Centrifugal Compressor Root Cause Analysis and Case Study (Pulsation and Vibration Issues)

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BETA
MACHINERY ANALYSIS

AGENDA

A) Excitation forces on centrifugal compressors

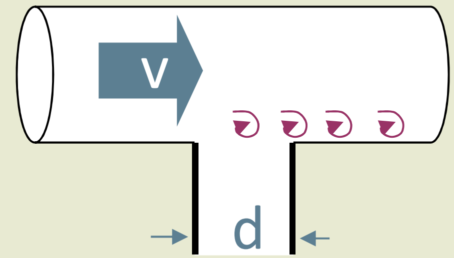
B) Root cause analysis approach

C) Case study:

- Pulsations, Blade Passing Frequency, Vibration
- Pipe shell mode & acoustic resonance
- Lessons learned

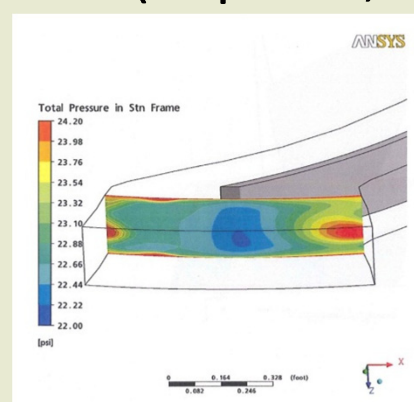
A) Forces Acting around Centrifugal Compressors

1. Mechanical Forces (imbalance of rotor, whip/whirl)
   - Low frequency (≤ run speed)



2. Flow Induced Turbulence
   - Created by flow past openings or obstructions
   - Can result in broad band energy and discrete frequency excitation

3. Pulsations from Compressor Impeller
   - Excitation frequency based on number of blades (impeller, diffuser, shaft speed)
   - Pulsation strength related to geometry and operating point

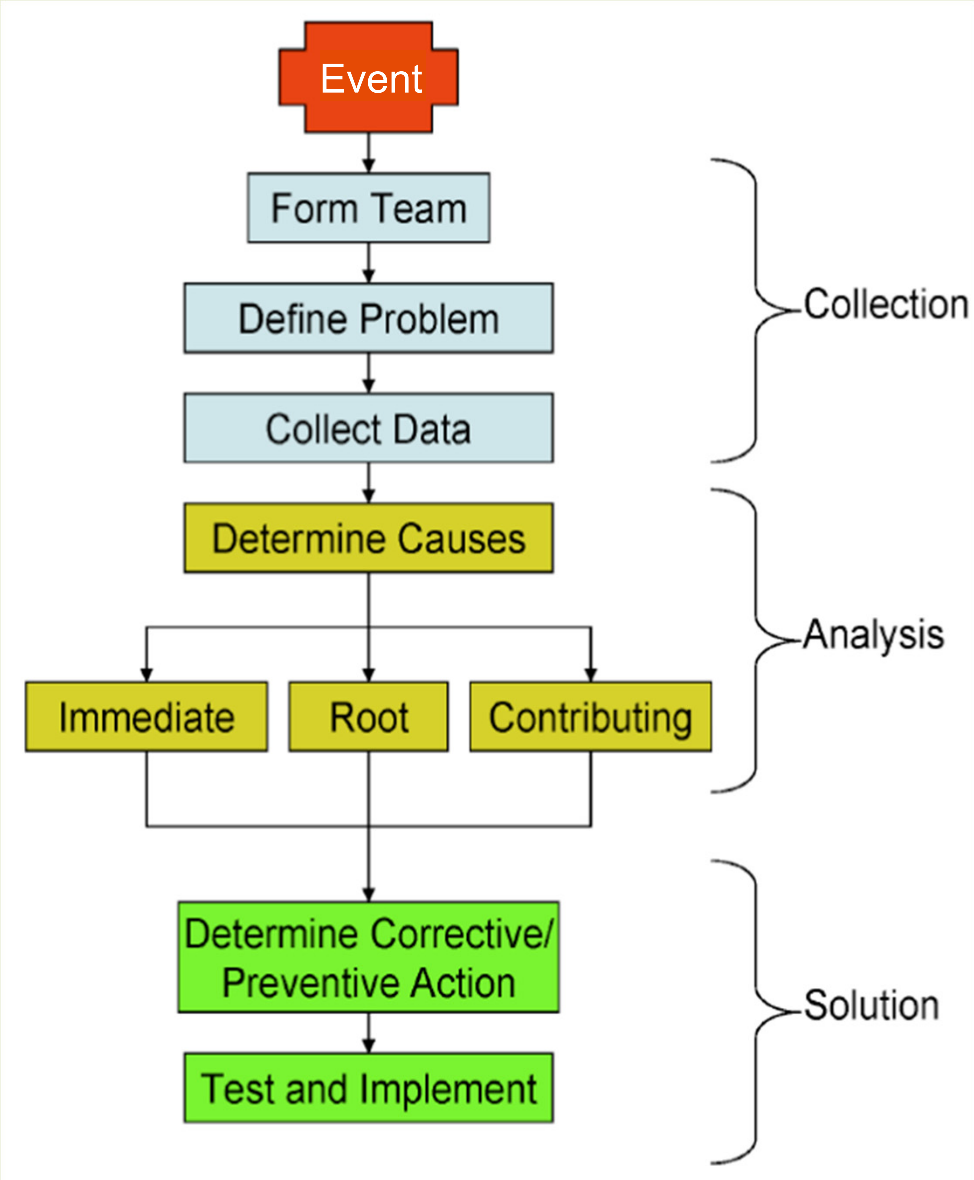


A) Forces Acting around Centrifugal Compressors

4. Rotating Stall Forces
    - Flow instabilities (at low flow)
    - Excitation at sub-synchronous compressor speeds (< run speed)

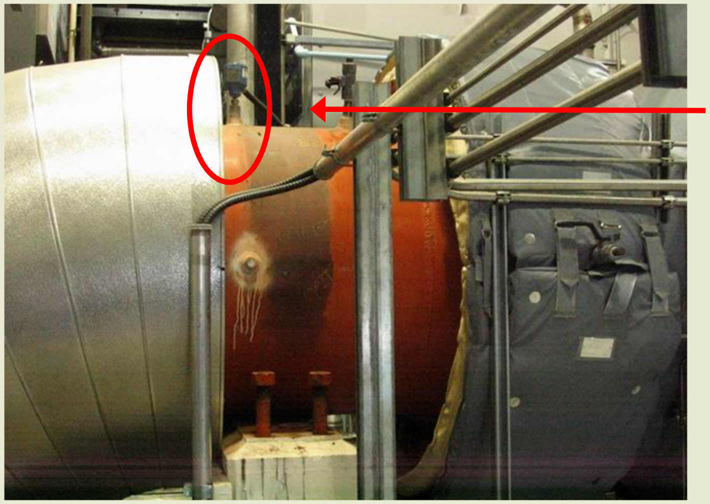
5. Momentum Changes
    - Due to rapid opening/closing of valves = transient pressure waves
    - Discontinuous, not steady-state
    - Can generate significant forces on the piping system

B) Root Cause Analysis Approach



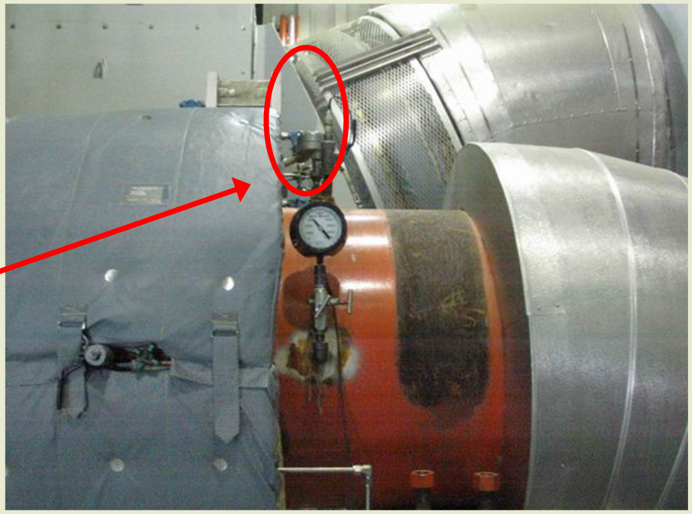
C) Case Study. Collection – Event, Team, Problem

- **Event** Revamp of 3 Compressors at Pipeline Station in 2008
  - Operating Speed 3600 to 5000 rpm
  - Ps = 42.4 barg (615 psig) ; Pd = 57.2 barg (830 psig)
- **Team** Field Troubleshooter and Client Representatives
- **Problem** Numerous failures of instrumentation
  - Primarily RTD/thermocouples on Discharge side, but some Suction too
  - Modifications made….but unsuccessful

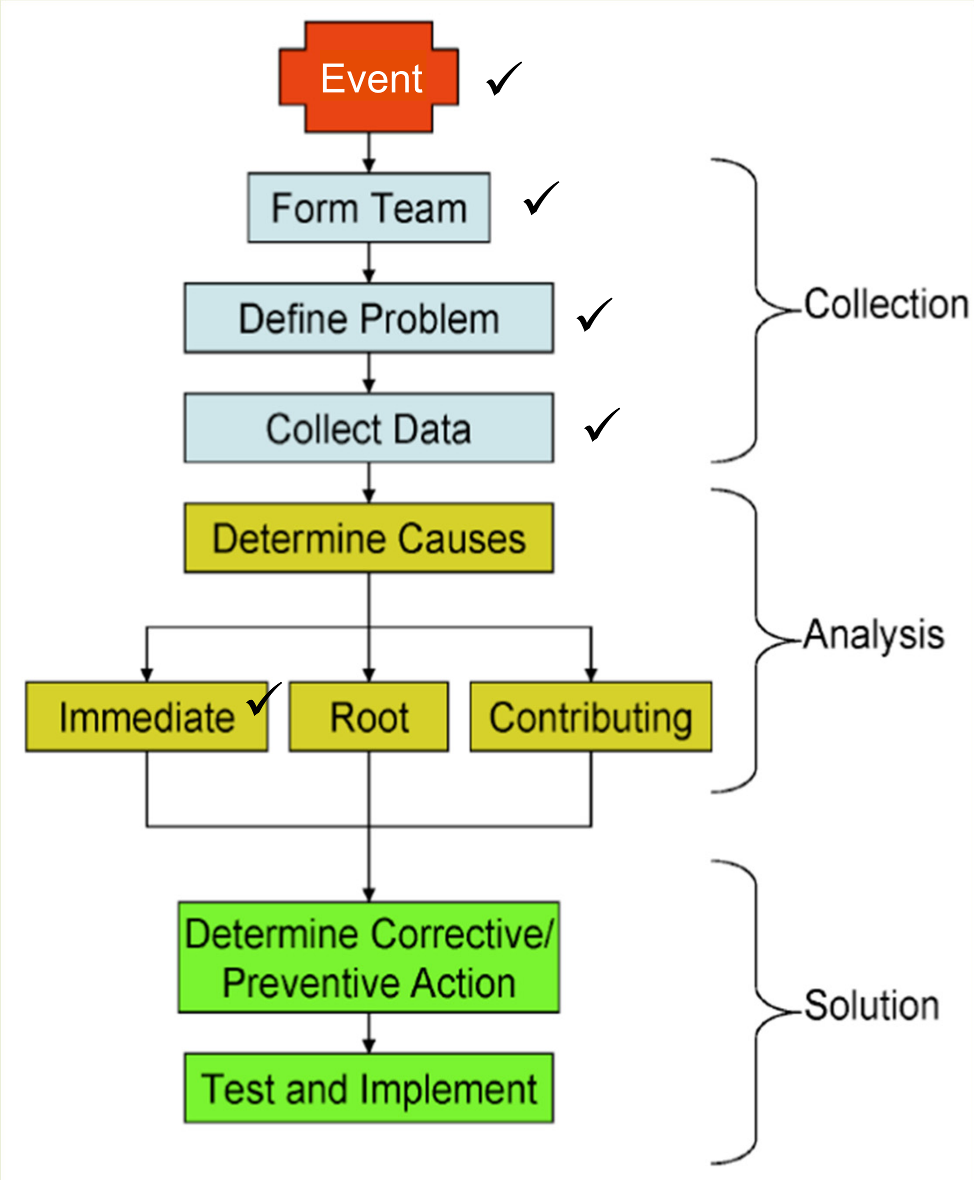


Discharge RTDs

Suction RTDs



C) Case Study - Root Cause Analysis Approach



C) Case Study. Collect Data - Initial Investigation

Background

- Client had changed thermowells (6” to 3”) with no improvement
  - Used this data to calculate thermowell frequencies:

	Original 6” Thermowell	Replacement 3” Thermowell
Mechanical Natural Freq. (1st Bending Mode of a Cantilevered Beam)	715 Hz	2840 Hz
Vortex Shedding Freq. at Suction Conditions (54° F, 615 psig, and 2500 mmSCFD)	370 Hz	
Vortex Shedding Freq. at Discharge Conditions (100° F, 826 psig, and 2500 mmSCFD)	310 Hz	

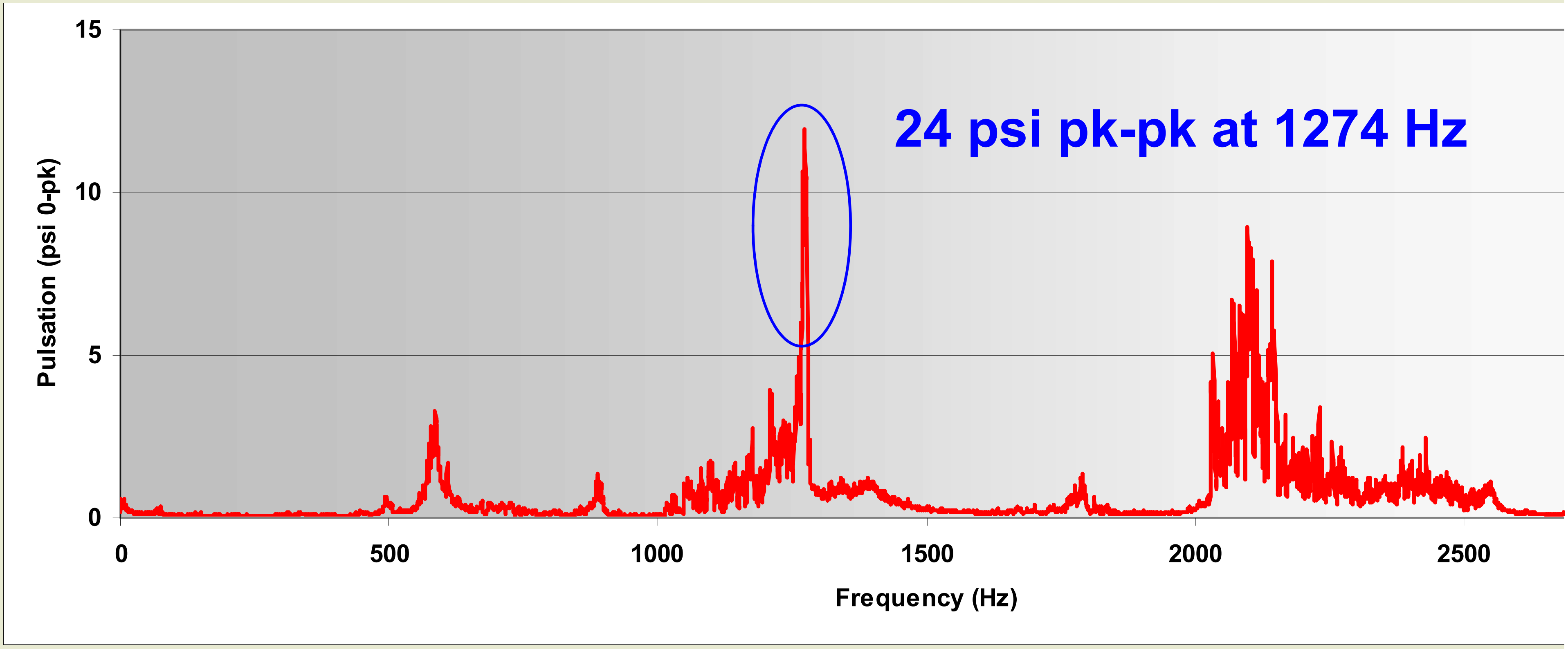
C) Case Study. Collect Data - Initial Investigation

- Field measurements of:
  - Pulsations in suction, suction eye and discharge
  - Vibration on suction and discharge piping

C) Case Study. Collect Data – Pulsation

Discharge Piping (3600 – 4500 rpm)

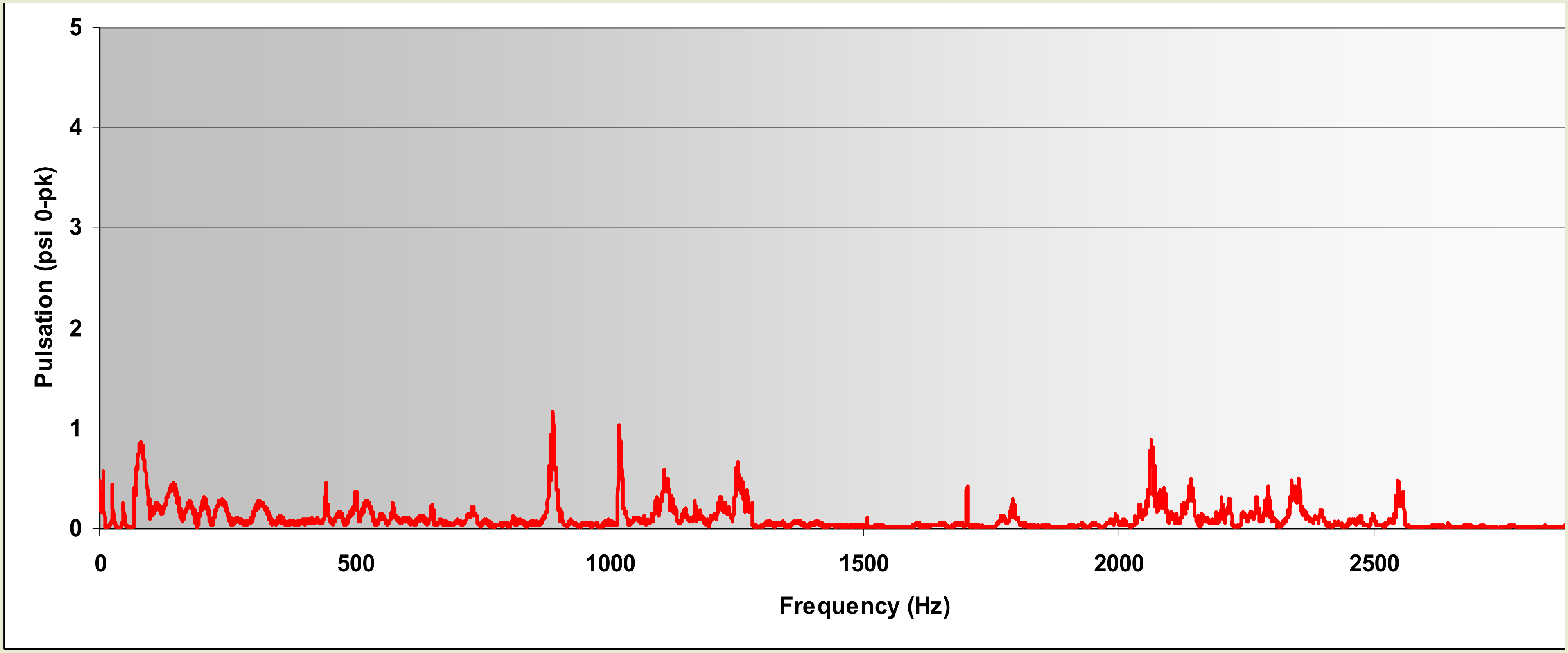
- Peak pulsation = 1.65 bar pk-pk (24 psi pk-pk) @ 1274 Hz
- About 3% of mean pressure
- 1274 Hz = blade pass frequency at 4500 RPM (17 X Run Speed)



C) Case Study. Collect Data – Pulsation

Compressor Suction Eye (3600 – 4500 rpm)

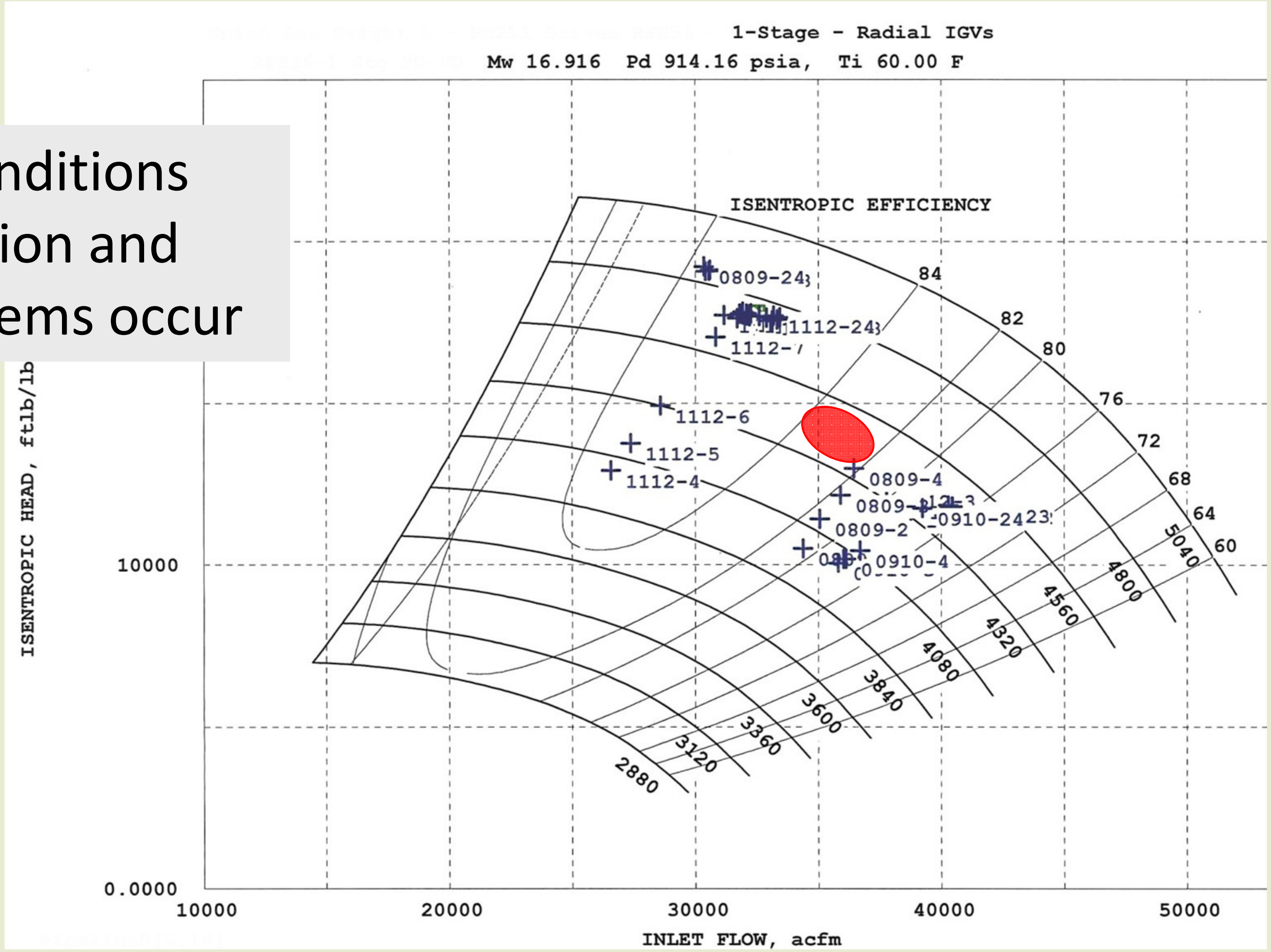
- Peak pulsation = 0.17 bar pk-pk (2.4 psi pk-pk) @ 830 Hz
- About 0.4% of mean pressure
- No Blade Pass Frequency found



C) Case Study. Collect Data – Source Strength

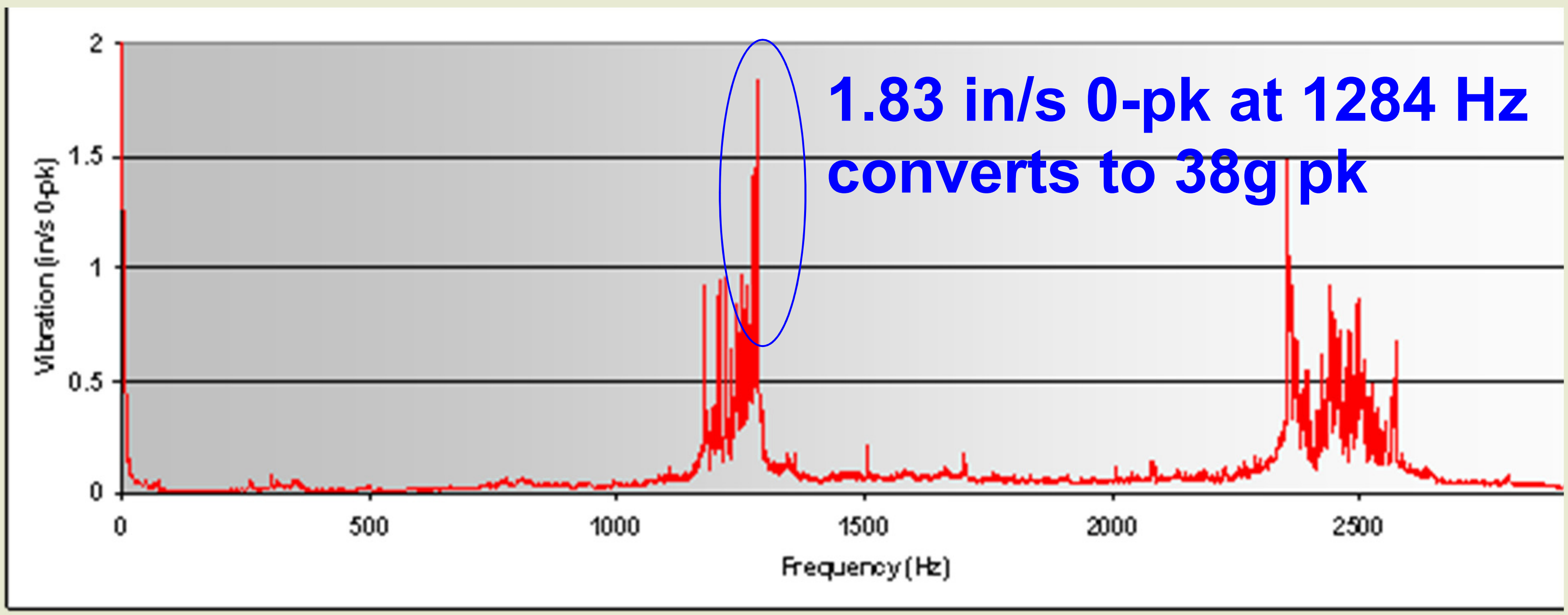
- Pulsation strength varies with operating conditions.
- Problems occur when compressor operates off its best efficient point

Operating conditions where vibration and pulsation problems occur



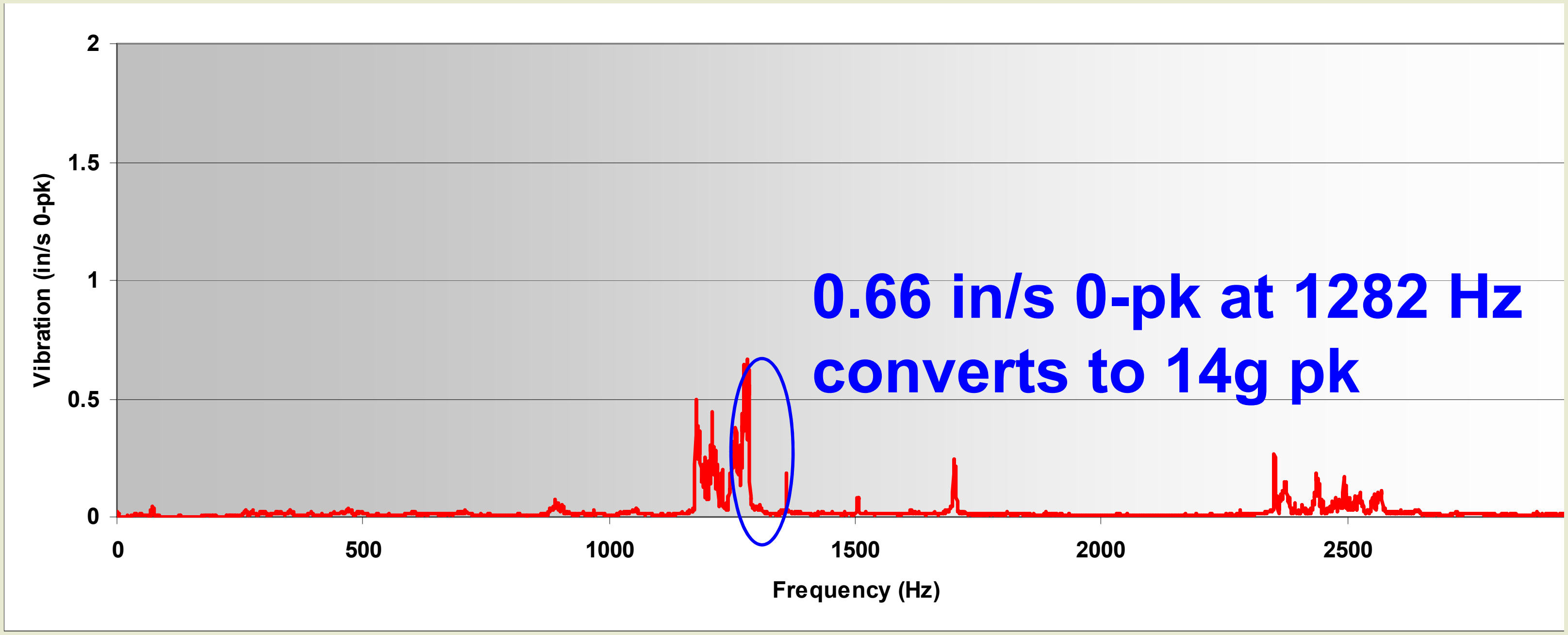
C) Case Study. Collect Data – Discharge Vibration

- Peak hold vibration @ speed range (4100 – 4510 rpm)
- 46.5 mm/s pk (32.9 mm/s rms, 1.83 in/s pk) @1284 Hz
- Discharge pipe (36" OD; 0.75" wall thickness)
- 1278 Hz = blade pass frequency at 4510 rpm
- Vibration equivalent to 38 g's pk (instruments rated for 10 g's)….no wonder RTDs failed



C) Case Study. Data – Suction Vibration

- Peak hold vibration @ speed range (4100 – 4510 rpm)
- 16.8 mm/s pk (11.9 mm/s rms, 0.66 in/s pk) @1282 Hz
- Suction pipe 914.4 mm OD (36” OD); 19 mm wall (0.75”)
- 1278 Hz = blade pass frequency at 4510 rpm
- Vibration equivalent to 14 g’s pk



C) Case Study. Analysis – Determine Causes

Evaluate Possible Root Causes of Vibration

- Peak Pulsation Frequency = Peak Piping Vibration Frequency (17X run speed)
  - Impeller Blade Pass = 17X run speed
- Thermowell Vortex Shedding and Natural Frequencies ≠ Problem Frequency
- Occurs at Steady State. Worst when not operating at Best Efficiency Point
- No likely Side Branch Sources
- Most likely source was Pulsation at Impeller Blade Pass

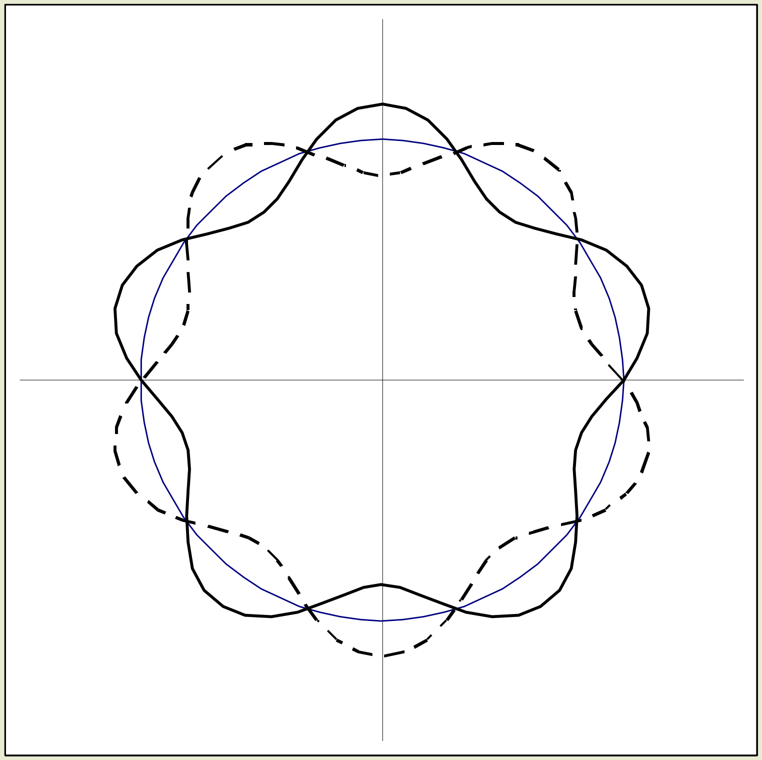
C) Case Study. Analysis – Determine Causes

Evaluate Possible Root Causes of Vibration

- Conduct an "Operating Deflected Shape (ODS)" analysis
  - Identify relative vibration (compared to reference point)
  - Requires field test
- Goal:
  - Define piping mode shape
  - Compare to calculated mechanical and acoustical mode shape

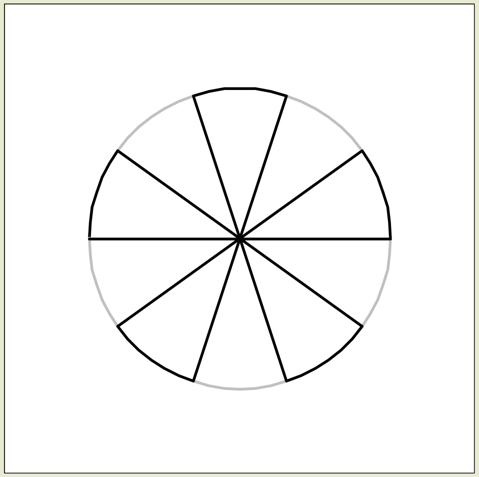
C) Case Study. Analysis – Piping Vibration ODS Results

- 5 lobe, circumferential mode shape of 36” discharge pipe
- Measured the amplitude and phase of vibration

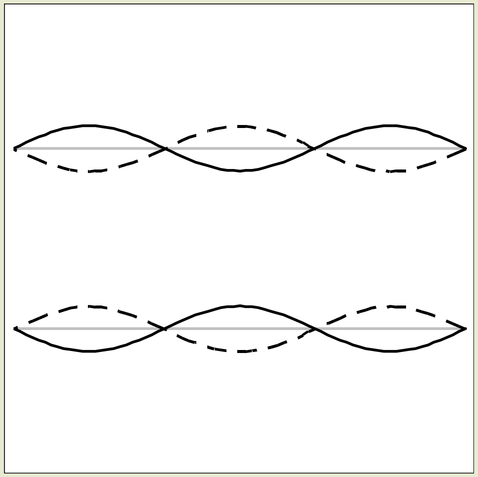


C) Case Study. Analysis – Acoustic Natural Frequencies (ANF)

- Calculated ANF
- High frequency – not a simple plane wave
- Many possible ANFs with transverse and axial components
- ANF closely matched measured vibration/ODS
  - 1261 Hz vs. measured 1274 Hz
  - 5 pressure lobes in transverse mode
  - 3 nodes along axial length



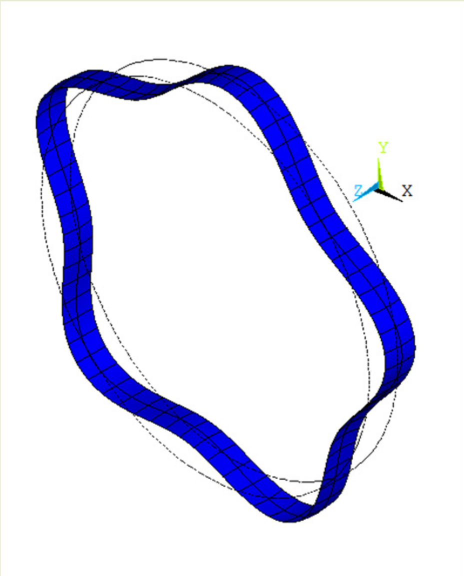
5 lobe plane acoustic mode

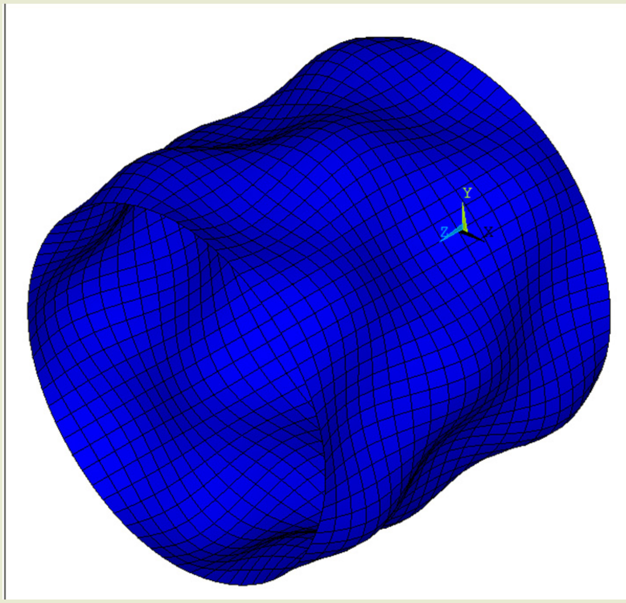


Axial acoustic mode

C) Case Study. Analysis – Piping Mechanical Natural Frequency (MNF)

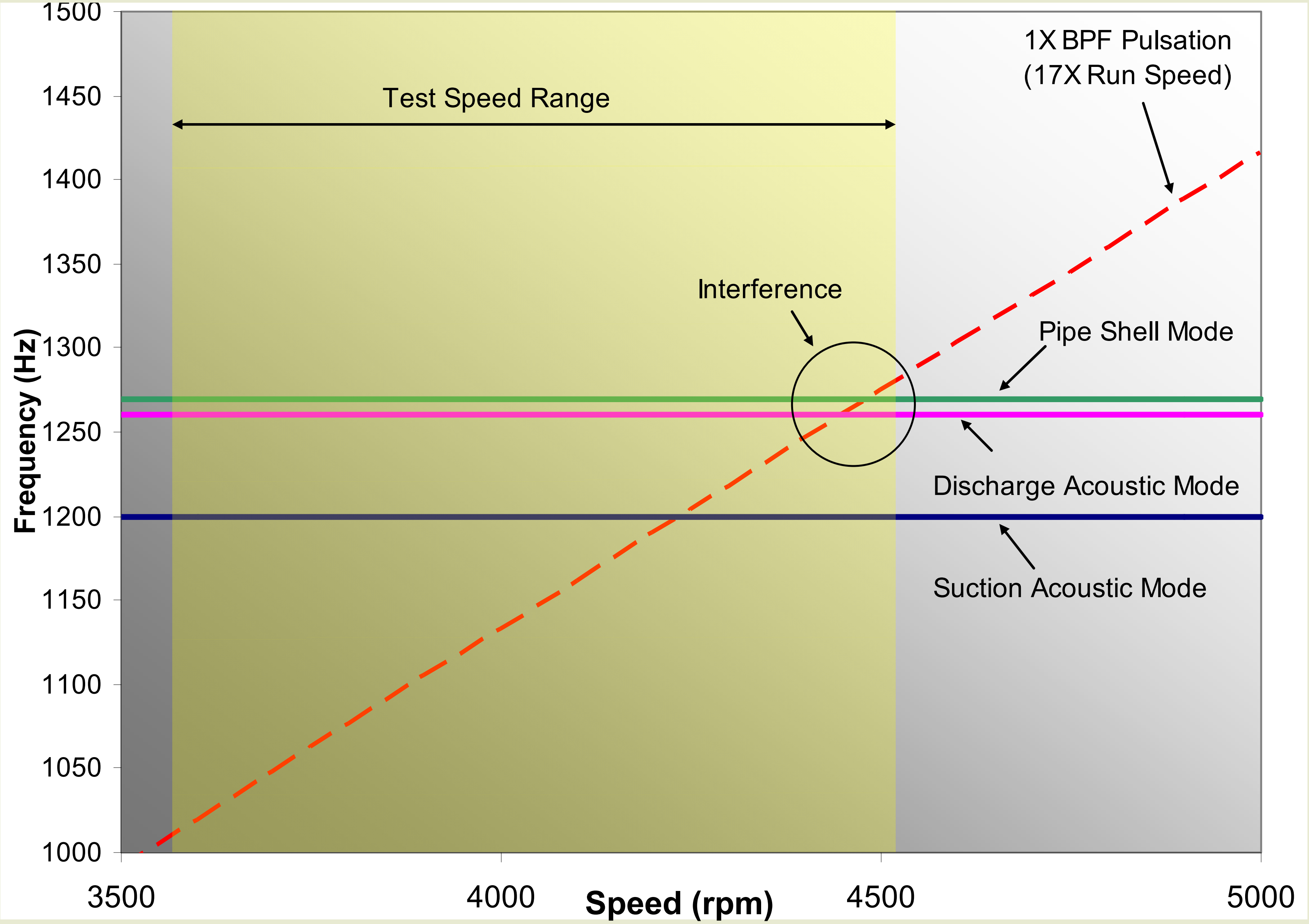
- Finite Element Model – discharge pipe
- Matching mode shape @ 1266 Hz
- Implications:
  - Coincident ANF and MNF with matching mode shapes
  - These would be highly coupled



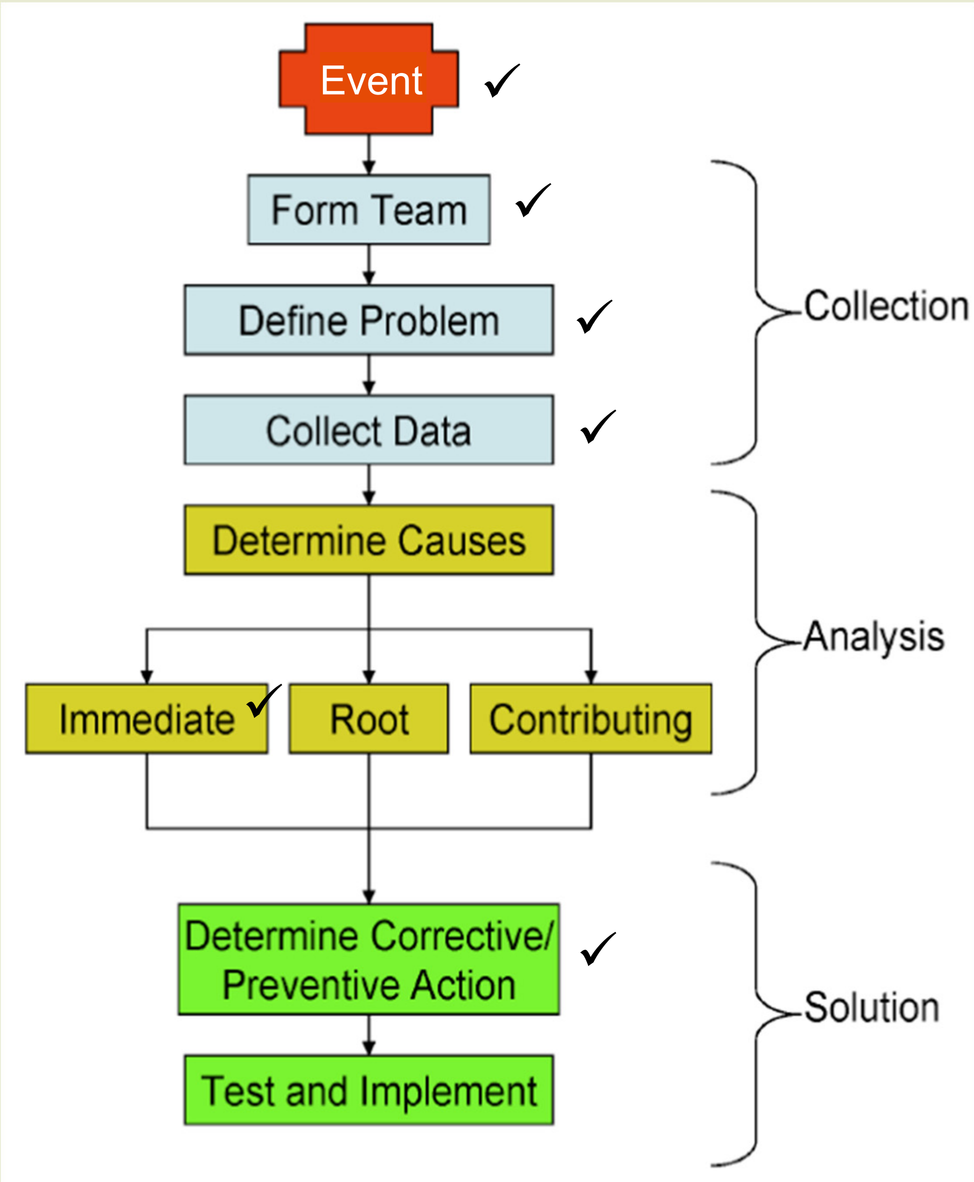


1266 Hz shell mode of the pipe with 5 lobes circumferentially and 3 nodes axially

C) Case Study. Analysis – Interference Plot…the "perfect storm"



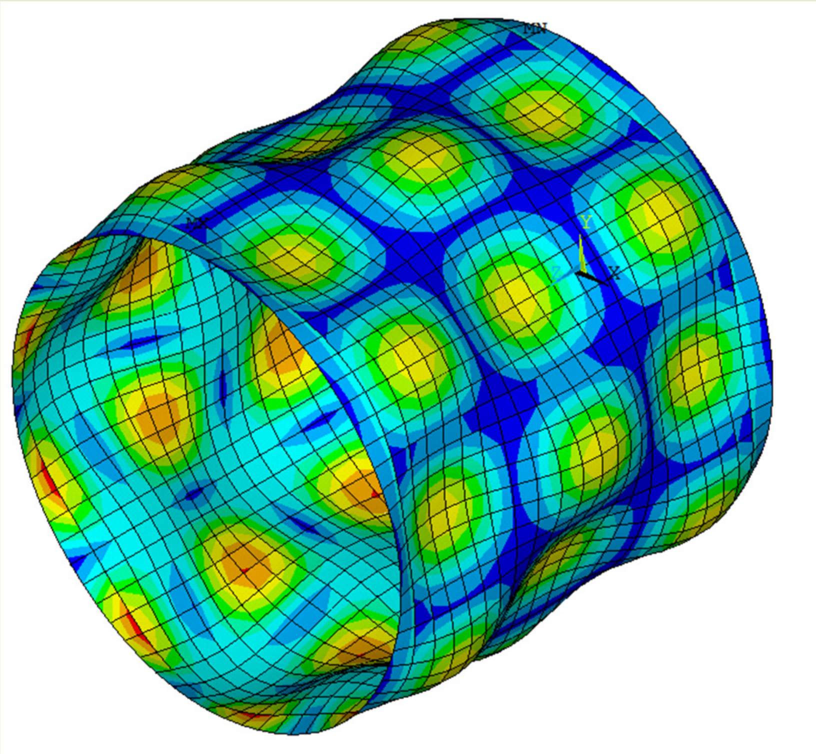
C) Case Study - Root Cause Analysis Approach



C) Case Study. Solution – Corrective/Preventive Action

What is immediate fatigue failure potential?

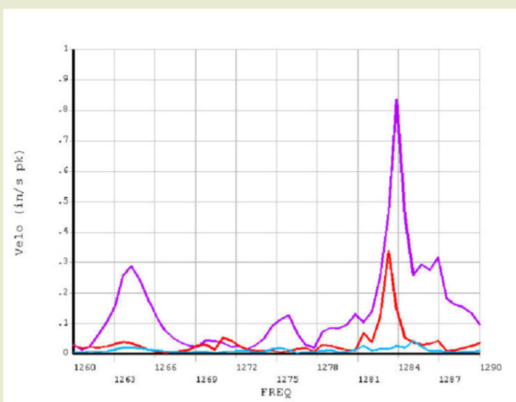
- Stress calculated using measured displacement , 4.9 MPa pk-pk (710 psi pk-pk)
- Pipe stress is well below endurance limit
- Small bore connections still at risk

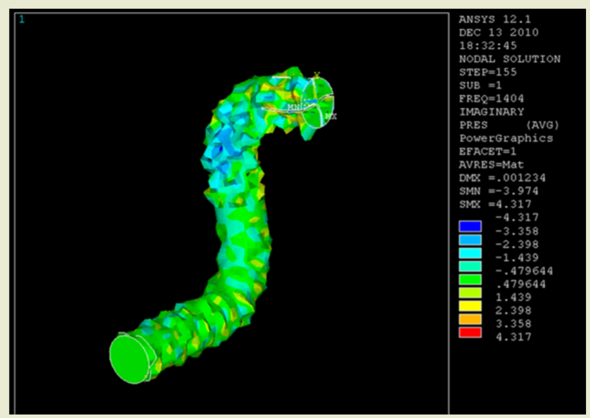


Stress plot for the 1266 Hz pipe shell mode

C) Case Study. Solution – Corrective/Preventive Action

- Recommendations to fix the problem (ongoing resolution)
  - Reduce or Eliminate Blade Pass Frequency Source:
    - Change Impeller – not likely
    - Remove vane diffusers – tested, but no significant improvement
  - Change Piping Design:
    - Add Internal Splitters or External Stiffeners
    - Thicker Wall – Finite Element Model Predicts most Effective





  - Eliminate Small Bore Connections Near Compressor
  - Improve Pipe Clamps

C) Case Study. Solution – Test and Implement

- Implementation
  - Relocation/removal of small-bore connections near compressor
  - Possible future discharge piping wall thickness change, but cost is a major factor

- Lessons Learned
  - Energy Institute (EI) Standard: good for screening problems such as small bore, flow induced vibration (FIV), acoustical induced vibration (AIV), and transients
  - Screen or analyze at Design Stage
  - Include pulsations at Blade Pass in the scope of work