714. Research of electric motor-generator vibrations

Remigijus Jonušas, Kazimieras Juzėnas, Egidijus Juzėnas
Faculty of Mechanical Engineering and Mechatronics
Kaunas University of Technology, Kaunas, Lithuania
E-mail: *remigijus.jonusas@ktu.lt*, *kjuzenas@ktu.lt*, *egidijus.juzenas@ktu.lt*

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Abstract. Vibrations of industrial rotary systems, what depends on many factors, have a significant negative influence on the reliability of those machines. Although the fundamentals of vibrations monitoring, fault diagnostics and methods of vibrations reduction are known, many practical cases related with complex rotary systems still need comprehensive experimental and analytical research.

This paper presents results of experimental research and numerical modelling of vibrations of middle size (0.8 MW) electric motor-generator. This motor is installed in a complex rotary machine, consisting of several sub machines: electric motor-generator, steam-fusion gas turbine-axial compressor and centrifugal compressor, connected through a mechanical reducer. The nominal rotation of the electric motor is 3000 r/min. Diagnostics of the motor have showed the increased level of vibrations. Modelling of the motor dynamics helped in determination of possible faults and finding solutions for the improvement of motor condition.

Keywords: vibrations monitoring, reliability, complex rotary systems, rubbing.

Introduction

Fault diagnostics of complex rotary systems and control of their vibrations are very important scientific and practical problems. Insufficient reliability of technological rotary machines potentially leads to significant financial losses, problems of safety and environmental protection.

Rotors of modern rotary machines have become more flexible and operate in conditions of high (often variable) loads and tight clearances. That leads to increased probability of rotor to stator contacts, cracking and failure of rotor, bearings and sealing elements. Therefore rubbing and cracking of rotors together with faults of bearings are quite frequent defects that may lead to the catastrophic failure of machines [1-7].

Recent research presents some tendencies. The first is application of more sufficient methods of measurement data processing and fault detection. However practical applications of those methods are still complicated and those methods are adapted for diagnostics of relatively simple elements (e.g. rolling bearings).

Another tendency is research related with analytical and numerical modelling of rotary systems. Modelling of dynamical situations allows fault detection, foreseeing of their development and analysis of machines behaviour in some extreme situations that can not be analysed experimentally. A. Muszynska, F. Chu and W. Lu, T. H. Patel and A. K. Darpe and other authors [1-7, 12] have made significant efforts and showed important results in the modelling of rubbing, cracking of rotors and faults of bearings. In the most cases partial or constant rubbing is modelled by introducing non linear forces of excitation [5] and [12] that are generated by impacts of rotor to machines body elements. Analysis of similar numerical models may be useful also for detection of specific initial defects (e.g. defects of bearings, cracks or disbalancies of rotors). However precise modelling of real rotary machines requires applications of complicated, often hybrid models and practically are used for design of new rotary systems or for behaviour modelling of very important ones (e.g. turbomachinery of big power plants).

Rotary system

Research was carried out at a chemical processing plant in on a turbine compressor (designed and produced in the former USSR). This medium power rotary machine consists of synchronic electric motor-generator (Fig. 1), fusion gas turbine-axial air compressor, centrifugal compressor and four-step reducer with chevron gearwheels. Problems of such machines dynamics are related with faults of their design and lack of modern build-in system of condition monitoring.

Fig. 1. General view of the electric motor-generator

In case of the regular operational conditions (between the first and the second critical speed), rotary speed of the electric motor-generator is near 3000 r/min, rotary speed of the centrifugal compressor is near 7500 r/min and rotor of the turbine rotates at speed of 5200 r/min. There are used two types of couplings: semi rigid and flexible ones.

All bearings of all machine rotors are pressurised hydrodynamic bearings. Closed two loops lubrication system is designed with coolers where oil is cooled by running water. Temperature of oil is measured using thermocouples inserted into antifriction layer of sliding bearings. Lubrication system may be filled by the ISO VG 36 viscosity class oils. Various defects of compressors, emerging during their exploitation cycle, were experimentally detected by means of vibromonitoring. The most common defects are faults in rotor’s sliding bearings (some might be caused by improper lubrication of those bearings), faults of centring, defects of turbine and axial compressor’s blades. Thermal deformations of rotors and machines bodies, disbalancies, defects of bearings and insufficient lubrication often lead to rotor to stator rubbing. However, specifics of the compressor exploitation cause some difficulties in diagnostics of such primary defects, especially in the early stage of their development. In some cases only secondary defects (rubbing) have been clearly detected (Fig. 2). Detection of rubbing allows establishing of machines reliability. However it does not help in determination of those initial defects. Exploitation modes and loads vary depending on manufacturing necessity. Therefore determination and localisation of defects, based on experimental data, is not always successful and problems of maintenance planning remain.

Experimental measurements of vibrations of the electric motor supports (oil film bearings) have shoved presence of some defects. Source of vibrations with frequencies equal or repetitive to the frequency of motor rotation has been determined easily. Slight rubbing of the motor rotor was the main source of those vibrations. However source of the 33 Hz vibrations component was not identified clearly, because of complexity of the rotary systems and possible influence of other machines. Fig. 2 shows spectra of the motor support vibrations in the vertical direction. Fig. 2 also shows changes of spectrum of electric motor-generator rotor support vibrations during it acceleration in case of partial radial rubbing. Frequency of the rotor rotation (x) changes from 8 to 50 Hz. At the same time development of higher modes vibrations with frequency 2*x*, 3*x*, 4*x* and even 5*x* to 8*x* (from the 6th curve) may be noticed. There is also component of vibrations (100 Hz), caused by the electric field. Spectra components, repetitive to frequency of rotation (50 Hz, 100 Hz, etc.), partial frequencies (25 Hz, 75 Hz, etc.) may be noticed. There is also component of 33 Hz showing higher speed of vibrations in the vertical direction.

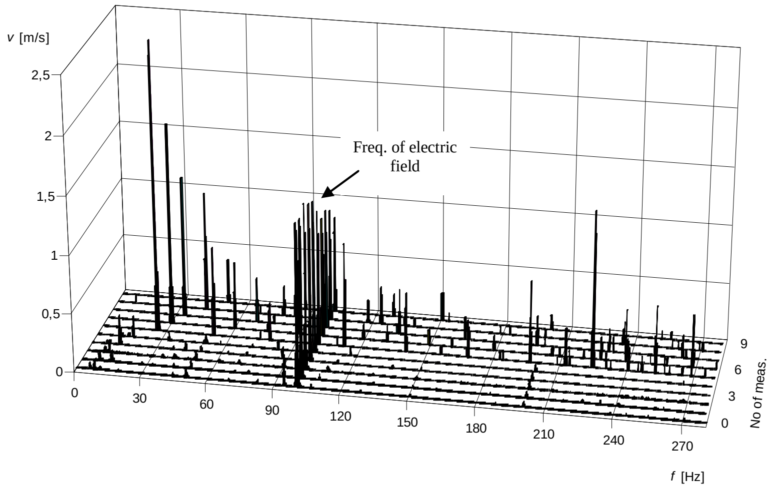

Fig. 2. Changes of spectrum of bearing vibrations caused by development of radial rubbing

Modelling of rotor dynamics

Proposed solution of this problem is application of relatively simple (in order to save time of model composition and analysis) finite element (FE) model of electric motor-generator (Fig. 1) rotor. The aim of this modelling is determination of defects (located on practically predicted elements) impact on dynamical condition of the rotary system.

Composed model (Fig. 3) allows changing of various parameters of dynamical system. Characteristics of vibrations of any element of compressor's rotor can be obtained for various conditions of exploitation (different properties of oil, defects of the rotor or bearings, etc.). Those characteristics can be used in evaluation of rotor dynamic state and for the modelling of influence of various parameters on the level of vibrations [14]. Here elements 1, 4, 6 and 8 represent location of labyrinth sealing of the rotor, where rubbing the most likely appears.

Dynamical equation of such rotor, evaluating the gyroscopic effect and forces, generated in bearings, can be written in the following way [1] and [14]:

$$(\mathbf{M}+\mathbf{M}')\ddot{\mathbf{U}}+\omega\mathbf{G}\dot{\mathbf{U}}+\mathbf{C}\dot{\mathbf{U}}+\mathbf{K}\mathbf{U}=\mathbf{F}+\mathbf{F}_{\mathrm{K}}+\mathbf{F}_{\mathrm{C}} \tag{1}$$

here **M** – matrix of rotor masses, **M'** – matrix of masses characterizing the revolutions of rotor cross-sections around their axes, **G** – gyroscopic matrix, **C** – damping matrix, **K** – stiffness matrix, **U** – matrix of rotor elements displacements, **F** – matrix of acting forces on a rotor and includes forces of rubbing f_r and f_t in the case of rub analysis, $\mathbf{F_K}$, $\mathbf{F_C}$ – matrix estimating hydrodynamic forces, ω – angular frequency or rotor revolution.

Partial rubbing is modelled as radial forces of rubbing [5, 9, 12] appearing on the certain element of the rotor when it is hitting stator (the radial displacement $e=\sqrt{x^2+y^2}$ of the rotor element becomes larger when the clearance between rotor and stator δ). Therefore, if $e \geq \delta$:

$$f_r = (e-\delta)k_s, \quad f_t = \mu(e-\delta)k_s. \tag{2}$$

There μ is coefficient of friction k_s is stiffness of stator. Forces in the x and y directions:

$$f_{rx} = -f_r \cos\phi + f_t \sin\varphi,$$
$$f_{ry} = -f_r \sin\phi - f_t \cos\varphi, \quad \tan\phi = \frac{x}{y}. \tag{3}$$

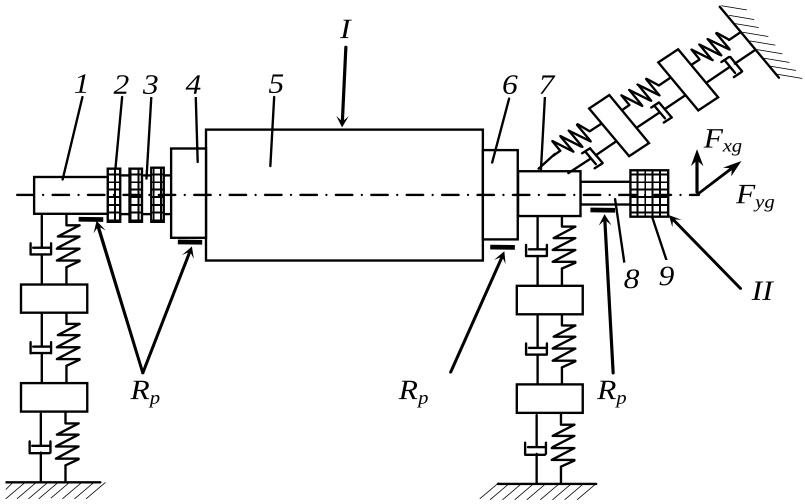

Fig. 3. Dynamic model of the modelled rotor (supporting structure of the 1st element in the y direction is not shown): I – rotor of the electric motor-generator, II – coupling, R_p – modelled locations of rubbing (locations of the labyrinth sealing), F_{xg} and F_{yg} – forces generated by the reducer, 1 – 9 structural elements of modelled rotor

The hydrodynamic forces acting in sliding bearing journals in x and y directions are calculated as follows:

$$\begin{aligned} F_{KCx} &= -K_{\mathrm{xx}}x - K_{\mathrm{xy}}y - C_{\mathrm{xx}}\dot{x} - C_{\mathrm{xy}}\dot{y} \\ F_{KCy} &= -K_{\mathrm{yy}}y - K_{\mathrm{yx}}x - C_{\mathrm{yy}}\dot{y} - C_{\mathrm{yx}}\dot{x} \end{aligned} \tag{4}$$

here $K_{\mathrm{xx,\,xy,\,yx,\,yy}} = \omega\frac{\mu l}{\psi^3}I_{1,2,3,4}$ and $C_{\mathrm{xx,\,xy,\,yx,\,yy}} = \frac{\mu l}{\psi^3}I_{5,6,7,8}$ are dynamic characteristics of a bearing, l – length of a bearing, $\mu = \nu\rho$ – dynamic viscosity of lubricant, ν – kinematic viscosity of lubricant, ρ – density of lubricant, $\psi = \Delta/R$ – relative clearance of bearing, R – radius of shaft journal, Δ – relative clearance, $I_1,...,I_8$ – coefficients of bearing dynamics (those coefficients are obtained from manufacturers specifications and experimental measurements). Values coefficients K and C are used for forming matrixes $\mathbf{F_K}$ and $\mathbf{F_C}$.

Amplitude-frequency and phase-frequency characteristics of any rotor element can be obtained. Some numerically modelled amplitude-frequency characteristics of rotor bearings vibrations are presented in Fig. 4.

It was assumed in the modelling that rubbing is developing on the 4th element when the frequency of rotation rises to 7 Hz (values of various parameters are taken form experimental research of the machine). Rubbing on the 8th element appear when amplitude of this element vibrations rises till 0,15 mm and becomes constant when the amplitude riches 0,2 mm.

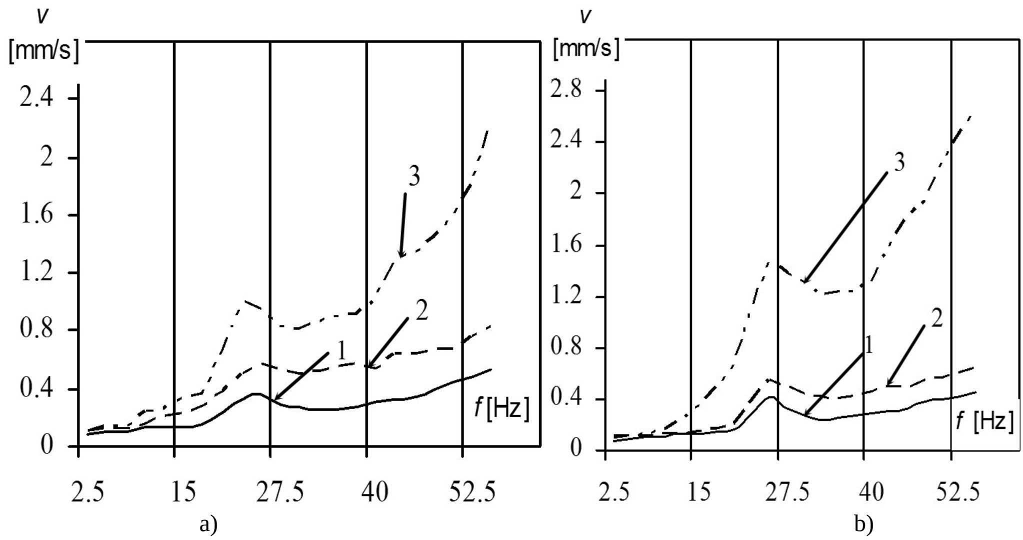

Fig. 4. Modelled amplitude-frequency characteristics of the a) 1st support: 1 – there are no rubbing, 2 – rubbing in the 4th element, 3 – simultaneous rubbing in the 4th and 8th elements, b) 2nd support: 1 – there are no rubbing, 2 – rubbing in the 4th element, 3 – simultaneous rubbing in the 4th and 8th elements

Results of the modelling together with experimental results allow detection of machine defects that caused rise of 23.3 Hz vibrations (Fig. 2). Explanation of this peak is origination of rubbing in a labyrinth sealing of the rotor when speed of rotor reached 1400 rotations per min. Further decrees of the 1st and other harmonics vibrations (Fig. 2) may be explained by transition of the constant rubbing into the partial one caused by increasing gap between the rotor and stator elements. Therefore modelling of the rotors dynamics allowed simulation of dynamic situations in cases of different defects. Modelling has revealed that increased vibrations are caused by instable oil film in the rotors supports. Instability of the oils film has been caused by imperceptible axils movement of the rotor generated by axial forces of rubbing.

Significant traces of very intensive wear of aluminium elements of the labyrinth sealing founded during maintenance of the machine proved presence of rubbing in this element and explained why amplitudes of vibrations were decreasing.

Dynamic model of the motor-generator has been also used for prediction of rotor behaviour in case of varying density of bearings oil. It has been revealed that insignificant change of oil density allows reducing of above mentioned vibrations. This presumption was tested practically by increasing oil temperature. Amplitude of vibrations has been reduced that allowed further exploitation of the machine.

Conclusions

Some cases of diagnostics of complex rotary systems bring complicated problems of defects detection and prognostics of machines reliability when traditional analysis of data of condition monitoring does not give reliably results. FEM modelling is useful tool for analysis of various dynamic situations of rotary systems and impact of different defects on machines vibrations. However composition of detailed FE models and parametrical analysis demand for significant human and computational time. Therefore it can not be practically performed for every system.

Applications of presented relatively simple model require less of recourses for it composition and parametrical analysis. This model gives results acceptable for many practical cases if they are analysed together with relevant experimental data. So, effective applications of the model are possible only having deep knowledge of the analysed systems.

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References

[1] **Muszynska A.** Rotordynamics. Boca Raton: Taylor & Francis, 2005.
[2] **Heng A., Zhang S., Tan A. C. C., Mathew J.** Rotating machinery prognostics: State of the art, challenges and opportunities. Mechanical Systems and Signal Processing, No. 23, 2009, p. 724–739.
[3] **Less A. W., Sinha J. K., Friswell M. I.** Model-based identification of rotating machines. Mechanical Systems and Signal Processing, No. 23, 2009, p. 1884–1893.
[4] **Bachschmid N., Pennacchi P., Chatterton S., Ricci R.** On model updating of turbo-generator sets. Journal of Vibroengineering, Vol. 11, No. 3, 2009, p. 379–391.
[5] **Pennacchi P., Bachschmid N., Tanzi E.** Light and short arc rubs in rotating machines: Experimental tests and modelling. Mechanical Systems and Signal Processing, No. 23, 2009, p. 2205–2227.
[6] **Jonušas R., Juzėnas E., Juzėnas K.** Analysis of some extreme situations in exploitation of complex rotary systems. Mechanika, No. 1 (81), 2010, p. 53–56.
[7] **Barzdaitis V., Mažeika P., Grigonienė J., Didžiokas R., Kartašovas V.** Modeling and diagnostics of the gear power transmissions. Mechanika, No. 17 (1), 2011, p. 71-77.
[8] **Triesz P.** Stability of the rubbing phenomenon with the gyroscopic effect, An International Journal for Engineering and Information Sciences, No. 1, 2010, p. 65–72.
[9] **Chu F., Lu W.** Stiffening effect of the rotor during the rotor-to-stator rub in a rotating machine. Journal of Sound and Vibration, No. 308, 2007, p. 758–766.
[10] **Chang-Jian C.-W., Chen C.-K.** Non-linear dynamic analysis of rub-impact rotor supported by turbulent journal bearings with non-linear suspension. International Journal of Mechanical Sciences, No. 50, 2008, p. 1090–1113.
[11] **Chang-Jian C.-W., Chen C.-K.** Couple stress fluid improved rub-impact rotor-bearing system – Nonlinear dynamic analysis. Applied Mathematical Modelling, No. 34, 2010, p. 1763–1778.
[12] **Patel T. H., Darpe A. K.** Coupled bending-torsional vibration analysis of rotor with rub and crack. Journal of Sound and Vibration, No. 326, 2009, p. 740–752.
[13] **Roques S., Legrand M., Cartraud P., Stoisser C., Pierre C.** Modelling of a rotor speed transient response with radial rubbing. Journal of Sound and Vibration, No. 329, 2010, p. 527–546.
[14] **Jonušas R., Juzėnas E.** The investigation of steam turbine flexible rotor vibrations by means of FEM. Proc. of 11th World Congress on Mechanisms and Machine Science, China Machine Press, Vol. 5, 2004, p. 2143–2147.