Static Frequency Test and Vibration Safety Analysis of Turbine Blades in a Power Plant

Baotong Chai*, Zhengfeng Wu, DongXing Zhang
Huadian Electric Power Research Institute Co., LTD. Hangzhou, Zhejiang, China
*523893780@163.com

Abstract. During the overhaul period of steam turbine in some plant, the blades’ vibrational security had been analyzed through the stationary frequency on the spot from the blades by using spectrum analyzer of Nanjing trued software engineering co., Ltd. The results of the test showed the turbine’s blades of next to last stages (stage groups) were safe when they were in operation with the working rotating speed.

1. Introduction
In the steam turbine equipment of a power station, the blade is one of the key rotating parts, and it is also a part where the accident is prone to occur. According to statistics from relevant agencies, in the accidents of steam turbines, the failure caused by blade factors accounts for 30% - 40%, most of which are caused by the inherent vibration characteristics of steam turbine blades[1]. Due to the long blade shape of the turbine rotor, especially the low-pressure rotor, the blade will be subject to steam and the centrifugal force generated by the blade itself. The natural frequency of the blade itself will be affected by some factors (such as blade cracks, blade looseness, etc.) influences.

The rotor blade dynamic frequency test is objectively complex. Therefore, the blade static frequency test can be used to check whether the actual assembly condition of the blade and the vibration characteristics of the individual blade itself are close to the resonance frequency, so that accidents of the unit equipment can be avoided in time. Through the on-site tap test of the turbine frequency modulation blade to determine whether the blade meets the requirements of long-term safe operation.

During the overhaul of Unit 2 of a power plant, the final two stage blades of the rotor was subject to a striking frequency test.

2. Blade frequency test method, principle and test instrument
There are many methods for determining the static natural frequency of the blade. In this paper, the self-vibration method is used for blade frequency measurement. Firstly, on the macroscopic level, the last two stages of the unit are inspected on site to observe whether there are surface cracks or the like; secondly, the tapping test is carried out, and each blade is rhythmically tapped by using a rubber hammer. The blade generates a freely attenuated vibration signal, which is then converted into an electrical signal input into the oscilloscope through a vibrator, and is compared with a known frequency signal generated in the signal generator to obtain a Lisharu diagram, thereby determining the natural frequency of the rotor blade. The natural frequency of the blade is measured by the self-vibration method. This method is fast, accurate and simple. The main mode shapes of the medium and long blades monitored are the most dangerous type A0 (first order), and the high-order vibration of the
blade, such as A1 type (second order), A2 type (third order), belong to fast disappearing, small amplitude vibration.

The on-site blade frequency measurement test uses the spectrum analyzer produced by Nanjing Anzheng Co., Ltd. The frequency measurement principle of the instrument is that the sensor converts the mechanical vibration signal of the blade after being impacted by an external force into an electrical signal and inputs it into the spectrum analyzer, and then passes through the filtering and amplification of the spectrum analyzer. After conversion, the computer analyzes the blade spectrum, and the accurate value (Hz) corresponding to the peak value at the maximum amplitude is the natural frequency of the blade. Striking frequency test, as shown in Figure 1.

3. Unit overview and On-site test data

3.1 Unit overview
The No. 2 steam turbine has a total of 18 flow stages, of which the 17th stage (the penultimate stage) of the rotor is a set of blades connected by a shroud, and the 18th (last stage) set of blades is a composite structure of a pull-gold and a shroud. This striking frequency test only conducts on-site tests on the 17th and 18th frequency modulation blades. The relevant parameters are shown in Table 1.

| Blade series | Total number of blades | Blade height / mm | Average diameter / mm | Blade type                  | Number of blade sets |
|--------------|------------------------|-------------------|-----------------------|-----------------------------|---------------------|
| 17           | 78                     | 448               | 1765                  | variable section twisted blade | 26*3                |
| 18           | 90                     | 792               | 2060                  | variable section twisted blade | 30*3                |

3.2 On-site test data
Draw the test data collected on the 17th and 18th frequency modulation blades, as shown in Figure 2 and Figure 3.
3.3 Blade vibration safety analysis

By analyzing the natural frequency of the last two stages and the A0 (first-order) vibration mode by on-site striking test, it can be further inferred whether the blade can avoid the resonance zone and can operate safely[2]. The safety of the blade is generally judged by the following two indicators, the blade safety override and the blade frequency dispersion [3].

3.3.1 Blade safety override. Only the blade safety override is not less than its corresponding allowable resonance safety override value to ensure reliable use of the blade. When the blade natural frequency is close to or the same as the excitation force frequency, the blade will resonate.

Blade dynamic frequency $f_d$ and resonance speed $n_1$ are calculated as:
\[ K n_i = \sqrt{f_s^2 + B_b n_i^2} \quad (K=1,2,3,\ldots,K) \]  
(1)

\[ f_i = K f_i = K n_1 \quad (K=1,2,3,\ldots,K) \]  
(2)

\[ n_i = \frac{f_s}{\sqrt{K^2 - B_b}} \]  
(3)

In the formulas (1) to (3), \( K, B_b, f_i \), and \( f_r \) respectively represent the blade resonance magnification, the blade dynamic frequency coefficient, the blade static frequency, and the blade excitation force frequency.

Since the blade dynamic coefficient \( B_b \) is very complicated to solve, engineering practice often uses empirical formulas to calculate.

\[ B_b = 0.72 \frac{D}{L} - 1 \]  
(4)

In the formula (4), \( D \) and \( L \) respectively represent the average diameter of the stage and the working height of the blade.

The kinetic coefficient \( B_b \) can be calculated by the blade structural parameters listed in the above formula (4) and Table 1, and the calculated values are as shown in Table 2 below.

| Blade series | Blade height \( L \) (mm) | Average diameter \( D \) (mm) | Dynamic frequency coefficient \( B_b \) |
|--------------|--------------------------|-----------------------------|----------------------------------|
| 17           | 448                      | 1765                        | 1.83                             |
| 18           | 792                      | 2060                        | 1.6                              |

The formula for calculating the resonance safety ratio \( \Delta \) of the frequency modulation blade:

\[ \Delta = \frac{n - n_c}{n} \times 100\% \]  
(5)

In equation (5), \( n_c \) and \( n \) respectively represent the critical speed and operating speed of the unit.

The safety rate of the last two stages is calculated by the first-order frequency value collected by the tap test, as shown in Table 3.

| Blade series | Resonance ratio \( K \) | Critical speed r\( \cdot \)s\(^{-1} \) | Resonance safety override calculation | Allowable resonance safety override % |
|--------------|------------------------|------------------------------------------|-----------------------------------|------------------------------------|
| 17           | 2                      | 69                                       | 38                                | 5                                  |
| 17           | 3                      | 38                                       | 24                                | 4                                  |
| 18           | 2                      | 137                                      | 174                               | 5                                  |
| 18           | 3                      | 75                                       | 50                                | 4                                  |

The standard of resonance safety rate increases as the blade resonance ratio \( K \) decreases. This is because the smaller the \( K \) value is, the shorter the period of free vibration of the blade is under the action of two adjacent excitation disturbances. At the same time, the common amplitude value and stress will increase accordingly [4]. It can be obtained from Table 3 above that the low-frequency allowable resonance safety override of the last two stages of the blade is much smaller than its corresponding calculated value, indicating that the blade can be safe, stable and reliable [5].

3.3.2 Blade frequency dispersion. The blade frequency dispersion \( \Delta f \) is:

\[ \Delta f = \frac{f_{\text{max}} - f_{\text{min}}}{(f_{\text{max}} + f_{\text{min}})/2} \times 100\% \]  
(6)

In equation (6), \( f_{\text{max}} \) and \( f_{\text{min}} \) represent the maximum and minimum static frequency values of the first-order A0 type vibration of the blade, respectively. In the regulation, the blade frequency dispersion degree \( \Delta f < 8\% \) is acceptable.
The static frequency values of the last two stages measured by the test are brought into the formula (6), and the frequency dispersion of each blade is calculated as shown in Table 4 below.

| Blade series | Minimum static frequency Hz | Maximum static frequency Hz | Frequency dispersion % |
|--------------|-----------------------------|-----------------------------|------------------------|
| 17           | 101                         | 106                         | 4.8                    |
| 18           | 202                         | 209                         | 3.4                    |

4. Conclusion

Through the static frequency test and vibration safety analysis of the last two stages of the No. 2 steam turbine of a power plant, the following conclusions are drawn:

(1) The Nanjing Anzheng spectrum analyzer was used to conduct on-site blade striking test to obtain the first-order A0 natural frequency of the 17th and 18th grade blades.

(2) The frequency dispersion degree of the last two stages of Unit 2 is $\Delta f < 8\%$, and the allowable resonance safety override value of the blade is much smaller than the experimental calculation value, and the blade meets the requirements of safe operation.

Acknowledgements

I would like to show my deepest gratitude to my wife, Cao xue, who has provided me with valuable guidance in every stage of the writing of this thesis. Without her enlightening instruction, impressive kindness and patience, I could not have completed my thesis. Her keen and vigorous academic observation enlightens me not only in this thesis but also in my future study.

References

[1] Bachschmid N., Salvini G., Tanzi E., Pesatori E. The Influence of Blade Row Dynamics on Lateral and Torsional Shaft Vibrations in Steam Turbines. In: Pennacchi P. (eds) Proceedings of the 9th IFToMM International Conference on Rotor Dynamics. Mechanisms and Machine Science, vol 21. Springer, Cham. (2015).

[2] Nordmann R, Krueger, T. Impact of rotordynamics on blade vibrations. In: Blade mechanics seminar, Winterthur. (2011).

[3] Mazur Z, Garcia-Illescas R, Aguirre-Romano J, et al. Steam turbine blade failure analysis[J]. Engineering Failure Analysis, 2008, 15(1-2): 129-141.

[4] Chai B T, Wu Z F, Zhang D X. Static Frequency Test and Leaf Fracture Analysis of Turbine Blades in a Power Plant[C]//Applied Mechanics and Materials. Trans Tech Publications Ltd, 2019, 893: 45-51.

[5] Mundt G, Neidel A, Matijasevic-Lux B. Moving blade failure in the low-pressure turbine of a steam turbo set[J]. Practical Metallography, 2012, 49(12): 782-791.