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Research on Modal Analysis of Rotor Shaft System of Hydro-generator

Litao Niu[1, *], Qi Sun[2, a], Chenyuan Ma[1, 3, b], Junlin Liao[1, c] and Jianguo Pu[1, d]

[1]Xi'an Thermal Power Research Institute Co. Ltd., Xi'an, China
[2]The 54th Research Institute of China Electronics Technology Group Corporation, Shijiazhuang, China
[3]Xi'an University of Technology, Xi'an 710048, China

*Corresponding author e-mail: niulitao1980@126.com, [a]469578358@QQ.com, [b]machenyuan@tpri.com.cn, [c]liaojunlin@tpri.com.cn, [d]pujianguo@tpri.com.cn

Abstract. In this paper, the influence degree of eccentricity distance and guide bearing stiffness change on the vibration characteristics of the unit under the condition of rotor mass misalignment is discussed respectively, and the modal characteristics of the unit are studied for the problem of generator mass eccentricity. First of all, the guide bearing is parameterized, the whole axle system modeling is calculated by FEM modal analysis method, the results of the analysis obtain the first 10th order critical frequency and vibration mode analysis results, further determine the danger point of displacement deformation concentration. Comparing the vibration type, critical frequency and center vibration trajectory of the guide bearing under different constraint conditions, the characteristics of the unit vibration caused by the rotor quality is: the axis tile gap is gradually enlarged due to the eccentric rotation of the rotor of the unit, and the vibration characteristics of the main frequency in the unit vibration signal are unchanged. The results are confirmed by field inspection data.

1. Introduction

Modal analysis is a method to study the dynamic characteristics of structures. Mode refers to the natural vibration characteristics of the mechanical structure [1]. Each mode has a specific natural frequency, damping ratio, and mode shape. The process of analyzing these modal parameters is called modal analysis. Hydropower unit belongs to large rotating machinery, which has the characteristics of low speed, large volume and large moment of inertia. The eccentric operation of the generator rotor will inevitably cause the vibration and swing value of the unit to rise. In serious cases, it will cause the structural connection to be loose, cracked, deformed, dropped, displaced, overheated, worn, damaged surface and other problems. If not intervened in time, it will also lead to dangerous accidents such as bore cleaning. [2-4]

In recent years, many scholars have carried out in-depth research on this problem. Li [5] used Riccati transfer matrix method and Wilson numerical integration method with good stability to establish the calculation model of the nonlinear transient response of the main shaft system of the hydraulic turbine generator unit. Considering the nonlinear oil film force and the unbalanced magnetic pull nonlinearity of the guide bearing, the influence of each factor on the main shaft system of the unit is analyzed. Yang [6] established a mechanical model of the transverse vibration of a single disk rotor, and studies the influence of gyroscopic moment, electromagnetic tension stiffness and other factors on the stability of the Three Gorges hydroelectric generating unit. The dynamic stiffness curve of the shafting is obtained. The reason why the stiffness of the guide bearing has an important influence on the stability of the unit is revealed based on the actual measurement data. Xiao [7] used ANSYS software and MATLAB software to compile an interface program that takes into account the system's continuous mass, moment of inertia, and gyro effect, and calculates and analyzes the vibration problems of the unit.

The above method does not analyze the vibration characteristics of mass eccentricity. In this paper, based on FEM analysis software, the stiffness coefficient and eccentricity of guide bearing are calculated and solved in batches. The calculation method not only overcomes the tedious and inflexible disadvantages of traditional matrix method, but also can more intuitively reflect the vibration characteristics of the model.

2. The hardware scheme FEM software modeling

FEM model calculation is divided into five parts: modeling, meshing, given boundary conditions, debugging and calculation, and results post-processing. The modeling work is imported into three-dimensional software. The meshing is divided by the default size of the system, and the given and post-processing parts of boundary conditions are the focus. The model is built based on the real machine parameters, the material density, Young's modulus and other physical parameters of each part are given respectively. According to the material, the generator model divides the grids of each part and gives the acceleration of gravity. Local coordinate system is established at guide bearing section. The plane geometric center of the above guide bearing and water guide bearing is the center of the circle. The spring element is constructed by picking up the nodes in six characteristic directions evenly distributed around. The guide bearing is composed of 14 units, and the whole machine model is composed of 187 units. The bearing base and frame are simplified as rigid fixed bodies, and the full constraint boundary conditions are applied to the degrees of freedom at the end of the spring. The grid division results of the shafting are shown in Figure 1.

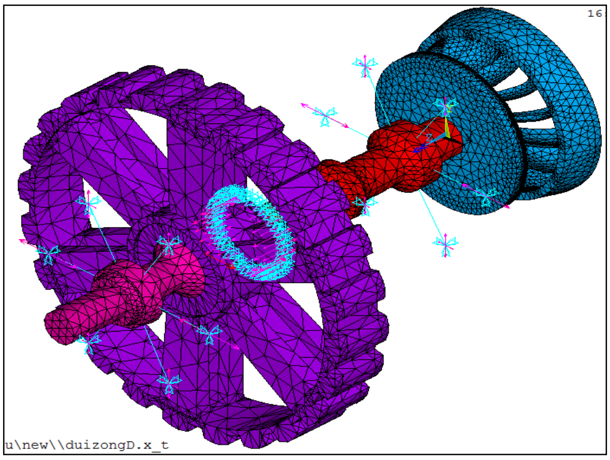

Figure 1. Shaft model mesh figure.

For any dynamic model, some of the elements of the system matrix can be accurately determined, while some can only be estimated or roughly calculated, especially for the sliding bearing part itself is a nonlinear system with temperature change [8]. In this paper, the real machine size model in FEM is used to simulate the actual fit conditions of block tile by changing the elastic modulus and damping coefficient. The unit parameters are shown in Table 1.

Table 1. Hydro-generator model parameter table.

Serial number	Name	numerical value
1	Generator rotor diameter (m)	12.00
2	Rated speed of generator (r/min)	107.9
3	Number of the upper guide bearing shells (block)	6
4	Number of water guide bearing shells (block)	6
5	The layout of generator set	Vertical half umbrella
6	Turbine model	HL220-LJ-550
7	Large shaft material	20SiMn
8	rotor material	0Cr13Ni15Mn
9	Guide bearing length (m)	0.45
10	Model of lubricating oil	46 turbine oil
11	Rated lubricating oil temperature (°C)	50

3. Comparative calculation of the influence of eccentricity of rotor and clearance of guide bearing bush on modal analysis

In order to find out the influence degree of the eccentricity distance of the rotor on the modal vibration mode, the five models of the eccentricity distance of the rotor are selected as 0.2cm, 0.4cm, 0.6cm, 0.8cm and 1cm respectively for calculation, and compared with the model of the complete alignment of the rotor; in order to further explore the influence degree of the change of the bearing parameters on the vibration characteristics of the shafting, the shafting of the unit must be considered as a linear system. The guide bearing is simplified as a spring damping system, in which multiple linearized stiffnesses and damping coefficients are used to replace the boundary conditions of the bearing. The gap of the guide bearing bush under the calculation condition 1 is 0.02CM in the FEM model, the guide bearing parameters are: $K_{xx} = 2.6\times10^{9} N/m$, $C_{xx} = 2.27\times10^{8} N{\bullet}S/m$; The clearance of guide bearing bush under calculation condition 2 is 0.03cm, and the parameters of guide bearing are as follows: $K_{xx} = 1.82\times10^{9} N/m$, $C_{xx} = 2.04\times10^{8} N{\bullet}S/m$.

3.1. Calculation of critical frequency for different eccentricity distance of the rotor

The mode solver in FEM software is used to calculate the critical frequency and the vibration pattern of each stage. In post-processing, read and store by order, observe and record the first 10 frequencies. See Table 2 for the calculation results.

Table 2. Different eccentricity and critical frequency value table.

Serial number	Eccentricity 1cm	eccentricity 0.8cm	eccentricity 0.6cm	eccentricity 0.4cm	eccentricity 0.2cm	Perfect alignment	average value	Maximum relative deviation (%)
1	0.04	0.04	0.04	0.04	0.04	0.04	0.04	0.00
2	8.48	8.48	8.48	8.48	8.48	8.48	8.48	0.04
3	8.89	8.89	8.89	8.89	8.89	8.89	8.89	0.01
4	8.98	8.98	8.98	8.98	8.98	8.98	8.98	0.01
5	12.38	12.37	12.38	12.39	12.38	12.38	12.38	0.01
6	12.40	12.41	12.40	12.41	12.40	12.40	12.40	0.01
7	22.21	22.21	22.22	22.20	22.18	22.21	22.21	0.02
8	22.30	22.33	22.30	22.32	22.30	22.27	22.30	0.04
9	30.70	30.70	30.70	30.70	30.70	30.70	30.70	0.13
10	34.23	34.24	34.24	34.23	34.23	34.22	34.23	0.00

From the statistical data, it can be concluded that the eccentricity of rotor mass within a certain range will not change the natural frequency of the generator shaft system when the bearing bush clearance is constant, and only the fault characterization caused by the rotor mass misalignment will not have obvious performance in the monitored system characteristic frequency.

3.2. Calculation of critical frequency for different guide bearing parameters
Select the basic vibration model with rotor eccentricity of 1cm, and simulate the actual situation of bearing clearance expansion by changing the bearing parameters. Carry out the modal analysis and calculation of the rotor system with two kinds of the guide bearing clearance. See Table 3 for the value of critical frequency.

Table 3. Comparison table of critical frequencies under different guide bearing clearances (rotor eccentricity 1cm).

Order number	Guide bush clearance 0.02cm	Guide bush clearance 0.03cm	Relative deviation (%)
1	0.039	0.038	0.40
2	8.483	8.438	0.54
3	8.893	8.652	2.71
4	8.983	8.785	2.20
5	12.380	12.066	2.56
6	12.420	12.087	2.54
7	22.214	21.970	1.10
8	22.270	22.093	-0.79
9	30.702	30.702	0.00
10	34.227	34.236	0.03

The critical frequency of shafting decreases slightly with the increase of bearing clearance. The maximum difference between the two calculation results is 2.71%, the influence of the reduction of the bearing stiffness coefficient on the frequency characteristics of the shafting can also be ignored.

3.3. Comparison of vibration modes of the different guide bearing parameters
In the post-processing module of FEM, the cloud map of displacement change is extracted, and the vibration modes of each stage and the maximum points of displacement change are observed. Comparing the two working conditions, it can be seen that after the parameter change of guide bearing, the vibration modes corresponding to the critical frequency of each stage are slightly different. The dangerous points are mainly at the outer edge of the rotor and the lower ring of the runner, and the vibration modes are shown in Fig. 2.

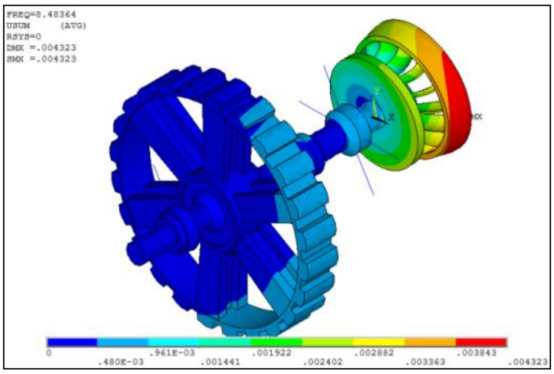

Figure 2. Eccentric 50%e second-order vibration pattern (case 1).

According to the statistics of the maximum modal displacement position and displacement value of shafting corresponding to each critical frequency, the maximum displacement relative deviation of shafting corresponding to the 5th and 6th critical frequency is 11.09%, and the data comparison is shown in Table 4.

Table 4. Maximum displacement position and value table under different guide bearing clearance.

Order number	Mode location	Guide bush clearance 0.03cm	Guide bush clearance 0.02cm	Relative deviation (%)
1	Rotor outer edge	1.694	1.693	0.06
2	Runner rim ±X	4.320	4.323	0.07
3	Runner rim ±X	2.917	3.097	5.81
4	Runner rim ±Y	3.947	3.733	5.73
5	Runner rim ±X	2.385	2.147	11.09
6	Runner rim ±Y	2.456	2.208	11.23
7	Rotor outer edge ±Y	2.615	2.638	0.49
8	Rotor outer edge ±X	2.588	3.595	0.27
9	Runner upper crown lower ring	3.273	3.276	0.09
10	Rotor outer edge	1.927	1.928	0.05

3.4. Comparison of vibration mode trace offsets of the different guide bearing parameters

Through calculation, we can not only get the modal shape of shafting, but also analyze a certain element separately. Using the FEM post-processing module, mark point A as the center of the upper guide bearing and point B as the center of water guide bearing. Trace the mode locus of the bearing center point in the 10th order mode in the plane coordinate system, find the maximum vector position of the mode locus offset and mark it with a straight line. The results are shown in Fig. 3 and Fig. 4. In condition 1, the displacement of vibration mode of upper guide bearing and water guide bearing is in direct proportion to the order within the first 7 critical frequencies, the maximum displacement of upper guide bearing is 4.15×10^{-4}m, and the maximum displacement of water guide bearing is 3.55×10^{-4}m; in condition 2, the displacement of upper guide bearing and water guide bearing is in direct proportion to the order within the first 6 critical frequencies, and the maximum displacement of upper guide bearing is 4.75×10^{-4}m The maximum displacement of water guide bearing is 3.65×10^{-4}m. The change of bearing clearance has a great influence on the vibration mode deviation of the guide bearing section center point, which will also be reflected in the vibration spectrum response amplitude caused by the mass eccentricity.

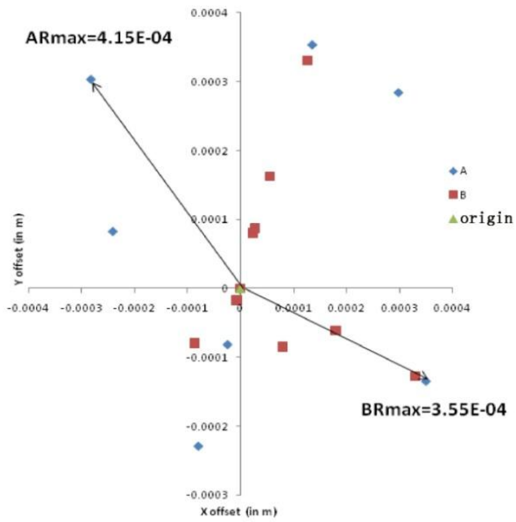

Figure 3. Guide bearing center 10th order displacement point map (condition 1).

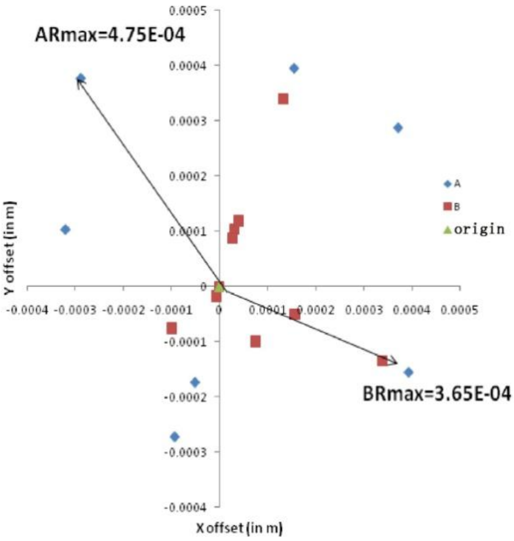

Figure 4. Guide bearing center 10th order displacement point map (condition 2).

Note: A is the center displacement point of the upper guide bearing section, and B is the center displacement point of the water guide bearing section. Mark the maximum point position deviated from the center position with a straight line.

4. Actual measurement results on site

During the start-up of unit 6 of a hydropower station, the swing of upper guide bearing and vibration value of upper frame of the unit continue to increase, the main frequency component rises from 0.04 mm to 0.12mm, and the main frequency rate of vibration is 1 time of the frequency, and the clearance of bearing bush is expanded from 0.3mm to 0.4mm through shutdown inspection. It is judged that the abnormal operation of the unit is due to the serious unbalance of the rotor quality, which eventually results in the expansion of the clearance between the upper guide bearing shells. After the treatment of the dynamic balance test, it recovered to normal, and the test results were consistent with the simulation results. The swing spectrum of the upper guide bearing during startup is the shown in Figure 5, and that before the shutdown is the shown in Figure 6.

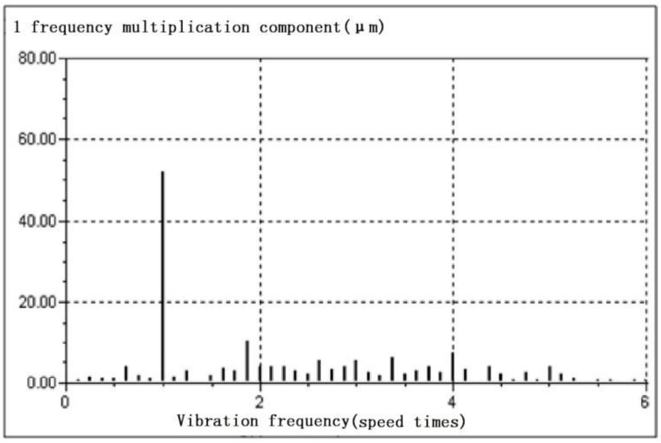

Figure 5. Frequency spectrum of swing of upper guide bearing at the beginning of unit startup (peak value in time domain 0.067cm).

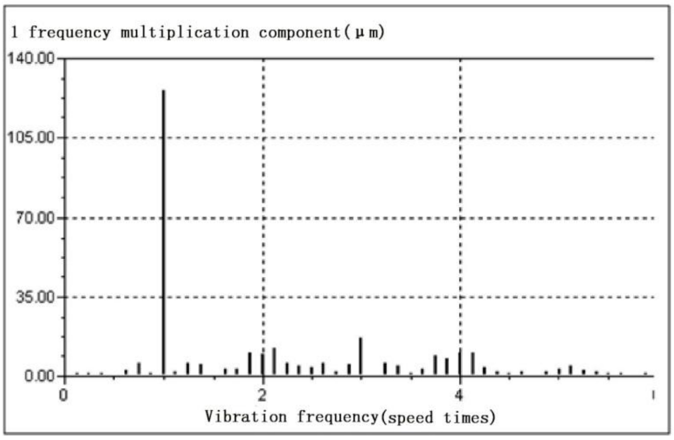

Figure 6. Swing spectrum of upper guide bearing after 20 minutes of operation (peak value in time domain 0.138cm).

5. Conclusion

In this paper, the parameter modelling is built according to the size of the real machine, the modal analysis and calculation are carried out for 5 kinds of rotor eccentricity distance and 2 kinds of bearing clearance conditions by using the finite element software, the characteristics of the track of vibration mode in the center of guide bearing journal are analyzed, the method of using the FEM theory as the modal analysis calculation mode is established, Using this method, the vibration characteristics of the model with the change of rotor eccentricity and guide bearing parameters are calculated respectively. The critical frequency and mode shape of each stage are extracted by the post-processing module and compared with the test results, the following conclusions are obtained

(1) By analyzing the characteristic modes of each order, it is found that the change of the eccentricity distance of the rotor and the clearance of the guide bearing bush has little influence on the critical frequency and the shafting vibration mode of each stage. the dangerous points of shafting vibration mode are mainly at the outer edge of rotor and the lower ring of runner, this part is easy to cause destructive problems such as stress concentration.

(2) Serious quality eccentricity of the generator will not only cause abnormal vibration of the unit operation and increase the value of the swing of the guide bearing shaft collar, but also reduce the support stiffness of the guide bearing. Long time operation will damage the support structure, which has a strong risk.

(3) The abnormal operation caused by the quality eccentricity of the generator rotor can be significantly reflected in the frequency domain signals monitored. With the increase of eccentricity, the characteristic frequency band of the main frequency in the monitored signal is still the unit frequency conversion, but the amplitude component of the main frequency signal will continue to increase. The experimental characteristics are consistent with the results of numerical simulation.

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