MODELLING RESEARCH OF THE CHARACTERISTICS OF GASTURBINE AXIAL COMPRESSOR WITH CHANGEABLE FLOW PASSAGE GEOMETRY

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Abstract

In real exploitational conditions marine gasturbine engine works at different rotation speeds determined by the required parameters of ship's movement. Conversion from one rotation speed to another is related to realization of unsteady energetic processes, determined by the gasodynamic mutual influence of compressor configurations. The values of working medium flow thermodynamic parameters undergo significant changes in time. To work out the qualitative and quantitative estimation of these changes it is necessary to define the dynamic equations, describing working medium flow through the gas passage of engine. In the next step it is necessary to resolve the defined equations for disturbances of steady cooperation of compressor configurations.

This paper concerns the application of mathematical modelling methods to analyzing gas-dynamic processes in marine gas turbines. The influence of geometry changes in axial compressor flow passage on kinematical air flow characteristics is presented. The elaborated mathematical model will make it possible to realize – in the future – simulative investigations of gas-dynamic processes taking place in a compressor fitted with controllable guide vanes. Individual parametric features of every engine in service are identified by using expensive experimental tests, which are conditioned by many constructional and operational limitations. The dynamic development of computer technology, enables using it for numerical simulation of changeable technical state processes.

Keywords: *gas turbine, axial compressor, variable stator vanes*

1. Introduction

An important problem of operation of ship gasturbine engines is to know the influence of changeable technical state of engine on its working parameters. Compressor is a unit of gasturbine engine especially sensitive on changes in its technical state during operation. The more-or-less-contaminated atmospheric air flowing into the compressor causes a.o. continuously changing form of blade passages, increased roughness of blade surfaces as well as change of mass of compressor rotor. It seriously impacts on compressor's operation stability, changes its characteristics as well as performance and efficiency of the entire engine. If the compressor design contains a control system for setting controllable blades of guide vanes (of initial swirl guide vane or/and first stage guide vanes) to make cooperation of all engine's units optimal by continuous correction of the compressor characteristics, then the disturbances occurring in work of the system will result in changes in operation of the compressor and engine, of character similar to those due to rotational speed changes or contaminated blade passages of compressor.

Individual parametric features of every engine in service are identified by using expensive experimental tests, which are conditioned by many constructional and operational limitations. The dynamic development of computer technology, which has been already applied to the design stage of engine's units, enables using it also for numerical simulation of changeable technical state processes. Such approach greatly shortens the time necessary for diagnostic tests aimed at determining a set of "defect-symptom' relations, in comparison to the time-consuming and expensive investigations carried out on real objects. Having at one's disposal an appropriate computer software one is able to elaborate the models simulating operation of engine's units, verified within an allowable range of static and dynamic loads. The computer software based on the mathematical models makes it possible to realize the numerical experiments, which consist in putting-in real variables and hypothetical technical states of engine.

2. Characteristics of axial compressors working in engine systems

The universal characteristics of compressor (Fig. l), which shows the dependence of the compression π_S and efficiency η_S of compressor on the air mass flowing through it, $\dot{m}$, and the rotational speed n, make it possible to determine the most favourable conditions of cooperation of compressor with other units of the engine under the assumption that parameters of the sucked -in air are values complying with the so-called ISA standard atmosphere (p_{ot} = 1.013 bar, T_{ot} = 273.15 K, φ = 0%). The characteristics serve for the selection of optimal conditions for air flow control and assessment of influence of operational factors on the parameters of the compressor.

Figure 1 highlights the occurrence of unstable work phenomenon, on which a schematic diagram is presented of flow round a blade of axial stage rotor under motion with the constant rotational speed n, for which a change in the air flow rate $\dot{m}$ is effected. Fig. 1a shows the schematic diagram of the flow in the conditions for which the air flow rate corresponds with optimal efficiency of the stage. The relative velocity vectors w_1, w_2 are parallel to camber line of blade profile that is conductive to laminar flow through the blade passages. The decrease of air flow (Fig. 1b) as compared with calculation conditions at the circumferential velocity u maintained constant makes the axial component of absolute velocity, c_a, lower, what results in the increase of the inlet angle i of air flow onto rotor blades. It is conductive to boundary layer separation out of concave surfaces of blades and to generation of whirl zones. A similar phenomenon occurs on the convex surface of blade (Fig. 1c), when air flow rate becomes greater at the circumferential velocity maintained constant. Then the air flow rate $\dot{m}$ reaches its maximum values.

At critical values of angles of flow through the produced whirl zones which constitute circumferentially displacing low-pressure zones, a sudden air flow reversal (pumping) towards compressor's inlet can happen, what results in the violent flow pulsations transferred onto engine's structure. The phenomenon is detrimental and dangerous because of the thermal and mechanical overload of engine's structure. The occurring vibrations of significant amplitudes may cause fatigue cracks in blades.

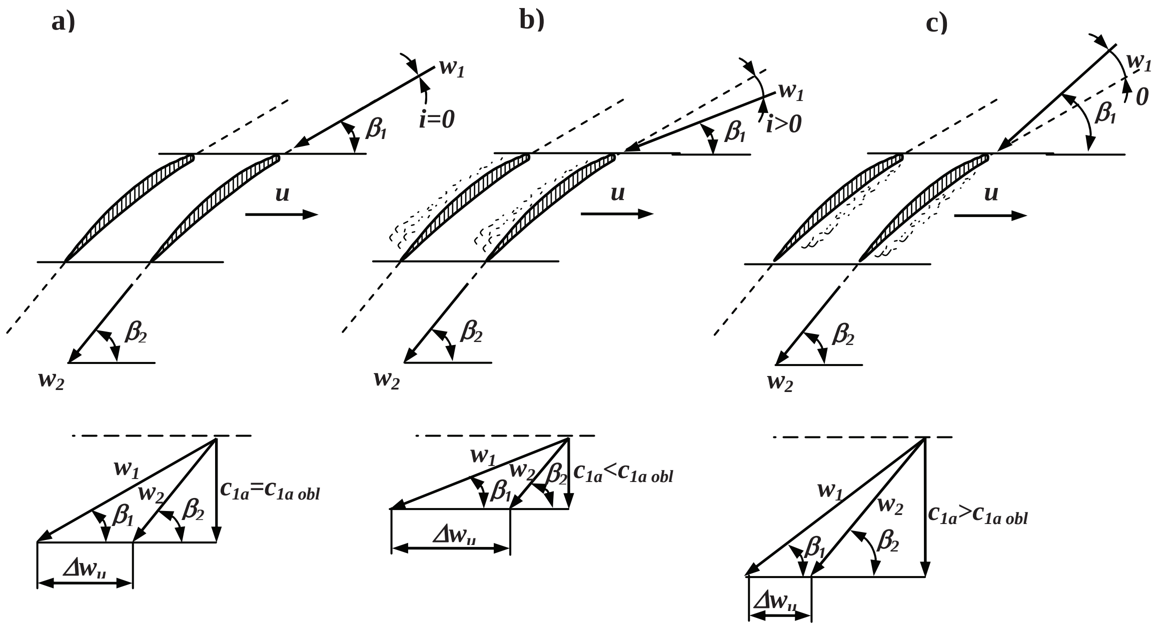


Fig. 1. Schematic diagram of the flow round blades of compressor axial stage ring at the constant rotational speed of rotor and changeable angles of air flow inlet; a) calculated angle of air flow inlet, b) positive angle of air flow inlet, c) negative angle of air flow inlet

In this connection the compressor should be - within the range of service rotational speeds - so adjusted as to place the line of compressor-network co-operation with certain stable work margin.

During the engine's operation, rotational speed of rotor, air flow rate and optimal form of kinematics of air flow through the stage blade passages, determined by the air flow inlet angle *i* onto the blades, have the greatest impact on the compressor's performance and efficiency. The main principle of compressor control during changing its rotational speed or flow rate is to maintain values of the air flow inlet angles *i* close to zero. One of the used methods for control of axial compressor is to change the geometry of its flow passage by applying a controllable inlet guide vane or controllable guide vanes of a few first stages of compression.

3. Controllable inlet guide vanes and guide vanes of first stages of compressor

The application of controllable blades of inlet guide vane and guide vanes of particular compressor’s stages enables the simultaneous change of inlet angles of flow onto blades of rotor rings of the stages by changing the setting angles of blades of guide vanes during the compressor’s rotational speed changing. Fig. 2 shows the essence of control of blades of controllable guide vanes by using single compression stage as an example. The situation, which is presented in Fig. 2b where velocity directions and values are indexed “1”, corresponds with average values of rotational speed service range of compressor’s rotor. In this case takes place an average angular setting of blades of guide vane ring, at which the inlet angle of flow onto blades of rotor ring does not generate disturbances in the flow through the blade passages. In the case of lower values of rotational speed of compressor, i.e. appearance of a lower value of the axial component of absolute velocity, c_{1a}’, it is necessary to decrease the outlet angle of flow from the controllable blades of guide vane ring (Fig. 2a) to such extent, which is necessary to maintain the same value of inlet angle of flow onto rotor blades. The analogous situation takes place during the operation of compressor at greater values of rotational speed of rotor, for which the axial component value of absolute velocity, c_{1a}”, increases. Then it is necessary to increase the outlet angle of flow from guide vane blades (Fig. 2c) to maintain the stable work of compressor hence and constant value of inlet angle of flow onto rotor blades.

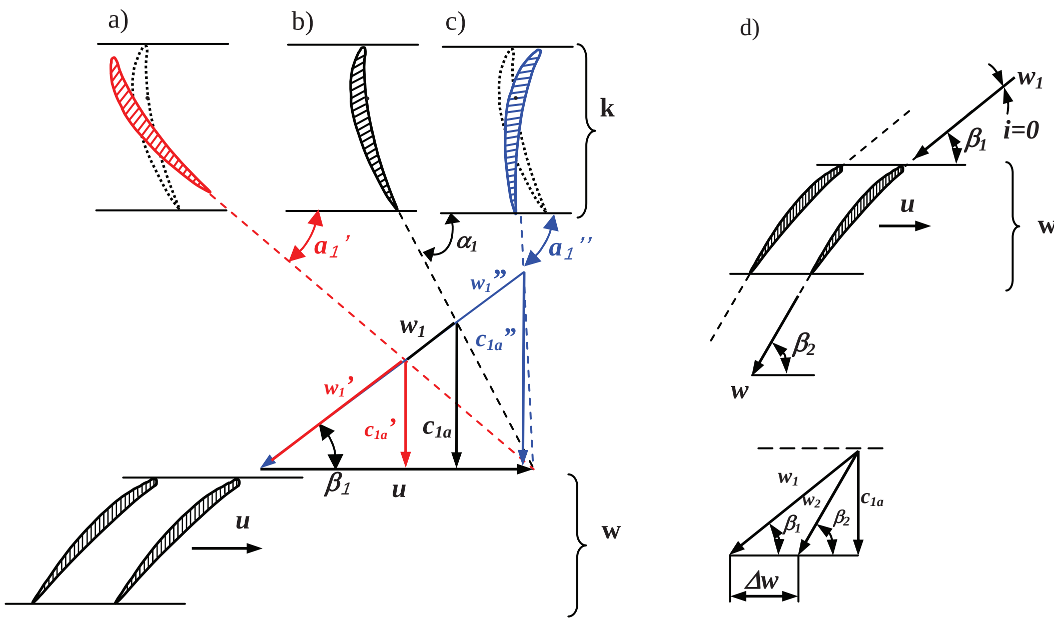


Fig. 2. Essence of control of compressor’s axial stage by changing the setting angle of stator vanes ring at changeable air flow velocity; a) decreased axial velocity, b) calculated axial velocity, c) increased axial velocity, d) schema of flow round of axial compressor rotor blades during the constant rotor speed and constant air stream inlet angles; k – variable stator vanes ring, w – rotor vanes ring

Application of a control system of geometry of flow passages to the gas turbine engine of a given design type significantly influences on the unsteady processes. In the multi-stage axial compressors of compression value exceeding 8-10 the design solution which ensures stable operation, is to apply a.o. controllable blades of inlet guide vane. In Fig. 3 are presented compressor characteristics of the engine fitted with controllable inlet guide vane.

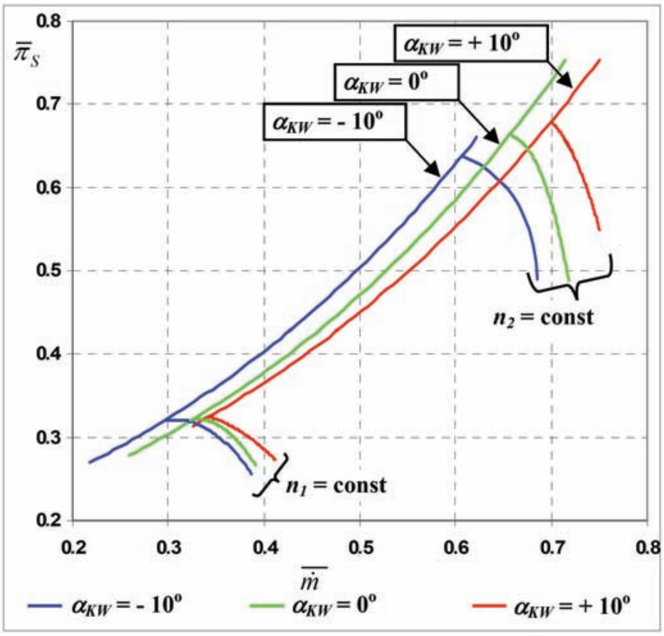


Fig. 3. Compressor characteristics – change of compressor operation range resulting from interaction of controllable inlet guide vanes

The collared lines on the characteristics represent the compressor's operation at three angular settings of inlet guide vane blades, α_{KW}, and the compressor rotational speed, n1, which is constant.

From the run of the lines shown on the characteristics it can be concluded that in the engine fitted with controllable inlet guide vane, it is possible to control the output parameters of its compressor at constant rotational speed of rotor unit.

The DR76 and DR77 triple-shaft engines operating within COGAG propulsion system installed on naval ships of "Tarantula" class are equipped with such design solution of inlet guide vanes. In modern solutions of currently manufactured marine engines their compressors are fitted with controllable inlet guide vanes and controllable guide vanes of their first stages in order to ensure sufficient stable work margins. Such compressors are characterized by the high compression values which exceed 20. From that type are the objects being a part of e.g. LM 2500 engines used for propelling the frigates of Oliver Hazard Perry class, as well as TW-3 aircraft engines applied on Mi helicopters.

The characteristic feature of the sixteen-stage axial compressor of LM 2500 engine is the possibility of changing the setting of outlet flow angles a1 of the blades of inlet guide vane and those of its first six stages in function of engine load. The solution prevents from the unstable work of compressor during the fast realization of transient processes from one stable state to another. In the case of aircraft version of the engine such solution makes it possible to transit from "low gas" state to "full load" within only five seconds not going beyond stable operation zone.

4. Mathematical model of characteristics of axial compressor with controllable blades of inlet guide vane

The problems of elaboration of sufficiently accurate mathematical models of axial compressors are associated with range of simplifying assumptions which determine the accuracy of the numerical modelling of real object. Simulating the investigations requires a.o. to convert the usual graphical form of compressor characteristics (Fig. 3) into functional one suitable for numerical calculations, i.e. to the following form:

$$\pi_s^{\ *} = f\left(\dot{m}_{zr}, n_{zr}, \alpha_{KW}\right), \tag{1}$$

$$\eta_s^* = f(\dot{m}_{zr}, n_{zr}, \alpha_{KW}). \tag{2}$$

The obtaining of an analytical form of the functional relations (l) and (2) which model real characteristics of compressor, at maintained minimum approximation error, is associated with the difficulties, which result from the complex form of functions. In the range of low rotational speeds, isodroms of the characteristics exhibit moderate slopes which correspond with $\pi_S \approx$ idem, whereas in the range of high rotational speeds they show steep sections corresponding with $\dot{m}_{zr} \approx$ idem. Hence it seems that the least squares method and multidimensional polynomial-based regression can be an effective way for determining the analytical description of axial compressor operation, which guarantees that deviations of the model from reality would be contained within the limits of measurement error.

The overall model of compressor is searched for by means of the set of regression equations which approximate its universal characteristics:

$$\pi_s^* = a_0 + a_1 \dot{m}_{zr} + a_2\left(\dot{m}_{zr}\right)^2 + a_3 n_{zr} + a_4\left(n_{zr}\right)^2 + a_5 \dot{m}_{zr}\, n_{zr} + a_6\alpha_{KW} + a_7\left(\alpha_{KW}\right)^2 + $$
$$+ a_8 \dot{m}_{zr}\, \alpha_{KW} + a_9 n_{zr}\alpha_{KW}, \tag{3}$$

$$\eta_s^* = b_0 + b_1 \dot{m}_{zr} + b_2\left(\dot{m}_{zr}\right)^2 + b_3 n_{zr} + b_4\left(n_{zr}\right)^2 + b_5 \dot{m}_{zr}\, n_{zr} + b_6\alpha_{KW} + b_7\left(\alpha_{KW}\right)^2 +$$
$$+ b_8 \dot{m}_{zr}\, \alpha_{KW} + b_9 n_{zr}\alpha_{KW}, \quad . \tag{4}$$

Values of the regression coefficients a_i, b_i are determined on the basis of the Gauss-Markov theorem when searching for minimum values of the functions:

$$J_{\pi_s^*}\left(a_0, a_1, a_2, a_3, a_4, a_5, a_6, a_7, a_8, a_9\right) = \sum_{k=1}^{n}\left(\pi_{sk}^{\ *} - \overline{\pi}_{sk}^{\ *}\right)^2, \tag{5}$$

$$J_{\eta_s^*}\left(b_0, b_1, b_2, b_3, b_4, b_5, b_6, b_7, b_8, b_9\right) = \sum_{k=1}^{n}\left(\eta_{sk}^* - \overline{\eta}_{sk}^*\right)^2. \tag{6}$$

They constitute sums of squares of deviations of the values obtained by using the model $\pi_s^{\ *}, \eta_s^*$ from real values $\overline{\pi}_s^{\ *}, \overline{\eta}_s^*$. Adequacy of matching the mathematical description of characteristics of a given compressor and its real run can be assessed on the basis of following factors :

a) value of the multi-dimensional correlation coefficient R expressed as follows:

$$R = \frac{\sum_{n=1}^{N}\left(\left(Y_n - \overline{Y}_r\right)\cdot\left(\hat{Y}_n - \overline{Y}_m\right)\right)}{\sqrt{\sum_{n=1}^{N}\left(Y_n - \overline{Y}_r\right)^2 \cdot \sum_{n=1}^{N}\left(\hat{Y}_n - \overline{Y}_m\right)^2}}, \tag{7}$$

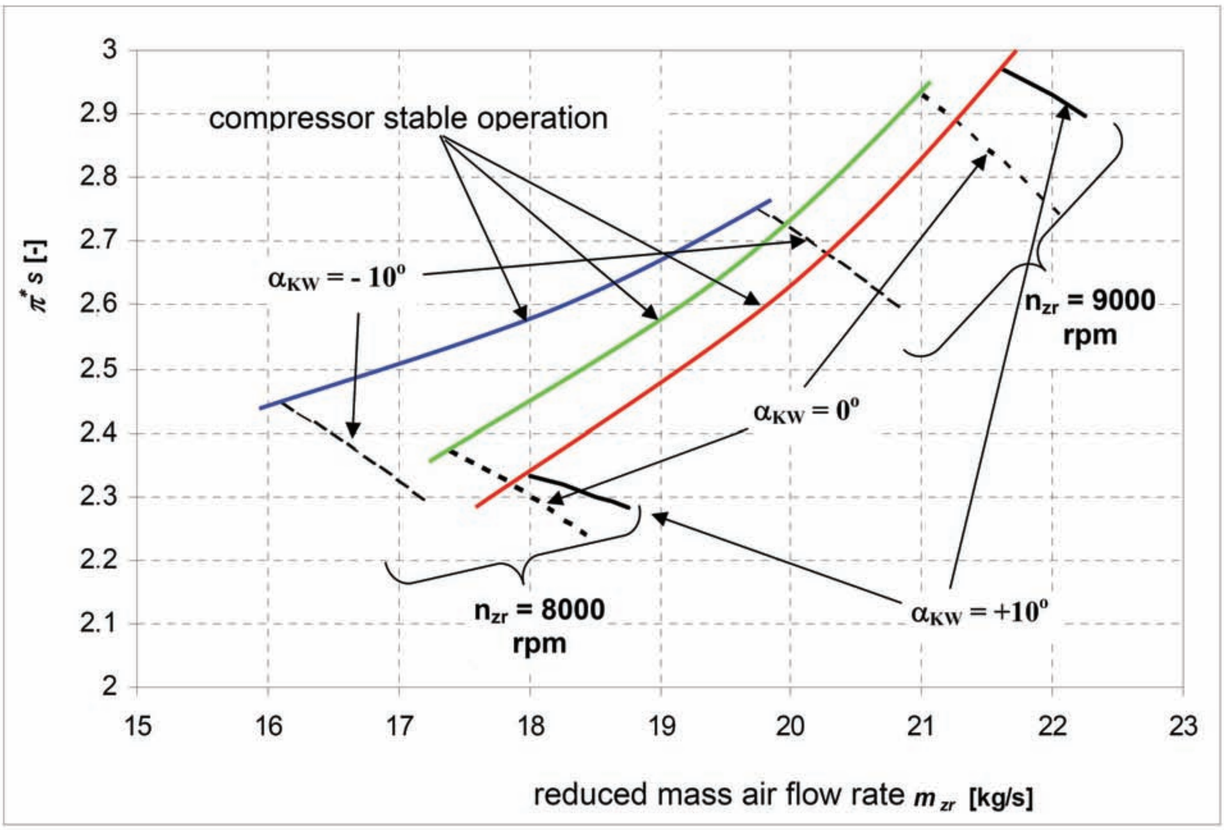
b) value of the remainder variance $\hat{\sigma}^2$ expressed as follows :

$$\hat{\sigma}^2 = \frac{1}{N - K - 1}\sum_{n=1}^{N}\left(Y_n - \hat{Y}_n\right)^2 \cdot \tag{8}$$

Value of R should be as high as possible, and value of $\hat{\sigma}^2$ - as low as possible. Then it speaks for an insignificant deviation of the model from reality.

The mathematical model of compressor can be improved by adding new terms or replacing the existing ones in the equations (3) and (4). Solution of the mathematical model of compressor consists in determining values of the coefficients in the elaborated regression equations.

The low pressure compressor of DR76 engine was used to confirm the usefulness of the least squares method and multi-dimensional polynomial-based regression method for building the mathematical model of characteristics of the compressor with changeable flow passage geometry. In Fig.4 a part of the compressor model characteristics is presented for three angular settings of controllable inlet guide vane: α_{KW} = -10°, α_{KW} = 0°, α_{KW} = +10°. Generally, for each angular position of guide vane blades the compressor characteristics take the form described by three values: rotational speed of compressor rotor n, mass air flow rate $\dot{m}_{zr}$ and the compressor compression π_S. In Fig.4 can be observed the changes of the compressor operation range for two values of reduced rotational speed of compressor, n_{zr} = 8000 and 9000 rpm, in function of the angle of setting of controllable blades of initial whirl guide vane.



stable operation limit line for the angle of setting of controllable guide vanes α_{KW} = -10°,
stable operation limit line for the angle of setting of controllable guide vanes α_{KW} = 0°,
stable operation limit line for the angle of setting of controllable guide vanes α_{KW} = +10°

Fig. 4. Compressor model characteristic for changeable angles of setting of controllable blades of initial whirl guide vanes

Values of regression coefficients of the equations, which approximate characteristics of the compressor in question, are given in Tab. l and in Tab. 2 - values of statistical parameters show the degree of adequacy of values obtained from model investigations in comparison to experimental ones.

Tab. 1. Values of regression coefficients for particular equations

No.	Regression coefficient	Value	Regression coefficient	Value
1	a_0	2.099955	b_0	-0.583216
2	a_1	-0.312917	b_1	-0.272626
3	a_2	-0.022847	b_2	-0.017450
4	a_3	0.000047	b_3	0.000976
5	a_4	0.000000	b_4	0.000000
6	a_5	0.000115	b_5	0.000113
7	a_6	0.059954	b_6	0.068760
8	a_7	-0.003556	b_7	0.000929
9	a_8	0.016743	b_8	0.009397
10	a_9	-0.000038	b_9	-0.000032

Tab. 2. Degree of adequacy between the model and real object

No.	Characteristic	R	$\hat{\sigma}^2$
1	$\pi_s^{\ *} = f\left(\dot{m}_{zr}, n_{zr}, \alpha_{KW}\right)$	0.99939	0,00628
2	$\eta_s^* = f(\dot{m}_{zr}, n_{zr}, \alpha_{KW})$	0.99782	0.00298

5. Conclusions

- Presented mathematical model of axial compressor working in the gas turbine engine system makes it possible to numerically investigate the gas-dynamic processes, which take place in its flow passages.
- The changeable setting angles of blades of controllable guide vanes of particular compression stages, taken into account in the model, significantly widens possible identification of unserviceability states in the system of flow passage geometry control.
- Simulative investigations of gas-dynamic processes, which take place in the compressor in the conditions of introduced changes in set-up values of the input quantities of the presented model, enables determining the diagnostic relations of "defect-symptom" used for assessing technical states of gas-turbine engines.

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