Dry friction damping blade structural vibration analysis of ocean nuclear power platform steam turbine

J Wu¹,², X S Lao¹, C H Dai¹, Y Liu¹ and W J Lv¹

¹Science and Technology on Thermal Energy and Power Laboratory, Wuhan Second Ship Design and Research Institute, Wuhan, Hubei Province, China

E-mail: wu_jun1986@126.com

Abstract. The ocean nuclear platform turbine blade is studied. Based on the nonlinear finite element method, the contact pressure load on the friction contact surface is analyzed. The three-dimensional finite element vibration characteristic numerical model is established. The spring damping element is applied to describe the friction damping characteristics between the shroud and snubber contact surfaces of adjacent blades. The equivalent stiffness and damping coefficients of the shroud and snubber contact surfaces are calculated based on the experiment results of dry friction dynamic characteristics with blade material. The structural vibration response under different contact pressure load is analyzed by harmonic balance method. Furthermore, the simulation results are compared with the experiment results. The results show that the blade structural vibration response decreases significantly as a result of the damping between friction contact surfaces. The relative error of the resonance frequency for free blade is only 0.53%. The relative errors of modal damping ratio for the free blade and constrained blade are less than 2.65% and 20% respectively. The relative error of the constrained blade resonance amplitude is less than 15%. The reliability of the numerical analysis method is verified by comparison with experimental data.

1. Introduction
Ocean nuclear power platform is an organic combination of nuclear reactor and ship engineering. The Rankine cycle has been utilized in the ocean nuclear power platform to convert thermal energy into electrical energy efficiently. It provides a safe and efficient energy supply for offshore oil exploration and remote islands. It has great significance and far-reaching impact on the development and utilization of new energy and global energy. The steam turbine is a key component in the procedure of thermal power conversion of the ocean nuclear power platform. The last stage blades tend to be longer, which reduces the stiffness and thus weakens the blade's ability to resist vibration stresses. It greatly increases the possibility of high cycle fatigue failure of the blade. For the purpose of blade vibration reduction and safety improvement, the dry friction damping structure is utilized in the last stage long blade to improve its stiffness and damping. As a result, the combination of shroud and snubber structure has been adopted during the design process of the steam turbine last stage blade. When the blades vibrate, vibration energy is consumed by the relative motion between the contact surface of adjacent shroud and snubber. The shroud and snubber structural can not only provide friction damping, but also can connect the entire circle of blades which effectively increase the stiffness of the blade and reduce the vibration response of the blade.

The solution of the vibration characteristics of dry friction damper is a complex nonlinear problem due to the inherent non-linearity of the friction law. The main calculation methods are time domain
method, frequency domain method and time-frequency conversion method at present. The time domain method includes the analytical method and numerical integration method. The frequency domain method includes the harmonic balance method.

The analytical method is mainly utilized to obtain the exact solution of the steady-state response of simple models such as single-degree-of-freedom system or two-degree-of-freedom system. However, the analytical method is not applicable to complex multi-degrees-of-freedom systems, multiple nonlinearities, and multi-harmonic excitations [1]. The advantage of the numerical integration method is the ability to accurately track the time history of the system response. Dry friction damping blades belong to weak damping systems. In order to ensure the accuracy of the calculation, the integration time step is often very small and requires a lot of calculation time due to the nonlinear characteristics of friction [1]. Compared with the numerical integration method, the harmonic balance method proposed by Nayfeh and Mook greatly improves the calculation efficiency [2]. Therefore, this method has been widely used for the nonlinear vibration response characteristics calculation of blades with dry friction damping structure [3,4]. Menq et al use harmonic balance method to analyze the harmonic vibration of dry friction damping blades [5]. Moreover, the dynamic response characteristics of the system are obtained and verified by numerical integration method [6]. Petrov et al conduct the blade vibration simulation of by multi-harmonic balance method combined with three-dimensional finite element model [7]. A three-dimensional friction contact model for solving the nonlinear friction force between contact surfaces is proposed by Gu et al [8]. In summary, the harmonic balance method in the frequency domain can greatly improve the computational efficiency and increase the accuracy of the computation.

The ocean nuclear power platform steam turbine last stage blade with shroud and snubber structural is studied here. According to the blade material friction experimental results, the dry friction damping blade vibration characteristics analysis method is proposed. The blade vibration response under different shroud and snubber normal contact force with different rotational speeds are computed. Moreover, the accuracy of the calculation results is verified by the corresponding experimental results.

2. Three-dimensional model of the dry friction damping blade
As shown in figure 2, the blade is composed of root, body and friction damping structure. The friction damping structure includes the shroud structure at the top of the blade body and the snubber structure at a height of about 50% of the blade body.

The total number of the whole circle blades is 76. When the whole circle blades are in the state of initial installation, there is a certain installation clearance both between the adjacent shroud contact surfaces and between the adjacent snubber contact surfaces. When the whole circle blades are in the state of working condition, the clearance between the contact surfaces are closed and a certain contact pressure load is produced under the effect of torsional recovery due to the blade structural
deformation. On the one hand, the blade stiffness increases due to the friction effect between the contact surfaces. It plays a role in adjusting the frequency and avoiding resonance. On the other hand, the friction effect consumes vibrational energy and increases damping of the structure. The combined effect of the two aspects effectively reduces the vibration amplitude of the blade and improves its operation safety.

Figure 2. The three-dimensional model of the blade and the friction surface.

3. Analysis method of the dry friction damping blade vibration response characteristics

The figure 3 shows the vibration response characteristic analysis process of the dry friction damping blade. The entire calculation process consists of static analysis and vibration analysis. Finite element method is applied to establish the numerical model for the static analysis and vibration analysis. The nonlinear contact static analysis is conducted by ANSYS. The contact pressure loads between friction surfaces are calculation by integral operation based on the numerical results.

Figure 3. The vibration analysis method of the dry friction damping blade.

The three-dimensional finite element vibration characteristic numerical model of the dry friction damping blade is constructed. The spring damper element is adopted in the model to describe the friction damping characteristic between the contact surfaces of adjacent shrouds and snubbers. The equivalent coefficients of stiffness and damping on the friction surfaces are calculated according to the blade material dry friction damping dynamic characteristics experimental results. Based on the harmonic balance method, the nonlinear vibration response characteristics under different normal contact forces are obtained by iteration calculations.

4. Nonlinear contact analysis

In order to improve the accuracy of the contact pressure load between friction surfaces, the whole circle blade model shown in figure 4 is adopted during the static analysis. The blade material is stainless steel with density of 7810 kg/m³, elastic modulus of 2.01e11 Pa and Poisson's ratio of 0.282.
Appropriate boundary conditions are applied to the whole circle blade model, such as displacement constraints, contact relationships and angular velocity load.

![Figure 4. The whole circle blades model.](image1)

![Figure 5. Contact pressure load results.](image2)

A total of 18 speed conditions were selected in the range of 2100~3000 rpm for detailed analysis. The distributions of displacement, von-misses stress and contact pressure are obtained under different rotational speed. Figure 5 shows the contact region of friction surface under the rotational speed of 2100 r/min, 2413 r/min and 3000 r/min. The size of the friction contact area increases with the increase of the rotational speed. Contact pressure loads between contact surfaces are calculated by integrating the area of the contact pressure under each condition. It provides basic data for the blade vibration response characteristics analysis.

5. Vibration response characteristics analysis

Figure 6 shows the three-dimensional model of the blade. Figure 7 shows the finite element model of the blade. The entire blade is meshed with the 8-node hexahedral element. The spring damping element is established on the contact surface to simulate the friction damping effect between the adjacent blade contact surfaces. The total number of nodes and elements of the blade are 10388 and 7428 separately.

![Figure 6. Geometry model.](image3)

![Figure 7. Finite element model.](image4)

The equivalent coefficients of stiffness $K_e$ and damping $C_e$ are computed by the linearized formula (1) and formula (2) according to the blade material dry friction damping dynamic characteristics experiment results [9].

$$K_e = \begin{cases} -0.2794\tilde{A} + 1 & 0 < \tilde{A} < 2 \\ -0.1293\tilde{A} + 0.6998 & 2 < \tilde{A} < 4.5 \end{cases}$$

$$C_e = \begin{cases} 0.4610\tilde{A} & 0 < \tilde{A} < 2 \\ -0.1335\tilde{A} + 1.1890 & 2 < \tilde{A} < 4.5 \end{cases}$$

Where $\tilde{A}$ denotes the displacement amplitude between contact surfaces [9].
On this basis, the blade vibration numerical model is established by the ANSYS Parametric Design Language. The dry friction damping blade vibration characteristics are calculated under different contact pressure loads. The blade vibration response curves are obtained under free and constraint state for two sampling points respectively. The position of sampling point 1 is at the blade section middle where about 80 mm away from the shroud bottom. The position of sampling point 2 is at the tailing edge position of the blade section where about 60 mm away from the snubber top.

The dimensionless variable $\gamma$ is defined during the result analysis.

\[
\gamma = \frac{A}{A_\gamma}
\]

Where $\gamma$ denotes the dimensionless vibration response amplitude, $A$ denotes the blade vibration response amplitude, $A_\gamma$ denotes the free blade resonance response amplitude on sampling point 1 with the excitation force amplitude of 2 N.

Figure 8. Blade vibration response curves of the sampling point 1. (a) Overall view and (b) Partial view.

Figure 9. Blade vibration response curves of the sampling point 2. (a) Overall view and (b) Partial view.

The dry friction damping blade vibration response curves of two sampling points are shown in figures 8 and 9.

The calculation results imply that the blade resonance vibration amplitude is significantly smaller than that of the free blade when the friction contact surface begins to contact. Compared with the free blade, the maximum reductions value of blade resonance amplitude at monitoring point 1 and point 2 are 98.63% and 92.36% respectively. Under the effect of damping between the shroud contact surfaces and snubber contact surfaces, the vibration energy is consumed effectively.

The contact pressure load increases as the blade changes from the free state to the state when only shroud contacts. During this period, the blade resonance vibration amplitude first decreases and then
increases. The resonance frequency increases gradually. Moreover, the blade changes from the state when only shroud contacts to the state when all the damping structures contact as the increase of the normal contact force. Similar to the previous process, the blade resonance vibration amplitude also first decreases and then increases. The resonance frequency is further increased. This is mainly caused by the nonlinear frictional damping between contact surfaces when contact pressure load continuous increases.

6. Verification of the numerical simulation results
The vibration response characteristic of the real blade with dry friction damping structure is measured by a test rig in [10]. The numerical results in this paper are compared with the experimental results in [10] to verify the reliability of the numerical analysis method.

Six working conditions are selected for the comparison, including condition A to condition F. The corresponding rotational speeds for the working conditions are 0 r/min, 2200 r/min, 2300 r/min, 2413 r/min, 2600 r/min and 3000 r/min respectively.

Figure 10 shows that the numerical result of the amplitude-frequency response curve of the free blade agrees well with that of the experimental results. \( \gamma_n \) denotes the blade resonance amplitude and \( \zeta \) denotes the modal damping ratio.

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\begin{array}{c}
\text{Figure 10. Results for the free blade.} \\
\text{Figure 11. Results for the constraint blade.}
\end{array}
\]

The experimental value of the free blade resonance frequency is 57.0 Hz and the numerical value is 56.4 Hz. The relative error of the resonance frequency is 0.53%. At this point, energy is primarily consumed by material damping rather than frictional damping. Furthermore, the value of \( \zeta \) is calculated based on the vibration response data. The experimental value of \( \zeta \) is 0.00543 and the numerical value is 0.00529. The relative error of \( \zeta \) for the free blade is just 2.65%.

As the shroud contact pressure load increases, the blade resonance response amplitude first decreases and then increases when only the shroud contacts. At the same time, the \( \zeta \) changes exactly the opposite. As the shroud and snubber both contact, the changing patterns of the resonance frequency and \( \zeta \) are similar to the situation when only shroud contacts. Figure 11 shows that the changing patterns of the blade resonance frequency and \( \gamma_n \) on the contact pressure load for numerical results and experimental results are similar.

Results in figures 12 and 13 imply that the difference of the \( \gamma_n \) and \( \zeta \) between the numerical results and experimental results are relative small. The relative errors for the \( \gamma_n \) and the \( \zeta \) are less than 15% and 20% respectively. Therefore, the calculation results agree well with the experimental results. The energy consumption process is a typical nonlinear process when the normal contact force increases. The value of the energy consumption is determined by the combination of the contact pressure load and the relative displacement. The energy consumption process of the friction contact surfaces between the adjacent shrouds and snubbers can be effectively predicted by the harmonic balance method combined with the friction damping dynamic mathematical model.
7. Conclusions
The vibration response characteristics method of the blade with shroud and snubber is calculated by the combination of finite element method, harmonic balance method and friction damping model. Vibration response curves of the blade are obtained under different normal contact force. The blade resonance vibration amplitude first decreases and then increases. The resonance frequency increases gradually. Compared with the experimental results, the relative error of the resonance frequency for free blade is only 0.53%. The relative error of resonance frequency and $\zeta$ is 0.53% and 2.65% respectively for the free blade. The relative errors of the $\gamma_n$ and the $\zeta$ are less than 15% and 20% respectively. The accuracy of the calculation results is verified by the experimental results.

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