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PREDICTIVE MAINTENANCE IN EQUIPMENT TROUBLESHOOTING

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Abstract: *The paper contains theoretical substantiation for machines and equipment troubleshooting, machine bodies in rotational motion. It analyses the dynamic rotary machines, fitted with elastic rotor in rigid bearings without damping, as well as the rotor symmetrically rotational movement with the elastic bearings without depreciation. The main dynamic rotating equipment defects are removed by static and dynamic balancing, which apply directly to predictive maintenance.*
Key words: *predictive maintenance, troubleshooting, dynamic rotary machines.*

1. INTRODUCTION

Corrective action to eliminate the main defects that occur in dynamic rotating machines are one of the most important components of the implementation of predictive maintenance and these are designed to bring the machines in a dynamic mode compatible with the system and it is necessary to work at a *Good grade* or *Utilisable*. The main defects that occur in dynamic rotating machines are imbalance and misalignment

In this stage, of particular importance is the diagnosis, which involves studying them from the point of view of the components, i.e. the type of Foundation work camps, the response of the rotor, the role of the machine in the process.

The rotor is a subset of these machines, consisting of a shaft which is one or more discs and executing a movement of rotation around its own axis. As a form, they can be simple or complex, but, regardless of type, being a moving element rotation determines the dynamic properties specific to rotor machines, which do not occur in the other types of machinery or structures[Fre 99].

From great machine rotor class belong the following subclasses of machines: motors, generators, turbines, compressors, pumps and blowers.

2. DYNAMIC ROTARY MACHINES

In machine operation, the rotor is subjected to vibration of bending and twisting. These vibrations are dependent on the geometry of the rotor and bearing type, and the excitatoare forces. The rotor, the precession, he turns his own Foundation. Complexity of dynamic phenomena is increased if it takes into account the fact that the hydro and aerodynamic forces can act upon the rotor, with variable gradient of temperature and pressure fields, electromagnetic fields etc. [Arg 02].

The main features of dynamics of rotor machines, compared with those without rotor systems are:

- All dynamic phenomena that occur during the operation of the rotor machines, are related to the movement of rotation of the rotor, with an energy transfer from the direction of rotation by the movement of precession.
- While, in the case of passive structures, a mode of vibration is characterized by its shape, the active structures, vibration of the rotor movement is defined by the precession mode. Therefore, the movement of vibration of the rotor comprises two lateral parts, inseparable, it has been agreed

to be referred to as vertical-horizontal component of precession.

- The machine rotor dynamics, generally due to the existence of some small differences, without symmetry, system characteristics on both vertical and horizontal directions, precession modes appear in pairs-for example: the first horizontally and the first vertically.
- Another unique feature of the rotor is the fact that they have their own disruptive force, which occurs as a result of the existence of unbalanced masses in rotary motion. This is due to the fact that they correspond to the modes of the rotor itself, and the fact that they are generally poorly written off. As a result, in the study of machines with rotor dynamic interest is granted with priority to the first own modes [Urs 98].

Types of bearings used in rotor machines are [Don 02]: bearings with ball bearings, bearings, sliding bearings, bearing with gas.

The machines with large power, the most commonly encountered are the berings with sliding, because of their special features: high capacity, high durability, high depreciation, which is the study of this paper.

2.1 The elastic rotor elastic with rigid bearing without depreciation

There is a disk of mass "m" fixed on a shaft that rotates with constant angular velocity in two rigid bearings (Fig. 1). Elastic constant of the shaft is considered to be less than 10% of the constant elasticity of the concentration bearings.

It is considered also that the gravity centre of the disc, the G point, does not coincide with its geometric center, the point C, which coincides with the centre of the cross section of the shaft. It is noted with “ē” the distance between the two points (which is noted only in the relations, because it need to b deffee about the “e” base of the neper logaritm) and aply the following assumptions: it neglects the mass of the shaft and all the forces of friction; when Ω=0 (the shaft does not rotate), the rotor axes does not deformează; the elastic constant of the shaft is k.

Neglecting the gravitational forces, it is observed that there are two forces acting on the disc: first elastic force of the shaft, which has a tendency to recover it; and then centrifugal force, which is the point of application in the centre of gravity (G), which describes the circle with radius “$r_c + \bar{e}$” (Fig.2).

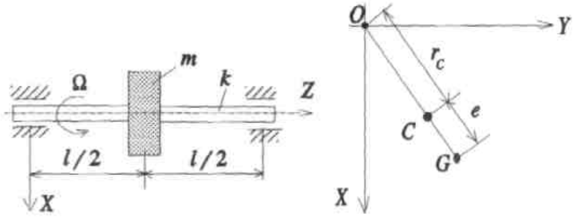

Fig. 1. The rotation notion of the shaft in two rigid bearings [STA 83]

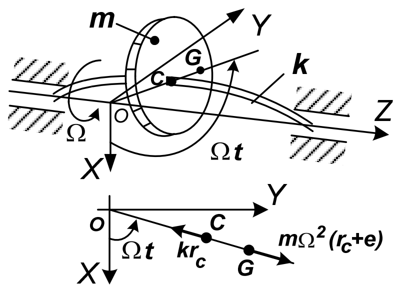

Fig. 2. The forces act on the disc

The C point is geometrical center of the shaft; the G point is the gravity centre of the disc ; ē is the excentricity, hence the distance between them is a constant, and $r_c = OC$ is the arrow shaft in the disc plane. The first of these forces depends on the elasticity of the shaft and is proportional to the arrow. She does so the value “kr_c”, turning to the rotation center, the O point.

Centrifugal force value is $m\Omega^2(r_c + \bar{e})$ and is directed outwards. These two forces must be equal:

$$kr_c = m\Omega^2 m\Omega^2(r_c + \bar{e}) \tag{1}$$

Finally

$$r_c = \bar{e}\frac{\Omega^2}{\frac{k}{m} - \Omega^2} = \bar{e}\frac{\left(\frac{\Omega}{\omega}\right)^2}{1 - \left(\frac{\Omega}{\omega}\right)^2} \tag{2}$$

It analyses the shaft using the D'Alembert principle. It represents the forces on the rotor and writes the differential equations of dynamics of it on the plane along the two directions. They are represented in Figure 3.

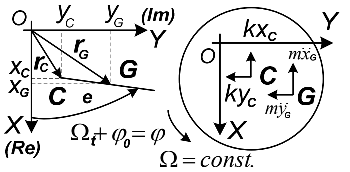

Fig. 3. The active forces on the rotor

The system of scalar differential equations is:

$$\begin{cases} m\ddot{x}_G + kx_c = 0 \\ m\ddot{Y}_G + ky_c = 0 \end{cases} \tag{3}$$

Between the C and G points there are:

$$\begin{cases} x_G = x_c + \bar{e}\cos\phi \\ Y_G = Y_c + \bar{e}\sin\phi \end{cases} \Big| \times i \tag{4}$$

in which i= $\sqrt{-1}$.

It applyse the second relation of the system (4) multiply with "i" and adding with the first, the result is:

$$r_G = r_c + \bar{e}e^{i\phi} \tag{5}$$

but:

$$\begin{cases} r_G = x_G + iy_G \\ r_c = x_c + iy_c \\ e^{ix}\phi = \cos\phi + i\sin\phi \end{cases} \tag{6}$$

If you want to replace the system with current angle, angle which depends on the initial phase, and the angle made in time with the angular constant speed, then the relationship (5) becomes:

$$r_G = r_C + \bar{e}\,e^{i(\Omega t + \varphi_0)} \tag{7}$$

The (4) relations will be deriving in time:

$$\begin{cases} \ddot{x}_G = \ddot{x}_c - \bar{e}\left(\dot{e}\right)^2 \cos\phi - e\ddot{\phi}\sin\phi \\ \ddot{y}_G = \ddot{y}_c - \bar{e}\left(\dot{e}\right)^2 \sin\phi - e\ddot{\phi}\cos\phi \end{cases} \tag{8}$$

But $\ddot{\varphi}$=0 because $\dot{\varphi} = \Omega$=constant.

With (8) relations, the (5) system becomes :

$$\begin{cases} m\ddot{x}_c + kx_c = m\bar{e}\Omega^2\cos\phi \\ m\ddot{y}_c + ky_c = m\bar{e}\Omega^2\sin\phi \end{cases} \Big| \times i \tag{9}$$

Hence:

$$m\ddot{r}_c + kr_c = m\bar{e}\Omega^2 e^{i(\Omega t + \varphi_0)} \Big| \div m \tag{10}$$

Noted with: $\omega = \sqrt{k/m}$, the vectorial sollution of the (9) system is:

$$r_c = \frac{\bar{e}\left(\dfrac{\Omega}{\omega}\right)^2}{1 - \left(\dfrac{\Omega}{\omega}\right)^2} e^{i(\Omega t + \varphi_0)} \tag{11}$$

Relationships (2) and (11) give the law of motion of mass center of the rigid rotor in noelasic bearings without depreciation. In the relationship (11) the first term represents the amplitude of the movement, while the second is a harmonic function corresponding to the pulsation movement, so his Ω .

2.2. Symmetric rotor in elastic bearing no damping

It is considered a rotor sitting between two symmetrical berings perfectly elastic, no amortization, as represented in Figure 4.

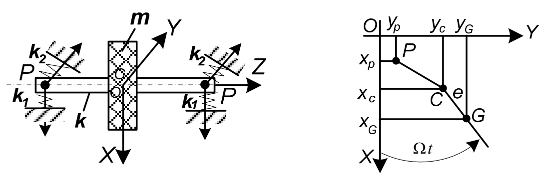

Fig. 4. Symmetric rotor in elastic bearing no damping

It is considered that the axis OX and OY are the main directions of rigidity of bearings, the elastic constants are k_1 and k_2 of the main directions in the bearings, and x_P and y_P are the movements on the two axes of the center on the shaft time in fixed coordinate system XOY.

The rotor charge in given in the figure 5.

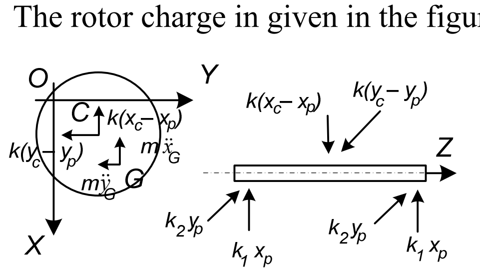

Fig. 5. Mechanical scheme of the rotor and shaft charge

The principle of D'Alembert's is applyed for the fictional dynamic equilibrium and obtain the scalar differential equations system:

$$m\ddot{x}_G + k(x_C - x_P) = 0$$
$$m\ddot{y}_G + k(y_C - y_P) = 0 \qquad (12)$$

in which they give the notation:

$$2k_1 x_F = k(x_C - x_P)$$
$$2k_2 y_P = k(y_C - y_P) \qquad (13)$$

Between the C and G point there are the linkage relations (4), but the inside the (12) and (13) they realize the elimination of x_P, y_P, x_G, y_G coordinates. The result is:

$$m\ddot{x}_C + k_X x_C = m\bar{e}\,\Omega^2 \cos\Omega t$$
$$m\ddot{y}_C + k_Y y_C = m\bar{e}\,\Omega^2 \sin\Omega t \qquad (14)$$

The elastic constants along the two perpendicular directions are:

$$k_X = \frac{2k_1 k}{2k_1 + k};\quad k_Y = \frac{2k_2 k}{2k_2 + k} \qquad (15)$$

The general solution of the (14) system is given by:

$$\begin{cases} x_c(t) = x_c \cos(\omega_x t + \phi_{ox}) + \dfrac{\bar{e}\left(\dfrac{\Omega}{\omega_x}\right)^2}{1 - \left(\dfrac{\Omega}{\omega_x}\right)^2}\cos\Omega t \\ y_c(t) = y_c \sin(\omega_y t + \phi_{oy}) + \dfrac{\bar{e}\left(\dfrac{\Omega}{\omega_y}\right)^2}{1 - \left(\dfrac{\Omega}{\omega_y}\right)^2}\sin\Omega t \end{cases} \qquad (16)$$

Because the mechanical system is anisotropic, hence $k_1 \neq k_2$, results as the system has two owner pulsations.

3. CONCLUSIONS

Where is studied the elastic rotor dynamics in bearing no damping, the system rigid presents same movement both for point C and point G, which is called the precession synchrony motion (Fig. 6.b.) with graphics representation of position vectors with the implementation of own pulsation retail system synchronously (Fig. 6.a.), and in Figure 7 are played the pulsations for symmetric rotor attached to elastic bearing, no damping, but anisotropic.

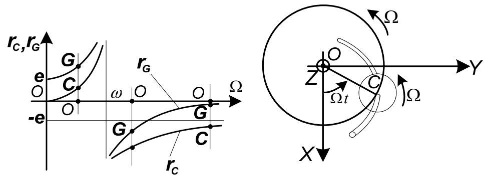

Fig. 6. The precession synchrony motion

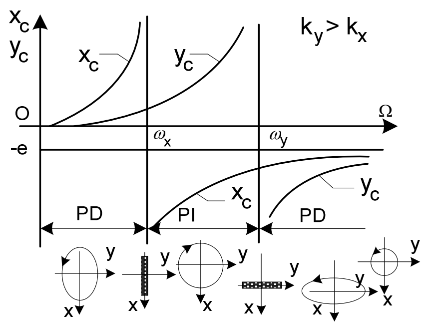

Fig. 7. Rotor on the elastic bearings, anisotropic

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Mentenanța predictivă în eliminarea defectelor echipamentelor

Rezumat: *Lucrarea conține fundamentarea teoretică pentru eliminarea defectelor masinilor si utilajelor, cu organe de masini in miscare de rotatie. Se analizează masinile dinamice rotative, prevazute cu rotor elastic in lagare rigide fara amortizare, precum si rotorul simetric cu miscarea de rotatie in lagare elastic fara amortizare. Principalele defecte la echipamentele dinamice rotative sunt eliminate prin echilibrarea statica si dinamica, prin care se aplica direct mentenanta predictivă.*

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