DYNAMICS OF THE ROTOR SYSTEM SUPPORTED BY THE ACTIVE MAGNETIC BEARINGS

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Abstract
The article surveys the problems related to the use of the active magnetic bearings in rapidly rotated rotors of turbosets. The results of the dynamic analysis of the rotor supported by the active magnetic bearings are given. The possibility to control the rotor system changing the magnetic bearings stiffness and damping is considered. The behaviour of the rotor system during the rotor drop on the auxiliary bearings and the following runout are investigated.

Key words: turbosets, active magnetic bearings, rotor dynamics, unsteady and non-linear rotor systems, clearance, whirling motion, backward precession

Introduction

The Active magnetic bearings (AMB) is the technology that has been well-known since the 1930-s. It has been widely used in space industry [1]. Furthermore, during latest decades, with the microprocessor technology and electronics development its application in different turbosets has become possible.

The AMB work is based on the principle of an active magnetic suspension of a ferromagnetic body, Fig.1Fig. 1. The body remains stable in the prearranged position by the forces of magnetic attraction affecting the body from the part of steering magnets. Currents are moved into magnets windings by the means of an automatic control system consisting of displacement pickups (sensors), an electronic controller and power amplifiers.

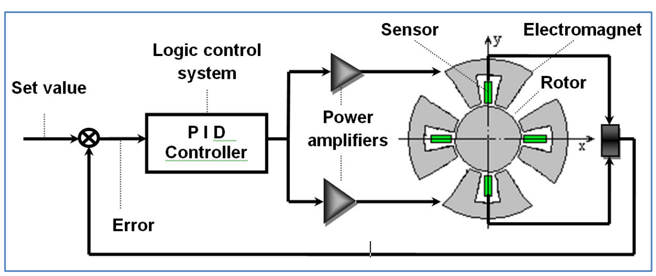


Fig. 2 Scheme of AMB operation

A proportional-integral-derivative controller (PID controller) became widely used as the basis of control system. About 90-95% of turbosets controllers [2, 3] being in service nowadays use PID-algorithms. The reasons for such popularity are simplicity of its simulation and industrial use, clearness of operation, possibility to solve a majority of practical problems and low cost.

A PID controller is a mechanism in a feedback loop that is used in automatic control systems for a control signal creation. It includes 3 independent loops: the proportional, the integral and the derivative one. The control equation in current terms is given by:

$$I(t) = K_p e(t) + K_i \int_0^t e(t)\,dt + K_d \frac{de}{dt} \tag{1}$$

where: $I(t)$ – change in current intensity; K_p, K_i, K_d – the proportional, integral and derivative terms; $e(t)$- error.

Changing the values of the controller terms the necessary parameters of the magnetic support can be obtained. FigFig.3 Impact of the control factors on the dynamic system when the static force appears

shows the dynamic reactions of the one-mass system with one degree of freedom, influenced by abrupt static load at different control factors.

$K_i = const; K_d = const;$ $K_p = var;$	$K_p = const; K_d = const;$ $K_i = var;$	$K_p = const; K_i = const;$ $K_d = var;$

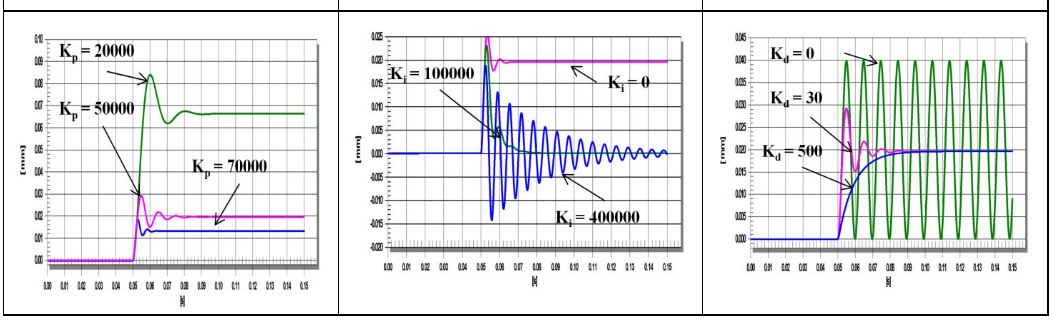


Fig.3 Impact of the control factors on the dynamic system when the static force appears

Many works are devoted to investigations and development of active magnetic bearings for various turbosets: from tiny turbomolecular pumps to big million-kilowatt turbogenerators [4].

Designing the rotor system mounted on AMB a series of problems related to rotor dynamics should be solved. They include:

- the rotor alignment in AMB clearance;

- passing through the resonance regimes;

- limitation of oscillations amplitudes of the rotor system ;

- the rotor system's going into running-in or backward precession.

The special feature of constructions supported by AMB is the necessity to mount auxiliary bearings because of the possible rotor's drop on account of the magnetic supports failure. The failure can happen as result of the power failure, the feeder cable breakdown, etc. So the following questions should be answered: if bearings withstand the loads, and how the rotor supported by auxiliary bearings will behave on impact and the following runout.

Auxiliary bearings endure a few emergency stops, after that they should be replaced. The other problem concerns industrial design of auxiliary bearings. For instance, if journal bearings are used as auxiliary ones, the rotor system going into whirling motion and backward precession at worst. This is related to appearance of high friction when the contact between the rotor and the auxiliary bearing takes place. The rotor's whirling motion may be also accompanied by the high resonance loads and the displacements.

All these problems can be worked out using specially for it created software packages and simulating various emergencies at the stage of designing. One of such software packages is Dynamics R4 [5] (Fig. 4 The main window of the Dynamics R4 software with the investigated rotor supported by AMB

), which is used by a significant quantity of Russian and foreign companies producing turbosets for different purposes. The developer of the software is the Russian consulting and engineering centre Alfa-Transit Co.,Ltd. (www.alfatran.com).

The software is used for solving various practical tasks of the multiple shaft unsteady linear and non-linear rotor dynamics. The subjects of inquiry are the power and electric plants, the gas-turbine engines, the turbine pumps, the accessory gearboxes with different types of gearing, etc. This software is used at different stages of the construction's designing, development and service. Besides, its developers do research in the direction of "vibration diagnostics through simulation".

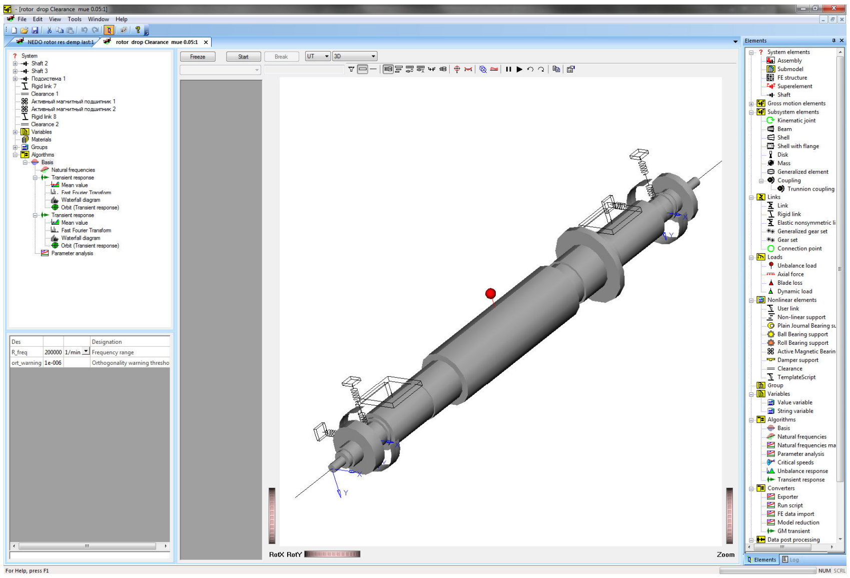


Fig. 4 The main window of the Dynamics R4 software with the investigated rotor supported by AMB

Method of non-linear rotor systems analysis

Any rotor supported by magnetic bearings represents a considerably non-linear dynamic system including a significant quantity of the constructive elements with the non-linear characteristics. That is magnetic bearings themselves, journal bearings or rolling bearings and clearances between them. Moreover, the rotor can be influenced by unstable loading appearing during its operation.

The most general method of dynamic analysis of the non-linear unstable rotor systems' is integration of the coupled equations of motion [6,7]. Dynamic behaviour of the system is counted for a number of sequential time ranges as for the linear steady dynamic system with dynamic characteristics set at the beginning of the range under consideration. At the end of every range the characteristics change according to the present strained state. The system's loading can also change. Thus, non-linear unsteady analysis is considered as the sequence of the uninterruptedly changing linear stable systems calculation.

New dynamic characteristics are obtained at the beginning of every integration step with the help of the nonlinear elements models. Input parameters of these models are the displacement and the velocity of the points through which the non-linear elements are connected to the rest of the dynamic system; output parameter is the dynamic reaction. So, non-linear effects are taken into account in the right side of movement equations.

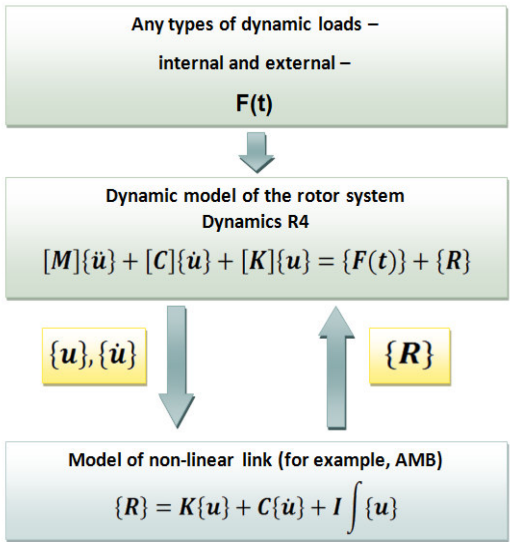


Fig.4 Scheme of interaction between the rotor system's dynamic model and non-linear elements

Отформатировано: Название объекта, По центру

Ошибка! Источник ссылки не найден. Fig.4 shows the general scheme of an interaction between the linear model of the rotor system and the models describing its non-linear elements such as journal bearings, hydrodynamic dampers, magnetic supports, etc. **Ошибка! Источник ссылки не найден.**

It is obvious that this process can also be carried out for the linear system if stiffness and damping are assumed to be constant.

Simulation and linear analysis of behaviour of the rotor supported by AMB

As an example for investigation of the rotor's dynamic behaviour the rotor from the international standard on magnetic bearings ISO 14839-3:2006(E) [8] is used. The rotor is supported by two radial AMB.

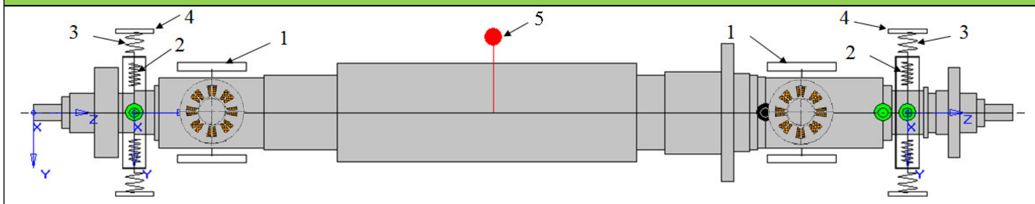
The Dynamics R4 software system has a significant quantity of special elements by the means of which the simulation of various unsteady and non-linear rotor systems can be carried out. In the example the following non-linear elements are used:

- The spring link set by stiffness and damping coefficients;
- The deflection limiter (clearance) having two elastic-damper packets set by stiffness and damping coefficients, friction coefficients in contact, sliding velocity of rolling elements, etc.;
- The active magnetic support with its geometry and control system;
- The rolling bearings set by their geometry, a number of their rolling elements, clearances in a bearing, contact stiffness;

- Different types of the journal bearings, etc.

Fig.5 shows the scheme of the rotor investigated and its parameters.

Scheme of the rotor and its main parameters



Left support	Rotor mass, kg	23.76	Right support
	Rotor length, mm	851	
	Diametral moment of inertia, kg m^2	0.92	
	Operating rotation, Hz	500	
	Rotor unbalance, g cm	15	
	Nominal stiffness, N/m	0.5e +8	
	Nominal damping, N s/m	100	
	Radial clearance, mm	0.5	
	Contact stiffness, N/m	1e+10	
	Auxiliary clearance, mm	0.25	
	Stiffness, N/m	1e+8	
	Damping, N*s/m	100	

Fig. 6 Scheme of the rotor and its main parameters

The elastic-inertial rotor model includes two AMB (1). In two auxiliary supports two elements – the clearance (2), the ball bearing (3) connected with pedestal (4) – are used for simulation. If the auxiliary bearing is the journal one the bearing is excluded from the rotor model.

Linear calculations of the rotor supported by the bearings with nominal stiffness and damping coefficients show that in the investigated range of the rotor's rotating speeds up to 50000 rpm there are three critical speeds, Fig. 7. On the natural frequencies map the points of critical speeds are highlighted and the corresponding mode shapes are shown. It means that going into the operating range (from 25000 up to 30000 rpm) the rotor meets two resonances which are desirable to pass without high vibration loads and displacements.

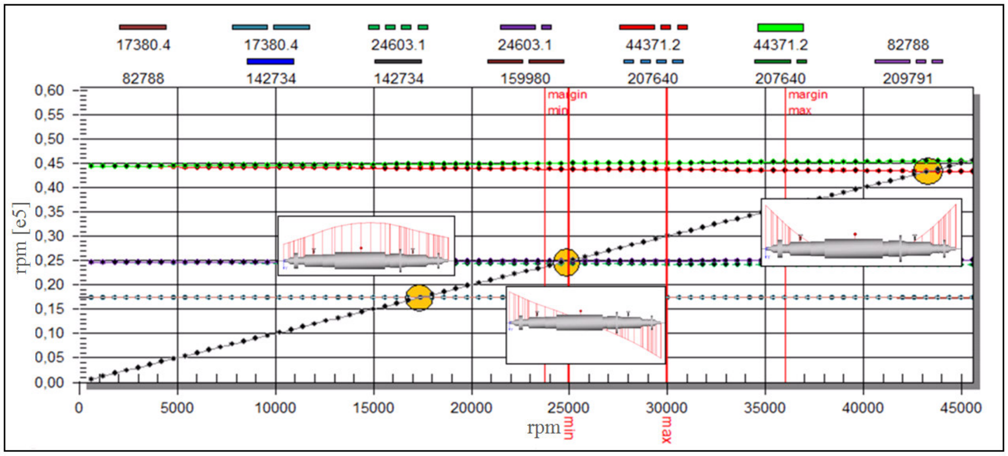


Fig. 7 Natural frequencies map

The possibility to influence the vibration amplitude by the support's damping characteristics may be estimated by the potential energy distribution between the shaft and the rotor supports, Table 1. As the table shows, the energy of two first mode shapes are concentrated in the supports. It means that changing their stiffness and damping we can influence the rotor dynamic characteristics. The rotor at the third shape virtually doesn't displace in supports, so it seems not possible to influence this shape efficiently.

Table 1

Unit	Critical rotating speeds, rpm		
	17373	24637	45303
	Potential energy, %		
Rotor shaft	15.506	3.843	87.558
AMB_1	48.745	43.710	3.602
AMB_2	35.749	52.447	8.839

Fig**Ошибка! Источник ссылки не найден.** shows the rotor length amplitude-frequency response. As the 3D diagram displays the main resonance regime is related to the first mode shape which can lead to the significant rotor displacements with the stator's contact. The second mode shape is not virtually excited by the radial unbalanced force. It is related to the fact that it is close to the crossing point between the precession line and the central axis.

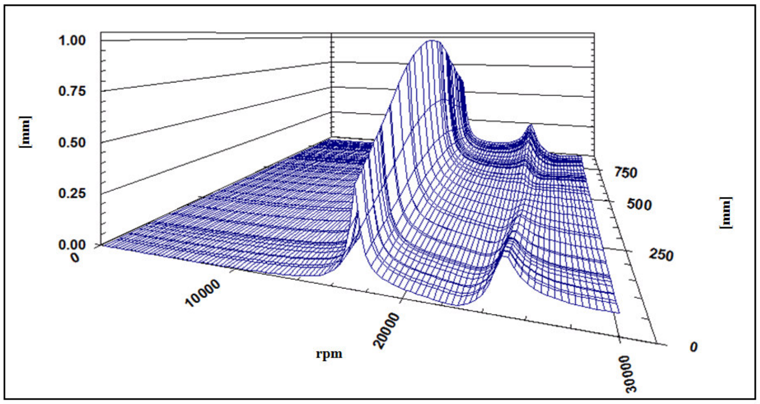


Fig.7 The rotor's amplitude-frequency response

As follows from the diagram for the investigated rotor the accepted values of stiffness and damping do not secure the normal rotor operation.

AMB stiffness and damping control

Taking into account the value of the auxiliary clearance it is essential that peak amplitudes should not exceed 0.25 mm when the rotor reaches the effective speed. The task may be solved by changing stiffness or damping of the magnetic supports, applying the corresponding principle of the stiffness and damping control for passing the resonances.

Stiffness and damping of the magnetic support assign controller's coefficients K_p and K_d correspondingly. The equations for the central rotor's position in the direction of one of the control axes (the magnetic pole) allow to estimate their influence:

$$K = \frac{\mu_0 N^2 A_g I_b}{\delta_0^2}\left(\frac{I_b}{\delta_0} + K_p\right); \qquad (2)$$

$$C = \frac{\mu_0 N^2 A_g I_b}{\delta_0^3} K_d\,, \qquad (3)$$

where: μ_0- vacuum permeability; N – number of circuits; A_g - area of electromagnet poles; I_b - displacement current; δ_0 – clearance between electromagnet stator and the rotor.

If the resonance characteristics are known it is possible to set the law of stiffness or damping variation in order to change the dynamic system for the amplitudes decrease. Fig.Fig. shows the transient response obtained for the rotor while its going into the operating mode at the left bearing section. Fig.Fig. 8 represents the chosen law of the damping control. It was taken into account that the calculated critical damping of the rotor system accounts for ~87000 Nsec/m. If damping in the supports exceeds the critical one then the rotor system will stop being oscillating.The nominal damping in AMB at resonance passing has been increased up to 12000 Nsec/m.

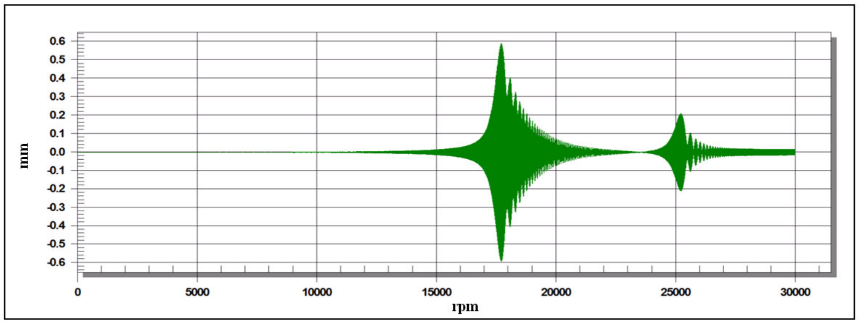


Fig. 8 Amplitude-frequency response at unsteady analysis, left AMB section

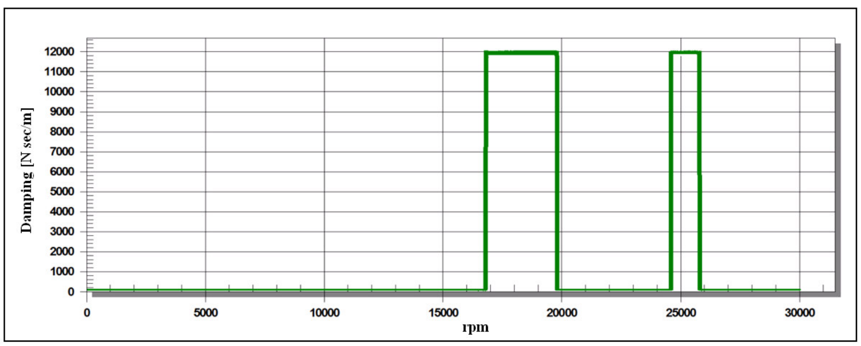


Fig. 8 The law of damping change

Fig.Fig. 9 depicts the results of going into the regime with the chosen program of the damping control. At the chosen damping the main level of vibration signal (blue colour) does not exceed 0.1 mm. However, at the moment of current strength change in AMB (the rotor's rotating speed is 16800 rpm) a certain increase in vibration amplitudes up to 0.12mm takes place.

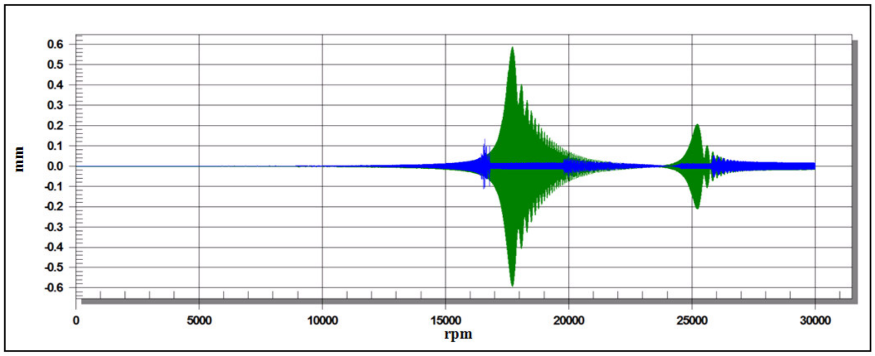


Fig. 9 Results of achieving the regime with the chosen program of damping control

Similar result can be achieved by the change in the magnetic supports stiffness,fig. 10 и fig. 11.

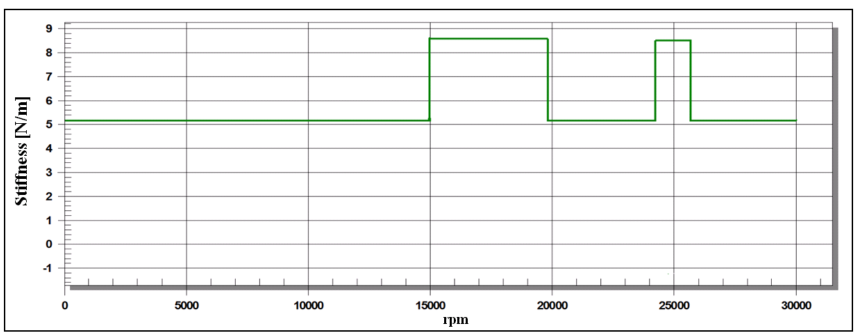


Fig. 10 The law of the stiffness change

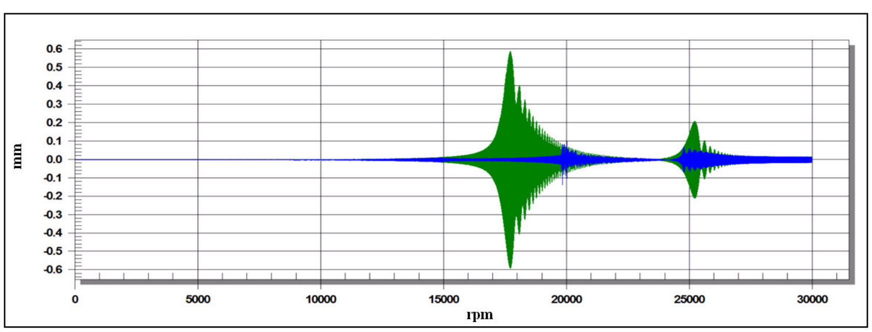
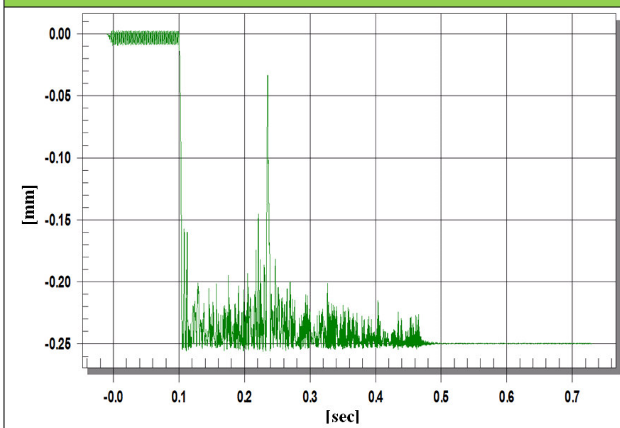


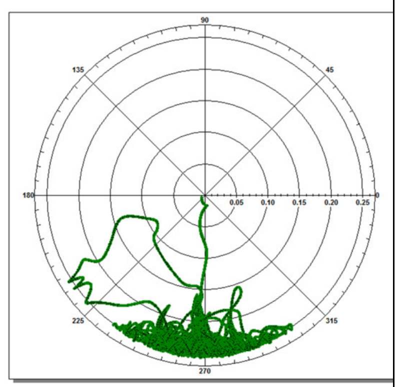
Fig. 11 Results of going into the regime with the chosen program of the stiffness control

The rotor drop on the journal bearings

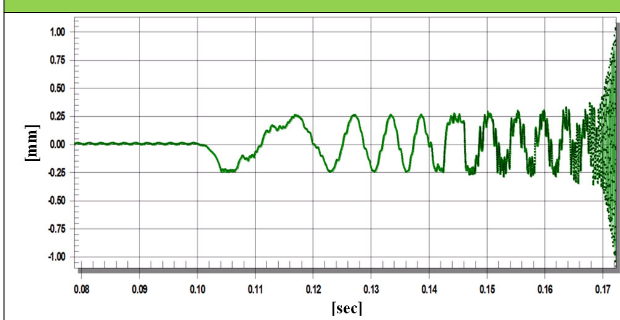
The use of the journal bearings as the auxiliary ones is possible only if they have low enough friction coefficient. Rather high friction coefficient may lead to the backward precession appearance. This motion results in the rapid rotor breakdown, Fig.Fig. 12.

Steel on babbit, friction coefficient μ=0.005. Smooth characteristics





Steel on graphite, friction coefficient μ = 0.1. Going into backward precession



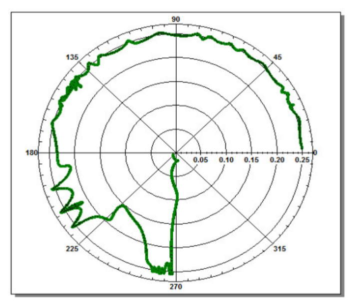


Fig. 12 The rotor drop on the journal bearing. Left bearing

The investigated rotor is centered by the magnetic supports in the time range from 0 to 0.1 sec. The rotating speed is 30000 rpm. The clearance in the magnetic supports is 0.25 mm. The magnetic supports are switched off at t = 0.1 сек.

At the orbits plots direction of the rotor rotation is the counterclockwise ones.

The rotor drop on the rolling bearings

More often the rolling bearings are used as auxiliary supports. Dropping on such support the rotor begins to rotate the bearing inner race because of the friction force . In this case the dangerous moment is when the rotor touches the fixed inner race. At this moment a significant friction force appears that can lead to the rotor's going into backward precession. So the spring element is set between the rolling bearing and the stator. This allows the bearing inner race to start spinning fast without high friction appearance [4].

Fig.14 shows the unsteady process of the rotor drop from the centered position on the rolling bearings at time t = 0.5 sec.

Left ball bearing

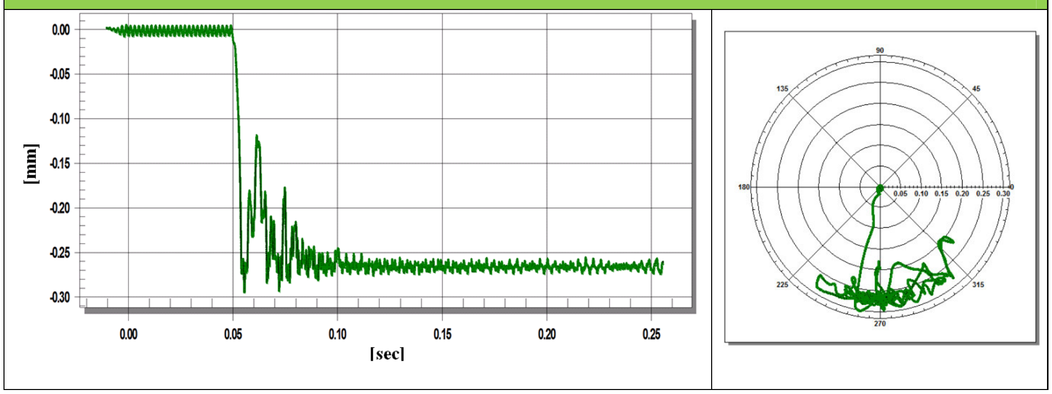


Fig. 13 The rotor drop on the rolling bearings

The rotor's runout after its drop on the rolling bearings

After the rotor's drop on the auxiliary bearings it should be stopped if it is not resume work rapidly. This may lead to the effect of the auxiliary supports' whirling motion by the rotor, Fig.15

Fig.. The whirling motion is accompanied by clearance adjustment, rolling bearings defects and possible damage of the shaft journals details.

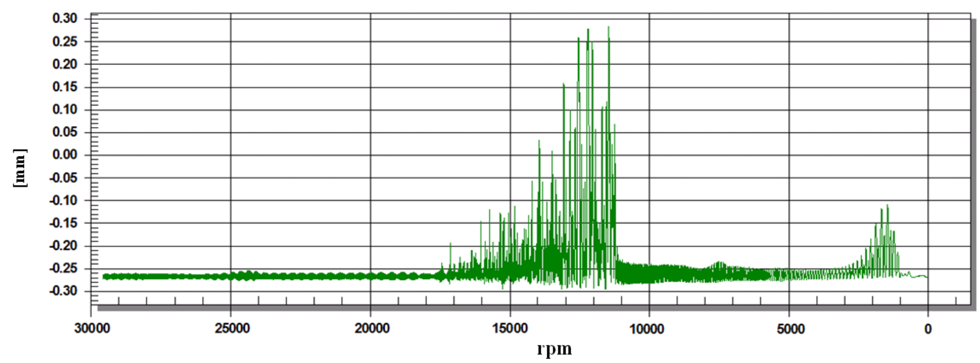
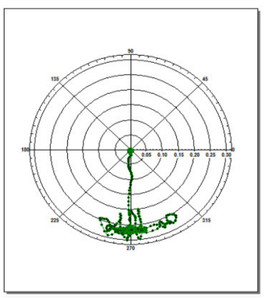
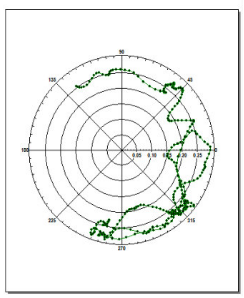
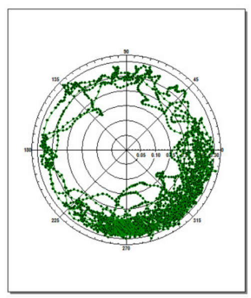
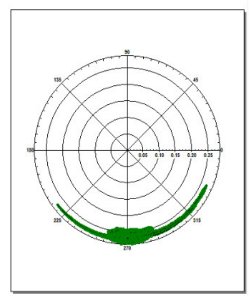
The rotor’s stop. Resonance regime at about 17000 rpm with going into whirling motion			
			
Movement before whirl 30000…17000	Going into whirling motion 17000…14000	Movement along the clearance 14000…11000	Close 11000…3000
			

Fig.15 The rotor’s runout and whirl effect

Conclusion

As a result of the works carried out the methodology of simulation and analysis of the rotor systems supported by AMB was developed. It can be used by the organizations designing turbosets or the ones producing AMB for concrete objects, Table 2.

Table 2

Determination of necessary values of AMB stiffness and damping
Estimation of AMB control coefficients and adjustment of the control system
Determination of the rotor's damped critical speeds and mode shapes – places of prospective resonances
Calculation of forced oscillations and amplitude values of vibroparameters : displacements, vibrovelocity, reactions in supporting units…
Development of the program of current – AMB stiffness and damping – control for passing systems resonances and going into the operating range
Unsteady analysis of the rotor supported by AMB with the aim of more precise definition of the rotor's characteristics at transient regimes
Simulation of the rotor supported by backup bearings. The non-linear unsteady rotor analysis. The aim is to determine the possibility of the rotor's going into whirling motion or backward precession
The unsteady rotor analysis after switching off AMB supported by non-linear rolling bearings while the rotor's deceleration

Using it the developers of the rotors with AMB should pay attention to both possibility of passing resonance regimes with the aim of decreasing vibration amplitudes and the supports design in order to exclude the possibility of the rotor's going into whirling motion or backward precession.

The possibility of accurate simulation and analysis of the proposed rotors design may be received with the use of special bundled software intended for the nonlinear and the non-stationary rotor dynamics analysis.

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