ISSN 1392 - 1207. MECHANIKA. 2010. Nr.1(81)

Analysis of some extreme situations in exploitation of complex rotary systems

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1. Introduction

Complex rotary systems (CRS), such as "flexible rotor on sliding bearings connected through gearbox", are used in many technological systems of industry and energetics. Their reliability is crucial for quality and productivity of various technological processes, as well as safety. An axial compressor (Fig. 1) from a chemical industry plant [1, 2] can be presented as a typical example of such system.

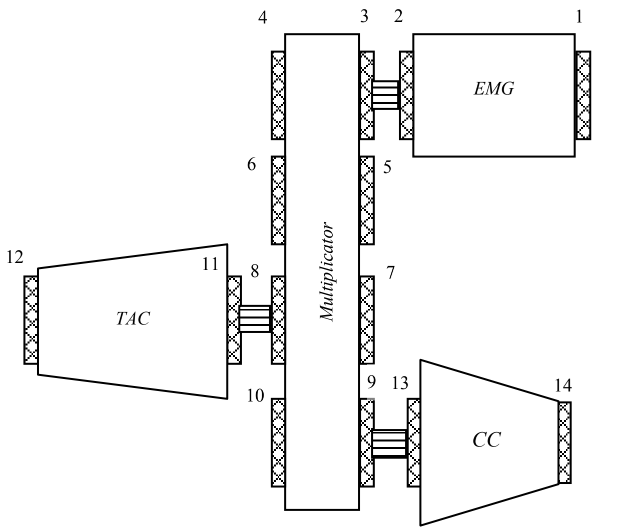


Fig. 1 Scheme of *GTT3* compressor. *EMG* – electric motor-generator, *CC* – centrifugal compressor, *TAC* – turbine-axial compressor, *1, 2,..., 14* – sliding bearings

Present complex rotary systems have longer, more flexible rotors and operate in high speeds, having very small allowed clearances (especially in case of labyrinth or dry sealing). Exploitation speed higher than the first critical and lower (in most cases) than the second critical speed is the typical feature of such rotary systems. Therefore some extreme situations can appear during exploitation of CRS in cases of specific conditions: high amplitude components of subharmonic vibrations may by generated, rubbing of rotor and housing elements caused by increased eccentricity of shafts may appear, etc. All those situations negatively affect reliability of complex rotary systems exploitation. In the most cases, such situations are caused by several defects [2-5].

Similarly rubbing of rotors has to be considered as the secondary phenomenon, resulting from other defects. However it has specific features and is highly dangerous. That makes this phenomenon very important object for research together with other typical sources of subharmonic and superharmonic vibrations.

2. Significance of the subharmonic vibrations

Influence of subharmonic vibrations on dynamics of complex rotary systems has been analysed in various aspects [2-5], however this area still is under intensive research [6-10]. It is established [2-4] that the influence of subharmonic vibrations may be described as "positive". Those vibrations cause reduction of amplitudes of the resonance vibrations during transitional processes of rotary systems (running up and running down when the first resonance frequency is passed). However in the most cases this influence is "negative", especially if high amplitude subharmonic vibrations may be generated for longer periods of time. Those vibrations can cause rubbing, fatigue and cracking of rotors elements. In this case reliability of such rotary system would be reduced significantly and fracture could lead to serious breakdowns.

Therefore two opposite trends are faced. On one hand, the situation of intensive subharmonic vibrations of significant amplitudes leads to reducing of amplitude of the first resonance vibrations. This phenomenon can improve dynamical situation of a vibrating rotor in specific period of time. On another hand, significant subharmonic vibrations (many subharmonic components of considerable amplitudes) can lead to an intensive fatigue of rotary system which (under additional influence of the first resonance vibrations) may cause cracking of the rotor and breakage of its elements. In the most practical cases, reduction of the second tendency is more important. Therefore sources of generation of high amplitudes subharmonic vibrations should be eliminated or reduced.

There are known various sources of subharmonic vibrations the frequencies of which are lower (1/2X, 1/3X, 1/4X) than the frequency of the first form vibrations (considering rotor as a flexible element). In the most cases [2, 4, 11] those vibrations are caused by:

1) variable rigidity of rotor elements in different directions of their cross-sections;

2) unsatisfactorily provided mounting of rotors on sliding bearings (when rotors are eccentric in the respect of bearing axes);

3) trends of gradual degradation of mounting quality during exploitation of a CRS;

4) significant thermal deformations of machines bodies what can cause robbing of rotors and body elements in different directions;

5) peculiarities of technological processes (e.g. pulsation of gas flows, etc.).

One of the most important causes is the first one, because it is determined by inappropriate design or manufacturing quality of a rotor. Therefore this source of vibrations can appear at the very beginning of machines exploitation and can not be controlled technologically. The rest of causes of subharmonic vibrations develop gradually during exploitation of a CRS and their influence may be controlled in a certain level.

Although it is practically impossible to eliminate all subharmonic vibrations, conditions of machines elements mounting and adjustment should ensure that amplitudes of those vibrations would be as low as possible.

In the most cases, subharmonic resonances of frequencies equal to one half of frequency of the rotor first form vibrations are registered. However other significant amplitude subharmonic vibrations can also appear in certain cases [2, 4].

3. Object of the research

Dynamical characteristics of chemical plant compressor *GTT3* (Fig. 1) are analysed. Rotor of the turbine-axial compressor (TAC) of this machine is mounted on hydrodynamic bearings. The specific feature of this turbine design and exploitation is that measurements of machines vibrations are practically feasible only on rotor supports 11 and 12. Characteristics of the machines rotor: working speed is approximately (it is slightly changing due to technological needs) 5040 r/min (84 Hz), the first critical speed is 3276 r/min (54.6 Hz), mass of the rotor is 2500 kg. The general view of TAC rotor in opened body is presented in Fig. 2. This rotor may be formally divided into two zones of gas turbine and axial compressor. Rigidity of the rotors elements is the same in different directions.

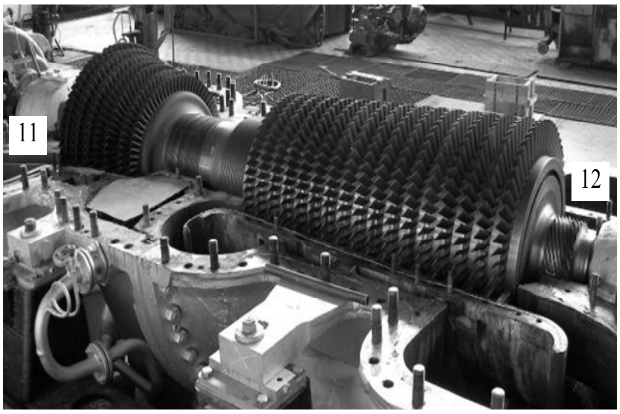


Fig. 2 General view of the TAC rotor. *11*, *12* – places of supports (bearings)

Generally, vibrations of the compressor are not high and correspond to norms of the *ISO 10816-1*, when its assemblage is made appropriately and conditions of exploitation are acceptable. Influence of various components of vibrations is insignificant. However there are specific problems of experimental research of such machines, related with precise determination of vibration sources. Those difficulties are related with the variety of defects what could possibly appear during exploitation: defects of bearings, oil supply systems, sealants, etc. Different parts of this complex machine (Fig. 1) are connected through gear reducer (rigid and semirigid couplings are used). Therefore there may be additional components of vibrations caused by defects of the reducer or even other parts of the machine (compressor or electric motor). Thus it is quite complicated to achieve the main task of condition monitoring – to determinate causes of increased vibrations and predict reliability of such machine.

Analytical dynamical model of this machine has been composed and numerical modelling of machines dynamics, applying the methods of finite elements has been made aiming to model dynamical situations in the presence of various defects. Such models allow modelling of defects development and helps in foreseeing of machines reliability [2, 6, 12, 13].

4. Modelling of rotor dynamics

Rubbing of rotor to the machine body is one of the common defects of such machines. Well known scientists (A. Muszynska, R.F. Benlty, F. Chu and W. Lu, T.H. Patel and A.K. Darpe [2, 6-8], etc.) have made significant efforts and reached important results in modelling of rub phenomena. In the most cases partial or constant rub is modelled by introducing non linear forces of excitation that are generated by impacts of rotor to other elements of the machine. Therefore stiffness of a rotor and tangential forces of friction between the rotor and elements of machine body are changing. Radial force of rubbing can be expressed [6-8]

$$F_r = \begin{cases} 0 & (\text{for } e \langle \delta) \\ (e-\delta)k_s & (\text{for } e \geq \delta) \end{cases} \quad (1)$$

$$F_t = \mu F_r \quad (2)$$

here F_r is radial forces of rubbing (impacting), F_t is tangential force (forces of friction), e is radial displacement of the rotor, δ is the gap between the rotor and other elements, k_s is stiffness of machines body and μ is coefficient of friction. Values of those forces are included into the right side of generalised dynamical Eq. (3). Certain values of tangential and radial forces are placed for those rotor elements which could have impacts with certain elements of machine body. Therefore changes of rotor elements stiffness are modelled during its rotation (this corresponds to the appearance of some virtual temporal supports of the rotor while it is rubbing).

The rotor of centrifugal compressor and the adjacent gear shaft of the reducer (Fig. 1) have been divided into 18 elements and analysed as a system of flexible rotors on four permanent supports. Each element has 4 degrees of freedom. The general equation characterizing forced vibrations of the modelled rotor is described in earlier woks [1, 13]

$$(\boldsymbol{M}+\boldsymbol{M}')\ddot{\boldsymbol{U}}+(\omega\boldsymbol{G}+\boldsymbol{C})\dot{\boldsymbol{U}}+\boldsymbol{K}\boldsymbol{U}=\boldsymbol{F} \quad (3)$$

here ***M*** is the matrix of rotor masses; ***M’*** is the matrix of masses characterizing rotation of the rotor cross-sections around the axes of a coordinate system; ***G*** is gyroscopic matrix; ***C*** is damping matrix; ***K*** is stiffness matrix; ***U*** is the matrix of rotor elements displacements; ***F*** is the matrix of forces affecting the rotor (forces of excitation); ω is angular velocity of the rotor. ***M*** matrix represents the masses of beam elements and matrix ***M'*** allows evaluating the rotation of their cross-sections. The structure of matrix ***F*** depends on the type of exciting forces [7, 13] and includes F_r and F_t elements in the case of rub analysis. Elements of ***F*** are functions of time. Solution of the dynamical equation is obtained applying small steps of time in order to avoid divergence of points where calculated variables are discontinuous.

Such model allows formation of machines amplitude - frequency characteristics (Fig. 3) the changes of which correspond to changes of the machines conditions of operation (appearance of certain defects of chosen rotors elements). Results of the modelling bring valuable information concerning vibrations of internal elements of the rotor what complements experimental data of vibrations of rotor supports.

However, the comparison of obtained results with experimental data (Figs. 4 and 6) shows insufficient coincidence of those results, though reliability of the modelling method is proved [7, 8] with some simplified experimental rotary systems.

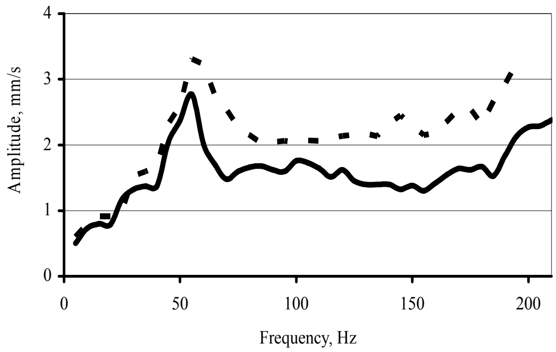


Fig. 3 Modelled amplitude – frequency characteristic of the 11th support vibrations. Solid line – machine is working in good conditions. Dashed line – rubbing occurs

Results of the modelling and their adequacy to exploitation conditions of a real rotary system strongly depend on the validity of initial parameters of the numerical model and their adequacy to physical parameters of a machine. The possibility of alternations of various parameters and presence of various (more or less significant) defects makes it really complicated task to ensure this adequacy.

Therefore a comprehensive experimental analysis of such real CRS and its exploitation conditions is needed. The most significant defects and presumptive locations of those defects should be evaluated as well as real physical parameters of machine elements (e.g. location and direction of possible rubbing, type of contact, etc.).

5. Experimental research of rotor to stator rubbing

Tendencies of degradation of the compressor rotor conditions at exploitation show up in certain duration of time. Specific components of vibrations appear and rise as well as general level of vibrations, becoming significant to machines reliability. Fig. 4 presents spectrum of turbine rotor 11th support (Fig. 1) vibrations.

The method of vibrations measurements applying proximity sensors and analysing orbits of the rotor elements is used in many cases of diagnostics [2, 5, 7]. However, in some practical cases there no possibilities to use this methods and vibration are measured applying seismic sensors. This method does not require specific installations, but has some specific limitations.

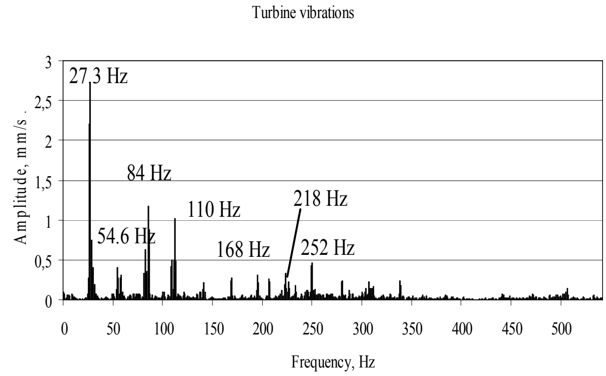


Fig. 4 Spectrum of the compressors 11th support vibrations when duration of the machine exploitation is similar to its typical interservice period

Experimental results present dynamical situation of the CRS when duration of the machine exploitation is similar to its typical interservice period. The second type of subharmonic vibrations (component of 27.3 Hz) can be seen as well as significant components of rotors rotation frequency 84 and 110 Hz. Rubbing of labyrinth sealing caused by thermal deformations of machine body is the possibly cause of the last component of vibrations. In this case, increased component of the rotation frequency is also caused by rubbing.

Spectrum of the machines 11th support vibrations after the breakdown of several turbine blades (as a possible result of rubbing) is presented in Fig. 5. Abnormally increased component of rotor rotation frequency is caused by significant unbalance of the broken turbine. Exploitation of the CRS is impossible in such conditions.

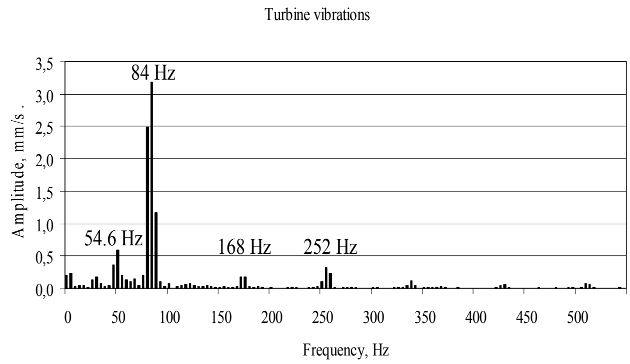


Fig. 5 Spectrum of the compressors 11th support vibrations after failure of some rotor blades

Such emergency situation could be foreseen analysing subharmonic vibrations of the machine, because it had been forming gradually in certain duration of time. Presumptively this situation was preceded by impacting of the CRS rotor blades to machines body that should be noticeable on the spectrum of vibrations as some increased subharmonic components. Accordingly rotor to stator rubbing was preceded by increased eccentricity of the rotor in bearings.

Fig. 6 presents spectrum of the 11th support vibrations in the case of rotor and stator blades partial rubbing. Very complex spectrum with multiple components of frequency of rotation as well as natural frequency (27.3, 54.6, 218 Hz, etc. and 42, 84, 252 Hz, etc.) can be noticed. There are also other subharmonic and super-harmonic, as well as chaotic components. Exploitation conditions of the CRS are unacceptable. Results of such machine failures (traces of rubbing) are presented in Fig. 7.

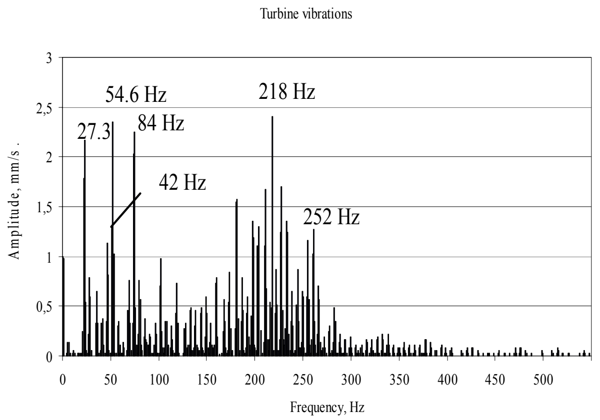
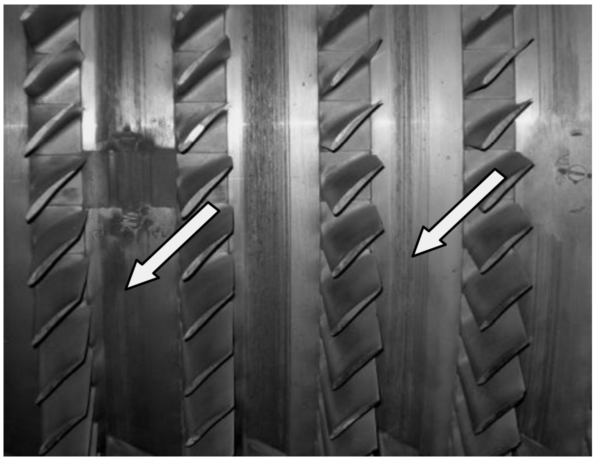
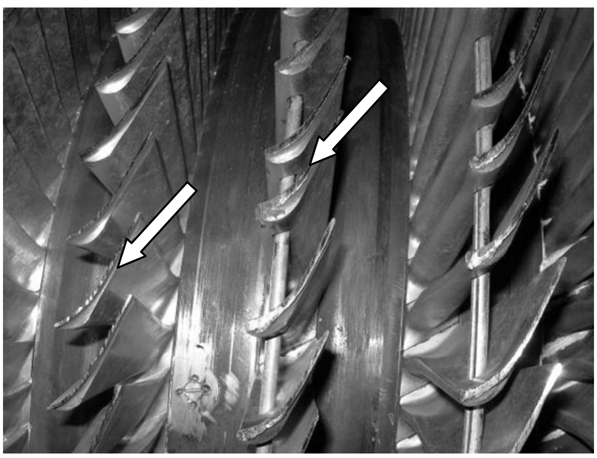


Fig. 6 Spectrum of the compressors 11th support vibrations in case of intensive rubbing



a



b

Fig. 7 Traces of rubbing between rotor and stator blades: a - traces of rubbing between rotor and blades of stator, b - traces on rotor blades left by broken blades

Identification of the primary defects of the rotary system is very important aim of diagnostics, because, as it was mentioned, rubbing of rotors has to be considered as the secondary phenomenon, resulting from other defects. In this case, rubbing was caused by the poor quality of the machine assemblage. However those primary defects were relatively insignificant and were not detected during condition monitoring of the machine.

Such experimental research of the rubbing cases allows determination of rotor elements that can be contacting with other elements of the machine and are affected by additional forces. This research also showed that rubbing appeared in the both – radial and axial direction that should be evaluated in further modelling of the machines dynamics. Only those complicated adaptations of the numerical model can lead to satisfactory adequacy of modelled results and real conditions of machines exploitation, helping determine its defects and foresee the reliability.

6. Conclusions

Certain actions should be initiated in order to avoid extreme situation of rotary system exploitation and in order to increase reliability of a CRS.

1. To determinate and eliminate possible causes of generation of subharmonic vibrations of significant amplitudes. Numerical modelling of machines dynamical situations can be applied, however it is quite complicated because of complexity of the problem and thorough understanding of machines properties is needed.

2. To provide comprehensive analysis of CRS structure and elements as well as conditions of its exploitation, determining possible locations and elements of rotor to stator rubbing, type (direction, duration, etc.) of contact and generation of additional forces.

Improved knowledge of machines dynamics should allow adapting of dynamical model and evaluation of real conditions of machine exploitation.

References

1. **Juzėnas, E., Jonušas, R., Juzėnas, K.** Research of complex rotary systems vibrocondition based on analysis of dynamical processes and spectrum of vibrations. -Mechanika. -Kaunas: Technologija, 2008, Nr.1(69), p.42-45.
2. **Muszynska, A.** Rotordynamics. -CRC Press, Taylor&Francis, 2005.-1128p.
3. **Erich, F.F.** Phenomena of chaotic vibrations in dynamics of high speed rotary systems. -Sovremennoe mashinostroenie, part B, 1991, No.5, p.72-80 (in Russian).
4. **Samarov, N.G., Gusarov, A.A.** Subharmonic vibrations as a factor of reliability of rotary machines. -Izvestia VUZ Mashinostroenie, 1998, No.1-3, p.36-39 (in Russian).
5. **Bently, D.E., Hatch, CH. T.** Fundamentals of Rotating Machinery Diagnostics. -Bently Pressurized Bearing Company, printed in Canada, 2002.-726p.
6. **Chu, F., Lu, W.** Stiffening effect of the rotor during the rotor-to-stator rub in a rotating machine. -J. of Sound and Vibration, 2007, 308, p.758-766.
7. **Patel, T.H., Darpe, A.K.** Vibration response of a cracked rotor in presence of rotor-stator rub. -J. of Sound and Vibration, 2008, 317, p.841-865.

8. **Patel, T.H., Darpe, A.K.** Coupled bending-torsional vibration analysis of rotor with rub and crack. -J. of Sound and Vibration, 2009, 326, p.740-752.
9. **Schweizer, B., Sievert, M.** Nonlinear oscillations of automotive turbocharger turbines. -J. of Sound and Vibration, 2009, 321, p.955-975.
10. **Djebala, A., Ouelaa, N., Hamzaoui, N., Chaabi, L.** Detecting mechanical failures inducing periodical shocks by wavelet multiresolution analysis. Application to rolling bearings faults diagnosis. -Mechanika. -Kaunas: Technologija, 2006, Nr.2(58), p.44-51.
11. **Jonušas, R., Juzėnas, E., Juzėnas, K.** Research of rub phenomena in industrial rotary systems. -Mechanika 2009: Proceedings of 14th International Conference. -Kaunas: Technologija, 2009, p.162-165.
12. **Vasylius, M., Didžiokas, R., Mažeika, P., Barzdaitis, V.** The rotating system vibration and diagnostics. -Mechanika. -Kaunas: Technologija, 2008, Nr.4(72), p.54-58.
13. **Jonušas, R., Juzėnas, E.** The investigation of steam turbine flexible rotor vibrations by means of FEM. -In Proc. of 11th World Congress on Mechanisms and Machine Science, China Machine Press, 2004, v.5, p.2143-2147.

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DALIES EKSTREMALIŲ SITUACIJŲ, SUSIDARANČIŲ SUDĖTINGŲ ROTORINIŲ SISTEMŲ EKSPLOATACIJOS METU, ANALIZĖ

R e z i u m ė

Sudėtingos rotorinės sistemos (SRS) yra eksploatuojamos chemijos pramonės, energetikos įmonėse. Jų darbo patikimumo užtikrinimas yra vienas iš svarbiausių techninės priežiūros uždavinių, nes dėl šių mašinų remonto ir prastovų patiriama didžiulių nuostolių.

SRS būklę labai padeda nustatyti mašinos virpesių diagnostika. Tačiau dėl tokių sistemų kompleksiškumo, įvairių defektų tikimybės ne visada galima tiksliai nustatyti virpesių šaltinius. Papildoma priemonė, leidžianti tiksliau nustatyti sistemos defektus yra jos dinaminės būklės skaitinis modeliavimas.

Šiame straipsnyje yra pateikiami SRS, kurioje reiškiasi rotoriaus ir statoriaus kabinimasis ir sužadinami subharmoniniai virpesiai, eksperimentinių tyrimų ir skaitinio modeliavimo duomenys. Realios SRS modeliavimas (ypač kai vienu metu yra keli defektai) yra sudėtingas uždavinys, todėl būtina tikslinti matematinį modelį, panaudojant eksperimentinių tyrimų duomenis.

R. Jonušas, E. Juzėnas, K. Juzėnas

ANALYSIS OF SOME EXTREME SITUATIONS IN EXPLOITATION OF COMPLEX ROTARY SYSTEMS

S u m m a r y

Complex rotary systems (CRS) are widely used in energetic, chemical industry. Their reliability is one of the main tasks of plant engineering, because costs of maintenance and production loses are really significant.

Vibrodiagnostics of such systems is one of the main tools applied for the evaluation of machines condition. However complexity of those systems, probability of various defects condition that reliable determination of sources of vibrations is not always achievable. Therefore additional mean for defects determination may be applied – numerical modelling of systems dynamical condition.

This article presents data of experimental and numerical research of a CRS, where the phenomenon of rubbing occurs and subharmonic vibrations are generated. Modelling of such realistic system is also a highly complicated problem, therefore improvement of numerical model, based on experimental data is needed.

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ИССЛЕДОВАНИЕ НЕКОТОРЫХ ЭКСТРЕМАЛЬНЫХ СЛУЧАЕВ В ЭКСПЛУАТАЦИИ СЛОЖНЫХ РОТОРНЫХ СИСТЕМ

Р е з ю м е

В химической и энергетической промышленности эксплуатируются сложные роторные системы (СРС). Одна из главных задач технического надзора – обеспечить надежность этих систем, так как ремонты и простои несут существенный ущерб.

Одним из способов, позволяющих определить состояние СРС, является вибродиагностика машин. Однако комплексность таких систем, вероятность разных дефектов часто не позволяют точно определить все источники вибраций. Дополнительным средством, позволяющим более точно определить дефекты системы, является численное моделирование ее состояния.

В статье представлены данные о экспериментальных исследованиях и численном моделировании СРС, в которых наблюдаются задевание ротора и статора и возбуждаемые субгармонические колебания. Моделирование реальной СРС (особенно если возникают несколько дефектов одновременно) является сложной задачей, поэтому необходимо уточнять математическую модель, применяя данные экспериментальных исследований.

Received November 18, 2009
Accepted February 10, 2010