

DIAGNOSTICS OF THE TURBOGENERATOR SHAFT LINE MISALIGNMENT BASED ON THE BEARING TRAJECTORY PATTERNS

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Summary

The paper aims at illustrating effects of computer simulated bearing misalignment defect in a power turbogenerator. The results are presented in the form of journal trajectories of all turbo-set bearings as the effect of displacing two most vulnerable machine bearings. The analysis is limited to misalignment of the bearings in horizontal and vertical directions by the maximum acceptable range calculated with regard to permissible bearing vibration. Shape and dimensions of bearing trajectories are interpreted basing on theory of hydrodynamic lubrication of bearings. It was shown that relative journal trajectories carry much important information about dynamic state of the machine, indicating also the way in which bearings are loaded. The article shows the potential of using trajectory patterns for diagnosing misalignment defects in rotating machines and suggests including sets of trajectory patterns to the knowledge base of the machine diagnostic system.

Keywords: rotor-machine, vibration, misalignment, bearing trajectory, technical diagnostics.

DIAGNOSTYKA ROZOSIOWANIA LINII WAŁÓW TURBOGENERATORA NA PODSTAWIE OBRAZÓW TRAJEKTORII ŁOŻYSK

Streszczenie

W artykule przedstawiono rezultaty komputerowej symulacji defektu rozosiowania łożysk turbogeneratora wielkiej mocy. Wyniki zostały zaprezentowane w formie trajektorii czopów wszystkich łożysk turbozespołu jako skutku przemieszczenia dwu najbardziej newralgicznych łożysk tej maszyny. Analiza jest ograniczona do rozosiowania tych łożysk w kierunku poziomym i pionowym o maksymalną dopuszczalną wielkość obliczoną ze względu na drgania łożysk. Kształt i wymiary trajektorii łożysk są interpretowane na podstawie hydrodynamicznej teorii smarowania. Pokazano, że względne trajektorie czopów niosą wiele ważnych informacji o stanie dynamicznym maszyny, wskazując także na sposób obciążenia łożysk. Artykuł wskazuje na możliwości użycia obrazów trajektorii do celów diagnostyki defektu rozosiowania w maszynach wirnikowych i sugeruje włączenie obrazów trajektorii do bazy wiedzy systemu diagnostycznego maszyny.

Słowa kluczowe: maszyna wirnikowa, drgania, rozosiowanie, trajektoria łożyska, diagnostyka techniczna.

1. INTRODUCTION

Displacing any of turbo-set bearings from its optimum position, defined by the designed shaft catenary, changes conditions of operation of the oil bearings and consequently rotors supported on them. The distribution of the static load of bearings and shafts changes distribution of unbalance along the shaft line and as a following consequence vibrations can be generated [1-4].

The motivation for undertaking the investigations were own observations [5] and the information in the literature on a remarkable effect of bearing misalignment on the dynamic state of a rotating machine with a long shaft line supported in numerous bearings [1-4, 6-9]. According to [7] "Misalignment of multi-bearing rotor systems is one of the most common fault conditions yet it still not fully understood". When the frequency of occurrence is taken into account, bearing

misalignment is the second most common type of failure of a rotating machine, after unbalance [8]. According to [9], misalignment may cause even 70% of vibration problems, which are observed in rotating machines. It is noteworthy, that the literature on bearing misalignment is anyway rather poor. This fact is underlined by majority of authors of papers on the subject [1, 4, 9]. A. Muszynska in [4] stresses it very strongly: "There is, however very little published on misalignment malfunction, its destructive, overloading effects on rotor and bearings, physical phenomena involved, and how to diagnose misalignment by using vibration monitoring". All this can testify to some underestimation of the phenomenon, but also to certain difficulties in analysing it. This situation makes undertaking the research in this area more justified.

The present work aims at illustrating and analysing changes of bearing trajectory attributes, as a result of the appearance of the bearing misalignment defect. The analysis is based on the results of computer simulation of the malfunction in numerical model of the turbogenerator, as this kind of experiments on a real object is, of course, inadmissible. The illustration of effects has a form of complete sets of trajectories of journal centres of all bearings, being the response to a dislocation of one particular bearing. The article interprets relations between scale, location and direction of bearing misalignment on the one hand and shape and dimension of bearing trajectories on the other hand, basing on operational principles of a hydrodynamic bearing. It was revealed, that trajectories of bearings, which express dynamic state of bearings, might play role of an indicator of bearing misalignment in multi-support rotating machine. Specifications of relations between bearing dislocations and the machine state expressed by its vibrations can be treated as sets of diagnostic relations linking bearing misalignment defect with symptoms of the defect [4, 10-12]. The work indicates great potential of using trajectory patterns for diagnosing misalignment defects in rotating machines.

2. RELATIONS BETWEEN BEARING DISLOCATIONS AND BEARING LOADS

The way and scale to which a machine reacts to bearing misalignment depends, first of all, on the type and magnitude of misalignment, but also on many other factors, especially structure of the machine. These are, among others, relative positions of particular bearings, mechanical properties of the bearings, especially stiffness characteristics of the oil film as well as stiffness characteristics of the rotor [3, 4, 11, 13].

If a rotating shaft is supported in two bearings static reaction forces are strictly defined, as the mechanical system description is trivial. A shaft supported in 3 bearings theoretically is statically indeterminate, however the system is simple for analysis and the computational problem disappears after taking into account strains of the shaft and support structure. Considering stiffness and mass of the rotors and the oil film stiffness characteristics it is possible to determine the static shaft shape and load of the bearings. In this situation dislocation of any of the three bearings can be reduced to the same case: to equivalent dislocation of the middle bearing, which then allow to calculate bearing loads.

The situation becomes much more complicated if number of bearings increases. Lets consider the shaft line supported in 4 bearings shown schematically in Fig. 1. If all bearings are positioned on the properly designed catenary line, as shown in Fig. 1a, bearing load distribution and operation

conditions are optimal. Lets assume that bearing No. 2 (denoted as B2) is at certain circumstances relocated up (Fig. 1b). In this situation B2 gets additional load but at the same time neighbouring bearings are unloaded. If the B2 is relocated enough far off, the upper-half of the B1 and B3 can be loaded and also B4 is loaded additionally at its bottom. Fig. 1c presents a different case, when B3 is moved down far enough to load its upper half-shell. In this situation the B2 and B4 are loaded additionally on their low half-shells. Besides the low half-shell of the B1 is unloaded or even is loaded upper half-shell of this bearing. It is easy to note, that bearing load distributions are very similar in the two cases: relocation of the B2 up (Fig. 1b) and relocation of the B3 down (Fig. 1c).

These two cases are really not absolutely equivalent; in some extend for reason of bearing clearances and oil film stiffness characteristics, but first of all because the shaft axis in the base case (shown in Fig. 1a) is not a straight line. Preliminary displacement of the bearings 2 and 3 down with respect to the geodesic line results in lack of symmetry in the shaft shape after displacing the bearing 2 up and after displacing the bearing 3 down. Although the shaft shapes are not identical in the two cases, their effects on the bearing load distributions are very similar. Since the vibration effects can be also similar and this fact will be shown in par. 5. Degree of similarity of the effects depends also on homogeneity of shaft stiffness and on mutual bearing distances, anyway the observation is true in principle. The above analysis is adequate also in relation to horizontal bearing dislocations and is even easier because of lack of the gravitational shaft deflection. Corresponding conclusions regarding similarity of the effects, however with more limitations, can be drawn with reference to dislocations of the terminal bearings: B1 and B4.

3. INVESTIGATION PROCEDURE

The analysis presented in the paper is limited to displacing only two most vulnerable bearings in the machine, i.e. displacing bearings 5 and 6 with respect to their base position on the catenary. These bearings were selected because the turbo-set turned out to be very sensitive to misalignments of the two bearings. What is more, their misalignments are extremely dangerous, as the ranges of acceptable dislocations of these bearings are small. The graphical illustration of the analysis is limited to the effects of the two bearing dislocations in the horizontal direction to the right and to the left, and in the vertical direction up and down and additionally were limited to only one dislocation range in every of the four direction: to a maximum acceptable distance.

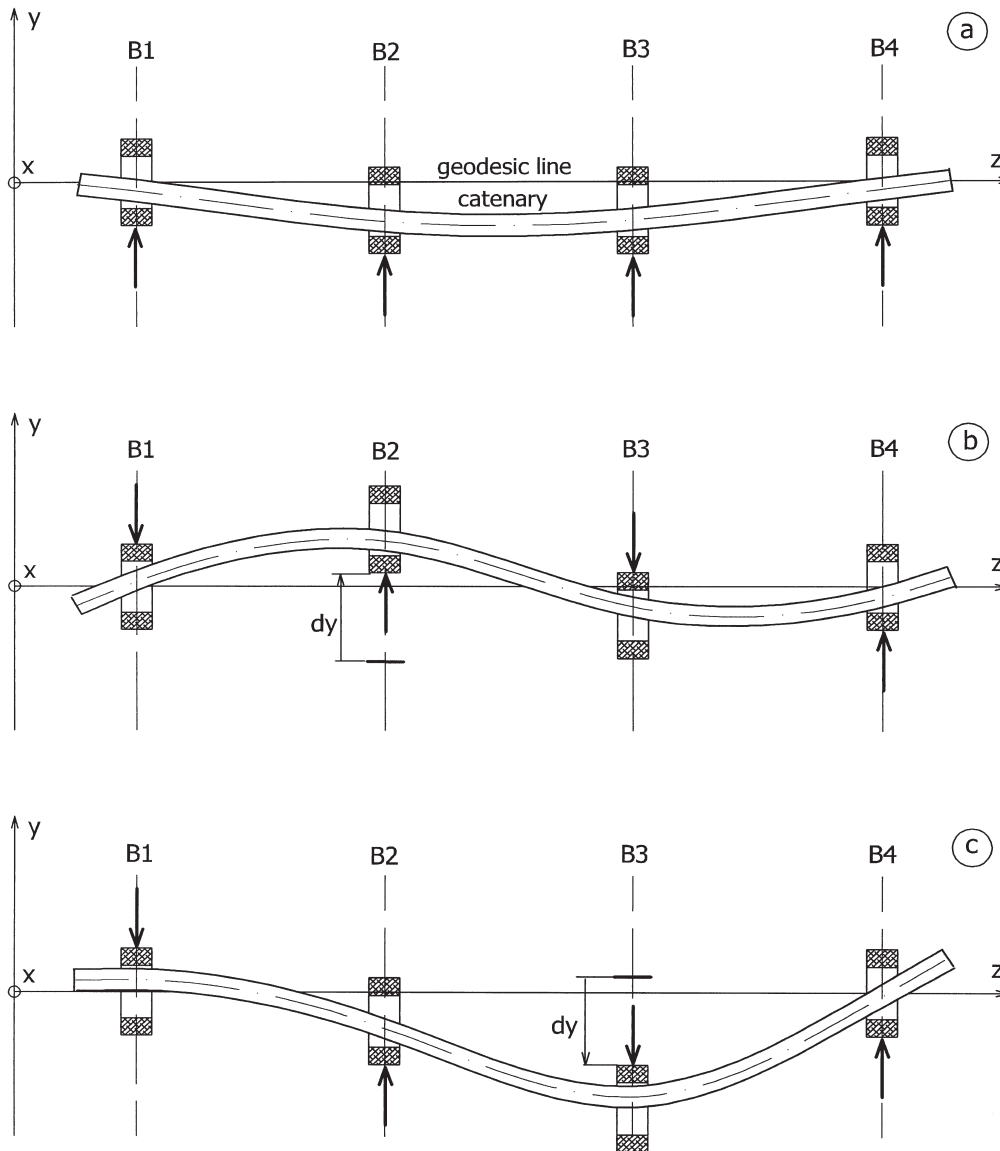


Fig. 1. Scheme of a shaft line bow and bearing loads depending on bearing dislocations:
 a – the bearings in catenary line; b – bearing 2 moved upward by dy ; c – bearing 3 moved down by dy

The ranges of acceptable bearing dislocations were determined taking into account the adopted criteria, which determine turbo-set operation as permissible [12, 14]. The criterion, which allows the state of the entire turbo-set to be assessed as permissible, is simultaneous meeting of two vibration conditions in all seven turbo-set bearings:

- relative journal-bush vibrations, expressed by the p-p displacement amplitudes A in two directions, inclined by 45° to the vertical:

$$A < A_{lim} = 165 \mu\text{m},$$

- absolute bearing vibrations, expressed by the RMS vibration velocities V in horizontal and vertical directions:

$$V_{RMS} < V_{RMS\ lim} = 7,5 \text{ mm/s}.$$

The limiting vibration parameters, values and directions of their measurements, were taken from ISO standards: relative vibration amplitudes from the standard 7919-2, while the absolute vibration velocities – from the standard 10816-2. The limits

correspond to the „warning state” for the turbo-set. It is noteworthy that the two conditions in the two directions are to be met in all seven bearings, and not only in the bearing in which the defect took place. The ranges of tolerable bearing dislocation in horizontal and vertical direction have been determined by author and are subject of the papers [15, 16]. The calculated values of tolerable misalignment of bearings 5 and 6 to the right, to the left, upward and downward are collected in Tab. 1.

All calculations were performed using a set of computer codes composing the system MESWIR. It is a package of codes developed and used in the Institute of Fluid-Flow Machinery for calculating dynamics of rotors supported on oil bearings. Its more detailed description and features are published e.g. in [10-12, 14, 15].

Table 1. The calculated maximum tolerable displacement ranges of bearings 5 and 6 in horizontal and vertical directions

No. of bear.	Direction of dislocation	Permissible dislocations $\Delta x, \Delta y$ [mm]	Vibration exceeded in:
5	right →	$\Delta x_R = 1.4005$	B 4 – dir. u
	left ←	$\Delta x_L = -1.0352$	B 6 – dir. u
	up ↑	$\Delta y_U = 0.8694$	B 4 – dir. u
	down ↓	$\Delta y_D = -1.0841$	B 5 – dir. u
6	right →	$\Delta x_R = 1.2918$	B 4 – dir. u
	left ←	$\Delta x_L = -1.7913$	B 4 – dir. u
	up ↑	$\Delta y_U = 1.2334$	B 5 – dir. u
	down ↓	$\Delta y_D = -0.8706$	B 6 – dir. u

The research consisted in calculating the dynamic state of a turbo-set numerical model with simulated dislocations of selected machine bearings, i.e. with a defect implemented to the base model of the machine. The essential calculations of the machine dynamic condition give at first the time-dependent dislocations of selected nodes, which then allow the trajectories of these nodes to be drawn. The trajectories are composed of instantaneous node positions and can be analysed with respect to their shape and dimensions.

The starting point for the calculations and for further analyses of the results is the “base case” – the numerical model of the machine free of defects [12, 14-16]. The model was created and tuned based on the results of measurements done in a power plant by turbogenerator diagnostic system in steady state, nominal conditions of operation. The data correspond to nominal rotational speed 3000 rev/min and full power output 211 MW. Tuning the numerical model consisted in selecting secondary machine properties and parameters, for instance, residual unbalance distribution, or material damping, in such a way that the results of calculations of the base case were the closest to the turbo-set characteristics measured in the power plant.

The object of examination was a large power turbogenerator 13K215, most commonly used in Polish power industry. The turbo-set consists of a 200 MW turbine and a generator. The turbo-set is a four-body machine, the rotors of which are supported on seven oil bearings. Four shaft segments are linked together by three rigid couplings. Noteworthy is distribution of two pairs of machine bearings, namely bearings 3 and 4 located between the MP and LP turbine parts, and bearings 5 and 6 located between the LP turbine case and the generator. The two bearings composing each pair are located close to each other and are supported on a common foundation block, as a result of which the dislocation of one bearing significantly affects the operation of both of them. Since the presented results and conclusions are

adequate for the pair of bearings 3 and 4, although the analysis in this paper is limited to the pair of bearings 5 and 6. All bearings are of hydrodynamic type, with elliptical clearance and two lubricating pockets in the horizontal division plane.

4. PRESENTATION OF THE MACHINE DYNAMIC STATE BY MEANS OF BEARING TRAJECTORIES

Figs 2 and 3 present tables containing sets of trajectories of journal centres of all bearings for the turbo-set with bearings dislocated from their base position and, for comparison, with bearings in the base position. Fig. 2 refers to misalignment of the bearing 5, while Fig. 3 refers to misalignment of the bearing 6. These diagrams are based on data recorded during 12 rotor revolutions. Successive rows in each table contain graphs of trajectories observed in seven consecutive turbo-set bearings. The first column in each table present a collection of trajectories for the base case, which characterise the state of the turbo-set without defects. They are a reference for further analysis. It results from Figs 2 and 3 that in the base case vibrations of journals and bushes in all bearings are much smaller than the permissible values stated in par. 3. Vibrations in a certain direction are represented by a projection of the trajectory on this direction. The columns 2, 3, 4, 5 present analogous trajectories for the cases when bearings are dislocated to the right, left, upward and down by maximum acceptable distance calculated with respect to permissible bearing vibration. The distances are collected in Tab. 1.

A general remark resulting from these diagrams is negligible effect of misalignment of bearings 5 and 6 on the trajectories of bearings 1, 2, and 3, and small effect on the trajectory of bearing 7. This tendency can be explained by a relatively large distance of those bearings from the displaced bearings, and relatively big shaft flexibility between those bearings. At the same time, the trajectories of bearings 4, 5, and 6 significantly increase in case of bearing 5 or bearing 6 dislocations in an arbitrary direction. A characteristic feature is that in the majority of cases the trajectories of those bearings stratify. In particular the bush trajectories do not coincide with each other during 12 shaft revolutions, which suggests significant contribution of subharmonic components and an analysis reveals that there are the 1/3X and 1/2X subharmonics. The bush vibrations are forced vibrations, which depend on the resonance frequency of the bush and supporting structure. The vibrations in the bearings are excited by forces generated by rotating shaft and by oil film reactions. More precise explanation of the nature of these vibrations requires more detailed analysis based on the vibration spectrum and modal analysis of the turbo-set structure but this problem will be the object of author's future investigations.

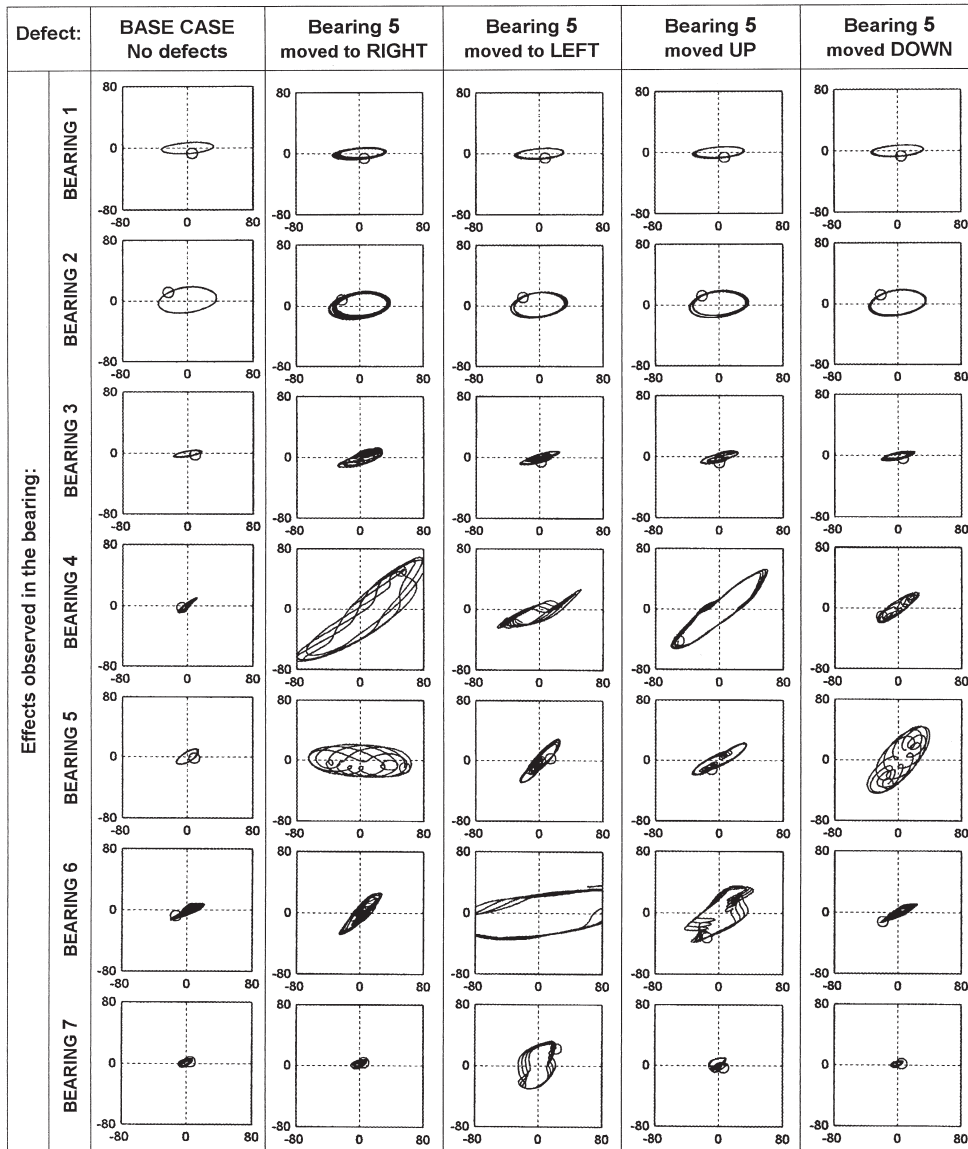


Fig. 2. Changes in relative trajectories of all bearing journals as the effect of dislocation of the bearing 5 by maximum tolerable distance in horizontal and in vertical direction

The diagrams in Figs 2 and 3 reveal that it is bearing 4, which is most sensitive to misalignment with respect to relative journal vibrations, although it is not the bearing to which the defect was introduced. It results from the Tab. 1 that in as many as 4 cases in this particular bearing took place exceeding of the permissible level of relative vibrations. At the same time the bush of this bearing does not get in high vibration. Bearing 5 behaves in an opposite way, characterised by moderate relative journal vibrations accompanied by very high bush vibrations in almost each case of misalignment of bearings 5 and 6. It is characteristic that in 75% of the examined cases the vibration limits are exceeded in the bearing next to the displaced one and not in this bearing itself.

5. INTERPRETATION OF BEARING TRAJECTORY FEATURES AND RELATIONS

Diagrams of the bearing journal trajectories shown in Fig. 2 and 3 reveal close symmetry of the effects of dislocation of the neighbouring bearings 5 and 6 to opposite directions. For instance, displacing bearing 5 down produces similar effect as displacing bearing 6 up. In both cases the pattern of vibrations observed in the two bearings is almost identical and exceeding of the permissible vibration level takes place in the same bearing 5. Also displacing bearing 5 right produces in both bearings identical effects as displacing bearing 6 left, and displacing bearing 5 left results in the same effects as displacing bearing 6 right. In the latter case exceeding of the permissible vibrations does not take place in the same bearing, and vibration amplitudes.

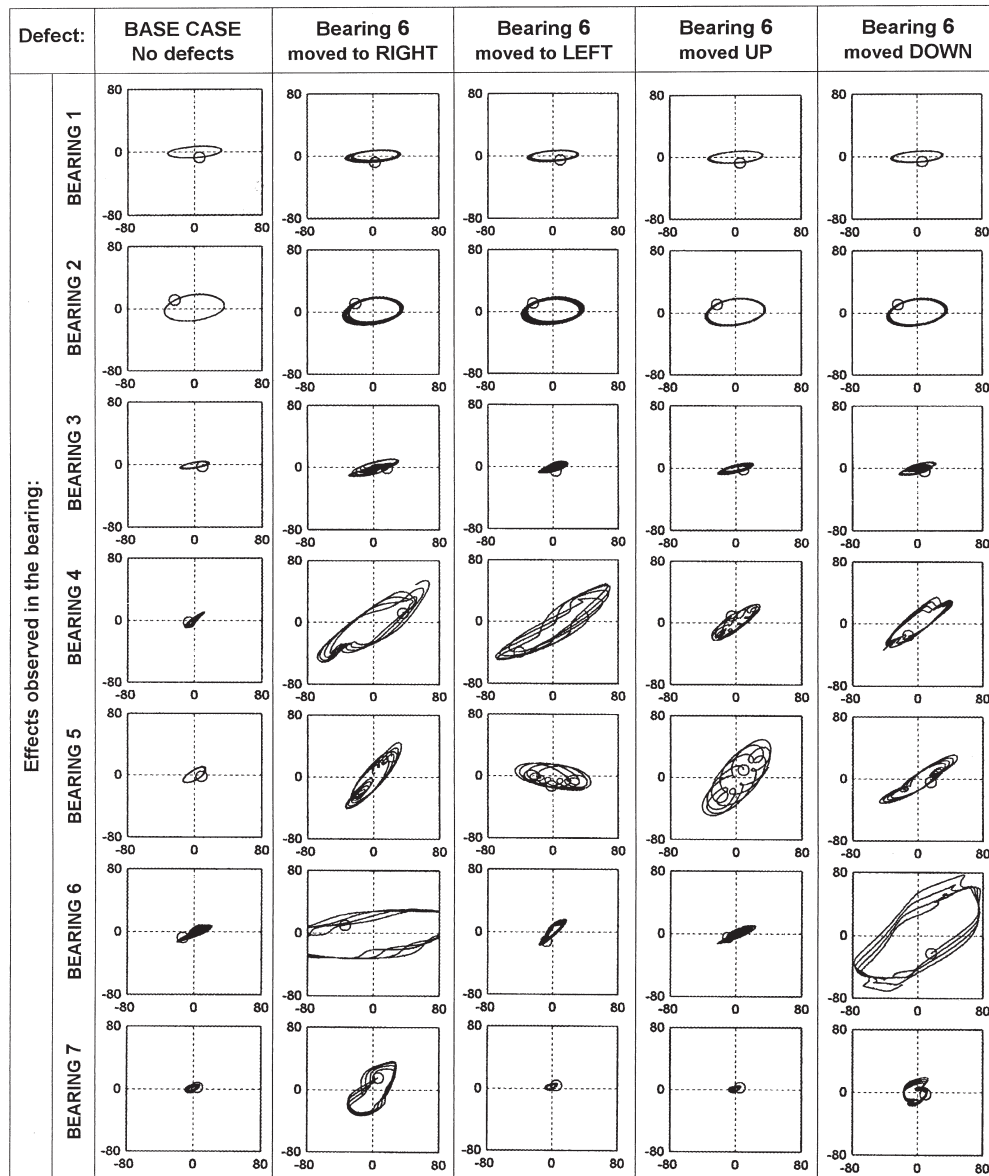


Fig. 3. Changes in relative trajectories of all bearing journals as the effect of dislocation of the bearing 6 by maximum tolerable distance in horizontal and in vertical direction

also differ, but still the diagrams of journal trajectories preserve geometrical similarity. Even in the pair of cases: dislocation of bearing 5 up and dislocation of bearing 6 down, we can observe certain similarities in the trajectory shape in every bearing, although the amplitude values of journal and bush vibrations differ considerably.

The described above similarity of the effects of dislocation of bearings 5 and 6 in opposite directions indicates that the operation of those bearings is closely related and cannot be analysed individually. This similarity also testifies to the same nature of phenomena taking place in the rotor-bearings system in the both cases. The opposite dislocations of the neighbouring bearings produce similar shaft deflection, and, as a consequence, similar change of bearing static load accompanied by similar rotor unbalance change, i.e. similar

dynamic load of the bearings. The symmetry in the bearing load and in the shaft deflection effects generated by displacing the neighbouring bearings to opposite directions was illustrated in the Fig. 1 and was discussed in par. 2.

The shape and dimensions of relative journal trajectories in particular cases of bearing misalignment can be explained basing on the hydrodynamic theory of lubrication and on the resultant mechanism of force generation in oil bearings [17]. Fig. 4 shows the scheme of a hydrodynamic bearing operation in the same coordinate system as implemented for the trajectories in Figs 2 and 3. The direction of rotor revolution (here, counterclockwise) makes the journal centre in the working bearing move to the right, as shown in Fig. 4. In these conditions the the bearing misalignment, which has the form of bush

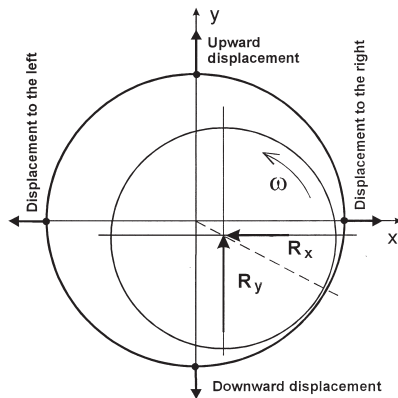


Fig. 4. Effect of the bearing bush dislocations on oil film reaction forces

displacement to the right, is equivalent to relative displacement of the bearing journal centre towards the bush centre. As a result, the main horizontal component of the bearing reaction decreases and the bearing has “more space” for horizontal vibrations. In this situation the neighbouring bearings must take over additional load from this unloaded bearing and it is the cause that horizontal journal vibrations in those bearings are likely to decrease.

The discussed situation is clearly illustrated by trajectories of bearings 5 and 6 after their horizontal dislocation. Displacing the bush of bearing 5 to the right (Fig. 2) removes some load from this bearing in horizontal direction, as a result of which its trajectory in horizontal direction enlarges. At the same time bearing 6 is given extra load as it takes over part of force previously carried by bearing 5, and, consequently, the dimensions of the journal trajectory of bearing 6 decreases. But displacing bearing 5 left pushes the journal towards the bush in this bearing, thus giving it extra load, which stabilises horizontal vibrations in this bearing and, at the same time, enlarges horizontally trajectory of the bearing 6 due to reduction of its load. Similar effects are generated while displacing bearing 6 (Fig. 3). Its dislocation to the right results in enlarging the trajectory of this bearing in the horizontal direction and stabilisation of horizontal vibrations in bearing 5. Similarly, displacing bearing 6 left leads to almost entire disappearance of vibrations in this bearing, accompanied by simultaneous horizontally enlarged journal trajectory in the neighbouring bearing 5.

Relations between the bearing trajectories and misalignment in vertical direction are similar, but even easier for interpretation. Fig. 2 reveals that displacing bearing 5 up extends the bearing trajectory in the horizontal direction and makes it narrower in vertical direction. At the same time dimensions of the journal trajectory of bearing 6 increase. Opposite effects are generated by moving bearing 5 down. The journal trajectory in bearing 5 becomes high, while that in bearing 6 - low. It is

quite understandable, as the dislocation of bearing 5 up gives it extra load, at the same time unloading the neighbouring bearings. The effects of vertical misalignments of the bearing 5 are shown in Fig. 3. Relations between trajectories and vertical misalignment of the bearing 6 can be interpreted in the same way as misalignment of the bearing 5.

6. UTILISATION OF BEARING TRAJECTORY FOR TURBO-SET DIAGNOSTICS

Understanding of the above-described mechanism of change of journal trajectory dimensions and shapes can be useful in using the trajectories for diagnosing large, multi-support and multi-rotor machines, in which particular rotors are linked by means of rigid couplings. The relations between bearing dislocations and machine vibration characteristics represented by bearing trajectories can be considered diagnostic relations and included to the base of diagnostic knowledge of the machine of concern. The concept of the present article was shown based on sample set of trajectories obtained after introducing to a machine the misalignment defect in the form of displacing two bearings by the maximum acceptable range. Similar data sets can be prepared also for misalignments introduced to bearings in an arbitrary range and even in arbitrary direction. Moreover, the analysis can be extended to dislocation of all bearings. However, it would result in great extending the amount of data and then increasing the base of knowledge of a diagnostic system.

Another limitation of possibility of diagnostics based on trajectories is the problem discussed in paragraph 6. There was found, that bearing trajectories and then vibration patterns are very similar in the two cases: dislocation of one bearing creating the pair in a certain direction and dislocation of the other bearing in opposite direction. The similarity results in situation that an attempt to identify a reason of observed vibration can lead to two alternative diagnoses: it can testify to the dislocation of one bearing in one direction, or to the dislocation of the other bearing in the opposite direction.

It is necessary to remember that the trajectory shape is not a characteristic symptom of misalignment of particular bearing. Therefore the diagnostics based on individual trajectories is hard or even not possible in practice. Chances for an accurate diagnosis increase with the number of available trajectories that can be analysed altogether. Also noteworthy is fact, that the presented defect-symptom type relations are strictly valid only for the machine for which the calculation model was worked out and tuned. However, certain remarks and conclusions can be generalised and applied to other similar machines. Such generalized deductions are set up in “conclusions”.

7. CONCLUSIONS

- The effect of misalignment of a bearing on trajectories of distant bearings is negligible, while the effect on trajectories of bearings located close is strong. In the majority of cases the permissible vibration level is exceeded not in the displaced bearing but in a neighbouring bearing.
- Symmetry exists between the effects of dislocations of two neighbouring bearings composing the pair in opposite directions. This testifies to the similarity of phenomena generated in the rotors-bearings system as a result of dislocation of two neighbouring bearings in opposite directions. This effect can be explained by similarity of bearing loads and resulting similar shaft deflection in the both cases.
- Mechanism of changes of journal trajectory shape and dimensions due to bearing misalignment can be explained based on the hydrodynamic theory of lubrication. Bearing trajectory patterns depend on direction and range of displacement and on relation of bearing displacement and direction of shaft revolution.
- Dimension and shape of a trajectory carry important information on the dynamic state of the machine: on the way in which the bearings are loaded and on the location and direction of bearing misalignment. A set of trajectories can be used for diagnostic purpose, however with limitations resulting from similarity of dislocation effects of neighbouring bearings to opposite directions.

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