Feature Article

Vibration Characteristics of Overhung Primary Air Fans

Summary. Vibration characteristics of several primary air fans are presented in this article. Reasons were sought for high rotor and pedestal vibrations that caused bearing failures and for difficulties encountered when balancing the fans.

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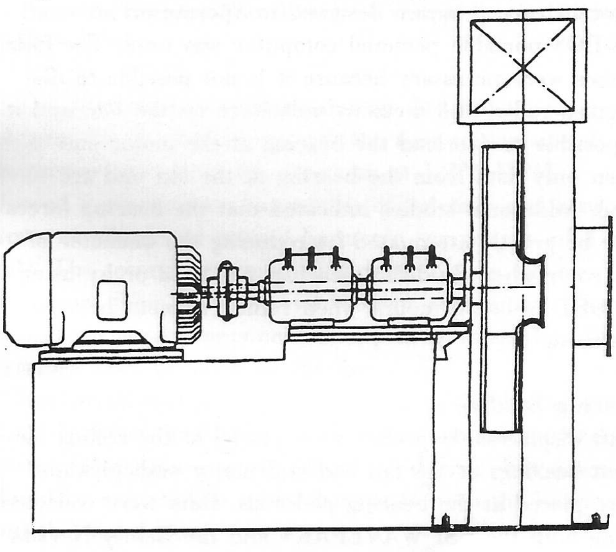
Four overhung primary air (p/a) fans were a continuous operational problem. High rotor and pedestal vibrations caused numerous bearing failures; in addition, the fans were difficult to balance. The fan assembly, which weighs about 1,500 pounds and operates at 1,800 RPM, supplies air to boilers.

Early Studies

The fan was designed with fluid-film journal bearings. Figure 1 is a schematic outline of a typical installation with the bearings mounted on a concrete foundation. Figure 2 is a schematic cross section of the standard Sturtevant fluid-film bearing arrangement and positive disc lubrication. The bearing span of the original primary air fan was 22 in.; the overhang distance was about the same. The weight of the fan wheel, 1,000 pounds, caused a static loading of 2,000 pounds and an uploading of 1,000 pounds on the bearing at the motor end. This bearing, which was not designed for an uploading, had reduced load capacity when uploaded because of the large oil supply groove.

Figure 1. Schematic Outline of Overhung Primary Air Fan.



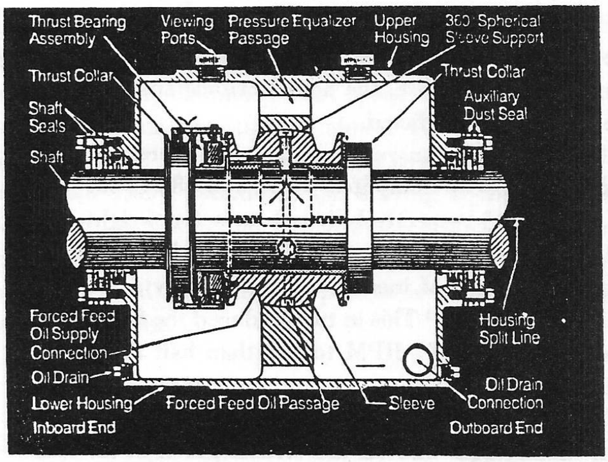
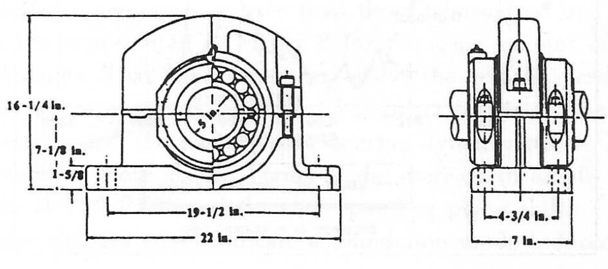


Figure 2. Sturtevant Fluid-Film Bearing with Standard Thrust Arrangement and Positive Disc Lubrication.

The original fluid-film bearings caused high pedestal vibration that ruptured cooling water lines to the bearing caps. The bearings were replaced with the antifriction rolling element bearings shown in Figure 3. It was assumed when the bearings were replaced that the fans were operating in the subcritical speed region. After the rolling element bearings had been installed, however, one

Figure 3. Cooper Split 02 Series Rolling Element Bearings.



fan became very sensitive to minor unbalance. This high sensitivity occurred as a result of the low damping in the system due to the rolling element bearings and the fact that the fan was operating close to a critical speed.

It is standard practice in the design of this class of fan that the critical speed be 20% above the operating speed range. But tests on the balance-sensitive fan showed that, with rolling element bearings, it was operating slightly above the first critical speed.

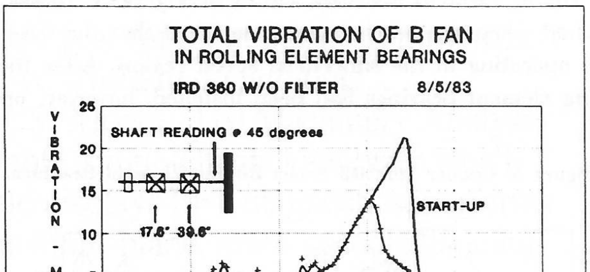
In a critical speed analysis undertaken in 1983 a bearing stiffness value of 2 E6 lb/in. was assumed. The calculated first critical speed of 3,577 RPM was based on the assumption that the fan wheel was rigid. However, this fan impeller was known to be very flexible. Because a flexible disc cannot transmit the theoretical gyroscopic moment of a rigid disc, the actual critical speed was lower than that predicted.

The polar and transverse moments of inertia for the rigid fan had been computed to be 583,685 lb-in.^{2} and 330,733 lb-in.^{2} respectively based on a fan weight of 1,010 pounds. In order to simulate a flexible fan disc, the polar moment of inertia was reduced by a factor of ten, to 58,368 lb-in.^{2} This in turn reduced the first critical speed from 3,577 RPM to less than half that value, or 1,723 RPM.

This calculation shows dramatically the importance of the flexibility of a disc on the critical speed of an overhung rotor. Because pedestal mass and support stiffness play an important role in reducing overall bearing impedance, the flexibility of the pedestals also had to be considered.

At the time of the first critical speed analysis in 1983, no experimental data were available to establish the critical speed of the fan in rolling element bearings. Vibration data were later obtained with an IRD 360 analyzer with the filter out during start-up and shut down of one fan (see Figure 4). Vibration was recorded with an IRD shaft rider located between the fan bearing and the air

Figure 4. Total Vibration of Fan Using IRD 360 Analyzer.



seal at a 45° angle from the vertical. Note the critical speed of 1,720 RPM on start-up and the amplitude of 22 mils.

The rapid decrease in amplitude after the fan passed through the first critical speed is typical of nonlinear bearing behavior under high rotating loads. On shut down the critical speed decreased to 1,520 RPM because the motor-driven drive torque had been removed and loading effects were being caused by a misaligned motor fan. No phase angle measurements were taken at this time.

In an attempt to overcome the balance sensitivity of the fans due to the rolling element bearings, the overhang of the fan was reduced to raise the first critical speed of the rotor so that the fan would operate below the critical speed. In several fans the bearing span was increased from 22 in. to more than 26 in. These fans had high vibration levels and greatly reduced bearing life – bearings on two fans were replaced on average every six months. Thus, even though the rotor might have been operating below its first critical speed, high forces could still occur on the bearing because the dynamic transmissibility ratio – the ratio of transmitted bearing forces to rotating unbalance forces – was always greater than one. This ratio can vary from 20 to 50 or more with this fan under certain conditions. The implication is that the transmitted bearing forces could exceed the rotating unbalance load by 20 times or even more.

In another attempt to decrease the balance sensitivity, fluid-film bearings were re-installed in one fan, and the bearing span was reduced to the original 22 in. Experiments indicated that the fluid-film bearings actually provided greater stiffness than the rolling element bearings. The fan was operating above the first critical speed, but the excellent damping provided by the fluid-film bearings allowed the rotor to be balanced more easily than rotors in rolling element bearings.

A single plane balance was successfully carried out on the fan using vertical and horizontal measurements – four probes – on the motor and the fan. A least-squared error balance program designed to operate on an MS-DOS portable personal computer was used. The four probes were necessary because it is not possible to distinguish radial and moment unbalance on the fan, and it is possible to overload the bearing at the motor end when only data from the bearing at the fan end are used. Additional studies indicated that the bearing forces may be greatly attenuated by reducing the diameter of the rotor when fluid-film bearings are used or by using a LORD isolation mount when rolling element bearings are used.

Bearing Studies

Shaft displacement probes were placed at the rolling element bearings at the fan end and motor end; pickups were placed at the bearing pedestals. Data were collected using both the CSI WAVEPAK® and the Bently Nevada

Digital Vector Filter II. Data from the vertical displacement probe at the fan bearing during a coast down are shown in the cascade diagram (Figure 5). Total motion is plotted against time in Figure 6. (The signal is clipped because the maximum amplitude for the wave form was set at 12 mils.)

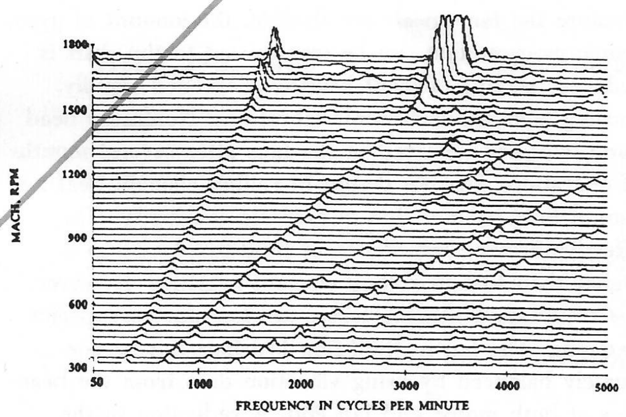


Figure 5. Frequency Spectrum of Vertical Bearing During Coast Down.

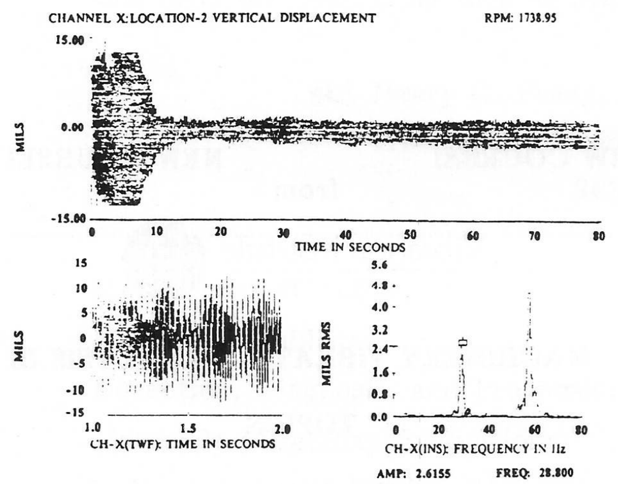


Figure 6. Transient Time History of Vertical Bearing During Coast Down.

The increase in amplitude during shut down occurred because the fan passed through the first critical speed. A frequency analysis of the spectrum at 1,728 RPM showed a synchronous magnitude of 2.4 mils and a two times amplitude, or second harmonic component, of 5.6 mils. The second harmonic component on the fan was thus more than twice the synchronous component. The second harmonic component disappeared when fluid-film bearings were installed on the fan.

Horizontal motion of the bearing pedestal during shut down is shown in Figure 7. The very high instantaneous horizontal motion – up to 50 mils – during shut down would be expected to cause damage to both the bearing and the foundation.

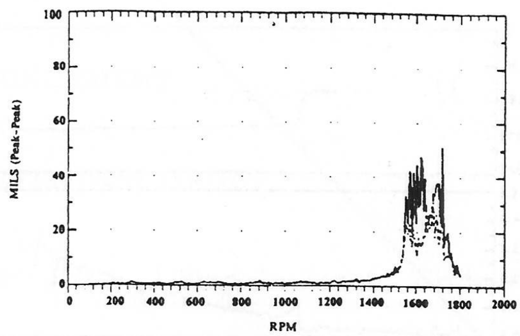


Figure 7. Motion of Horizontal Probe at Bearing Pedestal During Coast Down.

Synchronous amplitude and phase caused by rotor unbalance recorded during shut down are given in Figure 8. Note the distinct critical speed at 1,680 RPM and the synchronous amplitude of almost 10 mils. Such a high amplitude could easily cause a dynamic load of more than 4,000 pounds on the bearing at the fan end. Balancing this fan would be extremely difficult for two reasons – the fan is operating close to a critical speed and the second harmonic component is greater than the synchronous one.

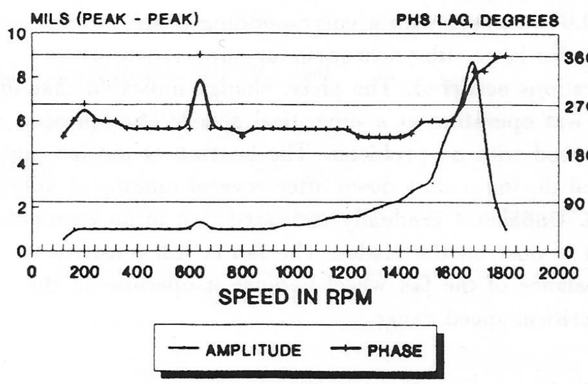


Figure 8. Synchronous Amplitude and Phase of Bearing During Coast Down Measured with Horizontal Probe.

The synchronous motion of another fan in rolling element bearings, in which the bearing span had been increased from 22 in. to 26.75 in., is shown in Figure 9. The span change reduced the fan overhang, thus raising the critical speed above the operating speed range. The amplitude caused by unbalance increased with speed. At 1,800 RPM the amplitude was almost 4.0 mils. The dynamic forces of the bearing at the fan end caused by unbalance are much greater than those transmitted to the fan represented in Figure 8 for the same amount of unbalance. That fan is operating above the critical speed. The fan represented by Figure 9 is operating below the critical speed and has a higher bearing dynamic transmissibility. Note also in Figure 9 the increase in amplitude at 720 RPM and the corresponding phase shift. These changes could indicate a foundation mode induced

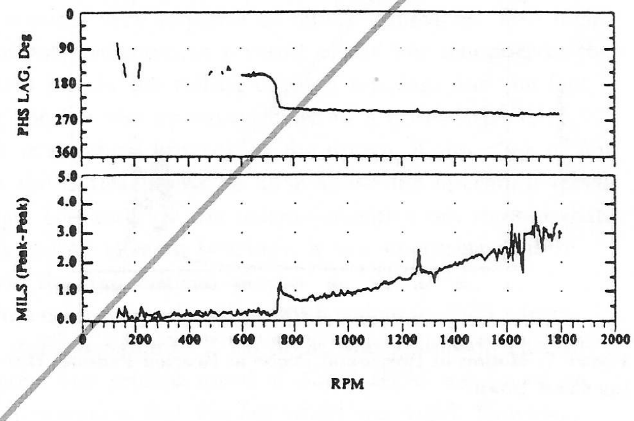


Figure 9. Synchronous Amplitude and Phase of Bearing During Coast Down Measured with Vertical Probe.

by the failure of the grouting under the bearing pedestals of the fan.

For another study rolling element bearings were removed from the fan with the extended bearing span and replaced with fluid-film bearings. The span was returned to the original condition of 22 in. Velocity measurements on the bearing cap were recorded during start-up (Figure 10). Pedestal motion increased gradually to 0.076 in./sec with a corresponding phase increase of 60°. No large subsynchronous or supersynchronous vibrations occurred. The phase change indicated that the fan was operating at a subcritical speed. The fan was balanced with no problems. The bearing at the fan end failed during a shut down after several months of operation. Unbalance gradually increased due to an accumulation of dust on the blades. The fan is still sensitive to unbalance of the fan wheel because it operates in the subcritical speed range.

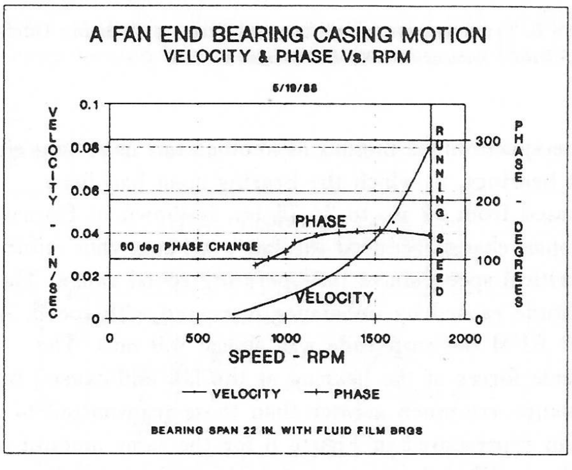


Figure 10. Bearing Housing Measured with Horizontal Probe During Start-Up.

Conclusions

The overhung primary air fans were designed to operate at subcritical speed rotors. However, they do not operate 20% below the first critical speed as originally designed. Accurate critical speed calculations could not be made on the basis of rigid disc theory with this class of fan wheel. Because the fan wheels are flexible, the amount of gyroscopic moment that can be transmitted to the shaft is reduced. Rolling element bearings degraded rapidly, causing high superharmonic vibrations. Nonlinear dead band behavior occurred in the fans after several months of operation. The fan design in rolling element bearings cannot hold balance due to extreme sensitivity. The effective bearing impedance in fluid-film bearings is higher than that in rolling element bearings. However, the transmitted bearing forces are less due to superior damping. The fan in fluid-film bearings can be adequately balanced by using vibration data from the bearings at both motor and fan end. A reduction in the sensitivity of a fan to unbalance and in transmitted bearing forces can be achieved by reducing the diameter of the overhang.