THE ANALYSIS OF THE ROTOR'S LONGITUDINAL VIBRATIONS WITH LARGE MISALIGNMENT OF SHAFTS AND ROTEX TYPE COUPLING

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Summary

It is very difficult to unambiguously define the symptoms of misalignment of rotating elements of machine on the basis of vibration spectrum. With numerous states of inefficiency the picture of amplitude-frequency characteristic which is the basic tool of vibrodiagnostics is similar. Loose coupling or rubbing of the rotor's moving parts against fixed elements cause large vibration velocity amplitudes for the sequence of ultra-harmonics of the rotational frequency, similarly as in the case of drive system misalignment.

The author analyzes the character of the rotor's longitudinal vibrations in which there is a large misalignment of shafts joined by coupling with spring elements in a form of a plastic insert with high hardness. It is also showed how the response of the system to the axial force changes in the case where spring element is damaged.

Keywords: misalignment, vibrations in the direction of the rotor's axis, rotor dynamics, vibration spectrum, rotor modelling, numerical computation.

ANALIZA DRGAŃ WZDŁUŻNYCH WIRNIKA PRZY DUŻEJ NIEWSPÓŁOSIOWOŚCI WAŁÓW I SPRZĘGLE TYPU ROTEX

Streszczenie

Jednoznaczne określenie symptomów występowania niewspółosiowości wirujących elementów maszyny na podstawie widma drgań jest zagadnieniem bardzo trudnym. Przy licznych stanach niesprawności obraz charakterystyki amplitudowo-częstotliwościowej, będącej podstawowym narzędziem wibro-diagnostyki, jest podobny. Luz w łożysku lub ocieranie części ruchomych wirnika o elementy nieruchome są przyczyną dużych wartości amplitud prędkości drgań dla kolejnych ultraharmonicznych częstotliwości obrotowej, analogicznie jak w przypadku niewspółosiowości w układzie napędu.

Autor analizuje charakter drgań wzdłużnych wirnika, w którym występuje duża niewspółosiowość wałów łączonych sprzęgłem z elementem sprężystym w postaci wkładki wykonanej z tworzywa sztucznego o dużej twardości. Pokazano również jak zmienia się odpowiedź układu na wymuszenie siłą osiową w przypadku, gdy element sprężysty ulega uszkodzeniu.

Słowa kluczowe: niewspółosiowość, drgania w kierunku osi wirnika, dynamika wirnika, widmo drgań, modelowanie wirnika, obliczenia numeryczne.

1. INTRODUCTION

The diagnosis of misalignment of shafts joined by coupling based on amplitude-frequency characteristic of a particular parameter of their vibrations is burdened with a high degree of uncertainty. Usually, it is stated that the dominant value in vibration spectrum for frequencies 1x and 2x constitutes believable symptom of this inefficiency. Rotor's vibration spectrum during resonance and with clearance between coupling and casing has often similar shape. This is understandable because in each case we are dealing with non-linear stiffness of the system.

The machines' rotors very often operate at rotational speeds close to critical velocities. The analysis of the character of the rotor's resonance vibration during misalignment of shafts becomes particularly important. Misalignment of the joined rotating elements, to greater or lesser extent, occurs almost always. That is why properly designed joint should show flexibility in a degree allowing the compensation of the effects connected with it.

The influence of the joint type and contact interaction between stiff and elastic elements on the character of the rotor's vibrations was discussed in the thesis [1]. The identification of main factors which affect transverse vibrations of the rotor was made by Woodcock [2]. Bloch and Geitner [3] estimated maximum values of misalignment for the joints because of the stress state in a coupling. Grigoriev [4] in theoretical and experimental way determined and then compared amplitude-frequency characteristics of bearing bodies' vibrations for different types of joints of cooperating shafts. Gibbons proved that misalignment causes the forces in a coupling which extort the vibration of the system [5].

Numerous researchers [6-7] have tried to define the symptoms of the system's misalignment by showing on the basis of the character of rotor's vibrations that most often the symptom is a huge value of amplitude for frequencies twice bigger than the rotational frequency of the shafts. Dewell and Mitchel [8] proved that vibration spectrum of the system with misalignment consists of components with broad frequency spectrum, however the components 2x and 4x of the rotational speed experience the greatest changes when the misalignment rises.

The shafts' misalignment, especially of an angular character, is accompanied by a time-varying axial force which composes a force for rotor's longitudinal displacements. This effect influences the form of amplitude-frequency response of the vibration's velocity. Therefore, an examination of the rotor's longitudinal vibrations character is as much important as analyzing its lateral vibrations.

2. THE ANALYSIS OF THE CHARACTER OF LONGITUDINAL VIBRATIONS OF THE ROTOR WITH MISALIGNMENT IN THE DRIVE SYSTEM.

The object of the research was twin-disk stiff rotor joined with the shaft of the motor by ROTEX type coupling with intermediary element in a form of plastic insert of 92ºSh hardness. The resulting misalignment of the system was superposition of parallel shift of shafts' axis and their relative rotation. The values of parameters describing reciprocal position of axis were depicted for three cases for which the research was conducted. They are presented in table 1.

Table 1. Parameter values of joined shafts misalignment

Surface	Type of misalignment	Parameter value of misalignment		
		state 1	state 2	state 3
horizontal	parallel	0.01mm	0.04mm	0.05mm
	angular	0.083%	0.133%	0.183%
vertical	parallel	0.01mm	0.13mm	0.47mm
	angular	0.017%	0.717%	2.333%

State 1 corresponds with the situation when misalignment of the rotor's and motor's shafts is little. State 3 relates to the case of very big misalignment. This applies mainly to relative angular position of the shaft's axis which amounts to 2.33%. As for the diameter of the coupling equal to 60mm it corresponds with the angle of 1.4º. State 2 defines the intermediate case between states 1 and 3.

Fig. 1 presents the way of measuring deviation of the rotor's and motor's shafts misalignment. On one of them there is an emitter/detector of laser beam reflected by the mirror placed on the second shaft. Measurement error is 0.001mm which affects precision of the shift of the regulated element in both surfaces of 0.01mm. Relative placement of the shafts is determined during their simultaneous rotation by angle the minimum value of which is 60º.

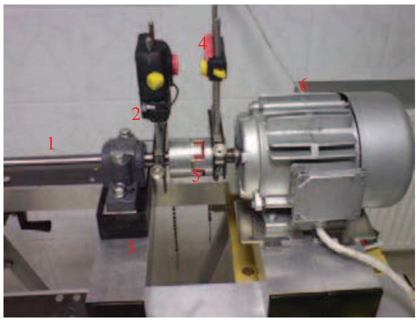


Fig. 1. The way of measuring misalignment of the rotor's and motor's shafts by using laser tool:
1 - rotor's shaft, 2 - bearing with the casing,
3 - rubber pad, 4 - laser tool, 5 - coupling, 6 - motor

Measurements of longitudinal vibration parameters of the motor and rotor's bearing located by the coupling were conducted for the rotational speed of 1200r.p.m, which is equal to frequency of 20Hz. It was also checked by determination of resonance curve of bearings' vibrations that the exploitative rotational speed of the rotor is not in the resonance area.

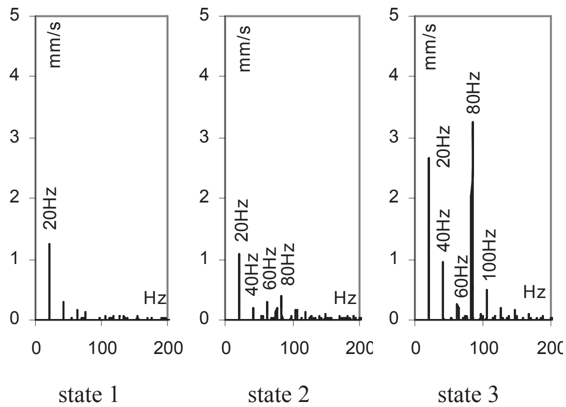


Fig. 2. Longitudinal vibration velocity spectrums of the rotor's bearing by the coupling

Fourier transforms determined on the basis of the time course of displacement speed prove that longitudinal vibrations of the rotor's bearing with bigger misalignment are characterized by the growth of components' amplitude with frequency of 1x, 2x and in particular 4x (Fig. 2).

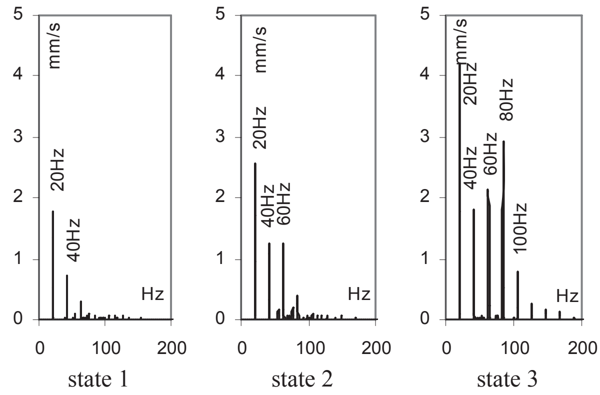


Fig. 3. Longitudinal vibration velocity spectrums of the motor

In he case of longitudinal vibrations of the motor which feet were lowered compared to the rotor's axis of 3.5 mm at the front and 5.83 mm at the back, the growth of the amplitude of components 1x, 2x, 3x, 4x, and even 5x was observed (Fig. 3). The characteristic thing is that high value of vibration amplitude of ultra-harmonic frequencies 1x and 4x occurs for numerous rotational speeds of the rotor (Fig. 4). This conclusion results from the analysis of vibration spectrum of the system of 1.64% misalignment.

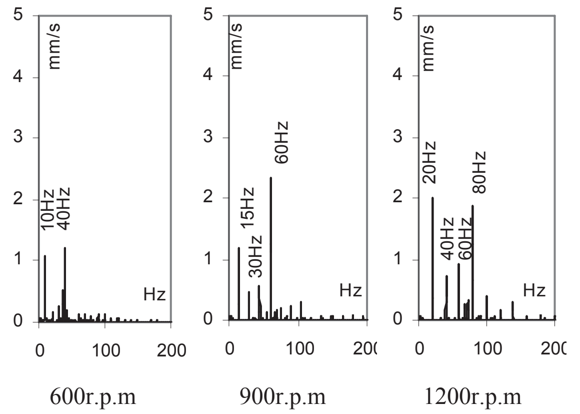


Fig. 4. Longitudinal vibration velocity spectrums of the rotor with angular misalignment of 1.64%

This kind of distribution of amplitude values results from the construction features of coupling. Its each half has four interlocking during the rotation centres. Contact force extorts longitudinal vibrations of frequency which is the product of the amount of pairs of elements and rotational frequency of the rotor. Interesting is the fact that if the geometry of coupling allows temporary contact between one centre and the socket so in fact the occurrence of high amplitude values in longitudinal vibration spectrum should be expected, also with the frequency of 8x. The correctness of this assumption was verified on the basis of the numerical analysis of the rotor's movement.

3. NUMERICAL ANALYSIS OF LONGITUDINAL VIBRATIONS OF THE ROTOR

The rotor which is a model for calculation is presented in Fig. 5. It can be used for the analysis of transverse vibrations as well as longitudinal vibrations and also torsional vibrations.

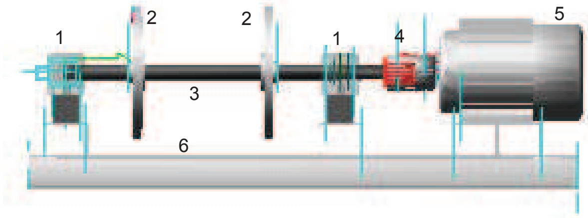
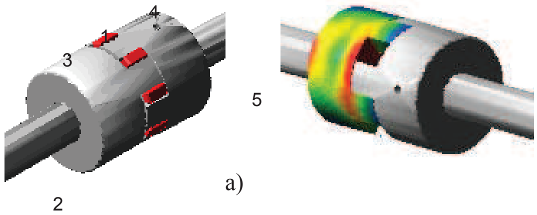


Fig. 5. Numerical model of the rotor: 1 - bearings ans casings, 2 - discs, 3 - rotor's shaft, 4 - coupling, 5 - motor, 6 - frame

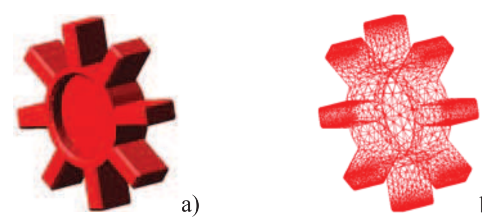
The model reflects the features of the stand presented in Fig.1 in the field of interial properties (mass and moments) and stiffness and damping of the support. The most crucial, from the point of view of the problem analysis, is the applied ROTEX model coupling. The halves of the coupling are treated as rigid blocks (Fig. 6), while the intermediary insert is a deformable element (Fig. 7a) which was discretized into finite elements in a way showed in Fig. 7b.



a) b)

Fig. 6. a) The model of the rotor used for calculations: 1 - intermediary element, 2 - rotor's shaft, 3 - the element of coupling connected with the rotor, 4 - the element of coupling connected with the motor, 5 - motor's shaft
b) Stress distribution in a half of the coupling from the rotor's side without the elastic insert

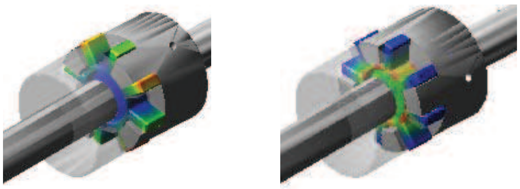
Thanks to that it was possible to determine the method of element's deformation during rotation of the rotor as well as stress distribution in its sections.



a) b)

Fig. 7. Intermediary element: a) look, b) the method of finite elements' discretization

Twisting moment by affecting the halves of the coupling causes bending of pads in a shape of cuboid placed symmetrically on the perimeter. The pads act identically as the beam clamped by one end and loaded in a continuous manner. Therefore, the greatest deformation occurs at the ends of elements (Fig. 8a).



a) b)

Fig. 8. a) Deformation of intermediary element in the next phase of rotation, b) stress distribution in the element

Maximum value of the stress occurs at the base of the element (Fig. 8b). For the torsion moment transferred by the coupling the coupling's material effort is small and amounts to only a few megapascals. The elastic intermediary element was also used in order to determine its damping properties. Fig. 9 presents longitudinal vibration velocity spectrums of the rotor for different stiffnesses in the contact area of coupling's metal elements separated by the susceptible insert.

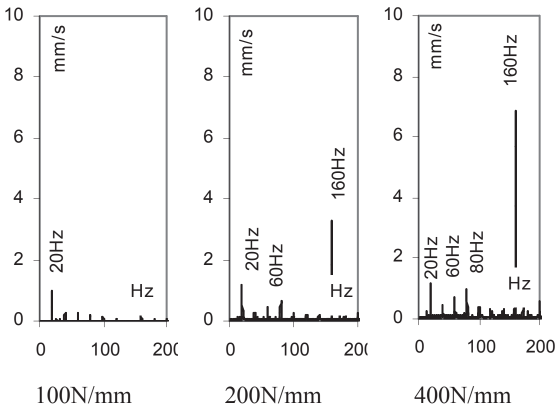


Fig. 9. The character of longitudinal vibration velocity spectrum of the rotor for different stiffnesses in the contact area of coupling's halves with elastic insert – numerical computations results

The analysis of vibration spectrums of the simulated rotor allows to draw a conclusion that along with the growth of stiffness in the contact area the value of vibration amplitude with frequency of 8x is growing. Therefore, it can be assumed that the actual stiffness in the contact area of coupling's elements is low and that is why we observe the dominant value of velocity amplitude of vibrations at the frequency of 4x. The correctness of this assumption can be proved by the picture of velocity spectrum of rotor's vibrations with coupling without the susceptible element (Fig. 10) for the stiffness of the contact between coupling's elements amounting to 100N/mm. The comparison of Fig. 9 and 10 gives an overview on the role of the insert in damping coupling's vibrations.

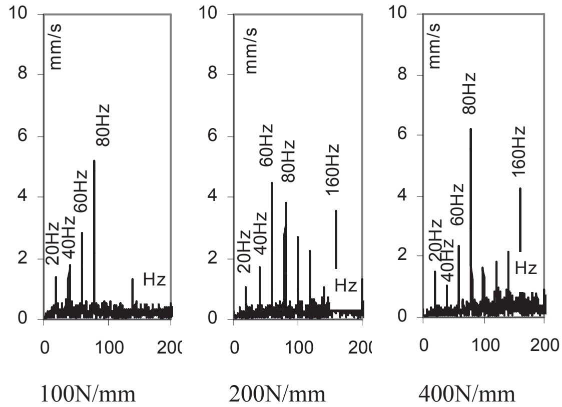


Fig. 10. Longitudinal vibration velocity spectrums of the rotor for different stiffnesses in the contact area of coupling's halves without the elastic insert – numerical computations results

Stress distribution in coupling without the intermediary element is presented in Fig. 6b. This state corresponds with the position where there is no axial play. For greater clarity, the stress map was visualized only in one part of coupling.

4. CONCLUSIONS

The research as well as numerical analysis of the system's vibrations confirmed the thesis presented at the beginning that unambiguous definition of the symptoms of misalignment is almost impossible. The picture of vibration spectrum depends on the size of misalignment, rotational speed of the rotor, construction features of coupling as well as stiffness in the contact area between its elements.

ROTEX type coupling with susceptible elements which was used in the research does not allow to obtain large angular misalignment of the shafts. For the value over 1.5º, as the rotational speed of the rotor was growing, the temperature of the elastic element was growing, and acoustic effects accompanied the work of the system. Durability of the insert in such conditions is rapidly declining. This situation is extremely unfavourable because susceptible element has damping properties and its damage causes the growth of vibration level of the rotor.

REFERENCES

[1] Mancuso J. R.: *The manufacturers world of coupling potential unbalance*. Proceedings of the Thirteenth Turbomachinery Symposium, 1985, s. 97-103.
[2] Woodcock J. S.: *The effects of couplings upon the vibration of rotating machinery*. Proceedings of the International Conference on

Flexible Coupling University of Sussex, Brighton U.K, Farnham U.K. Michael Neale and Associates Ltd., 1977, s. 1-20.

[3] Bloch H. P., Geitner F. K.: *An Introduction to Machinery Reliability Assessment*. New York, 1990.

[4] Grigoriev N. V.: *An investigation into the vibration isolation properties of elastic couplings of connected shafts under flexural vibration*. Soviet Engineering Research 2(12), 1976, s. 42-43.

[5] Gibbons C. B.: C*oupling misalignment forces.* Proceedings of the Fifth Turbomachinery Symposium, Gas Turbine Laboratories, Texas A&M University, College Station, Texas, 1976, s. 111-116.

[6] Schwerdlin H.: *Reaction forces in elastomeric couplings*. Machine Design 51(16), 1979, s. 76-79.

[7] Maten S.: Program machine maintenance by measuring vibration velocity. Hydrocarbon Processing, 49(9), 1970, s. 291-296.

[8] Dewell D. L., Mitchell L. D.: *Detection of a misaligned disk coupling using spectrum analysis*. American Society of Mechanical Engineers Journal of Vibration, Acoustics, Stress and Reliability in Design, 106, 1984, s. 9-15.

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