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POLISH ACADEMY OF SCIENCES

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Aims and Scope

Transactions of the Institute of Fluid-Flow Machinery have primarily been established to publish papers from four disciplines represented at the Institute of Fluid-Flow Machinery of Polish Academy of Sciences, such as:

- Liquid flows in hydraulic machinery including exploitation problems,
- Gas and liquid flows with heat transport, particularly two-phase flows,
- Various aspects of development of plasma and laser engineering,
- Solid mechanics, machine mechanics including exploitation problems.

The periodical, where originally were published papers describing the research conducted at the Institute, has now appeared to be the place for publication of works by authors both from Poland and abroad. A traditional scope of topics has been preserved.

Only original and written in English works are published, which represent both theoretical and applied sciences. All papers are reviewed by two independent referees.

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Editorial

These Special Issues of the *Transactions of Fluid-Flow Machinery*, Nos. 113 and 114 contain selected papers from the International Conference on *Turbines of Large Output* devoted to commemorate 100$^{\mathrm{th}}$ Birthday Anniversary of Prof. Robert Szewalski. The conference was held in Gdańsk, Poland on September 22-24, 2003.

The Conference is a continuation of previous conferences held at the Institute of Fluid-Flow Machinery PAS in former years dedicated to technology of steam turbines. Series of conferences bearing the same name took place in the years 1962, 1965, 1968, 1993. In 1997 organised has been a conference on steam turbines and related topics, but with a slightly amended title – *Problems of Fluid-Flow Machinery.* At present at the Institute of Fluid Flow Machinery there are conduced research of fundamental character encompassing both the issues related to steam turbines and fundamentals of power engineering.

Organisers of the present conference have returned to the traditional name, i.e. Conference on *Turbines of Large Output* to mark the respect to the memory of Professor Robert Szewalski (1903-1993), the founder and a director of the Institute of Fluid-Flow Machinery for several years, and initiator of the conference series devoted to the problems of steam turbines.

The Editors are very grateful to the referees of the papers presented in this issue of the *Transactions of Fluid-Flow Machinery*: J. Badur, W. Batko, E. S. Burka, J.T. Cieśliński, P. Doerffer, A. Gardzilewicz, B. Grochal, J. Kiciński, G. Kosman, T. Król, J. Krzyżanowski, J. Mikielewicz, A. Neyman, W. Ostachowicz, R. Puzyrewski, R Rządkowski, J. Świrydczuk, M. Trela, T. Uhl and Z. Walczyk.

We appreciate cordially the manifested authors interest in our conference. Special thanks are conveyed to Ms 'Maya' Bagińska, for her fruitful editorial help related to directing the papers to reviewers and correspondence with the authors of contributions.

Prof. Jarosław Mikielewicz
Editor-in-Chief

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Managing Editor

TRANSACTIONS OF THE INSTITUTE OF FLUID-FLOW MACHINERY
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On the applicability of bar models for estimation of the set support structure oscillations

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Abstract

In this paper the problems of the construction of numerical models of different complication using simple-profile bars of the support structure of the laboratory rotor are considered. The presence in the structure of the thin-walled elements of the opened profile (double-T shape element), for which the Saint-Venant principle is not applicable and deformation of the cross-section takes place, does not allow to carry out the simple change of such elements to separate bars. For the adequate description of the double-T shape elements deformation they are modeled by bar frame structures. Results of calculations by proposed method of forced vibrations of the all rotor support structure in the frequency range till 200 Hz are in good agreement with results of experimental investigations [1], and the results of calculations of free vibrations of the same structure without foundation – with results of analysis by more exact models using plate and spatial elements.

Keywords: Rotor dynamics; Support construction; FEM; Vibrations

Vibrational state of the power set is determined mainly by dynamic behaviour of the rotor. Because oscillations of the rotor and its support structure are interconnected, it is necessary to solve the problem of rotor vibrations analysis in two stages. At first it is necessary to determine support structure dynamic characteristics, then to estimate the rotor oscillations using them. This paper is devoted to the solution of the first part of the problem.

Estimation of the support structure dynamic characteristics is possible using the finite element method. It is possible to construct the numerical model using

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bar, plate and three-dimensional elements. Because the calculation of dynamic characteristics in wide frequency range requires large computational expenses, in the low-frequency region it is necessary to use simplified bar models.

At the Institute for Mechanical Engineering Problems of the National Academy of Sciences of Ukraine the authors developed and realized in the form of the software system for PC a method for calculation of free and forced oscillations of spatial bar structures using the finite element method. High speed of calculation is realized by means of using of effective algorithms for sparse matrices of special structure which are fully stored in the computer working storage. Post-processing of results permits to carry out animation of shapes of free and forced oscillations as well as graphic representation of amplitude-phase-frequency characteristics or elements of the matrix of dynamic compliance of selected points of structure.

At the calculation of the matrix of dynamic compliance in given frequency range for given points of structure and directions the system of equations with necessary number of right sides is solved.

Development and verification of the method has been carried out in collaboration with Institute of Fluid-Flow Machinery of Polish Academy of Sciences in Gdańsk at the analysis of the laboratory rotor support structure [1], presented in Fig. 1.

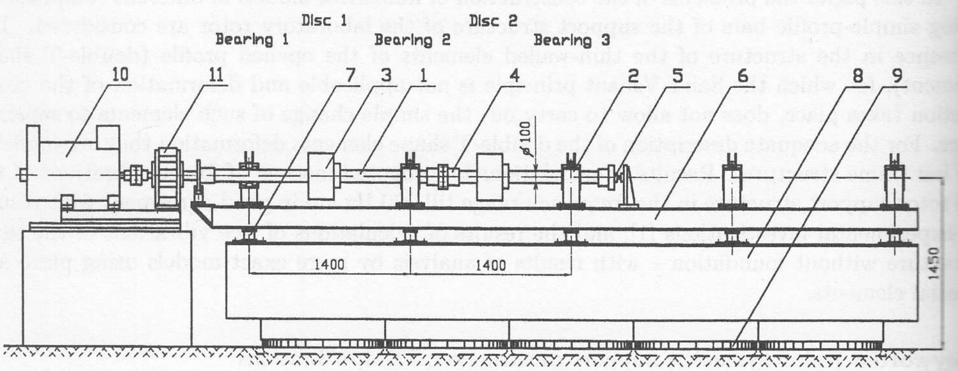

Figure 1. Rig for investigations of rotor and bearings dynamics; 1 – shaft, 2 – bearings stand, 3 – bearings loading disc, 4 – coupling, 5 – resistance bearing, 6 – frame, 7 – frame supports, 8 – foundation block, 9 – pneumatic absorbers, 10 – propelling engine, 11 – gear box.

The research rig shown is present at the IFFM PASci. Five bearings pedestals are fixed to a steel frame founded on a heavy foundation. Slide bearings supporting the rotor are founded in three of them. The rotor consists of two parts of

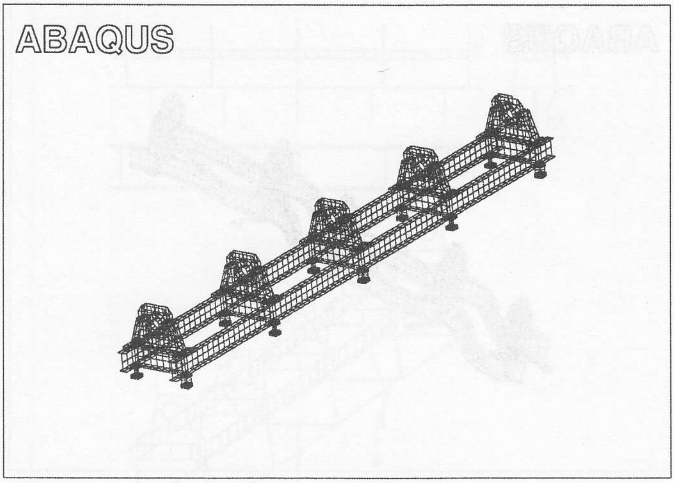

Figure 2. FE mesh of the supporting construction.

shaft connected by a stiff coupling. In the middle of each part of a shaft a disc with the mass of 185 kg is mounted with the aim to load the bearings statically. The foundation block with the mass of 11000 kg and 6 m length serves as a foundation. Five pumped bus tires are horizontally placed between the foundation block and soil to prevent the rig from soil vibrations.

The rotor is propelled by an electric engine through an elastic coupling. The motor and the gearbox are mounted on separate foundation. Maximum rotational speed of rotor is 5000 rpm. Static load of bearings results from a total mass of rotor. Dynamical load of bearings results from centrifugal force acting on the weights mounted on discs and producing the rotor imbalance. The rig is equipped with a measurement system enabling measurement of its dynamical state in characteristic points.

Supporting construction of the rotor-bearings system (without foundation) was modeled at IFFM PASci with FE method using the computer code ABAQUS. It's element mesh is shown in Fig. 2, [2]:

The frame of double-T bars was modeled with 8-node plate elements, as well as bearing pedestals. 20-nodes brick elements were used to model bearing bushes and some elements fixing bearing bushes and frame. The total number of the elements in the model is 2436 giving in total 11 182 nodes. The model is described with the system of 60 672 equations.

The first 5 natural frequencies are in the range of $0 \div 125$ Hz. The first form is shown in Fig. 3.

The presence in the structure of thin-walled elements of opened profile (double-T shape elements) for which the Sain-Venant principle is not applicable and the

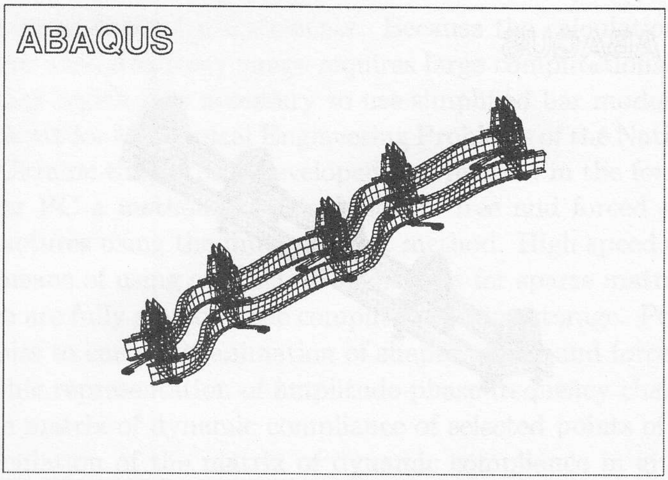

Figure 3. Free vibrations of support frame – the 1^{st} form at 67.5 Hz.

cross-section deformation takes place does not allow to realize simple change of such elements by separate bars. For adequate describing of thin walled double-T shape elements' deformations they have been modeled by bar frame structures.

The simplest numerical model which requires a minimum number of bars is constructed by means of the change of double-T shape element by two equal bars of T-shape which are formed as a result of dissection of the double-T shape element along the axis and have stiff and mass characteristics of double-T shape element. To guarantee the joint of their deformations T-shape bars are connected by weightless cross bars describing stiff characteristics of the double-T shape wall. Torsional rigidities are selected on the condition of the correspondence of frequencies and shapes of the frame structure free oscillations calculated by the theory of plates. The bedplate has been represented in the form of two bars of rectangular profile connected by weightless cross bars. For conservation of joint points of structure weightless bars of the corresponding rigidity have been added.

Bar finite elements have 6 degrees of freedom in the node describing flexural-torsional-longitudinal strains. Account of the shear deformation in bending and inertia of turn of cross-sections has been carried out without introducing of additional degrees of freedom. Bar numerical model of the rotor support structure including 594 bars depicted in three projections and axonometry in Fig. 4. Rigidities of support bars of the bedplate have been selected from the condition of coincidence of calculated and experimental first frequencies describing horizontal and vertical oscillations of the structure [1].

Matrix of dynamic compliance has been determined for horizontal and vertical rotor reactions in bearings. In Fig. 5 the quantities of its diagonal elements

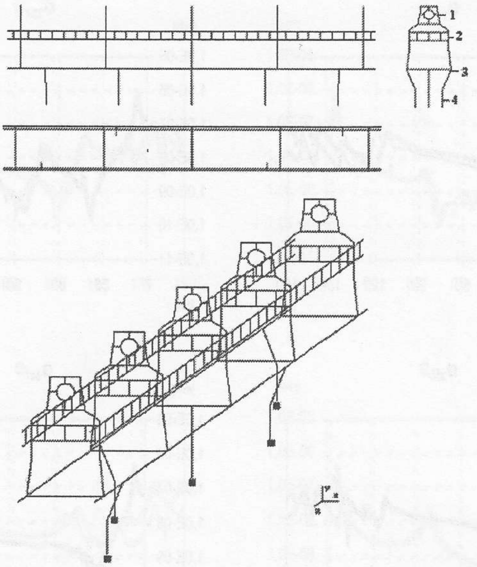

Figure 4. Bar numerical model of the support structure of the rotor, By black squares places of the stiff fixing of the structure are indicated. 1 – bearings' pedestals, 2 – double-T-shape elements frame, 3 – bedplate, 4 – pneumatic absorbers (bus wheels).

depending on the driving frequency in the frequency range till 180 Hz are presented. In Fig. 5 the following notation has been used: G_{11}, G_{33}, G_{55} – the decimal logarithms of compliances [m/N] in horizontal direction from the effect of horizontal loads in the 1^{st}, the 2^{nd} and the 3^{rd} bearings, respectively; G_{22}, G_{44}, G_{66} – the analogous quantities in vertical direction from the effect of vertical loads in the 1^{st}, the 2^{nd} and the 3^{rd} bearings, respectively. Some non-diagonal elements of the matrix of compliance are presented in Fig. 6. Results are in satisfactory agreement with experimental data obtained in [1].

It is important to note that shapes of forced oscillations in the bigger part of the frequency range differ essentially from the shapes of free oscillations. Thirty-two frequencies hit in the frequency range of $0 \div 180$ Hz. The initial 6 frequencies correspond mainly to the movement of structure as a rigid body on supports (wheels). The 7^{th} frequency is the first bending frequency of the bedplate in the plane of minimal stiffness. The 8^{th} frequency, the shape of oscillation of which is depicted in Fig. 7, is the first frequency of oscillation mainly of the frame of double-T shape elements.

After replacing of double-T shape elements with ordinary bar groups of very close frequencies took place. It is explained by very small torsional stiffness of profiles in the comparison with bending one, and in consequence of it some frame

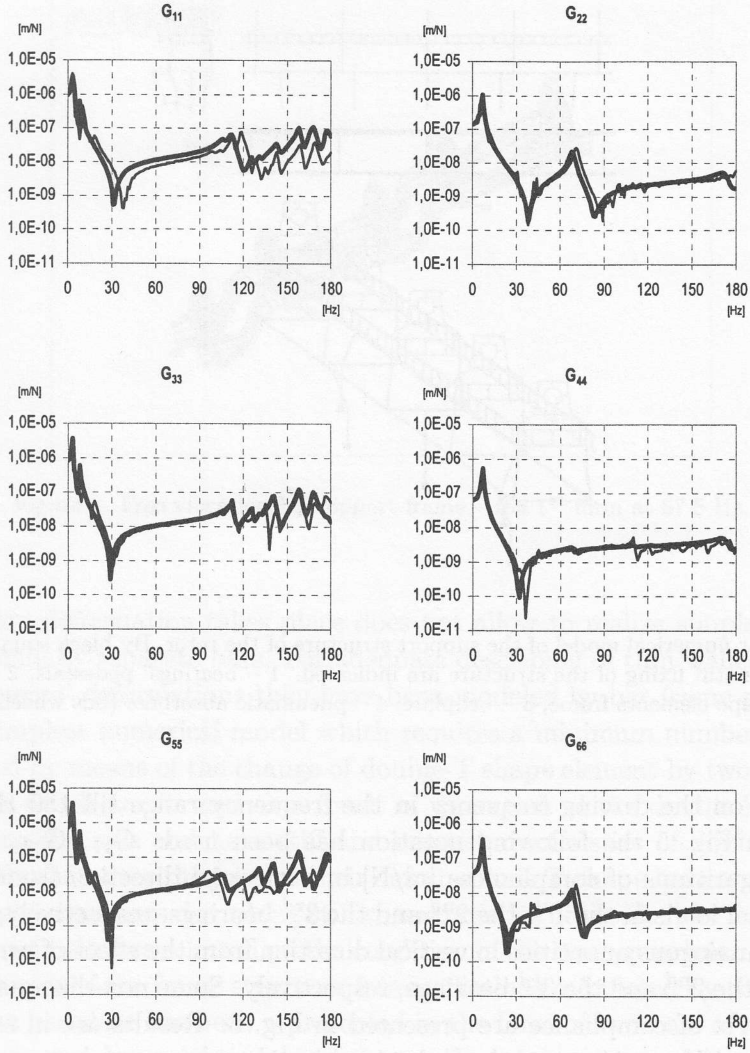

Figure 5. Graphs of frequency dependence of diagonal elements of matrix of dynamic compliance of rotor support structure bearings: thick line – calculation, thin line – experiment.

spans become practically isolated in torsional strains. However it is important to note that these and some other 'frame' frequencies are badly excited by loads applied in the rotor bearings. Due to those reasons probably matrices of dynamic compliance obtained for the simplest numerical model with 267 bars are lightly differ from results for more precise numerical model.

Developed method, in contrast to other known software systems, can be used for quick, determination of amplitude-phase-frequency characteristics of compli-

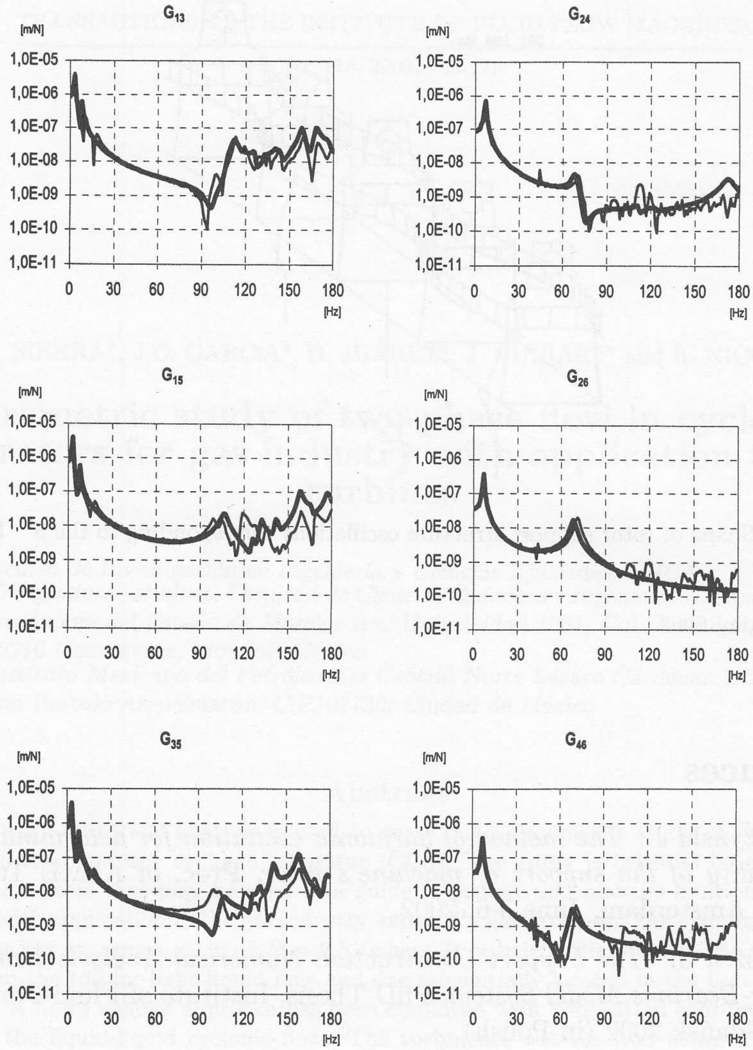

Figure 6. Graphs of frequency dependence of non-diagonal elements of matrix of dynamic compliance of rotor support structure bearings: thick line – calculation, thin line – experiment.

cated spatial structures of power machines with small time and resources expenses.

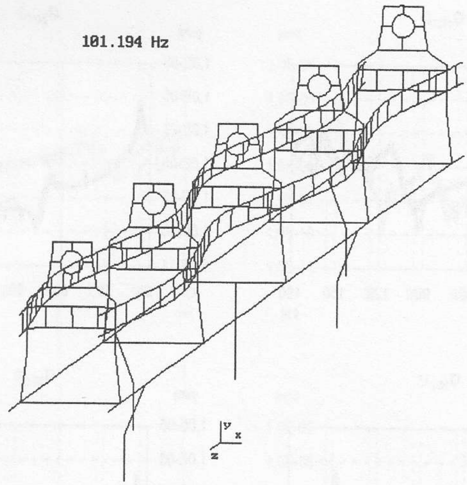

Figure 7. Shape of rotor support structure oscillations corresponding to the 8^{th} frequency.

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