A group of overhung seal air fans in a coal-fired generating plant experienced high rate of bearing failures. Balancing these fans proved to be very difficult. Run-up and coast-down data indicated that the fans were operating near the 1st critical speed. Although impact tests performed on the fan shafts showed a natural frequency more than 25% below the running speed, but due the gyroscopic stiffening effect, the 1st critical speed of the fan actually fell within 6% of the running speed. Insufficient bearing support stiffness was found to be the primary cause of the problem. The proposed solutions included lighter impeller wheel and stronger bearing support and frame. Both of these options were implemented and tried out. The latter option was preferred due to lower costs.

1. INTRODUCTION

Seal air fans are required to provide filtered air (at a higher pressure than that inside the mill) to the mill labyrinth seal to prevent pulverized coal from escaping to the atmosphere. There are three seal air fans in each unit of the multi-unit coal-fired power plant. Two are required to provide seal air for the unit five mills. The overhung centrifugal fan is driven by a 100 HP, 3600 RPM motor through a flexible coupling (Falk Steelflex). The fan impeller wheel is approximately 30" in diameter and 6" in width, and weights approximately 140 lb. The fan shaft is 34 1/2" long and 2 11/16" in diameter. The fan is running on self-aligning ball bearings with 16" bearing separation and 9 3/8" overhang distance. Both the fan and motor are mounted on a steel frame consisting of 1/4" thick steel plates and angle braces (Figure 1).

Figure 1: Seal air fan arrangement
High vibrations had caused regular bearing failures on a group of these fans (let’s call them “bad fans”). The highest bearing vibration of up to 25 mm/s pk (1 in/s pk) or more occurred on the fan out board (OB) in the horizontal direction (Figure 2) and was at least twice higher than that in the vertical direction. The predominant component was 1xRPM. However, strong harmonics of up to 4th order were also present. In some cases the 2x line frequency (120 Hz) also showed up.

These fans were very sensitive to unbalance. Balancing correction weights were often in the range of 2 to 4 grams. Maintenance good alignment was also proven to be difficult. In the early days, various modifications on the frame were carried out with little success. Switching positions between floating and fixed bearings sometimes improved the vibrations, but only for a short period. Motor soft-foot correction could only reduce to the 2xLF component.

The remaining fewer fans (let’s call them “good fans”) were running trouble free with bearing vibrations remained below 4 mm/s pk (Figure 3). There were no apparent differences between the bad and good fans.

An investigation into the cause of the high vibrations was carried out in 2000. The purpose of the investigation was to recommend a long-term solution for the problem. This paper presents the testing and analysis done during the investigation, as well as the implementation and results.
2. IMPACT TESTS

Figure 3: Velocity spectra taken on the fan OB bearing of a “good fan”

Figure 4: Compliance FRF curves on different frames at the bearing OB (top half) in the horizontal direction
Impact tests were done on the frames, motors and bearing casings on a number of bad and good fans. There was no indication of natural frequencies (structural resonances) in the vicinity of 60 Hz or 120 Hz. However, it was noticed that the horizontal stiffness on the OB bearing housing of a good fan was generally higher than that of a bad fan (Figure 4). The difference in stiffness can be attributed to the frame structure integrity. Although all frames were of the same design, they have not deteriorated equally over the more than 20 years in service.

Impact test results in dynamic stiffness (1/Compliance) are presented in the below table:

**Table 1:** Bearing support dynamic stiffness $K_d$ (E6 lbf/in) @ 59.5 Hz in the horizontal direction

<table>
<thead>
<tr>
<th>Location</th>
<th>Bad fan</th>
<th>Good fan</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fan Brg OB (top half)</td>
<td>0.283</td>
<td>0.545</td>
</tr>
<tr>
<td>Fan Brg OB (bottom half)</td>
<td>0.484</td>
<td>0.737</td>
</tr>
<tr>
<td>Fan Brg IB (top half)</td>
<td>0.604</td>
<td>1.01</td>
</tr>
<tr>
<td>Fan Brg IB (bottom half)</td>
<td>0.905</td>
<td>1.65</td>
</tr>
</tbody>
</table>

Impact tests were also performed on the fan shafts. It was again noticed that the horizontal 1st (static) mode of the shaft of a bad fan was generally 6 to 10 Hz lower than that of a good fan (Figure 5). It was clear that the higher horizontal stiffness of frame at the OB bearing of a good fan contributed to its higher 1st mode of the fan shaft.

![Figure 5: Compliance FRF curves on the good and bad fan shafts in the horizontal direction](image)

### 3. 1X RPM OPERATION DEFLECTION SHAPE (ODS) TESTS

Three directional ODS tests were carried on a bad and a good frames. Measuring points were chosen to be the same on both fans (Figure 6). The total number of points was 54.
Figure 6: ODS measuring points

Point Description | 1 & 10 | Fan OB & IB bearings, respectively
| 2/45 & 11/54 | Fan OB & IB bearing base, respectively
| 49 & 25 | Motor OB & IB bearings, respectively
| 50, 51, 52, 53 | Motor feet
| Remaining points | on the frame as shown

The results are shown below. Figures 7 and 8 each displays two successive 3-D displacement frames, all of the same scale. It was noticed that the fan bearing support plate enclosed by points 3, 12, 38 and 46 was much more flexible on the bad frame than that on the good frame. It appeared that the stiffness of this plate on the bad frame was insufficient due to degraded structure assembly.

Figure 7: 1X RPM ODS on a bad frame
4. RUN-UP AND COAST-DOWN TESTS

Figures 9 and 10 below show the shutdown 2\textsuperscript{nd} order responses of the two fans. These indicated that the 1\textsuperscript{st} critical speed of bad fan was 3360 RPM (less than 6\% away from the running speed) and that of a good fan was 4000 RPM (12\% away from the running speed). It was noticed that these resonant frequencies were higher than those obtained from impact tests (2580 CPM on the bad fan and 3180 CPM on the good fan) due to gyroscopic (stiffening) effect. Both of these fans just had the bearings replaced, so the difference in critical speeds could only attributed to the difference in the bearing support conditions between the two fans.

![Figure 8: 1X RPM ODS on a good frame](image)

![Figure 9: 2\textsuperscript{nd} order coast-down on the shaft in the horizontal direction of a bad fan](image)
5. CRITICAL SPEED ANALYSIS

A rotor model (Figure 11) was created to study the fan dynamic responses. A rotor dynamics program based on Finite Element Analysis (FEA), DyRoBeS Rotor, was used. The model consisted of 20 station rotors with the assumed rigid disc mounted at station 19. The disc polar and transverse moments of inertia were 37.34 lbf.s².in and 20.25 lbf.s².in, respectively as provided by the fan manufacturer. The fan IB and OB bearings were located at stations 6 and 14, respectively. In addition, the bearing supports shown as station 20 and 21 were included in the model. The bearing stiffness was assumed to be 1.5e6 lbf/in based on 500 lbf of bearing load. If damping is negligible, bearing support stiffness $K_s$ can be estimated based on the formula $K_s = K_d + m_s w^2$ where $K_d$ is the dynamic stiffness, $m_s$ is the mass of the support (approx. 200 lbs) and $w$ is the fan speed. Based on the dynamic stiffness results (Table 1), bearing support stiffness is estimated as follows:

Table 2: Bearing support horizontal stiffness $K_s$ (E6 lbf/in)

<table>
<thead>
<tr>
<th>Location</th>
<th>Bad fan</th>
<th>Good fan</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fan Brg OB (top half)</td>
<td>0.286</td>
<td>0.548</td>
</tr>
<tr>
<td>Fan Brg OB (bottom half)</td>
<td>0.487</td>
<td>0.740</td>
</tr>
<tr>
<td>Fan Brg IB (top half)</td>
<td>0.607</td>
<td>1.04</td>
</tr>
<tr>
<td>Fan Brg IB (bottom half)</td>
<td>0.908</td>
<td>1.69</td>
</tr>
</tbody>
</table>

Furthermore, the top and bottom bearing halves were considered springs in series. Thus, the following bearing stiffness values were used for the analysis.
Table 3: Effective bearing support horizontal stiffness $K_s$ (E6 lbf/in)

<table>
<thead>
<tr>
<th>Location</th>
<th>Bad fan</th>
<th>Good fan</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fan OB</td>
<td>0.180</td>
<td>0.315</td>
</tr>
<tr>
<td>Fan IB</td>
<td>0.364</td>
<td>0.644</td>
</tr>
</tbody>
</table>

It was clear that the system effective stiffness was governed by the bearing support stiffness.

Analysis without gyroscopic effects done on the bad fan resulted in 2514 rpm for the first mode (Figure 11). This is within a few percentages error from the impact test result (2580 rpm). As well, analysis without gyroscopic effects done for the good fan resulted in its first mode of 2960 rpm. This also agrees well with the impact test result (3120 rpm).

**Figure 11**: First mode of a bad fan, WITHOUT gyroscopic effects

With gyroscopic effects, the model was modified to have the IB and OB effective bearing stiffness values of $0.3\times10^6$ lbf/in and $0.2\times10^6$ lbf/in, respectively. This would yield a first critical speed of 3141 rpm (Figure 12) and is closely matched with the case of the bad fan for the rundown test result (3180 rpm).
6. MODIFICATION OPTIONS

Various fixed options were considered. Bearing relocation was ruled out as it was deemed to be unpractical. The one obvious option was to stiffen the fan OB bearing support. This idea was thought of before, however, the stiffening had been done by adding steel plates to the sides of the frame (enclosed by points 3, 5, 15 & 12 and 46, 48, 40 & 38 of Figure 6). It was clear from the ODS results that the horizontal plate supporting the OB bearing was the weakest section of the frame. Adding steel members to the side of the frame (as previously done) would not increase the horizontal stiffness at the OB bearing. Stiffeners should have been added horizontally and vertically beneath the fan bearing support plate. However, it was determined that a new and stronger frame would be more cost effective than frame modification. Stiffness test (Figure 4) on the new frame indicated more than 4 times higher in horizontal stiffness value at the OB bearing when compared to that on a bad frame. So far, only one frame was replaced. The fan with the new frame has been running trouble free for almost 2 years.

The impeller wheel of another fan was also replaced with an aluminum construction wheel with less than 1/3 of the steel wheel weight. A critical speed analysis was done for the same model, but with the revised disc weight of 45 lbf and polar and transverse moments of 12.01 lbf.s².in and 6.51 lbf.s².in, respectively. The analysis indicated that the aluminum wheel would allow the fan to run at least 12% below the first critical speed for an effective bearing stiffness greater than 0.135e6 lbf/in (Figure 13). This means that the aluminum wheel fan would run well even on a bad frame. However, a bad frame tends to get worse with decreasing bearing support stiffness. This eventually brings down the fan first critical speed near its running speed. As such, the aluminum wheel was not recommended for fans with bad frames.
7. CONCLUSIONS

A group of overhung seal air fans operated with high vibrations. The high vibrations generated large dynamic loading on the bearings that cause them to fail prematurely. Although these fans were designed to operate well within the sub-critical region, insufficient bearing support had put them to run within 6% of the critical speed. With very little damping, the amplification factor was as much as 30. As such, these fans were very sensitive to imbalance (< 1 gram/100 µm pk-pk at the OB bearing). It was determined that the bearing support plate was too weak, especially at the OB. Solutions found included lighter impeller wheel and stronger frame. The following points are noted:

- Due to the gyroscopic (stiffening) effect, the first critical speed of an overhung fan can be much higher than the result obtained from an impact test.
- In many cases where equipment is mounted on steel frame, the bearing effective stiffness is governed by the bearing support stiffness.
- When involved in critical speed problem, dynamic analysis should be done to ensure proper and effective solutions.

8. ACKNOWLEDGEMENT

The author would like to thank John Macfarlane for his valuable contribution in all the testing conducted.