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Vibration Analysis of the Steam Turbine Shafting caused by Steam Flow

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Abstract

This thesis analyzes vibration test signal of TUOKETUO Power Plant 600 MW steam turbine unit through vibration monitoring and signal analysis on the basis of TN8000 Steam Turbine Vibration Analysis Software. Fault characteristic which is raised by Steam Flow Excitations is reproduced by acceleration constant speed test and load test. Steam flow mechanism of excitation caused by vibration fault and fault-sensitive parameter are analyzed, measures reducing unit vibration has been proposed in line with the conditions. Test results show that: the vibration caused by the vapor stream excitation occurs mainly in the high-pressure rotor steam inlet end. However, the vibration signal, which occupies a large percentage of the rotor frequency of a first critical speed are sensitive to the changes in the load. Problems can be early identified; the maintenance program and maintenance means can be determined in the plant operation through analysis of vibration mechanism and sign. Security and reliability of the steam turbine running should be guaranteed.

Keywords: *steam turbine unit, shafting vibration, signal analysis, steam flow excited vibration*

1. Introduction

The turbine rotor is a rather complicated structure, possessing a continuous plaster of mass distribution. Due to such reasons as the manufacture, installation and operation, all may make bending vibration and torsional vibration caused by the shaft vibration. Vibration problem at No.1 and 2 bearing of No. 6 power unit in TUOKETUO Power Plant occurs during turbine operation. Shaft vibration value fluctuates from 40μm～100μm when the load is more than 550 MW (Figure 1). Waveform of accident can be seen obviously, the reason why shaft vibration occur large fluctuations lies in the violent change of low-frequency components of the vibration signal at an approximately 28.15Hz

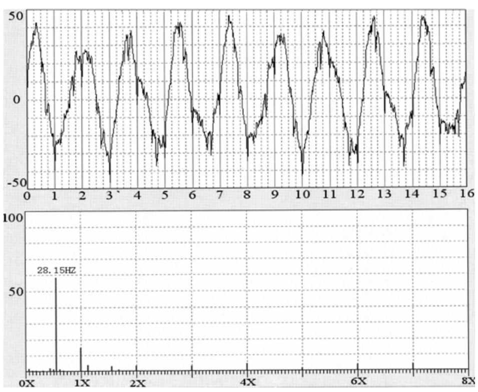

Figure 1(a). Time Domain Waveform and Spectrum of 1X Shaft Vibration

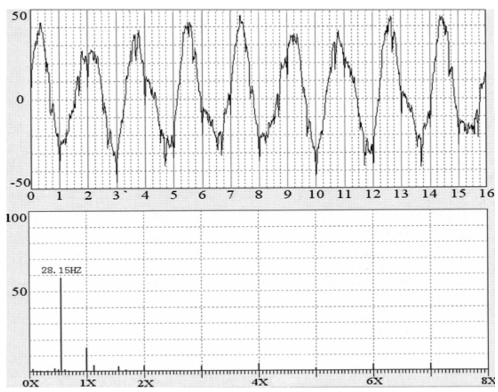

Figure 1(b). Time Domain Waveform and Spectrum of 1Y Shaft Vibration

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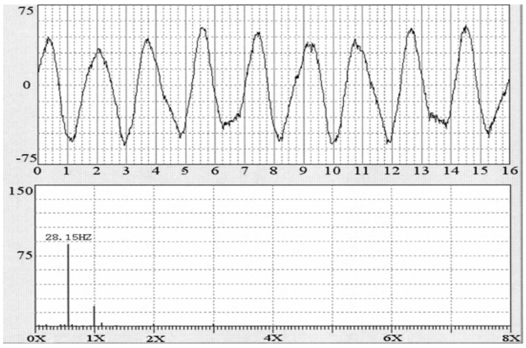

Figure 1(c). Time Domain Waveform and Spectrum of 1X Shaft Vibration

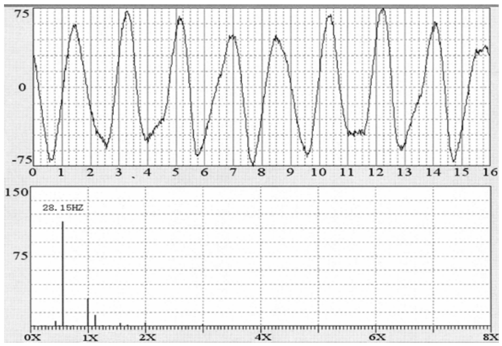

Figure 1(d). Time Domain Waveform and Spectrum of 1Y Shaft Vibration

2. The Relevant Technical Parameter of Units and Testing System

2.1. The Relevant Technical Parameter of Units

The fault unit is designed and manufactured by DONGFANG steam turbine plants, which is subcritical, impulsive, axial with triplex Four exhaust, the double back pressure condensing steam turbine and an intermediate reheat modeled N600-16.7/538/538. The high and medium pressure cylinder possesses co-cylinder, two-tier structure and the low-pressure cylinder is divided into A, B, two-cylinder. Turbine totals three whole forging solid rotor, each supported by two bearing. Of which, NO.1, 2 bearing are tilting pad bearings with six tilting pad and NO. 3, 4, 5, 6, 7, 8bearing are elliptical bearings. Thrust pad is on the back of exhaust of the intermediate pressure and working face on the side of the generator. critical speed value of steam turbine rotors are shown in Table 1 [1]. Vertical (Y) and horizontal (x) double amplitude vibration value measured in any journey of the steam turbine unit shall not be over 0.076mm. Bearing Vibration (w) peak shall not be greater than 0.05mm [1].

2.2. Testing System

The main supervision chart of steam turbine is shown in Figure 2. The monitoring chart is shown in Figure 3 due to characteristics of axis oscillation. The system is composed of: Steam Turbine Shafting, sensor, TSI instrument (MMS 6000), TN8000 system, computer, etc.

Table 1. The Critical Speed of the Turbine Rotator

Name of shaft section	First critical speed (r/min)	The second critical speed(r/min)
High and medium pressure rotator	1692	>4000
low pressure rotator A	1670	>4000
low pressure rotator B	1697	>4000
Generator rotator	933	2691

2.3. The Selection of Sensor

In order to accurately measure the real-time situation of turbine rotor vibration, the selection of the sensor should consider the following two aspects, on the one hand, the characteristics of the measured signal, on the other hand, the performance of the sensor [2]. PR6423 displacement sensor made by EPRO with range 400μm and sensitivity 8mv/μm is used for probe of shaft vibration and PR9268 speed sensor made by EPRO with range 100μm and sensitivity 28.5mv/μm for the tile cap vibration (W) through comparing the characteristics and functions of the oscillation sensor and combining the features of the turbine rotators.

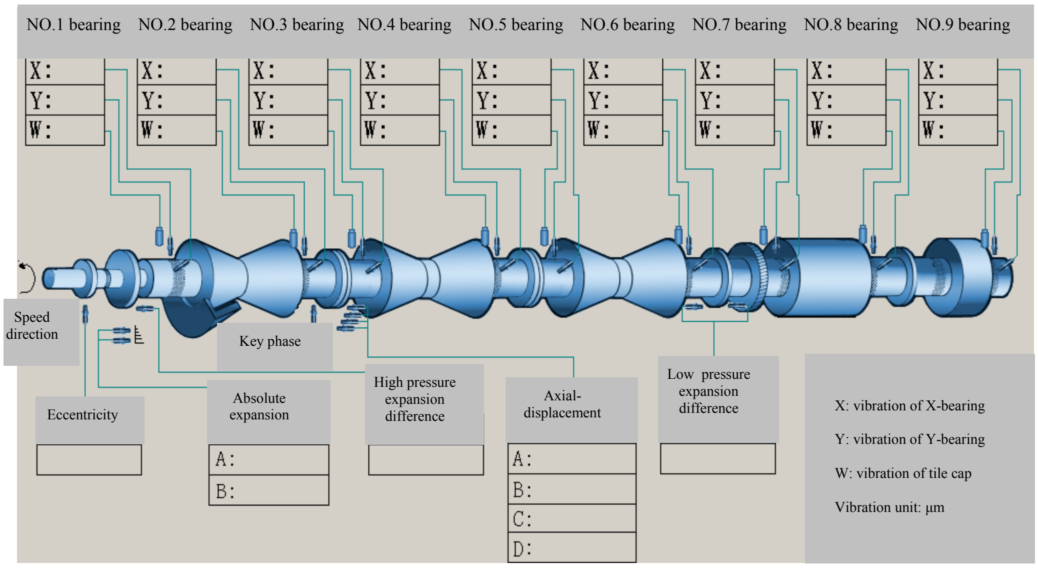

Figure 2. The Main Monitor Diagram

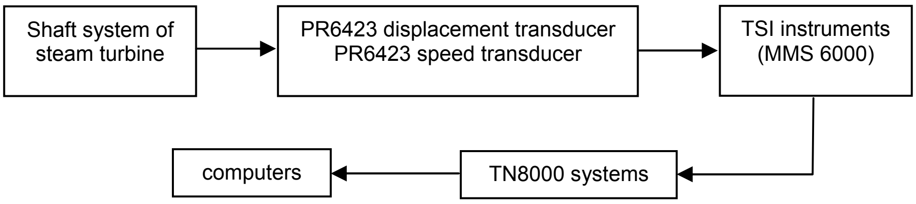

Figure 3. The Testing System

3. Test Analysis of Vibration Signals

According to the failure characteristics, the cause of the malfunction is excitation of steam flow at the first thought. Acceleration constant speed test and the load test are operated in order to verify the accuracy of the judgment and to find the source of the fault respectively.

3.1. The Acceleration Process of Start Machine and the Vibration of the Constant Speed Test

Waveform of direction shaft vibration when Unit starts up and accelerates from No. 1 to No. 8 watts X (horizontal), Y (vertical) is shown in Figure 4 with the abscissa in the figure being the rotator speed, the vertical axis being the Shaft vibration amplitude value and phase. From the Shaft vibration waveform, the entire acceleration process can be seen from constant speed to the rated speed 3000r/min with each vibration being small. When passing through their critical speed (each rotor order critical speed is shown in Table1), the peak value is small (the critical speed at a vibration amplitude are shown in Table 2) within the allowable range, indicating that each tile vibration properly.

Table. 2 Vibration Peak Value of Each Bearing when Raising Speed Over a Critical Value

Shaft vibration	1X	1Y	2X	2Y	3X	3Y	4X	4Y	5X
μm	27	31	24	32	20	39	22	32	18

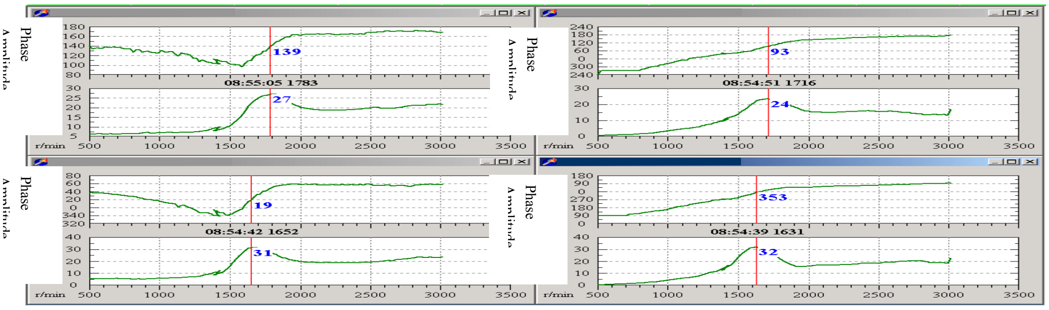

Figure 4(a). Waveform of shaft vibration of No.1 and 2 along x-y direction

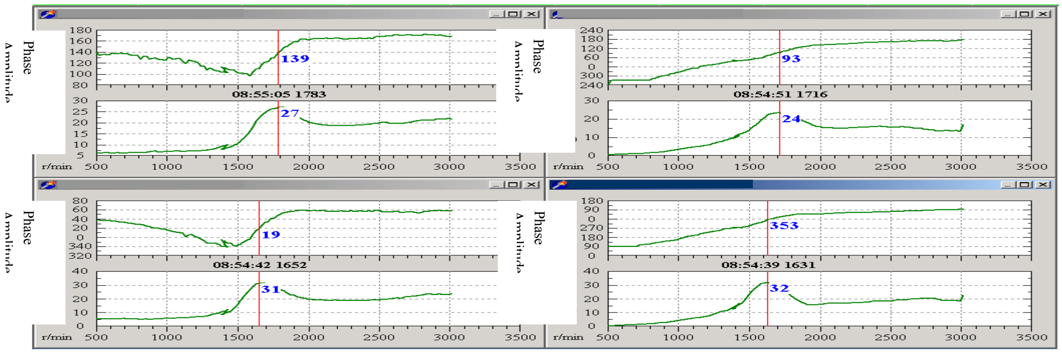

Figure 4(b). Waveform of shaft vibration of No.3 and 4 along x-y direction

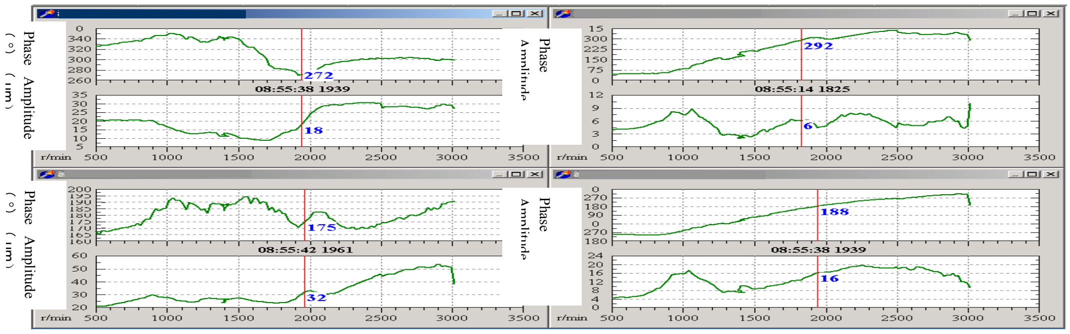

Figure 4(c). Waveform of shaft vibration of No.5 and 6 along x-y direction

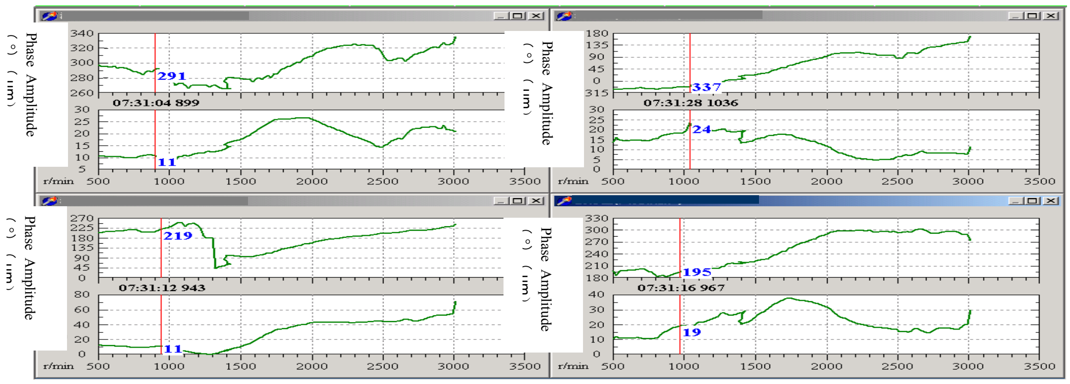

Figure 4(d). Waveform of shaft vibration of No.5 and 6 along x-y direction

3.2 Vibration with Load

With a load, watt-axis vibration has some changes to some degree. One of the most interesting things is NO. 2 and NO. 3-watt of the high and medium-pressure rotor as well as NO. 7 and NO. 8-watt vibration of the rotor of generator. There is a big fluctuation in the first and second ZWZ vibration when full capacity is 600 MW. There is a much more increase in No. 7, No. 8 ZWZ vibration than that without load. Given the speed and load watt vibration-pass frequency data are shown in Table 3. we will analyze in the following paragraphs.

Table. 3 Vibration Peak Value of Each Bearing when Raising Speed Over a Critical Value

Shaft vibration	Constant speed （3000 r/min）	100MW	300MW	450MW	600MW
1x	40	38	40	40	24~126
1y	40	34	40	36	22~132
2x	25	27	28	30	30~124
2y	30	30	35	36	40~149
3x	43	46	44	48	58
3y	52	54	43	37	55
4x	28	28	32	36	34
4y	35	32	39	47	39
5x	43	33	36	34	46
5y	50	39	36	30	49
6x	16	18	25	28	28
6y	20	22	38	38	33
7x	32	32	30	32	61
7y	57	58	59	66	76
8x	12	18	22	22	28
8y	40	45	47	51	57
9x	41	45	58	56	60
9y	43	50	55	56	64

3.2.1. The Analysis of the Vibration for the High and Medium-pressure Rotor

When No. 1 and No. 2 the ZWZ vibrations are 600 MW, there are large fluctuations. Take 2Y as an example, the minimum value is only about 40μm and the maximum value of more than 120μm. The vibration spectrums of three typical situations are shown from Figure 5 to Figure 7.

Figure 5 is a vibration spectrum without low-frequent components, of which the 1X vibration-pass frequency value is about 24μm and one octave amplitude of about 16μm. The signal amplitude of 28.13 Hz is about 2μm. Pass-frequent value of 1Y vibration is about 22μm, one octave amplitude of 14μm and signal amplitude of 28.13Hz is approximately 3μm, pass-frequent value of 2X vibration is about 30μm, one octave amplitude of 21μm and signal amplitude of 28.13Hz is approximately 4μm. Pass-frequent value of 2Y vibration is about 40μm, one octave amplitude of 33μm and signal amplitude of 28.13Hz is approximately 4μm, and the NO. 1 and NO. 2 watt vibration focus on octave ingredients.

Figure 6 is a vibration spectrum with evident low-frequency components of which the 1X vibration-pass frequency value is about 28μm and one octave amplitude of about 16μm. The signal amplitude of 28.13Hz is about 11μm pass-frequent value of 1Y vibration is about 35μm, one octave amplitude of 15μm and signal amplitude of 28.13Hz is approximately 16μm, pass-frequent value of 2X vibration is about 35μm, one octave amplitude of 21μm and signal amplitude of 28.13Hz is approximately 16μm pass-frequent value of 2Y vibration is about 60μm, one octave amplitude of 33μm and signal amplitude of 28.13Hz is approximately 21μm. At this time, the low-frequency components are close to or over one octave component.

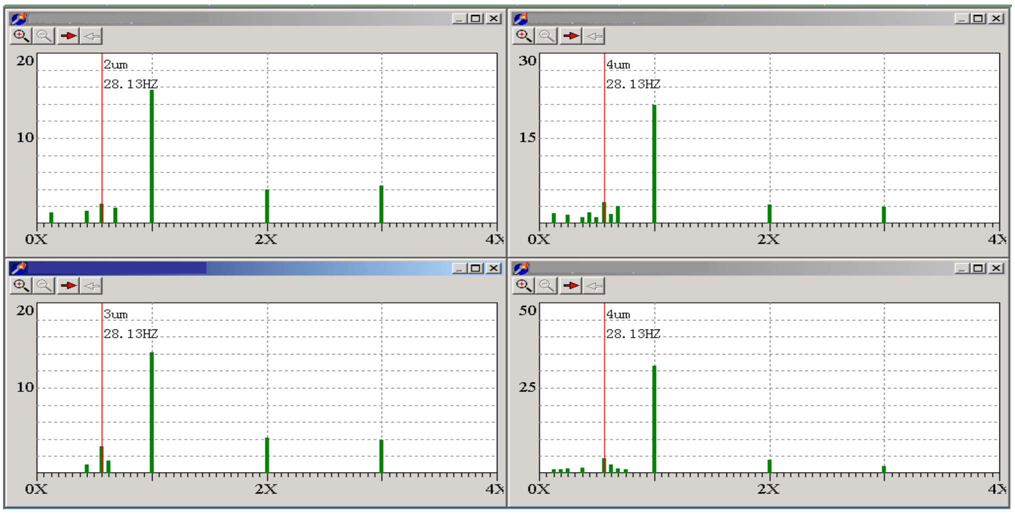

Figure 5. Vibration spectrum diagram of No.1 and 2 bearing when low frequency component is small

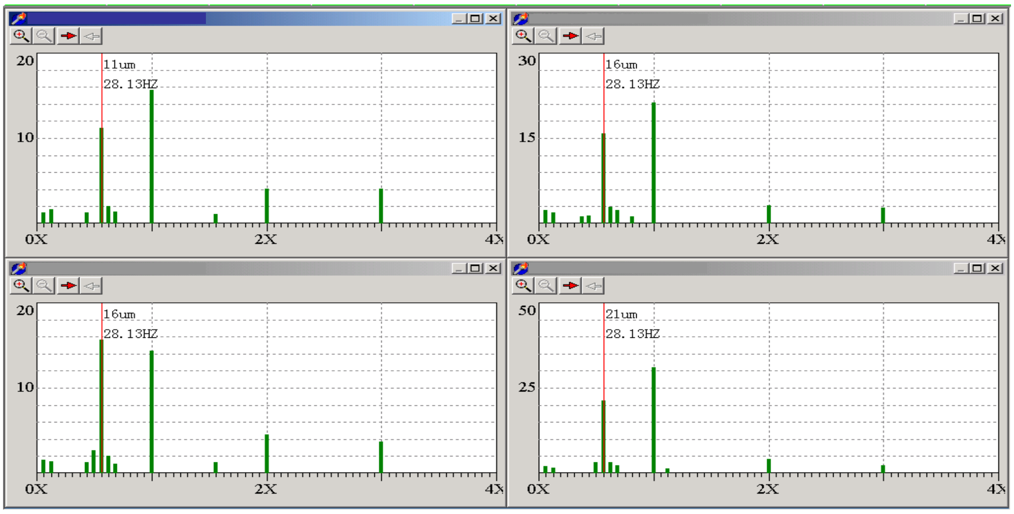

Figure 6. Vibration spectrum diagram of No.1 and 2 bearing when low frequency component is obvious

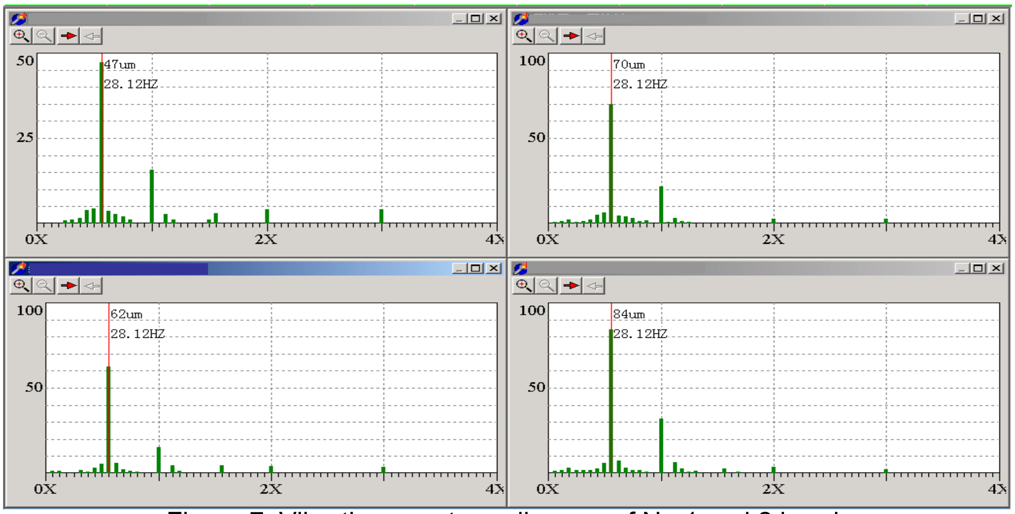

Figure 7. Vibration spectrum diagram of No.1 and 2 bearing when low frequency component is great

Figure 7 is a vibration spectrum with evident low-frequency components. of which the 1X vibration-pass frequency value is about 70μm and one octave amplitude of about 16μm. The signal amplitude of 28.13H_Z is about 47μm. Pass-frequent value of 1Y vibration is about 90μm, one octave amplitude of 17μm and signal amplitude of 28.13Hz is approximately 62μm, pass-frequent value of 2X vibration is about 95μm, one octave amplitude of 22μm and signal amplitude of 28.13Hz is approximately 70μm. Pass-frequent value of 2Y vibration is about 120μm, one octave amplitude of 32μm and signal amplitude of 28.13Hz is approximately 84μm. At this time, the low-frequency components ARE far much more than one octave component.

Through spectral analysis, we can find that the reason why the 1st and the 2nd tile of X, Y-direction shaft vibration occurs large fluctuations lies in violent vibration of an approximately 28Hz low frequency components. As you can see, however 1, 2 corrugated frequency value changes, the 1 octave amplitude and phase are very stable with few changes (Table 4). The axis track of the vibration (Figure 8) is in disorder, which is caused by the complexity of the frequency component of the vibration signal [3, 4], precession direction is positive.

Table 4. Amplitude and Phase of 1 Octave Vibration Signal of No. 1 and 2 Bearing

Test point	1x	1y	2x	2y
Amplitude(μm)	16	15	21	33
Phase (°)	161	49	196	81

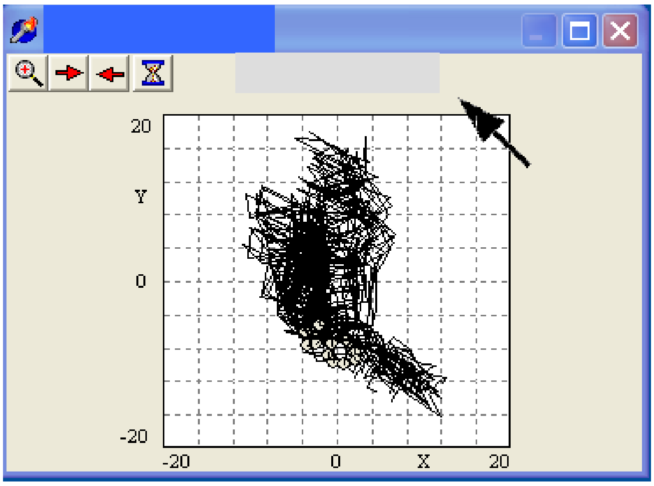

Figure 8. Vibration Axis path diagram of No.1 bearing

The basic characteristics of malfunction are as follows from the above analysis:
(1) Malfunction occurs to the inlet steam end of high pressure turbine, because vibration amplitude fluctuations very instantly, there will be a severe vibration.
(2) When P is no less than 500MW, vibration is intensified. Once there is a threshold load value, this malfunction occurs.
(3) In the vibration signal, the first critical speed (frequency) of the signal of the malfunction rotor is the main component of and amplitude occurs volatility, accompanied by a large number of low-frequency component and high-order harmonic components.
(4) Failure occurs, even if the frequency of vibration fluctuates volatility, but the basic frequency in vibration signal remains stable.
(5) The frequency components of the vibration malfunction are abundant with volatile vibration amplitude; therefore, orbit track is in disorder in positive precession.
(6) The vibrations have a fine reproducibility.

3.2.2. The Analysis for Reason of Vibration in the High and Medium-pressure Rotators

According to analysis of the mechanism of rotor dynamics , unit vibration phenomenon is closely related with critical speed frequency of the high pressure rotor, which means the vibration frequency does not match with the rotor frequency, but complies with the critical speed of the rotor and contains a low-frequency harmonic [5, 6, 7, 8, 9]. There are other factors causing instable sports vibration frequency 28.13Hz is close to critical speed 1692r/min of the high pressure rotor by comparing fault unit in low-frequency components of the high pressure rotor there are not obvious low-frequency components No. 1 and No. 2ZWZ vibration before 450pw by tracking the process of the liter load. A significant low frequency component occurs after 500kw, up to 10μm, a significant increase happens in the low-frequency component, up to 20μm, the value of which can not be overlooked. Values in the low-frequency vibration and occurrence frequencies have greatly increased in the full load 600MW. The vibration phenomenon is difficult to define as a random friction forced vibration with exclusion of low-frequency oscillation caused by motion of oil whirl (oil whirl vibration signal frequency is 0.5 times slower than unit operation frequency). All of this suggests that instable vibration of the high and medium-pressure rotor belongs to the excitation of the steam flow. Steam Flow Excitations belongs to a self-excited vibration (or negative damping vibration). Damping generated movement of vibration itself exacerbate the movement [5, 9, 10] rather than stop it. Steam Flow Excitations generally occurs to high-pressure (or high and medium-pressure) rotors of the turbine in high power under high load, the main features of which is that vibration is sensitive to load. Sudden vibration has a threshold load, which stimulates the excitation of the steam flow when vibration exceeds the load. While the load lowers to some certain values, vibration is resumed with good repeatability. The main reason causing the malfunction is steam flow excitation through the analysis of vibrant signals and features of steam flow excitation.

3.2.3. The Analysis of Mechanism of the Steam Flow Excitation

It must have two conditions to result in steam flow excitation, one is uneven radial distribution of the pressure within seal gap of the high pressure rotor; the other is imbalance in rotor torque radial [5].The type of excitation force:

The exciting force caused by weekly changes of the steam seal chamber pressure. Vapor pressure of the high-pressure rotator of the large turbine is high with large amounts of leakage in the steam seal of blades. When the temperature is constant, the pressure of the shaft seal chamber is proportional to the flow rate of the chamber. The radial clearance (Figure 9: δ1 = δ2) of the front and rear teeth is equal when rotor is in the rest position. The steam inflow is equal to the outflow without circulation in the chamber. On the premise of the outlet gap δ2 less than the inlet gap δ1, When the rotor radial displaces (which is the prerequisite of self-excited vibration), the relative change in exports teeth flow area is larger than that in the entrance of the Tooth flow area if the radial displacement of the rotor makes the shaft seal gap increase, steam outflows outnumber inflows with reduced pressure in the shaft seal chamber. In the other way round, the results are on the contrary. Rotor pressure and displacement are not synchronized due to the inertia of the rotor, i.e. when the rotor is displaced upward to the highest position, the upper gap is the smallest. However the pressure in the chamber room is not the highest.

When the rotator goes back to the vicinity of the rest position from the upper, the upper chamber pressure is the highest. Thus, top and bottom of rotor will form a pressure differential urging the rotor continuing its downward movement from the rest position so that the rotor can not stay in a stationary position .this inertia hysteresis effect makes the pressure in the lower chamber room increase when the rotor continues its downward movement. And this vapor pressure difference will enable the rotor to produce a displacement and forms a eddy due whirl. Because it is caused by the vapor system, therefore termed as the steam flow excitation.

The stress analysis of rotor eddy caused by pressure changes in shaft seal chamber is shown in Figure 10. pressure differential in the chamber makes the angle 90 between the steam force F_1 and rotor elastic restoring force F_2, both of which enables the rotor displace. ($0<\Phi<90$) At this time, the steam force F_1 can be decomposed into a rotor elastic restoring force with the same direction of the force F_{11} and another opposite to the direction of the damping force F_3, which is F_{12} functioning as the negative damping. When F_{12} is greater than F_3, the rotor will generate self-excited vibration (F_4 is centrifugal force) [11].

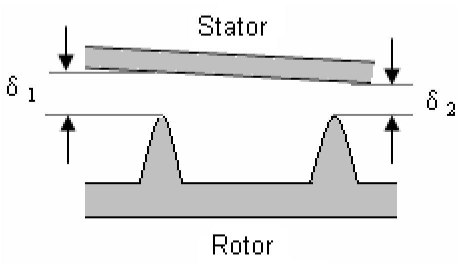

Figure 9. Rotor eddy caused by pressure changes in shaft seal

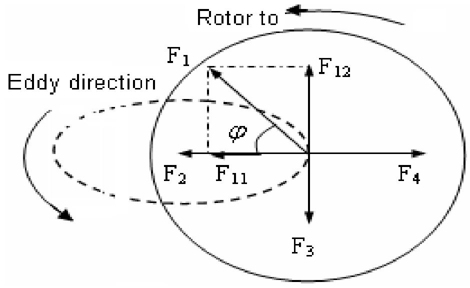

Figure 10. Pressure analysis in shaft seal

The exciting force caused by imbalance of the torque of the rotor:

Due to unit installation, cylinder deviation in operation and radial displacement of the rotor, rotor will deviate comparing with cylinder, causing uneven radial distribution when steam acts on the rotor and the rotor whirl. The decomposition of the eddy momentum is shown in Figure 11. Because the loss is smaller on the radial gap side, the power acted by steam is lager than that on the larger gap side. Whereby, torque formed by rotor is unbalanced. The unbalanced torque force can be divided into force causing rotation of the rotor in the circumference and unbalanced force Ft rotating with the rotor and opposite to the damping force. Ft acts on the rotor center and functions as a negative damping. When the force is greater than the damping force of the system, the rotor will generate self-excited vibration [12]. The unbalanced force is:

$$F_t=F_{t1}-F_{t2} \tag{1}$$

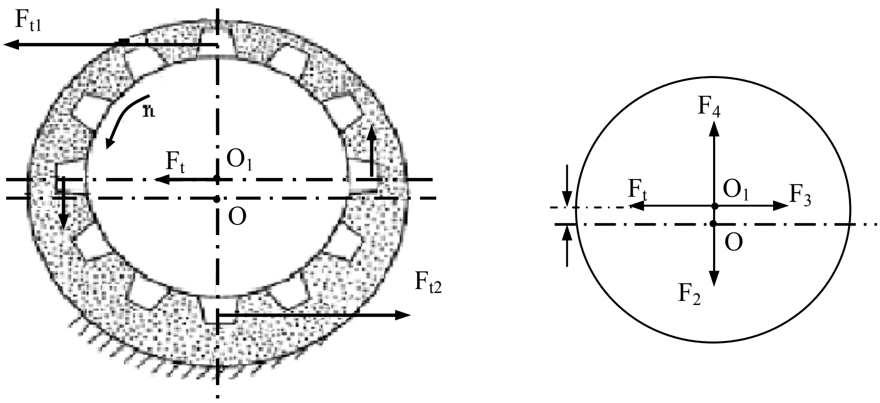

Figure 11. Rotor Eddy caused by Torque Imbalance of the Rotor

As shown in Figure 12, Ft is projected on the axis with x, y as the centre position of the wheel.

$$\left.\begin{array}{l} F_{tx}=-K_1x \\ F_{ty}=K_1y \end{array}\right\} \tag{2}$$

K_1 (unit:N/m) is the coefficient of gap exciting force, the calculation of which is:

$$K_1 = \frac{9549 P_i b}{D_i H_i n} \tag{3}$$

Formula:
D_i: impeller section circular diameter, m;
H_i: leaves, m;
β: Gap excitation factor;
P_i: class power, KW;
n: a rotor speed, r/min.

As Equation 3 shows that K_1 stage power P_i is proportional to, disproportional to the blade height H_i when certain speed n, as a result, power is increased, K_1 increases, the rotor is easy to instability. There is not vibration malfunction in the units when the load power stays low. Only on the condition of power of the generating units being greater 450MW, can the vibrant malfunction occur. Furthermore, the unstable condition of units is more serious with a further increase of the load of the random group. The high pressure rotor steam in the first order of bending natural frequency will be subjected to a large amplitude when the total in a cut perpendicular to the eccentric direction of the high-pressure rotor between the exciting force of the steam seal and exciting force of unbalanced torque exceeds the damping force of the bearing oil film.

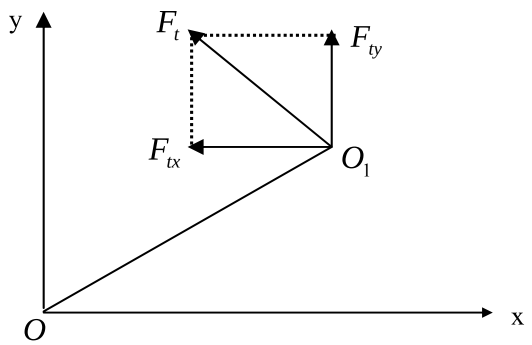

Figure 12. The Unbalanced Force

4. Conclusion

Steam flow excitation processing program of the high and medium-pressure torque is closely linked with its vibrant mechanism. According to analysis for the situation of rotor vibration and mechanism of the flow excitation as well as in-depth research of the vibration data and vibration graph, reducing excitation force and increasing the system damping are two methods to eliminate the vibration. The measures reducing steam flow excitation are as follows:
(1) The adjustment of the cylinder and the center of the rotor to avoid the evident displacement of the rotor and the cylinder center in operation.
(2) Change open procedures of the governor valve, thus avoiding unbalanced torque on the rotor caused by radial displacement of unilateral steam force. The adjustment methods can be adopted: inlet steam way is changed into a single valve, reducing steam parameters and changing the valve sequence;
(3) increasing bush damping. The steam flow excitation belongs to the negative damping vibration which plays a positive role in improving its damping suppression toward low-frequency vibration. Measures can be taken: improving lubricant temperature, reducing clearance of the top bearing, adjusting coordinates of the bearing, and increasing the length of the bush, adopting lubricant with, a high viscosity ,etc.;
(4) Bush with fine stability;
(5) To improve the critical speed of the rotor.

Acknowledgements

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