ANALYSIS OF ACCEPTABLE NONALIGNMENT OF BEARINGS OF LARGE POWER TURBOSET

Jozef RYBCZYNSKI

Institute of Fluid-Flow Machinery, Polish Academy of Sciences
Department of Machinery Dynamics, Fiszera 14, 80-952 Gdansk, Poland
Tel.: (+48-58) 3460881 ext. 273, fax: (+48-58) 3416144, e-mail: ryb@imp.gda.pl

Summary

Dislocation of bearings in a turbine set with respect to their basis location, determined by a kinetostatic chain line of the shafts, changes the dynamic state of the entire machine and the loads of machine's bearings. Numerical analysis of a digitised model of a large-power turbine set allowed the determination of permissible dislocation ranges of machine's bearings. The maximal dislocations were calculated of all bearings in vertical and horizontal directions taking into consideration criteria of the permissible load, and permissible vibrations of the bearings. The results were graphically illustrated in the form of areas of permissible bearing dislocations. The investigations have revealed an asymmetry of those areas, which may suggest that the constructional kinetostatic line of rotors is not optimal.

Keywords: technical diagnostics, rotating machine, slider bearing

ANALIZA DOPUSZCZALNEGO ROZOSIOWANIA ŁOŻYSK TURBOZESPOŁU DUŻEJ MOCY

Streszczenie

Przemieszczenie łożysk turbozespołu w stosunku do ich położenia bazowego, wyznaczonego przez kinetostatyczną linię łańcuchową wałów powoduje zmianę stanu dynamicznego maszyny i obciążeń łożysk. W drodze analizy numerycznej dyskretnego modelu turbozespołu dużej mocy wyznaczone zostały zakresy dopuszczalnych przemieszczeń poszczególnych jego łożysk. Obliczono zakresy maksymalnych przemieszczeń wszystkich łożysk w poziomie i w pionie ze względu na kryteria dopuszczalnych obciążeń i dopuszczalnych drgań łożysk. Wyniki zilustrowano w formie obszarów dopuszczalnych przemieszczeń łożysk. Badania wykazały niesymetrię obszarów dopuszczalnych przemieszczeń łożysk mogącą świadczyć o tym, że konstrukcyjna, kinetostatyczna linia wirników nie jest optymalna.

Słowa kluczowe: diagnostyka techniczna, maszyna wirnikowa, łożysko ślizgowe

1. INTRODUCTION

The constructional kinetostatic chain line of shafts in a rotating machine consisting of more than one rotor is determined theoretically during the design phase and set practically in the process of machine assembly. The shaft line should be designed in a way securing optimal operating conditions for particular bearings and rotors. Dislocating any bearing with respect to its basic position, changes operating conditions of individual slider bearings and, as a consequence, the operating conditions of the entire machine. Changed is the static load of the shafts and bearings, and, as a further consequence, the dynamic state of the entire machine. Vibrations of rotors and bearings can be generated in those circumstances [1], [2], [3].

The reasons of bearing dislocations with respect to the designed chain line can be of assembly, operating, or emergency nature. From the point of view of possible consequences of those types of defects, it is preferable for the ranges of permissible bearing dislocations to be approximately symmetrical in two opposite directions with respect to the basic location.

2. THE AIM AND RANGE OF ANALYSIS

The present investigations aimed at determining the ranges of permissible dislocations of bearings in the turbine set, taking into account certain criteria defining the acceptable regime of turbine set's operation [3], [4]. The permissible dislocations were determined as the dislocations, at which the permissible parameters of machine's operation were being surpassed. The ranges of permissible dislocations were determined for all bearings in the turbine set, separately in four directions: to the right, left, up, and down with respect to the basic location.

The most general criterion used for assessing the state of the entire turbine set as acceptable is simultaneous fulfilment of all three below named conditions for all seven bearings in the turbine set:

– relative journal/bushing vibrations, determined by the displacement amplitudes p-p,
  $s < s_{gr} = 165$ μm,
– absolute vibrations of the bearing, determined by mean square velocities of vibrations:
  $v_{RMS} < v_{RMSgr} = 7{,}5$ mm/s,
– load of the bearing determined by average pressures on the bush surface:
  $p < p_{dop} = 2$ MPa.

The object of investigations was a large power turbine set consisting of a 200 MW turbine and a generator. The examined object is a four-cylinder set, whose rotors are supported in seven slider bearings. Four shaft sections are linked together by three couplings.

The investigations consisted in calculating the kinetostatic and dynamic states of the machine using a set of computer codes developed and used in IFFM [5], [6]: KINWIN-60, KINWIN-I-LEW, NLDW-75-LEW. Scripts written in the package MATLAB were used for automation of the calculation process. Their task was to supervise the iterative process of calculations, then calculation of the displacement amplitudes of relative vibrations and the velocities of absolute vibrations in relevant nodes, as well as the bearing reacting forces.

The starting point for the calculations and the reference material for further analyses was a "basic case" [5], created on the basis of data measured on the real turbo set. For the purpose of calculations, a digital physical model of the turbine set was developed, along with corresponding numerical model adapted to the calculations making use of the fine element method [5], [6].

3. METHODOLOGY OF RESEARCH

The shape of the line of rotors and an analysis concept of bearing dislocation with respect to the kinetostatic line are schematically shown in Fig. 1. The investigations were carried out by moving particular bearings with respect to their basic location, i.e. introducing a certain defect to the basic model of the machine. The effects of this defect were studied in the form of changes of bearing loads and the development of relative and absolute vibrations of the bearing.

In the codes used for calculations, the locations of bearings were introduced as their dislocations with respect to the geodesic line. In the basic case those dislocations resulted only from the assumed kinetostatic line of rotors (denoted as *b* in Fig. 1). Additional dislocations of the bearings, representing the "defects" of the machine, Δx Δy, were added to the basic dislocation values. The general bearing dislocation acceptance area was created taking into consideration the three criteria mentioned above altogether.

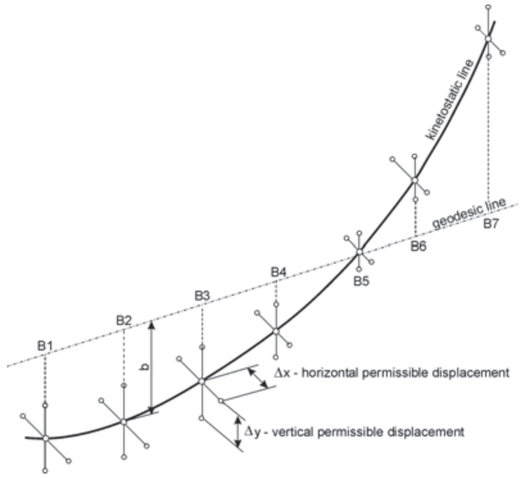


Fig. 1. The base kinetostatic shaft line and the concept of analysis of permissible bearings displacements.

The calculated ranges of permissible bearing dislocations are shown in Figs. 2-4. Points were marked on the coordinate axes, which correspond to the calculated permissible bearing dislocations in four directions. At each point, the value of maximal dislocation was given. Framed is the number of bearing (B1 – B7) in which the limiting value of load or vibrations has been surpassed.

The extreme points on four axes were connected with lines. The area created in this way defines the complete set of real permissible bearing dislocations. The origin of the coordinate system represents the initial location of the centre of bearing, with respect to which the bearing was dislocated. Bearing numbers are given near their centres. A dotted/broken line represents an axis crossing the geodesic line, which stands for the reference level for the locations of all bearings in the turbine set. The centre of the geodesic line is marked as *SG*. Quantities b1... b7 stand for dislocations of bearings No. 1... 7 resulting from the kinetostatic line.

4. ANALYSIS OF THE RESULTS

The maps of permissible bearing dislocations are shown in Figs. 2 – 4. The maps shown in the figures reveal that the ranges of horizontal dislocations of bearings nr 1, 2 and 7 are very wide (5mm – 20mm) and approximately symmetrical with respect to the basic location, i.e. they are almost the same to the left and right. The figures 3 and 4 show, that two pairs of bearings located close to each other have much narrower (1.2mm – 2.1mm) areas of permissible dislocations and reveal no symmetry in horizontal direction. This refers to the pair of bearings No. 3 and 4, and the pair of bearings No. 5 and 6, mounted in one casing and linked by quite stiff couplings. The permissible dislocation of the bearing No. 3 is by about 0,45 mm higher to the right than to the left, while the permissible dislocation of bearing No. 4 is by about

0,4 mm higher to the left than to the right. The permissible dislocation of bearing No. 5 is by about 0,52 mm higher to the right than to the left, while that of the bearing No. 6 is by about 0,62 mm higher to the left than to the right.

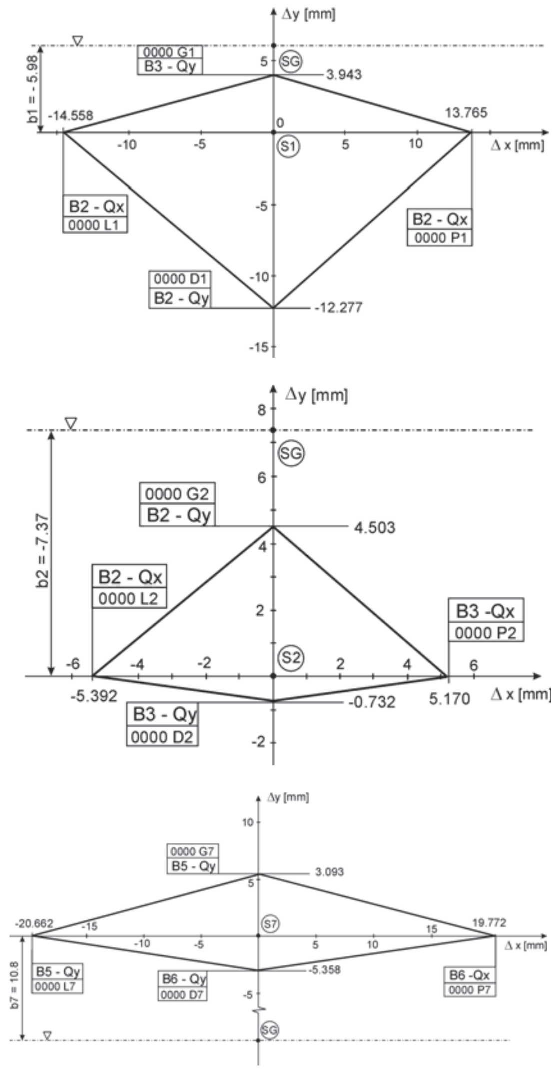


Fig. 2. The map of permissible displacements of bearings nr 1, 2 and 7.

Fig. 3 show, that no matter which bearing, No. 3 or No. 4, is displaced and in which horizontal direction, it always results in surpassing permissible horizontal reacting force in bearing No. 3. Fig. 4 reveals that dislocating bearing No. 5 to the right leads to the same effect as dislocating bearing No. 6 to the left and that amplitude of relative vibrations is surpassed in bearing No. 4. Similarly, dislocating bearing No. 5 to the left produces exactly the same effect as dislocating bearing No. 6 to the right. In this case the permissible vertical load is surpassed in bearing No. 4.

Vertical dislocations of all bearings reveal very strong asymmetry of permissible areas with respect to the basic location. The permissible dislocations in one direction are from 2 to 6 times as high as in the opposite direction. The relatively weakest asymmetry is observed for permissible vertical dislocations of bearing No. 7 (Fig. 2), while the strongest – for bearing No. 3 (Fig. 3).

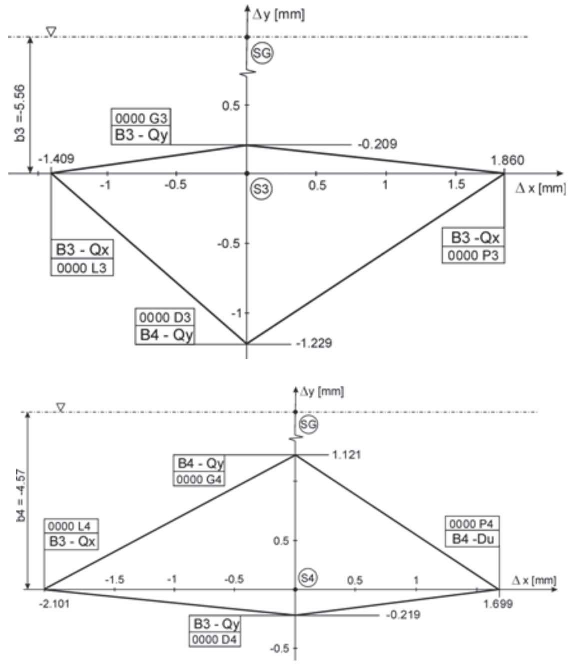


Fig. 3. The map of permissible displacements of bearings nr 3 and 4.

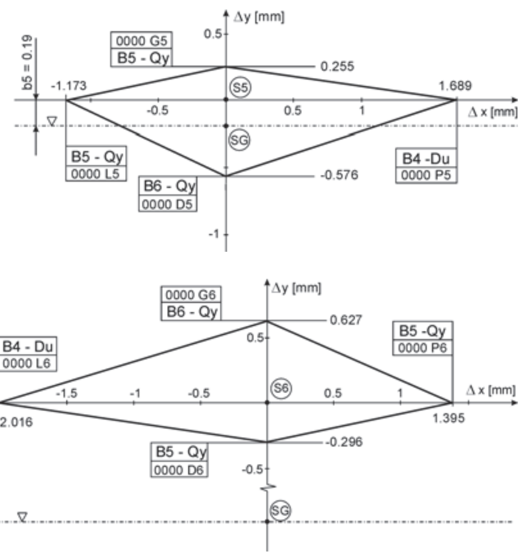


Fig. 4. The map of permissible displacements of bearings nr 5 and 6.

Comparing ranges of permissible vertical dislocation of bearings No. 1, 2 and 3 leads to the conclusion that these bearing impose load to each other. A hypothesis can be formulated that bearing No. 2 is located too high with respect to the bearings No. 1 and 3. One can expect that moving bearing No. 2 down by 1 to 2 mm would result in higher symmetry of diagrams for bearings No. 1, 2, 3 and, as a consequence, wider tolerance of the machine to possible emergency dislocations of those bearings.

It is noteworthy that, almost perfect symmetry is observed in areas of vertical dislocations for pairs of bearings mounted on the same bearing supports. The ratio of permissible vertical dislocations of the pair of bearings No. 3 and 4 equals about 5, while that for the pair of bearings No. 5 and 6 is approximately equal to 2. In each pair the first bearings (No. 3 and 5, respectively) reveal higher permissible dislocations downward, while the other bearings (No. 4 and 6) – upward. Also the absolute values of corresponding vertical dislocations of those bearings do not differ much.

Fig. 3 leads to the conclusion that moving bearing No. 3 up generates the same effect as moving bearing No. 4 down, namely surpassing permissible vertical load of bearing No. 3. At the same time moving bearing No. 3 down has the same result as moving bearing No. 4 up, which is surpassing permissible vertical load of bearing No. 4. The identical situation is in case of pair of bearings No. 5 and 6, which directly results from Fig. 4. Moving bearing No. 5 up has the same effect as moving bearing No. 6 down and moving bearing No. 5 down leads to the same effect as moving bearing No. 6 up.

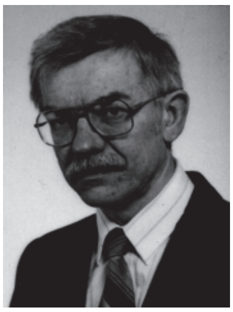
The symmetry of ranges of permissible dislocations of pair of bearings No. 3 and 4, and No. 5 and 6 leads to similar conclusions as in case of the system of bearings No. 1, 2, and 3. In each pair, the bearings impose mutual load to each other. One can suspect that bearings No. 3 and 5 are moved to high with respect to the locations of bearings No. 4 and 6, respectively. Moving bearing No. 3 down by about 0,5 mm, or moving bearing No. 4 up by the same distance would result in larger symmetry of the diagrams. The same effect would be obtained by moving bearing No. 5 down by about 0,2 mm, or moving bearing No. 6 by the same distance in opposite direction. Such correction of the bearing locations would result in higher tolerance of the machine to possible emergency dislocations of those bearings.

6. CONCLUSIONS

1. The bearings located at a relatively large distance from other bearings (nr 1, 2, 7) reveal a high range of permissible dislocations in horizontal direction. These are approximately equal to the right and to the left. This suggests the lack of mutual interaction between those bearings in horizontal direction.
2. For the pairs of bearings located close to each other (bearings No. 3 and 4, and No. 5 and 6), one bearing in the pair reveals the range of permissible dislocations in a given horizontal direction close to that revealed by the other bearing in the opposite direction. As a result, the two bearings in the pair impose the load to each other.
3. The ranges of permissible vertical dislocations of all bearings are highly asymmetric with respect to the basic location. The ratio of permissible upward and downward dislocations is between 2 (for bearing No. 7) and 6 (for bearing No. 3).
4. All bearings reveal alternate directions of the high and low ranges of permissible vertical dislocations. This testifies to strong vertical load imposed to each other by the adjacent bearings.
5. The asymmetry in the areas of permissible dislocations of individual bearings suggests that the constructional kinetostatic line of rotors is not optimal from the point of view of machine's resistance to bearing dislocations and the bearing location would be corrected.

REFERENCES

[1] Vance J. M., 1985, "Rotordynamics of Turbomachinery", A Wiley – Interscience Publications, New York.
[2] Hamrock B.J., 1994, "Fundamentals of Fluid Film Lubrication", McGraw-Hill Inc, New York.
[3] Rybczynski J., 2001, "Analysis of additional vibrations encountered during investigations of rotor dynamics", Transactions of the Institute of Fluid-flow Machinery, nr 108, p. 95-111.
[4] Rybczynski J., Luczak M., 2001, "Determination of the Acceptable Area of the Mutual Displacements of the Turboset Bearings Regarding Vibration and Loading", Polish Maritime Research No 1 (27), March 2001, p. 7-10.
[5] Kiciński J., Prońska A., 2004, „Identyfikacja modelu obliczeniowego Turbozespołu 13K215", Oprac. wew. IMP nr 4068/04.
[6] Kiciński J., Drozdowski R., Materny P., 1997, "The non-linear analysis of the effect of support construction properties on the dynamic properties of multi support rotor systems", Journal of Sound and Vibration, 206(4), pp. 523-539.



Jozef RYBCZYNSKI
Graduated from Gdansk University of Technology at Mechanical Engineering Faculty in 1971. Doctor of Philosophy in discipline mechanics. Employed since 1972 in The Institute of Fluid Flow Machinery, PAS in Gdansk in Mechanics of Machines Department. Area of interest: dynamics of rotors supported in slider bearings, journal bearings, machine diagnostics, experimental investigations into the rotor and foundation vibration, measurement techniques.