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OPTIMIZATION OF VIBRATION AMPLITUDES OF THE DYNAMIC ROTORS BY INTRODUCING HYSTERESIS PARAMETERS OF MATERIALS.

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Abstract

The aim of this work is the study of behavior of rotor dynamics of industrial turbines, using numerical simulation. Finite element model was developed by introducing a new hysteresis parameter to control more precisely the behavior of rolling bearings. The finite element model is used to extract the natural frequencies and modal deformed rotor vibration, as it identifies the constraints acting on the system and predict the dynamic behavior of the rotor transient. Results in Campbell diagram and those relating to the unbalance responses show significant amplitude differences in the parameters of hysteresis imposed.

Key words: *rotor dynamics, hysteresis, finite element, rotor vibration, unbalance responses, Campbell diagram.*

INTRODUCTION

The rotors are defined like any element turning around a fixed axis. The field of the rotors is vast, for example the animated geostationary satellites of a free rotation being in accordance with the definition of the rotors, such rotors are called free rotors, in opposition to the fixed rotors which are supported by stages. The fixed rotors are present in many fields: aeronautics, the car, nuclear power or oil industry. In our study, one will focus oneself on the fixed rotors, more precisely on those with horizontal axis applied in particular to the rotors of turbines of electrical production. The axial vibrations and of torsion are uncoupled from the vibrations more of inflection in which one is interested more particularly in overall dynamics. Essential studies of the dynamics of the rotors concerning the layout of the diagram of Campbell who represents the evolution of the Eigen frequencies according to the number of revolutions and the calculation of the answers to unbalance mainly at the time of the passage the critical velocities. The existence of the gyroscopic moments responsible for the variation of the Eigen frequency according to the number of revolutions or that of circulatory forces, make the movement unstable starting from a certain speed in the linear field. The set of themes approached within the framework of this work relates to the study of the dynamic stability of horizontal rotors, and more precisely the rotors of turbines industrial.

MODELING MATHEMATICAL FORMULATION

Rotor systems usually consist of many disks attached to a shaft that are constrained by bearings, dampers, seals etc. to a small lateral motion. For simplicity, a horizontal Jeffcott rotor with a disk of mass m mounted at mid-span of a mass less elastic shaft is considered in this study. The coordinates x, y and ξ, η represent the stationary and rotating axes. The eccentricity of the mass center of the disc

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from the geometric centre of the disc is *e*. The rotational speed and the damping coefficient are Ω and *c*, respectively.

Since the local flexibility induced by the transverse crack generates equations of motion with periodic stiffness coefficients in the fixed reference frame, the equations of motion of Jeffcott rotor are expressed in a rotating frame as follows:

$$m\left(\ddot{\xi}-2\Omega\dot{\eta}-\Omega^2\xi\right)+c\left(\dot{\xi}-\Omega\eta\right)+k_{\xi}\xi+k_{\xi\eta}\eta$$
$$=me\Omega^2\cos\alpha-mg\cos\Omega t$$
$$m\left(\ddot{\eta}+2\Omega\dot{\xi}-\Omega^2\eta\right)+c\left(\dot{\eta}+\Omega\xi\right)+k_{\eta\xi}\xi+k_{\eta}\eta$$
$$=me\Omega^2\sin\alpha+mg\sin\Omega t$$

where $k_{\square}$, $k_{\square}$ are direct stiffnesses along ξ, η directions, and $k_{\xi\eta}$, $k_{\eta\xi}$, are cross-coupled stiffness's.

The setting in equations of a system of solids, connected by flexible elements (of known stiffnesses) and of the shock absorbers (of constant data) provides one exact modeling of the system. The finite element method (MEF) breaks up a structure in small simple elements, discretized, of standard beam, plate, rotor…etc. the vibratory behavior of the element is brought back to that of the nodes. Then one connects the various elements (grid) by observing the conditions of continuity.

we can extract the matrices from mass and stiffness of the structure complete and deduce the own pulsations from the system not deadened and its normal modes. This tool for simulation of mechanical systems allows to better understand the behavior of a rotor. The gyroscopic effect must be taken into account and of the methods of resolution specific can be advantageously employed.

For the disk :

$$T_d=\frac{1}{2}M_d(\dot{u}^2+\dot{w}^2)+\frac{1}{2}I_{dx}(\dot{\theta}^2+\dot{\psi}^2)+\frac{1}{2}I_{dy}(\Omega^2+2\Omega\dot{\psi}\theta)$$

$$\frac{d}{dt}\left(\frac{\partial T}{\partial\dot{\delta}}\right)-\frac{\partial T}{\partial\delta}=\begin{bmatrix}M_d&0&0&0\\0&M_d&0&0\\0&0&I_{dx}&0\\0&0&0&I_{dx}\end{bmatrix}\begin{bmatrix}\ddot{u}\\\ddot{w}\\\ddot{\theta}\\\ddot{\psi}\end{bmatrix}+\Omega\begin{bmatrix}0&0&0&0\\0&0&0&0\\0&0&0&-I_{dy}\\0&0&I_{dy}&0\end{bmatrix}\begin{bmatrix}\dot{u}\\\dot{w}\\\dot{\theta}\\\dot{\psi}\end{bmatrix}$$

For the shaft:

$$T_a=\frac{\rho S}{2}\int_0^L\left[\delta\dot{u}^t N_1^t N_1\delta\dot{u}+\delta\dot{w}^t N_2^t N_2\delta\dot{w}\right]dy+\frac{\rho I}{2}\int_0^L\left[\delta\dot{u}^t\frac{dN_1^t}{dy}\frac{dN_1}{dy}\delta u+\delta\dot{w}^t\frac{dN_2^t}{dy}\frac{dN_2}{dy}\delta w\right]dy$$

$$-2\rho I\Omega\int_0^L\delta\dot{u}^t\frac{dN_1^t}{dy}\frac{dN_2}{dy}\delta w\,dy+\rho I L\Omega^2$$

The deformation energy is obtained from the expression :

$$U_a=\frac{EI}{2}\int_0^L\left[\delta u^t\frac{d^2N_1^t}{dy^2}\frac{d^2N_1}{dy^2}\delta u+\delta w^t\frac{d^2N_2^t}{dy^2}\frac{d^2N_2}{dy^2}\delta w\right]dy$$

For the bearings:

The characteristics of stiffness and damping connect the forces to displacements and speeds. The influence of the slopes and the moments is usually neglected. :

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$$\begin{bmatrix} F_u \\ F_\theta \\ F_w \\ F_\psi \end{bmatrix} = -\begin{bmatrix} k_{xx} & 0 & k_{xz} & 0 \\ 0 & 0 & 0 & 0 \\ k_{zx} & 0 & k_{zz} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u \\ \theta \\ w \\ \psi \end{bmatrix} - \begin{bmatrix} c_{xx} & 0 & c_{xz} & 0 \\ 0 & 0 & 0 & 0 \\ c_{zx} & 0 & c_{zz} & 0 \\ 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} \dot{u} \\ \dot{\theta} \\ \dot{w} \\ \dot{\psi} \end{bmatrix}$$

The first matrix is a matrix of stiffness, the second a matrix of viscous damping. These matrices are generally not symmetrical (hydrodynamic bearings) and the terms can vary in a important manner according to the number of revolutions .

For the mass unbalance :

The general expression of the unbalance is given below, the of Lagrange gives:

$$\frac{d}{dt}\left(\frac{\partial T}{\partial \dot{\delta}}\right) - \frac{\partial T}{\partial \delta} = -m_b\, d\, \Omega^2 \begin{bmatrix} \sin \Omega t \\ \cos \Omega t \end{bmatrix}$$

kinetic energy due to an application of the equations

RESULTS AND DISCUSSION

The model of compressor object of studied is modeled and discretized by the computation software of the dynamic rotors of the laboratory of Dynamics of the Machines and the Structures INSA de Lyon, by considering a rotor discretized in 31 nodes and three stages of comparable turbines to full discs, the complete rotor is supported by four elastic bearings with damping. The rotor is supposed to be coupled with a motor reducer with an elastic joining with variable, compensation stiffness of stability.

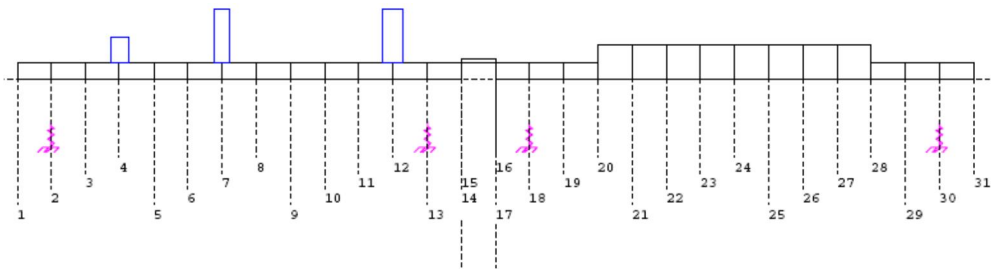


Figure 1: discretization of rotor by 31 nodes and 4 elastic bearings

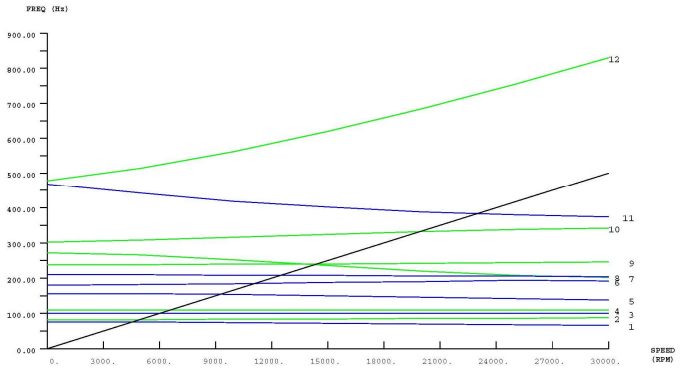


Figure 2: Campbell diagrams, rotor with surigidity

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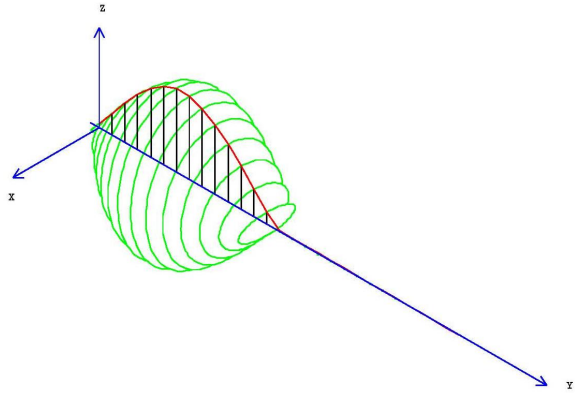


Figure 3: modes in rotation, mode 2 with surigidity

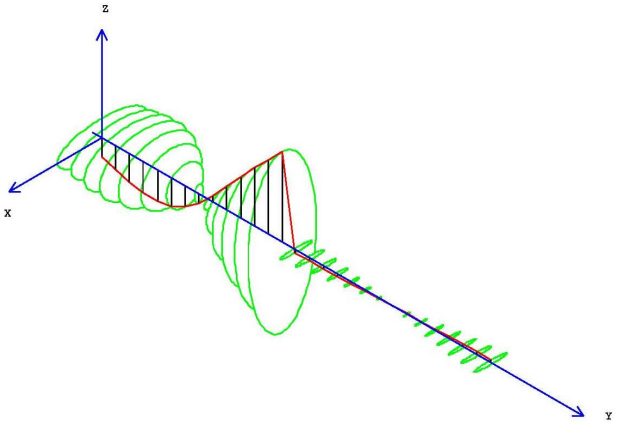


Figure 4: mode 6 in rotation with surigidity

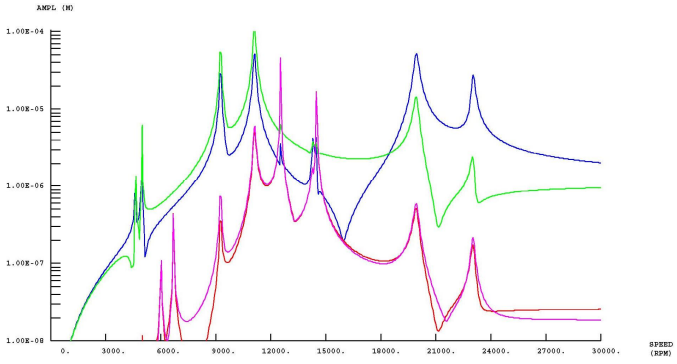


Figure 5: mass unbalance, with surigidity

DISCUSSIONS

a) Diagram of Campbell

One visualizes the whole of these calculations using a diagram of Campbell associated with the deformation with the clean modes, such as represented on the figures 2. Frequencies of excitation taken into accounts are from 0 to 900 Hz. We notice the extent of stability for all the beaches number of revolutions from 0 to 30000 tr/min, and that in the case of direct or opposite precession. The same remarks and observations are to be emitted for case of the rotor with additional rigidity.

b) Positioning critical velocities

The critical velocities must be far away from the normal beach of operation of the line of trees. It is understood easily that them desirable margins of separation between critical velocities and beach of operation are function of the damping of the modes considered and that they increase when damping decreases. One will refer usefully to the API codes (American Petroleum Institute) [5] [6] [7], which defines, according to damping, them minimal margins of separation between a critical velocity and the beach of operation. In conclusion, a critical velocity, result of a first study, not answering this criterion, must then be either moved, or deadened more, while intervening on the geometry of the mobiles and on the characteristic of the stages.

c) Modes of rotations

Curves represented in figures 3 and 4, in the case of surigidity of the rotor, the deformations of the rotor represent in space in rotation, more precisely the trajectories of the centers gravities of various points of the rotor. In both cases, namely normal rotor or rotor with additional rigidity, the curves represent 2^{nd} mode of inflection with 6^{th} mode of inflection, present elliptic trajectories because of gyroscopic coupling, we note that the ellipses are all the more flattened since the anisotropy of the stages is high. We also note that the amplitudes of vibration varies proportionally with rotational frequencies for each mode of rotation.

d) answer to the mass unbalance

Software used for the determination of the clean modes of inflection also make it possible to carry out calculations of answer to one unbalance. This analysis consists in determining, for each one of critical velocities, the response of the rotor to preset unbalances and judiciously placed in order to have an important answer of rotor.

Thus, to evaluate the incidence of an unbalance on the first mode of inflection, we will place this unbalance to the maximum of the deformation, that is to say with vicinity of the center of the rotor. On the other hand, for a mode of end of tree (figure 4), the unbalance will be placed at the end of this end of tree, we will then obtains the amplitude of the vibrations in each point of rotor for a given unbalance. These calculations are to be realized on all modes of vibrations.

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