

Fan Diagnosis in the Field

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SYNOPSIS

Fans are probably the simplest types of rotating machinery. However, the diagnosis of Fan problems in the field may require the use of elaborate techniques for the proper diagnosis of fan malfunctions. These techniques range from the use of spectral analysis to the use of operating deflection shape analysis. This paper presents a diagnosis procedure and several case studies for the field diagnosis of fans from the cement, fertilizer, and petrochemical industries. These cases include resonance, rotor unbalance, gearbox wear, flexible supports, and bearing faults. The cases illustrate the use of particular diagnosis technologies to identify fan malfunctions.

1 INTRODUCTION

Fans are used extensively in many industries, and in some cases a fan stoppage would cause a complete plant shutdown. This is the case with the kiln ID fan in the cement industry, or the boiler FD fan in the power industry. Therefore an accurate diagnosis of fan faults becomes imperative in these cases.

Fans are probably the simplest types of rotating machinery. However, even with this simplicity, the diagnosis of fan faults can be an elaborate exercise that may require the use of more advanced diagnosis tools. This paper introduces the diagnosis of fans as a step-by-step procedure describing the diagnosis process to guide the direction in performing the diagnosis and reaching an accurate and reliable diagnosis in a reasonable time frame to minimize production disruption. This step-by-step approach in a flow-chart format has been adopted by the author for a long time, and was published in conjunction with the diagnosis of installation faults [1]. More recently, this approach has been adopted by ISO TC108/SC2/WG10 in developing subsequent parts of the international standards ISO 13373 [2]. This step-by-step diagnosis procedure is illustrated by a number of case studies from different industries, where the diagnosis was performed according to the procedure outlined. These cases illustrate the effectiveness of the diagnosis procedure.

2 FAN DIAGNOSIS PROCEDURE

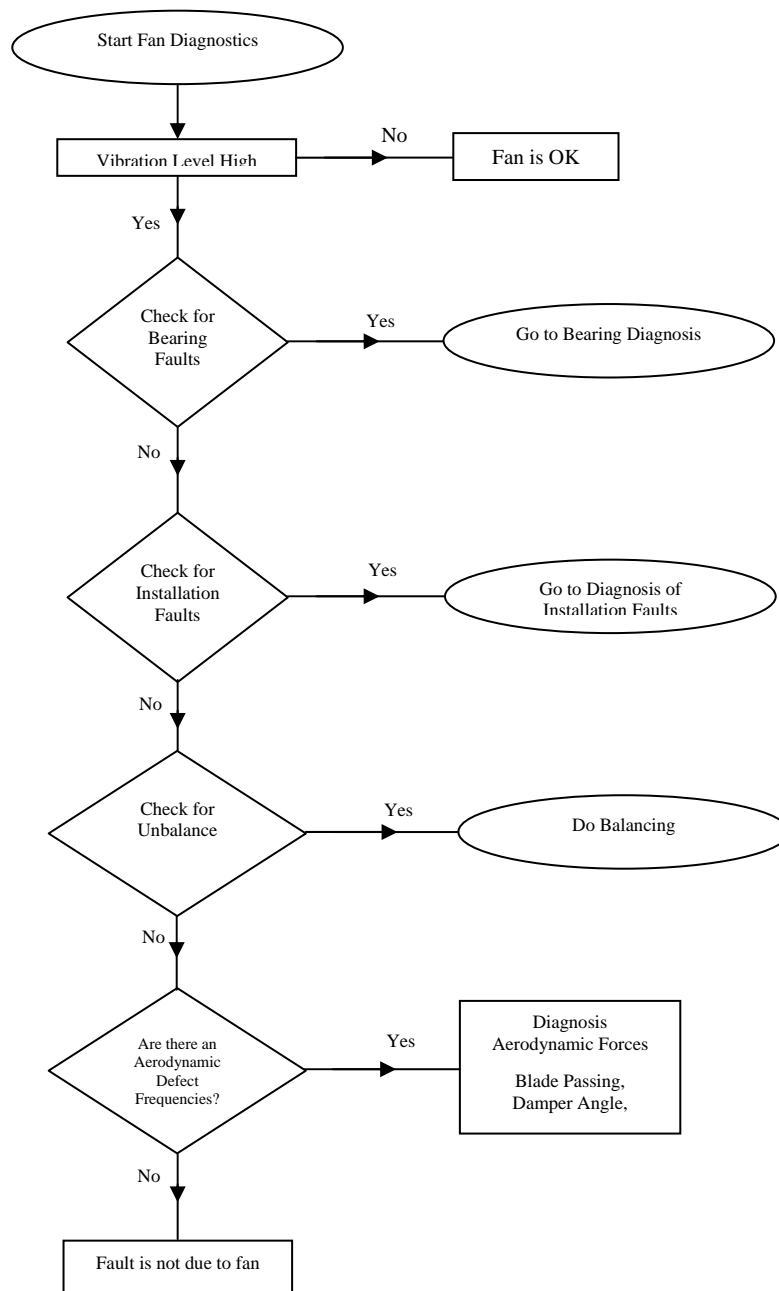


Figure 1 Fan Diagnostics

Figure 1 illustrates the fan diagnosis procedure. The most common faults in fans are bearing faults, installation faults, unbalance and aerodynamic excitation. The diagnosis procedure first asks if the vibration level in any band is high. If not, the fan is acceptable, otherwise check for bearing faults. The flow chart for bearing faults is not shown for brevity, but it includes check for the rolling element bearing frequencies: BPFO, BPFI, BSF, and FTF [3], as well as checking for bearing looseness or excessive bearing clearance. Also checks for faults in fluid film bearings are included. If the bearings are normal, then the user needs to check for installation faults. The flow chart for installation faults is illustrated in Figure 2.

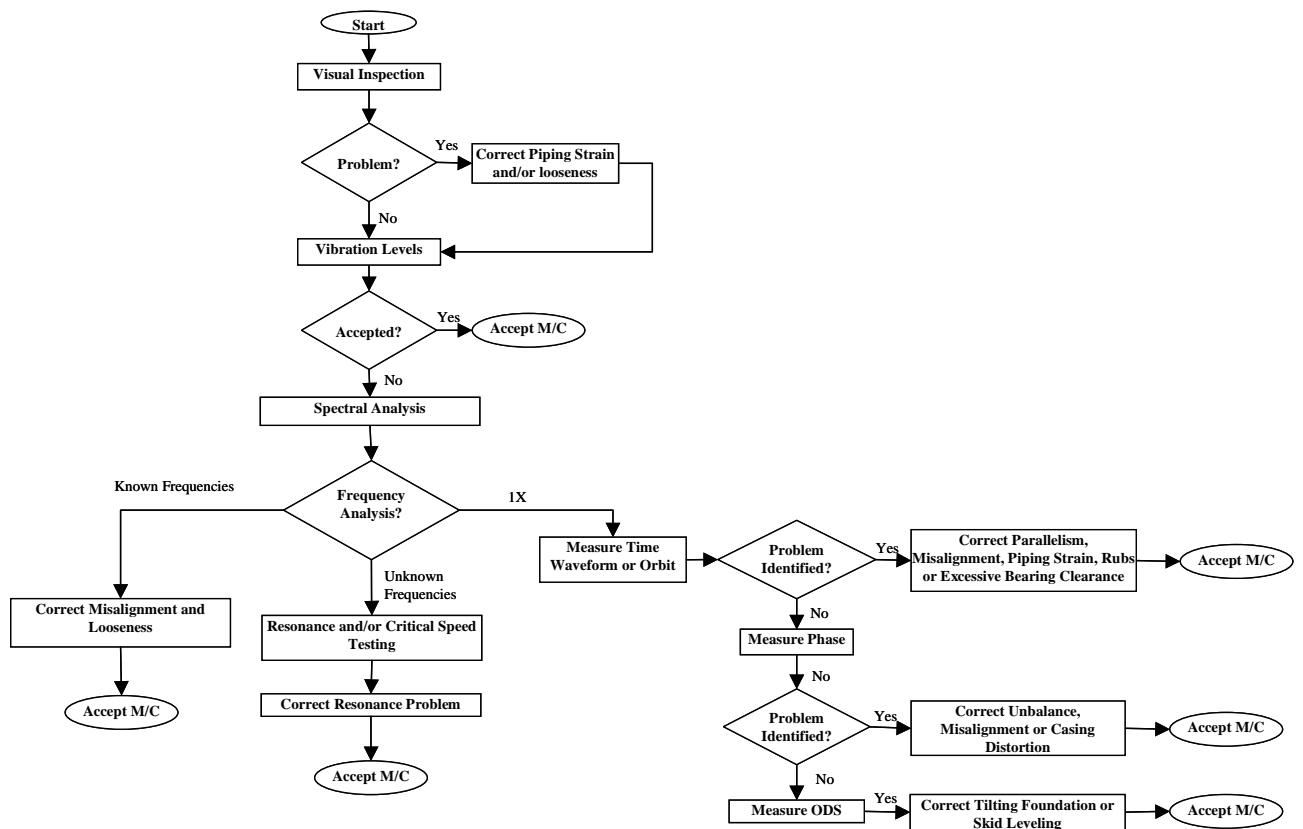


Figure 2 Flow Chart for Diagnosing Installation Faults

Basically, it is suggested that before any testing of installed machinery be performed, a visual inspection of the machine and the site be completed. In many cases, the presence of skid looseness would be evident to the naked eye. Actually, it is suggested that all skid and anchor bolts be tightened before testing an installed machine.

Spectral analysis is the core of the diagnosis of rotating machinery. Spectral data are usually taken as velocity data, which should be measured on all bearings on the fan, in all three directions, horizontal, vertical and axial. The purpose of the spectral analysis is to identify the frequencies causing the machine to vibrate. If all vibration amplitude levels are within acceptable limits [4], then the machine would be accepted as normal. However, if any of the spectral components has a high amplitude, then spectral analysis is used to correlate the frequency of the high amplitude vibration to a machine frequency.

The result of the spectral analysis of the high amplitude vibration is one of three cases: a) at a known frequency, b) at an unknown frequency, or c) at the running speed. By a known (unknown) frequency it is meant that the reason of the presence of this frequency in the spectrum is known (unknown). The first case is the easiest to analyze. If the high amplitude vibration is at a known frequency, then the problem is correlated to that known frequency. For example, if 2x vibration is identified then this is usually correlated to misalignment. If decreasing harmonics of the running speed are present in the spectrum, then this spectrum shape is usually correlated with looseness in the bearings or the skid. If however, there were unknown frequencies in the spectrum, then additional testing would be required to determine the source of the unknown frequencies. Amongst the additional testing that may

be required are: resonance testing (including impact (bump) test and critical speed test), modal testing, and flow characteristics testing. The purpose of the resonance testing is to correlate the unknown frequency to natural frequencies (stationary components) or critical speeds (rotating components) of the machine. Modal testing is a more advanced form of resonance testing, where all the modal characteristics of the machine are determined, including natural frequencies, damping ratios, and mode shapes. Modal testing is rarely used in the field, as it is an elaborate testing method, and is usually time consuming and costly. However, when justified, it can be a very powerful tool to obtain the machine characteristics and both identify clearly the unknown frequency in the spectrum, and suggest a solution to the problem.

The most difficult case occurs when the spectral analysis reveals high 1x vibration. There are many faults, related to installation problems, that lead to high 1x vibration. Amongst these faults are unbalance, misalignment, casing distortion, tilted foundation, skid leveling, piping strain and excessive bearing clearance. In this case, special vibration measurements have to be conducted on the machine to describe the nature of this 1x vibration, and to distinguish between the different 1x faults. These measurements include: time waveform measurement, phase measurement, and measurement of the Operational Deflection Shape (ODS).

The time-waveform measurement can be used to distinguish between misalignment, piping strain and excessive bearing clearance. For piping strain, it will be quite clear that the forcing on the machine is directional, usually in the horizontal direction, and this directional force would be acting on the whole machine. Inappropriate bearing clearance also results in directional forces, however this would be localized at the bearing with the inappropriate clearance. This is particularly true for special geometry bearings, such as lemon bore or multi-lobe bearings.

The phase analysis is quite important to diagnose unbalance, misalignment, bent shaft and casing distortion. In many cases, misalignment (the main installation anomaly) manifests itself as vibration at 1x only. One of the best ways to distinguish between 1x vibration due to unbalance and 1x vibration due to misalignment is to measure phase across the coupling. If there is a 180° phase shift across the coupling, then the problem is misalignment [5]. If no phase shift occurs across the coupling, then the problem is unbalance. A bent shaft would produce a 180° phase shift in the axial direction across the machine (end-to-end), corrected for transducer orientation. Casing distortion can be easily identified by 180° phase shift across the machine (side-to-side or end-to-end) in the horizontal, vertical and/or axial directions. A cocked bearing can be identified by measuring phase around the bearing housing and noticing the phase shift due to the wobbly action of the cocked bearing [6]. In many cases a coupled time-waveform-phase analysis is quite useful in visualizing the vibration pattern and identifying the problem.

If the 1x vibration problem is still not solved after the timewaveform and phase analysis, then an ODS should be measured. The ODS is useful in identifying problems of tilted foundation, skid leveling, skid looseness, and shaft parallelity. In the ODS measurement, phase-referenced 1x vibration is measured at grid points on the machine structure or skid. This reveals the actual deflection shape of the machine under the operating load, and at the operating speed. Note that the ODS is not a mode shape of the machine or structure, unless the machine is in resonance, but it can be

considered as a summation of the contribution of all of the modes of vibration. ODS analysis can be quite useful in identifying installation problems, as it provides a visualization of the actual vibration pattern of the machine and /or skid. In particular, if a machine skid exhibits a node in its ODS, then this is a clear indication of a tilted foundation or a leveling problem in the skid. Accurate measurement of skid and/or foundation levels would then be required to confirm the results of the ODS analysis.

The above steps should guide the user to diagnosing installation faults. If no installation fault is found then the next step according to the flow chart of Figure 1 is to determine if the fan is unbalanced. If the fan exhibits high 1x vibration with in-phase vibration at the coupling, then the fan needs cleaning and/or balancing. The final step in the step-by-step procedure shown in the flow chart of Figure 1 is to check for aerodynamic defect frequencies, usually the blade passing frequency. In this case the user needs either to correct for the controlling damper angle or check for appropriate axial clearances in the fan.

If the user follows the procedure in Figure 1, and no defect has been identified, then most probably the problem is not with fan, perhaps with the driving motor, and the user should check the motor diagnosis procedure.

The remaining sections of this paper describe cases where the diagnostic procedure of Figure 1 was used. These cases illustrate the application of the diagnostic procedure and clearly show the effectiveness of the step-by-step approach.

3 4000 RPM AIR BLOWER, FERTILIZER PLANT: SPECTRAL ANALYSIS

The main air blower in a sulfur plant was experiencing frequent motor bearing failure for the last two years. The 600 KW AC motor operating at 1500 rpm, was mounted on two rolling element bearings. The axial load was taken on the non-drive end angular contact bearing, while the drive-end bearing was the one experiencing the frequent failures. Most failures were in the form of a shearing cage. The motor was torn apart several times, without finding any cause for this problem.

On investigating this machine, the bearing failure rate was at a rate of once every two to three weeks. Spectral measurement on the machine confirmed that the motor had no malfunction, but the axial vibration is relatively high at the speed of the blower. Figure 3 shows the spectrum of the motor drive end bearing with high vibration at the speed of the blower. The motor was driving through a flexible coupling, a speed increaser gearbox mounted on journal bearings. The gearbox was driving the blower at 4000 rpm through a flexible coupling. The blower was overhung and mounted on two journal bearings. Survey of the blower and the gearbox showed a high axial component at both the gearbox and blower bearings at the speed of the blower. Figure 4 shows a typical gearbox spectrum. The gearbox showed a high gear mesh frequency of 1966 Hz and a high twice gear mesh frequency. Also, impacting was quite clear at a natural frequency of 2365 Hz with side bands. The highest vibration levels were at the vertical and axial directions of the blower bearings and were reading 5.2 mm/s.

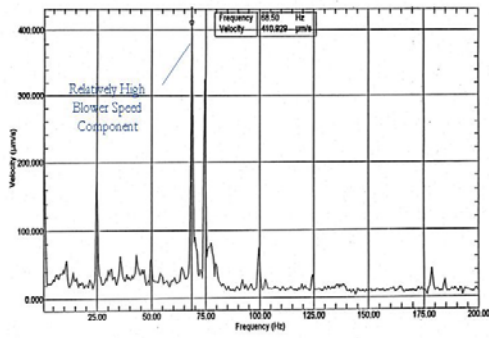


Figure 3 Motor Drive End Spectrum

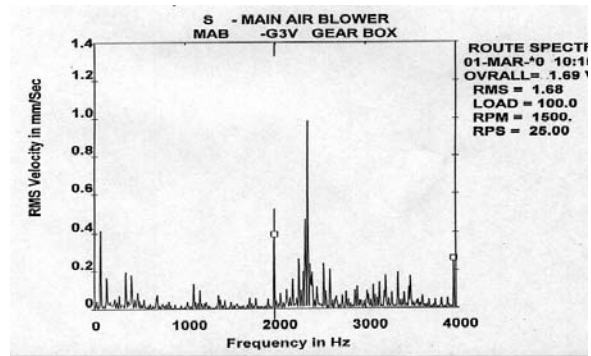


Figure 4 Gearbox Spectrum

The diagnosis of this sensitive machine was as follows: The blower is experiencing a couple unbalance producing an axial force that is providing an impacting on the gearbox and deteriorating the gear meshing. This axial force is then transmitted to the electric motor causing the bearing failure. The recommendation was to inspect the gearbox, correct any problems, and to balance the blower.

The gearbox was dismantled; the drive end journal bearing on the gearbox was completely destroyed. The axial thrust collar on the gearbox and on the blower bearings showed clear signs of impact. The blower bearings showed excessive clearance and rubbing on the rotor and bearings.

The gearbox was replaced, the blower bearings and shaft were replaced, and the motor bearings were replaced. The machine showed reduced overall vibration, but balancing was still recommended. The vibration vectors showed the need for a couple shot. A trial weight of 10 gm was used with this sensitive machine that was trim balanced with a 3 gm shot. The highest vibration level was reduced to 1.2 mm/s. No motor bearing failures have been reported since.

This case history illustrates the need to clearly identify the cause of vibration from spectral analysis. These problems should be corrected before attempting to balance the machine. The diagnostic procedure in Figure 1 was followed. No bearing frequencies were identified, however spectral analysis indicated deterioration of the gearbox, and a clear 1x indicated unbalance in the blower. 1x axial vibration due to unbalance occurs only in overhung machines. This axial vibration was transmitted to the motor, and its presence at the blower speed on the motor drive side was key to diagnosis of this machine train.

4 10.5 TON OVERHUNG CEMENT KILN ID FAN: BALANCING AND ODS

A Cement factory south of Cairo had a kiln fan that was experiencing high vibration. The fan has been installed some 12 years earlier, and had since experienced one catastrophic failure, and a major design modification to overcome its problems. In recent years, the fan has been experiencing high vibration resulting in the need to reduce its speed to reduce the vibration. This adversely affected production output.

On inspection of the machine it was found to be a huge 10.5 ton overhung fan on rolling element bearings with an impeller of diameter over 5 meters, and driven by a

1600 KW DC motor. The maximum speed is 490 rpm. Production required the machine to be operating at 95 % of its max. speed, while because of the high vibration the machine was operating at 82 % of its max. speed. The 4 meter high foundation showed visible cracks. Vibration measurements on the non-drive bearing showed a whopping 290 mm/s level, mainly at 1x.

Applying the diagnostic procedure of Figure 1, no bearing faults were found. Clearly the machine foundation required attention, however due to production requirements it was determined to balance the machine. This is unusual, and the author recommends that other problems should be resolved before balancing, but in this case the plant had requested a temporary solution, pending the resolution of the problem of the skid and foundation.

Before balancing it was requested to clean the fan blades, the vibration levels were reduced to 50 mm/s. A run-up test was performed to try to determine if any resonant conditions occurred and to mitigate any undesirable effects from the skid and foundation. Table 1 illustrates the results of the run-up test.

Table 1 Run-up Test on Kiln ID Fan

Speed rpm	Drive-end bearing		Non-Drive-end-bearing	
	Magnitude (mm/s)	Phase (deg.)	Magnitude (mm/s)	Phase (deg.)
284	6.74	-160	7.27	-159
309	3.36	-149	5.67	-144
355	3.52	173	2.72	172
380	3.74	162	5.2	155
415	3.73	140	10.9	135
425	5.8	126	16.5	125
440	10.6	112	31.8	111

From this Table it is clear that the machine is approaching resonance as its speed increases. A computer model was developed on RIMAP, a rotordynamics package [7], and it was determined that resonance occurs at the maximum speed of the machine. The mode shape is shown in Figure 5. Yet, it was decided to balance the machine. Inspection of the results of the run-up test show that the machine exhibits vibration that is in phase at both bearings at nearly all speeds, even though it exhibits a conical mode at resonance.

It was decided that the machine only needs a Static balance because of the in-phase measurement. A trial weight of 2 kg barely made the machine respond, but was used to obtain the balance sensitivity. The machine was finally balanced by a 6.1 kg weight, as shown in Figure 6, after removal of the trial weight. The vibration was reduced to 2.7 mm/s, and has been performing satisfactorily for over a year now. Production can now attain whatever speed they wish to load the kiln. Figure 7 shows the vibration spectra before and after balancing.

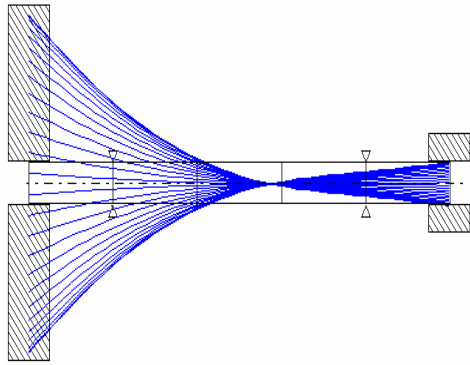


Figure 5 Mode Shape of the Fan.

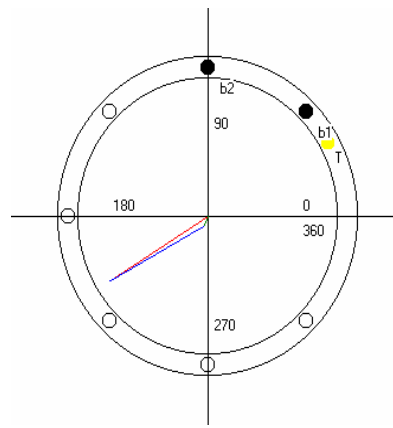


Figure 6 Static Balancing of the Fan

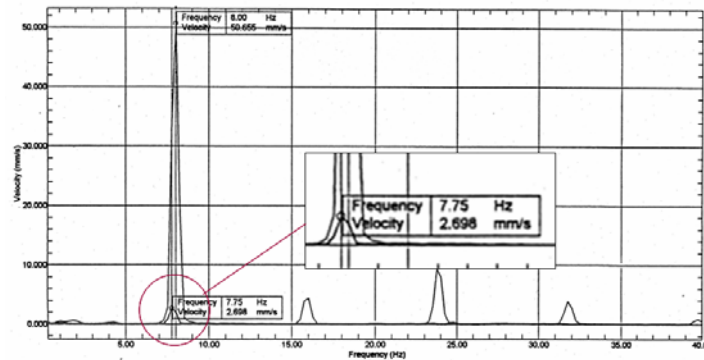


Figure 7 Spectra before and after balancing showing significant reduction

Upon successful balancing, it was required to investigate the skid resonance problem. An ODS of the skid was measured. Figure 8 shows a clear rocking motion for the skid in the vertical direction, while Fig. 9 shows a clear swaying in the skid in the horizontal direction. These measurements increase in amplitude as the machine speed increases and approaches resonance. Apparently the long period at which the fan was operating with severe unbalance resulted in the I-beams, embedded 0.8 m in the foundation and carrying the skid, separating from the concrete. A finite element model of the skid and foundation was developed and a suitable supporting structure was designed and implemented to prevent the rocking and swaying and to increase the natural frequency well beyond the operating speed of the fan.

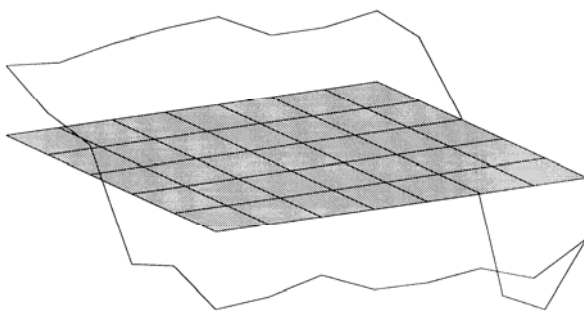


Figure 8 Vertical ODS Showing Rocking of Skid



Figure 9 Horizontal ODS Showing Swaying of the Skid

5 VIBRATION OF FAN SUPPORTING STRUCTURE: ODS ANALYSIS

This case study is from the petrochemical industry, and is for the supporting frame of a vertical air cooling fan supported on a horizontal steel frame with a horizontal motor and a bevel gearbox (see Figure 10). The unit is one of 12 units supported on an

elevated steel frame, used to cool air for 3 Gas Turbines 9 MW each. The units were experiencing frequent shear failures of the shaft connecting the motor and the gearbox at the input shaft of the gearbox. The manufacturer suggested the use of a larger diameter shaft. However, this resulted in accelerated wear in the gearbox.

Upon investigation of the problem, and applying the diagnostic procedure shown in Figure 1, a spectral analysis indicated that no bearing faults were present, and no unbalance, but the main exciting frequency is at 6x (blade passing frequency). However, investigating the operating conditions, it was clear that the blade angles were correct and the blade clearance was also set correctly. Therefore aerodynamic excitation was eliminated as a cause of the problem. A bump test was conducted, but was not conclusive because of transmitted vibration and it was not possible to shut down all coolers at the same time. An ODS was conducted at 6x, the exciting frequency, and the results are shown in Figures 11 and 12. It is seen quite clearly that the main deflection occurs in the middle of the frame, right where the repeated failure occurs, both in the horizontal and vertical directions. The reason of the failures is quite clear: the flexibility of the supporting structure at the gearbox input shaft position results in shaft failure. Increasing the shaft diameter only transmitted the problem to the gearbox, resulting in the accelerated wear. The root cause of the problem is the flexibility of the supporting structure. This supporting structure should be stiffened.

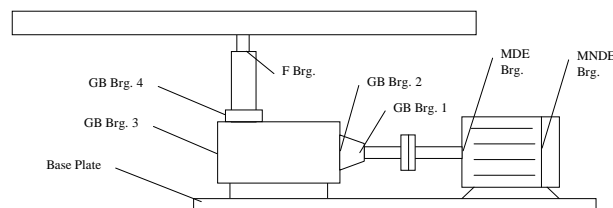


Figure 10 Schematic Layout of the Machine

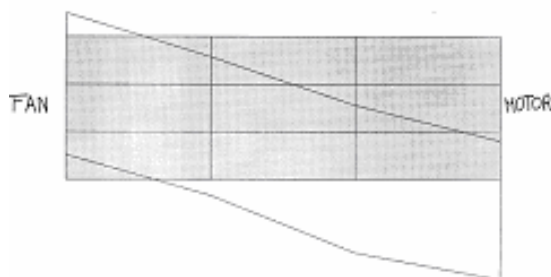


Figure 11 Base Plate ODS-Horizontal Direction

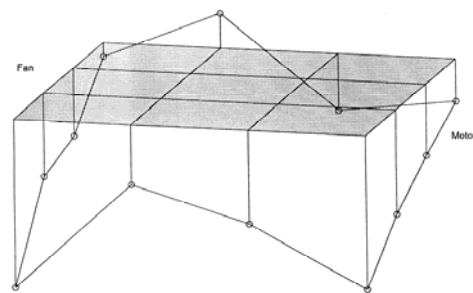


Figure 12 Base Plate ODS-Vertical Direction

6 BEARINGS PROBLEM

This is an unusual case. During acceptance testing of a cement plant, the vibration levels were relatively high for a new fan operating on rolling element bearings. The spectra are shown in Figures 13 and 14.

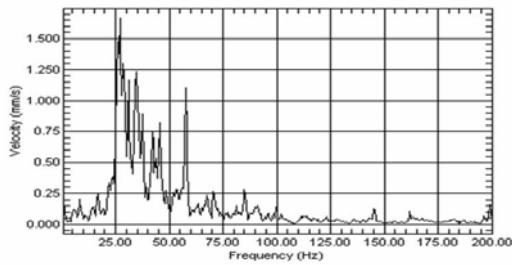


Figure 13 Drive End Bearing Spectrum

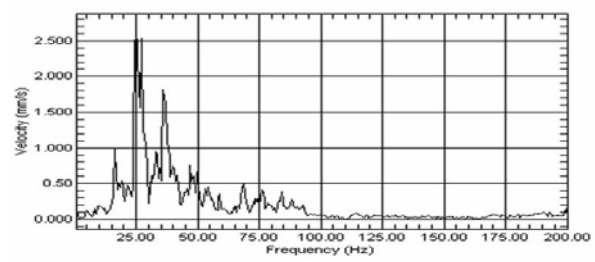


Figure 14 Non-Drive End Bearing

Applying the diagnosis procedure shown in Figure 1, no bearing frequencies were found, no installation faults were found, and the fan was well balanced. Also no vibration at the blade-pass frequency was seen. However, the fan bearings were noisy. The vibration pattern shown in Figures 13 and 14 is not usually associated with bearing faults. Although the spectrum is noisy with significant energy and has many frequencies, yet none of the frequencies matches with the bearing frequencies, namely BPFI, BPFO, BSF and FTF. However, discussion with plant personnel revealed that welding was performed near the fan. This immediately made us reconsider our diagnosis. The pattern shown in Figures 13 and 14 is an indication of electric discharge on the bearings, and occurs when welding is performed without proper electrical isolation. The fan was stopped and the bearings were removed, and clearly the discharge marks were evident. The fan was running smoothly, after the bearings change, allowing its acceptance by the customer.

7 CONCLUSIONS

A step-by-step diagnostic procedure for the diagnosis of fans was presented in this paper. The flow chart of Figure 1 is the basic tool for the diagnosis procedure. A user can follow the application of this procedure to diagnose fan problems. The case studies presented cover the full application of the diagnosis procedure, including installation faults, balancing and bearing faults. Various technologies are used including spectral analysis, phase analysis and ODS analysis.

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