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DEFECTS DIAGNOSIS IN ROTATING MACHINERY: VIBROACOUSTIC ANALYSIS

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1. INTRODUCTION

The objective of our work is the association of vibratory diagnosis knowledge with an acoustic part including frequential and spatial criteria for the identification of noise sources in rotating machinery. This paper gives the theoretical modelisation of a rotor on a bearings system, with a simulation of unbalance and defective bearings. The acoustic part is elaborated from two approaches:

- the first one, based on a simplified integral formulation, allows the acoustic pressure radiated in the free field to be predicted; and characterizes the acoustic behaviour of the rotor with or without defects.
- the second one permits from vibratory and acoustic measurements the construction of the vibroacoustic transfer function of the structure on its site, and thus the calculation of the acoustic radiation for any other vibratory state.

Then, we analyze from numerical simulations, a vibroacoustic diagnosis procedure contributing towards some defect detection in rotating machinery. The first approach consists of a validation reference for the second one.

2. VIBRATING MODEL OF ROTOR DEFECTS

The system consists of a rotor and journal bearings [1]. The journal bearings are modeled by springs which are of two types: springs in translational motion and springs in rotational motion.

The rotor is made of rigid disks and the shaft is modeled by a beam in bending. Journal bearings stiffnesses are known in both directions X and Z. Ω stands for the rotor angular velocity and θ and Ψ for the angles of the shaft around X and Z due to the beam bending deflection (Figure 1).

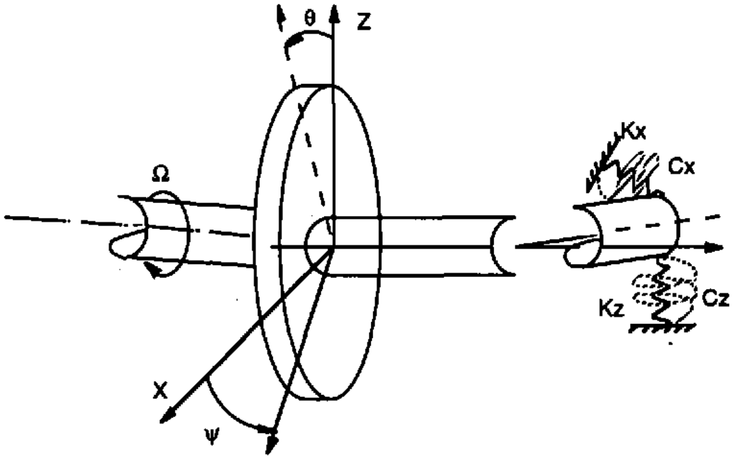

Figure 1: Rotor model

Displacements and rotations are given by the following expressions:

$$X(y,t) = \sum_{i=1}^{N} q_i(t)\sin(i\pi y/L) + \sum_{i=0}^{N} p_i(t)\cos(i\pi y/L) \qquad (1)$$

$$Z(y,t) = \sum_{i=1}^{N} h_i(t)\sin(i\pi y/L) + \sum_{i=0}^{N} t_i(t)\cos(i\pi y/L) \qquad (2)$$

$$\theta(y,t) = \frac{\partial Z(y,t)}{\partial y} \quad (3) \quad \text{and} \quad \Psi(y,t) = -\frac{\partial X(y,t)}{\partial y} \qquad (4)$$

Lagrange's equations associated with the Rayleigh Ritz method yield a system of differential equations that describe motion and whose degree of freedom N is limited:

$$\begin{bmatrix} [M] & O \\ O & [M] \end{bmatrix} \left\{\ddot{\delta}\right\} + \begin{bmatrix} O & [-C(\Omega)] \\ [C(\Omega)] & O \end{bmatrix} \left\{\dot{\delta}\right\} + \begin{bmatrix} [K_x] & O \\ O & [K_z] \end{bmatrix} \left\{\delta\right\} = \left\{F\right\} \qquad (5)$$

where [M], [C], $[K_x]$ and $[K_z]$ represent mass, gyroscopic, and stiffness matrices, and $\delta = \left\{q_i \quad p_j \quad h_i \quad t_j\right\}$.

The dynamic effects of some defects (such as unbalance or defects in bearings) are introduced in the second member {F} [1] of Eq. (5). Then, the modelling of different coupled defects may be carried out by applying the superposition principle in the expression of {F}. The vibration response is a spectrum with the same shape as the one characterizing the discrete system transfer function. One can therefore analyze the rotor vibrations along axes X and Z as a function of the shaft running speed, the magnitudes and positions in space of the different excitations whether or not they are coupled.

3. NOISE PREDICTION

The procedure consists in calculating the acoustic pressure in the outer field, according to the knowledge of the structure as a vibrating surface. The fluid medium is air, we can neglect its effects on the structure response. From vibratory computation carried out all over the rotor surface, the radiated acoustic pressure calculation constitutes the reference value of the acoustic system response for different simulations whether or not the defects are taken into account. The vibroacoustic transfer function construction, worked out from this acoustic reference value and vibratory velocities defined on the non-rotating surfaces of the system (such as the journal bearings), permits the spatial and frequential evolution prediction of the sound radiated from any vibratory state modification.

First Approach: free field

To solve this problem, we use the integral formulation for the acoustic pressure; we make some simplifying assumptions that allow us not to affect noise prediction from the qualitative point of view and with a satisfying accuracy for the quantitative predictions.

For low frequencies, surface pressure can be determined by introducing the hypothesis concerning dynamics of incompressible fluids. This hypothesis is valid when the acoustic wavelength ($\lambda=2\pi/k$) is greater than the biggest dimension L of the structure (S). It is given by the following expression: $P(M_0)=j\rho_0\omega V_n(M_0)$ (6)

The acoustic pressure at a point M of the acoustic medium is:

$$P(M)=\int\int_S ((\frac{L}{r}+jkL)\cos\theta+1)j\rho_0\omega V_n(M_0)\frac{e^{-jkr}}{4\pi r}dS \qquad (7)$$

with $\cos\theta=\frac{\overrightarrow{M_0M}}{r}\overrightarrow{n_{M_0}}$

In the high frequencies domain and when the sound wavelength is smaller than the structure curvature, the radiation impedance tends towards $\rho_0 c$. Therefore, the surface pressure $P(M_0)$ may be related to the normal surface velocity:

$$P(M)=\int\int_S (\cos\theta+1)j\rho_0\omega V_n(M_0)\frac{e^{-jkr}}{4\pi r}dS \qquad (8)$$

Second Approach: with the construction of the vibroacoustic transfer function in situ [3]

The method consists in constructing Green's function of the vibroacoustic system (the vibrating structure (S) in its site bound by a surface (Σ)). The vibrating surface is modelled by a distribution of point sources with an unknown density function allocated. The latter is calculated from acoustic pressure measurements in terms of modulus and phase. This density function combined with vibratory velocity measurements on the structure surface, permits the vibroacoustic transfer function $G_s(M,M_0)$ to be numerically constructed; we introduce the following assumptions:

1- the Green's function G_S verifies on (S) and (Σ) boundary conditions of type: $\frac{\partial G_s(M,M_0)}{\partial n_{M_0}} = 0$ on (S) and (Σ).

2- the normal vibratory velocities on (Σ) are negligible compared with the ones on (S).

A mesh of n acoustic pressures is achieved around the vibrating surface; for each receiver point M of the acoustic mesh, the construction of the transfer function $G_s(M,M_0)$ leads to minimize the following expression:

$$\iint_S (j\omega\rho_0 v_n(M_0)G_s(M,M_0) - \tilde{\mu}(M_0)g(M,M_0))dS \qquad (9)$$

where $V_n(M_0)$ is the normal vibratory velocity measured at each point source M_0 on the vibrating surface (S), $g(M,M_0)$ the free-space Green's function. The density function $\tilde{\mu}$ is calculated at each point source M_0 by resolution of the inverse problem from acoustic pressures measured around the vibrating structure.

Then, for any other vibratory state of the surface (S), the acoustic pressure can be worked out at a point M' of the real acoustic medium:

$$P(M') = \iint_S j\omega\rho_0 v_n(M_0)G_s(M',M_0)dS \qquad (10)$$

4. RESULTS AND CONCLUSIONS

The system choosen for this analysis is made up of a rotor with two disks and two journal bearings. The presence of two disks permits to increase the eigenfrequencies variation according to the angular velocity. The unbalance is simulated by placing some light masses on the disks.The bearings defect is simulated by changing the stiffness of the journal bearings and by applying a periodic excitation.

From the linear dynamic model given by Eq. (5), the vibratory velocites are worked out at every point of the rotating system. Next, the radiated acoustic pressure is calculated from Eqs (7) and (8). These computations are carried out for different defects configurations, and the acoustic pressure obtained in each case constitutes the reference value for the second approach using the transfer function construction in situ (9). In practice, vibratory velocities can not be necessarily measured all over the structure surface (for example, because of some rotating elements). Then, vibratory velocities defined only on the journal bearings, and the acoustic reference pressure are used in order to build this transfer function. This construction completes the missing vibratory information. The use of these vibratory velocities in Eqs (7) and (8) should yield erroneous pressure values beyond the first vibratory mode of the rotor. Figure 2, illustrating our remarks, displays the spatial variations of the sound pressure level along the rotor, at the rotational frequency when an unbalance is simulated (two light masses placed in opposite phase on the two disks).

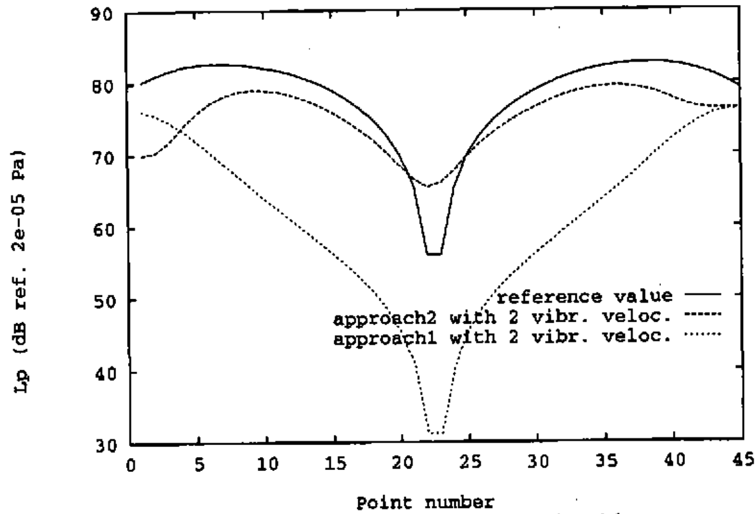

Figure 2: Spatial variations of sound pressure level for unbalance defect at the rotational frequency.

In conclusion, this study shows the interest of the acoustic information in defects detection. This simulation permits to free the values from measurements errors (due to background noise, random variations, ...), and to analyze the method quality through different validation procedures. An industrial case trial of this method is essential to complete the validation.

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