MAPS OF TOLERABLE MISALIGNMENTS OF BEARINGS APPLICABLE IN DIAGNOSTIC SYSTEM OF THE 13K215 TURBOSET

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Summary

The article presents results of research oriented on creating a pattern of the dynamic state of a turboset in the presence of a defect consisting in the misalignment of its bearings. The work contains the exemplary set of three-dimensional graphs, which illustrate vibration levels of all bearings as a function of displacement of each turboset bearing in any direction in relation to its base position. Graphs showing division of the area of the expected bearing dislocations into the sub-areas of permissible and prohibited dislocations have been then created. The work was done by means of computer simulation of bearing misalignments in the numerical model of 13K215 turboset by means of MESWIR computer code package. Relations between bearing dislocations and the dynamic state of the machine compose a set of diagnostic relations, which can be included to the base of diagnostic knowledge on the machine and used in its diagnostic system.

Keywords: technical diagnostics, vibration, bearing, misalignment.

MAPY DOPUSZCZALNYCH PRZEMIESZCZEŃ ŁOŻYSK DLA POTRZEB SYSTEMU DIAGNOSTYCZNEGO TURBOZESPOŁU 13K215

Streszczenie

W artykule przedstawiono wyniki badań mających na celu utworzenie obrazu stanu dynamicznego turbozespołu w obecności defektu polegającego na rozosiowaniu jego łożysk. Praca zawiera przykładowe zbiory trójwymiarowych wykresów, które ilustrują poziom drgań wszystkich łożysk turbozespołu jako funkcje przemieszczenia jego wszystkich łożysk w dowolnym kierunku w stosunku do ich bazowego położenia. Stworzone zostały wykresy pokazujące podział obszaru możliwych przemieszczeń łożysk na podobszary dozwolonych i zabronionych przemieszczeń. Prace wykonano poprzez symulacje komputerowe przemieszczeń łożysk w numerycznym modelu turbozespołu 13K215 pakietem oprogramowania MESWIR. Związki między przemieszczeniem łożysk i stanem dynamicznym maszyny tworzą zbiór relacji diagnostycznych, który może być włączony do bazy wiedzy diagnostycznej maszyny i użyty w jej systemie diagnostycznym.

Słowa kluczowe: diagnostyka techniczna, drgania, łożysko, rozosiowanie.

1. INTRODUCTION

The first investigations of the effect of misalignment of turboset shaft line on dynamic performance of the machine were carried out according to a commercial order of power turbine producers who needed these data for diagnostic purposes. The results of these investigations turned out very attractive and initiated further, more detailed and systematic investigations of the problem [1, 2, 3].

The bearing misalignment is a defect frequently observed in rotating machines. According to [4, 5, 6], when taking into account the frequency of appearance the bearing misalignment is the second type of inefficiency, after unbalance, recorded in large machines of this type. Numerous phenomena observed in turbosets, in particular vibrations of shafts, bearings and the foundation, are expected to have their origin in bearing misalignment. These vibration phenomena are difficult to explain otherwise, and frequently grow stronger or vanish without a visible reason.

The problem of bearing misalignment is of particular importance in great rotating machines of continuous operation, such as large power turbosets. Those turbosets are usually multi-casing and multi-rotor machines in which shafts are linked together via rigid and semi-rigid couplings and which are supported in numerous oil bearings [4, 5, 6, 7]. The bearings in multi-bearing machines are arranged in such a way that the shaft line constitutes a catenary, which makes it possible to eliminate shaft bending on the couplings, the most vulnerable elements of multi-rotor machines [6, 8, 9, 10]. Dislocating any bearing with respect to its optimum position, determined by the designed shaft catenary, changes operating conditions of the bearings and rotors supported in them. Static loads of particular bearings and shafts change, provoking changes in the dynamic state of the entire machine, which mainly manifests itself as the vibrations. The way and scale of machine response to bearing misalignment depend first of all on the scale and type of the misalignment, but is also affected by numerous other agents, such as: relative bearing positions, bearing oil film properties, and stiffness characteristics of the rotor supported by the bearings [4, 6, 11, 12, 13].

2. BACKGROUND AND MOTIVATION

The present work is a continuation and extension of author's earlier research activities in the area of bearing misalignment defect. The results of those activities were presented in series of publications. Early publications [2, 3] describe the procedure and initial results of calculations of acceptable turboset bearing dislocation ranges taking into account the criterion of permissible vibrations in all bearings, and the criterion of permissible static load of the bearings. In [2, 3] the permissible bearing dislocation ranges in vertical and horizontal directions were determined. In [11] these investigations were generalised by determining the ranges of permissible bearing dislocations in an arbitrary direction. These investigations revealed strong interaction between two bearings located close to each other. Ref. [9, 13, 14] analyses the effect of misalignment of two most vulnerable bearings in the turboset on its dynamics in transient states, i.e. during turboset start-ups and run-downs. It was concluded that bearing misalignment considerably changes resonance characteristics of the machine, affecting both critical speeds, and vibration amplification factors. All this creates failure threats during start-ups and/or run-downs of a machine with misaligned bearings due to the appearance of dangerous resonances at such rotational speeds, at which the same machine without the misalignment defect would work correctly.

All the above cited research activities have revealed that the relations between the bearing misalignment and the dynamic state of the machine are not simple and straightforward. It was found that limited bearing misalignment in some direction may increase the vibrations to the level exceeding the permissible vibration range, while a larger bearing misalignment in the same direction may make the vibrations vanish again [9, 11]. That is why the determined ranges of permissible bearing dislocations are very irregular and difficult to interpret. It is impossible to predict, even intuitively, which bearing dislocations will intensify turboset vibrations and which will make them vanish.

The motivation for undertaking the reported investigations was also the information found in the literature on a remarkable effect of bearing misalignment on the dynamic state of a large rotating machine with a long shaft line supported in numerous bearings [4, 5, 6]. A general theoretical approach to the problem of shaft misalignment was presented by A. Muszynska in „Rotordynamics" [4]. From the engineering point of view, the most comprehensive elaboration of the problem of rotating machine shaft misalignment is „Shaft Alignment Handbook" by J. Piotrowski [6]. Based on the abovementioned publications we can assume that the mechanism that generates shaft vibrations and resulting stresses due to bearing misalignment is briefly the following: Bearing dislocation results in the generation of static loads in the bearings and shaft. These loads differ from nominal values, which provokes shaft bending and changes unbalance distribution. As a consequence, rotor vibrations and disturbances in bearing operation are generated, the latter manifesting themselves in oil vibrations. The rotation and vibrations of the shaft generate time-dependent shaft stresses, which interfere with the static stresses. Moreover, the vibrations are transmitted via rotor elements and the bearings to the remaining parts of the machine and can negatively affect its operation, performance and even safety.

Advanced and expensive machines, including power turbosets, require effective protection in case of possible failure. Extended diagnostic systems are developed for this purpose. To act properly, the diagnostic systems require sets of relations linking the type and intensity of defects with their observable (measurable) effects. Maps of bearing dislocations, the result of the here reported activity, are in fact relations which link the bearing misalignment defects with their symptoms – vibrations. This way the relations between possible bearing dislocations and the dynamic and kinetic state of the machine compose a set of diagnostic relations, which can be included to the base of diagnostic knowledge on the machine and used in the diagnostic system [9, 15].

3. METHODOLOGY

The results of earlier studies on the subject and difficulties with their interpretation suggested the need for more comprehensive investigations of the effects of turboset bearings misalignment. Systematic studies were performed, the object of which was chosen the turboset 13K215. Scheme and specific data characterizing the machine are described in particulars in previous publications in the subject [1, 2, 10, 15]. The machine shaft line consists of 4 shafts connected by 3 couplings and supported in 7 oil bearings. The calculation model was created for a selected real turboset. The model was tuned using the results of measurements recorded in a power plant by its diagnostic system in steady state and nominal operation conditions, at the speed equal to 3000 rev/min and full power output 211 MW [10, 12].

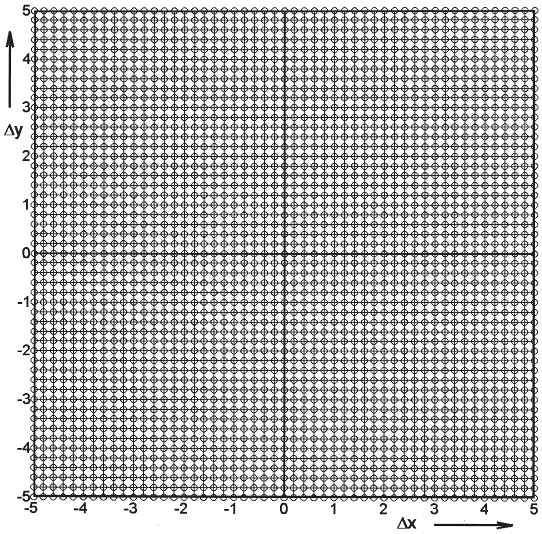

Fig. 1. The area of possible bearing dislocations – grid 51 x 51 points for which calculations were carried out

It was decided to study the response of this machine to dislocations of all bearings in all directions, and to record this response in the form of vibrational symptoms observed in all bearings. For this purpose a plane of possible bearing dislocations was defined, the centre of which is the basic, designed position of the bearing with respect to other bearings. Assuming this position as a base is justified by the fact that bearing dislocations reveal random nature, as a result of which dislocation of an arbitrary bearing in arbitrary direction is equally possible. Performed were systematic, detailed calculations of the effects of bearing dislocations within the ranges:

– $-5mm < \Delta x < 5mm$, with 0,2mm step in horizontal direction,
– $-5mm < \Delta y < 5mm$, with 0,2mm step in vertical direction.

It was assumed that simultaneous dislocations in vertical and horizontal direction are possible. Consequently, a 51 x 51 grid was defined on the dislocation plane, which gave 2601 possible positions of the bearing centre. The distribution of the calculation points within the 10x10 mm square is schematically shown in Fig. 1.

For each point marked with a small circle on the dislocation plane in the Fig. 1, the calculations were performed using the programme package MESWIR. For the purposes of the present publication the following quantities were calculated and analysed:

– amplitudes of relative vibrations in two directions A_x, A_y, perpendicular to each other
– RMS velocities of absolute vibrations in two directions V_x, V_y, perpendicular to each other.

The reported research is of typical theoretical and numerical nature. All calculations were performed using a set of computer codes composing the system MESWIR [1, 12, 15]. The system was developed and is in use in IF-FM for calculating the dynamics of rotors supported on oil bearings. The rotors are modelled using finite elements. Bearing characteristics (reactions, stiffness and damping properties) are obtained as the numerical solution of the two-dimensional Reynolds equation with the Reynolds boundary conditions. The main code package is written in FORTRAN. The work consisted in calculating the dynamic state of the machine with the simulated misalignment of a selected bearing from its base position. The misalignment defect was introduced to the base model of the machine as the dislocation of a relevant node representing the bearing, in the form of dislocation components: Δx in the horizontal direction and Δy in the vertical direction. An arbitrary bearing dislocation is the geometric sum of the above elementary dislocations. The results of calculations are sequences of instantaneous positions of selected system nodes as a function of time, which makes it possible to calculate vibration amplitudes and velocities, and draw trajectories of those nodes.

In order to determine the areas of permissible bearing dislocations with respect to allowable bearing vibrations, the limits of vibrations were assumed [10, 11]. When these limiting values were exceeded, the technical state of the turboset was considered unacceptable. The following vibration criteria were assumed:

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

where A_{lim} is the limit of relative journal-bush vibrations, expressed by the *p-p* dislocation amplitude in two directions inclined by 45° to the perpendicular,

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

where $V_{RMS\ lim}$ is the limit of absolute bearing vibrations, expressed by *RMS* vibration velocities in the horizontal and vertical directions.

The vibration limits were taken from the standards: ISO 7919-2 for relative vibration and from ISO 10816-2 for absolute vibration. The limits correspond to the machine warning state.

It is worth stressing that for the machine to be considered ready for operation as a whole, all these two conditions are to be simultaneously met in all bearings, and not only in the bearing in which the defect was recorded. In case any of these limits is exceeded, the turboset should be stopped from operation. There should be also underlined that type of vibration and location of points where the vibration are analysed in the machine model correspond to those measured by the diagnostic system in the real machine. That is why these maps are compatible with the recordings of the machine's diagnostic system and can be easily compared with the vibrations recorded by the system. Thus they can be used by the diagnostic system.

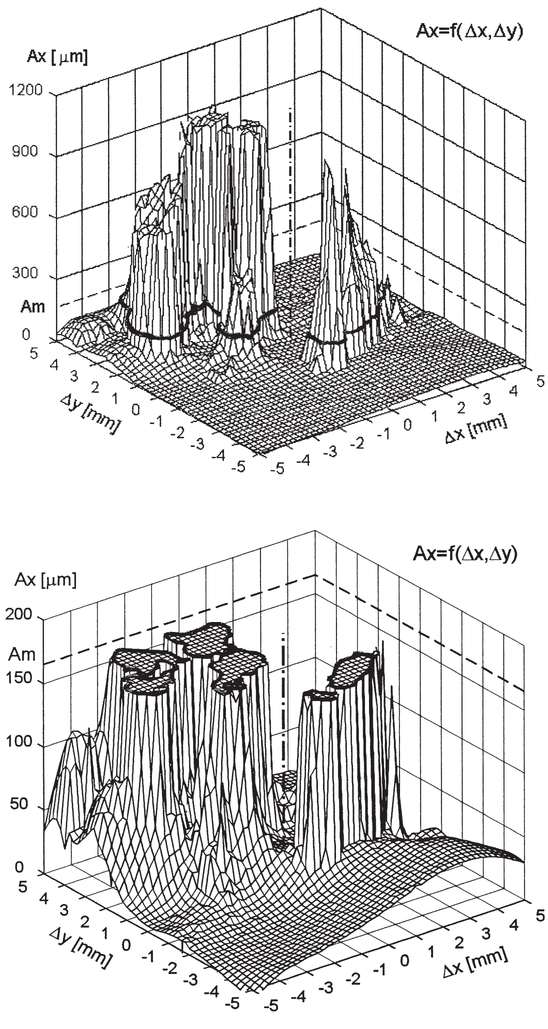

Fig. 2. The graphs showing vibrational response in the bearing 5 to displacements of the same bearing 5. Up: full scale of bearing vibration. Down: vertical axis magnified and cut at a parameter permissible level. Well visible are areas of prohibited bearing dislocations (shaded dark) at the limit level

4. VIBRATIONAL MAPS OF BEARING DISLOCATIONS

The response of the machine to the bearing misalignment primarily is presented in the form of three-dimensional graphs showing the intensity of vibrations of machine elements as a function of bearing dislocation. In the most voluminous form the results of the research are presented in 49 sets of graphs of this type. Each set presents the distributions of all analysed parameters in one of the seven bearings as functions of individual dislocations of one of the seven bearings. Thus the effects of misalignment of 7 bearings on those 7 bearing vibrational characteristics can be presented using 49 sets of diagrams, having the form of a function: *effect* $= f(\Delta x, \Delta y)$. By standard every set of the results contains four function graphs:

$A_x=f(\Delta x, \Delta y)$ – relative vibration amplitudes in the direction of the first quarter bisector,

$A_y=f(\Delta x, \Delta y)$ – relative vibration amplitudes in the direction of the second quarter bisector,

$V_x=f(\Delta x, \Delta y)$ – RMS velocities of absolute vibrations in the horizontal direction,

$V_y=f(\Delta x, \Delta y)$ – RMS velocities of absolute vibrations in the vertical direction,

Sample diagrams of this type are presented in Fig. 2. The figure shows the sample function $A_x=f(\Delta x, \Delta y)$, presenting the effect of dislocation of the bearing 5 on the relative horizontal vibration amplitude observed in the same bearing 5.

In the diagrams, the base bearing position on the catenary, (i.e. when the misalignment defect is missing), is marked with a vertical line starting from the centre of the dislocation plane, point Δx=0, Δy=0. An arbitrary bearing dislocation by a vector $[\Delta x, \Delta y]$ corresponds to certain values of the parameters A_x, A_y, V_x, V_y which can be read from the diagrams. On the *y*-axis, the level representing the permissible limit for the earlier defined parameter A_{lim} =165μm is marked. The crossing of the surface presented in the diagram with the plane parallel to the *x-y* plane and situated at the permissible limit level gives the level line shown in the upper part of the Fig. 2. Figures of this type were also prepared in a modified form, which cuts off the diagrams at the level of the parameter limit. As a result of this action the prohibited bearing dislocations are better visible and the distribution of the analysed parameter on the *x-y* plane is more readable. The lower graph of Fig. 2 presents sample of this type, which correspond to those shown in the left graph.

The projection of a cut of the vibration graph on the limiting level onto the dislocation plane defines the region of prohibited bearing dislocations from the viewpoint of a given criterion. When, in any way, the bush centre falls into this region, the machine should be stopped from operation as it means that the permissible limit for one of operating parameters was exceeded in one of the bearings.

5. MAPS OF PERMISSIBLE BEARING DISLOCATION AREAS

From the practical point of view, of high usability are simplified figures showing the division of the area of the expected bearing dislocations into the sub-areas of permissible and prohibited dislocations. Diagrams of this type may be regarded as maps of permissible/prohibited bearing dislocation. They are presented in Figs 3 – 6. Figs 3 and 4 are calculated with respect to the limit of relative bearing vibration, A_{lim} = 165 μm, and Figs 5 and 6 are calculated with respect to the limit of absolute bearing vibration, $V_{RMS\ lim}$ = 7,5 mm/s.

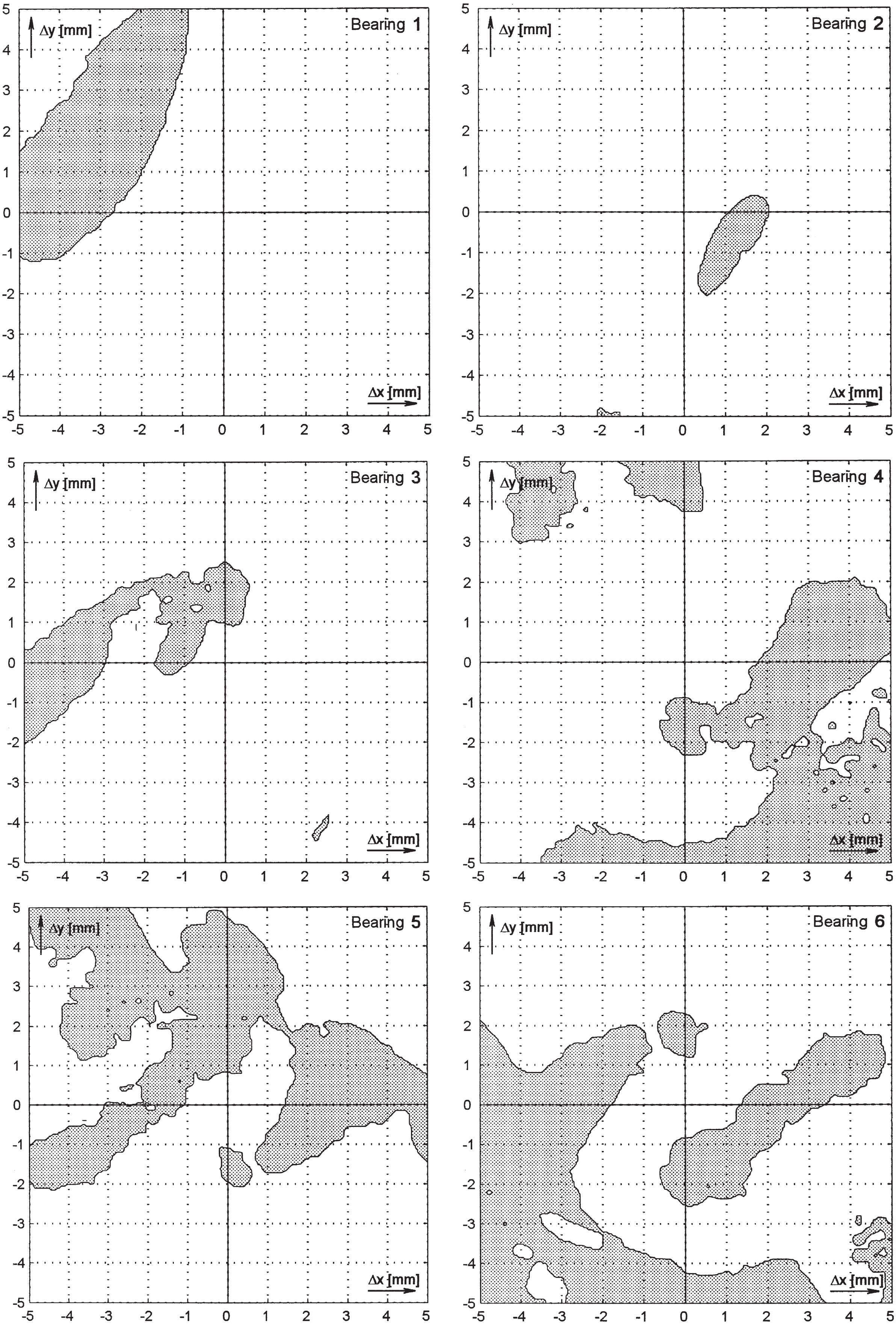

Fig. 3. Division of the area of possible dislocations of the bearings 1 - 6 to permissible (white) and prohibited (marked dark) areas calculated with regard to RELATIVE HORIZONTAL shaft vibrations

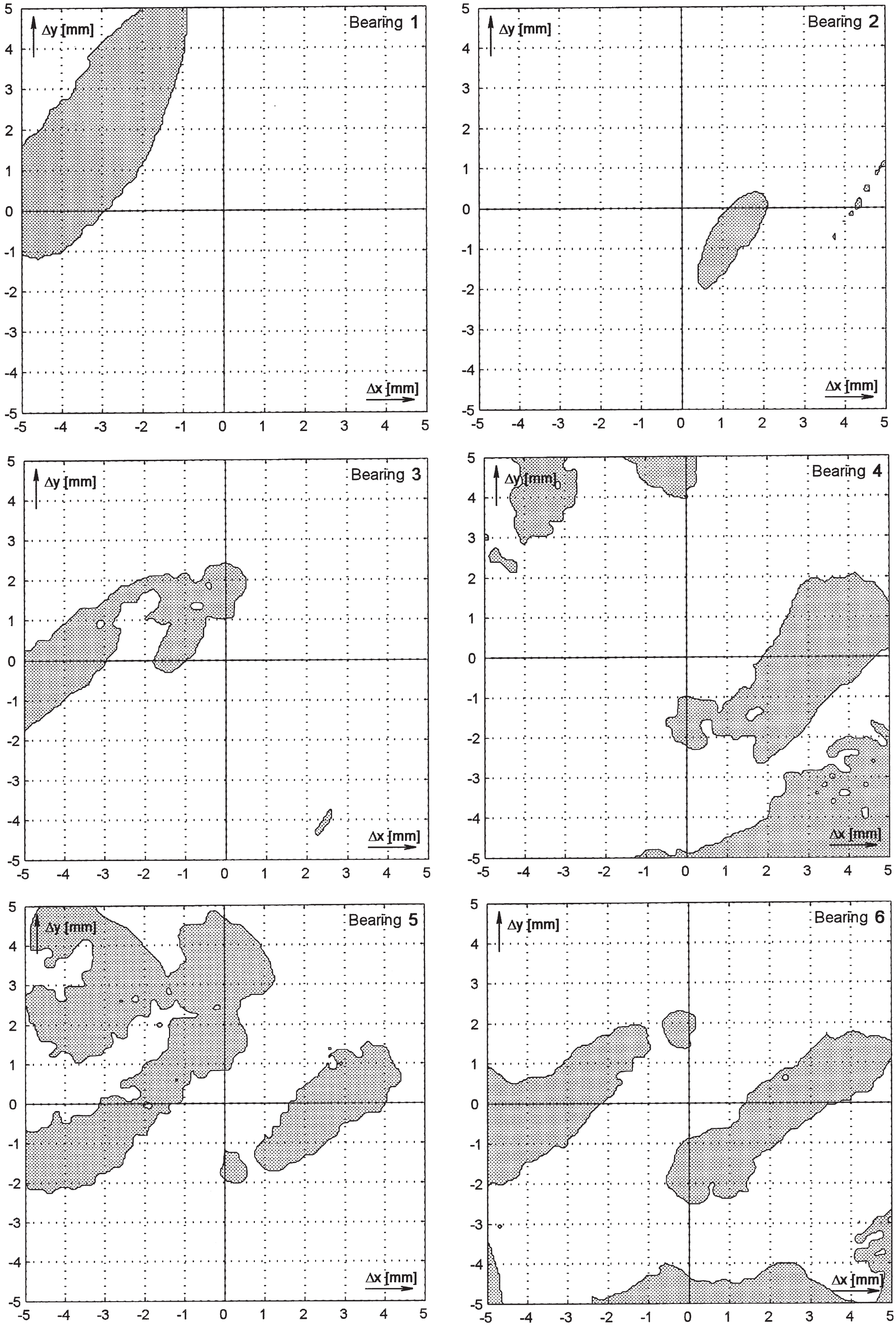

Fig. 4. Division of the area of possible dislocations of the bearings 1 - 6 to permissible (white) and prohibited (marked dark) areas calculated with regard to RELATIVE VERTICAL shaft vibrations

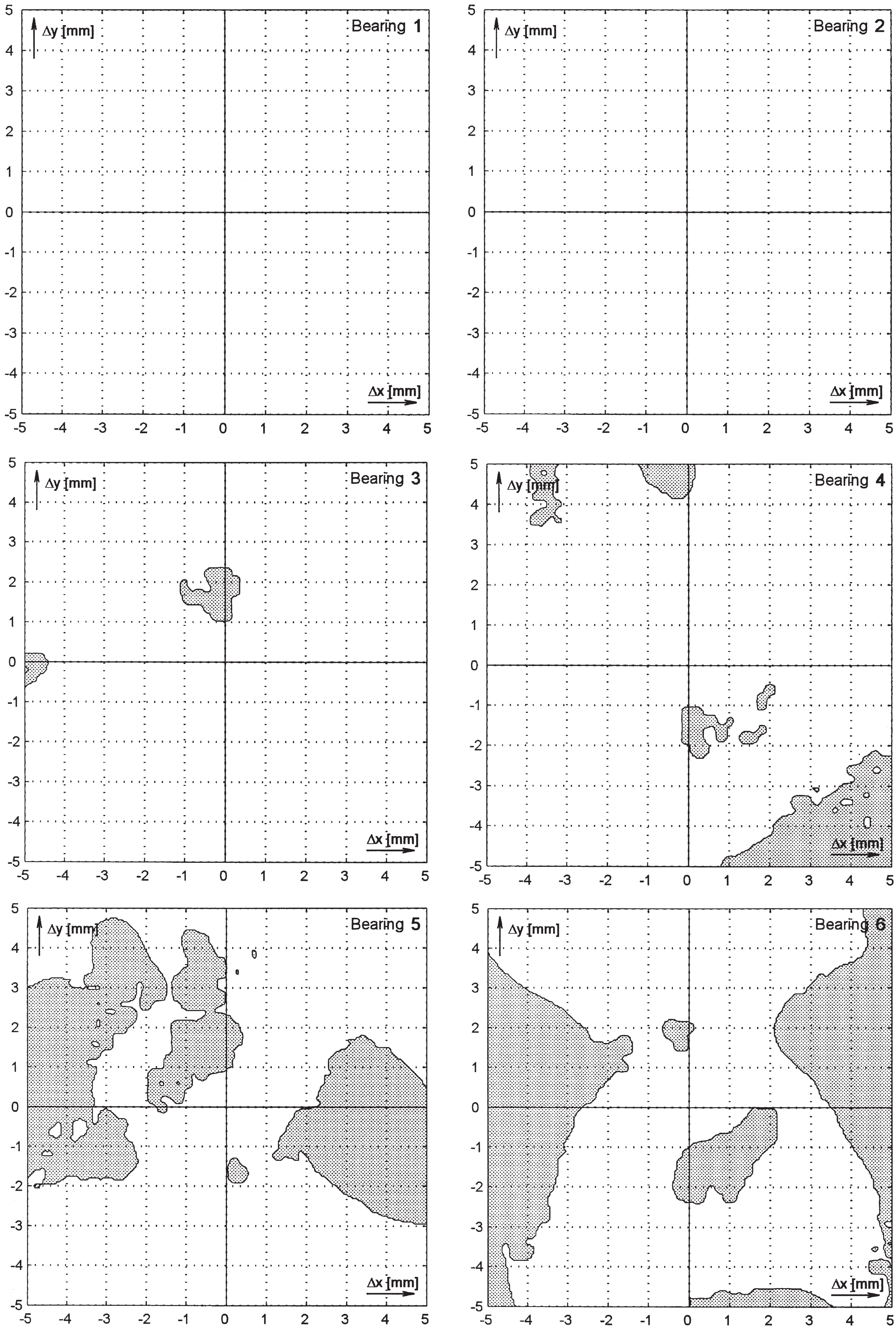

Fig. 5. Division of the area of possible dislocations of the bearings 1 - 6 to permissible (white) and prohibited (marked dark) areas calculated with regard to ABSOLUTE HORIZONTAL shaft vibrations

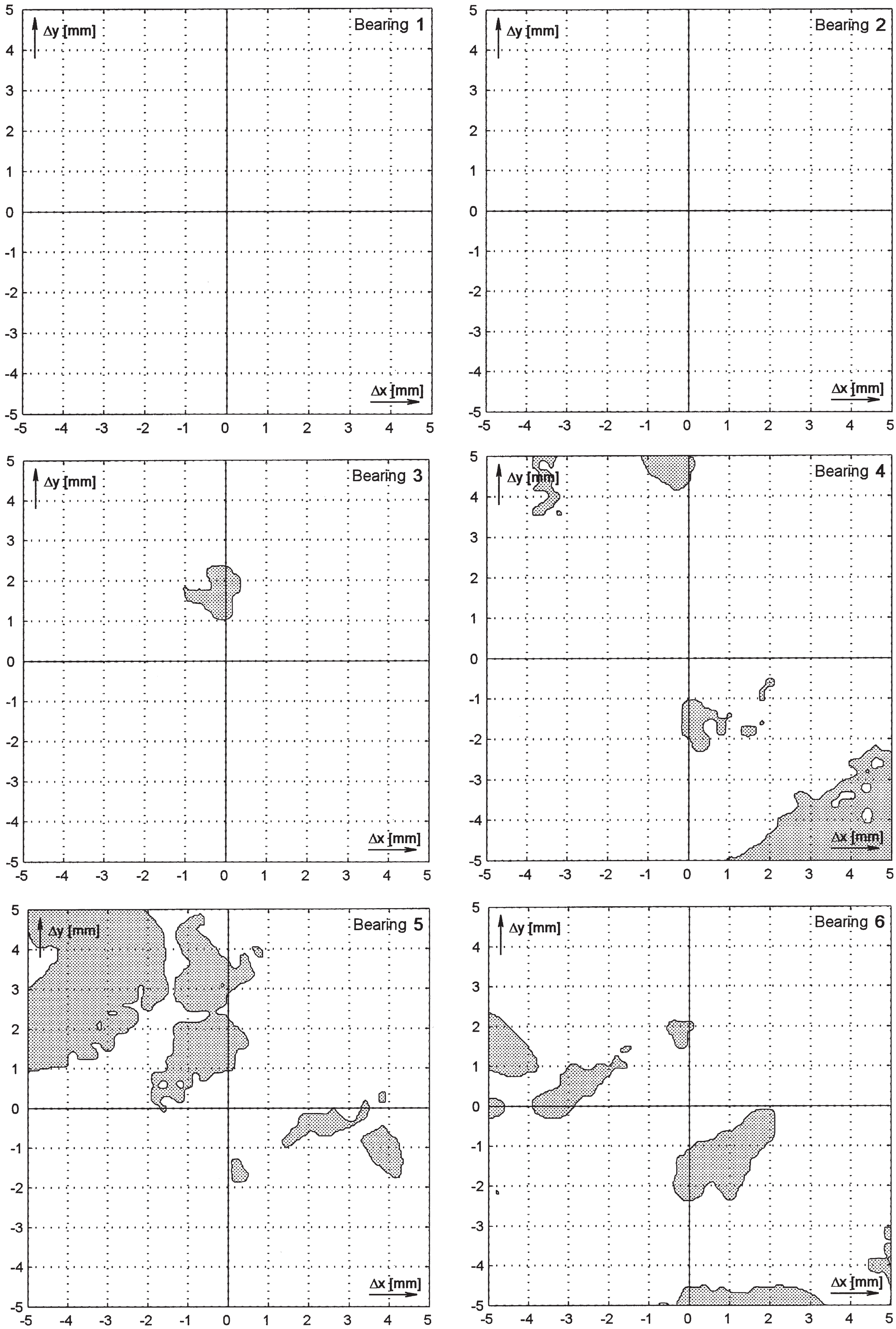

Fig. 6. Division of the area of possible dislocations of the bearings 1 - 6 to permissible (white) and prohibited (marked dark) areas calculated with regard to ABSOLUTE VERTICAL shaft vibrations

Figs 3 and 5 concern vibration in x-direction, and Figs 4 and 6 concern vibration in y-direction. In every figure the maps of permissible dislocations of six consecutive turbogenerator bearings No. 1 - 6 are presented. The diagrams connected with misalignment of the bearing 7 have not been included in the Figs 3 – 6 as they are completely empty, both with respect to relative and absolute vibration in x-direction as well as in y-direction. This means that arbitrary dislocations of bearing 7 within the area -/+ 5 mm do not generate dangerous vibrations in any bearing.

In the Figs 3 – 6 the entire area of regarded bearing dislocations is divided by lines, which are the projection onto the *x-y* dislocation plane those lines, which are the result of intersection of the 3-dimensional vibration function with plane on the limiting level. The areas of prohibited bearing dislocations are marked dark in the figure. At the same time dislocating the bearing to an arbitrary point located within the white area does not provoke those vibration effects, which could be considered unacceptable from the point of view of machine operation. The white area represents permissible dislocations. It should be stressed, that dark area on the maps present those forbidden positions of bearings, which cause dangerous vibration in at least one of the turbogenerator bearing, no matter in which one. It does not result from the maps in Figs 3 – 6 in which bearing the limit of vibration is exceeded. To have an information in which bearing dangerous vibration are present, additional maps should be prepared, separately with respect to vibration in every particular bearing.

The Figs. 3 – 6 reveal that the areas of prohibited bearing dislocations are limited by irregular close curves. The areas are chaotically distributed on the surface of the dislocation square, within $-5\text{mm}<\Delta x<5\text{mm}$, $-5\text{mm}<\Delta y<5\text{mm}$. Depending on the analysed case the areas have different sizes and locations, impossible to predict intuitively.

Comparison of the Fig. 3 that presents areas of prohibited bearing dislocations determined when taking into account the horizontal relative vibrations and the Fig. 4 that presents areas determined when taking into account the vertical relative vibrations are very similar to each other. Similar observation can be made also in relation to absolute bearing vibration: in horizontal direction shown in Fig. 5 and in vertical direction shown in Fig. 6 (in the last case remarkable differences are observed only in case of bearing 6 dislocations). That is why the areas of prohibited dislocations of all bearings were determined also taking into account a combined criterion of horizontal and vertical vibrations, however separately for relative and absolute vibrations. Graphs of this type (not included in the paper) may have direct application in diagnostic systems. The figures show that, generally, the prohibited dislocation areas calculated taking into account the relative vibration criterion are wider and those calculated at the absolute vibration criterion are situated almost entirely within them.

The machine as a whole should be considered unserviceable when permissible parameter limits are exceeded in any bearing due to misalignment of any bearing.

Therefore from the point of view of a certain criterion, the area of prohibited dislocations of a certain bearing is the sum (in the sense of the arithmetic of sets) of the areas of prohibited dislocations of this one bearing from the viewpoint of fulfilment this criterion in any single bearing. For instance, the area of prohibited bearing dislocations taking into account the relative vibration criterion covers all dislocations of the examined bearing which generate unacceptable relative vibrations in any of machine's bearing. The machine evaluation criteria can be linked together in an arbitrary way via the abovementioned operations on sets. Thus, based on numerous detailed maps we can construct areas of bearing dislocations provoking certain generalised effects. It is worth mentioning that the area of prohibited bearing dislocations, coloured grey in the Figs 3 - 6, is the complement to the permissible dislocation areas (white colour).

6. SUMMARY AND CONCLUSIONS

Based on the maps of permissible bearing dislocation following conclusions can be formulated:

- From the point of view of misalignment effects, of highest importance are bearings 3, 4, 5, and 6. Misalignment of these bearings threatens most severely the safety of machine operation, and in these bearings the effects of misalignment of each of them are most remarkable.
- Dislocating bearings 3, 4, 5, 6 within the range -/+5 mm always leads to exceeding relative vibration limits, both in horizontal and vertical direction, in all seven turboset bearings. Moreover, it leads to exceeding permissible absolute bush vibrations in bearings 4, 5, and 6.
- The turboset is, in practice, unaffected by dislocations of bearings 1, 2, and 7 within the range of -/+5 mm. Even if misalignment of these bearings leads to the exceeding of certain parameters, the exceeding is insignificant and does not provoke vibrational instability of the entire system.
- There is no threat with exceeding vibrations in bearings 1, 2, and 7 caused by dislocation of any turboset bearing. The recorded cases of exceeding of permissible bearing vibrations are insignificant and can be neglected in practice.

Observations and detail conclusions based on the presented investigations can be generalized as follows:

- The maps of permissible bearing dislocation gives a general view on the machine resistance to misalignment of particular bearings from the point of view of different vibration criteria.
- Vibrational response of the machine to misalignment is difficult to predict intuitively, and can be rapid and unexpected.
- A complete set of maps presenting the effects of machine bearing misalignment can be interpreted as the base of knowledge on the effects of the bearing misalignment defect and used in the diagnostic system of the analysed machine.
- The here presented base of knowledge concerning the bearing misalignment defect is an example of practical realisation of a concept of creating a pre-defined base of diagnostic knowledge using model based computer simulations.

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