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GAS TURBINE RESPONSE ON SEISMIC LOAD

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Abstract

Investigation of stationary gas turbine power unit behavior at seismic load is performed. Methodology for developing of gas turbine power unit system “rotor – fluid film bearings – casing – foundation frame” reduced finite-element (FE) models is presented. Nonlinear stiffness and damping characteristics of lubrication layer in fluid film tilting-pad and multilobe bearings are taken into account in presented models. Calculation of bearing stiffness and damping characteristics involves coupled solution of problem of incompressible lubrication flow in gap between shaft journal and pad surfaces and problem of each pad equilibrium position definition with elastic deformations of sliding surfaces taken into account. Natural frequencies and modes of vibration for gas turbine power unit are determined for two cases: with and without gyroscopic moments taken into account. Verification between different reduced FE models of rotor is carried out. Investigation of gas turbine power unit with rotating rotor dynamic response at seismic load is carried out by direct integration using Newmark scheme. Rotor orbits in bearings and disks orbits are calculated both at operational loads and at seismic load applied. Calculated orbits full spectrum analysis is performed. Conclusion regarding to fluid film bearings working capacity at seismic load is performed. Rotor and casing parts stress-strain state is calculated. During analysis of gas turbine power unit response on seismic event with amplitude of 9 by the MSK-64 scale the evaluation of static and dynamic reactions in rotor supports and casing mounting points is performed. Rotor orbits and casing parts deformations at seismic load in comparison with similar results for stationary operation conditions allows to evaluate gaps changing in gasdynamic path and in bearings and make a conclusion regarding to working capacity of gas turbine power unit at seismic load. The evaluation of lubrication layer stiffness in fluid film bearings in rotor supports influence on gaps in gasdynamic path at seismic load is carried out. Shown that gyroscopic moments of rotating rotor changing gas turbine natural frequencies and leads to sufficient rotor and casing reciprocal displacements that changing values of gaps in gasdynamic path at seismic load.

Keywords: gas turbine, rotordynamics, seismic load

1. Introduction

It is a common practice of stationary gas turbine unit usage as secondary or alternative power generation or in gas compressor units in seismic active regions. The main requirement in Russian Code [1] for gas turbine under the action of earthquake excitation is to guarantee of its working capacity at earthquake for more than one minute. On the base of this requirement the analysis of gas turbine load-bearing scheme should be carried out and alternating stresses acting in casings and bearing frames should be examined. The stress level (summary of operating and dynamic stresses) caused by earthquake should not exceed allowable in critical structural elements that identifying gas turbine robustness and durability. Correspondingly deformations and displacements of gas turbine parts should not exceed values specified in operational manual. Rotor and casing have relative displacements at dynamic load. These relative displacements could lead to rotor blade tips to casing contact and shaft journal to bearing race contact in journal bearings and further unit breakdown. Analysis of relative displacements in gasdynamic path and in rotor supports are one of most important problems for evaluation of gas turbine unit behavior at seismic load.

Gas turbine behavior investigation using special experimental equipment at load similar to seismic is possible in limited number of cases and for structures with limited inertia characteristics. Numerical investigation of gas turbine unit response at seismic load and evaluation of its seismic capacity should be performed at design stage. Mathematical simulation is also required when experimental equipment is unavailable or it is not possible to carry out the experiment. Russian Codes are missing the special requirements for conditions that should be taken into account at gas turbine unit dynamics investigation while simulating seismic response. Therefore, the structural characteristics and operating conditions of gas turbine should be taken into account for gas turbine unit seismic capacity investigation. Major input in stationary gas turbine unit dynamics contributes by rotating rotors that have big inertia and gyroscopic moments and by nonlinear hydrodynamic journal bearings.

2. Stationary gas turbine mathematical model

Stationary gas turbine with flexible power turbine rotor supported in hydrodynamic pad bearings (Fig. 1) is investigated in paper. Presented rotor models allows to estimate a total gas turbine response at seismic load with rotor gyroscopic moments and nonlinear stiffness and damping in journal bearings taking into account.

FE models of gas turbine unit including models of cases, rotors and frames are used in investigation. Level of detail for gas turbine mathematical model depends upon required calculation accuracy. Therefore, structural elements with natural frequencies sufficiently greater than frequencies of earthquake excitation can be simulated as rigid bodies with their masses and inertia moments included in total finite element model.

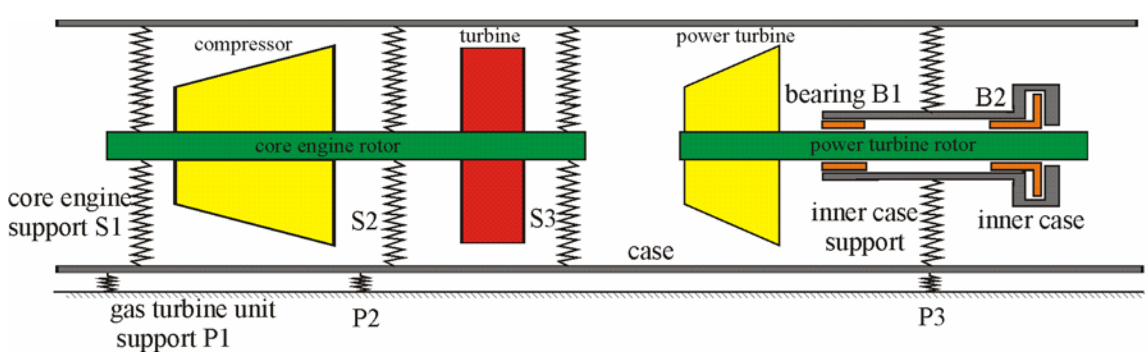


Fig. 1 – Gas turbine unit rotors model

Stationary gas turbine unit, which model presented in Fig. 1, consist of core engine and power turbine. Core engine rotor supported in rolling element bearings. Power turbine rotor supported in hydrodynamics pad bearings. For current investigation the two finite element models were developed: shell-beam model and beam model. Models detailed description presented in [2-4]. In shell-beam model (Fig. 2) core engine casing parts, outlet and internal casing of power turbine parts are simulated by shell finite elements. Core engine and power turbine bearing frames simulated in both finite element models by beam finite elements. Shell model is used for calculation of structure stress-strain state under the action of gravity and rotating inertia forces and for determining of natural frequencies and modes without taking into account of rotors rotation and nonlinear stiffness of hydrodynamic bearings. Beam model is used for calculation of natural frequencies and investigation of gas turbine behavior at seismic load with rotor rotation and nonlinear stiffness of hydrodynamic bearings taking into account. Beam model is used instead of shell-beam model due to huge size of the shell-beam model and excessive calculation time.

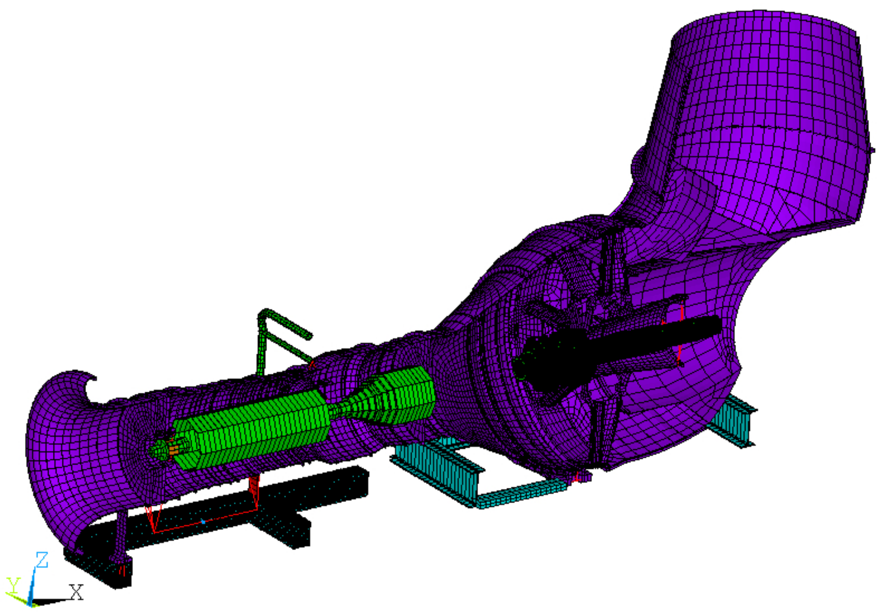


Fig. 2 – Stationary gas turbine unit shell-beam finite element model

For dynamics simulation the beam rotor model is used. This model verificated relatively to equivalent solid model in one of common commercial finite element analysis programs and taking into account experimental data if any. Beam model comparison to solid model is carried out for natural frequencies and modes in range 0-30 Hz [2, 5]. It is shown that beam model with less degrees of freedom one can use for calculation of gas turbine response at seismic load.

Table 1 shows the natural frequencies for further analysis that have been calculated using two models. For calculation of nonrotating rotors both models natural frequencies are close to one another. That confirms the correctness of simplifications introduced in developing of the beam model.

Table 1 – GTU natural frequencies, Hz (with gravity forces taken into account)

<table>
<tr><th rowspan="2">Natural frequency</th><th colspan="2">Shell-beam finite element model</th><th colspan="2">Beam finite element model</th></tr>
<tr><th>without gyroscopic moments</th><th>with gyroscopic moments</th><th>without gyroscopic moments</th><th>with gyroscopic moments</th></tr>
<tr><td>1</td><td>15.78</td><td>15.75</td><td>15.69</td><td>15.67</td></tr>
<tr><td rowspan="2">2</td><td rowspan="2">19.88</td><td>17.78(backward precession)</td><td rowspan="2">19.99</td><td>17.72</td></tr>
<tr><td>22.55 (forward precession)</td><td>21.62</td></tr>
</table>

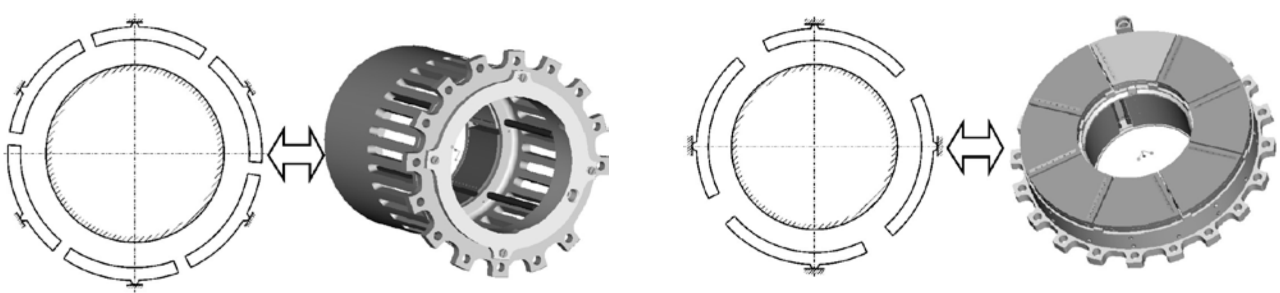
Analysis of seismic load influence at rotor-bearings system dynamics is performed by direct integration of rotor motion equations with gyroscopic moments and nonlinear journal bearings taking into account. Direct integration is carried out using Newmark scheme with iterative refinement of stiffness and damping matrices at each step. The harmonically changing displacements are applied as excitation at bearing frames supporting nodes. Rotor model is described by finite element equations in dead coordinate system as follows:

$$[M]\{\ddot{U}\}+[C(\omega)]\{\dot{U}\}+([K_R]+[K_S])\{U\}=\{F_0\}+\omega^2[M_1]\{E_0\}, \tag{1}$$

where $[M]$ is a matrix of rotor rotating parts inertia characteristics; $[C]$ is a matrix that taking into account of rotor parts gyroscopic moments, structural damping and damping into supports; $[M_1]$ is a part of matrix $[M]$ corresponded to rotor nodes linear displacements; $\{U\}$, $\{\dot{U}\}$ and $\{\ddot{U}\}$ are vectors of rotor displacements, velocities and accelerations correspondingly; $[K_R]$ is a rotor stiffness matrix; $[K_S]$ is a stiffness matrix that taking into account influence of supports and seals; $\{F_0\}$ is a vector of external forces acting at rotor; $\{E_0\}$ is a vector of initial imbalances of rotor cross-sections that are defined by initial imbalances $\{\varepsilon_0\}$ in each rotor cross-section. The special finite element, which characteristics determined using nonlinear model of support with hydrodynamic bearing, is used for taking into account of bearing stiffness characteristics in rotor model.

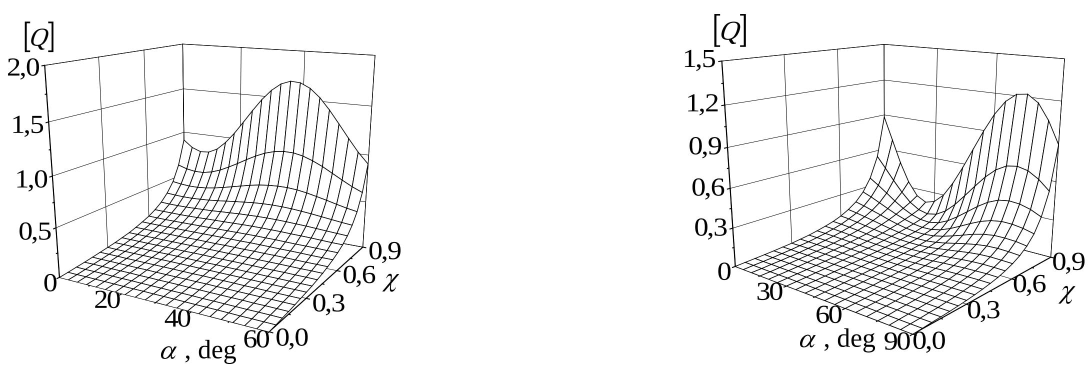
3. Support with hydrodynamic pad bearing model

One of the steps for creation of rotor mathematical model is a problem of hydrodynamic pad bearing mathematical model development and its integration in total gas turbine unit finite element model. Mathematical models for two pad bearings: with six pads and elastic support element like a squirrel wheel (Fig. 3a) in rotor support B1 and with four pads combined with thrust bearing (Fig. 3b) it rotor support B2. For calculation of gas turbine rotor supports nonlinear characteristics the support with hydrodynamic bearing mathematical model is applied [6]. This model taking into account stiffness and damping characteristics of oil layer, bearing parts and support casing.



a) bearing B1 b) bearing B2

Fig. 3 – Hydrodynamic pad bearings



a) 6-pad bearing b) multilobe bearing

Fig. 4 – Bearing carrying force versus α and χ

Lubrication characteristics in journal bearing are calculated by Reynolds equation [6]. In case of pad bearing the equilibrium position for all pads in bearing were determined for each shaft journal displacement in bearing and contribution of each pad in lubrication gap is calculated. Equilibrium position for each pad is calculated from pad equilibrium equation of moment from lubrication pressure relatively to pad pivot and friction moment in pad pivot. Calculation algorithm for pad bearing, that taking into account both pads angular displacements and pads deformations, is presented in [3-6]. Resulting system of equations allows to determine pressure distribution in bearing for arbitrary changing lubrication layer thickness. This allows to calculate stiffness characteristics of lubrication layer, hydrodynamic forces and moments in bearing depending upon shaft linear and angular displacements. Bearings parameters and detailed results for stiffness characteristics are presented in [2-4]. Results for 6-pad bearing and multilobe bearing with six lobes are presented in Fig. 4.

Casing stiffness characteristics are determined in common commercial finite element program during calculation of casing response for unit forces and moments applied in support installation places. Depending upon casing geometrical characteristics the shell or solid elements are used for these calculations.

Support model incorporated in rotor dynamics Eq. (1) uses method of sequential including of elastic elements in total model. These elements simulating elastic properties of lubrication layer, bearing parts and support casing. Thus support reactions vector $\{R^u_{\sup}\}$ acting at rotor is associated with shaft displacement vector $\{U_j\}$ by following equation:

$$[K_{\sup}]\{U_j\} = \{R^u_{\sup}\}, \tag{2}$$

where $[K_{\sup}] = \left([K_b]^{-1} + [K_c]^{-1}\right)^{-1}$ is a special finite element matrix simulating support elastic properties and included in matrix $[K_S]$; $[K_b]$ is a lubrication layer stiffness matrix taking into account displacements and deformations of bearing details; $[K_c]$ is a casing stiffness matrix. Support model is taken into account in Eq. (1) by means of special finite element which coefficients are determined for current rotor position in bearing and its rotating frequency.

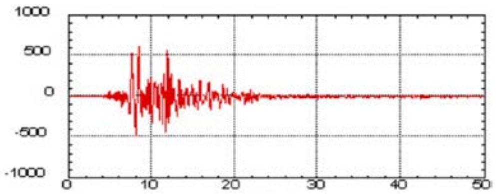
4. Seismic load response

Changing by harmonic law displacements were applied to bearing frames supporting nodes for simulating of seismic load. Typical acceleration amplitudes spectrograms of earth points depending from vibrational frequency are presented in Fig. 5.

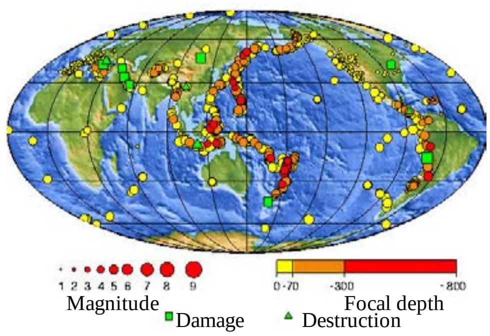
Displacements amplitudes corresponding to excitation frequency for structures height above the ground from 0 to 10 meters on the basis of accelerations amplitudes shown in GOST 30546.1-98 [1] are determined as follows:

$$A_u(f) = A_a(f)/(2\pi f)^2 , \tag{3}$$

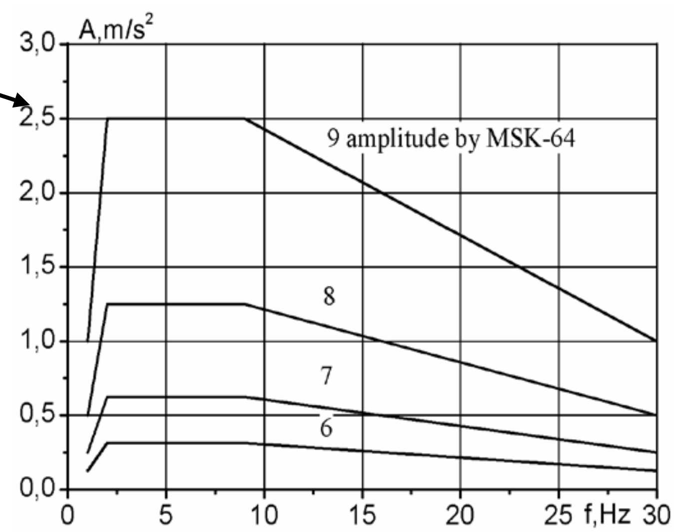
where A_u and A_a are amplitudes of displacements and accelerations correspondingly, f is an excitation frequency. Gas turbine unit foundation excitation applied in series in following directions: in vertical plane; in horizontal plane in gas turbine axial direction and in direction perpendicular to gas turbine axis. Displacements amplitudes for vertical excitation are applied with coefficient 0.7 in comparison with amplitudes in horizontal plane. Solution was performed for whole range of excitation frequencies region (0-30 Hz) with step of 1 Hz and separately for excitation frequencies equal to gas turbine natural frequencies if they exist in excitation frequency range. Structural damping was taking into account in the model. Dissipation energy is calculated in accordance with [1] in parts of structure elastic energy amplitude values of during vibrations. Calculated for case of vertical excitation the gas turbine parts displacements and deformations are smaller than similar for horizontal excitation because foundation excitation amplitudes were recalculated with decreasing coefficient of 0.7. Thus the results for gas turbine unit vertical excitation are not shown in presented results.



a) earthquake registered data



b) earthquake activity from 24.09.1998 till 15.11.1998

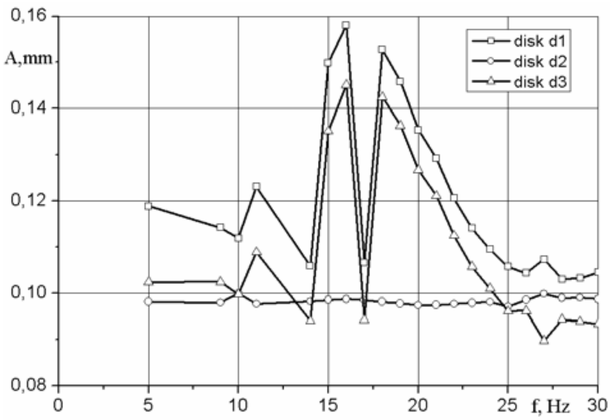


c) foundations accelerations for different earthquake amplitude by MSK-64 [1]

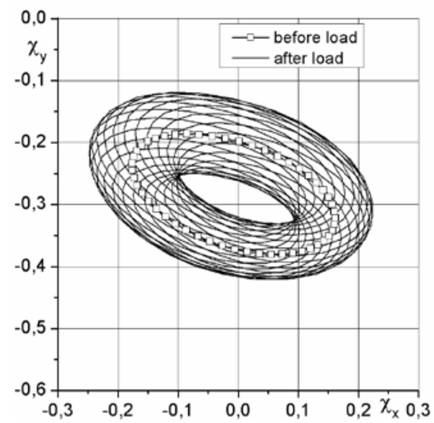
Fig. 5 – Earthquake data

Results for gas turbine unit with flexible rotor response at harmonic foundation excitation in lateral and axial directions in horizontal plane while gyroscopic moments taken into account presented in Fig. 6 for power turbine rotor supported in hydrodynamic pad bearings and in Fig. 7 for power turbine rotor supported in multilobe bearings with equal quantity of pads and lobes. Maximal displacements for each excitation type corresponds to gas turbine natural frequencies. Rotor orbits in disks and bearings cross-sections for whole range of harmonic excitation frequencies are obtained as results of mathematical simulation. In Fig. 6a and Fig. 7a the maximal relative change of gap in gasdynamic path at axial excitation for disks cross-sections (A, mm) are shown. Similar results for lateral excitation are presented in Fig. 6c and Fig. 7c. Rotor orbits in bearing before and after excitation in axial direction at gas turbine natural frequency (f = 18 Hz) are shown in Fig. 6b and Fig. 7b. Similar results for lateral excitation at gas turbine natural frequency (f = 16 Hz) are presented in Fig. 6d and Fig. 7d. Comparison of results for taking into account different effects in rotor model are presented in Fig. 8. This comparison demonstrates sufficient influence of rotor gyroscopic moments and nonlinear stiffness in supports with hydrodynamic bearings on gas turbine unit with flexible power turbine rotor response at seismic load. Gyroscopic moments providing high relative amplitudes of rotor and casing at harmonic excitation. Gas turbine unit natural frequency corresponded to disks angular displacements is divided for two frequencies of forward and backward precession in case of gyroscopic moments taking into account. As follows from obtained results the nonlinear hydrodynamic bearing stiffness provide greater amplitudes of displacements in comparison with rigid supports for gas turbine unit dynamic response at harmonic excitation (Fig. 8). Rotor supported in multilobe bearings have bigger values of cross-coupling stiffness coefficients in comparison with pad bearings and therefore demonstrates greater amplitudes of displacements at harmonic excitation. Full spectrum analysis demonstrates that rotor in pad bearings at axial excitation vibrates on excitation frequency and rotating frequency with big amplitude of forward precession and only small amplitude of backward precession at rotor speed. But for lateral excitation rotor in pad bearings have forward and backward vibrations on excitation frequency with almost same big amplitudes. Rotor in multilobe bearings have same picture of vibrations for both cases. However, for multilobe bearings the small amplitude vibrations on many other frequencies are exist in full spectrum.

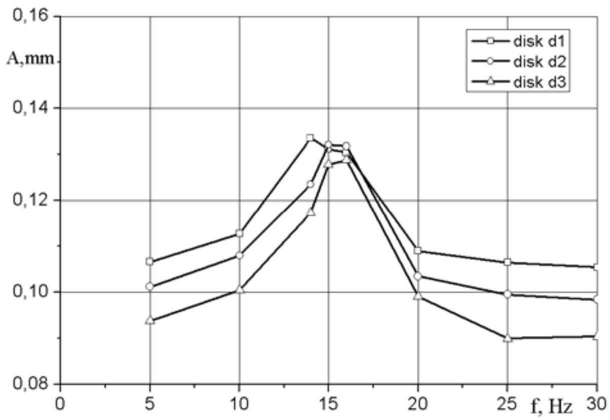
Using developed mathematical model of gas turbine unit, the maximal values of stresses in rotor, casing and bearing frames parts can be evaluated (Table 2).



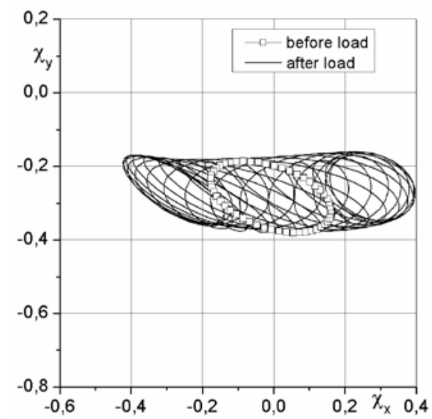
a) maximal relative change of gap in gasdynamic path at axial excitation



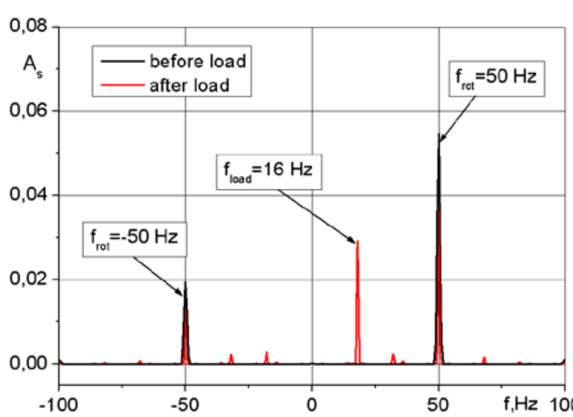
b) rotor orbits in bearing B1 for excitation frequency *f* = 18 Hz (axial excitation)



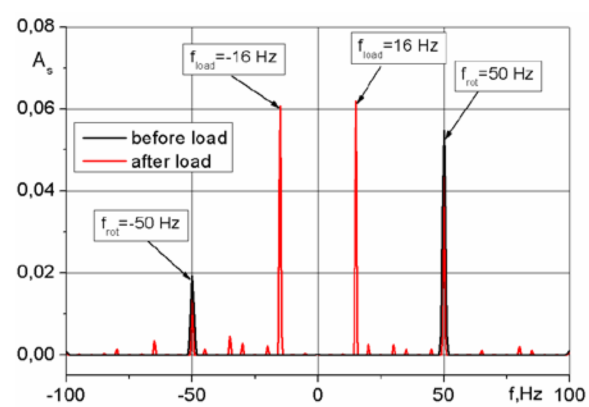
c) maximal relative change of gap in gasdynamic path at lateral excitation



d) rotor orbits in bearing B1 for excitation frequency *f* = 16 Hz (lateral excitation)

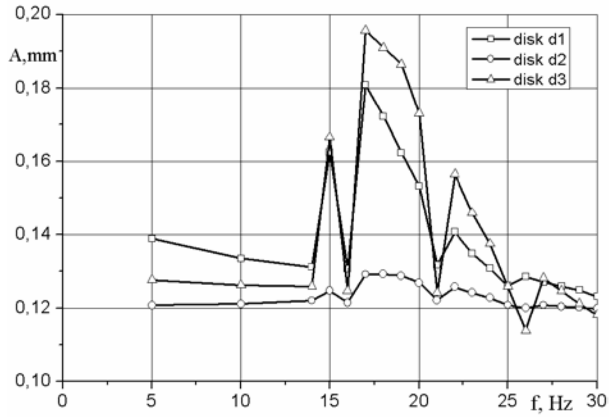


e) full spectrum of orbit in bearing B1 for excitation frequency *f* = 18 Hz (axial excitation)

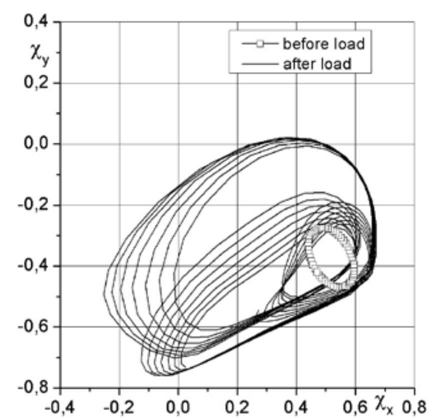


f) full spectrum of orbit in bearing B1 for excitation frequency *f* = 16 Hz (lateral excitation)

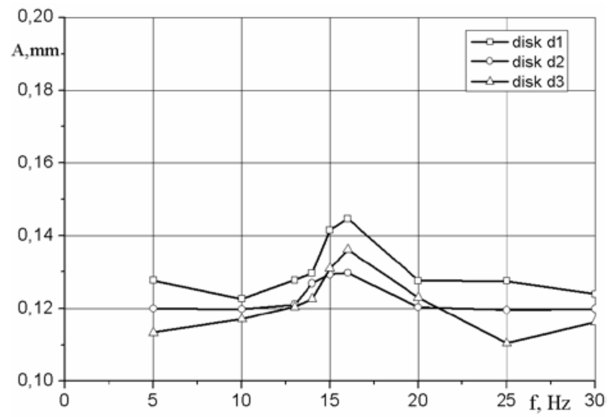
Fig. 6 – Gas turbine unit with power turbine flexible rotor in hydrodynamic pad bearings



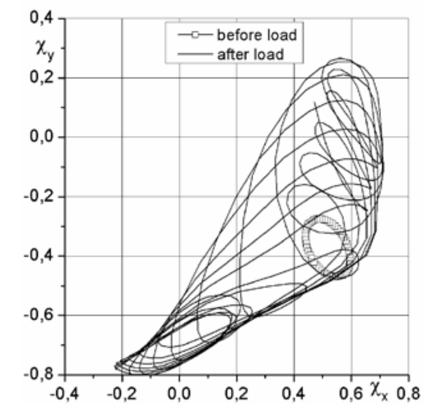
a) maximal relative change of gap in gasdynamic path at axial excitation



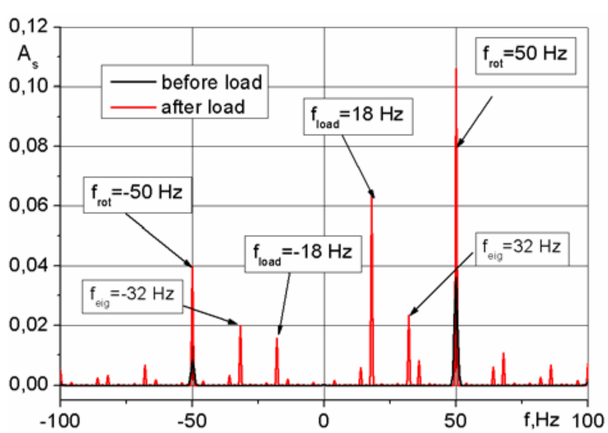
b) rotor orbits in bearing B1 for excitation frequency f = 18 Hz



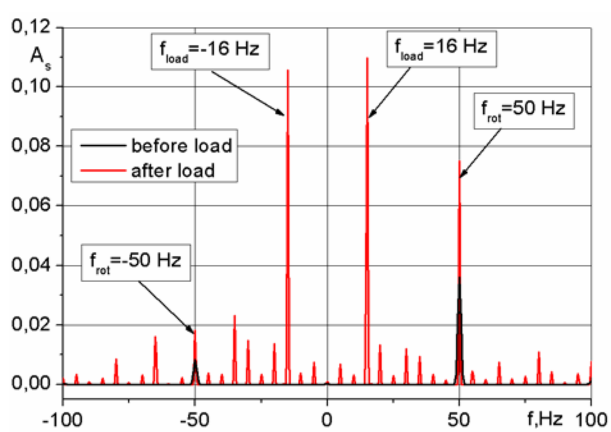
c) maximal relative change of gap in gasdynamic path at lateral excitation



d) rotor orbits in bearing B1 for excitation frequency f = 16 Hz

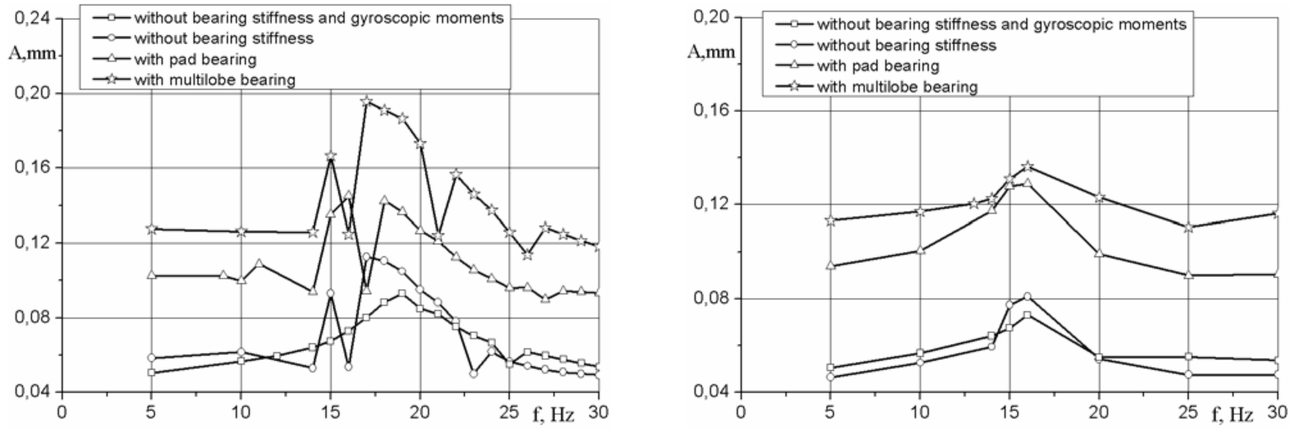


e) full spectrum of orbit in bearing B1 for excitation frequency f = 18 Hz (axial excitation)



f) full spectrum of orbit in bearing B1 for excitation frequency f = 16 Hz (lateral excitation)

Fig. 7 – Gas turbine unit with power turbine flexible rotor in hydrodynamic multilobe bearings



a) maximal relative change of gap in gasdynamic path at axial excitation

b) maximal relative change of gap in gasdynamic path at lateral excitation

Fig. 8 – Disk 3 relative displacements at earthquake excitation for different models

Table 2 – Gas turbine parts stresses at seismic load

Element of gas turbine	Equivalent stress in element of structure, MPa
Power turbine casing (side fixation to bearing frame)	151
Core engine casing	115
Power turbine rotor (disks flange)	3.45
Power turbine bearing frame (side support of power turbine casing)	39.6
Core engine bearing frame (side support of core engine)	56.9
Core engine bearing frame (front support of core engine)	17.9

5. Conclusion

Gas turbine unit dynamic behavior investigation methodology under the seismic load is developed. Presented mathematical model taking into account rotor gyroscopic moments and nonlinear characteristics of hydrodynamic bearings. Estimation of earthquake excitation on rotor orbits in disks cross-sections and on gas turbine unit parts stress-strain state is carried out for rotor supported in different hydrodynamic bearings types. Shown that rotor gyroscopic moments and hydrodynamic bearings nonlinear characteristics defining stationary gas turbine unit response at seismic load in general. Maximal displacements and deformations in gas turbine unit are appear at excitation on frequencies close to natural frequencies. Proposed methodology allows to find reactions in foundation mounting points, also forces and moments acting in critical structural parts and in bolt connections. The stress-strain state is sufficiently changed in critical elements evaluating gas turbine unit working capacity due to gyroscopic moments. It is most important for bearing frames mounting points and rotor supports. For bearing frames mounting points the six-fold overload is possible. Stresses in stress concentration zones are also growing in rotor supports structural elements.

Rotor radial displacements in bearings and disks cross-sections are determined. That allows to make a conclusion about hydrodynamic bearings working capacity and relative gaps in gas dynamic path changing. Comparative estimation of earthquake excitation influence on rotor orbits supported in hydrodynamic bearings with self-positioned pads and fixed lobes is carried out. For earthquake amplitude of 9 the maximal gap changing in gasdynamic path could reach to tenth of millimeter. However, taking into account that earthquake is a random process, the gap changing need to be corrected in assumption of simultaneous action of several frequencies in possible earthquake spectrum. This will lead to growth of relative rotor-casing displacements in gasdynamic path.

For guarantee stationary gas turbine unit working capacity at seismic load with big magnitude the special structural solutions need to be developed. The most sufficient improvement could be providing by removing of gas turbine unit natural frequencies from earthquake frequency spectrum range.

5. References

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