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Diagnosing turbine set bearing dislocation defects based on cascade rundown characteristics

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Abstract

The article presents results of the vibration analysis of the rundown process of a large-power turbine set at the presence of the bearing lateral dislocation defect. Numerical calculations were carried-out, with the aid of a package of MESWIR codes, on a model turbine set in which the bearings had been dislocated by the maximum permissible range with respect to the kinetostatic catenary of rotors. The results are presented in the form of cascade diagrams of the relative and absolute vibrations of an examined bearing set vs. rotational speed during the machine rundown. The research makes it possible to formulate a system of diagnostic relations connected with the bearing lateral dislocation defect, which can make the knowledge basis for a diagnostic system. It has been found that the dislocations of certain bearings in certain directions may make vibrations go beyond permissible regimes during machine rundown, even in cases when the vibrations have been within the limits at nominal operating conditions. Determined were most vulnerable location points and dislocation directions of the bearings, along with accompanying symptoms.

Keywords: Machine diagnostics; Mechanical vibrations; Fluid-flow machine

1 Introduction

Big and responsible fluid-flow machines, in particular large-power turbine sets, usually consist of a number of rotors linked together by couples. The system of shafts formed in this way has to be supported in a number of slide bearings, so it is statically undeterminable from the mechanical point of view. For such a machine to operate safely the bearings have to be set in a precisely defined manner

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with respect to each other, i.e. particular shafts should create a line defined in the design state, approximately similar to catenary. Desired shape of a shaft line is formed by appropriate positioning of particular bearings in relation to a geodesic line, i.e straight, horizontal reference line. Unavoidable in practice deviations of bearing locations with respect to their design location can result from assembly errors, thermal dilatations of bearing pedestals and foundation, ground settlement, and possible failures. Moving any bearing in the turbine set from its basic location, determined by the design shaft catenary, changes operating conditions in other slide bearings, thus also altering the operating conditions of the entire machine. The static load of the shafts and bearings is changed, which leads to the change in the dynamic status of the machine. As a consequence, vibrations of bearings and rotors can be generated [1-5]. This results in change rundown characteristics of the machine. That is why bearing vibrations, among other sources, can deliver information on the presence of bearing lateral dislocation defect in the turbine set.

The motivations for examining the effect of bearing dislocation defect on turbine set rundown vibration characteristics originated from the earlier research activities in the subject, which resulted in determining permissible displacements of bearings in the 13K215 turbine set from the point of view of the acceptable status of the turbine set. Exceeding the permissible bearing dislocations leads to inevitable stop of the machine, as its technical condition makes further operation impossible [2, 6-8]. The reason for making decision about this type of detailed analysis of the impact of bearing dislocation on turbine set characteristics in transient states is the following.

Let us assume that an arbitrary bearing in the multi-shaft fluid-flow machine has been dislocated from the designed optimal kinetostatic catenary. A possible reason can be the failure of bearing, its pedestal, or foundation, as well as thermal dilatations of those elements. Displacing any bearing in the machine in operation cannot exceed permissible values; otherwise it results in immediate machine stop. Therefore let us assume that the current displacement is equal to the maximal permissible dislocation, which leads to machine stop with resultant rundown. The knowledge on possible vibrations, to be recorded during the rundown process, is of high importance. The operating staff should know whether those vibrations are likely to exceed permissible values at certain speeds during the rundown, which may happen even if the vibrations have been within permissible ranges at nominal rotational speed.

The investigations consisted in calculating static and dynamic states of a large-power turbine set, with bearing lateral dislocation defect simulated using a set of MESWIR series computer codes. Introducing the defect to the basic model of the machine consisted in dislocating particular bearings by the maximal permissible range, a priori calculated taking simultaneously into account the vibration and bearing load criteria [7, 8]. The calculated vibrations of the bearing centres in the turbine set during its rundown were shown in the form of vibration cascade diagrams, prepared individually for each bearing and for each state of possible limiting displacements. Basing on the cascade diagrams the collective diagrams of maximal amplitudes of the vibrations vs. rotational speed were created. Detailed observations concerning the effect of dislocation of particular bearings on the dynamic state of the turbine set have been collected in Tables. The cascade diagrams and the maximum vibration diagrams made the basis for a detailed analysis of the relations between the dislocations of particular bearings and the rundown characteristics of the turbine set, and then for further conclusions of more general nature.

It is noteworthy that the relations between bearing dislocations and the dynamic and kinetostatic state of the machine, expressed by bearing vibrations and loads, make in fact a set of relations linking the defects and their symptoms. Therefore the rundown characteristics can be considered diagnostic relations, and included to the diagnostic knowledge basis for the examined machine. Moreover, some of these relations can be generalised to be used for a class of similar machines.

2 The range and object of investigations

2.1 Research goals

The object of investigations was a 13K215 turbine set, in operation in one of Polish power plants. Turbine sets, consisting of 200 MW turbines and generators, are in common use in Polish power engineering. A general diagram of such a turbine set, presenting the system of shafts and locations of bearings, is given in Fig. 1. The turbine set is a four-body machine, whose rotors are supported in seven slide bearings. Four shaft sections are linked together using three couples. Noteworthy are the locations of two pairs of bearings, namely bearings Nos. 3 and 4 between MP and LP turbines, and bearings Nos. 5 and 6 between LP turbine and generator. Each two bearings constituting those pairs are located close to each other and are supported by a common pedestal. This factor will be taken into account during the discussion of the obtained results.

The investigations focused on analysing the dynamic state of the turbine set at the presence of the bearing lateral dislocation defect in unsteady conditions, in particular during machine start-up and rundown periods. In real execution, due to the method of operation of the main tool used for the calculations, the start-up and rundown processes were modelled by series of steady states generated with proper step within the entire range of the rotational speed of the

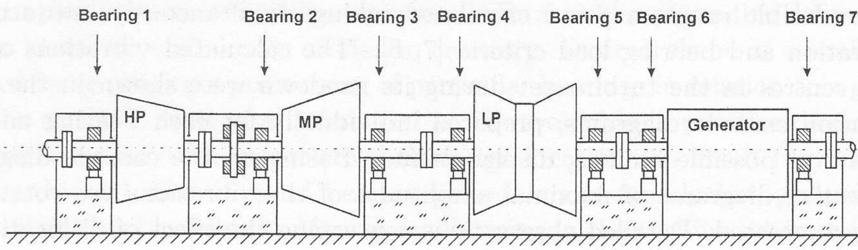

Figure 1. Illustration of the rotor line supported in slide bearings in the 13K215 turbine set.

machine. Therefore the results may contain some miscalculations, as the factor connected with shaft acceleration in the equations of a shaft movement has not been taken into consideration. However the errors are very small because the angular acceleration of the shaft of the heavy machine like a turbine set during its run down is relatively small. Thus time intervals of the vibration phenomena are very short in comparison to the intervals of time necessary for significant change of rotational speed.

Maximal permissible dislocations of all seven bearings in four directions: right, left, up and down, determined in earlier research activities, were assumed as the defect in the investigations [7, 8]. Analysed were machine responses to 28 elementary defects simulated in the above manner. The symptoms of each defect were relative and absolute vibrations of all bearings in vertical and horizontal directions.

2.2 Machine state evaluation criteria

Large and responsible fluid-flow machines are operated using extremely severe procedures oriented on avoiding possible threat of any failure. All parameters determining the thermal or dynamical state of the turbine set must be kept within permissible limits. The dynamical status is characterised by dislocation amplitudes of the relative vibrations of the bearing journals and the absolute bearing vibration velocities [2, 3, 9, 10]. The range of permissible vibrations is standardised for a given type of machine. For the purpose of the present work it was assumed that the qualification criterion that allows considering the work status of the entire turbine set acceptable is simultaneous fulfilment of the following three conditions for all seven turbine set bearings:

- permissible relative journal-bushing vibrations, limited by the amplitudes of p-p dislocations in two mutually perpendicular directions: $s < s_{gr} = 165\,\mu\text{m}$,

- permissible absolute bearing vibrations, limited by RMS vibration velocities in horizontal and vertical directions: $v_{RMS} < v_{RMSgr} = 7{,}5$ mm/s,
- permissible bearing load, limited by the average pressure on the bearing shell surface: $p < p_{dop} = 2$ MPa.

The limiting values were taken from ISO standards: for relative vibration dislocations – from ISO 7919-2, and for absolute RMS vibration velocities – from ISO 10816-2. These values correspond to the warning states. The limiting pressure distributions on the bearing shell surface were patterned on the values calculated by the turbine designers for the constructions of large slide bearings.

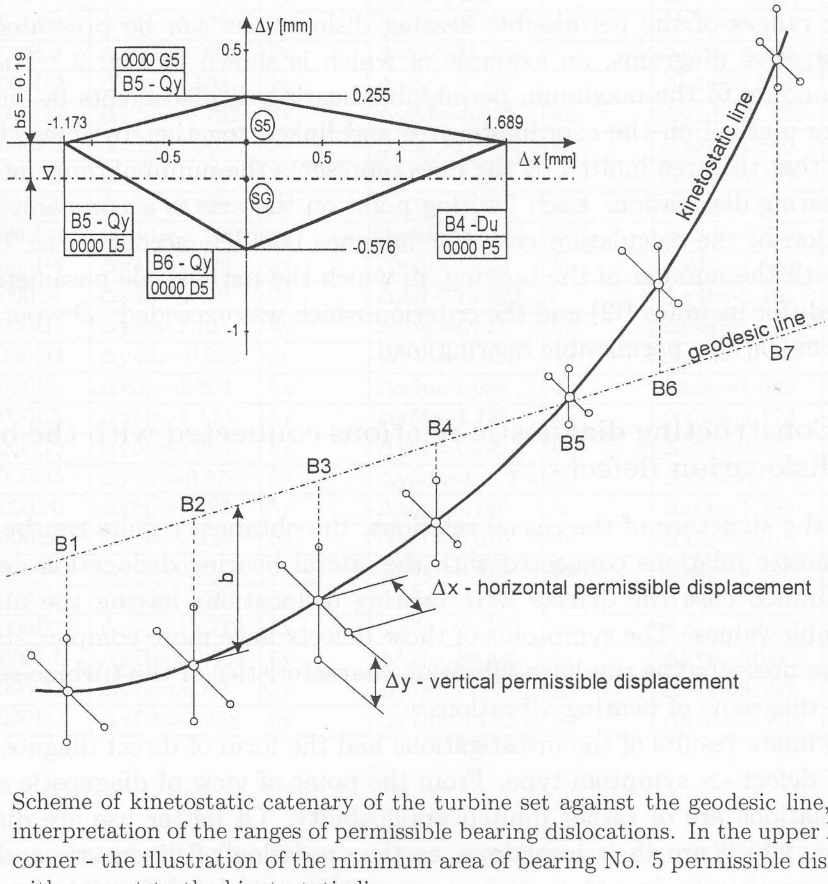

Figure 2. Scheme of kinetostatic catenary of the turbine set against the geodesic line, and the interpretation of the ranges of permissible bearing dislocations. In the upper left-hand corner - the illustration of the minimum area of bearing No. 5 permissible dislocations with respect to the kinetostatic line.

On the basis of the above assumptions, the ranges of permissible dislocations of the turbine set bearings were determined. The methodology and results of these calculations were presented in [7, 8], among other references. The concept of determining permissible bearing dislocations is illustrated in Fig. 2. It is noteworthy that the permissible displacements Δx, Δy refer to the basic positions of the bearings along the catenary kinetostatic line, which is displaced by b with respect to the geodetic line in bearing locations. The part of those results which was used in the present article has been collected in Table 1 presenting permissible dislocations of all seven bearings in four directions separately taking into account the permissible vibration and permissible bearing load criteria. The last column presents the values of the permissible dislocations with regard to the two criteria taken into account simultaneously. In each row, it is the lowest value from the two previous columns (calculated with regard to two separate criteria, vibration and load) as it is the actual limit value, which guarantees save operation of the entire machine.

The ranges of the permissible bearing dislocations can be presented in the demonstrative diagrams, an example of which is shown in Fig. 2. The points corresponding to the maximum permissible bearing displacements in four directions are marked on the coordinate axes and linked together by lines. One can assume that the area limited by the lines represents the minimal range of permissible bearing dislocation. Each limiting point on the axis is accompanied by the denotation of the calculation case (for instance 0000P1, according to Table 1), along with the number of the bearing, in which the permissible parameters were exceeded (for instance B2) and the criterion which was exceeded: D – permissible vibrations, or Q – permissible bearing load.

2.3 Constructing diagnostic relations connected with the bearing dislocation defect

Due to the structure of the casual relations, the obtained results can be treated as diagnostic relations connected with the lateral bearing dislocation defect. In the examined case the defects were bearing dislocations having the maximum permissible values. The symptoms of those defects have more complex structure. They are presented as rundown vibration characteristics of the turbine set in the cascade diagrams of bearing vibrations.

Proximate results of the investigations had the form of direct diagnostic relations of defect -> symptom type. From the point of view of diagnostic systems, these relations are of rather limited applicability. Of better use are diagnostic relations, which are their inversions, as the operation of diagnostic and expert systems consists in formulating diagnosis on the type of defect on the basis of observed symptoms. At the same time, experiments, simulations, and observations only deliver, as a rule, relations of defect -> symptom type. Obtaining diagnostic relations of symptom -> defect requires reversing the machine model. In case of one-value and one-to-one relations the process of reversing the model

Table 1. Permissible bearing dislocations due to two criteria taken into account separately and together

No.	Case mark	Criterion of maximum dislocation calculations:					
		Permissible pressure		Permissible vibrations		Together: pressure+vibrations	
		delta [mm]	exceeded in bearing	delta [mm]	exceeded in bearing	delta [mm]	exceeded in bearing
1	0000P1	$\Delta x1p=13.765$	2x	$\Delta x1p>389.239$	-	$\Delta x1p=13.765$	2x - Q
2	0000L1	$\Delta x1l=-14.558$	2x	$\Delta x1l=-248.681$	4v	$\Delta x1l=-14.558$	2x - Q
3	0000G1	$\Delta y1g=3.943$	3y	$\Delta y1g=83.694$	4u	$\Delta y1g=3.943$	3y - Q
4	0000D1	$\Delta y1d=-12.277$	2y	$\Delta y1d<-259.43$	-	$\Delta y1d=-12.277$	2y - Q
5	0000P2	$\Delta x2p=5.170$	3x	$\Delta x2p=20.018$	4v	$\Delta x2p=5.170$	3x - Q
6	0000L2	$\Delta x2l=-5.392$	2x	$\Delta x2l<-51.258$	-	$\Delta x2l=-5.392$	2x - Q
7	0000G2	$\Delta y2g=4.503$	2y	$\Delta y2g>76.887$	-	$\Delta y2g=4.503$	2y - Q
8	0000D2	$\Delta y2d=-0.732$	3y	$\Delta y2d=-6.381$	4u	$\Delta y2d=-0.732$	3y - Q
9	0000P3	$\Delta x3p=1.860$	3x	$\Delta x3p>17.086$	-	$\Delta x3p=1.860$	3x - Q
10	0000L3	$\Delta x3l=-1.409$	3x	$\Delta x3l=-3.387$	4v	$\Delta x3l=-1.409$	3x - Q
11	0000G3	$\Delta y3g=0.209$	3y	$\Delta y3g=1.080$	4u	$\Delta y3g=0.209$	3y - Q
12	0000D3	$\Delta y3d=-1.229$	4y	$\Delta y3d=-12.981$	5u	$\Delta y3d=-1.229$	4y - Q
13	0000P4	$\Delta x4p=1.699$	3x	$\Delta x4p=2.649$	5u	$\Delta x4p=1.699$	3x - Q
14	0000L4	$\Delta x4l=-2.101$	3x	$\Delta x4l<-19.222$	-	$\Delta x4l=-2.101$	3x - Q
15	0000G4	$\Delta y4g=1.121$	4y	$\Delta y4g=4.127$	5u	$\Delta y4g=1.121$	4y - Q
16	0000D4	$\Delta y4d=-0.219$	3y	$\Delta y4d=-0.881$	4u	$\Delta y4d=-0.219$	3y - Q
17	0000P5	$\Delta x5p=1.804$	5x	$\Delta x5p=1.689$	4u	$\Delta x5p=1.689$	4u - D
18	0000L5	$\Delta x5l=-1.173$	5y	$\Delta x5l=-3.788$	5x	$\Delta x5l=-1.173$	5y - Q
19	0000G5	$\Delta y5g=0.255$	5y	$\Delta y5g=1.290$	6u	$\Delta y5g=0.255$	5y Q
20	0000D5	$\Delta y5d=-0.576$	6y	$\Delta y5d=-1.212$	5u	$\Delta y5d=-0.576$	6y - Q
21	0000P6	$\Delta x6p=1.395$	5y	$\Delta x6p=3.746$	5x	$\Delta x6p=1.395$	5y - Q
22	0000L6	$\Delta x6l=-2.124$	5x	$\Delta x6l=-2.016$	4u	$\Delta x6l=-2.016$	4u - D
23	0000G6	$\Delta y6g=0.627$	6y	$\Delta y6g=1.499$	5u	$\Delta y6g=0.627$	6y - Q
24	0000D6	$\Delta y6d=-0.296$	5y	$\Delta y6d=-1.338$	6u	$\Delta y6d=-0.296$	5y - Q
25	0000P7	$\Delta x7p=19.772$	6x	$\Delta x7p=27.869$	5x	$\Delta x7p=19.772$	6x - Q
26	0000L7	$\Delta x7l=-20.662$	5y	$\Delta x7l=-39.445$	5x	$\Delta x7l=-20.662$	5y - Q
27	0000G7	$\Delta y7g=3.093$	5y	$\Delta y7g=11.551$	6u	$\Delta y7g=3.093$	5y - Q
28	0000D7	$\Delta y7d=-5.358$	6y	$\Delta y7d=-14.854$	5u	$\Delta y7d=-5.358$	6y - Q

can be automated with the aid of relevant methods and algorithms [11, 12]. As the present stage of development of those methods, such automation seems practically impossible in the examined case, in which the symptoms have the form of complicated vibration spectrum distributions, changing with changes of rotational speed of the rotor. What is only possible is visual comparison of changes in vibration pattern against the basic case, using the cascade diagrams. The work reported in the present article aimed at, among other goals, creating a set of diagnostic relations of symptom – defect type on the basis of the cascade diagrams and on the basis of the maximal vibration diagrams prepared from the cascade diagrams.

2.4 Methodology of calculations

The calculations were carried out using a set of MESWIR series computer code system developed and used in IFFM, Gdańsk, for non-linear analysis of rotors [13, 14, 15]. Three codes of the system were used for simulation calculations. They were:

- KINWIN-60 – calculations of kinetostatics of a rotor supported in slide bearings, assuming a diathermic model of oil film in the bearings,
- KINWIN-I-LEW – calculations of kinetostatics of a rotor, assuming isothermal model of oil film in the bearings and taking into account the presence of pressure pockets on bushing surface,
- NLDW-75-LEW – calculations of rotor dynamics; among others the results describe motions of particular nodes of the rotor-bearings system, thus making the database for preparing cascade diagrams.

These codes were started sequentially in the above order, in a controlled loop. Due to huge number of calculations to be carried out, with the resulting need for preparation of correspondingly high number of data files and processing a similar number of files with calculation results, the entire process had to be automated. The calculation control and file management were made with aid of auxiliary codes written in FORTRAN programming language. Processing of measurement results, their further analysis, and preparation of cascade diagrams and maximum vibration diagrams were carried out using scripts written in MATLAB environment [16, 17]. The basic calculations were made on the 256-processor "holk" computer cluster, of high calculating power, in use at the TASK Computer Network in Gdańsk.

The algorithm of calculations was the following: to the model turbine set, defined in the basic case [5], a defect was introduced in the form of bearing dislocation by the scale equal to maximum permissible dislocation taken from Table 1 [7, 8]. With the aid of a code written in the FORTRAN environment, the calculations were started for a series of rotation speed values ranging from 1000 rev/min to 3000 rev/min and differing by a 50-rev/min step. Thus, 41 elementary cases differing by rotational speed values were calculated for each of 28 bearing dislocation defects.

Since the performed analysis was of comparable nature, it was necessary to prepare a reference case for analysing the effects of the bearing dislocation defect on the machine rundown characteristics [14]. The basic case was represented by a defect-free numerical model of the turbine set, in which all bearing centres were located along the kinetostatic catenary. This case was prepared and tuned using the data recorded by the diagnostic system DT200 working on the 13K215-type turbine set operating in the Kozienice Power Plant. The data were recorded for the turbine set in continuous operation at nominal rotational speed equal to 3000 rev/min., real power equal to 211.25 MW, reactive power equal to 73.75 MW and phase factor $\cos\varphi = 0.94$ [18]. Tuning of the model consisted in selecting second-rate characteristics and parameters of the machine to get the basic case calculations as close as possible to the measured parameters of the real machine [14].

3 Analysis of the results

3.1 Presentation of the results

The effect of maximum permissible bearing dislocations on the pattern of turbine set vibrations during its rundown was analysed with respect to all possible individual dislocations. In practice, this analysis was of significant value mainly to bearings Nos. 3-6. Bearings Nos. 1, 2, and 7 are distant from each other and from other bearings. Moreover, they are located in separate cases and they support relatively long and slender shafts. As a result, the ranges of permissible dislocations of those bearings were so large that in operational practice such dislocations can perhaps happen during a failure, but almost never due to thermal and/or assembly effects. At the same time the ranges of permissible dislocations of bearings 3-6 were at the realistic level of about 2 mm in horizontal direction and even fractions of millimetres in vertical direction.

The effect of the bearing dislocation defect on the machine vibration pattern in unsteady and transient conditions can be assessed in the most comprehensive way using cascade diagrams. The frequency distribution of vibration amplitudes, shown in these diagrams, facilitates formulating conclusions on possible origins of the vibrations and applicable methods of their eliminations. Making use of the codes used for drawing spectrum diagrams [16] and diagnostic cards [17], additional codes were prepared for automatic generation of the cascade diagrams from the results of calculations performed by the NLDW code. Each defect symptom was presented on an individual cascade diagram, therefore the set of symptoms consisted of 784 cascade diagrams. In practice, each diagram comprised a set of 41 amplitude spectrum diagrams, which were the diagrams of vibration amplitudes vs. frequency for digitised values of rotational speed within the range from

1000 to 3000 rev/min with 50-rev/min resolution. The figure prepared for each bearing includes four cascade diagrams presenting:

- relative journal-bushing vibrations in horizontal direction,
- relative journal-bushing vibrations in vertical direction,
- absolute bearing vibrations in horizontal direction,
- absolute bearing vibrations in vertical direction.

The interpretation of such a big number of causal relations in order to create a system of diagnostic relations is extremely laborious. In order to simplify this work, from the cascade diagrams prepared were maximum spectrum amplitude diagrams as a function of the rotational speed. These diagrams lose the information on vibration frequency, but allow a preliminary assessment whether the vibrations recorded during the rundown are kept within the standards. Additionally the diagrams allow the selection of cases of certain significance for further detailed analysis performed with the aid of the cascade diagrams.

As it is not possible to present all prepared diagrams only a set of selected examples is shown here to illustrate the conclusions formulated and to discuss the diagnostic relations in the further part of the article. The diagrams show the vibrations of bearings Nos. 5 and 6, as resulting from the dislocation of bearings Nos. 4, 5, and 6 to the left. The corresponding results calculated for bearings No. 4 and 6 in the basic case are included for comparison. The maximum vibration amplitudes vs. rotational speed are given in Figs. 3-6, out of which Fig. 3 refers to the basic case, and Figs. 4, 5, 6 to the dislocations of bearing Nos. 4, 5, and 7, respectively. The cascade diagrams are shown in Figs. 7-14. Each of them presents a set of four diagrams:

- Figs. 7 and 8 refer to the basic case and present vibrations of bearings Nos. 5 and 6, respectively,
- Figs. 9 and 10 illustrate the effect of bearing No. 4 dislocation to the left on the vibrations in bearings Nos. 5 and 6,
- Figs. 11 and 12 illustrate the effect of bearing No. 5 dislocation to the left on the vibrations in bearings Nos. 5 and 6,
- Figs. 13 and 14 illustrate the effect of bearing No. 7 dislocation to the left on the vibrations in bearings Nos. 5 and 6.

A comparison analysis of similar diagrams for all bearings and all cases of dislocation, between each other and with the basic case, allowed making observations collected in Tables 2-5, and formulating conclusions. Table 2 presents the

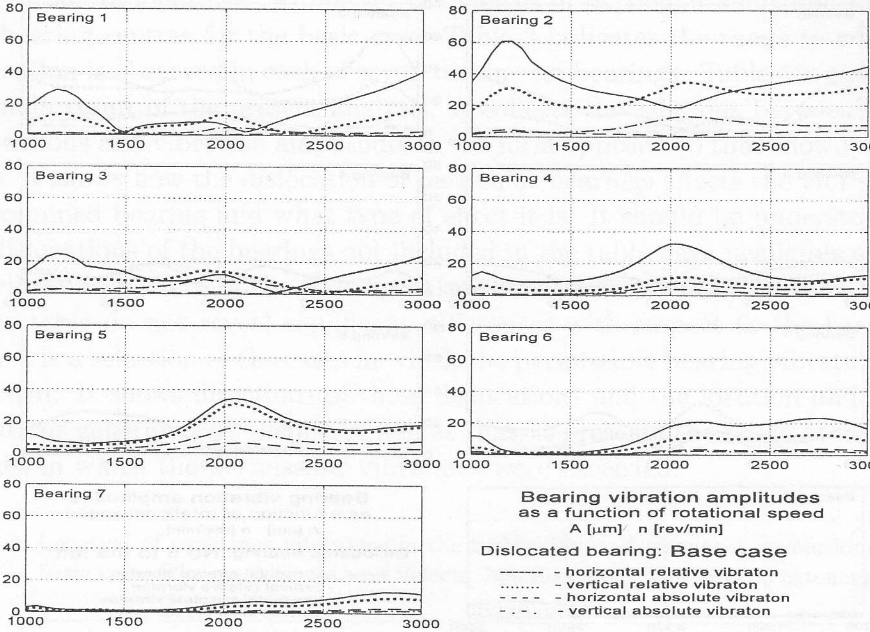

Figure 3. Maximum vibration amplitudes for basic case.

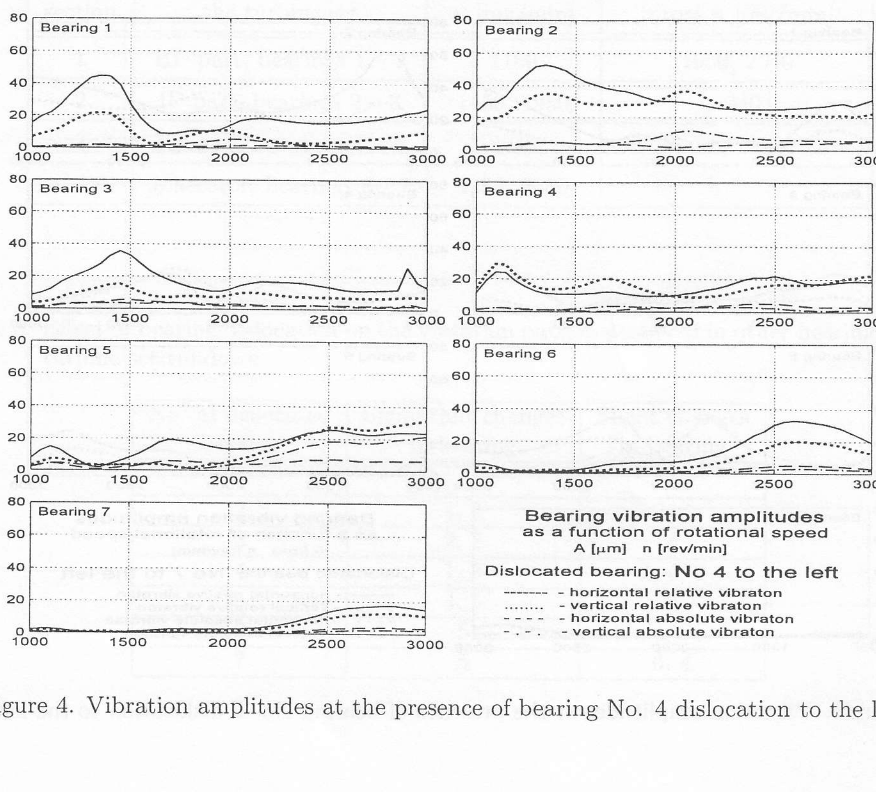

Figure 4. Vibration amplitudes at the presence of bearing No. 4 dislocation to the left.

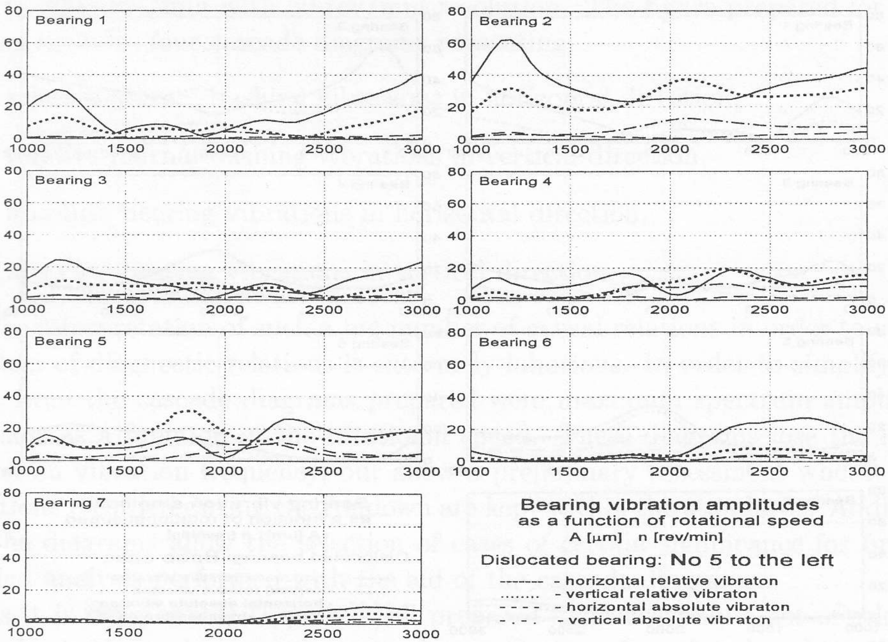

Figure 5. Vibration amplitudes at the presence of bearing No. 5 dislocation to the left.

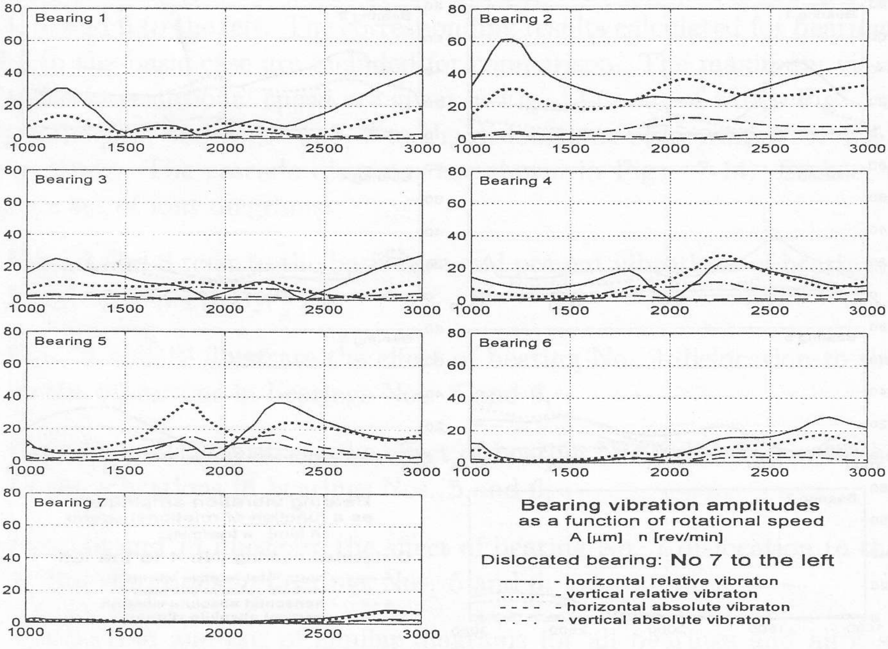

Figure 6. Vibration amplitudes at the presence of bearing No. 7 dislocation to the left.

distribution of identified turbine set resonances in particular shaft line segments and bearing centres for the basic case. Table 3 indicates the range to which the dislocation is observed in each of seven turbine set bearings. Table 4 is considered the main result of the present analysis. It collects the relations between bearing dislocations and vibration amplitudes in the form opposite to that shown in Table 3, i.e. it shows how the dislocation of particular bearings affects the vibrations of an examined bearing and what type of effect it is. It should be understood that the dislocations of the bearings not included in the table have negligible effect on the vibration pattern of the bearing. Also the vibration types that were omitted in the table do not reveal significant differences with respect to the basic case. Table 5 is a selection of the cases in which the permissible bearing vibrations were exceeded. It shows directions of those dislocations and the location and type of vibrations generated by them. As well as that, it presents the range of rotational speeds, in which the permissible vibrations were exceeded.

Table 2. Location of resonance vibrations in the turbine set and resonance revolutions for the basic case. The machine without defects, bearings along kinetostatic catenary.

No. of rotor section	Location of vibrations in the turbine set	Rotor section resonance n [rev/min]	Reflection of resonances of adjacent rotors n [rev/min]
1	HP part, bearings 1 – 2	1160	1950, 2040
2	IP part, bearings 2 – 3	1450, 1950	2040
3	LP part, bearings 4 – 5	2040	
4	generator, bearings 6 - 7	2200, 2800	

Table 3. Effect of bearing dislocation on the vibration pattern observed in other bearings during turbine set rundown.

No. of dislocated bearing	Significant changes in bearing	Slight changes in bearing
1	1, 2, 3	4
2	1, 3	2, 4
3	2, 3	4
4	1, 3	2, 4, 5
5	4, 5	6
6	5	4, 6
7	5	6, 4

Table 4. Relations between vibration pattern of bearings in column 1 and dislocations of bearings in column 2 during turbine set rundown – obtained from maximal vibration diagrams and from cascade diagrams.

Bearing No.	Directions of bearing dislocations:	Effect of bearing dislocations (listed in column 2) on the vibration pattern observed in bearing listed in column 1
1	1 right, 2 left	– increase of vibrations within the range of $n = 1300 - 2000$; strong resonance at $n = 1700$; decay of resonance at $n = 1160$ and $n = 2040$
	1 left, 2 right	– increase of resonance at n=2040; decay of resonance at $n = 1160$; complete disappearance of vibrations at $n > 2200$
	1 down, 2 up	– increase of relative horizontal vibrations
	1 left, 2 right	– stabilizes vibrations in the entire range of rotational speeds
	2 up	– strong resonance 1/2X at $n = 2200$; resonance 1X for $n = 1160$
	3 left	– stabilizes vibrations at the entire range of n
	3 right	– generates vibrations 2X within the range of $n = 1300 - -2600$
2	1 up	– strong relative horizontal vibrations 1/2X, 3 times as big as 1X within the range of $n = 2500 - -2800$; vibrations exceed permissible levels within the range of $n = 2300 - 2700$
	1 and 2 horizontally	– suppresses resonance at n=1160; intensifies resonance within the range of $n = 1500 - -2000$; decreases vibrations at n>2400
	1 right, 2 left	- generates resonance at $n = 1900$
	1 left, 2 right	– generates resonance at n=1700
	2 and 3 vertically	– reduces vibrations within the entire range of rotational speed
3	1, 2, 3 horizontally	– suppresses vibrations at $n = 1160$; generates or intensifies vibrations within the range of $n = 1500 - 2000$; significantly reduces vibrations at $n > 2400$
	1 right, 1 and 2 left	– generates resonance at $n = 1700$
	2 right, 3 left	- suppresses vibrations in all directions
	1 down, 2 up	– significantly intensifies resonance at $n = 1160$
	1 up	– generates strong relative horizontal vibrations 1/2X within the range of $n = 2500 - 2800$
	4 left and up	– generates increased vibrations in the vicinity of n=1400

continuation of Tab. 4

4	4 horizontally 5 and 7 right, 6 left 2, 4, 6 right, 3, 5, 7 left 5 left, 6 right 5 right	– cancels resonance at $n = 2400$ – generates stronger vibrations at $n > 2500$ – reduces vibrations and cancels resonance at $n = 2000$ – changes in horizontal vibrations; suppresses resonance at $n = 2040$; generates resonance at $n = 1750$ and at $n = 2300$ – generates strong vibrations; at n>2700 makes permissible relative horizontal and vertical vibrations 1/3X, 1/2X, 2/3X to be exceeded
5	5 and 7 right, 6 left 5 and 7 left, 6 right 5 right 5 left, 6 right 6 left, 7 right 2 right, 3 left 3 left 3 down 4 right 4 left	– increase of vibrations at $n > 2500$; cancels resonance at $n = 2000$ – slightly reduces amplitude of vibrations – generates strong vibrations; at $n > 2700$ makes permissible relative horizontal and vertical vibrations 1/3X, 1/2X, 2/3X to be exceeded – changes in horizontal vibrations; suppresses resonance at $n = 2040$; generates resonance at $n = 1750$ and $n = 2300$ – increase of vibrations within the range of $n > 2500$; at $n = 2650$ generates resonances of all types of vibrations – suppresses resonance for $n = 2040$ – generates resonance at $n = 1500$ – increases absolute vertical vibrations of the bushing at $n = 2040$ – resonance decay at $n = 2040$ and appearance at $n = 1450$ for relative vertical and horizontal vibrations – generates relative and absolute vibrations for $n = 2300–3000$
6	4 and 5 horizontally 5 right 6 left, 7 right	– cancels resonance for $n = 2200$ and for $n = 2800$; induces relative vibrations at $n = 2600$ – increases relative vibrations for $n > 2600$ and induces resonance at $n = 2950$ with frequencies 1/3X and 2/3X – resonance decay at $n = 2200$ and intensification at $n = 2800$
7	4 left 5 right	– increased amplitude of relative vibrations at $n = 2800$ – generates slight absolute and relative horizontal and vertical 1/2X and 2/3X vibrations at $n > 2800$

Table 5. Reason and place where the permissible bearing vibrations were exceeded during the turbine set rundown, at the presence of defect of rotor line dislocation.

No. of bearing and dislocation direction	No. of bearing and nature of vibrations which were exceeded in that bearing	Range of dangerous rotational speed
No. 1 upward	No. 2, relative horizontal	n = 2300 – 2700 rev/min
No. 2 upward	No. 1, relative horizontal	n = 2100 – 2400 rev/min
No. 4 right	No. 2, relative horizontal	n = 2550 rev/min
No. 5 right	No. 4 and No. 5, relative horizontal and vertical	n > 2700 rev/min
No. 6 left	No. 4, relative, exceeded in direction of bisector of the coordinate system	n >2500 rev/min

3.2 Turbine set rundown characteristics for the basic case

Below presented are the characteristics of the basic case, prepared under an assumption that the machine does not reveal defects. These make the reference material for further comparison study.

The distribution curves along the rotational speed axis, n, shown in the maximum vibration diagrams (Fig. 3), can be interpreted as the superposition of various types of vibration:

- monotonic increase in vibration amplitude relating to the increased rotational speed; the nature of those curves reveals that these vibrations were generated by unbalance forces,
- resonance curves with local maximums, corresponding to the mode shapes of particular shaft line segments; these vibrations are of resonance nature,
- uniform vibration increase after certain rotational speed has been exceeded; these vibrations are connected with the instability of the oil film in the bearings.

The rate of increase of the forced vibrations connected with the rotational speed increase differs for particular bearings and vibration directions. This can be interpreted as a result of different unbalance distribution along the rotor line and different characteristics of bearing supports. Good examples of this type of forced vibrations are seen in relative vibration diagrams for bearing No. 7, as they are not obscured by the rotor vibration resonance curves. Such a flat curve testifies to good balancing of the generator rotor, especially in the direct vicinity of bearing No. 7.

Table 2 presents the locations of all resonances, which can be found in the maximum vibration diagrams (Fig. 3) and cascade diagrams (Figs. 7 and 8) for the basic case. The resonance velocities are linked in the Table with particular segments of the shaft line taking into account the resonance structure in the vibration pattern of the bearings supporting the examined segment. The last column presents the vibration frequencies, which are transmitted to the rotor segment of interest from the adjacent rotor. The resonance speeds indicated in Table 2 preserve, in general, their values after the bearing has been dislocated, although they reveal limited shift with respect to the basic case. On the other hand, in numerous cases the bearing dislocations make some resonances decay and other resonances appear. This testifies to some differences introduced to the nature of the phenomenon. They are undoubtedly connected with certain changes in the vibrating structure due to relatively strong deformation of the shaft, or change of support (oil film) stiffness. These remarks concern both the relative and absolute vibrations, although for the journal-bushing absolute vibrations they are more pronounced. In the basic case, the vibrations synchronised with the revolutions are almost exclusively recorded. Higher harmonics and sub-harmonics are negligibly small.

3.3 Interpretation of the relations between machine rundown characteristics and bearing dislocations

The analysis of the maximum vibration curves, complemented by corresponding cascade diagrams prepared for start-up or rundown of the turbine set with bearing displaced by maximum permissible ranges, and their further comparison with similar curves for the basic case allowed preparing the setting-up shown in Table 4. These data make the set of diagnostic relations of symptom -> defect type, opposite to those directly obtained from the calculations. From this table, and from the complementing Tables 3 and 5 one can make the following generalisations of the obtained diagnostic relations.

The bearings that are the most vulnerable for changes in the vibration pattern due to bearing dislocation are bearings Nos. 3 and 5. This means that those bearings reveal the highest threat of vibration development due to bearing dislocation during the machine start-up or rundown. Bearing No. 7 is resistant to dislocations of all bearings, from the point of view of the vibration development. This means that the dislocation of any bearing in the turbine set does not introduce changes to the vibration pattern of bearing No. 7 obtained from its basic case. It is noteworthy that dislocating one bearing provokes, as a rule, the biggest changes in the vibration pattern of one of adjacent, or even more distant, bearings to the moved one, and not in this bearing itself. Only in the cases of bearings Nos. 1 and 3, significant changes are observed in the vibration patterns after their dislocation.

The following general conclusions concerning the effect of bearing dislocation on the vibrations of particular bearings can be formulated:

- Vibrations in bearing No. 1 are affected by dislocations of bearings Nos. 1-4, but only the effect of bearing Nos. 1, 2, and 4 dislocations can be considered significant.
- Vibrations in bearing No. 2 are significantly affected by dislocations of bearings Nos. 1 and 3.
- Vibrations in bearing No. 3 are significantly affected by dislocations of bearings Nos. 1-3. Of lower significance is the effect of bearing No. 4 dislocation.
- Vibrations in bearing No. 4 are significantly affected only by horizontal dislocations of the bearings. Out of them, the effect of bearing No. 5 dislocation is the largest, that of bearing No. 6 is smaller, and those of bearings Nos. 1, 2, 3, and 4 are very limited.
- Vibrations in bearing No. 5 are significantly affected by dislocations of bearings Nos. 5, 6, and 7, while the effects of bearing No. 3 and 4 dislocations are limited.
- Vibrations in bearings Nos. 6 and 7 are visibly affected only by dislocation of bearing No. 5.

The overwhelming majority of cases collected in Table 4 reveal stable operation of bearings. In the vibration spectra shown in the cascade diagrams dominating is the first harmonics. Only in few cases the operation of bearings is highly instable, and strong sub-harmonic vibrations are dominating or at least clearly recognisable. Sometimes the sub-harmonic 1/3X is several times as high as the first harmonics. This effect may result form the instability of the oil film in the bearings. Between two extreme states, several cases reveal slight instability. In those cases the sub-harmonics 1/3X and 1/2X have amplitudes comparable with that representing the first harmonics. When the permissible bearing dislocations are calculated taking into account the vibration criterion, the vibrations, as a rule, cannot reveal amplitudes visibly larger than the assumed criterion values. Analysing a series of diagrams for different dislocations around the limiting value calculated from the vibration criterion leads to a conclusion that the instability usually builds up rapidly in a bearing when the dislocation nears the limiting value. The development of those vibrations is not in accordance with classical principles of the bearing instability development. Therefore one can expect that the vibrations observed in those cases are most of all connected with rotor vibrations generated by the cambered shaft line due to bearing dislocations.

Taking into account the entire range of the rotational speed of the rotor during the rundown, from the nominal speed to zero, the permissible bearing vibrations are only exceeded in five cases, namely when bearing No. 1 is dislocated upward, No. 2 upward, No. 4 right, No. 5 right, and No. 6 left. These cases have been collected in Tab. 5, together with accompanying rotational speed ranges for which this effect took place. In each case the relative vibrations in the journal-bushing system are exceeded. In bearings Nos. 5 and 6 it takes place within the regime of high rotational speeds, including the nominal rotational speed, n=3000 rev/min. Table 1 reveals that only in two cases the vibrations were the criterion of significance for calculations of permissible bearing dislocation. In 26 remaining examined cases, the criterion was the pressure on the bushing surface. In those cases, the permissible bearing loads, which limited their maximum dislocations, were exceeded prior to the permissible vibrations. That means that in the majority of cases the permissible bearing load criterion turned out more restrictive that the permissible vibration criterion. This situation is unfavourable from the point of view of operational safety. Exceeding the permissible pressure criterions does not generate, as a rule, vibrations that are sufficiently high to stop the machine. At the same time, direct observation of pressure changes in the bearings is difficult and not used in practice. However, this does not create considerable threat of rapid failure, as, on the one hand, the assumed design values of the permissible pressure are very low and can be significantly exceeded in practice. On the other hand, long-lasting bearing overloads are usually connected with the rise of oil and bushing temperature, which are monitored.

4 Conclusions

- Dislocation of turbine set bearings changes the levels of relative and absolute vibrations of the bearings during machine rundown; these changes depend on the direction and range of the dislocation.

- For maximum permissible bearing dislocations, the pattern and range of bearing vibrations during turbine set rundown depend on whether the limiting vibrations were determined taking into account the vibration or the bearing load criterion.

- When the bearing is dislocated by a range corresponding to the permissible bearing load, the vibration spectrum is dominated by the first harmonics and the operation of the bearings is stable, while in cases of permissible bearing dislocations with respect to the vibration criterion the vibration spectrum is dominated by sub-harmonic frequencies 1/3X, 1/2X, 2/3X, and the operation of the bearings becomes unstable.

- During the rundown of the turbine set with dislocated bearings, the threat of exceeding permissible relative vibrations of the journal-bushing system is much higher than that of the exceeding the absolute bushing vibrations.

- Cascade diagrams prepared for different bearing dislocations of a near permissible range with respect to the vibration requirement allow a conclusion that the instability symptoms, manifesting themselves by strong vibrations, become, as a rule, reached rapidly when the dislocation nears the limiting value.

- As far as the development of strong vibrations during start-up or rundown of a machine with dislocated bearings is concerned, of highest susceptibility are bearings Nos. 3 and 5, while bearing No. 7 reveals resistance to dislocations of all bearings in the entire rotational speed range.

- The most pronounced changes in the vibration pattern are, as a rule, recorded in the bearings adjacent or even more distant to the dislocated bearing, and not in this bearing itself.

- The obtained results allow one to prepare a set of diagnostic relations referring to machine start-up and/or rundown, which can be applied in diagnostic systems.

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