Strategy and Technique of High Efficiency Balancing in Field for Turbo-Generator Units with Large Capacity

Wenfang Cai\textsuperscript{1*}, Songyuan Lu\textsuperscript{2}, Zhengfeng Wu\textsuperscript{3}, Guangyao Ying\textsuperscript{1} and Wenjian Wu\textsuperscript{1}

\textsuperscript{1} State Grid Zhejiang Electric Power Research Institute. Hua Dian Nong 1, Gongshu District, Hangzhou 310014, Zhejiang, China
\textsuperscript{2} School of Energy and Environment. Southeast University Nanjing, Jiangsu, China
\textsuperscript{3} Huadian Electric Power Research Institute Co., Ltd Hangzhou, Zhejiang, China
Email: cwf1777@126.com

Abstract. This paper aims at the high efficiency of field balancing for turbo-generator with large capacity currently, and introduces the strategies and key techniques of the rotor system balancing with practical cases of power plant in field. The acquisition, analysis and former processing of the original vibration data for balance calculation are included. Furthermore, they involve complete measuring points and conditions, accurate judgment for the types of unbalance exciting force and selection of stable vibration data, all these could reduce the blindness of balancing effectively. The strategies and techniques also contain the determination for axial plane of unbalance by the modal method, then the optimal steps and the plane of adding weight are chosen during the implementation of balancing. Besides, this paper also introduces the analysis and selection of influence coefficients and the phase of trial weight, these can help determine the final correction weight accurately in order to guarantee the balancing process prompt and efficient. Meanwhile, the restriction of practical location for adding weight and construction period of maintenance and production for the units should be considered during the high efficiency balancing in field. These strategies and techniques of high efficiency balancing have practical application value in promoting the technology of field balancing for turbo-generator units with large capacity.

Keywords. Large capacity turbo-generator, balancing, strategy and technique.

1. Introduction
Unbalance is the major factor that can lead to high vibration of turbo-generator units. If the inertia axis of the rotor deviates from its spin axis, the unbalance force will cause the vibration of the rotor, bearings and foundation, and lead to fatigue damage of rotating components, rubbing between rotor and static parts, loosing of connection parts, etc. Moreover, it can result in bearing damage and even the catastrophic accidents that the whole rotor system would fracture under some certain conditions.

Unbalance may come from the manufacturing process of the rotor, maybe from the operation and maintenance of units. High-speed balancing in field is a conventional method to reduce the 1X vibration (i.e. the vibration at 1X running speed) caused by the unbalance effectively [1]. The field balancing involves a series of skills about vibration measurement, diagnosis, balance theory, balance calculation, etc. There are many balancing schemes for a certain unit, but the process and effect may differ greatly among different methods. These differences usually occur as a result of the selection of...
the balancing methods, consideration of some strategies and grasp of the key technologies and techniques.

For economy, deadline, enterprise benefit and other reasons, the power plants often require the field balancing to achieve the best results within the least times of startup. At present, these requirements even come to be more rigorous: the times of start-up during balancing is required to be no more than twice in general, or even only once; besides, the amplitude of bearing vibration is required to be less than 30, 25 or 20 μm, shaft vibration less than 76 or 50 μm. Certainly, all these have increased the difficulties of balancing to a large extend. Based on field experience and practical examples, this paper introduces the formulation of the balancing scheme and discusses the application of key technologies about the high efficiency balancing in field for turbo-generator units with large capacity.

2. Formation of Balancing Scheme and Determination of Steps

In order to ensure that the rotor runs smoothly in working conditions, the balancing for flexible rotor is carried out to eliminate the flexural deformation and unbalanced force on the bearing. Although each of the rotors had been balanced individually at manufacturer, the vibration characteristics would be changed when they were connected with each other as a rotor system [2]. Hence, sometimes it is necessary for the rotor system to be balanced in field.

The principle of rotor balancing is not very complicated, however, there are many schemes for a case, especially in field balancing, the schemes are even more variable and contain many key steps and skills. Based upon the rich experience of balancing and accumulation of large amounts of data, sometimes the steps can be greatly simplified and the final result can be improved significantly.

In general, the steps of high-speed balancing for turbine-generator units in field are implemented as followed:

1. Measurement and analysis of original vibration data;
2. Formulating the balancing scheme and determining the steps of adding weight;
3. Obtaining influence coefficient by trial adding (unnecessary step);
4. Final adding.

Due to different balancing requirements for various rotors, the implementation process may be slightly different, but the overall method would be generally consistent with the process above. In addition, the step (2) above is the key of the whole process of balancing. An incorrect program will cause the useless adding weight and startup, and it will result in economic losses and construction delay, especially for the units with large capacity.

Through the analysis and processing of original data, the best scheme for balancing should be chosen from the various ones. Therefore, the principles should be followed as fewer times for adding weight and startup, lower residual vibration; meanwhile, the process and results of the balancing should be with a larger rate of success and a smaller risk of failure. All of these often conflict with each other, and the best scheme should reach a better effect of the balancing with smaller adding weight at fewer planes. The program will be adjusted and revised sometimes during the implementing, and there may be a number of different results contrary to the expectation after a trial or correction weight is placed on the shaft. So, a perfect balancing program should contain an alternative scheme in order to take effective steps immediately for an unexpected situation.

3. Acquisition and Processing of the Original Data before Balancing

The vibration measurement is an important work before balancing. Regardless of the operation of a new turbo-generator or startup of the unit after major maintenance, accurate and complete vibration data is the basis of the analysis, diagnosis and processing for the unit, and also is the precondition of the balancing. The structural features of the unit, measurement equipment, the former and present vibration of the unit should be realized in detail before measuring, only based on these can the pertinent measurement scheme be made.
3.1. The Arrangement of Measuring Points

The measurement of original data before balancing should include the amplitude and phase of high vibration of the bearing and shaft. Although the measurement can only have bearing vibration or shaft vibration, both of them should better be included, especially for the important measuring points of the units with large capacity. And for each bearing or shaft, the data from sensors installed in mutual perpendicular directions will play an important role in the process of balancing.

It is not clear whether a new turbo-generator needs to be balanced or not because the vibration of the unit can’t be realized before the startup for the first time, so the measuring points should be complete and it should be ensured as far as possible that each main bearing install one sensor at least. For the unit after major maintenance, it should be ensured that bearings which vibrated seriously in the past have sensors installed, as well as those of which the shaft was treated during the process. Once the vibration of a bearing or shaft is high, it is necessary for the point to be monitored intensively to guarantee the integrality of the data. The measuring points should be installed not only at the bearings whose vibration need to be reduced but also at the adjacent ones. The more data can be obtained by taking full advantage of instrument channels and arranging measuring point reasonably, and it may be helpful for vibration analysis and treatment because it is hard to estimate how difficult the balancing will be.

3.2. Determination of measuring Conditions

The measuring conditions for the amplitude and phase of bearing or shaft with high vibration should be as complete as possible. The Bode plot or record of speed / amplitude / phase during the process of run-up will help to analyze the axial position and the mode of unbalance in the subsequent balancing. Vibration of measuring points at medium-speed for warming-up need be recorded for some intervals. Because the vibration data at 3000 rpm is often used as the basis for balancing, it is necessary to record the vibration of all measuring points, as well as the vibration of each bearing in three directions i.e. vertical, horizontal and axial. Discontinuous record (or Trend plot) at the rated speed can show the changes of vibration with time, and provide evidence for the vibration fault probably caused by thermal deformations of the rotor. The vibration at load-up and rated load is also very important, because the unit operates at full load most of the time, it is a primary task and ultimate objective of the balancing to ensure that the unit operates smoothly at rated conditions. The rotor vibration at low-speed should be measured in case that the eddy current sensor had been installed, and the vibration data should also be recorded in the process of over-speed test.

Besides what is described above, the basic data should also contain the vibration of the points which are close to the high vibration at the same conditions. It could not be ignored in the rotor system balancing that the correction weight will change the vibration within not only the rotor where the weight locates but also the adjacent ones.

The recorded data in field balancing are precious, more data should be taken for future analysis and comparison during the first startup of the unit, and they can help to improve the success rate of balancing. For this point, more data will be better than fewer, but they are often limited by the amount of sensors and instrument channels. So, the measuring points should be pertinently arranged, with more measuring points at the positions in high vibration and less at the positions in low vibration. Vibration measurement before the balancing should be arranged elaborately in advance to make full use of existing sensors and instrument channels, and the key measurement should be distinguished from the ordinary ones in order to master the critical data by limited measurement.

3.3. Analysis of the Original Vibration Data

The original vibration data obtained will be analyzed as followed before balancing:

1) Confirming the high vibration is constituted by 1X vibration mainly;

2) Determining the 1X vibration is caused by the stable or unstable unbalance, this directly relates to the following method and steps of balancing.
(3) Choosing the method and steps of the balancing, they include determining the location of adding weight directly for a simple unbalance, and deciding to deal with the complex multi-span unbalance simultaneously or step by step: determining the times of startup and using the past influence coefficient or obtained by a new trial adding:

(4) Determining the final correction weight and phase angle.

4. Analysis of Exciting Force and Stability

4.1. Analysis of Exciting Force

Unbalance on the rotor is the main reason for the growth of 1X vibration, so generally, the balancing is always thought to be the method solving this problem, but sometimes there are other reasons for the vibration increment and balancing is not the optimal choice [3, 4]. In order to avoid economic losses and time delay by the blind balancing, the type of exciting force causing high 1X vibration must be distinguished first when dealing with such vibration problems.

Some exciting forces are related to the mechanical characteristics and defects of the unit and supporting system, as well as the types and severity of the defects, and these forces are irrelevant to the unbalance. They also cause a high 1X vibration but could not be eliminated by balancing. Rotating radial force caused by the defect of a flexible coupling or a joint shaft is related to the severity of defect and the torque transmitted by the rotors. Though the rotor has been balanced successfully in a certain torque or speed, the vibration may increase largely during another torque or speed. Radial force engendered by dynamic decantation in air-gap of alternator is the same with the inertial force of rotor or stator in direction, and against with the other one [5]. So once the balancing diminishes one of the forces, the vibration of the other mechanical component increases. The exciting force caused by rubbing between rotor and static parts is unstable, and can not be eliminated by adding a correction weight. Poor supporting structure often shows a high vibration throughout the whole speed range, and balancing can not achieve a significant effect. Mechanical defects illuminated as above should be diagnosed before balancing. Distinguishing the type of exciting force accurately and grasping the essence of defects will help to anticipate the balancing effect and reduce the blindness.

4.2. Stability of the Data

It must be noticed that the unbalance should be classified into the stable and unstable ones. The original eccentricity of mass on rotor is stable unbalance, while the thermal deformations of the rotor and displacement of moving parts are common unstable unbalance. The effect of balancing is obvious for the stable unbalance, but the unstable unbalance must be considered cautiously and can’t be blindly treated by adding a weight simply.

The stable unbalance has the following characters:

(1) The original unbalance will emerge in a new unit or an unit after a major maintenance with the first time startup and the early operation at the working speed, the amplitude and phase angle change a little in the subsequent startup, and will not be influenced by the way of startup.

(2) Amplitude and phase angle will be almost the same at the same speed during the process of run-up and coast-down.

(3) Amplitude and phase angle will stay stable at constant speed such as at medium-speed for warming-up and 3000 rpm, regardless of the time extending and the whole process with load, as well as the change in parameters of the unit.

In some special occasions, temporary faults of the rotor system may result in significant changes of the amplitude and phase angle in a few hours or even more, such as the rubbing between rotor and static parts and obstructed expansion of the cylinder. When the temporary fault eliminated by itself or artificially, the amplitude and phase angle would restore as ever. It still should be balanced as a stable unbalance in despite of the process.

The unstable unbalance includes the thermal deformations of the rotor and displacement of moving parts. The displacement of moving parts will cause the high 1X vibration in the process of run-up,
constant speed or with a load, and the vibration will be fluctuating, the balancing is not effective for this case generally. The amplitude and phase of 1X vibration caused by thermal deformations of the rotor change with the time, but this change tends to slow down after a while and stabilize finally. Therefore, the balancing is still used to eliminate the unbalance of the rotor caused by thermal deformations. While the fault of the thermal deformations of the generator rotor happens, the cooling system and floating oil block etc. of the rotor should be checked first usually. Balancing will be implemented only after excluding the faults such as blocked cooling channel, interturn short circuit, rubbing between floating oil block and rotor and so on.

5. Selection of Trail Weight and Processing to the Influence Coefficient

5.1. The Selection of Trail Weight

Without influence coefficient for the location of adding weight in advance, it can be obtained by the placement of one (or a group of) trail weight on the rotor, and then the final weight and phase angle could be calculated. Two parameters of the trail weight, the weight and phase angle, need to be decided. Experiences show that the phase angle is more important than the weight, it should be ensured that the correction weight is not close but opposite as far as possible to the original unbalance mass. The balancing is successful depending on whether or not the trail weight is installed at the right plane and phase, and the determination of trail weight is an important step during the balancing.

Certainly, it is most successful that the vibration is reduced to the satisfactory level by adding a trail weight only once, but it is achieved difficulty. If the original vibration changes significantly either in the magnitude or direction, the trail weight is added successfully, because the accurate influence coefficient can be calculated through the data obtained before and after the adding, and the vibration can reach a good level generally by adjusting the weight only once. For the measuring points with high original vibration, the situation should be avoided that the vibration increased so high by adding a trail weight that the rotor cannot pass through the critical speed or rise to the working speed, otherwise the influence coefficient can not be obtained and the equipment could be damaged. At the same time, it also should be avoided that the vibration has changed so little by adding a trail weight, otherwise the influence coefficient calculated by the data will have great error and poor credibility, and then this trail weight is not successful.

5.2. Analysis and Selection of Influence Coefficient

The purpose of adding a trail weight is to obtain the influence coefficient. Actually, the influence coefficient can be obtained by other ways, for example, the influence coefficient of the same type units can be utilized, and the influence coefficient of the unit’s own in the past can be used directly, and also, it can refer to the relevant technical information or the influence coefficient of the similar units balanced by the similar method.

Accurate influence coefficient is the key of the high efficient balancing. The influence coefficients in the same plane of adding weight may show much difference when they were collected from the same type unit or the unit in different periods. These influence coefficients are required analysis and selection under the principle as followed:

1) The influence coefficients obtained by heavy trail weight should be reserved, those obtained by light ones should be rejected.

2) The magnitude of influence coefficient (including the vertical and horizontal directions) should decrease generally from the plane of adding weight to a further plane, and the influence coefficients which deviate from this trend should be rejected.

3) The phase angles of influence coefficients in the two perpendicular directions on the same plane should differ about 90 degrees, and the credibility is low if it diverges too much.

4) Lag angle calculated from the influence coefficient by adding weight should be consistent with vibration theory, that is, the lag angle is less than 90 degrees when the rotor runs below critical speed,
and greater than 90 degrees when the rotor runs above the critical speed. So, the influence coefficient should be used especially careful when it is inconsistent with this rule.

6. Determination of Axial Location of Unbalance and Selection of the Plane of Adding Weight

The keys of the entire program are to determine the axial location of unbalance on the shaft and select the plane of adding weight before balancing. They affect the effect a lot and even determine whether the balancing is successful or not.

Theoretically, the vibration could still be reduced to zero, though the correction weight is inconsistent with the axial location of the unbalance, but apparently it could not achieve. The vibration of rotors could transfer and effect each other, and the axial locations for correction weight are limited, so when the correction weights are added at these limited positions to reduce the vibration at another position far away, it sometimes requires the heavier weight or often reduces the vibration of the original measuring points at the cost of increasing the vibration of the adjacent ones greatly. The unbalance of the component should be calibrated at the component itself, and the correction should be given in the same span with the unbalance during the rotor system balancing. It can reduce the flexural deformation which is caused by the unbalance force and correction force that are not in the same plane. Meanwhile, the differences among bearing chocks in dynamic characteristics should be taken into account [6], as well as the effect of thermal deformations of the rotor. The determination of the location for adding weight must be cautious. Accurate selection of the axial location is very important, and it could avoid the vast economy losses caused by startup for many times of the unit with large capacity.

In practice, the axial location of unbalance can’t be measured directly, but the approximate location can be determined through the vibration indirectly, the principle is described as follows:

1) If the bearing vibration at both ends of a rotor or shaft segment is high at the same time, the unbalance usually locates between the two bearings.

2) If only one bearing vibrated seriously and there is no bearing with a close range, the unbalance locates in the vicinity of the bearing, and furthermore it should determine which side the unbalance locates in.

3) If only one bearing shows high vibration and there is a nearby bearing with low vibration at one side, it is likely that the unbalance locates at the other side.

4) The position of the unbalance can be judged from the phases of the working speed and the critical speed, when the vibration of extended-end bearing is high.

The example of balancing for #2 unit of Jinghai power plant is taken to illuminate the process of analysis for determining the plane of adding weight. It is a domestic 600MW supercritical unit. It has revealed many defects in vibration during the process of run-up to 3000 rpm for the first time on June 5, 2017. In order to solve the problem of high vibration at #5 bearing and #6 bearing, the operator has made a variety of parametric experiments about lubricating oil temperature, as well as the hydrogen temperature and pressure of cooling system, and the vibration of two high-load test with a detailed analysis is listed in table 1.

| Table 1. Vibration of the main measuring points with high-load. 1X vibration: (μm p-p) |
|---------------------------------------------------------------|
| Testing time  | Load  | Shaft vibration | Bearing vibration |
|               |       | 5Y  | 6Y  | 7Y  | 8Y  | 5⊥  | 6⊥  | 7⊥  |
|----------------|-------|-----|-----|-----|-----|-----|-----|-----|
| 06-26 14:56    | 576MW | 55  | 39  | 150 | 72  | 50  | 75  | 62  |
| 06-28 21:57    | 576MW | 51  | 46  | 149 | 68  | 50  | 66  | 63  |

nY: the shaft vibration in Y direction at #n bearing;  
n⊥: the bearing vibration at #n bearing.

Tests showed that the major frequency component of 7Y was 1X frequency, so were 5⊥ and 6⊥. Phases of 7Y and the vibration of adjacent measuring points were stable. The phase of 7Y approaches
that of 8Y. The phase of 6⊥ was also equal to that of 7⊥ nearly [7]. Analysis showed that the possibility of electrical fault of generator rotor such as inter-turn short circuit could be excluded, because exciting current didn’t affect the vibration obviously. While the phases of 6⊥, 7⊥ and 8⊥ were normal and approached, and this showed that the fault of rubbing between the rotor and static parts did not exist. Then it could be judged that defects of assembly of rotor and balancing in manufacturer caused the poor thermal stability and high residual vibration of the rotor. At the same time, 5⊥ and 6⊥ change a little during the tests in parameter, the major frequency component of the vibration was 1X frequency during the two tests with a load, and the vibration occurred identically, these were the evidences of the unbalance. The high vibration at 3000 rpm illuminates a large unbalance on the rotor. In addition to the causes from the manufacturers, the faults might be caused by the improper connection of the LPB/GE coupling in field. The low support stiffness is caused by the defect of the low-pressure cylinder in dynamic characteristics during the design stage for this type unit.

With the vibration source closer, the measuring point shows the higher vibration. The data revealed the high 6Y, 7Y and 8Y of this unit, so it could be determined that the unbalance located at the LPB rotor and/or at the LPB/GE coupling. Then the corresponding steps of adding weight were determined and divided into two steps. In the first stage, the 930g was placed at the LPB/GE coupling, and with the 1320g added to LPB rotor, these could ensure the normal operation of the rotor at working speed. Then, in the second stage, in order to reduce the vibration at full load, the 430g was installed at the LPB/GE coupling, and with the 466g added to LBP rotor.

The vibration is listed in table 2.

Just like balancing for any other units, schemes could be quite different for this unit, but the result mentioned above would be the best. The comprehensive analysis of data and accurate selection of the plane of adding weight are the keys of successful balancing to the unit.

Table 2. Vibration of the main measuring points after adding weight. 1X vibration: (μm p-p)

| Time                  | Operating condition | Shaft vibration | Bearing vibration |
|-----------------------|---------------------|-----------------|-------------------|
|                       | 5Y  | 6Y  | 7Y  | 8Y  | 5⊥  | 6⊥  | 7⊥  |
| Vibration of the main measuring points about adding weight for the first time |             |              |                  |       |     |     |     |
| 2017-07-12 08:23 (Before adding weight) | 3000 rpm | 76  | 44  | 83  | 49  | 71  | 74  | 52 |
| 2017-07-16 21:56 (After adding weight) | 3000 rpm | 21  | 36  | 43  | 36  | 14  | 17  | 30 |
| 2017-07-17 08:53 (After adding weight) | 150 MW | 26  | 41  | 63  | 54  | 26  | 29  | 36 |
| 2017-07-18 15:20 (After adding weight) | 600 MW | 52  | 57  | 111 | 85  | 27  | 46  | 59 |
| Vibration of the main measuring points about adding weight for the second time |             |              |                  |       |     |     |     |
| 2017-08-08 05:38 (Before adding weight) | 3000 rpm | 18  | 24  | 34  | 48  | 20  | 28  | 16 |
| 2017-08-08 11:00 (After adding weight) | 200 MW | 29  | 15  | 24  | 43  | 22  | 24  | 19 |
| 2017-08-09 10:26 (After adding weight) | 500 MW | 52  | 31  | 50  | 62  | 28  | 35  | 38 |
| 2017-08-09 14:48 (After adding weight) | 610 MW | 43  | 28  | 58  | 54  | 19  | 9   | 32 |

7. Modal Method Assisting to Determine the Balancing Steps
The vibration of flexible rotor has following characters:

(1) The deflection curve of a rotor is formed by a number of mode shapes of the lateral natural frequencies, and because each mode shape varies nonlinearly with the rotational speed, the complex deflection of the flexible rotor shows nonlinearity [8].

(2) When a rotor runs near a critical speed, the deflection of the rotor will usually occur in accordance with this mode shape, and the other modes do not make effects.

(3) As the excitation source, the unbalance can also be decomposed to a series of unbalances corresponding to each main mode.
Based on the characters of flexible rotor mentioned above, it can determine the mode shapes of unbalance before balancing by the Bode plot during run-up or coast-down: the first mode, the second or third mode. The mode analysis can assist to determine the procedure of adding weight. When the #1 unit of Yuhuan power plant was starting-up after major maintenance in June, 2019, the high vibration at #3 bearing and #5 bearing at 3000 rpm is listed in table 3.

Table 3. Bearing vibration of #1 unit during start-up for the first time. 1X vibration: (μm p-p ∠°)

|    | #1    | #2    | #3    | #4    | #5    | #6    | #7    |
|----|-------|-------|-------|-------|-------|-------|-------|
| 06-25 12:23 | ↓ 3.6∠315 | 3.8∠236 | 27∠167 | 15∠233 | 48∠206 | 10∠59 | 14∠256 |
| 3000 rpm | ↓ 7∠233 | 9∠74  | 25∠108 | 18∠112 | 39∠142 | 14∠41 | 14∠207 |

For the phases of #4 bearing and #5 bearing are approximate, it was determined that the unbalance of the first mode leaded to the vibration of #4 bearing and #5 bearing. So the 516g were placed in phase at the both sides of the low-pressure rotor. However, the vibration of #4 bearing and #5 bearing was not reduced after adding the weights, and the judgment was denied that there was an unbalance of the first mode between the #4 bearing and #5 bearing. The single calculation of the vibration at #4 bearing or #5 bearing showed that the location of the trial weight should be adjusted to the opposite direction. So it was determined to decompose the vibration of #4 bearing and #5 bearing during run-up in such case of contradiction (figure 1).

Figure 1 The Bode plots of vibration at #4 bearing (a) and #5 bearing (b) in vertical direction during run-up after adding weight to the low-pressure rotor of the #1 unit in Yuhuan power plant.

If there was a large unbalance of the second or third mode on the low-pressure rotor, the components of unbalance at #4 bearing and #5 bearing in phase or out-of-phase would increase during run-up. However, according to the calculation, the components in phase or out-of-phase didn’t increase obviously, while the components of #5 bearing vibration and #6 bearing vibration in phase increased with the speed close to 3000 rpm. According to this result, the vibration of #5 bearing at 3000rpm was caused by the unbalance of the first mode on the shaft between #5 bearing and #6 bearing, and the optimal location for adding weight to reduce the vibration at #5 bearing was on the LP/GE coupling. Finally, the weight was added at the LP/GE coupling, as well as at the last stage of medium-pressure rotor, and the vibration at #3 bearing, #4 bearing and #5 bearing was reduced to desired level.

When it is required to determine the modes and whether the unbalance locates in the span or not, the modal analysis is a quite effective tool by using the data of run-up or coast-down.
8. Conclusions
The determination of axial location of unbalance and the selection of the plane for adding weight are keys of high efficiency balancing in field for turbo-generator units with large capacity, only based on this can the prefect scheme of the balancing be formed successfully. The techniques and experiences, such as the full collection and thorough analysis of the data before balancing, determination for the types of exciting force, obtaining and processing for the accurate influence coefficients, modal analysis for vibration mode etc. can effectively improve the accuracy of balancing, lessen the times of adding weight, and reduce the risk of failure.

References
[1] Bently D E and Hatch C T 2014 *Fundamentals of Rotating Machinery Diagnostics* (Beijin: Chnia Machine Press) pp 228.
[2] Shen X Y, Jia J H, Zhao M and Jing J P 2008 Experimental and numerical analysis of nonlinear dynamics of rotor-bearing-seal system *Nonlinear Dynamics* 31 31-44.
[3] Feese T D and Grazier P E 2004 Balance this! Case histories from difficult balance jobs Proceedings of the Thirty-third Turbo-machinery Symposium Houston, Texas p 193-212.
[4] Kim J S and Lee S H 2003 The stability of active balancing control using influence coefficients for a variable rotor system *Int. J. Adv. Manuf. Technol.* 22(7-8) 562–567.
[5] Zhang W J, Gao X, Zhang Y and Li H 2021 Analysis and treatments for abnormal vibration of turbine generator unit by bearing elevation *Turbine Technology* 63(4)293-296
[6] Rodyushkin V M and Shishkin V I 1995 High-speed measuring-computing system for balancing rotors on a production line *Measurement Techniques* 38(6) 645-648.
[7] Ma G P, Lu S Y and Feng Zeng 2008 Analysis and disposal of abnormal vibration of domestic supercritical 600MW units *Guangdong Electric Power* 21(3) 54-56.
[8] Ehrich F F 2004 *Handbook of Rotordynamics* (Florida: Krieger Publishing Company) chapter 3 pp 3