EPRI SYMPOSIUM ON
POWER PLANT FANS: THE STATE OF THE ART

CONTROLLING FAN VIBRATION - CASE HISTORIES

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Sponsored by

ELECTRIC POWER RESEARCH INSTITUTE

October 14-18, 1981
Hyatt Regency Indianapolis
Indianapolis, Indiana

Key Words:
Vibration
Foundation Natural Frequency
Dynamic Soil Properties
Multi-plane Balancing
Lateral Critical Speed
Disc Resonance
Inlet Vortex
Rotating Stall

CONTROLLING FAN VIBRATION - CASE HISTORIES

ABSTRACT

Donald R. Smith and J. C. Wachel

Fan vibration problems have been a serious cause of plant unreliability in large fossil-fired power plants and have resulted in operational problems, shutdowns, and reduced generation. The basic causes of most problems are dynamic resonances associated with the system. These have to be identified before practical and effective recommendations can be made for corrective action or design modifications. The most effective solutions can best be determined from computer models which match the measured field data.

This paper discusses several case histories and illustrates methods and instrumentation for analyzing existing fan problems. Experimental techniques are described for defining problem symptoms, and these are related to root causes by the use of data analysis and canputer simulation techniques. Computer modeling techniques can then be used to evolve reliable fixes or used in the design stage to simulate interaction between rotor dynamic response and complete system response to prevent potential dynamic problems.

Section 1

INTRODUCTION

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J. C. Wachel

Multiple fan systems are typically complex and vibration problems can be difficult to define due to dynamic acoustical and mechanical interaction between fans [1]. When fans are installed in multistoried buildings, the problems are further complicated due to structure-borne vibrations. Sanetimes it appears that "everything shakes," including the structure, floor, fans, ducts, and motors. Plant personnel are often frustrated in attempting to correlate the countless number of variables in the system to determine the exact cause of increases in vibration levels. Often the vibration increases are correlated to seemingly unrelated variables such as time of day, rainfall, or load changes on other units.

In the analysis of severe fan vibration problems, it is often found that the high vibrations are due to coincidences of one or more natural frequencies which amplify low level excitations. A list of possible excitation sources and natural frequencies generally found in fan systems is presented below:

EXCITATION SOURCES

Mechanical Defects: misalignment, improper tie-down, bad grout, bent shaft, shaft rub, warped thrust collar, loose rivets on wheel, bad shrink fit between wheel and hub, etc.

Unbalance: dust or ash buildup on blades or inside airfoil blades, blade erosion, thermal distortion (shaft bow, wheel warpage), inadequate balancing at low speeds

Pulsation: flow excitation across obstructions, vortex shedding, inlet box vortex, rotating stall, blade passage

NATURAL FREQUENCIES

- shaft lateral critical speed
- torsional critical speed
- pedestal, foundation, floor/support structure
- disc wobble (wheel rocking)
- acoustical

SYSTEM ANALYSIS

The first step in defining and solving vibration problems is to determine if the vibrations are due to (1) high level excitations, or (2) low level excitations which are amplified by the coincidence of one or more natural frequencies. If the excitations are high, the solution is generally to reduce the energy level. If coincidences of resonance exist, the solution is to modify the system to change the natural frequencies away from the excitation frequency. The most difficult vibration problems to analyze and solve are those which have several resonances at the same frequency which increases the cross coupling and interaction between the resonances.

Detailed field tests coupled with analytical analyses are generally required to separate and identify the excitation sources and their amplitudes and the system natural frequencies. The analysis techniques and field instrumentation have been greatly improved in the last few years. Equipment which used to be considered for laboratory use only can now be easily transported to the field for on site data analysis. A typical field instrumentation setup is shown in Figure 1-1 which includes a minicomputer, real time analyzer, trim balance analyzer, multi-channel FM recorder, oscilloscopes, signal conditioning amplifiers and integrators, and X-Y plotters. In complex problems minicomputers are used to gather and process several data channels simultaneously and plot the vibration mode shapes. Multi-channel telemetry systems can be used to collect data such as strain and temperature on the shaft and blades during on-line conditions.

The objective of this paper is to illustrate techniques to identify and solve fan vibration problems; however, these techniques are also routinely applied to solve vibration problems on other types of rotating and reciprocating equipment. Sane of the more unique fan vibration problems analyzed by SwRI have been selected for this paper to illustrate the diagnostic procedures and solution development.

Section 2

CASE HISTORIES.

CASE NO. 1

During the initial startup of the ID fans at a large power plant, the fan and motor bearing housing vibrations were greater than the specified maximum allowable level of 2 mils peak-peak at the running speed of 900 rpm.

The following symptoms were observed:

1. During the field balancing by the fan manufacturer, the fans were sensitive to small balance weights.

2. The vibrations were greater on the motor than the fan.

3. When the motor was run uncoupled, the motor vibrations were less than 1 mil but the fan outboard bearing housing vibrations were greater than 3 mils.

4. The horizontal vibrations at the top of the foundation were also above the desired limit.

5. The fan vibrations were sensitive to the inlet guide vane setting.

6. The vibrations of each fan were affected by the vibrations of the adjacent fans even though they were not installed on a common foundation mat.

7. Temperature changes in the flue gas had a significant effect on the vibrations.

8. The vibrations on one of the fans increased after a heavy rain.

Detailed investigations revealed that the fans were overly sensitive to small changes in unbalance, load, and temperature due to amplification from operating near two resonances. The first foundation natural frequency and the fan shaft installed resonant speed were both near the fan running speed. The natural frequencies could not be adequately separated without major modifications; therefore, the vibration levels were reduced by hot balancing each fan.

FOUNDATION NATURAL FREQUENCY

Vibration levels measured on the fan bearing housings during startups and coastdowns revealed a natural frequency just below the running speed (Figure 2-1). The vibration mode shape was obtained by measuring the vibrations on the bearing housing,. bearing pedestal, and several locations on the side of the concrete foundation. Vibration data were plotted on a scaled drawing (Figure 2-2). The entire foundation

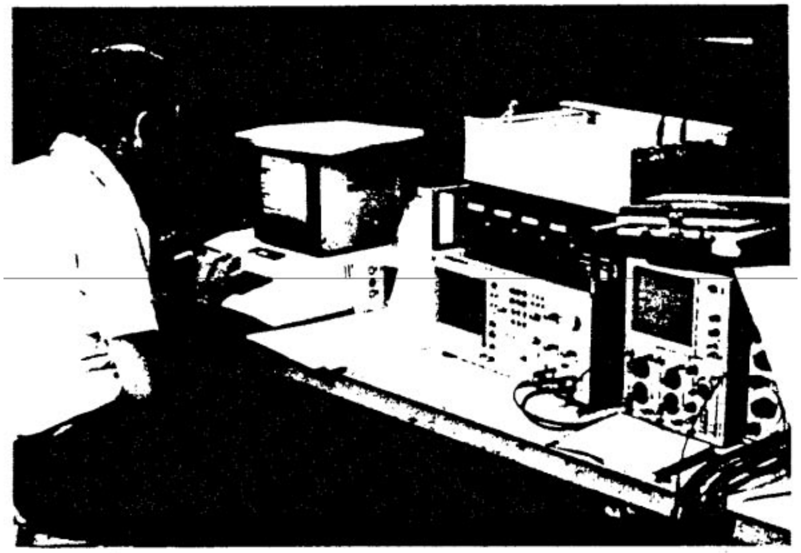


Figure 1-1. Typical Field Equipment Used in Vibration Analysis

and bearings moved together as a rigid body and rocked about a point several feet below the foundation which is characteristic of a foundation resonance.

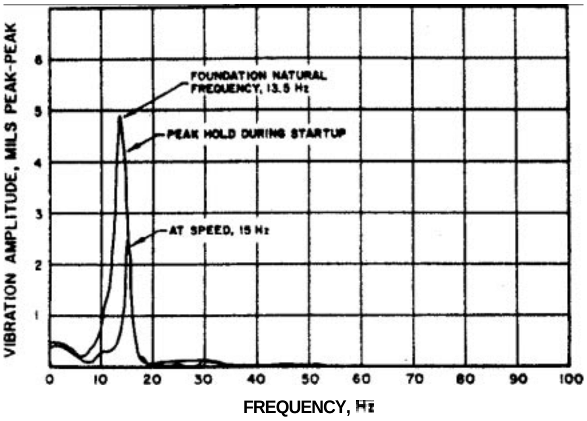
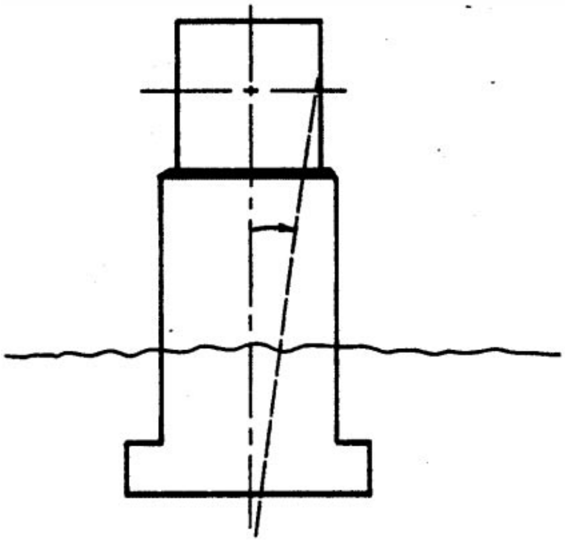


Figure 2-1. Spectral Analysis of Bearing Housing Vibrations During Startup



FOUNDATION ROCKING MODE (θy)

Figure 2-2. Fan Foundation Vibration Mode Shape

On three of the fans, the foundation natural frequency was just below the running speed, but on the fourth fan, the resonance was slightly above the running speed. To verify that the measured responses were foundation resonances, a variable speed, unidirectional, mechanical shaker was attached to the foundation and run through a speed range of 0 to 1800 rpm. The shaker-excited vibration response verified that the first foundation natural frequency ranged from 12 to 16 Hz for these four "identical" fans. The fan with the foundation resonance at 16 Hz was the most sensitive to small changes and had the greatest amplification at the running speed of 900 rpm (15 Hz).

These foundations were designed to have their lowest foundation natural frequency at approximately 1500 cpm (70% above running speed). Foundation calculations were made in the design stage using a value of Young's Modulus (E) of 500,000 psi for the sandstone beneath the foundations.

To evaluate the discrepancy between the measured and calculated foundation natural frequencies, the foundation-soil dynamic system was modeled on a computer program developed by SwRI. Foundation natural frequencies were calculated for a range of soil moduli and compared with the shaker data. As shown in Figure 2-3, an effective soil modulus of 80,000 to 120,000 psi was required to match the measured foundation natural frequencies.

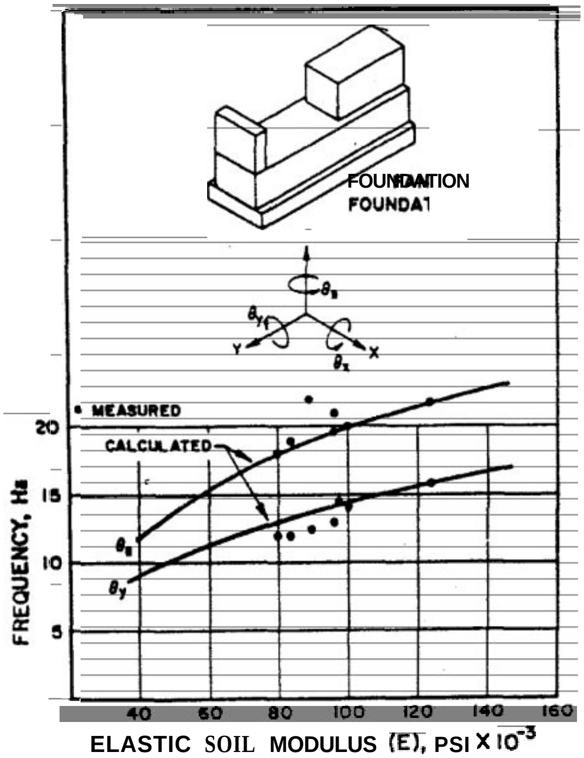


Figure 2-3. Comparison of Measured and Calculated Foundation Natural Frequencies

These effective E values as empirically defined are much lower than the typical values given for sandstone which can range from 500,000 to 3,000,000 psi. The original soil analysis report revealed that the soil was highly stratified, and the depth to each layer varied considerably over short distances. The soil modulus beneath the foundations could also have been reduced due to blasting and over-excavation when the foundations were constructed.

This case history illustrates that the major unknown in predicting the resonant frequency of a fan-foundation system is the effective soil modulus. In the design of these fans, a minimum value for the soil modulus was obtained from soil bore tests, but even this minimum value was approximately five times too large.

The field data indicated that fan vibration levels were also affected by moisture content of the soil because the vibrations would change significantly after a rain. Moisture changes the soil modulus and can shift the foundation natural frequency closer to the running speed which amplifies the vibrations. Also, the damping can be reduced due to the cohesion of the soil and separation from the concrete caused by vibration.

Since the foundation dynamic design is highly dependent upon the E value of the soil, every effort should be made in the design stage to obtain effective dynamic soil modulus directly below the foundation. There are several methods which can be used to obtain more accurate dynamic soil data, as given below:

1. A large output, low frequency shaker can be used to excite Rayleigh ground waves at the foundation elevation. The soil modulus can be calculated from this data [2]. The effective soil modulus should be obtained after the site has been excavated and the fill added.

2. The soil modulus can also be obtained by using cross hole tests. Two holes are bored to the bottom of the foundation [2]. A shear wave is created by an impact at the bottom of the hole and measured with a transducer in the other hole which is a known distance away. This method could also be used to determine the effective modulus at the bottom of piles.

3. A shaker can also be used to excite the natural frequencies on a small or partially completed foundation. The partially completed foundation can then be modeled on a computer and the effective soil modulus can be determined by comparing the calculated natural frequencies with the measured frequencies.

A shaker was used on a partially completed foundation block of an adjacent unit (Figure 2-4) to determine the effective soil modulus. The tests indicated that the effective soil modulus would cause the foundation natural frequency to be near the fan running speed. Foundation modifications to reduce the vibrations were analyzed using the computer program (e.g., increase mat size, piles, added mass, etc). Most of the foundation modiflcations investigated were ineffective, were not cost effective or were space limited. Other methods were then investigated to reduce the vibrations.

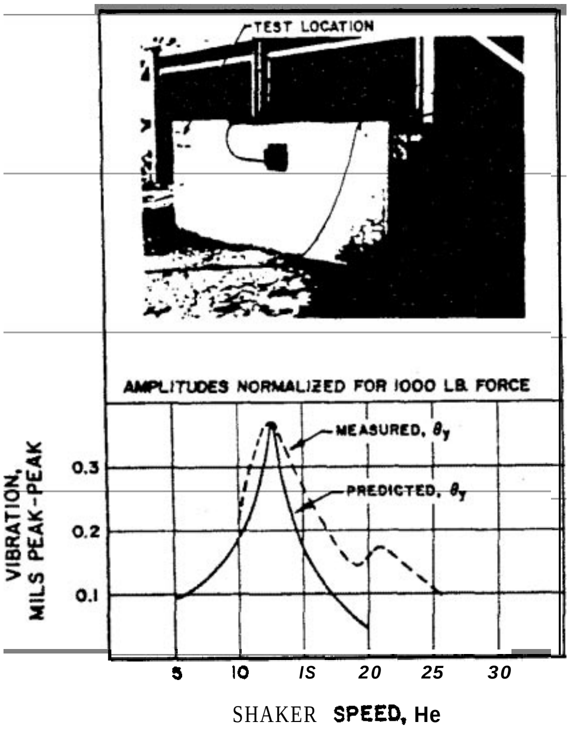


Figure 2-4. Comparison of Measured and Calculated Foundation Vibration Response

Shaker tests on one foundation revealed that the vibrations were increased by a factor of six when dirt was removed from the side of the foundation. Based upon this test, sand bags were temporarily placed against the side of the foundation to increase the damping and lateral restraint. Vibrations were reduced by a factor of approximately two to one.. These tests indicated the significant effect of soil damping in the overall design of foundations. Foundation vibrations can also be reduced in some cases by shortening the height of the foundation above the ground or designing 'the foundation in the shape of a truncated pyramid instead of a tall rectangular block as is usually done.

FAN SHAFT LATERAL CRITICAL SPEED

The fan shaft lateral natural frequency was measured to be 960 cpm using a variable speed shaker attached to the foundation. For these tests accelerometers were attached to the fan shaft and proximity probes were mounted at the bearings to separate the rotor response from the foundation response. The fan installed resonant speeds were also calculated as shown in the critical speed map (Figure 2-5) where the shaft natural frequencies are plotted versus the effective support stiffness. The effective support stiffness includes the stiffnesses of all the springs from the rotor to ground including the oil film, bearing pedestal, foundation, and soil.

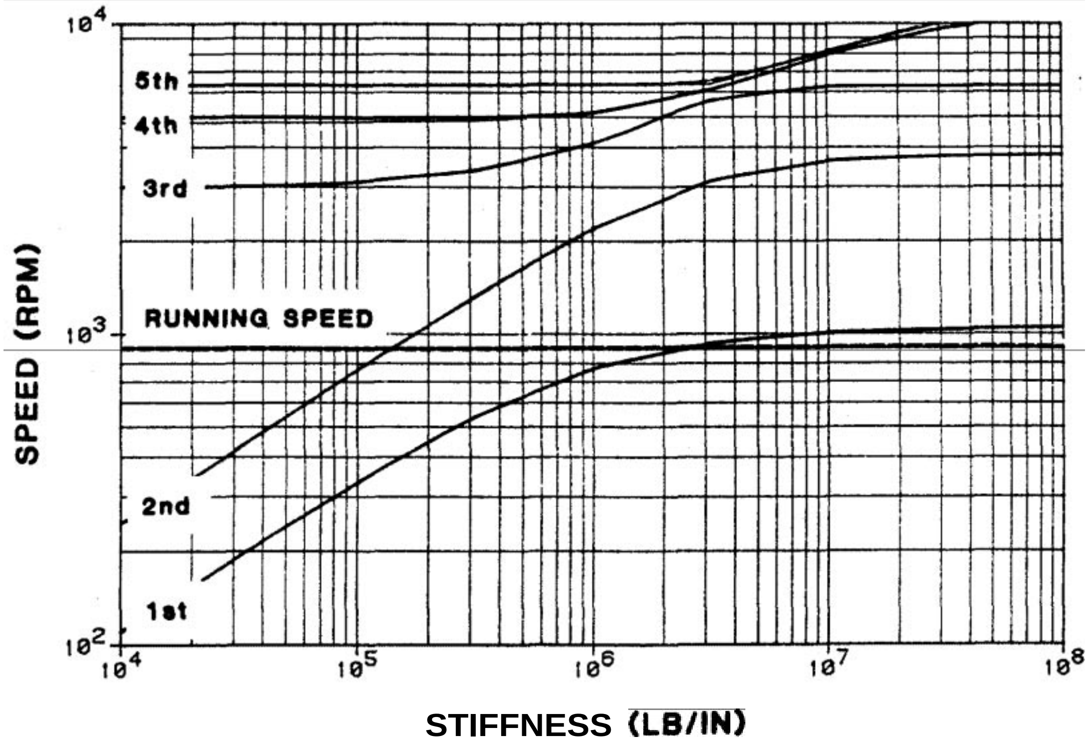


Figure 2-5. ID Fan Critical Speed Map

The fan manufacturer had calculated the critical speed to be 1180 rpm; however, the calculations were based on a "rigid bearing critical". For many years fan manufacturers used this calculation which assumes the rotor is mounted on rigid supports because many of the supporting stiffness values were not known. As a result, there has been some confusion in the fan industry as to the exact definition of critical speed and resonant speed. To clarify the problem, the Air Movement and Control Association (AMCA) [3] adopted the following definitions:

> Critical Speed: A critical speed is that speed which corresponds to the natural frequency of the rotating element (impeller and shaft assembly) when mounted on rigid supports. (Note: This is generally referred to as the rigid-bearing critical speed.).
>
> Design Resonant Speed: Design resonant speed is that speed which corresponds to the natural frequency of the combined spring-mass system of the rotating element, oil film, bearing housing, and bearing supports but excluding the foundation (foundation stiffness is considered as infinite).
>
> Installed Resonant Speed: Installed resonant speed is that speed which corresponds to the natural frequency of the combined spring-mass system of the rotating element, oil film, bearing housing, bearing supports, and includes the effect of foundation stiffness.

The calculated rigid-bearing critical was 1180 rpm and the calculated installed resonant speed was 960 rpm. Normally the installed resonant speed should be at least 20% from the running speed to prevent excessive vibration amplification, In this case the calculated installed resonant speed was only 7% above the running speed; therefore, amplification could be expected. The rigid-bearing critical speed is not applicable because in the real world the rotor is not rigidly supported. The actual stiffness of the concrete and pedestal for this system was approximately 5×10^6 lb/in which is much less than rigid.

A forced vibration response analysis is normally required to accurately determine the rotor installed resonant speed. The rotor should be modeled using the oil film and the combined spring-mass system of the bearing housing, bearing supports, and foundation. The bearing oil film for journal bearings is usually represented by horizontal, vertical, and cross coupling stiffness and damping terms. The cross coupling terms significantly affect the calculated rotor response, particularly if the horizontal stiffness is much less than the vertical stiffness. in these cases the rotor installed resonant speeds cannot be accurately calculated if the oil film is represented as a single stiffness and damping value and the effects of the cross coupling terms are omitted.

Therefore, it is important that fan users be aware of the critical speed definitions used in the fan industry and refer to the installed resonant speed when writing design specifications.

For this systen major modifications such as shortening the bearing span would have been required to increase the installed resonant speed to 20% above the running speed. Since these modifications were not practical, it was decided to reduce the shaft vibrations by improving the fan balance.

SYSTEM BALANCE

It was desired to reduce the vibrations on the fan and motor bearing housings by adding balance weights only in the fan. This means that the vibration levels on a11 four bearing housings must be considered when calculating the balance weights.

To determine if the vibration amplitudes were stable and repeatable, the vibration amplitudes were plotted versus the phase angles as shown in Figure 2-6. The variation in amplitude and phase angle for each bearing created elliptical patterns due to the interaction from adjacent fans. The best balance was obtained by reading the vibration amplitude and phase angle at the center of each elliptical pattern.

On one fan a particularly strong interaction from an adjacent fan was observed and a stable vibration phase angle could not be obtained. When the adjacent fan was tripped, the fan vibrations immediately stabilized (Figure 2-7). To further document this interaction, the fan vibrations were plotted with the key phaser installed on the adjacent fan which showed that the fan vibrations were more closely related to the adjacent fan.

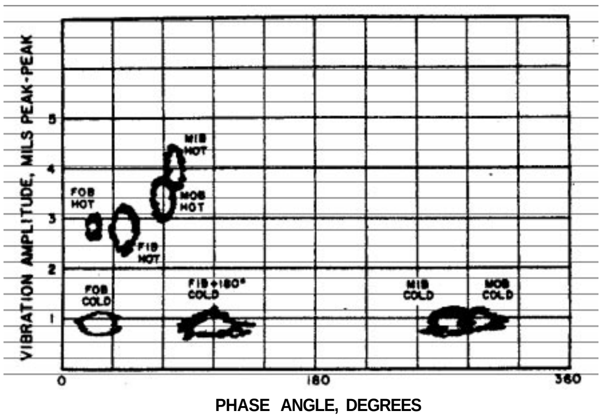


Figure 2-6. Variation of Motor and Fan Bearing Vibration During Hot and Cold Conditions

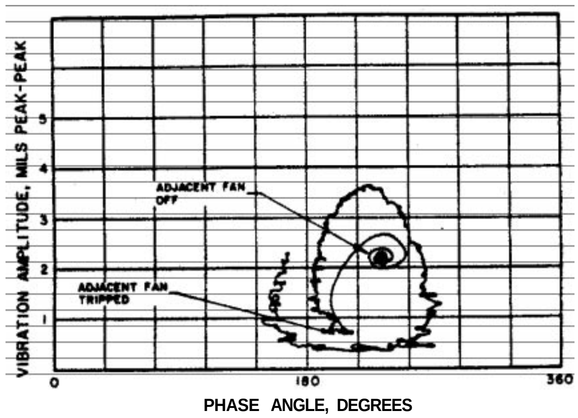


Figure 2-7. Fan Vibration Interaction with Adjacent Fan

Even with the interaction between the fans and the amplification due to the foundation natural frequency and installed resonant speed, the vibrations on the fans and motors were reduced below 2 mils cold using a two-plane balance (Figure 2-6). In the balancing procedure, a least squares balancing technique was used which includes the vibrations at all four bearing locations for the two balance planes that were available.

HOT BALANCE

When the boiler was fired during the initial steam blows and the temperature of the exhaust gases in the ID fan reached 250° F, the vibrations on the bearing housings immediately increased and the phase angles changed (Figure 2-6). The balance of a large fan normally changes when the fan is heated due to thermal distortion; however, if the fan installed resonant speed and foundation natural frequency are near the running speed, the small changes in unbalance can result in significant increases in vibration amplitudes.

It was therefore recommended that the system be "hot balanced" to reduce the bearing housing vibrations to meet the required vibration limits. Hot balancing fans with the plant at full load is a difficult, time consuming process. One of the major problems was that the fans did not have turning gears and the rotors bowed due to thermal stratification of the gases in the fan housing. When the fans were restarted with a bowed shaft, the vibrations were excessive and the fans were immediately tripped, Each startup and coastdown reduced the bow due to more uniform heating of the rotor, Sometimes after two or three starts and coastdowns, the vibrations would be reduced to levels where the fan could be operated.

The shaft thermal bow problem was later solved by installing a temporary turning gear to rotate the shaft when the fan was off. It is highly recommended that turning gears be included in the original design on all hot fans because it is difficult and expensive to retrofit turning gears.

It took approximately 10-12 hours of running time before the fan thermally stabilized and the vibration and phase angle quit changing. Balance data taken before this time could result in inaccurate balance weights.

The fan and motor vibrations were reduced by hot balancing; however, the fans remained very sensitive to small changes. The balance weights used were much less than the allowable residual balance as specified for fans of this type [4].

The bearing housing vibrations could not be maintained below the specified level of 2 mils; however, there was no bearing damage because the foundation moved with the shaft. It was found that the bearing housing vibration levels could be increased to 5 mils peak-peak without causing excessive shaft-to-housing vibrations. The plant operated for several years with these vibration levels without bearing damage. This illustrates that allowable vibration levels must be determined for each individual piece of equipment and published vibration criteria should be used as a guide or reference.

CASE NO. 2

Two boilers were converted from forced draft systems to balanced draft systems with the addition of two large ID fans on each unit. The ID fans were originally installed with louver controls which were later changed to variable inlet vanes to eliminate high pulsations in the ducts. The variable inlet vanes apparently reduced the pulsations in the fan discharge; however, the fan and motor bearing vibration increased to unacceptable levels when the units were run above half load.

The following symptoms were observed:

1. After a few days, the fan bearing housing vibrations would increase in amplitude from 2 mils to over 5 mils and the phase angle would shift over 180 degrees.

2. Hot balancing the fans was not effective in reducing the vibrations for a long time period.

3. The groundborne interaction between the fans was excessive and only one fan could operate at a time.

It was determined that the fan wheel disc wobble natural frequency was slightly above the running speed which amplified low level excitations and resulted in excessive vibrations. The problem was corrected by raising the disc wobble natural frequency further above the running speed by adding a stiffener plate to the center plate.

DISC WOBBLE

The bearing housing vibrations and shaft vibrations were recorded during several startups and shutdowns (Figure 2-8) to determine if the fan was operating near a mechanical natural frequency. It is known that vibrations due to unbalance alone (discounting resonances) will increase as a function of the speed squared. However, the vibration at 900 rpm was significantly greater than the level due to a pure unbalance, which indicated that the fan was operating near a natural frequency. The lack of a 180 degree phase shift on the bearing housing vibrations indicated that this resonance was probably not a shaft critical speed.

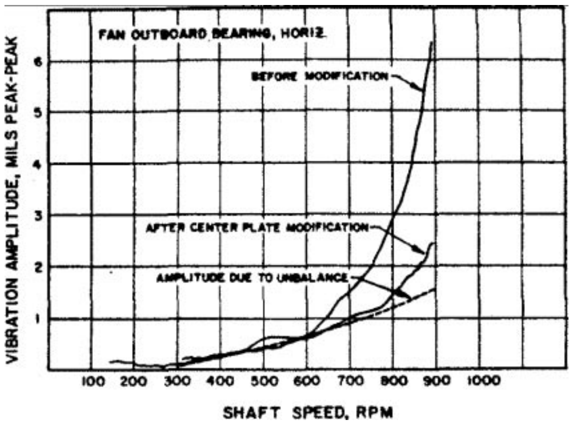


Figure 2-8. Bearing Housing Vibration Response During Startup

Pulsations measured in the discharge duct were synchronous and phase coherent with shaft speed. The pulsation amplitudes were approximately 0.7 psi at the fan running speed and produced a force equivalent to a 10 oz unbalance at the outside radius of the fan. The fans were apparently sensitive to small pulsations or aerodynamic excitations because changing the shaft dust covers had a significant effect on the fan vibration characteristics. It is difficult to reduce low level pulsations; therefore, it was felt that the long range solution would be to identify and modify the mechanical natural frequency which was amplifying the vibrations.

To identify the resonant frequency, a large variable speed shaker was bolted to the concrete foundation with the shaking force applied in the horizontal direction and run through a frequency range from well below the fan running speed to 50% above. Vibrations were measured on the foundation, bearing housings, shaft, and fan wheel to identify the resonance just above the running speed.

The foundation response was found to be highly damped and offered no significant amplification of the vibrations at the running speed. The major effect of the foundation was as a vibration transmission path between adjacent fans.

The fan shaft natural frequency was measured to be 1200 cpm with the shaft resting on the bearings. Calculations indicated that the installed resonant speed when the shaft was running on the oil film would be lowered to 1080 cpm which was still adequately above the running speed.

Vibrations measured on the fan wheel indicated a disc resonance at 930 cpm (Figure 2-9). There was a classic 180 degree phase shift in the vibrations as the shaker speed increased above the resonant frequency. The vibrations were measured at several locations on the wheel to determine the vibration mode shape.

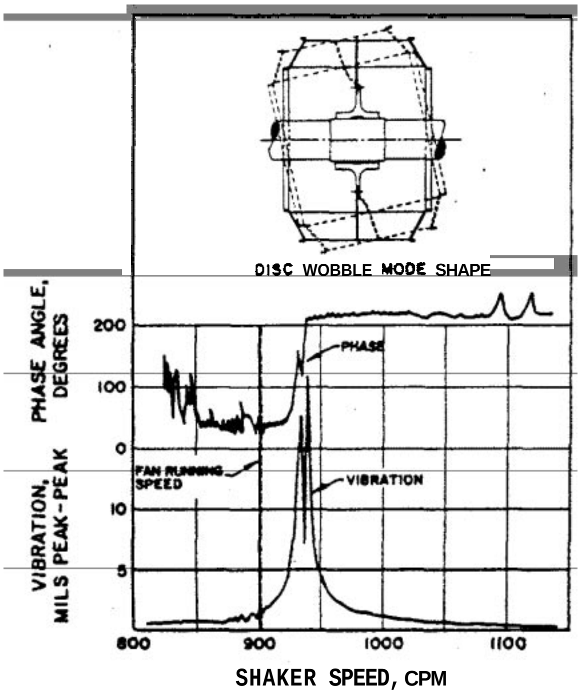


Figure 2-9. Disc-Wobble Mode Vibration Response to Shaker Excitation

The shaker test illustrated that the axial wheel vibrations are easily coupled into horizontal vibrations on the bearing housing because the wheel resonance was excited by a horizontal shaking force on the foundation.

It was felt that the bearing housing vibration amplification at running speed could be reduced by raising the disc resonant frequency. To evaluate the effect of stiffening the center plate, wedges were driven between the fan impeller and the inlet cone to restrain the wheel. When the shaker test was repeated, the response at 930 cpm was eliminated on the fan wheel and the bearing housings.

An evaluation was made of a plate stiffener that could be fabricated and bolted to the center plate in the field to raise the disc resonant mode. Since it is difficult to mathematically model a complex bolted joint, a small 1/10 scale model was fabricated and tested. The model test confirmed that the center plate stiffness could be adequately increased by bolting a reinforcement plate to the center plate.

After a stiffener plate was installed in one fan, a shaker test was performed which showed that the wheel natural frequency increased 25%. When the fan was restarted, the bearing housing vibrations were significantly reduced as shown in Figure 2-8. The other fans were then similarly modified and have been in operation for several years and are no longer sensitive to low level excitations even at full load.

CASE NO. 3

During the initial startup of two ID fans at a power plant, high vibration levels were observed on the fan inlet housing which resulted in structural damage requiring extensive repairs. One of the inlet cones was broken away from the common sheet and several of the pipe braces which extend between the common sheet and the side sheets were also broken. The unit had been in operation for only 11 days during the initial firing and steam blows of the boiler. The second fan had run for only 4 days and it too had suffered extensive damage similar to the first fan.

The following symptoms were observed:

1. The vibrations on the fan inlet housing were excessive and increased as a function of the inlet damper position.

2. The vibrations on the fan bearing housing (pedestals) and foundation were low and were not affected by the inlet housing vibrations.

The problem was due to an inlet vortex which created a rotating stall condition and generated high pulsations at multiples of 20 Hz. The pulsations at 40 Hz matched the mechanical natural frequency of the inlet cone and resulted in high vibrations and fatigue failures of the inlet cone and the internal pipe braces. The inlet vortex was eliminated by adding splitter plates in each inlet.

Accelerometers, strain gages, and pressure transducers were installed at several critical locations on the fan housing to identify the source of excitation. It was found that the pulsation, vibration, and strain levels increased as a function of the damper position and the levels were excessive at a damper position of approximately 70% open. Strain levels of 350 in/in x 10^{-6} peak-peak were measured which were higher than the allowable for this material and configuration.

The predominant pulsation and strain frequencies were at multiples of 20 Hz (Figure 2-10). Rotating stall generally occurs at multiples of 1/3 shaft speed. This fan ran at 15 Hz and the pulsations at 20 Hz were at 4/3 shaft speed. The stall was due to marginal flow or improper preswirl conditions at the fan inlet which caused the air to impinge on the fan blade at a poor angle of attack.

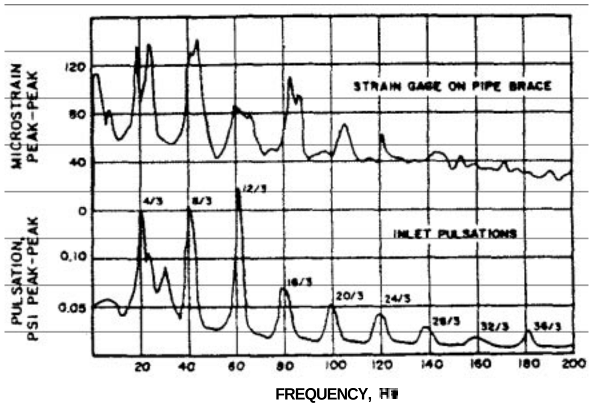


Figure 2-10. Inlet Pulsation and Strain Spectra Showing Multiples of Rotating Stall Frequencies

Several modifications suggested by the manufacturer were made to the ducts and turning vanes upstream of the fan, but no reduction of pulsation or vibration occurred. Testing was then conducted to determine the effect of discharge throttle control. Flow through most fans is controlled on the inlet side and outlet dampers are used for fan isolation. For this test, the outlet dampers were temporarily set up for manual control. The inlet dampers were opened to 45% which resulted in high pulsations and vibrations. The inlet dampers were left on automatic control and the outlet dampers were partially closed, keeping the flow rate constant. When the outlet dampers closed to 55%, the inlet opened to 51% and the vibrations and pulsations immediately reduced. This test suggested that a fixed orifice in the discharge duct could reduce the pulsations and vibrations, but its effects at full load were not known. A properly designed damper to control discharge flow was impossible to obtain and install in a reasonable time period; therefore, testing was continued to obtain another solution.

It was felt that an inlet box vortex could be created by the inlet dampers and the inlet duct configuration [5]. The inlet vortex could cause improper inlet flow conditions and result in a rotating stall condition. In order to destroy any vortex forming tendencies, splitter plates were installed in each inlet box (Figure 2-11) directly opposite the inlet dampers. The vibrations, pulsations, strain, and noise levels were greatly reduced. In addition, the air flow at comparable vane settings was increased by a factor of 1.7.

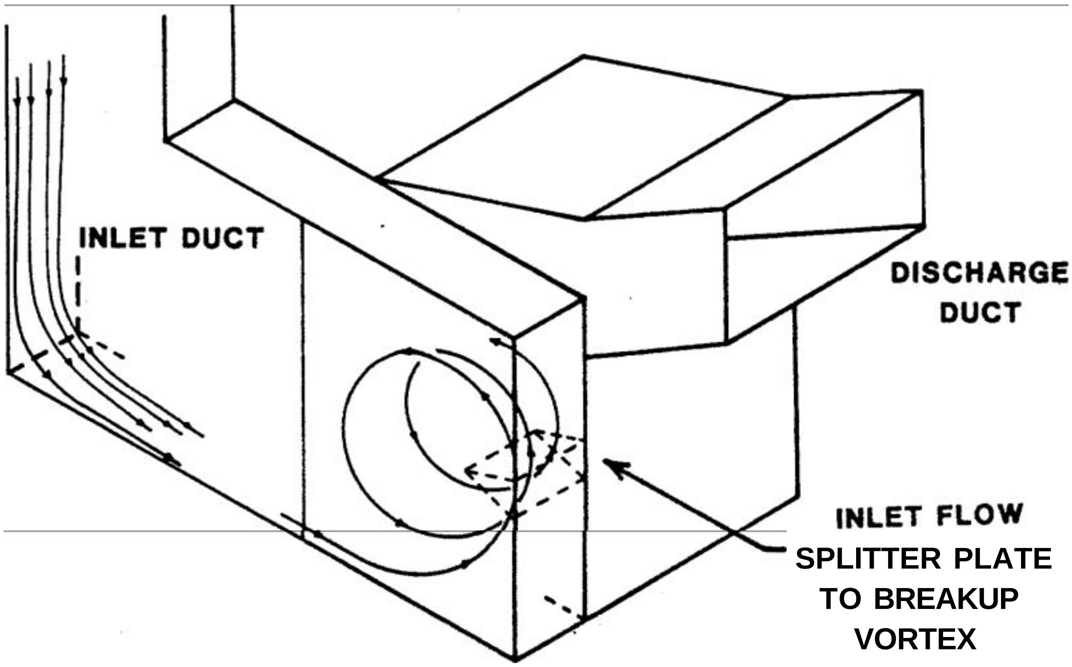


Figure 2-11. Inlet Flow Splitter Used to Reduce Inlet Vortex

Section 3

CONCLUSIONS

The case histories discussed illustrate that fan vibration problems are usually system related and caused by mechanical or acoustical interaction between the fan, motor, foundation, and ducts. Systematic diagnostic field test procedures and computer analyses of individual components are usually needed to define exact causes and optimum solutions. Several problems common to fans were identified and guidelines for analyzing and solving the problems are summarized below.

FOUNDATIONS

1. Foundation natural frequencies should be at least 20% away from the fan running speed because they can amplify fan and motor vibrations.

2. Accurate dynamic soil properties must be known to properly simulate foundation-soil systems.

3. The dynamic soil modulus should be experimentally measured at the site of each foundation after the excavation has been completed and the backfill has been added.

4. Most foundation vibration problems are due to a rocking mode about a point below the foundation. Foundations are often designed as tall rectangular blocks with the height greater than the width. By increasing the width of the foundation block and mat, the lateral stability of the entire system could be improved.

5. The outboard concrete pedestal is typically underdesigned for the dynamic radial and axial forces from the fan. The dynamic stiffness should be canparable with the inboard pedestal.

SHAFT LATERAL CRITICAL SPEED

1. The fan shaft "installed resonant speed" as defined by the AMCA should be used in the design or specification of a fan system.

2. The installed resonant speed should be at least 20% removed from the running speed.

3. To accurately calculate the rotor installed resonant speed, a forced vibration response analysis should be performed modeling the bearing oil film stiffness and damping characteristics with the eight coefficient representation.

BALANCINE

1. When balancing, extraneous influences such as excessive misalignment, fan interaction, etc., need to be minimized to obtain the proper balance data.

2. For sensitive systems alignment specifications may have to be tightened.

3. Hot balancing often requires waiting for 10-12 hours for a large fan to stabilize and quit shifting. The fan is generally stabilized when the rotor stops growing in the axial direction. Balancing with data taken before thermal stabilization may result in ineffective correction.

4. The least-squares multiplane balancing method allows minimizing vibration at any number of points. This method has advantages for systems operating near resonance and can minimize the number of balancing runs which reduces the required downtime.

5. All hot fans should have turning gears.

WHEEL RESONANCES

1. The disc wobble resonance should be at least 20% removed from the running speed. Fans with the disc resonance close to running speed can be overly sensitive to low level excitations such as pulsation and unbalances.

2. The disc axial resonant vibrations can cause amplification of horizontal vibrations on the bearing housings.

3. Disc resonances can be experimentally determined by impact tests or shaker tests. All centrifugal fans should be checked during manufacturing to determine disc resonant frequencies.

4. The disc wobble resonant frequency in the example was increased by stiffening the center plate. A stiffening plate could provide a long-term fix for existing fans with this problem.

ACOUSTIC EXCITATION

1. The inlet ducts should be designed to prevent forced inlet vortices (spin-swirl). The inlet vortex causes improper flow conditions and results in reduced fan performance and possibly rotating stall on the fan blades.

2. Pulsations generated by rotating stall usually occur at multiples of 1/3 shaft speed and, when sufficiently large, can cause structural damage.

3. Splitter plates can be installed in the fan inlet to break up the inlet vortex.

REFERENCES

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