# Appendix B. Chapter 11.

# Fan Housing Vibration Caused by Resonance

## Application of Modal & Vibration Analysis

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**KSC Consulting LLC**

# Background

Four FD fans were installed at the site to meet environmental requirements of the process.

The fan design details were as follows:

- 1500 HP Direct Drive AMCA Arrangement 8
- Inlet Guide Vane Control
- Spherical Roller Bearings, Grease Lubricated
- PLC Monitoring and Control

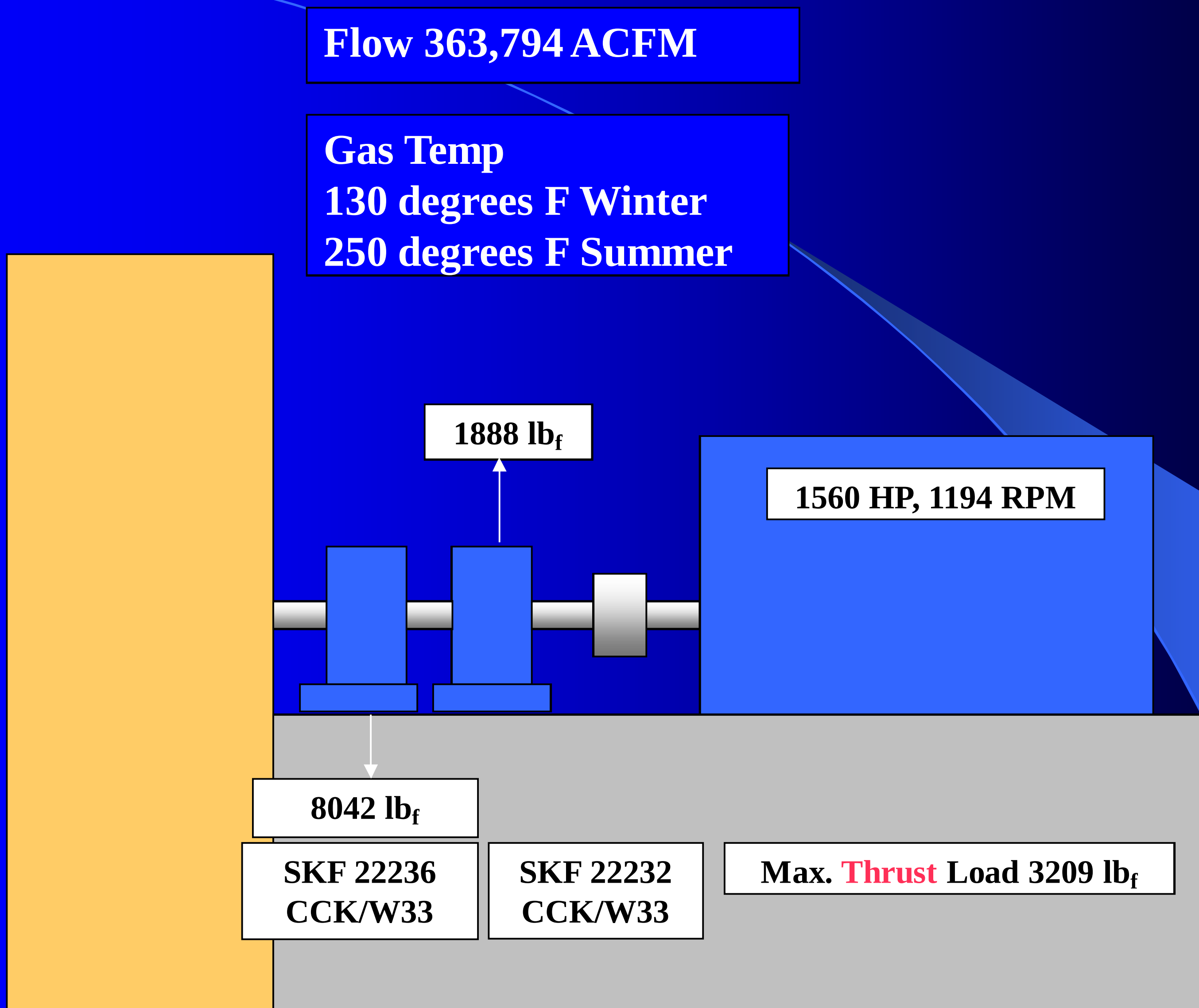



**Figure 1. Schematic of Fan with some Parameters Labeled**

# Background

The fans experienced many reliability problems during the 1st year of operation. These problems included:

- Bearing failures on each fan
- A catastrophic bearing failure on #1 Fan resulting in a broken motor shaft and damaged fan shaft, which required replacement.
- High bearing operating temperatures.
    - Several lubrication greases were tried including varying quantity and interval of greasing as well as grease removal from the bearing housings.
    - A small oil recirculation system was installed on one fan, but only provided mixed results
- Component fatigue failures of discharge vane assembly,

# Initial Assessment

**The initial assessment of the fans was made in Dec 2001. Numerous deficiencies were identified:**

- **The Fan End Bearing was operating at 92% of maximum recommended speed with grease lubrication.**
- **The spherical roller bearings calculated to be marginal for grease lubrication for the fan operating speed of 1194 RPM.**
- **Fan housing high amplitude vibration (access to housing limited by insulation.)**
- **The discharge vane housings exhibited high levels of vibration and cracking of welds at the linkage/shaft interfaces.**
- **Vibration analysis indicated lubrication-related high energy vibration (CSI PeakVue data) caused by bearing roller skidding** (Bearing load less than specified minimum load.)

# Initial Assessment

## Initial Recommendations included the following:

- **Perform a modal test of one fan housing.**
- **Change bearing lubricant from grease to oil and Install an oil re-circulating system with filtering and temperature control (The fan end bearings were operating at 92% of recommended maximum speed with grease lubrication (borderline for grease; this design truly needed oil lubrication).**

During second visit to site Feb 2002. Bearing & fan housing vibration was measured. A single accelerometer was attached to the fan housing though a hole drilled in the insulation. **Minimum** 1.2 in/sec was present.

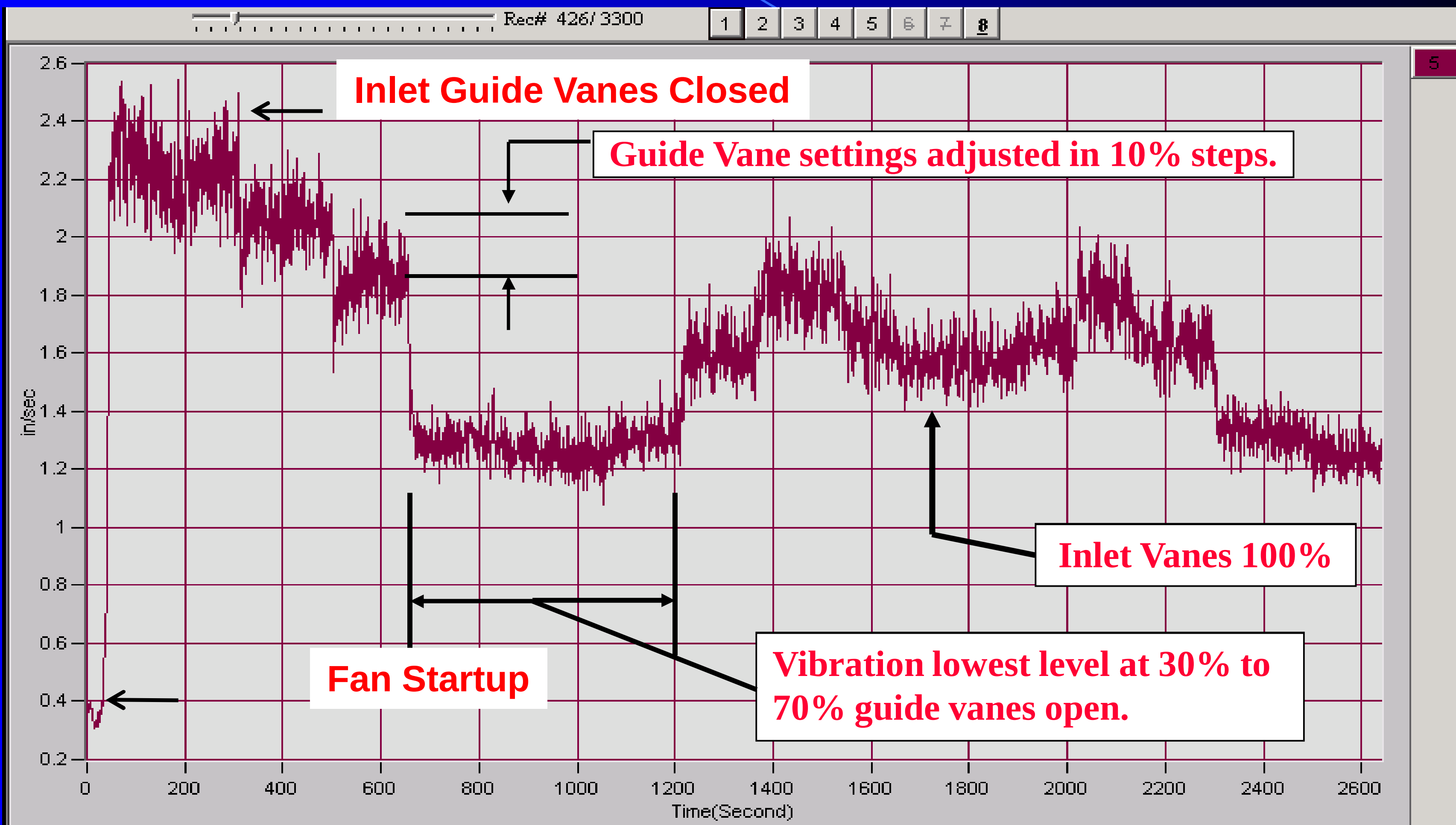



Figure 2. Overall Vibration (in/sec) of Fan #4 Housing Measured Prior to Startup of the Fan and During Inlet Guide Vane Adjustments to Full Open, Closed and Shut Down.

# Frequency Spectrum in February 2002 on Fan #4 Housing

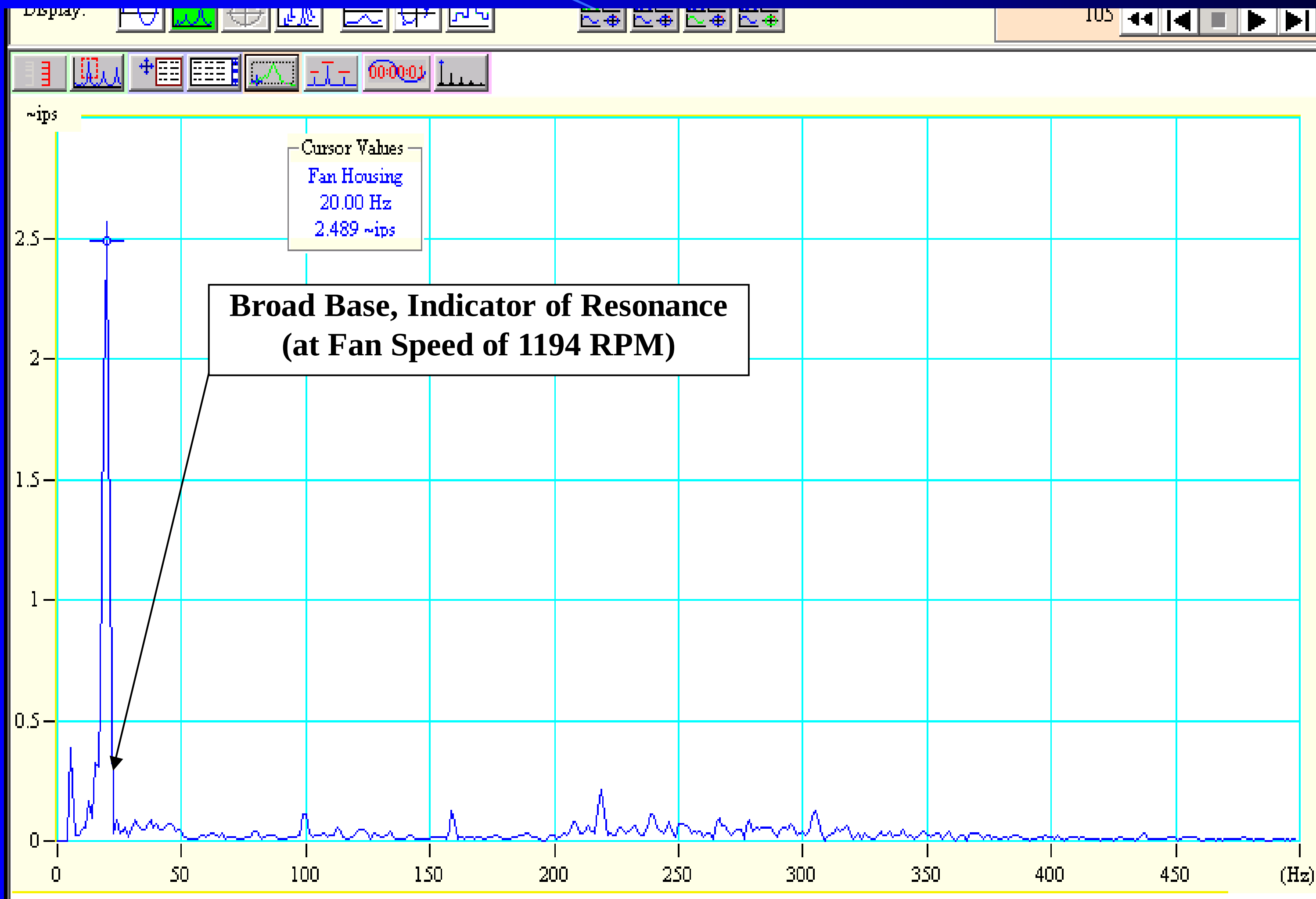



Figure 3. Fan #4 Housing Vibration Spectrum. Most Vibration Energy was at about 20 Hz (1200 CPM), or 1 X Fan Running Speed (2.5 in/sec).
The Response Appeared to be Caused by Resonance of the Housing.

# Fan Inspection After Insulation Removed

- Recommendation to perform fan housing modal test was accepted and scheduled for Sept 11-12, 2002.

- Insulation was removed to provide access to the fan housing.
- The fan housing was inspected for cracking as well as general design and fabrication methods.
- No cracks were found.


Illustration A. Overview of Scrubber Fans Evaluated

# Fan Inspection After Insulation Removed

- The housing was a three piece design to permit shipping and removal of the fan wheel.

- The two upper sections were bolted to the lower half at the centerline and to each other at a vertical flange.

It was learned that sections of the housing flanges (4 inch Channel) had been removed using a cutting torch to provide clearance for the shaft. The removal of these stiffeners in the center of the housing significantly reduced the rigidity of the housing.


**Figure 5. Fan Housing Showing Vertical and Horizontal Joints and Intersection Near the Fan Centerline**

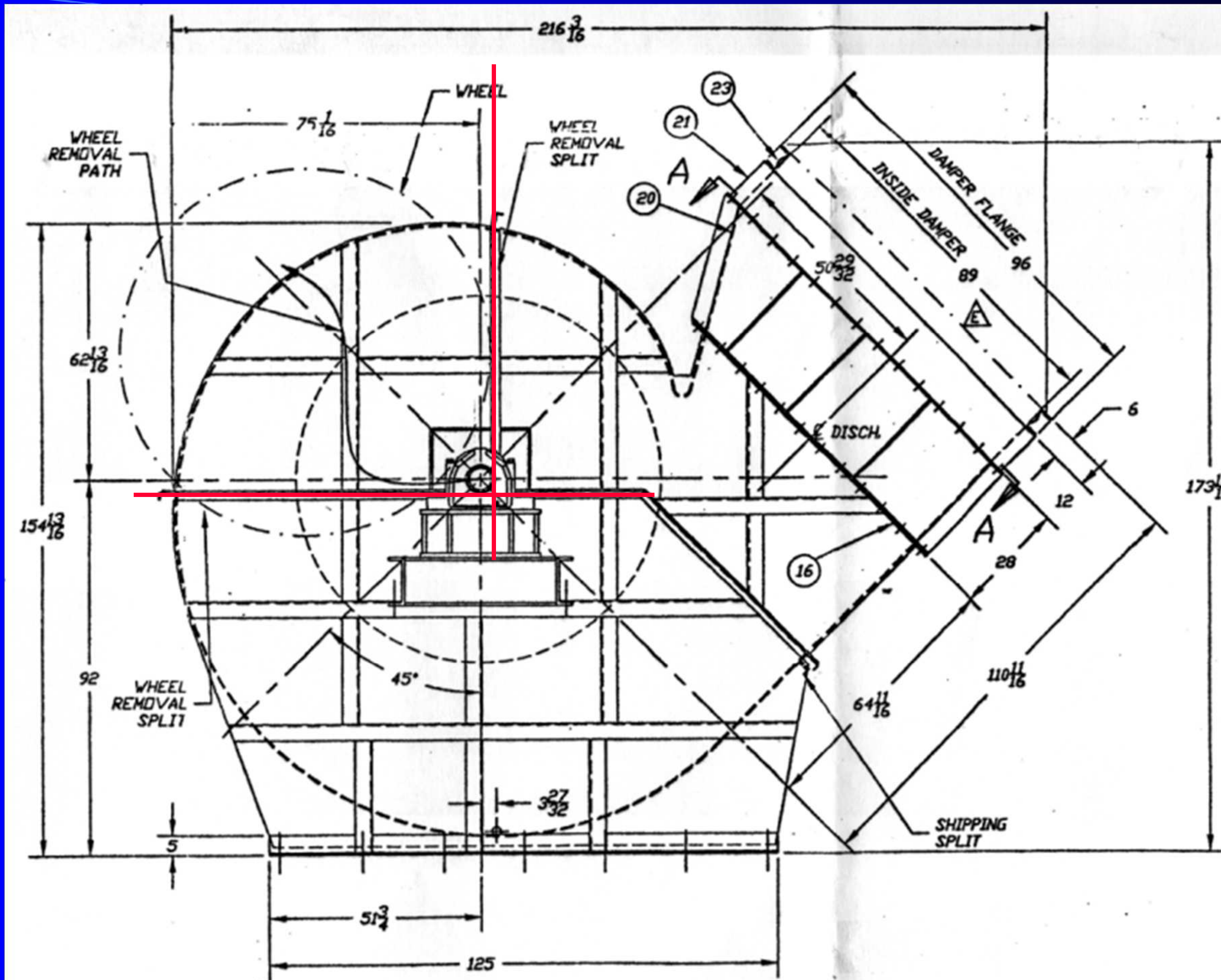



Figure 4. The Drawing by the Fan OEM Indicates the Vertical and Horizontal Flanges were Designed to be Very Close to the Shaft. Extending Lines for the Vertical and Horizontal Flanges Indicate Potential Shaft Interference.

Figure 6. Groove Worn In Fan Shaft Caused by Continuous Contact With Rubber Air Seal.

Note Fan Housing Structural Members of 4” Channel Cut with Torch to Provide Fan Shaft Clearance.

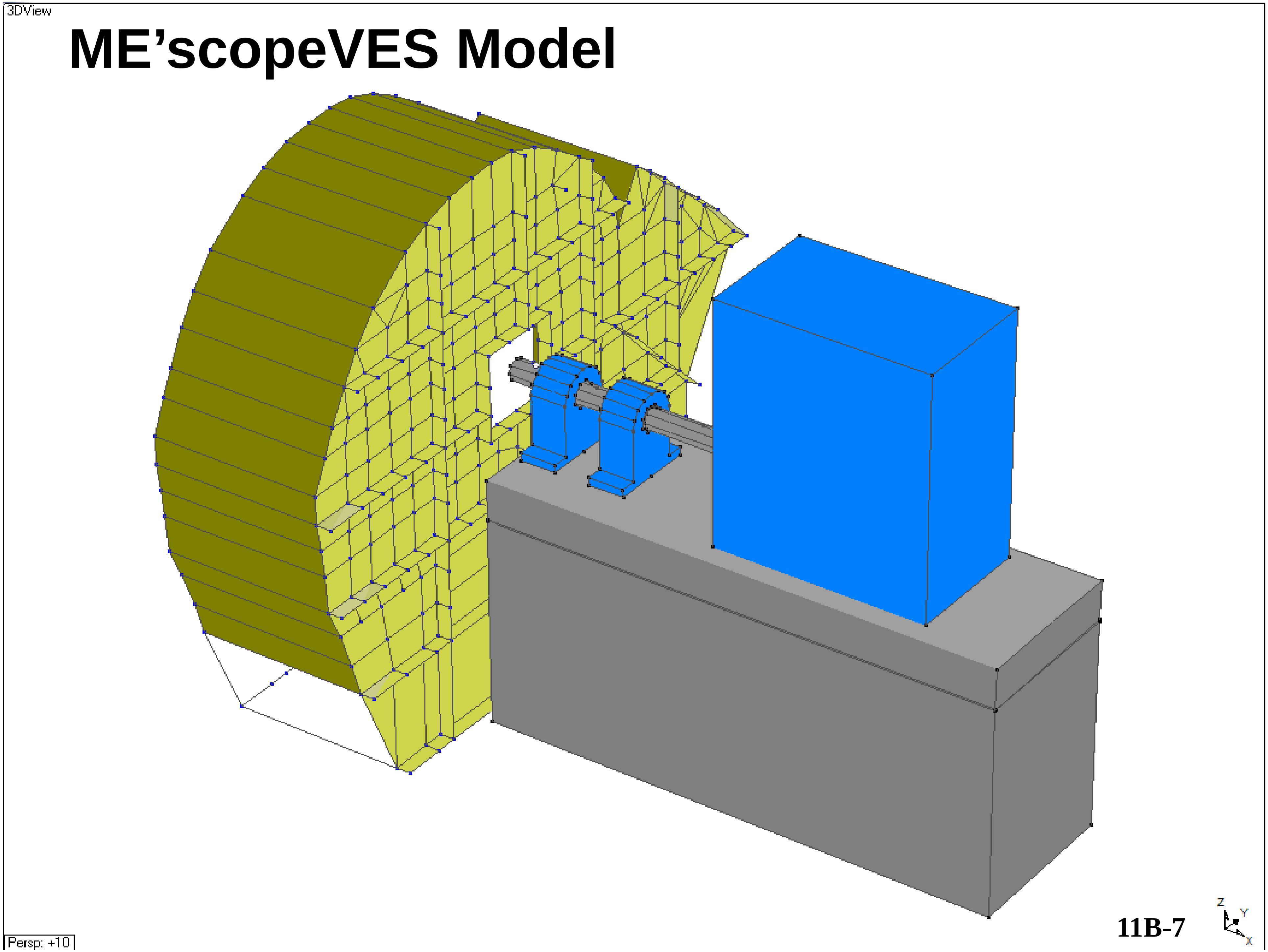
# Modal Test

- 195 Test measurement points (DOF) were laid out in a mesh as shown in the photos.
- Excitation was by impact using a PCB Instrumented 12 lb Sledge Hammer.   5 Accelerometers were roved to each DOF.

Figure 7.

Mitch France - Laying Out Modal Test Points on Fan Housing, Shaft Inlet Side

# ME’scopeVES Model



# ME'scopeVES Model DOF's (195 DOF Measured)

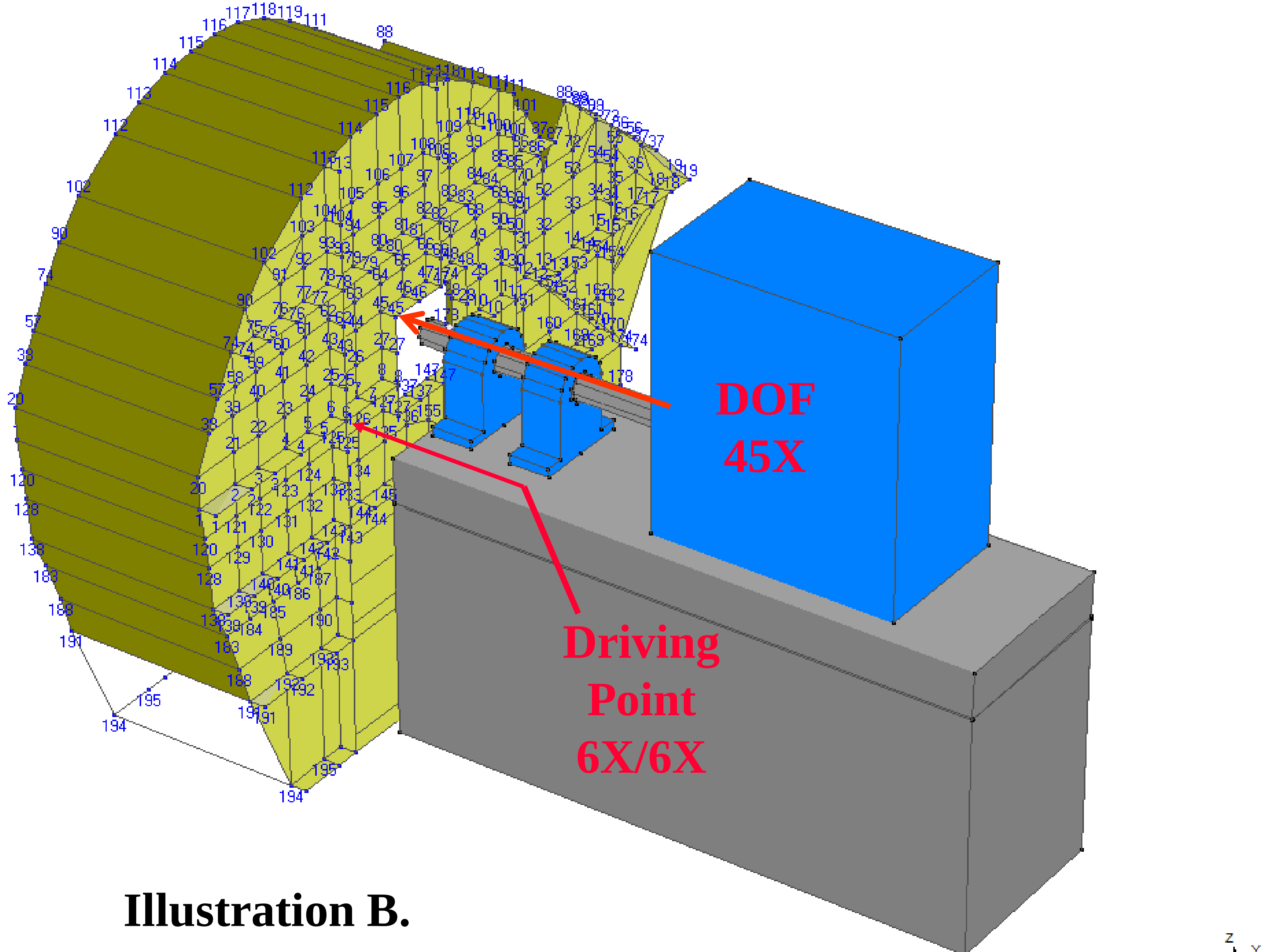



Illustration B.

**Driving Point Frequency Response Function (FRF) taken on the fan housing showed many Natural Frequencies.**

**There was a very responsive natural frequency at 1290 CPM (21.5 Hz) near Fan 1X running speed, which turned out to be the primary problem.**


**Figure 9. Frequency Response Function (FRF) of Housing Inertance Magnitude Plot ($g/lb_f$)**

## Figure 10 shows a Spectrum taken on Driving Point 6X with the fan running.

**Showed high vibration of 1.10 in/sec at 1 X Fan speed on fan housing.**

**Also showed a notable level of .384 in/sec at 3150 CPM (near the 2nd Mode identified in Figure 9).**

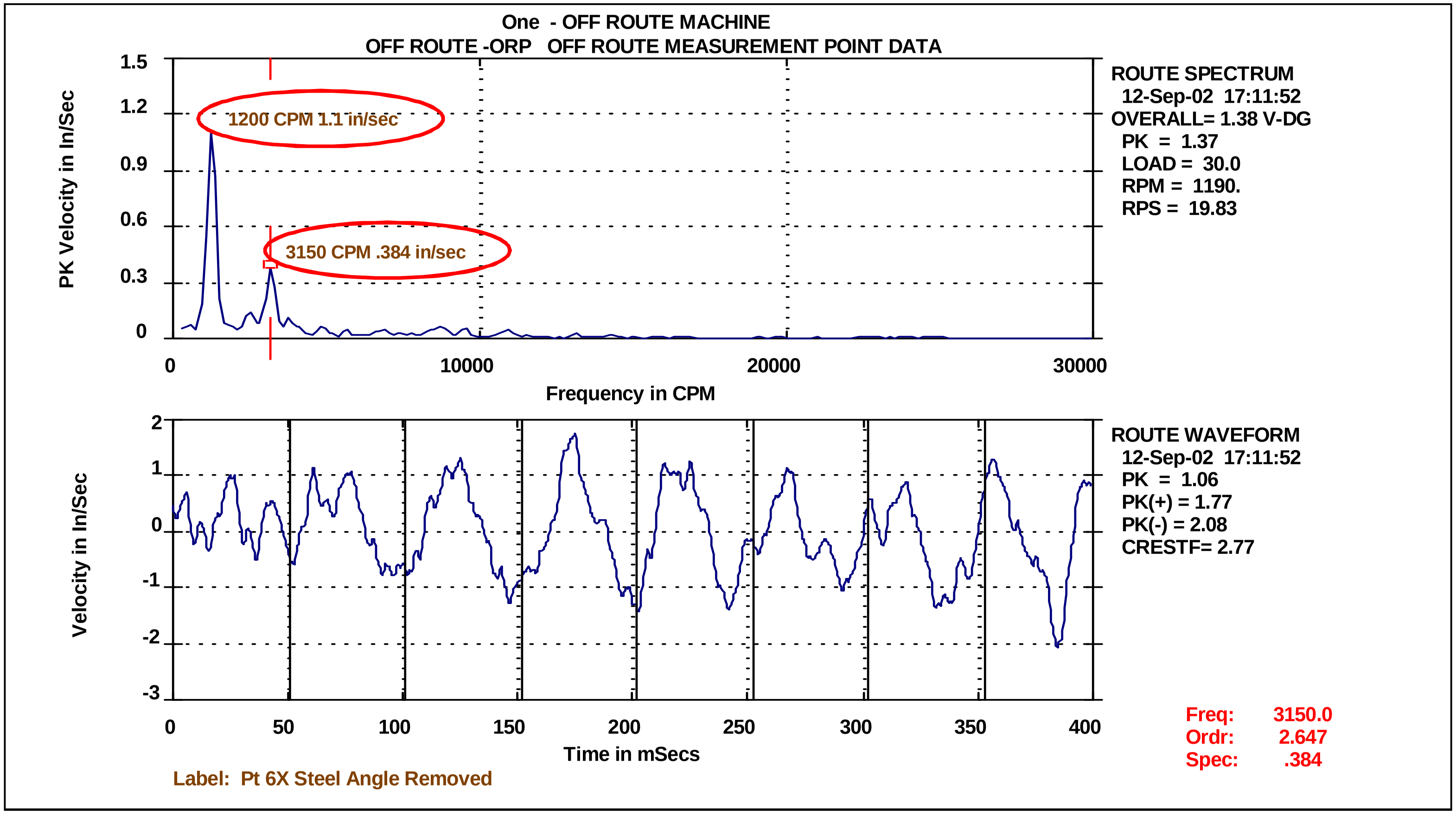



**Figure 10. Vibration at DOF 6X which was the Modal Test Driving Point Location. Most Vibration was Found at 1200 and 3150 CPM (20.0 and 52.5 Hz).**

# Fan Housing Mode Shapes

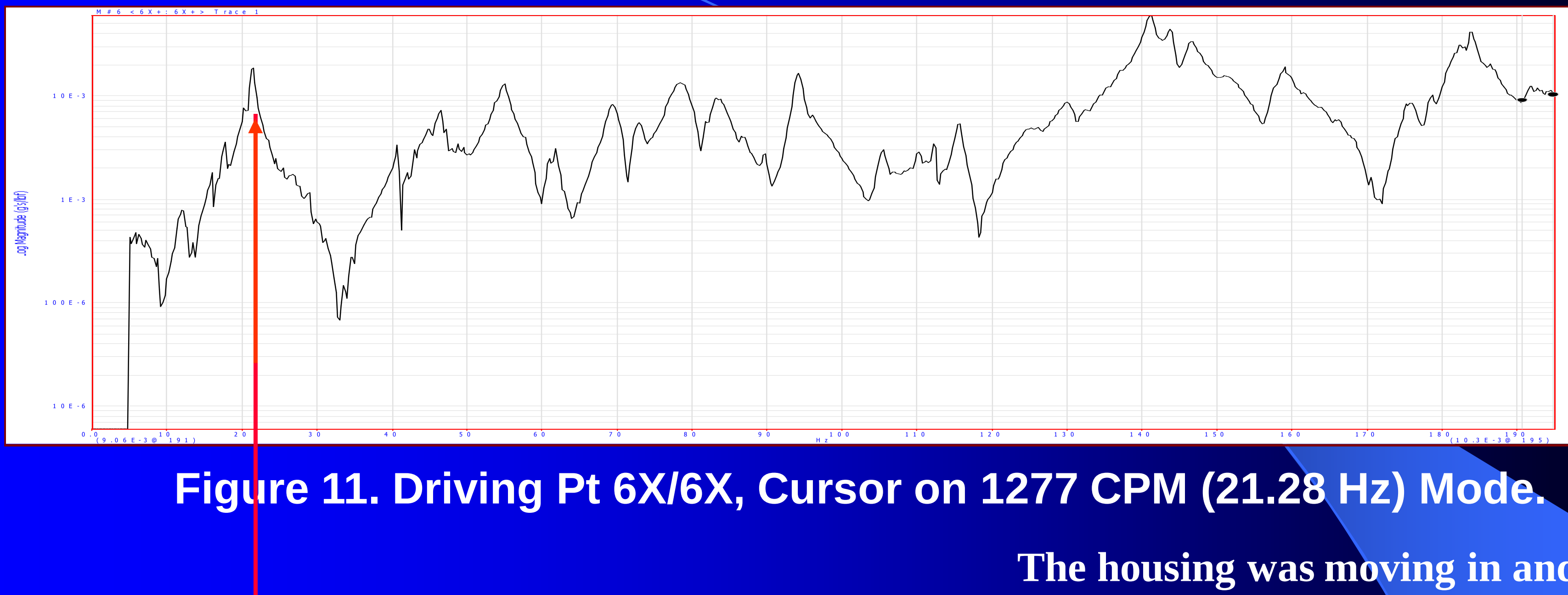



Figure 11. Driving Pt 6X/6X, Cursor on 1277 CPM (21.28 Hz) Mode.

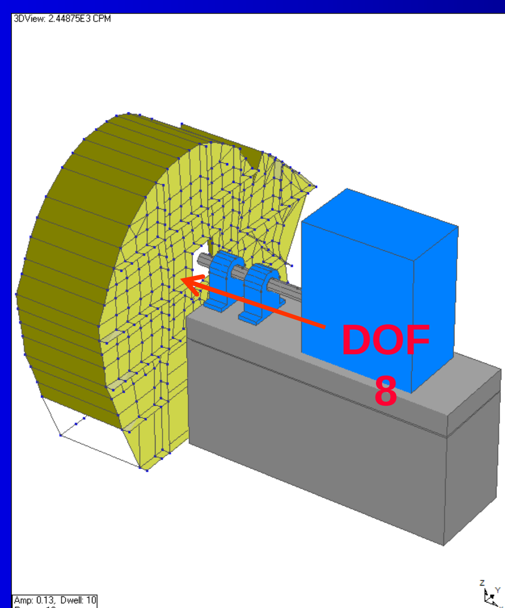



**Figure 12. Mode Shape at 1277 CPM (21.28 Hz).**

***This Circular Mode was the Primary Contributor to Fan Housing Vibration.***

**The housing was moving in and out perpendicular to the shaft axis due to cut stiffeners.**

**1st Circular Mode Shape (near 1194 RPM fan running speed).**

**Max vibration. measured during the test was DOF 8X at 1.72 ips (near the fan housing cutout where the shaft passes through the housing - see Illustration B).**

# Fan Housing Mode Shapes

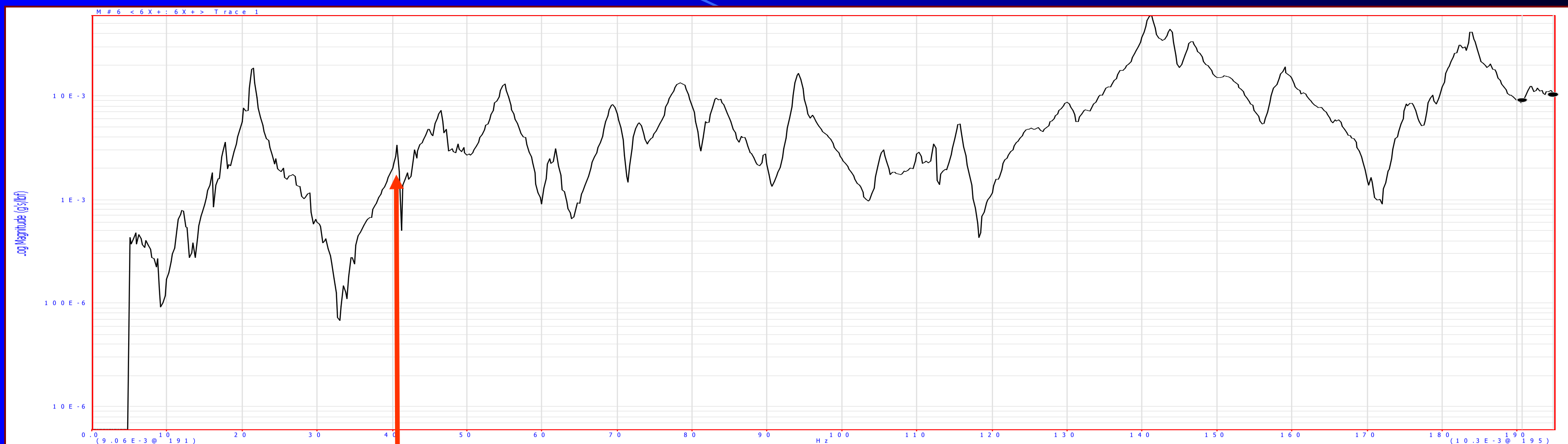



Figure 13. Driving Pt 6X/6X, Cursor on 2449 CPM (40.8 Hz) Mode.

2449 CPM (40.8 Hz)

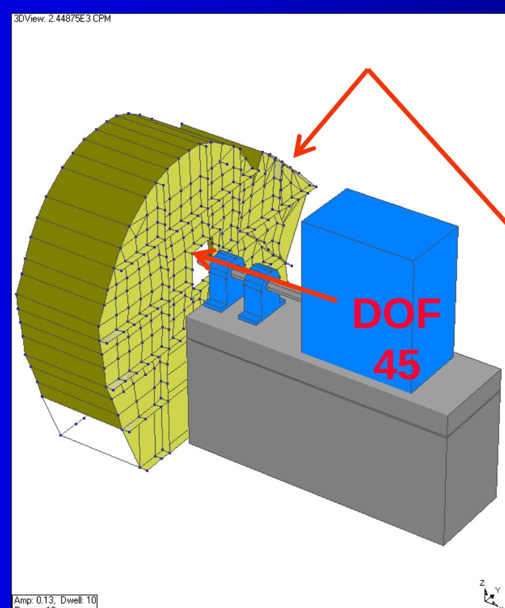



Figure 14. Mode Shape at 2449 CPM (40.8 Hz) is near 2 X Fan Speed.

The housing motion is out of phase on each side of the housing centerline.

Discharge nozzle participates in the mode – fatigue failure of welds.

Max vibration. measured during the test was DOF 45 at 0.155 ips (see Illustration B).

# Fan Housing Mode Shapes

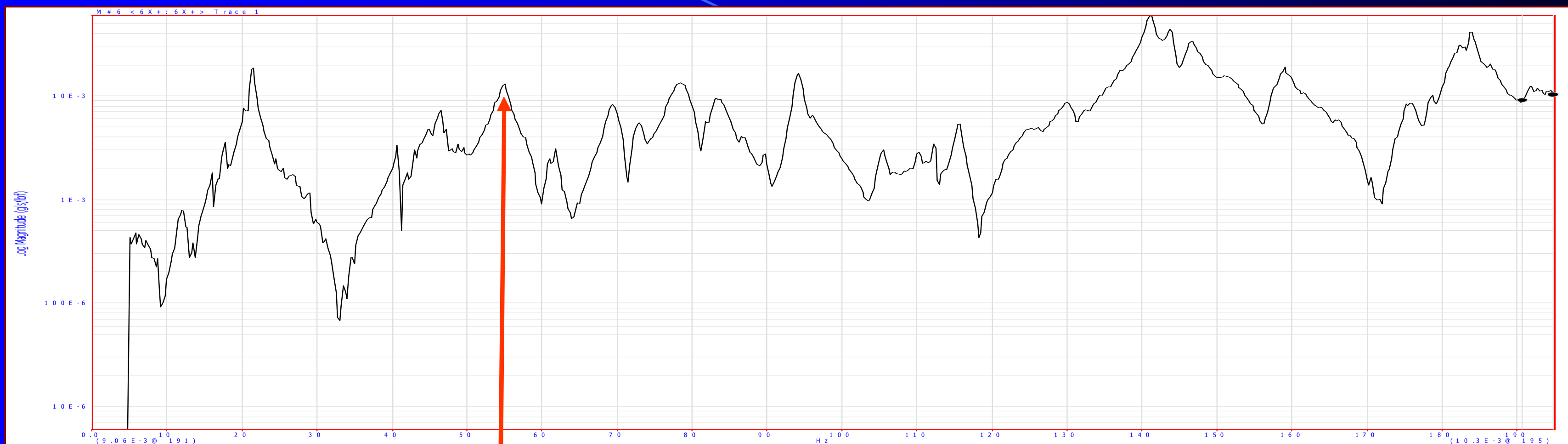


Figure 15. Driving Pt 6X/6X, Cursor on 3299 CPM (55.0 Hz) Mode.

3299 CPM (55.0 Hz)

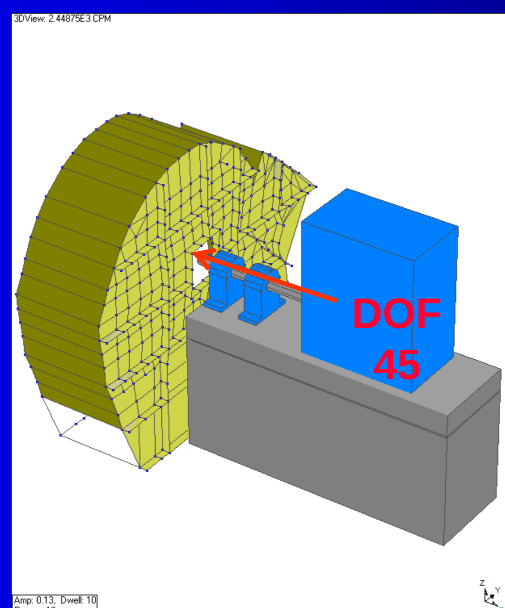



Figure 16. Mode Shape at 3299 CPM (55.0 Hz). This was the 2nd Most Responsive Mode.

The housing motion is out of phase each side of the centerline.

Discharge nozzle does not participate in the mode.

Max vibration. measured during the test was DOF 45 at 0.455 ips.

# Fan Housing Mode Shapes

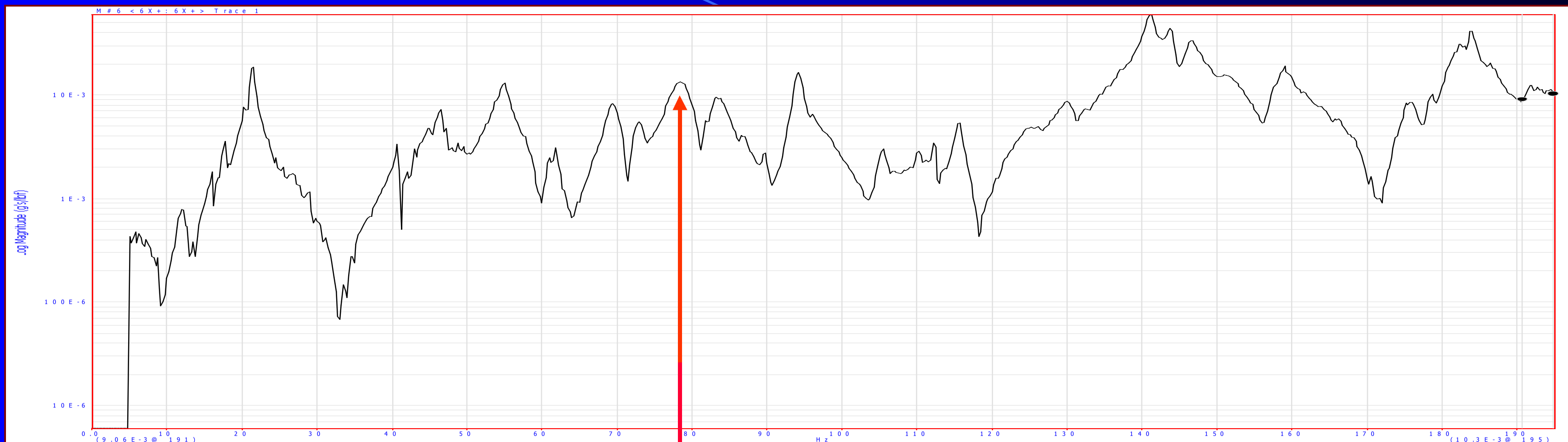



Figure 17. Driving Pt 6X/6X, Cursor on 4713 CPM (78.55) Hz Mode.

4713 CPM (78.65 Hz)

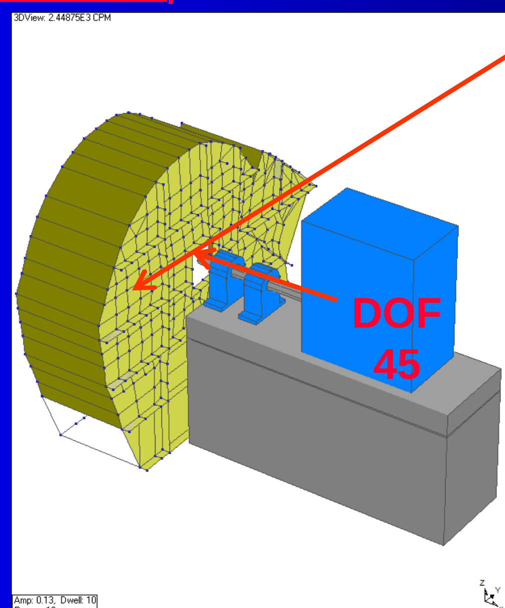



Figure 18. Mode Shape at 4713 CPM (78.65 Hz). Note Motion on Left Side of Blower Housing.

Primarily panel circular mode left side of housing.

Max vibration measured during the test was DOF 45 at 0.10 ips.

# Fan Housing Mode Shapes

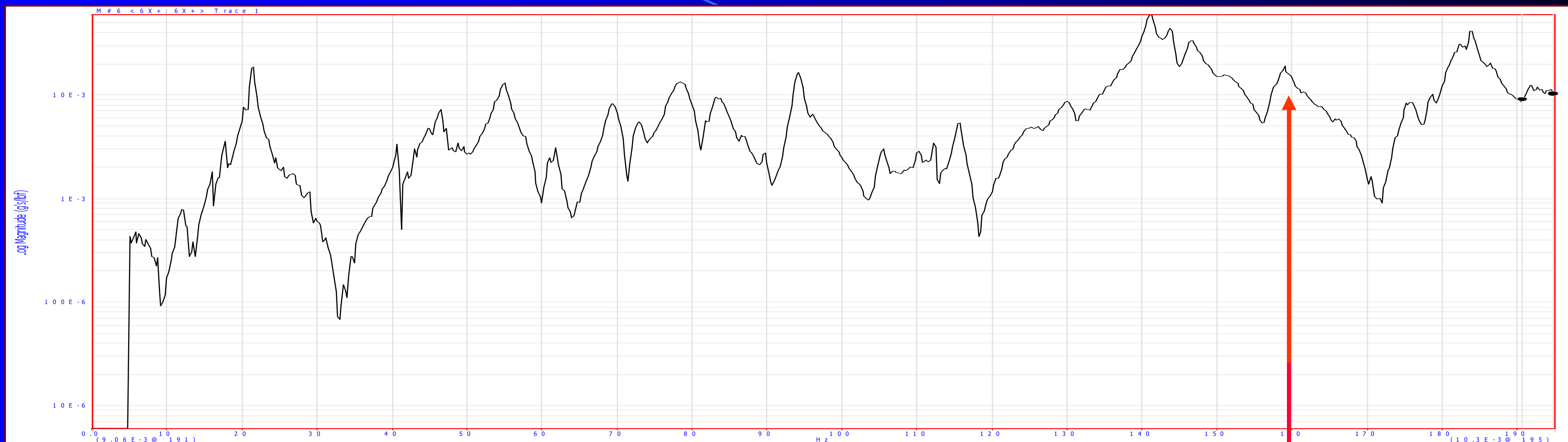



Figure 19. Driving Pt 6X/6X, Cursor on 9580 CPM (159.7 Hz) Mode.

9580 CPM (159.7 Hz)

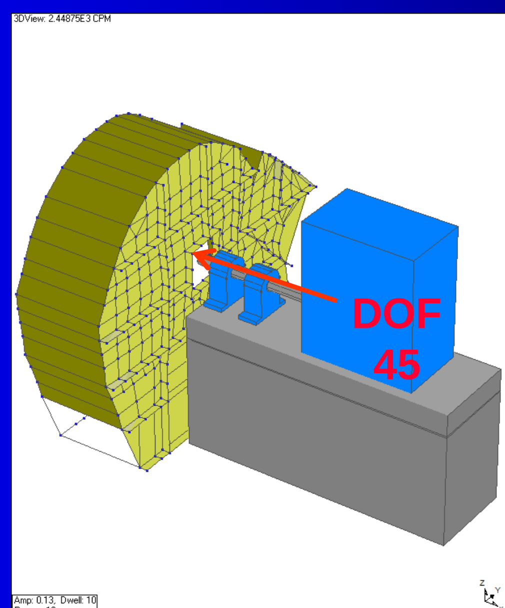



Figure 20. Mode Shape at 9580 CPM (159.7 Hz). Primarily Panel Bending Modes of Fan Housing.

Primarily panel circular modes of the fan housing.

Max vibration measured during the test was DOF 45 at 0.144 ips.

## After the Modal Test, a Temporary 4” x 4” Angle was Clamped to the Fan Housing and the Fan was Restarted.

4”X4” Angle C-Clamped to Fan Housing Channel

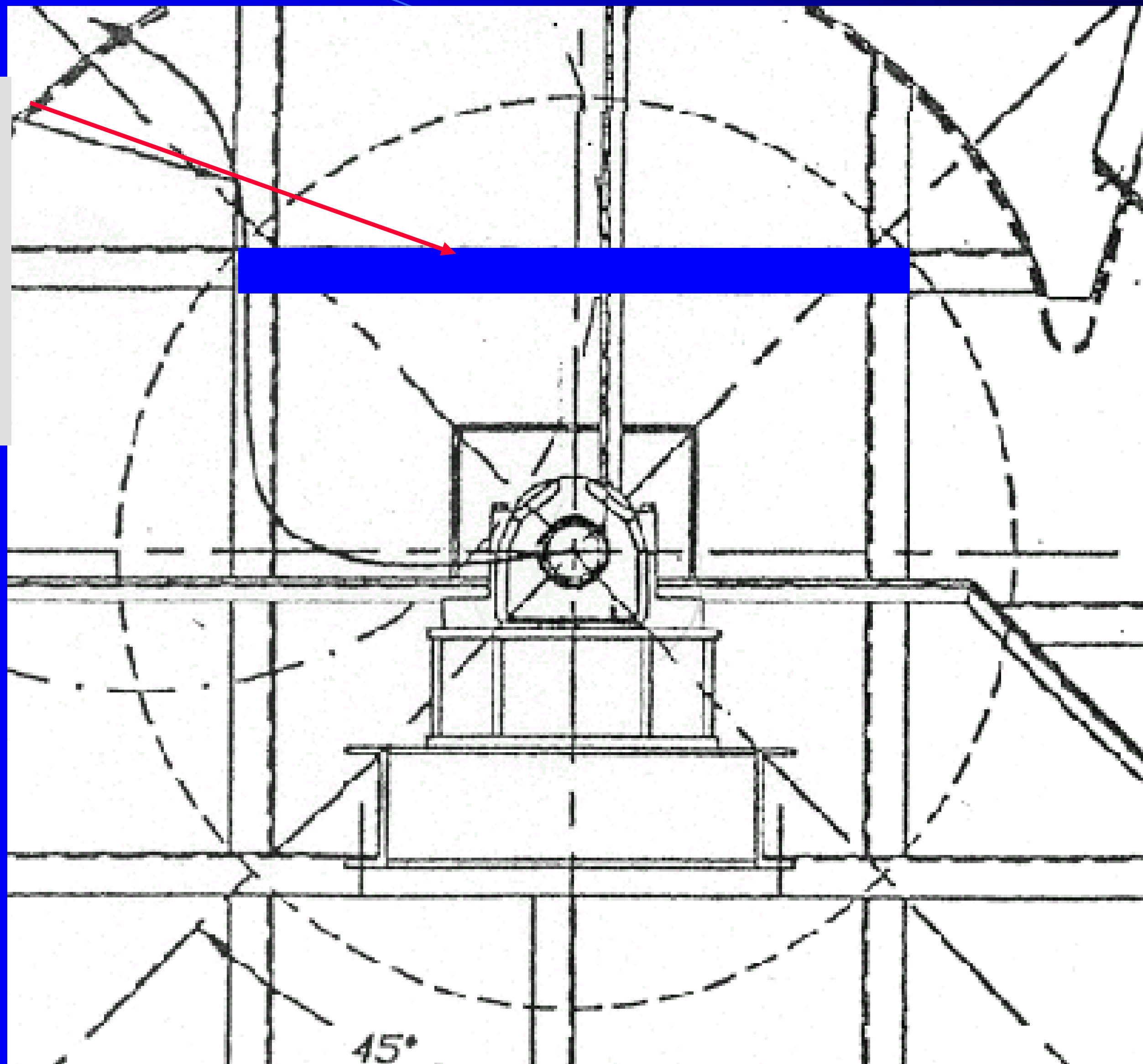
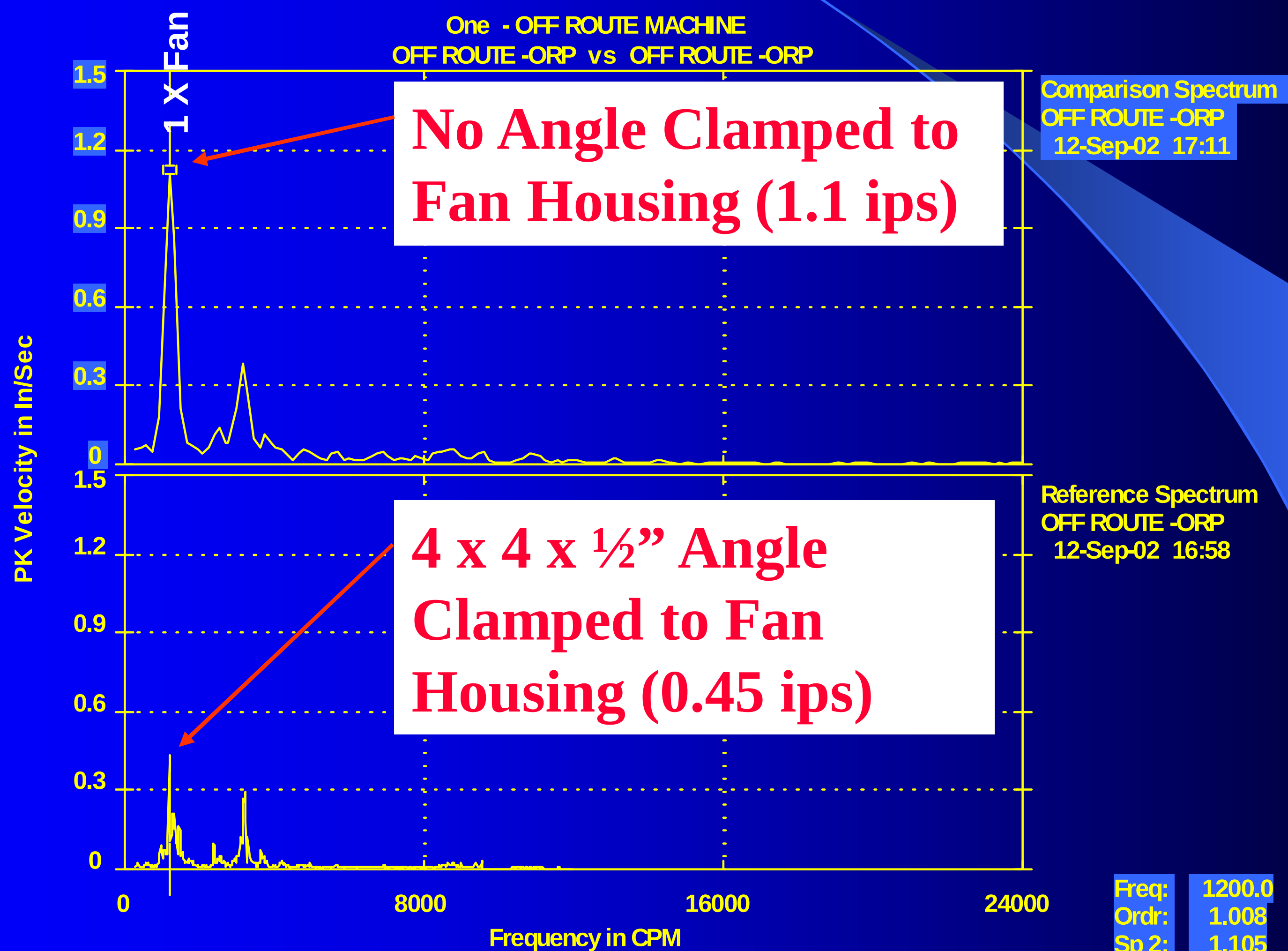



Figure 21. Vibration was Taken With & Without the Angle Attached to the Housing.

## Fan Running with Temporary 4 x 4 Angle clamped to the fan housing. Vibration was predominantly at DOF 6X (Illustration B).

## Temporary stiffener did prove to be effective in reducing vibration



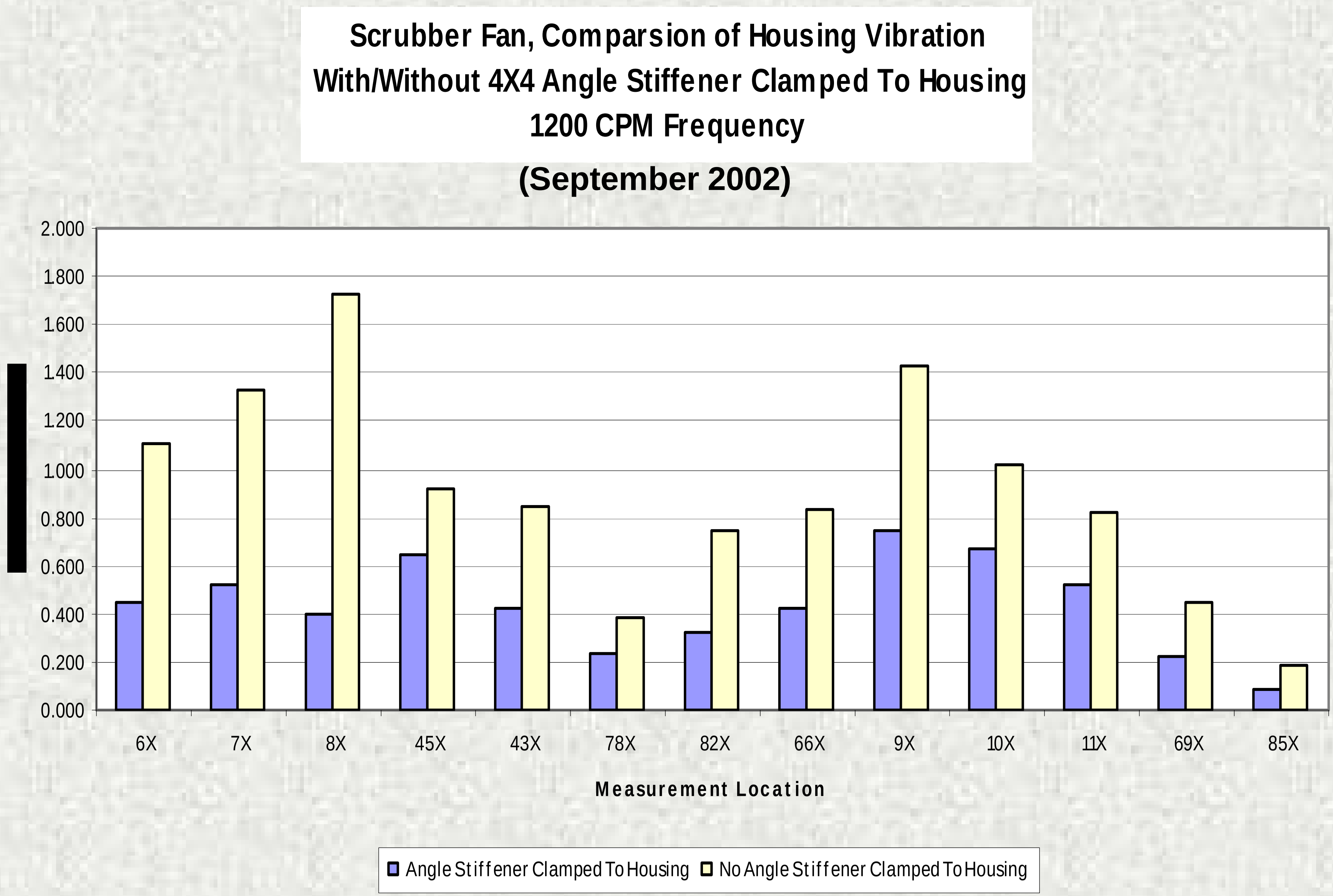



**Chart 1. Vibration at 1200 CPM Frequency with Temporary Angle Clamped to Housing Acting as a Stiffener (Blue) and the Angle Removed (Yellow).**

# Recommendations

Stiffening the fan housing to control the primary resonant amplification at 1277 CPM (21.28 Hz) was the major focus.

Structural steel stiffeners bolted and welded to the housing were proposed as follows:

1. Bolt Two 4” x 4” Angle to the housing above the bearings to control the primary mode. Use bolted attachment to permit removal of the housing section for fan wheel access.
2. Weld a 4” x 4” Angle to the housing below the centerline to help control the primary mode at 1277 CPM (21.28 Hz).
3. Weld ½” x 4” Bar stiffeners to control panel modes.

# Fan Housing Stiffening Recommendations.

4” x ½” Plate Stiffeners welded to the housing and existing angle.

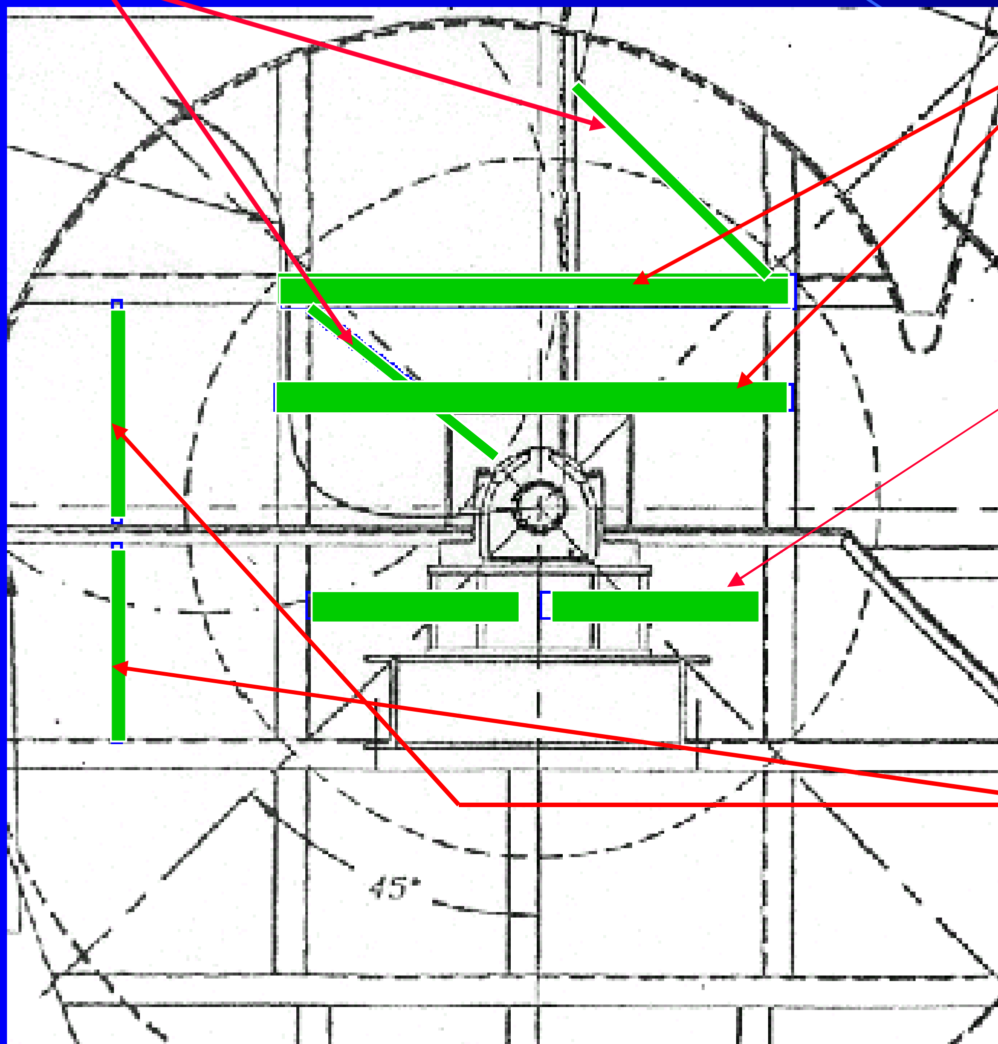



Figure 22.

Two 4” x 4” x 86” angle bolted to the existing vertical angle and channel.

Two 4” x 4” pieces of angle welded to the housing and existing channel.

4” x ½” Plate Stiffeners welded to the housing and existing angle.

A site visit in Dec 2004, two years after the modal test, provided an opportunity to recheck vibration and the structural modifications that were actually made to the fan.

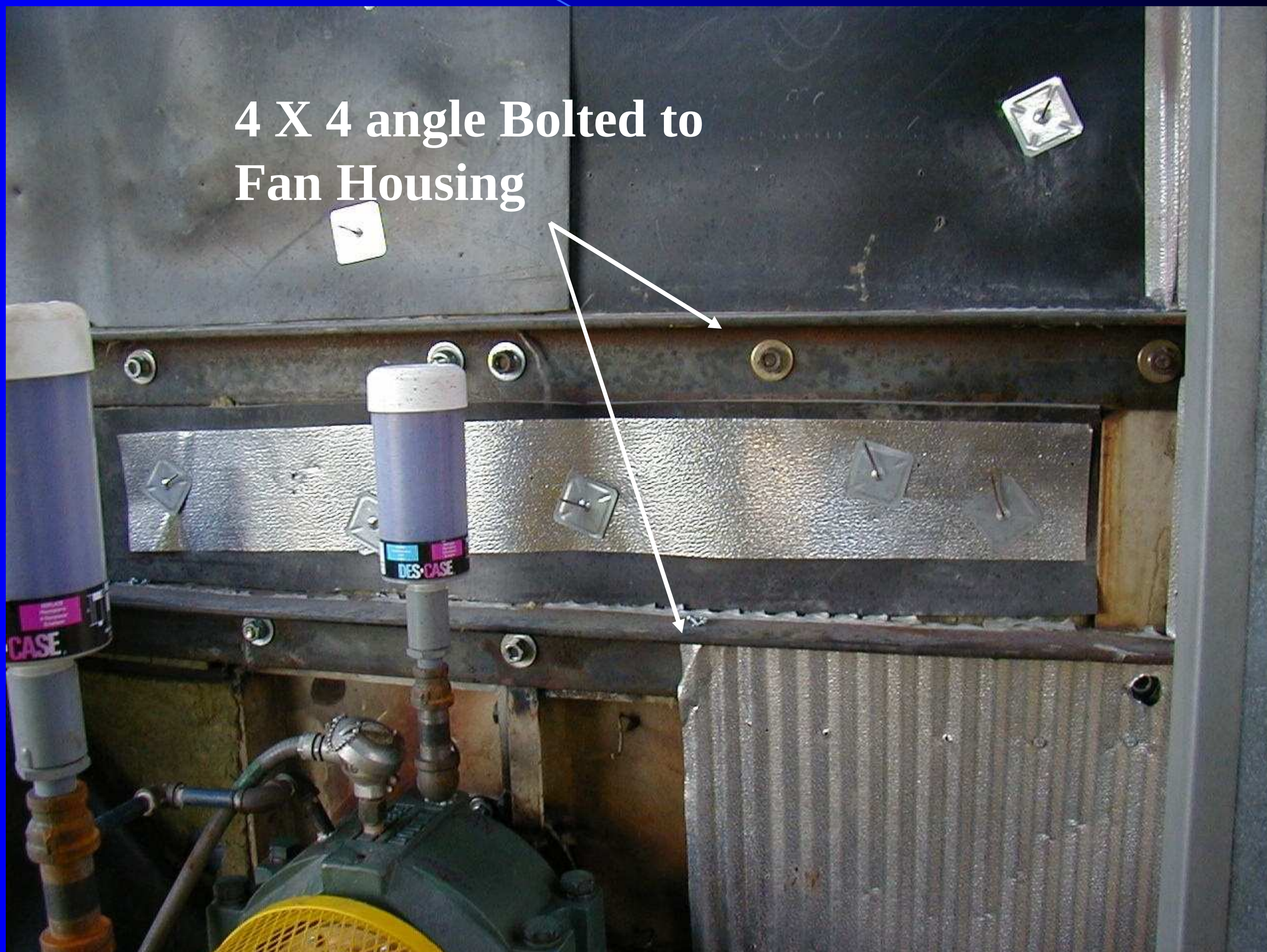



Figure 23.

Initial vibration measured 1.6 in/sec, pk at 40% inlet guide vane setting.

After angle was bolted to the housing, vibration was reduced to 0.15 in/sec, Pk at 1X, showing a 10.7:1 reduction (91% decrease).

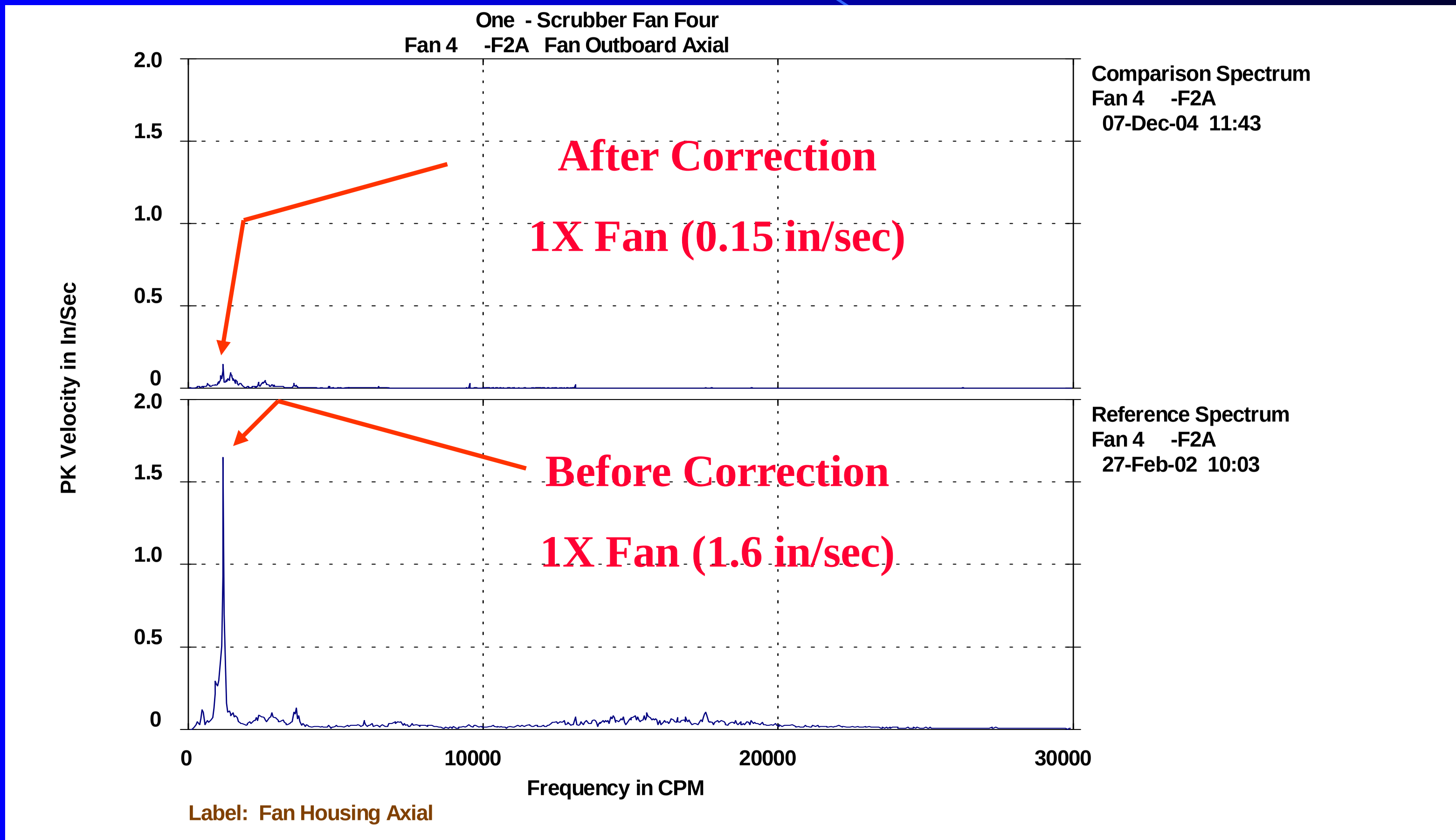



Figure 24. Comparison of Before and After Vibration Spectra Measured on Fan #4 Housing, DOF 6X at Fan Housing Split Line Structural Member.

Initial Fan 1X vibration measured 1.6 in/sec, pk at 40% inlet guide vane setting.

After angle was bolted to the housing, vibration was reduced to 0.15 in/sec, Pk at 1X, showing a 10.7:1 reduction (91% decrease).

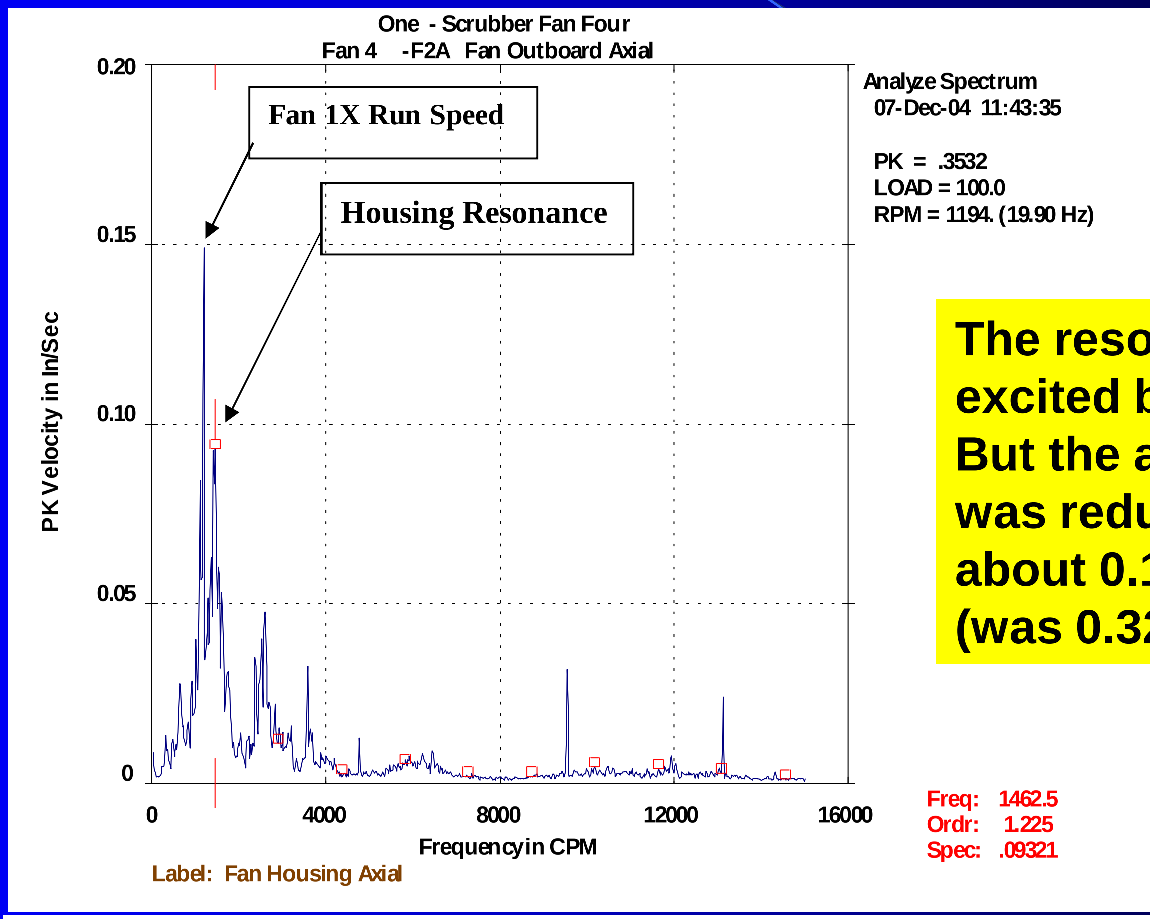



Figure 25. The Housing Vibration after Angle Bolted to Housing. Spectrum Shows the Housing Natural Frequency has Increased from Approximately 1277 CPM to 1462 CPM (20.0 to 24.4 Hz).

**The resonance is still excited by air flow; But the amplitude was reduced to only about 0.1 in/sec, pk (was 0.32 in/sec, pk).**

**The resonance is still present but is shifted 14.5% higher in frequency away from fan running speed.**

**After modifications, the Mode measured 1462 CPM or 22.0% from Fan 1X. Prior to modifications, the Mode measured 1277 CPM or 6.5% from Fan 1X.**

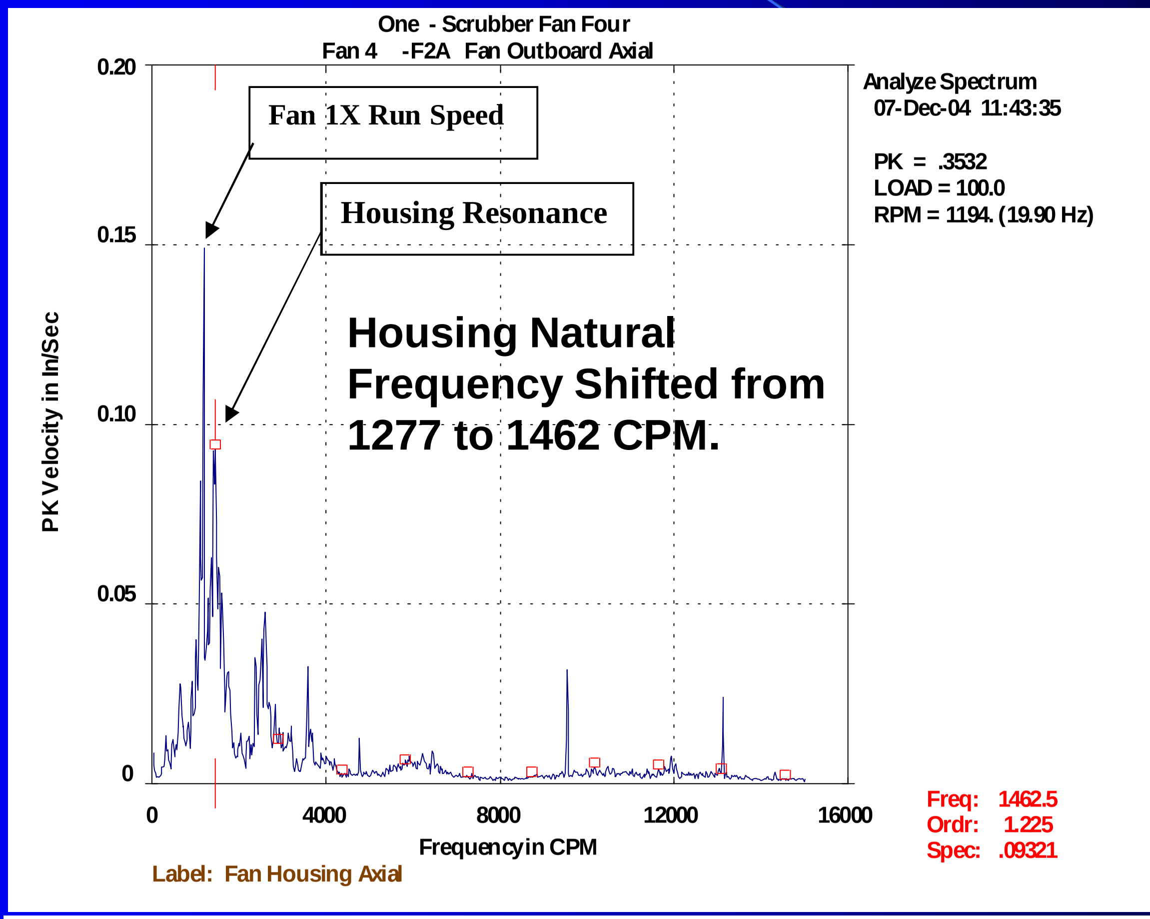



**Figure 25. The Housing Vibration after Angle Bolted to Housing. Spectrum Shows the Housing Natural Frequency has Shifted from Approximately 1277 CPM to 1462 CPM (21.3 to 24.4 Hz).**

## CONCLUSIONS:

1. The vibration and modal test data confirmed the suspected structural resonances of the fan housing.

2. During original installation of the fans, the mating joint flanges at the center of the housing had been removed by a cutting torch to provide clearance for the shaft. The removal of these stiffeners significantly reduced rigidity of the shaft entry side of the fan housing.

3. Most housing vibration response was at five natural frequencies – 1277, 2449, 3299, 4713 and 9600 CPM. Excitation of the 1277 CPM mode was the primary contributor to housing vibration. The 3299 CPM mode was the 2nd most responsive.

4. During the modal test, the inability to isolate the fan from air leakage caused significant housing vibration. This background vibration made it more difficult to obtain high quality modal test data. A medium sledge modal hammer was too small to input enough force to excite the housing above the background vibration. An instrumented sledge hammer was required to input the 1500 to 2000 $lb_f$ necessary to excite the housing.

## CONCLUSIONS:

5. The modal test data were curve fit to determine the mode shapes of the housing natural frequencies. After reviewing the mode shapes, it was decided to attach a temporary stiffener of 4 x 4 inch angle to the vertical flanges of the housing using heavy duty C clamps. Vibration data taken with the angle stiffener clamped to the housing showed that vibration of the 1277 CPM resonant response would be lowered significantly if stiffening at the proper locations was added.

6. Recommendations were made to bolt and weld structural members to the fan housing based on review of the mode shapes (see Figure 23).

7. After the stiffeners were installed, the vibration levels were reduced from about 1.6 in/sec to 0.15 in/sec, (91% reduction). The fan housing $1^{st}$ mode shifted from about 1277 CPM to 1462 CPM providing a separation margin of 22.5% from running speed (see Figures 24 and 25).

8. This project took approximately three years from initial inspection of the fans, recommendation to perform modal testing, conducting the modal test, installing the housing stiffeners and rechecking vibration.

# Acknowledgement:

Thanks to Mitchell France of Technical Associates of Charlotte, PC for assistance with the modal test, data reduction and formulating a solution.

The End