Vibration Institute Piedmont Chapter 2018 Training Event

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Vibration Institute

Vibration Institute

Founded in 1972

Currently:

Bob Sayer, PE (President)

Bill Pryor (Vice President)

Michael Long (Executive Director)

Dr. Ron Eshleman (Technical Director of Training)

Dave Corelli (Technical Director of Certification)

VI Training & ISO Certification

Introduction to Machinery Vibration (IMC – CAT 1)

Basic Machinery Vibration (BMV –CAT 2)

Machinery Vibration Analysis (MVA – CAT 3)

Basic Machinery Balancing (CAT 3 & CAT 4)

Practical Rotor Dynamics & Modeling (RDM –CAT 4)

Advanced Vibration Analysis (AVA – CAT 4)

Advanced Vibration Control (AVC – CAT 4)

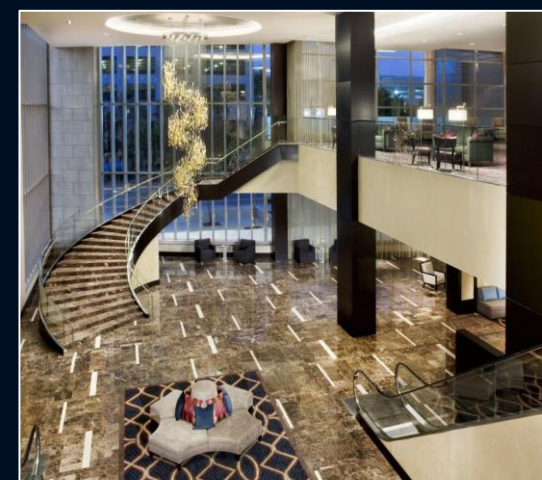
Modal Analysis 2-Part Series: (NEW!!!)

Practical Modal Analysis with ME' Scope

Vibration Diagnostics using Modal & ODS

VI ATC 2018
VIBRATION INSTITUTE ANNUAL TRAINING CONFERENCE

42nd Annual Training **Conference**



NEW ORLEANS, LA • JULY 17-20
HYATT REGENCY – NEW ORLEANS

42nd Annual Training Conference New Orleans (July

Tuesday: Pre-Conference Training

Rotor Dynamics in Rolling Element & Journal Bearings

Pump Performance, Reliability & Repair

Wednesday – Friday Conference

Over 50 Presentations & Over 50 Vendors

Co-Located with Reliability-Web IIoT Conference

42nd Annual Training Conference
New Orleans (July

<u>Wednesday – Friday Conference</u>

Keynote: Monster Pumps of New Orleans

2 Balancing Workshops

Motion Amplification w/Demonstration

Wireless Condition Monitoring

Pump Vibration

HI Vibration Spec Review

Torsional Vibration

MEMS Sensors

Case Studies

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42nd Annual Training Conference
New Orleans (July

<u>Wednesday – Friday Conference</u>

<u>Complimentary Technologies</u>

New Shaft Alignment Standard

Electric Current & Signature Analysis

Development of Multi-Technology Monitoring Program

Design & Implementation of Oil Analysis Program

<u>Friday Afternoon: Post-Conference Training</u>

Road Map to Effective Vibration Diagnostics

Review of Vibration Diagnostic Techniques

Robert J. Sayer, PE

President, The Vibration Institute

Vibration Analysts Toolbag

Hardware:

- FFT Analyzer (Smaller & More Powerful)
- ICP Sensors (All Types)
- Modal Hammers
- Motion Amplification Video

Software:

- Modal/ODS Programs
- FEA Programs

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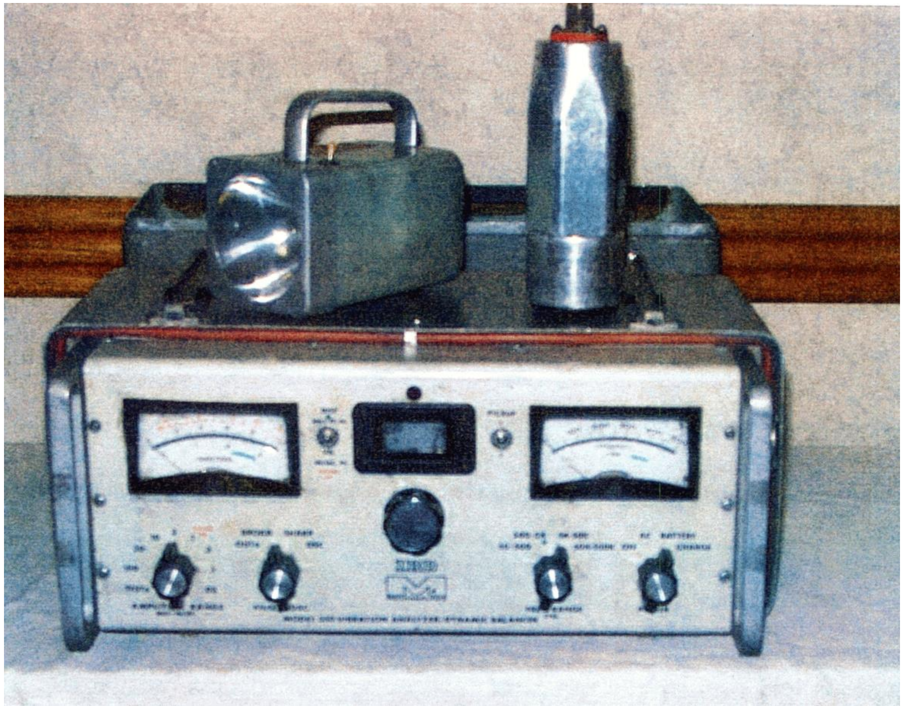
Pre-FFT Analyzers



IRD 350 Vibration Analyzer Shown in Photo. Pre FFT – Analog Tuneable Filter Analyzer.

Art Crawford, together with Ted Ongaro and Walter Leukhart, founded International Research and Development in 1952, which later became IRD Mechanalysis.

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Real Time Analyzer

1965: Technical Paper by James Cooley (IBM) & John Tukey (Bell Labs & Princeton U.)

“An Algorithm for the Machine Calculation of Complex Fourier Series”

This Paper set forth the details of the Fast Fourier Transform (FFT) Algorithm that is the basis for today’s Analyzers.

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1980's FFT Analyzer

2-Channel Scientific Atlanta FFT

400 lines of Resolution - Rather Large and Heavy. Small Display Screen. Internal Memory.

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Introduction of Personal Computer

IBM PC released in 1981.

DOS Operating System- 64 kb RAM, 32 Mb Hard Disk

Windows Operating System released in 1985

1990’s PC-based FFT Analyzer

Lunchbox & Laptop PC’s

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Current PC-based FFT Analyzers

24 bit (25,600 line) pocket sized DFT Analyzers. Easy export to ME'scope & Star.

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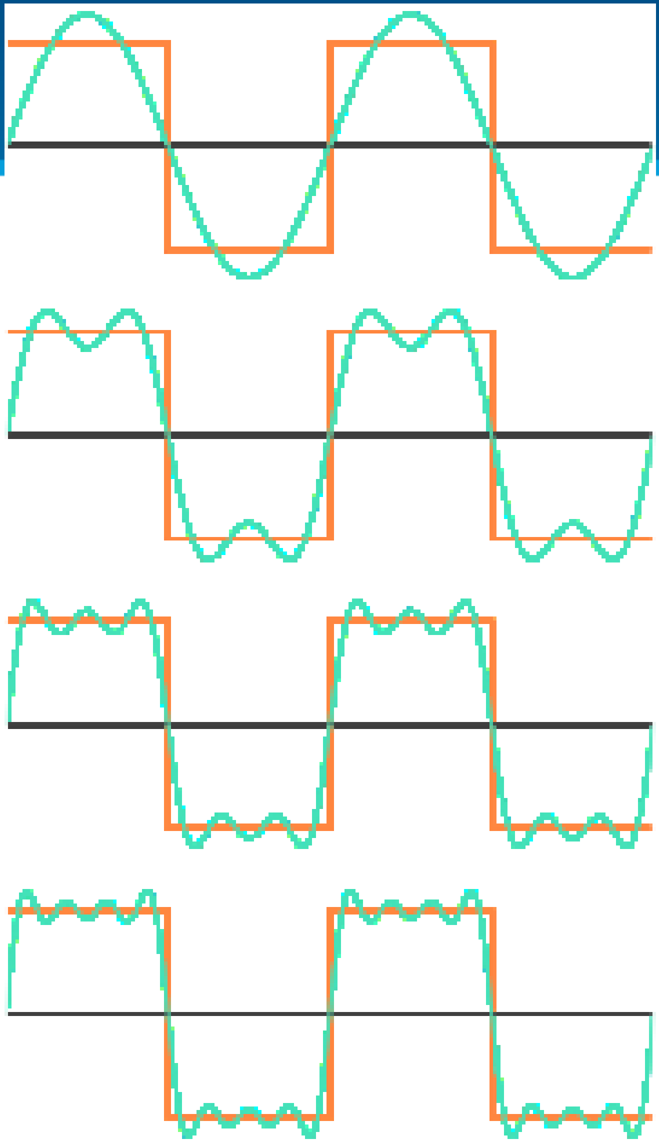
History of FFT Algorithm

Fourier Series & Fourier Transforms are named after Jean Baptise Joseph Fourier, a French mathematician, (March 1768 - May 1830), who initiated the investigation of the Series and their application to heat transfer and vibrations.

A Fourier series decomposes any periodic function into a sum of simple sine and cosine functions.

The DFT is a digital solution to the Fourier transform (FFT) made possible by the advent of the micro- processor.

Fourier Series



Square Waveform (orange) approximated by sine waves (green) @ 1x, 2x, 3x and 4x of sawtooth frequency.

An FFT frequency spectrum of the above sawtooth waveform would then have responses at all of these harmonic multiples even though the signal repeats only 1x per revolution.

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Operating Deflection Shape (ODS)

- Testing & Analysis Procedure that provides an Animation of the response of a Mechanical System @ a Discrete Frequency (4.9 Hz, 24.9 Hz, 29.7 Hz, 59.4 Hz). The animation provides a display of information that might otherwise be difficult to relay to persons that are not conversant in vibration analysis.
- The Test is performed with the equipment operating. It provides a **linearly exaggerated** animation (at a slower speed) of the relative movement of all structural and/or mechanical components tested based upon Transfer Function and Phase.

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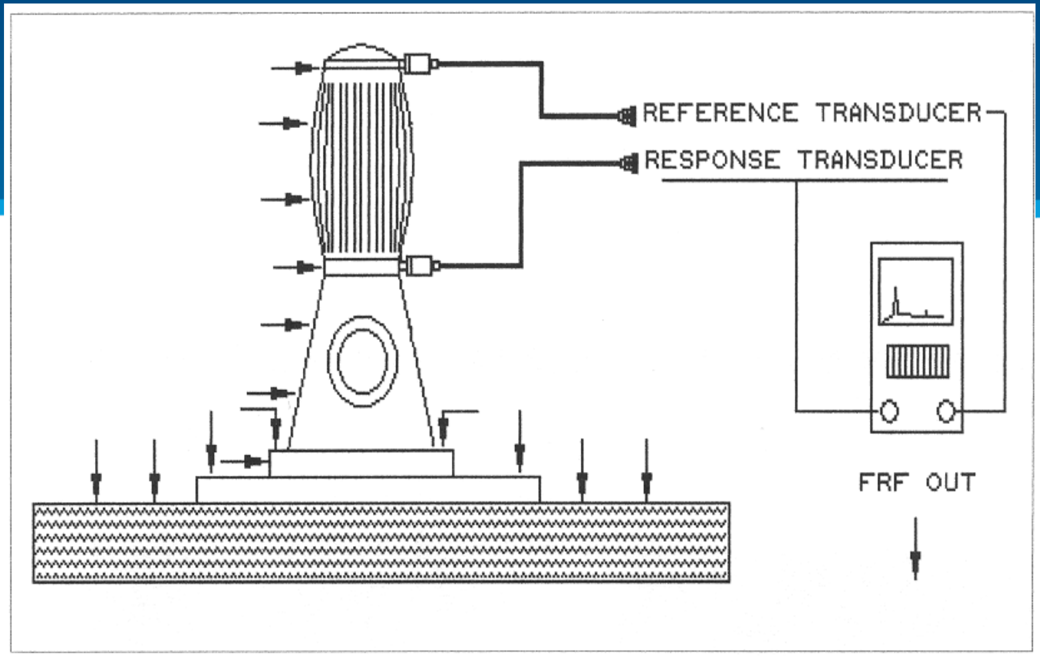
ODS Test Procedure

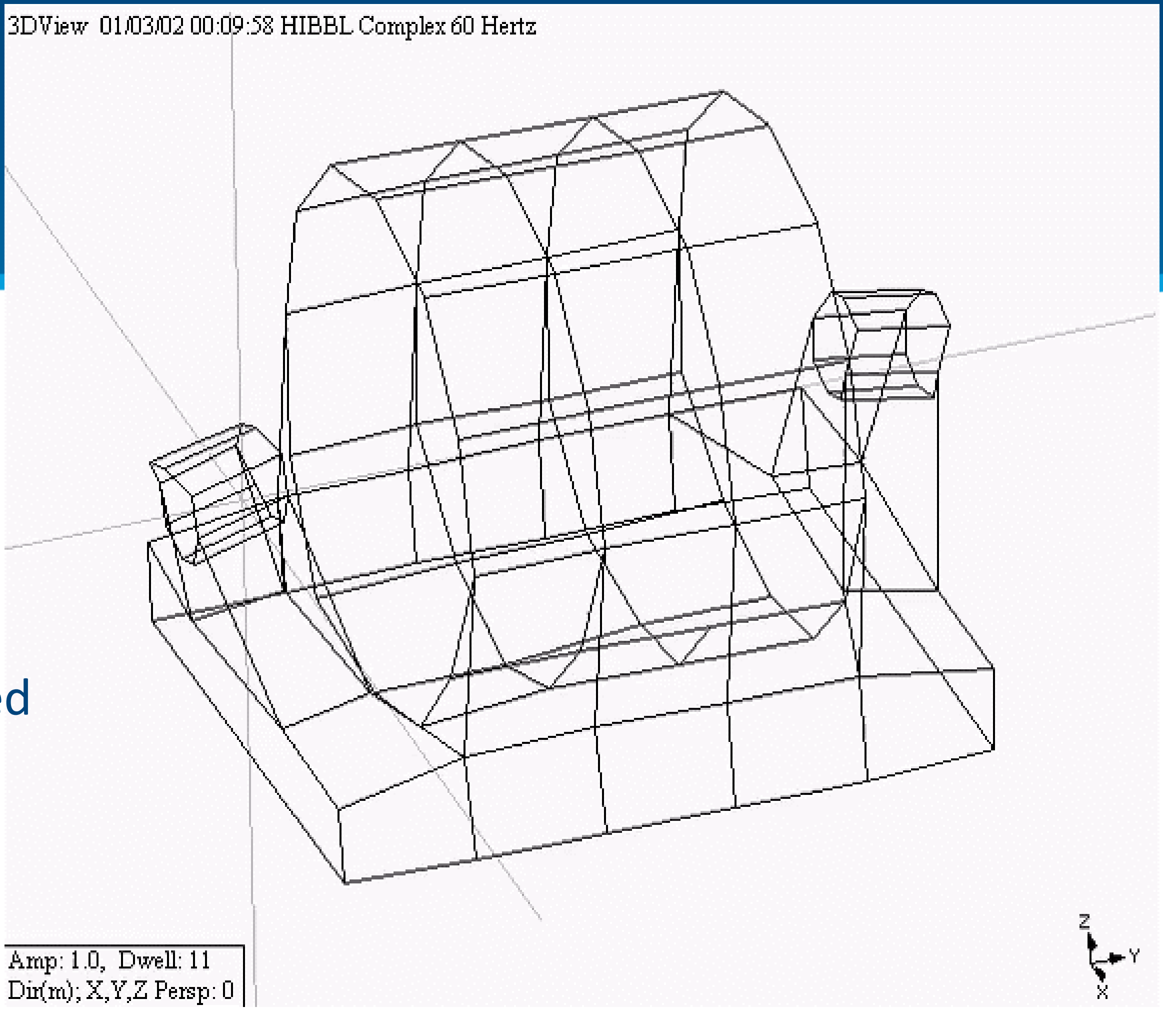


- A minimum of two transducers are used for data acquisition. One Transducer remains stationary during the entire test as a Reference. The other transducers are used as Response Transducers.
- The Animation is based upon the Relative Magnitude of the Response Transducer as normalized by the Reference Transducer. In most cases, not affected by variations in vibrations during the test. **(Unless APS curve-fit used)**

Operating Deflection Shape (ODS)

Animates only Data that is Measured or Points that are Extrapolated or Interpolated from Measured Data.

Otherwise Data Point is assumed not to have any Motion!



ODS Animation of Motor Frame and Support.

Suspicious Possible Lack of Data in this ODS!!!

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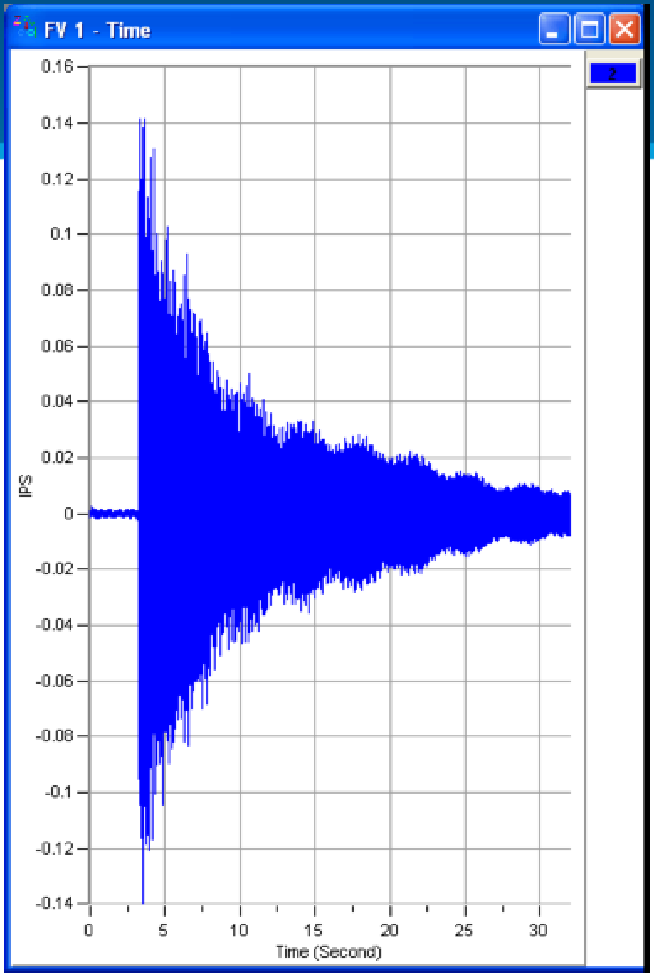
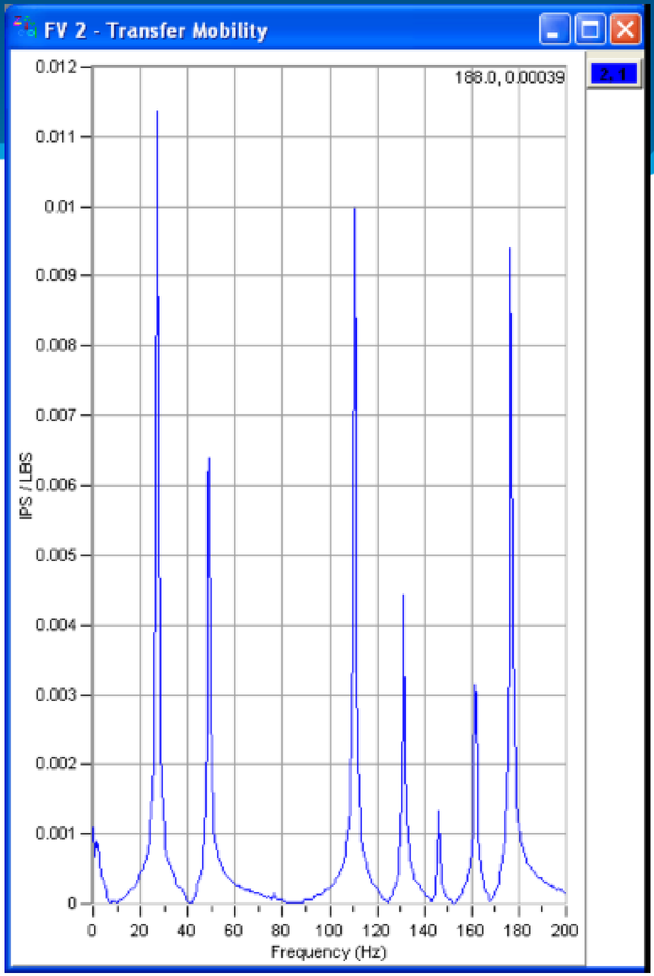
Natural Frequency Test





Ringdown Response & Transfer Function

Instrumented Force Hammer used to excite Natural Frequencies of structural-mechanical system.

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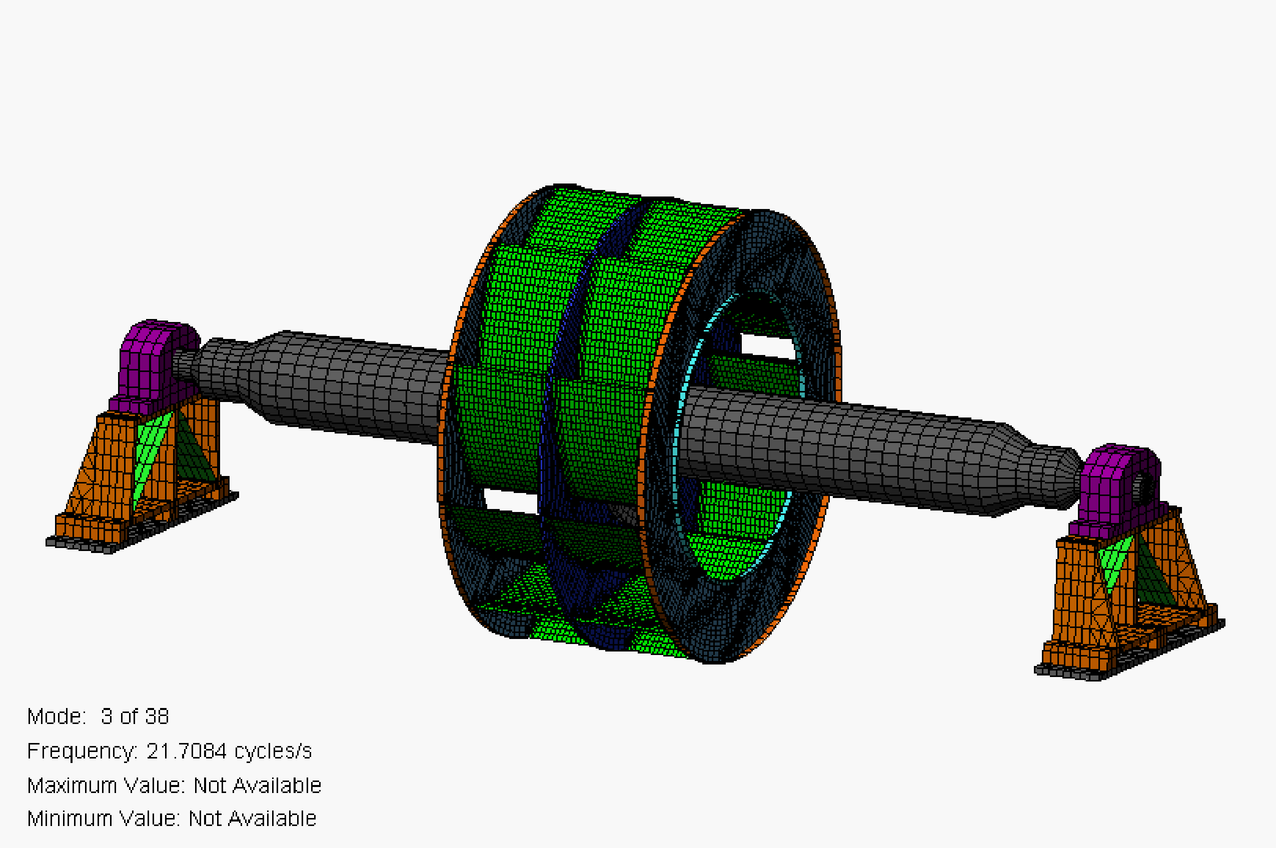
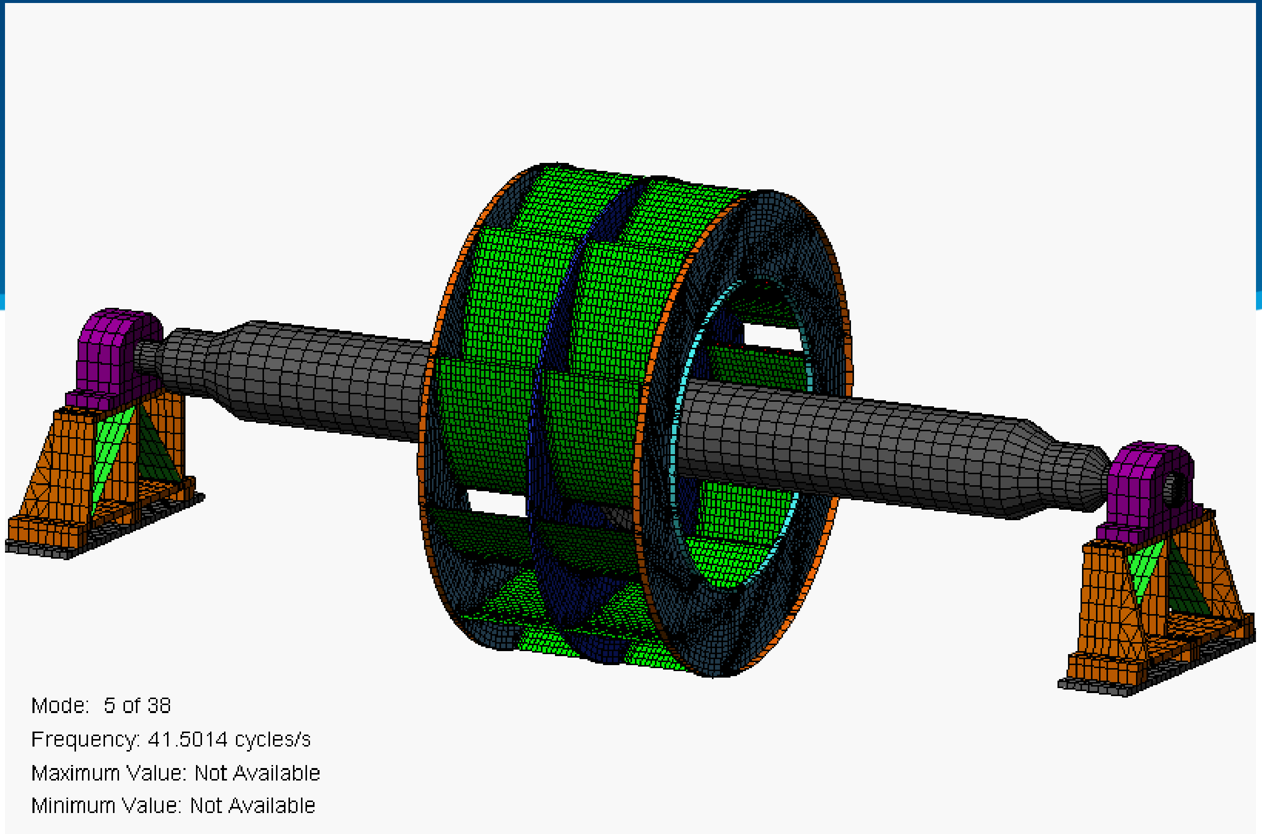
Finite Element Analysis



Finite Element Analysis (FEA) is a **numerical** technique that can be used to **approximate** the structural dynamic characteristics of vibrating mechanical systems. FEA models contain many more dof's than EMA models and are more descriptive. Better suited for SDM studies.

Finite Element Analysis



Previous Slide showed shaft critical mode of a centrifugal fan. This Slide is an animation of wheel wobble mode. The FEA model contains bearing pedestals; It could have included foundation, floor slab, etc. Boundary conditions (rigid constraints, spring constants) are place at the terminal point of the FEA model).

Finite Element Analysis

FEA method focuses on calculating the behavior and response of a continuum that consists of an infinite number of points. In a continuum problem, a field variable such as displacement or velocity contains an infinite number of possible values, since it is a function of each point in the continuum. This task is simplified using a finite element representation that divides the continuum into a finite number of subdivisions called *elements*. The elements are connected at *nodal points* into a mesh or finite element model. The process of dividing the continuum into a finite number of elements makes the solution provided by the finite element model an **approximation** to the theoretical solution.

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Review of Some Digital Signal Analysis Basics

Robert J. Sayer, PE

President, The Vibration Institute

Digital Signal Analysis Basics

Modern fast Fourier transform (FFT) analyzers are digital instruments. A block of vibration data is digitized in an analog-to-digital converter and then processed using a fast Fourier transform algorithm.

Fan on Isolator 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?

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Fan on Isolator Base

Fan speed controlled by VFD. At normal operating conditions:

1x Fan = 45.3 Hz

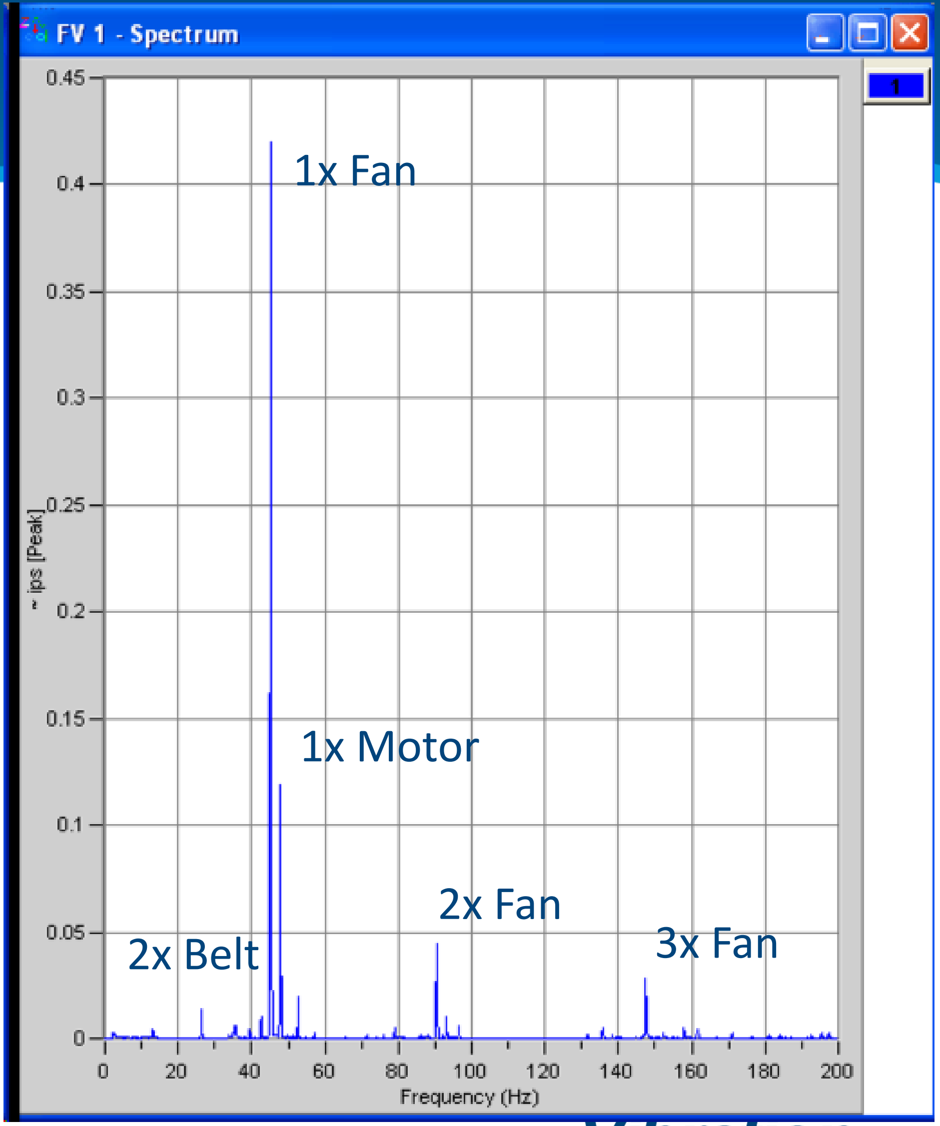
1x Motor = 47.9 Hz

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.



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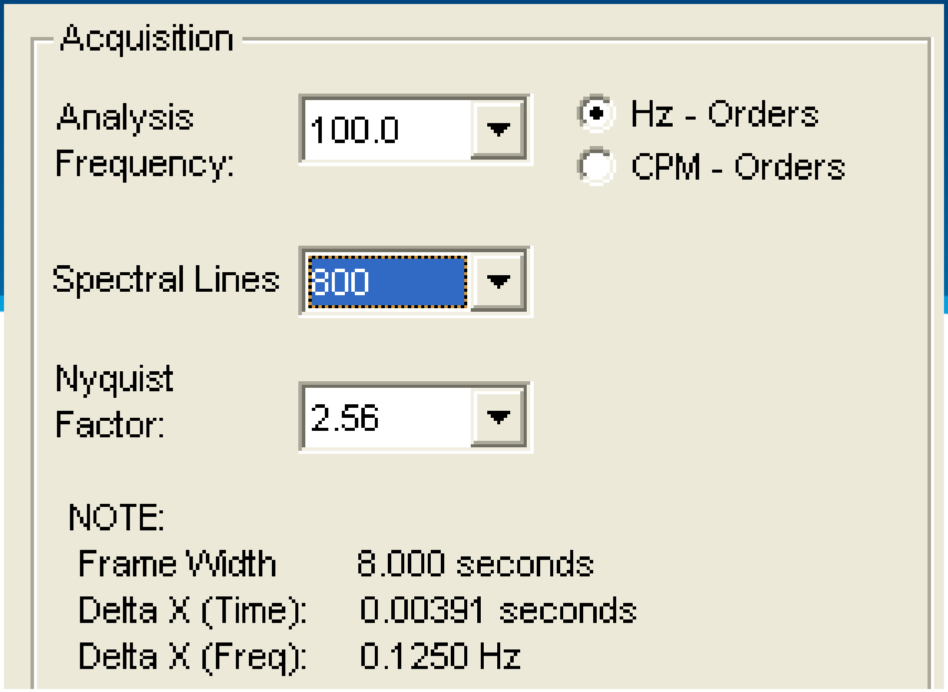
FFT Resolution



Acquisition parameters that must be defined prior to acquiring data are F_{max} (maximum analysis frequency) and N (spectral lines of data). These parameters dictate the sampling rate and resolution of the digitized data.

For the example:

F_{max} = 100 Hz, N = 800, Freq Resolution = 100/800 = 0.125 Hz

Time Req'd per Sample = 1/Freq Resolution = 8 seconds

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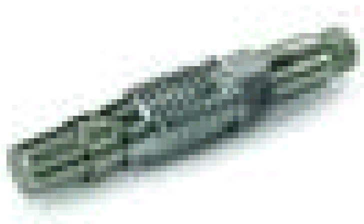
Vibration Analysis of Centrifugal Fans

Definition - Any device that produces a current of air by movement of a broad surface can be called a fan. Industrial/commercial fans fall under the general classification of turbomachinery. They have a rotating impeller and are at least partially encased in a stationary housing.

Fans are similar in many respects to pumps and compressors.

Common Analyst Tools
For Fan Vibration Analysis

- Vibration Analyzer
- Accelerometer/Velocity Sensor
- Proximity Probes
- Dynamic Pressure Sensor
- Microphone
- Shaft Stick



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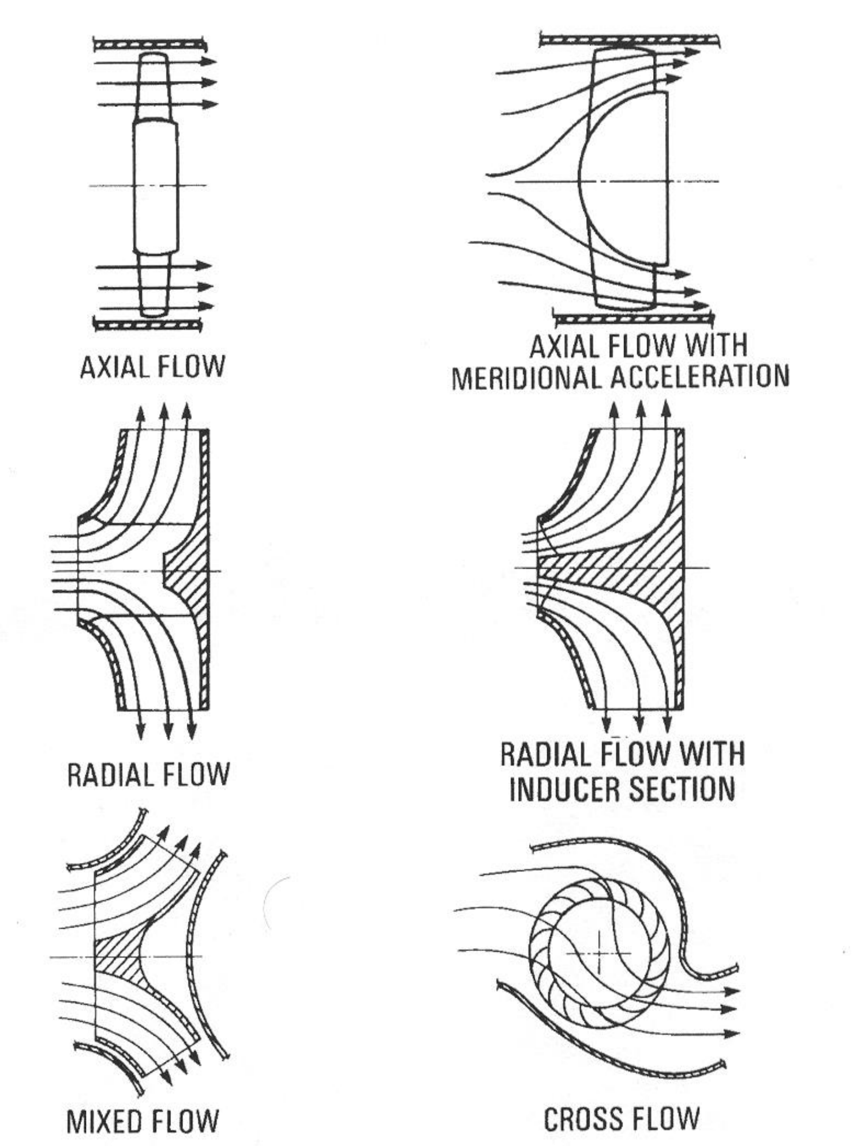
Aerodynamic Classification of Fans

There are many types of Fans.

This Presentation concentrates on Radial Flow (Centrifugal) Fans



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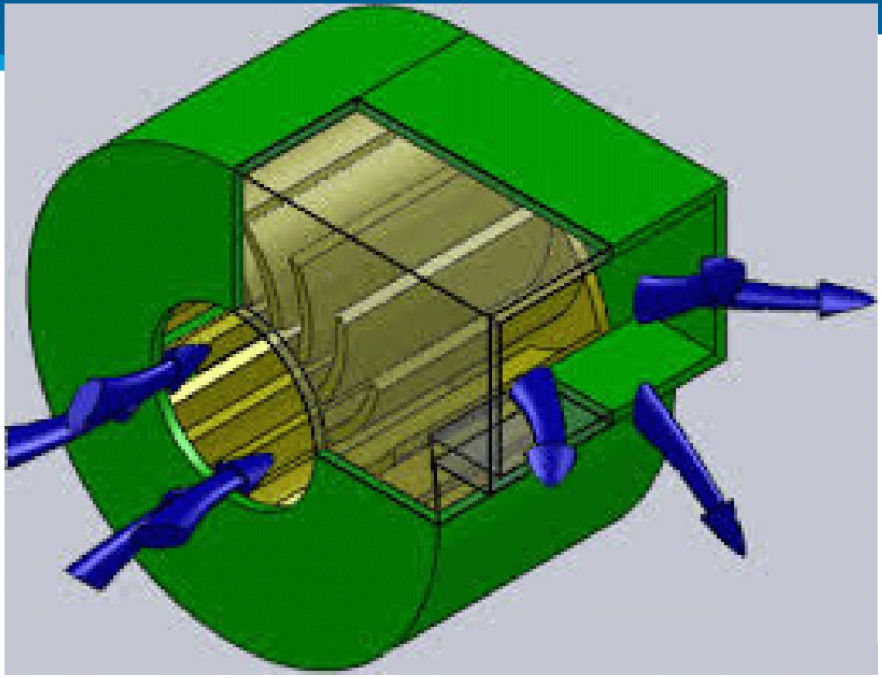
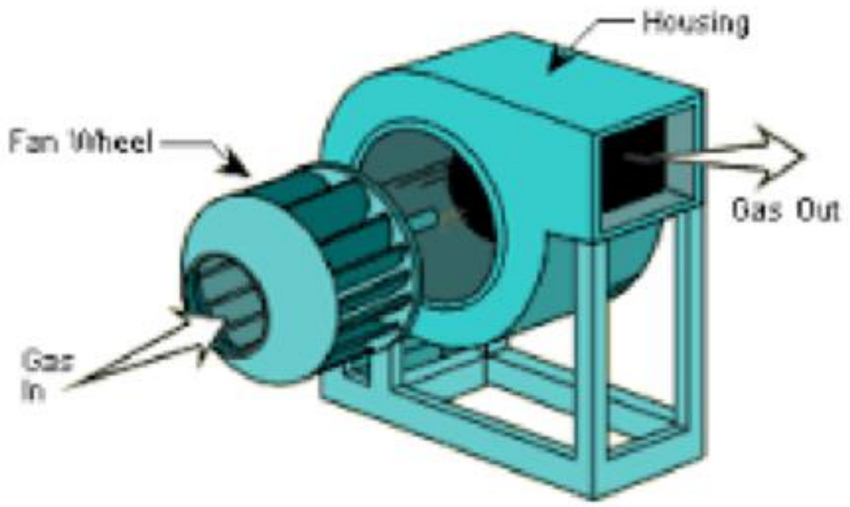
Air Flow Through Centrifugal Fan

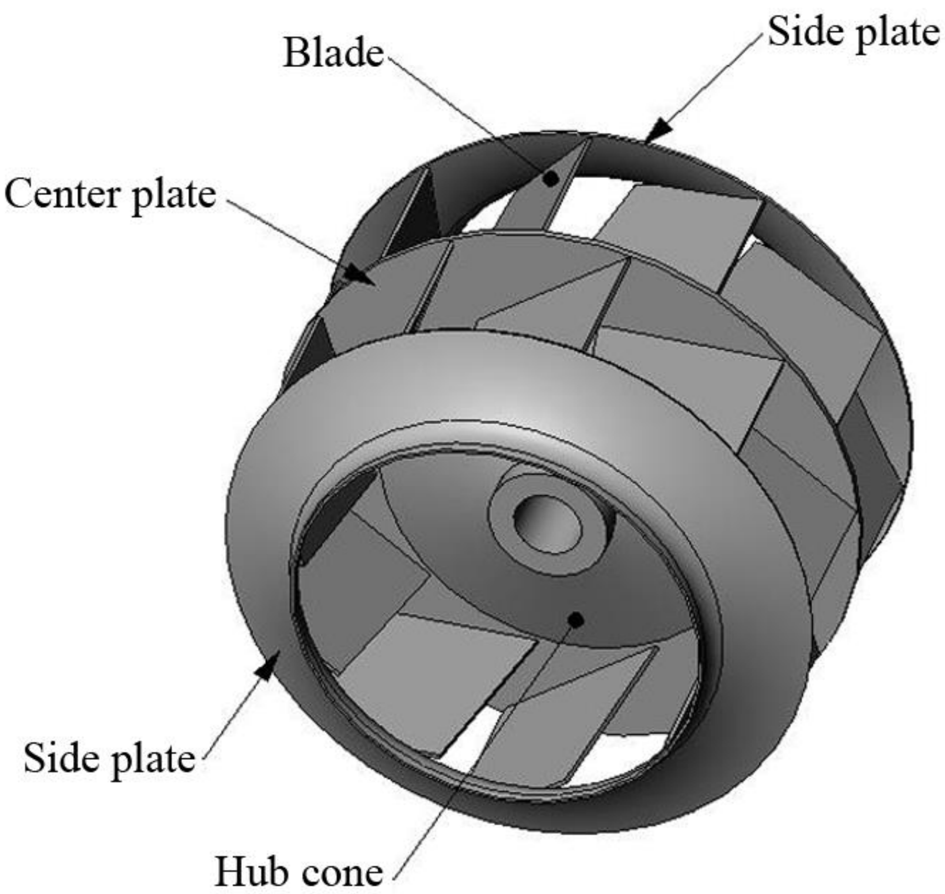




Air enters in center of Fan Wheel

Air leaves at Outer Diameter of Fan Wheel

Double-Wide Double Inlet DWDI Centrifugal Fan Wheel



DWDI Fan Rotor

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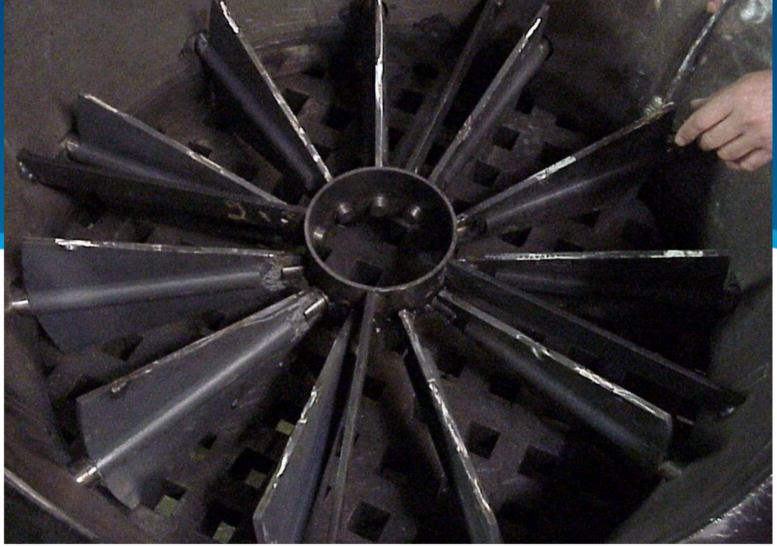
Sources of Dynamic Pressure Pulsations

- Blade Pass Pulsation Pressure (Normal for all Fans)
- Transient Process Pressures (Dependent upon Application)
- Rotating Stall
- Surge
- Inlet Box Vortex Shedding
- Oulet Box Vortex Shedding
- IVC Vortex Shedding

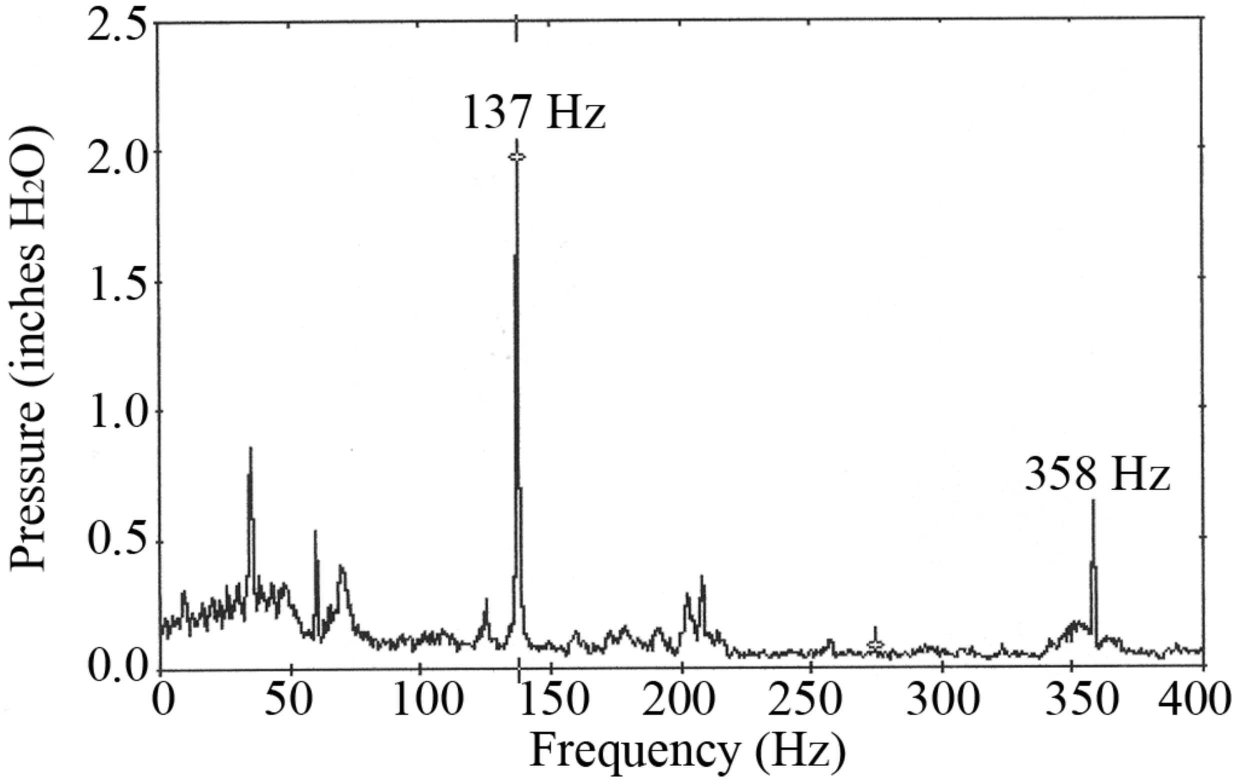
Note: Rarely detected as high vibration at bearing. Methods of Detection- transducer on duct, dynamic pressure sensor, and/or microphone.

IVC Vortex Shedding



IVC Damper Control – Can Throw Vortices @ Certain Opening Angles

IVC Vortex Shedding



- Detected by spectral analysis of dynamic pressure data.
- 12- Bladed, 1800 rpm SWSI Fan with IVC Damper.
- BPPF = 358 Hz; Pressure ~ 0.60 inches
- IVC Vortex Freq = 137 Hz (~4.57x Fan Speed or 38% of BPPF)
- Vortex Pressure = 2.0 inches > BPP ~ 0.65 inches

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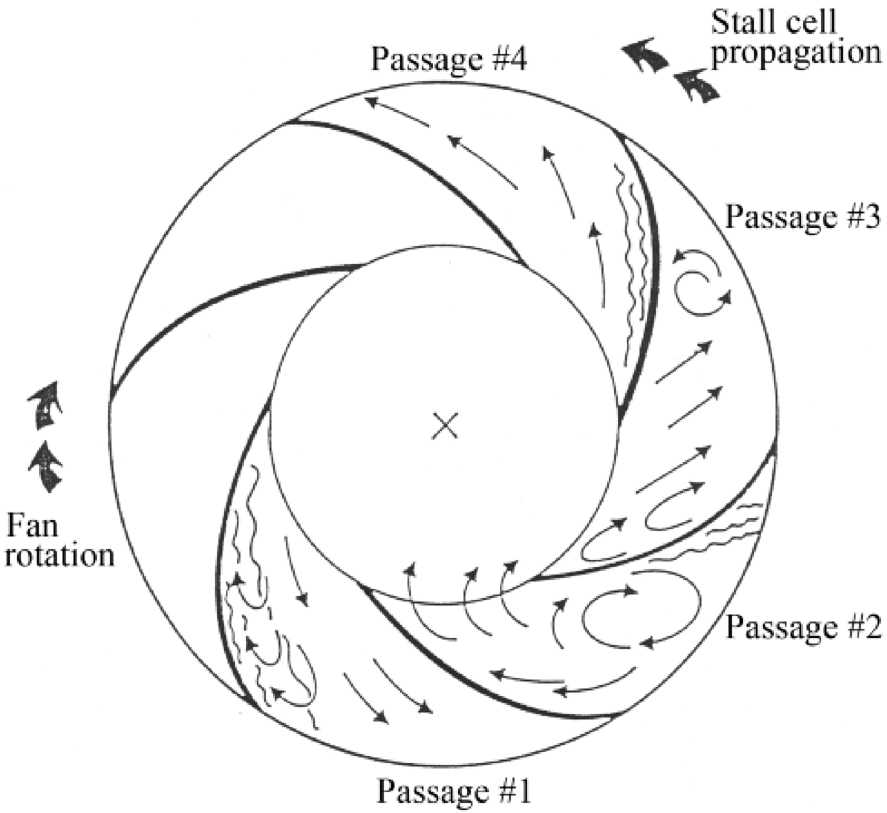
Rotating Stall

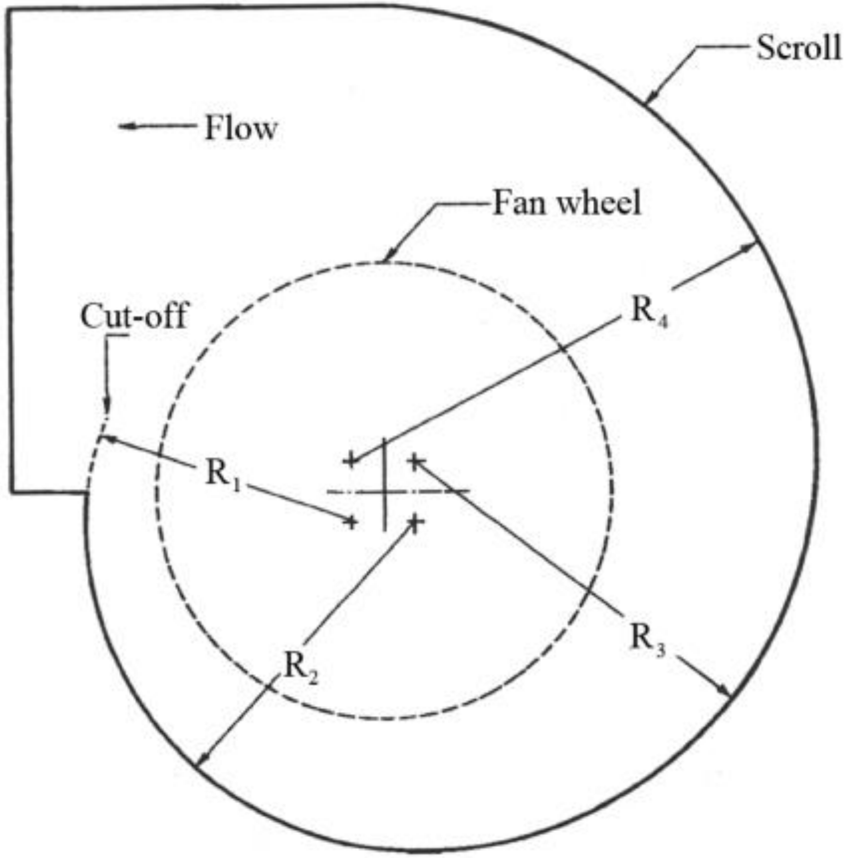


- Rotating Stall is caused by steep incident angle at low flow conditions. This produces a boundary layer separation on suction side of blade in Passage #1.
- Rotating Stall Cell rotates opposite to the direction of rotation of the fan wheel.

Rotating Stall

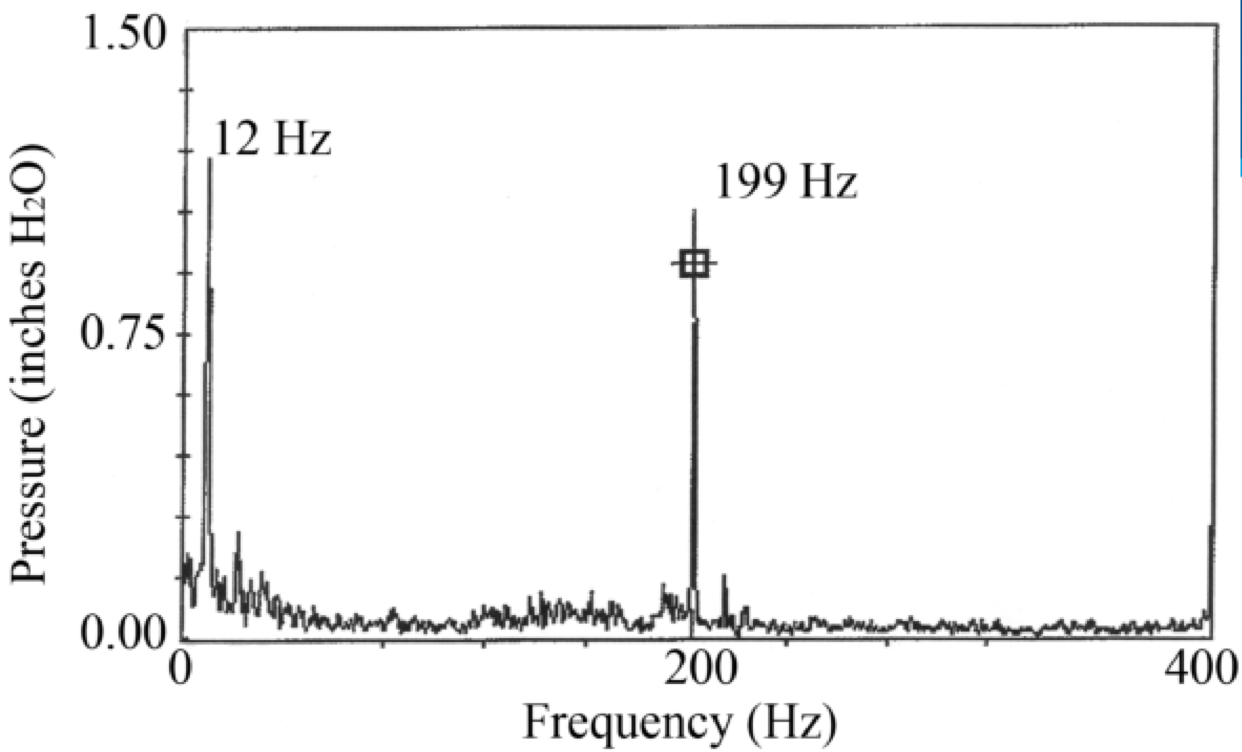
- Low frequency and low pressure rarely cause a problem in the fan wheel. A fully developed rotating stall cell typically occurs at a frequency within 2/3 -3/4 x fan speed. However, stall pressure pulsations have been documented between 0.60 - 1.0 x fan speed.
- Rotating stall is periodic, but not exactly harmonic. Thus, it frequently produces forces at harmonic multiples of principal stall frequency.
- It is possible that a single fan wheel can have multiple stall cells, which results in larger force at a higher harmonic.

Blade Pass Pulsation Frequency



- Fans produce pulsations @ BPPF as blades pass cut-off point in the scroll.
- BPPF = No. Blades x rpm

Pulsation Spectrum (BPPF & Stall)



Identified by Spectral Analysis using a Dynamic Pressure Sensor.

Spectrum is for a 10-bladed DWDI fan rotating at 1195 rpm (19.9 Hz)

BPPF = 10 blades x 19.9 Hz = 199 Hz; Stall Freq = 12 Hz which is 0.60x rotational speed

Average Stall Pressure = 1.2 inches = .043 psig; Avg BPPF Pressure = .98 inches = .035 psig

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Example of Misinterpretation of FFT Data

Fan Failure (10=bladed fan @ 1190 rpm).

Theory #1: 2-Nodal Diameter Mode excited by 1/2xBPPF

Theory #2: 5-Nodal Diameter excited by BPPF

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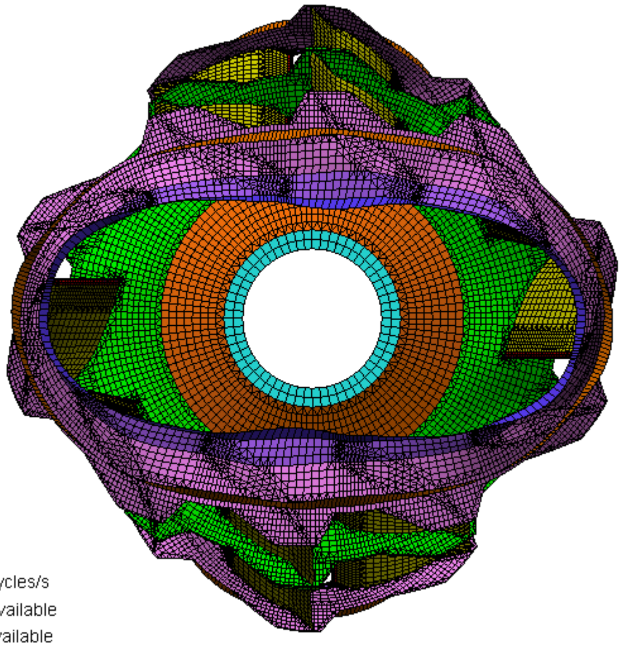
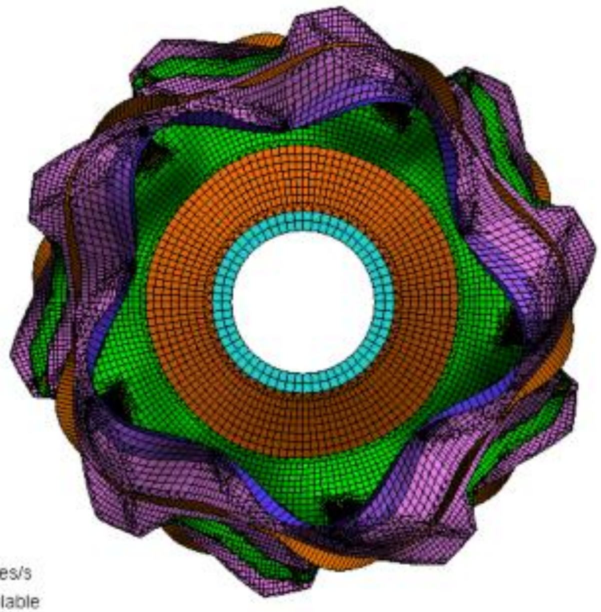
FEA Analysis of Fan Wheel





2 Nodal f_n = 98.6 Hz ~ ½ BPPF

5-Nodal f_n = 199.6 Hz ~ BPPF

FEA results support both theories.

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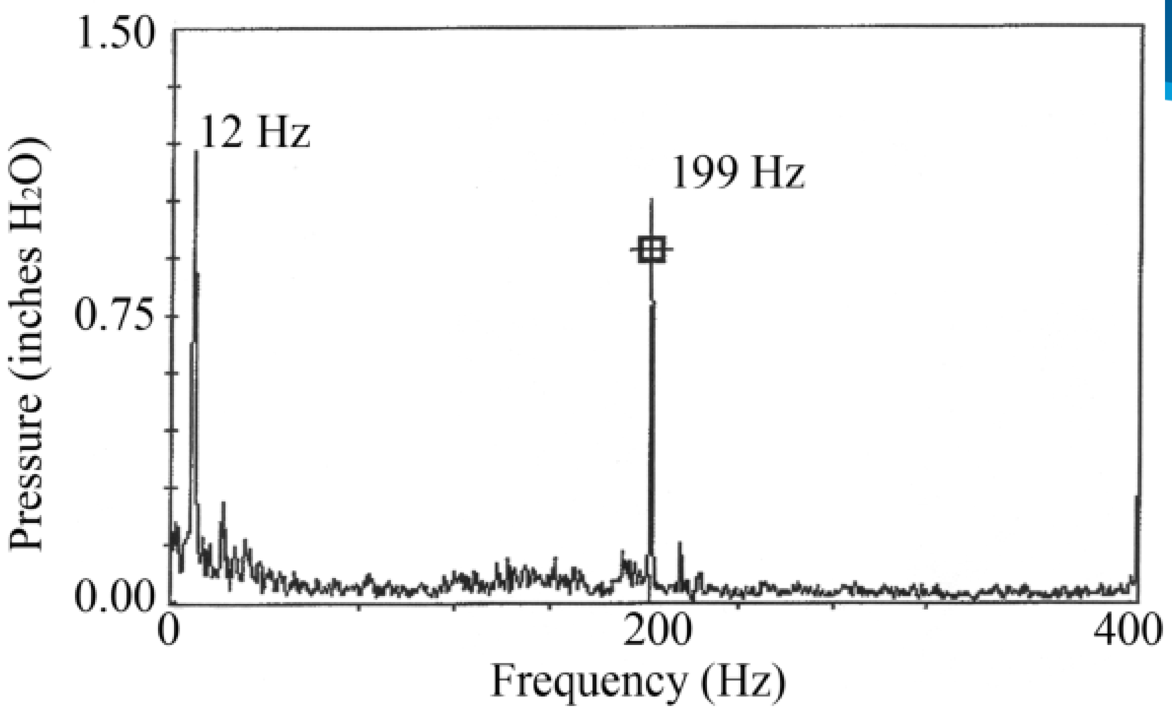
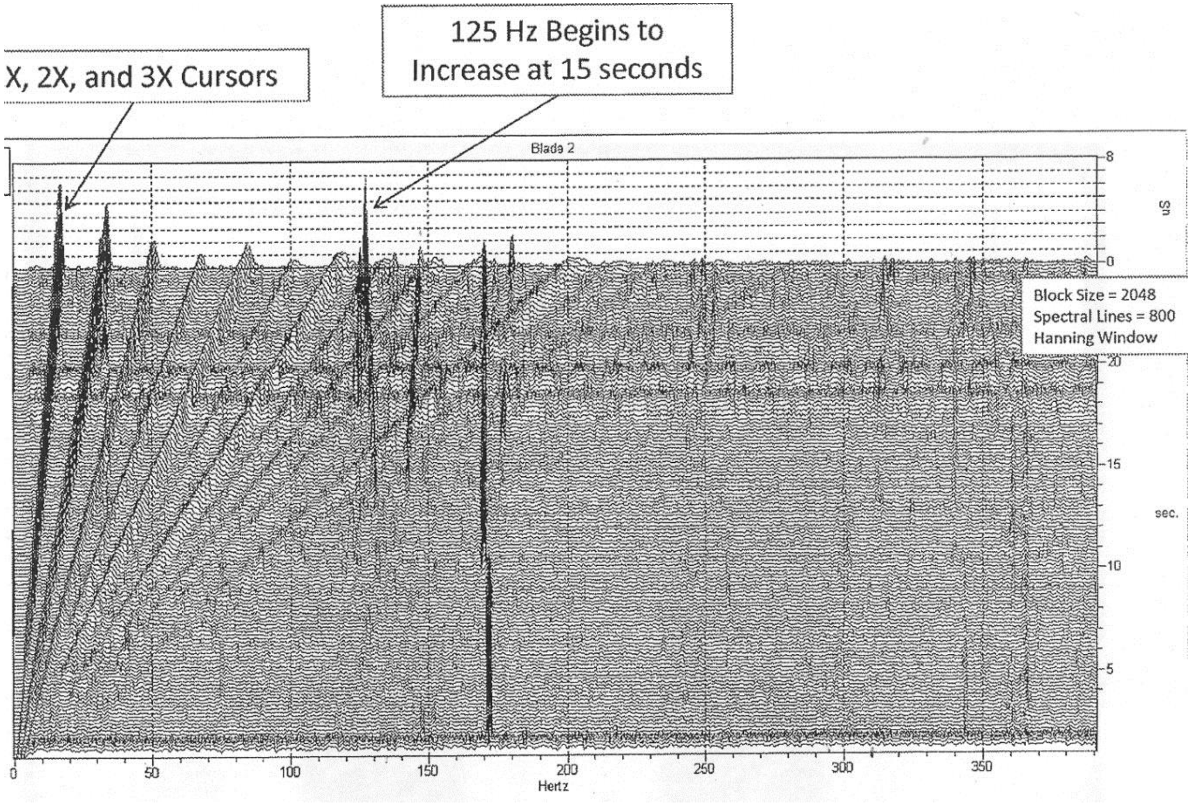
Dynamic Pressure Data from Fan



Dynamic pressure data clearly indicates presence of pulsation at BPPF which is not unusual for a fan.

Dynamic pressure data does not show any pulsation at 1/2xBPPF. There must be a force for resonance to occur.

Start-Up Strain Data (New Fan Design)



This data was used to argue that a 5x or 1/2xBPPF pulsation existed and that it was around twice as large as the BPPF at 10x.

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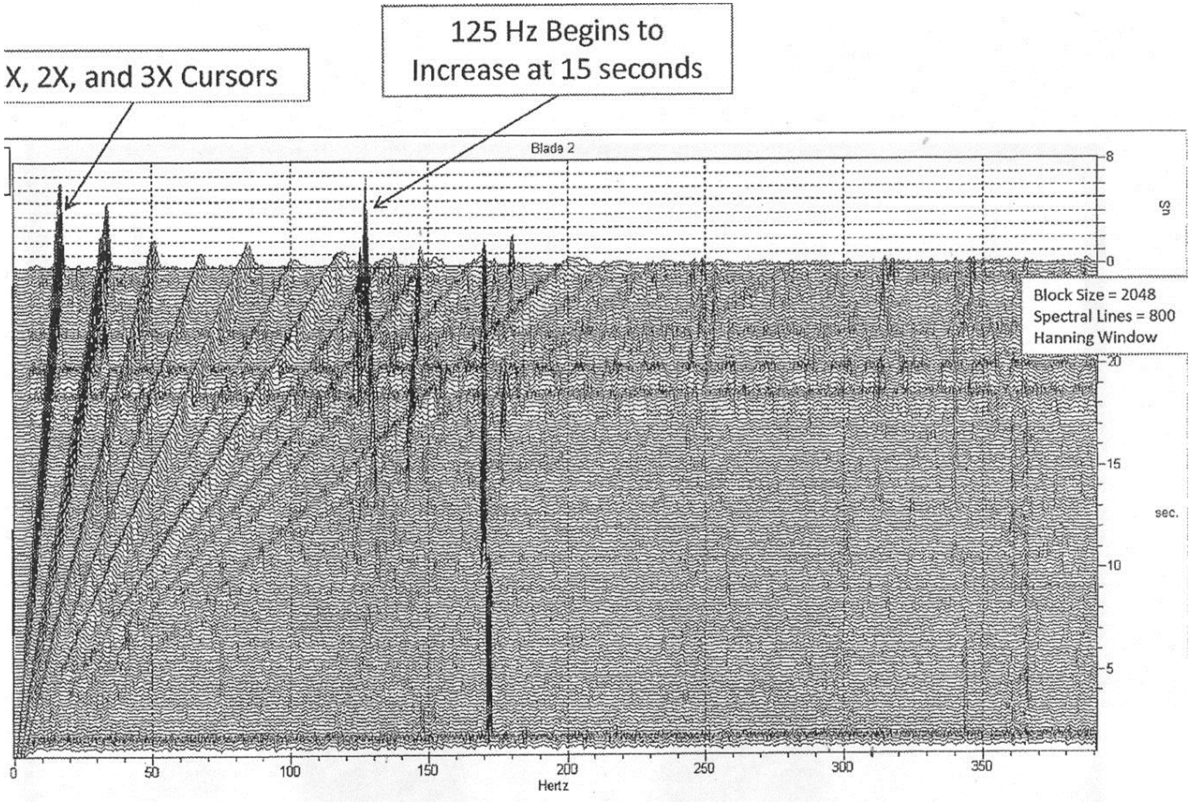
Start-Up Strain Data (New Fan Design)

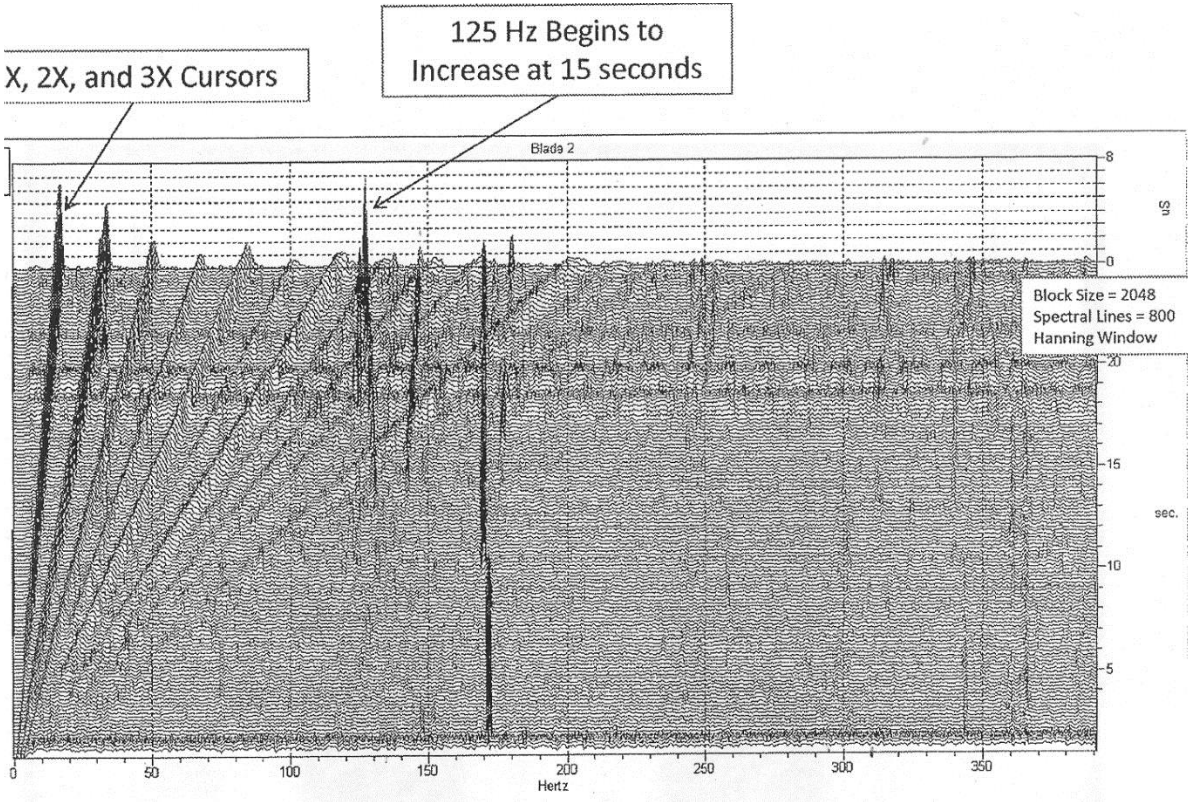


Problem #1: F_{max} = 400 Hz; 800 lines; requires 800/400 =2 seconds of data for each spectrum on the waterfall. The speed increase rate = 1200 rpm/30 sec = 40 rpm/sec.

Change in 10x frequency = 10 (40/60)x2 sec = 13.33 Hz

Change in 5x frequency = 6.67 Hz

Start-Up Strain Data (New Fan Design)



- Problem #2: Strain Gages record strain, not force. This data does not confirm presence of 5x pressure pulsation.
- Start-up of induction motor will contain torsional pulses due to pole slipping or soft-start harmonic distortion.

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Case History - Fan Duct Support Vibration/Noise Issues

Photo During Construction

Large ID Fan Exhaust Duct Noise & Vibration Problem

Site suspected aerodynamic excitation from unusual placement of outlet damper (some distance from fan).

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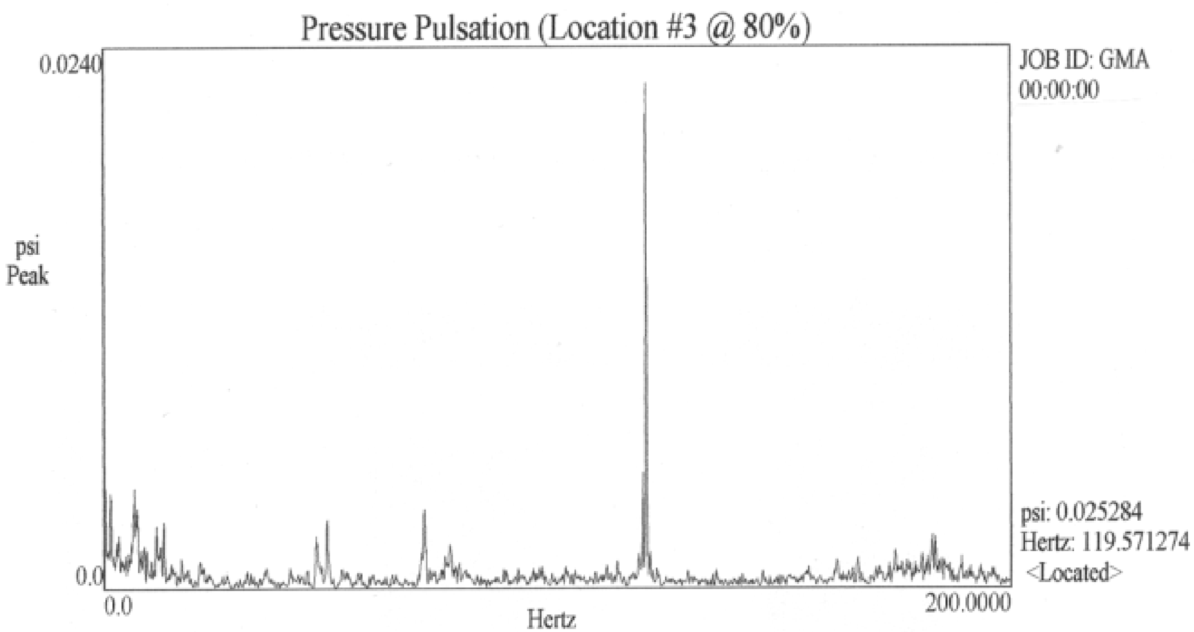
Case Study

Photos after Construction

Pressure Pulsation Data



- Frequency Spectrum of Pressure Pulsations
- 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.

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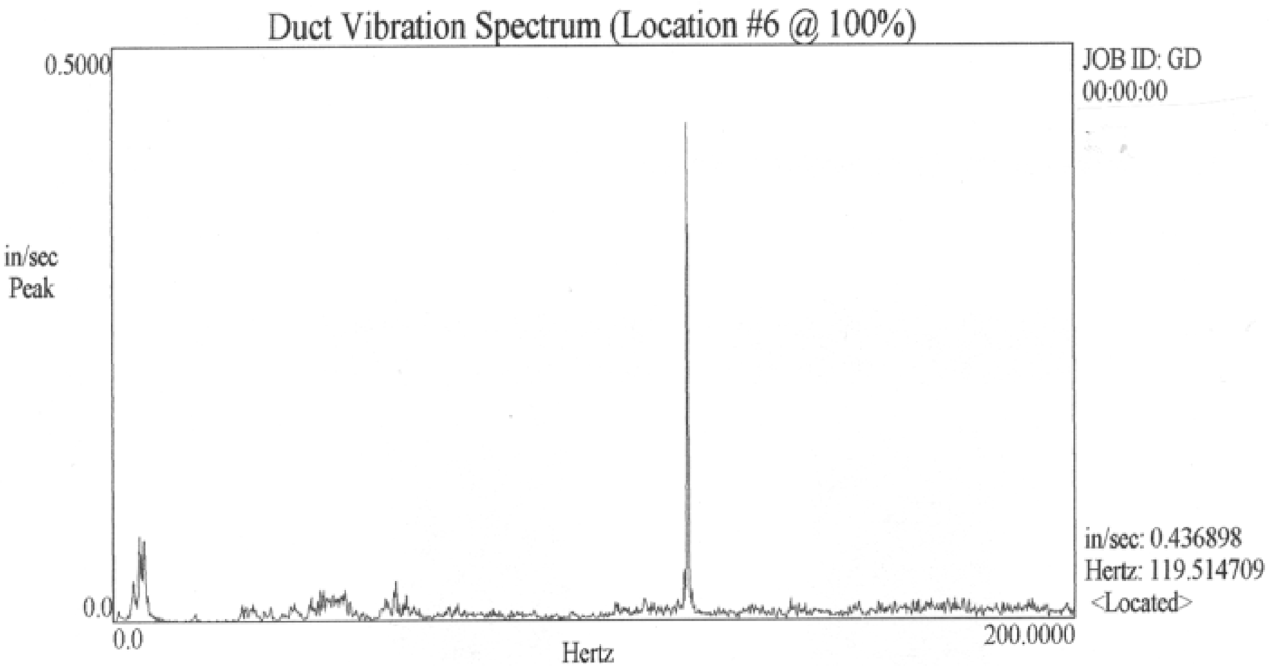
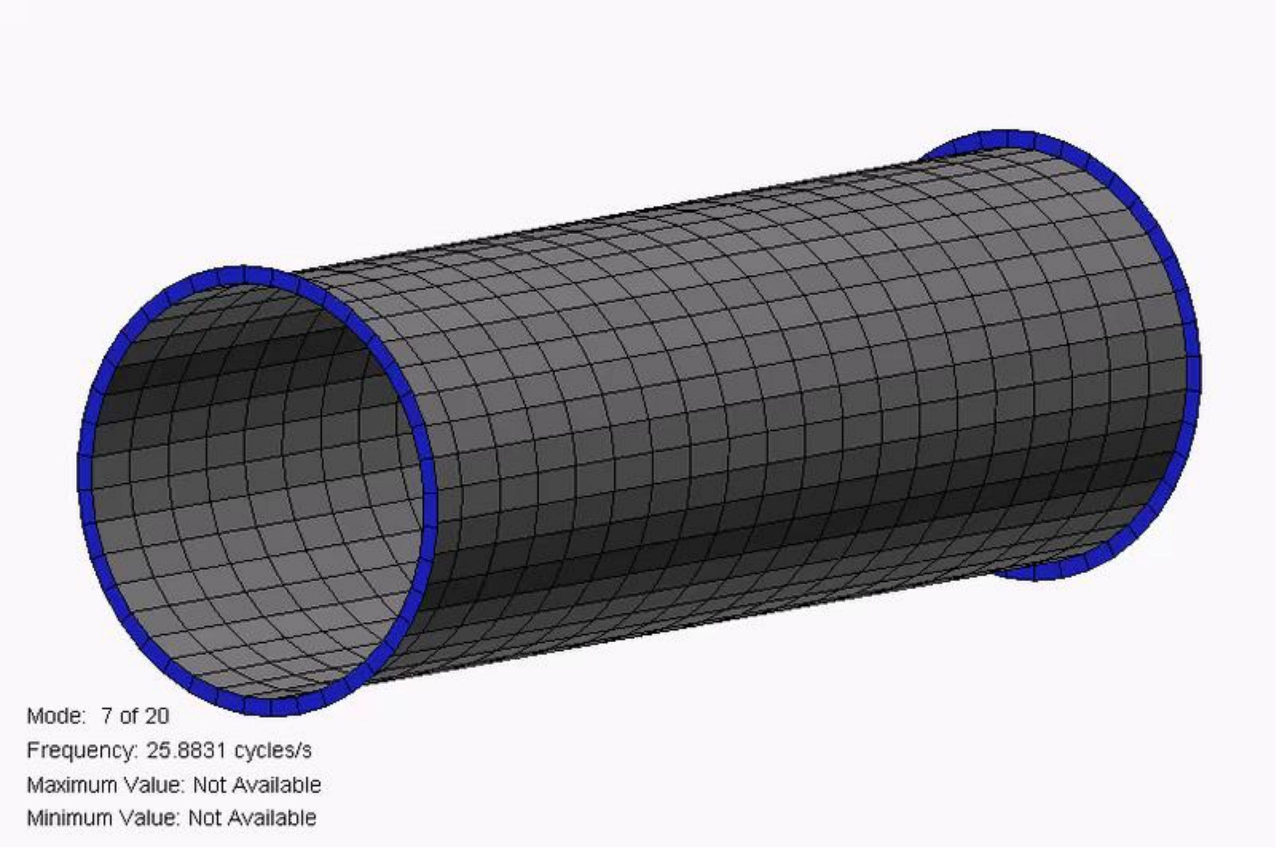
Duct Vibration



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

Natural Frequency of Duct



fn (25.8 Hz) is not even close to BPPF (119.6 Hz). Natural Frequency not the problem. However, Duct is very flexible.

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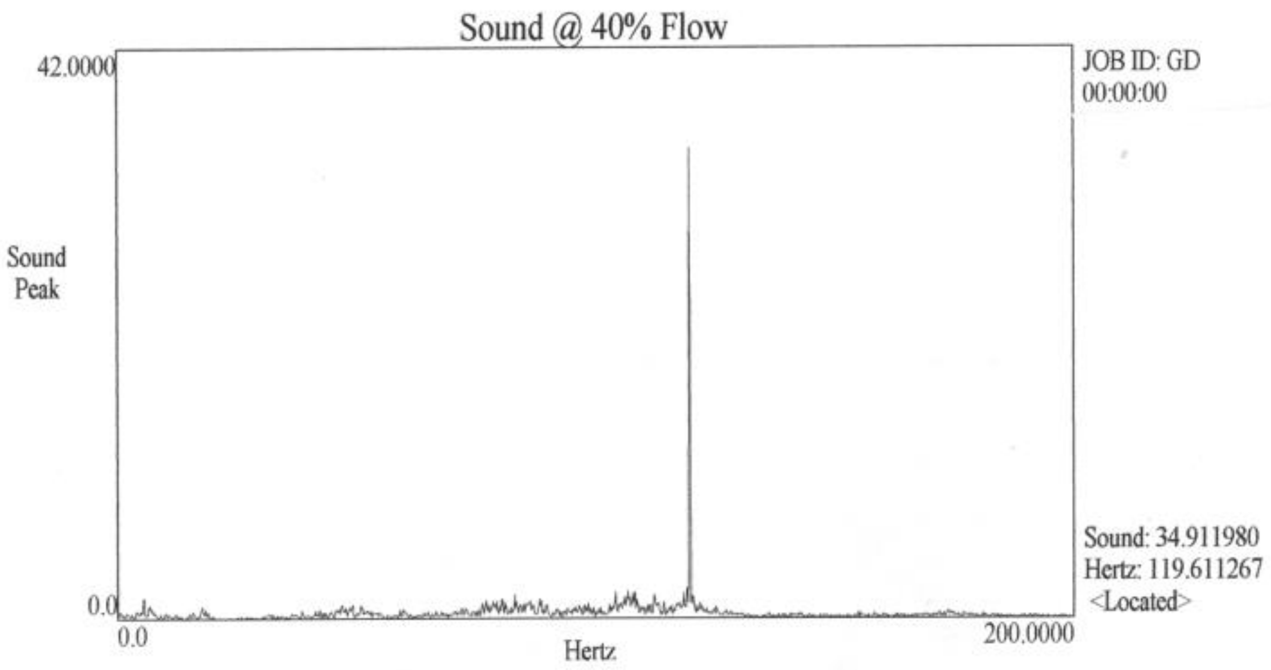
Sound Data



- Frequency Spectrum of Noise.
- Sound Pressure related to Duct Vibration which is caused by BPPF pulsations. Moving outlet damper will not effect duct vibration and noise.

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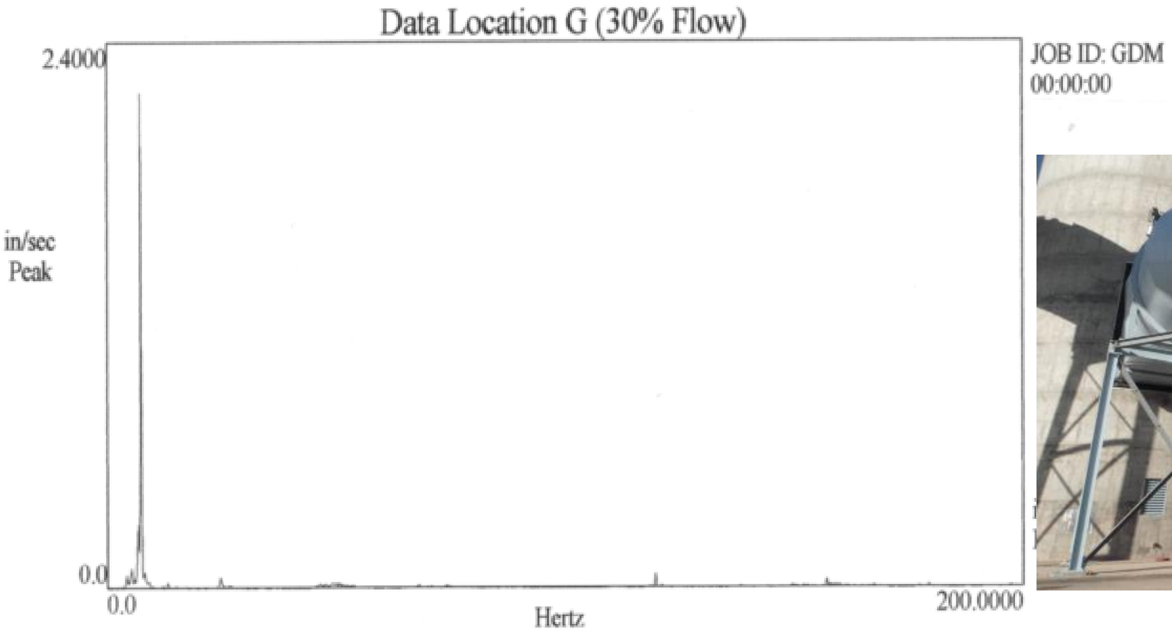
Structural Vibration Data



- Frequency spectrum of structural vibration dominated by subharmonic 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.

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Fan Ductwork

Fan Ducts come in a variety of Sizes, Shapes & Stiffness; Light Gage versus Thick Plate

Circular vs. Rectangular

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Fan Duct Work

- Sensitivity of Ductwork is dependent upon:
- The Source of Pulsation (and it's frequency content) &
- The Stiffness of the Duct (and it's Natural Frequency)

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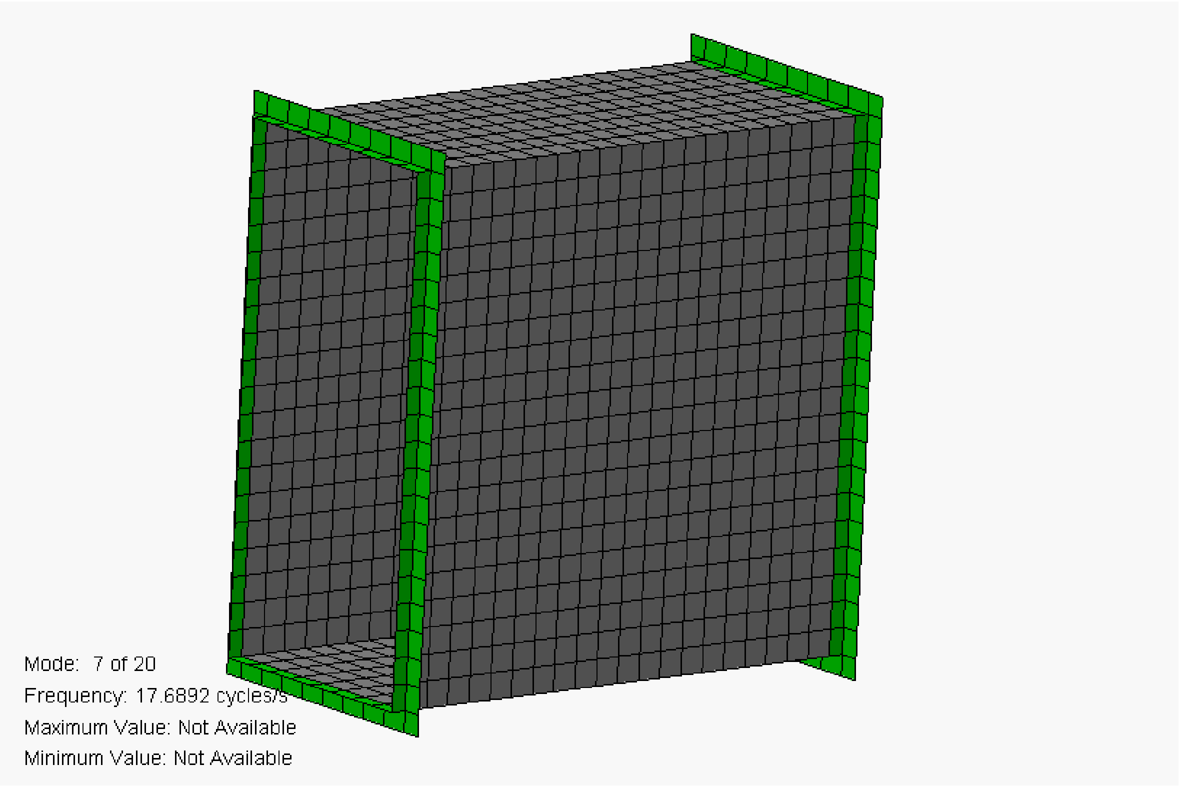
Flexible Rectangular Duct



- Natural Freq = 17.7 Hz
- Since Stall occurs between 0.60 - 0.75X Fan Speed,
- Susceptible to Resonance @ Fan Speeds between 1420 - 1770 rpm

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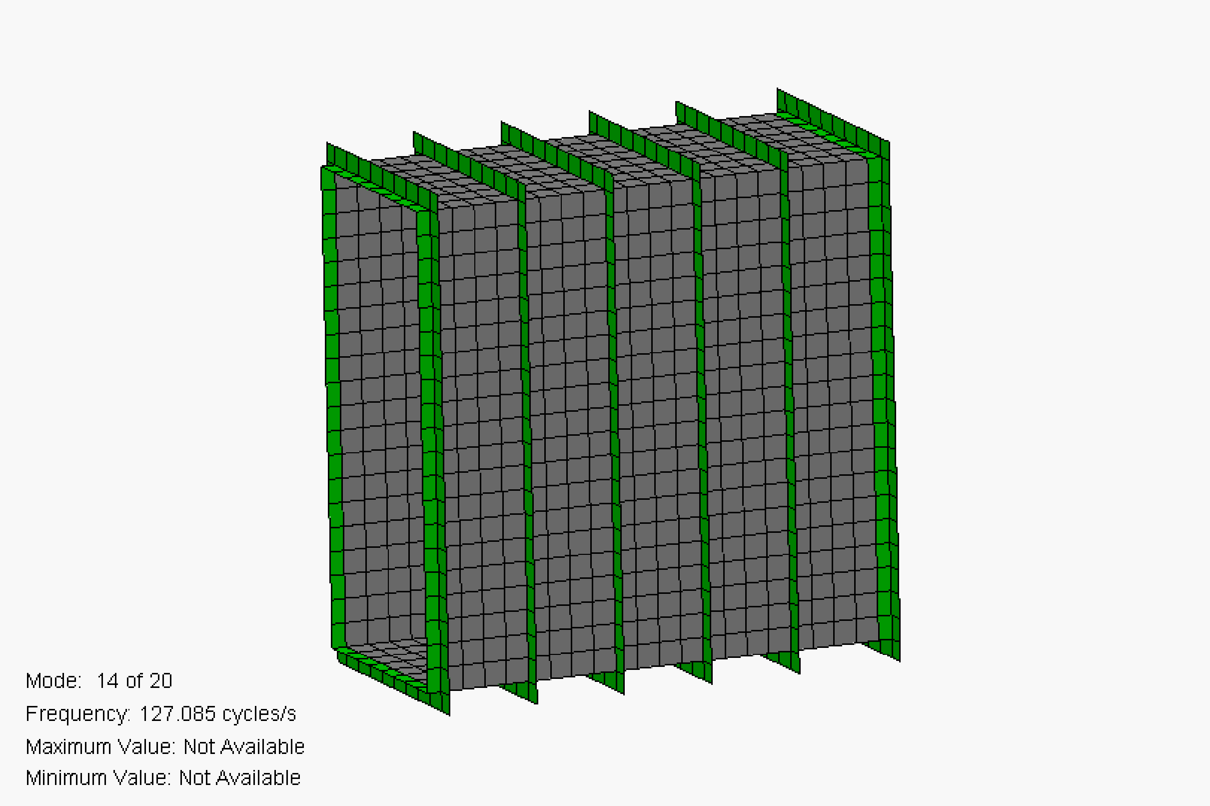
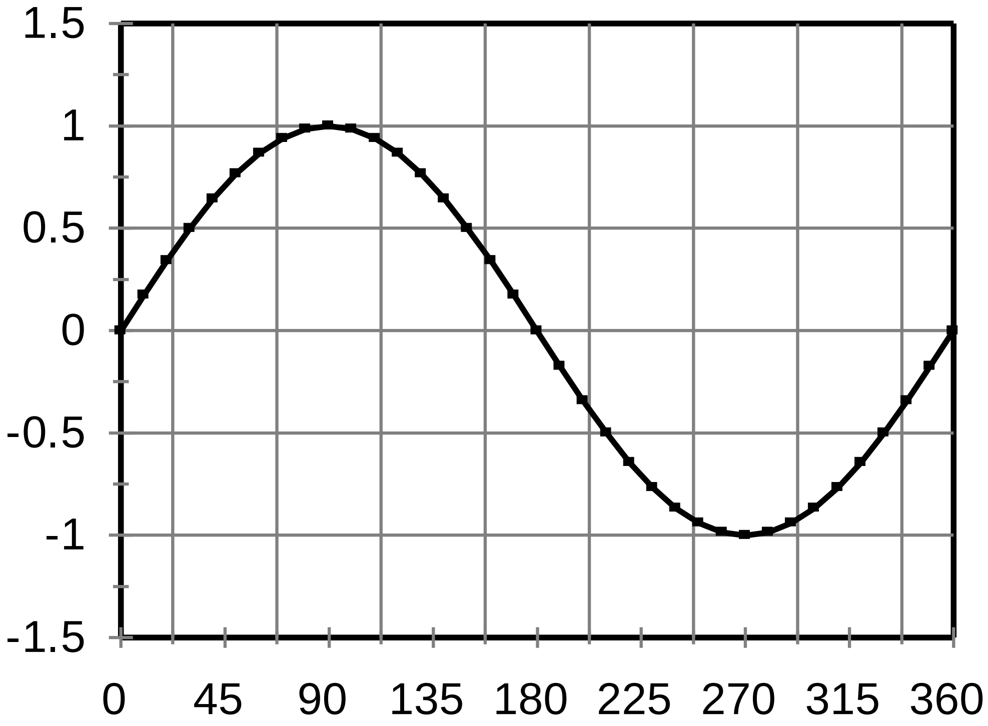
More Rigid Rectangular Duct



- Closely Spaced Stiffeners
- Natural Freq = 127 Hz
- 8-bladed, 900 rpm Fan; BPPF = 120 Hz
- 10-bladed, 720 rpm Fan; BPPF = 120 Hz
- Could also be sensitive to pulsations from inlet damper vane pulsations.

Resolving the Waveform

- Consider a pure sine waveform, where sampling is triggered to start at the very beginning of the sine wave (pure academic exercise).
- The above shows the approximation with 36 samples per cycle (sampling every 10 degrees).



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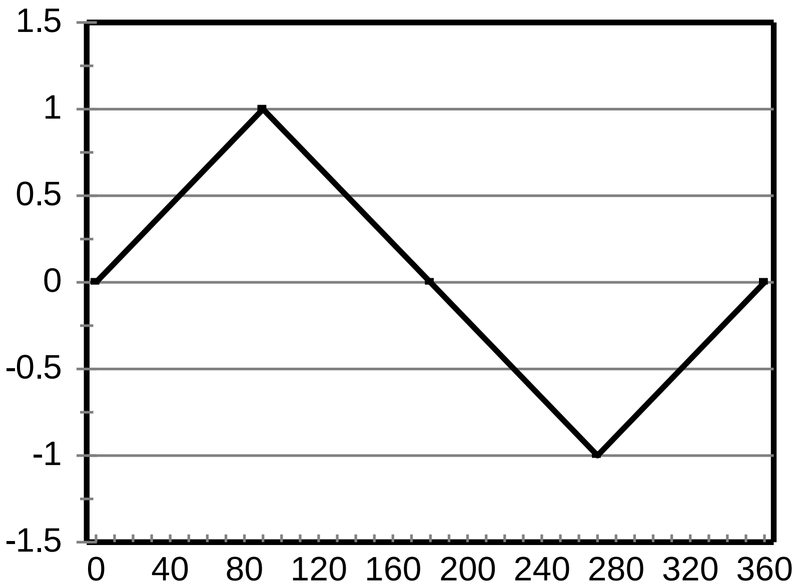
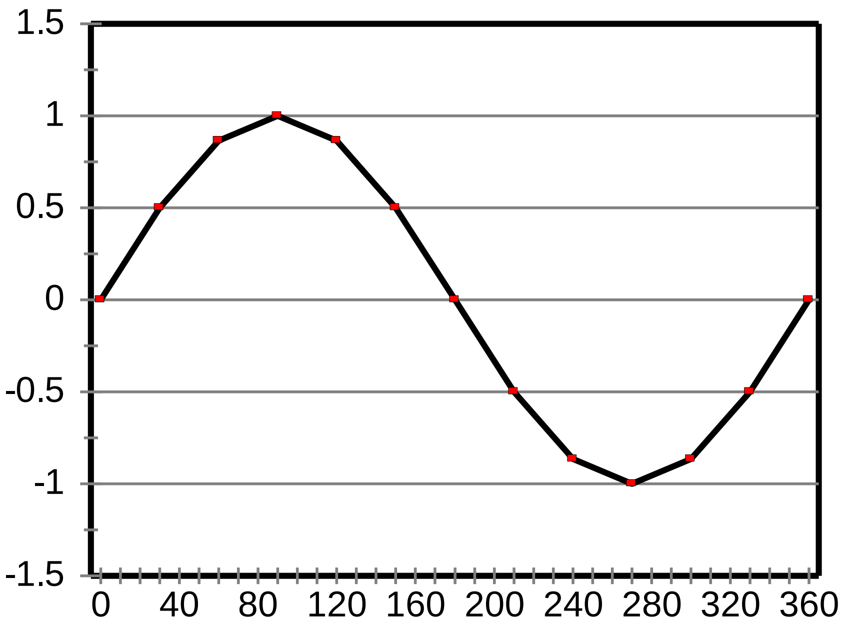
Resolving the Waveform





Sampling with 12 samples versus 4 samples per cycle.

Both provide Max = 1.0

Both repeat at the same frequency.

FFT of 4 samples will have harmonics.

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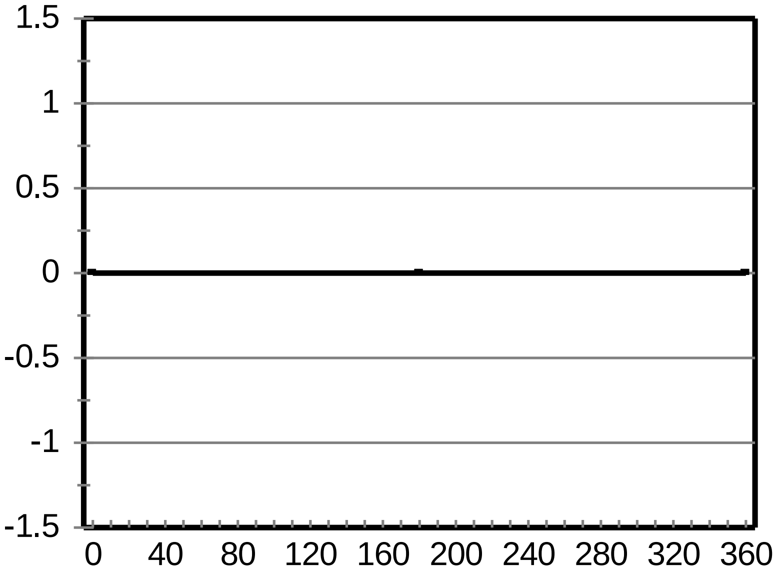
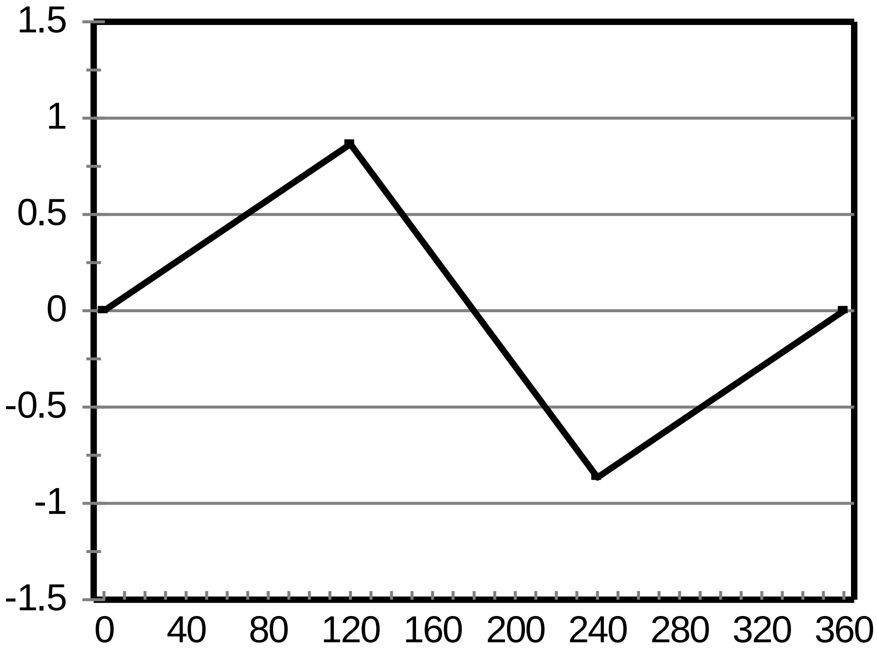
Resolving the Waveform





Sampling with 3 samples versus 2 samples per cycle.

Both miss the Max value 1.0.

3 samples Max = 0.866

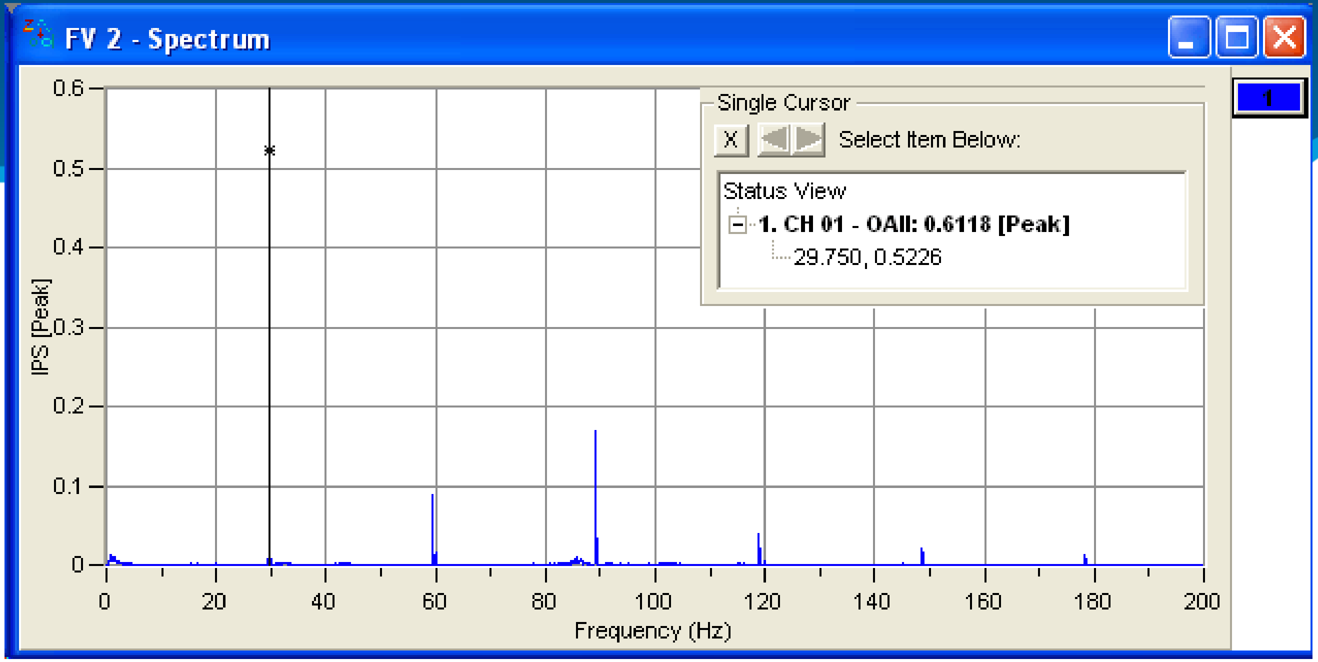
2 samples Max = 0 DC.

Vibration Institute

SWSI Fan

- 1780 rpm Fan
- Directly-Driven by Motor
- Low MTBF Rate of OB Bearing
- Apparent Excessive OB Vibration
- Apparent Large Foundation Vibration

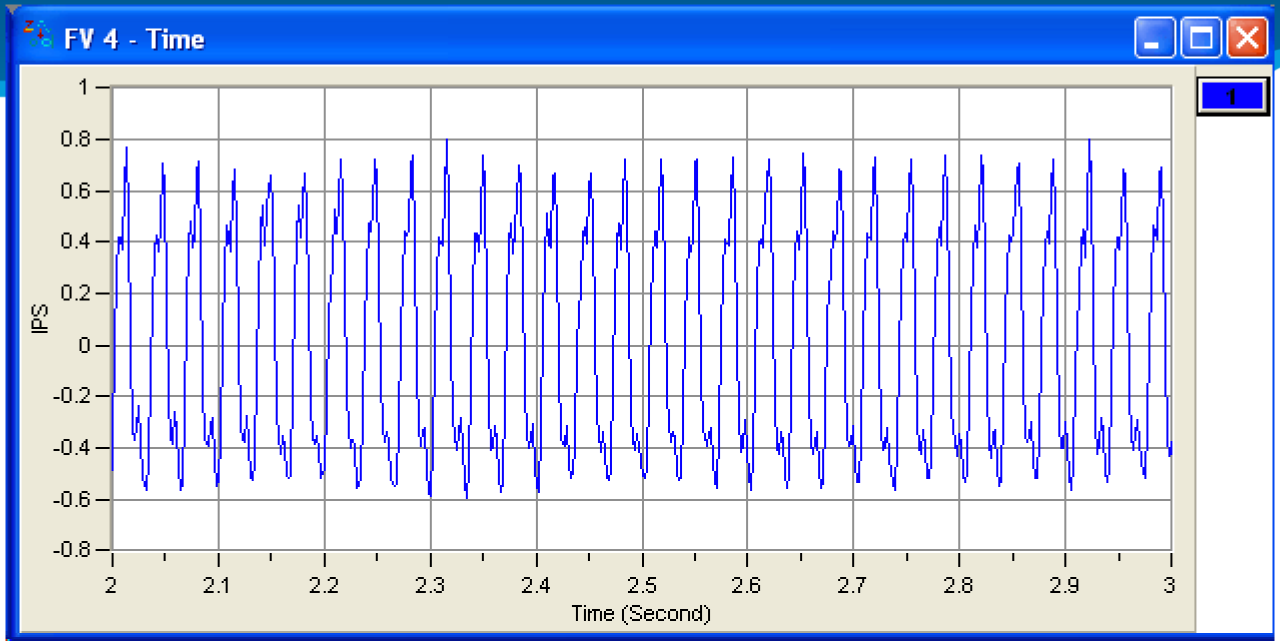
SWSI Fan



Overall Vibration = 0.612 ips (15.5mm/sec)

Highest Component = 0.523 ips (13.3 mm/sec)@ 1x

SWSI Fan Waveform



Fan Speed = 29.67 Hz, used F_{max} = 200 Hz.

Could have used F_{max} = 100 Hz , but sampling rate would have dropped.

Waveform shows 2 distinct events, closely spaced (phase) in time. This is not a truncated waveform. The 2 events would not have been clear at a lower sampling rate. This will be important in the root-cause analysis.

Vibration Institute

AMCA 204 Vibration Severity Criteria

Condition	Fan Application Category	Rigidly Mounted mm/s (in./s)	Flexibly Mounted mm/s (in./s)
Start-up	BV-1 BV-2 BV-3 BV-4 BV-5	14.0 (0.55) 7.6 (0.30) 6.4 (0.25) 4.1 (0.16) 2.5 (0.10)	15.2 (0.60) 12.7 (0.50) 8.8 (0.35) 6.4 (0.25) 4.1 (0.16)
Alarm	BV-1 BV-2 BV-3 BV-4 BV-5	15.2 (0.60) 12.7 (0.50) 10.2 (0.40) 6.4 (0.25) 5.7 (0.20)	19.1 (0.75) 19.1 (0.75) 16.5 (0.65) 10.2 (0.40) 7.6 (0.30)
Shut-down	BV-1 BV-2 BV-3 BV-4 BV-5	NOTE 1 NOTE 1 12.7 (0.50) 10.2 (0.40) 7.6 (0.30)	NOTE 1 NOTE 1 17.8 (0.70) 15.2 (0.60) 10.2 (0.40)

Value shown are peak velocity, mm/s (inches/s), Filter out.

Note 1: Shutdown levels for fans in Fan Application Grades BV-1 and BV-2 must be established based on historical data

Shutdown Level Rigid Mount = 0.40 ips

Shutdown Level Flex Mount = 0.60 ips

Measured Vibration = 0.61 ips exceeds both Shutdown Levels.

Vibration Institute

Fan Case Study

OB Bearing H Vibration = 0.612 ips

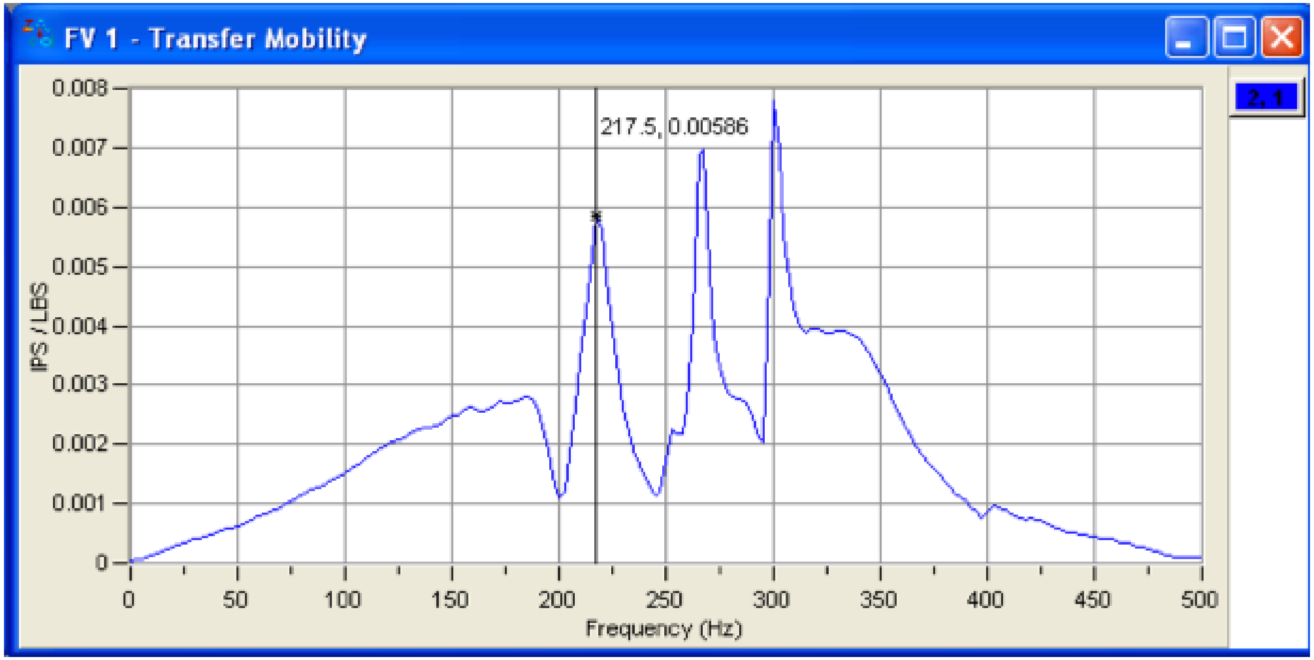
OB Bearing V Vibration = 0.116 ips

H/V Ratio = 0.612/0.116 = 5.3

Foundation Resonance? Need to perform Impact Test?

Vibration Institute

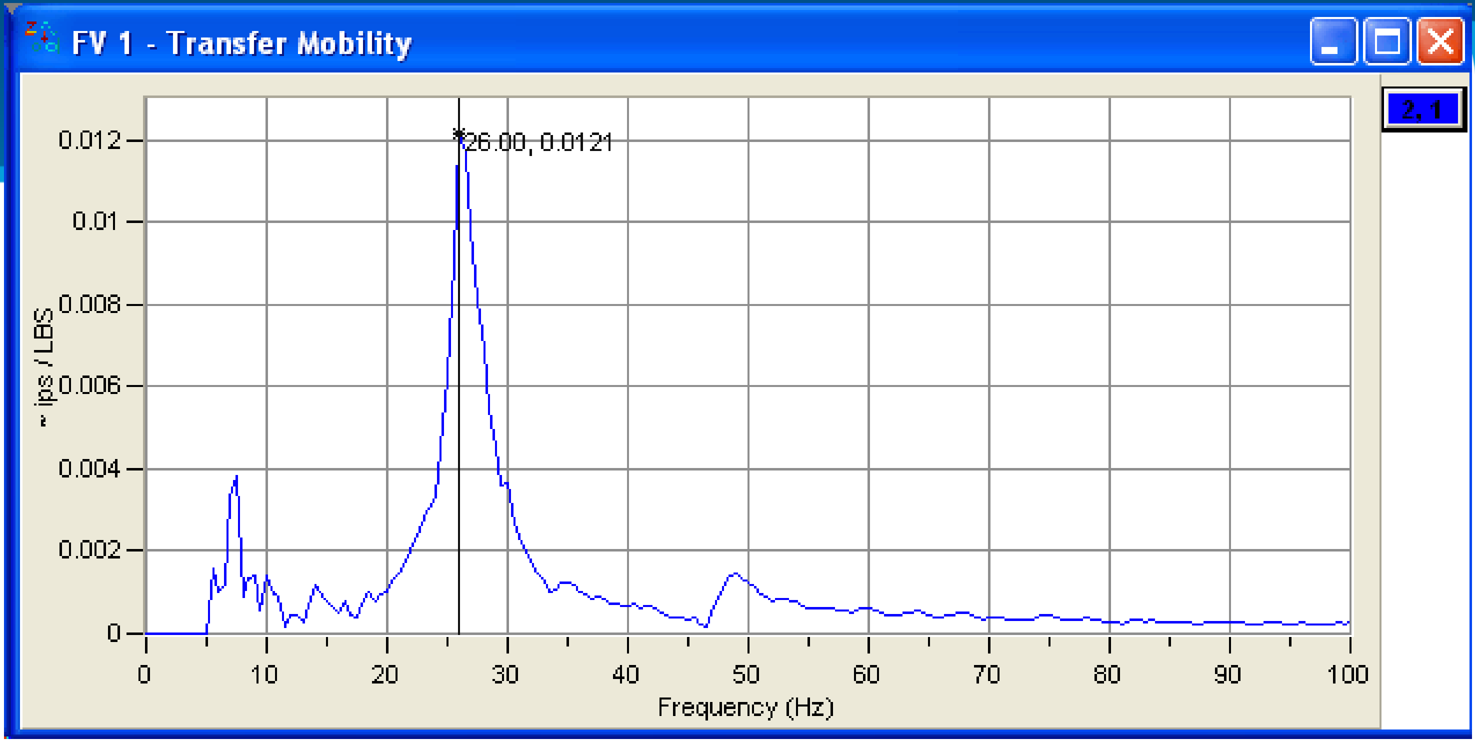
Natural Freq Test of Foundation



Natural Frequency Check of OB Bearing Support did not find any Natural Frequency near operating speed (1785 rpm = 29.75 Hz)

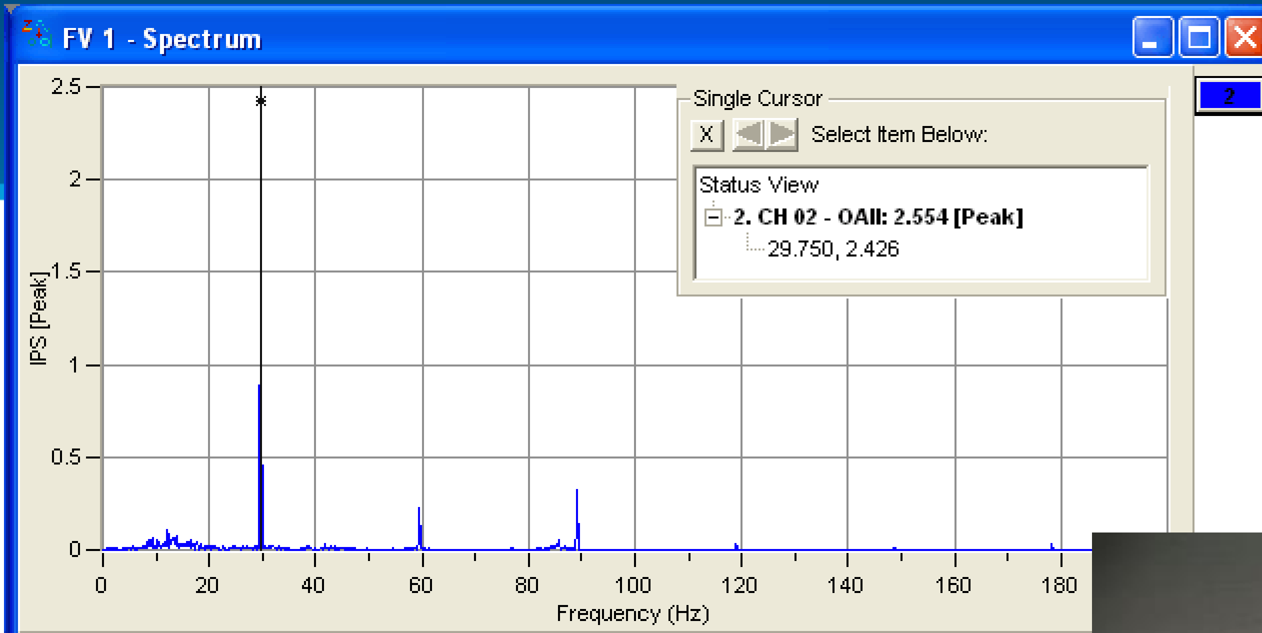
Vibration Institute

Natural Freq Test of SWSI Fan Rotor



- Natural Frequency Test Result ~ 26.0 Hz
- Operating Speed = 29.75 Hz
- Stress Stiffening Effects moves fn close to fo.

SWSI Fan

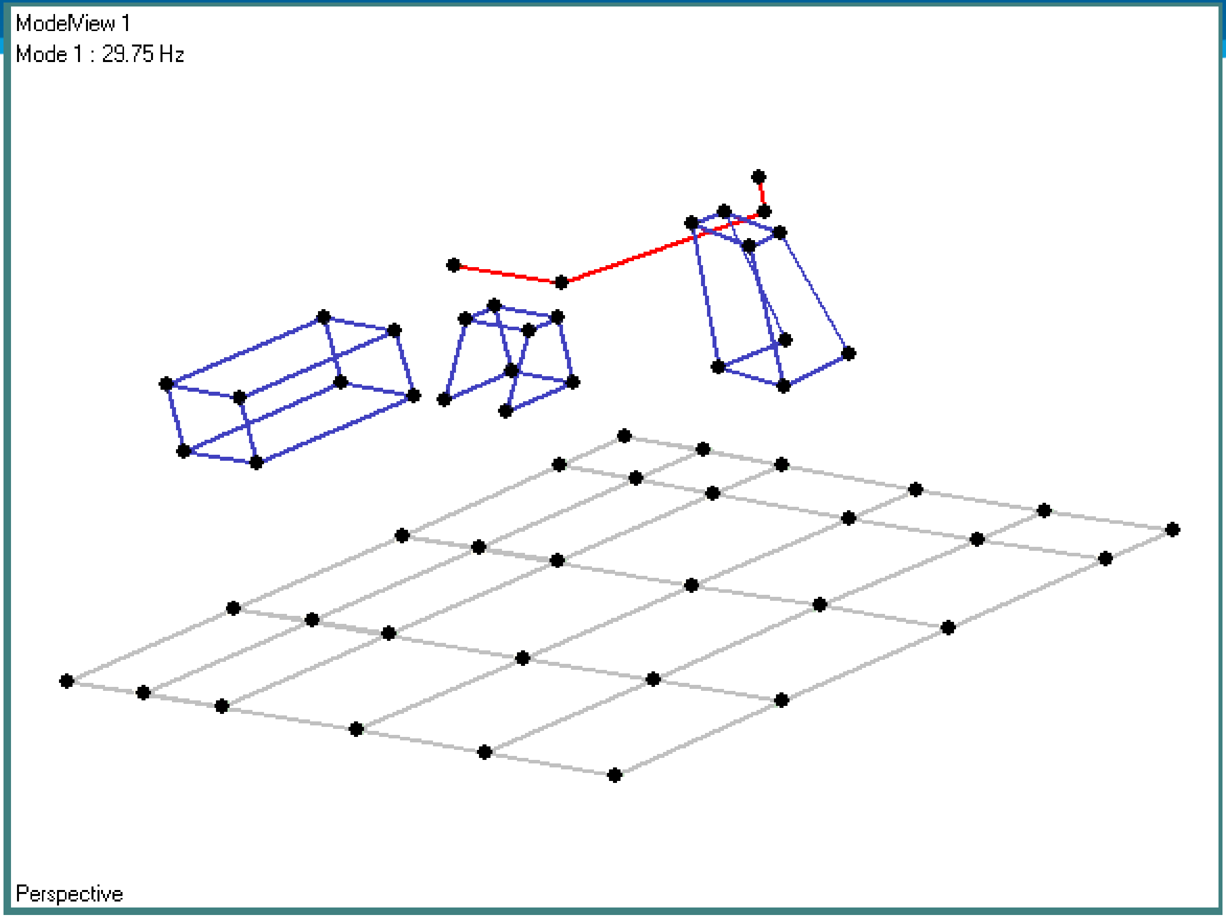


Frequency Spectrum of Shaft Vibration

V = 2.5 ips (63.5 mm/sec)

Compared to 0.612 ips (15.5 mm/sec) vibration level of Bearing.

ODS Testing to Diagnose Rotor Resonance in Anti-Friction Bearings



ODS clearly shows vibration response dominated by Fan Rotor.

Vibration Institute

Fan Case Study

What would have happened if a Motion Amplification Video were used to investigate the foundation issue?

The rotor would not have been part of the video. The video would have been similar to an ODS without the rotor.

Vibration Institute

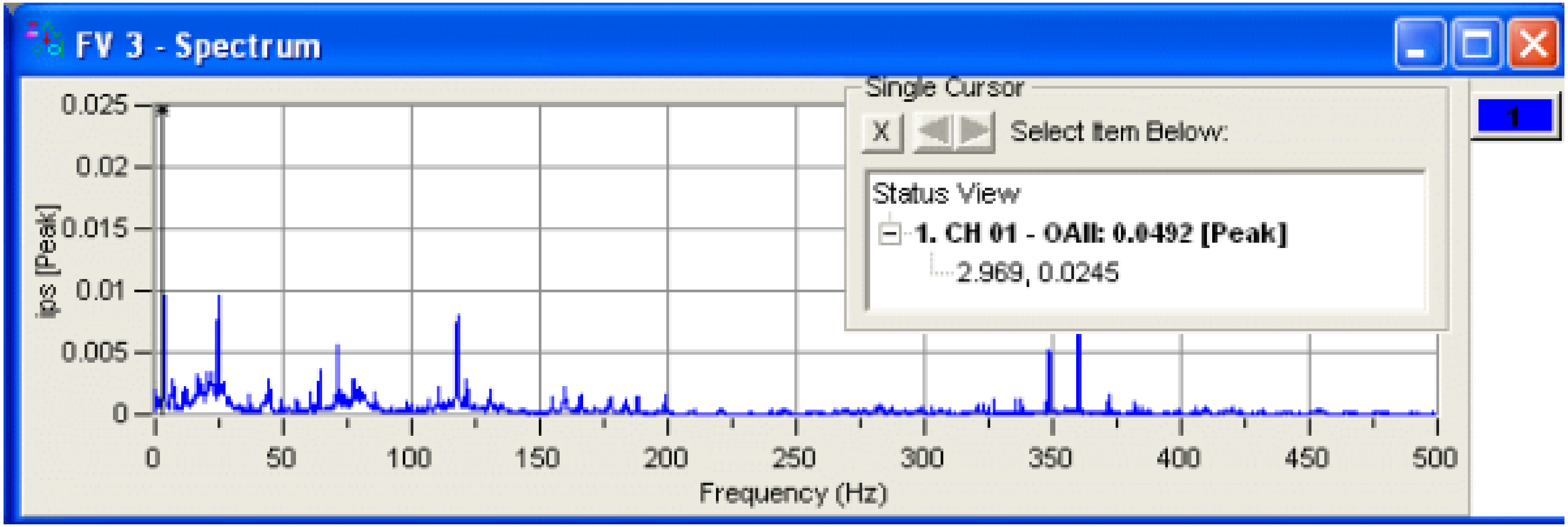
Banbury Mixer

Motor Bearing Race

Banbury Mixer

DC Motor & Gear Box

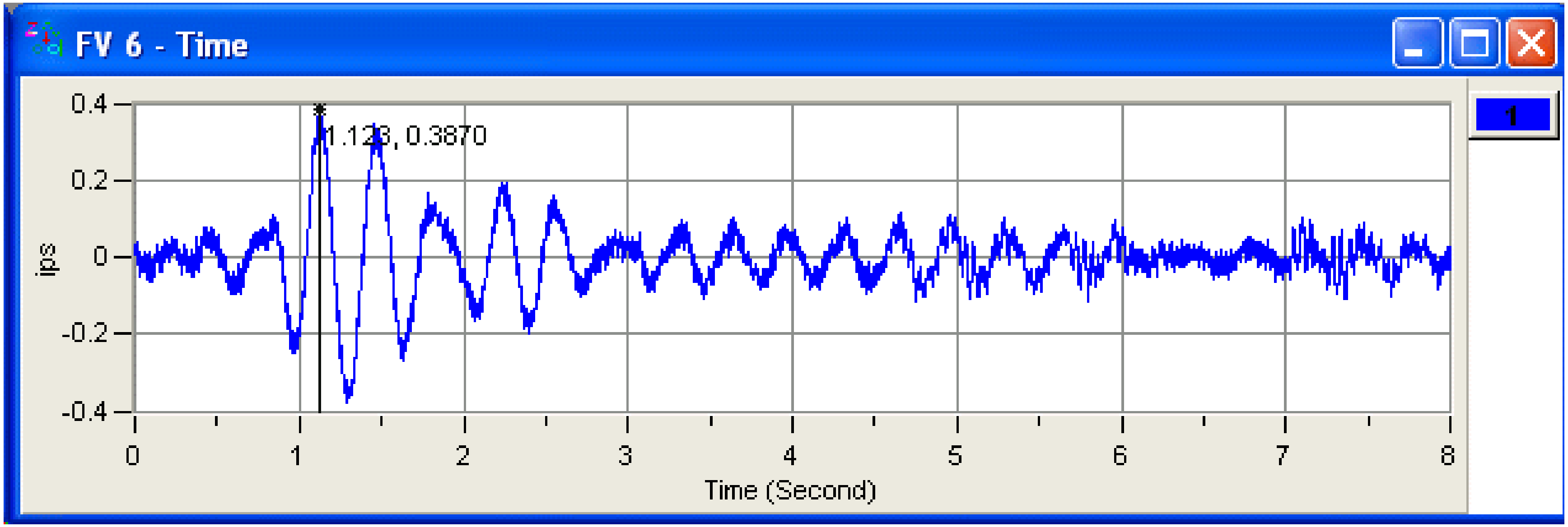
Banbury Mixer Case Study



FFT of Motor Vibration (Horizontal) indicates low vibration level (0.025 ips) dominated by 4x Mixer Frequency (2.96 Hz).

Vibration Institute

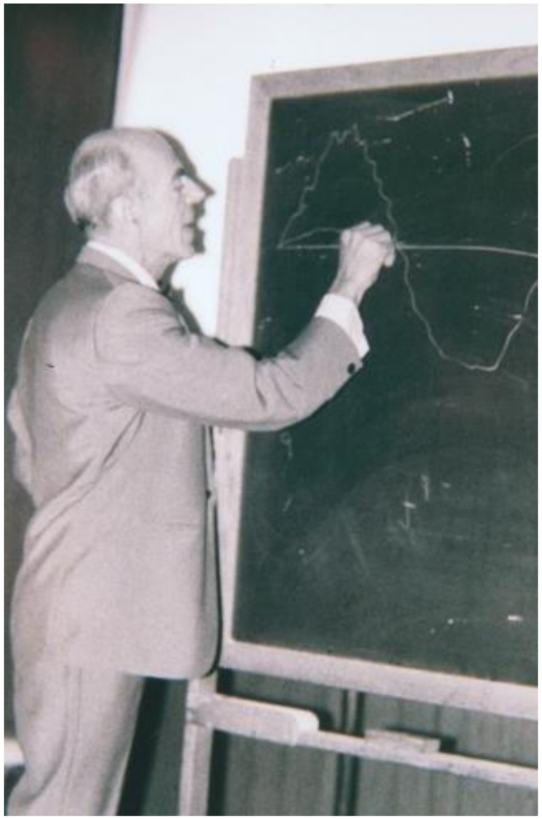
Banbury Mixer Case Study



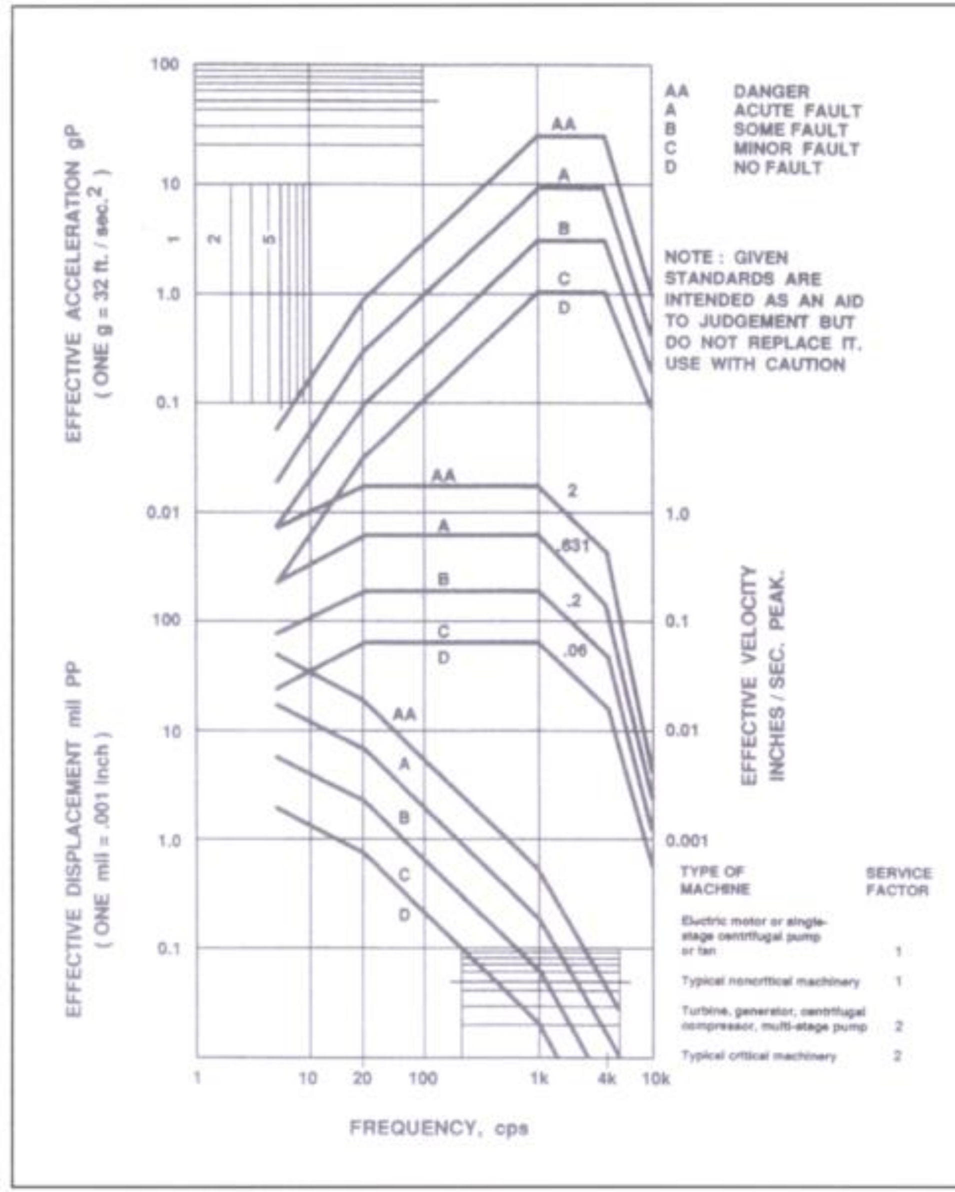
Waveform during Gate Opening approaches 0.40 ips, sometimes reaching 0.60 ips. Response @ 2.7 Hz.

Vibration Institute

Low Frequency Vibration Severity Criteria – Blake Chart 1972



Michael Blake (Original Founder of VI)



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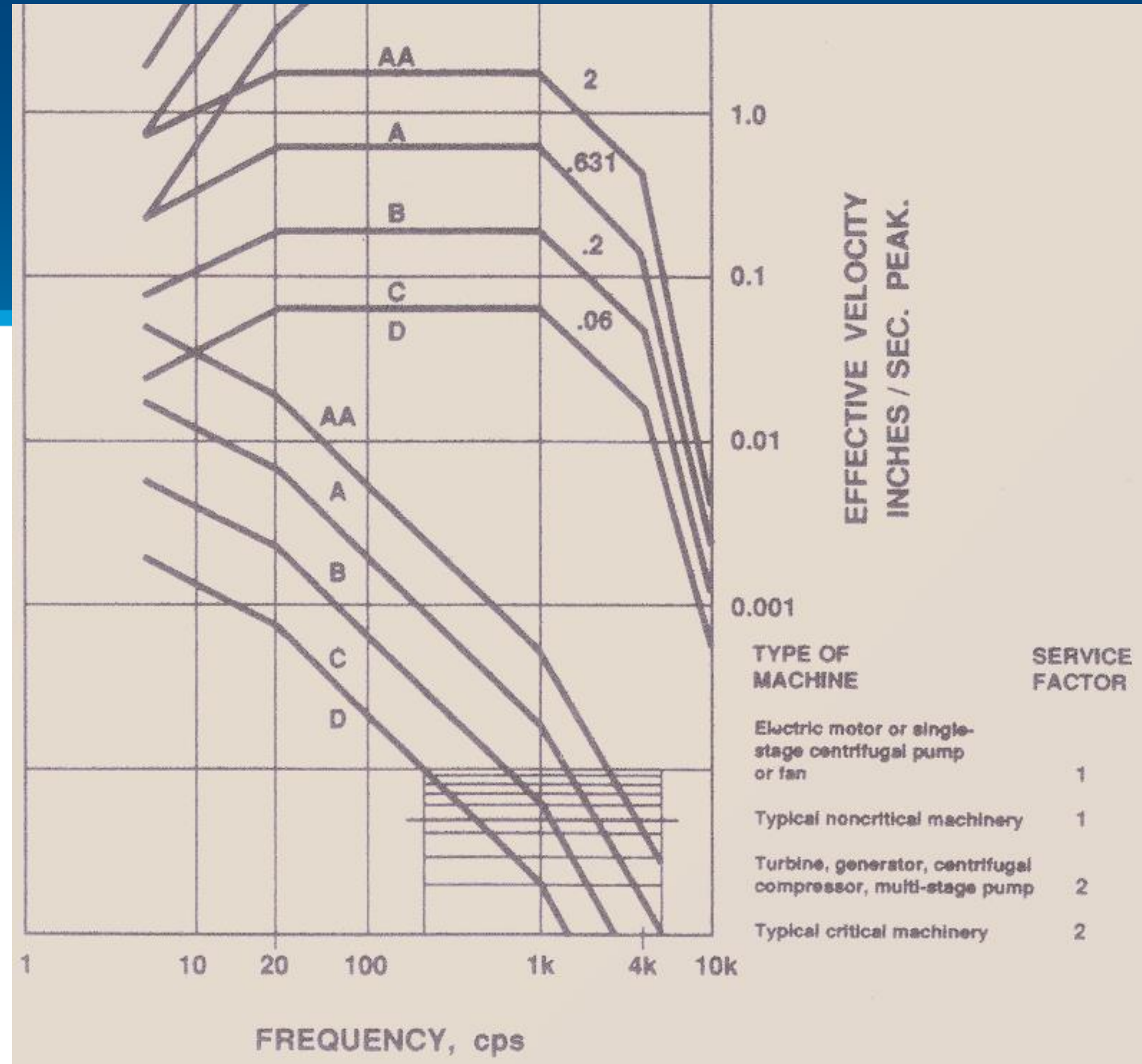
Low Frequency Vibration Severity Criteria – Blake Chart 1972

AA	DANGER
A	ACUTE FAULT
B	SOME FAULT
C	MINOR FAULT
D	NO FAULT

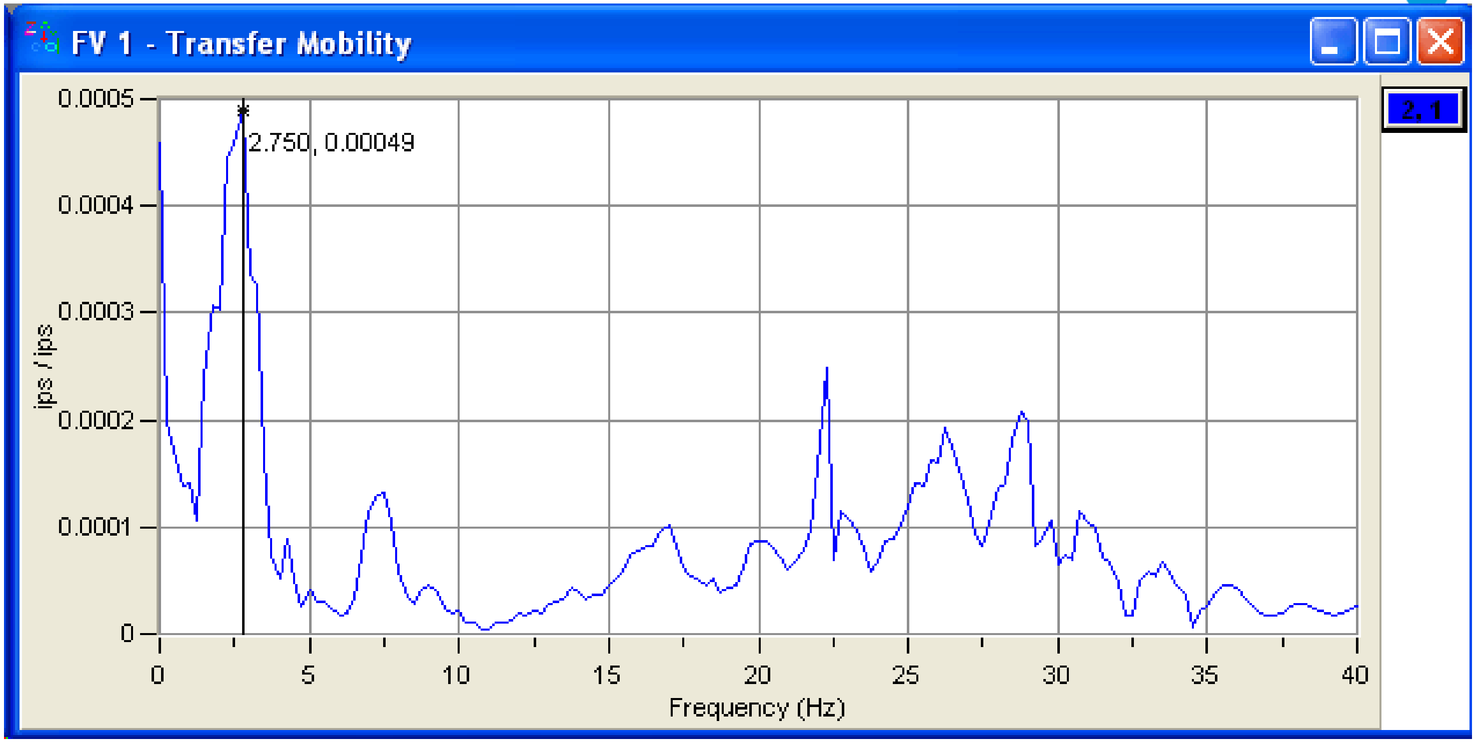
NOTE : GIVEN STANDARDS ARE INTENDED AS AN AID TO JUDGEMENT BUT DO NOT REPLACE IT. USE WITH CAUTION



A Line @ 5 Hz; V < 0.35 ips; Critical Equipment has Service Factor = 2

A Line @ 20 Hz – 1 kHz; V < 0.63 ips

Banbury Mixer Case Study



Natural Frequency Test of Motor Support Structure identifies fn ~ 2.7 Hz.

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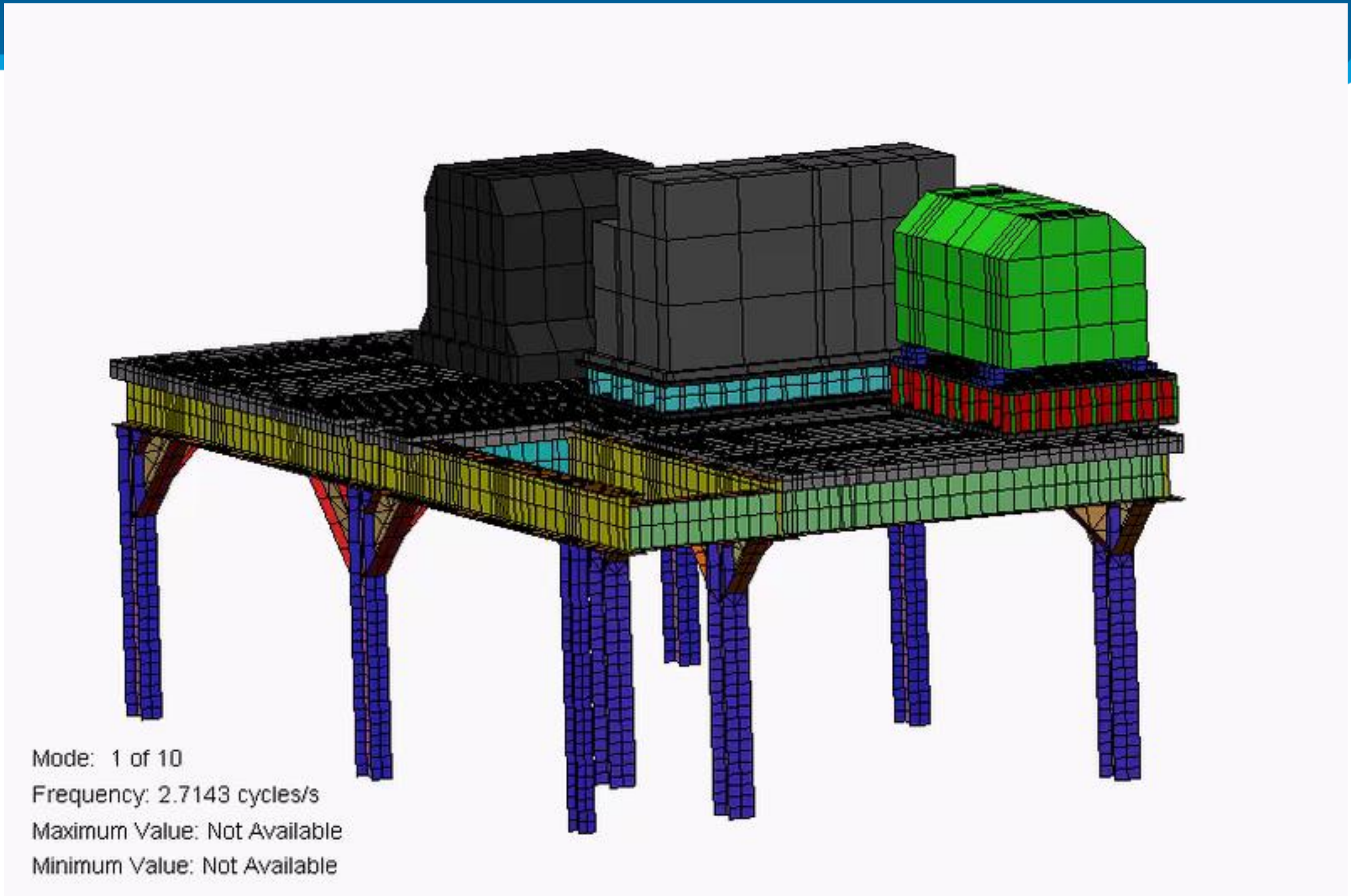
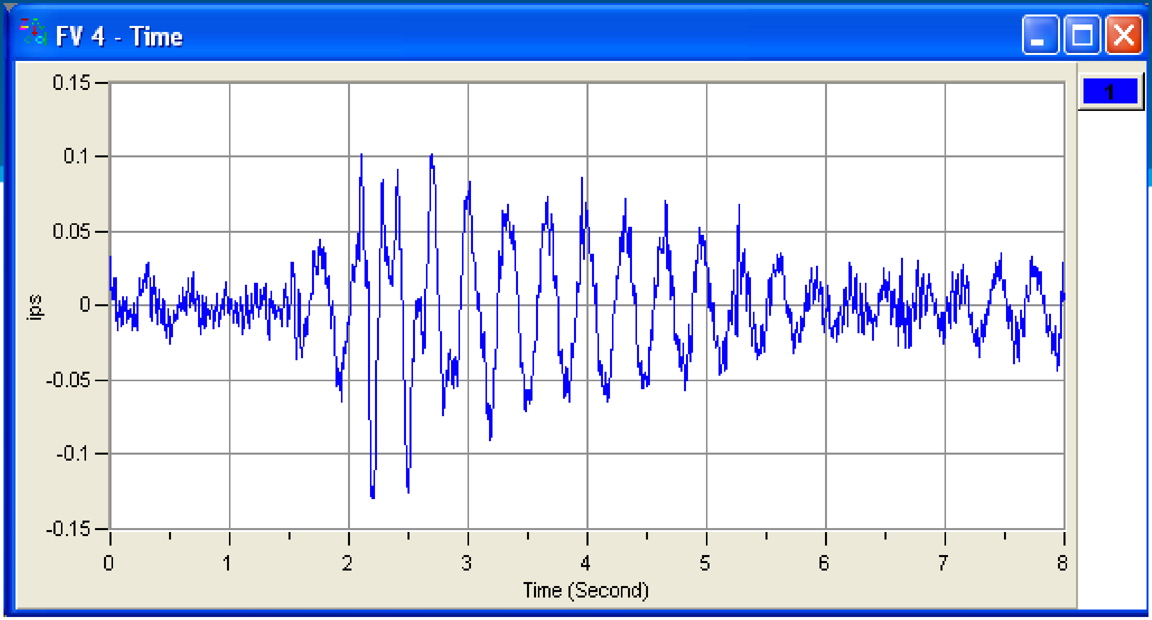
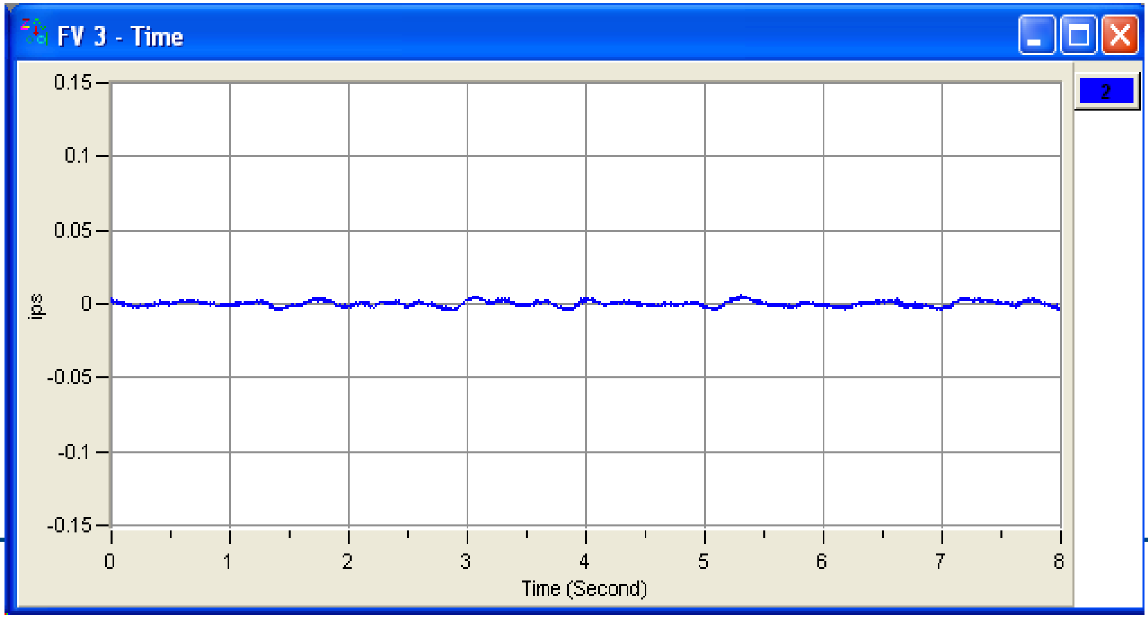
Banbury Mixer Case Study



FEA shows mode shape of Very Flexible Support System.

Vibration Waveforms





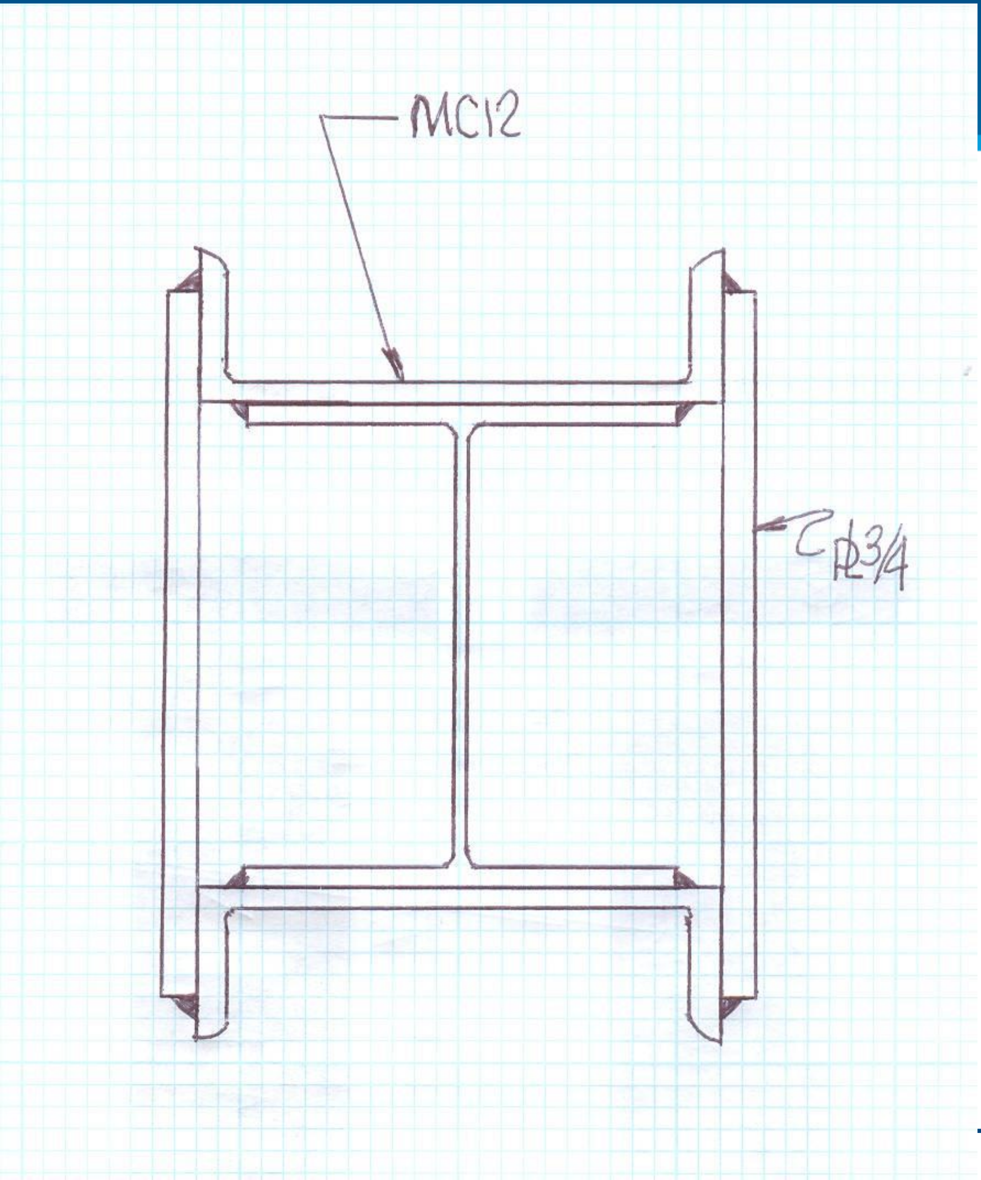
Waveform @ Top of Column versus Waveform @ Bottom of Column

Vibration Institute

Modification Objective

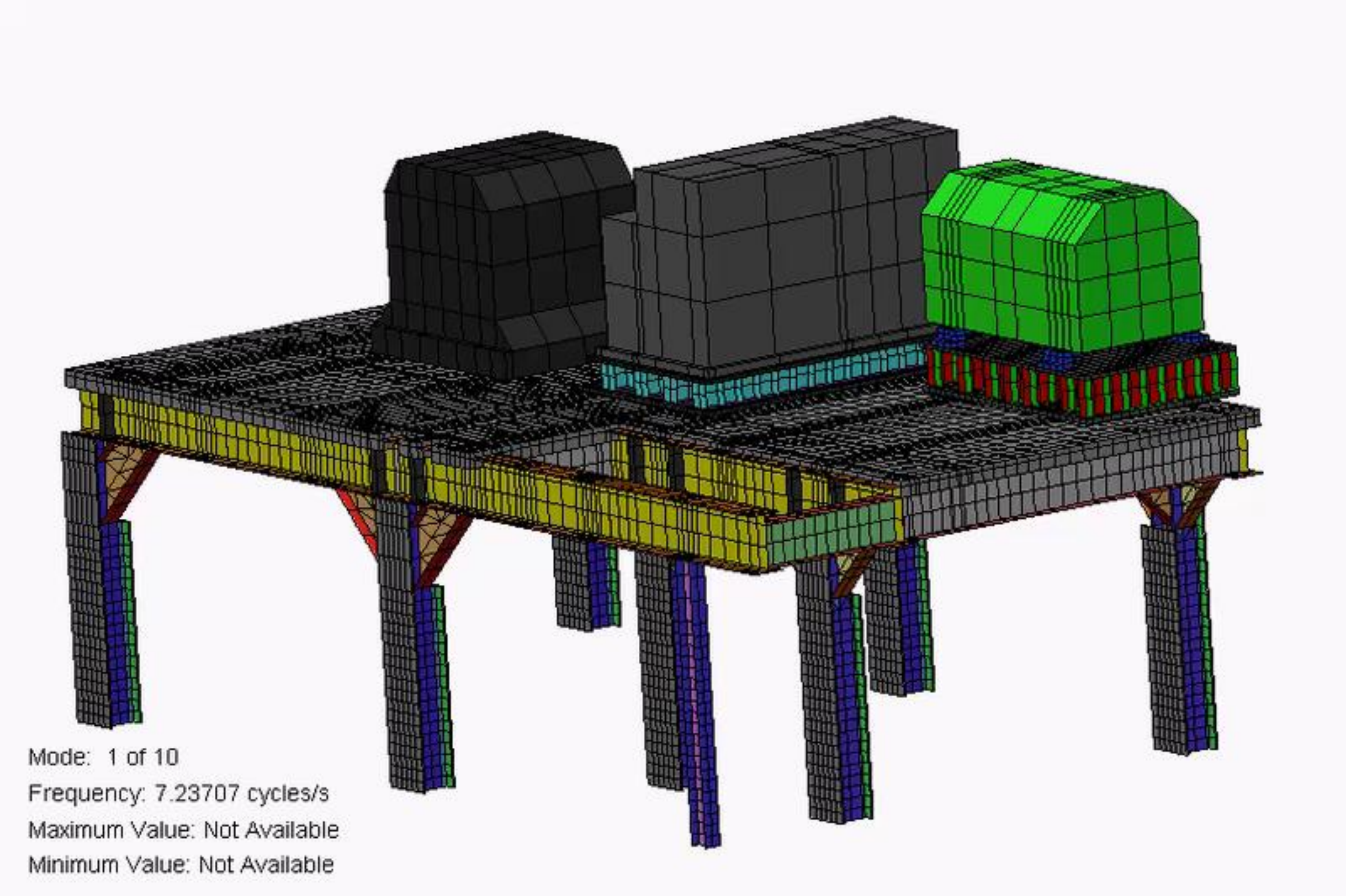
- Current Natural Freq ~ 2.7 Hz
- Increase as much as possible without getting close to Motor Speed (12 – 13 Hz; 720 – 780 rpm)
- Target Natural Freq ~ 7.5 Hz; ratio = 7.5/12 = 0.63

Modification Try #1



- MC12 Channels welded to Exist Column Flanges
- Plate welded to Flanges of MC12
- Cover Plate(s) @ Top of MC12 to Prevent Buildup of Material between Exist Col & Plate

Modified Column



Natural Frequency increases to 7.3 Hz

Close to Objective

Basics of Signal Analysis

The DFT wants a record (data sample) to start and finish with a value of zero (0.0).

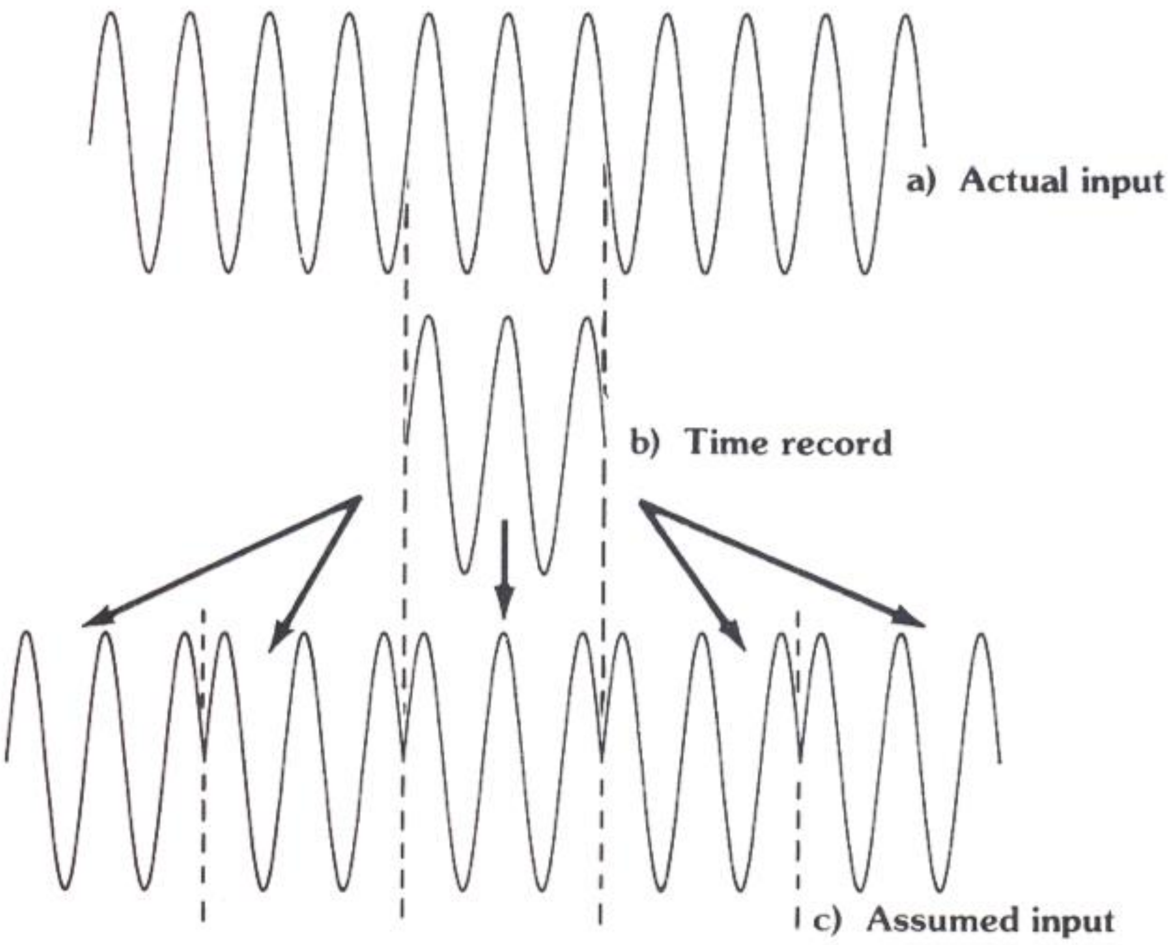
For most real signals, this is not the case. If an FFT is performed on a raw signal that starts and ends with a value other than 0.0, fictitious peaks will occur in the spectrum that are not real (picket fencing).

For this reason, data conditioning windows are typically applied to the raw data prior to performing the FFT.

The Hanning Window is most commonly used to acquire Vibration Data.

Vibration Institute

FFT Algorithm – Single Sample



FFT is a Batch Process

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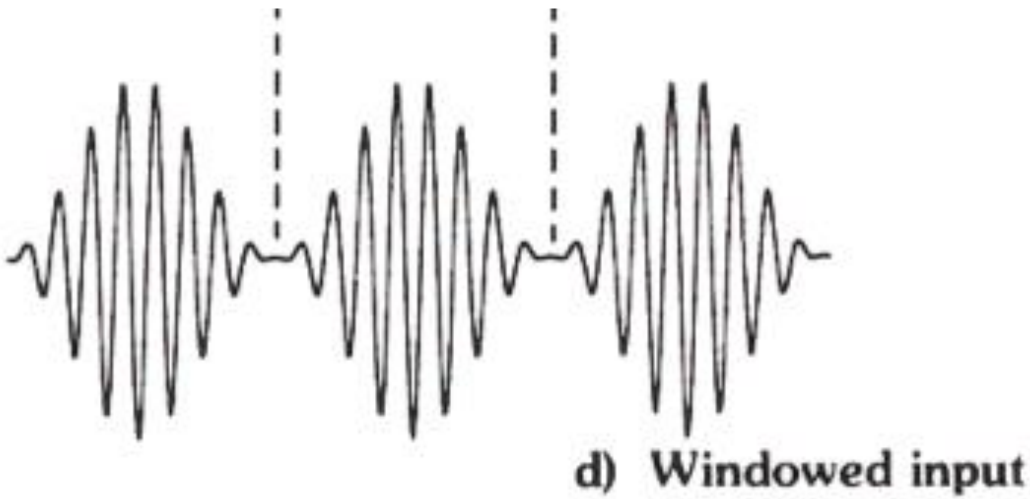
FFT Algorithm w/Hanning Window



Windowed Data starts and finishes @ 0.0 for each sample.

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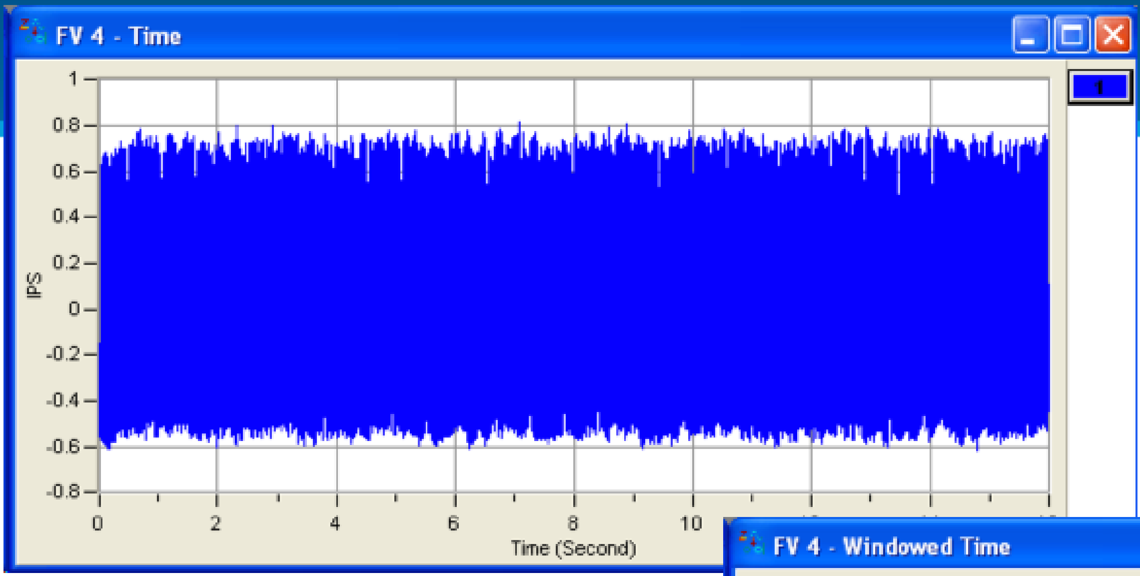
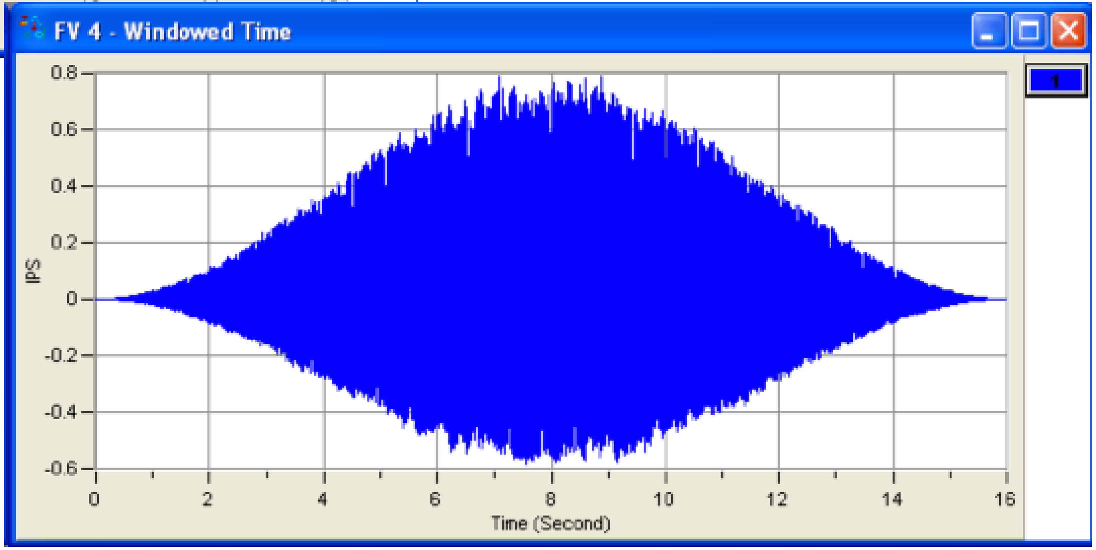
Hanning Window



Raw Data & Windowed Data



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Window Selection

Window	Purpose	Amplitude Uncertainty	Window Factor (WF)
Uniform	impact tests	56.50%	1.0
Hanning	fault analysis	18.80%	1.5
Flat Top	condition evaluation	1.00%	3.8

$$\text{bandwidth} = \frac{\text{frequency span}}{\text{number of lines}}[\text{WF}]$$

(V/I) Resolution = 2x Bandwidth

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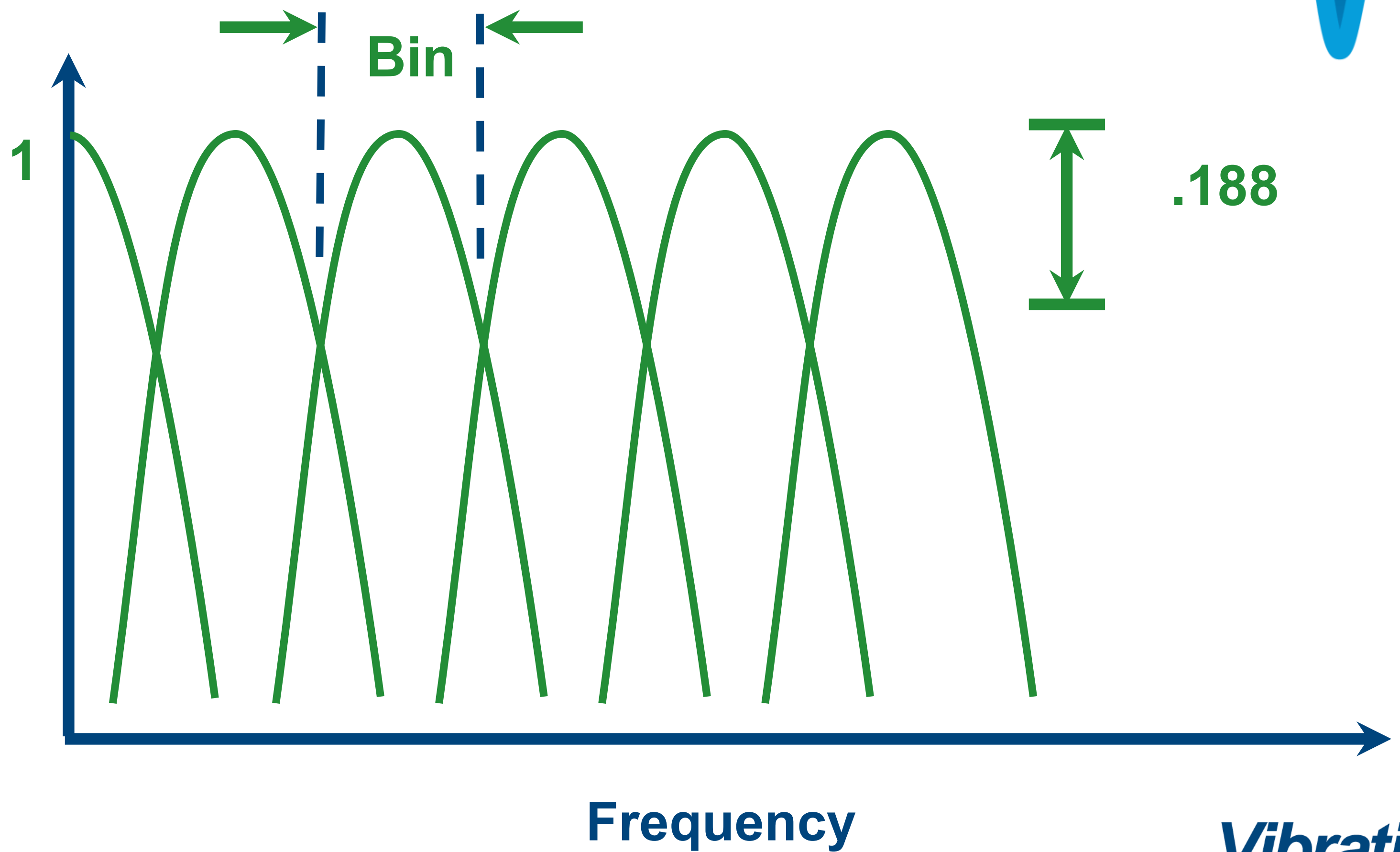
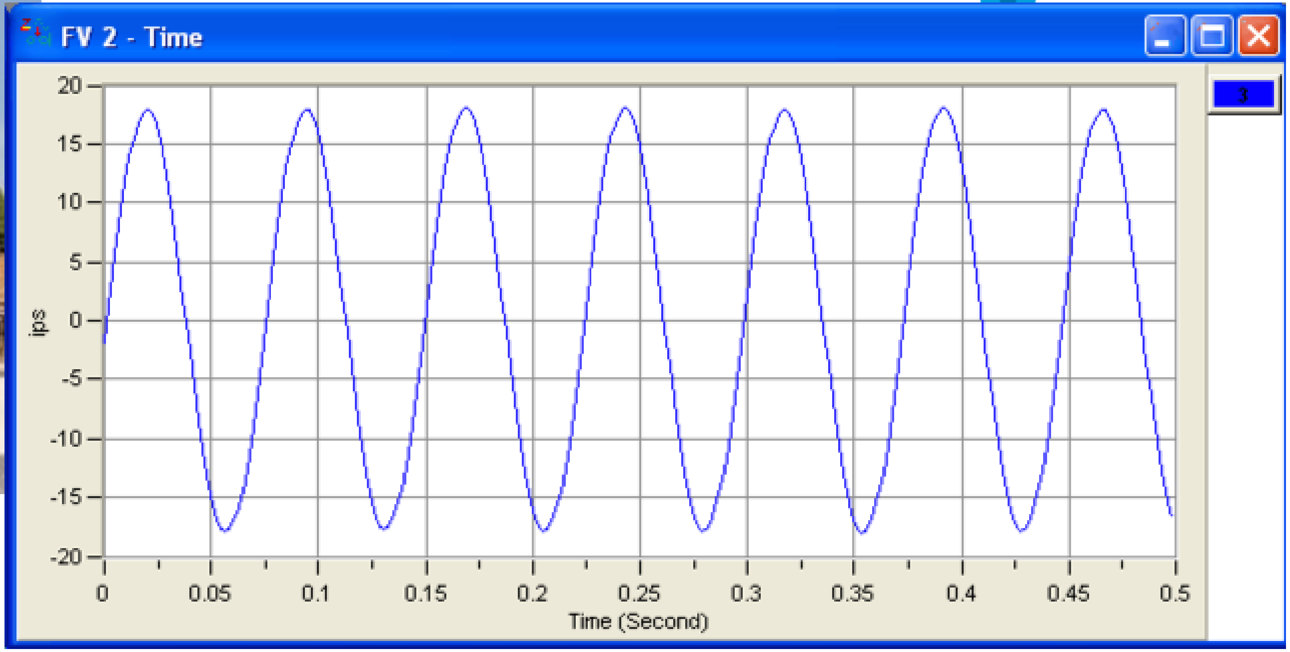
Hanning Bins



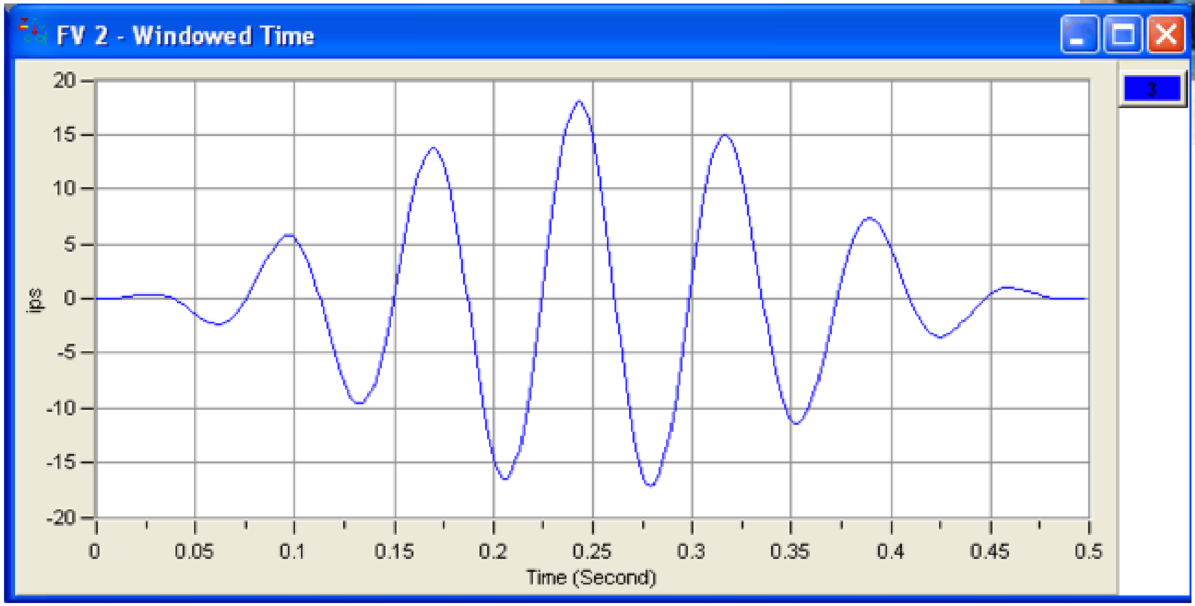
Hanning Window (Bin Centered Effects)



Peformance Test : Required Throw = 18.0 ips

Raw Waveform = 18.05 ips meets performance requirement

Hanning Window (Windowed Data)



Peformance Test : Required Throw = 18.0 ips

Windowed Waveform = 18.05 ips meets performance requirement

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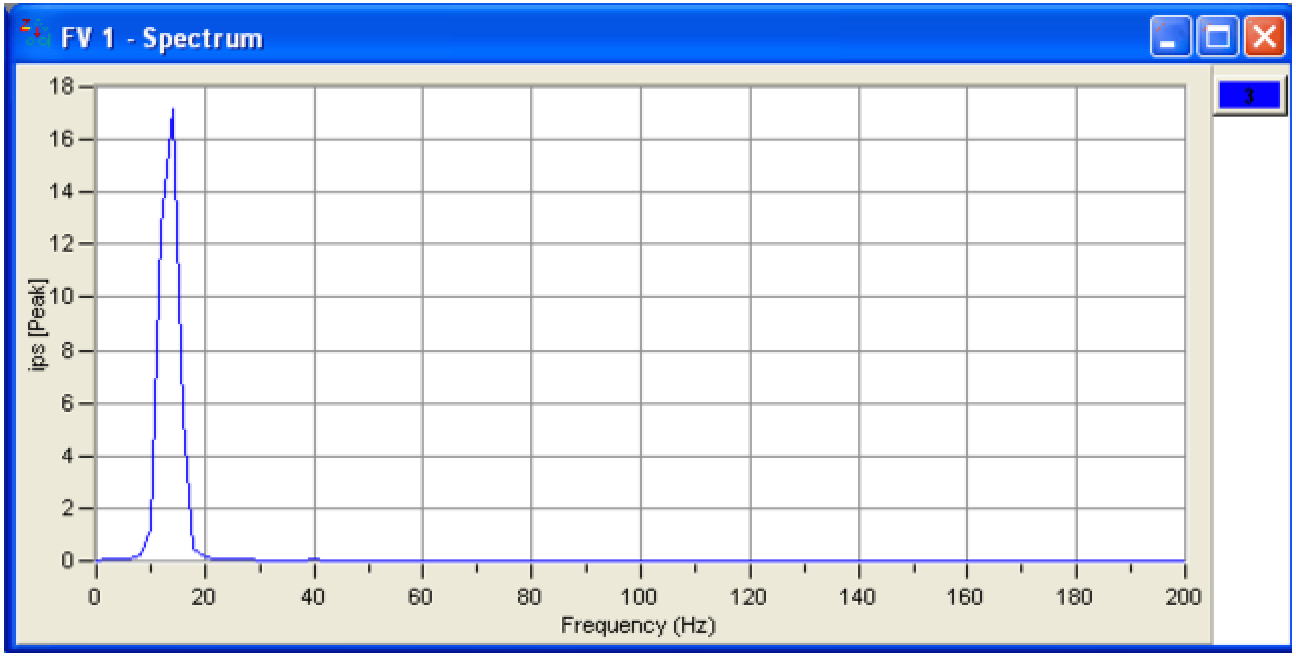
FFT – Non Bin Centered



F_{max} = 200 Hz
Lines = 100
Resolution = 2 Hz

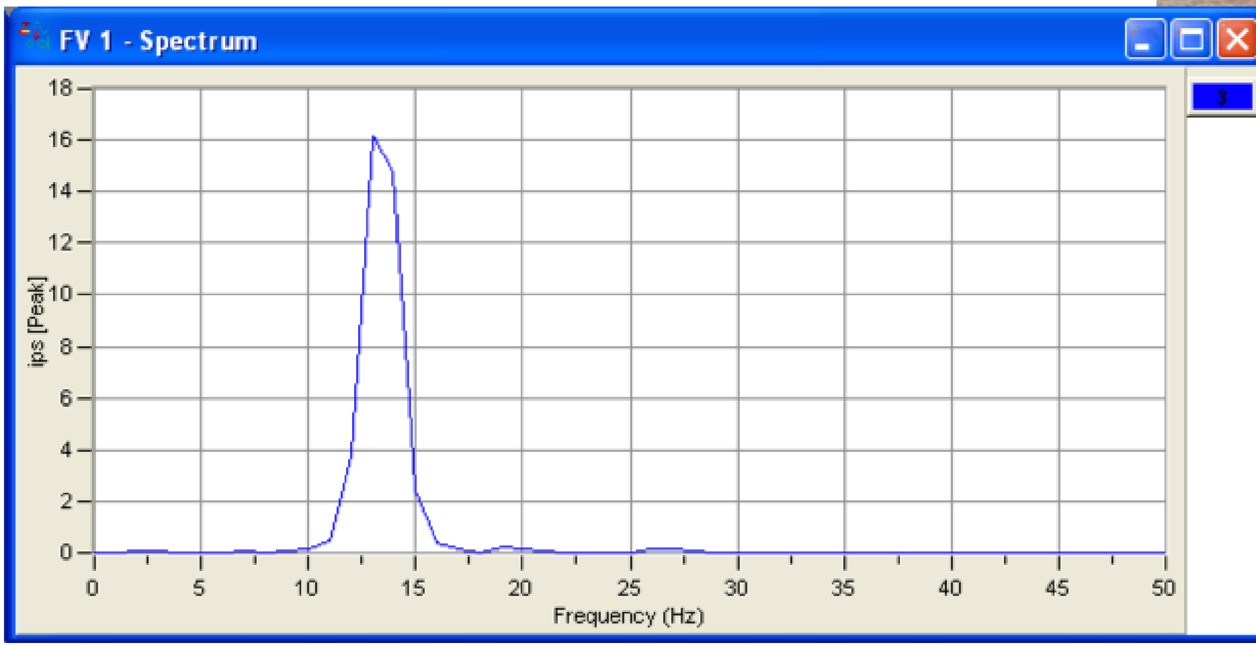
Freq Spectrum shows 17.14 ips @ 14.0 Hz (Bin Center)

Spectrum understates Max Vibration by 0.91 ips, Actual Speed = 13.4 Hz

FFT Windowed V/Actual V = 17.14/18.05 = 0.95

FFT values = 12.57 ips @ 12.0 Hz, 17.14 ips @ 14.0 Hz and 5.52 ips @ 16.0 Hz

FFT Non Bin Centered



NOTE: Many Analyzers have the capability to estimate the actual Peak. [PEAK LOCATE FUNCTION]

Consider FFT for Fmax = 50 Hz, N = 50 lines

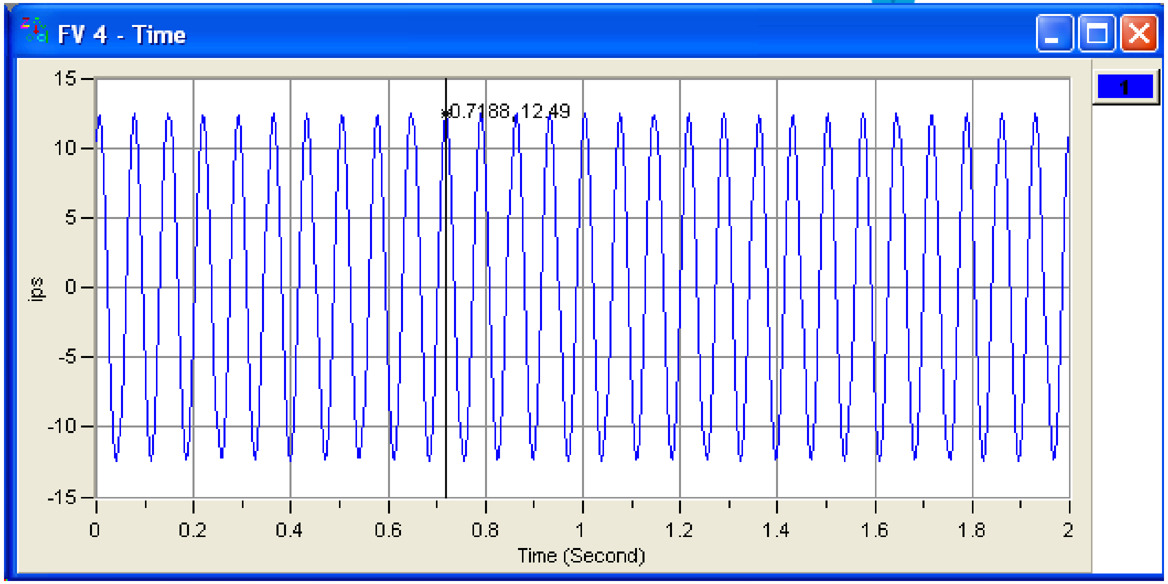
Freq Spectrum shows 16.19 ips @ 13.0 Hz (Bin Center)

Spectrum understates Max Vibration by 1.86 ips (16.19/18.05 = 0.90)

Actual Speed = 13.4 Hz

Vibration Institute

Hanning Window (Bin Centered Effects) Another Screen Example



Peformance Test : Required Throw = 12.5 ips

Raw Waveform = 12.49 ips meets performance requirement

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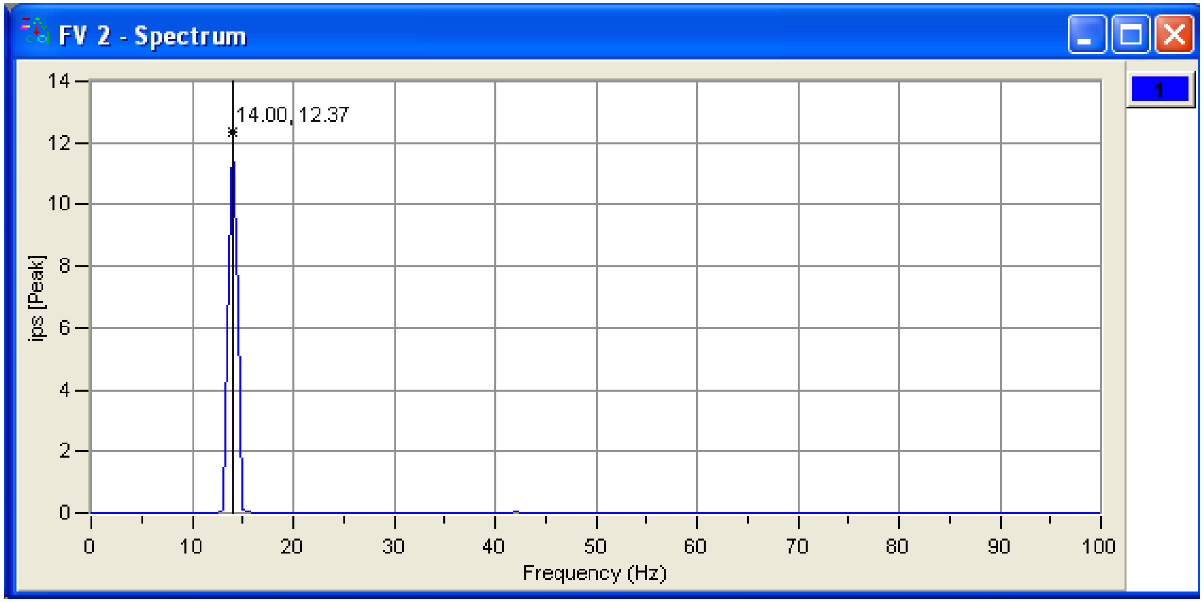
FFT – Non Bin Centered



F_{max} = 100 Hz
Lines = 200
Resolution = 0.5 Hz

Freq Spectrum shows 12.37 ips @ 14.0 Hz (Bin Center)
Actual Speed = 14.032 Hz
FFT Windowed V/Actual V = 12.37/12.49 = 0.99
Bin Freq Range = 13.75 Hz - 14.25 Hz

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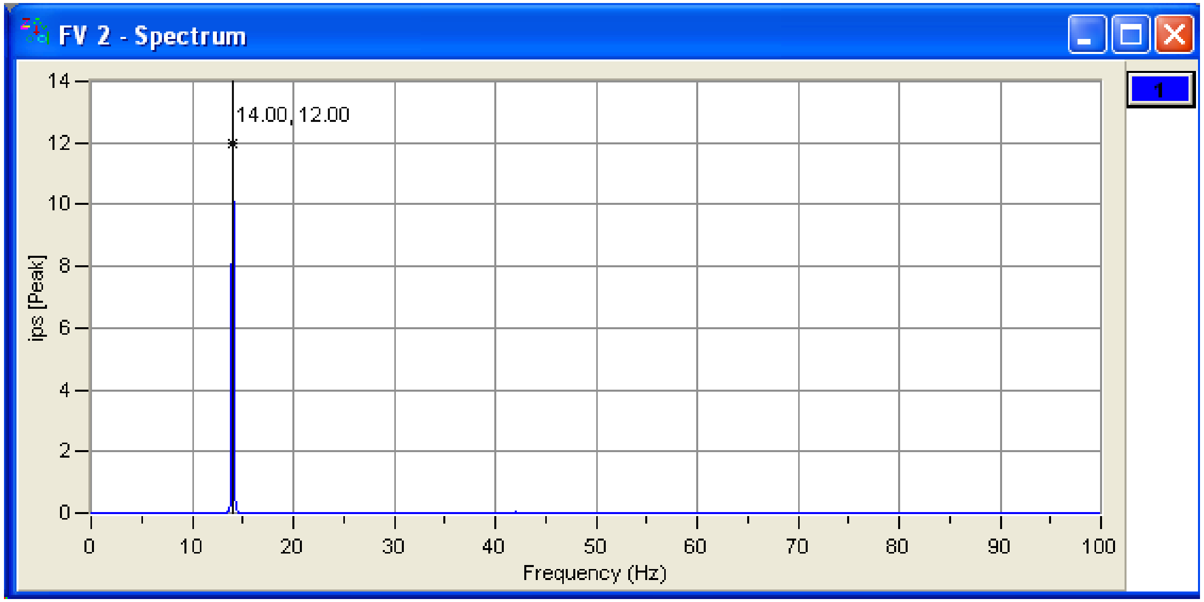
FFT – Non Bin Centered



F_{max} = 100 Hz
Lines = 400
Resolution = 0.25 Hz

Freq Spectrum shows 12.00 ips @ 14.0 Hz (Bin Center)
Actual Speed = 14.032 Hz
FFT Windowed V/Actual V = 12.00/12.49 = 0.961
Bin Freq Range = 13.875 Hz - 14.125 Hz

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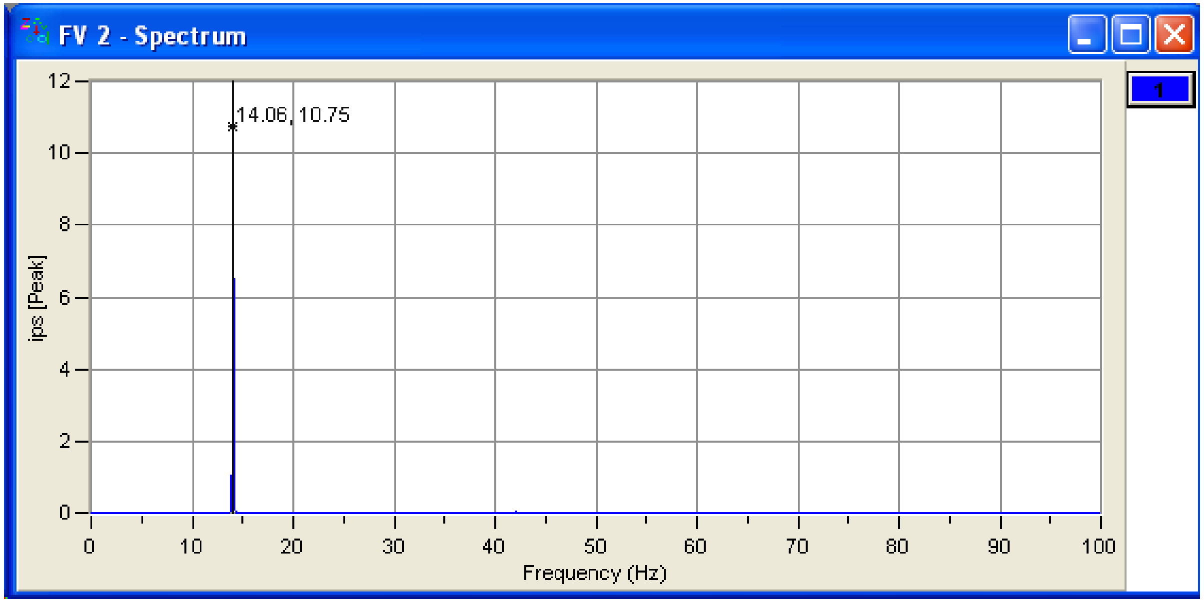
FFT – Non Bin Centered



F_{max} = 100 Hz
Lines = 1600
Resolution = 0.0625 Hz

Freq Spectrum shows 10.75 ips @ 14.0625 Hz (Bin Center)
Actual Speed = 14.032 Hz
FFT Windowed V/Actual V = 10.75/12.49 = 0.861
Bin Freq Range = 14.3125 Hz - 14.09375 Hz

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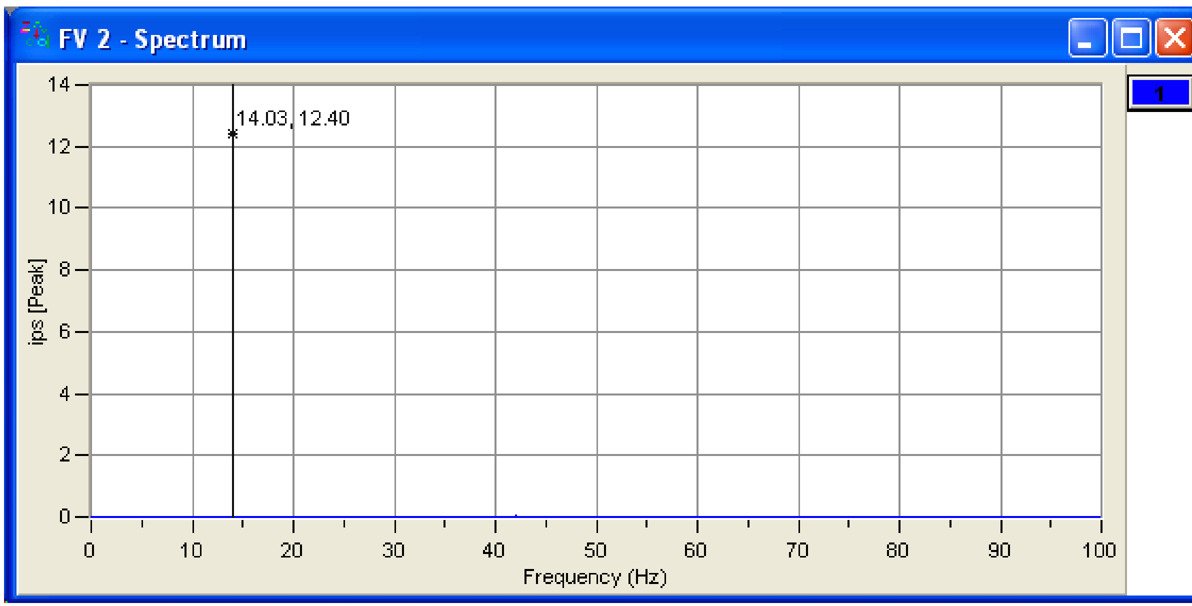
FFT – Non Bin Centered



F_{max} = 100 Hz
Lines = 3200
Resolution = 0.03125 Hz

Freq Spectrum shows 12.40 ips @ 14.03125 Hz (Bin Center)
Actual Speed = 14.032 Hz
FFT Windowed V/Actual V = 12.4/12.49 = 0.993
Bin Freq Range = 14.015625 Hz - 14.046875 Hz

Vibration Institute