ROTORDYNAMICS OF COMPRESSOR WITH ELECTROMOTOR SUPPORTED BY AMB

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The article presents the results of modeling and analysis of dynamic characteristics of the radial-flow compressor of the "Turbopneumatic" Co Ltd. with the electromotor produced by the SKF company and supported by magnetic bearings. This modeling was done with the help of the Dynamics R4 program system. The tasks of the compressor rotor system optimization are solved by control of tightening torques, interferences and clearances. The results of the compressor analysis as nonlinear dynamic system are given at unsteady statement. The compressor rotor drop on the auxillary rolling bearings is considered when current is switched off.

The subject of inquiry is the high-speed radial flow compressor supported by the active magnetic bearings (AMB) with the electromotor drive. The compressor is used to build the perspective system of air launch of stationary turbounits. It was developed in the Turbopneumatic" Co Ltd. Fig.1 gives its general view in the test rig.

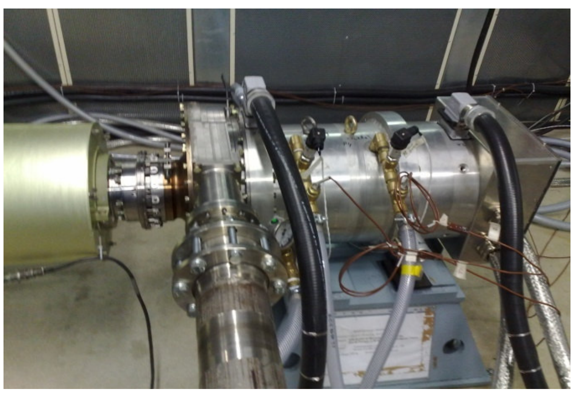

Fig. 1 Compressor in test rig

The main compressor characteristics are given below.

Compression ratio	$\pi^*_{к} = 4{,}81$.
Efficiency	$\eta^*_{ад.к} = 81$ %.
Air consumption	$G_{air} = 1{,}42$ kg/sec.
Compressor shaft power	$N_{k\ phys.} = 270$ kwatt.
Rotating speed	$n_{k\ phys.} = 39\ 000$ rpm.

The increased vibrations appeared while carrying out the compressor tests. Supposed reason for this is appearance of resonance frequency that corresponds to the first bending mode shape and that is close to the operating range, in spite of the fact that design calculations showed the sufficient frequency reserve. The reason for the resonance frequency decrease is loss in tightness between the compressor details – the impeller, the shaft, the labyrinth disc, etc.

The purpose of this work is to check the design solutions of the compressor by dynamic analysis of its rotor system, to recommend the methods for vibrations decrease and improvement of its characteristics.

The general direction in investigation of the compressor design with consideration of the dynamic characteristics obtained at modeling is decrease in the compressor mass and increase of its resource at no-failure operation.

Assigned tasks have been solved by leading of variants dynamic calculations. The main factors influencing the compressor dynamic characteristics are obtained by their results.

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General description of model

The model of the rotor dynamic system is made in the Dynamics R4 program system used to analyze the dynamic characteristics of the rotating machines. The system is developed by the "Alfa Transit" Co Ltd.[1]. Fig.2 shows the virtual rotor model.

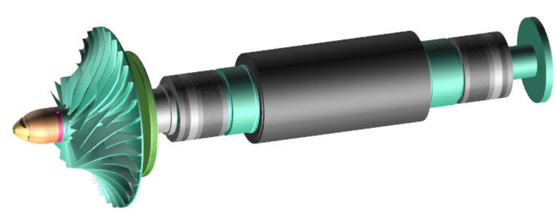


Fig. 2 Virtual model of compressor rotor

Fig.3 gives the main program window with the model including the compressor part and the electromotor supported by the AMB. The case details influence on the compressor characteristics insignificantly, so it is excluded from the model.

In the program system the model is created using the cane segments called "beam", the point inertia elements and rigid links between the subsystems of the whole rotor system, Fig.4.

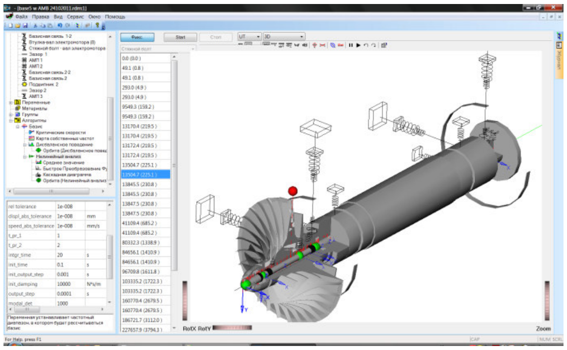


Fig.3 Main Dynamics R4 window with rotor model

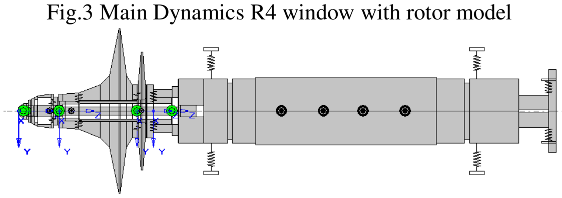


Fig. 4 Two-dimensional rotor model

Model is level-based and it includes five subsystems, Fig. 5:

- compressor impeller;
- impeller bush;
- labyrinth disk;
- electromotor shaft;
- pinch bolt.

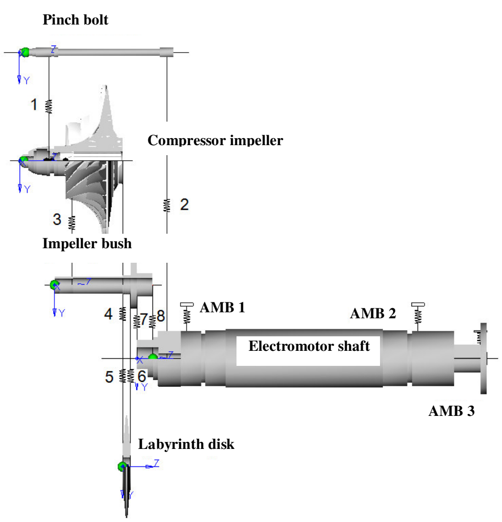


Fig. 5 Level-based rotor model

The electromotor shaft is supported by two magnetic bearings – AMB 1 and AMB 2 and it is fixed in axial direction by the AMB 3 bearing. Magnetic electromotor bearings are shown in Fig. by elastic links with the basement.

In linear analysis elastic links are set by radial stiffness and damping coefficients. Elastic links are set between the model subsystems describing the details of the rotor compressor part. These links are used to simulate clearances and fits between details. Links in the picture are numbered.

Link 1 – models loading in thread connection of the impeller with the pinch bolt.

Link 2 – models loading in thread connection of the pinch bolt with the electromotor shaft.

Link 3 – models fit between the impeller and the wheel bush (centering spigot).

Link 4 – models loading in the junction between the impeller and the labyrinth disk.

Links 5 and 6 - model tightness between the impeller bush and the labyrinth disk.

Links 7 and 8 – model tightness between the wheel bush and the electromotor shaft.

Coefficients of the stiffness matrixes are obtained with consideration of the actual construction characteristics along all degrees of freedom. Coefficients which are equal to the contact stiffness (~1e10 N/m) correspond to the stationary fixing of one detail relative to the other one. Loss in tightness and appearance of the local flexibility is simulated by decrease in the stiffness coefficient; appearance of clearance is simulated by the link deletion.

For linear analysis stiffness of the elastic links modeling radial magnetic bearings AMB 1 and AMB 2 are chosen in such way that the shaft displacement from its own weight accounts for ~5% from the value of the auxillary bearings clearance (0.125 mm).

In the nonlinear rotor model the magnetic bearings are described by the special element included into the library of the program system. In the "Magnetic bearing" element of the Dynamics R4 program system the coefficients for the control current equation of PID-controller of the control system are set. Varying their values, it is possible to obtain the necessary forces and stiffness coefficients of the magnetic bearings [2].

Fig. 6 shows the result of the non-stationary analysis of the rotor supported by the non-linear AMB 1 and AMB 2. On the basis of the analysis the corresponding stiffness coefficients of the supports are obtained; they are further used in linear analysis.

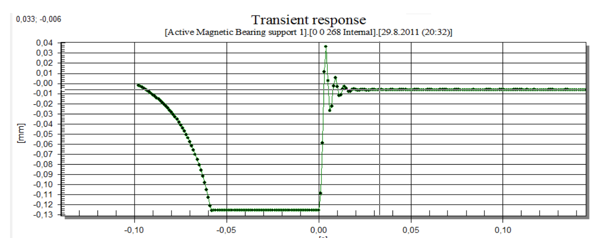


Fig. 6 Transient response (drop of rotor on clearance, rise after AMB switching on)

Let us give some explanatory notes to the Fig.6. At the time moment of -0.1 sec there is start of integrating of the obtained at modeling the movement equations of the rotor system with zero initial conditions at speed and displacement. The rotor under weight force drops down and removes clearance of 0.125mm. It is quiescent from -0.04 sec to 0 sec. At 0.0 sec current in windings of the magnetic supports is switched on, and the rotor rises up to the value of the displacement from weight force that accounts for 0.00625 mm. Stiffness coefficients of magnetic bearings for this case are k = ~ 3e7 Н/м.

Thrust AMB 3 is simulated by the elastic link with the stiffness of 1e10 N/m limiting only axial displacement. All other coefficients are obtained as a result of identification of the rotor model according to the SKF experimental data, Fig. 7.

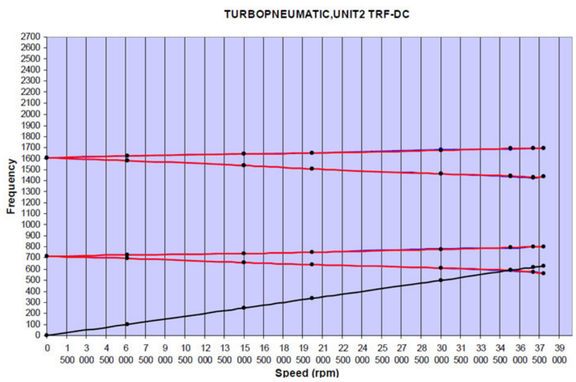


Fig.7 Experimentally obtained Campbell diagram of rotor supported by AMB (frequency in Hz)

Identification was done at main mode shapes highlighted in the diagram and corresponding to the frequencies of non-rotating and undamped compressor rotor: ~ 700 Hz (first bending mode shape); ~ 1600 Hz (second bending mode shape).

Fig.8 presents the identification results (Natural Frequency Map obtained in DYNAMICS R4).

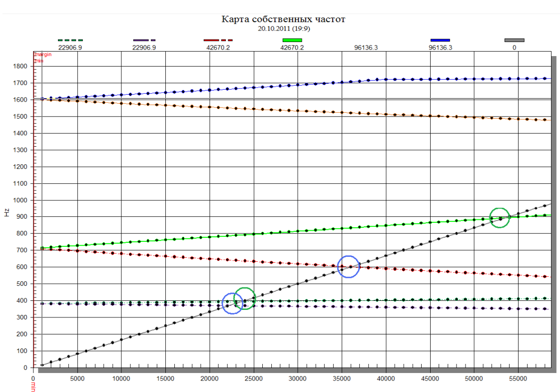
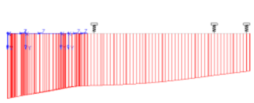
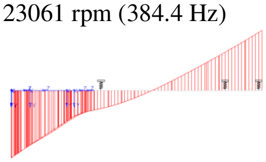
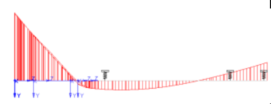
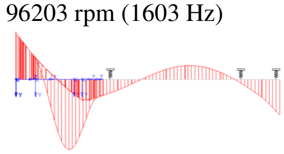


Fig. 8 Campbell diagram from Dynamics R4

Table 1 shows frequencies and mode shapes of the rotor of the compressor. They are obtained after the model identification. Close coincidence of the obtained frequencies with the experimental ones should be noticed. Error does not exceed 1…2 %.

Table 1

11794 rpm (196.6 Hz)	23061 rpm (384.4 Hz)
	
42901 rpm (715 Hz)	96203 rpm (1603 Hz)
	

Distribution of deformation potential energy on the model elements – subsystems and links – is used for identification of frequencies and mode shapes. Distribution knowledge allows highlighting the elements, by change in which stiffness the position of the oscillations natural frequencies may be controlled, Table 2.

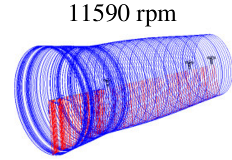
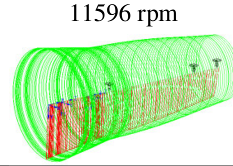
Table 2

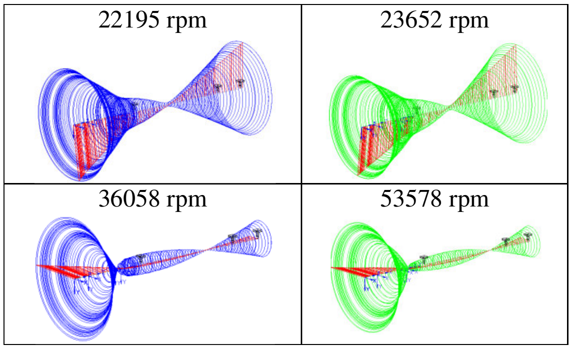
Subsystems and links	Natural frequencies, rpm			
	11794	23061	42901	96203
Pinch bolt	0.010	0.068	0.405	3.861
Impeller + Screw nut	0.032	0.277	2.704	1.803
Impeller bush	0.168	1.504	14.16	5.720
Electromotor shaft	2.680	31.20	46.13	68.736
Labyrinth disk	0.011	0.091	0.735	0.326
AMB 1	58.74	34.99	4.183	12.740
AMB 2	37.89	28.09	1.623	0.064
Impeller-bush (3)	0.000	0.000	0.000	0.000
Bush-electromotor shaft (7)	0.434	3.560	28.59	6.243
Pinch bolt-electromotor shaft (2)	0.000	0.000	0.000	0.002
Bush-labyrinth disk (6)	0.010	0.058	0.209	0.064
Bush-electromotor shaft (8)	0.011	0.100	0.986	0.420
AMB 3	0.009	0.056	0.246	0.020
Impeller-labyrinth disk (4)	0.000	0.001	0.003	0.001
Bush – labyrinth disk (5)	0.000	0.000	0.000	0.000
Screw nut – pinch bolt (1)	0.000	0.000	0.000	0.000
All subsystems	2.90	33.14	64.15	80.45
All links	97.1	66.86	35.85	19.55
Total	100.0	100.0	100.0	100.00

Critical rotating speeds

In Fig.8 critical speeds corresponding both forward and backward precessions are highlighted by circles. Fig.3 presents frequencies and mode shapes corresponding to them.

Table 3

11590 rpm	11596 rpm
	



Critical speed n=53578 rpm, the most dangerous for operating capacity of the rotor of the compressor, is far from the operating regime (39000…40000 rpm). Under conditions of saving of tightening forces and tightness, the mode shape corresponding to this frequency can't lead to the increased vibrations level.

Forced oscillations of rotor

The important part of the rotor analysis is estimation of its forced oscillations under unbalanced forces. To carry it out, the calculation of vibration speed from the impeller unbalance equal to 10 g*cm was carried out. Fig. 9 shows amplitude-frequency rotor characteristics in the AMB 1 section.

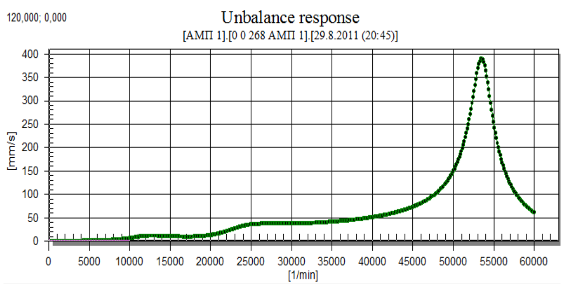


Fig. 9 Amplitude-frequency rotor characteristics in AMB 1 section. Stable tightness.

It should be noticed that in operating modes up to 40000 rpm, obtained vibration speeds do not exceed 50 mm/sec for the given unbalance and damping – this is acceptable value for high-speed rotors.

Influence of tightening torque and tightness

Slackening of axial tightening is simulated by decrease in moment stiffness of links 3,4,5,6 up to 1e5 N*m. In this case the critical rotating speed n = 53578 rpm drops to the compressor operating mode n = 442843 rpm. In forced oscillations such drop is accompanied by increased amplitudes of vibration speed for AMB 1 (Fig. 10) and AMB 2 (Fig. 11) at operating mode 39000…40000 rpm to 125 and 90 mm/sec correspondingly.

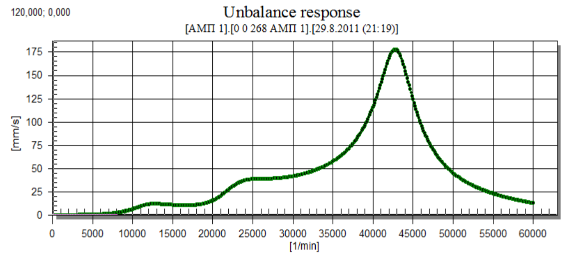


Fig. 10 Amplitude-frequency rotor characteristics in AMB 1 section. Slackening of axial tightening

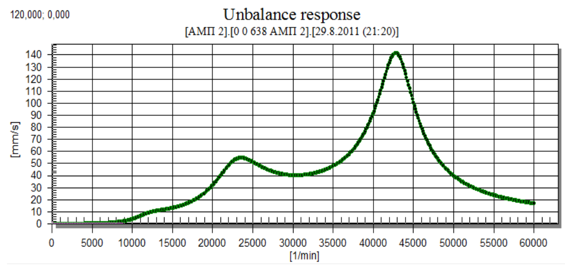


Fig. 11 Amplitude-frequency rotor characteristics in AMB 2 section. Slackening of axial tightening

Tightness slackening is the most influencing factor considering the rotor vibrations at operating mode. It is simulated by decrease in radial stiffness of the links 7 and 8 or by reduction of moment stiffness of the link 7. Simultaneous decrease in radial stiffness of the links 7 and 8 to 1e8 H/м results in reduction of critical speed n=53578 rpm to n=40872 rpm and vibrations growth. Fig.12 and 13 give amplitude-frequency rotor characteristics for this case.

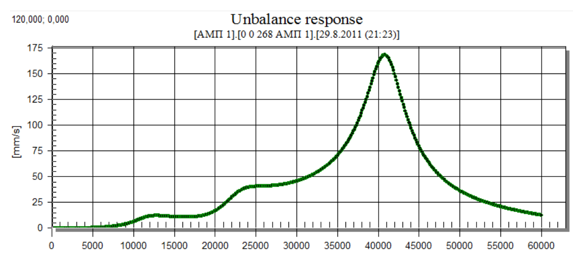


Fig. 12 Amplitude-frequency rotor characteristic in AMB 1 section. Tightness slackening

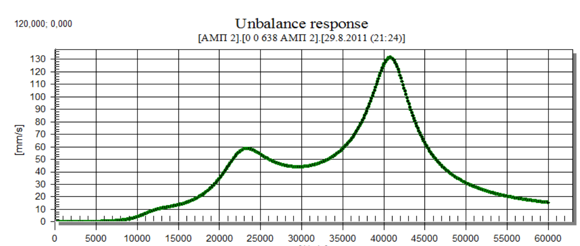


Fig. 13 Amplitude-frequency rotor characteristic in AMB 2 section. Tightness slackening

Simultaneous decrease in radial stiffness to 1e9 N/m and moment stiffness from 450000 Hм to zero of the link 7 results in reduction of critical rotating speed n=53578 rpm to n=36687 rpm. Fig. 14 and 15 give amplitude-frequency rotor characteristics for this case.

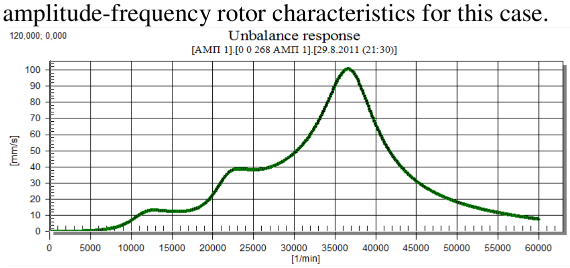


Fig. 14 Amplitude Frequency characteristics in AMB 1 section. Tightness inequality along fit length of compressor bush on electromotor shaft

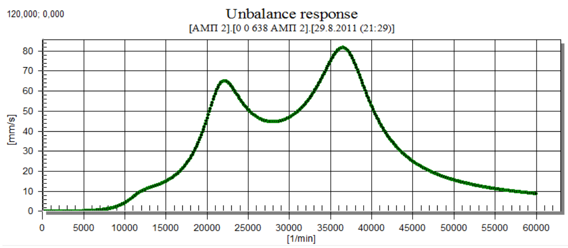


Fig. 15 Amplitude Frequency characteristics in AMB 2 section. Tightness inequality along the fit length of compressor bush on electromotor shaft

Vibrations amplitudes are slightly lower than in the previous case but their values exceed allowable values significantly.

Clearances influence

Clearances simulate full loss in tightness between details of the compressor construction. It means that the most sensitive to tightness loss links, which model them, may be found. The following variants of the full tightness loss are investigated.

Variant 1. The link 5 between the labyrinth disk and the impeller bush is deleted. In this case the impeller is supported only by the labyrinth disk. Here is an initial variant of fixing and tightness.

Variant 2. The link 4 connects the compressor impeller and the impeller bush. The link 5 connects the impeller bush and the labyrinth disk. The additional fit of the impeller on the bush is simulated.

Variant 3. The impeller is connected with the impeller bush by the link 4, the link 5 is deleted. Weakening of the radial tightness between the labyrinth disk and the impeller bush is modeled.

Variant 4. The link 4 is deleted. The impeller is fixed in cantilever. Weakening of the axial tightness is simulated between the impeller and the labyrinth disk.

Table 4 shows spectra of critical speeds at different variants of link. Frequency of the first bending mode shape is highlighted in red.

Table 4

Initial variant	Var.1	Var.2	Var.3	Var.4
11590.0	11590.0	11589.8	11565.7	11509.1
11595.9	11595.8	11595.6	11575.9	11524.2
22195.6	22194.8	22192.3	21162.0	12062.8
23652.0	23650.9	23647.8	23309.8	16515.1
36058.0	36055.9	36045.9	28575.7	21123.3
53578.8	53547.3	53455.2	46722.8	32693.1
80332.3	80286.4	80300.8	49994.2	34142.5
-	-	-	-	75819.1

In initial variant the impeller has tightness at the left centering spigot. Variants analysis shows that inclusion of the impeller additional fitting in construction at the right part makes the system more stable at breach of the other conditions of assembling and may be recommended to add to the compressor construction.

Transient rotor analysis

The aim of the present research is to model the rotor drop on the auxillary bearings and rundown of the compressor rotor to the moment of its stop.

In the previous investigations the authors proved that rolling bearings are the best ones as auxillary bearings [2]. They preserve the compressor from possibility for the rotor to start the backward precession. This is the reason for their use in the investigated compressor design. At the same time the other problem relating to the rotor rundown arises. Clearances in rolling bearings may lower the resonance regime corresponding to the first bending mode shape; its passage may be accompanied by high vibrations.

To solve the task three new non-linear elements are added into the rotor model for every support unit. They are active magnetic bearing [2], clearance and rolling bearing [3] and also elastic link to calculate frequencies basis and natural mode shapes of the rotor], Fig. 16.

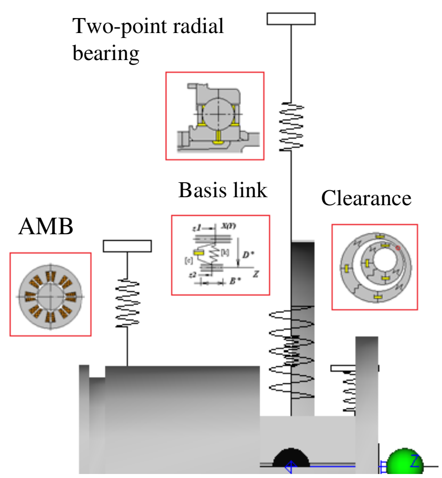


Fig. 16 Modeling of support unit

Four variants of the rotor system of the compressor supported by AMB are considered. AMB parameters (proportional and differential factors of PID-regulator) for all variants are constant. Unbalance of the compressor impeller is 1 g*cm. The rotor drop happens at time t=0. The results are given at the section of the AMB 1 auxillary bearing.

Variant 1. The rotor is supported by linear rolling bearings. The inner bearing race does not rotate. Sliding friction is equal to zero μ=0. Calculation results are presented in Fig.17 (amplitude-time characteristic) and Fig.18 (orbits of the rotor movement).

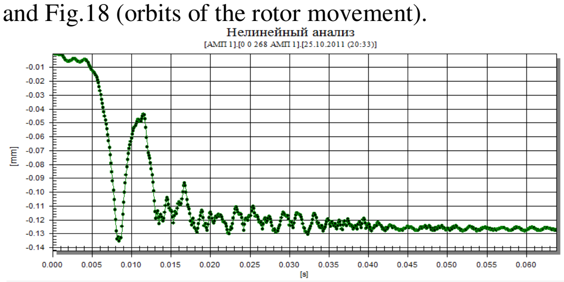


Fig.17 Amplitude Time characteristic. Variant 1

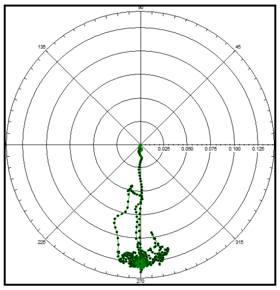


Fig.18. Orbits of movement. Variant 1

The rotor drops on the inner bearing race and then rundown takes place up to the full stop. The rotor has stable characteristics.

Variant 2. The rotor is supported by the linear rolling bearings. The inner bearing race does not rotate Sliding friction is μ=0.16. Fig. 19 and Fig.20 give the calculation results.

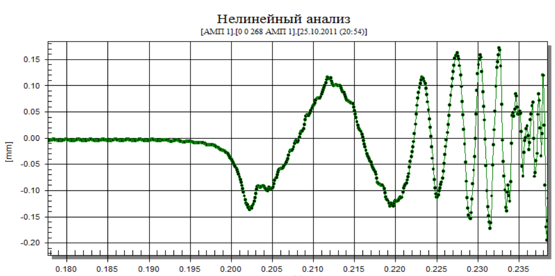


Fig. 19 Amplitude Time characteristic. Variant 2

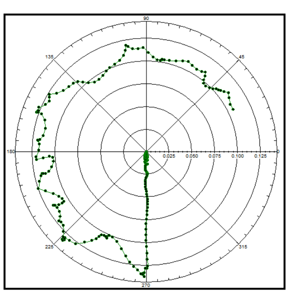


Fig. 20 Orbits of movement. Variant 2

Inner race works as a dry bearing. After rotor drop its backward precession (clockwise movement) starts. Such operating mode is not allowed.

Variant 3. The rotor is supported by linear rolling bearings. The inner bearing ring rotates in the whole range of the rotor rundown with its speed. Sliding friction is μ=0.1. Fig. 21 and Fig.22 give the calculation results.

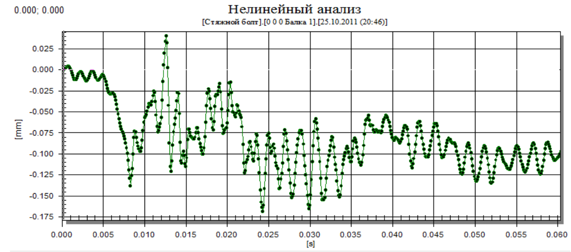


Fig. 21 Amplitude Time characteristic. Variant 3

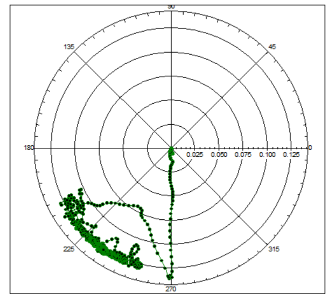


Fig. 22 Movement orbits. Variant 3

The rotor drops on the bearing ring without loss in stability in the whole rundown range.

Conclusion

The influence of tightness on the compressor characteristics is presented. Reasons for additional centering spigot of the rotor with tightness at it are given.

It is shown that jam of the auxillary bearings is not allowed: it leads to backward precession of the rotor.

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