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Comparison of rubbing induced vibration responses using varying-thickness-twisted shell and solid-element blade models

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ABSTRACT

In aircraft engines, small gaps between the rotating blade and casing can improve the overall efficiency, but may also cause blade-casing rubbings, which usually induce damages of the blade and casing, or generate excessive vibrations of rotor. Numerical simulation is the most commonly used method in analyzing blade-casing rubbings, but the simulation results are highly affected by blade modeling techniques. In this study, two finite element (FE) models of blade, i.e., a variable-thickness-twisted shell (VTTS) model and a solid element model are developed on the platform of ANSYS software, and rubbing induced vibration responses using the two FE models are compared. In these two models, the blade-tip is equally divided into 20 elements, and the corresponding casing is equally divided into 21 two-degree-of-freedom lumped mass points (LMPs) along the axis of rotation, which are rigidly connected (i.e., these mass points have the same vibration displacements) to describe the global casing vibration. Rubbing induced vibration responses of the blade and casing are investigated based on these models, where the angle misalignment, radial misalignment, radial elongation of the blade-tip and casing vibration are taken into account. Considering the effects of angle misalignment, blade-casing rubbing is simulated during the run-up process from 0 RPM to 10,000 RPM. The results exhibit that the VTTS model has higher calculation efficiency than the solid model in the rubbing simulation. For example, for the rubbing simulation during the run-up process, the calculation time using the VTTS model decreases by almost 23% comparing with the solid element model under the same element numbers (400 elements). In these two models, the rubbing-induced vibration characteristics, such as super-harmonic resonances and flexural-torsional coupled vibrations, are almost the same. The errors between the primary and super-harmonic resonance speeds are all less than 1%, and the maximum error between the amplitudes corresponding to these resonance speeds is about 1.23%. In addition, because the different centrifugal loads acting on model nodes lead to different blade-tip deformations rooting in the nonuniformity of the blade thickness along the chordwise direction, different rubbing positions are also detected by using these two models.

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Nomenclature

c_{ri}, c_{fi}	damping of lumped mass point (LMP) i of casing in radial and flexural directions
$\mathbf{D}$	global damping matrix of blade
e_c	radial misalignment
f_n, f_t	normal and tangential rubbing forces of the cantilever beam model
f_{n1}, f_{n2}	first and second natural frequencies of blade during rotation (Hz)
$\mathbf{f}$	excitation force vector of blade
F_e, F_{e1}	aerodynamic force (Pa) and amplitude of aerodynamic force (Pa)
F^i_{Zb}, F^i_{Xb}	normal and tangential rubbing forces of blade-tip node i
$\mathbf{G}(\Omega)$	global Coriolis force matrix of blade
g_{min}	minimal gap between the blade-tip and casing without considering angle misalignment
g^i_{min}	minimal gap between the blade-tip node i and the corresponding LMP i of casing considering angle misalignment
g^i	gap between blade-tip node i and corresponding LMP i of casing at rest
g^i_{rub}	gap between blade-tip node i and corresponding LMP i of the casing during rotation
$\mathbf{K}_{acc}(\dot{\Omega})$	stiffness matrix caused by acceleration
$\mathbf{K}_c(\Omega)$	stress stiffening matrix (also called centrifugal stiffening matrix)
$\mathbf{K}_e$	global structural stiffness matrix of blade
$\mathbf{K}_s(\Omega)$	global spin softening matrix of blade
k_{ri}, k_{fi}	casing stiffness of LMP i in radial and flexural directions
$\mathbf{M}$	global mass matrix of blade
m_c, $m\underline{i}$	total mass of casing and mass of LMP i of casing
R_c, R_d, R_g	radii of the casing, disk and blade-tip orbit at rest
$\mathbf{u}$	displacement vector of blade

Greek symbols

δ_i	penetration depth of blade-tip node i
δ	sum of the penetration depth of each blade-tip node
β_1	misalignment angle
β_2	stagger angle of blade
μ	coefficient of friction of Coulomb friction model
θ	angular displacement of spin (rad)
Ω	angular velocity of spin (rad/s)
$\dot{\Omega}$	angular acceleration of spin (rad/s^2)

Abbreviation

FE	finite element
VTTS	variable-thickness-twisted shell
LMP	lumped mass point
1F	first flexural mode of blade
1T	first torsional mode of blade

1. Introduction

In aeroengine, small gaps between the rotor and stator can increase the overall efficiency, but can also cause the blade-casing interaction which may induce complex dynamic excitations of the engine, and may threaten the engine structural integrity or weaken the system performance. The interaction between the rotor and stator has been drawn extensive attention, and detailed analysis and summary about the rotor-stator interaction are presented in Refs. [1,2]. In general, the interaction phenomena can be classified as rubbing [3–5], modal interaction (also known as travelling wave speed coincidence) [6–8] and whirl and whip [9]. In order to get a better understanding of dynamic behaviors during interaction, some numerical [3,4,10–19] and experimental investigations [20–23] have been proposed, as well as some developed numerical algorithms were introduced in Refs. [24,25].

According to different levels of complexity, blade models can be simplified as beam, plate and three-dimensional (3D) finite element (FE) models in many studies dedicated to the blade-casing rubbing and modal analysis. Many researchers simplified 3D blades as simple beam models due to the high simplicity of beam models [3,4,8,14,15]. Considering the flexural-torsional-axial motion under unilateral contact conditions, Sinha [14] proposed a mathematical model to analyze the vibration response of a rotating pretwisted Timoshenko cantilever beam under a blade-tip rubbing. Sinha [26] pointed out that