SIMULATION OF THE EXPANDER OF THE EXCESS GAS PRESSURE UTILIZATION PLANT

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ABSTRACT

The object of research: The expander of the excess gas pressure utilization plant.

Investigated problem: Increasing the efficiency of the process of utilization of excess gas pressure during its distribution from main high-pressure gas pipelines. Development of hardware and software tools that provide automation of the process of regulating flow parameters under the influence of time-varying disturbances.

The main scientific results: A mathematical model of the process of utilization of excess gas pressure has been developed, and an experimental assessment of its adequacy has been carried out. With the help of the Matlab application "Linear Analysis Points", the description of the object is linearized, which makes it possible to obtain its representation in the state space.

The area of practical use of the research results: The sphere of application of the results of the study is automation objects related to the regulation of gas flow parameters. The results of identification of model parameters provide an opportunity for the synthesis of the process controller.

Innovative technological product: In accordance with the task of increasing the efficiency of the process of utilization of excess gas pressure when it is distributed from the main high-pressure gas pipelines, a mathematical model of the channel for controlling the speed of rotation of the turbine (expander) has been developed, which serves as the basis for the synthesis of the regulator for the utilization process.

Scope of the innovative technological product: The developed model can be used in the construction of turbine speed controllers, which are used in the generation of electricity. The approach to the linearization of the description of the control object makes it possible to automate the process of identifying its parameters.

1. Introduction

A necessary condition for the functioning of the gas supply system is the use of gas distribution stations (GDS). Their main purpose is to reduce the pressure of gas, which is distributed to consumers from high-pressure gas pipelines.

Based on the relevance of introducing energy-saving technologies, the task is to use the energy of compressed natural gas, which is lost when the pressure reduces at the reduction points. These energy saving tasks are quite successfully solved with the help of excess gas pressure utilization plants – expander-generator units (EGU) [1].

The most efficient way to use the electricity produced by the EGU is to connect to the consumption network the technical means of automation of the GDS, lighting, pumps for forced circulation of water in the heating system and electric room heating.

Along with the classical design of the EGU scheme, in which the energy of the natural gas flow is converted into electrical energy, schemes have been implemented that use temperature fields differences. As a result of the pressure reduction of the gas stream, temperature differences in the gas stream occur. At the same time, the high-temperature heat energy that is generated during the operation of the EGU is used for heating, and the low-temperature one is used to create refrigeration units and air conditioning systems [2].

The equipment used for energy saving in gas supply systems is constantly being improved [3]. Modernization of equipment and optimization of energy saving processes is carried out in accordance with the chosen directions for the use of recycled energy. At the same time, the priority of research is the energy efficiency of the utilization plants used [4]. Analyzing the parameters that determine the efficiency of EGU, it can be stated that the main factors for reducing the efficiency of utilization are the change in the operating modes of the gas supply system, as well as the instability of pressure in the supply main gas pipeline [5].

When the gas flow rate in the gas pipeline changes, respectively, its pressure, as well as the operating modes of the EGU, the utilization parameters do not correspond to the optimal ones. Fluctuations in the power consumed by the load also reduce the efficiency of excess energy utilization, especially at relatively low gas flow rates and high gas pressures at the EGU inlet.

The direction of further research seems to be in the development of automation tools that increase the efficiency of utilizing the excess gas energy that is present during its transportation and consumption.

The task of the ongoing research is to develop a mathematical model of the dynamics of the expander of the plant for the utilization of excess gas pressure energy, taking into account the features of the modes of its transportation and consumption. The presence of such a model is the basis for creating a high-quality controller of the energy recovery process.

1. 1. The object of research

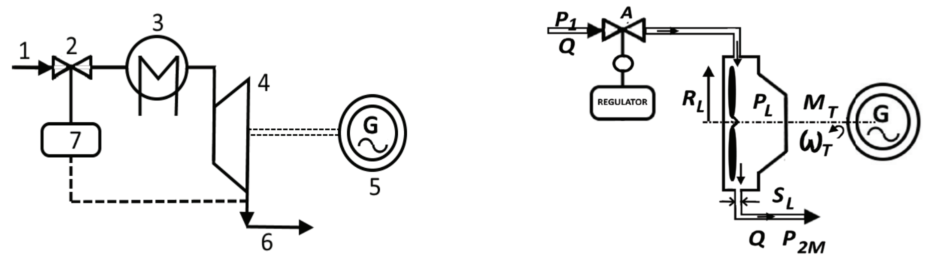
The expander of the excess gas pressure utilization plant.

1. 2. Problem description

A sufficient number of publications have been devoted to the study of the design and technological parameters of EGU and the modes of their operation. Only a few of them consider aspects of the control capabilities of the mentioned object, which ensure the maximum efficiency of the utilization unit.

The subject of research is represented by a mathematical model of the expander of the utilization plant of excess gas pressure.

The scheme of the utilization plant, in which a turbine is used as an expander, is shown in **Fig. 1, *a*.**



a *b*

Fig. 1. Expander-generator unit scheme: *a* – turbine is used as an expander; *b* – EGU efficiency by heating the gas flow and consider only the expander model

The stability of the EGU operation is ensured by the control valve 2, which sets the required pressure of the gas flow coming from the high-pressure gas pipeline 1. In the EGU unit, as a result of gas expansion after the valve, and in the expander 4, the temperature of the gas flow is significantly reduced. Therefore, to increase the efficiency of the EGU, the gas is heated in the heater 3. After the turbine 4, in which the kinetic energy of the flow is partially converted into the mechanical energy of rotation of the generator shaft 5, the gas enters the low-pressure gas pipeline 6. The required turbine rotation speed is achieved by changing the pressure using the regulator 7, which acts on the valve 2. Electric generator 5 is the mechanical load of the turbine, the value of which can vary over time.

If to postpone the solution of the problem of increasing the EGU efficiency by heating the gas flow and consider only the expander model, then the EGU scheme can be depicted as shown in **Fig. 1, *b***. Based on the analysis of the process of utilization of the excess pressure of the gas flow, the studied control object can be represented in the form of a diagram shown in **Fig. 2**.

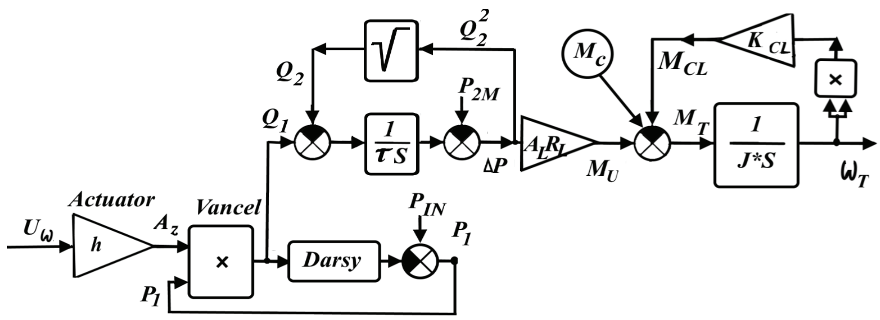


Fig. 2. Diagram of the control object model

In this scheme, the “Actuator” block simulates the operation of a controlled throttle drive, the “Vancel” block displays the change in gas flow parameters, and the Darsy block shows the pressure loss during the propagation of gas flow through the pipeline.

A control signal U_ω is applied to the electric drive of the controlled valve (throttle), under the action of which the position of its stem h and the cross section of the valve A_Z change. As a result of adjusting the cross section of the valve, the pressure in front of the valve P_1 and the flow rate of the throttled flow Q change.

The main parameter that determines the dynamics of the change in the cross section of the throttle A_Z is the time of movement of the gate from one extreme position to another. Accordingly, to simulate the valve drive, an integrator link with limitations that correspond to the extreme positions of the valve stem is used. In addition, the adopted model of the valve drive takes into account the transmission coefficient of the gearbox and its backlash, which affects the dynamics and positioning accuracy of the valve drive.

A feature of the description of the functional dependence of the cross-sectional area of the throttle A_Z is its non-linear dependence on the angle of rotation of the motor shaft (stroke of the valve stem). This dependence of the cross-sectional area of the throttle A of radius R on the stroke of the stem h is illustrated in **Fig. 3** and the following functions:

$$A_Z(\alpha)=0.5R^2(\alpha-\sin\alpha);$$

$$\alpha=\arccos\,(1-h/R), \tag{1}$$

where α – angle of the sector of the throttle section, which depends on the stroke of the stem.

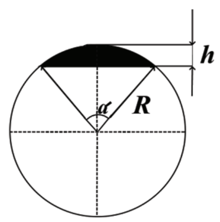


Fig. 3. Change in cross-sectional area depending on the stroke of the stem *h*

If to calculate the numerical values of the nonlinear function $A(h)$ (1), then an acceptable level of accuracy in the approximation of this function is achieved using a third-order polynomial [6].

Depending on the ratio of inlet P_1 and outlet P_2 pressures of the pipeline, the gas flow regime can be critical and subcritical. Criticality index Y_{cr}

$$Y_{cr} = \frac{P_2}{P_1} = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} = 0.546, \tag{2}$$

where k – adiabatic index for gas (for real k=1.3, for air k=1.4), defines the boundaries of the use of the Saint-Venant-Wenzel formula [7].

For $Y \leq Y_{cr}$, the Saint-Venant-Wenzel equation for the gas flow rate Q has the form:

$$Q = \varepsilon \cdot A_Z \cdot P_1 \sqrt{\frac{k}{RT_1}\left(\frac{2}{k+1}\right)^{\frac{k+1}{k-1}}}, \tag{3}$$

where ε – flow coefficient of the throttle section; A_Z – cross-sectional area of the throttle; T_1 – gas temperature in the supply pipeline; R – universal gas constant.

The pressure loss P_{PI} in the pipeline due to friction is estimated according to the Darcy-Weisbach formula (“Darsy” block):

$$P_{PI} = \frac{\lambda \cdot L \cdot Q^2}{2d \cdot \rho \cdot A_{PI}^2}, \tag{4}$$

where λ – coefficient of hydrodynamic friction losses; L – length of the pipeline; d – diameter of the pipeline; A_{PI} – cross-sectional area of the pipeline; ρ – gas density.

The gas flow rate Q_2 at the outlet of the turbine, in which there is a subcritical flow, is determined by the pressure drop ΔP between the inlet and outlet of the turbine. Neglecting pressure losses in the connecting pipeline, it is possible to assume that the pressure at the outlet of the turbine is equal to the pressure P_{2M} at the inlet to the low-pressure gas pipeline:

$$Q_2^2 = A_T\left(P_2 - P_{2M}\right), \tag{5}$$

where P_2 – pressure at the turbine inlet; A_T – turbine outlet cross section.

In turn, the dynamics of pressure P change in the turbine depends on the time constant τ_T of establishing equilibrium between the inlet and outlet gas flows:

$$\tau_T \frac{dP}{dt} = Q_{dr} - Q_2. \tag{6}$$

The description of the process of formation of the torque moment of the turbine M_T is based on the theorem of changing the moment of momentum of the gas flow relative to the axis of rotation [8]. This takes into account the process of interaction of the gas flow with the elements of the turbine and the pressure drop across the expander:

$$M_T = M_U + M_{\Delta P} - M_{CL} - M_C, \tag{7}$$

where M_U – moment due to the interaction of the gas flow with the turbine blades; $M_{\Delta P}$ – moment determined by the pressure drop ΔP across the expander; M_{CL} – moment depending on the aerodynamic resistance of the medium during the rotation of the rotor; M_C – moment of resistance to rotation.

At the same time:

$$M_U + M_{\Delta P} = (Q \cdot V + \Delta P \cdot A_L) \cdot R_L, \tag{8}$$

where V – linear speed of rotation of the turbine blades; Q – current gas flow rate; A_L – cross section of the gas flow; R_L – radius of the turbine blades.

Moment depending on the aerodynamic resistance of the medium:

$$M_{CL} = K_{CL} \cdot \omega_T^2, \tag{9}$$

where ω_T – angular speed of rotation of the turbine; K_{CL} – aerodynamic drag coefficient.

This coefficient depends on the density of the environment, the outer diameter of the rotor, the shape, number and relative position of the rotor arms, and the speed of the turbine shaft. The complexity of the analytical description of the dependences of the aerodynamic drag coefficient leads to the need for experimental evaluation of its values. These estimates are carried out under the conditions of operation of experimental bench installations [9].

The angular speed of rotation of the turbine ω_T, in turn, is determined after integrating the equation:

$$J \cdot d\omega_T/dt = M_U + M_{\Delta P} - M_{CL} - M_C, \quad (10)$$

where J – moment of inertia of the turbine.

As mentioned, when the cross-section A_Z of the controlled throttle changes, not only the pressure of the gas flow changes, but also its temperature. Temperature change is described by the Joule-Thomson formula [7]:

$$T_2 = T_1 - \mu\ \Delta P, \quad (11)$$

where T_1, T_2 – temperature at the inlet and outlet of the plant; ΔP – pressure drop across the throttle valve; μ – Joule-Thomson coefficient.

In industrial utilizing plants, the pressure and temperature drop are so significant that in order to increase the efficiency of the plant, it is necessary to use special heaters (**Fig. 1, *a***, pos. 3), which increase the flow temperature by 50–60 °C.

In the experimental installation for the utilization of excess gas pressure, which is created to assess the adequacy of the mathematical model, the pressure drop is much smaller, and the change in the flow temperature during throttling does not exceed 10 °C. Therefore, the errors in modeling the flow parameters according to formula (3), in which the flow temperature appears, do not exceed a few percent. This allows at this stage of research to consider the temperature of the flow constant and to simplify the mathematical model (MM) of the expander of the utilization plant.

Thus, in the considered MM of the expander, the process of controlling the speed of rotation of the turbine by changing the position of the throttle valve is displayed.

1. 3. Suggested solution to the problem

From the analysis of the system of equations (3)–(10), which represents the model of the expander of the excess gas pressure utilization plant, it can be seen that, in fact, the model is described by nonlinear functions. Therefore, the list of tasks to be solved in the development of the considered model in the form of a state space includes the stage of linearization of its description.

This description is built at a given operating point, to which corresponds a set of state variables, input and output values of the model:

$$\begin{aligned} \delta x(t) &= x(t) - x_0, \\ \delta u(t) &= u(t) - u_0, \\ \delta y(t) &= y(t) - y_0, \end{aligned} \quad (12)$$

where $x(t)$ – object states; $u(t)$ – input signals; $y(t)$ – output signals.

Then the states and output of the model at the selected operating point at time t_0 will be:

$$x(t_0) = f(x_0,\ u_0,\ t_0) = x_0;$$

$$y(t_0) = g(x_0,\ u_0,\ t_0) = y_0.$$

The set of mentioned variables $\delta x(t)$, $\delta u(t)$ and $\delta y(t)$ corresponds to the linearized system of equations in the state space:

$$\delta x'(t)=A\ \delta x(t)+B\ \delta u(t),$$

$$\delta y(t)=C\ \delta x(t)+D\ \delta u(t), \tag{13}$$

where *A, B, C, D* – matrices with constant coefficients. These matrices are the Jacobians of the system, evaluated at the operating point.

$$A=\frac{\partial f}{\partial x}\Big|t_0,x_0,u_0,\ \ B=\frac{\partial f}{\partial u}\Big|t_0,x_0,u_0,$$

$$C=\frac{\partial g}{\partial x}\Big|t_0,x_0,u_0,\ \ D=\frac{\partial g}{\partial u}\Big|t_0,x_0,u_0. \tag{14}$$

The accuracy of the linearization used depends on the distance of the evaluated variable from the selected operating point.

The selected operating point of the object corresponds to the transfer function of the linearized model, which is determined by the ratio of increments of the Laplace transforms from the output $Y(s)$ to the input $U(s)$ variable:

$$W_0(s)=\delta Y(s)/\delta U(s). \tag{15}$$

The process of linearization of a complex object leads to cumbersome expressions and a lot of boundary conditions, which stipulate the applicability of approximating expressions. This is due to the fact that different blocks of the model have different types of nonlinearities. Therefore, a justified approach to the linearization of an object is a combination of linearized blocks that form the general scheme of the model.

In this case, linearization involves determining the values of the input quantities and the state for each block from the operating point, which makes it possible to form the *Jacobian* for these values for each block. At the same time, it is difficult to obtain analytical linearization for some simulated blocks. This happens in the following cases:

– non-linearities do not have a definite Jacobian;

– Jacobians of discrete blocks, reflecting state diagrams, as well as switched subsystems tend to zero during linearization;

– custom S-Function blocks and Matlab Function blocks do not have analytic Jacobians.

Then the linearization of the blocks is performed based on the estimation of changes in the input data, block states and the response to these disturbances at the corresponding operating point.

Structural transformations that are carried out during the linearization of a dynamic model in the Simulink dynamic systems simulation package, in some cases, lead to the formation of an "algebraic loop". In our case, such a contour is formed when calculating the values of the gas flow rate *Q* according to relations (3) and (5).

Attempts to untie the algebraic loop by reducing the step size or including the *Memory* block of the Simulink system [10] in the block diagram do not give a positive result. A more efficient solution of the problem is achieved by dividing the model into subsystems and using pre-calculated Lookup Table linearization tables.

The main difficulties in the development of a mathematical model of the expander are associated with the insufficient amount of initial and experimental data that take into account the design and technological parameters of the utilization plant. In particular, the complexity of describing the functional dependences of the aerodynamic drag coefficient of a turbine (expander) on many design parameters leads to the need to create special automation tools when conducting research on an experimental setup.

2. Materials and Methods

The simulation scheme in the Matlab environment is shown in **Fig. 4**.

In accordance with this simulation scheme, the speed of rotation of the expander is controlled by changing the stroke *h* of the control valve stem.

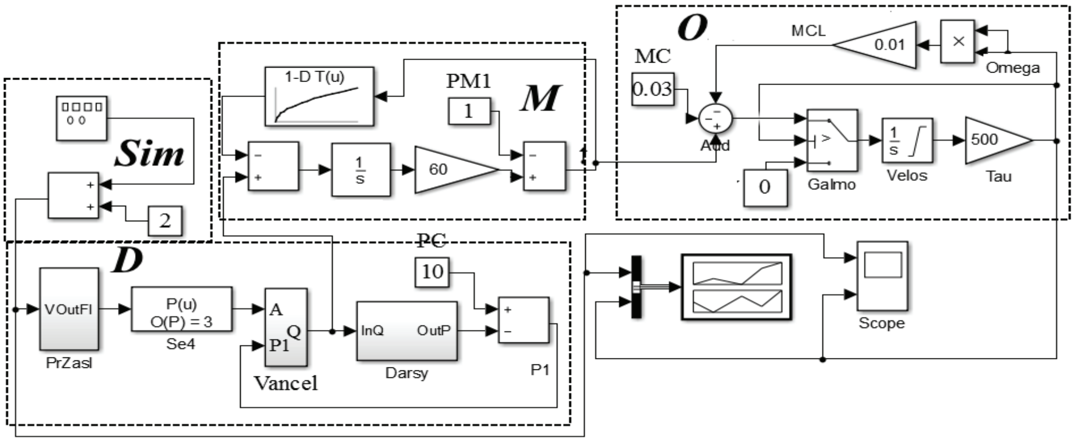


Fig. 4. Scheme of the expander model in the Matlab environment

The model scheme has three blocks. Block *D* simulates the control of gas flow by changing the position of the throttle valve (relation (3)). Block *M* displays the dynamics of changes in flow parameters (5), (6) and turbine torque, and block *O* forms the value of the turbine rotation speed according to relations (9), (10).

The use of the non-linear *Galmo* element in the last block fixes the process of stopping the turbine at Q=0.

In the model scheme, a simulator *Sim* for changing the position of the valve is provided, with the help of which the speed of rotation of the turbine is controlled.

The functioning of the model is illustrated by oscillograms of the change in the position of the valve *h* and the corresponding changes in the value of the turbine rotation speed ω*,* which are shown in **Fig. 5**.

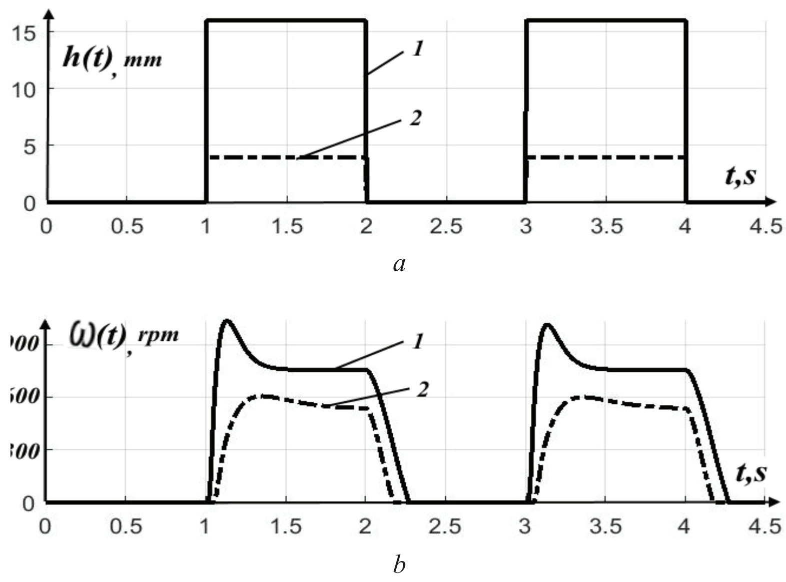


Fig. 5. Oscillograms of: *a* – valve position changes; *b* – changes in turbine speed; $1 - h_{max}$=16; $2 - h_{max}$=4

From the above oscillograms, it can be seen that for different values of the throttle valve cross section, the dynamics of speed change has a different character, which confirms the nonlinearity of the object under study.

Carrying out operations to identify an object involves obtaining its transfer function.

Due to the non-linear dependence of the rotation speed on the control actions, when constructing the model under consideration, the problem of object linearization is solved using the

Matlab application “Linear Analysis Points”. Linearization procedures are reduced to estimating the Jacobians (14) of the operating points of the selected range of values.

This application has a graphical user interface (**Fig. 6**) that allows to evaluate the parameters of the model states that correspond to the selected operating point (op_trim). As a result of the linearization procedures, it is possible to obtain the frequency response of the nonlinear model, which is used to check the quality of the linearization of the nonlinear model.

The identification of the system by parametric models is carried out on the basis of the “System Identification” functions, which makes it possible to form a set of transfer functions of the object that correspond to the selected operating points of the entire range of valve movements.

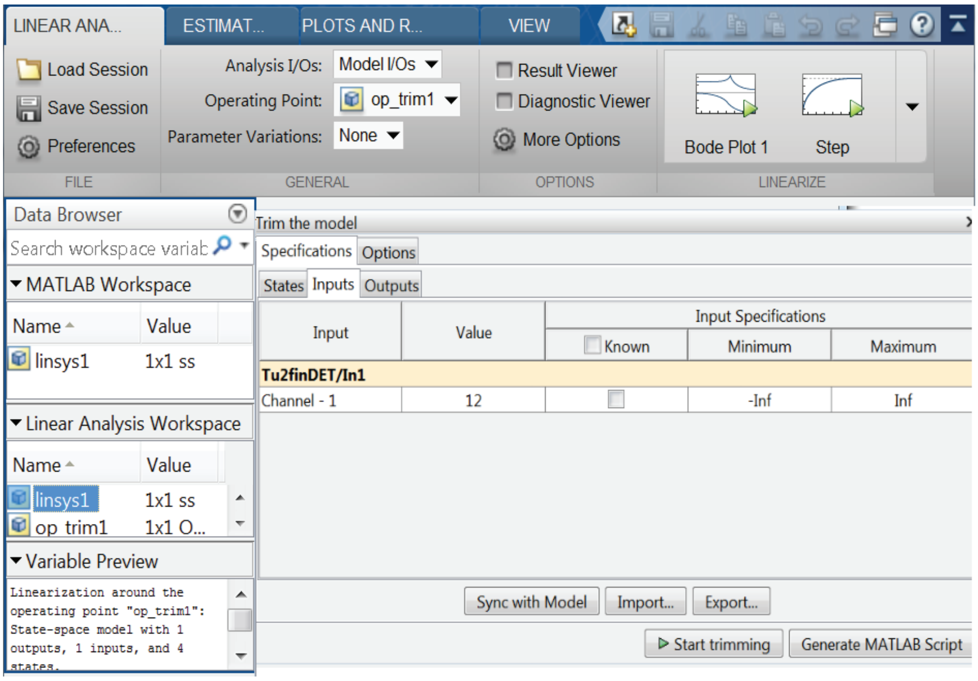


Fig. 6. View of the graphical user interface

Nevertheless, the implementation of the end-to-end linearization process in the model under consideration causes certain difficulties caused by the singularity of the calculations of the Jacobians of the operating points of the modeled object. These difficulties are due to the fact that some Matlab Function blocks do not have analytic Jacobians. Overcoming the existing difficulties is based on obtaining the transfer functions of individual blocks with their subsequent combination according to the modeling scheme (**Fig. 2**).

The first step is to obtain the transfer function of the “stem stroke/turbine torque” channel, for which the input is the input of the *D* block and the output is the output of the *M* block.

The user interface of the Matlab application “Linear Analysis Points” allows to get a family of operator equations of the third order:

$$W_{Mi}(s) = \frac{b_{1i}s + b_{0i}}{s^3 + a_{2i}s^2 + a_{1i}s + a_{0i}}, \qquad (16)$$

where i – number of the operating point; a_i, b_i – coefficients of polynomials; $W_{Mi}(s)$ – transfer operator function of the i-th operating point.

The family of characteristics corresponding to the family of transfer functions (16) is shown in **Fig. 7** as Bode diagrams.

As a result of using a similar procedure, let’s obtain a family of transfer functions for block *O*. They correspond to a first-order aperiodic link with different transfer coefficients and time constants:

$$W_{Oi}(s) = \frac{b_{0i}}{a_{1i}s + a_{0i}}. \tag{17}$$

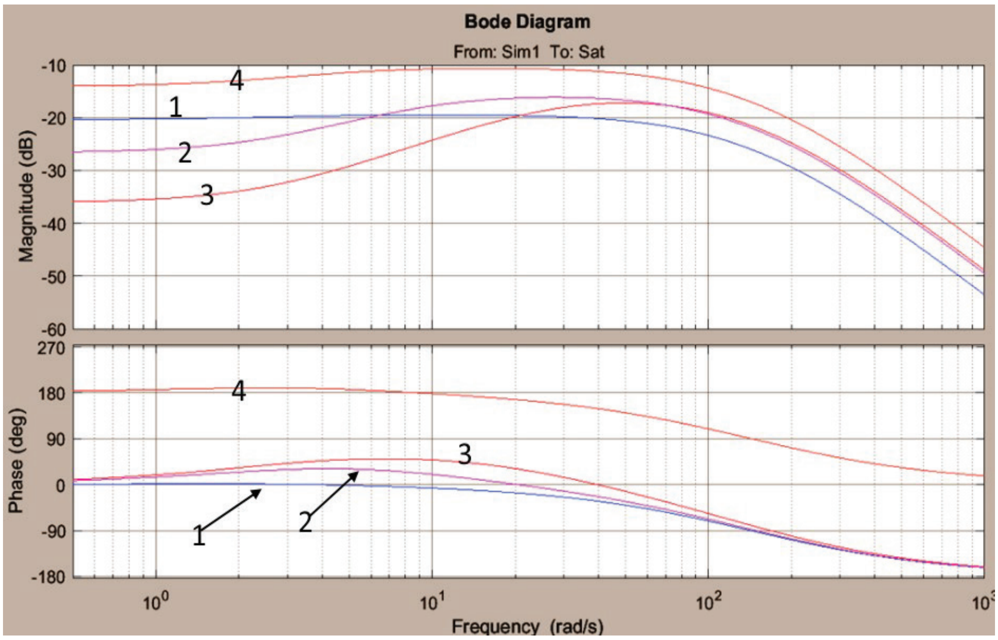


Fig. 7. Bode diagrams for different throttle positions: 1 – *h*=2; 2 – *h*=6; 3 – *h*=10; 4 – *h*=16

The resulting transfer function of the control object has the form:

$$W_{DTi}(s)=W_{Mi}(s)\cdot M_{Oi}(s) \tag{18}$$

The resulting analytical description of the utilization plant model, which is a non-linear control object, can be used to construct a regulator for the excess gas pressure utilization process. If necessary, description (18) can be represented in the state space (13).

3. Results

The specificity of the model of the studied control object is the difference in the types of simulated processes that ensure the utilization of excess gas pressure, respectively, and their dynamics. The need to take into account the difference in the dynamics of aerodynamic (turbine, pipeline) and electromechanical (valve drive) processes is due to the desire to ensure the stability of the model and the controller used to control the installation.

This need arises due to the fact that the utilization process is accompanied by external disturbances of various nature, both in the inlet gas flow and from the load side. To assess the relationship between the time constants of the object blocks and its model on an experimental set-up assembled according to the scheme of **Fig. 1**, the transient characteristics of the "stem stroke/turbine rotation speed" channel are taken.

According to the conditions of the experiment, the inlet pressure of the installation was 4 bar. The family of transient characteristics of the mentioned channel (**Fig. 8**) was obtained by changing the cross section of the throttle valve at the inlet to the expander (turbine). The values of the throttle section corresponded to certain values of the throttle stem travel. The duration of the valve opening was chosen arbitrarily.

In **Fig. 8** characteristic 1 corresponds to the value of the cross section of the full throttle opening; characteristic 2 – 75 % of the section; characteristic 3 – 50 % of the section, and 4 – 25 % of the section.

Comparison of the obtained characteristics (**Fig. 8**) with similar characteristics (**Fig. 5**) obtained as a result of modeling allows to state that the developed mathematical model reliably reflects the operation of the experimental gas pressure utilization unit.

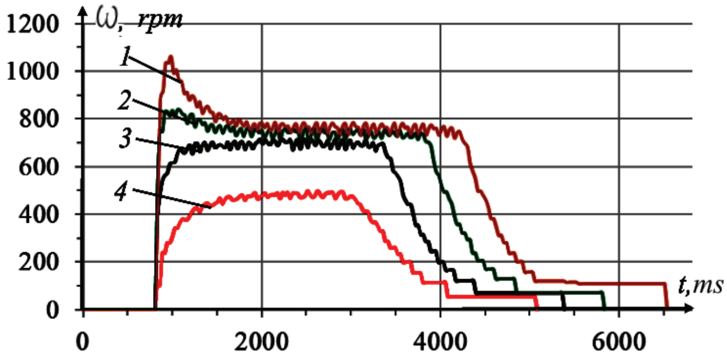


Fig. 8. The family of transient characteristics of the "stem stroke/turbine rotation speed" channel for different throttle valve positions

4. Discussion

Options for improving the developed model are determined by the possibilities of automating the procedures used, especially if this model is used when developing the object controller. In our case, the procedural task is to choose the linearization method. In this case, the criterion is traditionally used – the effectiveness of control / ease of implementation of the model.

In both global linearization [11] and local linearization [12], the task of simplifying the model is usually posed. Accordingly, the real basis for improving the model is the use of combined type linearization methods.

The combined approach in constructing the model makes it possible to take into account nonlinearities of various types. At the same time, as follows from the specifics of the object under study, its structural and technical parameters limit the use of the research results by the considered experimental setup. This is due to the fact that the speed of the valve electric drive, the length of the pipeline, the aerodynamic drag coefficient of the turbine of pressure utilization units differ from each other. The listed parameters are basic for constructing a model of a turbine expander and determine the nature and dynamics of the gas flow through the nozzles of a jet-reactive turbine [8]. Therefore, the obtained results cannot be extended to models of steam [13] or wind turbines [14], which are widely used in the production of electricity by installations of various power.

The prospect of further research is determined by the availability of hardware and software tools that make it possible to use the proposed model for:

– simulation of turbine speed stabilization systems;

– simulation of the processes of interaction of a turboexpander with a power consumption network;

– the structural-parametric synthesis of the EGU controller.

5. Conclusions

As a result of the analysis of the processes of the installation for the utilization of excess gas pressure, a mathematical description of the object under study was obtained. Its mathematical model is represented by non-linear differential equations.

Using the Matlab application "Linear Analysis Points" made it possible to obtain a family of linear transfer functions that can be represented in the state space.

As can be seen from the obtained Bode diagrams, the difference in the parameters of the transfer functions for different operating points of the operating range necessitates the adaptation of the controller parameters when the load changes and the action of parameter disturbances.

Conflict of interests

The authors declare that there is no conflict of interest in relation to this paper, as well as the published research results, including the financial aspects of conducting the research, obtaining and using its results, as well as any non-financial personal relationships.

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