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A new modal-based approach for modelling the bump foil structure in the simultaneous solution of foil-air bearing rotor dynamic problems

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ABSTRACT

Recently-proposed techniques for the simultaneous solution of foil-air bearing (FAB) rotor dynamic problems have been limited to a simple bump foil model in which the individual bumps were modelled as independent spring-damper (ISD) subsystems. The present paper addresses this limitation by introducing a modal model of the bump foil structure into the simultaneous solution scheme. The dynamics of the corrugated bump foil structure are first studied using the finite element (FE) technique. This study is experimentally validated using a purpose-made corrugated foil structure. Based on the findings of this study, it is proposed that the dynamics of the full foil structure, including bump interaction and foil inertia, can be represented by a modal model comprising a limited number of modes. This full foil structure modal model (FFSMM) is then adapted into the rotordynamic FAB problem solution scheme, instead of the ISD model. Preliminary results using the FFSMM under static and unbalance excitation conditions are proven to be reliable by comparison against the corresponding ISD foil model results and by cross-correlating different methods for computing the deflection of the full foil structure. The rotor-bearing model is also validated against experimental and theoretical results in the literature.

1. Introduction

A FAB (Fig. 1(a)) supports the shaft by means of a cushion of air bounded by a flexible foil structure. The introduction of the foil structure resolves the problems associated with the very tight radial clearance required by a plain air bearing. With a FAB, the hydrodynamic air film pressure generated as the shaft turns pushes the foil boundary away, allowing the shaft to become completely airborne [1]. The dynamics of FAB turbomachinery are governed by the interaction between the rotor, air films and the foil structures, and exhibit nonlinear effects. The governing system of equations comprises the state (i.e. time-based first order differential) equations of the air films, foils and rotor and can be expressed in state-space form as [2,3]:

$$\mathbf{s}' = \boldsymbol{\chi}(\tau, \mathbf{s}) \tag{1}$$

where $()'$ denotes differentiation with respect to the time variable τ, $\mathbf{s}$ is the state vector, containing the n_{state} state variables (respectively relating to the air film, foil and rotor domains) and $\boldsymbol{\chi}$ is a vector of nonlinear functions of the state variables.

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Nomenclature

$()'$	differentiation with respect to τ
B	foil width (along z_f-axis)
c	undeformed radial clearance of FAB
$\mathbf{D}$	diagonal damping matrix (Eq. (10))
E	Young's Modulus
f_{res}	fundamental resonance of single isolated bump in Hz
$\mathbf{f_p}$	vector of air pressure forces on bumps
F_x, F_y	Cartesian FAB forces
i, j	generic degrees of freedom
$\tilde{h}$	air film gap divided by c
$\mathbf{H_{x_f}}$	foil modal matrix whose columns are $\boldsymbol{\phi}_{\mathbf{x_f}}^{(r)}$, $r = 1 \ldots n_f$
$\mathbf{H_{y_f}}$	foil modal matrix whose columns are $\boldsymbol{\phi}_{\mathbf{y_f}}^{(r)}$, $r = 1 \ldots n_f$
k_{bump}	stiffness of single independent bump
$\mathbf{K}$	FE stiffness matrix of foil structure
l_b	length of one bump along x_f-axis (Fig. 2)
L	axial length of FAB
m_r	half-mass of rotor
$\mathbf{M}$	FE mass matrix of foil structure
n_{bumps}	number of bumps
n_f	number of foil structure modes considered
n_{state}	total number of state variables in Eq. (1).
N_z,N_θ	number of points of FD grid along ξ, θ directions
$p(\xi, \theta, \tau)$	absolute air pressure at (ξ, θ) for FAB
p_a, $\tilde{p}$	atmospheric pressure, p/p_a respectively
$\mathbf{q_f}$	$n_f \times 1$ vector of modal coordinates
r	counter for foil modes
R	undeformed radius of FAB
$\mathbf{R_{x_f y_f}}(0)$	static receptance matrix relating the elements in $\mathbf{u}$ with those in $-\mathbf{f_p}$
$\mathbf{R_{y_f y_f}}(0)$	static receptance matrix relating the elements in $-\mathbf{w}$ with those in $-\mathbf{f_p}$
$\mathbf{s}$	vector of state variables (Eq. (1))
S	static load in y direction per bearing
S_p	pitch of bumps (Fig. 2)
t	time in seconds
t_b	thickness of foil
u	radius of rotor unbalance
$\mathbf{u}$	$n_{bumps} \times 1$ vector of the circumferential displacements at the bump apexes
w	foil deflection in radial direction
$\tilde{w}$	$=w/c$
$\mathbf{w}$	$n_{bumps} \times 1$ vector containing the radial displacements at the bump apexes
x, y, z	Cartesian coordinate system (Fig. 1)
x_J, y_J	Cartesian displacements of journal centre J relative to (fixed) bearing centre
x_f, y_f, z_f	Cartesian coordinate system local to the foil structure (Fig. 2)
$\alpha_{ij}(\omega)$	receptance FRF relating generic degrees of freedom i and j
$\boldsymbol{\Delta}$	diagonal matrix of squares of foil natural circular frequencies
$\varepsilon_{x,y}$	x_J/C, y_J/C
ε	$=\sqrt{\varepsilon_x^2 + \varepsilon_y^2}$
ξ	$=z_f/R$
ζ_{f_r}	viscous damping ratio of foil mode no. r
θ	angular local bearing coordinate (Fig. 1)
$\phi_i^{(r)}$	mass-normalised modal displacement of foil in mode no. r at degree of freedom i
$\boldsymbol{\phi}_{\mathbf{x_f}}^{(r)}$	$n_{bumps} \times 1$ mass-normalised eigenvector in mode no. r containing the circumferential displacements at the apexes of the bumps
$\boldsymbol{\phi}_{\mathbf{y_f}}^{(r)}$	$n_{bumps} \times 1$ mass-normalised eigenvector in mode no. r containing the radial displacements at the apexes of the bumps
γ	attitude angle
Λ	bearing number [2,3].
μ	air viscosity
ρ	density of foil material
η	damping loss factor of foil structure
$\psi(\xi, \theta, \tau)$	$=\tilde{p}\tilde{h}$
τ	non-dimensional time ($=\Omega t/2$)
Ω	rotational speed in rad/s
ω	general circular frequency in rad/s
ω_{f_r}	natural circular frequency of foil mode no. r in rad/s
$\boldsymbol{\chi}$	nonlinear vector function of $\mathbf{s}$ in Eq. (1).

Prior to the work by the co-author [2,3], the simultaneous solution of the state equations in (1) was avoided due to the computational burden involved. As explained in [2,3], this was typically done by approximating the subset of state equations relating to the air film as algebraic equations, effectively decoupling them from the rest of the equations. A non-simultaneous ("time-lagging") solution scheme was applied wherein the *current* value of the local air film thickness, and its rate of change with time, were estimated based on the foil and rotor displacements at the *previous* time step (e.g. as used in [4]). The work in [2,3] has shown that an efficient simultaneous solution is possible in both the time and frequency domains, which preserves the state-space format of Eq. (1) (and thus the true simultaneous coupling of its components), alleviating the potential convergence issues suffered by decoupled integration schemes [5].

The basic strategy for the simultaneous time-domain solution introduced in [2] has been independently verified by other researchers in [6] and successfully used to correlate with experimental results in [7], [5] and [3]. However, the simultaneous solution schemes in [2,3,5–7] are based on a simplified spring-damper model of the foil structure (as described below). The present paper addresses this simplification by devising a means of including more advanced features of the foil (namely, bump interaction and inertia) into the simultaneous solution scheme. The foil structure considered will be of the "first generation" type shown in Fig. 1(a), consisting of a single corrugated strip (comprising a series of bumps) underneath a top foil covering.

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