A Method Based on Initial-phase-point Analysis for Determining the Fault Mode of the Unbalanced Vibration of Turbine Rotor

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Abstract. Rotor imbalance is one of the common faults of turbines, but traditional analysis methods are difficult to distinguish different fault modes, e.g. the original imbalance of rotor and the thermal bending of rotor. By introducing the concept of initial phase point, a rotor imbalance model is established and corresponding identification method for original imbalance and thermal bending is developed. Through the analysis and processing of practical cases, it is proved that this method is effective and can carry out the fault diagnosis of rotor unbalance accurately.

1. Introduction
Rotor imbalance is a common fault of various turbines such as expansion generator, turbo-generator, hydro-generator and gas turbine [1,2]. Unbalanced fault belongs to the common forced vibration according to the type of excitation force, which usually has large rotational frequency component on frequency spectrum diagram as the most significant characteristic. However, many fault modes has this characteristic, e.g. the original unbalance of rotor, the thermal bending of rotor, the rigidity change of rotor or bearing, excessive clearance or insufficient rigidity of bearing [3–6], etc. Therefore, it is difficult to distinguish these faults by using traditional FFT spectrum.

Based on operation experience, among all the rotation frequency faults, the rotor imbalance induced rotation frequency fault accounts for the largest proportion. Therefore, technicians usually hold the view that the emerged rotation frequency vibration is caused by the imbalance of rotor, and use high-speed dynamic balance method to solve this kind of vibration problem. However, in many cases the dynamic balance method can’t achieve satisfactory balance effect when dealing with the vibration problem of turbine unit, due to two aspect of reasons: (1) One is that the information of vibration response isn’t fully utilized; (2) The other is that the type of imbalance isn’t further identified even when imbalance is diagnosed as the dominant fault, and thus the corresponding balance scheme can’t be selected in a targeted way. Blindly adopting dynamic balance method to eliminate rotation frequency failures will not only bring huge economic losses due to multiple start up and shut down, but also bury potential safety hazards due to the true cause of failure isn’t identified. In order to improve the accuracy of
unbalanced fault diagnosis, it is necessary to introduce initial phase point (IPP) analysis method on the basis of traditional spectrum analysis, to make full use of the vibration information collected on site, and to comprehensively analyze and diagnose the cause of fault.

2. The physical meaning of IPP

When two displacement sensor signals arranged vertically on a section are processed through isolation and filtering, the frequency, amplitude and phase of spectral line can be accurately determined by using interpolation technique. In this paper, the phase is defined as the angle at which the leading edge of the bond signal pulse lags the first forward zero crossing. The power frequency components of both directions are extracted and synthesized to obtain:

\[ x_i = A_i \sin(\alpha_i + \alpha_i), \quad y_i = B_i \sin(\alpha_i + \beta_i) \]  

(1)

where, \( A_i \) and \( B_i \) are vibration amplitudes in X and Y directions respectively, \( \alpha_i \) and \( \beta_i \) are vibration phases in X and Y directions respectively, and \( \omega = 2\pi f \) (unit: rad/s) is the angular velocity of rotor rotation, and \( f \) is the rotation frequency (unit: Hz). Eq. (1) can be regarded as the parametric equation of the rotor frequency axis locus of this section. The resulting trajectory is an ellipse, which is called the power frequency ellipse, namely the two dimensional holographic spectrum of power frequency. When the rotor has only unbalanced faults, the physical meaning of Eq. (1) is essentially the same as the results of theoretical analysis, and both represent the unbalanced response of the rotor expressed in the form of complex vectors. This form of expression takes into account the vibration information in both directions of rotor and the factors of the anisotropic stiffness of rotor-bearing system, and describes the vibration characteristics of rotor more comprehensively.

Here we define that, when the key phase slot on rotor is facing the key phase sensor, the rotor precesses to the IPP. Based on Eq. (1), the expression of the vector (initial phase vector) from the center of the power frequency ellipse of the \( i \)th measuring surface to the IPP can be defined as:

\[ \text{IPP}_i = \sqrt{(A_i \sin(\alpha_i))^2 + (B_i \sin(\beta_i))^2} \]  

(2)

\[ R_i = \sqrt{(A_i \sin(\alpha_i))^2 + (B_i \sin(\beta_i))^2} \]  

(3)

\[ \theta_i = \arctan \left( \frac{B_i \sin(\beta_i)}{A_i \sin(\alpha_i)} \right) \]

The IPP on the power frequency ellipse is essentially the point \( \omega t = 0 \) on the axis track of rotor rotation frequency, as shown in Fig. 1.

It can be seen from its expression that it takes into account the two directions of vibration sensor, more comprehensively reflects the vibration of rotor, and the IPP can clearly indicate the size and orientation of the imbalance of rotor. When the size and orientation of the imbalance change, the position of IPP on the power frequency ellipse also changes.
3. The physical meaning of IPP

For the convenience of analysis, the turbine rotor is equivalent to a single-disk Jeffcott rotor. When the unbalanced amount of the disk is \( me_u e^{i\phi_m} \) and the static initial bending amount of the rotor is \( \lambda_b e^{i\phi_b} \), the vibration response of the unbalanced bending rotor is:

\[
R = r_u + r_b = \frac{\lambda_b e^{i\phi_b}}{\omega_{cr}^2 - \omega^2 + 2i\xi\omega_{cr}\omega}
\]  

Eq. (4) indicates that the response of the rotor \( r \) is the superposition of the unbalanced response \( r_u \) and the bending response \( r_b \). Applying the theory of modal analysis, the unbalanced bending response (i.e. Eq. (4)) of Jeffcott rotor is extended to multi-disks elastic support system. In this case, the center power frequency response of the \( m \)th turntable can be expressed as:

\[
r(m, \omega) = r_u(m, \omega) + r_b(m, \omega)
= \sum_{n=1}^{N} (u_n \omega^2 + \epsilon_n \omega^3) N_n(\omega) \phi_m^{(n)} e^{i\omega t}
\]  

where, \( N_n(\omega) = [(\omega_n^2 - \omega^2) + 2i\xi_n \omega_n \omega]^{-1} \).

Thermal bending usually has a certain directionality, that is to say, the phase remains stable even when the amount of rotor bending changes. Assuming that the rotor has a temporary thermal bending failure, the bending amount at the \( m \)th disc is \( E(m) \)

\[
E(m) = \sum_{n=1}^{N} \epsilon_n \phi_m^{(n)} e^{i\omega t}
\]  

where, \( \epsilon_n \) is the \( n \)th bending amount of the rotor, \( \phi_m^{(n)} \) is the \( m \)th element of the \( n \)th mode shape \( \{ \phi^{(n)} \} \). Based on Eq. (5), the power frequency response \( r(m, \omega) \) of the rotor now can be written as
Assuming that the rotor continues to undergo thermal bending and reaches the bending amount \( E'(m) = \partial E(m) \), it can be seen from Eqs. (6) and (7) that the power frequency vibration amount \( r'(m, \omega) \) of the rotor at the \( m \)th disc under the same speed is:

\[
E'(m) = \delta \sum_{n=1}^{N} e_n \phi_m^{(n)} e^{ij\omega t} = \sum_{n=1}^{N} \delta e_n \phi_m^{(n)} e^{ij\omega t} = \sum_{n=1}^{N} e_n' \phi_m^{(n)} e^{ij\omega t}
\]

(8)

\[
r'(m, \omega) = r'_n(m, \omega) = \sum_{n=1}^{N} e_n' \omega_n^2 N_n(\omega) \phi_m^{(n)} e^{ij\omega t}
\]

(9)

It can be seen from Eq. (8) that under different thermal bending amounts, the power frequency response of the rotor will change, but its phase is stable. In other words, when the size and orientation of the unbalanced mass of the rotor change, the IPP must change.

Take the derivative of both sides of the parametric equation with respect to time (for convenience, the subscript \( i \) is omitted here), we obtain

\[
\begin{align*}
\frac{dx}{dt} &= A \omega \cos(\omega t + \alpha) \\
\frac{dy}{dt} &= B \omega \cos(\omega t + \beta)
\end{align*}
\]

(10)

Therefore, the slope \( k \) of the tangent to the IPP on the power frequency ellipse of the holographic spectrum is

\[
k = \frac{dy}{dx}\bigg|_{t=0} = \frac{B \cos \beta}{A \cos \alpha}
\]

(11)

From the theoretical analysis formula (i.e. Eq. (11)) of the bending response, it can be seen that when the unbalanced state of the rotor does not change, in the case of different thermal bending amounts, although the size of the power frequency response of the rotor has changed, its phase is stable. Therefore, as the rotor continues to undergo thermal bending, the vibration signal becomes

\[
\begin{align*}
x' &= \delta \omega \sin(\omega t + \alpha) \\
y' &= \delta \omega \sin(\omega t + \beta)
\end{align*}
\]

(12)

Correspondingly, the slope \( k' \) of the tangent to the IPP is:

\[
k' = \frac{dy}{dx}\bigg|_{t=0} = \frac{\delta \omega \cos \beta}{\delta \omega \cos \alpha} = \frac{B \cos \beta}{A \cos \alpha} = k
\]

(13)
It can be seen from Eq. (13) that at a constant speed, when the unbalanced state of the rotor does not change and the amount of thermal bending continues to increase, the power frequency ellipse representing the power frequency response of the rotor continues to grow, while the tangent lines at the IPP on the power frequency ellipse remain parallel to each other. This feature of the IPP can be used as a diagnostic index to determine whether the rotor unit is out of balance or thermally bent.

As can be seen from above theoretical analysis and the characteristics of the IPP, when the rotor undergoes thermal bending, if the original unbalanced state of the rotor does not change (namely the imbalance size and orientation does not change) and the temporary thermal bending is the only reason for the increased power frequency vibration, the tangent line of the IPP on the power frequency ellipse should remain parallel and the initial phase angle that characterizes the imbalance orientation remains unchanged before and after the vibration increases.

4. Identification method for rotor imbalance and temporary thermal bending

When the turbine rotor is out of balance, the vibration will increase, and the position of the IPP on the power frequency ellipse will change due to the change of the key position of the rotor. The identification diagram is shown in Fig. 2.

![Figure 2. Schematic diagram of rotor imbalance.](image)

When the turbine rotor undergoes temporary thermal bending, the size of the power frequency ellipse will change, and the position of the IPP will also change, but the tangent of the ellipse past the IPP is parallel to the original one. The identification diagram is shown in Fig. 3.

![Figure 3. Schematic diagram of rotor temporary thermal bending.](image)

5. Case analysis

A turbine is undergoing a single unit test run, and the radial relative vibrations of the rotating shaft at the two bearings are within 25 μm when running at the rated speed without load, which is lower than the manufacturer's warning standard value (37 μm) for the radial relative vibration of the rotating shaft. Subsequently, during the pressurization test by gradually closing the bypass valve opening, it is found that the amount of vibration presents continuous increase as the pressure increasing. When the pressure reaches 0.4 MPa, the vibration of the free end of the turbine exceeds 100 μm. When the opening of bypass valve is gradually increased, the vibration decreases but the decreasing rate is slow. When the turbine is restarted after stopping for four hours, the vibration is still very good before pressurization, but above phenomena repeatedly emerge after pressurization.
Using the precision diagnostic toolkit to analyze the vibration data during the pressurization process, we select eight groups of vibration data at the free end as the trend graph of the two-dimensional holographic spectrum, as shown in Fig. 4. The tangents at the IPPs on the two-dimensional holographic spectrum of each power frequency are marked in the figure.

**Figure 4.** The power frequency change trend diagram of the free end of turbine during pressurization process

The analysis found that the power frequency ellipse becomes larger with the increase of pressure at the rated working speed, but the tangent line at the IPP remains parallel. According to the previous discussion, this phenomenon reflects the temporary thermal bending fault of the rotor. In order to verify the correctness of this conclusion, an open cylinder inspection was carried out on the second day of the test, and it was found that the surface of the rotor between the 3rd and the 4th impellers presented high-
temperature purple (see Fig. 5), and the diaphragm gas seal at the lower half of the place was damaged serious (see Fig. 6).

After inspection by the manufacturer’s technicians, it was found that the axial clearance of the lower half of the air seal was not suitable during installation, and the axial clearance on the high pressure side was too small. The reason for the failure was diagnosed to be: when the bypass valve was gradually closed, the pressure at the turbine outlet would continue to increase, and the pressure difference between the two sides of the impeller would push the rotor to move axially to the turbine inlet, and too small axial clearance causes friction between the high and low teeth on the rotor and the lower air seal. The friction only occurs on the outer surface of the rotor in a stretched state, and the heat generated by the friction caused the thermal bending of the rotor. After decreasing the pressure, the axial displacement of the rotor was reduced, the friction was weakened too, and thus the thermal bending of the rotor was reduced. However, the vibration would not drop rapidly due to thermal inertia. After four hours of shutdown, the rotor was cooled down, and the temporary thermal bending phenomenon disappeared, but the vibration returned to the normal value when the turbine was restarted to the working speed.

After returning the diaphragm air seal to the factory for reprocessing, the turbine vibrated below 30 μm in the subsequent pressure test, and it ran normally.

6. Conclusions

Rotor imbalance can be caused by many fault modes, and thereby determining specific fault mode is an important premise for adopting suitable vibration eliminating method. In order to raise the diagnosis rate of rotor vibration faults, an identification method based on initial-phase-point monitoring and analysis is proposed. Practical case studies indicate that, this method can effectively and accurately judge whether the large vibration fault of turbine unit is caused by the temporary thermal bending of rotor or the original imbalance of turbine unit. Therefore, it can avoid unnecessary economic loss caused by blindly adopting dynamic balance to eliminate vibration.

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