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CE-2017-106

Model of exhaust system of a traction diesel engine with pulse supercharging on transient modes

A mathematical model of joint operation of a piston engine with pulse turbine drive, compressor as well as air supply of a diesel locomotive is presented in the article. This model in an integral part of the complex engine-generator power unit of a diesel locomotive in steady states and transient conditions, developed by the authors.

The periodic process in the exhaust manifold is represented as two specific characteristic ranges which encompass the process of forced exhaust and the scavenging process. The scavenging calculation is carried out jointly with the processes taking place in the inlet manifold of the engine, including inlet devices of the locomotive. The process of forced exhaust is calculated separately from the inlet processes and doesn`t require usage of empirical coefficients of impulsiveness. Both processes are represented as systems of nonlinear algebraic equations describe in the quasi-steady flow of working fluid in the elements of the inlet and exhaust tracts. The gas-dynamic calculation of the turbine in the transition process is conducted while taking into account changes in flow angles on the rotor blades. As a result of numerical calculation of the a8S22W diesel operational indices in transient modes, a satisfactory compliance of the calculated and the experimental results was obtained.

Key words: *diesel engine, locomotive, mathematical model, pulse supercharging, exhaust system, transient process*

1. Introduction

The analysis of the operating modes of the diesel engine shows that the engine runs on transient mode for a considerable part of the operation time.The transient modes, accompanying acceleration of the engine, are characterized by sharp increasing of fuel supply into the cylinder of engine and as a result by a reduction of the air excess coefficient λ, reduction of indicated efficiency η_i, significant peak temperature increase of exhaust gases T_t, incomplete combustion of fuel and other negative effects. A lot of such effects are excluded by using pulse supercharging.

Experimental studies of an operation of the diesel generator with pulse supercharging when momentary changes of load showed that the ratio of pressure in the inlet manifold P_s to pressure in front of the turbine P_t, is practically always larger than one throughout in the whole period of transient process. This parameter is one of the determining factors of the preference of using pulse supercharging system. The advantages of the pulse supercharging system are a better use of exhaust gases energy, which facilitates the increase in the torque of the turbine and improving the air supply of the diesel engine in the transient processes.

One of the most effective methods of studying the performance of the diesel locomotive engine with pulse supercharging in operation is a mathematic modelling of its working processes with a computer. It allows analysis of the working parameters of a diesel locomotive in exploitation, to determine and indicate the most efficient factors for improvement of the parameters already at the stage of design, tests and modernisation of engine systems.

This task can be achieved if the complex model of the engine-generator unit operation of a diesel locomotive in steady and transient modes is developed. This model should include a traction diesel engine – the piston part with its systems – inlet, exhaust and a turbocharger, locomotive transmission: – electric or hydraulic as well as auxiliary equipment on the vehicle. Development of such a model requires creation of mathematical models for above-mentioned components and systems. The model of the diesel locomotive power unit (system) obtained in this way will be used for simulation research (optimization) of influence of the revolution and power control system of the power unit on the energy efficiency of power transmission. Compliance of the model with the real processes, and also compactness and versatility of the description of processes for different types of engines and air aggregates are significant criteria of applicability of the mathematic model.

In [29] the authors reviewed 155 foreign publications concerning diesel engine simulations under transient operating modes in the open literature in the years 1971-2006. Most of these works (over 90%) concern simulation models of automotive engines as the research object, and about 5% marine engines. In this article, the authors summarize basic equations and modelling aspects concerning in-cylinder calculations, friction, turbocharger, engine dynamics, governor, fuel pump operation, and exhaust emissions during transients. The various limitations of the models are discussed together with the main aspects of transient operation (e.g. turbocharger lag, combustion and friction deterioration), which diversify it from the steady-state.

The general classification of the operating models of the piston engine cycle, known from the Polish literature, is described in [30], and the theoretical basics of fuel and air supply modelling of engines – in [35]. Implementation of mathematical models of diesel engine processes for simulation research of exhaust emissions are described in [12, 22, 26], and in publication [36], the model of diesel engine heat circulation is used for optimization of the research engine operation.

In the field of mathematic modelling of operation of automotive engines, most papers relate to control in engine systems with taking into account the transient modes [7, 13, 18, 24, 39].

A representation of conditions of the joint piston part and turbocharger operation in the form of calculations is a difficult task. It requires an accurate determination of the conditions of airflow and exhaust gases flow through the turbocharger. In [6, 19, 20, 27, 32], some alternative methods of solving the task are presented. They are different both in essence and accuracy.

The design of a model of the diesel with a turbocharger requires a mathematical description of the compressor and turbine. For solving this task, analytic methods [28, 31], which allow for calculating the hydraulic and thermal losses in the flow part of the turbine and turbocharger, can be used. Another method uses experimental characteristics of turbocharges for modelling, as functional relations – approximation of characteristics by polynomials of highest degree [8, 14, 15, 40].

Some mathematic models of the transient processes of diesel engines are based on the assumption that the diesel engine operation can be presented with a system of five differential equations [37]. The authors represented general differential equations without analysing the equations which express the relationship between concrete parameters of the engine. In [16], the mathematic model of the transient conditions of a marine diesel with pulse supercharging was elaborated based on the system of seven differential equations. The flow impulsiveness of exhaust gases is taken into account here by means of the correction factors obtained from experimental studies. The mathe-matical model of transient processes represented in [8] is also of interest. This model is based on a system of n differential equations for the piston part and of 6 equations describing the processes in the exhaust manifold in the case of using pulse supercharging. The mathematical models of transient modes of diesels in [38, 41] are based on similar principles. However, it is necessary to state that these models require a lot of computations and the presence of standard subprograms in the computer database. The precision of the model depends on the solution of the system of differential equations which is connected with the time of calculation.

For taking into account the pulsation of the mass flow which is delivered to the turbine at pulse supercharging, in [42] a coefficient α was entered, and to take into account the energy delivered at changeable pressure of gases, a coefficient β was entered. In [14, 15], impulsiveness of the flow of exhaust gases is taken into account with correction coefficients k_G, k_H and k_S for determining values of momentary growth of consumption of gas and pressure and also of changes in turbine efficiency. These coefficients are obtained based on analysis of diagrams of changes in gas pressures in the exhaust manifold for a certain number of engines. These coefficients are represented as approximate relations dependent on the pressure ratio π_k.

A systematization of coefficients type α, β, k_G, k_H, k_S is complicated due to large number of criteria, which have different effects on the shape of the curve of gases pressure in the exhaust manifold.

The authors of this article analysed the possibility of application of the existent mathematic model of the transient process of a diesel with the isobaric supercharging

system [21] for the calculation of characteristics of the diesel with a pulse supercharging system.

The method of implementation static characteristics of diesel and supercharging aggregates in the mathematic model, as well as the calculation of a compressor and inlet system, are analysed in detail in [1, 21, 23]. A calculation of the transient processes performed in accordance with this method, while taking into consideration the pulse coefficients for determination of the power and the pressure of the turbine, showed that the condition $P_s/P_t > 1$ is not met, and the estimated time of transient process calculation is considerably prolonged [1]. The inclusion of calculations of the gas exchange process in this method, in order to achieve a more precise determination of the momentary values of gas parameters in the exhaust manifold, makes the mathematic model more complicated and useless for optimization calculations.

In [5, 9, 10, 25, 33], the authors described some interesting simulation models of diesel engine operation in steady conditions and transient states while using 1D models for turbocharger and chosen processes of diesel engines, and in publications [11, 17, 34] 3D CFD models. In these works, the research (modelling) object is, among others, a pulse-charged turbine in application for mainly automotive engines. In this scope of research, imple-mentation of 1D and 3D models allows for effective simulation of the diesel engine processes chosen by the researchers. The object of research (modelling) and optimization calculations done by the authors of this article is the diesel locomotive power system, and the model of the exhaust system with pulse turbine supply is only an element of the above-mentioned complex model. Development of such a model implementing the 1-D modelling suggested by the researchers is a complex problem due to the variety of processes in the individual elements and systems of the diesel locomotive power unit and due to the limitation in the technical capabilities of computers. As it was indicated in the introduction to the chapter, the mathematical model should be compact, universal and suitable for multiple optimization calculations.

Given the above, the authors proposed a different approach to modelling of diesel engine exhaust system with pulse turbine supply which does not require implementation of differential equations for calculation of gas exchange, using experimental pulse indices or creation of complex, 1D or 3D models, e.g. for the piston part of the traction diesel engine, turbocharger.

2. Modelling of a diesel engine exhaust system

Based on modelling of the traction diesel engine, a quasi-steady model of working fluid flow in engine and turbocharging system elements was used [21]. The quasi-steady flow model allows for achieving satisfactory correspondence of the numerical calculation results to the experimental results [2, 32].

While creating the model it was proposed that the change in gas pressure in the exhaust manifold of diesel was dependent on time or on crankshaft angular velocity ω_d represents a periodic function with the period:

$$T = \omega_d \cdot \varphi^* \qquad (1)$$

where ω_d, φ^* – crankshaft angular velocity and period of change in gas pressure.

The period of changes in gas pressure in the manifold φ^* will be equal to the value of phase shift of cylinder operation. We will assume that periodic process in the manifold may be represented in the form of some sequence of processes with the duration $\Delta\varphi$ inter-related by boundary conditions. There can be many such consecutive processes, however in relation with the processes in the inlet manifold, there are two specific periods: a process of forced exhaust (angle $\varphi_{ex}–\varphi_{in}$) and a scavenging process (angle $\varphi_{in}–\varphi_{exe}$, see Fig. 1a).

In the engine with pulse supercharging, the process of forced exhaust is completely isolated from the processes in the inlet manifold and it can be calculated based on the known methods if the parameters of the working fluid in the cylinder are set. The scavenging process is directly connected with the processes in the inlet manifold. In this work, it is suggested to calculate the scavenging process together with the processes in the inlet tract of the engine including the inlet equipment of diesel locomotive, air filter 1, compressor 2, intercooler 3, piston part 4, turbine 7 and exhaust equipment 8. (see Fig. 1b). The process of forced exhaust is calculated separately from the inlet processes. Both processes are presented by systems of nonlinear algebraic equations, which describe quasi-stationary flow of working fluid in the elements of the inlet and exhaust tracts. As in [21, 23], systems of algebraic equations are complemented by a system of differential equations, which describe changes in temperatures of heat carrier, of exhaust tract elements and of turbocharge rotor speed depending on time. The calculation of the flow of working fluid in the scavenging process and the exhaust process is carried out subsequently by taking the average values of pressure, temperature and working fluid flow rate in the corresponding periods with the assumed period $\Delta\varphi$ (see Fig. 1a). The initial parameters for a calculation of the exhaust tract with forced exhaust are the parameter values determined while calculating exhaust manifold during scavenging.

A working fluid flow rate (of air-gas mixture) in the diesel, G_{ag}, is represented by a sum of air flow rate in the scavenging process of cylinders, G_{sc}, and for filling cylinders from the moment of exhaust valves closure to bottom-dead-point, G_d.

It was assumed, that at the end of filling (in the cylinder) there will be a mass of working medium equal to the sum of the mass from the filling process and the mass present in the cylinders at the closure moment of the exhaust valves.

At pulse supercharging, $G_{SC} > 0$ which demonstrates why the pressure in the exhaust manifold behind the cylinder at the scavenging process will be determined by means of relation:

$$P_{10\ sc} = P'_6/\pi_d \qquad (2)$$

where P'_6 – pressure of working fluid before inlet valves π_d – ration of intake manifold pressure and exhaust manifold pressure.
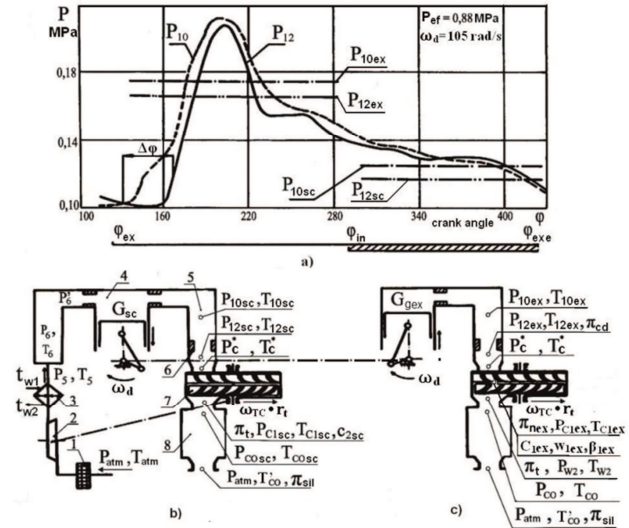

Fig. 1. Change in pressure in exhaust manifold (a) calculation model of a8C22 diesels with pulse supercharging at scavenging (b) and at free exhaust (c)

From the energy-conservation equation we determine the temperature of air-gas mixture in manifold at scavenging process (see Fig. 1b):

$$T_{10sc}\cdot(G_{sc}+G_{Vc})\cdot C_{pag}=T_6\cdot G_{sc}\cdot C_{pa}+T_{10ex}\cdot G_{Vc}\cdot C_{pex} \qquad (3)$$

where G_{Vc} – working fluid flow rate at exhaust, which corresponds with the mass, contained in cylinders at the moment of opening of the exhaust valves; C_{pag}, C_{pa}, C_{pex} – heat capacity at constant pressure of air-gas mixture, air dose, working fluid at exhaust process, respectively; T_6 – temperature of air dose;

Working fluid flow rate, G_{Vc}, can be determined from equation:

$$G_{Vc} = \frac{V_h \cdot \omega_d \cdot z_c}{(\varepsilon - 1) \cdot 2\pi \cdot \tau} \cdot \rho_{gex} \qquad (4)$$

where V_h, ω_d – cylinder capacity and crankshaft angular speed; z_c, ε, τ – respectively: the number of cylinders, compression rate and the number of crank revolutions per power stroke; ρ_{gex} – gas density at temperature T_{10ex} and pressure P_{10ex}.

If we assume that heat capacity at a constant pressure of air-gas mixture, air dose and exhaust gases are constant and equal ($C_{pag} \approx C_{pa} \approx C_{pex}$), then from the equation (3) and (4) we obtain:

$$T_{10sc} = \left(T_6 \cdot G_{sc} + G_{Vc} \cdot P_{10ex} \cdot R_g^{-1}\right)/\left(G_{sc} + G_{Vc} \cdot \rho_{gex}\right) \qquad (5)$$

In calculations it is assumed that at time $\Delta\varphi/\omega_d$ of scavenging, average gas flow rate G_g is determined by scavenging air flow rate G_{sc}. The pressure of working fluid before turbine P_{12sc} is determined from equation:

$$P_{12sc} = P_{10sc}/\pi_{cd} \qquad (6)$$

where π_{cd} – pressure differential in conditional diaphragm 6 (see Fig. 1b).

The average working fluid flow rate at time $\Delta\varphi/\omega_d$ of forced exhaust is determined as a sum of gas flow rate G_d for filling cylinders and fuel mass flow rate b_{fc}.

$$G_g = G_d + b_{fc} \tag{7}$$

The temperature of the exhaust gases T_{10ex} is determined by the heat balance while taking into account the heat losses from incomplete burning of fuel according to the following equation:

$$T_{10ex} = \frac{b_{fc} \cdot H_u \cdot (\xi - \eta_i - \xi_1)}{G_g \cdot C_{pex}} + \frac{G_d}{G_g} \cdot T_{10sc} \tag{8}$$

where H_u – calorific value of fuel; ξ – available heat factor at the end of expansion process; ξ_1 –coefficient taking into account share of unburned fuel; η_i – efficiency indicator of diesel engine, and the gas pressure before turbine P_{12ex} – according to equation (6).

The temperature of gases before turbine T_{12sc}, T_{12ex} is determined by taking into account the heat exchange in exhaust manifold by means of method described in [21, 23].

A calculation of the turbocharger turbine at transient modes is of particular interest as a change of working fluid parameters can significantly change flow angles on the rotor blades (see Fig. 2). The pulsation of pressure in manifold and partial gas supply lead to the fact that the gas turbine works at constantly changing parameters of the gas before nozzles. Only the turbocharger shaft speed remains constant due to relatively large inertia moment of the rotor.
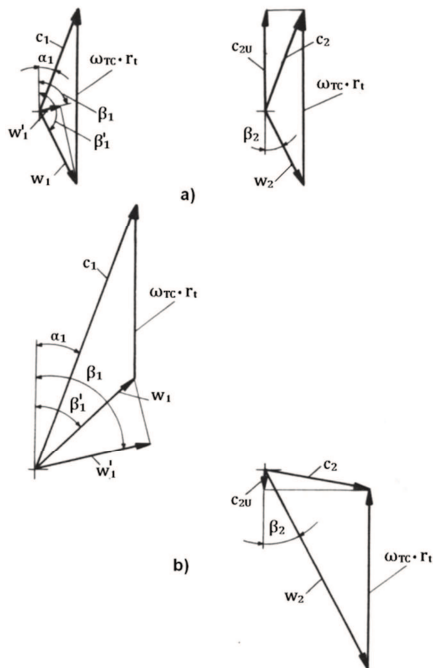

Fig. 2. Velocity diagrams at turbine rotor blades, inlet and exit at scavenging (a) and exhaust (b) – a8C22 diesel for ω_d = 52 rad/s, $\overline{P}_e$ = 0.25: α_1 – angle of flow exit from nozzle unit; β_1 – direction of shock-free inlet to the rotor blades; β'_1 – direction of relative velocity of flow at inlet of the rotor blades; β_2 – angle of flow exit from the rotor blades in relative motion

In this method, considering the gas-dynamic calculation of the turbine, it is assumed that the temperature of

stagnated gas stream at the inlet of the rotor blades is determined by the relative rate w_1, (see Fig. 2), and the pressure of stagnated gas stream – by velocity w_1', which is the projection of the relative velocity w_1 in the direction β_1 of the shock-free flow of gas into the inlet of the rotor blades. The difference of kinetic energy at speeds w_1 and w_1' represents energy losses and turns to potential energy at isobaric process, therefore, the pressure of stagnated flow is determined by speed w_1'. Speed losses in the blades of the nozzle unit and rotor blades are taken into account by the coefficient of losses, and their values are determined according to method [1, 23].

Taking into account the stated assumptions and chosen method of calculating algebraic equations, the order of calculating the turbine is accepted as follows:
a) for a given value of working fluid flow rate, the parameters of stagnated flow stream at inlet of nozzle unit are determined as follows

$$T_c^* = T_{12} + \frac{G_g^2}{2} \cdot \frac{R_g \cdot T_{12}^2}{P_{12}^2 \cdot f_{em}^2} \cdot \frac{k_g - 1}{k_g}$$

$$\tag{9}$$

$$P_c^* = P_{12} \cdot \left(\frac{T_c^*}{T_{12}}\right)^{\frac{k_g}{k_g-1}}$$

where f_{em} – cross-sectional area of exhaust manifold.
b) from the equation of gas flow rate at adiabatic flow, the flow function ψ_n can be determined

$$G_g = f_n \cdot \frac{P_c^*}{\sqrt{R_g \cdot T_c^*}} \cdot \psi_n$$

$$\tag{10}$$

$$\psi_n = \frac{G_g \cdot \sqrt{R_g \cdot T_c^*}}{f_n \cdot P_c^*}$$

where f_n – cross-sectional area of sector nozzle unit;

$$f_n = \frac{F_n}{z_{em}} \cdot \frac{\Delta\varphi}{2\pi \cdot \tau} \cdot z_c \tag{11}$$

where F_n – total cross-sectional area of nozzle unit; z_{em} – the number of exhaust manifolds which are connected to the turbine and have separate gas supply to the nozzle unit

$$\psi_n = \sqrt{\frac{2}{k_g - 1}\left[\left(\frac{P_{C1}}{P_C^*}\right)^{\frac{2}{k_g}} - \left(\frac{P_{C1}}{P_C^*}\right)^{\frac{k_g-1}{k_g}}\right]} \tag{12}$$

c) using approximation of the flow function, we determine:
– pressure reduction in nozzle unit and parameters of working fluid at the outlet of nozzle unit

$$\pi_n = \pi_n(\psi_n) \tag{13}$$

– pressure P_{c_1} and temperature T_{c_1}

$$P_{c_1} = P_c^*/\pi_n$$

$$\tag{14}$$

$$T_{c_1} = T_c^* \cdot \pi_n^{\frac{1-k_g}{k_g}}$$

– absolute c_1 and relative w_1 flow velocities

$$c_1 = \varphi_1 \cdot \frac{G_g \cdot R_g \cdot T_{c_1}}{f_n \cdot P_{c_1}},$$

$$w_1 = \sqrt{c_1^2 + (\omega_{TC} \cdot r_t)^2 - 2c_1 \cdot \omega_{TC} \cdot r_t \cdot \cos\alpha_1}, \quad (15)$$

where φ_1 – the speed loss coefficient in nozzle unit

– relative gas velocity w_1' by the shock-free inlet and parameters of stagnation at the inlet of rotor blades

$$\beta_1' = \pi/2 - \arcsin\left(\frac{c_1 \cdot \cos\alpha_1 - \omega_{TC} \cdot r_t}{w_1}\right),$$

$$w_1' = w_1 \cdot \cos(\beta_1' - \beta_1),$$

$$T_{w_1}^* = T_{c_1} + \frac{w_1^2}{2} \cdot \frac{k_g - 1}{k_g} \cdot R_g^{-1},$$

$$T_{w_1'}^* = T_{c_1} + \frac{w_1'^2}{2} \cdot \frac{k_g - 1}{k_g} \cdot R_g^{-1}, \quad (16)$$

$$P_{w_1}^* = P_{c1} \cdot \left(\frac{T_{w_1'}^*}{T_{c_1}}\right)^{k_g/(k_g-1)}$$

d) based on parameters of stagnation at inlet of rotor blades, we determine flow function ψ_t, turbine pressure ratio π_t, parameters of working fluid at the outlet of rotor blades, relative velocity of gas exhaust from rotor blades w_2 and components of absolute velocity c_2

$$\psi_t = \frac{G_g \cdot \sqrt{R_g \cdot T_{w_1}^*}}{f_t \cdot P_{w_1}^*}, \quad (17)$$

where f_t – cross-sectional area of sector rotor blades.

$$\pi_t = \pi_t(\psi_t),$$

$$P_{w_2} = P_{w_1}^*/\pi_t,$$

$$T_{w_2} = T_{w_1}^* \cdot \pi_t^{\frac{1-k_g}{k_g}}, \quad (18)$$

$$w_2 = \varphi_2 \cdot \frac{G_g \cdot R_g \cdot T_{w_2}}{f_t \cdot P_{w_2}},$$

$$C_{2u} = w_2 \cdot \cos\beta_2 - \omega_{TC} \cdot r_t,$$

$$C_{2o} = w_2 \cdot \cos(\pi - \beta_2), \quad (19)$$

$$C_2 = \sqrt{C_{2u}^2 + C_{2o}^2},$$

where φ_2 – speed loss coefficient in the rotor blades.

e) with the Euler's formula, we determine specific work on the rotor blades L_{ad}, adiabatic heat drop H_t and internal efficiency of the turbine η_t

$$L_{ad} = \omega_{TC} \cdot r_t \cdot \left(w_1 \cdot \cos\beta_1' - w_2 \cdot \cos(\pi - \beta_2)\right),$$

$$H_t = \frac{k_g}{k_g-1} \cdot R_g \cdot T_{12} \cdot \left(1 - (\pi_n \cdot \pi_t)^{\frac{1-k_g}{k_g}}\right), \quad (20)$$

$$\eta_t = L_{ad}/H_t.$$

Exhaust temperature and pressure at the outlet of turbine are calculated from the equation:

$$T_{co} = T_{w_2} + \frac{c_{2u}^2}{2} \cdot \frac{k_g - 1}{k_g} \cdot R_g^{-1}, \quad (21)$$

$$P_{co} = P_{w_2}$$

A calculation of the exhaust tract and turbine parameters during a forced exhaust was performed by determining the work fluid flow rate throughout the engine, described in the [21].

From a calculation of exhaust process, the average working fluid flow rate in period $\Delta\varphi$ is known, and by means of that calculation we determine pressure P_{10ex} in the manifold 5 (see Fig. 1c).

As a first step of calculation, we set starting value of pressure P_{10ex}^I in the first approximation. With equation (8), we can determine a value of temperature T_{10ex}^I and calculate parameters of exhaust tract and turbine; as a result we receive values of pressure reduction in the conditional diaphragm π_{cd}, nozzle unit π_n, rotor blades π_t, and silencer π_{sil}. Then, we calculate a product of pressure reductions of working fluid at P_{10ex}^I, T_{10ex}^I:

$$\prod_1^n \pi_i = \frac{P_{10ex}^I}{P_{atm}} \cdot \frac{1}{\pi_{cd} \cdot \pi_n \cdot \pi_t \cdot \pi_{sil}}, \quad (22)$$

and we check the condition

$$\left|\prod_1^n \pi_i - 1\right| < \varepsilon_2, \quad (23)$$

If the condition (23) is not met with a given accuracy of calculations ε_2, we set a new value of the current value of pressure P_{10ex}^I and repeat the calculation. Otherwise, the average value of pressure P_{10ex} in period $\Delta\varphi$ is the value searched.

An average value of the exhaust gas flow rate throughout the engine can be calculated from relation

$$G_g = \sum_{i=1}^{i=n} \frac{z_c}{2\pi \cdot \tau} \cdot \Delta\varphi_i \cdot G_{gi}, \quad (24)$$

where n – number of periods $\Delta\varphi$.

After the calculation of the inlet system, piston part and exhaust system of diesel, we solve the system of equations which describes the diesel generator fuel supply and power control at transient processes. The system of equations includes a differential equation of power balance and systems of algebraic equations for the determination of the dependence of diesel generator power and limiting parameters of operational process of the diesel on an angular velocity of the diesel crankshaft. The mathematic model simulates functional connections used in control systems of angular velocity of the crankshaft and power of a diesel locomotive engine [3, 4, 21]. A calculation of transient and steady modes of diesel operation is carried out using a program, common for any diesel type. A general block diagram of that program is represented in Fig. 3.
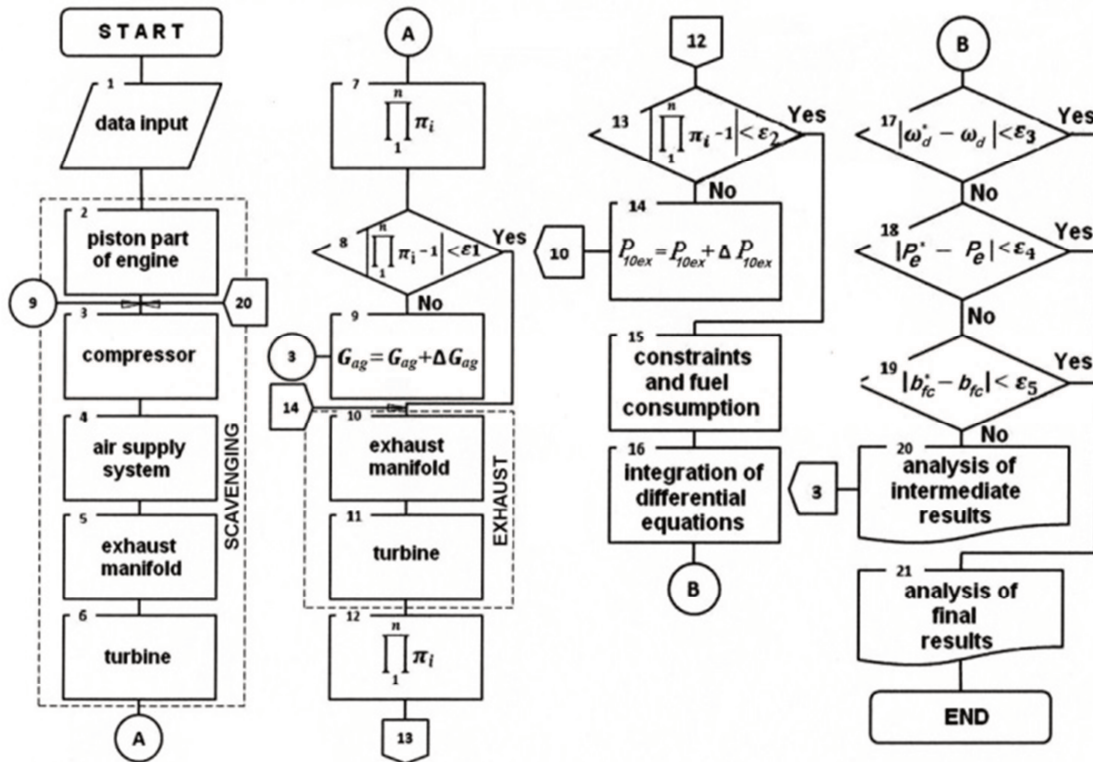
Fig. 3. Block diagram of calculation algorithm of transient process for diesels with pulse supercharging

The diesel operation mode and specific data are entered into the program in block 1 as an array of source data: array of constants, array of constants of the diesel engine and approximate experimental relations, like efficiency of diesel, characteristics of turbocharger compressors and turbines, etc.

In blocks 2–9, the calculation of air-gas mixture flow rate, G_{ag}, at set values of angular velocities of the diesel crankshaft and turbocharger rotor, temperatures and fuel supply is performed by means of system of nonlinear algebraic equations.

In blocks 10–14, the calculation of exhaust tract and turbine during the time of free exhaust is performed by means of system of nonlinear algebraic equations.

In block 15, a choice of possible increment of fuel consumption is performed according to system of differential and algebraic equations describing specific features of fuel feeding and load control system.

In block 16, an integration of differential equations describing changes the above-mentioned set values in time is done.

In blocks 17–19, actual values of angular velocity of crankshaft ω_d, effective power of diesel P_e and fuel mass flow rate b_{fc} are compared with the set ones ω_d^*, P_e^*, b_{fc}^*. In the case of their compliance with given accuracy, the transient process is considered to be finished. Otherwise, the calculation is continued with new values ω_d and ω_{TC}, starting from block 3.

In block 21, a print of output parameters is made and the calculation ends.

3. Validation of a diesel engine model

Using the demonstrated block diagram and method described in [21, 23], the performance calculation program of diesel locomotive engine with pulse supercharging on transient mode was made and debugged.

Before using the mathematic model for a calculation of transient modes of diesel engines, compliance of the model with the real objects on steady modes of operation was checked. To this purpose, the results of the numerical calculations were compared with the experimental data from the producer of the engines, HCP Cegielski, for exploitation and load characteristics of the a8C22W traction engines with the pulse turbine supply. Comparison of the calculations with the results of the experimental research showed a sufficient similarity of the results [2, 21]. The discrepancy between the calculated and experimental values is not greater than 2–4%. The differences occurring in the angular velocity of the turbocharger rotor, turbocharging pressure and air low rate under low load can be explained mainly by the shortage of experimental data of compressor (characteristics) within lower angular velocities of the turbocharger.

The evaluation of a mathematic model suitability at transient process was made by comparing operational indices of diesel obtained experimentally on the water braking resistor at rapid increase in the engine loading – switching of the driver controller from IV to XI position, to the numerical calculation data. Figure 4 presents a comparison of calculated and experimental time characteristics of the transient processes of an a8C22W diesel-generator unit for controlling revolutions and load realized on a SM31 diesel locomotive [2]. Figure 4 proves the sufficient

similarity of the results of simulation calculations to the results of experimental research. The greatest deviation of the calculated values is observed for the turbocharging pressure index between 4 and 6 second of the transient process period and is equal to 6%. The discrepancy between
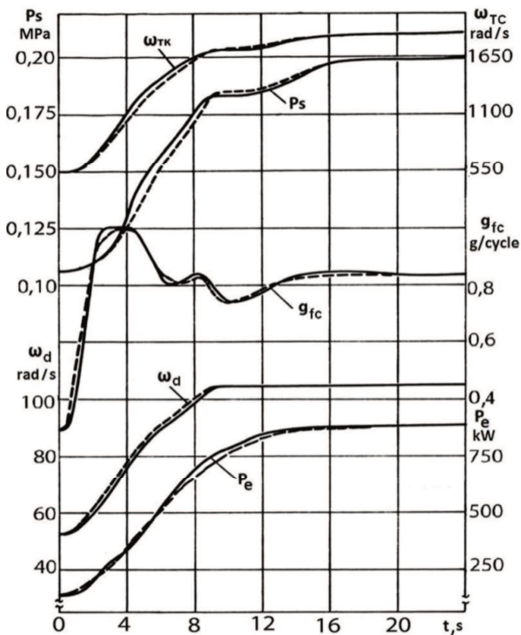

Fig. 4. Time characteristics of transient process in a8C22W diesel-generator unit: ω_d, ω_{TC} – angular velocity of crankshaft and turbine rotor; P_s – pressure of charge air; g_{fc} – fuel charge per cycle; P_e – effective power of diesel; continuous line – experimental data; dashed line – calculated data

the calculated and experimental values for the other performance rates of the diesel engine is not greater than 3%. These results allow the conclusion that the mathematical model reliably reflects the performance rate of the diesel locomotive traction engine both in quality and quantity. Summing up, the developed model can be implemented in optimization calculations of locomotive systems.

4. Conclusion

The developed mathematical models of interaction between the diesel traction engine and turbocharger supplied at variable pressure (including the authors' models), using the experimental pulse indices, provide a sufficient similarity of the numerical calculations to the experimental results in steady modes of the engine operation. In contrast, in transients modes the satisfactory similarity cannot be obtained. Inclusion of calculations of gas exchange processes into the models using differential equations or implementation of 1-dimension models, e.g. in the case of a turbocharger, makes the mathematical model complicated and unsuitable for optimizing calculations of locomotive systems.

The model of the exhaust system of the diesel traction engine with the pulsating supply of the turbocharger presented in this article was developed by the authors on the basis of a non-linear algebraic equations without the use of differential equations of the charge exchange process and gas exhaust, as well as of experimental, correction coefficients taking into account the pressure pulse of the exhaust gases. The assumption made by the authors for the development of the model are its specific features in comparison with existing methods of modelling of the exhaust systems of diesel traction engines. Confirmation of the validity of the adopted modelling assumptions is obtaining a satisfactory similarity of the results of numerical calculations to the actual runs, both in steady and transition stages. Moreover, the developed mathematical model gains compactness and versatility in terms of description of the processes in various types of traction engines and power aggregates.

The model of the traction engine exhaust constitutes an integral part of the complex model of the diesel locomotive power unit in steady operation and transient stages in exploitation [1, 3]. The complex model was used in the scientific-research works for simulation tests of influence of the power unit loading control on the energy efficiency of the power transmission. These research was aimed at evaluation of the modernisation effectiveness of the power unit loading control of diesel locomotive series SM31 [3, 4].

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