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An experimental and theoretical analysis of a foil-air bearing rotor system

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ABSTRACT

Although there is considerable research on the experimental testing of foil-air bearing (FAB) rotor systems, only a small fraction has been correlated with simulations from a full nonlinear model that links the rotor, air film and foil domains, due to modelling complexity and computational burden. An approach for the simultaneous solution of the three domains as a coupled dynamical system, introduced by the first author and adopted by independent researchers, has recently demonstrated its capability to address this problem. This paper uses this approach, with further developments, in an experimental and theoretical study of a FAB-rotor test rig. The test rig is described in detail, including issues with its commissioning. The theoretical analysis uses a recently introduced modal-based bump foil model that accounts for interaction between the bumps and their inertia. The imposition of pressure constraints on the air film is found to delay the predicted onset of instability speed. The results lend experimental validation to a recent theoretically-based claim that the Gümbel condition may not be appropriate for a practical single-pad FAB. The satisfactory prediction of the salient features of the measured nonlinear behavior shows that the air film is indeed highly influential on the response, in contrast to an earlier finding.

1. Introduction

The foil-air bearing (FAB) is a key enabler of oil-free turbomachinery technology and its environmental and technological benefits are well documented [1]. Breakthroughs in materials and manufacturing technology reported in the late 1990s/early 2000s have intensified research into its applicability to an ever increasing range of turbomachinery [2]. However, the analysis of FAB-rotor systems presents a challenging nonlinear problem that discourages their use [3]. As stated by Balducchi et al. [3], the designer is not only faced with a bearing that is considerably more complex than traditional journal bearings, but experimental results show nonlinear phenomena that cannot be predicted with traditional theoretical models based on linear rotordynamic coefficients.

The dynamics of FAB-rotor systems are governed by the nonlinear interaction between the air film, foil structure and rotor domains, where each domain is governed by time-based differential equations [4–6]. As discussed in Refs. [4–6], in order to reduce the computational burden, the problem has been subjected to simplification to one or more aspects. One major

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Nomenclature

$(\ )^{'}$	differentiation with respect to τ
c	radial clearance of FAB (undeformed foil)
$\mathbf{D}_\mathrm{f}$	diagonal foil damping matrix, eq. (11)
$\mathbf{f}_\mathrm{J}$	Cartesian FAB forces, eq. (5)
$\mathbf{f}_\mathrm{u}$	vector of unbalance forces
$\mathbf{f}_\mathrm{s}$	vector of static loads
$\mathbf{f}_\mathrm{p}$	vector of air pressure forces on bumps
$\mathbf{g}_\mathrm{RE}$	right-hand side of discretized Reynolds Equation (eq. (2a))
$\tilde{h}$	air film gap divided by c
$\tilde{h}_j$	$= \tilde{h}(\theta_j)$
$\mathbf{H}_\mathrm{P}$	rotor modal matrix, eq. (4)
$\mathbf{H}_{\mathbf{f}_\mathrm{u},\mathbf{f}_\mathrm{s},\mathbf{f}_\mathrm{J}}$	rotor modal matrices in eq. (2c)
$\mathbf{H}_{\mathbf{w}_\mathrm{p}}$	foil modal matrix whose columns are $\phi^{(r)}_{\mathbf{w}_\mathrm{p}}$, $r = 1\ldots n_\mathrm{f}$
i,j	counters for finite difference (FD) grid
L	axial length of FAB
n_bumps	number of bumps
n_f	number of foil structure modes considered
$N_z,\ N_\theta$	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
$\mathbf{q}$	vector of rotor modal coordinates
$\mathbf{q}_\mathrm{f}$	$n_\mathrm{f} \times 1$ vector of foil modal coordinates
r	counter for foil modes
R	undeformed radius of FAB
s	vector of state variables (eq. (1))
$\mathbf{s}_\mathrm{E}$	static equilibrium value of s
S	bump pitch
t	time in seconds
u	foil deflection in tangential direction
w	foil deflection in radial direction
$\tilde{w}$	$= w/c$
$\tilde{w}_j$	$= \tilde{w}(\theta_j)$
$\tilde{\mathbf{w}}$	$= [\cdots \quad \tilde{w}_j \quad \cdots]^\mathrm{T}$
$\mathbf{w}$	$n_\mathrm{bumps} \times 1$ vector containing the radial displacements at the bump apexes
x, y, z	Cartesian coordinate system
$x_\mathrm{J}, y_\mathrm{J}$	Cartesian displacements of journal centre J relative to (fixed) bearing centre
z_f	local axial coordinate for FAB
$\boldsymbol{\Lambda}$	diagonal matrix of squares of rotor natural circular frequencies
$\boldsymbol{\Lambda}_\mathrm{f}$	diagonal matrix of squares of foil natural circular frequencies
$\boldsymbol{\varepsilon}$	$= [x_\mathrm{J}/c \quad y_\mathrm{J}/c]$
$\Delta\xi$, $\Delta\theta$	FD grid spacings in ξ, θ
ξ	$= z_\mathrm{f}/R$
ξ_i	$\xi,\ i = 1, \ldots, N_z$
$\zeta_{\mathrm{f}r}$	viscous damping ratio of foil mode no. r
θ	angular local bearing coordinate (Fig. 1)
θ_j	$\theta,\ j = 1, \ldots, N_\theta$
$\boldsymbol{\phi}^{(r)}_{\mathbf{w}_\mathrm{p}}$	$n_\mathrm{bumps} \times 1$ mass-normalised eigenvector in mode no. r containing the radial displacements at the apexes of the bumps
$\mathit{\Lambda}$	bearing number [5,6].
$\psi(\xi, \theta, \tau)$	$= \tilde{p}\tilde{h}$
$\psi_{i,j}(\tau)$	$= \psi(\xi_i, \theta_j, \tau)$
$\boldsymbol{\psi}$	$= [\cdots \quad \psi_{i,j}(\tau) \quad \cdots]^\mathrm{T}$