FIN FAN VIBRATION REMEDIATION

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SUMMARY

Thirty fin fans in a large refinery were tested to solve structural support vibration problems. Operational deflection shape and impact modal testing were used in conjunction with finite element analysis to identify three troublesome modes of vibration. The FEA model was then used to design additional restraints to reduce vibrations. Twenty-three fans were selected for modification. Initial vibration levels were as high as 15 to 27 mm/sec rms. After modifications all vibrations were below 15 mm/sec rms and 19 of the fans had measurements below 10 mm/sec rms.

INTRODUCTION

Major refineries have a large number of fin fan coolers. Maintenance of so many individual machines becomes a considerable task. They need to have an inspection and testing program defined and performed periodically in order to find problems before they become critical and to further identify generic problems common to many of the units. In most cases the fans are similar units, but usually not all identical, thus causing differences in problems and the means by which they must be inspected and maintained.

The Kuwait National Petroleum Company (KNPC) Shuaiba Refinery near Ahmadi, Kuwait has approximately 760 fans of similar design, some of which experience vibrations at levels that cause concern. Three failures of similar mode have occurred on these fans in the past. The failures involve separation of the fan skid structure from the supporting plenum above, allowing the fan, skid, and motor to drop down on one end.

The fans of concern all have a similar structural design with significant variations, as listed in the following discussions. In the plant section in which the failures occurred the fans have larger motors than most other units in the plant.

In order to gain an understanding of the variety and causes of the vibration problems and failures, KNPC formed a task force with Southwest Research Institute (SwRI) to inspect and test about 30 of the fans identified as problem units.

TESTING

Several types of testing were performed. The initial tests consisted of (1) operational deflection shapes (ODS) to define the directions and frequencies at which the vibration occurs on operating fans and the excitation sources acting upon them, and (2) impact modal tests on the fan structure while it is shut down to determine the resonant frequencies and their mode shapes.

During the course of the testing, two temporary modifications were developed to add stiffness to the structure at locations expected to reduce the vibration amplitudes. Test measurements were taken on these modified fans and the data contributed significantly to the following analyses and recommendations.

Included in the list of fans tested were selected ones which had prior modifications installed, that sought to reduce the vibrations such as the one shown in Figure 1. Two of these were fans which had experienced failures, and one "good" fan.

Figure 1. Motor End of Fan with Vertical Support Added Previously.

IMPORTANT OBSERVATIONS AND FINDINGS

In an attempt to determine why some fans had vibration problems and others did not, the variations in fan construction were inspected. The following differences were found.

- Variable and fixed pitch fan hubs that were of different masses and configurations
- Wide and narrow fan blades
- At least three different sizes of motors
- Two different fan speeds
- Differences in the motor mount bracket and its attachment stiffness to the fan skid
- Various support modifications performed previously on selected fans

Each of these factors has an influence on the frequencies of the structural resonances or the energy imparted to the structure to drive the vibrations. They also contribute to the differences in vibration levels measured among what appear to be identical fans.

Analysis of the data show that there are three primary modes of vibration which contribute to the overall vibration levels measured on the various fans. In each individual fan, the design differences may cause a different result in the vibration levels and frequencies measured. The modes will be described shortly.

The structural support system is very flexible and subject not only to the driving forces of motor speed, fan speed, and blade-pass frequency, but also to random turbulence. The structure has very low stiffness as the fans are only supported from sheet metal above. This is further demonstrated by the fact that the fans were observed to vibrate in approximately the same mode shapes when they were not operating as when they were operating, albeit at lower amplitudes.

FAN VIBRATION CAUSES

Several contributing factors were found to cause the fan vibrations.

- There is insufficient support stiffness in the structure to resist any more than minimal vibratory energy. The entire fan skid is suspended from the ceiling (plenum), which is flexible and has only small vertical supports. Figure 2 shows that the ceiling plenum vibrates at about the same amplitude as the motor mount in the vertical direction. In the absence of stiffness in the plenum, support can be applied directly to the fan and motor skid. Prior vertical supports installed on the fan end of certain machines appear to address this need for greater vertical stiffness. The temporary modification tested also accomplished this vertical support objective, using somewhat less material.

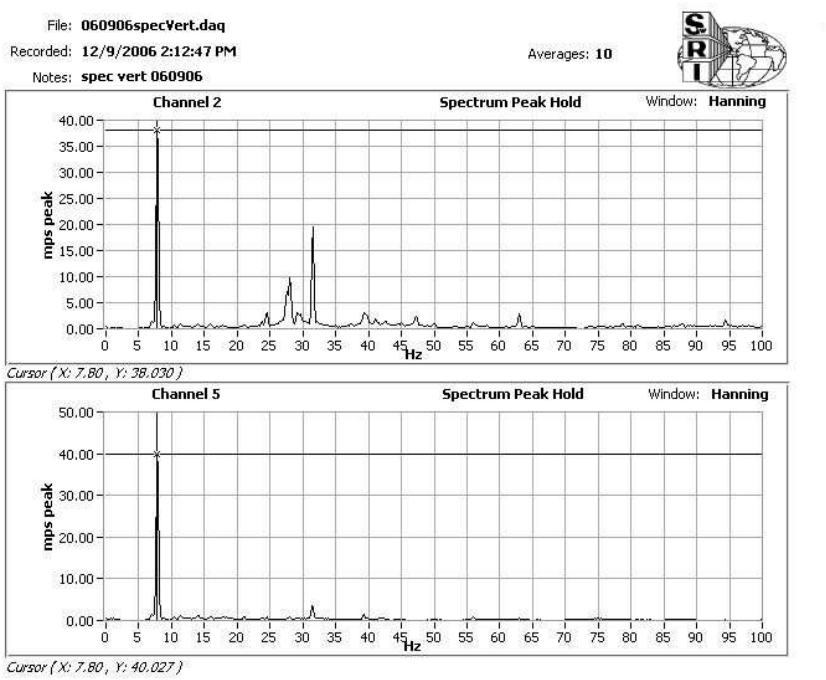

Figure 2. Comparison of Fan Skid (bottom – 40 mm/sec, 0-p) and Ceiling Plenum (Top – 38 mm/sec, 0-p) Vibration at Fan Speed (7.8 Hz.), Unbraced

- Ribbons tied to the fan screens indicated that there was recirculation of air around some fans due to blockage of flow through the cooling fins and/or excessive blade pitch angle, which tries to force more air through the coils than can be accommodated by their impedance to flow. When the air cannot flow out of the fan through the normal discharge path, it builds pressure and turbulence and forces its way back out at the blade tips and fan hub. Observations confirm that some fans did indeed have dirt collected in the cooling fins on the bottom (fan) side of the cooling coils. In units that had less obstruction to flow or where it is confirmed that the cooling thermodynamics are adequate and vibration and recirculation were also experienced, the solution is to reduce the blade angle of attack, or reduce fan speed to eliminate the recirculation and resulting turbulence. [1], [2]

- There are significant differences in the geometry, masses, and stiffnesses of the various motor support brackets on the various fans. There are three different sizes of motors identified and there are differences in both the size and the location of the motor mounts on the fan skids. This may be due in part to differences in drive belt lengths or hub diameters. There will also be inherent differences in the tightness of the adjustable lock bolts that hold the motor mounts in place on the skids. The vibration would affect their ability to stay as tightened. These differences provide significant variability in the ability of the motor to vibrate, either in resonance or in purely forced vibration. The right combination of variables places specific units in resonance.

- The channel beams that constitute the main structure of the fan skid are too flexible to resist the normal rotating forces placed at the center of the skid by the fan. The vibration is usually larger in the horizontal direction than the vertical direction, but both modes are significant in some fans. This vibration occurs at or near blade-pass frequency and, therefore, is not driven by rotating unbalance of the fan (see Figure 3). It can be influenced by blade asymmetries. Diagonal braces installed previously on a few fans appear to be intended to resist the horizontal component of this motion. The wooden beams installed between adjacent units during the current sensitivity testing also reduced the horizontal component of the vibration. Compare Figure 3 with Figure 4.

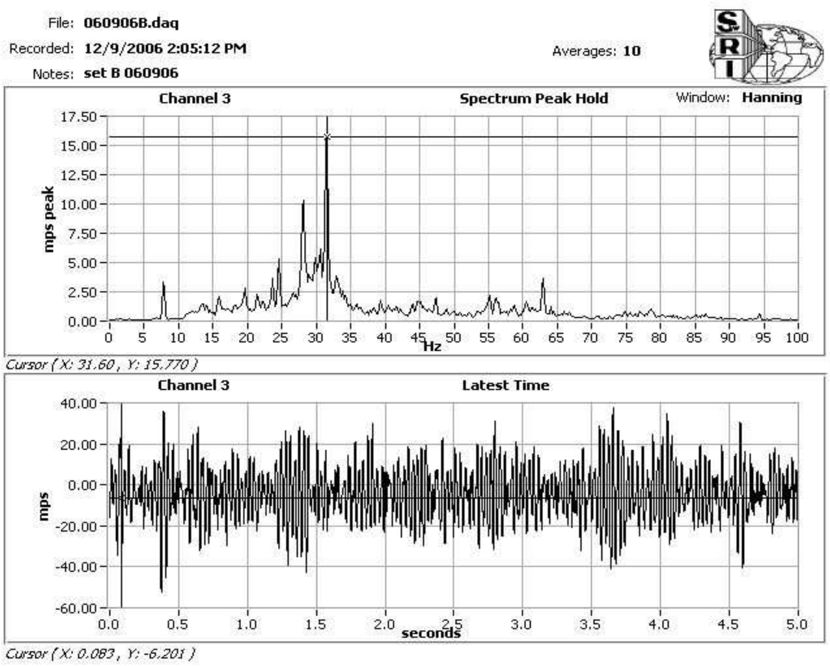

Figure 3. Horizontal Vibration at the Fan with NO Horizontal Bracing.
Note Blade-Pass Frequency (31.6 Hz) is 15.7 mm/sec, 0-p.

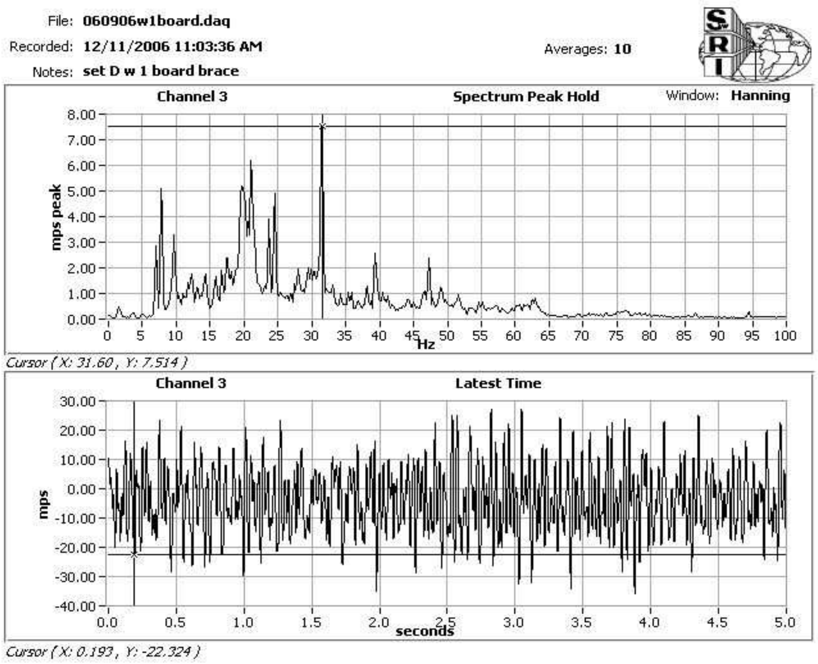

Figure 4. Horizontal Vibration at the Fan WITH Horizontal Wooden Bracing.
Note Blade-Pass Frequency (31.6 Hz.) is Reduced by Half to 7.5 mm/sec, 0-p.

DESCRIPTIONS OF MODES OF VIBRATION

The three primary modes of vibration of the fan system include:

- A vertical bounce mode of the fan skid with the leading motion at the motor end. This motion takes place at the fan speed of 7 to 7.8 Hertz (Hz). The resonance of this mode is about 10-12 Hz.
- A lateral vibration of the fan skid at its center location where the mass of the fan and its support structure are attached to the skid channel beams. This motion is predominately at blade-pass frequency of 27 to 32 Hz, while its resonance is 12-13 Hz.
- A mode of vibration of the motor and its mounting frame controlled by the stiffness of the bolts that attach it to the fan skid and by the motor mass. This mode occurs at or near motor speed of 24.4 Hz.

The failures experienced were probably the result of stress contributions from all three modes of vibration.

FINITE ELEMENT MODELING

The finite element model of the fan structure produced vibration modes very close in frequency to those measured by field impact testing. The vibration mode shapes at the excitation frequencies (off resonance) were also identical.By field measurements and modeling, the resonances of two of these modes are in the range of 10 to 13 Hz. There are no excitations from fan speed, motor speed, or blade-pass in this (10 to 13 Hz) resonance frequency range. Thus it is clear that the vibrations are off-resonance forced vibrations. The mode shapes are so flexible that they respond to energy at any frequency.

Figures 5, 6, and 7 illustrate the three primary modes of vibration. The end of the color spectrum towards the red end signify the largest vibration amplitudes, while the dark blue end of the color spectrum are points that are essentially static.

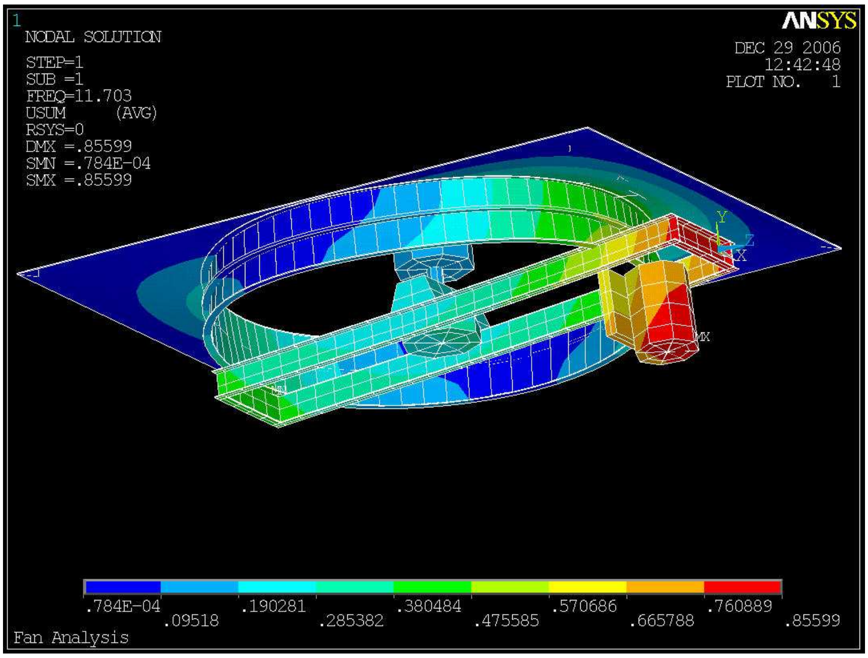

Figure 5. Modal Analysis of Motor Vertical Vibration Mode at 11.7 Hz.

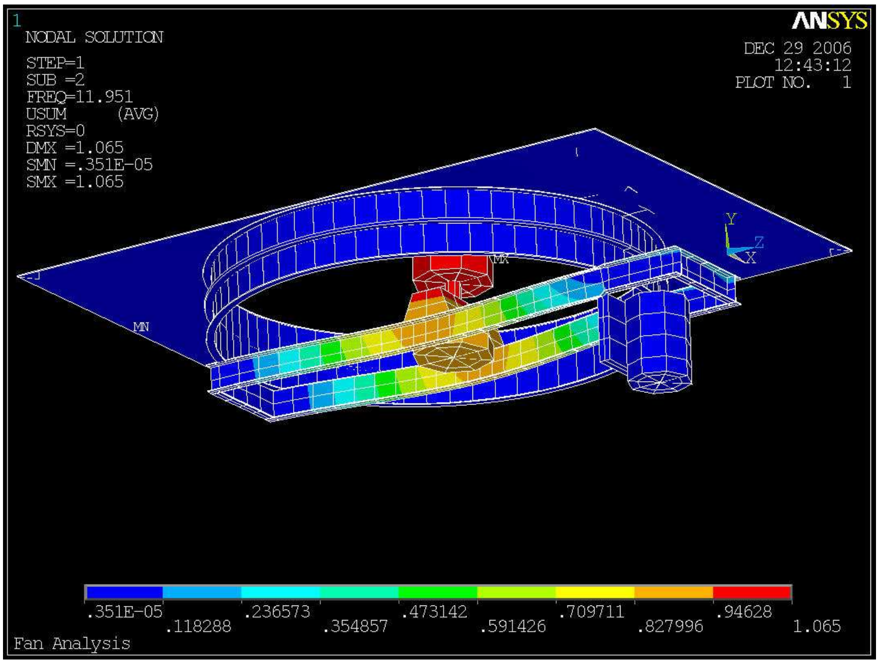

Figure 6. Modal Analysis of Horizontal Skid Vibration Mode at 12 Hz.

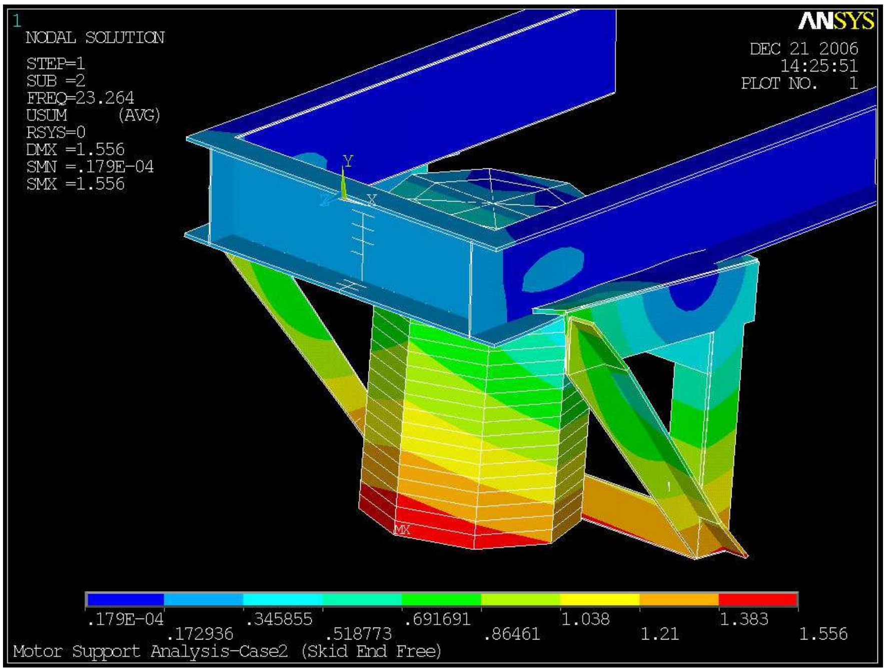

Figure 7. Modal Analysis of Motor Bracket Results (Skid End Free).Resonance at 23.3 Hz.

For the horizontal vibration mode of Figure 8, the finite element (FE) model was used to determine the force required to produce the (largest) measured response of 10 mils (23mm/s) at 27 Hz at the fan location in the horizontal direction. This force (137 lbs) was then applied to the model and modifications were identified which could control the vibration.

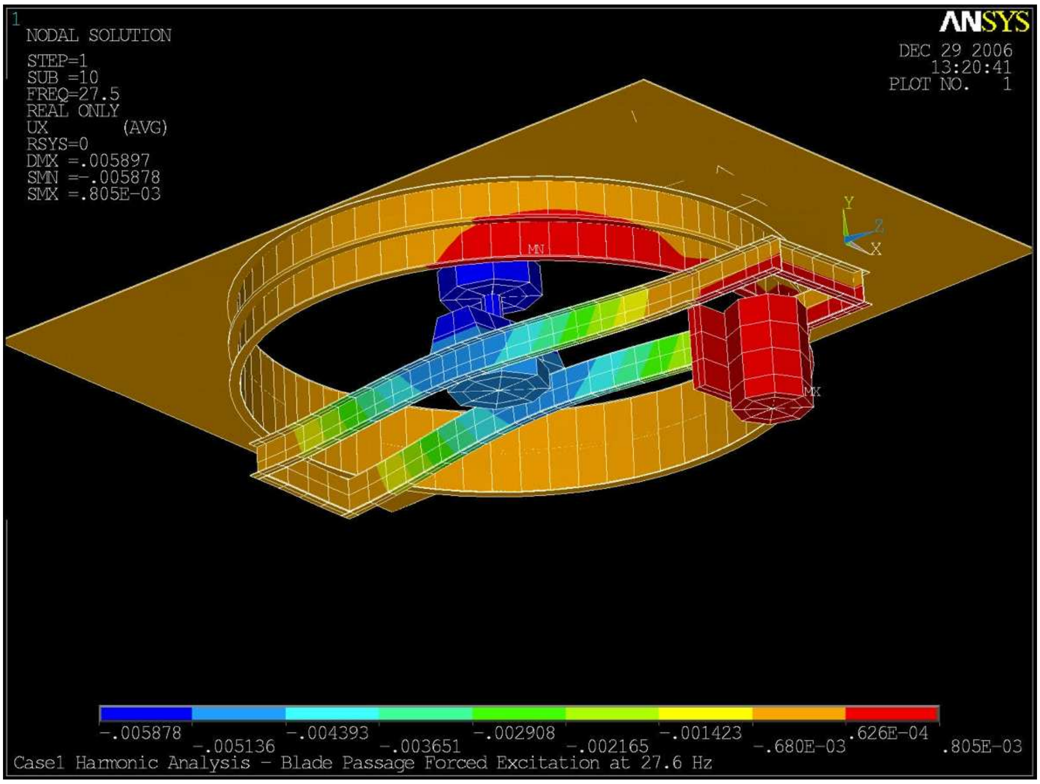

Figure 8. Harmonic Analysis Forced Excitation at 27.5 Hz (137 Lb 0-peak). Horizontal Vibration Amplitude Approximately 0.91 ips 0-pk (23 mm/s 0-pk) at 27.5 Hz (30 HP Motor, Fixed Pitch Hub).

The trial modifications tested in the field treated the first (bounce) and second (fan and channel lateral vibration) of these modes. The resulting increased stiffness allowed the third mode to be uniquely identified, as it was the remaining vibration mode, which apparently had become resonant as a result of the vertical stiffening of the skid. Figure 9 shows the motor frame vibration without vertical supports. The main motion is at fan speed 7.8 Hz. After vertical bracing, Figure 10 shows that the 7.8 Hz motion is removed, the system is stiffened, and the motor mounting mode is excited at 24.6 Hz.

The FE model was used to identify the motor bracket mode of vibration, as shown in Figure 7. The model shows that the expected resonance, for the large motor, is 23.3 Hz (just below motor excitation frequency of 24.4 Hz) and the motion is a rocking of the motor mount relative to the skid side channels. The model then applied the vertical stiffness caused by the supports added by modifications. This moved the mounting resonance up to 24 Hz, which coincides with motor running speed, thus amplifying that mode of vibration, while the vertical bounce mode is greatly reduced (see Figure 11). This is what was witnessed during the testing.

The model was then used to add a 12 mm thickening plate to the top of the angle bracket that attaches the motor mount to the channel skids. As a result, the model predicts a shift in the resonance up to 56.7 Hz, well away from motor speed.

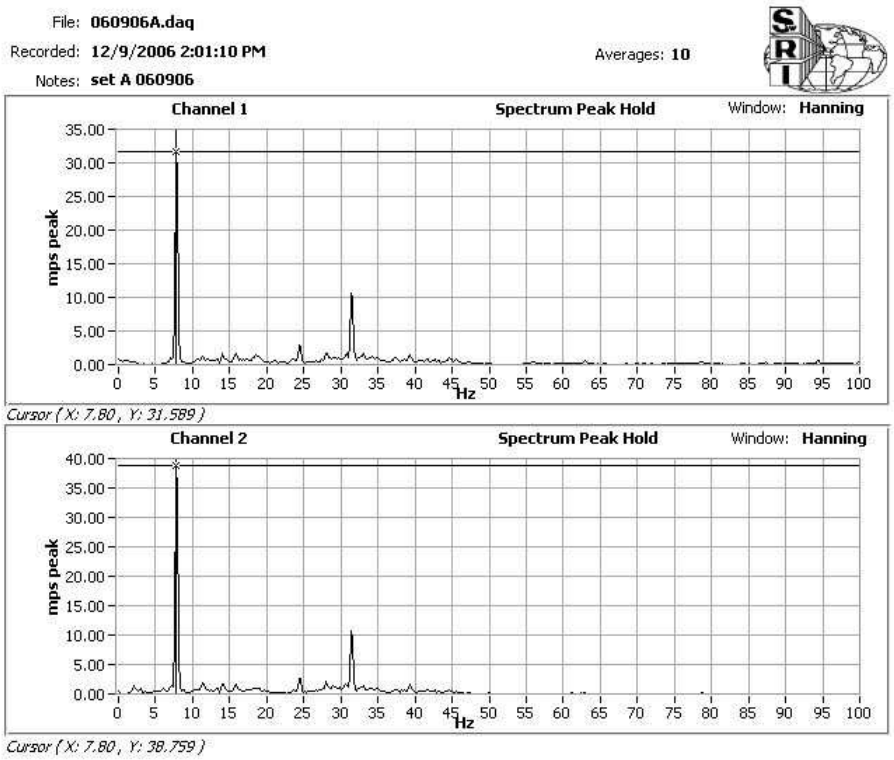

Figure 9. Vertical (top) and Horizontal (bottom) Vibration of the Motor and Mount of Fan E-06-09-M6 without Vertical Support Modification. Note Fan Speed Vibration at 7.8 Hz.

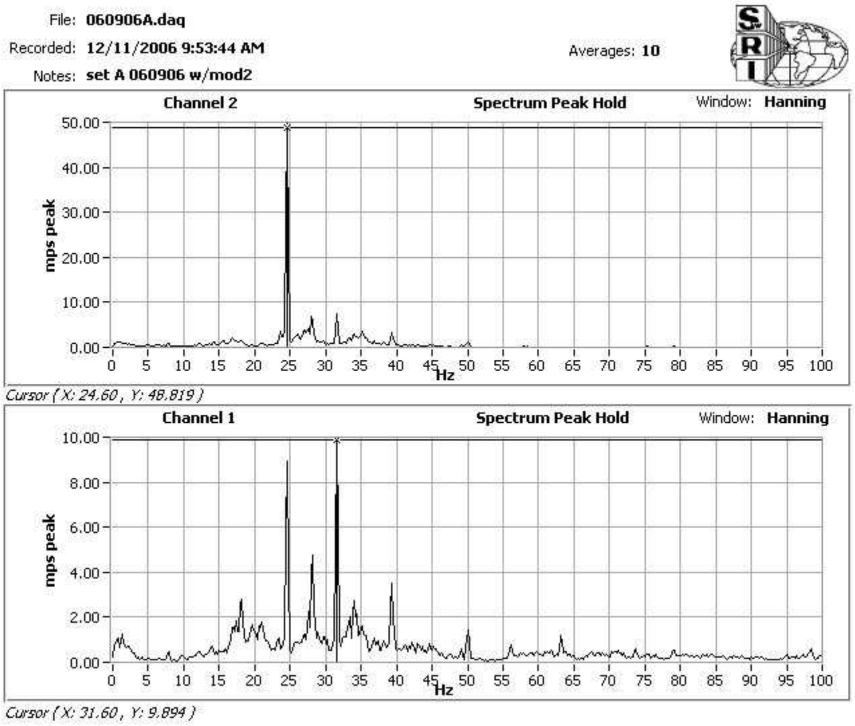

Figure 10. Vertical (bottom) and Horizontal (top) Vibration of the Motor and Mount of Fan E-06-09-M6 with Vertical Support Modification. Note Fan Speed Vibration at 7.8 Hz Removed and Motor Speed (24.4 Hz) Amplified.

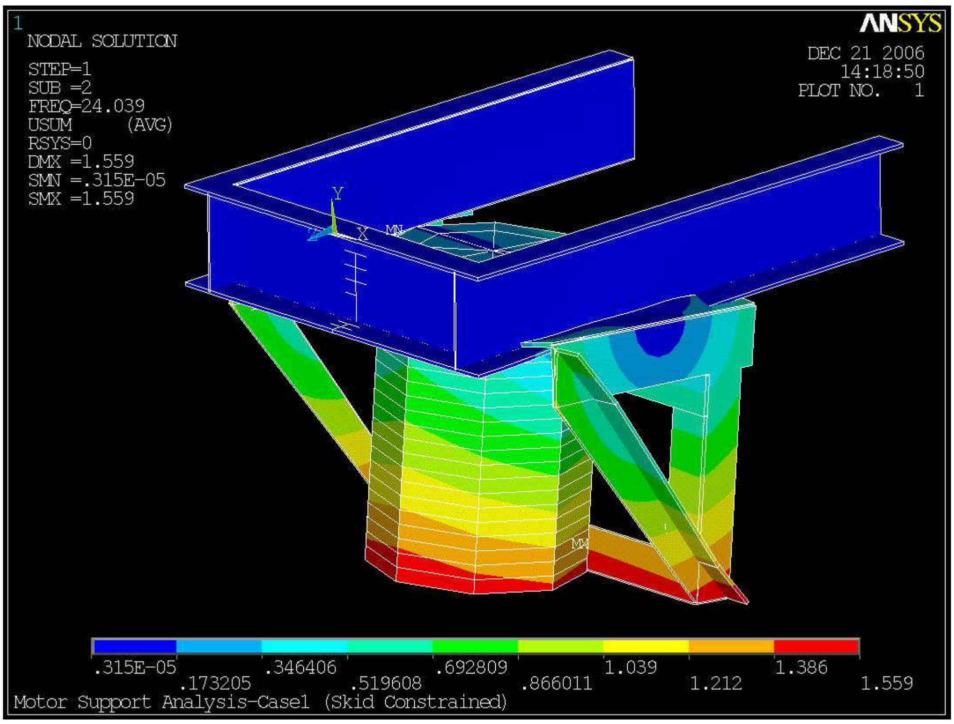

Figure 11. Modal Analysis of Motor Bracket (Skid End Fixed, 30 HP Motor) (Bracket Tied to Skid Channel at Two Bolt Locations Each Side) Results in 24 Hz Resonance- Motion In Line with Long Axis of Skid.

MODIFICATIONS

1. The vertical bounce vibration of the entire fan skid, identified by vertical vibrations at or near the fan speed (7-8 Hz), was addressed as shown in Figures 12, and 13.

Figure 12. Modification for Vertical Skid Restraint.

Figure 13. Modification for Vertical Skid Restraint.

2. The horizontal and vertical vibration mode of the center of the fan skid was braced to reduce the amplitude of that motion near blade-pass frequency (27-32 Hz). Figure 14 shows the design of a brace of steel construction that was used on those units that were found to have objectionable vibration in this mode. It is designed that the first modification, which addresses the vertical bounce mode of vibration, also provides additional help for the vertical motion at the fan position.

Figure 14. Cross Bracing Between Fans To Stiffen For Horizontal Skid Vibration due to 28-32 Hz Blade-Passage Excitation.

3. The motor mount mode of vibration identified at or near motor running speed (24.4 Hz) was restrained by stiffening the motor mount and its attachment to the fan skid. Figure 15 shows the stiffening plate for control of this mode of vibration. The frequency of this mode was increased from near 24 Hz to 57.2 Hz by combining the effects of both the vertical support and the motor mount modification.

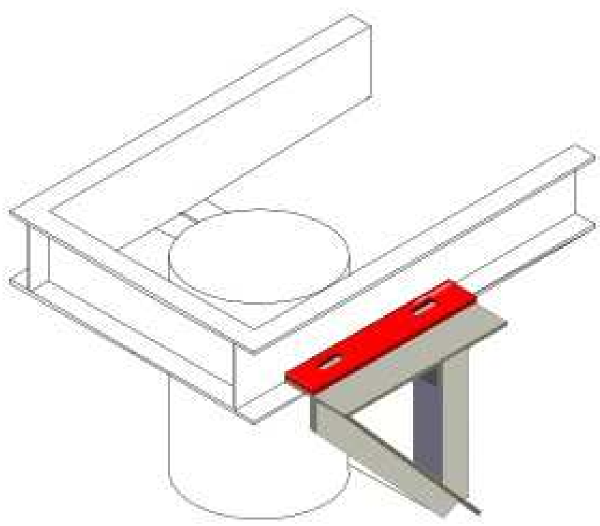

Figure 15. Illustration of 12 mm Thickening Plate for Attachment of Motor Mount to Fan Skid.

RESULTS

The plant selected 23 fans for modification using the several methods described above. Initial levels were as high as 15 to 27 mm/sec rms. After modifications, all but one fan had the higher levels lowered, and all measurements were below 15 mm/sec rms. Nineteen of the fans had all measurements below 10 mm/sec rms.

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