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Determination of the acceptable area of the mutual displacements of the turboset bearings regarding vibration and loading

SUMMARY

The paper presents results of a computer analysis of dynamic behaviour of 13K215/28 power-station turboset of 200 MW output. The aim of the analysis was to determine influence of bearing mutual displacements on vibration state of the entire turboset as its bearings sometimes work in displaced positions due to thermal effects or assembly errors.

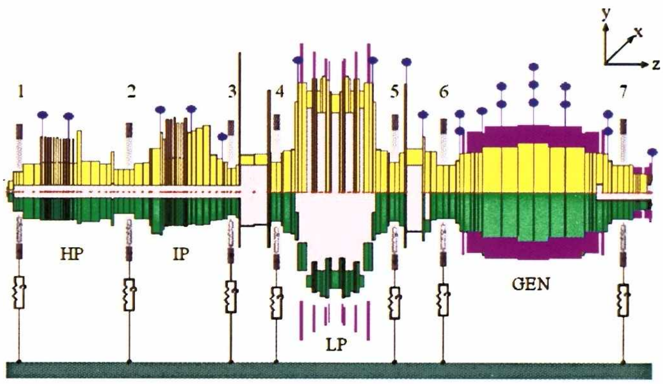
The analysis was carried out with the use of MESWIR computer software developed by the Institute of Fluid Flow Machinery, Polish Academy of Sciences, (IFFM-PAS), Gdańsk.

INTRODUCTION

Stable and safe work of a turboset is its main operational criterion. Vibrations excited by the turboset bearings seriously influence stable operation of the entire turboset. One of the causes of bearing vibrations are their displacements due to thermal effects or assembly errors. Knowledge of influence of mutual displacements of the bearings on vibrational properties of turboset is very helpful for diagnostics and exploitation technique of machines. Research on the topic was undertaken by the IFFM-PAS at the request of some industrial customers. This paper presents results of the dynamic behaviour calculations of the 13K215/28 power-station turboset of 200 MW output. The simulations were aimed at determining acceptable areas of the mutual displacements of the bearings with respect to the turboset's vibration and the loading transferred by its bearings. The calculations were performed by means of MESWIR computer software based on the finite element method, developed by the IFFM-PAS, Gdańsk [1,2].

A computation model of the 13K215 turbogenerator was prepared on the basis of its geometrical and operational data [3]. The calculation model was „identified" in order to obtain conformity of the results of the turbogenerator's dynamic calculations and the operational data achieved from measurements carried out on a real object being in a „very good" technical state. The identification consisted in performing series of calculations during which the initial calculation model and input data were step-by-step corrected [4,5]. The so determined calculation model was assumed the basic case, i.e. the reference basis for the results of the simulative calculations.

The calculation model of the machine's rotor is presented in Fig.1 which shows the division of the rotor into finite elements (FEs), and points of its support by the bearings no.1÷7. (Note : The drawing scale along the rotor axis and that transverse to it, are different).



Elements coloured in :
- elastic cylindrical FEs assumed with the respect of their mass
- elastic cylindrical FEs assumed with the respect of intertia moment of their cross-section areas
- rigid disc FEs *- nodes* *- residual unbalance masses*

Fig.1. FEM calculation model of 13K215 turboset rotor

A range of allowable displacements of the bearings both with respect to the turboset vibration and their loading can be determined from the two points of view :

- ➔ of the operation state of the entire turboset (its serviceability), and
- ➔ of the loading and stable operation of the individual bearings.

In the presented investigations results of calculations were analyzed with respect to the first of the cases, i.e. in order to assess influence of the diplacements of any of the bearings on serviceability of the entire machine.

To assess acceptable location areas of the bearings the numerical limits for the displacements of individual bearings were determined with respect to the following three criteria whose exceedance was assumed not permissible :

- absolute vibration criterion
- criterion of relative vibrations (of pin against sleeve)
- bearing sleeve loading criterion.

The acceptable area of bearing displacements with respect to turboset's serviceability is a common part of the three respective areas (this is the most inner area shown in Fig.2 to 7).

Vibration criteria

The allowable limits of the relative vibration displacements were specified in compliance with ISO 7919-2 standard, and for rms absolute vibration velocity – with ISO 10816-2 standard.

The following values were assumed limiting :

✦ *for the relative vibrations :*
the warning-state vibration displacement $Sx_p = Sy_p = 165$ μm, and

✦ *for the absolute vibrations :*
the rms vibration velocity $(Vrms)_{Sx_p} = (Vrms)_{Sy_p} = 7.5$ mm/s

The entire turboset is assumed serviceable if the above specified limits are not exceeded at any of its bearings.

Loading criterion

The average pressure value $p_p = 2$ MPa on the bearing sleeve surface was assumed limiting. On this basis the permissible values were determined for the reaction forces at each bearing, both for their horizontal and vertical components.

INVESTIGATION METHOD

One of the bearings, or a group of them were simulated systematically displaced against their basic positions to determine the limiting values of the displacements. The basic positions were assumed those resulting from a „hot" kinetostatic line. (i.e. a real form of shaftline axis at a steady-state temperature of the turbine in operation, resulting from gravity forces and thermal dilatations of supports).

In order to develop an entire set of inference maps it would be necessary to calculate mutual influence of the displacements of all bearings to each other. This appeared unrealistic due to time and technical reasons.

Therefore the following limitations were accepted :

➲ bearings no.1,2,3,4 and 7 to be individually displaced with basic positions of the remaining ones maintained

➲ bearings no.5 and 6 to be simultaneously displaced to each other and against their basic positions, because the two were known from service experience the most troublesome. They were located close to each other, and appeared also high-loaded ones; hence their mutual displacements would be very limited

➲ the bearings to be displaced only in the x-y plane, perpendicular to the shaft rotation axis : upwards and downwards, to the left and right, against their basic positions.

All the above specified simulations consisting in mutual shifting of the bearings, were carried out at the rated rotational speed of the turbine n = 3000 rpm, and at the same „residual" unbalanced mass distribution along its shaftline.

CALCULATION RESULTS

On the basis of the performed calculations the detail limiting areas were determined of the bearing displacements, acceptable from the point of view of the particular criteria (see Fig.2 to 9).

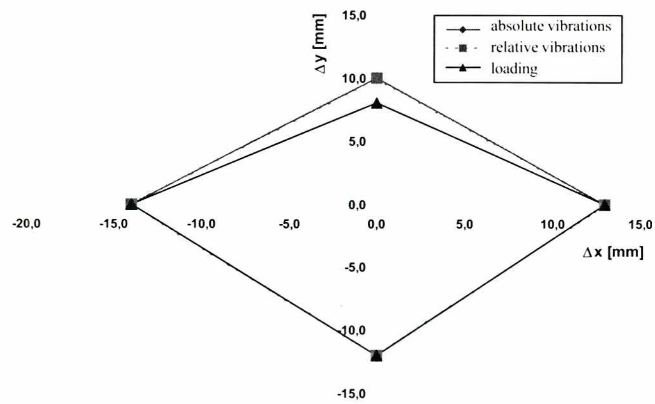


Fig.2. Permissible displacements of bearing no.1

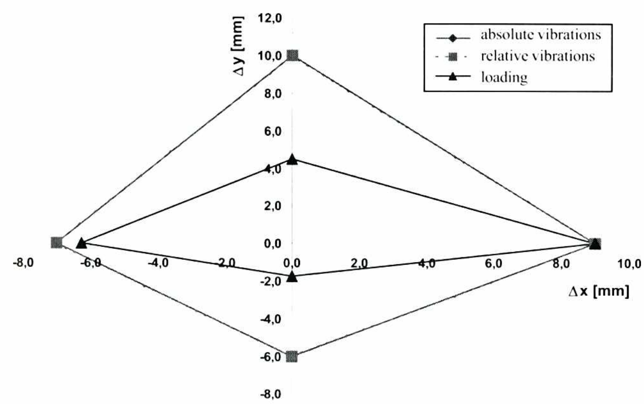


Fig.3. Permissible displacements of bearing no.2

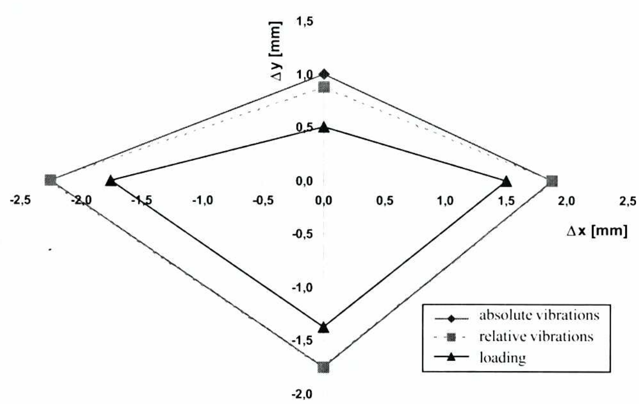


Fig.4. Permissible displacements of bearing no.3

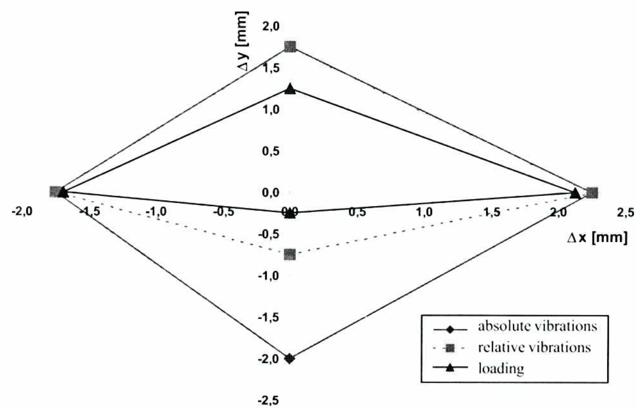


Fig.5. Permissible displacements of bearing no.4

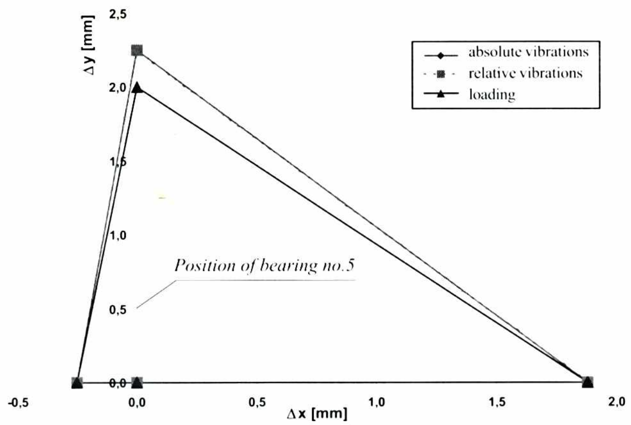


Fig.6. Area of the permissible displacements of bearing no.6 if bearing no.5 is shifted up by 0.5 mm

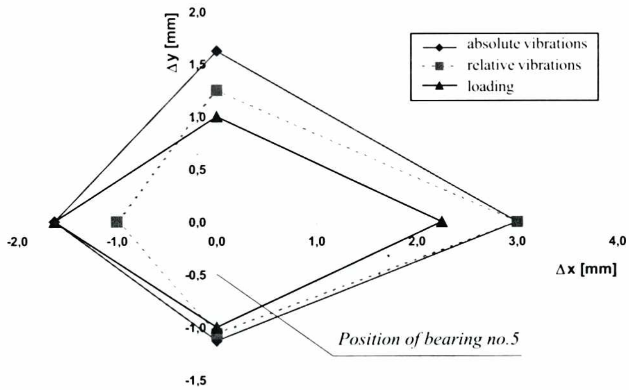


Fig.7. Area of the permissible displacements of bearing no.6 if bearing no.5 is shifted down by 0.5 mm

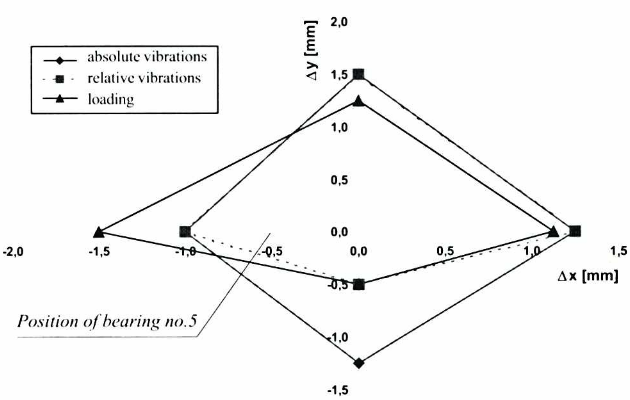


Fig.8. Area of the permissible displacements of bearing no.6 if bearing no.5 is shifted to the left by 0.5 mm

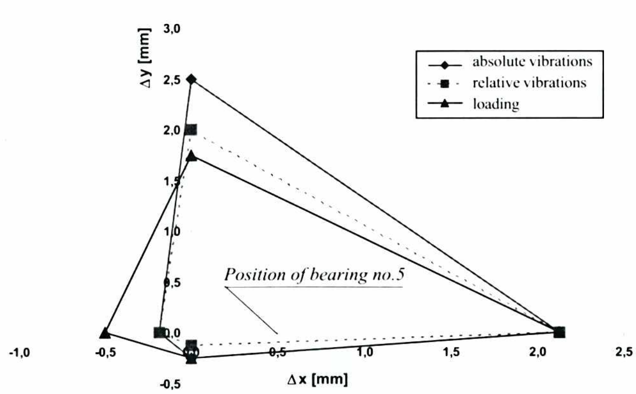


Fig.9. Area of the permissible displacements of bearing no.6 if bearing no.5 is shifted to the right by 0.5 mm

The areas of the permissible displacements of the bearings no. 1,2,3, and 4 are presented in Fig.2 to 5, and those of the pair of the bearing no.5 and 6 – in Fig.6,7,8 and 9.

With the use of the diagrams it can be determined by what value and in which direction a given bearing is permitted to be displaced without exceeding the limit values resulting from the particular criteria. For instance according to Fig.5 : bearing no.4 is permitted to be shifted down by :

- 2.00 mm - with respect to bearing sleeve loading
- 0.75 mm - with respect to relative vibrations, and
- 0.25 mm - with respect to absolute vibrations.

DISCUSSION OF THE INVESTIGATION RESULTS

In Fig.2,3,4, and 5 the following relationship can be observed : when shifting the bearings no.1,2,3 and 4 the permissible loading values are exceeded first, next the permissible values of the relative vibrations of the pin against the sleeve, and finally those of the absolute vibrations of the sleeve.

It can be explained by the fact that those bearings are highly lenticular, therefore those of antivibration properties. It simultaneously means that the bearings in question have a large safety margin regarding the displacements relative to the geodesic line. Bearing no.7 is of an even greater safety margin (the result is not presented in the attached figures).

The permissible displacement range for bearings no.1,2, and 7 is wider by one order than that for bearings no.3,4,5 and 6. Hence these bearings seem resistant to assembly random errors, foundation settlement, or thermal dilatations of bearing stands.

From Fig.6,7,8 and 9 it can be observed that the permissible ranges of the relative displacements of bearing no.6 against bearing no.5 with respect to the sleeve loading criterion, are close to each other, and they do not depend on the current position of bearing no.5 against its basic position.

The permissible values of the relative displacements of bearing no.6 against bearing no.5 are as follows :

- ✧ downwards : $\Delta y_p = -0.5$ mm
- ✧ upwards : $\Delta y_p = 1.5$ mm
- ✧ to the right : $\Delta x_p = 1.5$ mm
- ✧ to the left : $\Delta x_p = -1.0$ mm

Such small permissible areas of the mutual displacements of those bearings can be explained by the fact that the bearings are very close placed to each other. However the observation that the permissible displacement values of bearing no.6 against bearing no.5 do not depend on the absolute displacements of bearing no.5, results from their location faraway from the remaining bearings (see Fig.1).

The largest displacements of bearing no.6 are allowed in the case when bearing no.5 is shifted downwards by 0.5 mm against its basic position.

In Fig.6 (where bearing no.5 is shifted by 0.5 mm upwards against its basic position) it can be observed that no downward displacement is permissible. It can be explained by the fact that bearing no.6 has almost double load-carrying margin while that of bearing no.5 is of only 18% (it is result of detail calculations not presented here). Hence any downward displacement of bearing no.5 does not result in overloading of bearing no.6.

When considering displacements of bearings no.5 and 6 with respect to the absolute vibration criterion the similar relationship can be observed as in the case of the permissible loading criterion. Namely, the permissible displacements of bearing no.6 against bearing no.5 (irrespective of the location of the latter against its basic position) were approximately determined as follows :

- ⇒ downwards : $\Delta y_p = -0.5$ mm
- ⇒ upwards : $\Delta y_p = 1.5$ mm
- ⇒ to the right : $\Delta x_p = 2.0$ mm
- ⇒ to the left : $\Delta x_p = -0.7$ mm

In this case also the allowable displacement range of bearing no.6 is the largest if bearing no.5 is shifted downward by 0.5 mm.

An detail analysis of the calculation results reveals that if the vibration criterion of the allowable displacements is taken into account the allowable vibration levels are simultaneously exceeded in bearings no.4,5,6 and 7 when the allowable displacement of any of these bearings is exceeded. The vibrations then increase violently. It can be explained as follows :

Displacing any of the bearings changes the loading state of the shaftline, and causes its deflection against its basic form. At a sufficiently large deflection the centrifugal forces cause an additional change of loading of the bearings, which makes their loads increase or decrease. The relieved bearings are prone to self-excited vibrations. These vibrations can generate vibrations of other bearings. Initially they are moderate, oil-whirl vibrations which next turn into oil--whipping action.

In the consequence of the made assumption that:

any exceedance of any of the acceptance criteria in any of the bearings (not necessarily that displaced) makes the entire turboset unserviceable

it should be assumed that in practice the acceptable bearing displacements respective to the geodesic line are those contained within the smallest areas limited by the lines of the permissible absolute and relative vibrations and sleeve loading, shown in Fig.2 to 9.

FINAL REMARKS

- The performed calculations made it possible to determine the acceptable areas of mutual bearing displacements with respect to turboset vibrations and loading of its bearings.
- Computational simulations are suitable for generating many relationships between bearing displacements and dynamic and static states of particular bearings. The relationships in a form of diagnostic relations may enrich knowledge bases of expert systems.
- It is also possible to determine influence of displacing a particular bearing on the working state of the entire turboset and its other bearings (as the presented results are only a fragment of broad investigations of the diagnostic relations occurring in a turboset).
- The calculation results are deemed a very useful source of the knowledge applicable to service procedures of the power-station turbosets.
- It is also possible to extend the calculation range to reveal also other diagnostic relations interesting from the industrial application point - of -view.

Appraised by Jan Kiciński, Prof.,D.Sc.,M.E.

NOMENCLATURE

IFFM-PAS	-	Institute of Fluid Flow Machinery, Polish Academy of Sciences
Δx	-	value of bearing horizontal displacement with the respect of numerical convergence of the calculation algorithm
Δy	-	value of bearing vertical displacement with the respect of numerical convergence of the calculation algorithm
Sx	-	displacement values of relative horizontal vibration
Sy	-	displacement values of relative vertical vibration
$(V_{rms})_x$	-	rms velocity values of absolute horizontal vibration
$(V_{rms})_y$	-	rms velocity values of absolute vertical vibration
p	-	values of average pressure on sleeve surface

Indices

p	-	permissible value
rms	-	root mean square value

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Conference

AUTUMN SEMINARS

Two last-of-the-year 2000 seminars were lately held by the Regional Group of the Utility Foundations Section (ŚPE), Polish Academy of Sciences :

The first of them – on 16 November in Olsztyn, hosted by Mechanical Engineering Institute, Faculty of Engineering Sciences, Warmia-Mazury University. Three of the presented papers dealt strictly with agricultural engineering topics, and 7 remaining ones had a more general, scientific-technical character :

- *Influence of gaseous phase on propagation velocity of shock wave within multi-layer hydraulic pipe* (by T. Nałęcz and P. Pietkiewicz)
- *Numerical modelling of cavitation in fluid-flow phenomena* (by J. Badur and W. Sobieski)
- *Dynamics of rotary machines : rotor – supports system* (by W. Miąskowski)
- *Deterministic prediction of structural life by using numerical methods of mechanics* (by J. Badur and W. Dudda)
- *Advanced techniques of coal conversion in gas – steam systems* (by J. Badur and P. Szutkowski)
- *Diagnostic method of machines with the use of a hybrid expert system* (by A. Rychlik)
- *Diagnosing dynamic state equilibrium of machine aggregate* (by P. Szczyglak)

The first five papers were prepared by authors from the Department of Mechanics and Foundations of Machine Construction, and the two remaining ones – from the Department of Exploitation of Vehicles and Machines, Warmia-Mazury University.

The next seminar was held on 6 December 2000 in Mechanical Faculty, Technical University of Gdańsk.

Its numerous audience had the opportunity of being acquainted with four papers prepared by young research workers of the Faculty. It was the following papers dealing with interesting tribological topics :

- *Investigation methods of frictional wear under oscillatory vibrations in sliding plane* (by R. Gawarkiewicz)
- *Research problems of sliding materials under angular micro-oscillations* (by J. Gliwiński)
- *Investigation of temperature influence on fatigue strength of sliding layer of AlSn20Cul alloy* (by K. Kurzych)
- *Three – surface, anti-vibration bearing of Y –type* (by A. Olszewski)

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