Journal of Sound and Vibration 381 (2016) 179–191

Contents lists available at ScienceDirect

Journal of Sound and Vibration

journal homepage: www.elsevier.com/locate/jsvi

Stability of rigid rotors supported by air foil bearings: Comparison of two fundamental approaches

Jon S. Larsen *, Ilmar F. Santos, Sebastian von Osmanski

Department of Mechanical Engineering, Technical University of Denmark, 2800 Kgs., Lyngby, Denmark

A R T I C L E I N F O

Article history:
Received 24 January 2016
Received in revised form
3 May 2016
Accepted 16 June 2016
Available online 7 July 2016
Handling Editor: M.P. Cartmell

Keywords:
Air foil bearing
Rigid rotor
Transient simulation
Nonlinear analysis
Stability

A B S T R A C T

High speed direct drive motors enable the use of Air Foil Bearings (AFB) in a wide range of applications due to the elimination of gear forces. Unfortunately, AFB supported rotors are lightly damped, and an accurate prediction of their Onset Speed of Instability (OSI) is therefore important. This paper compares two fundamental methods for predicting the OSI. One is based on a nonlinear time domain simulation and another is based on a linearised frequency domain method and a perturbation of the Reynolds equation. Both methods are based on equivalent models and should predict similar results. Significant discrepancies are observed leading to the question, is the classical frequency domain method sufficiently accurate? The discrepancies and possible explanations are discussed in detail.

1. Introduction

For almost 150 years, bearing-influenced rotor vibrations have attracted the attention of engineers and researchers. In one of the first documents found in the literature, dated 1869, Rankine [1] reported the amplification of shaft lateral vibrations at different (critical) speeds. In 1894, Dunkerley [2] presented an approximate method for finding the natural frequencies or critical "whirling" speeds of simplified rotor systems. In 1919, Jeffcott [3] published one of the most significant contributions to rotor dynamics, in which the lateral vibration of loaded shafts in the neighbourhood of a whirling speed was addressed. With the significant developments of rotating machines in the 1920s, especially driven by steam turbines [4], the influence of fluid film bearings on the lateral shaft vibrations became a very important research topic. Motivated by the necessity of enlarging the operational range of steam turbines, the influence of higher angular velocities on the shaft lateral vibrations was investigated by Newkirk and Taylor [5] in 1925. They documented not only the increase of shaft lateral vibrations while crossing over the first critical speed but also, for the first time, the severe unexpected vibrations about twice the first critical speed. In 1927, with an iterative approach based on assumed shaft mode shapes and the new concept of "nonlinear oil springs", Stodola [6] linked rotor lateral vibrations, bearing stiffness and their dependence on journal eccentricity. Using a rigid rotor, Newkirk [7] experimentally proved in 1931 that half-speed whirl could occur over a wide speed range, well below twice the first critical speed. Using a simplified infinite-width bearing model and disregarding cavitation, Swift [8] analytically investigated the non-steady conditions in journal bearings. Taking into account bearing as well as rotor flexibility, Smith [9] reported the possibility of four critical speeds for non-symmetrical rotors, a remarkable achievement in the beginning of the 1930s, an era characterised by the absence of digital computer power.

* Corresponding author.
E-mail addresses: josla@mek.dtu.dk (J.S. Larsen), ifs@mek.dtu.dk (I.F. Santos), sebastian@osmanski.dk (S. von Osmanski).

http://dx.doi.org/10.1016/j.jsv.2016.06.022
0022-460X/

Nomenclature

$\mathbf{a}$	Nodal vector
a	Scaler field quantity
A, B	bearings
$\mathbf{A}$	state space matrix
b, $\tilde{b}$	structural damping per unit area, $\tilde{b} = C/(p_a\omega)b$
$\mathbf{B}$	shape function derivatives matrix
C	radial clearance
D, $\tilde{D}$	bearing damping, $\tilde{D} = \omega C/(p_a R^2)D$
$\mathbf{D}$, $\tilde{\mathbf{D}}$	damping matrix, $\tilde{\mathbf{D}} = \omega C/(p_a R^2)\mathbf{D}$
$\tilde{\mathbf{D}}_s$	first-order coefficients, $\tilde{\mathbf{D}}_s = \tilde{\mathbf{D}} - \tilde{\mathbf{G}}$
e, ε	journal eccentricity (or e = element), $\varepsilon = e/C$
E	modulus of elasticity of foil
f_γ	direction cosines
$\mathbf{f}, \tilde{\mathbf{f}}$	bearing force vector, $\tilde{\mathbf{f}} = 1/(p_a R^2)\mathbf{f}$
$F, \tilde{F}$	bearing force components, $\tilde{F} = 1/(p_a R^2)F$
$\mathbf{g}()$	nonlinear vector function
$\mathbf{G}$, $\tilde{\mathbf{G}}$	gyroscopic matrix, $\tilde{\mathbf{G}} = \omega^2 C/(p_a R^2)\mathbf{G}$
h, $\tilde{h}$	film height, $\tilde{h} = h/C$
h_b	bump foil height
$h_c, \tilde{h}_c$	film height correction, $\tilde{h}_c = h_c/C$
h_r, $\tilde{h}_r$	film height (rigid), $\tilde{h}_r = h_r/C$
$h_s, \tilde{h}_s$	slope height, $\tilde{h}_s = h_s/C$
$\tilde{\mathbf{h}}_c$	foil deformation vector
i	Pad, i=1,2,..., Np
I	mass moment of inertia
$\mathbf{I}$	identity matrix
k, $\tilde{k}$	structural stiffness per unit area, $\tilde{k} = C/p_a k$
K, $\tilde{K}$	bearing stiffness, $\tilde{K} = C/(p_a R^2)K$
$\mathbf{K}$, $\tilde{\mathbf{K}}$	stiffness matrix, $\tilde{\mathbf{K}} = C/(p_a R^2)\mathbf{K}$
l_0	bump half length
l_1,l_2	distance to bearings
L, $\tilde{L}$	bearing length, $\tilde{L} = L/R$
μ_f	Foil friction coefficient
m	mass
$\mathbf{M}$, $\tilde{\mathbf{M}}$	mass matrix, $\tilde{\mathbf{M}} = \omega^2 C/(p_a R^2)\mathbf{M}$
N	Number of fluid film nodes
N_p	number of pads
$\mathbf{N}$	shape function matrix
$\tilde{\mathbf{p}}_m$	Arithmetic mean pressure vector
p, $\tilde{p}$	film pressure, $\tilde{p} = p/p_a$
p_a	ambient pressure
p_γ, $\tilde{p}_\gamma$	perturbed pressure, $\tilde{p}_\gamma = Cp_\gamma/p_a$
$\tilde{\mathbf{p}}$	pressure vector
$\tilde{q}_c$	Structural flexibility complex form
q, $\tilde{q}$	structural flexibility per unit area, $\tilde{q} = p_a q/C = 1/\tilde{k}$
$\mathbf{r}$	residual vector
R	journal radius
$\mathbf{s}$	advection vector, $\mathbf{s} = \{S, 0\}^T$
S	compressibility number, $S = 6\mu\omega/p_a(R/C)^2$
S_b	bump foil pitch
t	time
t_b	thickness of bump foil
t_t	thickness of top foil
$x, y, z, \tilde{z}$	Cartesian coordinates, $\tilde{z} = z/R$
$\mathbf{y}$	state vector
$\mathbf{w}, \tilde{\mathbf{w}}$	load vector, $\tilde{\mathbf{w}} = 1/(p_a R^2)\mathbf{w}$
$W, \tilde{W}$	static load components, $\tilde{W} = 1/(p_a R^2)W$
$\mathbf{z}$	rotor state vector
$\mathbf{z}_1$	rotor displacement vector, $\mathbf{z}_1 = \boldsymbol{\varepsilon}$
$\mathbf{z}_2$	rotor velocity vector, $\mathbf{z}_2 = \dot{\boldsymbol{\varepsilon}}$
α	bearing position
$\boldsymbol{\Gamma}$	fluidity matrix
$\Delta\varepsilon$	perturbation amplitude
$\boldsymbol{\varepsilon}$	eccentricity vector
η	structural loss factor of foils
θ, $\tilde{\theta}$	circumferential angle, $\tilde{\theta} = \theta R$
θ_l	first pad leading edge angle
θ_s	first pad slope extend
θ_t	first pad trailing edge angle
λ	complex eigenvalue, $\lambda = \tilde{\omega}_n i + \tilde{\beta}$
μ	dynamic viscosity
ν	Poisson's ratio of foil
ξ_i, η_j	gauss points
τ	dimensionless time, $\tau = \omega t$
Φ	fluid domain
ψ	film state variable, $\psi = ph$
$\boldsymbol{\psi}$	film state vector
ω	angular speed of journal
ω_n, $\tilde{\omega}_n$	eigenfrequency, $\tilde{\omega}_n = \omega_n/\omega$
ω_s, $\tilde{\omega}_s$	excitation frequency, $\tilde{\omega}_s = \omega_s/\omega$
γ	Coordinates x, y
$\mathbf{0}$	zero matrix
CG	Center of Gravity
FE	Finite Element
LD	Logarithmic Decrement, LD $= -2\pi\,\mathrm{Re}(\lambda_i)\mathrm{Im}(\lambda_i)$
ODE	Ordinary Differential Equation
OSI	Onset Speed of Instability
PDE	Partial Differential Equation
$\nabla\cdot$	divergence
∇	gradient, $\nabla = \{\partial/\partial\theta, \partial/\partial\tilde{z}\}$
$\ddot{()}$	time derivative, $\mathrm{d}^2/\mathrm{d}\tau^2$
$\dot{()}$	time derivative, $\mathrm{d}/\mathrm{d}\tau$

The development of digital computers since the 1940s has strongly facilitated researchers to evaluate more complex and sophisticated mathematical models for describing rotor and bearing dynamics. It has encouraged the development of new alternative procedures to predict bearing-influenced rotor dynamic response and stability [10–13] in the 1950s and 1960s. From the viewpoint of solid mechanics, rotors with complex geometry could be refined using the transfer matrix method [14,15] and later on the finite element (FE) method [16] to account for distributed rotor mass, inertia, and gyroscopic effect. From the viewpoint of fluid film bearing dynamics, effects of fluid compressibility [17–19] and inertia [20–22], cavitation [23] and turbulence [24] could be numerically investigated and more accurately predicted. For example, lubricant compressibility strongly influences the static and dynamic behaviour of gas bearings as already reported by Harrison in 1913