FREQUENCY DEPENDENT DYNAMIC PROPERTIES OF TILTING PAD JOURNAL BEARINGS: EXPERIMENTAL RESULTS AND UNCERTAINTY ANALYSIS

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ABSTRACT
The paper presents experimentally obtained TPJB response to multifrequency excitation and its comparison with theoretically obtained data. Uncertainty considerations for the results obtained using the power spectral density method are also presented. It has been concluded that inertia forces and pivot flexibility effects are behind the variations of dynamic coefficients with frequency of excitation.

1. INTRODUCTION
It has been expected that the dynamic coefficients of tilting-pad journal bearings are frequency dependent ([1]). However, considering uncertainty of the results, the experimental evidence has been rather inconclusive (e.g. [2]). This paper presents experimental results of bearing response for a wide range of frequency of excitation. The test bearing was a five pad TPJB with load-on-pivot configuration.

It is well known that the measured bearing dynamic coefficients have relatively significant uncertainties. They are estimated based on the statistical analysis of the already obtained results. This paper presents an approach based on an analysis of error propagation for a frequency domain method.

2. EXPERIMENT
The test rig has been described elsewhere ([3]). The test bearing parameters are presented in Table 1.

Table 1 Test bearing parameters
| Pad length | 0.0396 m |
| Nominal bearing diameter | 0.0987 m |
| Bearing radial clearance / Preload | 0.088×10⁻³ m / 0.3 |
| Number of pads/Pad angular extension | 5 / 55° |

Advantages of using frequency domain have been known for many years ([4]). For the experimental setup with the floating test bearing the equation of motion in frequency domain and after introducing auto and cross spectral densities can be written as

\[ G_{F_x F_x}(\omega) - m_b G_{F_x A_x}(\omega) = H_{F_x}(\omega) G_{F_x A_x}(\omega) + H_{A_x}(\omega) G_{A_x F_x}(\omega) \]  
\[ G_{F_y F_y}(\omega) - m_b G_{F_y A_y}(\omega) = H_{F_y}(\omega) G_{F_y A_y}(\omega) + H_{A_y}(\omega) G_{A_y F_y}(\omega) \]

where

\[ H_x = k_x + i\omega c_x; \]

power spectral densities \( G_m(\omega) \) (autospectral density) and \( G_{xy}(\omega) \) (cross spectral density) are defined as

\[ G_m(\omega) = \lim_{T \to \infty} \frac{2}{T} \mathbb{E}[u^*(\omega) u(\omega)] \]  (* denotes complex conjugate)

\[ F_x, F_y, A_x, A_y, X, Y \] are Fourier transforms of excitation forces in the horizontal and vertical directions, accelerations in the horizontal and vertical directions, and displacements in the horizontal and vertical directions, respectively,

\( m_b \) - bearing mass
\( f_x, f_y \) - components of the dynamic force
\( x, y \) - shaft center coordinates in the rectangular system with the origin at the bearing equilibrium position
\( k_{xx}, k_{yy}, k_{xy}, k_{yx} \) - stiffness coefficients
\( c_{xx}, c_{yy}, c_{xy}, c_{yx} \) - damping coefficients.

4. UNCERTAINTY OF THE EXPERIMENTAL RESULTS
Typically, uncertainty of the measured bearing dynamic properties is evaluated based on statistical analysis of the obtained data, which is usually referred to as Type A analysis ([5]). Such an approach gives an idea about the consistency of the results. Here, the uncertainty analysis is aimed at estimating the potential effect of elemental uncertainties of individual measurements on the uncertainty of the evaluated coefficients, which is referred to as Type B analysis ([5]). In other words, a relationship has been sought linking time domain and frequency domain uncertainties.
The equation for uncertainty $u_i(y)$ of a result $y$, which is often referred to as the law of propagation of uncertainty, is as follows:

$$u_i(y) = \left[ \sum_{i=1}^{n} \left( \theta_i u_i(x_i) \right)^2 \right]^{1/2}$$

(4)

where $\theta_i$ sensitivity coefficients, $\theta_i = \frac{\partial y}{\partial x_i}$, $u(x_i)$ standard uncertainty associated with the input $x_i$, $n$ number of records, $\Delta t$ time increment for sampling data.

The elemental uncertainties $u$ were $2.5 \, \mu m$ for displacement and $0.01 \, g$ for acceleration.

5. CALCULATIONS

The calculations have been based on a thermoelastohydrodynamic model, which was described in [3]. The calculated dynamic coefficients include the effect of pivot stiffness.

6. RESULTS AND DISCUSSION

As an example, the frequency response functions $H_{ij}$ representing the bearing direct coefficients in the load direction are illustrated in Figure 1. For a very lightly loaded bearing (2.02 kN, 14876 rpm), the direct bearing stiffness and damping coefficients show very limited variations with frequency of excitation (Figure 1a,b). With increased load these variations become more evident, which is illustrated in Figure 1c. Although, for this bearing, the damping coefficients (represented by the slope of the line) have not varied with frequency of excitation, pad inertia effects became evident in the bearing stiffness at high frequencies. In the test conditions considered in this study, these effects have been significant at frequencies above that corresponding to the shaft speed.

The examples of the results’ uncertainties show yet another advantage of the PSD method. When compared to the time domain method [7], the frequency domain leads to lower uncertainty of the results. The uncertainties shown in Figures 1 c,d do not exceed 10%.

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REFERENCES


