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Characterisation and calculation of nonlinear vibrations in gas foil bearing systems–An experimental and numerical investigation

Robert Hoffmann *, Robert Liebich

Chair of Engineering Design and Product Reliability, Berlin Institute of Technology (TU Berlin), Str. d. 17. Juni 135, 10623 Berlin, Germany

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ABSTRACT

This paper states a unique classification to understand the source of the subharmonic vibrations of gas foil bearing (GFB) systems, which will experimentally and numerically tested. The classification is based on two cases, where an isolated system is assumed: **Case 1** considers a poorly balance rotor, which results in increased displacement during operation and interacts with the nonlinear progressive structure. It is comparable to a Duffing-Oscillator. In contrast, for **case 2** a well/perfectly balanced rotor is assumed. Hence, the only source of nonlinear subharmonic whirling results from the fluid film self-excitation. Experimental tests with different unbalance levels and GFB modifications confirm these assumptions.

Furthermore, simulations are able to predict the self-excitations and synchronous and subharmonic resonances of the experimental test. The numerical model is based on a linearised eigenvalue problem. The GFB system uses linearised stiffness and damping parameters by applying a perturbation method on the Reynolds Equation. The nonlinear bump structure is simplified by a link-spring model. It includes Coulomb friction effects inside the elastic corrugated structure and captures the interaction between single bumps.

1. Introduction

Gas foil bearings (GFBs) have successfully been introduced into small turbo machinery for more than 40 years, e.g. air cycle machines, turbo compressors, turbochargers and compressors of fuel cells. Major advantages of compliant foil bearings are low drag friction, high speed operation, high temperature endurability and the omission of an oil system, [1]. In a bump type gas foil bearing the elastic bearing wall comprises a bump and a top foil made of thin sheet metal. Both foils are fixed with the bearing sleeve, e.g. by spot welds. Due to the eccentrically rotating bearing journal a fluid dynamic pressure field $p(z, \theta)$ is generated in the aerodynamic wedge and deforms the elastic structure $h(z, \theta)$ and an optimal film thickness is achieved, see Fig. 1. Thus, higher load capacities compared to rigid gas bearings are generated, [2]. The deformation of the foils may activate sliding contacts inside the elastic structure delivers additional damping and improves the dynamic behaviour compared to rigid gas bearings. Nevertheless, the low viscosity of the air film results in an overall low damping level, which is still a key issue because the poor damping ability may result in nonlinear vibrations. Those can significantly affect the rotor dynamic perfor-

* Corresponding author.
E-mail addresses: robert.hoffmann@posteo.de (R. Hoffmann), robert.liebich@tu-berlin.de (R. Liebich).

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Nomenclature

Abbreviations

CG	Center of Gravity
BP	Balance Plane
FE	Finite Element
GFB	Gas Foil Bearing
ODE	Ordinary Differential Equation
OSSV	Onset Speed of Subharmonic Vibration
RE	Reynolds Equation

Latin

b_s	shim width
c	radial bearing clearance
c_0	nominal radial bearing clearance
c_t	Petrov-model parameter
c_d	structural damping
c_{ij}	linearised damping value $i, j = x, y$
c_t	slope parameter Petrov-Model
e_i	journal displacement $i = x, y$
e_0	zero order journal displacement
Δe_i	perturbed journal displacement $i = x, y$
f	frequency
f_i	eigenfrequency for mode $i = 1, 2 \ldots n$
f_{OSSV}	onset speed of subharmonic vibration; frequency
Δe_i	perturbed journal displacement $i = x, y$
$\mathbf{f}_B$	bearing reaction force vector
$\mathbf{f}_f$	friction force vector
$\mathbf{f}_p$	pressure force vector
$\mathbf{f}_U$	unbalance force vector
h	film thickness
Δh	vertical displacement link-spring-model
$\Delta \hat{h}$	dynamic vertical displacement link-spring-model
h_b	bump height
h_r	rigid term of the film thickness
h_c	compliant term of the film thickness
h_0	zero order film thickness
h_i	perturbed film thickness $i = x, y$
$h_{c,i}$	perturbed compliant film thickness term $i = x, y$
h_o	operation film thickness
j	complex number $\mathrm{j} = \sqrt{-1}$
k_1	interaction spring stiffness link-spring-model
k_2	spring stiffness link-spring-model
k_d	dynamic structural stiffness
k_{eq}	equivalent bump stiffness ($k_{eq} = A_b/K$)
k_{ij}	linearised stiffness value $i,j = x, y$
k_t	Petrov-model stiffness parameter
k_s	static structural stiffness
l	bearing length
l_b	half bump length
l_r	shaft length
l_s	shim length
$\Delta l_{CG,i}$	distance from CG to bearing midline $i = F, R$
m_r	rotor mass
n_{OSSV}	onset speed of subharmonic vibration; rotor speed
p	pressure
p_a	ambient pressure
p_0	zero order pressure
p_i	perturbed pressure $i = x, y$
r	rotor displacement (magnitude)
s_b	bump pitch
Δs	shim thickness
t	time
t_b	bump thickness
t_f	top foil thickness
$\mathbf{u}$	displacement vector
$\mathbf{w}$	loading vector $\mathbf{w} = \{W_x, W_y\}^T$
x	Cartesian coordinate
x_s	displacement Petrov-model
x_{bot}, x_{up}	horizontal displacements link-spring model; *bottom* and *up*
$\mathbf{u}$	system vector; dynamic structural model
y	Cartesian coordinate
z	Cartesian coordinate
A_b	Bump surface ($A_b = s_b l$)
$\mathbf{A}$	system matrix dynamic link-spring-model
$\mathbf{C}$	damping matrix
$\mathbf{C}_B$	linearised GFB damping matrix
D_a	nominal shaft diameter
$\mathbf{G}$	gyroscopic matrix
E	Young's modulus
F	force
F_r, F_l	right and left normal force link-spring-model
F_{bot}, F_{up}	friction force link-spring-model
$F_{b,x}$	horizontal beam lever force link-spring-model
F_s	interaction force link-spring-model
F_x	horizontal reaction force
F_p	bump load
$\hat{F}_p$	dynamic bump load (amplitude)
J_i	moment of inertia $i = x, y, z$
$\mathbf{K}$	stiffness matrix
K	bump compliancy $K = \frac{2s_b}{E}\left(\frac{l_b}{t_b}\right)^3 (1 - \nu^2)$
$\mathbf{K}_B$	linearised GFB stiffness matrix
$\mathbf{K}_{Bump}$	bump stiffness matrix
ΔL	horizontal displacement link-spring-model
L'	modified horizontal displacement link-spring-model
$\mathbf{M}$	mass matrix
N	normal contact force
N_{bot}, N_{up}	normal contact force link-spring-model
N_b	bump number
N_S	shim number
R	bearing journal radius
R_b	bump radius
T_a	ambient temperature
U	journal rotational speed $U = R\Omega$
U_F, U_R	unbalance front (F) and rear (R)
W_i	bearing load $i = x, y$