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A component damping identification method for mistuned blisks

D.E. Holland [a,*], B.I. Epureanu [b,**]

[a] Department of Mechanical Engineering, Cedarville University, 251 N. Main St., Cedarville, OH, 45314, USA
[b] Department of Mechanical Engineering, University of Michigan, 2350 Hayward St., Ann Arbor, MI, 48109, USA

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ABSTRACT

Different components of a structure can have different damping characteristics, and that affects the dynamic response of the overall system. Herein, we assume that damping has a simple form only at a component level. That leads to a complex damping model at a system level. A new method is introduced which identifies the component damping of a structure. This method is applied to mistuned blisks in regions of low and high modal density. The method incorporates reduced-order models, and remains accurate in the presence of measurement noise. Results are shown for a mistuned blisk with varying levels of measurement noise. The accuracy of damping identification is observed through a forced response prediction and amplification factor study. Also, a discussion on the effects of damping and stiffness mistuning on the maximum response is presented. Some differences between component and modal damping are highlighted.

1. Introduction

Wear or manufacturing processes or defects can cause slight variations in the mass or stiffness of a nominally cyclically symmetric structure such as an integrally bladed disk (blisk). These variations (referred to as mistuning) can cause the vibration energy to localize at certain regions of the structure resulting in larger than expected forced responses and stresses. Therefore, these mistuned systems are susceptible to high cycle fatigue. One method to compensate for the effects of mistuning is to apply damping coatings. However, Joshi et al. [1] found that the dynamics of the system can significantly change due to variations in the thickness of the applied coating. To determine the damping characteristics of coatings, a damping identification technique is needed which can determine the damping associated with individual components of a mistuned blisk (before and after the coating is applied). Currently, damping identification relies on one of the several common damping models, namely structural, viscous, and component (material) damping. In general, structural damping is defined for a full system, while viscous damping is defined for individual system modes.

Most current damping identification techniques assume that damping has a certain form at a system level. For example, the damping in complex structures is often assumed to be viscous, modal, or structural. Such assumptions provide accurate results for structures with relatively simple geometries and low modal density. However, these approximations can be cumbersome or inaccurate for structures with complex geometry and high modal density. For such systems, using a component-oriented model can be more effective. In particular, component damping may correspond to a proportional or

* Principal corresponding author. Tel.: +1 7176497614.
** Corresponding author.
E-mail addresses: deholla@umich.edu (D.E. Holland), epureanu@umich.edu (B.I. Epureanu).

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Nomenclature

α_{uv}	relative forcing coefficient for frequency pair u and v
$\overline{\mathbf{\Phi}}$	matrix of the mistuned modes
$\overline{\mathbf{\Psi}}$	matrix of the disassembled mistuned system modes
$\overline{\mathbf{p}}$	vector of the mistuned modal amplitudes
$\overline{b}$	reduced number of frequency pairs
$\overline{S}^{c_i}$	maximum sensitivity for component i
$\overline{S}^{c_i}_{uv}$	sensitivity of the target mistuned mode
β	blade portion of sector
$\delta\Gamma^{c_i}$	damping perturbation of component i
$\delta\boldsymbol{\kappa}^{c_i}$	stiffness perturbation matrix for component i
Δ	disk portion of sector
Γ^m	mean component damping
ω	excitation frequency
$\boldsymbol{\kappa}^{c_i}$	stiffness matrix for component i obtained using cyclic symmetry
$\mathbf{\Lambda}^i_{CB}$	diagonal matrix of the mistuned cantilevered blade eigenvalues for blade i
$\mathbf{\Phi}_{c_i,u}$	vectors containing the portion of the uth mistuned mode shapes corresponding to component c_i
$\boldsymbol{\sigma}$	Lagrange multipliers
$\mathbf{E}$	constraint matrix
$\mathbf{F}$	non-conservative generalized force vector
$\mathbf{I}$	identity matrix
$\mathbf{q}^{c_i}$	generalized coordinates
$\mathbf{Q}_{c_i}$	participation factors of mistuned cantilevered blade i onto the tuned system modes
$\mathbf{S}$	synthesis matrix
$\mathbf{X}$	independent generalized coordinates
$\mathbf{x}$	harmonic response vector
$\tilde{\boldsymbol{\eta}}^{c_i}$	damping matrix for component i
$\tilde{\boldsymbol{\kappa}}^{c_i}$	stiffness matrix for component i
$\tilde{\boldsymbol{\mu}}^{c_i}$	mass matrix for component i
$_{aug}$	augmented matrix
$_{n,u}$	mistuned natural frequencies of modes u
$_{u}$	frequency u
a	number of independent DOFs
b	number of frequency pairs
c_i	component i
D	Rayleigh damping function
e	combined number of independent and dependent DOFs for a given component
h	number of candidate vectors
N	number of disassembled DOFs
n	number of components
s	number of sectors
T	kinetic energy
T_m	dependence threshold
V	potential energy

structural damping applied to individual components of a structure (i.e., each component has an associated material damping). For instance, each blade of a blisk can be modeled as a separate component with an associated damping parameter (which is to be identified). The resulting damping is only approximately modal, and the corresponding modal damping values vary with different component damping properties. As a result, the component damping can represent the physical attributes of the structure more closely and may require less effort than identifying the modal damping for a structure with a moderate number of components and a large number of modes in the frequency range of interest.

While finite element modeling (FEM) methods incorporate component damping, most identification methods found in literature are difficult or impossible to use to identify this type of damping from experimental data[2–20]. Statistical energy analysis (SEA) is one method for determining component (subsystem) damping loss factors [21–27]. This method is useful for high frequency ranges. However, this technique is limited as the damping is assumed to be known either from the power injection method (PIM) [21–24], power modulation method [24], or a wave approach [25]. PIM and power modulation techniques require measuring energy for all the components, and that can be difficult or impossible to gather accurately. Also, wave theory applies to periodic systems, but mistuning destroys periodicity. Moreover, the SEA method can be time consuming if there are many components [24]. In addition, the accuracy of the SEA method depends on the modal density and the level of damping, as shown by Mace [27] and Yap et al. [26].

The work herein presents a novel component damping identification technique which can be applied to mistuned blisks and uses certain reduced-order models (ROMs) in regions of low and/or high modal density. In addition, a method for predicting the forced response of the system using ROMs is provided, and that can be used to decrease design time and enable statistical analyses of component damping scenarios.

2. Equations of motion

In this section, equations of motion are developed beginning with Lagrange's equation (closely following [28]). Component level damping is introduced to the disassembled system by including a Rayleigh damping function. Finally, the system-level structural equation of motion incorporating component damping is obtained.

The kinetic and potential energies and the Rayleigh damping function for a structure can be expressed as

$$
\begin{aligned}
T &= \tfrac{1}{2}\dot{\mathbf{q}}^{c_1^T}\tilde{\boldsymbol{\mu}}^{c_1}\dot{\mathbf{q}}^{c_1} + \tfrac{1}{2}\dot{\mathbf{q}}^{c_2^T}\tilde{\boldsymbol{\mu}}^{c_2}\dot{\mathbf{q}}^{c_2} + \cdots + \tfrac{1}{2}\dot{\mathbf{q}}^{c_n^T}\tilde{\boldsymbol{\mu}}^{c_n}\dot{\mathbf{q}}^{c_n}, \\
V &= \tfrac{1}{2}\mathbf{q}^{c_1^T}\tilde{\boldsymbol{\kappa}}^{c_1}\mathbf{q}^{c_1} + \tfrac{1}{2}\mathbf{q}^{c_2^T}\tilde{\boldsymbol{\kappa}}^{c_2}\mathbf{q}^{c_2} + \cdots + \tfrac{1}{2}\mathbf{q}^{c_n^T}\tilde{\boldsymbol{\kappa}}^{c_n}\mathbf{q}^{c_n}, \\
D &= \tfrac{1}{2}\dot{\mathbf{q}}^{c_1^T}\tilde{\boldsymbol{\eta}}^{c_1}\dot{\mathbf{q}}^{c_1} + \tfrac{1}{2}\dot{\mathbf{q}}^{c_2^T}\tilde{\boldsymbol{\eta}}^{c_2}\dot{\mathbf{q}}^{c_2} + \cdots + \tfrac{1}{2}\dot{\mathbf{q}}^{c_n^T}\tilde{\boldsymbol{\eta}}^{c_n}\dot{\mathbf{q}}^{c_n},
\end{aligned}
\tag{1}
$$