Application Case of NK50/56 Small Steam Turbine for Dynamic Balance

Qingliang Niu1,*, Yuli Gong2, Guocheng Tian1, Chuanmei Liu3 and Qian Wang3

1Shandong Branch, Huadian Electric Power Research Institute Co., LTD, Jinan, Shandong, China
2Cnooc Petrochemical Engineering Co., LTD, Jinan, Shandong, China
3Zouxian Power Plant, China Huadian Engineering Co., LTD, Jining, Shandong, China
*corresponding author’s e-mail: niuqingliang@foxmail.com

Abstract. Vibration anomaly occurred at the first start of the NK50/56 small steam turbine after repair A plant, and the vibration increased with the increase of rotational speed. Through data analysis, the vibration spectrum was mainly composed of power frequency and high frequency. The main reason was that there was mass imbalance and slight friction in the steam turbine. After two times of dynamic balance, the problem of abnormal vibration of small steam turbine was successfully solved.

1. Introduction
Small steam turbine (the full name "feed water pump driven steam turbine" ) is an important rotating machinery equipment of power plant, which of stable and reliable operation directly determine the safety and economic production of power generation enterprises. However, the vibration problem inevitably occurs during the operation of the equipment. The main causes of vibration problems of small steam turbines include rotor mass imbalance, rotor friction, coupling misalignment, looseness and oil film oscillation, etc. When the vibration of small steam turbine occurs, how to quickly identify the fault, find out the cause of vibration and accurate treatment is always a problem of concern to the majority of technicians.

Dynamic balance is the most extensive way to deal with the vibration problem of small steam turbines. The position of the unbalance of the rotating machine is measured in the state of rotation, and the position and size of the corresponding balancing block is determined accordingly. This balancing method is called dynamic balance Dynamic balance refers to the process of detecting and adjusting the mass distribution of the rotor when necessary to ensure that the residual unbalance or the vibration of the journal and the force acting on the bearing are within the specified range under the corresponding operating speed frequency[1]. Dynamic balance can eliminate both dynamic unbalance couple and static unbalance centrifugal force. So the dynamic balance is a practical and convenient method to eliminate vibration on site. Dynamic balance is divided into rigid rotor dynamic balance and flexible rotor dynamic balance according to different balancing objects. For the rotor whose working speed is far below the critical speed, the deformation caused by unbalance is very small. The rotor can be treated as a rigid body, and the dynamic balance can be carried out at a low speed, which is called the dynamic balance of the rigid rotor. In the dynamic balancing of the rigid rotor, the centrifugal inertial force system caused by the unbalance of each micro segment can be simplified to two arbitrarily selected sections, and the dynamic balancing can be completed by making the corresponding
correction (de-weighting or counterbalancing) on these two surfaces. When the rotor working at the supercritical speed starts and stops, the speed must pass the critical speed, then the unbalance will make the rotor produce obvious deformation. If the deviation of the center of mass of each segment of the rotor has obvious influence on the deformation of the axis of rotation, the rotor cannot be treated as a rigid rotor. Then the corresponding dynamic balance is called the dynamic balance of the flexible rotor.

In this paper, the vibration problem of NK50/56 small steam turbine in a factory was analysed and diagnosed by means of rigid rotor dynamic balance, combining the process of vibration increase, fault characteristics and equipment maintenance. The treatment effect was good, which can be used for reference for similar units.

2. Fault Situation

2.1. Introduction to Small Steam Turbine
The title is set 17 point Times Bold, flush left, unjustified. The first letter of the title should be capitalized with the rest in lower case. It should not be indented. Leave 28 mm of space above the title and 10 mm after the title.

The steam turbine of a power generation enterprise with 335 MW unit was equipped with two steam feed pumps, which were correspondingly equipped with nK50/56 small steam turbine produced by Hangzhou Steam Turbine Co., LTD. The equipment condition was as follows:

Rated speed: 5250r/min, rated steam inlet pressure: 0.7017mpa, rated steam inlet temperature: 328℃, rated steam outlet pressure: 0.0072mpa. The small steam turbine shaft and the feed water pump shaft were connected with a toothed coupling. The rotor was supported by two radial bearings. The front bearing was #1 and the back bearing was #2. The shafting structure was shown in Figure 1.

Vibration of #1 bearing and #2 bearing was measured by eddy current sensor, characterized by displacement. Vibration of #3 bearing and #4 bearing was measured by velocity sensor and characterized by velocity.

![Figure 1. Shafting structure diagram.](image)

2.2. The Failure Process
At 2:30 on March 13, 2019, the small steam turbine with the repair of Unit A began to run. When the speed reached 1000r/min, the axial vibration of #1 and #2 rose slowly. When it reached 4200r/min, the axial vibration of 1X, 1Y, 2X and 2Y reached 107.5 m, 25.3 m, 98.2 m and 34.0 m, respectively. If the value of 1X and 2X exceeds the unit alarm value of 90 m, the unit will no longer rise at a safe speed. In the course of running, the maximum horizontal and vertical vibration of #3 bearing is 1.7mm/s and 1.3mm/s respectively, and the maximum horizontal and vertical vibration of #4 bearing is 1.2mm/s and 1.1mm/s respectively, both of which are good vibration levels.

Before this repair, the small steam turbine operating in the 3800r/min-4800r/min range, 1X, 1Y, 2X, 2Y are all 50 m range, vibration level is good. Suspected due to the new adjustment of the steam seal occurred dynamic and static friction, decided to reduce vibration through the warm machine and friction.
At 7:40 on March 13th, respectively in 1000 r/min, 1800 r/min and 2400 r/min the warm-up phase, each phase warming-up 60 minutes vibration could reduce 10 microns, but it was no longer reduce the vibration with continuing to warm machine. The vibration of #1 and #2 axis increased as the rotating speed rises.

At 23:25 on March 13th, the seal clearance was adjusted again from 0.50mm to 0.65mm. The vibration of #1 and #2 axis increased with the increase of rotating speed. The vibration trend diagram in the punching process was shown in Figure 2.

3. Vibration Fault Diagnosis and Treatment

3.1. Style and spacing

In the process of impact rotation, the vibration of #1 and #2 axis changed significantly with the rotation speed, which was characterized by mass imbalance from the perspective of trend. To further investigate the reasons, 1X, 1Y, 2X and 2Y vibration spectrums were adjusted when the rotation speed reached 4200r/min, as shown in Figure 3-6.

It was suspected that there was a problem with the tooth coupling, because when the rotor on both sides produced relative displacement during operation, the tooth surface of the internal and external teeth periodically made axial relative sliding. Once this working state changed, abnormal vibration would be generated[2].

Understands from power plants, as long as the fit clearance between sleeve and the outer tooth coupling controlled within the standard value of 0.04 ~ 0.10 mm, would be able to make the inner gear sleeve in high speed runtime to automatically. Which prevented the gear sleeve from deviating from the center line instantly and having a large unbalance, thus preventing the shafting from producing a large forced vibration[3].
According to the above analysis, 1X, 1Y, 2X and 2Y were mainly in power frequency vibration with a small amount of high-frequency components at 4200r/min. The possibility of shaft coupling center deviation, oil film vortex and steam induced vibration was excluded, but it was impossible to further determine whether it was mass imbalance or frictional vibration caused by vibration anomaly. The appearance of the high frequency component indicated that there was friction vibration, but the vibration reduction was not obvious after the full warm-up, but the vibration still increased with the increase of rotating speed. It could be speculated that the vibration anomaly of small steam turbine was related to rotor mass imbalance. The central rotating part of the rotor falls off, and the first unbalanced component was mainly excited, which had a great influence on the vibration at the critical speed. The rotor end parts fallen off simultaneously excited the first-order and second-order unbalanced components, which had a great influence on the vibration of both the critical and working speeds[4]. Through the above analysis, the possibility of falling off the rotating parts was excluded.

Therefore, it was decided to try to adopt dynamic balance method to solve this problem. According to the operation rules, the critical speed of the small turbine is > 7000 r/min, and the working speed is 3800 r/min-5000 r/min. The small turbine motor balance belongs to the low-speed
balance of the rigid rotor. The rotor is required to show strict rigid behavior[5], which can be regarded as a rigid rotor when it is generally lower than 70% of the critical first-order speed of the rotor.

3.2. Style and spacing
At present, there are a lot of researches on dynamic balance algorithms, but the most commonly used methods are still the influence coefficient method and the mode of vibration method[6]. In the balance of rigid rotor, the influence coefficient method is generally adopted. The influence coefficient method of rigid rotor is mostly applied to single plane and biplane equilibrium. The basic principle of equilibrium is the same as that of equilibrium on a single plane. The double plane balance needs to be weighted on two planes respectively, while the single plane only needs to be weighted on one plane. If the weight loss surface of the rotor is in a certain plane, the single plane balance method can be adopted. At present, domestic turbine manufacturers still mainly adopt the traditional influence coefficient method to carry out high-speed dynamic balance of rotor. In the process of use, they mainly select the balance surface and the balance speed by experience, and calculate the influence coefficient and counterweight[7].

In this case, the rotor of the small and medium steam turbine is a multistage impeller, and in the field balance, the two stages of the impeller were aggravated simultaneously. The influence coefficient balancing method was described below.

- The original vibration $A_1$,
- The try weighting $P_1$,
- The weighted vibration $A_2$,
- The influence coefficient $C$:

$$ C = \frac{A_2 - A_1}{P_1} \quad (1) $$

The amount of correction $\overline{P_2}$:

$$ \overline{P_2} = -\frac{A_1}{C} \quad (2) $$

The phase difference (hysteresis Angle) between vibration and force is very important to reasonably determine the weighting Angle of dynamic balance test[8]. Taking the data in Table 1 as the original data of dynamic balance, it was calculated the weighted Angle according to the lag Angle of 30°. So the author added 100g $\angle 110°$ to each of the final stage blades at both ends of the exciter, and the vibration level after constant speed was shown in Table 2. The vibration frequency of #1 axis decreased by 20$\mu$m, and the vibration frequency of #2 axis decreased by 17$\mu$m. The first dynamic balance effect was not obvious.

The effect of the first dynamic balance was not obvious, which would be related to the inappropriate selection of lag Angle or the inappropriate aggravation of mass. For trial weighting, the rotor safety should be ensured first. On this basis, the Angle and quality should be adjusted according to the response of the rotor.

| Table 1. Vibration data of the first dynamic balancing front shafting. |
|---------------------------------------------------------------|
| Serial Number | #1x | #1y | #2x | #2y |
| Vibration Value/$\mu$m | 107.5 | 25.3 | 98.2 | 34.0 |
| Power frequency value and phase/$\mu$m | $94.5 \angle 275°$ | $24.4 \angle 355°$ | $89.0 \angle 264°$ | $30.7 \angle 353°$ |
Put the data in Table 1 and Table 2 into the formula in turn (3)–(4),
\[
\bar{a} = (74.6 \angle 223^\circ - 94.5 \angle 275^\circ) ÷ 100 \angle 110^\circ = 0.763 \mu \text{m/g} \angle 35^\circ \tag{3}
\]
\[
\bar{P}_2 = (94.5 \angle 275^\circ) ÷ 0.763 \angle 35 = -123.853 \mu \text{g} \angle 240^\circ \tag{4}
\]

The reverse weight was 123.853g\angle 60^\circ.

At 12:15 on March 14th, according to analysis and calculation, 125g\angle 60^\circ was increased at the bearings of #1 and #2 of the small steam turbine. At 21:20 on March 14, the set constant speed was 4200r/min, and the vibration data were shown in Table 3. After continuous operation for 60min, the rotational speed was rushed to 5000 r/min, thus the vibration problem of the small steam turbine was solved. The vibration data were shown in Table 4.

### Table 3. Vibration data of the second dynamic balance rear shafting.

| Serial Number | #1x    | #1y    | #2x    | #2y    |
|---------------|--------|--------|--------|--------|
| Vibration Value/μm | 16.3   | 26.8   | 9.5    | 12.1   |
| Power frequency value and phase/μm | 10.4 \angle 200^\circ | 22.3 \angle 280^\circ | 15.6 \angle 210^\circ | 12.8 \angle 283^\circ |

### Table 4. Vibration data of shafting at 5000 r/min.

| Serial Number | #1x    | #1y    | #2x    | #2y    |
|---------------|--------|--------|--------|--------|
| Vibration Value/μm | 20.5   | 31.6   | 22.1   | 18.9   |
| Power frequency value and phase/μm | 16.4 \angle 205^\circ | 26.2 \angle 289^\circ | 21.9 \angle 214^\circ | 16.3 \angle 286^\circ |

### 4. Conclusion
According to the operation parameters, vibration data and equipment installation of small steam turbine, the author analyzed the reason of abnormal vibration of small steam turbine and formulated the measures. The main reason for the vibration of the small steam turbine was the mass unbalanced. After the components of friction vibration reduced by heating machine and rubbing, the dynamic balance of the stable power frequency vibration part was carried out.

Rotor dynamic balance is an important method to deal with vibration faults of rotating machinery, which eliminates the unbalance (centrifugal force and centrifugal force couple) caused by rotor rotation. In practice, as long as the power frequency vibration stably, it can be solved by dynamic balance. However, if shafting adjustment occurs during the next overhaul, the dynamic balance may need to be redone. Therefore, it is required that the maintenance of the turbine body and bearing and the center adjustment must be serious and serious.

### Acknowledgments
The research work of this topic was supported by Research and Application of Intelligent Protection System for All Working Conditions of Shafting Motion Pair of Steam Turbine Generator Unit.

### References
[1] L B Li, B Y Zou and Y W Li 2012 Study on high-speed dynamic balance of low-pressure II rotor of 1000MW ultra-supercritical steam turbine Steam Turbine Technology vol4 pp317-320.
[2] Q S Gao, X W Deng and C Zhang 2014 Unbalance response analysis of single support 1000 MW USC steam turbine *Vibration and Impact* pp202-203.

[3] J M Xiao 2018 Vibration fault analysis and treatment of feed water pump turbine with 600 MW unit *Electromechanical Information* vol558 pp80-83.

[4] Y W Tian and P Yin 2014 Research on vibration Mechanism caused by falling off rotating parts of steam turbine *Steam Turbine Technology* vol2 pp145-149.

[5] W Q Deng, G Tang and D P Gao 2008 An review on dynamic characteristics and dynamic balance of rotors *Gas Turbine Test and Research* vol21 pp57-62.

[6] G Y Zhang 2018 Study on dynamic balance of Turbine rotor and its Evaluation criteria *Thermal Turbine* vol47 pp38-41.

[7] G F Bin, L D He and J J Gao 2013 High-speed dynamic balancing method for low-pressure rotor of large steam turbines based on modal analysis *Vibration and Shock* vol14 pp87-92.

[8] S C Ma, Y S Chen, L Shen and W J Li 2020 Analysis and treatment of abnormal vibration of 300 MW steam feed pump *Thermal Power Engineering* vol4 pp288-292.