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CONTROL SYSTEMS FOR TECHNOLOGICAL OBJECTS

Design of Diesel Engine Mathematical Model Oriented to Speed Control

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Abstract—We provide the synthesis procedure for a mathematical model of the 20ChN26.5/31 diesel engine treated as a part of 20EDG500 diesel-electric set (power 6.3 MW). The model is designed for synthesis of engine speed control and simulation of closed-loop control systems in generator sets. The model structure is based on the fundamental laws of physics, while its parameters and static functions are obtained using least squares approach and the data taken during experimental testing of the set. The results of model verification are presented, and the model outputs are compared with the experimental data. Simulation of the control system closed by proportional-integral-differential law is presented as an example of the model application.

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INTRODUCTION

Nowadays, diesel-power generator sets are widely used due to their independence from power supply and portability. Electrical power generation by autonomous source is actual where power grids are not accessible or where power backup is required for increase of customer electrosafety. Medium and high power generator sets used for power supply of hard-to-reach regions, in military and civil transport with electric powertrains, in electric generation stations as well as for energy backup in medical, governmental and other facilities are of primary concern.

Since modern diesel-electric sets are widely utilized and seriously affect the environment, high requirements to their operating are tightening. The requirements are increasingly restricted to reliability (especially for strategically important facilities), noise protection, precision, transients as well as fuel, economy and ecological quantities. For instance, in highly developed countries and in many developing countries such emission standards as *Tier* standards (Environment Protection Department in the United States) or *Stage* (European Union) are introduced and become progressively more stringent standards at the legislation level. In this regard and due to restrictive fuel source all worldwide diesel manufacturers are working on reduction of fuel consumption.

In general, there are two ways of diesel-electric sets development. The first way is related to improvement of the construction of engines and generators: the geometry of the air, fuel and oil delivery systems, exhaust gas disposal system, electric circuits and rotating parts of engines/generators, etc. The second way is involved with improvement of diesel-generators control systems regulating the output frequency and voltage, and indicating emergencies. A major breakthrough to development of diesel-generators was due to implementation and improvement of microprocessor-based systems that permitted to design arbitrary complex and efficient control laws taking into account construction features, generator dynamics and environment conditions. Besides, the use of modern computational systems offers high potentials for satisfaction of both present and future requirements and standards to diesel-generator sets.

It is known that diesel-electric generator sets are complex nonlinear systems consisting of many subsystems and units and operate in presence of loads and disturbances. A number of variables characterizing the motion of diesel-generator sets is subjected to amplitude and phase distortions. Some variables cannot be measured directly and can only be evaluated. In this regard, to increase the efficiency of diesel-generator sets we have to accurately predict its dynamics and evaluate the speed, voltage and frequency responses to input signals (controls, load and disturbances) and design features. This motivates the design of high-efficient mathematical models oriented to control algorithms synthesis and experimental testing of these algorithms in control systems of diesel-generator sets.

During the last 30 years, the intensive research in the area of diesel engine modeling was carried out basically for automotive industry, and this gained experience can be applied for energetic systems. Conditionally, the engine models presented in publications can be classified as follows.

1. Linear or piecewise-linear models (see [1–3]) are oriented to design of engine speed and air-to-fuel ratio control via well-known and highly developed classical theory of linear systems. As a result of these models applications piecewise linear local controls are designed. Each local control provides required performance within a regime corresponding to some range of pressures, speeds, delivery begin angles, etc. Despite the simplicity of control structure and relatively high performance of local controls, design of the whole sets of both local models and controls can be a laborious procedure and can be essentially complicated due to increasing requirements to control system performance.

2. Nonlinear physical models (see [4–10]) are based on fundamental physical laws and are represented in the form of nonlinear differential equations connecting control variables and disturbances together with regulated and state variables. Unlike linear models, nonlinear models are global, and control algorithms based on these models are global and can be applied for all the regimes of diesel-generator operating. At the same time the algorithms are derived relatively easy on the basis of nonlinear control theory. Since the model design is physically motivated, there is every reason to believe that the derived control will be implementable in practice. At the same time, it is not always possible to use physical laws for description of processes in the diesel. For example, volumetric efficiency depends on intake geometry, engine speed, air pressure after compressor [5, 11–14] and is usually evaluated empirically from approximating equalities. Turbine and compressor efficiency, delivery begin efficiency, fuel-to-air ratio, combustion efficiency, friction torque and other characteristics are evaluated in the same way. These evaluations require preliminary time consuming experiments, otherwise the characteristics are averaged and assumed constants what leads to the lost of model performance.

3. Empirical models (see [15, 16]) are represented as abstract structures with parameters calculated experimentally during preliminary or adaptive (online) identification. As an example, an artificial neural network or a linear regressive model can be considered. A suitable choice of the structure and parameters identification permit to improve the model performance without detailed investigation of engine processes. At the same time calculation of the model parameters requires teaching sample of experimental data that covers all possible regimes of diesel engine operating. Besides, the abstract structure usually does not permit direct derivation of physically implementable control law. Exception to this are so called inverse models [17, 18] that are also empirical and are taught over the same data sample. These models are usually used for implementation of feedforward loop in a control law.

Thus, it is quite important to find a tradeoff between the physically inspired choice of models structure, performance and simplicity, that allows us to directly derive a control. In this paper the model of diesel internal combustion engine of generator set is designed. The model is oriented to experimental research and design of engine speed control systems. The model structure is based on the fundamental laws of physics, while its coefficients and parameters are to be identified via experimental data obtained during the testing of the 20-cylinder V-type 20ChN26.5/31 diesel engine as a part of 20EDG500 diesel-electric generator set[1] of power 6.3 MW (see Fig. 1). The engine is equipped with two crankshafts, two turbochargers, two intake and exhaust manifolds.

The work of the turbochargers essentially influents on the engine speed. As a sequence, the model contains equations describing the dynamics of turbines and compressors together with the aerodynamics of airflows after the compressors and the exhaust gas flows before the turbines.

For the sake of simplicity of paper presentation all the variables and constants of the model are described in the Table 1.

The paper is organized as follows. In Section 2, input and output variables of the model are selected, and the engine is decomposed into several interconnected subsystems characterizing certain physical processes. Each subsystem is described by nonlinear differential and algebraic equations. Then the models are sampled and reduced to linear regression forms. Least squares approach is applied to identify the parameters of these regressions. In Section 3 an example of the model use is presented, and finally simulation results for the closed-loop control system are illustrated.

[1] 20EDG500 diesel-electric generator set is manufactured by OAO Kolomenskii zavod and is used as a part of the diesel-electric generator set in the safety train of atomic power plant.

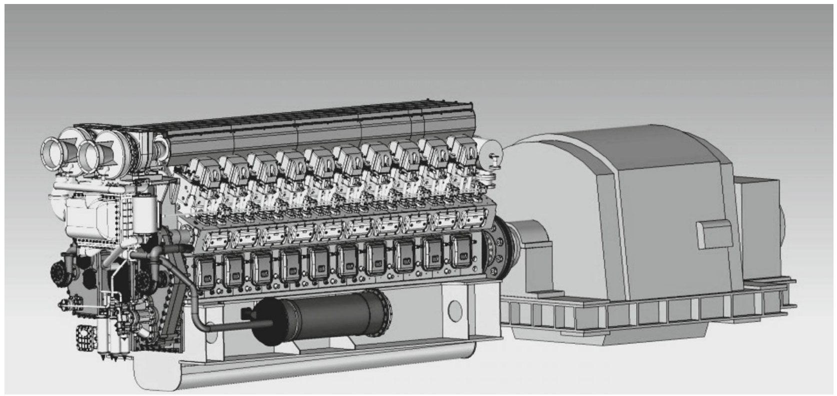

Fig. 1. Draft of the diesel-electric set 20EDG500.

1. PROBLEM STATEMENT

The problem is to design a mathematical model for a diesel engine (treated as a part of a diesel-electric generator set) oriented to the engine simulation and synthesis of crankshaft speed control. Required maximum steady-state error is to be less than 1 percent, while transient error is to be less than 15 percent. These boundaries are related to standard restrictions of stabilization of electrical frequency at the output of generator set.

2. SYNTHESIS OF MODEL

The engine model relates the regulated variable with control variables and disturbances via state variables. The control variables are the delivery begin angle ζ and the fuel pulse width Δ_f. The regulated variable is the engine speed ω. State variables are represented by the speed ω_c of the turbine rotation, the air pressure P_m in the intake manifold, and the pressure P_e of the exhaust gas after the cylinders. The active electric load power P_{gen} is defined as external disturbance. The model uses variables averaged per one engine stroke.

The model is designed under the following assumptions.

Assumption 1. The fuel pulse shape and the pressure of the injected fuel are constant.

Assumption 2. Variations of the air temperature in the intake manifold T_{im} and the exhaust gas temperature T_{em} are neglected.

Assumption 3. The air entering the cylinders and the exhaust gases are assumed to be ideal gases.

Assumption 4. As it is written before, 20ChN26.5/31 engine has two turbochargers, two intake collectors, and two exhaust manifolds. The engine is equipped with two crankshafts. To simplify the model, we assume that the variables ω, ω_c, P_m, and P_e are averaged over the pairs of the shafts, turbines, and the manifolds, respectively.

Assumption 1 allows us to simplify the computation of the mass of the fuel injected, which is supposed to be proportional to fuel pulse width Δ_f. This assumption is satisfied for perfectly stabilized fuel pressure in the injectors.

Assumption 2 implies that the temperatures T_{im} and T_{em} in the model equations are constant.

Assumption 3 permits to use the ideal gas equations in the model.

The model structure is presented in Fig. 2; it includes the following four interconnected subsystems:

—the crankshaft speed;

—intake air pressure after the compressor;

—the turbine speed;

—exhaust gas pressure.

Consider each subsystem and identify the unknown functions and parameters for it.

Table 1. Variables and constant parameters of diesel engine model

Designation	Notation
c_0, c_1, and c_2	Friction coefficients
J	Moment of inertia, reduced to motor shaft of diesel-electric generator set, kg m
J_c	Turbine moment of inertia, kg m
k , k_1, k_2, r, and c	Constant coefficients depending on design features of diesel-electric generator set, turbocharger and fuel efficiency
m_f	Fuel volume flow, mm^3/cycle
M_{fr}	Friction torque, N m
M_i	Indicator torque of the engine, N m
M_{pump}	Torque of pumping losses, N m
N_c	Power produced by compressor, W
N_t	Power produced by turbine, W
P_a	Ambient air pressure, bar
P_{gen}	Active generator power, kW
P_m	Air pressure in the intake manifold (after compressor), bar
P_e	Exhaust gas pressure after cylinders, bar
P_{em}	Exhaust gas pressure after compressor, bar
R_a	Ideal gas constant for air, J/kg K
R_e	Ideal gas constant for exhaust gases, J/kg K
R_c	Compressor blade radius, m
s	Function depending on compressor pressure ratio P_m/P_a
T_a	Ambient air temperature, K
T_{em}	Exhaust gas temperature, K
T_{im}	Intake manifold temperature after turbine and intercooler, K
V_{im}	Intake manifold volume (after compressor but before cylinders), m^3
V_{em}	Exhaust manifold volume (after cylinders but before turbine), m^3
V_d	Cylinders volume, m^3
W_c	Air-mass flow into cylinders, kg/s
W_{ei}	Air-mass flow after compressor, kg/s
W_{eo}	Mass flow of exhaust gases from cylinders, kg/s
W_t	Mass flow of exhaust gases into turbine, kg/s
Δ_f	Fuel pulse width, ms
Φ_c	Volumetric flow coefficient
φ	Function depending on turbine compression ratio P_e/P_a
ζ	Delivery begin angle, degree to the top dead center
η	Diesel efficiency
η_c	Compressor efficiency
η_m	Turbocharger efficiency
η_{tm}	Turbine efficiency
η_{vol}	Volumetric efficiency
ω	Crankshaft speed, rpm
ω_c	Turbine speed, rpm

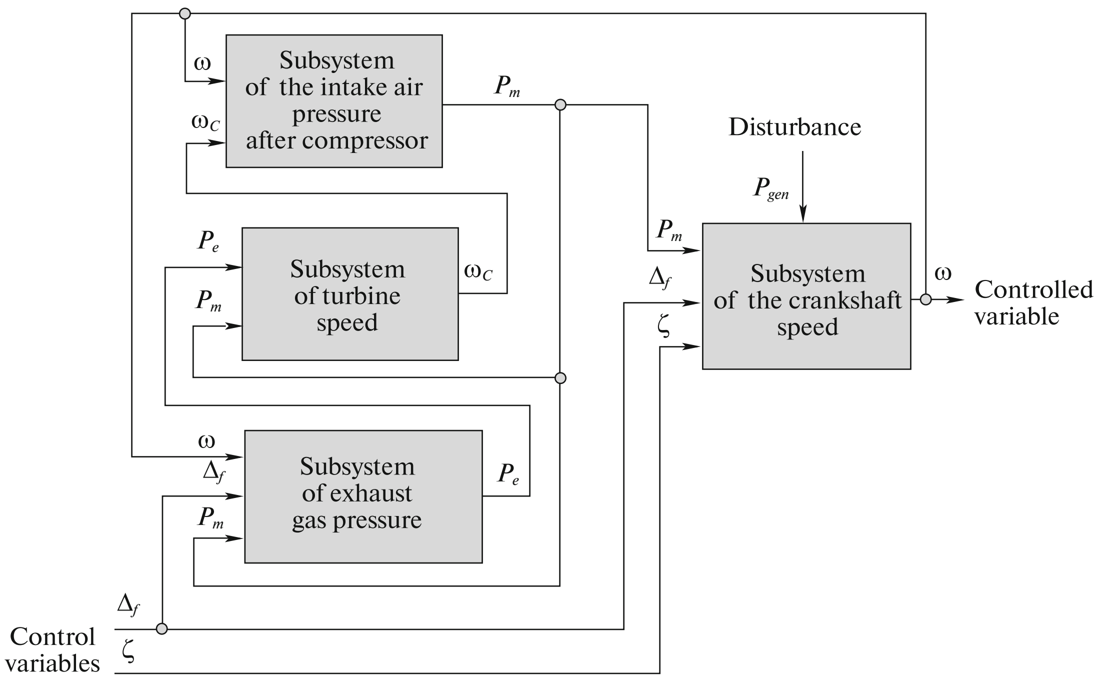

Fig. 2. Structure of the engine model.

2.1. Subsystem of the Crankshaft Speed

The subsystem model allows us to compute speed ω, using the fuel pulse width Δ_f, delivery begin angle ζ, the air pressure P_m, the exhaust gas pressure P_e and the generator power P_{gen} (load). The subsystem is described by the following equation (see [19–21]):

$$\dot{\omega} = \frac{1}{J}\left(M_i - M_{fr} - M_{pump} - \frac{P_{gen}}{\omega}\right), \tag{2.1}$$

where the indicator torque M_i, the friction torque M_{fr}, and the torque of pumping losses M_{pump} are defined by equalities

$$M_i = k_1\,\eta(\Delta_f, \zeta, \omega, P_m)\Delta_f, \quad M_{fr} = c_0 + c_1\omega + c_2\omega^2, \quad M_{pump} = k_2(P_m - P_e), \text{ respectively.} \tag{2.2}$$

The constants k_1, k_2, c_0, c_1, c_2 and the function $\eta(\cdot)$ are to be identified.

The function η is approximated by Taylor series with respect to all its arguments:

$$\eta(\Delta_f, \zeta, \omega, P_m) \approx \sum_{i=1}^{15} d_i\varphi_i = d^{\mathrm{T}}\varphi, \tag{2.3}$$

where $d_i, i = \overline{1,15}$ are constants,

$$\varphi = \left[1, \zeta, \zeta^2, \zeta\omega, \zeta P_m, \zeta\Delta_f, \omega, \omega^2, \omega P_m, \omega\Delta_f, P_m, P_m^2, P_m\Delta_f, \Delta_f, \Delta_f^2\right]^{\mathrm{T}}. \tag{2.4}$$

To identify all the constants in (2.1)–(2.3) we discretize the equations (2.1)–(2.4) by applying the Euler method and get:

$$\omega(k+1) = \omega(k) + \frac{\tau}{J}\left(M_i(k) - M_{fr}(k) - M_{pump}(k) - \frac{P_{gen(k)}}{\omega(k)}\right),$$

where τ is the sampling time.

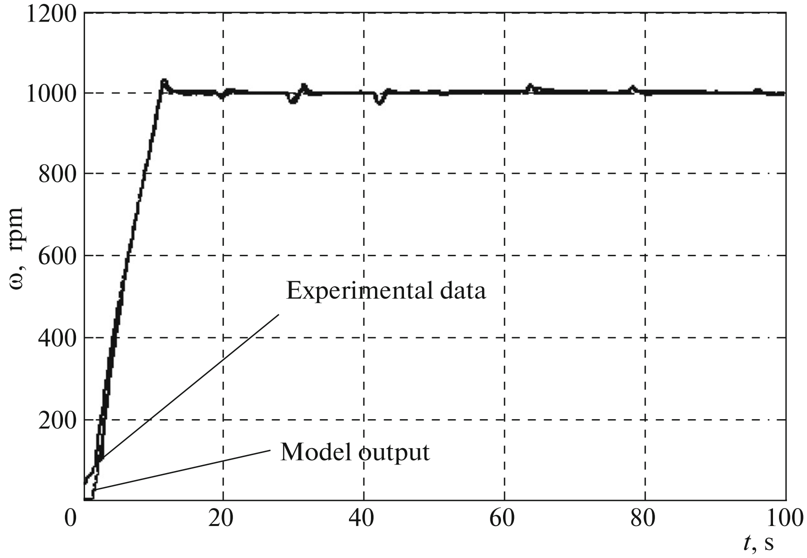

Fig. 3. The verification results of the engine speed model.

By replacing (2.2)–(2.4) in the last expression we obtain linear regression model

$$\omega(k+1) = \theta^{\mathrm{T}}\mu(k), \tag{2.5}$$

where θ is the vector of parameters to be identified given by

$$\theta = \left[\frac{k_1\tau}{J}d^T, -\frac{c_0\tau}{J}, 1-\frac{c_1\tau}{J}, -\frac{c_2\tau}{J}, -\frac{\tau}{J}, -\frac{\tau k_2}{J}\right]^{\mathrm{T}},$$

while

$$\mu = \left[\Delta_f, \zeta\Delta_f, \zeta^2\Delta_f, \zeta\omega\Delta_f, \zeta P_m\Delta_f, \zeta\Delta_f^2, \omega\Delta_f, \omega^2\Delta_f, \omega P_m\Delta_f,\right.$$

$$\left.\omega\Delta_f^2,\ P_m\Delta_f, P_m^2\Delta_f, P_m\Delta_f^2, \Delta_f^2, \Delta_f^3,\ 1, \omega, \omega^2, P_{gen}/\omega,\ P_m - P_e\right]^{\mathrm{T}}.$$

Representation (2.5) allows us to apply the algorithm of least squares to identify the parameters θ (see [22]):

$$\begin{cases} \hat{\theta}(k+1) = \hat{\theta}(k) + \gamma(k)\left(\omega(k) - \hat{\omega}(k)\right), \quad \hat{\theta}(0), \\ \gamma(k) = \dfrac{\chi S(k)\mu(k)}{1 + \chi\mu^{\mathrm{T}}(k)S(k)\mu(k)}, \\ S(k+1) = \chi(S(k) - \gamma(k)\mu^{\mathrm{T}}(k)S(k)), \quad S(0) = \sigma I \succ 0, \end{cases} \tag{2.6}$$

where $\chi \in (0,1]$ is the forgetting factor and σ is a positive constant.

Vectors μ and ω are recorded during test of diesel engine 20ChN26.5/31. The frequency of data acquisition corresponds to $\tau = 0.1$ sec. The testing time is 100 s.

The results of model verification via experimental data are presented in Fig. 3 and illustrate the model precision comparable with precision requirements for automatic control system of diesel-electric generator sets. The maximal error is 33.84 rpm (3.4% of maximum speed).

Note, if all the elements of the vector μ are measurable online, the model of speed subsystem is self-sufficient for design and implementation of the control law. To simulate processes in the speed control system, we have to (additionally) compute manifold air pressure P_m, exhaust gas pressure P_e, and the turbine speed ω_c (see Fig. 2). In this regard, we present the models of the corresponding subsystems.

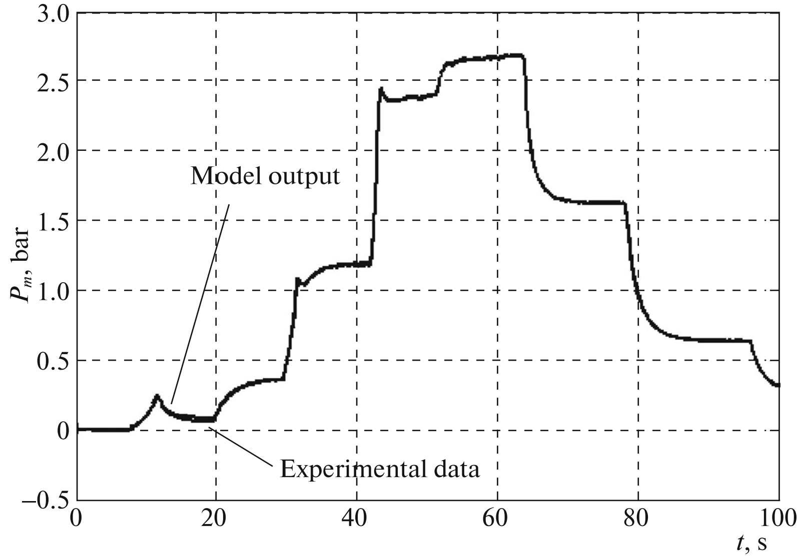

Fig. 4. The verification results of the intake air pressure model.

2.2. Subsystem of the Intake Air Pressure after the Compressor

The subsystem model makes it possible to compute the pressure using the crankshaft speed ω and turbine speed ω_c via Ideal Gas Equation differentiated with respect to time (see [23, 24]):

$$\dot{P}_m = \frac{R_a T_{im}}{V_{im}}(W_c - W_{ei}), \tag{2.7}$$

where the air-mass flow W_c after the compressor and the airflow W_{ei} to the cylinders are given by

$$W_c = \frac{P_a \pi R_c^3}{R_a T_a}\Phi_c(\omega_c)\omega_c \tag{2.8}$$

and

$$W_{ei} = \frac{\eta_{vol}(P_m, \omega)V_d}{120 R_a T_{im}} P_m \omega, \tag{2.9}$$

respectively.

Replacing (2.8) and (2.9) in (2.7) and substituting derivative $\dot{P}_m$ for the finite difference we obtain the discrete equation in the form

$$P_m(k+1) = P_m(k) + \frac{\tau T_{im}}{V_{im}}\left(\frac{P_a \pi R_c^3}{T_a}\Phi_c(\omega_c(k))\omega_c(k) - \frac{\eta_{vol}(P_m(k), \omega(k))V_d}{120 T_{im}} P_m(k)\omega(k)\right).$$

Then, expanding the right part of the equation in the Taylor series up to the second power, we get the regression model

$$P_m(k+1) = \theta_a^{\mathrm{T}} \mu_a(k), \tag{2.10}$$

where θ_a is the vector of the parameters to be identified and $\mu_a(k)$ is the vector of the measurable functions

$$\mu_a = \left[\omega_c, \omega_c^2, \omega_c^3, P_m\omega, P_m\omega^2, P_m\omega^3, P_m^2\omega, P_m^3\omega, P_m^2\omega^2, P_m\right]$$

obtained from the expansion. Representation (2.10) allows us to apply the algorithm of least squares analogous to the algorithm (2.6).

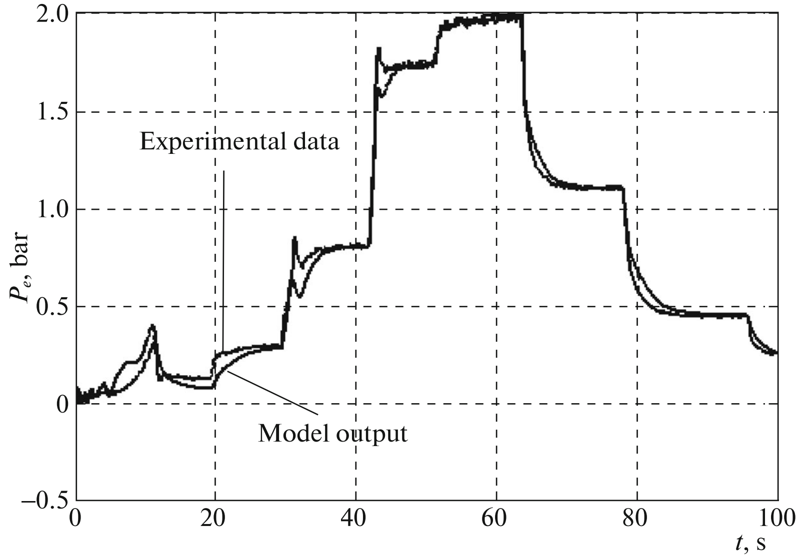

Fig. 5. The verification results of exhaust gas model.

The verification results for the subsystem model are presented in Fig. 4. It is seen from the figure that the maximal error of the model is 0.053 bar (2% of the maximal pressure).

2.3. Subsystem of the Exhaust Gas Pressure After the Cylinders

The pressure P_e depends on the crankshaft speed, the fuel pulse width Δ_f, and the intake air pressure P_m. We compute P_e via Ideal Gas Equation [25, 26]:

$$\dot{P}_e = \frac{R_e T_{em}}{V_{em}}(W_{eo} - W_t) \tag{2.11}$$

where the flows W_{eo} and W_t are calculated using equalities

$$W_{eo} = K\omega\Delta_f + \frac{\eta_{vol}(P_m, \omega)V_c}{120 R_a T_{im}} P_m \omega \tag{2.12}$$

and

$$W_t = f(P_e, T_{em}), \tag{2.13}$$

with the function f computed empirically.

Replacing (2.12) and (2.13) in (2.11) and substituting $\dot{P}_e$ for the finite difference we have:

$$P_e(k+1) = P_e(k) + \frac{\tau R_e T_{em}}{V_{em}}\left(K\omega\Delta_f + \frac{\eta_{vol}(P_m, \omega)V_c}{120 R_a T_{im}} P_m\omega - f(P_e, T_{em})\right).$$

By applying Taylor series to the last equality up to the second power we obtain the linear regression in the form

$$P_e(k+1) = \theta_e^{\mathrm{T}} \mu_e(k), \tag{2.14}$$

with the vector of parameters θ_e to be identified and vector of measurable functions

$$\mu_e = \left[1, P_e, P_e^2, \Delta f\omega, P_m\omega, P_m^2\omega, P_m^3\omega, P_m\omega^2, P_m\omega^3, P_m^2\omega^2\right].$$

By applying the algorithm of least squares we get the estimate of the vector θ_e and the model of subsystem.

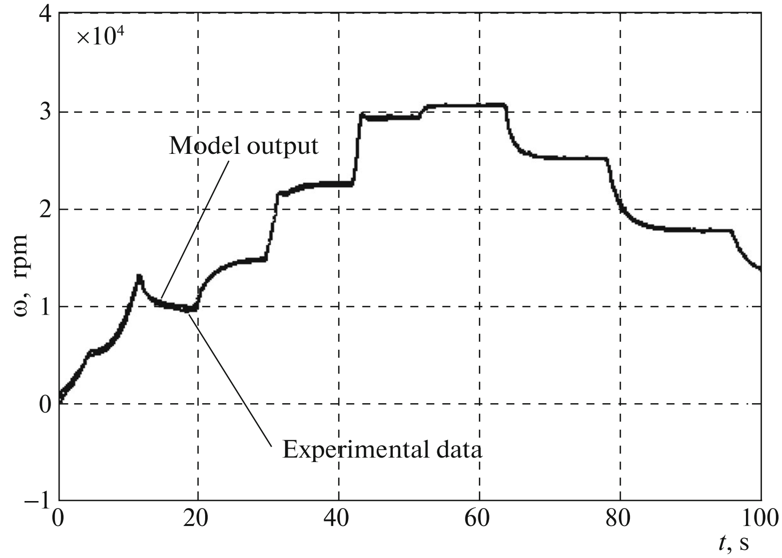

Fig. 6. The verification results of turbine speed model.

The verification results of the model are presented in Fig. 5 and show that the maximal error of the model is 0.24 bar (9% of the maximal pressure).

2.4. Subsystem of the Turbine Speed

The turbine speed ω_c is described by the second Newton's law

$$\dot{\omega}_c = \frac{P_t\eta_m - P_c}{J_c\omega_c}, \tag{2.15}$$

where the product $P_t\eta_m$ is the actual power of the turbine calculated by the equality (see [27–29])

$$P_t\eta_m = r\eta_{tm}(\omega_c, P_e, T_{em})\varphi(P_e)W_tT_{em}. \tag{2.16}$$

The compressor power P_c is given by

$$P_c = c\frac{W_cT_as(P_m)}{\eta_c(W_c)}. \tag{2.17}$$

Similarly to the previous subsystems, we replace (2.16) and (2.17) in (2.15) and substitute $\dot{\omega}_c$ for the finite difference:

$$\omega_c(k+1) = \omega_c(k) + \frac{\tau r\eta_{tm}(\omega_c, P_e, T_{em})\varphi(P_e)W_tT_{em}}{J_c\omega_c} - \frac{\tau cW_cT_as(P_m)}{J_c\omega_c\eta_c(W_c)}.$$

Expanding the right part of the last relation in the Taylor series and taking into account (2.8), (2.13) and Assumption 2 we obtain linear regression

$$\omega_c(k+1) = \theta_c^{\mathrm{T}}\mu_c(k) \tag{2.18}$$

with vector of unknown parameters θ_c and the vector of measurable functions

$$\mu_c = \left[1, \omega_c, \omega_c^{-1}, P_e\omega_c^{-1}, P_e^2\omega_c^{-1}, P_e, P_m, P_m^2, P_m\omega_c\right].$$

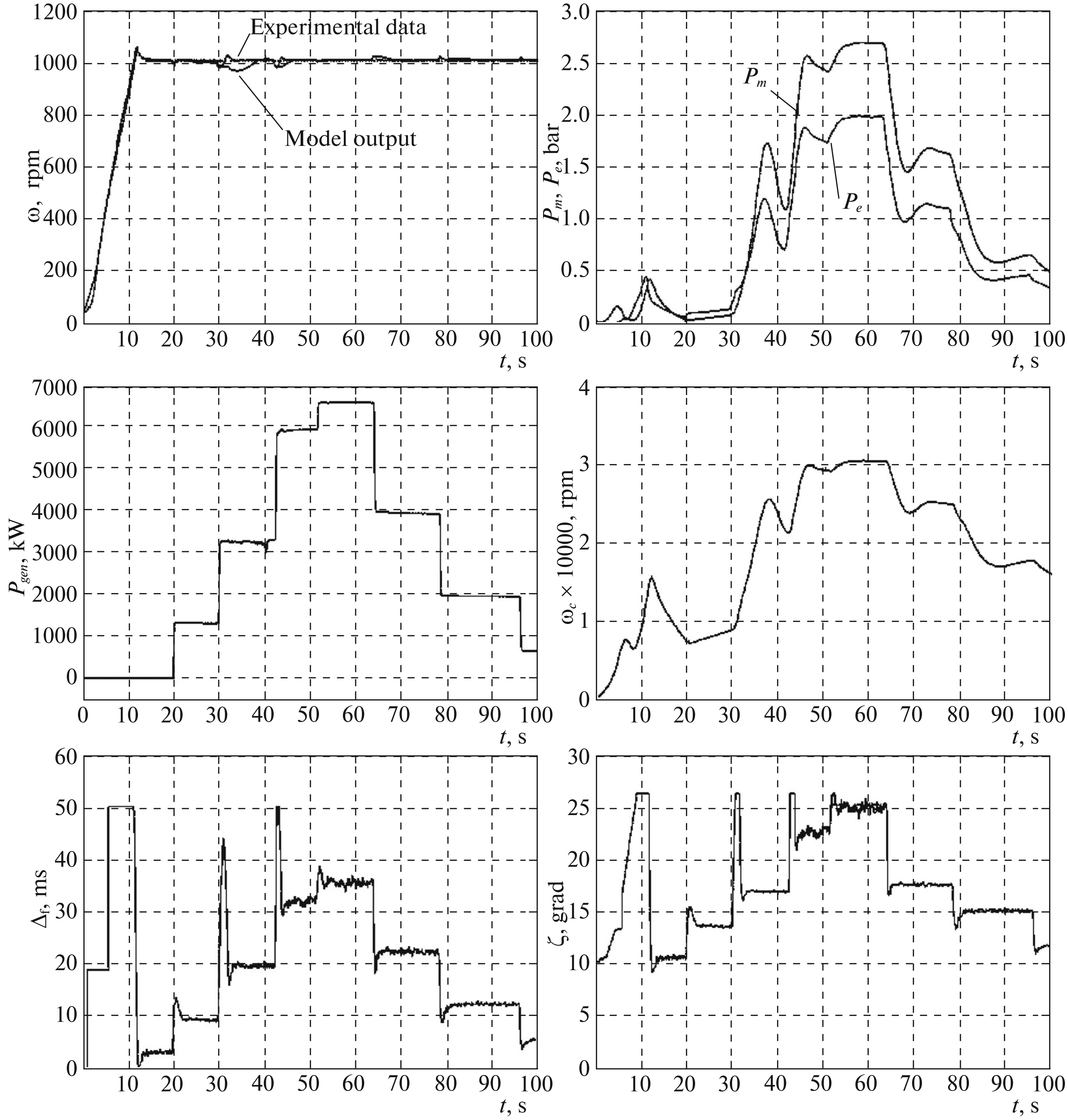

Fig. 7. The results of combined model verification.

The identification of θ_c is done with the least squares algorithm, and the verification results are presented in Fig. 6. It is seen that the model error is 850 rpm (2.8% of the maximal value of the turbine speed).

2.5. Combined Model

By combining recurrent equations (2.5), (2.10), (2.14), and (2.18) we obtain a model of the diesel internal combustion engine, transforming the control variables Δ_f and ζ and the disturbance P_{gen} into the crankshaft speed (frequency). Verification results of the model are presented Fig. 7 and show that the maximal error is 40 rpm (4.2% of the maximal speed). The model performance is comparable with the performance requirements for industrial speed controllers. As a sequence, the model can be used for both design of high-performance control laws and computer simulation of automatic high-performance control systems of diesel generator sets.

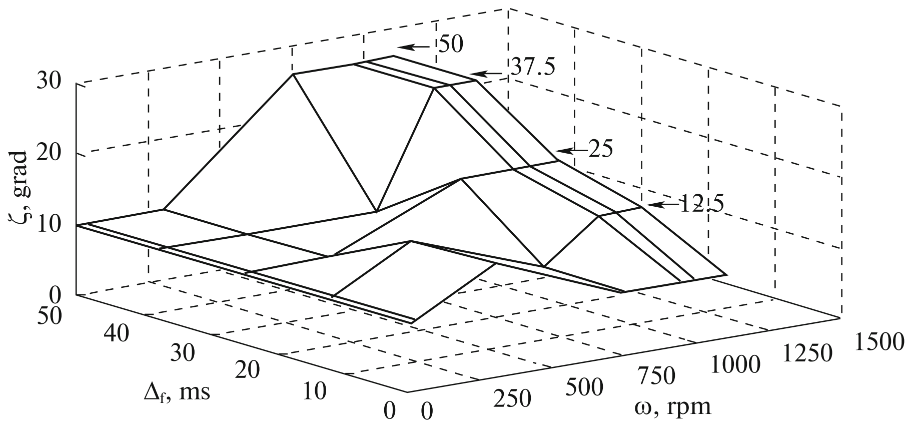

Fig. 8. Delivery begin map.

The following remarks regarding the model (2.5), (2.10), (2.14), and (2.18) are in order.

1. The model is highly nonlinear. On the one hand this property reflects complexity of the engine dynamics, but from the other hand requires application of nonlinear control methods.

2. The model performance is reasonable and corresponds to the precision required for speed stabilization in diesel-electric generator sets. The model response to the change of input signals corresponds to reality. So, the increase of active power leads to decrease of the speed and, vice versa, the power decrease leads to the jumps of the speed.

3. The application of least squares method for identification of the model parameters gives the optimal (in a sense of quadratic index) fixed estimate over sample between 0 and 100 s. At the same time, in order to improve the model performance and teach the model over other samples for different regimes it is reasonable to apply least squares algorithm (2.6) online.

3. EXAMPLE OF MODEL USE IN THE SYNTHESIZE OF ENGINE SPEED CONTROL

We consider

$$\Delta_f(k) = K_P\varepsilon(k) + K_I \frac{\tau}{z-1}[\varepsilon(k) - K_A(\Delta_f(k) - sat\{\Delta_f(k)\})] + K_D \frac{z-1}{\tau z}[\varepsilon(k)],$$

where K_P, K_I, K_D, and K_A are the coefficients of the proportional, integral, and differentiating components and the "anti-windup" coefficient, respectively, $\varepsilon = \omega^* - \omega$ is the control error, ω^* is the speed reference, and z is the forward shift operator,

$$sat\{\Delta_f(k)\} = \begin{cases} 50 \text{ ms}, & \text{if} \quad \Delta_f(k) \geq 50 \text{ ms}, \\ \Delta_f(k), & \text{if} \quad 0 \leq \Delta_f(k) < 50 \text{ ms}, \\ 0, \quad \text{if} & \Delta_f(k) < 0 \text{ ms}. \end{cases}$$

Controller design parameters are selected as follows: $K_P = 30$, $K_I = 80$, $K_D = 7$, and $K_A = 1$. Sampling time τ is equal to 0.1 s.

Delivery begin angle is regulated in accordance with look-up table 1 (see the three-dimensional map in Fig. 8) and the linear interpolation algorithm. The angle varies from 9° to 26° and depends on the fuel pulse width and the engine speed.

The simulation results are presented in Fig. 9 and illustrate the high performance of the speed control under different regimes of engine operating. The steady-state error under condition of constant load P_{gen} is less than 5 rpm (about 0.5%). This number corresponds to the strong requirements of voltage frequen-

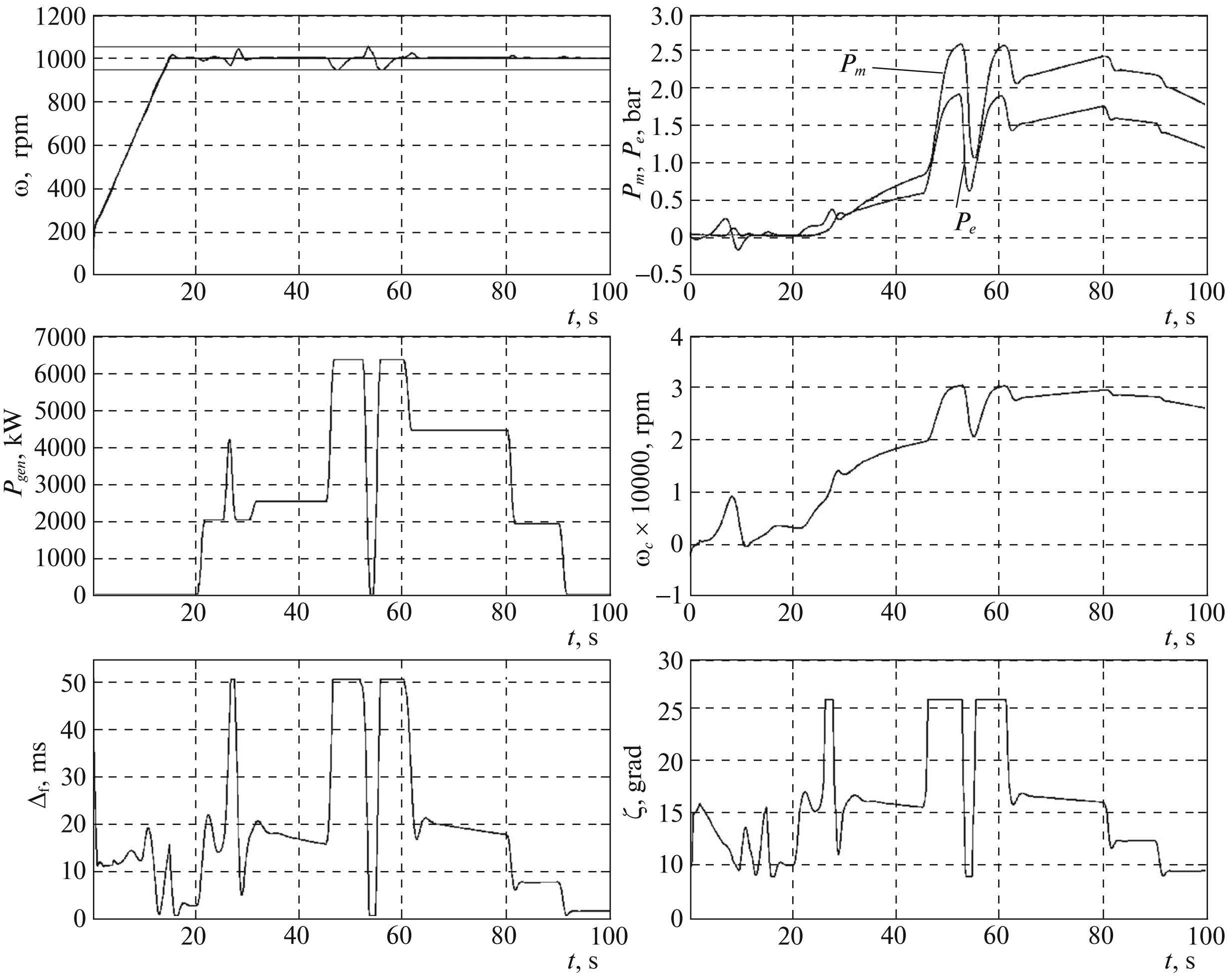

Fig. 9. Transients in the closed-loop system of engine speed control.

cies at the output of generator sets. The maximal control error is achieved within the interval from 52 to 55 s, when the load changes rapidly from the minimum value (0 MW) to the maximum value (6.4 MW).

Simulation result show that the system response to the loads increase or decrease corresponds to the "native" response of the plant. The short speed drawdowns of the speed are seen during load increases, while the short speed leaps can be seen during load decreases. Besides, the responses of controller as well as subsystems of the pressures and the turbine speed correspond to behavior of real physical systems.

Thus, the model representation of automatic control system has potentials for research of transients in engine and effective tuning of control.

CONCLUSIONS

The model of diesel engine of diesel generator 20EDG500 manufactured in OAO Kolomenskii zavod and oriented to simulation and high efficiency control law design is presented in the paper. The model permits to reduce the costs related to experimental research of automatic control systems of diesel generators and can be used in design of electronic benchmarks of diesel generators.

Algorithm used for identification of model functions and parameters can be utilized for the synthesis of diesel engines of other types (automotive, airplane, marine, etc.). Besides, the use of this algorithm online allows us to improve the performance of both model and the control based on this model.

Including thermodynamic equations and equations describing electrical part of diesel generators can be considered as the way of further development of the model.

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