Diagnosis and treatment of an abnormal vibration caused by oil whirl

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Abstract. During normal operation of a 680MW unit, the vibration of No.1 bearing frequently fluctuated greatly, which caused the unit to trip. Through the test and analysis of the vibration data, it was diagnosed that the cause of vibration was oil whirl caused by insufficient bearing stability. By optimizing the installation data of bearing and adjusting shafting center, adopting special method to strengthen the stiffness of No.1 bearing, the stability of bearing bush was improved. After vibration treatment, the vibration of No.1 bearing returned to normal value.

1. Introduction
The steam turbine of this unit is N680-25/600/600 type, ultra supercritical, once intermediate reheat, single shaft, four cylinder four exhaust and condensing steam turbine. The unit adopts the operation mode of full cycle steam admission. It takes the bearing vibration value as the vibration protection, and the tripping setting value is 11.8mm/s. After the unit had been put into commercial operation for 3 years, the vibration of No.1 bearing frequently fluctuated greatly and caused tripping. In order to find out the cause of the vibration, several variable load tests were carried out, and the variation trend and frequency components of the vibration were comprehensively tested.

2. Abnormal vibration data of the unit

2.1 Vibration data during tripping
Before the unit trip, the vibration of No.1 bearing at several load points increased sharply and fluctuated, and the abnormal collision sound from inside the bearing could be heard on site. The unit load was 656MW when the unit tripped. At the same time, the half frequency vibration increased greatly. The vibration data during tripping are shown in Table 1.

| Table 1. The vibration data during tripping. |
|-----------------------------------------------|
| Shaft 1X (μm) | Shaft 1Y (μm) | Bearing 1A (mm/s) | Bearing 1B (mm/s) |
| Direct | 301 | 203 | 16.1 | 12.3 |
| 1X | 54.5 | 53 | 12.1 | 10.9 |
| 0.5X | 181 | 112 | 10.8 | 11.9 |

2.2 Vibration data of the first variable load test
According to the variation of No.1 bearing vibration with load, the variable load vibration test was carried out, and the vibration analyzer was connected to the TSI signal output end to record the
vibration data of the whole test process: The load began to decrease from 600MW to 550MW and then increased to 670MW. Then the load went through a cycle of decreasing and increasing again.

According to the vibration data, the vibration of No.1 shaft fluctuated greatly in the range of 570MW~590MW, which was obviously different from the load when the turbine tripped. When the vibration increased, the half frequency vibration rose sharply again. The vibration data during first test are shown in Table 2.

|          | Shaft 1X (μm) | Shaft 1Y (μm) | Bearing 1A (mm/s) | Bearing 1B (mm/s) |
|----------|---------------|---------------|-------------------|-------------------|
| Direct   | 100~135       | 95~110        | 1.2~1.3           | 1.3~1.5           |
| 1X       | 90~120~190    | 90~100~120    | 0.9~1.1           | 1.0~1.2           |
| 0.5X     | 10~80         | 5~55          | 1.1~1.5           | 1.3~1.6           |

2.3 Shaft vibration data of the second variable load test
When the unit was in stable operation at 680MW full load, the No.1 shaft vibration was relatively stable with small fluctuation, and the shaft vibration of 1X and 1Y were about 110μm and 106μm respectively. The variable load test was started, and the load gradually decreased until the load stabilized at 400MW. The vibration data during this period are shown in Table 3.

|          | Shaft 1X (μm) | Shaft 1Y (μm) |
|----------|---------------|---------------|
| 680MW    | 90~107        | 92~111        |
| 540MW    | 87            | 112           |
| 400MW    | 103~162       | 122~180       |

It can be seen from the vibration data that the vibration changed obviously with the load in this test. When the load decreased, and the vibration tended to increase gradually. The change trend of the direct frequency and the power frequency was basically the same. When the load was 680MW, the vibration fluctuation was relatively small, and there was a certain low frequency component. When the load dropped below 560MW, the obvious low-frequency component began to appear. After the load dropped to 400MW, the load kept stable, and the vibration fluctuated greatly, the amplitude of sudden low-frequency component reached the maximum about 102μm.

3. Cause analysis of abnormal vibration
For a new unit which has been put into operation for only 3 years, its foundation settlement is relatively large, which will cause the deviation of rotor’s center in the flow passage and the change of load distribution. In addition, the high steam pressure of ultra supercritical unit and the uneven clearance of rotor steam seal will produce a great additional steam exciting force. The bearing stability is poor due to the small bearing load and poor self-aligning of bearing bush. The larger steam exciting force will cause oil film whirl of bearing bush[1]. The following is an explanation from several aspects.

3.1 Position change of rotor center
According to the metal temperature analysis of No.1 bearing, the temperature of upper pad was higher when the turbine tripped, the two measuring points were 77℃ and 72℃. The temperature of lower pad measuring points were 65℃ and 58℃. It shows that the force of the upper bearing pad was larger than that of the lower pad, and the position of the rotor was in the upper left part of the bearing bush[2-3].

The No.1 oil film pressure was 2.5MPa at 3000rpm and 1.2MPa at full load 680MW. It shows that the rotor position rose with the increased of load. The floating of the rotor center indicated that the rotor was subjected to radial force from bottom to top. Theoretically, the rotor with full cycle steam admission should have only axial force. But in hot state, due to the uneven expansion of the cylinder, the gap changed and the rotor was subjected to radial excitation force.
3.2 Impact phenomenon of bearing bush
When the bearing vibration was large, obvious knocking and collision sound could be heard on site. The analysis shows that under the long-term continuous action of large relative vibration of the rotor, the bearing was damaged and loosened, and the faults such as poor contact and poor self-aligning performance occurred. When the effective oil film could not be established, the stability of the bearing became worse and anti-interference ability was weakened. When the bearing was subjected to a large exciting force, the vibration increased rapidly, and there was a big knock and collision sound.

3.3 The source of low frequency oil whirl
The vibration data showed that the main component of No.1 bearing vibration fluctuation was low-frequency component, which indicated that the nature of vibration belongs to self-excited vibration. Because the vibration fluctuated greatly in a certain load range, it had a greater correlation with the load. The vibration fluctuation of No.1 bearing was caused by the steam exciting force, and the steam exciting force was produced by the deviation of rotor center. Moreover, the self-aligning performance of bearing bush was poor when the rotor center changed, and the oil film formation was not good, which exerted unstable oil film force on the bearing. In other words, the unstable low-frequency vibration of No.1 bearing was caused by the joint action of steam exciting force and unstable lubricating oil film force, which made the bearing bush produce low-frequency oil whirl.

4. Vibration treatment measures and effects

4.1 Disassembly and maintenance of No.1 bearing
When No.1 bearing bush was disassembled and inspected, it was found that the spherical gasket and the left spherical surface of the bearing bracket were seriously corroded and the spherical surface was not closely matched. In order to improve the bearing load and stability, the bottom of No.1 bearing bush was raised by 0.20mm. After grinding the pillow surface of the bearing bush, coating it with red lead, and then grinding them to make the contact area reach more than 75%[4-5].

4.2 Maintenance of cat claw support key of high pressure cylinder
During the inspection, it was found that the left and right cat claw support keys at the front end of the high pressure cylinder were worn. In order to prevent uneven change of clearance caused by blocked expansion and contraction of high pressure cylinder during operation, all worn supporting keys were replaced with new spare parts.

4.3 Bearing reinforcement measures
According to the vibration characteristics of No.1 bearing, the bearing bush had insufficient dynamic stiffness, and the bearing bush was easy to loose and shift during operation. Therefore, it was decided to install jacking screw on the upper cover of the bearing as an auxiliary measure to fasten the bearing during the operation. There was already a process hole on the upper cover of No.1 bearing box. Two jacking screws were added on both sides of the original process hole. The center distance between adjacent jacking screws was 55mm. The center line of the new jacking screw holes and the original process hole was parallel to the turbine rotor, and each jacking screw was perpendicular to the shaft center. The installation position of jacking screws were shown in Figure 1.
4.4 Vibration treatment effect
After above maintenance and treatment, no more abnormal low-frequency vibration of No.1 bearing of the unit had occurred during the loading process. The vibration of other bearings in the shafting were stable, and the abnormal sound of No.1 bearing disappeared. The unit could be put into normal commercial operation.

5. Conclusion
If the clearance of large steam turbine with full circle steam admission is not uniform, it can also produce greater steam exciting force. When the load of bearing is light and the stiffness is insufficient, the bearing bush will produce low-frequency oil whirl under the joint action of rotor exciting force and oil film force. This kind of oil whirl can be eliminated by properly raising the center of bearing and improving the stiffness of bearing bush.

References
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