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# Theoretical problems of the motion stability of turbine rotors

SUMMARY

*The theoretical motion stability problems of the rotors of great output turbosets are considered in the paper. The rotor motion equations are solved by using the transfer matrix method and applying complex variables. The motion stability is introduced according to the Lapunow idea. The theoretical results are applied to motion stability investigation of a 200 MW energy-plant turboset.*

## INTRODUCTION

For the dynamical analysis problems of the complex mechanical systems the division into relatively simple subsystems like: dimensionless (concentrated) inertial elements, rigid finite elements, massless elastic elements, damped deflection elements etc. have been applied. Those simple systems are known as modelling subsystems because a physical model of the whole system is defined by physical models of the simplest subsystems.

Mathematical model of the mechanical system consists of equations of motion for the modelling subsystems, equations describing mutual influence between those systems (the equation of balance of the influence forces, equations of the displacement compatibility of the subsystems), and the equations describing influence of the „environment" on the considered system (the equation of the constraints and external forces). The characteristic feature of the mechanical system is that all physical parameters of each simple modelling subsystem are simultaneously the model parameters.

In majority of cases the analysis of the big mechanical systems is accomplished by using the linear mathematical models having the constant parameters with respect to time. However in the case of vibration analysis of the mechanical systems their mathematical models contain the parameters dependent on vibration frequency or amplitude of the system elements, which conduct to a great number of the necessary variables describing the motion of the complex system.

Inconveniences are increasing when the algorithm requires multiple solution of the system model. Such situation occurs when the iteration methods are used to solve the problem. In such cases the necessary computation time to solve vibration problems of the mechanical systems may be considerably abbreviated. The realization of that target is achieved by applying another way, than the usual, of forming and solving the complex model of mechanical system.

## PARTIAL MACHINE SUBSYSTEMS

The machine which is a complex mechanical system has been divided into subsystems, however not the simple modelling subsystems mentioned before. Introduced subsystems can be relatively complex mechanical systems themselves and they are known as the partial subsystems. The next step is the definition of mutual influence between them. Those influences can be defined by adequate dynamic characteristics - the dynamical influence numbers or coefficients. The theory and the applications of those variables can be found in literature [1, 2, 3].

The dynamical influence numbers are not the variables which describe the features of the partial system arbitrarily, but they depend on the kind of movement of the system. Most frequently the dynamical influence numbers are used when the system is under stationary forced vibration. Then the numbers depend on the forced vibration frequency. More complicated case is when the free motion of the system is considered. In that case the dynamical influence numbers depend on two parameters: the vibration frequency, and damping decrement [1]. The result of the further consideration is that the damping of the partial system can be modelled without using two-parameter influence numbers. Hence each of the partial subsystems of the mathematical model can be represented by the appropriate set of dynamical influence numbers. In an extreme case, the model of the system can only consist of the dynamical influence numbers of the introduced partial subsystems. Then the characteristic equation (the equation of the system frequency) contains only dynamical influence numbers of the partial systems without any other physical and geometrical parameters of the system.

## THE TRANSFER MATRIX METHOD IN MODELLING OF THE ROTOR ELEMENTS

Forming the physical model of the rotor by the one-dimension element method may be combined with the transfer matrix method used to the appropriate mathematical rotor model. In the case when flexural vibrations are considered the components of the rotor cross-section state vector are: rotor deflection $G$, slope of the rotor axis $\phi$, internal bending moment $\mathfrak{M}$, and shearing force $\mathfrak{T}$. Each of those quantities can be expressed as a complex variable :

$$\begin{aligned} G(z,t) &= G^x(z,t) + jG^y(z,t) \\ \phi(z,t) &= \phi^x(z,t) + j\phi^y(z,t) \\ \mathfrak{M}(z,t) &= \mathfrak{M}^x(z,t) + j\mathfrak{M}^y(z,t) \\ \mathfrak{T}(z,t) &= \mathfrak{T}^x(z,t) + j\mathfrak{T}^y(z,t) \end{aligned} \tag{1}$$

The above specified variables are the components of the state vectors of the rotor cross-sections which are the basic quantities of the transfer matrix method. In the considered case they obtained the following form :

$$\{w(z,t)\} = \mathrm{col}\{G(z,t), \phi(z,t), \mathfrak{M}(z,t), \mathfrak{T}(z,t)\} \tag{2}$$

Then the motion equation of each k-th rotor element is as follows :

$$\{w(t)\}^-_{k+1} = \mathbf{D}_k \{w(t)\}^+_k + \{v(t)\}_k \tag{3}$$

where : $\mathbf{D}_k$ - the rotor element transfer matrix (Fig.1).

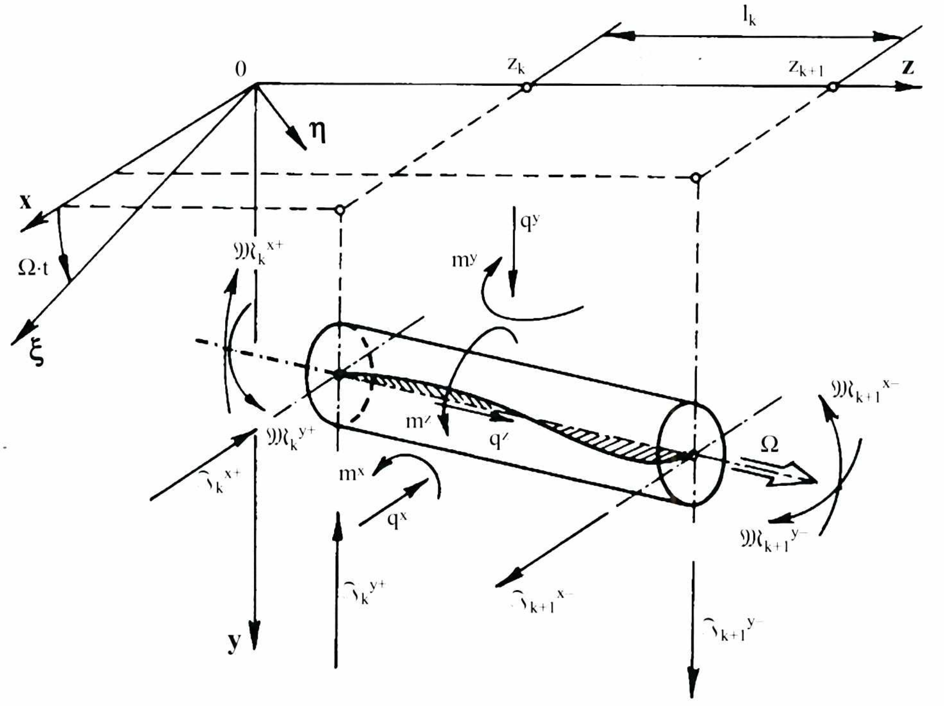


*Fig.1. Rotor elastic element*

In the frame of the linear theory of system vibration the state vectors can be expressed in the form :

$$\{w(t)\}_k = \sum_{h=1}^{h_s} e^{\lambda_h t} \{w\}_{k(h)} \tag{4}$$

where for the considered free vibrations of the system the complex eigenvalues are applied :

$$\lambda_h = \gamma_h + j\,\omega_h \qquad h = 1, 2, \ldots, h_s \tag{5}$$

For each rotor span the following relationship can be obtained :

$$\{w\}^-_{k^{i+1}(\sigma)} = \mathbf{D}^{k^{i+1}}_{k^i(\sigma)} \{w\}^+_{k^i(\sigma)} + U^{k^{i+1}}_{k^i(\sigma)} \tag{6}$$

where :

$$\mathbf{D}^{k^{i+1}}_{k^i(\sigma)} = \prod_{p=k^{i+1}-1}^{p=k^i} \mathbf{D}_{p(\sigma)} \tag{7}$$

is the transfer matrix for entire span of the rotor, and :

$$U^{k^{i+1}}_{k^i(\sigma)} = \sum_{q=k^i}^{q=k^{i+1}-2} \prod_{p=k^{i+1}-1}^{p=q+1} \mathbf{D}_{p(\sigma)} \{v\}_{q(\sigma)} + \{v\}_{k^{i+1}-1(\sigma)} \tag{8}$$

where: $\sigma = \pm h$

One should notice that using the transfer matrix for the rotor span leads to particular influence on increasing the errors of the numerical rounding because some of the elements of the transfer matrix are big (support stiffness values). The using of the transfer matrix method can limit the modelling of the construction supporting the rotor, namely, the model with mutual coupling of the supports thought as the foundation of the machine, cannot be used.

In the presented paper the transfer matrix method is limited only to the respective rotor spans. In that way the dynamical properties of the rotor are described by the (N+1) transfer matrix of the rotor spans for every harmonic component of the movement separately. This approach removes the traditional conception of the transfer matrix method errors. The transfer matrix of the rotor spans is the result of the multiplication of relatively small number of transfer matrices for the particular elements of the rotor. In addition, it is possible to use full model of the supporting construction involving the coupling between supports.

## MODELLING OF THE OIL FILM BEARING

With neglecting of the inertia forces of the bearing oil film, the bearing reaction forces depend on mutual position of the bearing journals in the bushings. Then the reaction which acts on the bearing journal and the reaction on the bearing bushing are the opposite forces (without the phase delay). Very important are the co-ordinates of bearing journals being in the steady-state position. Components of the whole bearing oil film reaction can be expanded into the Taylor's series in the neighbourhood of the point of journal steady-state :

$$\Re^x = R^x + \left(\frac{\partial \Re^x}{\partial x}\right)_0 u^x + \left(\frac{\partial \Re^x}{\partial y}\right)_0 u^y + \left(\frac{\partial \Re^x}{\partial \dot{x}}\right)_0 \dot{u}^x + \left(\frac{\partial \Re^x}{\partial \dot{y}}\right)_0 \dot{u}^y + \Delta\Re^x$$

$$\Re^y = R^y + \left(\frac{\partial \Re^y}{\partial x}\right)_0 u^x + \left(\frac{\partial \Re^y}{\partial y}\right)_0 u^y + \left(\frac{\partial \Re^y}{\partial \dot{x}}\right)_0 \dot{u}^x + \left(\frac{\partial \Re^y}{\partial \dot{y}}\right)_0 \dot{u}^y + \Delta\Re^y \qquad (9)$$

The co-ordinates of the linear parts of the expansion (9) are the dynamic characteristics of the oil film :

**elasticity characteristic**

$$C^{xx} = \left(\frac{\partial \Re^x}{\partial x}\right)_0 \qquad C^{xy} = \left(\frac{\partial \Re^x}{\partial y}\right)_0$$

$$C^{yx} = \left(\frac{\partial \Re^y}{\partial x}\right)_0 \qquad C^{yy} = \left(\frac{\partial \Re^y}{\partial y}\right)_0 \qquad (10)$$

**damping characteristic**

$$\lambda^{xx} = \left(\frac{\partial \Re^x}{\partial \dot{x}}\right)_0 \qquad \lambda^{xy} = \left(\frac{\partial \Re^x}{\partial \dot{y}}\right)_0$$

$$\lambda^{yx} = \left(\frac{\partial \Re^y}{\partial \dot{x}}\right)_0 \qquad \lambda^{yy} = \left(\frac{\partial \Re^y}{\partial \dot{y}}\right)_0 \qquad (11)$$

The constant parts of the extensions (9) are the component of the statical reaction of the bearing, dependent on the steady-state point co-ordinates $(x_o, y_o)$ :

$$R^x = R^x(x_0, y_0)$$
$$R^y = R^y(x_0, y_0) \qquad (12)$$

The above mentioned dependences are highly non-linear.

It should be taken into consideration, that the names of the co--ordinate groups are derived more from the tradition of using them than from the physical interpretation. Only the $C^{xx}$ and $C^{yy}$ co-ordinates are the actual elastic coefficients, and $\lambda^{xx}$ and $\lambda^{yy}$ are actual damping coefficients. The characteristics with mixed upper indices express coupling of the bearings motions in two mutually perpendicular directions. They have an essential influence on the mechanical energy balance of the bearing (oil film).

For the particular bearings the coupling characteristics determine the additional reason of using the non-conservative properties of the oil film, revealed by increased dispersion of the mechanical energy or cumulating of the energy, which can conduct to self-excited vibrations. The computation of the hydrodynamic characteristics is performed on the basis of an appropriate physical model of the oil film. At present the adiabatic or diathermic model of the oil film are used. In the diathermic model the heat exchange between oil film and bearing journal as well as bushing is taken into account. The elastic and damping characteristics of the bearing oil film are dependent on the journal bearing rotation velocity $\Omega$, but independent of the vibration frequency $\omega$.

## MODELLING OF THE BEARING SUPPORTS AND MACHINE FOUNDATION

The bearing reactions $\Re$ cause the displacement of the bearing supports, $\zeta$ , which can be expressed (within the frame of the linear vibration theory ) as follows :

$$\zeta_{k^i}(t) = \sum_{k^*=k^1}^{k^N} \Delta_{k^i k^*}\, \Re_{k^*}(t) + \Delta^*_{k^i k^*}\, \overline{\Re}_{k^*}(t) \qquad (13)$$

where : $k^i = k^1, k^2, \ldots\ldots, k^N$ are indices of supports.

In the above given expressions the complex dynamic influence numbers are applied which can be obtained by using the real dynamic influence coefficients :

$$\Delta_{k^i k^*} = \frac{1}{2}\left[\left(\Delta^{xx}_{k^i k^*} + \Delta^{yy}_{k^i k^*}\right) + j\left(\Delta^{yx}_{k^i k^*} - \Delta^{xy}_{k^i k^*}\right)\right]$$
$$\Delta^*_{k^i k^*} = \frac{1}{2}\left[\left(\Delta^{xx}_{k^i k^*} - \Delta^{yy}_{k^i k^*}\right) + j\left(\Delta^{yx}_{k^i k^*} + \Delta^{xy}_{k^i k^*}\right)\right] \qquad (14)$$

## FREE VIBRATIONS AND MOTION STABILITY OF THE ROTOR

Finding the complex eigenvalue $\lambda = \gamma + j\omega$ is equivalent to the necessity of computing two real variables $(\lambda, \omega)$ for which the characteristic determinant is zero. There are two methods to deal with that problem. One of them is the adaptation of the Muller's square approximation to determine zeros of the determinant of the complex matrix of the final simultaneous equations of free vibrations of the rotor.

In this paper the Muller's method has been successfully used to determine the complex roots of the complex characteristic determinant. In this case, the square approximation function is the complex function of the complex argument.

In the case of the rotor free vibration the movement trajectories of the rotor axial points which are placed at the cross-sections introduced to the analysis, are as follows :

$$G_{k(\sigma)}(t) = G_{k(\sigma)}\, e^{\lambda_\sigma t} + G_{k(-\sigma)}\, e^{\bar{\lambda}_\sigma t} \qquad (15)$$

or

$$G_{k(\sigma)} = e^{\gamma_\sigma t}\left[G_{k(\sigma)}\, e^{j\omega_\sigma t} + G_{k(-\sigma)}\, e^{-j\omega_\sigma t}\right] \qquad (16)$$

for $k = 1, 2, \ldots, n$ and $\sigma \in H = \{h_1, h_2, \ldots, h_s\}$

where :
- H - set of harmonic component numbers
- n - total number of rotor elements.

The expression represents spiral ellipsoidal trajectories whose dimensions increase when $\gamma_\sigma > 0$, or decrease when $\gamma_\sigma < 0$ (Fig.2).

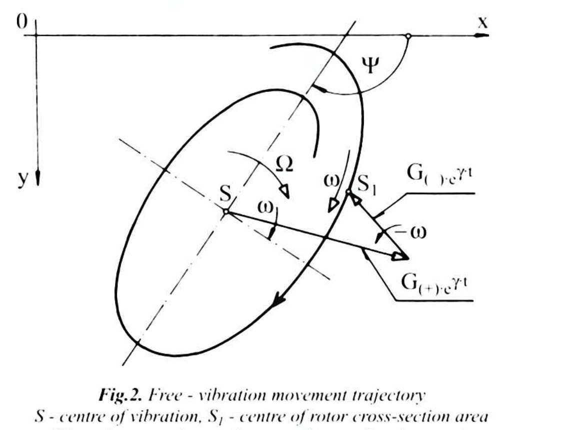


***Fig.2.** Free - vibration movement trajectory*
*S - centre of vibration, $S_1$ - centre of rotor cross-section area*
*Ψ - trajectory axis angular co-ordinate related to x-axis*
*$G_{(+)}$, $G_{(-)}$ - forward and backward component of rotor cross-section motion, respectively*

The problem of the eigenvalue of free motion is very often connected with the problems of the motion stability. Here, a few general remarks are offered about the motion stability of the dynamic systems in order to precisely characterise the way of the solution finding.

The motion stability idea depends on the adopted definition. The most often used definitions of the motion stability are the classic definition proposed by Lapunow, Lagrange or Poincare. On the basis of these definitions the dynamical systems were investigated in two situations: how the system would behave when the initial conditions are insignificantly changed, and what influence the „small" disturbances would have during all the movement.

The motion stability in Lapunow's meaning is the mathematical idea due to two reasons. First of all, it is defined for the solutions of the system motion equations, which means that this definition is on the mathematical level of the established model of the system. Secondly, in the definition the infinitely small quantities (infinitely small variable increments) and the infinitely large quantities (infinite increment of time) are introduced. Only such motion stability is identified with the stability of the physical model or real dynamical systems. It is significant that Lapunow's motion stability can not be examined in practice because it is impossible to run a dynamical system twice at the same conditions. Moreover it is practically impossible to realize the infinitely small and large quantities used in that motion stability concept, namely to obtain the technical steadiness. In the definition in question finite increments of solutions and flow of time are assumed. Investigation of the motion stability, according to its different definitions can even lead to opposite results. Hence it results that the stability is not the natural physical feature of dynamical systems but it depends on the way in which investigations of the system stability are performed.

The estimation of the rotating system stability is a difficult problem, especially when the models with the distributed parameters are used, like in this paper. Such research is continued for many years. As a good example of the newest achievements are some chapters of the work [4] , in which the Lapunow-Mowcan's idea of motion stability is used together with the second Lapunow's method applied to the discrete systems.

For investigation of the continuous systems the solution distance (metrics) of the integral type was used. The approach was successfully applied to simple continuous systems. However, its application to the complex system such as that considered in this paper, is difficult. First of all, it is very difficult to find the Lapunow's functional for the investigated system of the complex boundary conditions. Such functional should be given in an analytical form to be able to investigate their features.

The attempt to adapt the first Lapunow method to investigate a linear continuous system motion by using the one distance (or norm) [5] gave only partial results unsuitable for solving the rotor motion problem. On the other hand, a discrete method for the continuous systems (rotors) was tested, based on using the finite differences to investigate the motion stability. It was used to solve the generalized problem of the matrix eigenvalues. The obtained experience showed that this method can be practically applied only to the rotors with two supports. With a view of that, it was concluded that the motion stability of the rotor should be investigated by using a little different definition of stability in relation to the systems in question, called *the stability in the limited range.*

The investigation of the stability is limited to a few first forms of the rotor vibration. The limitation causes that the analysed continuous system may be treated as the discrete system (of a limited degree of freedom) and therefore the idea of the first Lapunow method can be used [6]. Rotor motion is stable if all the eigenvalues of the free motion equations, adequate to the vibration forms taken into consideration, have the negative real parts. Moreover the rotating systems described by the beam models obtained the feature of having one eigenvalue for every form of vibration.

The second kind of methods which allow to efficiently analyse free vibration of the rotors are those based on the finite elements: rigid or deformable FEs. Application of the FE method is equvalent to using the discrete model (linear in general) with the finite number of degrees of freedom. The theoretical solution of the Lapunow stability problem by using discrete model exists and it is reduced to solving the generalized problem of the matrix eigenvalues. One can say that the FE method used in solving the motion stability problem theoretically is a „complete" method, which means that after establishing the physical and mathematical model of a considered system, it is also possible to solve the problem according to the theory of the motion stability. However, the way proposed in the paper of solving the motion stability problem in a limited range by using the transfer matrix method can be called an „incomplete" method, because not all the eigenvalues can be determined (as the continuous systems have infinite number of eigenvalues). Those differences of both methods disappear in practice. Most often, for the complex dynamical system of the large number of degrees of freedom, solved by FE method, only a few eigenvalues (less than the number of freedom degrees) are usually determined. Therefore, both proposed methods of solving the motion stability problem of the rotors differ from each other only in the used models, but in both cases the motion stability is investigated on the basis of the incomplete number of eigenvalues. The method of solving the general problem with the eigenvalues, used in FE method and giving all the eigenvalues, is efficacious only for relative simple physical model subsystems of the machine (in particular with the respect to the supporting construction of the rotor).

Theoretically, in the transfer matrix method the possibility of determination of the large number of eigenvalues (like in FE method) exists. For the complex physical model of the rotor, the difficulties of the numerical processing would be comparable in both cases, but the computing cost will be large.

In the used physical model of the machine the following destabilizing factors were taken into consideration :

- internal material and structural damping in rotors
- hydrodynamic forces of the bearing oil film
- parametric excitations of the rotors with asymmetrical shaft
- steam seals of the turbine.

## THE MOTION STABILITY OF 200 MW TURBOSET ROTOR

The rotor line of the turboset was divided into 138 elements (defined by 139 rotor cross-sections) and supported on N = 7 bearings (Fig.3).

The problem of motion stability was solved by using the ROTSTAB computer program.

During calculations the influences of the following factors were taken into account :

⇨ internal and external damping
⇨ bearing oil film

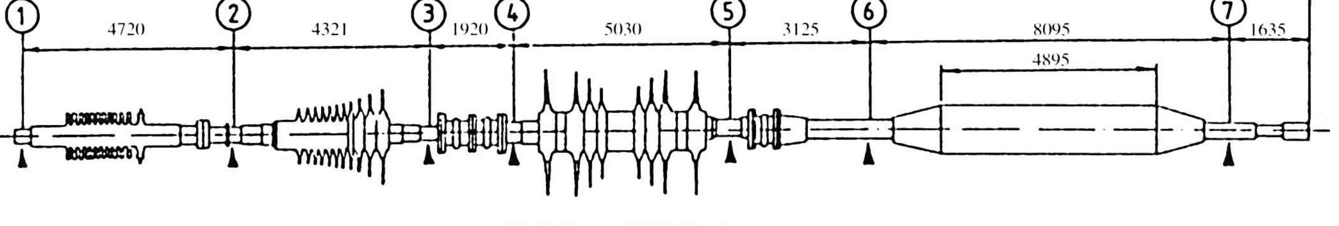


*Fig.3. Sheme of 200 MW turboset rotor*

- ⇨ elasticity and damping of bearing supports and foundation
- ⇨ rotational inertia, gyroscopic effect and shearing displacement of rotor element (according to the Timoshenko idea)
- ⇨ rotor rotational speed Ω
- ⇨ temperature influences on material modulus of elasticity
- ⇨ geodetic height of bearings (two variants - Table)

Some results of the calculations are shown in Fig.4 ÷ 6.

***Table.** Initial geodetic height of turboset bearings*

| Bearing No | | 1 | 2 | 3 | 4 | 5 | 6 | 7 |
|---|---|---|---|---|---|---|---|---|
| | | [mm] | | | | | | |
| Variant 1 | $\zeta^{px}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| | $\zeta^{py}$ | -8.25 | -3.05 | -1.23 | -0.69 | 0 | 0 | -2.0 |
| Variant 2 | $\zeta^{px}$ | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| | $\zeta^{py}$ | -8.25 | -3.05 | -2.23 | -0.69 | -2.0 | 0 | -2.0 |

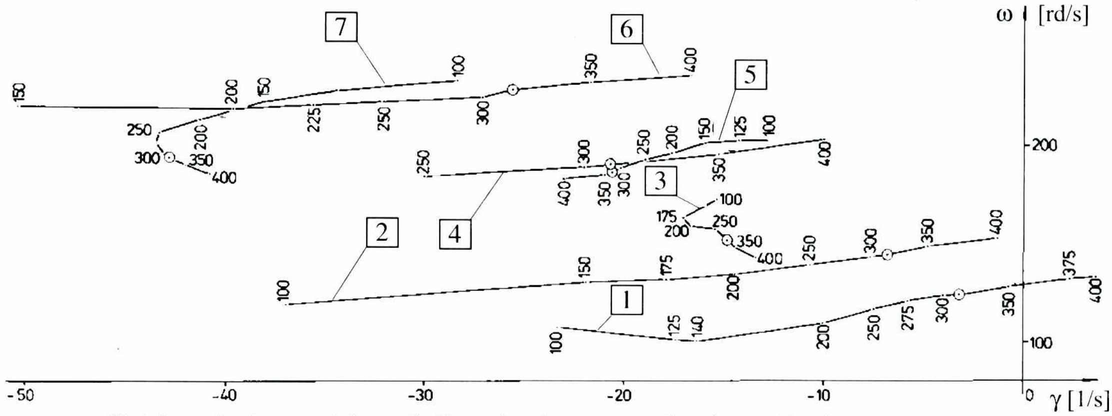


***Fig.4.** Rotor vibration spectral diagram for Variant 1 (without accounting for influence of foundation and support damping)*
***Note :** - Rotor speed Ω [rad/s] is plotted as the parameter*
*- Vibration mode number is given in □*

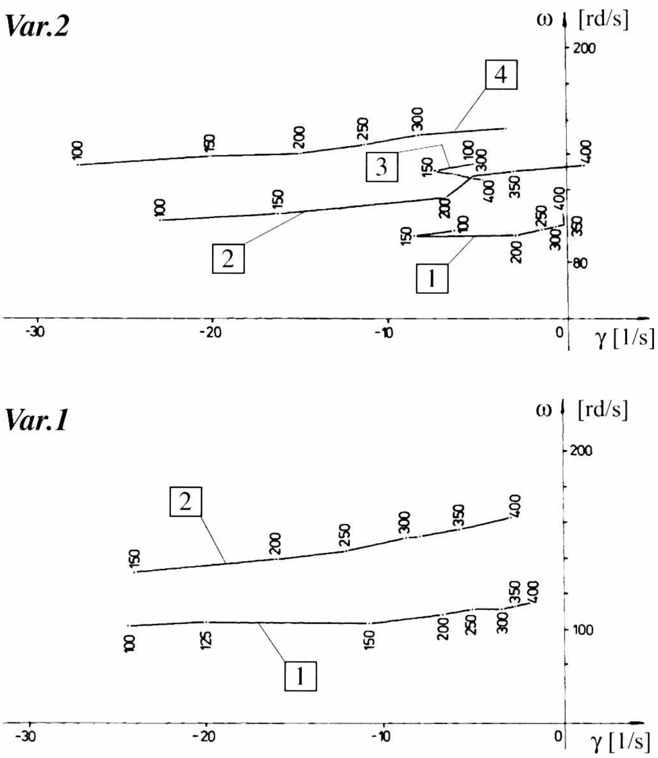


***Fig.5.** Rotor vibration spectral diagrams :*
*for Variant 2 (with accounting for influence of foundation and support damping)*
*for Variant 1 (without accounting for influence of foundation and support damping)*
***Note :** - Rotor speed Ω [rad/s] is plotted as the parameter*
*- Vibration mode number is given in □*

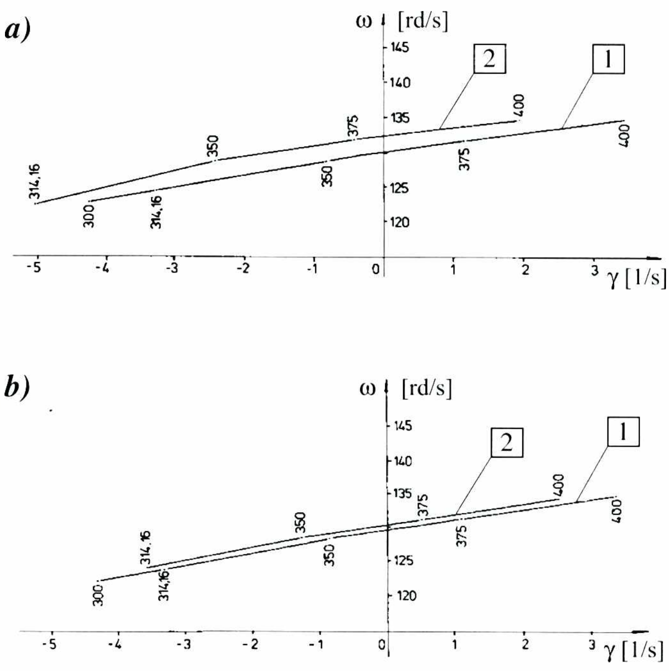


***Fig.6.** Spectral diagrams of 1st mode rotor vibrations for Variant 1:*
***a)** - influence of external damping :damping coefficient $b^{ex}$ = 1 kNs/m², for case 1*
*$b^{ex}$ = 1 kNs/m², for case 2*
***b)** - influence of internal damping :damping coefficient $b^{in}$ = 1 kNs/m², for case 1*
*$b^{in}$ = 1 kNs/m², for case 2*
***Note** : - Rotor speed Ω [rad/s] is plotted as the parameter*
*- Vibration mode number is given in □*

## CONCLUSIONS

- ◆ Increasing of the rotor speed is a source of the decrease of motion stability reserve for almost all vibration modes (see Fig.4) and it can even conduct to an unstable motion of the rotor (first vibration mode of Var.1-Fig.4).
- ◆ However, when the foundation damping (Var.2) is taken into consideration then the first mode of vibration is stable in a wide range of rotor speed (Fig.5b).
- ◆ Lifting the bearing Nr 5 (2 mm up) involves that the rotor motion is unquiet (several first modes of vibrations have very small stability reserve - Fig.5a).This phenomenon is known when the slide bearing is statically unloaded (lifting the bearing Nr 5 causes unloading of the bearing Nr 6 - Table and Fig.3).
- ◆ External damping acting on the rotor heightens the rotor motion stability (Fig.6a) but internal damping has opposite influence and it reduces the motion stability reserve (Fig.6b).
- ◆ The obtained calculation results are known in rotor dynamics, however in this paper the new aspects were given of the rotating machine modelling and the rotor motion stability.

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***Appraised by Jan Kiciński, Prof.,D.Sc.***

**NOMENCLATURE**

$b^{ex}$ - external damping coefficient
$b^{in}$ - internal (material) dumping coefficient
C - bearing oil film rigidity
**D** - transfer matrix
e - natural logarithm base
G - rotor deflection
h - harmonic component number
$j = \sqrt{-1}$
$\mathfrak{J}$ - shearing force
k - rotor element index
k' - rotor support index
m - bending moment intensity
$\mathfrak{M}$ - bending moment
n - total number of rotor elements
N - total number of support bearings
o - steady-state point
q - continuous transverse load intensity
R, $\mathfrak{R}$ - static and dynamic support reactions, respectively
t - time
u - bearing oil film displacement
U - vector of exciting forses of the rotor span
v - vector of exciting forces
{w} - state vector of rotor cross-section
x,y,z - horizontal, vertical and axial Cartesian co-ordinate, respectively
$\gamma$ - damping parameter $[\gamma = Re(\lambda)]$
$\Delta$ - coefficient of rotor support (dynamical influence number)
$\zeta$ - bearing support displacement
$\eta, \xi$ - rotating co-ordinate system
$\lambda$ - complex eigenvalue
$\Lambda$ - bearing oil film damping
$\sigma$ - harmonic component number
$\phi$ - angle of rotor axis slope
$\omega$ - free vibration frequency $[\omega = Im(\lambda)]$
$\Omega$ - rotor rotational speed
$(\bar{\ })$ - conjugate complex number

# *Miscellanea*

## GDYNIA MARITIME ACADEMY

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## A MEMBER OF IAMU AND IMLA

**IAMU** is the acronym of the International Association of Maritime Universities located in Istambul. Several tens of maritime universities of the world are its members. Among them Gdynia Maritime Academy, Poland is one of its active members. Under IAMU patronage common undertakings are arranged dealing with scientific and didactic cooperation between partners. The Polish university maintains especially fruitful contacts with Istambul Technical University,Turkey.

**IMLA** is the acronym of the International Maritime Lecturers Association placed in Malmöe, which associates many lecturers of maritme universities worlwide. Under its auspices scientific conferences and workshops are cyclically organized in which scientific workers of Gdynia Maritime Academy regularly take part. Their papers are published in proceedings of the conferences and IMLA journals.

# FOREIGN *conferences*

## PRADS 2001

From 16 to 21 September 2001 in Shanghai , China, it was held 8th International Symposium on :

***Practical Design of Ships and Other Floating Structures (PRADS)***

Arranging those cyclic conferences was initiated in Tokyo in 1977, and six years later the next conference was simultaneously organized in Tokyo and Seul. The successive meetings had place : in Trondheim (1987), Varna (1989), Newcastle (1992), Seul (1995) and The Hague (1998).

PRADS Standing Committee is the international elective body which fulfils the role of programming and steering the aim and topics of the current meeting. One out of 14 S.C. members is Dr. Tadeusz Borzęcki, the representative of Faculty of Ocean Engineering and Ship Technology, Technical University of Gdańsk.

In PRADS 2001 over 260 participants from 25 countries all over the world took part among which representatives from China, Japan, Korea, Germany, Norway, Denmark and Great Britain prevailed.

169 papers were approved for presentation during the meeting. One of the papers titled :

***Influence of journal bearing modelling method on shaftline alingment and whirling vibrations***

was prepared by Lech Murawski, D.Sc. from the Institute of Fluid Flow Machinery, Polish Academy of Sciences, Gdańsk.

The next participant of the meeting, Dr. T. Borzęcki was the chairman of the topical session on Design Optimization.

## NATO Advance Research Workshop

On 24÷26 October 2001 the University of Trás-os-Montes and Alto Douro w Vila Real (Portugal) was the venue of NATO Advanced Research Workshop on :

***Systematic Organization of Information in Fuzzy Systems***

37 participants took part in the workshop who represented scientific research centres of 14 countries. Two Polish scientists were among 15 special guests invited by the workshop organizers, who presented the following papers :

- *Certainty and sharpness of information* – by Prof. Walenty Ostasiewicz from Wrocław University of Economics
- *A method for incorporating human factors in fuzzy – probabilistic modelling and risk analysis of industrial systems* – by Prof. Mirosław Kwiesielewicz from Technical University of Gdańsk.