],
– characteristic tilt (e.g. resulting from sensor ageing):
$U_{sen} = U_{sen\,(m)} \cdot \alpha$,
where: U_{sen} - sensor voltage signal [V],
$U_{sen\,(m)}$ - sensor voltage signal coming from the model without failures [V],
α – coefficient of sensor characteristic tilt [V].

4.2. Controlling system

The engine control system model is presented in Fig. 14.

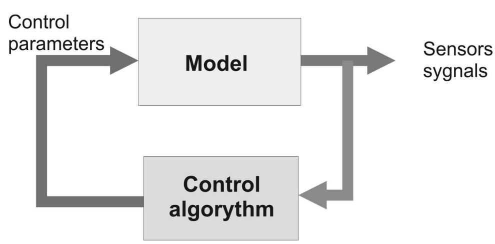

Fig. 14. Engine control system for chosen operation parameters

Signals and algorithm are misspelt fig 4.2 above.

In the range of studies, an algorithm of engine control based on maps and adaptation matching has been worked out. The control was carried out with three variables:

– ignition advance angle (constant value during tests);
– by-pass valve position: in this case, adaptation control, whose aim was to stabilize engine speed on the assumed level was used. The algorithm of control, described in the paper: “Idle speed stabilization using competitive adaptation control of by-pass valve in SI engine” by Czarnigowski, Wendeker, Jakliński SAE-NA 34-2005, was used.
– Injection time:
  In this case, the injection time is calculated according to the formula:
  $T_{inj} = k_{\alpha} \cdot t_{inj\,(m)} \cdot \alpha_{ET} \cdot \alpha_{MAT}$,
  where: T_{inj} - injection time [ms],
  k_{α} - correction factor calculated by adaptation algorithm,
  α_{ET} - correction factor from engine temperature,
  α_{MAT}- correction factor from the temperature of air in the suction manifold,
  $t_{inj\,(m)}$ - basic injection time determined from the map [ms],
  and: $t_{inj\,(m)} = f\,(n, LOAD)$,
  where: n – engine speed[rpm].
  LOAD - signal of engine load, depending on the tested sensor it is the degree to which the throttling valve opens, pressure in the suction manifold, the amount of air mass (mass flowmeter).

The disturbance of signals from sensors causes the change either of injection base dose or correction factor value (in case of temperature).

Figs. 15. – 20. present the course of rotational speed stabilization at the step increase and decrease of load in the case of efficient sensors.

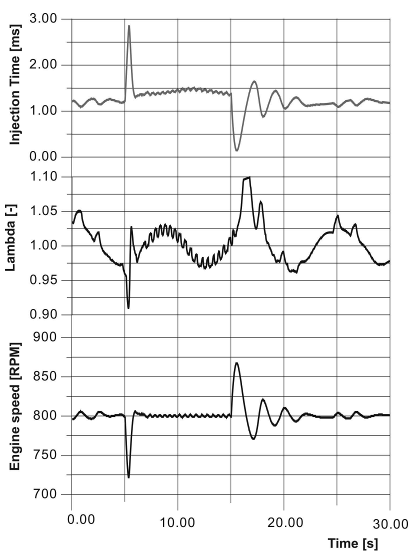

Fig. 15. Course of engine speed stabilization at the step increase and decrease of load in the case of efficient sensors

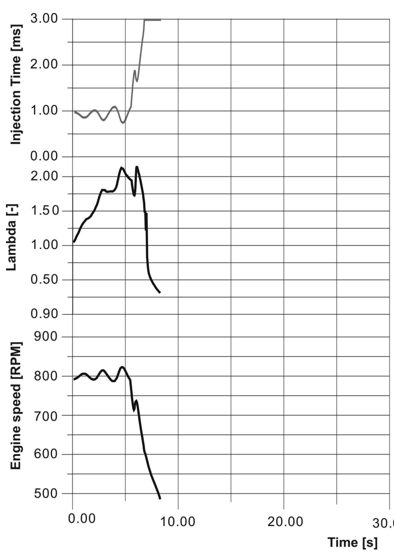

Fig. 16. Course of engine speed stabilization at the step increase and decrease of load for a pressure sensor failure in the suction manifold – fault to frame

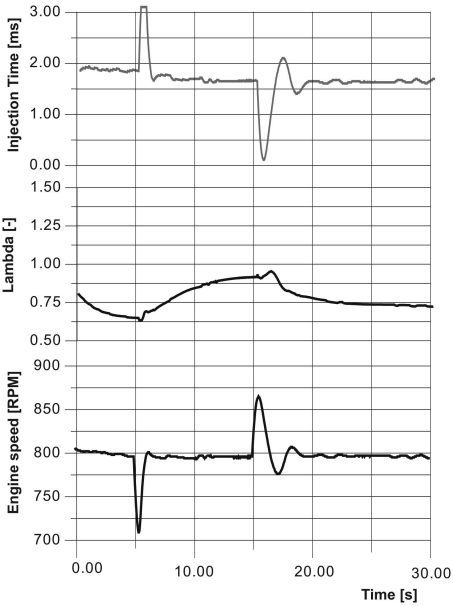

Fig. 17. Course of engine speed stabilization at the step increase and decrease of load for a pressure sensor failure in the suction manifold – fault to admission

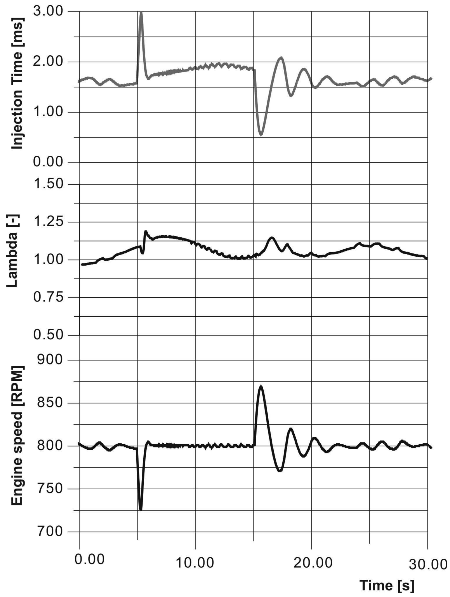

Fig. 18. Course of engine speed stabilization at the step increase and decrease of load for a pressure sensor failure in the suction manifold – characteristic shift

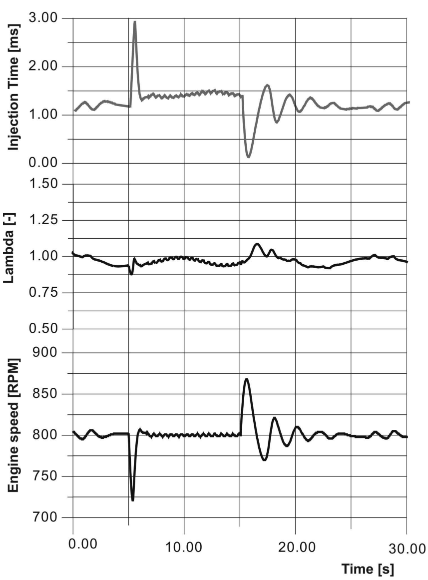

Fig. 19. Course of engine speed stabilization at the step increase and decrease of load for a pressure sensor failure in the suction manifold – characteristic tilt

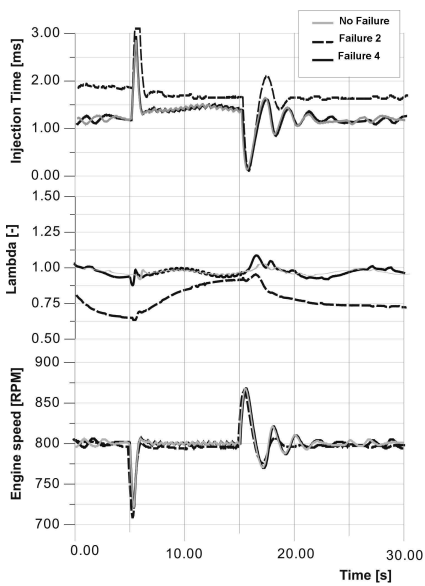

Fig. 20. Courses of engine speed stabilization at the step increase and decrease of load for the efficient sensor or its two failures

CONCLUSIONS

Applying control oriented modelling to the synthesis of control algorithms enables us to conduct simulation tests to identify settings of regulators. Moreover, verification and comparative tests of particular control algorithms. Advantages of models based on mean values are ease of identification of parameters by satisfying level of adequacy in both steady and dynamic states and fast calculations.

In the case of developing idle speed control algorithms, allowing for nonstationarity and nonrepeatability of the engine is well-founded. Therefore, a special submodule has been introduced into the presented model allowing the researchers to closely approximate the actual behaviour of the engine. The adequacy of the model has been proved in tests.

The courses of engine speed stabilization at the step increase and decrease of load in the case of the efficient sensor or its two failures are not identical.

REFERENCES

1. Ando H., Motomochi M.: Contribution of Fuel Transport Lag and Statistical Perturbation in Combustion to Oscillation of SI Engine Speed at Idle*, SAE Technical Paper* ,nr 870545, 1987.
2. Ashhab M.-S. S., Stefanopoulou. A. G, Cook J. A., Levin M. B.: Control-Oriented Model for Camless Intake Process (Part I*) ASME Journal of Dynamic Systems, Measurement, and Control*, Vol 122, 2000, pp. 122-130.
3. Bidan P., Kouadio L.K, Valentin M, Montseny G.: Electrical assistance for S.I. idle-speed control*, Control Engineering Practice*, 6, 1998, pp. 829-836.
4. Cassidy J.F. , Athans M., Lee W.H.: On the Design of Electronic Automotive Engine Control Using Linear Quadratic Control Theory*, Transaction on Automatic Control*, IEEE 1980.
5. Dotoli M.: Fuzzy Idle Speed Control: A Preliminary Investigation*, Report on the research* no. 97-E-858, Department of Automation, Technical University of Denmark 1997.
6. Dziubiński M.: Reliablity aspects in constructing vehicle electronic equipment*, Journal of Middle European Construction and Design of Cars,*pp. 2003, 44-48 Prague.
7. Dziubiński M.: Computer Control of Alternators Manufacture, *Manufacturing Engineering,*4, 2003, pp.57-61.
8. Dziubiński M.: Investigations of ignition systems reliability, *Journal of KONES Internal Combustion Engines,2004, ,*pp. 137-146 Warsaw.
9. Dziubiński M.: Reliability aspects in manufacturing of starter. *New Ways in Manufacturing Technologies, 2004, pp.385-390 .*
10. Jung M., Glover K.: Control-Oriented Linear Parameter Varying Modelling of a Turbocharged Diesel Engine*, Proceedings of the IEEE Conference on Control Applications*, Istanbul, Turkey, June 2003.
11. Gustaffson T.K., Makila P.M.: Modelling of Uncertain Systems with Application to Robust Process Contro*, Journal of Process Control*, No 11, 2001.
12. Han M., Loh R.N.K., Wang L., Lee A., Stander D.: Optimal Idle Speed Control of an Automotive Engine*, SAE Technical Paper* ,nr 981059, 1998.
13. Hawryluk B.: *Stochastyczny model silnika benzynowego w aspekcie stechiometrycznego składu spalin*, praca doktorska, Politechnika Lubelska, 2001.
14. Powell B.K., Cook J.A., Grizzle J.W.: Modeling and analysis of inherently multi-rate sampling fuel injected engin idle speed loop*, Proc. of 1987 American Control Conference*, Minneapolis, USA, 1987.

15. Powell B.K., Powers W.P.: Linear Quadratic Control Design for Nonlinear IC Engine Systems*, Proc: ISATA Conference*, Stockholm, Sweden, 1981.
16. Powell J.D.: A review of IC engine models for control system design*, Proc. of the 10th IFAC World Congress*, Munich, Germany, 1987.
17. Powers W.F., Powell B.K., Lawson G.P.: Application of optimal control and Kalman filtering to automotive systems*,* . Int. J. Of Vehicle Design, Special Publication SP4, 1983.
18. Pukrushpan J., Stefanopoulou A., Varigonda S.: *Control-Oriented Model of Fuel Processor for Hydrogen Generation in Fuel Cell Applications"* IFAC Symposium in Adavnces in Automotive Systems, Salerno, 2004.
19. Shim D., Khargonekar P.P., Ribbens W.: *Idle speed control for automotive engine".* Proc. of 1995 American Control Conference, Seattle, USA, 1995.
20. Smith R.S. *Model Validation for Robust Control: an Experimental Process Control Application.* Automatica Vol. 31. No. 11, 1995.
21. Stefanopoulou A.G., Cook J.A., Freudenberg J.S., Grizzle J.W.: Control-Oriented Model of a Dual Equal Variable Cam Timing Spark Ignition Engine*,* ASME Journal of Dynamic Systems, Measurement, and Control, vol. 120,1998, pp. 257-266.
22. Wendeker M., Czarnigowski J.: Adaptive Control of the Idle Speed, *ICES2003-646 Proceedings of ICES03 2003 Spring Technical Conference of the ASME Internal Combustion Engine Division Salzburg,* Austria, May 11-14, 2003.
23. Wedeneker M., Czarnigowski J.: Adaptive Control Of The Idle Conditions Of Spark Ignition Engine*, SAE_NA Technical Paper* SAE_NA 2003-01-15, 2003.

NOMENCLATURE

F_{flow} – air flow area,
h_z – by-pass valve position,
I_b – engine moment of inertia,
m_{air} – mass of air reaching the cylinder,
M_b – engine friction torque,
$\dot{m}_{bp}$ – by-pass valve air mass flow rate,
$\dot{m}_{cyl}$ – cylinder mixture mass flow rate,
M_{dod} – additional load torque,
m_{film} – fuel film mass,
$\dot{m}_{fuel}$ – mass flow rate of fuel reaching the cylinder,
m_{fp} – mass of fuel evaporated from fuel film,
M_b – engine resistance torque,
M_i – engine indicated torque,
$\bar{M}_i$ – mean value of indicated torque,
M_{istch} – indicated torque by stoichiometric mixture,
$M_{i\sigma}$ – indicated torque estimated by stochastic calculation,
m_{if} – mass of fuel deposited in fuel film by single injection,
m_{inj} – mass of fuel supplied by single injection ,
m_{ip} – mass of fuel vaporized directly by single injection,
m_{mf} – mixture mass in manifold,
$\dot{m}_{tr}$ – mass flow rate of air reaching the manifold by throttle,
n – engine speed,
p_o – atmospheric pressure,

R – gas constant of air,
R_{mix} – air/fuel mixture gas constant,
RND – random variable (0-1),
t_{inj} – injection time,
T_{mf} – manifold air temperature,
T_o – ambient air temperature,
V_{cyl} – cylinder volume,
V_{mf} – manifold volume,
X – fuel deposited coefficient,
α – throttle position,
$\beta(\lambda)$ – air/fuel ratio correction,
Δt – calculation period,
$\Delta\alpha_z$ – spark advance,
η_v – volumetric efficiency,
μ – air flow coefficient,
λ – relative air/fuel ratio,
ξ – back propagation coefficient,
ψ – flow type coefficient,
τ – fuel film vaporisation time constant,
σ_{Mi} – standard deviation of indicated torque.

MODELOWANIE I WERYFIKACJA USZKODZEŃ UKŁADU WTRYSKOWEGO SILNIKA SPALINOWEGO

Streszczenie. Rosnące wymagania stawiane silnikom spalinowym przyczyniają się do wprowadzania nowych systemów sterowania Wymagania te związane są z opracowaniem nowych algorytmów sterowania, które powinny być wyznaczane drogą eksperymentalną i w postaci modelowania. Artykuł przedstawia przyjęte założenia i wymagania dotyczące modelowania w aspekcie sterowania wolnych obrotów silnika. Wybrane badania identyfikacji i weryfikacji w postaci testów przeprowadzono dla modelu silnika 1.5 GLI. Na bazie modelu silnika przeprowadzono symulacje typowych uszkodzeń czujników.

Słowa kluczowe: silnik spalinowy, wolne obroty silnika, sterowanie, symulacja uszkodzeń czujników.