Dynamic Balancing Simulation Based on Virtual Prototyping Technology

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Abstract—Rotor unbalance is the main vibration excitation source of rotating machinery accounting for 60~70% of the total machine fault. In this paper, unbalance fault and dynamic balancing are simulated by the virtual prototyping technology based on the unbalance mechanism and balancing theory. The single-face and double-face dynamic balancing are studied. The results validate the correctness and feasibility of the proposed method. The contribution of this paper is to provide one new way to verify balancing method and analyze balancing error without real test.

Keywords- virtual prototyping; unbalance simulation; dynamic balancing

I.  INTRODUCTION

Rotating machinery is widely used in petroleum, chemistry, power, metallurgy and machine manufacturing fields. They mostly belong to power generating machine and are key equipment in factories. Abnormal vibration of rotating machinery can make noise, reduce work efficiency, even make whole product line breakdown that brings huge economic losses, and causes other serious damage.

Thus, reliability and safety of equipment is one important mission for equipment engineers. Plenty of engineering practice illustrates that rotor unbalance is the main vibration excitation source accounting for 60%~70%. People has paid more attention to how to solve rotor unbalance fault.. Many dynamic balancing methods have been proposed and successfully applied to real cases.

In this paper, the research focuses on unbalance fault and dynamic balancing based on virtual prototyping technology. The 3-D model of ZT-3 rotor test rig is established. Constrains among parts and motion excitation are defined to simulate the unbalance fault. The vibration signal acquired by virtual sensors, and motion and dynamic analysis are carried out using MSC software ADAMS. Based on acquired data, dynamic balancing is used to reduce vibration to fix unbalance fault.

II. BUILDING VIRTUAL PROTOTYPING MODEL

A. The Structure and Modeling of The Test Rig

The rotor vibration test rig is the equipment for simulating the typical rotor vibration. It is used to test the rotor characteristics under the condition of forced-vibration and self-excited vibration. It has simple structure, and wide speed range.

In the process of modeling, three assumptions are given to simplify the model according to the equivalence principle and ADAMS features.

- All parts are rigid components (without considering deformation), and the entire system is a rigid system;
- The size, quality, location of the center of mass, moment of inertia of modeling is the same as actual situation;
- Ignore friction between the two hinge points.

The simplified modeling of the test rig is shown in Fig.1.

Figure 1. The virtual prototyping modeling

B. Model Verification

In order to ensure the simulation analysis smooth, the virtual prototyping should be verified to check the implied mistakes during modeling before the simulation. The possible errors are shown as following:

- Incorrect connections and constraints, unrestricted components, massless components, and the degrees of freedom.
- All constraints are damaged or wrongly defined.

Figure 2. The verify information

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The verified information of the model displayed by ADAMS/View is shown in Fig.2 [1], that expressed the correctness of the model.

C. The Validation of Rig and Flexible Characteristic

The rotor can be divided into two types according to the work of the state and the mechanical characteristics: one is the rigid rotor, another is flexible rotor. There is no certain conception. In engineering, the rotor critical speed is the dividing line: the rigid rotor works slower than the critical speed; the flexible rotor works faster than the critical speed [2]. The critical speed of the test rig is 4400r/min. The speed of the single-face and the double-face dynamic balancing are lower than the critical speed. Thus, the rotor can be treated as a rigid rotor.

III. THE FAULT SIMULATION OF UNBALANCE

A. The Unbalance Fault Mechanism

For a rotor system, unbalance results from the fact that the center of gravity of the rotating member does not coincide with the center of rotation.

Assumption: eccentricity e, mass M, stiffness k, damping coefficient c, rotate speed n(r/min), angular velocity \( \omega = \frac{2\pi n}{60} \), centrifugal force \( F = Me^2 \omega \), the force is divided into two directions as follows:

\[
F_1(t) = F \cos \omega t = Me^2 \cos \omega t
\]
\[
F_2(t) = F \sin \omega t = Me^2 \sin \omega t
\]

90 degrees difference between two forces, the vibration equation is:

\[
My + cy + ky = Me^2 \sin \omega t
\]

Normalization:

\[
\ddot{y} + 2\zeta \omega_n \dot{y} + \omega_n^2 y = e \omega^2 \sin \omega t
\]

where the damping coefficient \( \zeta = c / 2M \omega_n \), \( 0 \leq \zeta \leq 1 \); natural frequency of vibration is \( \omega_n = \sqrt{k / M} \); exciting frequency \( \omega = \frac{2\pi n}{60} \).

General solution:

\[
y(t) = De^{-\zeta \omega_n t} \sin(\sqrt{1 - \zeta^2} \omega_n t + \phi) + Y \sin(\omega t - \Psi)
\]

The first part of the solution is the transient solution due to free vibration. It disappears soon:

\[
y(t) = Y \sin(\omega t - \Psi)
\]

The second part of the solution is the steady-state solution due to forced vibration:

\[
Y = cH(\omega) = \frac{e(\omega / \omega_n)^2}{\sqrt{[1 - (\omega / \omega_n)^2]^2 + (2\zeta \omega / \omega_n)^2}}
\]

B. Detailing The Unbalance Mode

The components of the test rig model will not produce deformation under the unbalance force since they are rigid. To obtain vibration signal, the some components should be transferred to flexible bodies. In this model, only the rotor shaft need be flexible. AutoFlex is used to build the flexible shaft, and replacing the rigid shaft. After the replacement the load and joints will transferred to the flexible body automatically. The flexible shaft is show in Fig.4.

C. The Simulation and Analysis of The Single-face Dynamic Balancing

1) The simulation of the single-face dynamic balancing

The single-face unbalance simulation settings are 600r/min motor speed, 0.05kg eccentric mass on 25mm radius of the disc, which is shown in Fig.5. The time waveform of unbalance condition is shown in Fig.6.
From the time domain waveforms, we can see that the unbalance increases the vibration amplitude. The phrase difference between direction X and Y is 90 degrees. These characteristics are in accordance with the unbalance phenomenon [3].

2) Balancing analysis of the single-face dynamic balancing

The idea of the single-face dynamic balancing is: at the selection speed, add a trial weight to the rotor and calculate the influence coefficient [4]. Then the balancing mass can be calculated based on the method of the influence coefficient. In the process of dynamic balancing, the speed must be a constant. The concrete steps are listed as follows:

- Measure the original vibration $A_v$.
- Add a trial weight to the disc $P$, then measure the vibration $A_v$.
- Calculate the influence coefficient:
  \[ \alpha = \frac{(A_v - A_{\theta})}{P} \]  
  where, $A_v - A_{\theta}$ is the vibration changes; $\alpha$ is the influence coefficient, which represents that in the direction of zero, one meter radius causes the vibration changes due to one kilogram weight.
- Calculate the rotor balance weight $Q_v$:
  \[ Q\alpha + A_v = 0 \]  
  The equation’s physical meaning is: the vibration changes $Q\alpha$ caused by the added balancing weight $Q_v$ should be able to eliminate the original vibration $A_v$. Thus we can obtain: $Q = -\frac{A_v}{\alpha}$

In this case, $A_v = 0.0024 \angle 215^\circ$, $A_{\theta} = 0.0021 \angle 180^\circ$, $\alpha = 0.046667 \angle 58.6^\circ$, the added weight $Q = -0.0514 \angle 121.4^\circ$, minus indicates that the adding weight is on the opposite direction. The vibration waveforms before and after dynamic balancing are shown in Fig.7.

From Fig.7, the amplitude of vibration reduces to 2.2177E-4mm after adding balancing mass, that is to say the vibration caused by the unbalance mass are eliminated. The error between the initial unbalance mass and the added mass is $\delta = (0.0514 - 0.05)/0.05 \times 100\% = 2.8\% < 5\%$, which is in accordance with the engineering requirements.

For the single-face dynamic balancing, the balancing result is satisfactory since the amplitude decreased to less than 2.2177E-4mm which is far from the initial fault vibration and close to zero.

D. The Simulation and Analysis of The Double-Face Dynamic Balancing

1) The simulation of the double-face dynamic balancing

Two discs are placed to simulate double-face unbalance at the speed of 3600r/min. 0.06kg unbalance mass is added on 25mm radius of one disc, while 0.01kg unbalance mass is added on the same place of another disc. The model is shown in Fig.8:

- Measure the original vibration of two sides of discs $A_v$, $B_v$.
- Add trial weight $P_1$ in the face I, then measure the vibration: $A_{\theta_i}$, $B_{\theta_i}$.
- Remove weight $P_1$, add weight $P_2$ in the face II, then measure the vibration: $A_{\theta_2}$, $B_{\theta_2}$.
- Calculate the influence coefficients of two points:
  - the influence coefficient to the face I:
    \[ \alpha_{v1} = \frac{(A_{v1} - A_{\theta})}{P_1} ; \alpha_{v2} = \frac{(B_{v2} - B_{\theta})}{P_2} \]  
  - the influence coefficient to the face II:
    \[ \alpha_{v2} = \frac{(A_{v2} - A_{\theta})}{P_2} ; \alpha_{v1} = \frac{(B_{v1} - B_{\theta})}{P_1} \]  

2) Balancing analysis of the double-face dynamic balancing

The procedure of the double-face dynamic balancing is the same as the single-face. It is just a promotion to the single-face. (a) the single-face dynamic balancing only needs to add weight in one face while the double-face dynamic balancing add weights in two faces.(b) the single-face dynamic balancing only considers one measured point while the double-face dynamic balancing considers two measured points [5].

- Measure the original vibration of two sides of discs $A_v$, $B_v$.
- Add trial weight $P_1$ in the face I, then measure the vibration: $A_{\theta_i}$, $B_{\theta_i}$.
- Remove weight $P_1$, add weight $P_2$ in the face II, then measure the vibration: $A_{\theta_2}$, $B_{\theta_2}$.
- Calculate the influence coefficients of two points:
  - the influence coefficient to the face I:
    \[ \alpha_{v1} = \frac{(A_{v1} - A_{\theta})}{P_1} ; \alpha_{v2} = \frac{(B_{v2} - B_{\theta})}{P_2} \]  
  - the influence coefficient to the face II:
    \[ \alpha_{v2} = \frac{(A_{v2} - A_{\theta})}{P_2} ; \alpha_{v1} = \frac{(B_{v1} - B_{\theta})}{P_1} \]
Different with the single-face dynamic balancing, the double-face dynamic balancing has four influence coefficients. It can be expressed as a matrix:

\[
A = \begin{bmatrix}
\alpha_{11} & \alpha_{12} \\
\alpha_{21} & \alpha_{22}
\end{bmatrix}
\]

- Arrange the balance weight \(\hat{Q}_i, \hat{Q}_j\), in the face I, II. Vibration changes caused by balance weight plus the original vibration equals zero.

\[
\begin{bmatrix}
\alpha_{11} & \alpha_{12} \\
\alpha_{21} & \alpha_{22}
\end{bmatrix}
\begin{bmatrix}
\hat{Q}_1 \\
\hat{Q}_2
\end{bmatrix}
+ \begin{bmatrix}
A_1 \\
A_2
\end{bmatrix} = \begin{bmatrix}
0 \\
0
\end{bmatrix}
\]

This is a quadratic equation with one variable, written in matrix form:

\[
\begin{bmatrix}
\alpha_{11} & \alpha_{12} \\
\alpha_{21} & \alpha_{22}
\end{bmatrix}
\begin{bmatrix}
\hat{Q}_1 \\
\hat{Q}_2
\end{bmatrix}
+ \begin{bmatrix}
A_1 \\
A_2
\end{bmatrix} = \begin{bmatrix}
0 \\
0
\end{bmatrix}
\]

- Solve the weight and phase angle from the equation sets.

<table>
<thead>
<tr>
<th>Point</th>
<th>Marker 1 (mm)</th>
<th>Marker 2 (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Original vibration</td>
<td>0.1227(\sqrt{256})°</td>
<td>0.0569(\sqrt{4})°</td>
</tr>
<tr>
<td>Test weight Face I (kg)</td>
<td>0.06(\sqrt{180})°</td>
<td>0.1124(\sqrt{340})°</td>
</tr>
<tr>
<td>Test weight Face II (kg)</td>
<td>0.01(\sqrt{240})°</td>
<td>0.1039(\sqrt{243})°</td>
</tr>
</tbody>
</table>

Calculate the influence coefficient of two points:

\[
\begin{bmatrix}
\alpha_{11} & \alpha_{12} \\
\alpha_{21} & \alpha_{22}
\end{bmatrix}
= \begin{bmatrix}
2.625\(\sqrt{211}\)° 3.02\(\sqrt{232}\)° \\
0.5616\(\sqrt{14}\)° 10.29\(\sqrt{309}\)°
\end{bmatrix}
\]

The balance weights: \(\hat{Q}_1 = 0.04326\(\sqrt{183}\)°, \hat{Q}_2 = 0.0282\(\sqrt{267}\)°.

The calculation balance weights are added to the virtual prototyping, the comparison waveform is shown in Fig.9:

![Figure 9. The vibration waveform comparison (double)](image)

From the Fig.9, the amplitude of vibration reduces from 0.0577mm to 0.0139mm from 0.0577mm after adding the weights.

The maximum allowable residual unbalance:

\[
U_{uw} = 1000 \times G \times m \times 30/(\pi \times n)
\]

where, \(m\) is rotor mass, \(G\) is rigid rotor balance quality grade \((G=6.3)\), \(n\) is rotor speed \((n=3600\text{r/min})\).

The maximum allowable eccentricity:

\[
e_{per} = U_{per}/m = 1000 \times 6.3 \times 30/(\pi \times 3600) = 16.7(\text{g.mm/kg})
\]

For the double-face dynamic balancing, if double face lie in the bearing span and have the approximately same distance from rotor center so each face will have the half of the allowable residual unbalance in the above the equation. That is to say: \(e_{per} = 8.35(\text{g.mm/kg})\).

Ignore the damping:

\[
Y_{max} = e_{per}H(\omega) = \frac{e_{per}(\omega/\omega_n)^2}{\sqrt{1-(\omega/\omega_n)^2} + \left(2\zeta\omega/\omega_n\right)^2}} = 0.01691\text{mm}
\]

After dynamic balancing: \(y = 0.0139\text{mm} < 0.01691\text{mm}\), it is satisfactory for balance quality.

3) Results analysis

Single-face and double-face dynamic balancing are simulated based on the theories using virtual prototyping technology. The results are satisfactory from the view of balancing. However, unbalance can be totally eliminated even under virtual environment.

The reasons of balance error between real unbalance mass and calculation mass can be summarized as following:

- Amplitude error

The waveform caused by unbalance mass is not pure sine waveform. In other words, the peaks of the vibration amplitude are not the same. Thus, average amplitude is used to calculate the balance mass that introduces the error.

- Phase error

Phase is another important factor during balancing. In the process of measuring the phase angle, the phase whose amplitude is close to the average amplitude, as a result, caused the accumulation of errors. So the final results are different from the normal situation. It is very difficult to do the double-face dynamic balance in site test since we may need a number of tests in order to achieve the requirement. In this paper, the simulation result by the virtual prototyping has been greatly improved, and a new direction to try in the site test has been provided.

IV. CONCLUSIONS

In this paper, virtual prototyping technology is used to simulate unbalance fault and research dynamic balancing of rotor. The fault characteristics reappear in the unbalance vibration signals obtained from virtual prototyping. Dynamic balancing processes are simulated; the balancing results verify the correctness of the simulation. At the same time, the whole
fault simulation process based on virtual prototyping is more intuitive. The repeatability is greatly facilitated by the experimental data collection, analysis and processing work. It improved the diagnostic accuracy and reliability in a great extent.

REFERENCES


