Identification of Unbalance in Rotating Machinery Using Vibration Analyse Solution

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Abstract. Vibrations in rotating machinery are normally the consequence of mechanical faults such as mass unbalance, manufacturing, mechanical looseness, assembly, installed machines, coupling misalignment, and many other causes. However, unbalance is the most popular reason of machine vibration. An unbalanced rotor always causes more vibration and generates excessive force in the bearing area and reduces the life of the machine. In this paper, experimental studies were performed on a rotor with two mounted disks to identify the unbalance in rotor using order tracking technique based on one key phasor. After calculation of the coefficients of influence $\alpha$ of each experimental test, the matrix of influence coefficient $[C]$ and the global matrix of coefficients $[H]$ can be obtained. Finally, the two suitable masses are identified in order to reduce the unbalance of the rotor systems.

1. Introduction

Rotor unbalance is the most popular reason in rotating machinery. A very small amount of unbalance may cause severe problem in high speed rotating machines. The vibration caused by unbalance may destroy critical parts of the machine, such as bearings, seals, gears and couplings. As a result of mass unbalance, a centrifugal force is generated and must be reacted against by bearing and support structures. The purpose of rotor balancing is to attain satisfactory running when installed on site. It means no more than an acceptable magnitude of vibration is caused by the unbalance remaining in the rotor. In the case of a flexible rotor, it also means that not more than an acceptable magnitude of deflection occurs in the rotor at any speed up to the maximum service speed. Some analytical methodologies have been carried out to unbalance response such as the transfer method. Further, the unbalance part of the rotor rotates at the same speed as the rotor and therefore the force caused by the unbalance is synchronous [1]. Vibration signatures are commonly used as a useful method for investigating progressive machine mechanical malfunctions and form the baseline signature for further comparative monitoring to detect mechanical faults [2]. G.K. Yamamoto [3] proposed a smart experimental setup for vibration measurement and imbalance fault detection in rotating machinery. By adding a correction mass at the location indicated by the system, the vibration of the machine will be effectively reduced. In this paper, experimental studies were performed on a rotor with two mounted disks to identify the unbalance in rotor using order tracking technique based on one key phasor. After calculation of the coefficients of influence $\alpha$ of each experimental test, the matrix of influence
coefficient $[C]$ and the global matrix of coefficients $[H]$ can be obtained. Finally, the two suitable masses are identified in order to reduce the unbalance of the rotor systems.

2. Test rig description

Figure 1 depicts an overview of the experimental apparatus, highlighting the main parts of the test rig. Basically, it is encompassed by a flexible rotor-bearing system. The test rig [4] consists of a rigid shaft of 10mm diameter supported by two rolling element bearings at the ends of shaft. The shaft is driven by a 150 W electric motor through a flexible coupling. The motor speed is adjusted by a speed controller with 2 rpm accuracy and can reach a maximum value of 5000 rpm. The disk with a diameter and thickness of 75 mm and 25 mm respectively is installed in the middle of the shaft for the balancing the system. In order to measure the vibration of the shaft, two proximity probes of 90° phase angle are installed at each bearing position in the horizontal and vertical direction. An additional proximity probe is used as a key phasor ([5]-[8]).

Figure 1. Picture of the test rig: (A) Bearing; (B) Proximity probes; (C) Shaft; (D) Disk; (E) Control box; (F) Electric Motor; (G) Keyphasor [4]

3. Description of experiment procedure

Figure 2 shows the methodology for identification the unbalance of the rotor system. Suppose that two unbalance disks are mounted on the rotating shaft.

Firstly, the rotor is run at a speed of 4000 rpm. Next, the run-down test is performed to speed of 500 rpm. This test is known as the reference test. After that, an arbitrary mass is added to the disk #1 (1° balancing plane) and disk #2 (2° balancing plane). The run-down tests are repeated. The following steps are the procedure of the experiment tests:

1. Measure the vibrations of the rotor for one or more unbalanced rotation speeds, on different planes of measurement in the $x$ and $y$ direction (reference tests).
2. Add a known mass arbitrary at the 1° balancing plane.
3. Measure the new vibrations $X^*(\Omega_j)$ at 1° balancing plane.
4. Calculate the coefficients of influence $a(\Omega_j)$ for the $j$-th rotation speed.
5. Repeat steps 2-3-4 for the 2° balancing plane.
6. Assembly the matrix of influence coefficient $[C]$.
7. Obtain a global matrix of coefficients $[H]$.
8. Estimate the 2 masses of imbalance that reproduce the experimental vibration of the rotor.
9. Place masses in opposite position to balance the rotor (add 180° to the phase).

**Step 1:** After measuring the rotor vibrations in the time domain, performing order tracking and extracting 1X component $X_i(\Omega_j)$ of the rotor. The velocity vector is $\{\Omega\} = \{\Omega_1 \ldots \Omega_i \ldots \Omega_N\}^T$ and the 1X component corresponding to each speed is $X(\Omega_j) = \{X_{NDEx}(\Omega_j), X_{NDEy}(\Omega_j), X_{DEx}(\Omega_j), X_{DEy}(\Omega_j)\}^T$.

**Step 2:** By adding a known mass to arbitrary $w_j$ balancing plane, the force due to the imbalance $U_w$ can be obtained by:

$$U_w = m_w r e^{i\phi}$$

$$F_w = m_w \Omega^2 e^{i(\phi+\Omega t)}$$

where $m_w$, $r_w$, $\phi_w$ is mass, radial distance from the axis and angular position (phase) of mass.

In this test rig, the masses available are 0.1, 0.2, 0.4, 0.8, 1.0, 1.2, 1.6, 2, and 2.2 grams and the positions available on the disks are 0° - 360° in steps of 22.5°.

**Step 3:** Measure the new vibrations $X^{**}(\Omega_j)$ with the added mass

$$X^{**}(\Omega_j) = \{X_{NDEx,w}(\Omega_j), X_{NDEy,w}(\Omega_j), X_{DEx,w}(\Omega_j), X_{DEy,w}(\Omega_j)\}^T$$

the difference $X^{**}(\Omega_j) - X(\Omega_j)$ vibrations due to the imbalance between the original plus additional mass $X^{**}(\Omega_j)$ and vibration due to imbalance only original $X(\Omega_j)$ represent the vibrations from one additional mass.

**Step 4:** Calculate the coefficient of influence $a_{zw}(W_j)$ for the $j$-th rotation speed:

$$a_{zw}(\Omega_j) = \frac{X^{**}(\Omega_j) - X(\Omega_j)}{U_w}$$

$$= \frac{m_w r e^{i\phi}}{m \Omega^2 e^{i\Omega t}}$$

The coefficient of influence is the z-th vibration 1X exclusively due to a unified force with zero phase acting on the $W$-th balancing plane. For the first plane, we have:

$$a_{NDEx,1}(\Omega_j) \quad a_{NDEy,1}(\Omega_j) \quad a_{DEx,1}(\Omega_j) \quad a_{DEy,1}(\Omega_j)$$

**Step 5:** Repeat the steps above to obtain the coefficients of influence for the second balancing plane

$$a_{NDEx,2}(\Omega_j) \quad a_{NDEy,2}(\Omega_j) \quad a_{DEx,2}(\Omega_j) \quad a_{DEy,2}(\Omega_j)$$

**Step 6:** Building the global matrix of influence coefficients for the $j$-th rotation speed $[C(\Omega_j)]$. Under hypothesis of linearity of the system, for the $j$-th rotation speed, we have:

$$\begin{bmatrix}
Y_{NDEx}(\Omega_j) \\
Y_{NDEy}(\Omega_j) \\
Y_{DEx}(\Omega_j) \\
Y_{DEy}(\Omega_j)
\end{bmatrix} = \begin{bmatrix}
a_{NDEx,1}(\Omega_j) & a_{NDEx,2}(\Omega_j) \\
a_{NDEy,1}(\Omega_j) & a_{NDEy,2}(\Omega_j) \\
a_{DEx,1}(\Omega_j) & a_{DEx,2}(\Omega_j) \\
a_{DEy,1}(\Omega_j) & a_{DEy,2}(\Omega_j)
\end{bmatrix} \begin{bmatrix}
m_1 r_1 e^{i\phi_1} \\
m_2 r_2 e^{i\phi_2}
\end{bmatrix}$$

where $Y_j(W_j)$ is the vibration due to the additional mass $W$-th applied measured in the z-axis.

**Step 7:** Repeat the previous steps at various speeds to obtain N matrices $[C(\Omega_j)]$, then be assembled into a single global matrix of coefficients $[H]$. 

[3]
\[ \{y\} = [H]\{\theta\} \]

\[ \{y\} = \{y(\Omega_1), \ldots, y(\Omega_i), \ldots, y(\Omega_m)\}^T \]

\[ [H] = \begin{bmatrix} C(\Omega_1) & \cdots & C(\Omega_i) & \cdots & C(\Omega_m) \end{bmatrix}^T \]

\[ \{\theta\} = \begin{bmatrix} m_1 r_1 e^{i\phi_1} \\ m_2 r_2 e^{i\phi_2} \end{bmatrix} \]

**Step. 8**: Matrix \([H]\) is the experimental model of the system for different speeds. This matrix allows to simulate the response 1X system for two different masses placed on two distinct planes balancing. The two balancing masses to be placed will have the same value of those of unbalance estimated, the same radial distance and an angular position offset by 180° from these positions. The estimation problem is overdetermined (more equations than unknown parameters):

\[ \{y_{\text{rotor to be balanced}}\} = [H]\{\hat{\theta}_{\text{unbalance masses}}\} \]

Assumed an estimate for the masses of unbalance \(\{\Theta\}\), vibration simulated obtained with the estimated model \([H]\) will be different from the imbalance measured experimentally \(\{Y\}\). The residual value is defined as the difference between quell vibration measured experimentally and the simulated:

\[ \{r\} = \{Y\} - \{\hat{Y}\} = \{Y\} - [H]\{\hat{\theta}\} \]

The solution of the problem is overdetermined obtained through a least squares approach, minimizing the sum of residues:

\[ \{\hat{\theta}\} = \min_{\{\theta\}} \left( \sum_{i=1}^{n} r_i^2 \right) = \min_{\{\theta\}} \{r\}^T \{r\} \]

\[ \{\hat{\theta}\} = (\{H\}^T [H])^{-1} \{H\}^T \{Y\} \]

4. Results

Figure 3 shows the 1X components of four proximity probes (NDEH, NDEV, DEH, DEV) of three tests (reference test and tests with added mass #1 and mass #2). It is clearly that the test with added mass #2 have very large 1X components for all proximity probes.

![Figure 3. 1X components of the rotor for unbalanced rotation speed](image-url)
In order to obtain this 1X components, the acquired signals were applied synchronous averaging after performing order tracking by using a tacho signal. The main advantage of this method is that the noise of 1X signal can be reduced by 70%.

Figure 4 shows the influence coefficients $\alpha_1$ and $\alpha_2$ at each proximity probes from speed of 500 rpm to 4000 rpm. Finally, by applying the algorithm presented above step, two suitable masses and their positions can be obtained as: mass #1 = 0.00063kg @ 147° and mass #2 = 0.00088kg @ 288°.

![Influence coefficients at four proximity positions](image)

**Figure 4.** Influence coefficients at four proximity positions

Figure 5 shows the 1X components of four proximity probes before and after balancing as a function of rotor speeds. It is clearly that the 1X components have been significantly reduced for all proximity probes, especially in the range of resonance.

![1X components of the rotor before and after balancing vs. rotation speeds](image)

**Figure 5.** 1X components of the rotor before and after balancing vs. rotation speeds

The proposed experimental setup for vibration measurement and imbalance fault detection in rotating machinery was successfully tested and is ready for application in “real world” systems.

5. Conclusions
An experimental test was performed on a small test rig with two mounted disks to identify the unbalance in rotor using order tracking technique based on one key phasor. The rotor is first run at speed of 4000 rpm and then, the run-down test is performed to 500 rpm. The coefficients of influence $\alpha$ of each experimental test, the matrix of influence coefficient [C] and the global matrix of coefficients [H] can be obtained in order to identify the suitable masses including the value and the position to reduce the unbalance of the rotor systems. This paper only presented the run-down test. The run-up test should be carried out to compare the results between two tests.
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