MODEL OF THE RECIPROCATING ENGINE USING ITERATIVE PROCEDURES OF THE TRANSIENT TORQUE CALCULATION

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Abstract

Process of development of vehicles is nowadays preceded by extensive modelling tests, which allow evaluating influence of applicable design solutions on future utilitarian parameters of the vehicle. Here, estimation of vehicle's dynamics by measuring of acceleration time from 0 to 100 km/h can be an example. Similar, maximal speed, pollutants emission or fuel consumption under specific conditions of vehicle's operation can be the example. Modelling research allow to shorten time of introduction of new vehicle for production significantly. Most eager group of model used in simulation tests of vehicles, according to short time of calculation, is black box models. They do not take into consideration they cyclic type of operation, however, it is possible to identify fast and simple their coefficients. This work involves description of reciprocating engine tests, which have been done under transient and steady-state conditions. Transient conditions were forced by rapid change of engine control signal. During the tests differences of torque generated by engine under transient and steady-state conditions have been registered for the same control signal. Basing on the received results a new algorithm of transient torque calculation has been proposed. In the presented model value of the transient torque has been combined with the engine control signal and others engine parameters, which describe transient history of the engine. Hence, iterative procedures of the transient torque calculation with declared time step have been implemented.

Keywords: *reciprocating engines, transient conditions, torque calculation, iterative procedures, black box model*

1. Introduction

Process of development of vehicles is nowadays preceded by extensive modelling tests, which allow evaluating influence of applicable design solutions on future utilitarian parameters of the vehicle. Here, estimation of vehicle's dynamics by measuring of acceleration time from 0 to 100 km/h can be an example. Similar, maximal speed, pollutants emission or fuel consumption under specific conditions of vehicle's operation can be the example. Model research allows shortening time of introduction of new vehicle for production significantly [6, 9, 18]. Models of reciprocating engines can be potentially used in Adaptive Cruise Control (ACC) systems also, which enable to keep constant distance from previous vehicle [7, 15]. ACC system gauges a distance from previous vehicle and according to the obtained results makes a decision about possible change of speed of riding. Realization of proper acceleration or deceleration of vehicle follows through change of torque generated by engine or through using of braking system. Selection of course of the engine control signal, which will assure set up course of the torque, bases on using of an engine model. ACC system, which is used by a car, must be equipped with reciprocating engine model, which appoints course of the torque after change of the engine control signal.

Obtainment of cause-effect model of engine is possible on two manners [5]:

a) Mathematical modelling of engine operation cycle is realized by recognition of physical and chemical phenomenon, which goes together with engine operation [8]. Complexity of that phenomenon in engine does not allow, on our present knowledge, to define sufficient relations on theoretical way only.

b) State equations, which are mathematical form of cause-effect model, are obtained measuring physical parameters of interest. Expected relations are then defined using proper tables, graphs or functions, which approximate those relations. Models of these groups are named “black box” [2, 5, 10].

Most eager group of model used in simulation tests of vehicles, according to short time of calculation, is black box models. They do not take into consideration they cyclic type of operation, however, it is possible to identify fast and simple their coefficients. For realization of the purpose, the presented work is concentrated on (real time model operation, model coefficient identification for each engine), using black box model is optimal solution.

2. General characteristic of the engine model

Parameters of engine measured under steady-state and transient conditions differ between them. Above-mentioned problems are presented in numerous publications [1, 6, 16, 17, 19]. Internal Combustion Engines and Compressors Institute owns rich tradition in range of engine testing under transient conditions [3, 4, 13]. On Fig. 1 a scheme of engine test bed used for engine testing under transient conditions has been presented. Applied methods allowed calculating transient torque M_d of the tested engine.

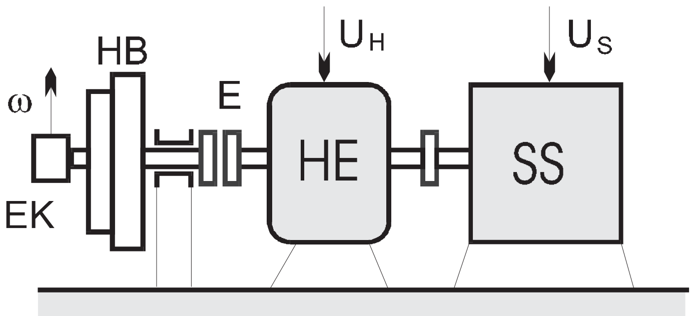

Fig. 1. Scheme of the engine test bed: SS – internal combustion engine, HE – electric brake, E – electromagnetic clutch, HB – inertia brake, EK – encoder

Test of engine’s crankshaft acceleration [3] follows possible fast change of engine control signal (Fig. 2). Engine’s crankshafts is braked by moment of inertia of rotating test bed elements (SS, HE, E, HB) only. During the test, angular speed of the engine’s crankshaft ω is registered and angular acceleration ε is calculated according to special procedure next [3]. For defined operation conditions (control signal U_s, angular speed ω, thermal state defined by cooling fluid and oil temperatures) engine generates under steady-state conditions a torque named static M_S. Sets of this way obtained points present static characteristic of the engine (Fig. 3):

$$M_S = M(\omega)\,\big|_{U_S = const}\,. \tag{1}$$

Transient torque M_d for engine’s crankshaft acceleration is calculated using Newton’s rule, assuming: electric brake, electromagnetic clutch are excluded and course of angular acceleration $\dot{\omega}$ is known (Fig. 3):

$$(J_{HB} + J_S)\dot{\omega} = M_d - M_{HB}(\omega), \tag{2}$$

where:
J_{HB} – moment of inertia of the inertia brake,
J_S – moment of inertia of the engine,
$M_{HB}(\omega)$ – torque of movement resistance as a function of angular speed.

Knowing relations (1) and (2) following equation has been defined:

$$\Delta M = M_s - M_d \text{, for US=USR.} \quad (3)$$

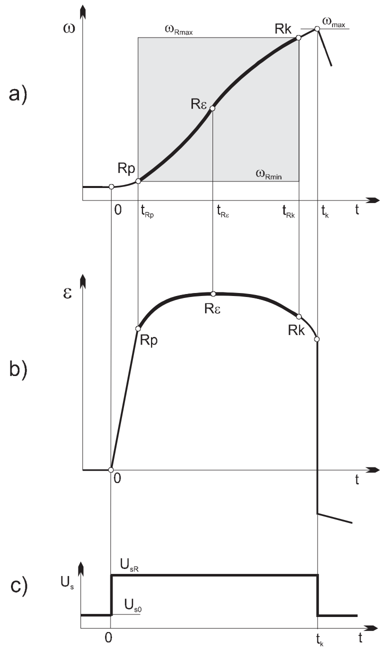

Fig. 2. Course of measured angular speed ω (a) as a replay of the engine, connected with inertia of rotating elements, to rapid change of the engine control signal Us (c); course of counted angular acceleration ε (b). Thick line shows part of the course between points Rp and Rk being taken into consideration

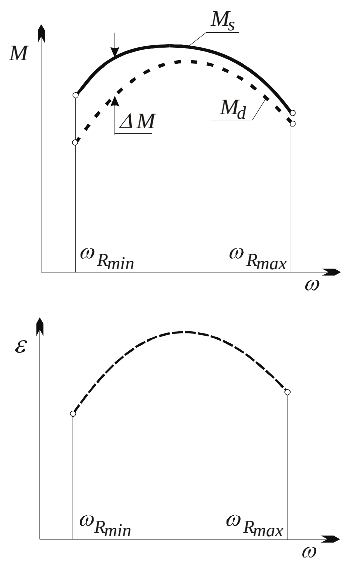

Fig. 3. Course of static MS and transient Md torque and their difference ΔM against angular speed of the engine's crankshaft (upper figure). Course of angular acceleration against angular speed of the engine's crankshaft (lower figure)

Algorithm of transient torque calculation has been proposed basing on the observation that transient torque is dependent on the engine control signal (after its rapid change) and pointed below parameters, which describe transient history of the engine. On Fig. 4 diagram of the course of the transient torque after rapid change the engine control signal has been presented. When $t \in \langle t_o ; t_1 \rangle$ it can be assumed that engine operates under steady-state conditions and angular speed is constant. Hence, the transient torque is approximately equal to the static torque:

$$Ms=f(Us, \omega) \quad (4)$$

for the same engine operation signal.

When $t \in \langle t_1; t_2)$, for the beginning of the acceleration process, angular speed is constant no longer and distinct difference between static M_S and transient M_d torque appears, here defined as $\Delta M|_{t=t_1}$, which following the acceleration process decreases. Consequently, static torque (calculated using static characteristic of the engine) is constant no longer, because angular speed of the engine's crankshaft has been changed. In this situation, it is logically theoretical parameter, which can be obtained under steady-state conditions (U_s = const, ω = const). For the final phase of the acceleration process $t \in \langle t_2; t_3 \rangle$ it can be seen that system stabilizes $\omega \cong$ const, $M_d \cong M_s$, $\Delta M \to 0$. Hence, it can be assumed, that transient torque, after rapid change of the engine control signal, can be calculated according to the relation:

$$M_d = M_s - \Delta M , \quad (5)$$

where:

$$\Delta M = \Delta M\big|_{t=t_1} \cdot k(t), \quad (6)$$

$$\Delta M\big|_{t=t_1} = M_s\big|_{\substack{U_S=U_{S_2}\\ \omega=\omega_1}} - M_d\Big|_{t=t_1}, \quad (7)$$

$$k(t) = e^{-\frac{t-t_1}{T_M(\omega_1, \Delta U_S)}}, \quad (8)$$

$$\Delta U_S = U_{S_2} - U_{S_1}. \quad (9)$$

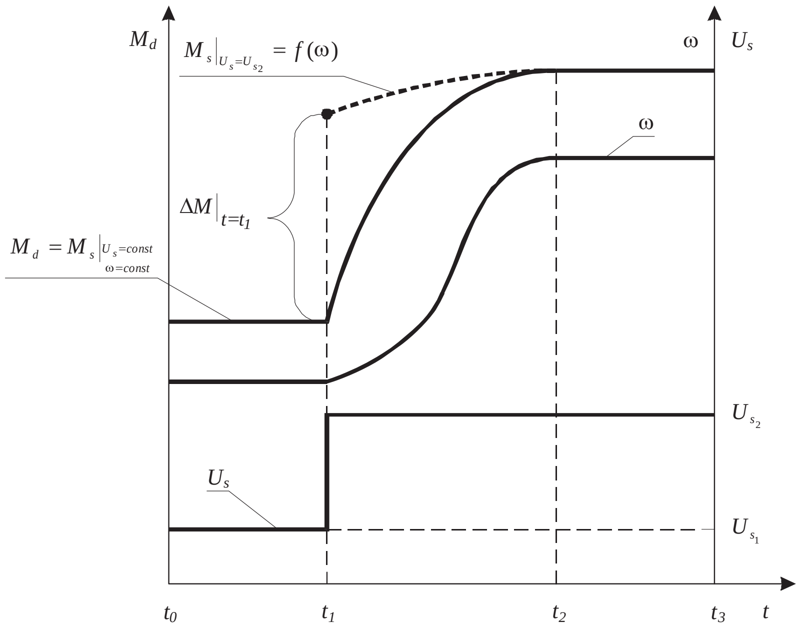

Fig. 4. Diagram of the course of transient torque Md and angular speed ω after rapid change of the engine control signal Us against time t

Value of the transient torque $M_d\big|_{t=t_1}$ (7), after next engine control signal change, should be taken from previous calculations according to the relation (5). According to the assumptions, which have been made above, the static torque M_s and the parameter T_M (8) are functions of two parameters: ω and ΔU_S. Both characteristics should be determined during engine tests. For their best approximation, three-dimensional functions can be used, which have been worked out in Internal Combustion Engines and Compressors Institute [12, 14] basing on the Spline function. Taking into account information, that reciprocating engine is dynamically nonlinear and nonsymmetrical object [4], second counting block should be prepared for events, when engine is used as a brake.

Figure 5 shows model of the reciprocating engine using iterative procedures of the transient torque calculation for condition $\frac{\Delta U_S}{\Delta t} > 0$, using equations (4) - (9).

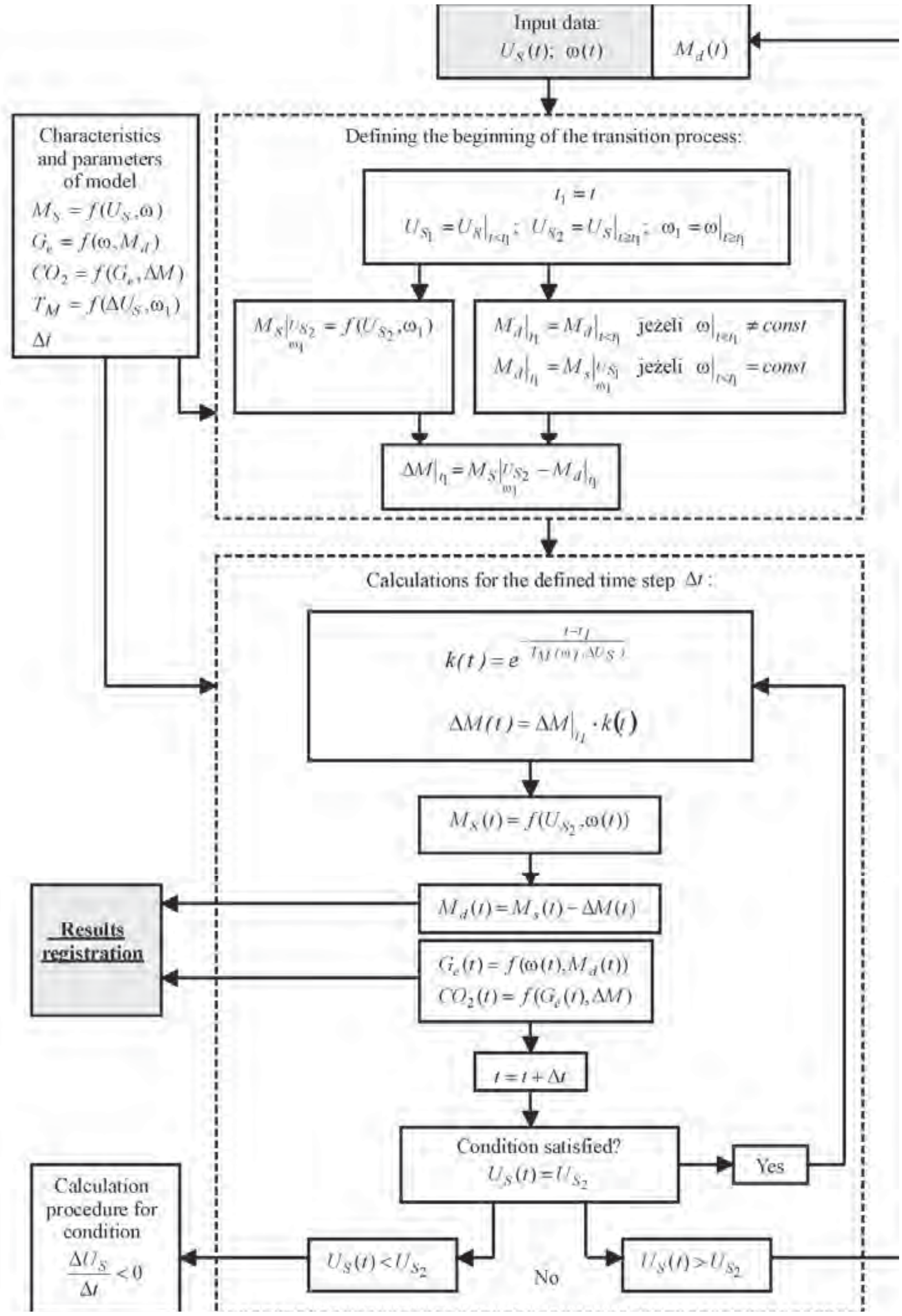

Fig. 5. Model of the reciprocating engine using iterative procedures of the transient torque calculation for condition $\frac{\Delta U_S}{\Delta t} > 0$

3. An example of the use of iterative procedures for determining the transient torque

The following results refer to tests of a passenger car equipped with an internal combustion engine spark ignition with a capacity of 1.8 dm^3 and a mass of 1343 kg. Condition of engine load resulted from the ongoing cycle of driving in traffic conditions and the control of propulsion system. Vehicle traction parameters were measured using a GPS system with a phenomenological correction of the height signal [11]. Selected parameters of the engine (among others: speed, fuel consumption, etc.) were measured using the system communicating with the onboard network CAN. The torque was calculated using the model in the form of bond graph [2, 14]. Fig. 6 contains a comparison of the torque generated by the engine after the change of control signal, while driving with 3rd, 4th and 5th gear. This situation corresponds to an intense acceleration on the road after a sharp pressing the accelerator pedal. Before accelerating the vehicle operates at a constant speed with the engine speed range: 900-1100 rpm. The initial value of control signal U_{S1} (Fig. 4) is slightly different for each gear and it is close to 0%. The value of control signal after a step change is the maximum for all gears (U_{S2} = 100%).

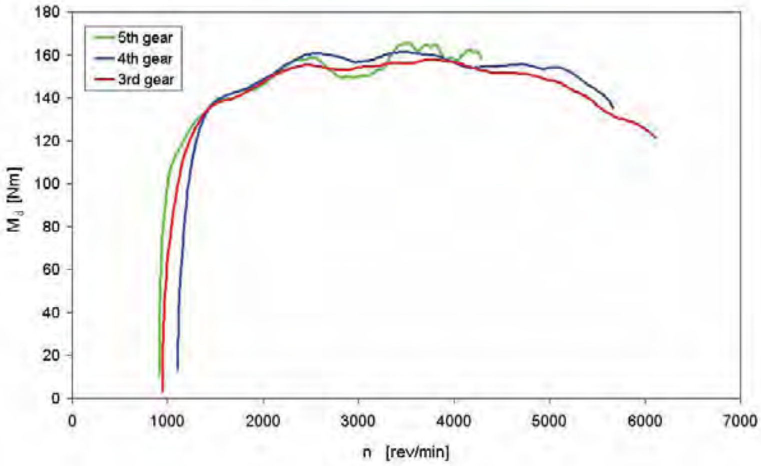

Fig. 6. Comparison of the transient torque generated by the engine after step change of the control signal (US2=100%), while driving with 3rd, 4th, and 5th gear

Assuming that the acceleration of the engine with 5th gear takes place the most slowly, it is closest to the static characteristic obtained at constant engine speed [3, 13]. In Fig. 6 can be observed the disappearance of the differences in torque values recorded at a speed of about 1400 rpm. It can therefore be assumed that some time after the start of the accelerating process the static and transient torque are equal. This observation is confirmed in earlier work [3, 13]. Observed at higher speeds, the differences come from the uniqueness of the settings used by the motor controller and measuring errors. In the rest of the work it has been decided to use the course obtained during acceleration process with 3rd gear after 1400 rpm as the external (U_S=100%) static characteristic of the engine. It was decided to use the characteristic as the static one due to the technical ability to achieve the maximum engine speed.

Figure 7 shows the changes of the transient torque during engine accelerating with 3rd gear after the step change of the control signal (U_{S2}=100%). Acceleration took place for different initial speeds: 1407, 1840, 2870 and 3280 rpm. The obtained courses have been marked in a simplified manner, respectively: Md 1400, Md 1800, Md 2900, and Md 3300. In Fig. 7 shows also the external static characteristic of the engine, indicated by Ms. This characteristic has been obtained from the process of the engine accelerating in the manner described above. In Fig. 7 it can be observed, as well as in Fig. 6, the balancing of static and transient torque after a certain period of time since the start of the accelerating process, for each test. In accordance with the adopted model (Fig. 5) the difference between static and transient torque is described by the relation (6) and it is a function of time elapsed since the control signal changes. Using the available results of tests the calibration of the engine model (Fig. 5) for the case when U_{S2} = 100% has been performed.

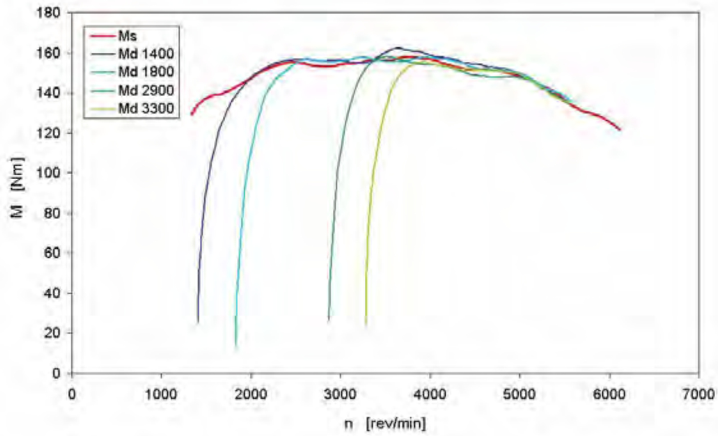

Fig. 7. Comparison of the transient torque generated by the engine after step change of the control signal (US2=100%), while driving with 3rd gear for different initial engine speeds

Figure 8 presents the courses of the measured difference between static and transient torque (indication: test) and calculated using the engine model (indication: model) for the engine acceleration with the 3rd gear for the initial speed 2337 rpm.

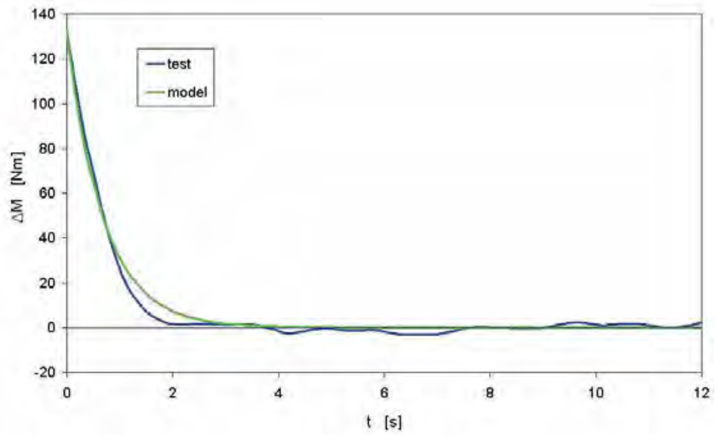

Fig. 8. Comparison of the static and transient torque generated by the engine after step change of the control signal (US2=100%), while driving with 3rd gear for the initial speeds 2337 rpm

The presented results allow to conclude that the proposed model has a high accuracy in the tested field (U_{S2}=100%). The difference in the T_M parameter values (8) calculated on the basis of the model and measured is 0.04 s (relative error of 6%). The results presented in Fig. 8 allow also to observe the high compliance of the course registered (indication: test) with the modelled one (indication: model). For both courses the balancing of static and transient torque after a certain period of time since the start of the accelerating process can be observed.

The presented study involved only selected processes (U_{S2}=100%), because of technical problems obtaining the rectangular course of motor control signal for values less than 100%. Accelerator pedal should stay at a predefined position, other than the extreme, after his stepwise push. The implementation of such trials will be possible using a mechanism, which limiting step of the accelerator pedal.

4. Conclusions

Presented in this work model of the internal combustion engine calculates the value of the transient torque using control signal and the dynamic history of engine operation. The presented results confirm the validity of such a description of the phenomenon being modelled. It can be also noted that some time after the start of the accelerating process the balancing of static and transient torque takes place. Using the method of acceleration of the engine under load resulting from the vehicle driving the approximate static characteristic of the engine can be calculated. The more this process will be slowed, for example by using 5th gear, the greater accuracy will be achieved with determination of static characteristics. The proposed model is characterized by good stability due to the use of static characteristics and the simplicity of calibration. In this work, to calibrate a simplified model (for U_{S2}=100%) only 5 tests of engine acceleration have been used.

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