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Vibration response comparison of twisted shrouded blades using different impact models

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ABSTRACT

On the basis of our previous work (Ma et al., 2016, Journal of Sound and Vibration, 378, 92–108) [36], an improved analytical model (IAM) of a rotating twisted shrouded blade with stagger angle simulated by flexible beam with a tip-mass is established based on Timoshenko beam theory, whose effectiveness is verified using finite element (FE) method. The effects of different parameters such as shroud gaps, contact stiffness, stagger angles and twist angels on the vibration responses of the shrouded blades are analyzed using two different impact models where the adjacent two shrouded blades are simulated by massless springs in impact model 1 (IM1) and those are simulated by Timoshenko beam in impact model 2 (IM2). The results indicate that two impact models agree well under some cases such as big shroud gaps and small contact stiffness due to the small vibration effects of adjacent blades, but not vice versa under the condition of small shroud gaps and big contact stiffness. As for IM2, the resonance appears because the limitation of the adjacent blades is weakened due to their inertia effects, however, the resonance does not appear because of the strong limitation of the springs used to simulate adjacent blades for IM1. With the increase of stagger angles and twist angles, the first-order resonance rotational speed increases due to the increase of the dynamic stiffness under no-impact condition, and the rotational speeds of starting impact and ending impact rise under the impact condition.

1. Introduction

Impacts with clearances exist widely in many mechanical structures with different applications. For instance, the shrouded blades with clearances are designed for the vibration reduction and avoiding the fatigue failure during the operation of the blade, because the vibration energy can be dissipated due to impacts and frictions between adjacent shrouded blades [1]. Consequently, the vibro-impact systems with clearances are of particular interest to many researchers, and fruitful results have been published in recent years. Many dynamic models of the rotating blades such as beam models [2–13] and plate models [14–16] and impact models such as the lumped mass models [17–29], beam-to-stop models [30–33] and beam-to-beam models [34] are established to describe the vibration characteristics of vibro-impact systems.

Many modeling methods for the rotating blades have been proposed by considerable researchers [2–16]. Bazoune reviewed the modeling methods and dynamic behaviors of rotational uniform beams [2]. As the research continued, a lot of

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Nomenclature

$\mathbf{A}_1$	transformation matrix from local blade coordinate to rotating coordinate
$\mathbf{A}_2$	transformation matrix from rotating coordinate to global coordinate
A	cross-sectional area of the blade
b	the width of the blade cross-section
$\mathbf{D}$	Rayleigh damping matrix
E	Young's modulus
$\mathbf{F}$	canonical external force vector without impacts
$\mathbf{F}^*$	canonical external force vector with impacts
F_e	uniformly distributed aerodynamic force per unit length
F_t	the impact force
$f_c(x)$	centrifugal force of the shrouded blade
$\mathbf{G}$	Coriolis force matrix
G	shear modulus
h	the thickness of the blade cross-section
I_y, I_z	area moment of inertias about y and z axes of the blade section
$\mathbf{K}_e$, $\mathbf{K}_c$, $\mathbf{K}_s$	structural stiffness matrix, centrifugal stiffening matrix, spin softening matrix
$\mathbf{K}_{acc}$	stiffness matrix caused by angular acceleration
k	contact stiffness
L	blade length
$\mathbf{M}$	mass matrix
m_s	shroud mass
N	the number of modal truncation
$\mathbf{q}$	canonical coordinates vector of the blade
R_d	the radius of disk
$\mathbf{r}_P$	the displacement vectors of an arbitrary point P of the blade
$\mathbf{r}_Q$	the displacement vectors tip-mass point Q
T_{total}, T_1, T_2	total kinetic energy of shrouded blade, kinetic energy of blade and tip-mass
u, u_L	radial displacements of arbitrary point P and tip-mass Q
V	the total potential energy of twisted shrouded blade
v, v_L	flexural displacements of arbitrary point P and tip-mass Q
v_{L1}, v_{L2}	tip-mass flexural displacement of passive blade 1 and passive blade 2
W'_{non}, W_{non}	work done by external force with/without considering impact force
w, w_L	swing displacements of arbitrary point P and tip-mass Q

Abbreviations

AM	analytical model
FE	finite element
FEM1, FEM2	finite element model 1, finite element model 2
IAM	improved analytical model
PB1, PB2	passive blade 1, passive blade 2
RMS	root mean square
TAM	traditional analytical model

Greek symbols

ρ	the density of the blade
θ	the rotational angle of the disk
κ	shear correction factor
β_0, β'	stagger angle and twist angle of shrouded blade
$\beta(x)$	the twist angle of an arbitrary point P of the blade
Ω	the rotational speed (rev/min)
ϕ, φ	rotational angle of the section of the blade around v and w axes
ω	the angular velocity of the blade
$\phi_{1i}(x)$, $\phi_{2i}(x)$, $\phi_{3i}(x)$	the ith modal shape functions in radial, flexural/swing and rotational angle directions
β_i, λ_i	eigenvalues of the ith modal shape function $\phi_{1i}(x)$ and $\phi_{2i}(x)$
Δ	shroud gap
Δ_1, Δ_2	shroud gap between passive blade 1 and active blade, passive blade 2 and active blade

attention was paid to the modeling of tapered blade and twisted blade on the basis of beam theory [3–13]. Based on Timoshenko beam theory and considering the effects of rotational speed, the dynamic equations of motion of a doubled-tapered blade with stagger angle were derived through Hamilton's principle [3], and the results demonstrated that the stagger angles have a significant effect on the natural characteristics for transverse modes. A twisted Timoshenko beam was modeled using Hamilton's principle [4], and a conclusion that the shear deformation has similar influences on the natural frequencies of twisted beam and untwisted beam was given. A coupled bending-bending vibration model of a twisted and tapered beam was established [5], and the parametric instability of the beam with non-constant rotational speed was analyzed by the multiple scales method. An analytical model (AM) of a twisted rotational blade subjected to contact and frictional loads was established to study the combined axial, transverse and twisting motions of the twisted blade [6]. A modeling method of the twisted blade with a concentrated mass was proposed in Yoo's work [7], and the effects of some parameters such as the concentrated mass, its location and the twist angles on the natural characteristics of the twisted blade were investigated. In addition, in order to avoid the weakness of beam model which can not consider the vibration in width direction of the blade, some researchers also studied the blade modeling methods based on the theory of plates and shells [14–16].

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