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Vibration response analysis of a rotational shaft–disk–blade system with blade-tip rubbing

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ABSTRACT

This paper aims at the blade-casing rubbing in a shaft–disk–blade (SDB) system including shaft, disk, blade and bearing, and focuses the effects of stagger angles of blades, rotational speeds and casing stiffness on the rubbing-induced vibration responses of the SDB system and casing. Firstly, a finite element (FE) model of an SDB system is developed, and the rubbing between the blade-tip and casing is simulated using contact dynamics theory. In the proposed model, Timoshenko beam elements are adopted to simulate the shaft and the blade, and shell elements to simulate the disk, and spring-damping elements to simulate the ball bearings. A point–point contact element is adopted to simulate the blade-casing rubbing. Moreover, the augmented Lagrangian method is utilized to deal with contact constraint conditions, and the Coulomb friction model is used to simulate the friction between the blade and casing. The proposed model is also validated by comparing the natural frequencies with those obtained from the published literature. The results indicate that (1) amplitude amplification phenomena can be observed when the multiple frequency components coincide with the torsional natural frequency of the SDB system and the bending natural frequencies of the blades under rotational state; (2) the torsional vibration features of the SDB system with blade-tip rubbing are more significant than the lateral vibration features of the shaft; (3) the torsional vibration of the SDB system increases, and the blade bending vibration reduces with the increase of the stagger angle of the blade; (4) period-2 motion may appear under the large casing stiffness and high rotational speeds, and the torsional vibration of the SDB system and blade bending vibration tend to increase with the increasing casing stiffness.

1. Introduction

Blade-casing rubbing has been regarded as a significant contributor to excessive maintenance and in general to engine failure. The rubbing may result in complicated vibration of the overall unit, and may reduce the system performance and the lives of the blade and the casing. Blade-casing rubbing is essentially nonlinear, which involves contacts, large displacements and deformations of the blade. Moreover, the rubbing may be characterized by significant interactions between the global dynamics of the shaft–disk–blade (SDB) system and local vibration of the blade [1].

Many researchers studied the blade-casing rubbing mechanisms and rubbing induced complicated nonlinear dynamic behaviors using cantilever beams to simulate the blades [2–6]. Padovan and choy [2] deduced the relationship between the normal contact force and the blade radial deformation by simplifying the blade as a cantilever beam. Considering the effect of the centrifugal force of the blade, Jiang et al. [3] derived the normal blade-casing rubbing force based on Padovan's model. Based on Jiang's model [3], Ma et al. [4] developed a revised model of the rubbing between the blade and flexible casing, and verified the revised model using experimental results. Sinha [5] presented many mathematical expressions about the impulse loading, such as half-sine wave, rectangular pulse or sawtooth pulse, and analyzed the vibration responses of the rotating Timoshenko beam under the impulse loading of the half-sine wave. Simplifying the blade and casing as the straight beam and curved beam respectively, Batailly et al. [6] analyzed the rubbing between the blade tip and the casing by adopting a combination of component mode synthesis methods with a contact algorithm based on the Lagrange multiplier technique.

Because cantilever beam models are difficult to describe the blade torsional vibration, many researchers adopted cantilever plate models to simulate blade-casing rubbing [7,8]. Kou and Yuan [7] simplified two types of functions (sine wave and sine pulse

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wave) as axial rubbing load, and studied the nonlinear vibration responses of a rotating large deflection plate subjected to rubbing. Yuan and Kou [8] analyzed the rubbing-induced dynamic responses of a rotating nonlinear plate under thermal shock, and discussed the influences of friction coefficient, contact stiffness and thermal transient on the vibration response of the plate.

In order to describe the blade vibration exactly, the real blade structures are also widely used to simulate blade-casing rubbing [9–12]. Simplifying the rubbing force as impulse loads, Turner et al. [9] developed a software package to simulate the blade-casing rubbing, and analyzed the transient vibration response of the blade under the rubbing condition. Ma et al. [10] established a finite element (FE) model of a cyclic sector corresponding to blade-disk structure with dovetail connection (1/38 blade-disk) based on ANSYS software, and adopted a pulse force model to simulate the local rubbing between the blade and elastic casing, and analyzed the effects of rotational speeds and penetration depths on the blade vibration and contact behaviors of the dovetail interface under the rubbing. Legrand et al. [11] presented a full three-dimensional (3D) contact algorithm to simulate the rubbing process between the blade and the casing, and analyzed the contact dynamic characteristics of the blade and the casing. Based on a 3D FE model of a blade clamped on the root, Batailly et al. [12] studied the unilateral contact-induced dynamics of a blade rotating within a perfectly rigid yet distorted casing.

Above researches mainly focused on the local vibrations of the blade and casing, and neglected the effects of the rotor whirl. Considering the effects of the rotor whirl, many researchers established different dynamic models of the rotor–blade system such as analytical models [13,14] and FE models [15–20], to analyze the system vibration responses caused by blade-casing rubbing. Simplifying the rotating blades as cantilever beams with uniform section, Sinha [13] derived the system equations for a fully-bladed flexible rotor (shaft and disk) supported by a set of bearings at multiple locations, and analyzed the vibration responses of the system under the rubbing between the blade tip and the rigid casing. Lesaffre et al. [14] considered the gyroscopic effect, the spin softening effect and the centrifugal stiffening effect by using a pre-stressed potential, and derived a dynamic model of a bladed rotor system by Lagrange equation. In their model, the dynamic contact problem between the blade and the casing are dealed with as a static contact problem in some rotational speed ranges by considering the casing as an elastic ring. Considering the casing deformation and shaft flexibility represented by spring, Salvat et al. [15] presented a two-dimensional (2D) FE model of the bladed-disk and casing coupling system, and studied the effects of the blade-casing rubbing on the shaft whirl. Based on a coupling model of flexible bladed shaft and casing, Parent et al. [16] analyzed the blade-casing rubbing phenomena, and pointed out that a combination is generated with one nodal diameter (1ND) interaction related to the whole model and a two nodal diameter (2ND) interaction involving the local models of the blades and casing. In their another paper [17], they underlined the effects of 3D kinematics and 3D local geometry on both rubbing detection and the system stability due to blade-casing rubbing. Their simulation results indicate that the inclination of the casing inner surface has a strong influence on the rubbing detection, and the 3D formulation affects the system stability. By using a developed contact element based on linear spring model (LSM) and non-linear spring model (NLSM), Petrov [18] proposed a multi-harmonic analysis method to simulate whole-engine vibration due to blade-casing rubbing. His research shows that the odd harmonics can be excited due to blade-casing rubbing. Moreover, the nonlinear vibration responses obtained by LSM and NLSM show a good agreement over the low rotating speed region, however, there is a large difference over the high rotational speed region. Thinery et al. [19] analyzed the dynamic behaviors of a misaligned Kaplan turbine with blade-to-stator contacts. In their model, the rotor is modeled using the FE method with beam elements while the rigid blades are adopted to deal with the contact between the rotor and casing.

Form above analysis, it can be seen that the current researches on blade-casing rubbing mainly focused on disk–blade or single blade structures. However, little attention has been devoted to the vibration responses of the SDB system with blade-tip rubbing. Moreover, the studies of the stagger angles of the blades on the rubbing are also limited. Aiming at this issue, this paper will focus on the effects of some key parameters such as stagger angles of the blades, rotational speed and casing stiffness, on the blade-casing rubbing and complicated nonlinear behaviors due to blade-tip rubbing. It is worth noting that a shorter version of this journal paper is included in the Proceedings of the 9th IFToMM International Conference on Rotor Dynamics [20]. On the basis of Ref. [20], this paper adopts the contact element to simulate the rubbing between the blade-tip and casing, and considers the effects of the stagger angle of the blades and casing vibration, and analyzes the effects of stagger angles on the system vibration responses.

The structure of the paper is as follows. After this introduction, a dynamic model of an SDB system is established based on FE method in Section 2.1, and an FE model with the blade-casing rubbing is described in Section 2.2, and in Section 2.3, the proposed model is validated by comparing the natural frequencies with those in published literature. The effects of rotational speed, stagger angle of the blade, and casing stiffness on the vibration responses of the SDB system are analyzed in Sections 3.1, 3.2 and 3.3, respectively. Finally, conclusions are drawn in Section 4.

2. Dynamic model of an SDB system

2.1. FE model of an SDB system based on ANSYS

The FE model of an SDB system is simplified according to the following assumptions:

(1) The shaft is divided into 14 Timoshenko beam elements (BEAM188 element in ANSYS) and 15 nodes. Every node has 6 degrees of freedom (DOFs) except the right-end node whose axial and torsional DOFs are restrained. The disk is simulated using 96 shell elements (SHELL181 element in ANSYS) and 335 nodes, and every node has six DOFs. Four identical blades are symmetrically installed on the disk by dovetailing, which are simulated using Timoshenko beam elements (BEAM188 element in ANSYS), and every blade is divided into 4 elements and 5 nodes.
(2) Two ball bearings are identical, and are simulated ideally by linear stiffness and damping, in addition the cross terms are neglected.
(3) Shaft and disk are rigidly connected, and the disk and blades are connected by sharing nodes.
(4) Gyroscopic effects of the shaft and disk are ignored.
(5) Only the rubbing between a blade and casing is considered which can correspond to the situation when the radial elongation of a blade is lightly greater than those of other blades.
(6) Lateral-torsional coupling effects due to the unbalances and blade-casing rubbing are ignored.

The model parameters of the SDB system are shown in Fig. 1a and Table 1, and the schematic of FE model of the system is shown in Fig. 1b. The symbolic meanings in Table 1 are shown in Appendix A.

Considering the centrifugal stiffening and spin softening effects of the rotational blades, the equations of motion of the SDB system