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An identification method for damping ratio in rotor systems

Weimin Wang [a], Qihang Li [a,*], Jinji Gao [a], Jianfei Yao [a], Paul Allaire [b]

[a] *Beijing Key Laboratory of Health Monitoring and Self-recovery for High-end Mechanical Equipment, Beijing University of Chemical Technology, Beijing 100029, PR China*
[b] *Rotor Bearing Solutions International, Charlottesville, VA 22901, USA*

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ABSTRACT

Centrifugal compressor testing with magnetic bearing excitations is the last step to assure the compressor rotordynamic stability in the designed operating conditions. To meet the challenges of stability evaluation, a new method combining the rational polynomials method (RPM) with the weighted instrumental variables (WIV) estimator to fit the directional frequency response function (dFRF) is presented. Numerical simulation results show that the method suggested in this paper can identify the damping ratio of the first forward and backward modes with high accuracy, even in a severe noise environment. Experimental tests were conducted to study the effect of different bearing configurations on the stability of rotor. Furthermore, two example centrifugal compressors (a nine-stage straight-through and a six-stage back-to-back) were employed to verify the feasibility of identification method in industrial configurations as well.

1. Introduction

The purpose of a compressor stability evaluation is to identify the rotor modal parameters in either the shop testing of turbo compressors or in field installations. The critical point is to make sure the compressor works stably in the designed operating conditions. The ideal identification method should: (1) eliminate or reduce the effects of mode overlap; (2) have the capability of working in a high noise environment; and (3) be able to work with a relatively low vibration level in the process of shop testing or field use. To try and achieve these goals, many researchers have reported a huge number of works considering both frequency and time domain approaches.

In the frequency domain, Lee [1] transformed the traditional frequency response function (FRF) from the real domain into the complex domain, forming the directional frequency response function (dFRF) with a set of complex numbers defining of vibration displacements and exciting forces. This method is able to separate the rotor forward and backward modes. Kessler [2] exported a similar expression of the dFRF from stability tests, as Lee [1] did. This approach used the forward and backward exciting forces as the inputs and the forward and backward vibration displacement response as the

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Abbreviations: AMBs, Active Magnetic Bearings; ARMA, Auto Regressive Moving Average; AVF, amplitude-versus-frequency; AVP, amplitude-versus-phase; BAR, backward autoregressive; dFRF, directional frequency response function; DOF, degrees of freedom; FE, finite element; FRF, frequency response function; LBP, Load Between Pads; LOP, Load On Pads; PEM, Prediction Error Method; MIMO, Multiple Input Multiple Output; MOBAR, Multiple Output Backward Autoregression; NSR, Noise to Signal Ratio; SISO, Single Input Single Output; OLS, Ordinary Least Square; RPM, Rational Polynomial Method; SNR, Signal to Noise Ratio; SVD, singular value decomposition; TEHD, thermo-elastic-hydrodynamic; IV, Instrumental Variable; WIV, Weighted Instrumental Variable

[*] Corresponding author.

E-mail address: liqihang2012@foxmail.com (Q. Li).

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Nomenclature

$\boldsymbol{a}$	numerator in rational polynomials
$\boldsymbol{b}$	denominator in rational polynomials
$\boldsymbol{C}$	damping matrix
c	complex number
c_{ij}	bearing damping
$\boldsymbol{e}$	error vector
$\mathbf{f}$	external forces applying on the rotor
$\mathbf{G}$	gyroscopic matrix
$\boldsymbol{G}$	directional FRF matrix
G_d	cross damping
$\boldsymbol{H}$	FRF matrix
$\boldsymbol{H}'$	measured frequency response function matrix
H_{mn}	frequency response in the m direction due to an excitation in the n direction
$\mathbf{J}$	constant transformation matrix
j	$\sqrt{-1}$
$\boldsymbol{K}$	stiffness matrix
$\boldsymbol{K}_c$	stiffness matrix caused by gyroscopic effect
K	weighting value matrix
k	weighting function value
k_{ij}	bearing stiffness
$\boldsymbol{M}$	mass matrix
m_s	shaft mass
$\boldsymbol{p}$	parameter vector
q	cross stiffness
$\boldsymbol{q}$	vector
$\mathbf{q}, \dot{\mathbf{q}}, \ddot{\mathbf{q}}$	rotor displacement, velocity, and acceleration
$\boldsymbol{R}$	complex displacement vector
S_N	number of support points
s	Laplace frequency point
$\boldsymbol{u}$	modal vector
$\boldsymbol{v}$	adjoint modal vector
$\boldsymbol{W}$	instrumental variable matrix
x	rotor horizontal displacement degree of freedom; modal displacement
y	rotor vertical displacement degree of freedom; modal displacement

Greek symbols

α	rotor rotation about x-axis
β	rotor rotation about y-axis
λ	eigenvalue
ω	angular frequency
Ω	shaft rotational speed
χ_c	parameter to adjust the level of horizontal to vertical anisotropy of damping
χ_k	parameter to adjust the level of horizontal to vertical anisotropy of stiffness

Subscripts

B	backward precession
b	bearing terms, finite element model
c	complex number symbol
F	forward precession
k	k-th measured point
m	lumped mass terms, finite element model
R	complex domain
s	shaft terms, finite element model

outputs of the rotor system respectively. They provided a major contribution to the mode separation in their complex modal analysis. However, they did not compare the modal parameters with the FRF and dFRF, and did not study how to improve the accuracy of the identified modal parameters. Takahashi et al. [3] used the Prediction Error Method (PEM) to fit the dFRF stability measurements for a practical centrifugal compressor with small vertical /horizontal stiffness anisotropies of bearings and seals in shop testing. The paper shows that the forward and backward modes can be separated well in the positive and negative frequency axis sides and the single input signal output (SISO) dFRF identification method was better suitable. However, they did not study the applicability of the identification method to further applications.

Later, Chouksey et al. [4] adapted the dFRF method presented by Lee [1] to estimate the modal parameters and investigate the influence of rotor–shaft material damping on the rotor–shaft system stability. Agneni et al. [5] applied Hilbert transforms to obtain FRFs with white-noise perturbations. The output signals were measured and then singular value decomposition (SVD) was used in the frequency domain to estimate the modal parameters of the rotor system. This approach was verified by the comparison with other well-established estimation methods.

In the time domain, Kumaresan and Tufts [6] developed the backward autoregressive (BAR) technique that was combined SVD to handle the noise effects, and can accurately identify the damping of close sister modes. Using a similar approach, Cloud et al. [7] developed the Multiple Output Backward Autoregressive (MOBAR) method and carried out significant work on the rotor stability investigation. This is a method of system damping using the output (measured rotor displacements) only method. Excellent results were developed. However, one weakness of the MOBAR approach is that the excitation method acquires 3–4 times the normal vibration amplitude due to the rotor unbalance. This high level is necessary to improve the signal to noise ratio (SNR) for modal parameter identification [7]. Furthermore, compared to the identification process in frequency, the decay process with impulse or step excitation limits the total sampling number of signals, and less useful data points must decrease the accuracy of identified results. Later Pettinato et al. [8] investigated shop acceptance testing of compressor rotordynamic stability and the theoretical correlation. The results indicate that the estimated results between the MOBAR and PEM method are in very close agreement. Finally, this BAR method does not employ the input data, such as the excitation force from the magnetic bearing exciter, which allows for an input/output measurement approach.

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