manufacturing industrial engineering

Journal of Manufacturing and Industrial Engineering (MIE)

Scientific and professional journal of Faculty of Manufacturing Technology of Technical University of Košice with a seat in Prešov

ISSN 1335-7972 (Print)
ISSN 1339-2972 (On-line)

Investigation of nonlinear axial rotor oscillations of the multistage centrifugal compressor with the automatic balancing device

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Keywords

centrifugal compressor, face gap, pressure difference regulator, axial force, amplitude frequency characteristic, transient process, dynamic stability

Abstract

In this paper is represented the method of dynamic analysis of rotor vibrations of multistage centrifugal compressor with automatic balancing device based on nonlinear mathematical model that determines the axial movement of the rotor and leakages through hydraulic throttling gaps with non-stationary components. To analyze nonlinear oscillations the method of linearization is used. Dynamic stability of the system was investigated by Hurwitz criterion.

Article		
	History	Received 25.09.2013 | Revised | Accepted
	Category	Original Scientific Paper
	Citation	Pavlenko I (2013) Investigation of nonlinear axial rotor oscillations of the multistage centrifugal compressor with the automatic balancing device *Journal of Manufacturing and Industrial Engineering*, 12(3-4):20-24, http://dx.doi.org/10.12776/mie.v12i3-4.

Introduction

Axial rotor balancing in multistage centrifugal air or gas compressors is carried out mainly by using the unloading plungers. The residual axial force is unloaded by end bearings. Leakage of working gas is limited by end seals. Such systems are the most often applied constructions, but they are complicated. They consist of piston, seal and axial bearing.

Paper [1] presents a new design of the axial forces balancing device of the multistage centrifugal compressor, which has automatic unloading disc working in sealing liquid. That balancing device completely closes the air or gas compressor. Regulator of flow rate of sealing liquid provides right quantum of liquid under the disc. Paper [2] presents the static and flow characteristics of closing automatic rotor balancing device (CARBD) in the multistage centrifugal compressor. These characteristics are obtained basing on equations of rotor axial equilibrium and flow balance. These devices are complex gas-liquid-dynamic or gas- dynamic system with feedback that under certain conditions may have intensive self-oscillations, which affects the vibration state of the compressor.

In papers [3-4] is represented methodology of static and dynamic calculations of the centrifugal machine rotor characteristics.

Paper [5] considers linearised dynamic equations of simplified device design for laminar gas flow regime.

The results of those articles require additional work which would take into account the turbulent gas flow regime, refinement of expressions for calculation leakages through throttling gaps. New design must include the regulator of pressure difference (RPD) to ensure the constant pressure difference between closing and working air or gas.

Material and Methods (Experiment)

Scheme of the closing automatic rotor-balancing device is represented in Figure 1.

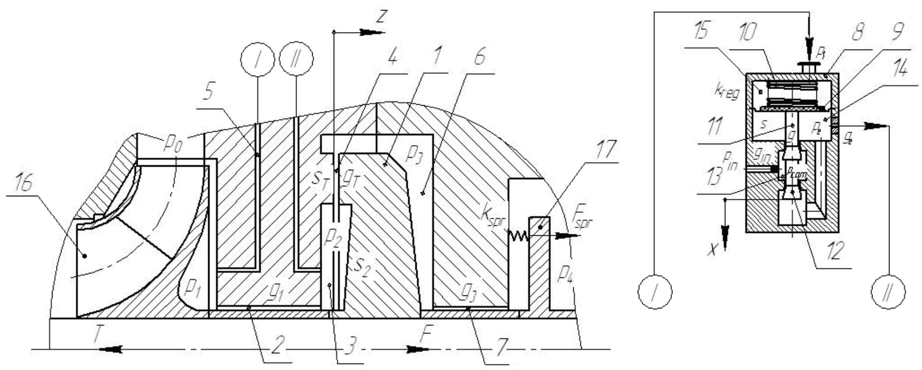

Figure 1 CARBD scheme: 1 – unloading disc; 2 – cylindrical gap; 3 – chamber on front of disc; 4 – face gap; 5 – pulse channel; 6 – chamber behind of disc; 7 – outlet axial gap; 8 – RPD housing; 9 – membrane; 10, 17 – spring elements; 11 – rod; 12 – valve head; 13 – RPD inlet chamber; 14, 15 – membrane chambers; 16 – last stage impeller

The principle of this device operation is: axial force T acting to the rotor is unloaded by disk 1. Closing air or gas is supplied to the chamber on front of disc, through the gap 2. Pressure in the chambers 3 and 6 depends on the axial gap z. Random change of axial force T changes the value of z. Thus, pressure difference $(p_2 - p_3)$ takes the value that provides equality of unloading force F and the axial force T.

CARBD of multistage centrifugal compressor is the automatic control system for which the axial gap z and closing air or gas leakages Q_e are controlled variables, axial force F is regulating action; axial force T, discharge pressure p_1 and closing air or gas pressure p_e are external factors.

MATERIAL AND METHODS (MODEL)

Dynamic analysis of the CARBD is to determine the dynamic characteristics of the system "rotor-CARB-RPD" based on the equations of axial movement of the compressor rotor and the RPD rod:

$$m_r\ddot{z}+c_z\dot{z}+k_{spr}z=F-T+F_{spr};\; m_0\ddot{x}+c_x\dot{x}+k_{reg}x=F_{reg}-F_m \quad (1)$$

and balance equations through available throttling gaps according to the hydraulic path (Figure 2) considering of non-stationary components: $Q_{in} = Q_{cam} + Q_{cam}^{p} + Q_{cam}^{v} = Q_e + Q_e^{p} + Q_e^{v} = Q_1 + Q_T + Q_T^{p} + Q_T^{v}$; $Q_T + Q_T^{p} + Q_T^{v} = Q_3 + Q_3^{p} + Q_3^{v}$, where z, x – axial movement of the rotor and RPD rod; T – total axial force acting to the rotor; $F = s_2p_2 + 0.5s_T(p_2 + p_3) - s_3p_3 = s_e(p_2 - p_3)$ – unloading force; $s_e = s_2 + 0,5s_T$ – effective area; s_2 – area under the face gap; s_T – face gap area; $F_{spr} = k_{spr}\Delta$ – spring force to prevent scoring of face surfaces; $F_{reg} = k_{reg}\Delta_{reg}$ – previous deformation force of the RPT spring elements; k_{spr}, k_{reg}, Δ, Δ_{reg} – stiffness coefficients and previous deformations of the spring elements; $F_m = s_m(p_e - p_1)$ – force acting to the work membrane area s_m; m_r, m_0 – rotor and rod masses; c_z, c_x – damping coefficients; Q_{in} – inlet gas leakage; Q_{cam} – leakage through throttling gap between the RPD valve head and housing; Q_e – closing air or gas leakage; Q_1, Q_3, Q_T – leakages through inlet, outlet and face gaps; Q_{cam}^{p}, Q_e^{p}, Q_T^{p}, Q_3^{p}, Q_{cam}^{v}, Q_e^{v}, Q_T^{v}, Q_3^{v} – compressing and displacement leakages:

$$Q_{cam}^{p}=V_{cam}\dot{p}_{cam}/E;\; Q_e^{p}=V_m\dot{p}_e/E;\; Q_T^{p}=V_2\dot{p}_2/E;\; Q_3^{p}=V_3\dot{p}_3/E;\; Q_{cam}^{v}=s_c\dot{x};\; Q_e^{v}=s_m\dot{x};\; Q_T^{v}=Q_3^{v}=s_e\dot{z}; \quad (2)$$

V_{cam}, V_m, V_2, V_3 – volumes of hydraulic path chambers; E – adiabatic modulus of closing air or gas; s_c – contact area of the RPD saddle.

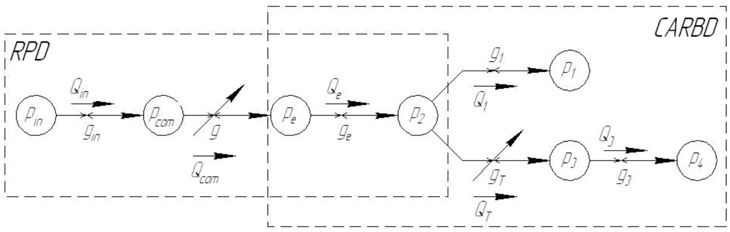

Figure 2 Hydraulic path scheme

Leakage through throttling cylindrical or face gaps are determined by dependences:

$$\begin{aligned}&Q_{in}=g_{in}(p_{in}-p_{cam});\; Q_{cam}=g\sqrt{p_{cam}^2-p_e^2};\; Q_e=g_e(p_e-p_2);\\&Q_1=g_1(p_2-p_1);\; Q_T=g_T\sqrt{p_2^2-p_3^2};\; Q_3=g_3\sqrt{p_3^2-p_4^2},\end{aligned} \quad (3)$$

where g_1, g_3, $g_T = g_{Tb}u^{1.5}$ – conductivity of throttling cylindrical and face gaps; g_{in}, g_e, $g = g_b\xi^{1.5}$ – conductivity of RPD throttling gaps; $u = z/z_b$; $\xi = x/x_b$ – dimensionless gap values; g_b, g_{Tb} – base conductivities for nominal values of x, z; p_1 – discharge pressure; p_{in} – inlet pressure in RPD; p_e – closing pressure; p_{cam}, p_2, p_3 – chambers pressure ; p_4 – outlet pressure.

Taking into account the expressions (1) – (4) is allows obtaining the system of equations for dynamic analysis:

$$\begin{cases}m_r\ddot{z}+c_z\dot{z}+k_{spr}z=s_e(p_2-p_3)-T+k_{spr}\Delta;\; m_0\ddot{x}+c_x\dot{x}+k_{reg}x=k_{reg}\Delta_{reg}-s_m(p_e-p_1);\\ g_{in}(p_{in}-p_{cam})=g_b\xi^{3/2}\sqrt{p_{cam}^2-p_e^2}+V_{cam}\dot{p}_{cam}/E+s_c\dot{x}=\\ =g_e(p_e-p_2)+V_m\dot{p}_e/E+s_m\dot{x}=g_1(p_2-p_1)+g_{Tb}u^{3/2}\sqrt{p_2^2-p_3^2}+V_2\dot{p}_2/E+s_e\dot{z};\\ g_{Tb}u^{3/2}\sqrt{p_2^2-p_3^2}+V_2\dot{p}_2/E+s_e\dot{z}=g_3\sqrt{p_3^2-p_4^2}+V_3\dot{p}_3/E+s_e\,\dot{z}.\end{cases} \quad (4)$$

MATERIAL AND METHODS (CALCULATIONS)

The system of nonlinear differential equations (4) can not be solved analytically. Further research is conducted for variations of time-variables parameters ("δ" is variation sign) by linearization $z = z_0 + \delta z$, $u = u_0 + \delta u$, $x = x_0 + \delta x$, $\xi = \xi_0 + \delta\xi$, $T = T_0 + \delta T$, $F = F_0 + \delta F$, $p_1 = p_{10} + \delta p_1$, $p_{cam} = p_{cam0} + \delta p_{cam}$, $p_e = p_{e0} + \delta p_e$, $p_2 = p_{20} + \delta p_2$, $p_3 = p_{30} + \delta p_3$ relatively to stationary values (with index "0") as a result of solving the system of algebraic equations of static analysis

$$\begin{cases} s_e\left(p_{20}-p_{30}\right)=T_0-k_{spr}\Delta;\ s_m\left(p_{e0}-p_{10}\right)=k_{reg}\Delta_{reg};\\ g_{in}\left(p_{in0}-p_{cam0}\right)=g_b\xi_0^{1,5}\sqrt{p_{cam0}^2-p_{e0}^2}=g_e\left(p_{e0}-p_{20}\right)=g_1\left(p_{20}-p_{10}\right)+g_{Tb}u_0^{1,5}\sqrt{p_{20}^2-p_3^2};\\ g_{Tb}u_0^{1,5}\sqrt{p_{20}^2-p_{30}^2}=g_3\sqrt{p_{30}^2-p_4^2} \end{cases} \quad (5)$$

relatively to parameters u_0, ξ_0, p_{cam0}, p_{e0}, p_{20}, p_{30}.

For further calculations are used dimensionless parameters: $\delta\psi_{cam}=\delta p_{cam}/p_b$; $\delta\psi_e=\delta p_e/p_b$; $\delta\psi_1=\delta p_1/p_b$; $\delta\psi_2=\delta p_2/p_b$; $\delta\psi_3=\delta p_3/p_b$; $\delta\tau=\delta T/(p_b s_b)$; $\delta\varphi=\delta F/(p_b s_b)$, where p_b is base pressure value corresponding to the nominal discharge pressure p_n of the compressor; $s_b=T_n/p_n$ – base area as the ratio of nominal axial force T_n to pressure p_n.

In parameters $\delta\tau=b\cdot\delta\psi_1$, $\delta\varphi=\sigma(\delta\psi_2-\delta\psi_3)$ (b – proportionality coefficient, $\sigma=s_e/s_b$ – dimensionless effective area) system of dynamics equations (4) can be represented in matrix and operator form

$$N(p)\delta U=B\delta\psi_1, \quad (6)$$

where

$$N(p)=\begin{bmatrix} K_1\left(T_1^2p^2+2\zeta_1T_1p+1\right) & 0 & 0 & 0 & -\sigma & \sigma\\ 0 & K_2\left(T_2^2p^2+2\zeta_2T_2p+1\right) & 0 & \sigma_м & 0 & 0\\ 0 & K_4(T_4p+1) & T_3p+1 & -K_3 & 0 & 0\\ 0 & -K_6(T_6p+1) & -K_5(\tau_3p+1) & T_5p+1 & -K_7 & 0\\ K_8(T_8p+1) & \tau_6p & 0 & -K_9(\tau_5p+1) & T_7p+1 & -K_{10}\\ -K_{12} & 0 & 0 & 0 & -K_{13}(\tau_7p+1) & T_9p+1 \end{bmatrix} \quad (7)$$

is matrix of differentiation operators; $\delta U=\{\delta u\ \delta\xi\ \delta\psi_{кам}\ \delta\psi_e\ \delta\psi_2\ \delta\psi_3\}^T$ is reaction of the system to the external action $B\delta\psi_1$, where $B=\{-b\ \sigma_м\ 0\ 0\ K_{11}\ 0\}^T$.

The matrix $N(p)$ contains 28 constant parameters: time constants $T_{1\ldots9}$, $\tau_{3,5,6,7}$, damping coefficients $\zeta_{1,2}$ and amplification factors $K_{1\ldots13}$:

$$T_1=\sqrt{\frac{m_r}{k_{spr}}};\ T_2=\sqrt{\frac{m_0}{k_{reg}}};\ T_3=\frac{V_{cam}/E}{g_{in}+\dfrac{g_Б\xi_0^{3/2}p_{cam0}}{\sqrt{p_{cam0}^2-p_{e0}^2}}};\ T_4=\frac{2s_cz_b}{3g_b\sqrt{\xi_0\left(p_{cam0}^2-p_{e0}^2\right)}};\ T_5=\frac{V_m/E}{\dfrac{g_b\xi_0^{3/2}p_{e0}}{\sqrt{p_{cam0}^2-p_{e0}^2}}+g_e};$$

$$T_6=\frac{2\left(s_c-s_m\right)z_b}{3g_b\sqrt{\xi_0\left(p_{cam0}^2-p_{e0}^2\right)}};\ T_7=\frac{V_2/E}{g_e+g_1+\dfrac{g_{Tb}u_0^{3/2}p_{20}}{\sqrt{p_{20}^2-p_{30}^2}}};\ T_8=\frac{2s_ez_b}{3g_{Tb}\sqrt{u_0\left(p_{20}^2-p_{30}^2\right)}};\ T_9=\frac{V_3/E}{p_{30}\left(\dfrac{g_{Tb}u_0^{3/2}}{\sqrt{p_{20}^2-p_{30}^2}}+\dfrac{g_3}{\sqrt{p_{30}^2-p_4^2}}\right)};$$

$$\tau_3=\left(1+\frac{g_{in}}{g_b\xi_0^{3/2}}\sqrt{1-\frac{p_{e0}^2}{p_{cam0}^2}}\right)T_3;\ \tau_5=\left(1+\frac{g_b\xi_0^{3/2}}{g_{in}}\frac{\sqrt{p_{cam0}^2-p_{e0}^2}}{p_{cam0}}\right)T_5;\ \tau_6=\frac{1,5s_m}{s_c-s_m}\frac{3g_b\sqrt{\xi_0\left(p_{cam0}^2-p_{e0}^2\right)}}{p_b\left(g_e+g_1+\dfrac{g_{Tb}u_0^{3/2}p_{20}}{\sqrt{p_{20}^2-p_{30}^2}}\right)}T_6;$$

$$\tau_7=\left(1+\frac{g_e+g_1}{g_{Tb}u_0^{3/2}}\sqrt{1-\frac{p_{30}^2}{p_{20}^2}}\right)T_7;\ \zeta_1=\frac{c_z}{2k_{spr}T_1};\ \zeta_2=\frac{c_x}{2k_{reg}T_2};\ K_1=\frac{k_{spr}z_b}{p_bs_b};\ K_2=\frac{k_{reg}z_b}{p_bs_b};$$

$$K_3=\frac{g_b\xi_0^{3/2}p_{e0}}{g_{in}\sqrt{p_{cam0}^2-p_{e0}^2}+g_b\xi_0^{3/2}p_{cam0}};\ K_4=\frac{3g_b\sqrt{\xi_0\left(p_{cam0}^2-p_{e0}^2\right)}}{2p_b\left(g_{in}+\dfrac{g_b\xi_0^{3/2}p_{cam0}}{\sqrt{p_{cam0}^2-p_{e0}^2}}\right)};\ K_5=\frac{g_b\xi_0^{3/2}p_{cam0}}{g_b\xi_0^{3/2}p_{e0}+g_e\sqrt{p_{cam0}^2-p_{e0}^2}};$$

$$K_6=\frac{3g_b\sqrt{\xi_0\left(p_{cam0}^2-p_{e0}^2\right)}}{2p_b\left(\dfrac{g_b\xi_0^{3/2}p_{e0}}{\sqrt{p_{cam0}^2-p_{e0}^2}}+g_e\right)};\ K_7=\frac{g_e}{\dfrac{g_b\xi_0^{3/2}p_{e0}}{\sqrt{p_{cam0}^2-p_{e0}^2}}+g_e};K_8=\frac{3g_{Tb}\sqrt{u_0\left(p_{20}^2-p_{30}^2\right)}}{2p_b\left(g_e+g_1+\dfrac{g_{Tb}u_0^{3/2}p_{20}}{\sqrt{p_{20}^2-p_{30}^2}}\right)}; \quad (8)$$

$$K_9=\frac{g_e}{g_e+g_1+\dfrac{g_{Tb}u_0^{3/2}p_{20}}{\sqrt{p_{20}^2-p_{30}^2}}};\ K_{10}=\frac{g_{Tb}u_0^{3/2}p_{30}}{\left(g_e+g_1\right)\sqrt{p_{20}^2-p_{30}^2}+g_{Tb}u_0^{3/2}p_{20}};\ K_{11}=\frac{g_1}{g_e+g_1+\dfrac{g_{Tb}u_0^{3/2}p_{20}}{\sqrt{p_{20}^2-p_{30}^2}}};$$

$$K_{12}=\frac{3g_{Tb}\sqrt{u_0\left(p_{20}^2-p_{30}^2\right)}}{2p_bp_{30}\left(\dfrac{g_{Tb}u_0^{3/2}}{\sqrt{p_{20}^2-p_{30}^2}}+\dfrac{g_3}{\sqrt{p_{30}^2-p_4^2}}\right)};\ K_{13}=\frac{g_{Tb}u_0^{3/2}p_{20}}{p_{30}\left(g_{Tb}u_0^{3/2}+g_3\sqrt{\dfrac{p_{20}^2-p_{30}^2}{p_{30}^2-p_4^2}}\right)}.$$

The matrix $N(i\omega)$ can be decomposed into real and imaginary parts (i is imaginary unit, ω is angular frequency of the rotor):

$$N(i\omega) = N_{\mathrm{Re}}(\omega) + i\omega N_{\mathrm{Im}}(\omega), \tag{9}$$

Real $\overline{U}$ and imaginary $\overline{V}$ parts of the vector of frequency transfer functions $W(i\omega) = [N(p)]^{-1}B = \overline{U}(\omega) + i\omega\overline{V}(\omega)$ are

$$\overline{U} = \left(N_{\mathrm{Re}} N_{\mathrm{Im}}^{-1} N_{\mathrm{Re}} + \omega^2 N_{\mathrm{Im}}\right)^{-1} N_{\mathrm{Re}} N_{\mathrm{Im}}^{-1} B; \quad \overline{V} = -\left(N_{\mathrm{Re}} N_{\mathrm{Im}}^{-1} N_{\mathrm{Re}} + \omega^2 N_{\mathrm{Im}}\right)^{-1} B. \tag{10}$$

Vector of amplitude and phase frequency characteristics consists of modules and phases of elements of vector of $W(i\omega)$:

$$A_i(\omega) = \sqrt{\overline{U}_i^{\,2}(\omega) + \overline{V}_i^{\,2}(\omega)}; \quad \varphi_i(\omega) = arctg\,\frac{\omega\overline{V}_i(\omega)}{\overline{U}_i(\omega)} \quad (i = u, \xi, cam, e, 2, 3). \tag{11}$$

To ensure dynamic stability of the system it is necessary to all real roots of characteristic equation $|N(p)| = a_0\lambda^8 + a_1\lambda^7 + \ldots + a_8$ were negative. By Hurwitz criterion this condition is satisfied if coefficients $a_{0\ldots8}$ and the main diagonal minors of the matrix Δ are positive:

$$\Delta = \begin{bmatrix} a_1 & a_3 & a_5 & a_7 & 0 & 0 & 0 & 0 \\ a_0 & a_2 & a_4 & a_6 & a_8 & 0 & 0 & 0 \\ 0 & a_1 & a_3 & a_5 & a_7 & 0 & 0 & 0 \\ 0 & a_0 & a_2 & a_4 & a_6 & a_8 & 0 & 0 \\ 0 & 0 & a_1 & a_3 & a_5 & a_7 & 0 & 0 \\ 0 & 0 & a_0 & a_2 & a_4 & a_6 & a_8 & 0 \\ 0 & 0 & 0 & a_1 & a_3 & a_5 & a_7 & 0 \\ 0 & 0 & 0 & a_0 & a_2 & a_4 & a_6 & a_8 \end{bmatrix}. \tag{12}$$

Results and Discussion

Numerical calculations are carried out for the compressor K 180-131-1. Initial data for dynamic analysis are parameters of the static calculations: p_b = 4.6 MPa, s_b = 0.039 m^2, z_b = 0.15 mm; u_0 = 0.88, ξ_0 = 0.59; s_c = 3.9·10^{-5} m^2, s_m = 3.9·10^{-4} m^2, s_e = 0.08 m^2; p_{e0} = 5.1 MPa, p_{20} = 4.8 MPa, p_{30} = 2.6 MPa, p_4 = 0; g_{in} = 2.8·10^{-6} m^3/(Pa·s); g_b = 2.3·10^{-7} m^3/(Pa·s); g_e = 2.7·10^{-6} m^3/(Pa·s); g_1 = 2.3·10^{-6} m^3/(Pa·s); g_{Tb} = 3.7·10^{-7} m^3/(Pa·s); g_3 = 4.7·10^{-7} m^3/(Pa·s) and following parameters: m_r = 350 kg, m_0 = 0.65 kg, E = 1.42·10^5 Pa; dynamic viscosity of closing gas μ=1.82·10^{-5} Pa·s; damping coefficients c_z=0.1$\pi\mu(d_3^2 - d_2^2)^2/z_b^3$ = = 6.4·10^3(N·s/m), c_x = c_z = 6.4·10^3 N·s/m; parameters (8) are T_1 = 5.4 ms, T_2 = 1.2 ms, T_3 = 8.7^{-3} ms, T_4 = 1.8·10^{-6} ms, T_5 = 5.4·10^{-4} ms, T_6 = 1.6·10^{-5} ms, T_7 = 4.8·10^{-4} ms, T_8 = 5.8·10^{-3} ms, T_9 = 7.5·10^{-3} ms, τ_3 = 0.02 ms, τ_5 = 5.4·10^{-4} ms, τ_6 = 4.9·10^{-6} ms, τ_7 = 6.9·10^{-3} ms; ζ_1 = 0.05, ζ_2 =0.22; K_1 = 0.01, K_2 = 3.5·10^{-4}, K_3 = 0.02, K_4 = 0.25, K_5 = 0.02, K_6 = 0.15, K_7 = 0.99, K_8 = 0.09, K_9 = 0.87, K_{10} = 0.04, K_{11} = 0.04, K_{12} = 0.68, K_{13} = 0.55; $\delta\psi_{1a}$ = 0.2. Angular velocity of shaft rotation ω_0 = 1480 rad/s. Amplitude frequency characteristic is represented on Figure 3.

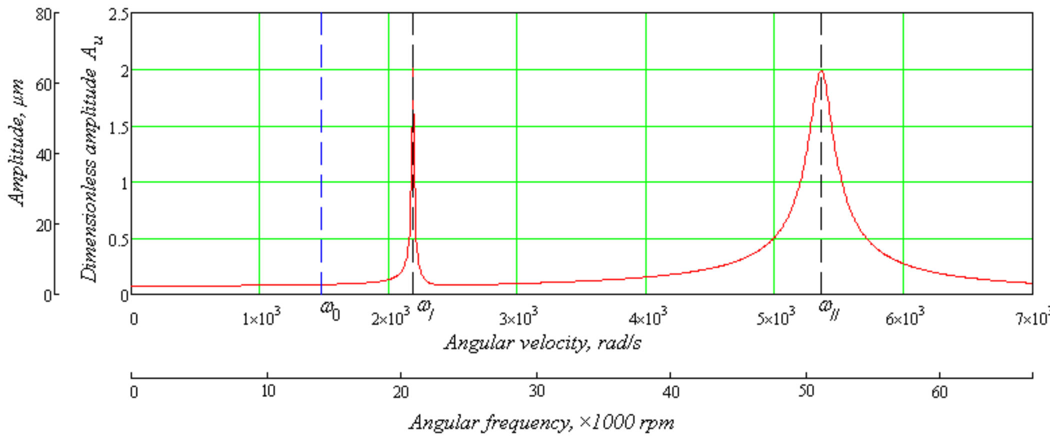

Figure 3 Amplitude frequency characteristic for axial oscillations of the rotor

The first and second resonance frequencies of axial oscillations of the rotor are ω_I = 2190 rad/s and ω_{II} = 5360 rad/s. Operating frequency ω_0 = 0,7ω_I and ω_{II} = 2,4ω_I. Resonance amplitudes of axial oscillations of the rotor are A_I = 62 µm, A_{II} = 60 µm. Amplitude on the operating mode A_0 = 3 µm corresponds to accident-free mode.

Positive values of characteristic equation coefficients a_0 = 2,65·10^{-40}, a_1 = 8,8·10^{-34}, a_2 = 8,4·10^{-28}, a_3 = 2,2·10^{-22}, a_4 = 1,6·10^{-17}, a_5 = 1,0·10^{-14}, a_6 = 5,4·10^{-10}, a_7 = 5,0·10^{-8}, a_8 = 2,2·10^{-3} and main diagonal minors of matrix (12) indicate stability of the dynamic system. In addition, numerical roots of characteristic equation (λ_1 = –1,8·10^6, λ_2 = –1,1·10^6, λ_3 = –2,2·10^5, λ_4 = –1,3·10^5, $\lambda_{5,6}$ = –6,0 ± 2,19i, $\lambda_{7,8}$ =–93,1 ± 5,36·10$^3 i$) have negative real parts. Modules of imaginary parts of roots $\lambda_{5,6,7}$ are equal to natural frequencies of the system: ω_I = 2190 rad/s, ω_{II} = 5360 rad/s.

Figure 4 represents transient characteristics of CARBD. Control time is t_0 = 40 ms, maximum over-control for the rotor is 11 µm.

 http://dx.doi.org/10.12776/mie.v12i3-4.

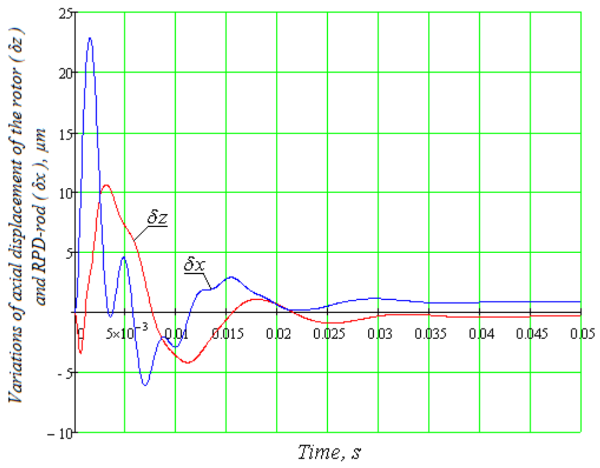

Figure 4 Transient processes

NEW APPROACHES FOR PROBLEM SOLUTION

The mathematical model, which is stated in this article, is a system of nonlinear differential equations of 8^{th} order that describes rotor and rod motions and leakages through laminar and turbulent cylindrical and face throttling gaps. Described method of dynamic analysis of the dynamic system "rotor – automatic axial balancing device – regulator of pressure difference" – is based on this model.

Research provides an opportunity to build amplitude and phase frequency characteristics, calculate natural frequencies of axial oscillations of the rotor and check dynamic stability of the system. The results can be used for design calculations of rotor vibration state of the multistage centrifugal compressor with automatic balancing device.

CONCLUSIONS AND FUTURE DIRECTION OF RESEARCH

The closing automatic rotor-balancing device of the multistage centrifugal compressor with regulator of pressure difference acts as the end seal and the hydrostatic bearing with self-regulating gap and gas leakages. The main advantages of this design are absence of end seals and bearings and leakage of working gas.

Amplitude frequency characteristic of rotor axial oscillations and transient processes and dynamic stability of the system are represented as example for compressor K 180-131-1

At this stage dynamic stability of the system is implemented by numerical verification by Hurwitz criterion. In perspective it is necessary to define conditions and boundaries for dynamic stability analytically and investigate nonlinear mathematical model without linearization on the basis of Runge-Kutta and Bulirsch-Stoer methods.

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