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Dynamic performance of foil bearings with a quadratic stiffness model

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ABSTRACT

Hydrodynamic foil bearings found various applications in high speed small turbomachinery due to several advantages compared with traditional oil lubricated bearing and rolling element bearings such as high speed, pollution free, high temperature, long service life, etc. Corrugated bump foils made of thin sheet metal is the most widely adopted foil structure used in many commercial foil bearings. However, due to various factors during manufacturing process of the foil structure and mounting methods of the foil structure on to the bearing sleeve, measured stiffness curves of the foil structure show very strong non-linearity. The non-linearity of the foil structure is often neglected in most simulation studies in the past, and single value of bump stiffness combined with appropriate structural loss factor were used in these simulation studies. Based on measured non-linear structural stiffness of entire foil bearing, a quadratic stiffness model of individual bump foil was derived and utilized in transient rotordynamic simulations in time domain by solving Reynolds equation, rotor equations of motion, and bump deflections together. The simulation results using the quadratic bump stiffness model were compared with those using linear bump stiffness models.

1. Introduction

Bump-type foil bearings are compliant, self-acting hydro-dynamic air bearings which are used in high-speed rotating machinery such as turbo expander etc [1]. Compared with oil bearings and rolling element bearings, these bearings have excellent reliability, environment durability and can be operated at high speeds and temperatures [2]. Moreover, the foil bearings have less environmental pollution because the lubricate media is gas. The main process of predicting bump-type foil bearings is modeling the foil structure and solving compressible gas lubricated Reynolds equation coupled with the compliant surface. From the idea of compliant foil bearing first mentioned in the beginning, modeling of foil structure always accompanies with bearing simulation.

Wallowit and Anno [3] first gave the theoretical model of a single bump static stiffness equation using mechanical beam model without inner friction consideration. The bump stiffness can be obtained easily and quickly only using bump geometry parameters. This model is so convenient that a lot of researches simulate the bearing performance using this model. In 1983, this bump static stiffness equation was used by Heshmat et al. to analysis gas lubricated foil journal bearing and gas lubricated compliant thrust bearing [4,5]. Even in 2000s, Peng and Khonsari [6–8] still employed this static stiffness equation to predict the hydrodynamic performance of a foil journal bearing such as load-capacity and film thickness. The model was established based on the compliance of the bearing surface.

Since more and more people found foil bearing structural stiffness and damping can be enhanced by the friction between top foil and bump foil, and bump foil and bearing housing, more attention was paid to bump structural modeling in consideration of friction insider. Roger and Heshmat [9] studied the stiffness characteristics of bump foil structure by introducing friction force between bump foil and the housing or the top foil. Then they used the perturbation method and closed hysteresis loop to evaluate the stiffness and damping of the bearing [10]. Carpino and Talmage [11] used equivalent viscous dissipation to model the foil structure as a simple elastic foundation via adding Coulomb friction into the viscous damping coefficient formula and the effects of friction, orbit size and frequency on rotor dynamic coefficients are simulated using finite element method. In 1999, Iordanoff [12] presented a very simple model for an aerodynamic compliant foil thrust bearing. The model considered the different stiffness of welded bump and free bump with consideration of Coulomb friction force between bump foil and top foil. The results showed that stiffness of welded bump is larger than free ends bump. Swanson [13] developed a new bump foil damping model via simplifying bump strips with two friction interfaces to a single bump with a single friction interface. The model comprised two springs with load dependent friction force and a mass. Then the multiple harmonic balance (MHB) method is applied to get the approach of characteristics bump foil. The results of static response match the experimental data trends due to the friction

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Nomenclature

h_b	bump height (mm)
l_0	half of the bump length (mm)
s	bump pitch (mm)
t	foil thickness (mm)
v	Poisson's ratio
k_{XX},k_{YY}, k_{XY},k_{YX}	bearing stiffness coefficients (N/m)
D_{XX},D_{YY}, D_{XY},D_{YX}	bearing damping coefficients (Ns/m)
A_0,A_1,A_2	bump coefficients
C	nominal bearing radial clearance (mm)
C_{00}	initial bearing radial clearance for simulation (μm)
D	bearing inner diameter (mm)
D_0,D_1,D_2	coefficients for bump stiffness
H	normalized film thickness ($=h/C$)
K_{Ni}	single bump non-linear stiffness (N/mm)
K_s	single bump linear stiffness (N/mm^3)
L	axial bearing length (mm)
N	number of bumps
P	normalized pressure ($=p/p_a$)
R	bearing radius (mm)
W_0	reference load ($W_0=2p_aRL$)
Z_{Ni}	deflection of Nith bump (mm)
X,Y,Z	Cartesian coordinates
$\varepsilon_{X,Y}$	normalized journal eccentricity in X and Y
Λ	Bearing number ($=6\mu\omega R^2/p_aC^2$)
μ	viscosity of air
θ	circumferential coordinate
θ_{Ni}	angle between single bump height direction and Y direction
τ	normalized time ($=\omega t$)
ω	rotation speed of a rotor

force in the model. Lez et al. [14] used a classical isotropic Coulomb friction model to establish the foil structure model. The finite element method was used to calculate the static and dynamic stiffness of foil structure and the influence of friction coefficient was studied. They found that the stiffness under different load cases had the same trends and tendencies as experimental results in literatures. They also developed a foil structural multi-degree freedom model in consideration of interaction between the bumps and the finite element method is used to calculate the static stiffness of bump foil. They also made an experimental procedure to validate their theoretical results [15]. In 2009, Lee et al. [16] used a transient analysis method to study the performance of foil bearings with the consideration of Coulomb friction. The stick-slip motion of bump foil is solved by using a direct implicit integration technique. The hysteretic behavior of the bump foil is observed and results showed that the foil bearing has a less an amplitude than the rigid bearing at the resonance frequency. They also developed a static foil structural model considering the hysteretic behavior of the friction [17]. Feng and Kaneko [18] presented a complete analytical model of bump-type foil bearing that took into account the effects of the stiffness of bumps, interaction forces, friction forces and local deflection of the top foil. The single bump is assumed to two rigid links and a horizontally spaced spring. Although friction is considered in foil structure modeling, the simulation result of bump stiffness doesn't agree well with test data of bearing stiffness which show non-linear features [19,20].

Ideal model is finite clearance and constant bump stiffness. This case is only possible when all the foils are manufactured ideally sitting ideally each other creating finite visible clearance. But it never happens because foils are not manufactured ideally they never sit together. So we see load-deflection curve similar to San Andres's paper [19].

If shaft is heavy and initial displacement of rotor is large close to the design clearance, once rotor spins, large pressure is built creating the clearance following the ideal model.

For three pad design with hydrodynamic preload, even if foil structures are loose and clearance is not visible during assembly, once rotor spins, clearance is immediately formed.

But for single pad design with very light rotor, rotor deflection is so small, not creating much hydrodynamic pressure, loose foil structure can be modeled as equivalently soft bump and small clearance (because loose foil structure does not allow any visible clearance). In this paper, a new approach to simulate this single pad bearing with light rotor using a quadratic bump stiffness model base on measured load-deflection curve, and simulation results such as dynamic coefficients using this model were compared with those using linear bump stiffness models.

2. Stiffness model and analysis method

2.1. Structural quadratic stiffness model

The foil structural quadratic stiffness model is developed using foil bearing static stiffness test data in [19], and the same bearing is used for simulations in this paper. The length and inner diameter of bearing are 38.10 mm and 38.17 mm, respectively. A shaft with 38.10 mm diameter is chosen to get load-deflection curve. The bearing parameters are summarized in Table 1 and bump configuration is presented in Fig. 1, respectively. The nominal bearing radial clearance is 35.5 μm.

From the original load-deflection curve with 38.10 mm diameter shaft, polynomial load-deflection curve was reconstructed as a polynomial function using commercial data extraction software as shown in Fig. 2 and Eq. (1). The curve shows non-linear characteristics of the entire bearing structure:

$$F(X) = 1.516e^5X^3 + 3010X^2 + 953.1X \tag{1}$$

where $F(X)$ is the bearing load and X is the displacement of shaft. Eq. (1) is force equation for entire bearing, not for single bump.

Non-linear stiffness of single bump can be derived through the process to be described herein. The bearing load-deflection testing process is shown in Fig. 3. During the loading process, half of bumps are deflected and reaction forces along the vertical direction of all the deflected bumps are added. The total added reaction forces from the bumps are equal to the bearing external load. The load-deflection equation of single bump can be derived by using

Table 1
Nominal dimensions and parameters of bearing [19].

Foil bearing parameters:	Values
Inner diameter of sleeve (mm), D	38.17
Axial bearing length (mm), L	38.10
Nominal clearance (mm), C	0.0355
Number of bump, N	25
Bump pitch (mm), s	4.572
Bump length (mm), $2l0$	4.064
Foil thickness (mm), t	0.102
Bump height (mm), hb	0.381
Poisson's ratio, ν	0.29
Modulus of elasticity (MPa), E	213.73

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