How Housing Dynamics Affect the Monitoring of Rotor Unbalance: A Case Study

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When monitoring faults in a rotating system, the rotor support and housing have an important effect on the measured dynamic response. Within this contribution, we show the effect of stiffness orthotropy in the supports when monitoring the unbalance of a rotor. The study is illustrated with an industrial case, namely a cooling blower for a drive system. The considered application was evaluated in a new and a defective state by measurements using radially distributed accelerometers to answer two questions:

1. May a suspected unbalance be detected at different operating conditions by measuring on the housing?
2. Which problem can appear in estimating the rotor unbalance when the support has orthotropic stiffnesses?

Thereby, we figured out that the suspected unbalance can be well detected only between global resonance frequencies of the application. The correlating modes superimpose the acquired signals, initially evoked by unbalance excitation. Due to these modes, a phase shift of the measured unbalance signals in correlating measurement directions occurs. Thus, forward and backward whirl expresses themselves within the rotational frequency in the sensor signals. Due to these effects, the knowledge of global modes is a requirement for unbalance monitoring.

1 Introduction

Vibration-based monitoring of rotating machinery by measuring on the housing gains in importance for industry. Especially when considering integrated sensor systems, due to wireless transmission capacity, external measurement points are mandatory. Particularly, within the scope of monitoring various rotor fault types, as described in [1], structural dynamics of the entire system must be well known. Figure 1 shows an exemplary application, where measuring at the outside housing is an industrial requirement due to wireless sensor data transmission of the vibration-based monitoring system. The depicted drive cooling blower sucks the air in axial direction, and expels it on the drive via a centrifugal principle. During operation, the rotor resonances are coupled with the struts and housing’s eigenfrequencies. Despite of occurring resonances at specific operational speeds, the feasibility of unbalance monitoring will be evaluated hereinafter.

2 Operational Measurements

In order to get an impression of the blower’s vibration resulting from a suspected unbalance at the whole operating range, we measured the system response during a rotor run up. For data acquisition, Bruel & Kjaer 4397 piezoelectric accelerometers in combination with a National Instruments frontend PXIe-4492 (8 kHz sampling frequency) were used. The waterfall diagram is acquired via Fast Fourier Transform (10 s measurement time for each rotational frequency, 10-60 Hz, 1 Hz steps) and shown in figure 3. The measurement position and direction is depicted in figure 2 (schematic top view of the blower housing), whereby the circle represents the housing’s top ring with centred rotor (not shown). Looking at the waterfall plot, the first
Fig. 2: Measurement position and direction.

Fig. 3: Run up waterfall diagram of a blower with suspected unbalance excitation (marked 1st, 11th and 22nd order).

Order resonance is well visible at about 42 Hz - excited by an expected unbalance. Furthermore, we see the 11th and 22nd order. These rotational speed-dependent amplitude super-elevation occurs due to 11 impeller blades and its twofold excitation frequency by reason of flow channel division.

Next, the sensor setup was changed as shown in figure 4 with two orthogonal measurement positions and directions. Within these measurements, we also evaluated a new blower in addition to the system with suspected unbalance. Therefore, we chose 35 Hz as operational speed, in order to stay far enough away from the first order resonance shown in figure 3. For all the following orbit plots, a FIR bandpass filter (+/- 5 Hz of the considered operational speed) was used, applying MATLAB’s filtfilt function to eliminate phase shift errors. The resulting acceleration orbits are plotted on the right.

Comparing the orbit plots in figure 4, the amplitude difference is remarkable. We also see that the circle is closed, containing only the operational speed frequency, which correlates with the unbalance excitation. At this point, our unbalance assumption is verified. A special feature emerges, if we have a look at the rotation direction of the signals. It is oppositely orientated to the sense of rotation. In section 3, the effect will be discussed. Before that, we consider two further operating conditions. First, the maximum operational speed at 60 Hz above the 42 Hz resonance is considered (figure 5). We recognize a rotation direction reversion of the acquired signal, because the circle is drawn in rotational sense of the rotor. Furthermore, a huge difference between the two elliptical shaped orbits occurs.

Fig. 4: Left: measurement setup top view with sensor positions and directions at the outer housing. Right: resulting acceleration orbits of two blowers at 35 Hz rotational speed (black: new blower, blue: blower with suspected unbalance).
This form stems from the bending direction associated to the modes. The resonance seems to have still an influence at 60 Hz, but we need an evidence. That prove is brought by the next orbit plot in figure 6. For this purpose, the measurement position and direction was rotated to the depicted x-y system. The y-direction was already considered within the waterfall plot (figure 3). Now, the orthogonal x-direction was added due to an assumption of two orthogonal mode shapes. The measurements were performed on the blower with unbalance using two additional operational speed conditions (22 Hz and 42 Hz).

These speeds were selected so as to validate the mode shape at 42 Hz and to show a second resonance, which must be located below due to the rotation direction reversion effect. The result is clear: we see two approximately orthogonal modes with different amplitude amplifications at the operating speeds. The plotted amplitude difference is justified by lower exciting unbalance force at 22 Hz. In addition to that, the great frequency difference between the two modes can be explained by lower x-direction stiffness of the overall system (considering the schematic drawing in figure 1, the geometry-dependent horizontal rigidity is evident). At these stiffnesses, we will have a closer look hereinafter.

3 Coupled Effects

The two different blower stiffnesses in the x-y plane are related to the rotor support. This can be understood from the transfer functions between rotor and measurement position at the outside housing, which represent the inverse dynamic stiffnesses. These differ from each other regarding x- and y-axis direction. Figure 7 shows a simplified analogous model of a flying supported rotor with two different bearing stiffnesses.

Shaft stiffness $c$ & bearing stiffness $k_x \neq k_y$

Fig. 7: Simplified analogous model of a flying, orthotropic supported rotor.

Fig. 8: Analogous model Frequency Response Function with neglected damping. Response amplitudes $\hat{w}_x(\Omega)$, $\hat{v}_x(\Omega)$ with excitation frequency $\Omega$, two resonances $\omega_x$, $\omega_y$ and eccentricity $\varepsilon$. 
This kind of support is orthotropic, whereby \( k_x \) and \( k_y \) represent the rotor suspension including blower’s struts and housing. For the simplified model, we assume a homogenous, symmetric shaft which can be represented by a lumped stiffness \( c \) and mass \( m \) connecting the supports. Concerning the mass position, figure 7 is just a schematic depiction with respect to the flying rotor support of the real application. If we perform an eigenanalysis of the model, we obtain the two resonances

\[
\omega_x = \sqrt{\frac{c}{m} \frac{2k_x}{2k_x+c}} \quad \text{and} \quad \omega_y = \sqrt{\frac{c}{m} \frac{2k_y}{2k_y+c}}.
\]

These eigenfrequencies expresses themselves as vertical lines in the Frequency Response Function (see figure 8). Here, The black line characterizes the frequency response amplitude \( \hat{w}_x(\Omega) \) in x-direction with resonance \( \omega_x \) whereby the blue line \( (\hat{v}_x(\Omega)) \) shows a corresponding response in y-direction with resonance \( \omega_y \). For the schematic representation in figure 8, we only consider unbalance excitation with an eccentricity \( \varepsilon \) and neglect damping effects. Up to \( \omega_x \), x- and y-direction respond in phase (operational speed of the application below 22 Hz). Above this first resonance, we can see a 180 degree phase shift of \( \hat{w}_x(\Omega) \) (operational speed of the application between 22 Hz and 42 Hz). Only after \( \omega_y \), the amplitudes respond in phase again (operational speed of the application above 42 Hz). These phase shifts express themselves in the sensor signal when measuring at the blower in x- and y-direction. This effect can also be explained based on [2] as an increasing and abating of again (operational speed of the application above 42 Hz). These phase shifts express themselves in the sensor signal when measuring at the blower in x- and y-direction. This effect can also be explained based on [2] as an increasing and abating of.

For the simplified model, we assume a homogenous, symmetric shaft which can be represented by a lumped stiffness. For that, we use \( \hat{w}_x \) and \( \hat{v}_x \) in the time-domain. Here, they describe the elliptical half-axes

\[
\hat{w}_x(t) = \frac{\varepsilon \cdot \Omega^2(t)}{\omega_y^2 - \Omega^2(t)} \quad \text{and} \quad \hat{v}_x(t) = \frac{\varepsilon \cdot \Omega^2(t)}{\omega_x^2 - \Omega^2(t)}
\]

and the complex value

\[
r_x(t) = \hat{w}_x(t) + j\hat{v}_x(t) = \hat{w}_x \cos \Omega t + j\hat{v}_x \sin \Omega t.
\]

whereby \( \Omega \) is the current rotational speed and \( \varepsilon \) the unbalance mass eccentricity. If we apply the Euler Equations, the shaft centre movement can be written as

\[
r_x(t) = \frac{1}{2} (\hat{w}_x + \hat{v}_x) e^{j\Omega t} + \frac{1}{2} (\hat{w}_x - \hat{v}_x) e^{-j\Omega t} = \hat{r}_+(t) e^{j\Omega t} + \hat{r}_-(t) e^{-j\Omega t}
\]

with the circular shaft movements \( \hat{r}_+(t) \) and \( \hat{r}_-(t) \) in sense and contrary to the rotor turning direction. Depending on these two superimposed orbits, the higher amplitude is responsible for the whirling orientation: If \( |\hat{r}_+(t)| > |\hat{r}_-(t)| \), the shaft orbit is elliptical and rotates in forward whirl. If \( |\hat{r}_+(t)| < |\hat{r}_-(t)| \), the shaft also moves on an elliptical shape but in backward whirl. If \( |\hat{r}_+(t)| = |\hat{r}_-(t)| \), we can observe almost oscillating linear motion (see figure 6). By inserting the half-axes (equation 2) into relation 4, we obtain the amplitudes of forward and backward whirling movement ratio

\[
\hat{r}_+(t) = \frac{\varepsilon \Omega^2}{2} \frac{\omega_x^2 + \omega_y^2 - 2\Omega^2}{(\omega_y^2 - \Omega^2)(\omega_x^2 - \Omega^2)} \quad \text{and} \quad \hat{r}_-(t) = \frac{\varepsilon \Omega^2}{2} \frac{\omega_x^2 + \omega_y^2}{(\omega_y^2 - \Omega^2)(\omega_x^2 - \Omega^2)}.
\]

4 Conclusion

We show the effect of an orthotropic elastic supported rotor on an industrial example and verify a suspected unbalance within a monitoring task. This phenomena has to be considered, if the whole operation range including appearing resonances is regarded. At these eigenfrequencies, the vibration alignment at the outer housing is only orientated in mode shape direction. Hence, unbalance can exclusively be monitored away from the resonances.

Acknowledgment

The contribution was funded by the Federal Ministry of Education and Research within the project "AMELI 4.0: Micro-Electro-Mechanical Electronics System for Condition Monitoring in Industry 4.0" (ID: 16ES0442).

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