Troubleshooting of a 330MW turbo generator unit during commissioning

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Abstract. During the commissioning stage of a 330MW turbo generator unit, several serious defects occurred, such as low vacuum of condenser, abnormal vibration of shafting and bearing damage. Through fault analysis and vibration test, the hidden danger in design, manufacture and installation were found out and solved, the unit successfully completed 168 hours full load test run and put into operation.

1. Introduction
The unit is N330-17.75/540/540 type single shaft three cylinder subcritical double exhaust condensing steam turbine. The generator model is QFSN-330-2 type water-hydrogen-hydrogen brushless excitation unit, with stator winding water cooling, rotor winding hydrogen internal cooling, iron core and other components hydrogen cooling. The shafting of turbo generator unit is composed of high pressure rotor, intermediate pressure rotor, low pressure rotor and generator rotor. All rotors are connected by rigid coupling. The steam turbine is equipped with six radial support bearings and one thrust bearing, all radial support bearings are spherical elliptical bush bearings. The type of generator bearings is cylindrical bush, and its inner side is annular hydrogen sealing pad. The shafting structure is shown in Figure 1.

Figure 1. The shafting structural diagram of the unit.

2. Analysis and treatment of low vacuum of condenser
In the test run stage, when the circulating water temperature was less than 29℃ and the unit load was only 273MW, the vacuum had been reduced to -87.78KPa, which affected the normal load of the unit. With the further increase of ambient temperature, the vacuum continued to decrease. If the normal vacuum could not be maintained, the unit would not be able to operate normally.

2.1 Cause analysis of low vacuum
There was no rubber ball cleaning equipment in the unit, and the condenser end difference was higher than 9℃, indicating that the heat exchange performance of the condenser was very poor, which was the main reason for the low vacuum. After washing the condenser, the vacuum could be increased by 3KPa under the same conditions, indicating that the accumulation of fouling on the condenser surface
had seriously affected the heat transfer of the condenser. However, because the vacuum value was very low at that time, it was impossible to carry out half cleaning during the operation of the unit. Moreover, the condenser still had the problem that the inlet and outlet valves of circulating water were not closed tightly, so the unit could only be shut down for cleaning[1].

The circulating water used by the unit was seawater. Since the water source was far away from the power plant and the waterway was relatively high relative to the sea level, new seawater would be added only when the sea water was at high tide twice a month. The normal seawater was basically internal circulation and can only use the surface heat dissipation, so the seawater temperature was high. This was another important reason for the low vacuum. In addition, there was no secondary filter screen in the circulating water, which could not filter out the impurities in the water, so it was easier to accumulate in the condenser.

2.2 Treatment measures for low vacuum
According to above analysis, the following solutions were taken: The rubber ball cleaning equipment and secondary filter screen were added, and ensure that they could operate normally after commissioning. Cooling water with lower temperature was used as working water of vacuum pump. A large capacity make-up pump was added at the seawater intake to timely replenish the seawater with lower temperature and change the water temperature and water quality. The designed circulating water flow of condenser was much higher than that of current circulating water pump, so it was replaced by circulating water pump with larger capacity. After the above treatment, the condenser vacuum of the unit could meet the requirements of normal operation.

3. Analysis and treatment of steam turbine shafting vibration
During the test run of the unit, when the load was 267MW, the main steam pressure was 16.47MPa, the main steam temperature was 547°C, and the vacuum was -90.5KPa, the shaft vibration of steam turbine bearings began to increase, among which 3X grew most rapidly. Half an hour later, 3X shaft vibration had reached 184μm, and the turbine tripped due to large vibration protection. The shaft vibration alarm value of the unit is 140μm, and the tripping value is 180μm.

3.1 Analysis of 3X abnormal shaft vibration
After analyzing the data of abnormal vibration of the unit, it was found that the main frequency component of 3X shaft vibration was 50Hz. In addition, it had the characteristics of vibration climbing under constant speed. The main causes of this abnormal shaft vibration were as follows: the abnormal change of operating parameters, the stress change of steam pipe, and the uneven material of intermediate pressure cylinder caused the uneven expansion of the cylinder block, which made the front side of the cylinder block deviate to one side, then the shaft seal and journal rubbed on the side with insufficient clearance[2-3].

Because the front shaft seal of intermediate pressure cylinder was an elastic steam seal with certain pre tightening force composed of cylindrical spring, it could yield after contact, so it was not easy to be worn off, which made the rubbing repeatedly.

On the other hand, due to the expansion deviation of the intermediate pressure cylinder, the oil baffle of No.3 bearing will also occur rubbing with rotor. However, the copper teeth could wear in a short time, the rubbing disappeared after the gap increased, so it was not the main reason for the abnormal vibration.

3.2 Vibration treatment measures
The clearance of the sliding pin system of the intermediate pressure cylinder was measured again, and the front side deviation of the intermediate pressure cylinder was compensated in reverse direction. According to the wear condition of the front shaft seal of the IP cylinder, the preload of the shaft seal spring was appropriately reduced and the yielding clearance of the steam seal body was increased. The floating oil baffle of No.3 bearing was checked and repaired according to the degree of wear, and the
clearance was enlarged to the upper limit. After maintenance, the unit was started again, and the vibration of steam turbine shaft system was qualified and stable. The BODE diagrams of 3X shaft vibration of IP rotor is shown in Figure 2.

![Figure 2. The BODE diagram of 3X shaft vibration.](image)

4. Vibration analysis and treatment of generator bearing

During the load adjustment of the unit, 8X shaft vibration of the generator gradually increased from the normal value 80μm. When the load was 320MW, 8X shaft vibration was 150μm. The relevant data of the abnormal vibration were analyzed. It was considered that the main reason for the abnormal vibration of 8X shaft vibration was the rubbing of the sealing pad. The cause of rubbing was poor tracking of balance valve, which reduced the inlet oil flow of hydrogen side. In addition, the large temperature difference between hydrogen side and air side led to poor expansion of sealing pad, resulting in deformation of sealing pad and rubbing between sealing pad and journal[4-5].

![Figure 3. The BODE diagram of 8X shaft vibration.](image)

Based on the above analysis, the sealing pad was disassembled and inspected. It was found that the clearance and contact surface on both sides of air and hydrogen were within the standard range. The following measures were taken in operation: keep the inlet oil temperature of sealing oil at hydrogen side and air side above 45°C, control the temperature difference within 1°C, and increase the differential pressure between oil and hydrogen to 60KPa -70KPa. After vibration treatment, the shafting vibration of generator was qualified and stable during the start-up and load stage. The BODE diagrams of 8X shaft vibration is shown in Figure 3.
5. Treatment of high temperature of No.4 bearing

During normal operation the temperatures of the two measuring points of No.4 bearing were 87°C and 99°C respectively. It shows that the self-aligning of bearing was not flexible in running state, resulting in large temperature difference between front and rear. After the bearing was disassembled and inspected, it was found that the lower bush was worn and the axial load of the bearing bush was uneven. Combined with the on-site maintenance, the reason for the high temperature of No. 4 bearing was analyzed, it was considered that under the hot state the No.4 bearing pedestal rose, it made the bearing bush overload and finally led to the increase of bearing bush temperature.

Based on above analysis, the bottom pad iron of No.4 bearing lower pad pillow was removed by 100μm to lower the center of No.4 bearing. And enlarged the top clearance of bearing bush to the upper limit of installation standard. After above treatment, the temperature of two measuring points of No.4 bearing decreased to 80°C and 85°C respectively when the unit load was 307MW. The metal temperature of relevant bearings are shown in Table 1.

| Table 1. The metal temperature of turbine bearings. |
|------------------|------------------|------------------|------------------|
|                  | 3A/3B (°C)       | 4A/4B (°C)       | 5A/5B (°C)       | 6A/6B (°C)       |
| 3000rpm          | 52/56            | 75/79            | 62/61            | 82/76            |
| 307MW            | 55/59            | 80/85            | 69/67            | 85/81            |

6. Conclusion

Due to the defects in design, manufacture and installation, several faults occurred during the test run of the unit. Through field test analysis and targeted maintenance treatment, these problems had been successfully solved. The analysis and treatment method used in this paper can be used for fault diagnosis of the same type of units during adjustment and test run.

References
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