Diagnosis and Treatment of the Shaft Vibration Problem for an Ultra Super-critical 700MW Unit

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Abstract: In this article, there existed a second order unbalance on LP1 shaft of a Super-critical 700MW unit. It can be derived from the mathematical model of flexible and symmetrical shaft vibration characteristics that there are two specific balancing methods, symmetrical for handling odd orders and symmetrical reverse for even orders. The symmetrical reverse balancing method is proved to be effective to solve the second order vibration problems.

1. Introduction

So far, super-critical 600MW turbine generators have become the major national units. According to incomplete statistics, the shaft vibration problems of domestic steam turbine generator are resolved by dynamic balancing on maintenance site. A turbine of a power plant is super-critical reheated, single-axis, four-cylinder, four row steam exhausted, condensing turbine, whose model is N700-24.2/566/566. The shaft system is formed by high pressure cylinder, intermediate pressure cylinder, low pressure cylinder A and B, generator and exciter rotors, connected by rigid coupling mutually, as shown in figure 1.

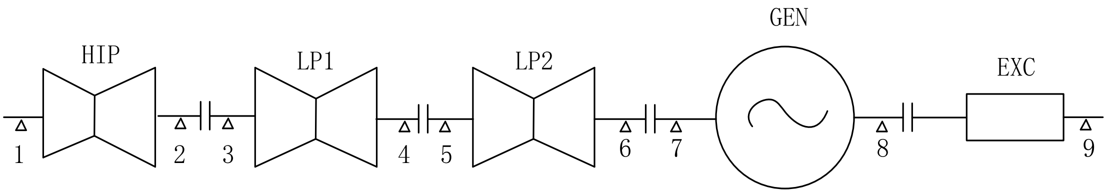

Figure.1 shaft system diagram

2. Mathematical model of flexible rotor vibration

It is known that there are two kinds of rotor, rigid rotor and flexible rotor, according to the comparison between the nominal speed and critical speed. Generally, the turbine generator shaft system belongs to flexible rotor, while auxiliary equipment such as fans and pumps belong to rigid rotor.

The vibration characteristic of the flexible rotor is shown in equation 1.

$$X_{(z)} = \sum_{n=1}^{\infty} \frac{U_{\mathrm{n}}}{1-(\omega/\omega_{\mathrm{n}})^2}\phi_n(z) \quad (1)$$

where: $\phi_n(z)$ -------------vibration mode characteristics

U_{n} --------------unbalanced vibration components

ω_n -------------Rotor nth order critical speed

ω --------------rotor speed

Z --------------the axial distance from the left journal bearing

By introducing a check quality w_j, equation 1 can be transformed equation 2.

$$X_{(z)} = \sum_{n=1}^{\infty} \frac{W_j \phi_n (z_j) + U_n}{1 - (\omega / \omega_n)^2} \quad (2)$$

While it is assumed that there are N of check quality on the rotor, the equation can be changed as 3.

$$X_{(z)} = \sum_{n=1}^{\infty} \frac{\sum_{j=1}^{N} W_j \phi_n (z_j) + U_n}{1 - (\omega / \omega_n)^2} \quad (3)$$

Flexible rotor dynamic balancing conditions can be shown as 4.

$$\sum_{j=1}^{N} W_j \phi_n (z_j) + U_n = 0 \quad (4)$$

Flexible rotor dynamic orthogonal conditions can be shown as 5 and 6.

$$\sum_{j=1}^{N} W_j \phi_s (z_j) = 0 \qquad s \neq r \quad (5)$$

$$\sum_{j=1}^{N} W_j \phi_r (z_j) + U_r = 0 \quad (6)$$

Where symmetrical flexible rotor has two journal bearings, they are symmetrical axially. It will bring out:

$$\phi_2 (z_1) = -\phi_2 (z_2), \quad \phi_1 (z_1) = \phi_1 (z_2)$$

Put into the orthogonal conditions, it will lead to dynamic balancing methods for symmetrical flexible rotor, listed in table 1.

Table 1. dynamic balancing methods for or symmetrical flexible rotor

Balancing methods	orthogonality	conclusions
Unilateral	Non-orthogonal	Activating vibration for all orders
Symmetrical	Orthogonal for even orders	Balancing only for odd orders
Symmetrical reverse	Orthogonal for odd orders	Balancing only for even orders

3. Problems and analysis

When the turbine-generator in this article was run to nominal speed 3000rpm after maintenance, vibration data of each journal bearing are listed in table 2, which shows that vibration problems have taken place at No.3 and 4 bearing of LP1 shaft.

Table 2. Vibration data of each bearing running at 3000rpm

bearing	1	2	3	4	5	6	7	8	9
x-dimension	36∠164 (56)	30∠344 (32)	83∠250 (87)	92∠70 (96)	50∠337 (60)	55∠84 (60)	9∠209 (46)	16∠240 (49)	25∠299 (56)
y-dimension	33∠75 (67)	23∠227 (27)	77∠126 (80)	83∠327 (89)	2∠313 (17)	45∠295 (62)	2∠95 (34)	13∠94 (53)	8∠157 (23)

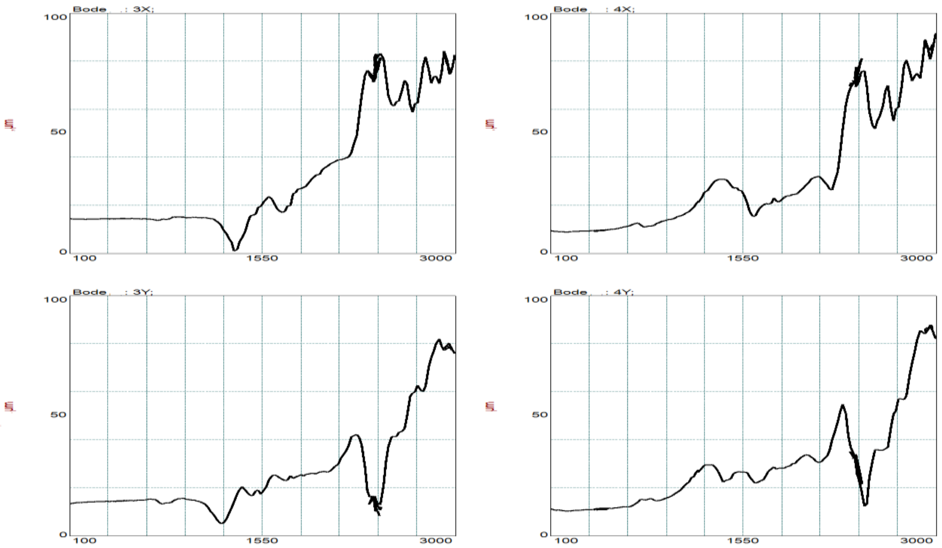

Figure 2. the bode chart of No.3 and 4 bearing

From table 2,vibration data of No.3 and 4 bearing of LP1 shaft happen to be the fundamental component, whose phase are approximately reverse. On the other hand, the bode chart(figure 2) has shown that vibration data of No.3 and 4 bearing of LP1 shaft increase dramatically after the critical speed of the shaft. It is can be concluded that there is a second-order unbalance of LP1 shaft to be balanced. According to table 1, the balancing method of vibration problems proposed in this article should be symmetrical reverse.

4. Treatment

After precise analysis and accurate calculation, 383g object should be added at 225° angle for No.3 bearing while 383g object should be added at 45° angle for No.4 bearing. Obviously they are opposite. When the turbine-generator was run to nominal speed 3000rpm once more after treatment in this way, vibration data of each journal bearing in table 3 and the bode chart of figure 3, both show the shaft vibration data decrease dramatically, which means that the shaft vibration problems presented in this article have been overcome and the symmetrical reverse balancing method is effective to solve the second order vibration problems.

Table 3. Vibration data of each bearing running at 3000rpm after treatment

bearing	1	2	3	4	5	6	7	8	9
x-dimension	22∠170 (44)	29∠3 (33)	40∠279 (44)	31∠67 (39)	59∠334 (60)	31∠88 (40)	12∠234 (48)	11∠155 (41)	15∠307 (50)
y-dimension	23∠79 (40)	26∠251 (28)	43∠149 (46)	44∠323 (49)	31∠215 (40)	34∠291 (55)	5∠158 (36)	20∠50 (56)	2∠127 (20)

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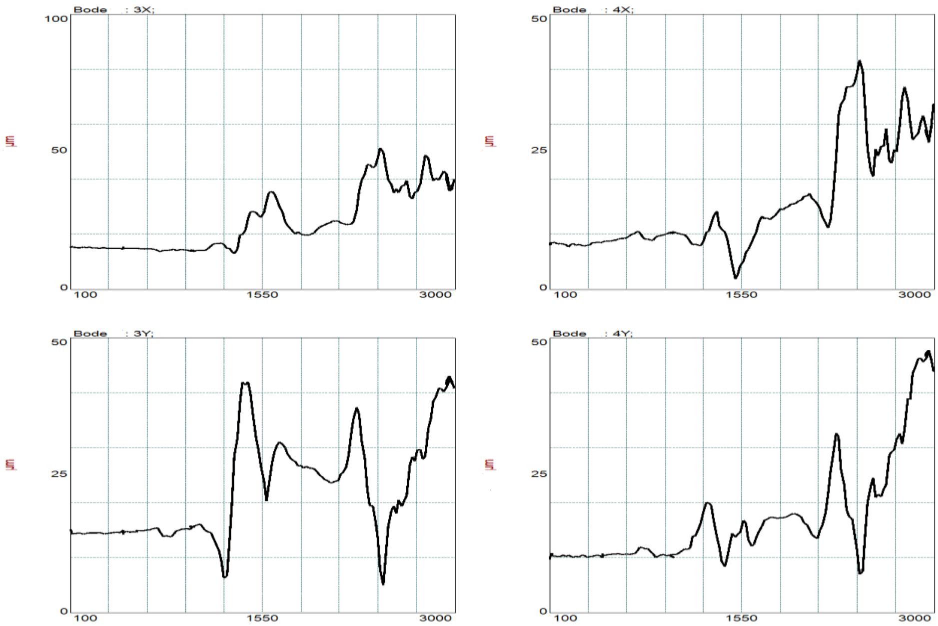

Figure 3. the bode chart of No.3 and 4 bearing after treatment

5. Conclusion

After precise analysis, there existed a second order unbalance on LP1 shaft. It can be derived from the mathematical model of flexible and symmetrical rotor vibration characteristics that there are two specific balancing methods, symmetrical for handling odd orders and symmetrical reverse for even orders. The symmetrical reverse balancing method is proved to be effective to solve the second order vibration problems in this article.

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