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Editorial

Application of Dry Friction Contact in Vibration Reduction in Engineering—A Short Review

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Dry friction contact is widely used to reduce vibration in engineering, especially in the vibration reduction of turbine blades, which are major components of aeroengines. Due to high vibrational stresses, high cycle fatigue failure of turbine blades occurs and may cause serious aeroengine incidents. Using dry friction contact to consume energy and adjust the natural frequency of the blades, dry friction dampers have been verified to have good performance in vibration reduction. A great deal of research has been carried out in this area, including dry friction contact models, dynamic modelling, solving methods and vibration reduction characteristics.

There are two typical types of dry friction contact models: the macro-slip model and the micro-slip model.

The macro-slip model is a single-point contact model which assumes that the normal load distribution on the contact interface is uniform, and that all contact points are sliding or sticking synchronously. When the normal load or the contact area is small, using the macro-slip model to characterize the dry friction contact seems to be appropriate. Two main macro-slip models, the Coulomb friction model and the bilinear hysteresis model, have been adopted frequently. Den Hartog [1] first introduced the Coulomb friction model to model the dry friction contact between the damping structure and the blade, and a single-degree-of-freedom blade damping system was studied. Oden and Pires [2] further extended the traditional Coulomb friction law by proposing a non-local friction law in integral form. In [3], the Coulomb friction model was improved by introducing the exponential velocity-dependent friction coefficient model. The difference between the static and dynamic friction coefficients was considered and the Coulomb stick–slip motion based on the velocity criterion was analyzed. For the rigid body assumption in Coulomb friction law, Goodman and Klummp [4] first considered the contact stiffness and deformation, and pioneered the bilinear hysteresis model, in which the sliding between the contact surfaces does not occur suddenly and there is a certain elastic deformation at the interface before sliding, which is determined by the sliding load and the critical friction force. Srinivasan and Cassenti [5] considered the elastic deformation at the contact point before relative sliding occurs and observed that the normal stress at the contact point has non-local characteristics by using anisotropic weighting functions.

The theory of micro-slip originated from the work of Mindlin et al. [6]. They established a micro-slip model related to the shear layer theory, using multiple contact points in a slip state or viscous state to describe the characteristics of the contact surface. Iwan [7] provided a series parameter model and a parallel parameter model to model the micro-slip at the interface. Menq et al. [8] established a micro-slip model in which the dry friction interface has an elastic shear layer with negligible thickness. Based on [8], Csaba [9] further proposed a micro-slip friction model with a quadratic normal load distribution type, in which the shear layer was removed for simplification. Petrov and Ewins [10] developed a

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Citation: He, S.; Zhang, W.; Li, Z. Application of Dry Friction Contact in Vibration Reduction in Engineering—A Short Review. *Coatings* **2023**, *13*, 910. https://doi.org/10.3390/coatings13050910

Received: 9 May 2023
Accepted: 11 May 2023
Published: 12 May 2023

micro-slip model by considering time-varying friction contact parameters and anisotropic contact stiffness. Liu et al. [11] proposed a friction contact stiffness model to describe the friction force at different rough interfaces and different normal loads, in which the friction contact surface is discretized to a series of friction contact pairs that each can experience stick, slip, or separate states.

As the two classical dynamic models of turbine blades with dry friction dampers, the lumped mass model and the finite element model both have their own advantages.

The lumped mass model has high computational efficiency and is suitable for studying the mechanism of vibration reduction. Griffin [12] established a single-degree-of-freedom model and verified its availability through experiments. In [13], a single-degree-of-freedom model that accounts for both normal load variation and stick–slip phenomenon was set up. Cigeroglu and Ozguve [14] simplified the dry friction system as a multi-degree-of-freedom lumped mass model for nonlinear vibration analysis. According to cyclically symmetric assumption, Kumar and Sarkar [15] proposed a three-degrees-of-freedom lumped mass model and ensured that their dynamic characteristics were equivalent to the first three eigenvalues of actual turbine blades.

The finite element model is not as efficient as the lumped mass model in calculation efficiency. However, its numerical results are closer to engineering practice. In [16], a pre-twisted blade with varying cross-sections was modelled as a Timoshenko beam, and the dynamic equation of the model was established based on finite element theory. In [17], a commercial turbine-bladed disk with an under-platform damper was set up by using the finite element analysis software, in which its first flap, first torsion, and first edgewise modes were considered. Ma et al. [18] established a finite element model by simplifying the rotating shrouded blade as a cantilever beam with a mass point at the free end and considering the effects of the centrifugal stiffening, spin softening, and Coriolis force. Pesek et al. [19] established a three-dimensional finite element model of a surface-to-surface dry friction contact model, and dry friction interblade damping was performed. In addition, in order to improve the computational efficiency of the finite element method, Alain [20] used the modal synthesis method to simplify the contact motion to be one-dimensional, then reduced the order of the large finite element equation and solved the dynamic response of blades with damping blocks.

The dry friction damping system has strong nonlinearity and often has no analytical solution. Numerical computation methods, including time domain methods, frequency domain methods, and time–frequency conversion methods, are frequently adopted.

Time-domain methods, such as the Runge–Kutta methods, Newmark-β method, and Wilson-θ method, can be chosen according the kind of the dynamic model used. They can accurately capture the state at each moment and have high computational accuracy. Wang and Shieh [21] used the Runge–Kutta method to solve a mass-spring system with a single degree of freedom. In [22], the Newmark-β method was introduced to solve a finite element model and the influence of shroud structure parameters on the damping effect of the shroud was studied.

The frequency domain methods, including the first-order harmonic balance method and the multi-harmonic balance method, often have higher computational efficiency, in contrast with the time-domain methods. Meanwhile, as the number of higher-order harmonics being considered increases, the computational efficiency may decrease. Lou et al. [23] analyzed the vibration reduction effect of cylindrical dampers and wedge dampers by the first-order harmonic balance method. In [24], the first-order harmonic balance method was used to transform the nonlinear differential equations into a set of nonlinear algebraic equations. Vibration characteristics of the single degree of freedom system under periodic excitation were studied. For the strong nonlinear system, higher-order harmonics should be included to ensure the solving accuracy. Lau and Cheung [25] proposed an incremental harmonic balance method by introducing incremental terms and found that it has higher computational efficiency compared to the time-domain method. Ren and Beards [26] considered higher-order harmonics and proposed a method based on the flexibility method, which improved the computational efficiency partly. Hall et al. [27] developed a high-order harmonic balance method using the Discrete Fourier Transform method for linearization, which has higher computational efficiency and equal computational accuracy compared with the time-domain method.

The time–frequency transformation method uses a fast Fourier transform and its inverse transform to repeatedly transform in the time–frequency domain until a convergent calculation result is obtained. It has both the computational accuracy of the time-domain method and the computational efficiency of the frequency-domain method. Laxalde et al. [28] proposed the dynamic Lagrangian method based on the time-frequency conversion method, and analyzed the integral blade disk structure system in some detail. In [29], the dynamic Lagrangian frequency time algorithm was employed to capture the nonlinear effects, and the effect of under-platform dampers on blade disk contact occurrence and damping efficiency was investigated. Herzog et al. [30] compared the higher-order harmonic balance method with the dynamic Lagrangian method. They identified that these two methods have their own advantages regarding computational efficiency and numerical iterative stability. Siewert et al. [31] calculated the contact force in the time domain and converted it into the frequency domain, then solved the dynamic response of the shrouded blade.

Recently, two kinds of dry friction damper, under-platform damper and shrouded blade damper, have been widely used in vibration reduction of turbine blades. Vibration reduction characteristics, the most important topic in this area, have been investigated in detail by many researchers and great progress has been made.

Under-platform dampers are installed at the root of the blades and use the dry friction contact between the blade and the damper to suppress the blade's vibration. Sayed et al. [32] studied the effect of energy consumption and the viscous and slip transitions of an under-platform damper system, which showed that the contact state of optimal vibration reduction was half-stuck and half-slip. He et al. [33] found that with the external excitation amplitude increasing, the vibration reduction of the transient response may be negative, and a larger normal load would be needed to maintain the vibration reduction efficiency of the damper. In [34], with uniform, linear and Hertzian normal load distributions, the dynamic characteristic of a one-dimensional friction system was discussed, which indicated that within a steady-state cycle, the energy dissipation of friction force of uniform distribution increases by 30% compared to that of linear distribution. In [35], the study of a rotating under-platform damper system showed that with the increase in rotating speed, the resonance amplitude first increases and then decreases, a resonance frequency shift occurs, and there was no energy dissipation shown under enough high rotating speed as the interface was in the stick phase. Hu et al. [36] analyzed a system including two blades with an under-platform damper. They found that when the two blades were subjected to the same excitation force, the energy dissipation of the system was higher than that subjecting to a single excitation force or a pair of reverse excitation forces.

While rub and impact are often applied to suppress the vibration of the shrouded blade, Xie et al. [37] found that the displacement amplitude decreases with the increase of stagger angles or twist angles, which is due to the fact that both angles can weaken the effects of the spin-softening matrix, while the twist angle strengthens the effects of the structural stiffness matrix. In [38], within a certain gap range (0.02 mm–0.1 mm), the amplitude of the blade tip decreases and the resonant frequency increases with the gap decreasing, and within the setting range, the mass of the shroud has little effect on the vibration reduction characteristics. Guo et al. [39] found that when the shroud position was closer to the blade tip, the vibration reduction effect of the shrouded blade was improved. In [40], the dynamics of a shrouded blade with an asymmetric gap was investigated, which showed that the displacement amplitude increases with the ratio of the gap of the right side to the left side increasing from 1 to 1.5.

To sum up, despite significant progress in the research of micro-scale dry friction contact, the precise modeling of friction contact remains a challenging task. Furthermore, there is a lack of efficient and accurate finite element analysis methods for the engineering design of dry friction dampers. To address these challenges, strong collaboration between industry and academia is necessary.

Author Contributions: Investigation, S.H.; writing—original draft preparation, S.H. and W.Z.; writing—review and editing, Z.L. All authors have read and agreed to the published version of the manuscript.

Funding: H.S. is grateful for the financial support from the National Natural Science Foundation of China (No. 51405452). Support from China Postdoctoral Science Foundation to L.Z. is also gratefully acknowledged (No. 2019M652564).

Conflicts of Interest: The authors declare no conflict of interest.

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