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Research Paper

Dynamic study of viscoelastic rotor: Modal analysis of higher order model considering various asymmetries

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A R T I C L E I N F O

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A B S T R A C T

The primary aim of this paper is to study modal analysis of higher order finite element model for a generally viscoelastic rotor supported by the journal bearing. The gyroscopic couple, internal damping and fluid film forces of journal bearings have a significant asymmetric influence on rotor dynamics. A complex modal coordinate creates a platform to indicate the directivity of modes and provides better information about the direction of the whirl. In which, study of natural modes and directional frequency response function are obtained for free and force vibration analysis. It also distinguishes the importance of higher order model over conventional second order model.

1. Preamble and motivation

The rotating machine is a mechanical device, that is used in many applications such as engines, electrical generators, hydraulic turbines, pumps, compressors, etc., well established in Vance [1], Lee [2] and Rao [3]. Since all dynamic behaviour of the rotor is interlinked with rotation Lee [4], the concept of directivity becomes crucial in rotor dynamics. Thus, modal analysis of such rotating system is an essential tool to get an insight view of dynamic behaviour. This paper employs the complex modal analysis for the generally viscoelastic rotor system, where complex refers to complex variable representation concerning either excitation or response [5] and Mesquita et al. [6]).

Modal analysis is a numerical technique to determine the modal parameter of a system. Mechanical models only describe the dynamic behaviour of the system and they are constrained by some assumptions and boundary conditions. Most of the vibration analyses in rotor-dynamics are using a traditional concept that is based on natural modes, natural frequencies, and critical speed, reported in many works of literature like Nakra [7], Genta [8] and Friswell et al. [9]. The major distinction comes into picture when the vibration analysis is done on a rotating system. Due to the existence of asymmetry, which arises from gyroscopic and circulatory forces, the rotor shaft system is non-self-adjoint nature. The equation of motion of such non-self-adjoint system is solved by after transforming it into state space form. Two kinds (left and right) eigenvectors are obtained through eigensolution, which help to proceed with modal analysis [10]. Firstly, it was notified by Lee [4] after considering asymmetric system due to the gyroscopic stiffening effect and explained the backward, forward whirling. Following the same theory Jei and Kim [11] started the modal analysis of a rigid rotor supported by a flexible bearing using a classical model with two displacements and two rotational degrees of freedom. Due to the conjugate even property of traditional coordinate directivity of modes is lost while it is protected under complex coordinate. Modal parameters like modal frequency, modal damping, mode shape and direction of whirl were well explained with the use of complex modal analysis by Joh and Lee [12], Kessler [13]. Mesquita et al. [6] did a comparative study between traditional frequency response function and directional frequency response function (dFRF) of the rotor shaft system. Kessler and Kim [14,15]

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Nomenclature

a, b	Coefficients of material property
e	Exponential
i	Iota (imaginary unit)
l	Length of element
m	Unbalance mass
$\{\mathbf{q}\}$	Total degrees of freedom
v, w	Mechanical displacement along y and z axis respectively
r	Radius of rotor
$\{\mathbf{u}\}$	Excitation force
t	Time in second
A	Cross sectional area
$[\mathbf{A}]$	System state matrix
$[\mathbf{B}]$	Input matrix
$C_{b_{yy}}$, $C_{b_{zz}}$	Direct damping coefficient of bearing
$C_{b_{yz}}$, $C_{b_{zy}}$	Cross coupled damping coefficients of bearing
D	Diameter
E	Modulus of elasticity
$[\mathbf{G}]$	Gyroscopic matrix
$[\mathbf{H}]$	Frequency response function matrix
I	Area moment of inertia
$\mathbf{I}$	Identity matrix
$[\mathbf{K}]$	Stiffness matrix
$K_{b_{yy}}$, $K_{b_{zz}}$	Direct stiffness coefficient of bearing
$K_{b_{yz}}$, $K_{b_{zy}}$	Cross coupled stiffness coefficients of bearing
L	Length of rotor
M_{yy}, M_{zz}	Bending moment about y-axis and z-axis respectively
$[\mathbf{M}]$	Mass matrix
$\{\mathbf{P}\}$	External nodal force vector
R	Deformation of the rotor center line
$\{\mathbf{x}\}$	State vector
σ	Mechanical stress
ε	Mechanical strain
φ, ϑ	Rotation about y and z- axis respectively
ρ	Mass density
$[\mathbf{\Lambda}]$	Eigenvalues
$[\mathbf{\Psi}]$	Right eigenvectors
$[\mathbf{\Phi}]$	Left eigenvectors
ω	Whirl speed
Ω	Spin speed
θ	Phase angle

Subscript

brg	Bearing
d	Disc
i, j, k	Indices
p, g	Complex displacement and force rotating in same direction of the shaft
$\bar{p}$, $\bar{g}$	Complex displacement and force rotating in opposite direction of the shaft
n	Total number of degrees of freedom
r	Rotor
x, y, z	Conventional coordinate axis

Superscript

T	Transpose

Abbreviation

FNF	First natural frequency
dFRF	Directional Frequency Response Function
r−dFRF	Reverse directional Frequency Response Function
SLS	Stability limit of spin speed
SOM	Second Order Model
SWL	Synchronous Whirl Line
TOM	Third Order Model

Operators

()	Operator
(•)	Differentiation with respect to time
(−)	Conjugate
(*)	Non dimensional term
(ˆ)	Assumed quantity

represented general planer motion and forces as the linear superposition of forward and backward rotating vectors. dFRF was interpreted as a composition of forward and backward component of complex displacement and excitation function.

Most of the researchers considered asymmetry due to gyroscopic effect and studied the modal behaviour. But another source of asymmetry is the internal friction of the rotor. A speed dependent tangential force is generated by this internal friction and acts towards the whirl direction. Many researchers like Tondl [16], Zorzi and Nelson [17], either they have considered a hysteretic or a viscous form of internal damping and included a skew-symmetric circulatory stiffness matrix with the bending stiffness matrix. Dutt and Roy [18] applied an operator based constitutive relationship to obtain the higher order equations of motion of a generally viscoelastic rotor-shaft continuum after discretizing it with finite Timoshenko beam elements. The order of differential equation depends on the inherent material behaviour. Later, Roy and Dutt [19] and Roy et al. [20] again utilized the same concept to develop the finite element model of the viscoelastic composite rotor. Chouksey et al. [21] obtained mode shape and dFRF for a flexible damped rotor shaft supported by antifriction (ball or roller) bearings. Shaft material damping was included through two elements Voigt model.

Journal bearing is frequently used to support heavy rotors in the power plant industry, railway car and others. The fluid film forces depend on many parameters like clearance, lubricant viscosity and spin speed. Detail derivation of these forces in terms of four stiffness and four damping coefficients are available in Rao [3], Friswell et al. [9]. It also provides asymmetry to the system model and has a significant effect on dynamics. Though these hydrodynamic forces are nonlinear functions of displacement and velocity of the journal on its bearing housing, but sometimes linear forcing function is assumed for simplicity [22]. A negatively cross couple term exists in the forcing function. Thus, it also provides a tangential force in the direction of the whirl and proportional to spin speed. This tangential force is dissipative in nature and has a destabilizing effect. After certain spin speed, this tangential force is