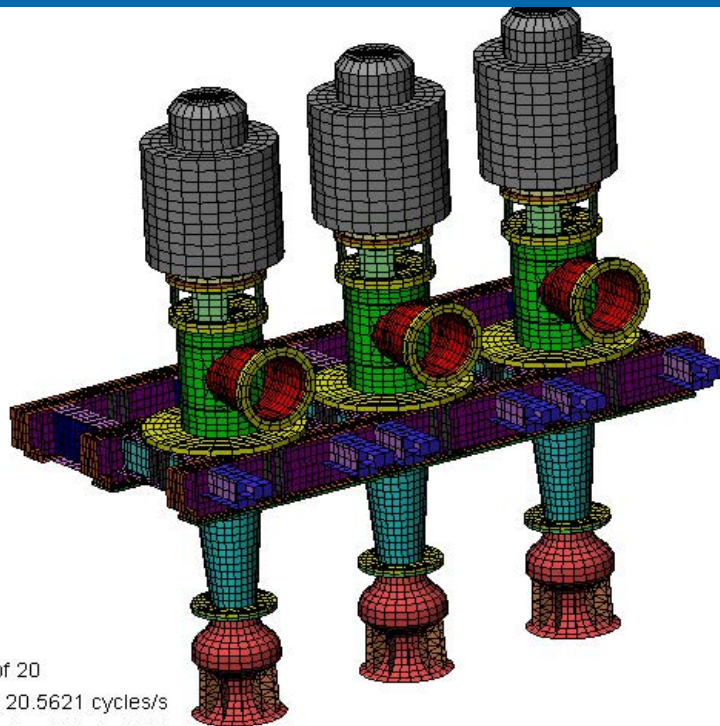


Vibration Analysis

by Robert J. Sayer, PE Applied
Structural Dynamics Medina, Ohio
Michigan Water Environment Assn



Mode: 12 of 20
Frequency: 20.5621 cycles/s
Maximum Value: Not Available
Minimum Value: Not Available

Vibration Analysis can be used for:

- Predictive/Proactive Maintenance & Operations: Determine if equipment requires maintenance and predict what & when such maintenance can be done such that impact on production is minimized.
- Troubleshooting: Determine root-cause of failures or reduced reliability.
- Design & Certification: Vibration levels can be specified to increase probability that equipment will meet it's intended reliability. Vibration and Modal Testing can be performed prior to shipment to insure that specifications are met. Numerical FEA studies can be used in Design Stage prior to manufacture of equipment.

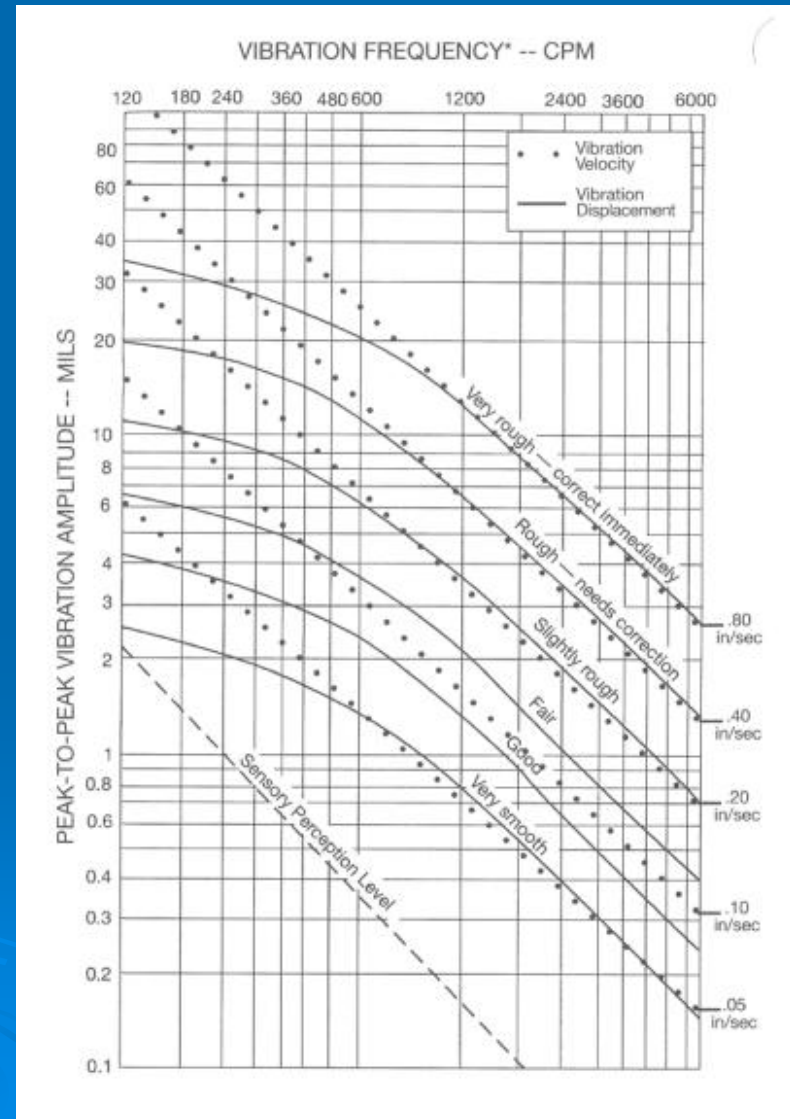
Vibration Trend

- Typical Overall Vibration Trend. Does not provide information as to the source of vibration.
- Indicates change in mechanical condition.



Vibration Severity - Mechanical Reliability

- Rathbone Chart
- This chart and others like it based upon Rathbone work in the 1940's. It is still used today. It does not distinguish between equipment types and sizes.
- Velocity = .05 ips is Very Smooth
- Velocity = .125 ips is Fair
- Velocity = .40 ips Rough (needs correction)
- Velocity = .80 Very Rough (correct immediately)

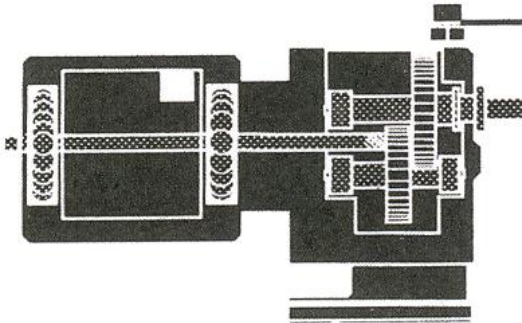


Vibration Severity - Mechanical Reliability

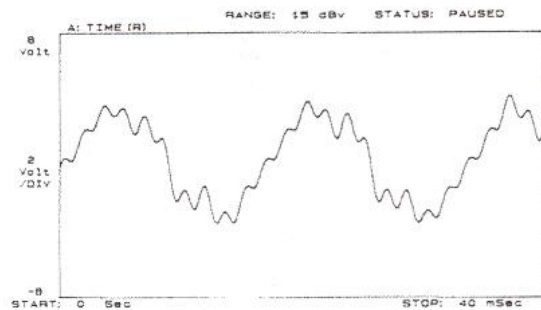
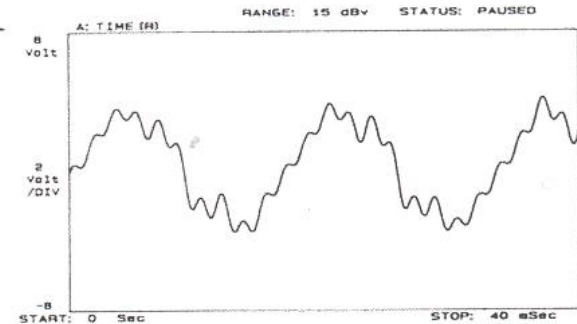
- ISO 10816-1
- Like Rathbone Chart is based upon allowable velocity.
- Distinguishes between size of equipment and accounts for support conditions of large machines.
- i.e. Large Machines (>400 HP) on Rigid Support: Satisfactory = 0.16 ips; Alert = 0.40 ips; Danger = 1.0 ips

Velocity Severity		ISO 10816-1 Velocity Range Limits and Machine Classes			
mm/s RMS	in/s Peak	Small Machines Class I (<20 HP)	Medium Machines Class II (20 HP – 400 HP)	Large Machines	
				Rigid Supports Class III	Less Rigid Supports Class IV
0.28	0.02	Good	Good	Good	Good
0.45	0.03				
0.71	0.04				
1.12	0.06	Satisfactory	Satisfactory		
1.80	0.10				
2.80	0.16	Unsatisfactory (alert)	Unsatisfactory (alert)	Satisfactory	Satisfactory
4.50	0.25				
7.10	0.40	Unacceptable (danger)	Unacceptable (danger)	Unsatisfactory (alert)	Unsatisfactory (alert)
11.20	0.62				
18.00	1.00			Unacceptable (danger)	Unacceptable (danger)
28.00	1.56				
45.00	2.51				

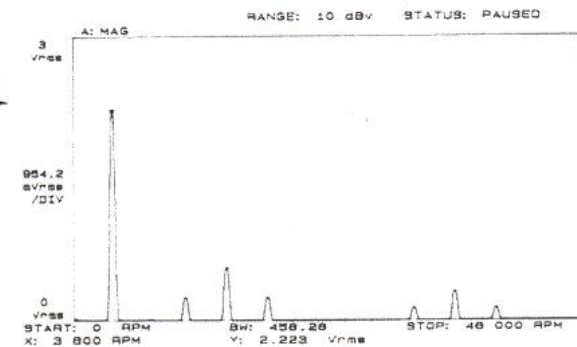
Vibration Analysis Process



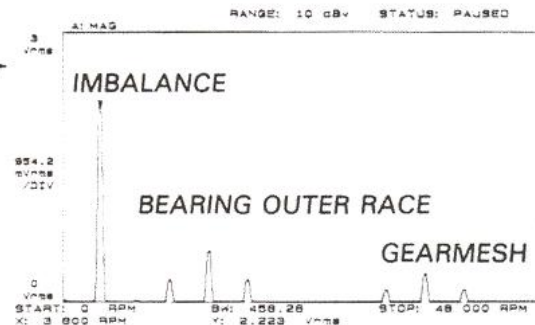
1) CONVERT VIBRATION
TO AN ELECTRICAL SIGNAL



2) REDUCE THE SIGNAL
TO ITS COMPONENTS



3) CORRELATE SIGNAL
COMPONENTS WITH
MACHINERY

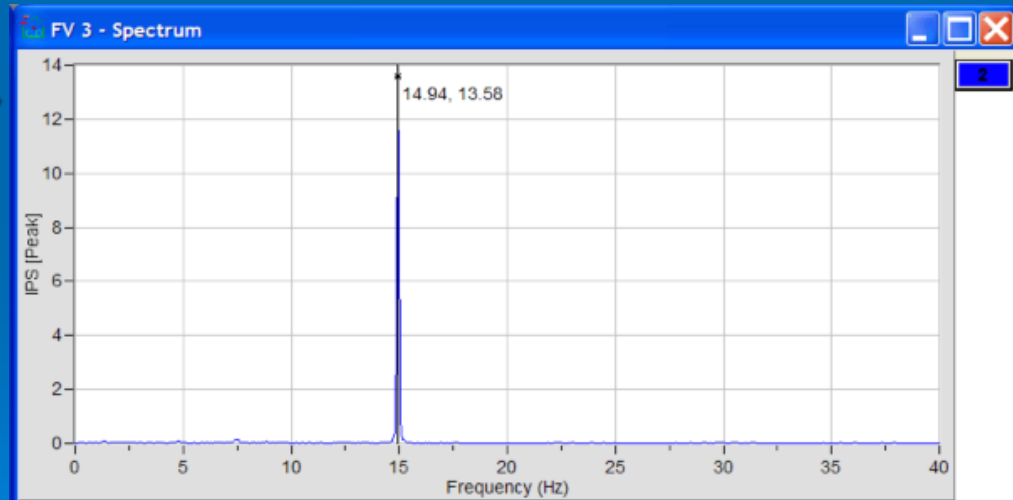
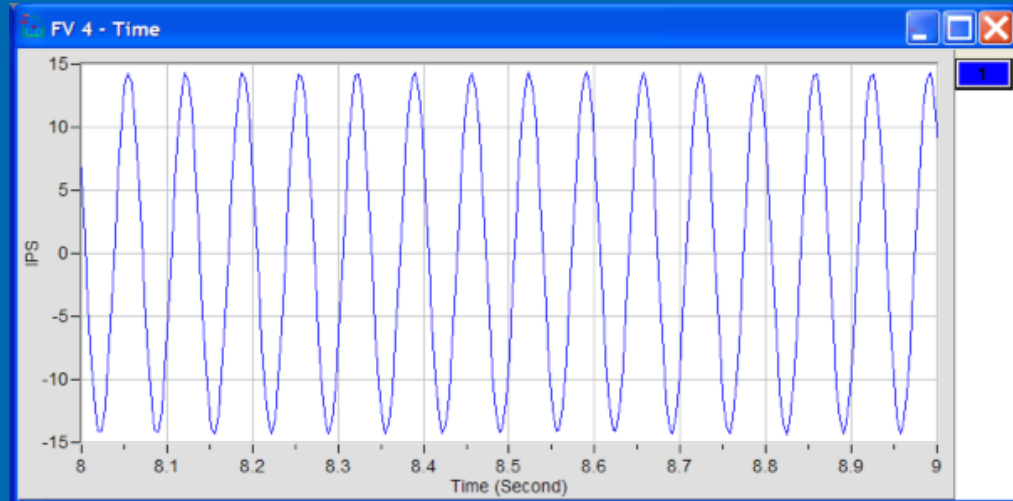


4) DOCUMENT,
ANALYSE
AND ARCHIVE
RESULTS

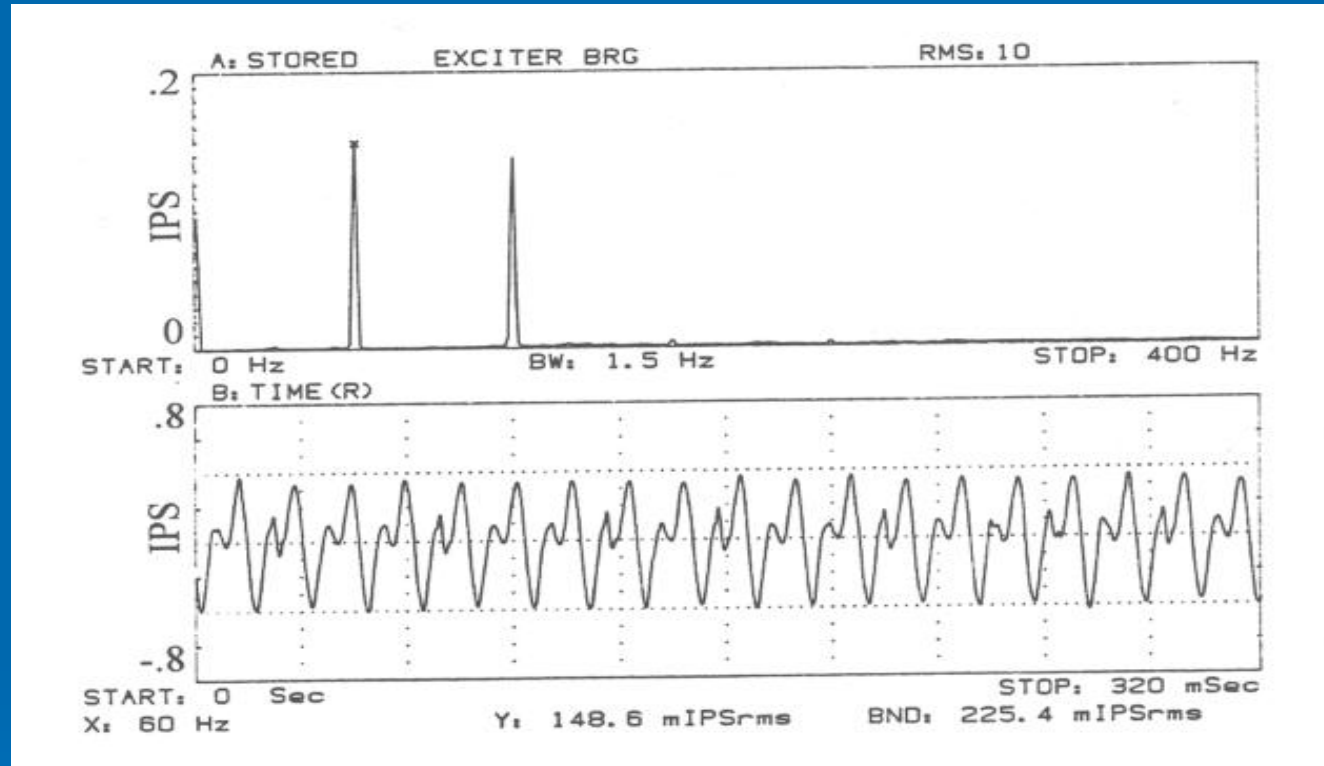
MAINTENANCE
RECORD/TEST
/ANALYSIS
REPORT, ETC.

FFT of Classic Unbalance

- Waveform & Frequency Spectrum of Pure Unbalance.
- Waveform is a Pure Sine Wave.
- Frequency Spectrum has only one indicator and it is at the speed (frequency) of the machine.



FFT of Classic Misalignment

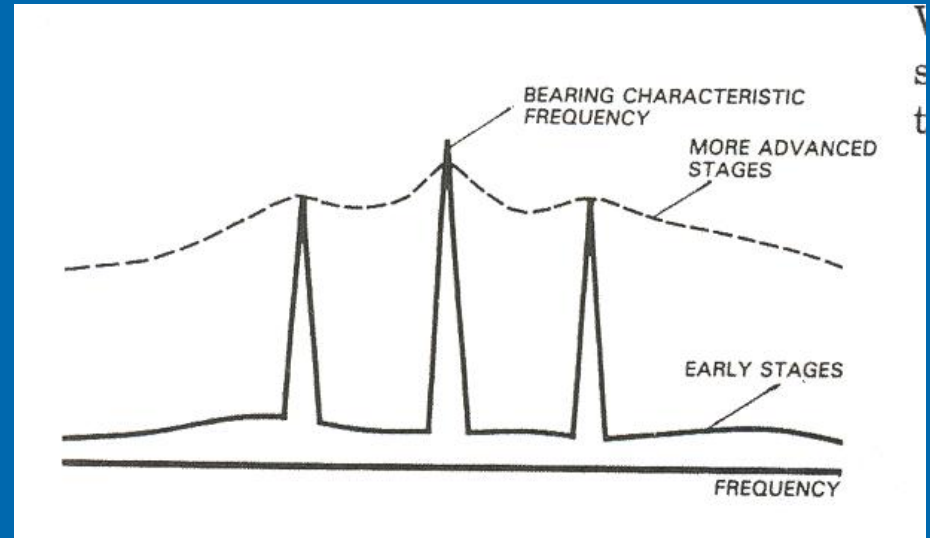
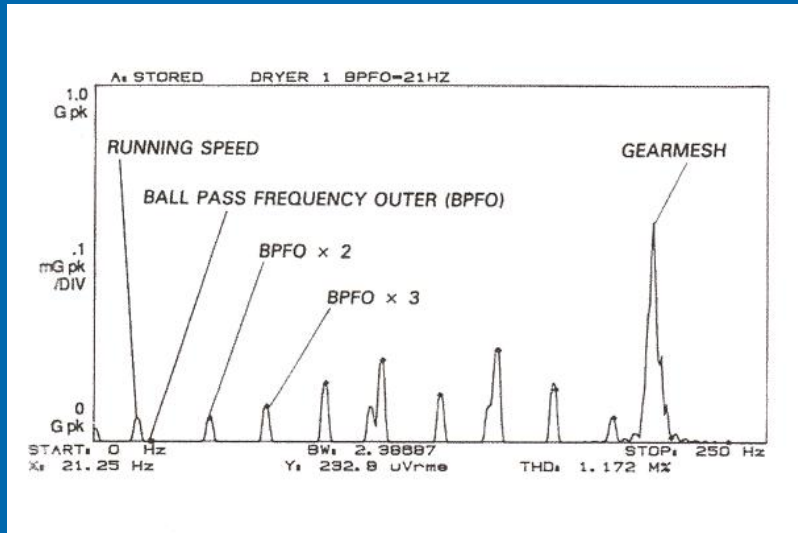


- Waveform & Frequency Spectrum (Misalignment)-High 2X Vibration
- From Eshlemen "Basic Machinery Vibrations"

Bearing Fault Analysis

- The most common use of vibration analysis for predictive/proactive maintenance is roller element bearing fault analysis.
- Components of a roller element bearing can develop faults which produce impact forces as a ball passes faults in the outer race (BPFO), inner race (BPFH) or faults in the roller elements (BSF) or cage faults. All of these conditions occur at distinct frequencies which can be determined for each bearing.
- The frequency spectrum is key in tracking bearing faults. The objective of this analysis is to determine the optimal time for bearing replacement.

Bearing Fault Analysis



- Frequency Spectrum can be used to identify the fault type, then to track it's progression with time.
- Historic bearing failure/vibration data can be used to specify vibration level at which bearing should be removed.
- Vibration trend at fault frequency(ies) can be used to predict future vibration.

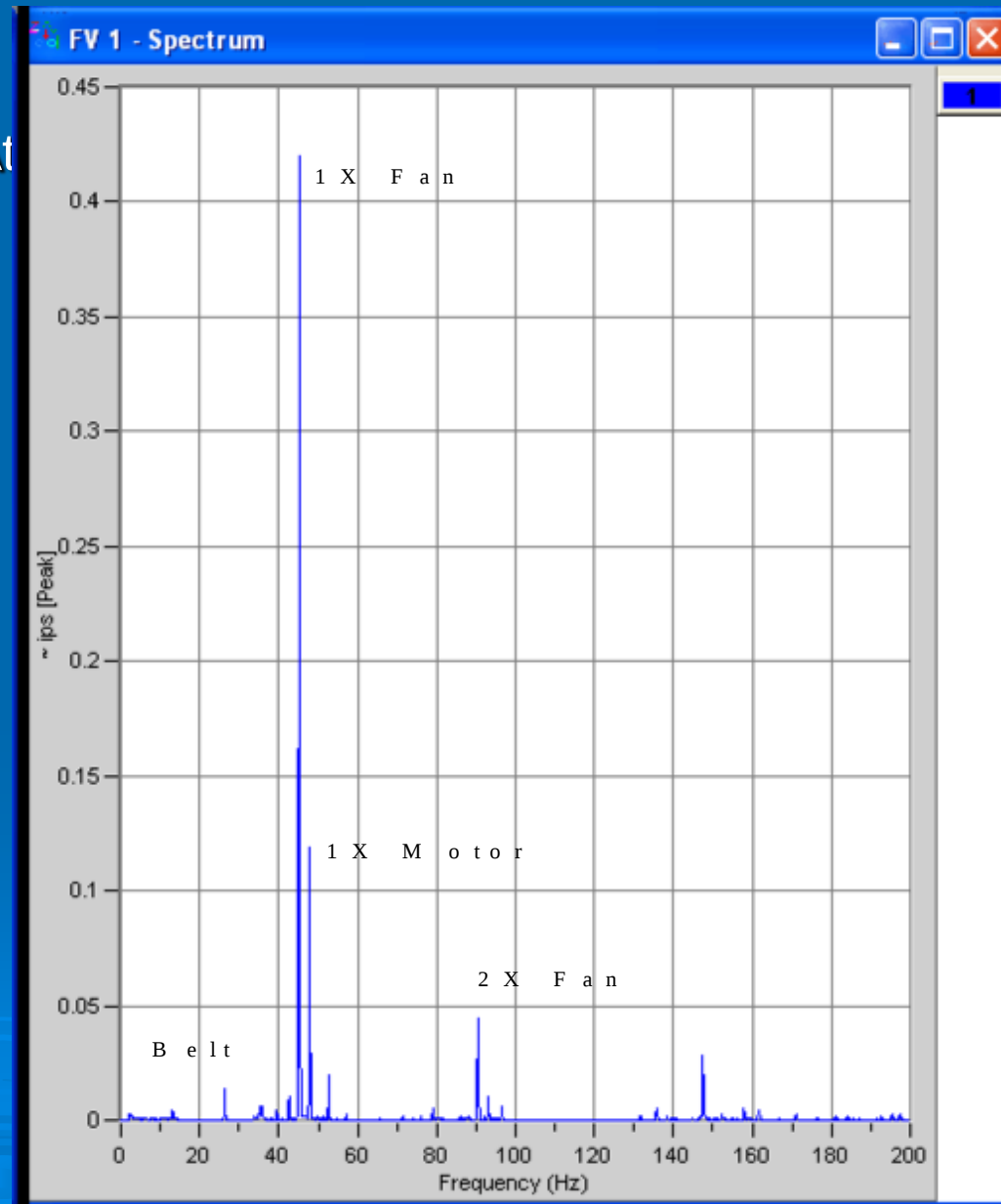
Fan on Isolation Base

- Vibration Level excessive.
- Are the Vibrations a result of a Mechanical Source or Aerodynamic Source?
- Do we:
 - Balance the Fan?
 - Send the Motor out for Repair?
 - Change the Belts?
 - Change Operating Characteristics of the Fan?
 - Change the Isolator Springs?
 - All of the above & hope for the best?



Vibration Data vs Sources

- Fan speed controlled by VFD. At normal operating conditions:
- 1x Fan = 45.3 Hz
- 1x Motor = 47.9 Hz
- 1x Belt = 13.1 Hz
- Most of the energy is associated with vibration tied to the fan.
- Thus, maintenance on motor or belts would not be productive.
- The vibration is not associated with aerodynamic source.



Case History



Large ID Fan Exhaust Duct Noise & Vibration Problem
Site suspected aerodynamic excitation from unusual
placement of outlet damper.

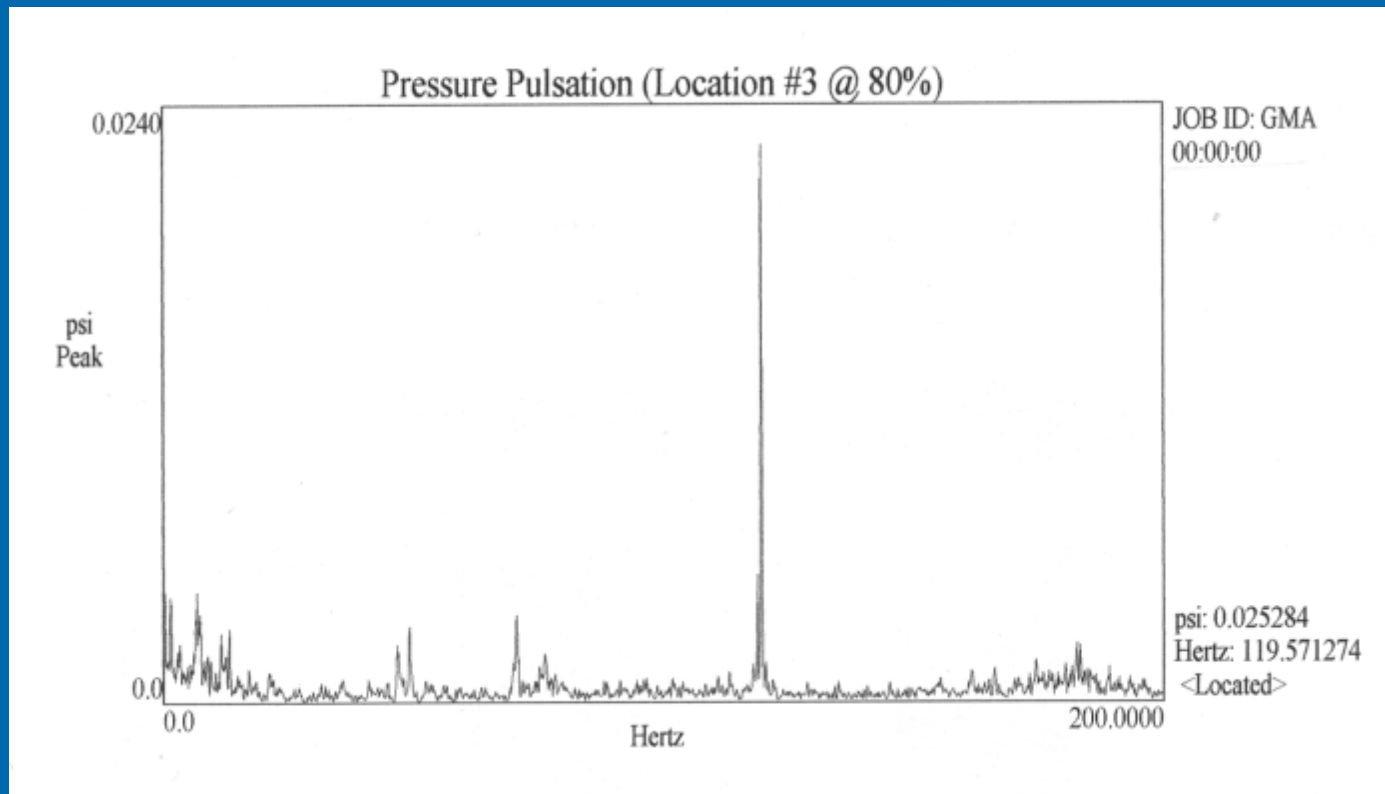


Ductwork had 90 degree bend, made up of two 45 degree corners.
Structural Steel vibrated excessively.
Very Noisy under Duct.



Sources of Dynamic Pressure Pulsations

- Blade Pass Pulsation Pressure (Harmonic; Normal for all Fans)
- (Freq = Number of Blades x Rotational Speed)
- Turbulence (Non-Harmonic)
- Rotating Stall (Non-Harmonic)
- Surge (Non-Harmonic)
- Inlet Box Vortex Shedding (Non-Harmonic)
- Outlet Box Vortex Shedding (Non-Harmonic)
- IVC Inlet Damper Vortex Shedding (Non-Harmonic)
- Outlet Damper Vortex Shedding (Non-Harmonic)

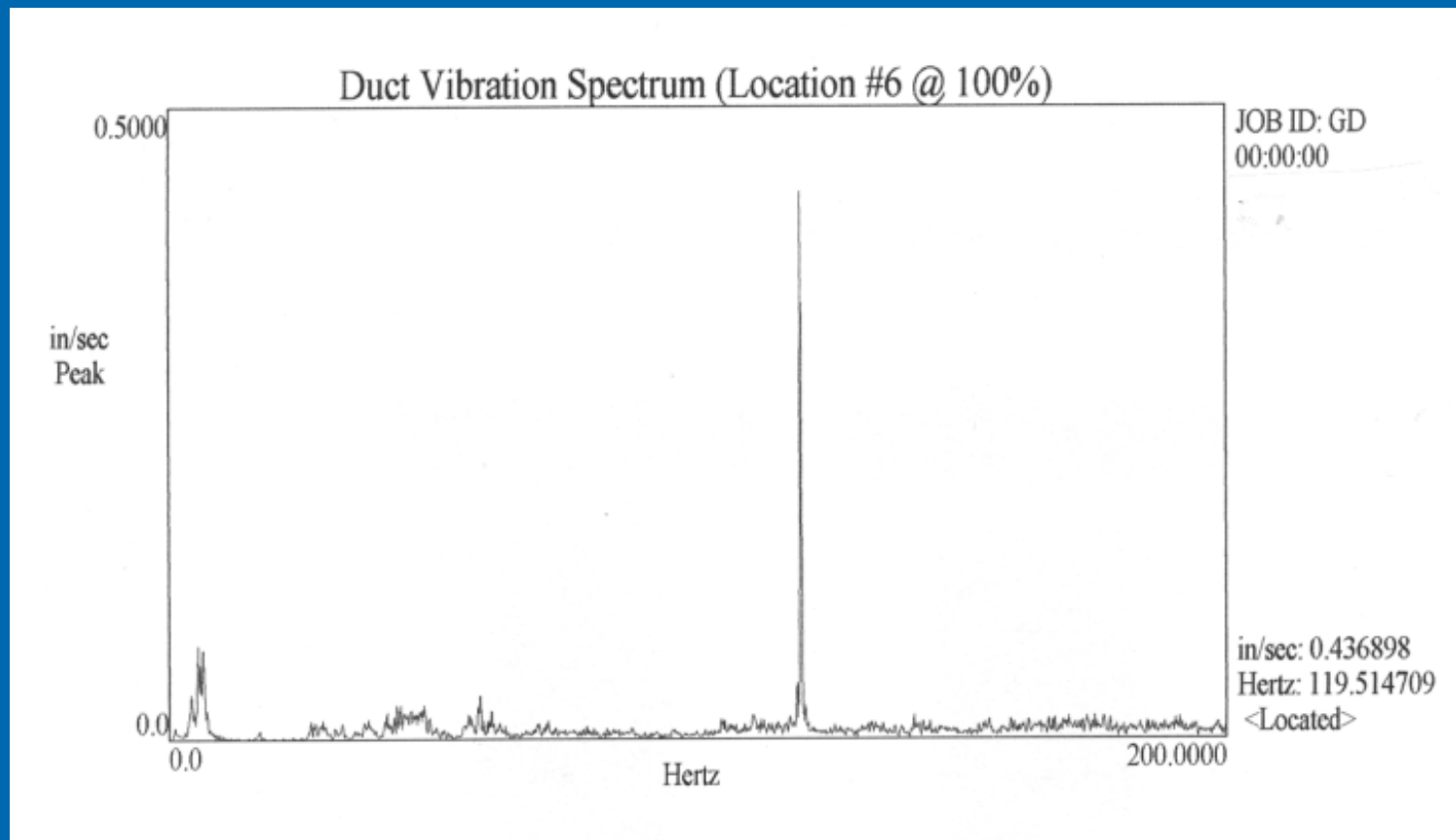


Frequency Spectrum of Pressure Pulsations
Obtained using Dynamic Pressure Sensor in Duct
Spectrum dominated by Pulsations @ 119.6 Hz.

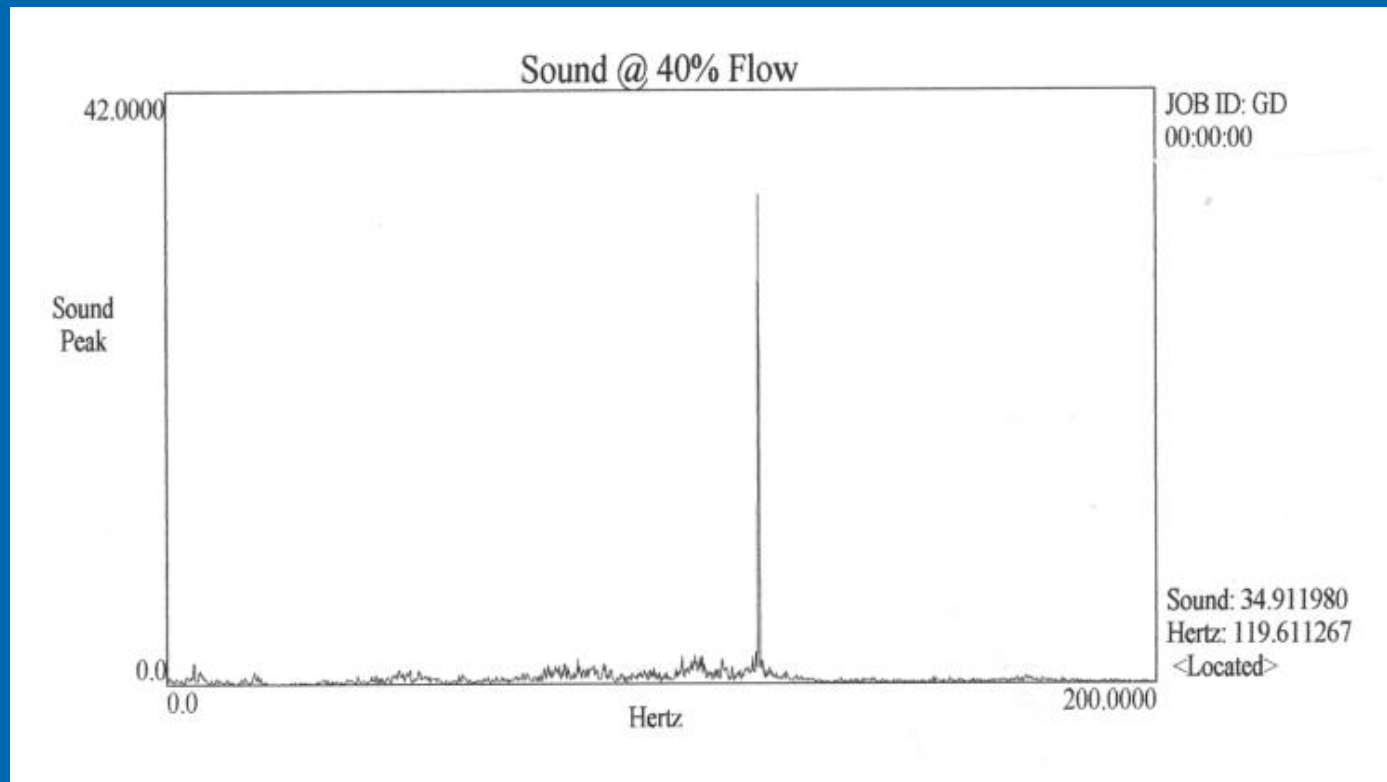
Fan Speed = 897 rpm = 14.95 Hz

BPPF = 8 blades x 14.97 = 119.6 Hz

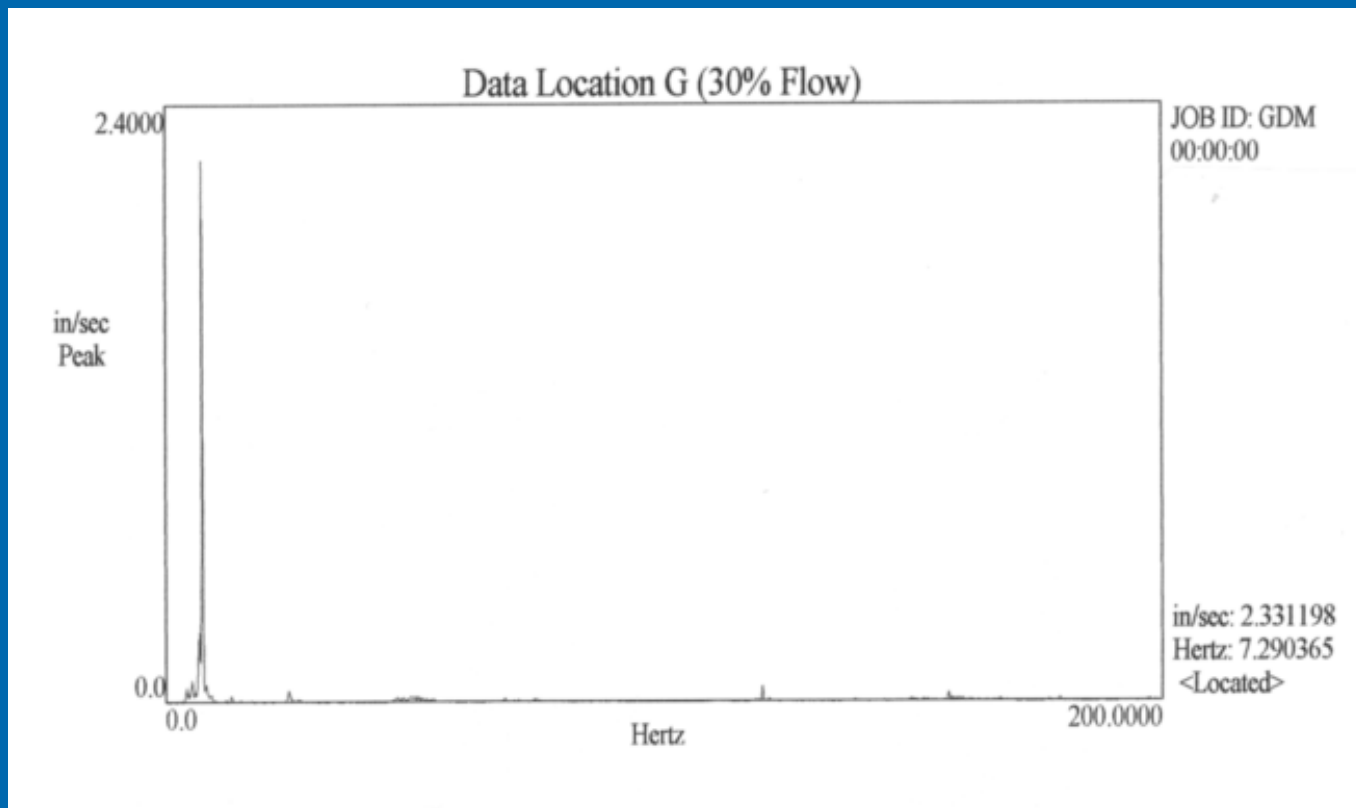
There wasn't any indication of vortex shedding or stall.



- Frequency Spectrum of Duct Vibration.
- Spectrum dominated by Pulsations @ 119.6 Hz.
- Duct vibration directly related to BPPF pulsations.
- Outlet Damper has no effect.

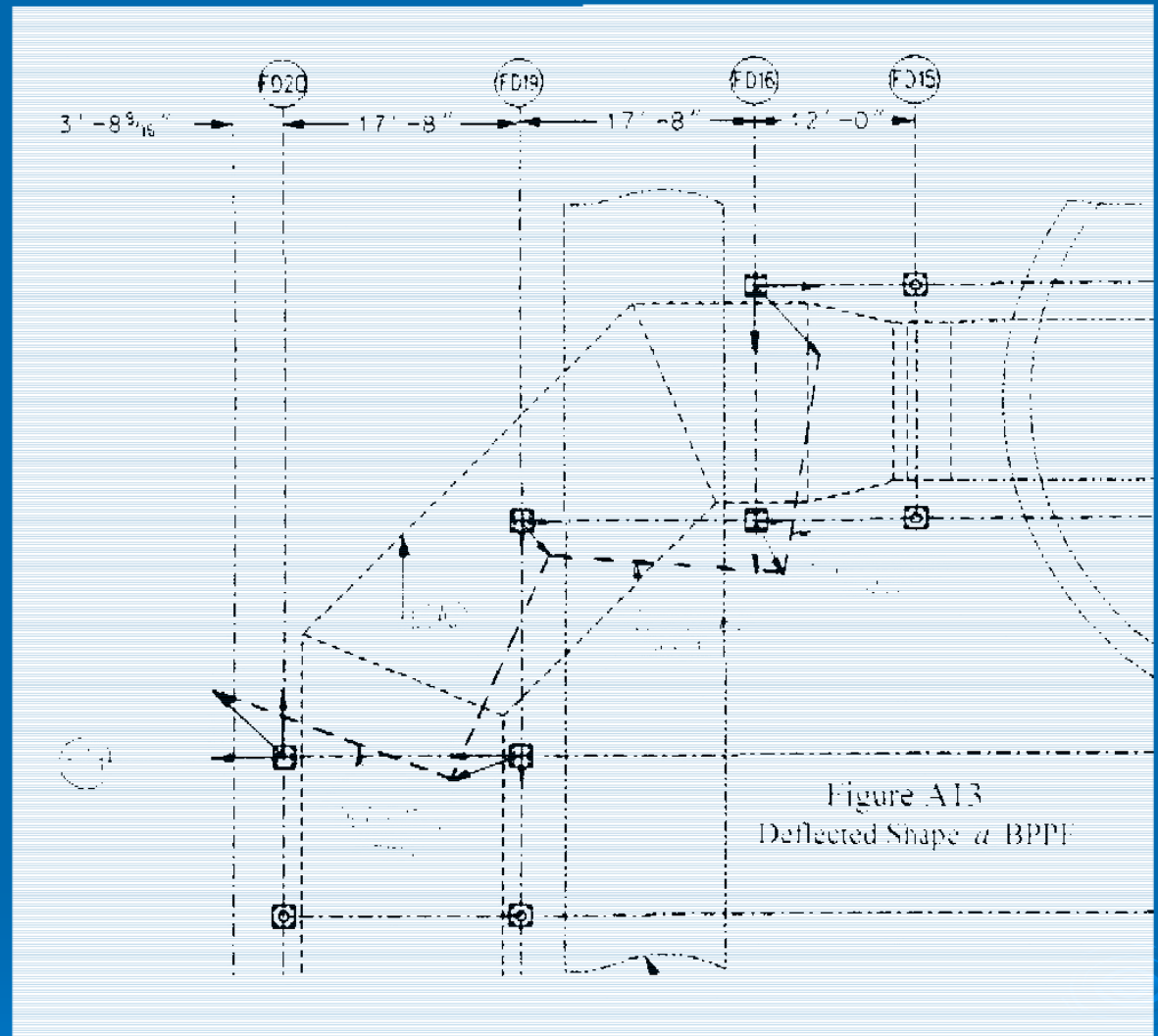


- Frequency Spectrum of Noise acquired with Data Microphone.
- Sound Pressure related to Duct Vibration which is caused by BPPF pulsations. Moving outlet damper will not effect duct vibration and noise. However, it was structural vibration, not duct vibration, that was a concern.

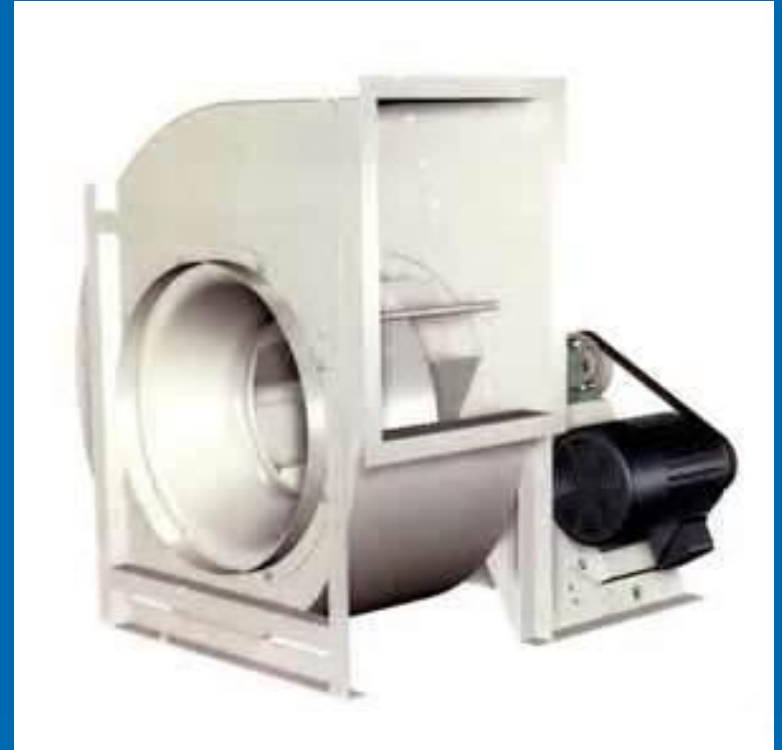


- Frequency spectrum of structural vibration dominated by sub-harmonic response @ 7.3 Hz. This frequency did not show up in pulsation data, and thus, it was concluded that it was not associated with pressure pulsations.
- Structure did not respond to BPPF and, thus, structural vibration and noise issues were not directly related.

- Structural Vibration due to broad band (non-harmonic) force at duct elbow acting on a un-symmetric structure.

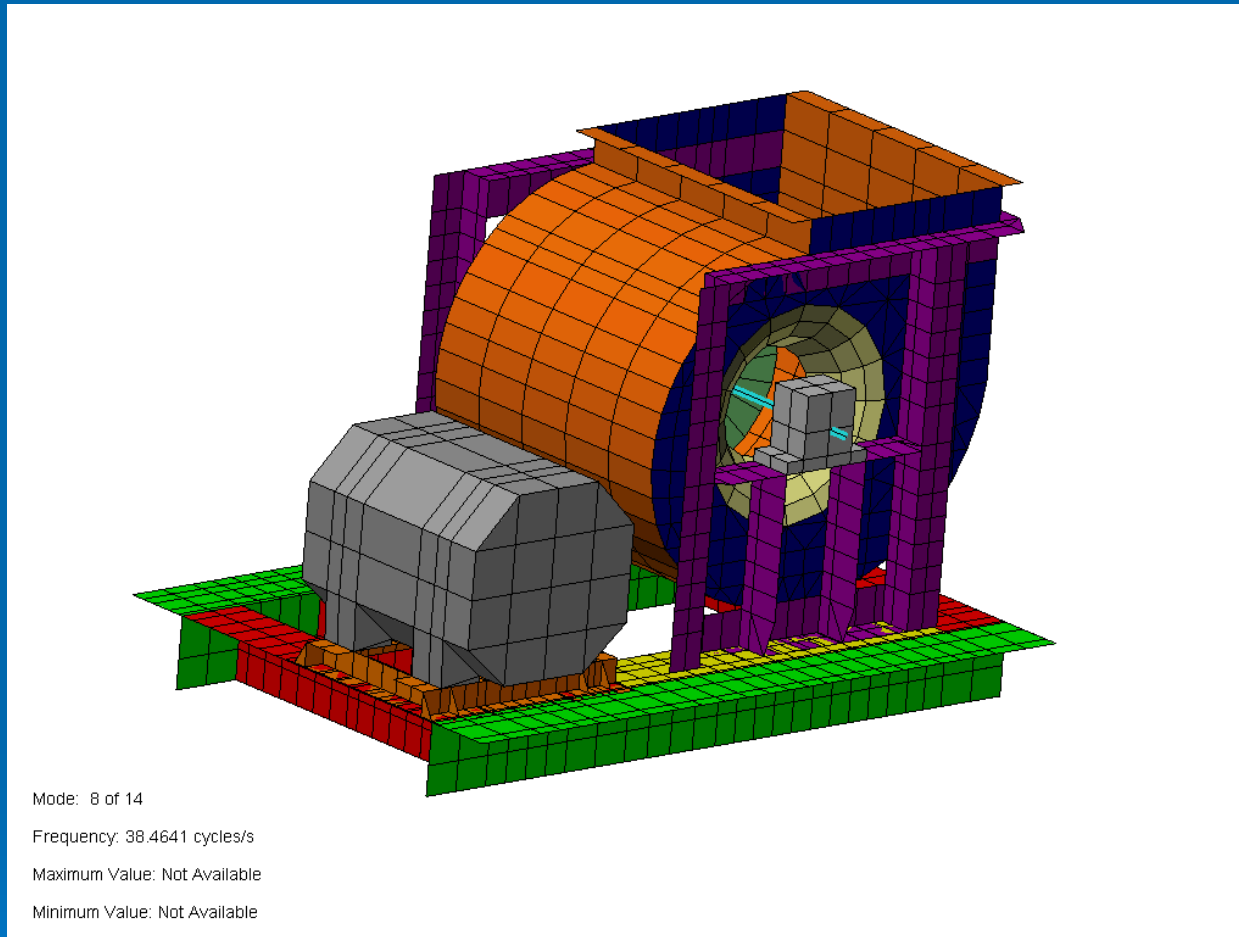


Vibration/Modal Specification



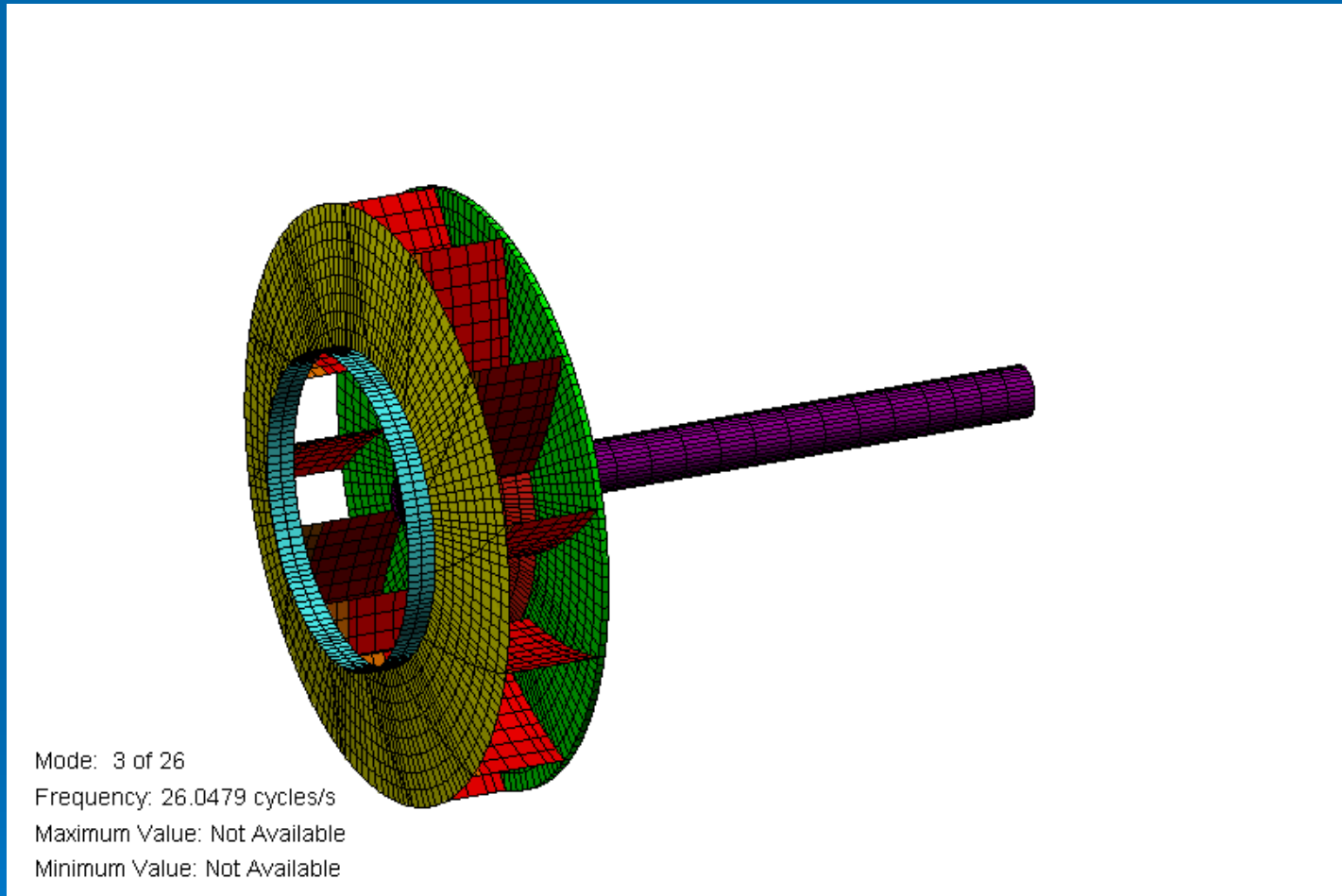
- WWTP Specifications for Fans are more frequently requiring analysis and test to minimize resonance (natural frequency excitation) of fan and/or foundation.

Resonance of Fan Base



- Natural frequency excitation can cause fan base to vibrate such that the motor is out-of-phase with the fan. Tough on belts and bearings.

Resonance of Fan Shaft



- Natural frequency excitation of fan shaft results in excess stress in shaft, bearings and belts.

FRP Fan Test

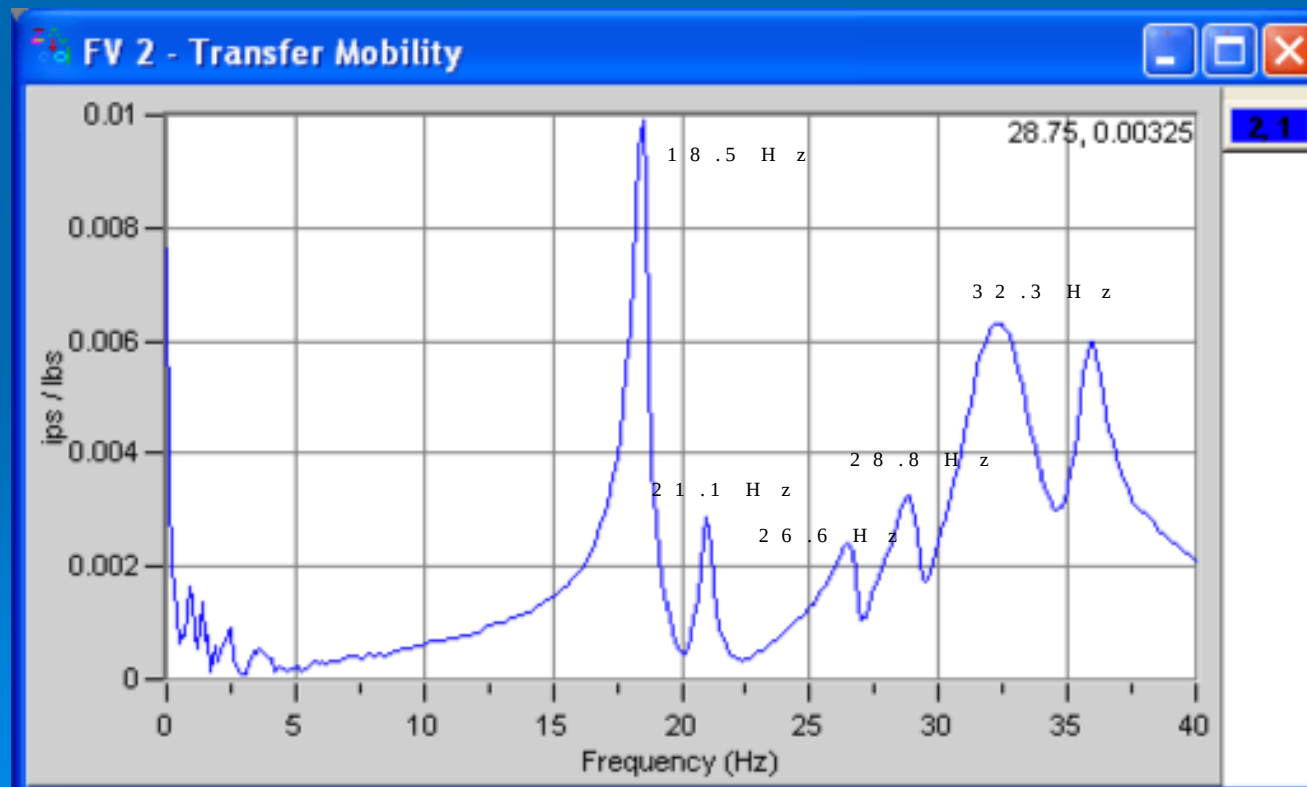
Fan Wheel diameter = 54"

Fan Speed = 962 rpm (16.0 Hz)

Motor Speed = 1785 rpm (29.8 Hz)

Natural Frequency @ 18.5 Hz (16% above Fan Speed)

Natural Frequency @ 28.8 Hz too close to Motor Speed



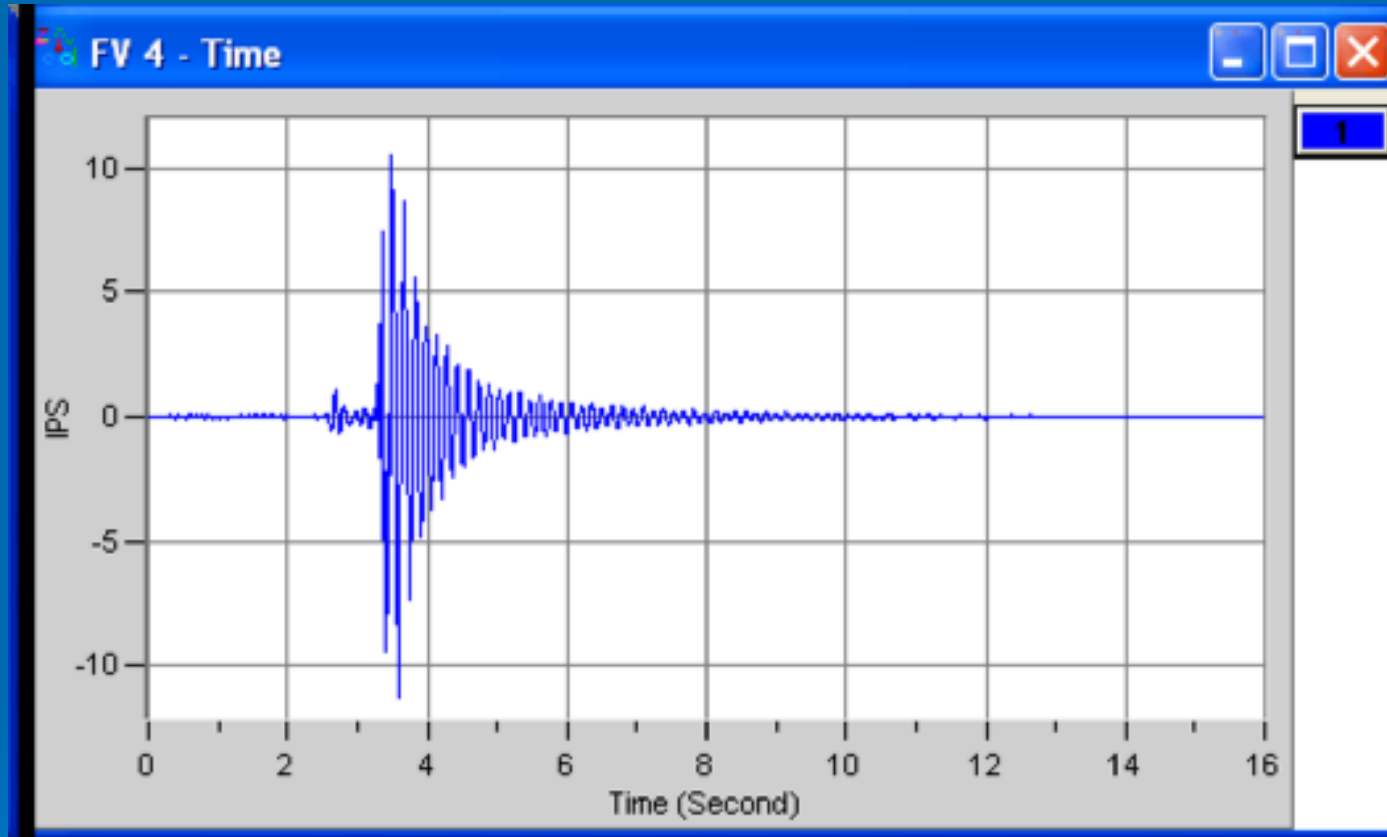
Finite Element Analysis (FEA)

- FEA is a numerical technique to approximate the structural dynamic characteristics and vibration response of a machine, structure and/or foundation.
- FEA can be performed in design stage prior to manufacture to insure that vibration problems will not occur due to design deficiencies increasing equipment reliability.
- FEA can be used to develop and evaluate structural or mechanical modifications prior to implementation increasing the probability of success.

Example of FEA used to modify Pump Support

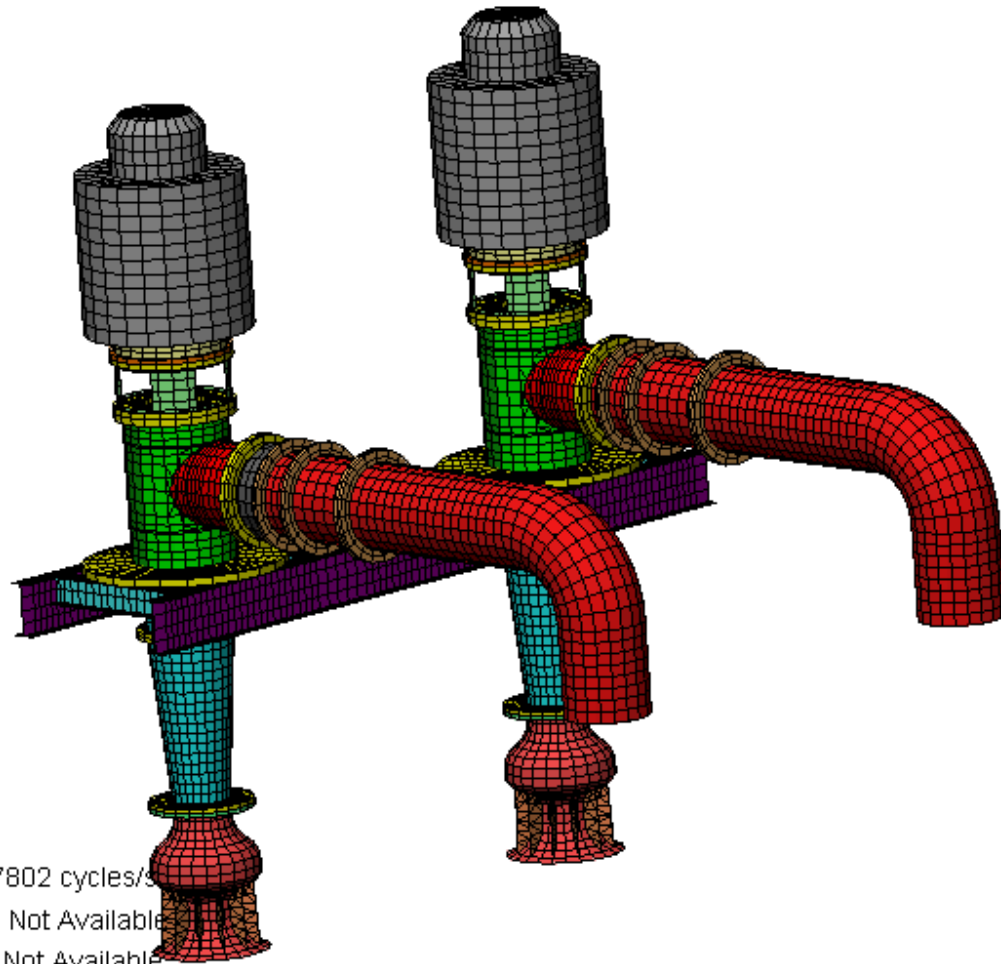


Vertical Pump – Vibration @ Shutdown

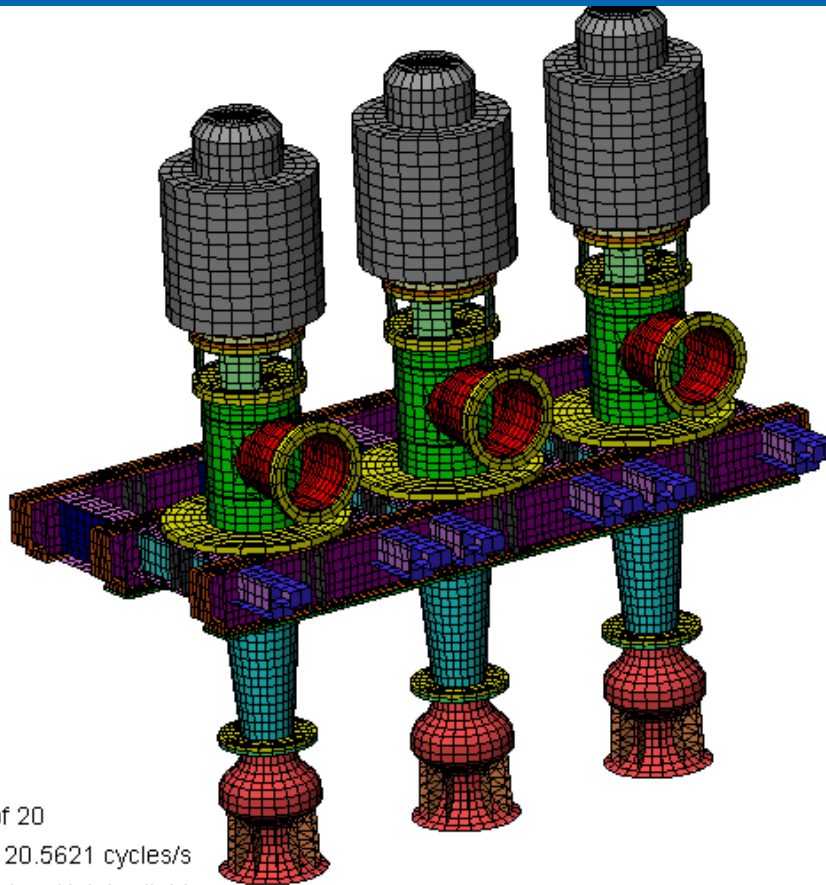


- Impulse occurs as check valve closes (slams shut)
- Impulse excites system natural frequencies (ring-down response)

FEA of Pump, Piping & Original Support



Flexible Spool Piece (Isolator) & Structural Modification

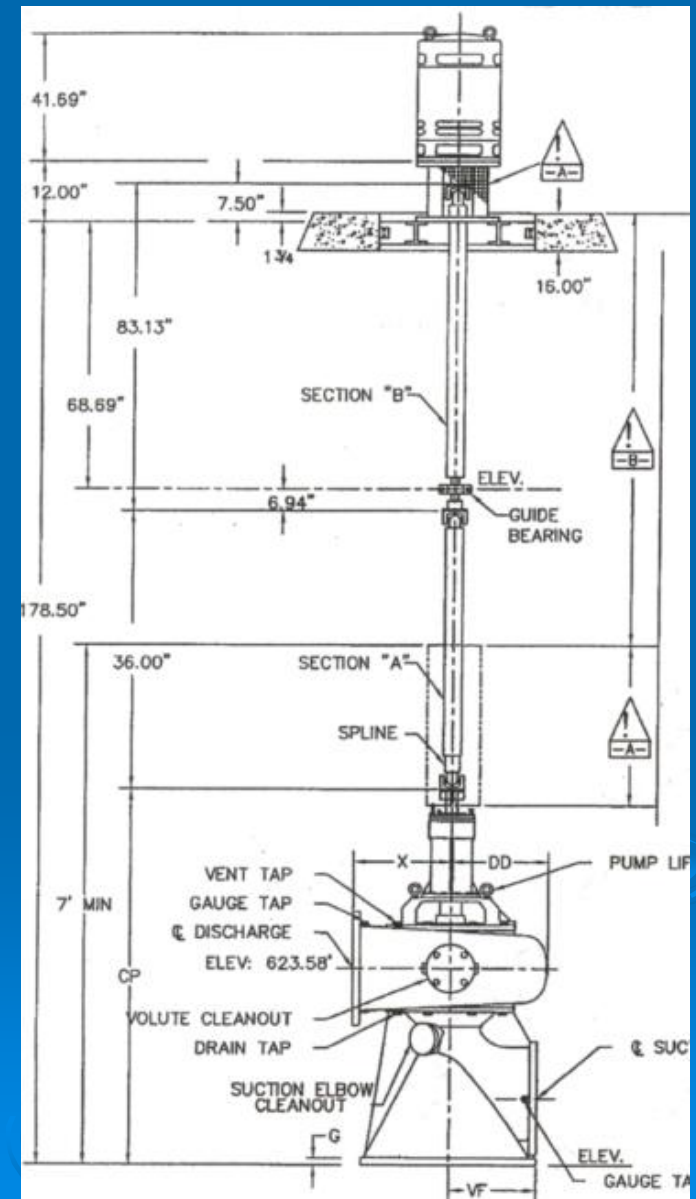


Mode: 12 of 20
Frequency: 20.5621 cycles/s
Maximum Value: Not Available
Minimum Value: Not Available

- Uncoupled Pumps from Piping.
- Structural Mod to Supports.
- Evaluate with Future Center Pump

Case Study - WWTP Sludge Pumps

- Vertically Mounted Centrifugal Sludge Pumps @ WWTP
- Facility has 4 Pumps
- Primary Pump = 40 hp/900 rpm
- Peaking Pumps = 100 hp/1200 rpm
- Motor located on elevated floor.
- Pump located in Dry Well.
- Two Drive Shafts (Upper & Lower) between Motor & Pump.
- Problem: Multiple Premature Guide Bearing Failures & Excessive Vibration of 100 hp motors at full speed.

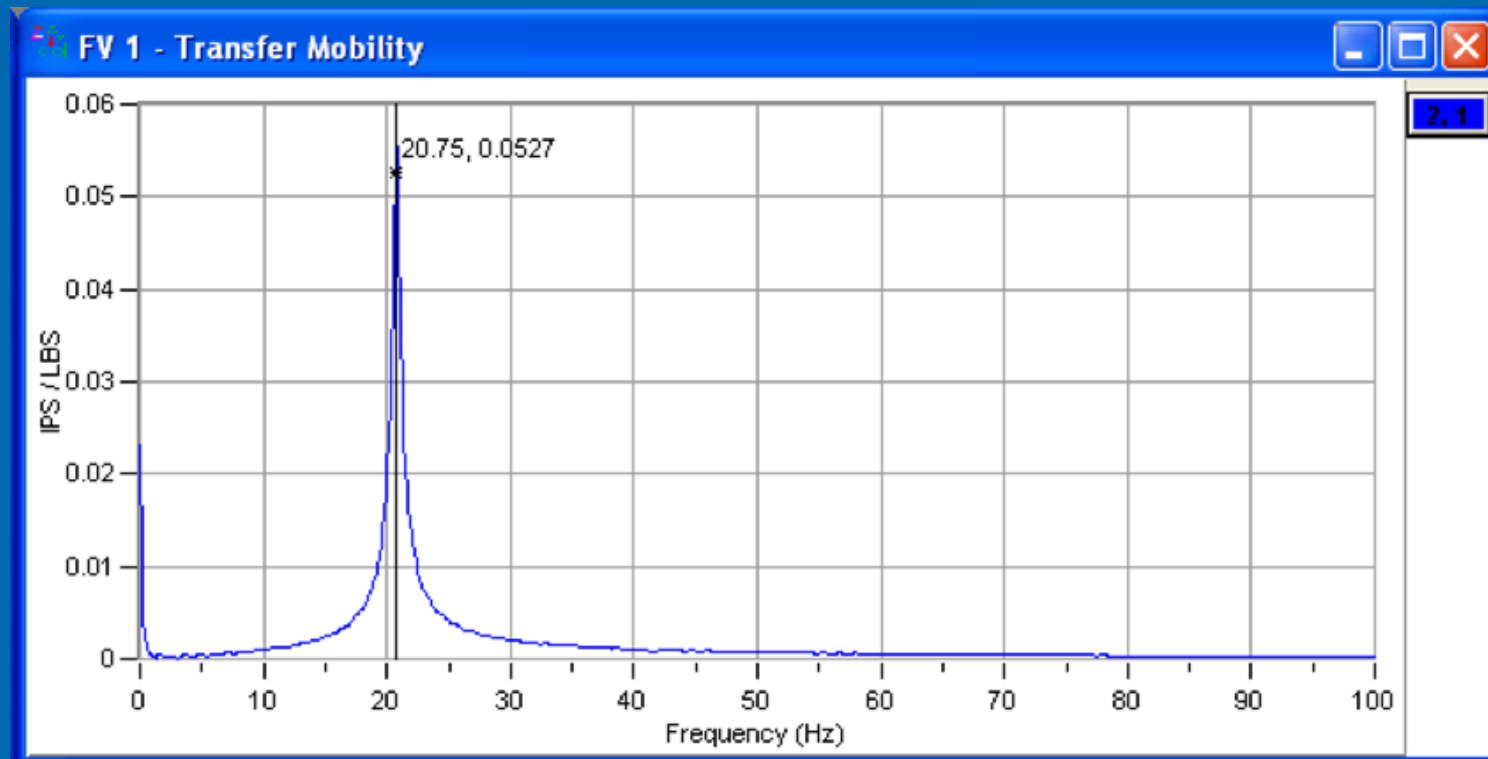


Case Study – WWTP Pump



- (1) 40HP Motor and (3) 100 HP Motors
- And VFD Controls on Upper Floor.
- Motors on Pedestals similar to Vertical Turbine Pumps except there isn't any discharge pipe.

Natural Frequency of 100 HP Motor



- Natural Frequency = 20.8 Hz.
- Motor and floor vibration excessive at max speed
- Pump Speed = 1200 rpm = 20 Hz @ VFD Setting = 100% = 60 Hz

Case Study - Motor Vibration Solution

- Placed restriction on Operating Range.
- Allowable VFD setting = 46 Hz - 56 Hz or 76% - 93% of Full Range.
- This eliminated the excitation of Motor Reed Frequency mode @ VFD = 100%.
- The minimum VFD setting of 46 Hz (76%) was placed on the system to eliminate the onset of cavitation in the pump.
- The motor resonance could also have been solved by increasing the stiffness of the support pedestal, thus, increasing the natural frequency. However, the pump capacity above VFD = 93% was not needed.

Case Study – Pump Bearing Problems

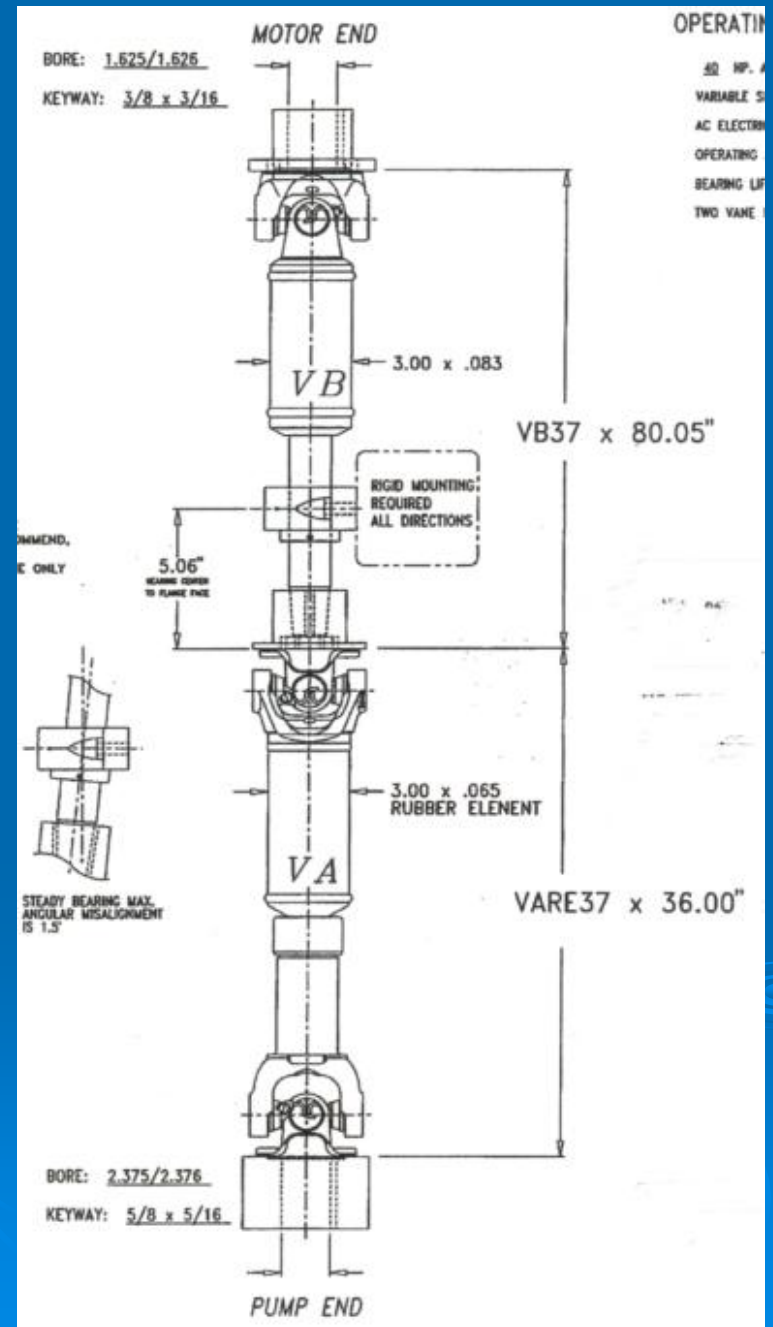


- Views of Pump Arrangement in Dry Well

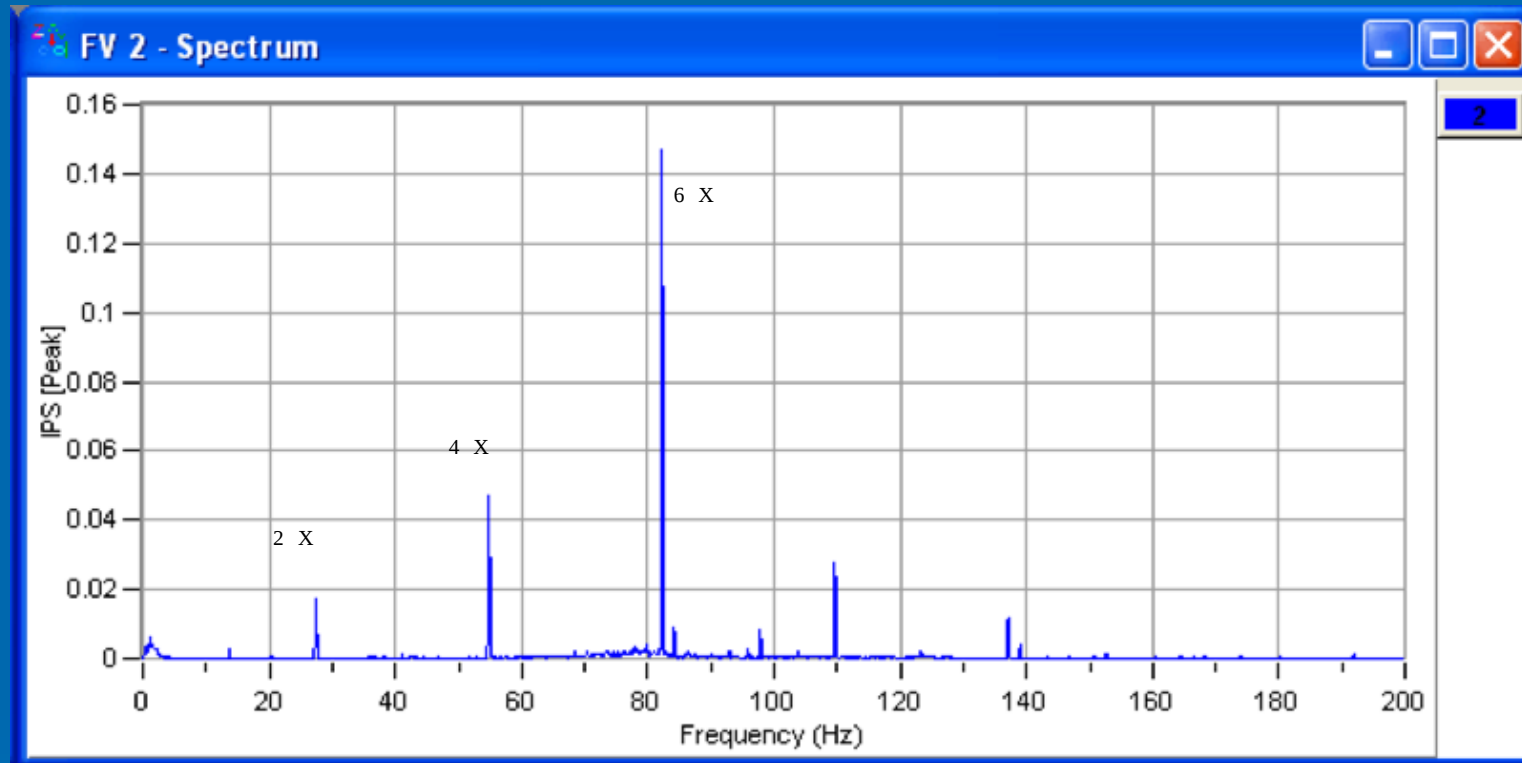


Case Study – Pump Bearing Problems

- The Upper Shaft is coupled to the Motor with U-Joint.
- The Upper Shaft is coupled to the Lower Shaft with U-Joint.
- The Lower Shaft is coupled to the Pump with U-Joint.
- Intermediate guide bearing located at the Upper Shaft. Bearing specified to be rigidly mounted.

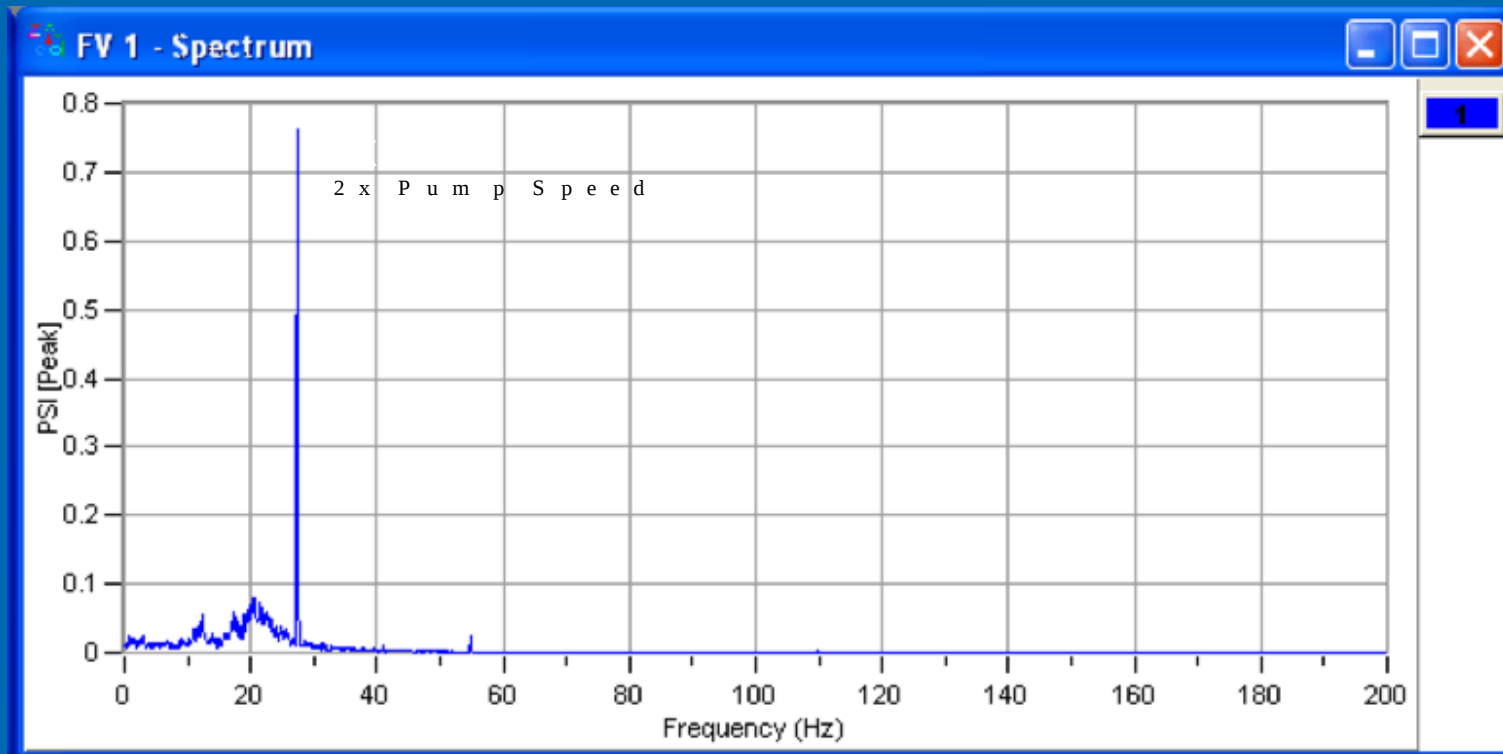


Pump Bearing Vibration Data



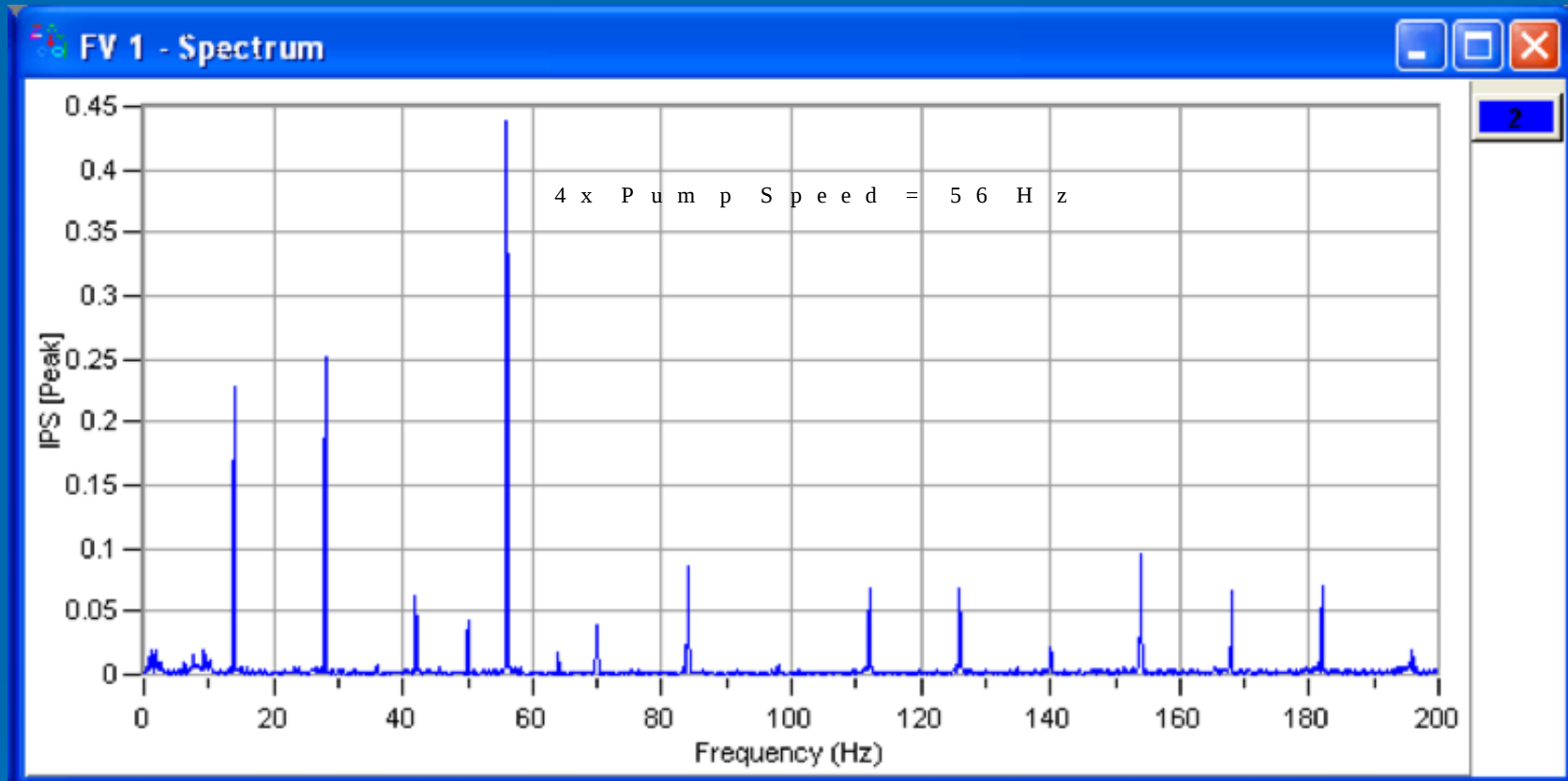
- Pump bearing vibration not excessive @ VFD = 56 Hz. Vibration @ 1x almost non-detectable. Vibration @ 2x (vane pass) very low (0.02 ips). Vibration @ 4x and 6x higher than 1x and 2x.

Dynamic Pressure Pulsation



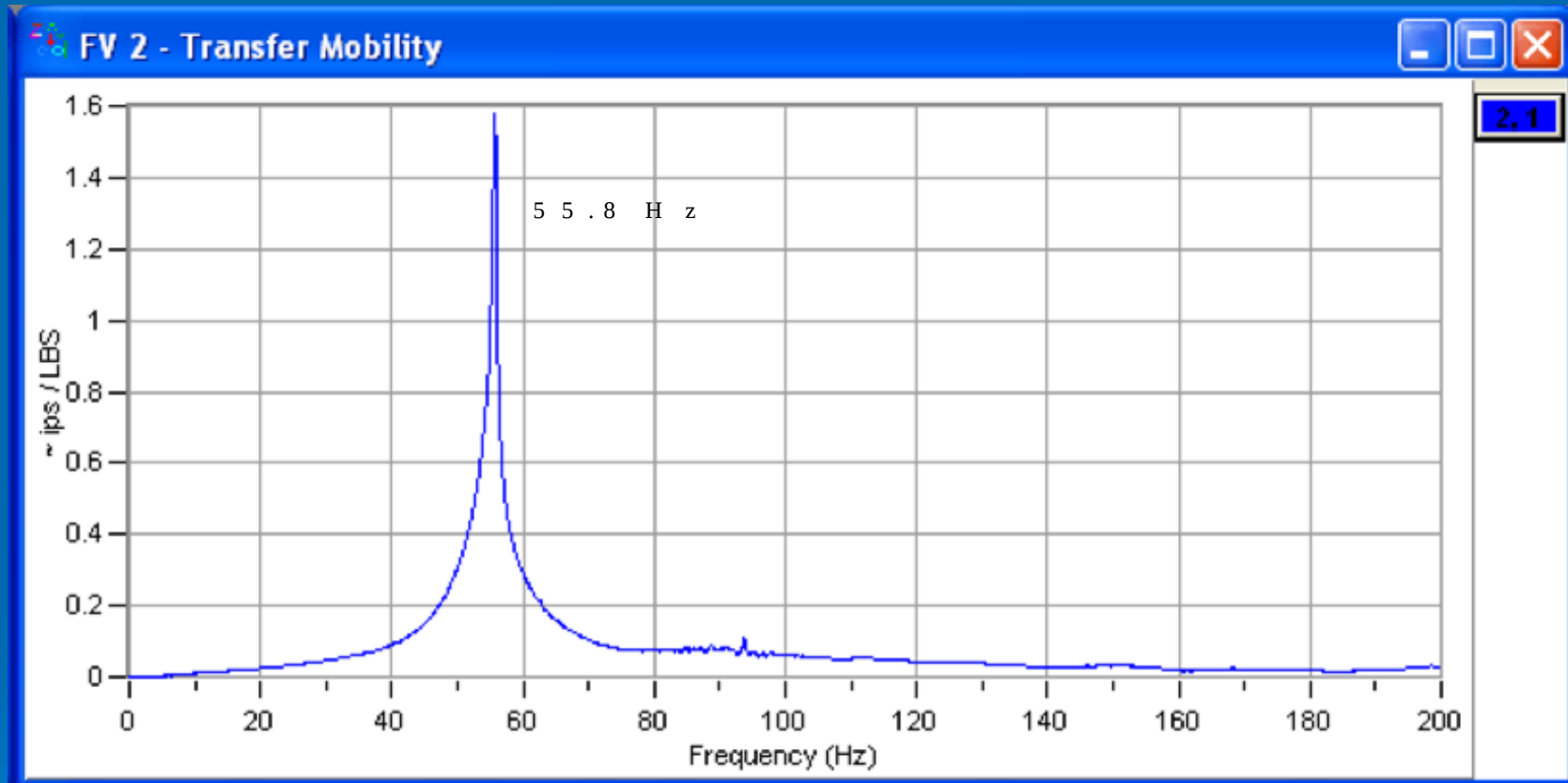
- Pressure Pulsation data acquired at Pump discharge that indicated 2x pulsations due to vane pass. All other harmonics were insignificant. This led to conclusion that harmonics in the vibration data were transmitted mechanically instead of acoustically.

Shaft Vibration



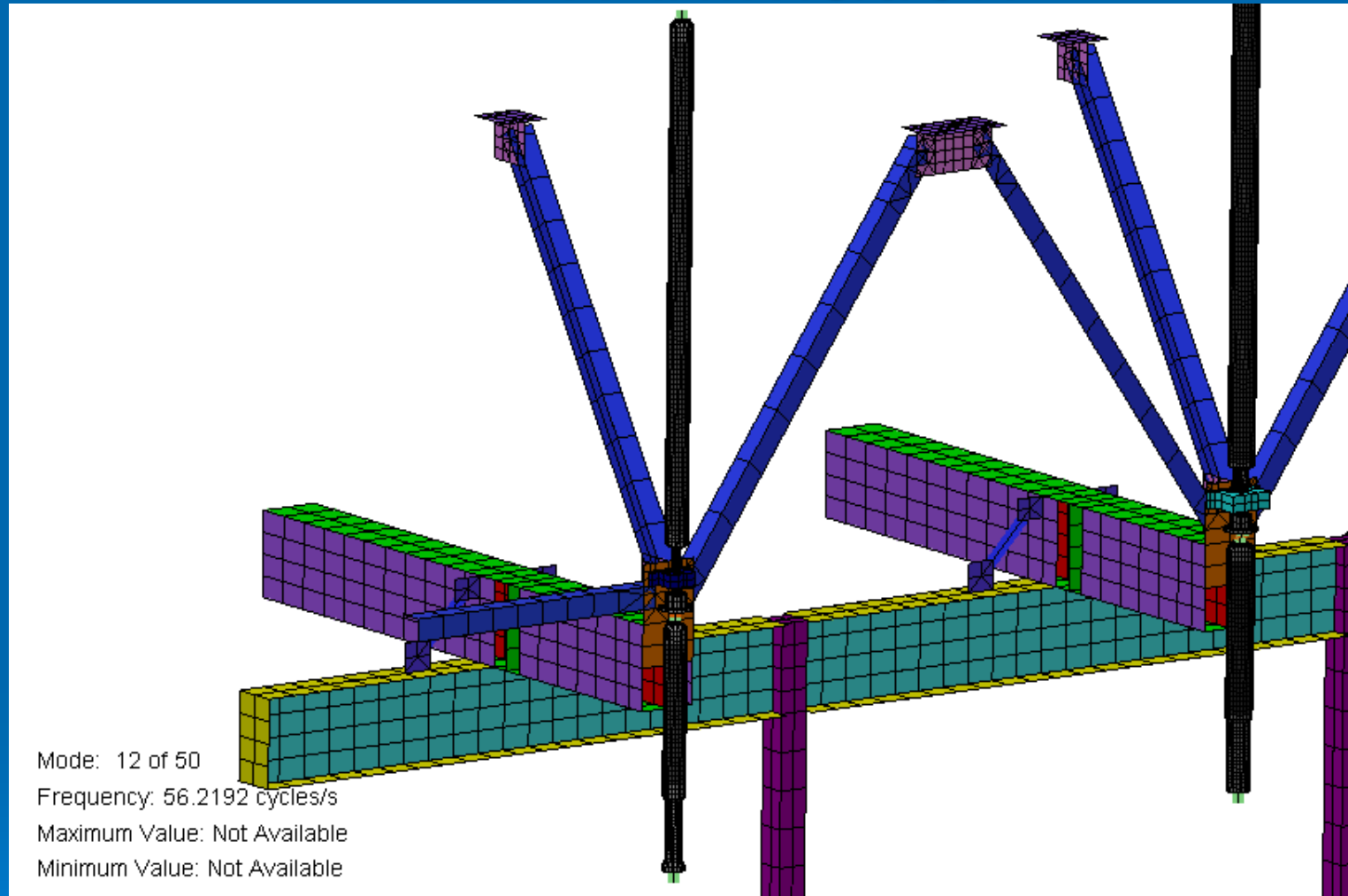
- Shaft vibration, obtained with a shaft stick, indicated that shaft motion dominated at 4x when VFD ~ 56 Hz or 93%.
- Elevated response of Shaft due to excitation of it's natural frequency.

Drive Shaft Natural Frequency Test



- Natural Frequency of Drive Shaft = 55.8 Hz

FEA Animation



- FEA prediction = 56.3 Hz (Impact Test = 55.8 Hz)
- Excitable @ VFD = 93% by 4x Force

Solution to Bearing Problem

- Exclusion Zones:
- Desired Pump Speed Range = 11.0 Hz - 14.8 Hz
- VFD control range = 45 - 60 Hz
- Approx (1X) Exclusion Zone = 11.0 - 14.8 Hz
- Harmonic (2X) Exclusion Zone = 22.0 - 29.6 Hz
- Harmonic (4X) Exclusion Zone = 44.0 - 59.2 Hz
- Harmonic (6X) Exclusion Zone = 66.0 - 88.8 Hz
- Changed Diameter of Upper Drive Shaft.
- Natural Frequency = 64 Hz
- Good for Operation of VFD 45 - 60 Hz; Pump Speed = 11.2 - 14.7 Hz.