JAN KICIŃSKI, Assoc.Prof.,D.Sc.,M.E.
PIOTR MATERNY, D.Sc.,M.E.
Polish Academy of Sciences
Institute of Fluid Flow Machinery
Gdańsk

Examples of driving system structures modification by using mode analysis method

SUMMARY

Dynamic characteristics of a modernized low pressure casing of the 13K215 turbine and a foundation frame of the large gabarite stand for testing multi-span rotor systems as well as the bending stiffness of a flexible coupling which connects rotors of the 13K215 turbine set were investigated with the use of mode analysis approach. The ABAQUS FEM analysis computer software system was used.

The presented examples of the mode analysis application refer to the real objects. In all the cases interesting information was gathered and utilized to modify dynamic characteristics of the investigated structures. It speaks for practical applicability of the analysis method particularly while connected with FEM application.

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OBJECT AND SCOPE OF INVESTIGATIONS

Objects of investigations were the following structures:

- a modernized low pressure casing of the 13K215 turbine and
- a foundation frame of the large gabarite stand for testing characteristics of the rotor-bearings-supports model systems.

In both cases the main task was to determine critical frequencies, natural vibration modes and course of forced vibrations of the structures in question, as well as to calculate the dynamic stiffness coefficients of bearing supports within the structures. Additionally, the bending stiffness analysis is presented of a flexible coupling which connects rotors of the13K215 turbine set.

The only applicable theoretical analysis method is the FEM due to complexity of the structures. Therefore the FEM analysis computer software systems being actually at disposal of the authors (a.o. ABAQUS of Hibbitt, Karlsson & Sorensen) were applied. In both cases the structures were discretized with an appropriately dense FEM mesh (of several tens of thousands of d.o.f.' s). The mode analysis is very useful in this case mainly in terms of computation time. Particularly it can be applied in a situation where linear range of the considered relations is satisfactory.

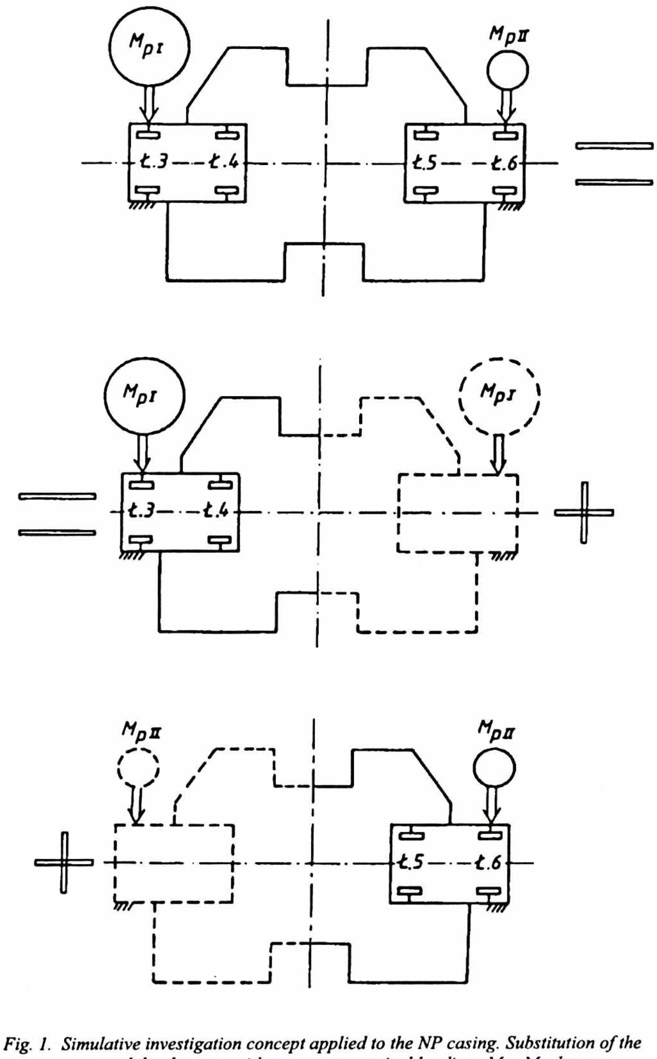


Fig. 1. Simulative investigation concept applied to the NP casing. Substitution of the real load system with two unsymmetrical loadings M_{pI}, M_{pII} by sum of two load systems symmetrical in respect to M_{pI} and M_{pII} loadings

DYNAMICS OF NP CASING

The NP casing, in spite of its symmetrical structure, is characteristic due to rather substantially diversified loads applied to each of its halves (the left half of the SP part side - and bearing no. 3, and the right half of the generator side - and bearing no. 6). The difference is caused by a greater load applied to the casing by the rear part of SP casing (the left half) or by the turning engine (the right half).

In dynamic calculations the influence was assumed possible to be substituted by the masses M_{pI} and M_{pII} concentrated at selected control points. Magnitudes of the masses are equivalent to the forces causing the NP casing load unsymmetry. The approach is highlighted in Fig. 1. It means the amplitudes, phase angles, and stiffness and damping coefficients of bearing supports nos. 3 and 4 are determined with the M_{pI} mass taken into account, but those of bearing supports nos. 5 and 6 with the M_{pII} mass accounted for. Due to this the FEM mesh was prepared for one half of NP casing only, but with calculation results valid for the entire casing.

It is hard to assess how far is the assumed model close to reality. Certainly a better approach is to discretize all the NP casing with FEM mesh and to carry out calculations with the unsymmetrical loading accounted for. However, the selected approach, based on FEM modelling applied to one symmetrical half of the casing only, seems to be the best solution.

Second assumed idealization deals with the stiffness of the foundation regions which support the NP casing. The foundation was assumed perfectly stiff in the regions supporting the casing, as the data about the dynamic stiffness and damping coefficients of the sole foundation were not available. This is a rather important simplification because the bearing supports of no. 3 and 6 are located very close to the foundation modelled in the simplified way. It means that the values of the to-be-calculated stiffness coefficients of the bearing supports (particularly their vertical components) could be unrealistically overestimated.

A sinusoidal variable exciting force was assumed to act in the parting planes of the bearing supports in the x (horizontal) and y (vertical) directions.

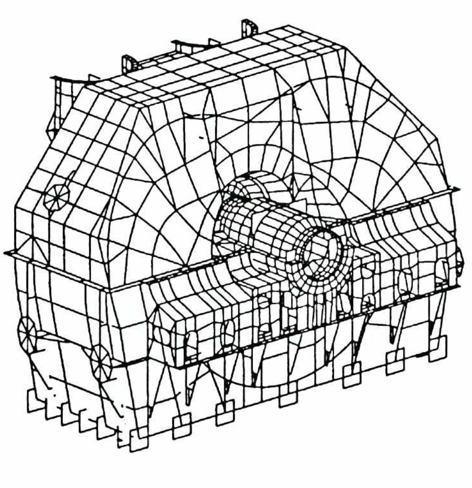
Components of the exciting forces being applied successively to the supports no. 3, 4, 5, 6 generate forced vibrations of the entire casing. Displacements of the respective supports only were registered in view of the aim of the investigations.

If amplitude and phase angle values are known the stiffness coefficients $(c_{11})_j$, $(c_{22})_j$ and damping coefficients $(d_{11})_j$, $(d_{22})_j$ can be calculated in function of excititation frequencies.The amplitude and phase angle courses make determining critical velocities of the bearing supports of the NP casing possible.

In the calculations structural damping was assumed in a form of the so-called direct modal damping defined by the coefficient ξ_m which is the ratio of the actual damping for a given vibration mode and the critical damping. For all vibration modes $\xi_m = 0.03$ was assumed (in compliance with [1]).

The symmetrical half of NP casing was given FEM mesh on the basis of casing's technical drawings. 3339 plate and beam finite elements as well as point masses were used and in result 10545 nodes and 59406 d.o.f.'s obtained. This large number of d.o.f.'s for one half of NP casing only was assumed to obtain possible accurate modelling of the supports and the so-called bearing basket which are the particularly important structural regions in view of the investigation purpose. Fig. 2 shows the assumed FEM mesh for the half of NP casing.

a)



b)

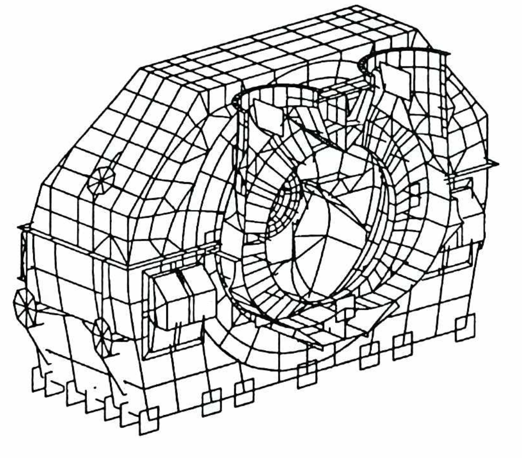


Fig. 2. Discretization of the symmetrical half of NP casing with the use of FEM mesh a) outer view, b) inner view

RESULTS OF CALCULATIONS

In the mode analysis method natural vibration frequencies and modes first are determined and forced vibration then are calculated within the frequency ranges corresponding to the modes and assuming an external, sinusoidally variable force to be applied.

In Fig. 3, 5th, 6th and 7th natural vibration modes of NP casing are shown where characteristic vibrations of the bearing basket can be observed. It may imply the stiffness of supports no. 4 and 5 to be too low in relation to that of outer supports no. 3 and 6.

MODE 5

MODE 6

MODE 7

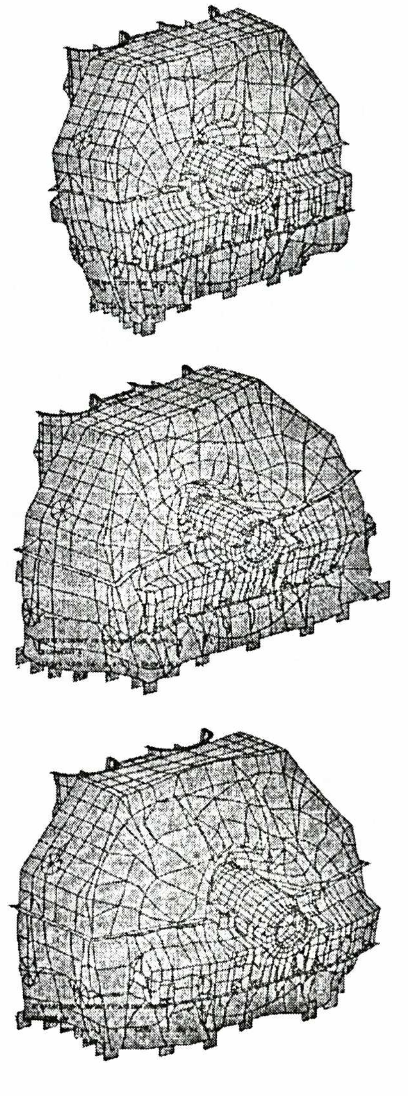


Fig. 3. Selected modes of natural vibration of NP casing

Calculation results of the NP casing forced vibration revealed at the bearings no. 3, 4, 5 and 6 are given in Fig. 4. The results are composed of the symmetric and antisymmetric cases because the calculations have been performed utilizing the concept of load and structure symmetry highlighted in Fig.1. Vector sum of the partial solutions provides the searched final result and is valid for the entire NP casing.

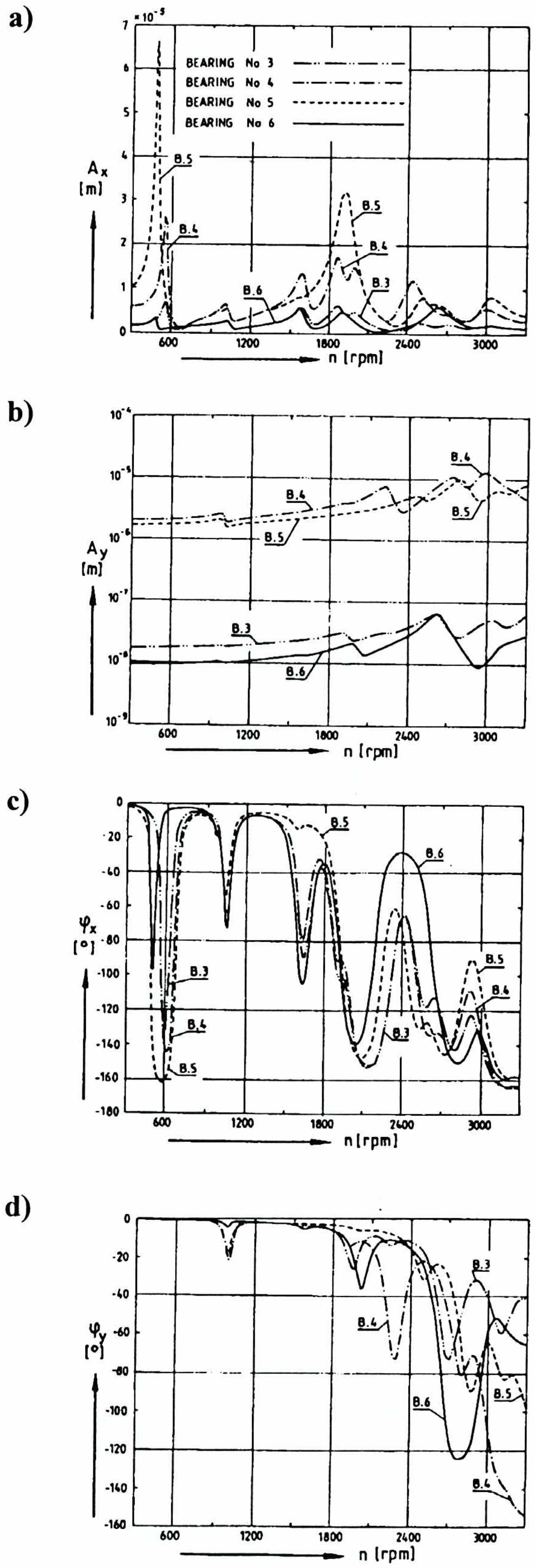


Fig. 4. The vibration amplitudes (A_x , A_y) and phase angles (φ_x , φ_y) versus the rotation speed (n) at supports no. 3, 4, 5 and 6

In Fig. 4 the symbols A_x , A_y , φ_x , φ_y denote the horizontal and vertical components of the vibration displacement amplitudes and phase angles respectively. The inscription BEARING No. 3 or B.3 etc denotes both the point where the excitation force components were applied and the amplitudes and phase angles were registered. The response courses are shown for four supports simultaneously and in the common scale to ease making direct comparisons.

In the figures several resonances (within 5 to 55 Hz range) of the horizontal component vibration displacement amplitudes A_x at the bearing supports no. 4 and 5 can be observed. The first, most powerful can be found within 8 to 9 Hz range, the second, equally distinct within 32 to 33 Hz. Several less distinct resonances also can be observed e.g. that within 50 to 51 Hz range. The horizontal vibration resonances at the bearing supports no. 3 and 6 are much less developed.

The courses of the vertical vibration displacement amplitudes A_y at the rotor supports of NP casing greatly differ from those horizontal. First of all a higher vibration level of supports no. 4 and 5 than that of supports no. 3 and 6 can be observed obviously in consequence of the perfectly stiff foundation assumed within the NP casing seating regions. Resonance velocities of the vertical vibration components, especially of the supports no. 4 and 5, underwent a shift towards higher frequencies with a simultaneous increase of vibration level.

In Fig. 5 courses of the dynamic stiffness coefficients c_{11} and c_{22} for support no.5 versus frequency are presented when transforming the results given in Fig. 4 on the basis of the following definition of the dynamic stiffness coefficients:

$$\left(c_{i,i}\right)_j = \frac{\left(Q_i\right)_j}{\left(A_i\right)_j}$$

where:

$\left(Q_i\right)_j$ - the amplitude of the force exciting vibrations at the support j in the direction i (i = x , y)

$\left(A_i\right)_j$ - the displacement amplitude of vibrations at the support j in the direction i (i = x , y)

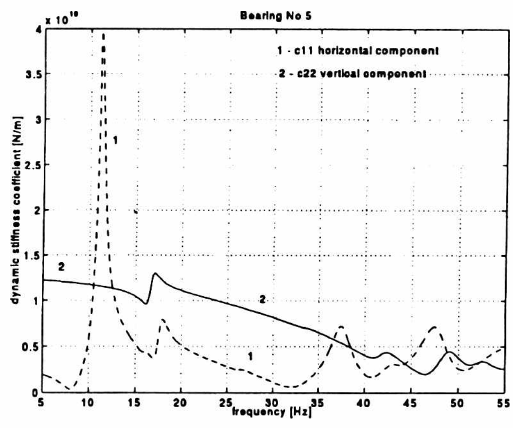


Fig.5. Courses of the dynamic stiffness confficients c_{11} and c_{22} for support no. 5 versus frequency

It can be stated from the example given in Fig. 5 that the dynamic stiffness coeficient c_{22} (vertical) continuously decreases as the excitation frequency increases to reach its minimum at about 50 Hz (nominal speed).This is accompanied by the horizontal stiffness coefficient c_{11} dropping. Their cumulated effect around 50 Hz frequency could be detrimental to dynamic characteristics of the rotor seated onto the so designed bearing baskets. The investigated casing may therefore appear to be the resonance structure around the nominal speed of 3000 rpm and its stiffening necessary. Searching for different variants of stiffening the NP casing is out of the scope of this paper.

MODIFICATION OF TESTING STAND FOUNDATION FRAME

The large gabarite stand for testing characteristics of the multi-span rotors with slide bearings was built in the vibration diagnostics laboratory of the Fluid Flow Machinery Institute, Gdańsk. Its foundation frame is an

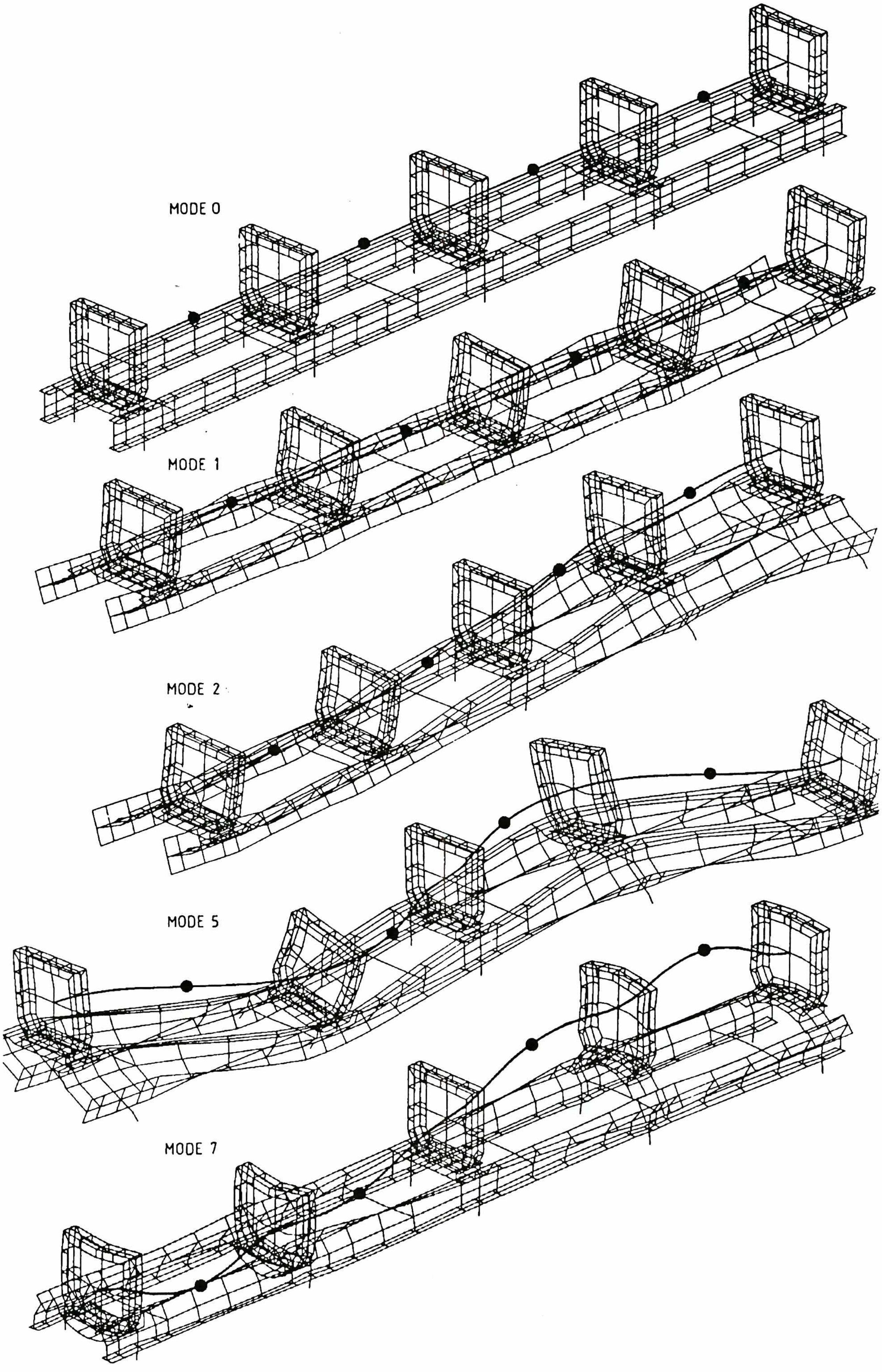


Fig. 6. Natural vibration modes of the first variant of the foundation frame

important element. Frame structure is to have its first critical frequency of a value exceeding 100 Hz, viz. above the frequency range to be investigated. The first variant of the structure was designed and its modal analysis performed. The analysis results are presented in Fig. 6.

It results from the figure that the first natural vibration mode is 35.2 Hz, therefore it is placed within the assumed test range. The computed vibration modes imply an improper design of many important structural components (too flexible bearing sleeve footing, unsufficient stiffening of the main frame etc). After the performed analysis a modified structure was proposed and its modal analysis carried out again.The results are shown in Fig. 7. The first natural vibration frequency of 187 Hz was obtained, i.e. out of the assumed test range. This design was implemented in practice.

Only 1/8 part of the object was modelled when utilizing structural symmetry. It made applying a relatively dense FE mesh possible with an acceptable calculation time. The model is presented in Fig. 8. The flexible coupling bending stiffness was determined when applying relevant loads to the model (analogical to those in the case of the casing) and relating them to the resultant displacements. It was astonishing to find the obtained coupling stiffness value equal to the bending stiffness of the rotors connected by the couplings. Having in view bending properties only one can consider the flexible coupling as a rotor segment of a constant diameter. In Fig. 9 an equivalent stress distribution on coupling's surface is exemplified.

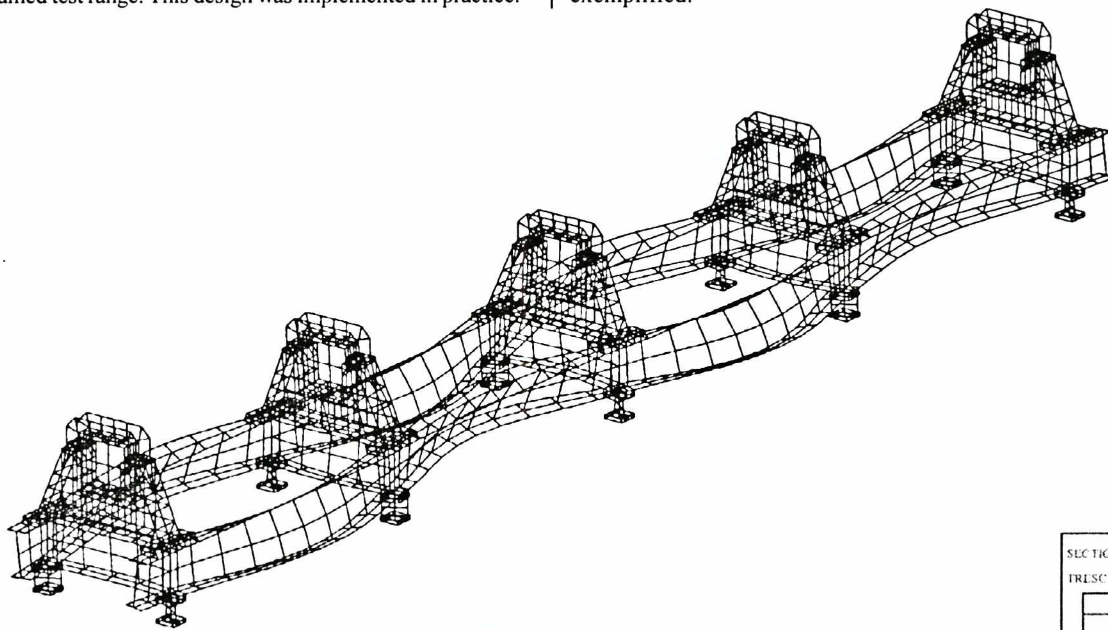


Fig. 7. One of the natural vibration modes of the modified foundation frame

DETERMINATION OF FLEXIBLE COUPLING BENDING STIFFNESS

While carrying out works aimed at dynamic properties determination of the modernized 13K215 turbine, a part of which were the presented earlier investigations of the NP casing of the turbine, it was necessary to model theoretically and calculate bending stiffness of the flexible couplings connecting rotor line segments of the turbine. Without doing this it would not be possible to perform simulative calculations of the entire rotor line of the turbine set in question.

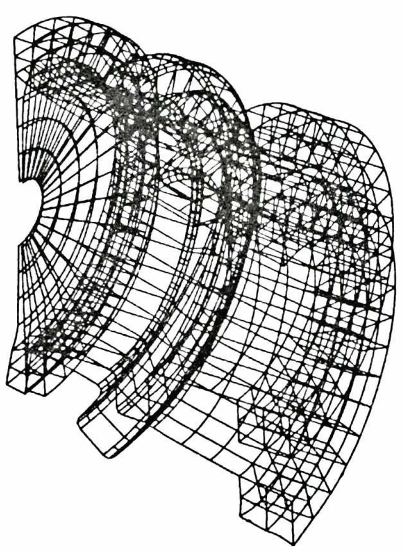


Fig. 8. The model of the 13K215 turbine flexible coupling

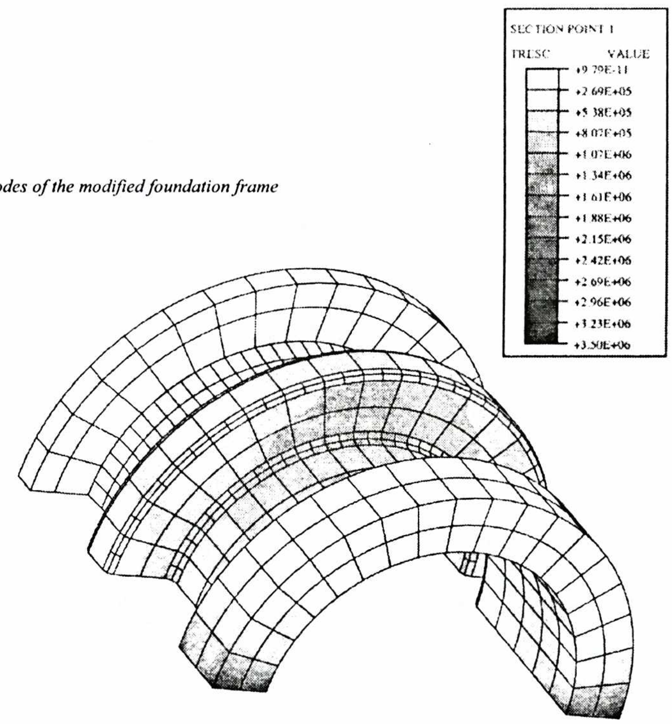


Fig. 9. Example of the equivalent stress distribution on coupling's surface

FINAL REMARKS

The presented examples of the mode analysis application refer to the real objects. In all the cases interesting information was gathered and utilized to modify dynamic characteristics of the investigated structures. It speaks for practical applicability of the analysis method particularly in connection with FE method. On the other hand however, to carry out such an advanced analysis appropriate computer systems of high quality (including work stations) and the modern software systems for solid structures FEM analysis are required.

BIBLIOGRAPHY

1. Hibbit, Karlsson & Sorensen: „ABAQUS -Theory Manual ".