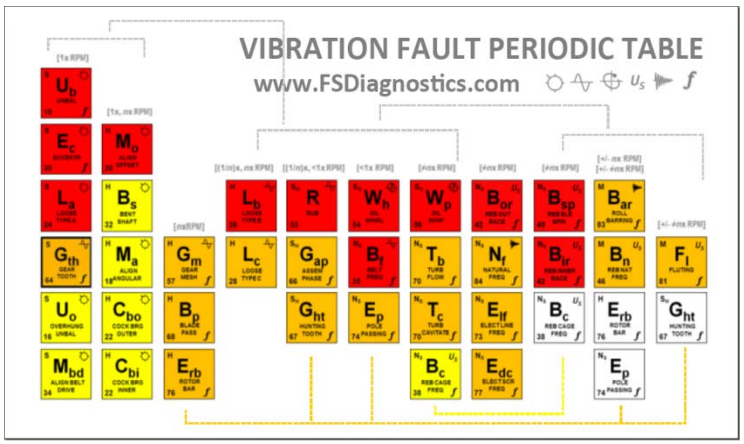
VIBRATION ANALYSIS CASE HISTORY



Dan Ambre is the founder of Full Spectrum Diagnostics, (June of 2000) providing the finest in Machinery Condition Monitoring, Specialized Analyses, and Vibration Training for Industrial, Manufacturing, Aerospace, and Automotive applications.

Dan is a graduate of ***The University of Iowa*** with a Bachelor's degree in Mechanical Engineering, and has completed additional graduate level course work in Engineering Dynamics from ***The University of Illinois at Chicago***, and ***Florida Atlantic University***.

Dan has over 30 years experience in the aviation and aerospace industries with a background in Vibration & Acoustic testing, Dynamic analysis of high speed rotor systems, Experimental Modal, and Finite Element Analysis. His consulting vibration experience base comes from positions at Sundstrand Aviation Corporation, Pratt & Whitney (United Technologies Corporation), and Technical Associates of Charlotte.

Dan is a Certified Level III / CAT IV. Vibration Training Instructor and a registered Professional Engineer in Minnesota.

The TEST ARTICLE

Machine Class: Pulp Mill Boiler Recovery Fan
Rotating Speed: 4,430 RPM
Overall Alarm: 0.400 ips

PROBLEM STATEMENT:

This machine was experiencing Elevated Vibration response at 1x RPM following an over-speed test of the turbine. The dominant response was an out-of-phase motion in the Horizontal Direction across the turbine bearing coupling with the gearbox.

PRELIMINARY ASSESSMENT (Historical):

1. Coupling Alignment Error (Horizontal Offset)
2. Natural Frequency near Turbine rotating speed.

RECOMMENDATIONS / OPTIONS:

- De-rate Fan operating speed to reduce vibration response (until next outage cycle).
- Perform (ODS) Phase Analysis Across Coupling.
- Perform Experimental Modal Analysis (EMA).
- Document Natural Frequencies & Access Feasibility of Speed Change and/or Structural Modification.

The NATURAL FREQUENCY & RESONANCE PHENOMENA

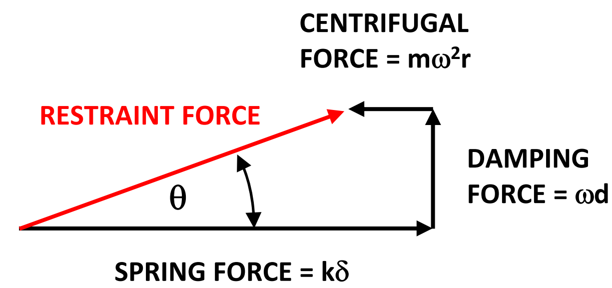
The Resonance condition creates a Phase response that is a function of the forces generated in a linear “balanced” system.

The **Physical** “structure” includes **Mass**, **Stiffness**, and **Damping**.

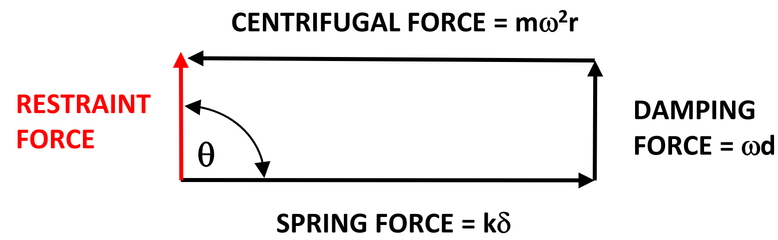
The **Dynamic** system includes motions in the form of **Displacement**, **Velocity** and **Acceleration**.

The **Forces** generated are an **Inertial** (centrifugal) force, a **Spring** (stiffness) force and a **Damping** (dissipation) force.

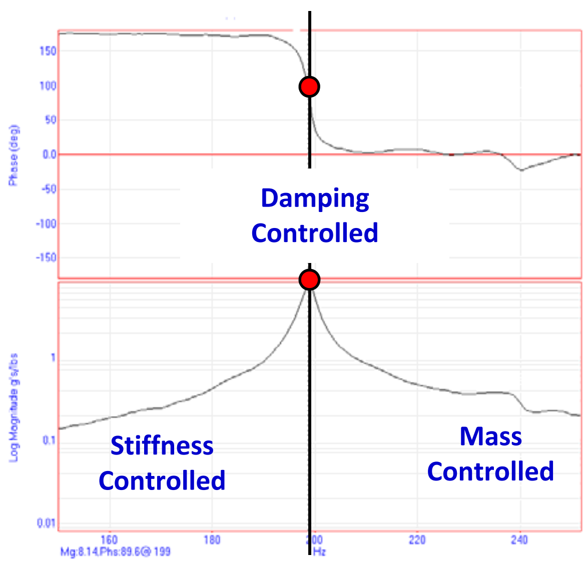
The simplified diagram to the right shows the balance in forces as this system transitions into a resonance condition. Note the initially small phase angle shifts 90 degrees at resonance and includes another 90 degree shift (approaching a total 180 degrees above resonance).

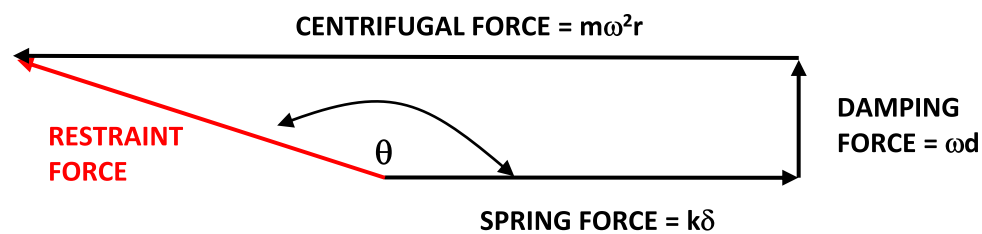


BELOW RESONANCE: STIFFNESS CONTROLLED



AT RESONANCE: DAMPING CONTROLLED





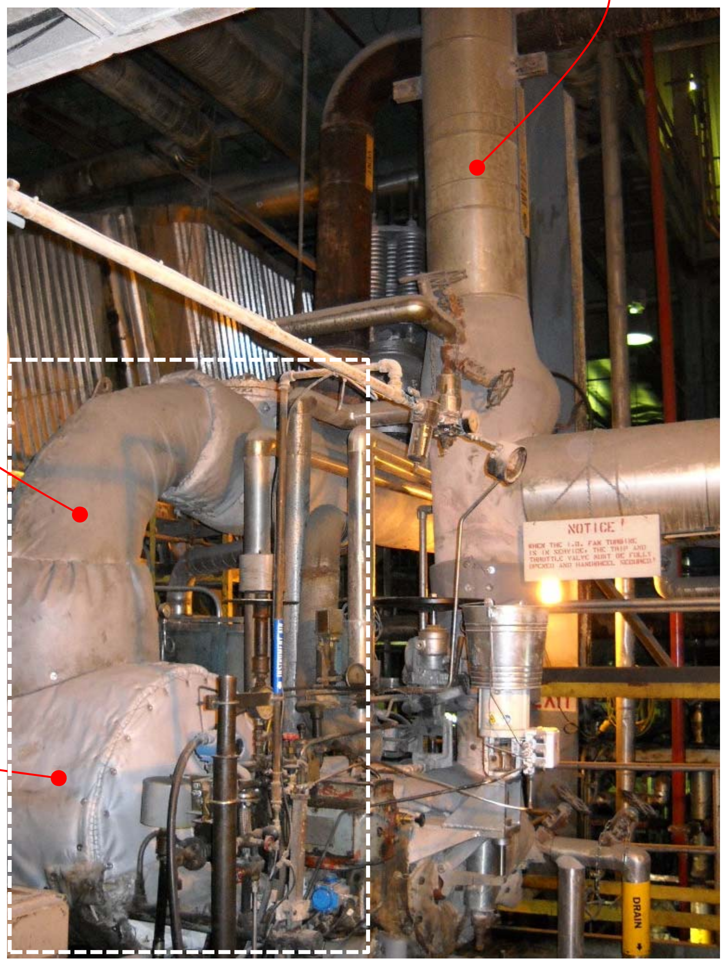
ABOVE RESONANCE: MASS CONTROLLED

Natural Frequency Testing: The Test Attributes

Turbine Exhaust Line (Insulated)

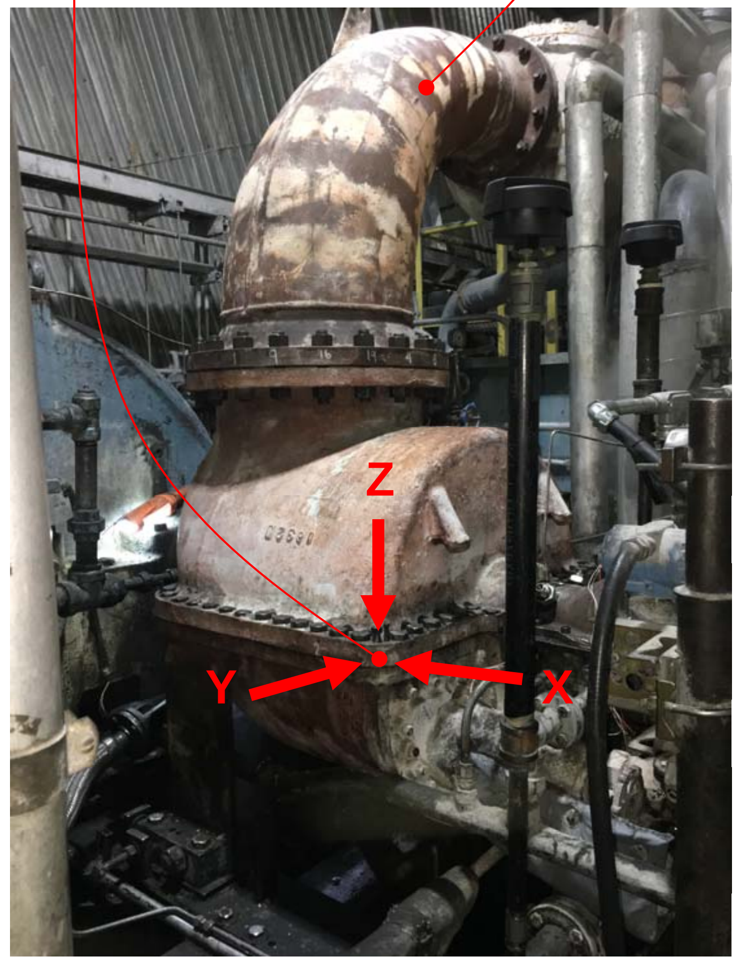
900# Steam Supply Line

Turbine



Turbine Exhaust Line

Impact Reference

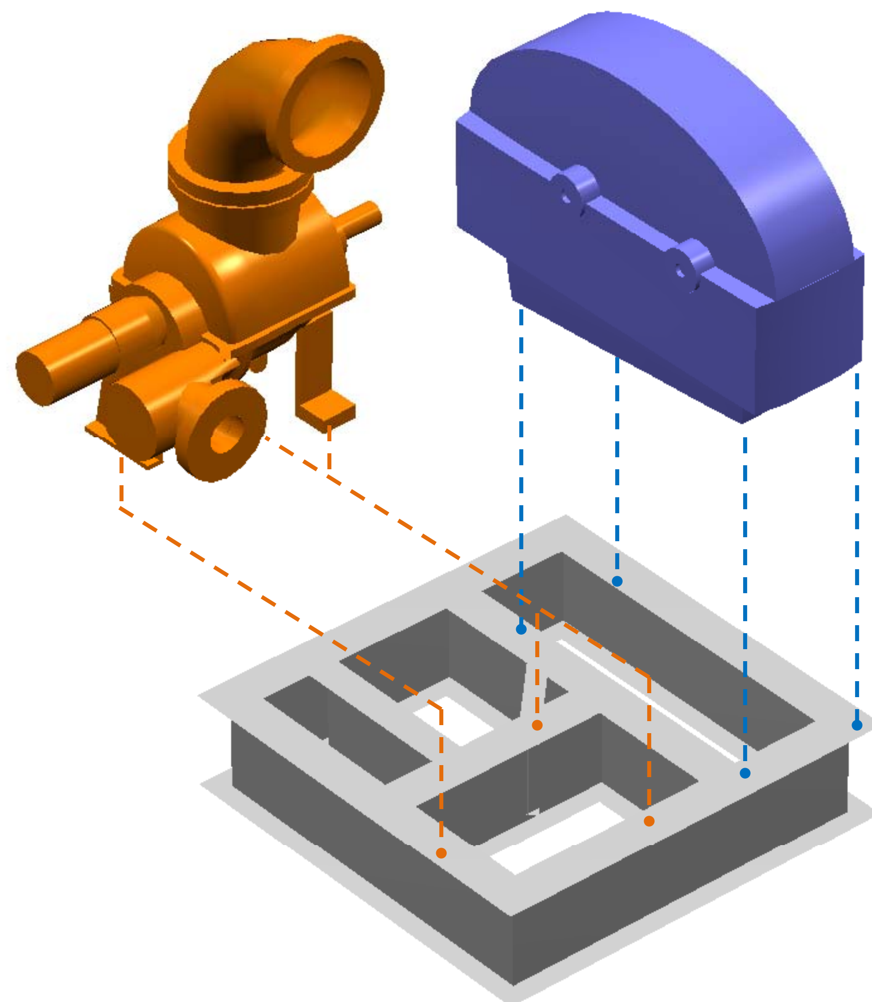


Natural Frequency Testing: The Structural Model

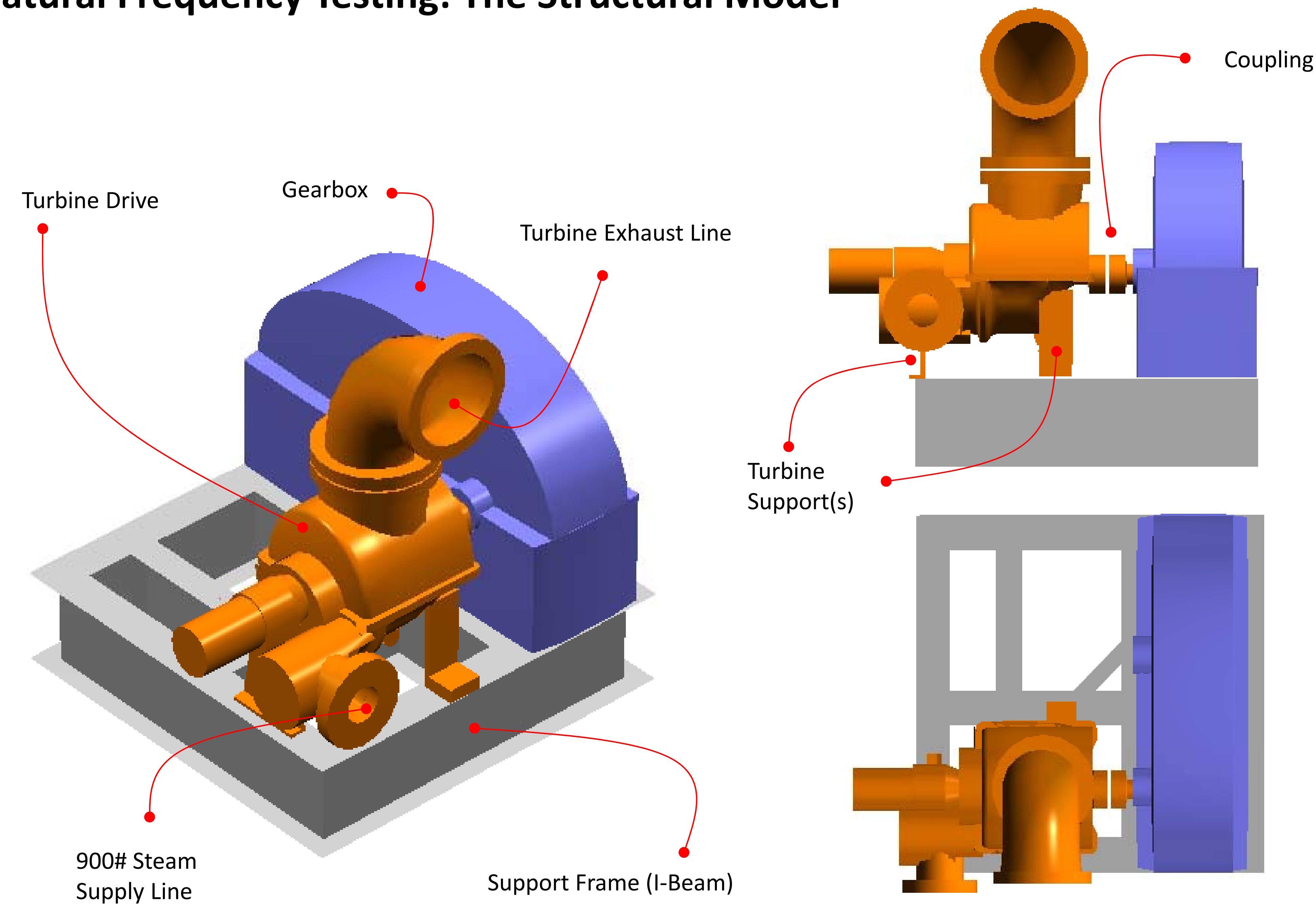
The Structural Model is constructed to approximate the physical structure. Remember that this model is **NOT** a Finite Element Analysis (FEA) model. The Experimental Modal Analysis (EMA) is a purely empirical analysis. That is, the results are a result of entirely "measured" test data.

Software drawing tools allow relatively quick construction of complicated structures. In this case, The turbine is constructed from cylinder and cube sub-structures. The base frame I-Beam structure was extruded from simple cross-section profiles. The gearbox was constructed from "re-sized" or stretched cylindrical and cube structures.

In this analysis, the 3-point turbine mount was important. These mounts are relatively flexible, but designed for a foundation (seismic) mount. Instead, the turbine was mounted on an elevated I-beam frame. This tends to increased the flexibility in the turbine mount by placing accentually "springs" in series.



Natural Frequency Testing: The Structural Model



Natural Frequency Testing: The Measurement Grid

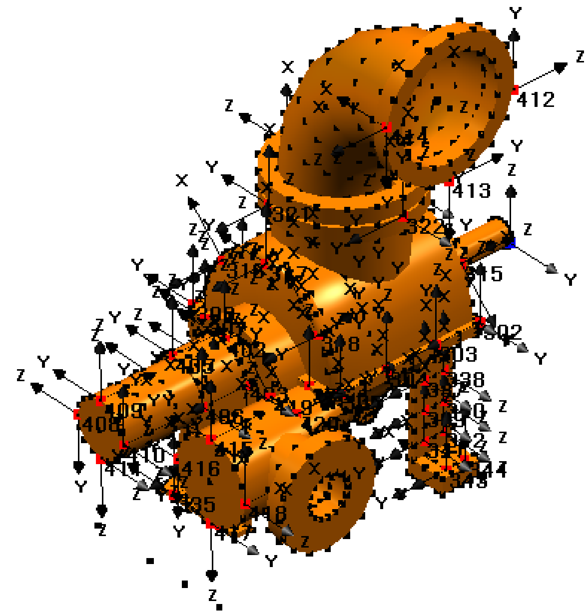
The model construction phase is a good time to consider the appropriate Measurement Grid for analysis.

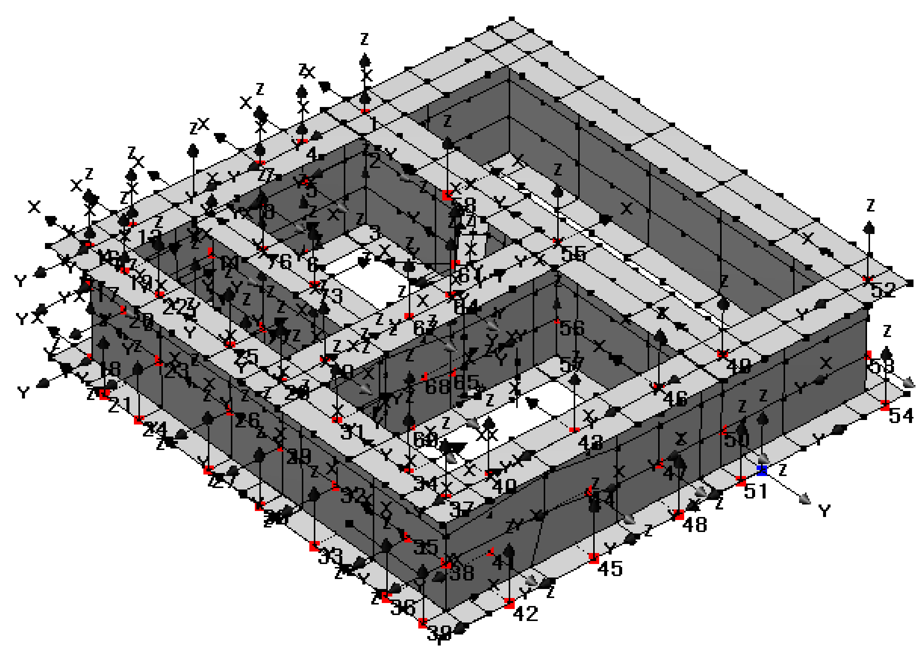
Measuring a Natural Frequency in a physical structure requires only one measurement.

Measuring a Mode Shape in a physical structure requires a enough resolution (measurement points) to resolve the shape.

Higher natural frequencies can be measured by adjusting the frequency range, but higher frequency (complexity) Mode Shapes requires additional physical measurements.

This measurement set was collected as a “Roving Transducer” test.   Separate sets of data were collected in each (x, y, z) measurement direction.

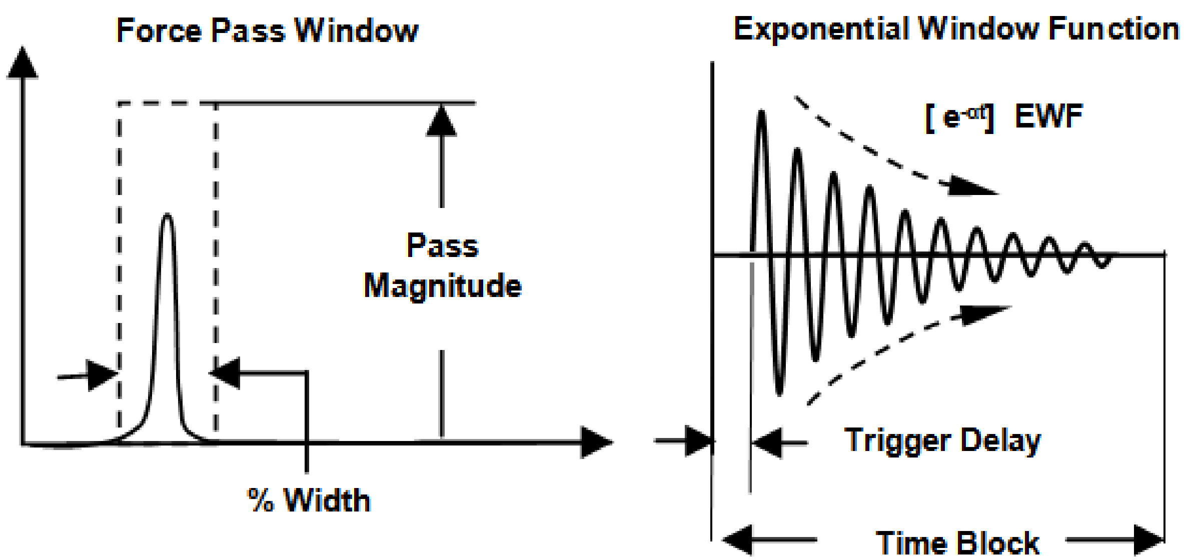




Experimental Modal Analysis: IMPULSE Method Testing

Experimental Modal Analysis accuracy relies on the application and capture of a single properly scaled impact from the test article. The size of the hammer (mass), the type of tip (hardness) used will determine the usable frequency range and the ability to detect lightly or heavily damped natural frequencies. The length of the time block will determine the frequency range in the analysis. The time block can be modified by changing the F_{MAX} and # of FFT Lines in the analysis setup.

The impact transient must be wholly contained within the time block, which will include a trigger delay and the "ring-down" of the transient response. For lightly damped systems the ring-down can be artificially forced to decay by use of an Exponential Window Function (EWF). The added damping should be considered when the final data is curve-fit and damping estimates are made. A Force Pass Window is a narrow band pass function used to reduce or eliminate noise in the Impact time waveform.



Experimental Modal Analysis: Instrumentation

Massive Structures:
Lower Frequency / Heavy Hammer / Soft Tip
Floors, Walls, Building Frames, Inertial Blocks & Pedestals

Intermediate Structures:
Mid-range Frequencies / Medium Hammer / Medium-Tip
Framework, Fabricated Machine Bases, Heavy Components and Sub-Assemblies.

Light-weight Structures:
High Frequencies / Small Hammer / Hard-tipped
Turbine Blades, Gears, Piping, Light-weight Components and Sub-Assemblies

The **Turbine Drive, gearbox, support frame and piping** was considered an intermediate to massive structure. The frequency range of interest was low with more "rigid-body" motions (translation & rotation). The best instrumentation choice was a larger-sized hammer with a soft rubber tip.

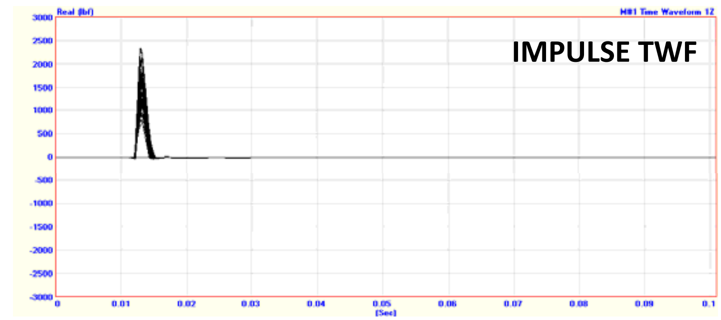
The transducer of choice was a everyday size (100 mv/g) tri-axial accelerometer with a magnetic mount. This testing was a "Roving Transducer Test".

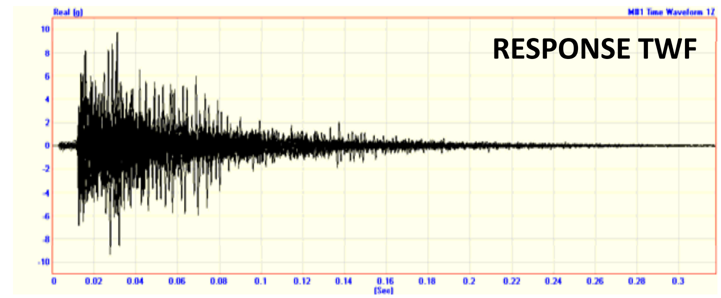
SPECIFICATIONS
Sensitivity: 100 mV/g
Frequency Response (±3dB): 60-390,000 CPM
Dynamic Range: ± 50 g, peak
Temperature Range: -58 to 250°F (-50 to 121°C)
Connector: 4 Pin J Connector

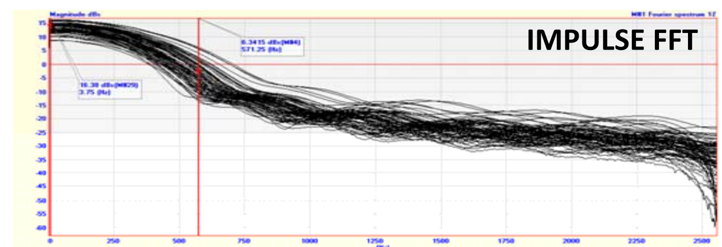
SPECIFICATIONS
Sensitivity: (±15%) 1 mV/lbf (0.23 mV/N)
Measurement Range: ±5000 lbf pk (±22240 N pk)
Hammer Mass: 12.1 lb (5.5 kg)

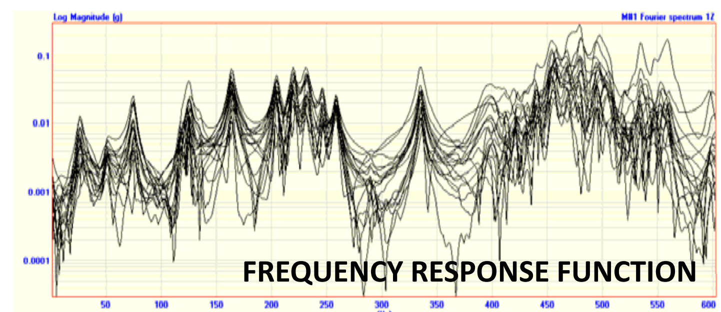
Experimental Modal Analysis: MEASUREMENT CRITERION

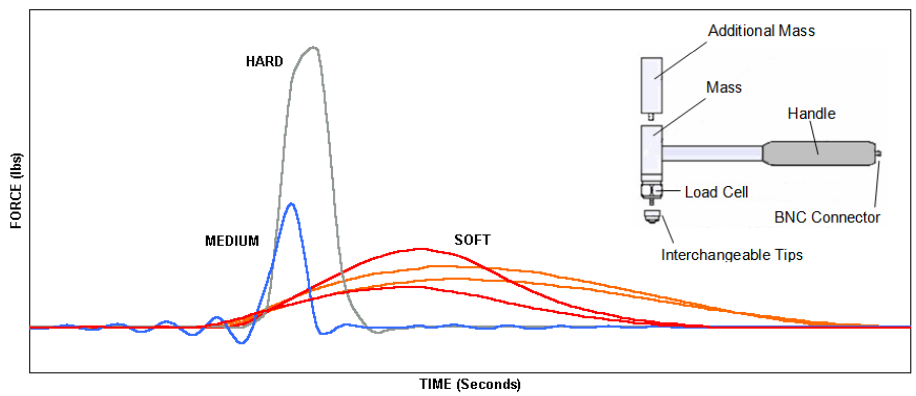
- ✓ SINGLE REPEATABLE IMPULSE TIME WAVEFORM
- ✓ RESPONSE "RING-DOWN" within the TIME BLOCK
- ✓ IMPULSE FFT Indicates USEABLE FREQUENCY RANGE
- ✓ FREQUENCY RESPONSE FUNCTION Reveals SUSPECT MODES



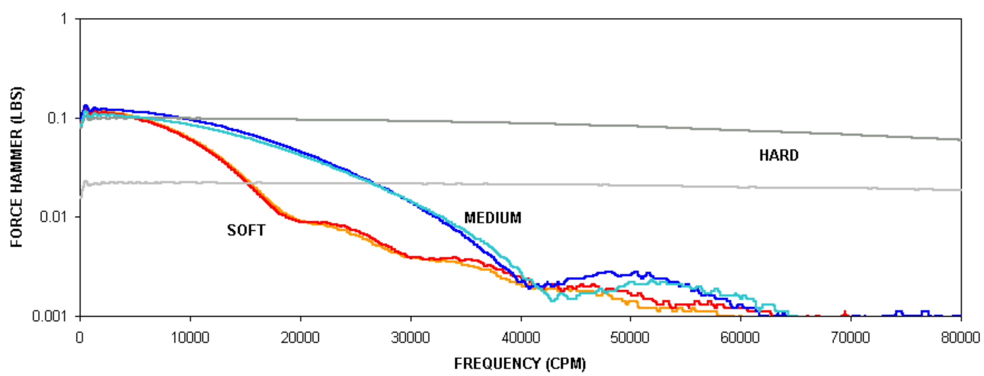








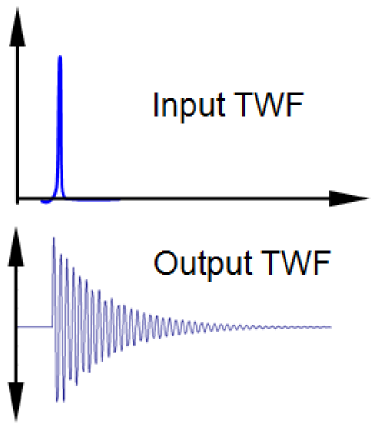
Time & Frequency Response for Various Tip Hardnesses

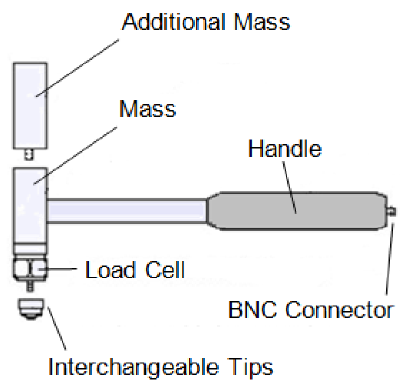


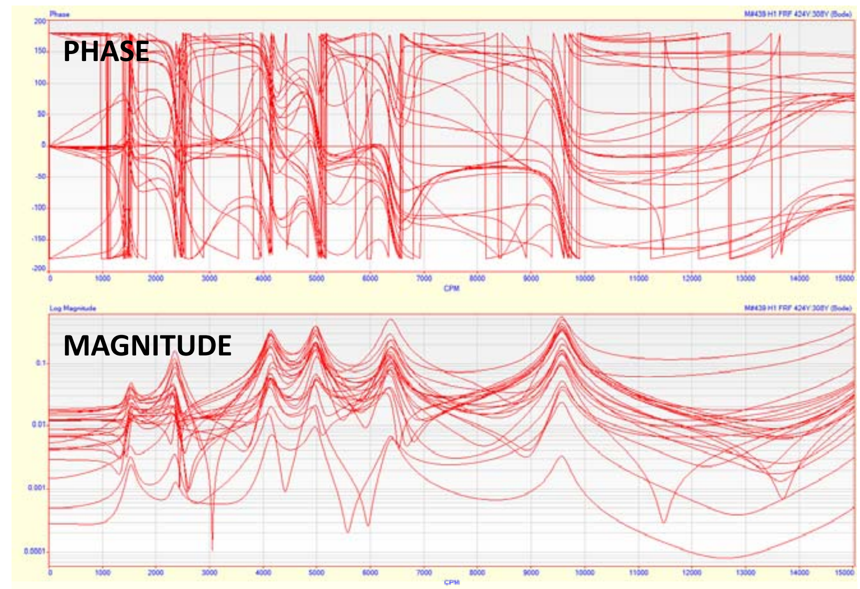
Experimental Modal Analysis: Acquisition Setup

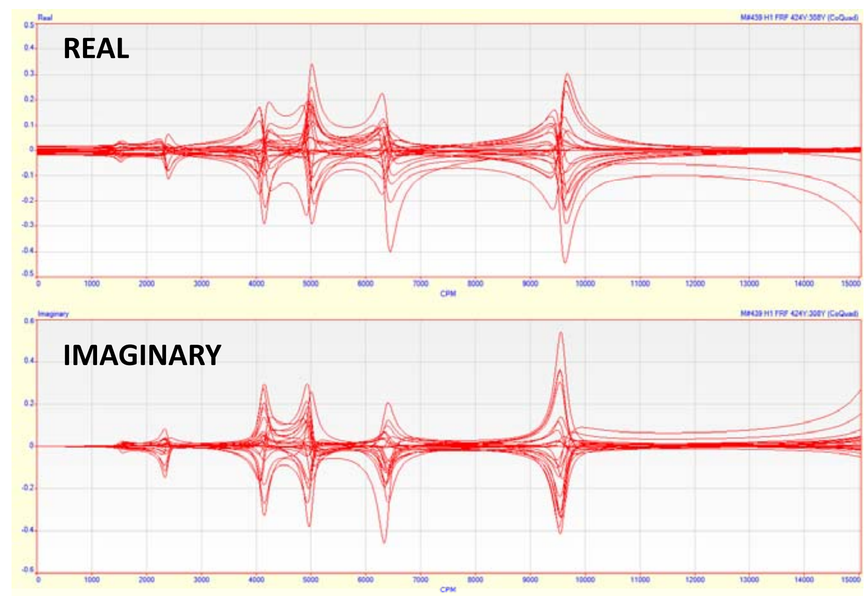
The setup parameters are defined by the desired frequency range. It is recommended to define a time block in the 2 second range to reduce background noise.

There are also options in many acquisition software for reducing noise on the input and/or output channels (H1, H2). Spend some time in the setup phase for better measurements.









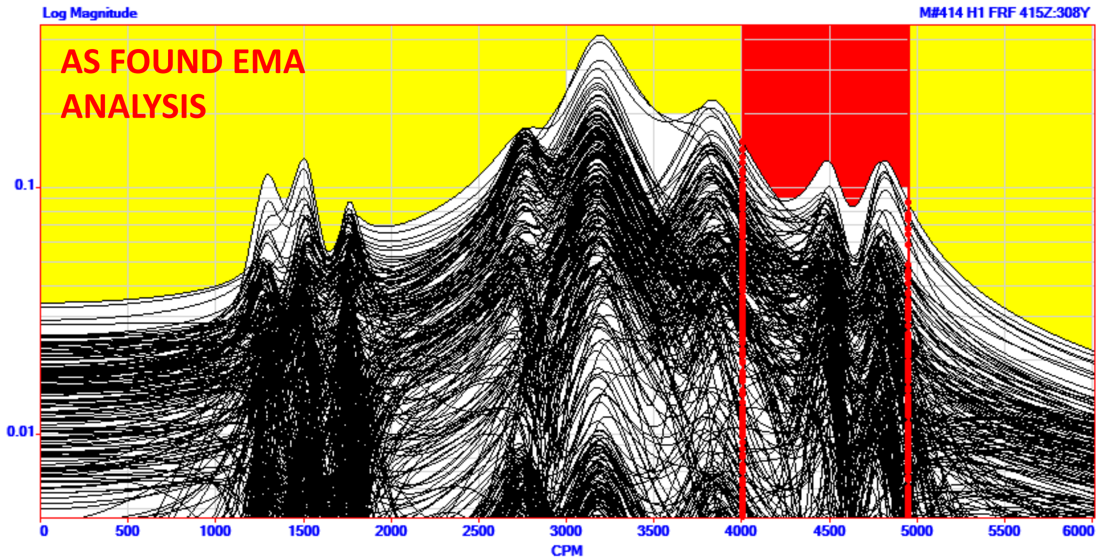
Experimental Modal Test Instrument Setup Parameters

F_{MAX} [CPM / Hz]	#LINES	SAMP RATE	#AVE	OVER LAP	RESOLUTION [CPM/LINE / Hz/LINE]	T_{MAX} [SEC]	DELAY [SEC]
12,000 / 200	400	1,024	4 - 6	0%	30.0 / 0.500	2.00	-0.050
30,000 / 500	800	2,048	4 - 6	0%	37.5 / 0.625	1.60	-0.050
60,000 / 1000	1,600	4,096	4 - 6	0%	37.5 / 0.625	1.60	-0.050
90,000 / 1500	3,200	8,192	4 - 6	0%	28.1 / 0.468	2.13	-0.050

Note: Use of Trigger Delay is dependent on the Analyzer Setup Parameters.

The Experimental Modal Analysis: AS FOUND

- The Modal Frequency
- The Modal Damping
- The Mode Shape(s)



The CAMPBELL Diagram: Turbine Measurements (AS FOUND)

A **Campbell Diagram**, also know as an **Interference Diagram**, allows the user to visually plot rotating orders of machinery versus natural frequencies in the system to determine if a coincidence situation exists, or more plainly, to determine potential resonance drivers in the system.

This diagram is a plot of Frequency (Natural Frequency) in CPM units versus Frequency (Rotational Speed) in RPM units.

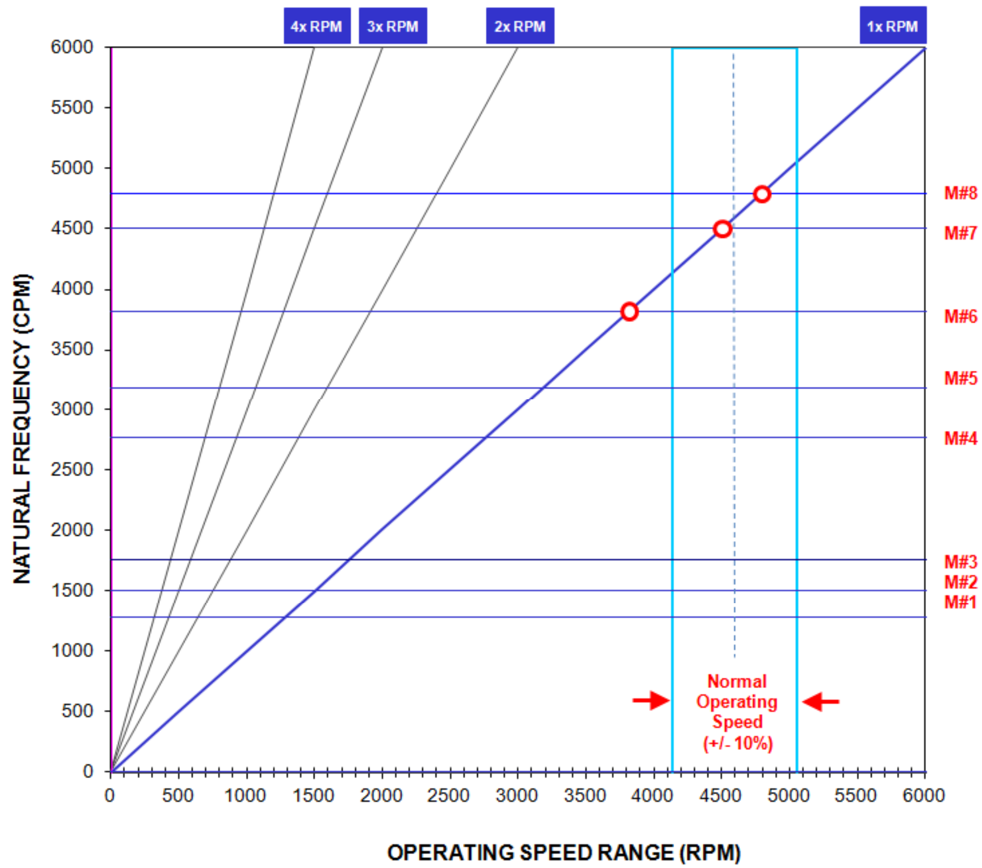
The natural frequencies of the Machine or Structures are defined from Experimental Modal testing and plotted as horizontal lines on the diagram.

The operational speed frequency ranges are defined and plotted as a vertical range on the diagram.

Rotating orders of running speed (1x, 2x, 3x, ….nx RPM) are plotted as sloping lines. The 1x RPM line has a 1:1 slope, the 2x RPM line has a 2:1 slope, and each higher harmonic follows the same characteristic.

Crossings of the rotating orders (1x, 2x, …RPM) and the plotted natural frequencies within the operating speed ranges indicate possible resonance concerns that may impact the vibratory amplitudes in the system

Typically 10% speed margin is required for structural natural frequencies, and 20% speed margin (minimum) for rotor critical speeds.



Select Shape	Frequency (or Time)	Damping	Units
1	1286.7	70.876	CPM
2	1511.3	65.645	CPM
3	1754.7	62.04	CPM
4	2767.3	106.88	CPM
5	3180.7	149.45	CPM
6	3823.9	158.35	CPM
7	4507.2	86.324	CPM
8	4792.6	94.714	CPM

EMA: Turbine Measurements (AS FOUND)

Modal analysis is used to characterize resonant vibration in mechanical structures. Each resonance has a specific "natural" or modal frequency, a modal damping or decay value, and a mode shape. FRF-Based parameter estimation (or curve fitting) is used to estimate the modal parameters of a structure from a set of FRFs.

1284 Discharge Piping Lateral mode

1506 Discharge Piping Axial mode

1752 Inlet Piping Lateral mode

2766 Rigid Body Turbine Assembly Lateral mode

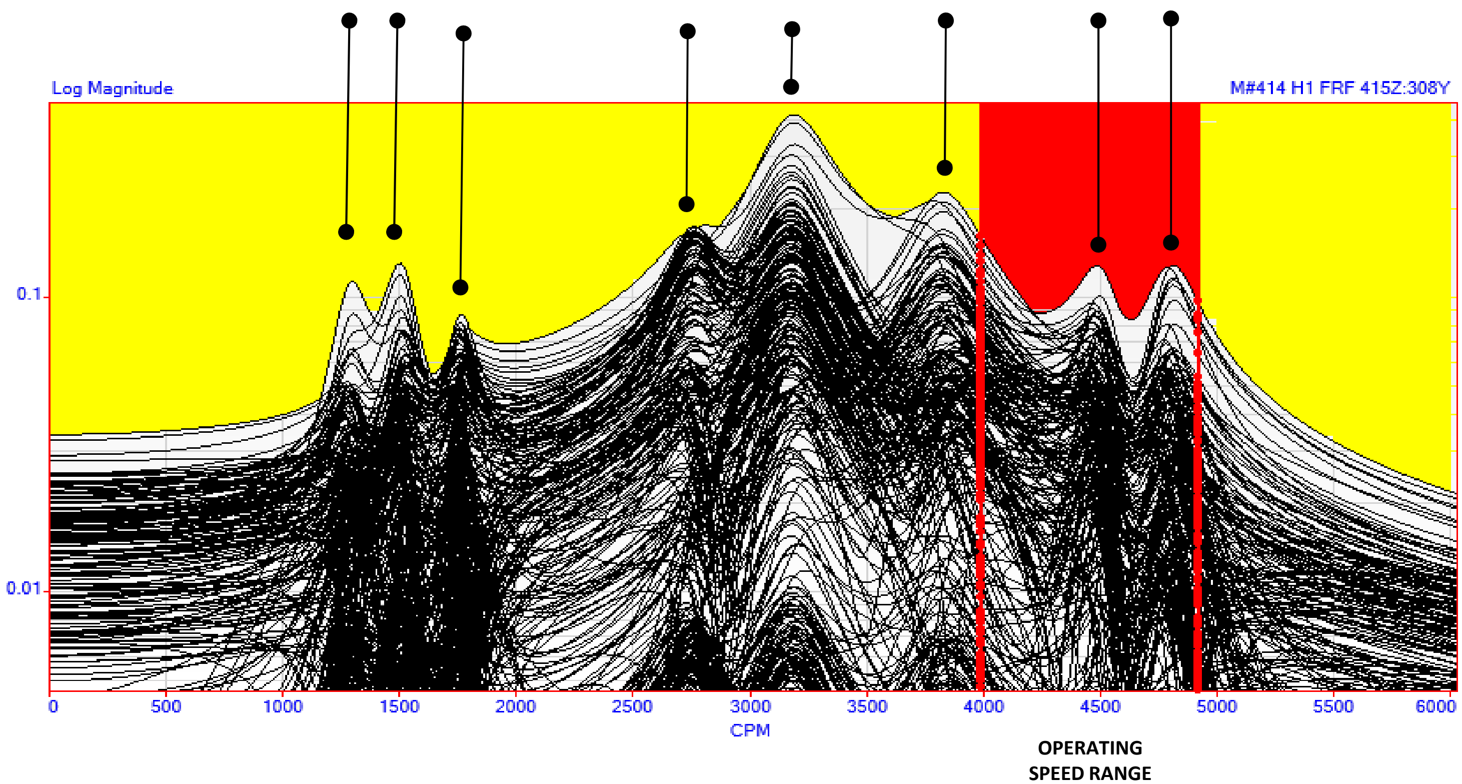
3180 Turbine Inboard Mount Flexural mode

3822 Turbine Outboard Mount Vertical Pitch mode

4506 Turbine Torsional mode

4788 Turbine Torsional mode

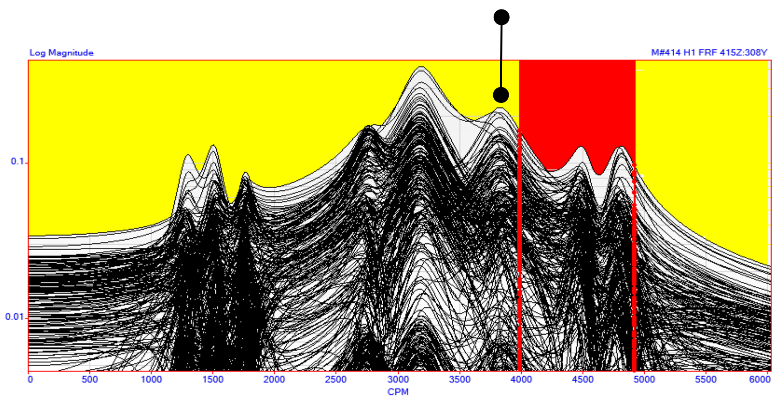
ME'scope Curve-Fitting Table



EMA: Turbine Measurements (AS FOUND)

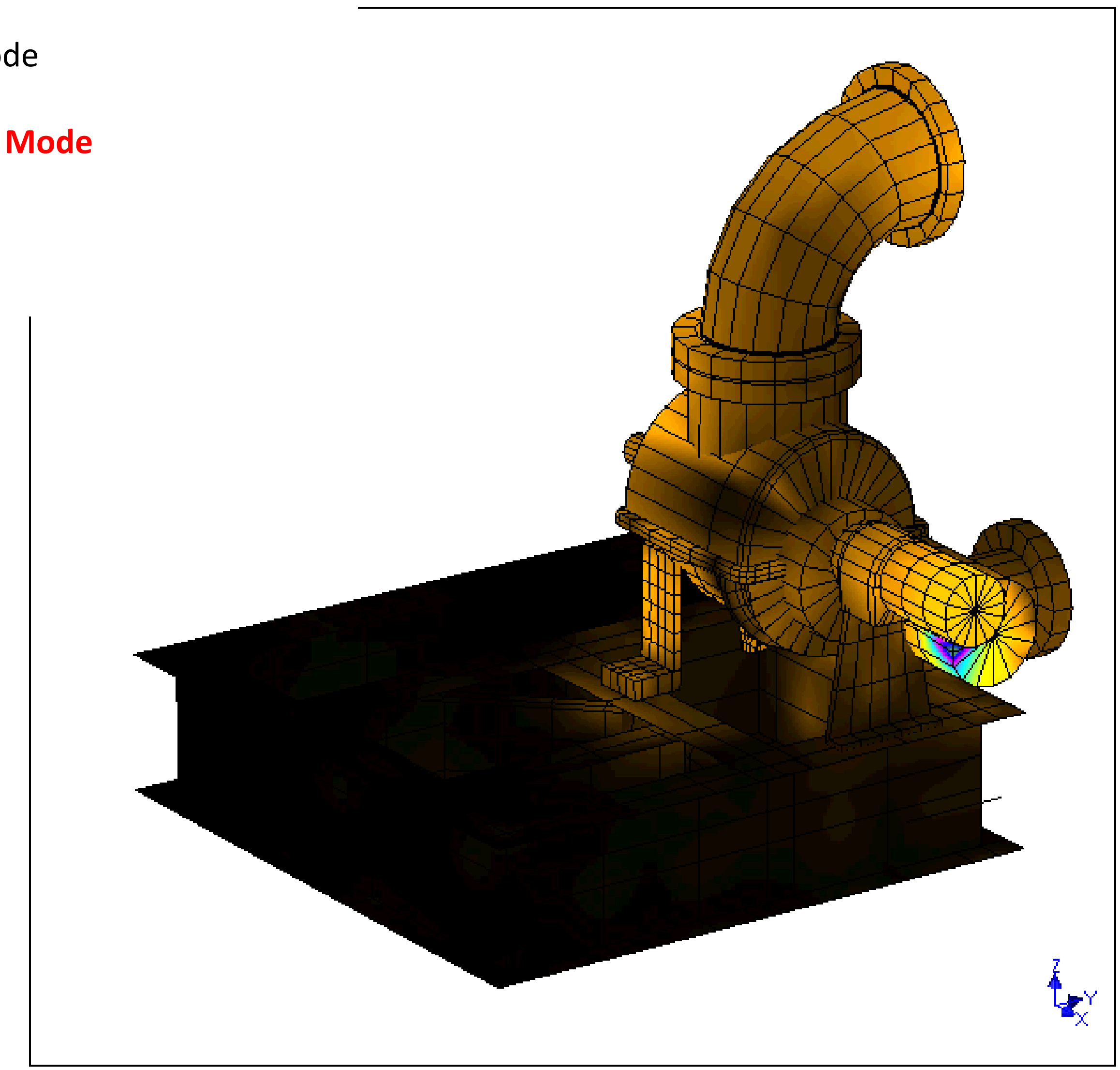
AS FOUND Natural Frequency Testing :

1284 CPM Discharge Piping Lateral Mode
1506 CPM Discharge Piping Axial Mode
1752 CPM Inlet Piping Lateral Mode
2766 CPM Rigid Body Turbine Assembly Lateral Mode
3180 CPM Turbine Inboard Mount Flexural Mode
3822 CPM Turbine Outboard Mount Vertical Pitch Mode
4506 CPM Turbine Torsional Mode
4788 CPM Turbine Torsional Mode



OPERATING
SPEED RANGE

3822 CPM

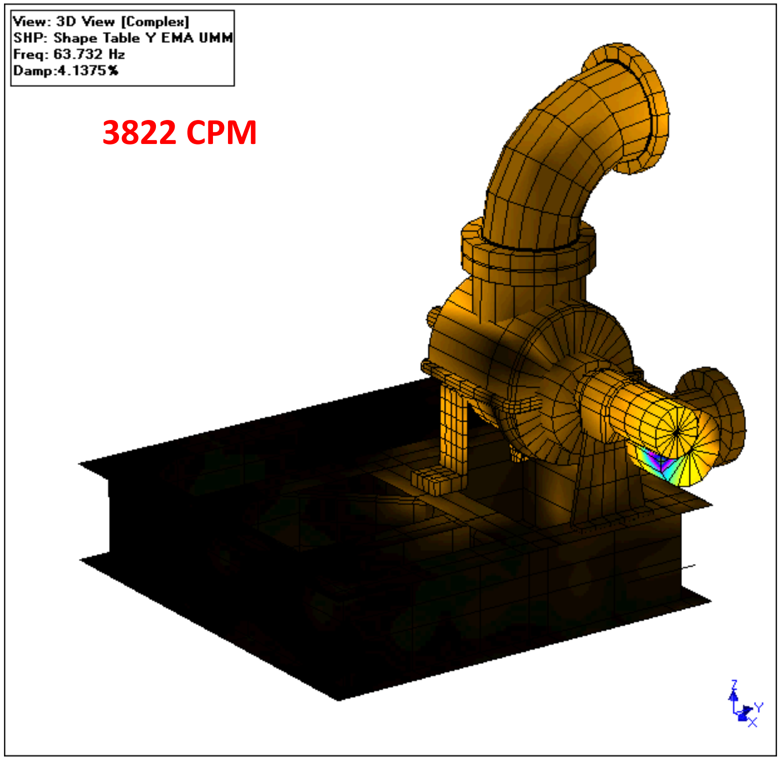


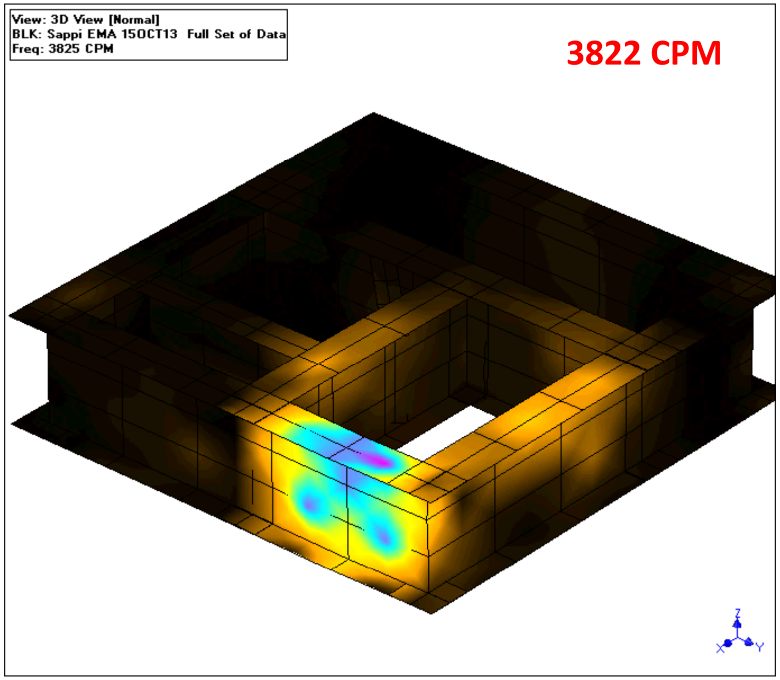
EMA: Turbine Measurements (AS FOUND)

The analysis includes three modes of interest in the 4000-4800 RPM desired operating range. The animations show that the modes of interest include lateral bending and torsional pitching of the turbine in the lateral direction. These modes were deemed "potentially excitable" from residual unbalance and misalignment in the rotor system.

The mode at 3822 CPM was not in the 4000-4800 RPM range, but the unit may have to operate through this range. The mode shape shows a dominant distortion in the outboard tri-angular turbine mount. This mount is designed to flex for alignment and thermal growth. The animation indicates that the framework supporting this mount is also flexing. This creates a "springs-in-series" effect that effectively creates a softer overall mount system. This location was a targeted for stiffening.

The lower animation was constructed without the turbine to better enhance the weak area in the support frame.

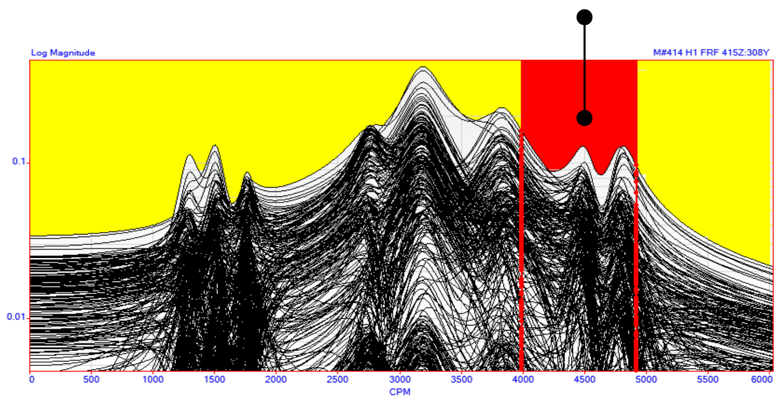




EMA: Turbine Measurements (AS FOUND)

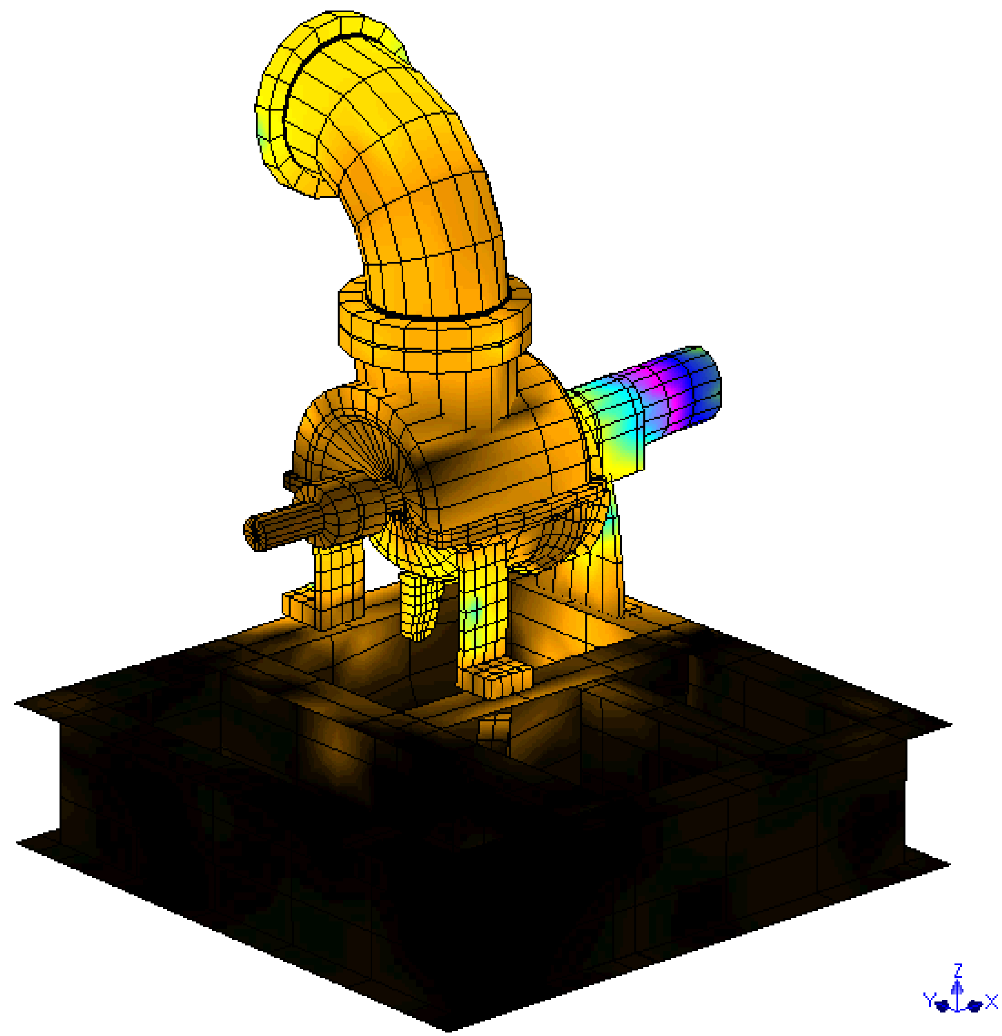
AS FOUND Natural Frequency Testing :

1284 CPM Discharge Piping Lateral Mode
1506 CPM Discharge Piping Axial Mode
1752 CPM Inlet Piping Lateral Mode
2766 CPM Rigid Body Turbine Assembly Lateral Mode
3180 CPM Turbine Inboard Mount Flexural Mode
3822 CPM Turbine Outboard Mount Vertical Pitch Mode
4506 CPM Turbine Torsional Mode
4788 CPM Turbine Torsional Mode



OPERATING
SPEED RANGE

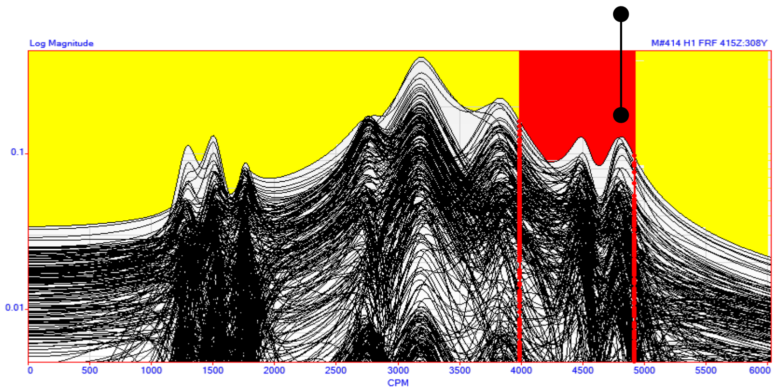
4506 CPM



EMA: Turbine Measurements (AS FOUND)

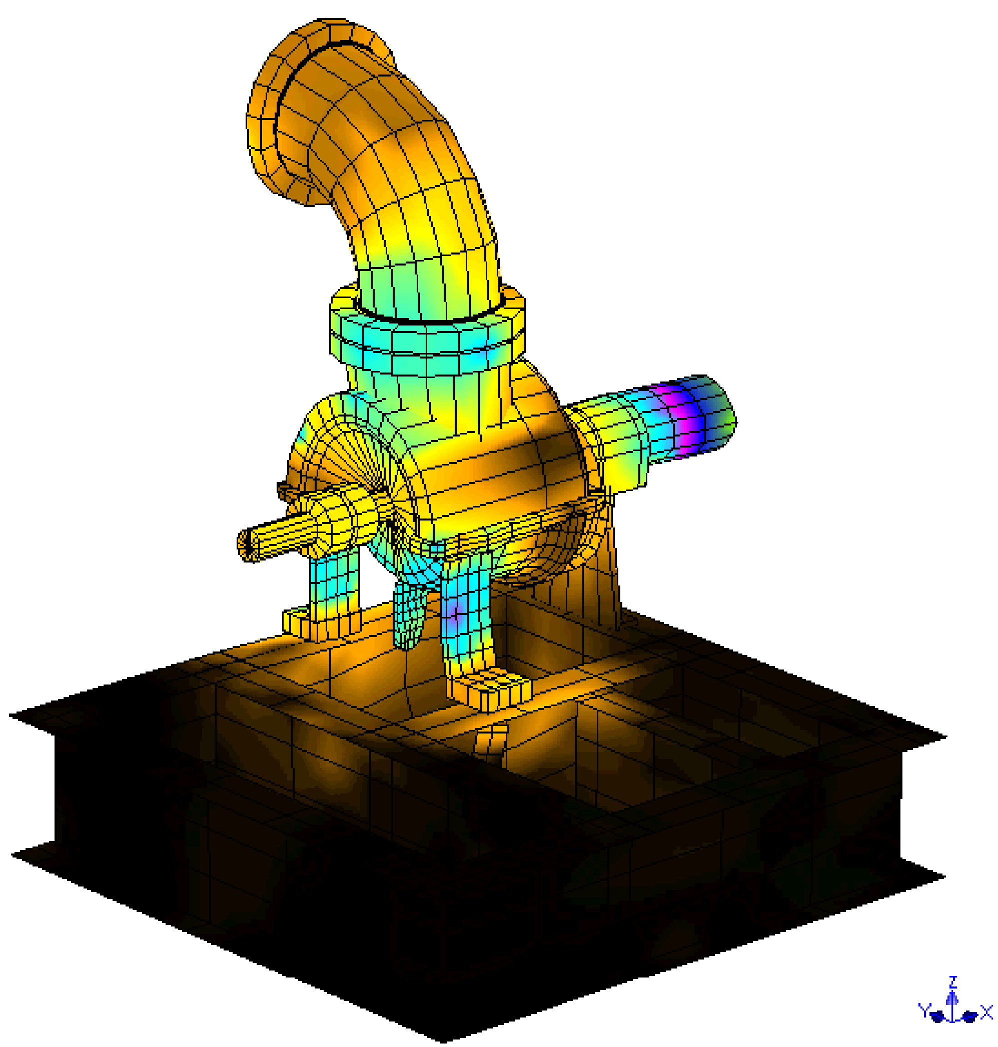
AS FOUND Natural Frequency Testing :

1284 CPM Discharge Piping Lateral Mode
1506 CPM Discharge Piping Axial Mode
1752 CPM Inlet Piping Lateral Mode
2766 CPM Rigid Body Turbine Assembly Lateral Mode
3180 CPM Turbine Inboard Mount Flexural Mode
3822 CPM Turbine Outboard Mount Vertical Pitch Mode
4506 CPM Turbine Torsional Mode
4788 CPM Turbine Torsional Mode



OPERATING
SPEED RANGE

4788 CPM

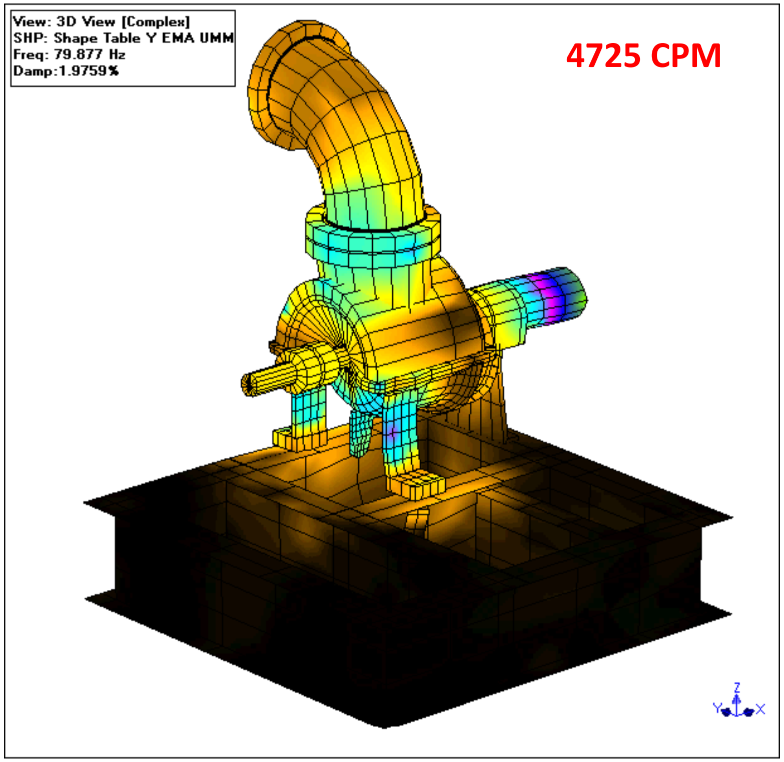


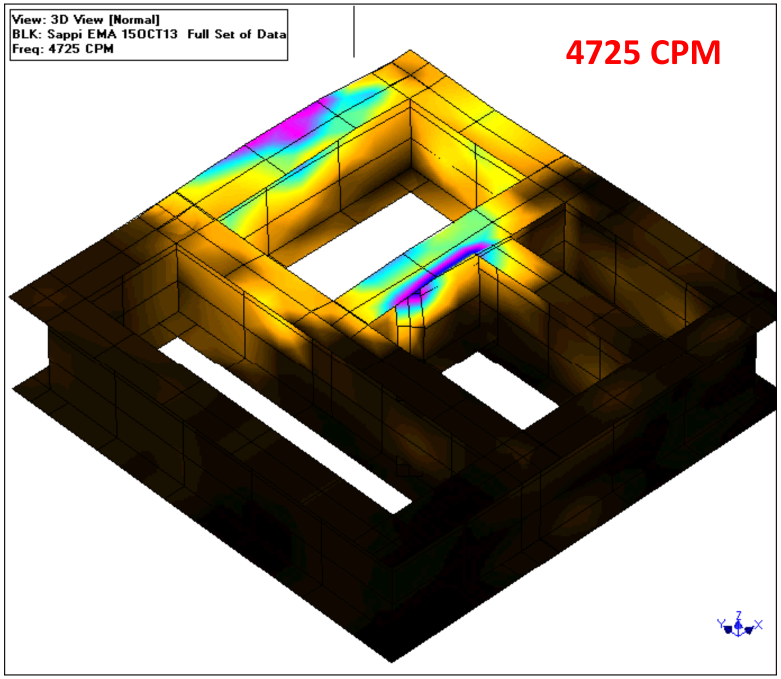
EMA: Turbine Measurements (AS FOUND)

The analysis includes three modes of interest in the 4000-4800 RPM desired operating range. The animations show that the modes of interest include lateral bending and torsional pitching of the turbine in the lateral direction. These modes were deemed “potentially excitable” from residual unbalance and misalignment in the rotor system.

Modes at 4506 and 4788 CPM indicated torsional twisting and lateral deflection in the inboard turbine mounts. Again, these mounts are designed to flex to accommodate thermal growth in the turbine. But, just as before, the modes indicated distortions in the I-beam framework and were suspected to be reducing the overall stiffness in the mounts.

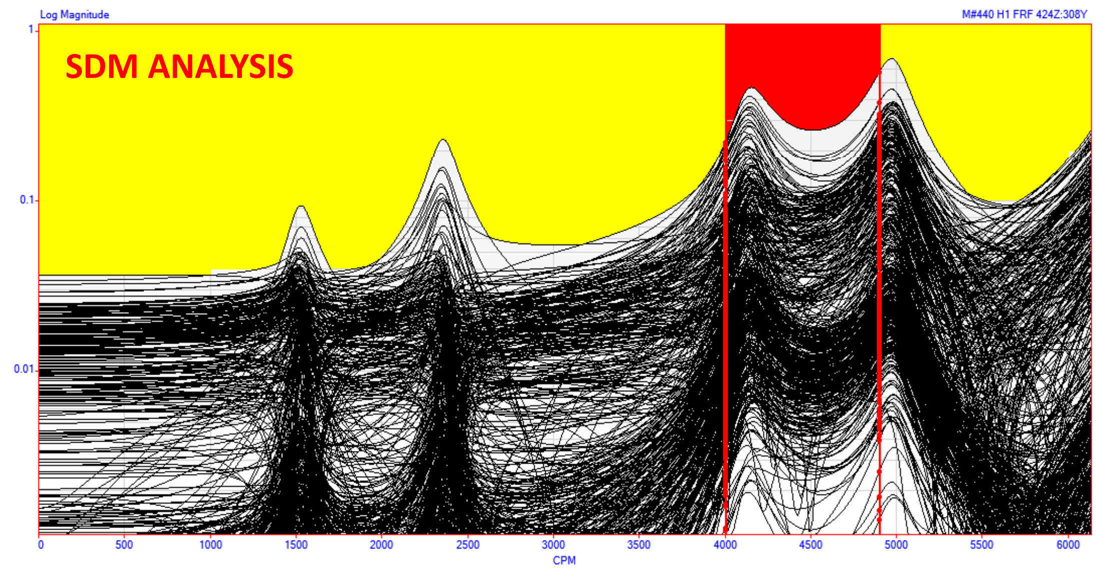
The fundamental Torsional mode of the turbine is shown to the right. The lower animation was constructed without the turbine to better enhance the weak area in the support frame.





The Structural Dynamic Modification Analysis:

- The Modal Frequency
- The Modal Damping
- The Mode Shape(s)



Structural Dynamic Modification (SDM): Analytical Modifications

The **Structural Dynamic Modification** analysis is a combination of empirically collected Experimental Modal Analysis measurements and analytically calculated Finite Element Analysis changes to be applied to those measurements.

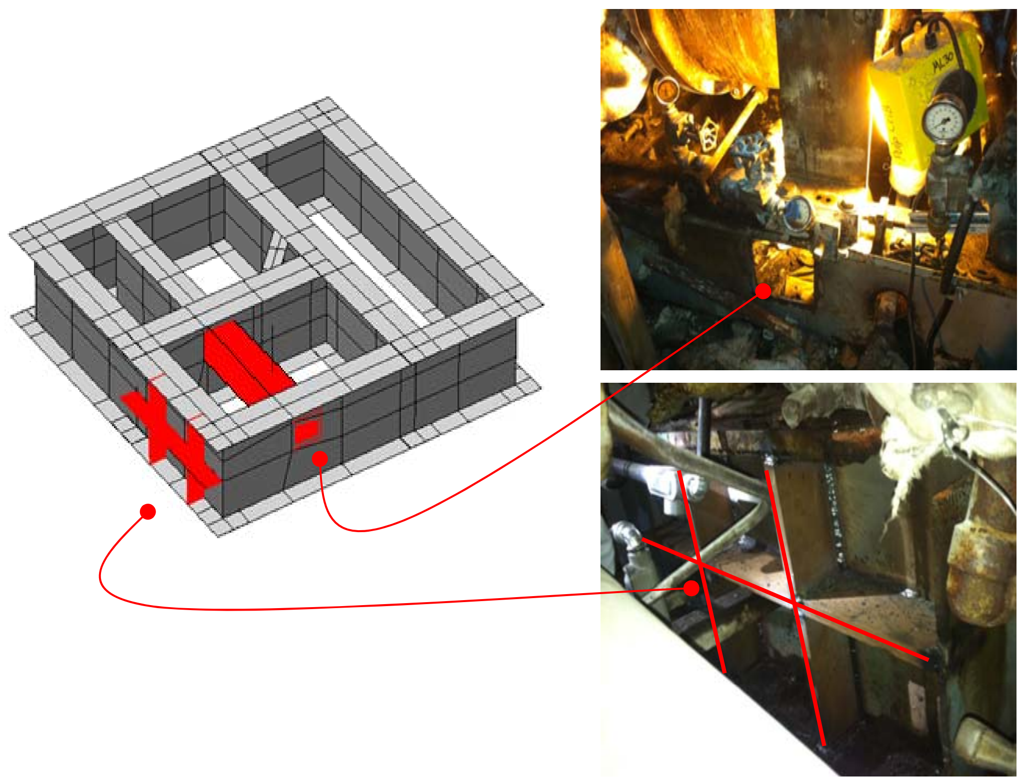
The Experimental Modal Analysis provides Natural Frequencies, Mode Shapes, and Damping Estimates for each mode found in the structure. The SDM program works in reverse to generate an equivalent Mass and Stiffness matrix in the computer that fits the empirical (frequency) measurements.

The analyst can then add potential modifications to the analytical structure via Finite Elements that alter this Mass and/or Stiffness Matrix. The program then re-calculates the Natural Frequencies of the "analytically modified" structure. The SDM analysis allows multiple modification trials to be performed in the computer simulation prior to any physical changes to the structure.

SDM #1
Installed 8" x 8" Steel Tube in Support Frame between Inboard Turbine Mounts.

SDM #2
Installed two Vertical Struts and one Horizontal strut at Support Frame under Outboard Turbine Mount

SDM #3
Welded Previous (bolted) I-beam modifications in place.

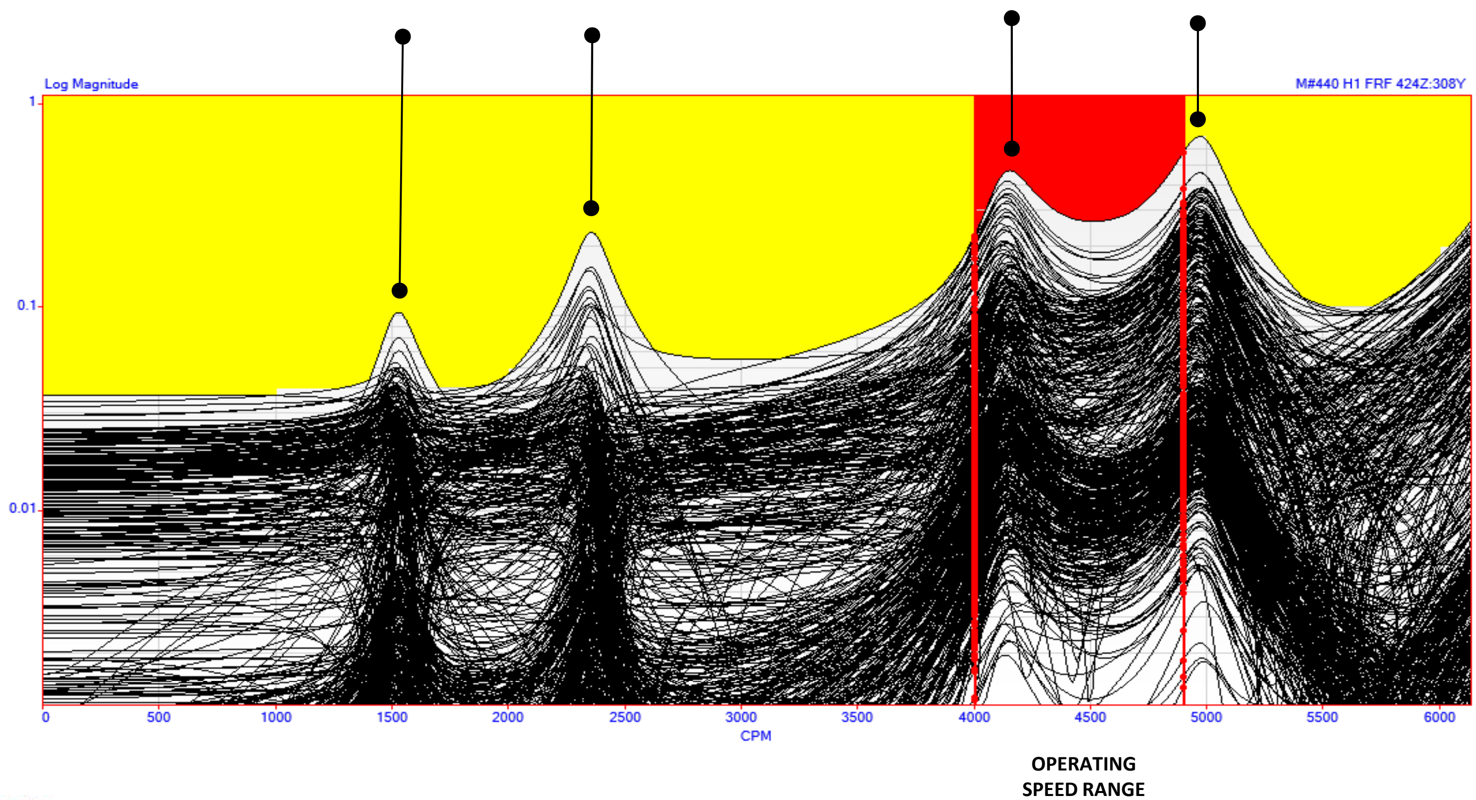


Structural Dynamic Modification (SDM): Analytical Predictions

Modal analysis is used to characterize resonant vibration in mechanical structures. Each resonance has a specific "natural" or modal frequency, a modal damping or decay value, and a mode shape. FRF-Based parameter estimation (or curve fitting) is used to estimate the modal parameters of a structure from a set of FRFs.

1524 CPM Discharge Piping Lateral mode
2355 CPM Discharge Piping Axial mode
4133 CPM Inlet Piping Lateral mode
4978 CPM Inlet Piping Axial mode
6373 CPM Turbine Outboard Mount Vertical Pitch mode
9571 CPM Turbine Torsional mode
15,771 CPM Turbine Torsional mode

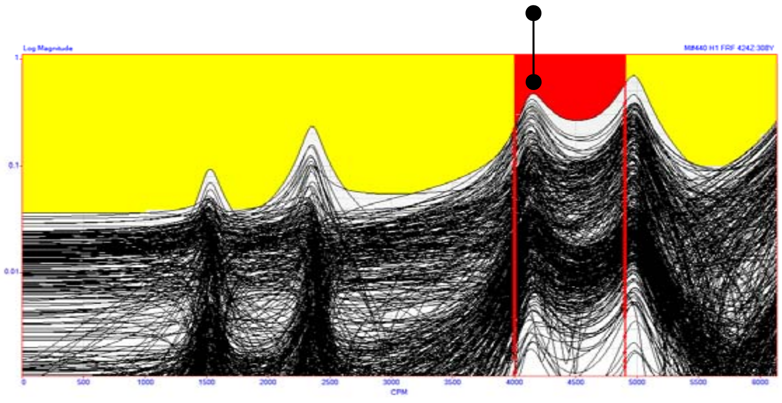
ME'scope Curve-Fitting Table

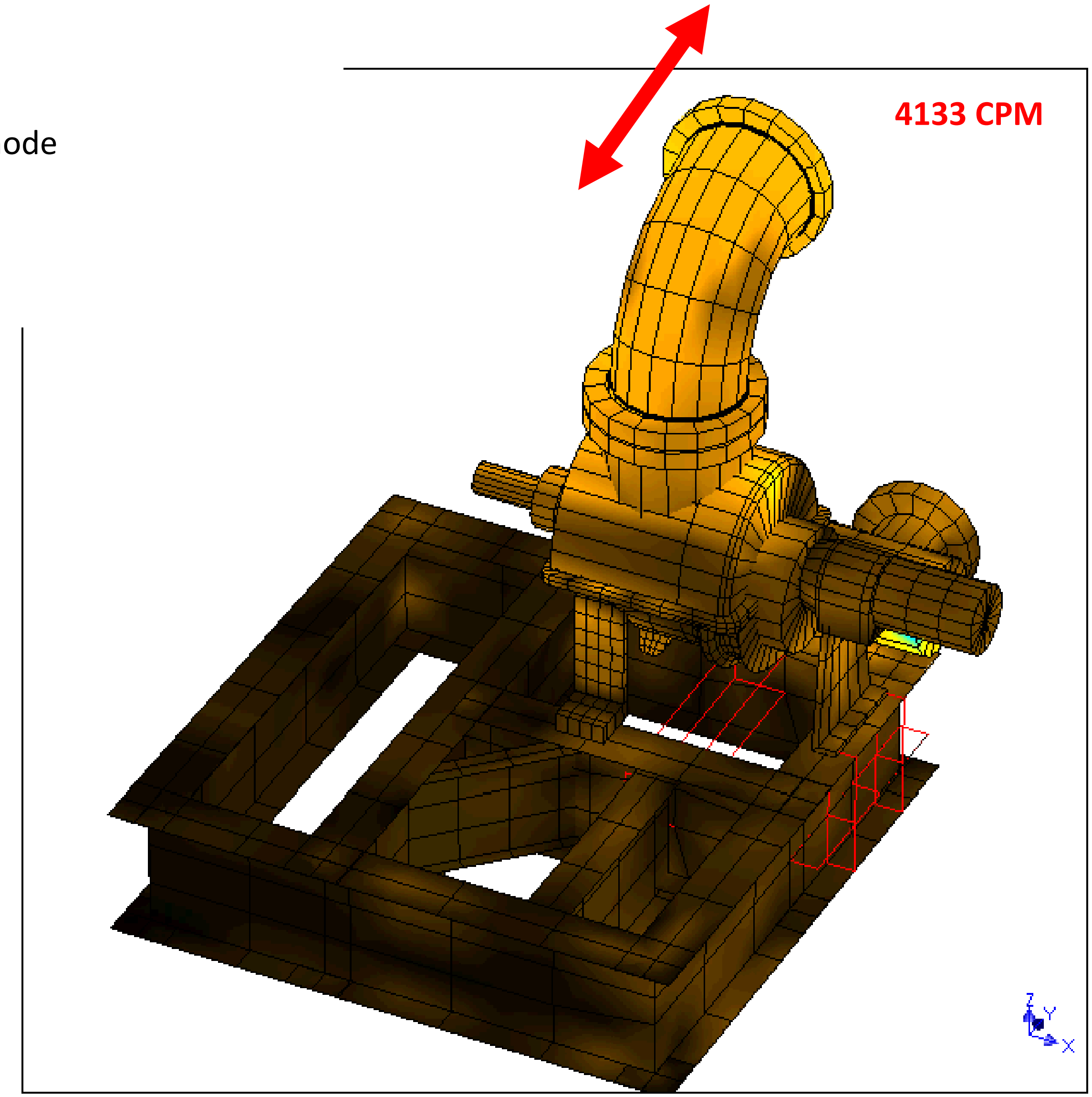


Structural Dynamic Modification (SDM): Analytical Predictions

SDM ANALYSIS:

1524 CPM Discharge Piping Lateral mode
2355 CPM Discharge Piping Axial mode
4133 CPM Inlet Piping Lateral mode
4978 CPM Inlet Piping Axial mode
6373 CPM Turbine Outboard Mount Vertical Pitch mode
9571 CPM Turbine Torsional mode
15,771 CPM Turbine Torsional mode

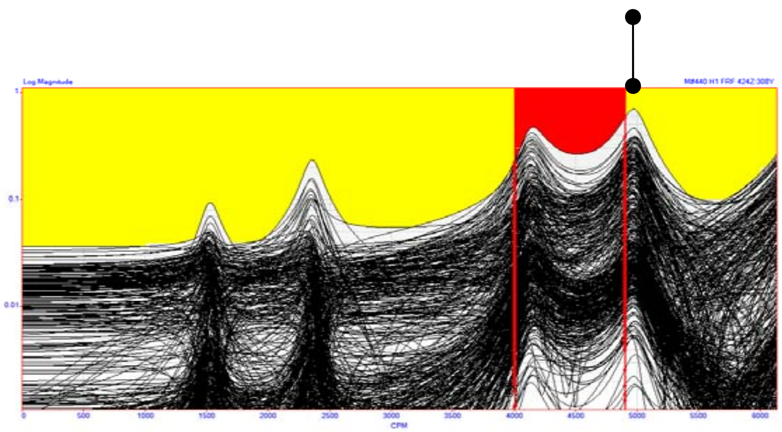




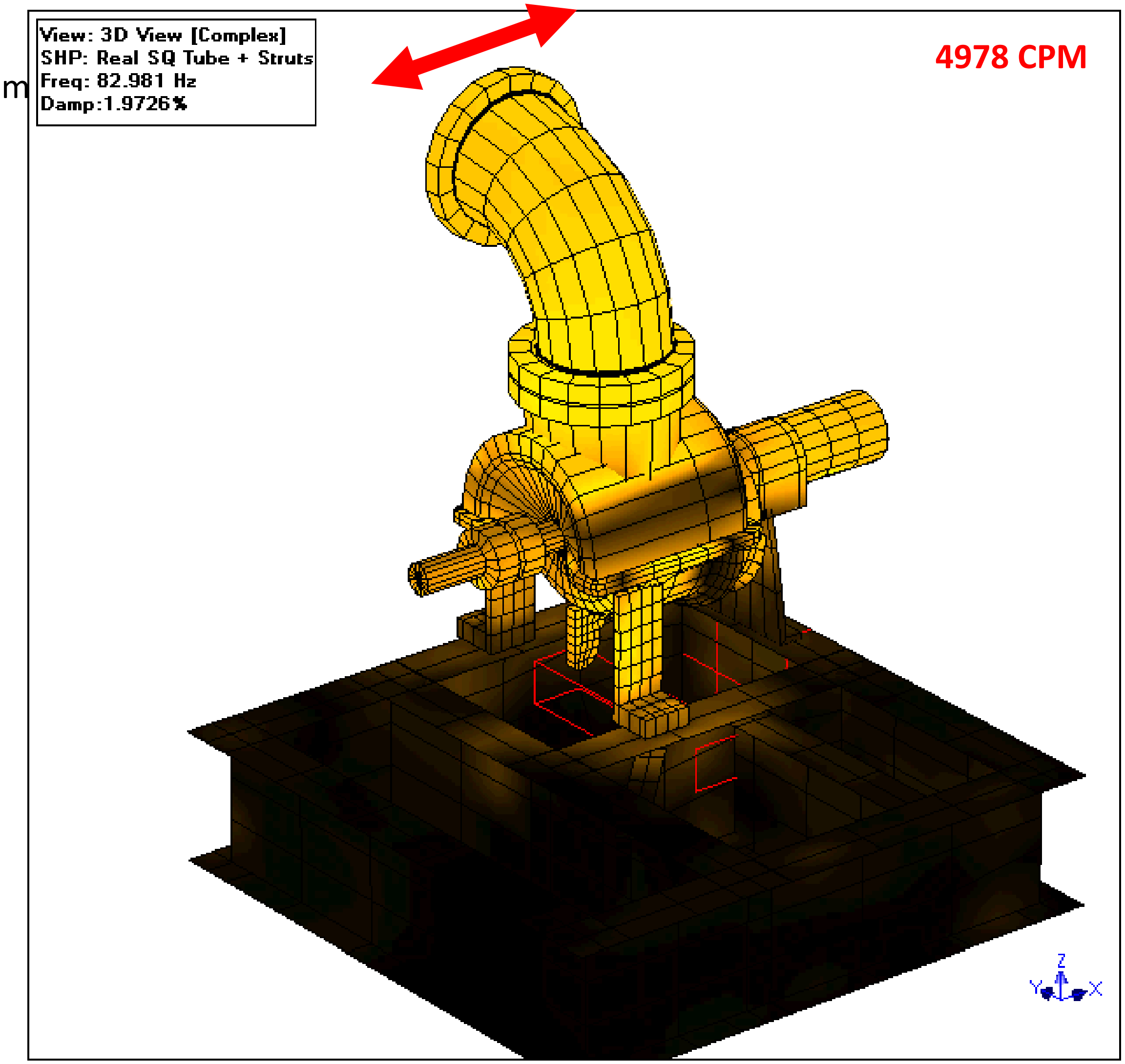
Structural Dynamic Modification (SDM): Analytical Predictions

SDM ANALYSIS:

1524 CPM Discharge Piping Lateral mode
2355 CPM Discharge Piping Axial mode
4133 CPM Inlet Piping Lateral mode
4978 CPM Inlet Piping Axial mode
6373 CPM Turbine Outboard Mount Vertical Pitch m
9571 CPM Turbine Torsional mode
15,771 CPM Turbine Torsional mode

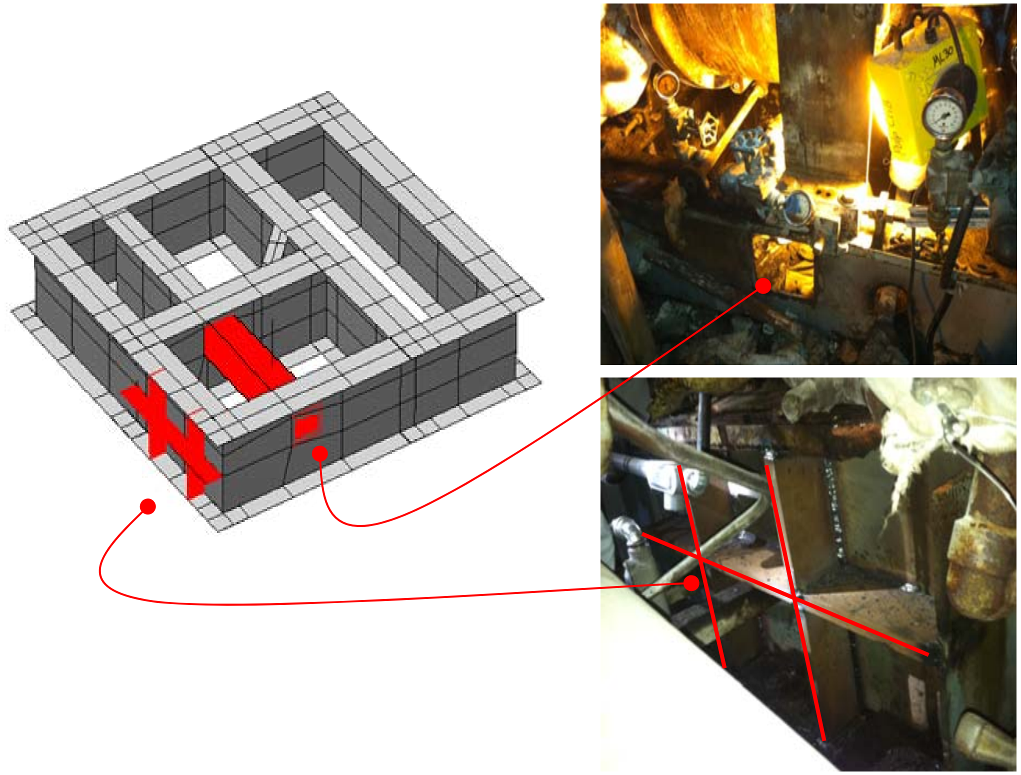


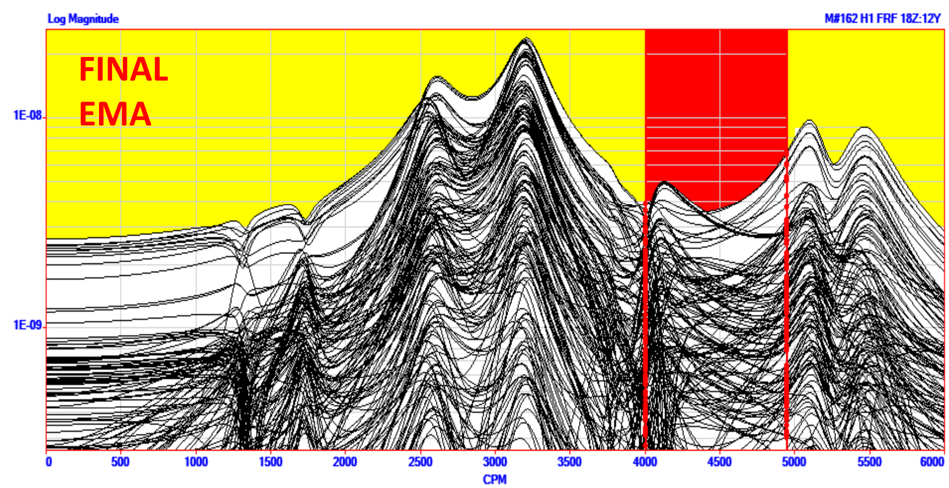
OPERATING
SPEED RANGE



Physical Modifications: EMA Verification Analysis

- The Modal Frequency
- The Modal Damping
- The Mode Shape(s)





Physical Modifications: EMA Verification Analysis

Modal Analysis is used to characterize resonant vibration in mechanical structures. Each resonance has a specific "natural" or modal frequency, a modal damping or decay value, and a mode shape. FRF-Based parameter estimation (or curve fitting) is used to estimate the modal parameters of a structure from a set of FRFs.

RAW DATA

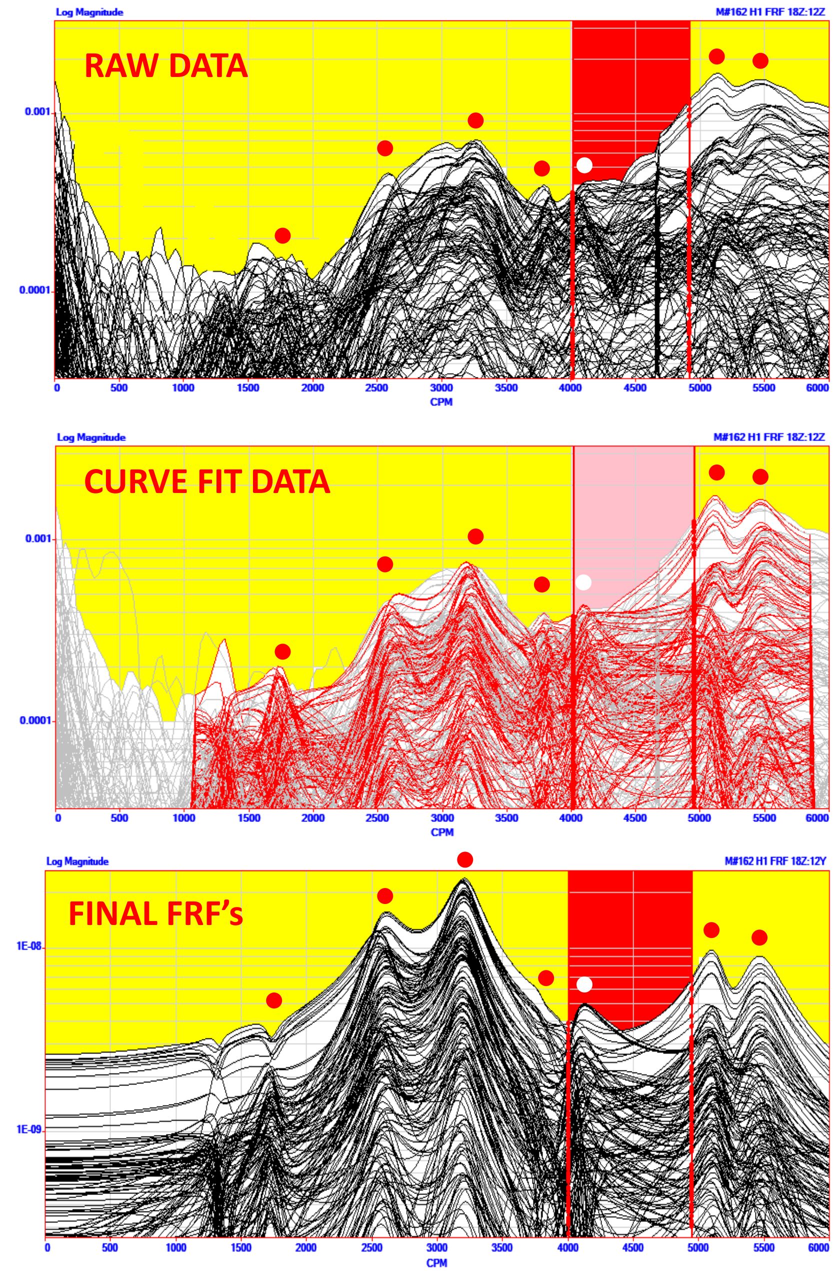
The top plot shows an overlay of the raw FRF measurements from the EMA analysis following structural modifications.

CURVE-FIT DATA

The middle plot shows the curve-fit data and "Residue" or "best-fit" approximation (red) of each resonance based on the Modal Frequency, Modal Damping and proximity to adjacent modes.

FINAL FRF's

The bottom plot shows the Synthesized FRF's from the Curve-Fitting analysis, or the Residues minus the Raw Data signatures.



The Verification Analysis:

"AS FOUND" EMA Analysis

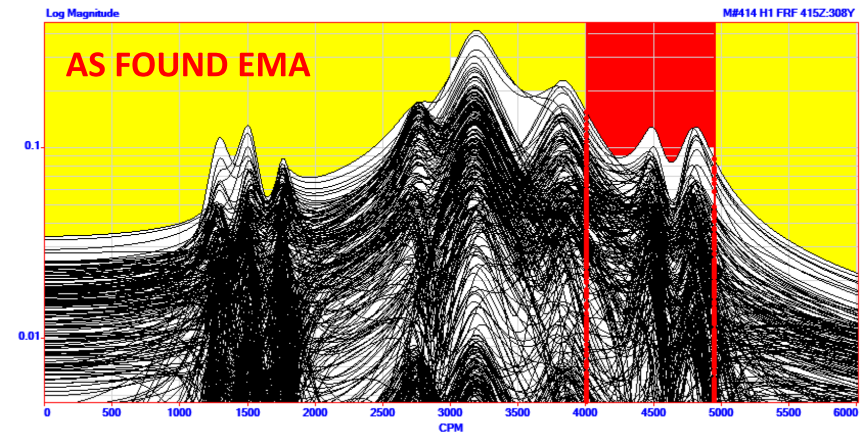
The initial EMA analysis of the Steam Turbine and its support structures indicated two dominant natural frequencies within the operating speed range of the unit.

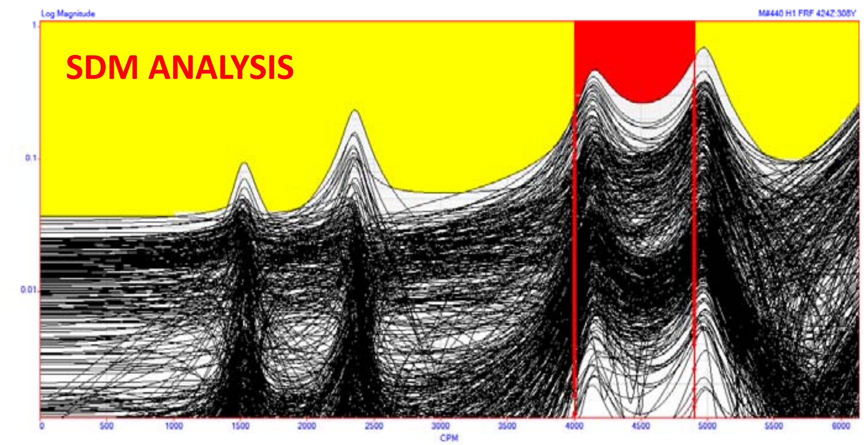
SDM Analysis / Analytical Predictions

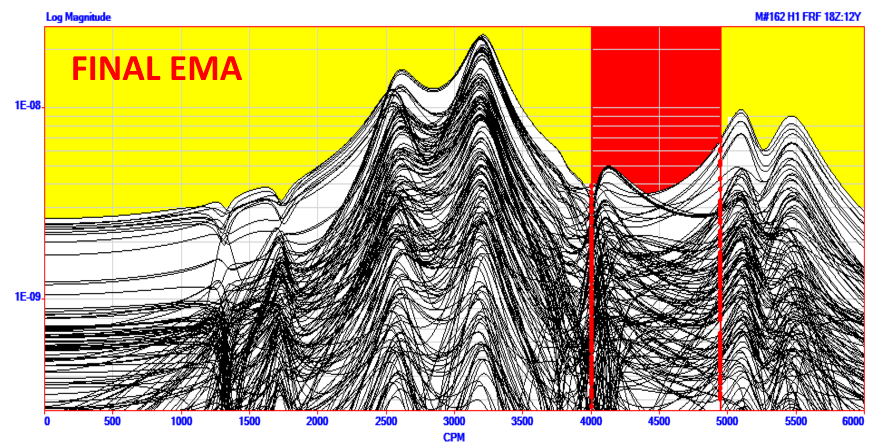
A series of structural modifications were performed in an attempt of removing the problem modes from the operating speed range. Limitations on the extend of modifications permitted a reasonable acceptable solution.

FINAL EMA Verification Analysis

A final post-modification EMA analysis verified the Analytical Predictions from the SDM Analysis. Analysis indicated a relatively resonance-free operating range for the Turbine.







The Verification Analysis:

"AS FOUND" EMA Analysis

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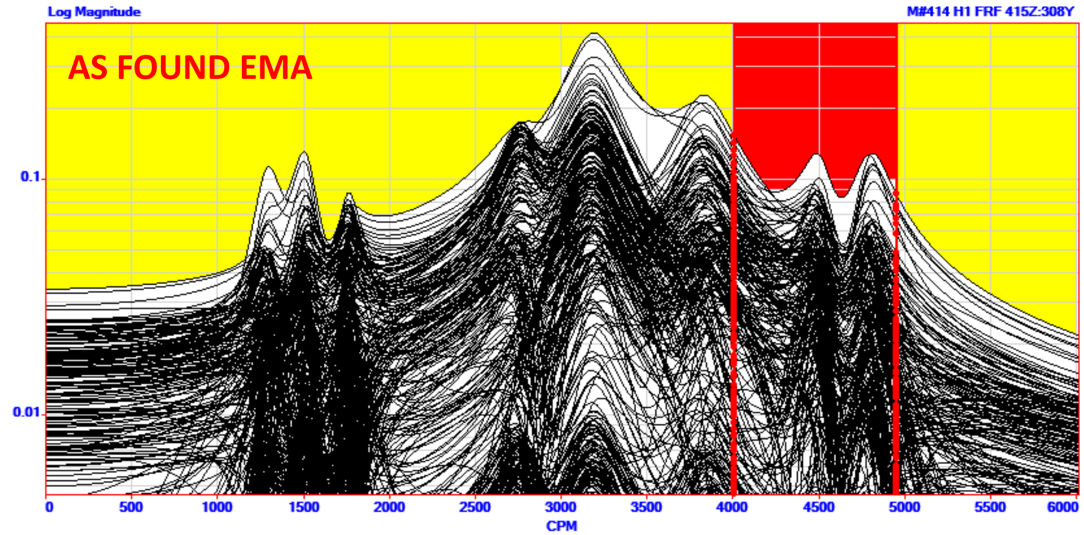

FINAL EMA Verification Analysis

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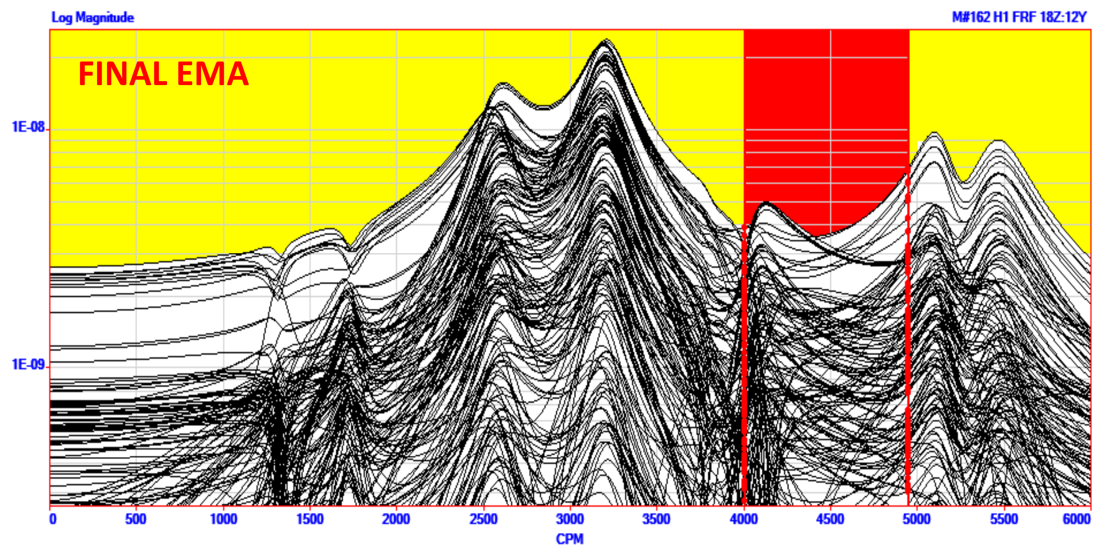
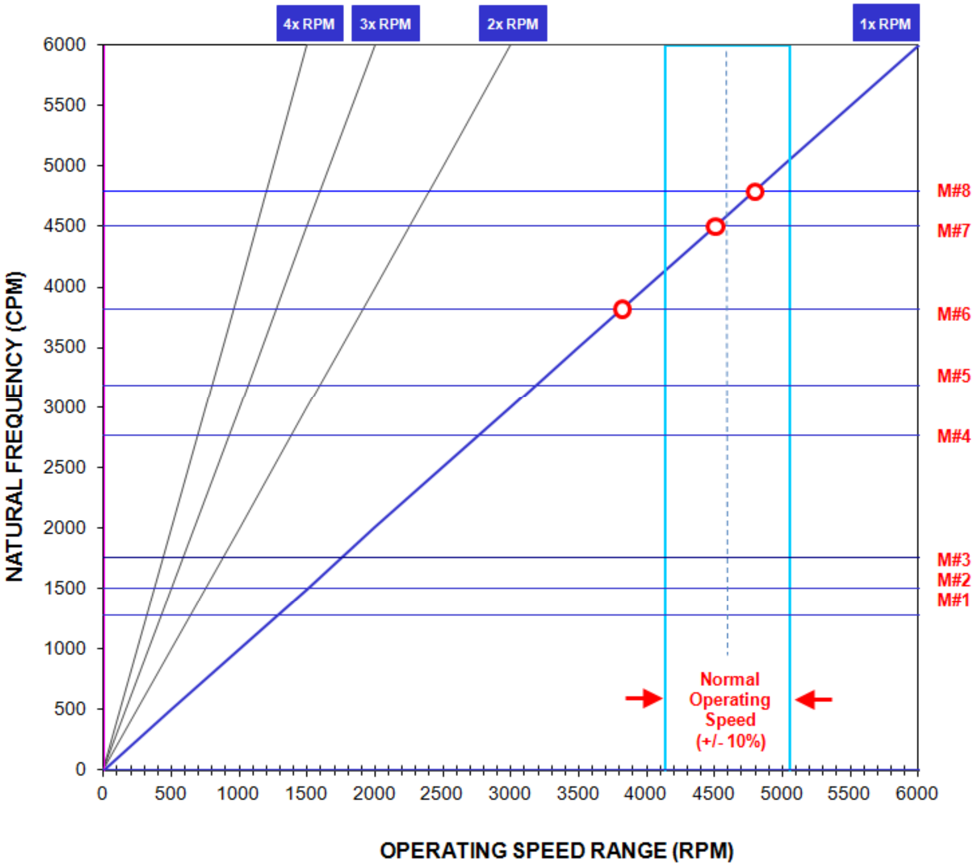
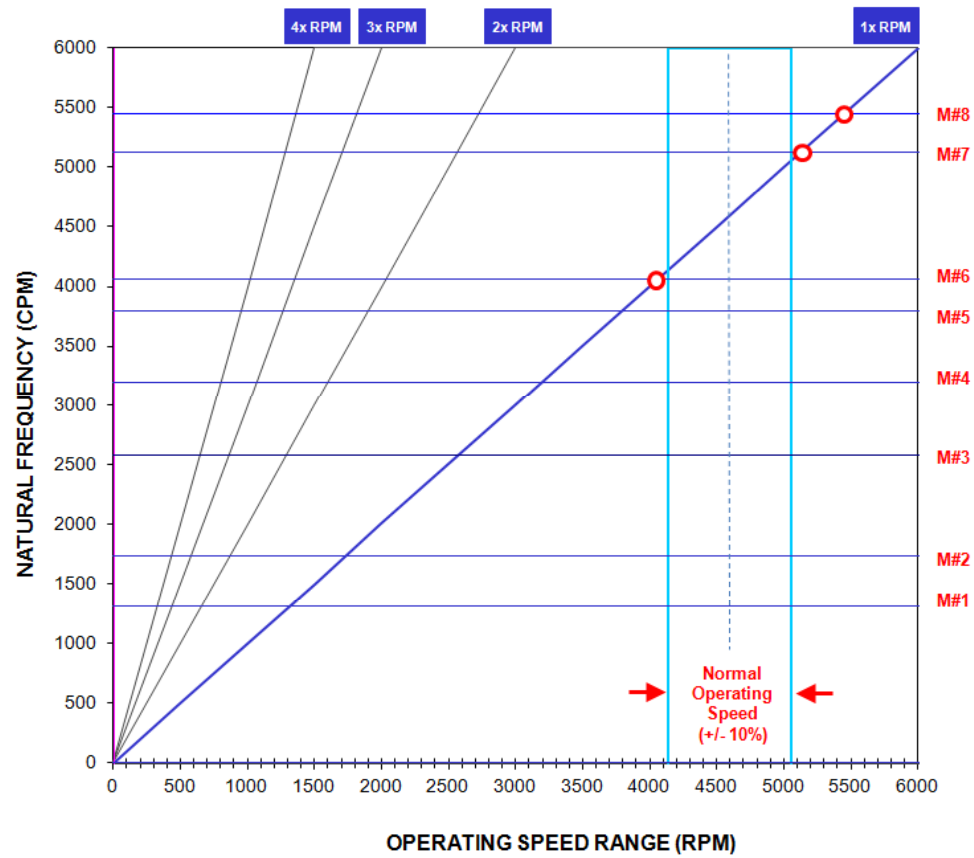


The CAMPBELL Diagram: As Found Analysis vs. Verification Analysis





Select Shape	Frequency (or Time)	Damping	Units
1	1286.7	70.876	CPM
2	1511.3	65.645	CPM
3	1754.7	62.04	CPM
4	2767.3	106.88	CPM
5	3180.7	149.45	CPM
6	3823.9	158.35	CPM
7	4507.2	86.324	CPM
8	4792.6	94.714	CPM

As Found
EMA Analysis

Select Mode	Frequency CPM	Damping CPM	Damping (%)
1	1310.9	64.61	4.9228
2	1730.1	79.171	4.5713
3	2579.2	134.73	5.2164
4	3200.6	120.33	3.7568
5	3794.9	64.535	1.7003
6	4063.7	108.52	2.6696
7	5122.3	114.12	2.2273
8	5452.6	163.8	3.0027

Post Modification
EMA Analysis

Operating Verification Transient Analysis

Following the Structural Modifications, the turbine was aligned and re-started. The turbine startup was hindered by elevated vibration levels. Vibration Phase analysis indicated a significant residual misalignment in the turbine. Dial indicators were placed on the turbine and gearbox couplings to better define the thermal growth in the unit. The unit was taken off-line and re-aligned based on the better thermal information.

The turbine was restarted and the speed was increased to nearly 4,700 RPM. The amplitude and phase response was now acceptable, indicating that the alignment problems were eliminated. The amplitude versus speed relationship was plotted. The curve followed a "speed-squared" profile indicating the response was due to residual unbalance. A resonance condition will follow an exponential profile with respect to speed.