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THE MODELING OF DEFECTS IN THE ROTOR-TRAINS OF TURBOMACHINERY – SIMULATION-BASED DIAGNOSTICS

The problem of the turbomachinery defects modeling is considered. The numerical analysis of the proper model and its defect can give the symptoms of the defect. The presented computer software MESWIR can generate the kinetoststic characteristics of bearings and vibration of the entire system (the shaft and the bearings), that allow carry out model-based diagnostics of a several classes of defects. A large power turbo-set is taken for example calculations. A two different defects were modeled: the bearing misalignment and the crack of the shaft.

Рассматривается проблема моделирования дефектов роторных машин. Показано, что с помощью численного анализа модели ротора и его дефекта можно определить признаки этого дефекта. Представлено программное обеспечение MESWIR, которое может генерировать кинетостатические характеристики подшипников и вибрации всей системы (вала и подшипников), что позволяет проводить диагностику дефектов различных классов. В качестве примеров приведены расчёты турбоагрегата большой мощности. Показаны результаты моделирования двух дефектов: смещение подшипника и поперечная трещина вала.

Розглядається проблема моделювання дефектів роторних машин. Показано, що за допомогою чисельного аналізу моделі ротора та його дефекту можна визначити ознаки цього дефекту. Наведено програмне забезпечення MESWIR, що може генерувати кінетостатичні характеристики підшипників і вібрації всієї системи (вала й підшипників), що дозволяє проводити діагностику дефектів різних класів. Як приклади наведені розрахунки турбоагрегату великої потужності. Показано результати моделювання двох дефектів: зсув підшипника й поперечна тріщина вала.

1. The numerical model of rotating shaft founded on slide bearings

Knowledge acquisition for critical machinery diagnostics means defining and describing symptoms of defects - response of the machine to particular malfunctions. Relations between defects and their symptoms are called “diagnostic relations” (fig. 1).

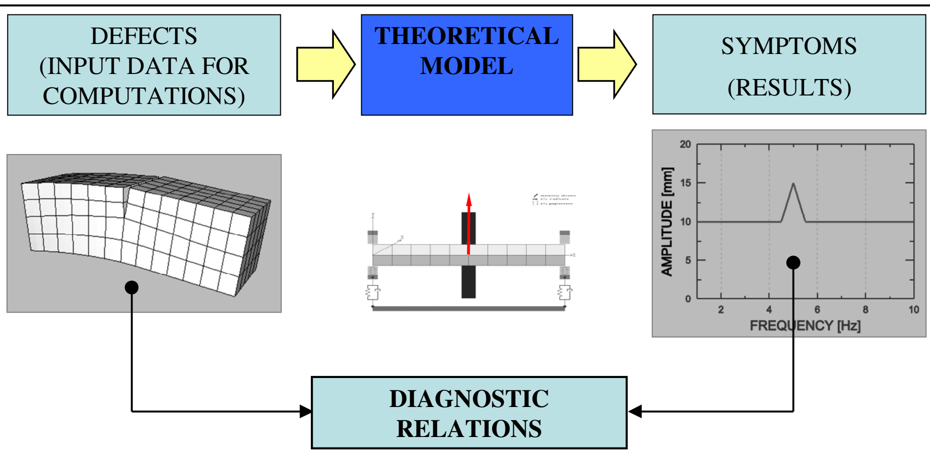

Fig. 1. The concept of model (computer simulation) based diagnostic relations.

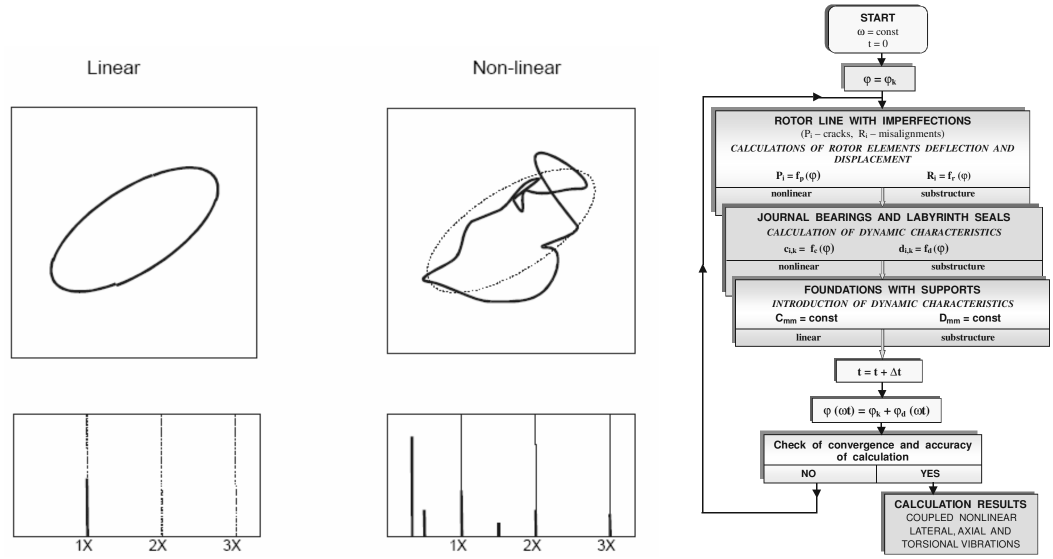

Fig. 2 The linear versus non-linear modeling. The algorithm of the dynamic's calculation

The computer code, originally invented in The Institute of Fluid-Flow Machinery, PASci. in Gdansk, is called NLDW (first letters of Polish phrase "Non-Linear Rotor Dynamics"). It is the effect of many years of experience of the research group under a leadership of Prof. Kicinski. The entire model is described in details and verified in the monograph [1].

From the mathematical point of view, the dynamics of rotating machine is described by the non-linear differential equations in a matrix form:

$$M\ddot{x} + D(x,\dot{x})\dot{x} + K(x,\dot{x})x = P(t) \tag{1}$$

where: M – global matrix of inertia; D – global matrix of damping; K – global matrix of stiffness; $x, \dot{x}, \ddot{x}$ – generalized vectors of displacements, velocities and accelerations; P – generalized vector of external excitations; t – time.

Equation (1) is the equation of motion of entire system, composed of a number of DOFs. The rotor line is being modeled with beam finite elements with 6 DOFs per node. The properties of the rotor depend on its geometry and material. The fig 2. shows the algorithm of the calculation.

The stiffness and damping of slide bearings are being calculated with the use of non-linear model which encompasses 3D Reynolds equation, energy equation, conductivity equation, etc. (fig. 3) The stiffness and damping coefficients can be defined as:

$$c_{11} = \frac{\partial W_x}{\partial x}, \qquad c_{22} = \frac{\partial W_y}{\partial y}, \qquad c_{12} = \frac{\partial W_x}{\partial y}, \qquad c_{21} = \frac{\partial W_y}{\partial x},$$

$$d_{11} = \frac{\partial W_x}{\partial \dot{x}}, \qquad d_{22} = \frac{\partial W_y}{\partial \dot{y}}, \qquad d_{12} = \frac{\partial W_x}{\partial \dot{y}}, \qquad d_{21} = \frac{\partial W_y}{\partial \dot{x}},$$

where W_x, W_y are the components of oil film reaction and a dot over the symbol indicates the time derivative.

The 3D Reynolds equation in the following form is the main equation describing the properties of bearings [2]:

$$\frac{\partial}{\partial x}\left(\frac{\partial p}{\partial x}A\right) + \frac{\partial}{\partial z}\left(\frac{\partial p}{\partial z}A\right) = -\left[U_1\left(\frac{\partial h}{\partial x} - \frac{\partial B}{\partial x}\right) + \frac{\partial h}{\partial \tau}\right],$$

where: p – oil pressure, h – oil film thickness, μ – the lubricant viscosity, U_1 – linear velocity of bearing journal, τ – time, A, B – functions of the viscosity and the local film thickness.

The dependence of stiffness and damping coefficients of oil film on the instantaneous position and transverse velocity of the vibrating bearing journal is the essence of slide bearing's non-linearity. That's why the stiffness and damping coefficients, instantaneous positions of bearing

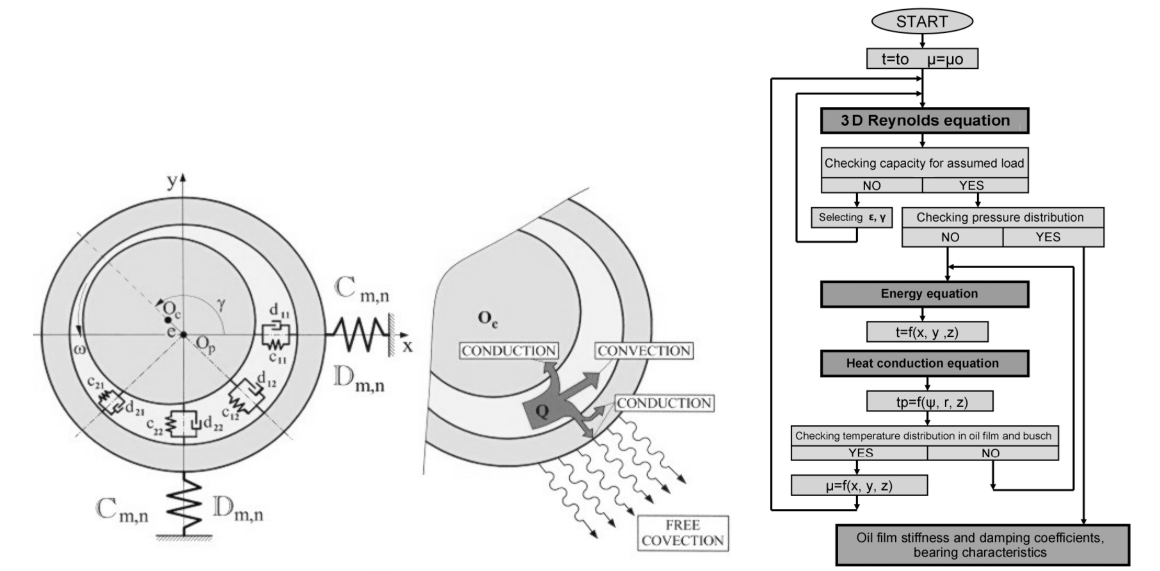

Fig. 3. The slide bearing as a mechanical system: the stiffness and damping coefficients, thermal flux, the algorithm of the slide bearing's properties' calculation

journals and instantaneous shape of the shaft line are being calculated iteratively for each time-step, and for each time step the appropriate matrices are being actualized. Carrying out the calculations for a number of subsequent revolutions of the shaft gives a result the time history of vibrations.

2. The example simulations of defects

The model described above allows to simulate the following defects:

- imbalance
- rotor misalignment
- load modification
- transverse crack
- support stiffness loss
- displacements of bearings (misalignment, skewness)
- thermo-elastic deformations of the bearing shells
- hydrodynamic instability
- modification of bearing properties (e.g. caused by the altered lubrication conditions).

First, one have to build a so-called "base-case" – the model and the response of the healthy rotor. It is the reference model for results of further calculations when the defects would be simulated.

Let's take as an example, the 200 MW turbo-set. Fig. 4 presents the FE mesh of its model. The length of the rotor is about 28 m.

The dynamic state of the object is presented in the form of so-called "diagnostic cards", which show the orbits and spectra of the bearings' vibrations. Following figures present the diagnostic cards for the example defects. The fig. 5 presents the diag-

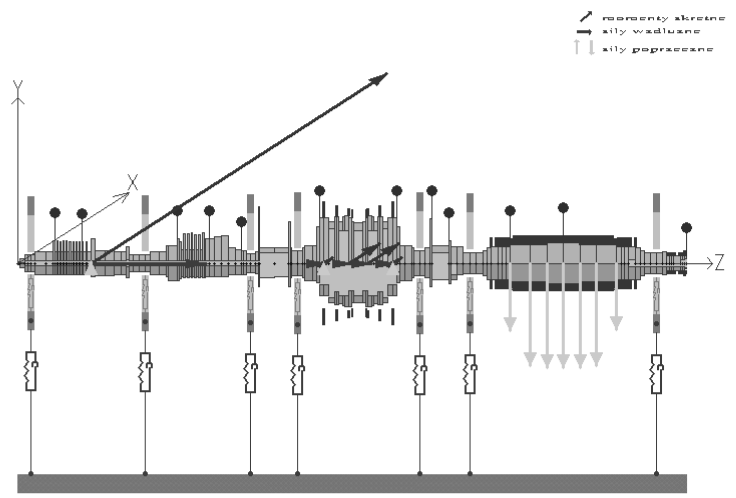

Fig. 4. The FE mesh of the turbo-set

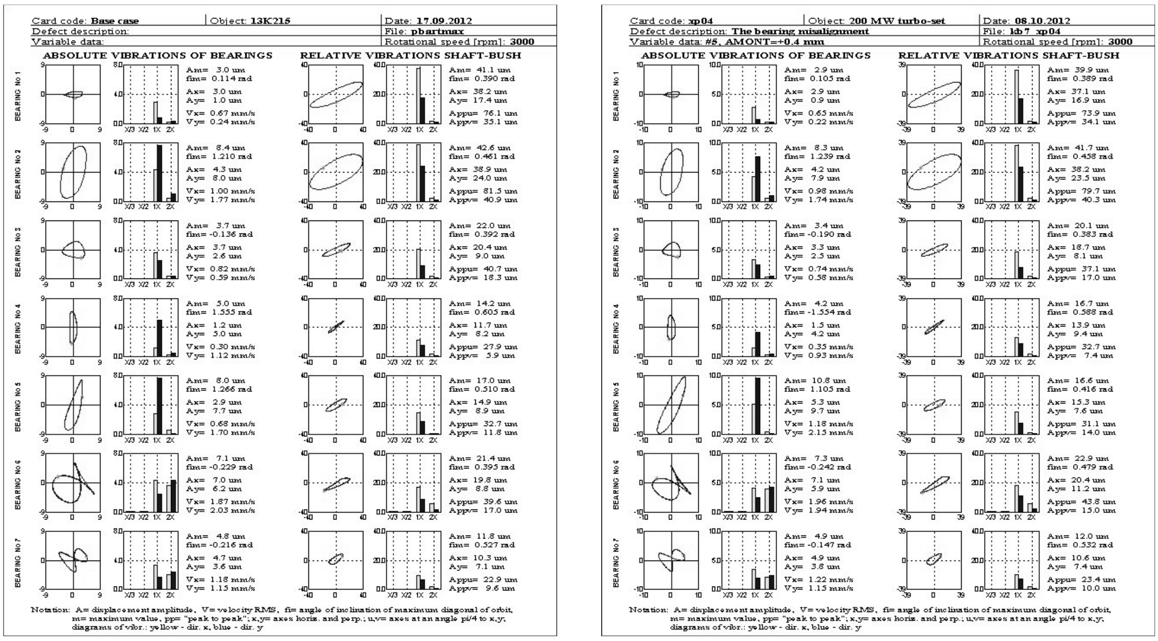

Fig. 5. The diagnostic cards for the 200 MW turbo-set:
left – the base case, right – the misalignment of the bearing No.5

nostic card of the base case (healthy rotor) of the turbo-set and the case of the misalignment of the bearing No.5 [3, 4]. As we can see, such misalignment (0.4 mm into the right) didn't change the vibration state of the machine significantly.

The following examples will testify, how the response of the system can depend on the position and orientation of the defect in the machine. The crack in the shaft will be the example defect of the machine. The model of the crack in the MESWIR system is built with the use of simple binary model by Knott [1, 5, 6].Geometry of the element with a crack is presented in Fig. 6.

The stiffness of the element with the crack is modified for the additional flexibilities induced by a crack and calculated from following relation:

$$K_e = T_t \cdot (L + L_d)^{-1} \cdot T,$$

where: K_e – the stiffness matrix of the element, L – the flexibility matrix of intact element, L_d – the matrix of additional flexibility induced by the crack, T – the matrix of transformation, T_t – transposed matrix of transformation, –1 – index denoting the matrix inversion.

The way of the crack incorporation into the program allows to model the influence of the rotor spinning, the kinetostatic deflection of the shaft and the presence of the lateral-axial-torsional vibrations' couplings. The fig. 7 presents the two example cases of the crack in the generator: close to the bearing and in the midspan.

We can observe the rise of sub-harmonic components of 1/3 X, which usually testify the

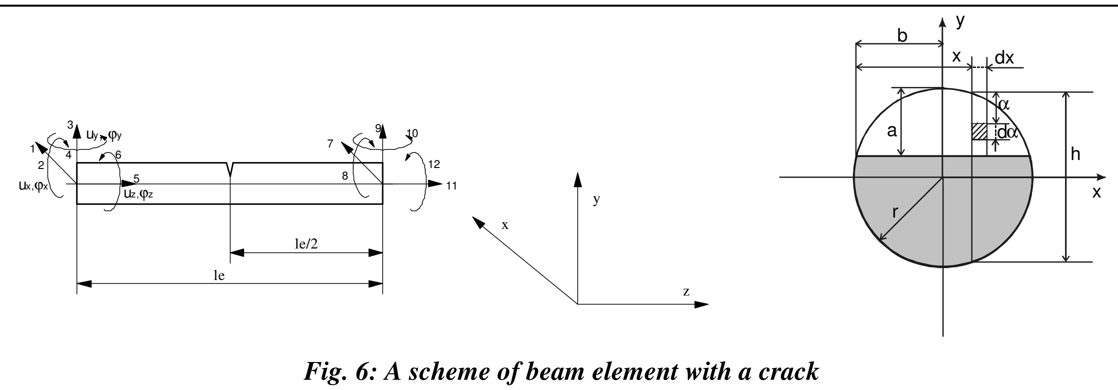

Fig. 6: A scheme of beam element with a crack

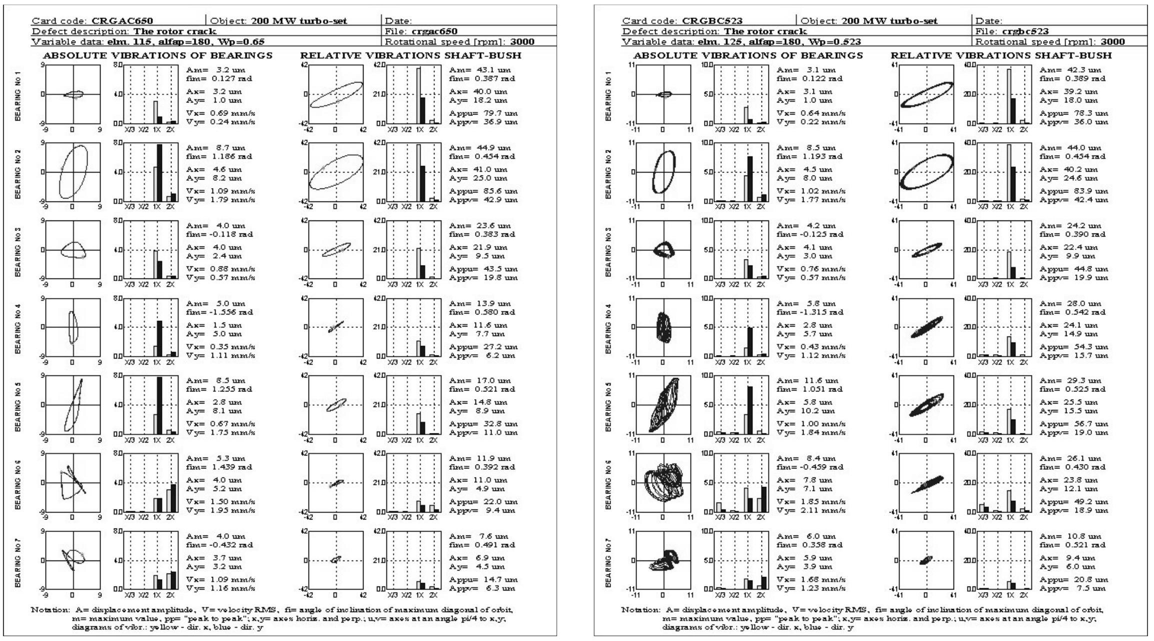

Fig. 7. Diagnostic cards for the example cases of the crack in the rotor:
left – crack close to the bearing No. 6, right – crack in the midspan

hydrodynamic instability of the bearings. It is also interesting that the influence of the crack on the absolute vibrations of the bush is stronger than the influence on the relative vibrations (journal-to-bush).

3. Closing remarks

The paper shown only a very few examples of building the diagnostic relations in a way of computer simulations. If a proper model of the machine is built, such a relations can be utilized in the intelligent supervisory system in the power plant (as they were invented). They can also be utilized in the design process of the new machinery (units) or at the modernization. They can bring us an information how can we affect the dynamic state of the machine at its exploitation.

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