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Vibration Analysis and Dynamic Balance Treatment of Condensate Pump Motor of 390MW Gas Turbine Unit

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Abstract. The on-site vibration test and diagnostic analysis of the condensate pump motor vibration of a 390 MW gas turbine unit in a power plant were carried out. It is considered that the serious imbalance of the rotor of the condensate pump motor is the main cause of the vibration abnormality of the motor. The vibration anomaly is eliminated by the on-site dynamic balance test.

1. Introduction

After the overhaul of a power plant unit, the 3A condensate pump has a large vibration fault when it is tested. The drive motor is a four-stage vertical motor. The free end of the motor has a cooling fan, which is directly connected to the motor shaft through a single button. The motor is connected to the condensate pump through a coupling, and the vibration measuring point arrangement of the condensate pump is as shown in Fig. 1.

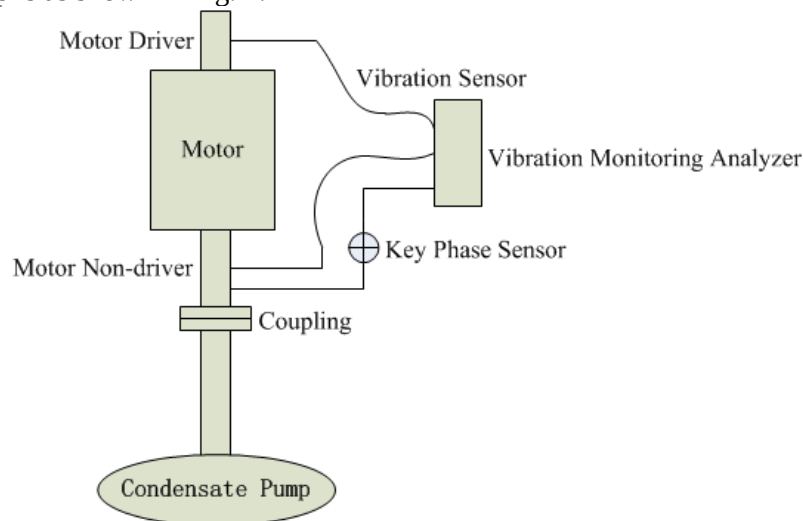


Figure 1. 3A Condensate Pump Vibration Measuring Point Arrangement

The on-site vibration data acquisition uses the EVM-8 vibration monitoring analyzer produced by Nanjing Dongzhen Measurement and Control Technology Co., Ltd. The condensate pump is monitored by two magnetolectric speed sensors with a sensitivity of 19.7 mV/mm. The two sensors are located in the vertical direction of the non-drive end of the motor and the vertical direction of the drive end. The photoelectric key sensor is located at the motor drive end and is arranged in line with the speed sensor.

2. Phenomenon and cause of failure

After the 3A condensate pump is overhauled, the vibration of the power plant during commissioning is large. After the condensate pump motor key phase sensor and vibration measuring point are arranged in the scene as shown in Fig. 1, the raw vibration data of the motor is measured and tested. See Table 1 for details. The three sets of vibration data listed in Table 1 are measured at different times, and the average value can be taken in the calculation.

Table 1. 3A Condensate Pump Motor Vibration Data

No.	Position	Pass-through amplitude/ μm	1X component/ μm	1X vibration phase/ $^{\circ}$	Speed / ($\text{r}\cdot\text{min}^{-1}$)
1	Motor non-driver	140.6	138.8	213	1493
	Motor drive	53.2	53.1	211	
2	Motor non-driver	138.9	137.1	211	1493
	Motor drive	48.8	47.9	209	
3	Motor non-driver	142.7	140.9	216	1493
	Motor drive	56.3	56.1	214	

From the vibration data measured in Table 1 above, the ratio of the fundamental frequency component of the drive end and the free end of the condensate pump motor is very large. The phase and amplitude of the 1X component are relatively stable at the rated speed, which is judged to cause equipment vibration. The cause of the abnormality is mainly caused by the rotor imbalance, which can be solved by the on-site high-speed dynamic balance test.

3. Dynamic balance theory and experiment

Dynamic balancing work requires a test weighting, and the weight should be calculated based on the vibration changes before and after the weighting. In theory, the weight and angle of the test weight can be determined casually. In fact, the test weight is very important for the dynamic balance work. Whether the test is appropriate or not is directly related to the efficiency and precision of the dynamic balance work.

When the influence coefficient is known, the test weight and angle can be directly calculated from the influence coefficient and the original vibration, which is relatively simple. When the influence coefficient is unknown, it is necessary to estimate the magnitude and angle of the original vibration estimation, and there are many factors to consider.

3.1. Reasonable determination of the angle of trial weight

In the case of on-site high-speed dynamic balance test, the test angle is especially important compared with the test weight [1]. This is because:

- Accurate test angles will reduce vibration and provide favorable conditions for subsequent dynamic balance tests.
- If the test angle is appropriate, even if the weight is too heavy or light, the vibration after the weighting will change significantly, and it is beneficial to obtain a more accurate dynamic balance influence coefficient.

After determining the on-site phase detector, the actual position of the sensor and the phase angle of the vibration, the factors affecting the angle of the unbalanced force during the dynamic balance test mainly depend on the mechanical lag angle of the equipment [2].

The mechanical lag angle is not only related to the rotor support characteristics of the equipment, but also affected by the critical speed of the rotor itself and the distance between the required speeds. Through the combination of theoretical knowledge and experience, the following points are summarized in the selection of equipment mechanical lag angle:

- For a rigidly supported rotor system, when the critical speed is greater than the equilibrium speed, the mechanical lag angle is taken from 0° to 90° ; when the equilibrium speed is close to the

critical speed, the mechanical lag angle is taken as 90° ; when the critical speed is less than the equilibrium speed, the mechanical lag angle is $90^\circ\sim 180^\circ$. The higher the rotational speed, the greater the mechanical lag angle.

- For power plant auxiliary equipment, such as motor rotor, fan rotor, pump rotor, etc., it is generally regarded as a rigid rotor, and its mechanical lag angle is selected from 0° to 45° . Determining the test weighting position in this way has been successfully verified in the actual device processing application in the field.

- For equipment with a working speed between the first-order/second-order critical speed, the unbalanced fault in this state is usually caused by the second-order imbalance of the rotor, and the mechanical lag angle is generally less than 90° .

- With the development of power generation equipment in the direction of large-scale, flexible support has emerged, that is, the critical speed of the support system is lower than the working speed. In this case, the selection of the lag angle is more complicated. In addition to the above factors, it is also necessary to consider the effect of the working speed on the distance from the critical speed of the support system. The lag angle of the bearing vibration under the working speed of the flexible support system is generally $60^\circ\sim 180^\circ$ larger than the rigid support.

The test aggravated angle can be reversed by the vibration phase and the mechanical lag angle. The meter phase usually refers to the angle from the leading edge of the pulse to the first positive peak of the vibration signal (commonly known as the high point of vibration). Therefore, when the key phase mark on the rotating shaft is aligned with the key phase sensor, a pulse signal is generated. After the phase angle is turned from this moment, the high point is just at the sensor position. It can be known from the vibration theory that the difference in angle between the unbalanced force and the high point of vibration is the mechanical lag angle. Therefore, after finding the high point of vibration, the angle of unbalanced force can be found.

3.2. Reasonable determination of the weight

The test weight is lighter, the vibration changes before and after the increase is small, and the calculated influence coefficient error is large. Excessive weighting may result in excessive vibration of the unit and damage to the equipment. In the absence of reference data, the test weight can be determined according to the principle that the centrifugal force generated by the weighting is approximately equal to 10% of the rotor weight, and the correction is performed on this basis. When you are sure to reduce the vibration, the weight of the test can be biased. If you are not sure, the weight can be lighter. The weight of the trial weight is also related to the weighted form. A large number of engineering practices have shown that at working speeds, antisymmetric weighting is usually more sensitive than symmetric weighting. Therefore, under the same vibration amplitude, the antisymmetric weight can be smaller than the symmetrical weight. [3].

3.2.1. Dynamic balance test. The on-site dynamic balance test of the condensate pump motor rotor can be performed by the influence coefficient method, that is, the original vibration of the condensate pump motor at the rated speed is first tested. Technical staff perform vibration analysis, select the test weighting plane, and start the condensate pump motor after the test is added, and then measure the vibration of each bearing. After calculating the dynamic balance influence coefficient by the vibration data before and after the weighting, technical staff adjust the weight according to the obtained result. After adjusting the weight, start the vibration measurement. If the vibration value falls to the allowable range, the on-site dynamic balance test ends, otherwise the calibration will continue. [4].

By analyzing the raw data of the condensed water pump motor vibration in Table 1, it is found that the vibration of the free end and the driving end of the motor are basically same in phase. According to the weighting plane and the allowed weighting conditions, the weight is determined on the fan wheel of the free end of the motor. Considering that the motor operating speed is $1493\text{r}\cdot\text{min}^{-1}$, it is below the first-order critical speed. According to the discussion of the mechanical lag angle in Section 3.1, the mechanical lag angle can be tested as 30° . According to experience, the first test weight is about 25g.

That is, the fan wheel on the free end of the motor is weighted by 25g $\angle$ 45° on one side. After the weighting, the vibration is as shown in Table 2.

Table 2. Vibration data after first dynamic balance test

No.	Position	Pass-through amplitude/ μm	1X component/ μm	1X vibration phase/(°)	Speed / (r·min ⁻¹)
1	Motor non-driver	105.4	104.5	229	1493
	Motor drive	40.5	39.9	227	
2	Motor non-driver	96.8	95.8	226	1493
	Motor drive	36.7	35.9	223	

Compared with before the dynamic balance test, the condensate pump motor vibration decreased. The dynamic balance adjustment test plan is: under the premise of retaining the first counterweight, the weight is again 25g $\angle$ 90° at the fan end of the free end of the motor. The vibration data after the second dynamic balance test is shown in Table 3.

Table 3. Vibration data after second dynamic balance test

No.	Position	Pass-through amplitude/ μm	1X component/ μm	1X vibration phase/(°)	Speed / (r·min ⁻¹)
1	Motor non-driver	45.7	44.2	208	1493
	Motor drive	17.5	15.8	193	
2	Motor non-driver	42.3	40.4	206	1493
	Motor drive	15.8	14.9	195	

3.2.2. Test result. The vibration data analysis after the above start-up shows that the vibration has been improved after two dynamic balance tests on the site, so that the condensate pump motor can run safely and stably at the rated working speed. At present, the operation of the motor has been in line with China's relevant regulations on the safe and stable operation of large rotating equipment [5].

4. Conclusion

Through on-site diagnostic analysis of large vibration faults of the condensate pump motor, it is found that the imbalance is the main cause of abnormal vibration of the equipment. According to the principle of large-scale rotating machinery dynamic balance test, combined with the actual situation of the site, the on-site dynamic balance test was carried out on the condensate pump motor, which finally improved the vibration of the condensate pump motor and ensured the safe and stable operation of the equipment.

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