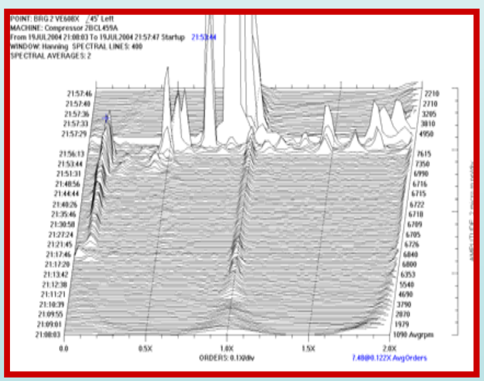
Investigation of a Rotordynamic Instability in a High Pressure Centrifugal Compressor Due to Damper Seal Clearance Divergence



J. Jeffrey Moore, Ph.D. ('91)

Southwest Research Institute

April 11, 2019

Moore, J.J., Camatti, M., Smalley, A.J., Vannini, G.V., Vermin, L.L., 2006, **Investigation of a Rotordynamic Instability in a High Pressure Centrifugal Compressor Due to Damper Seal Clearance Divergence**, *7th International Conference on Rotor Dynamics*, September 25-28, 2006, Vienna, Austria.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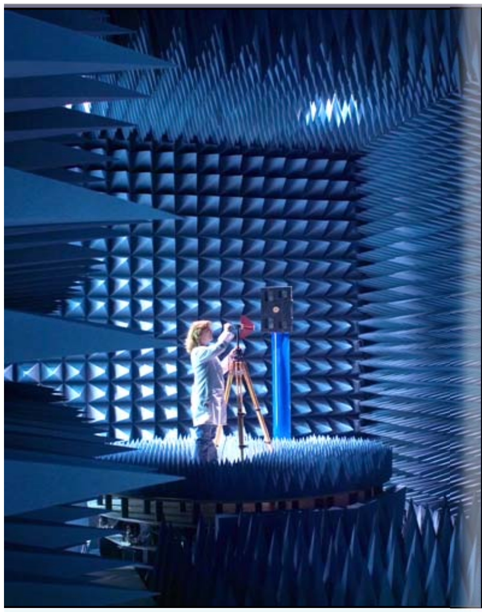

Applied Physics

Chemistry & Chemical Engineering

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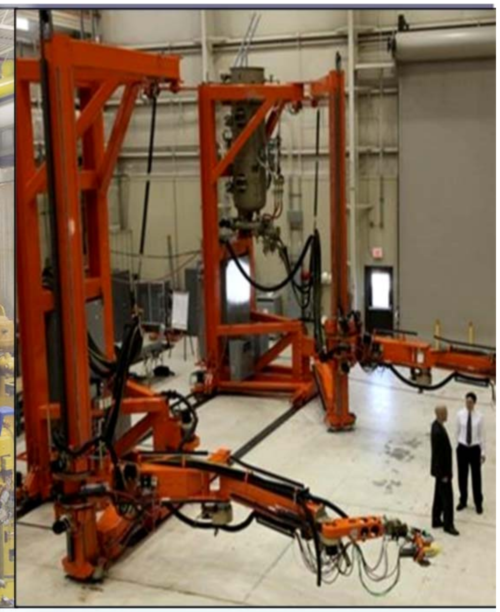
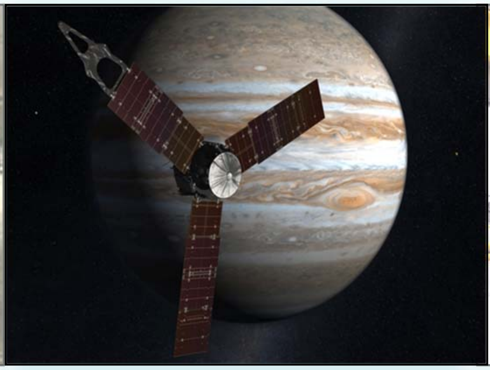


Intelligent Systems

Mechanical Engineering

Machinery Department

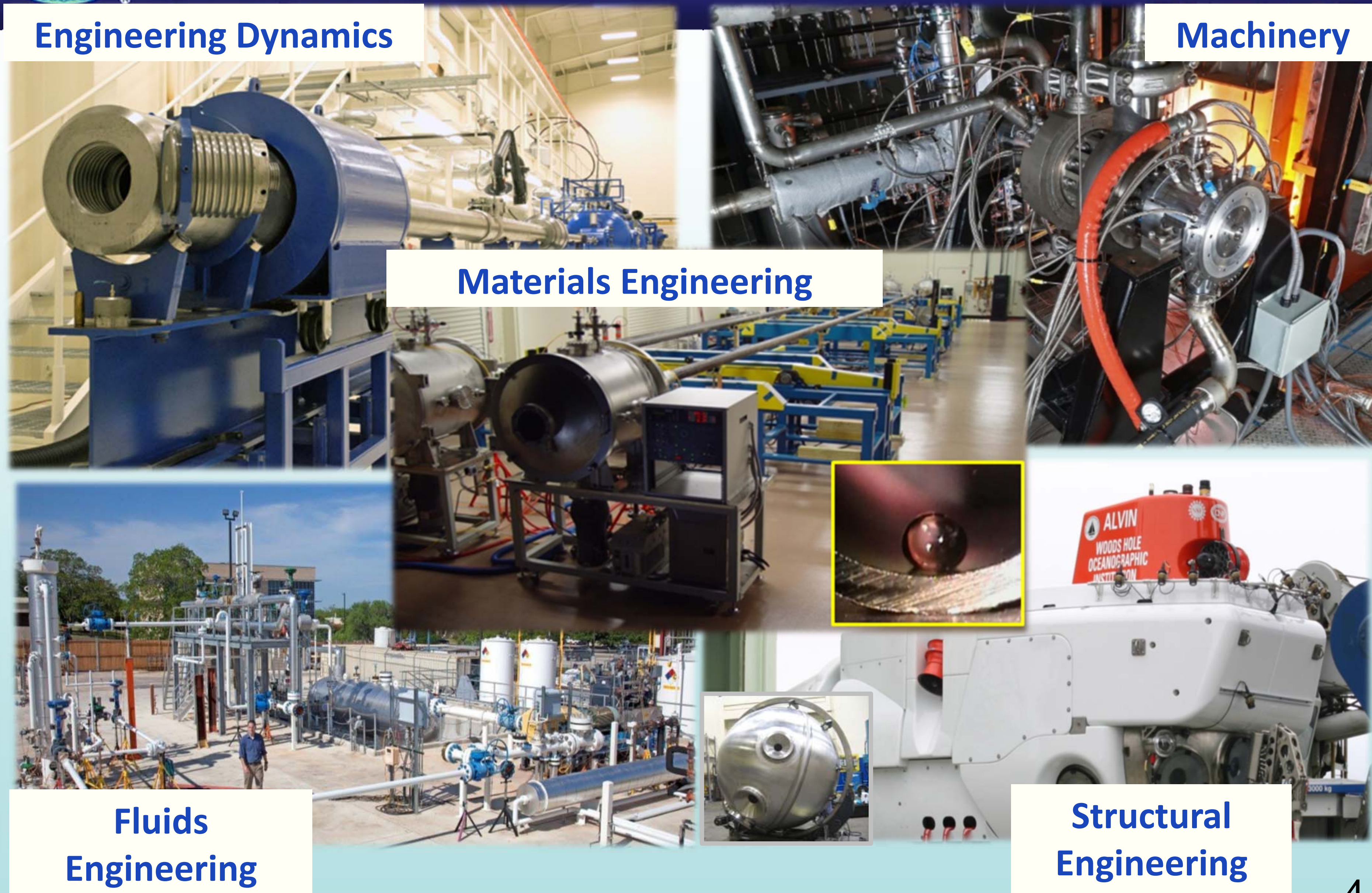
Defense & Intelligence Solutions



Space Science & Engineering

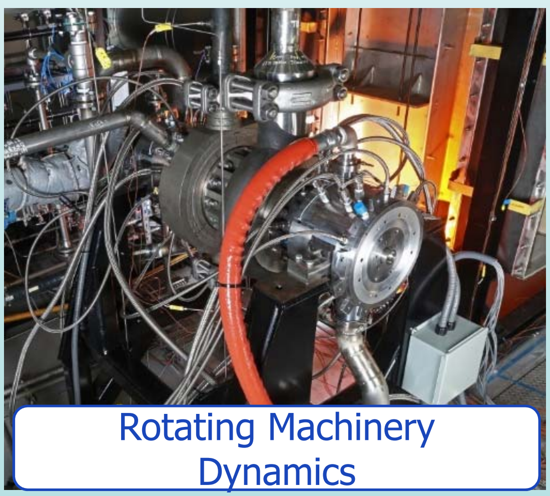
Powertrain Engineering

Mechanical Engineering Division

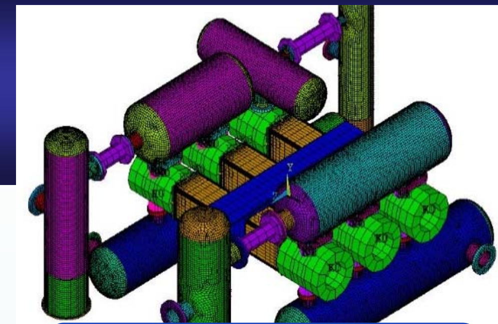


Machinery Department

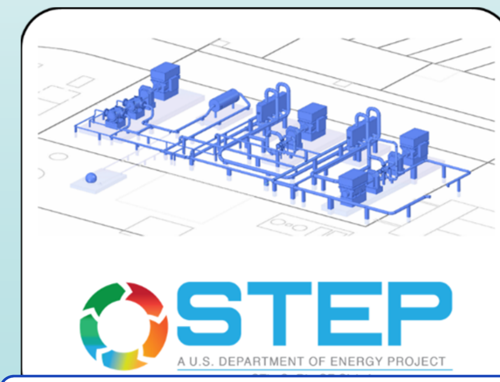

Propulsion & Energy Machinery


Rotating Machinery Dynamics

- 60 engineers/technicians
- Design, analysis and testing
- Turbomachinery
- Thermodynamic cycle
- Performance testing
- Life cycle analysis/design
- Mechanical systems (bearings, seals, etc.)
- Prototype development
- Computer aided engineering
  - CFD, FEA, CAD
- Rotordynamics
- Fluid/thermal systems


Fluid Machinery Systems


Machinery Services

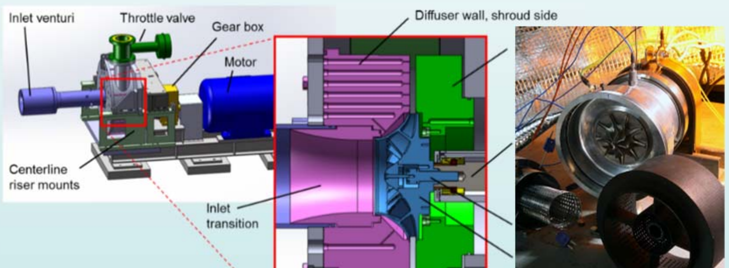


Power Cycle Machinery

Machinery Department Test Facilities

Multiphase Machinery Laboratory (B287)

Turbomachinery Laboratory (B278)



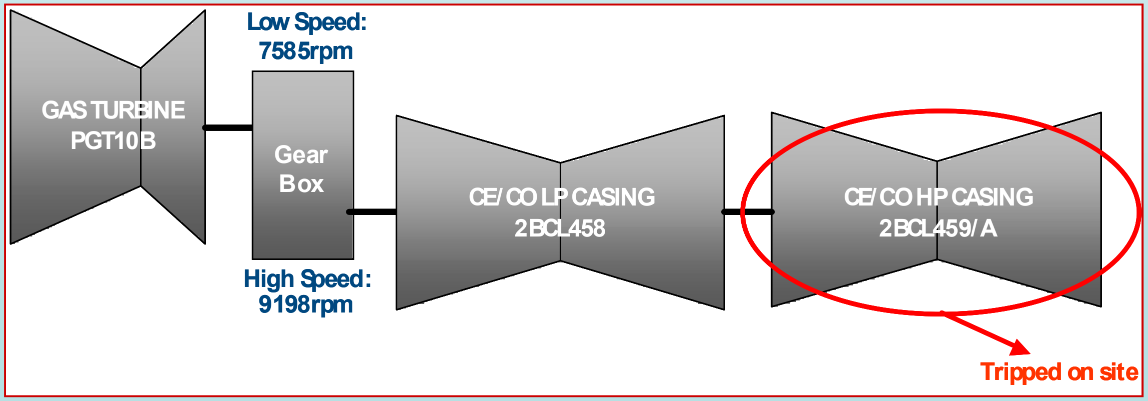
Gas Turbine Laboratory (B129)

Recip Compression Laboratory (B77)

- Operating Fluids: Air, CO_2, N_2
- Multiphase with Air/H_2O
- Drive power up to 4 MW
- Shaft speeds up to 140,000 rpm
- Machinery: centrifugal pumps & compressors, reciprocating compressors, and small gas turbine engines

Case Study: Rotordynamic Instability

- Two body compression train driven by 10 MW Gas Turbine through a gearbox
- Gas gathering application that feeds large LNG plant in Nigeria
- LP compressor is a 8 stage back-to-back design and is drive-through
- HP compressor is a 9 stage back-to-back design operating at about 10,000 rpm
- Total train pressure ratio is 48:1
- Instability experience on HP compressor during field start-up

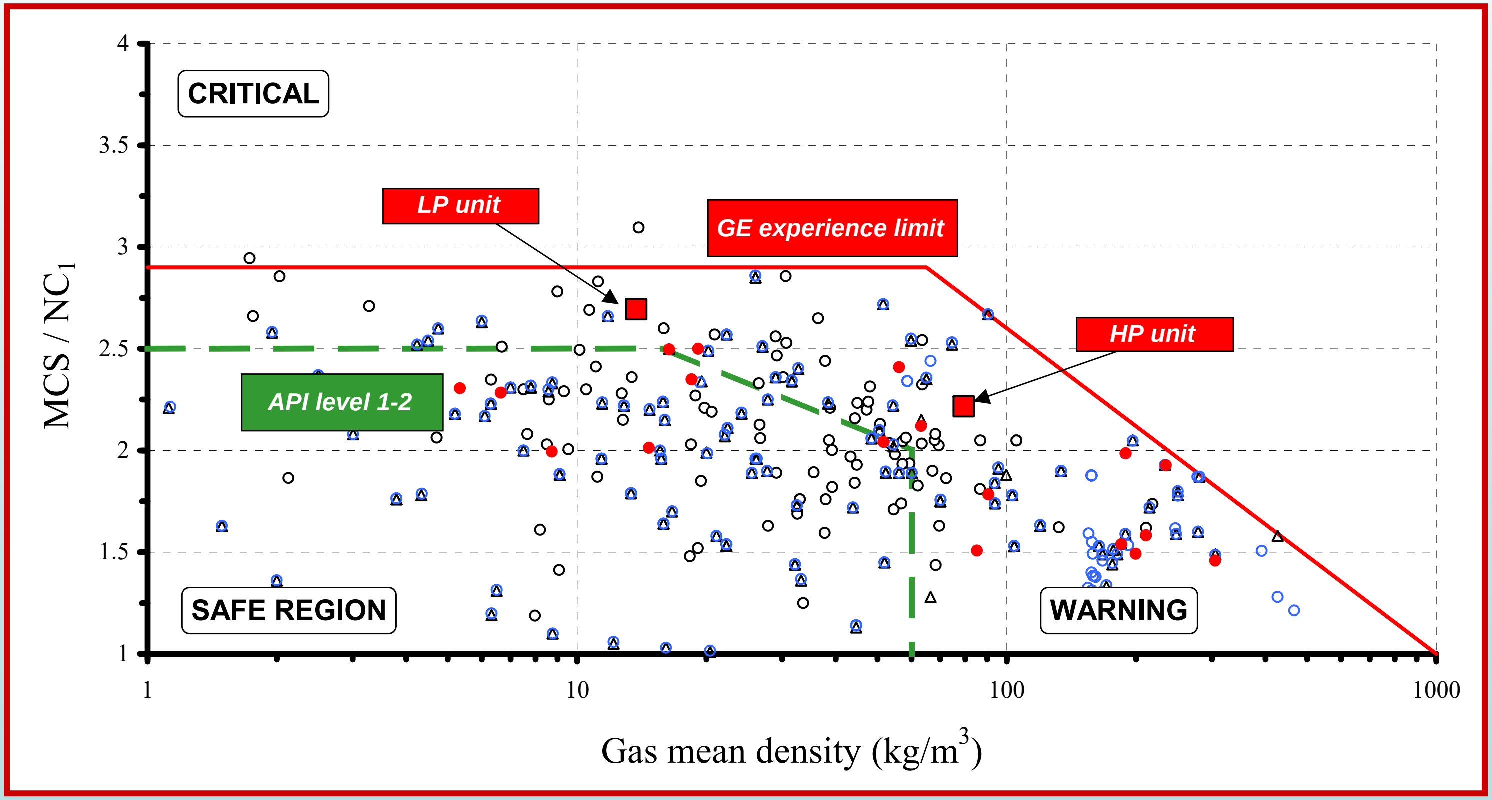


Original Compressor Seal Designs

- LP Compressor:
  - Tooth-on-stator labyrinth seal at both impeller eye and center balance piston locations
  - Shunt injection on center balance piston
  - No swirl brakes
- HP Compressor
  - Honeycomb damper seal used at center balance piston with
  - No swirl brake or shunt injection on any seal
- Design of compressor predated more recent experience with adverse effects of damper seals
- Design Pressure
  - P1=22 bar (319 psi)   P2=133 bar (1930 psi)
- Maximum Discharge Pressure
  - 189 bar (2740 psi)   (Maximum Continuous Speed & Near Surge)

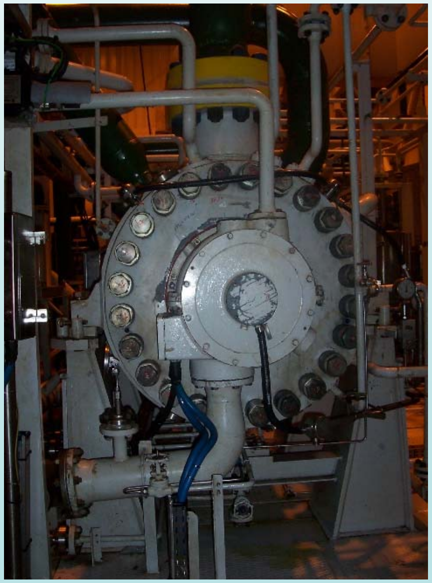
Experience Chart

- LP and HP compressor plotted on Fulton experience chart
- Both machines within experience limits



Site Description

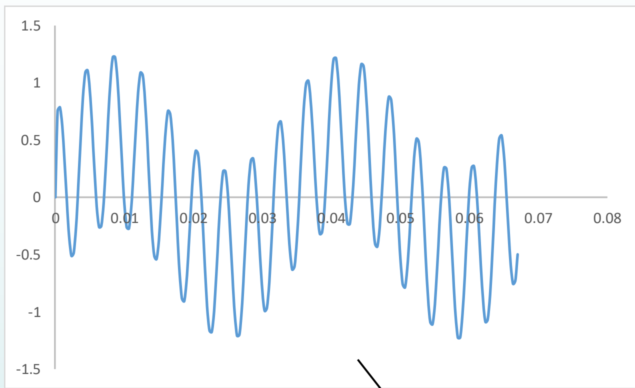
- Site located near the Niger Delta in Nigeria
- Gas is used to feed large LNG plant
- Instability not discovered until field start-up since units were not full load tested at the factory

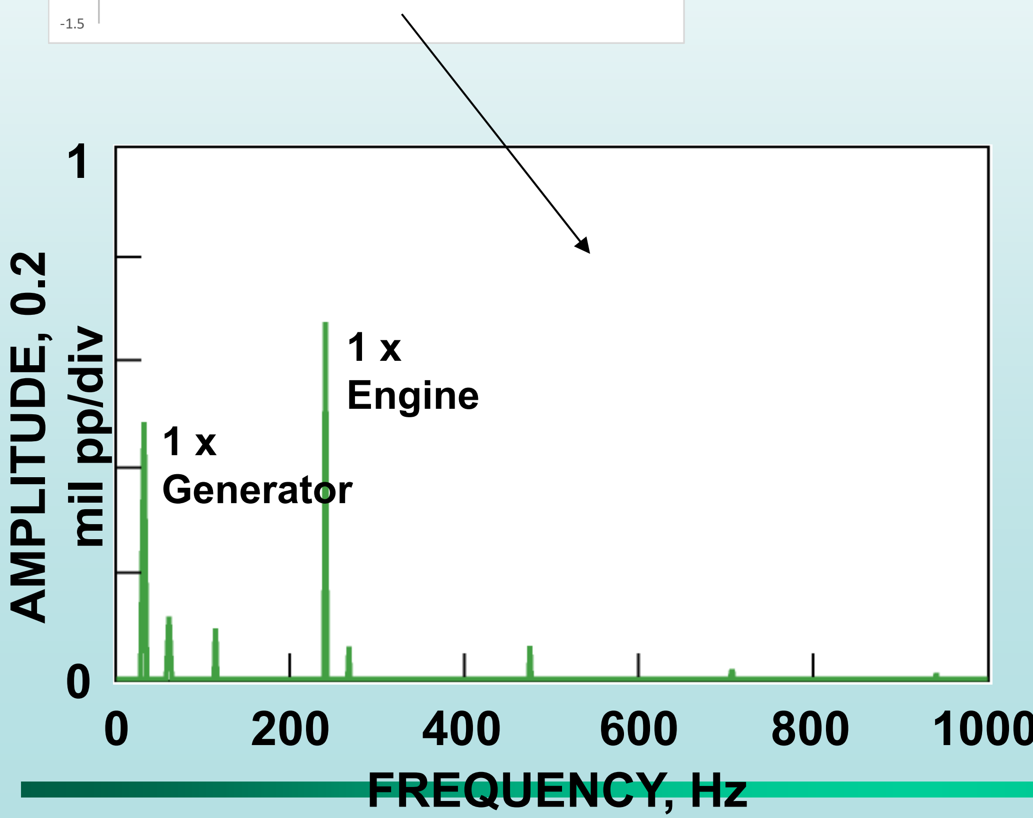


Displacement Probe System

- Measures displacement
- 0 – 2,000 Hz range
- Gain
  - 200 mV/mil or 7.87 V/mm
  - 192 mV/mil or 7.56 V/mm with Intrinsically Safe barrier
- Most effective for pk-pk measurements from 0 – 1,000 Hz

Typical Spectrum Plot

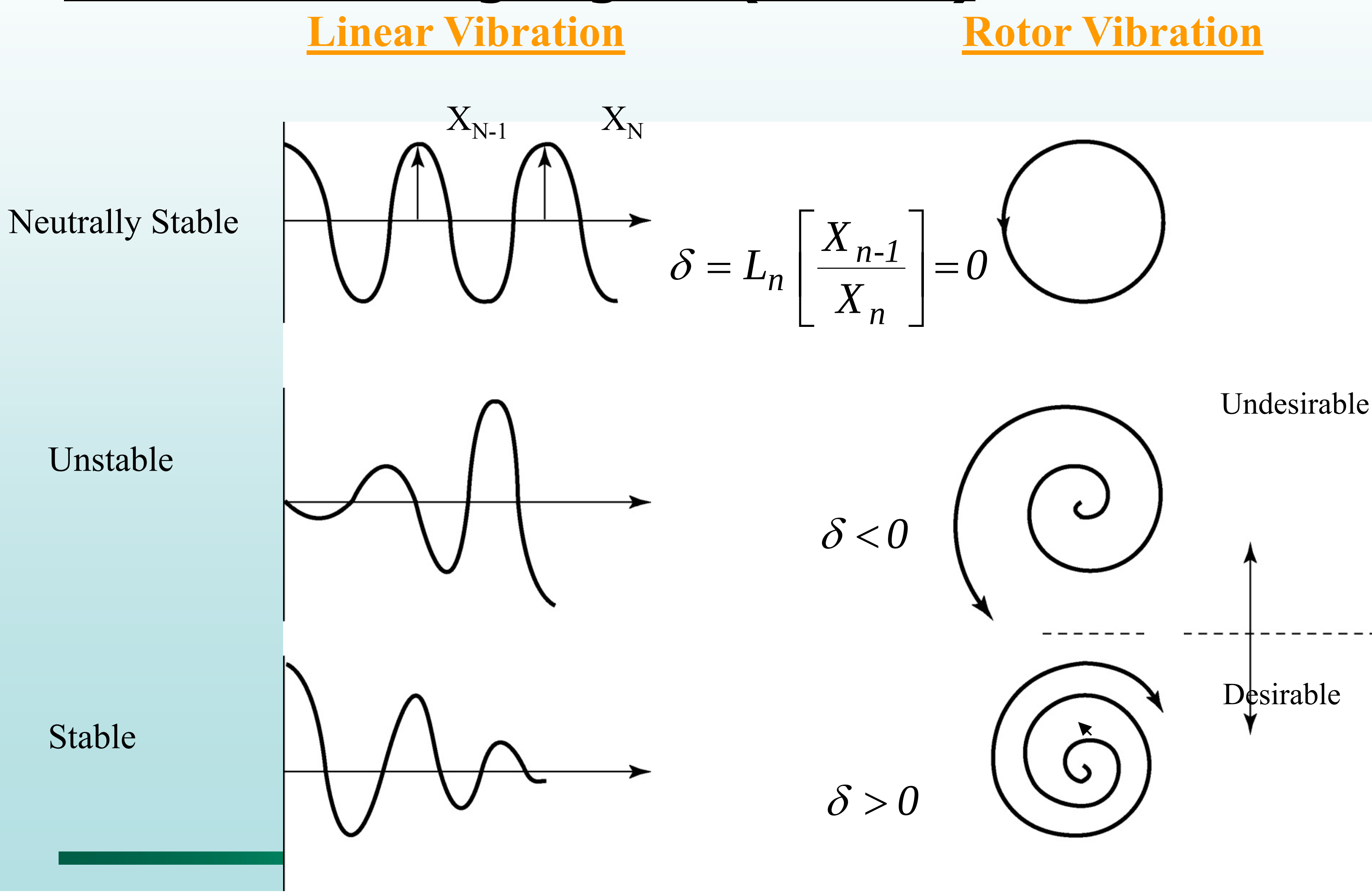




Typical spectral plot identifies

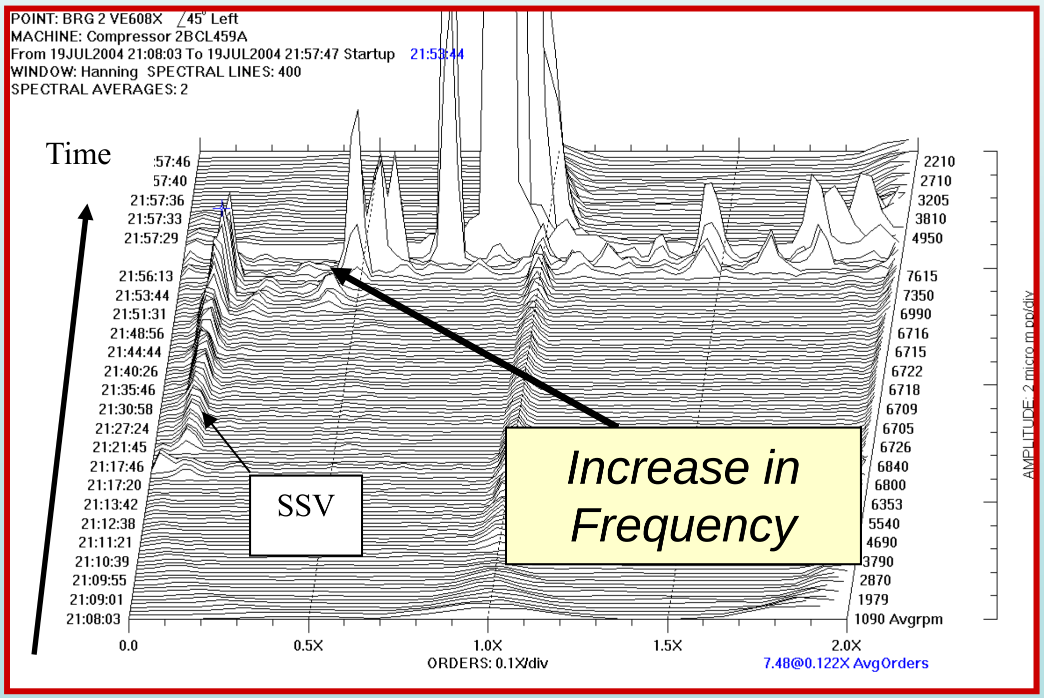
- Synchronous (1X) vibration
- Subsynchronous vibration
- Supersynchronous vibration
- Relative magnitudes of the discreet vibration frequencies
- Signal noise
- Random vibration
- Frequency domain data

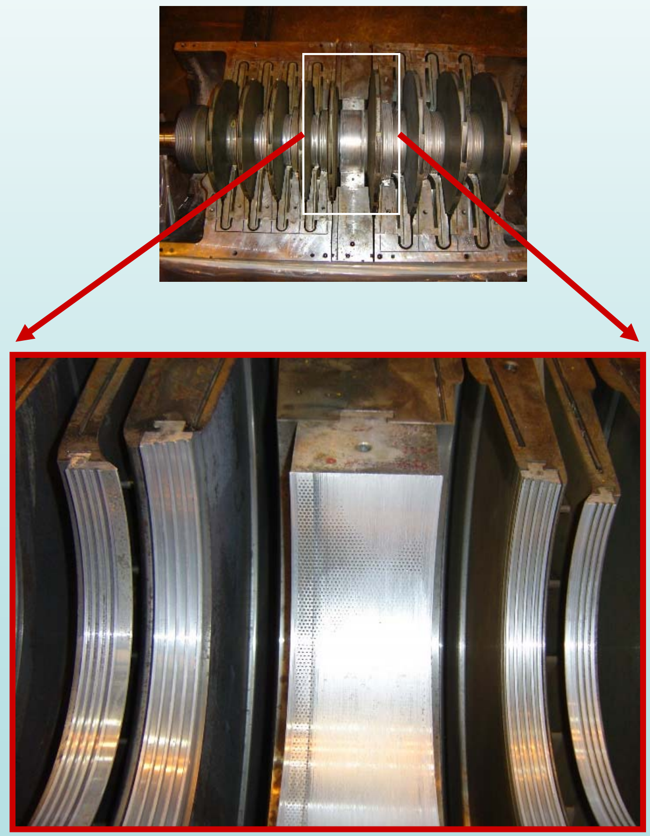
Evaluation Using Log Dec(rement)



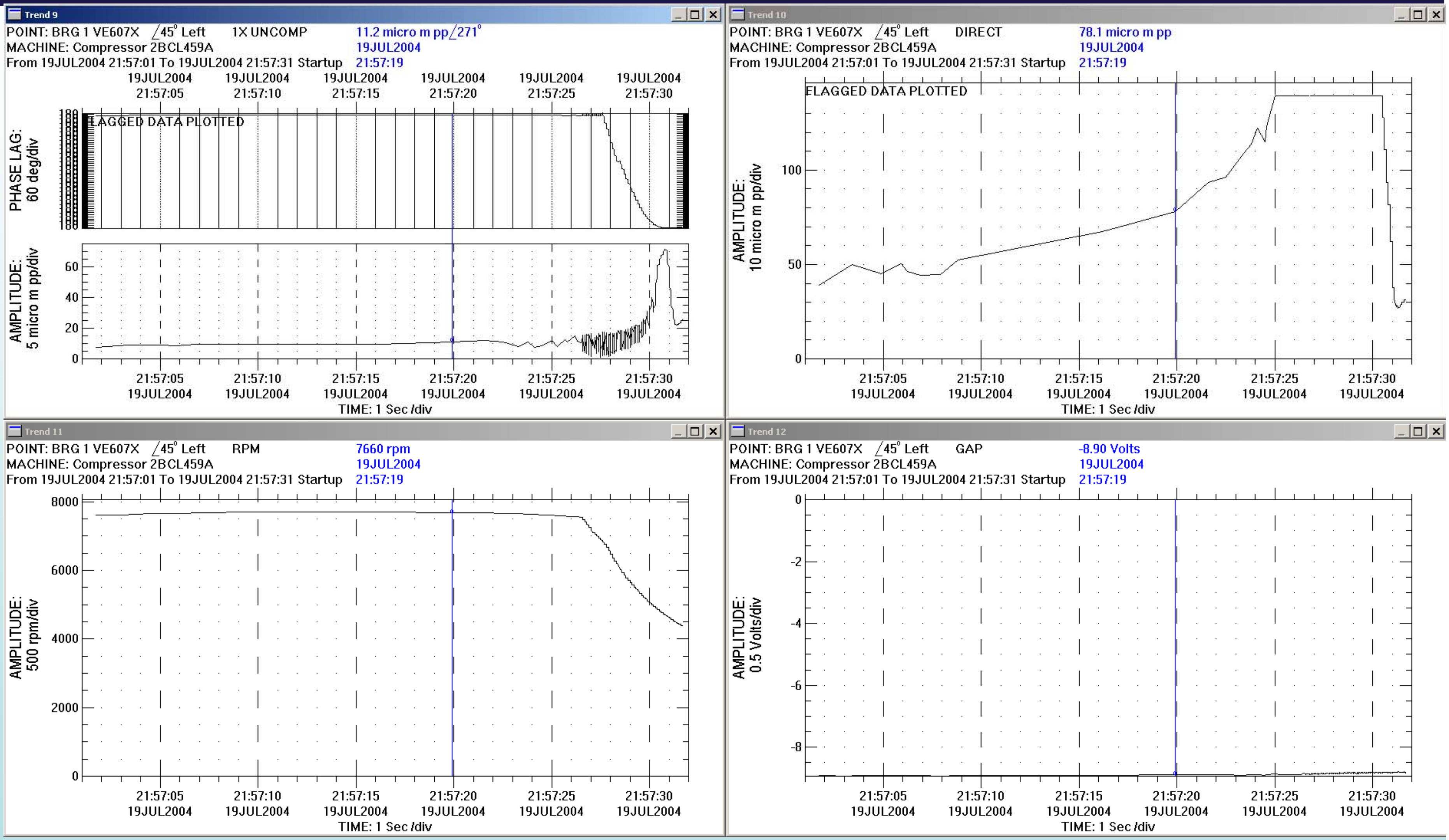
Field Vibration Measurements

- Subsynchronous Vibration (SSV) First Appeared at 12% of Running Speed
- Once Subsynchronous Amplitude Increased, Seal Rubbing Occurred Causing an Increase in Frequency
- Unit tripped out on high vibration
- All seals found to be heavily rubbed

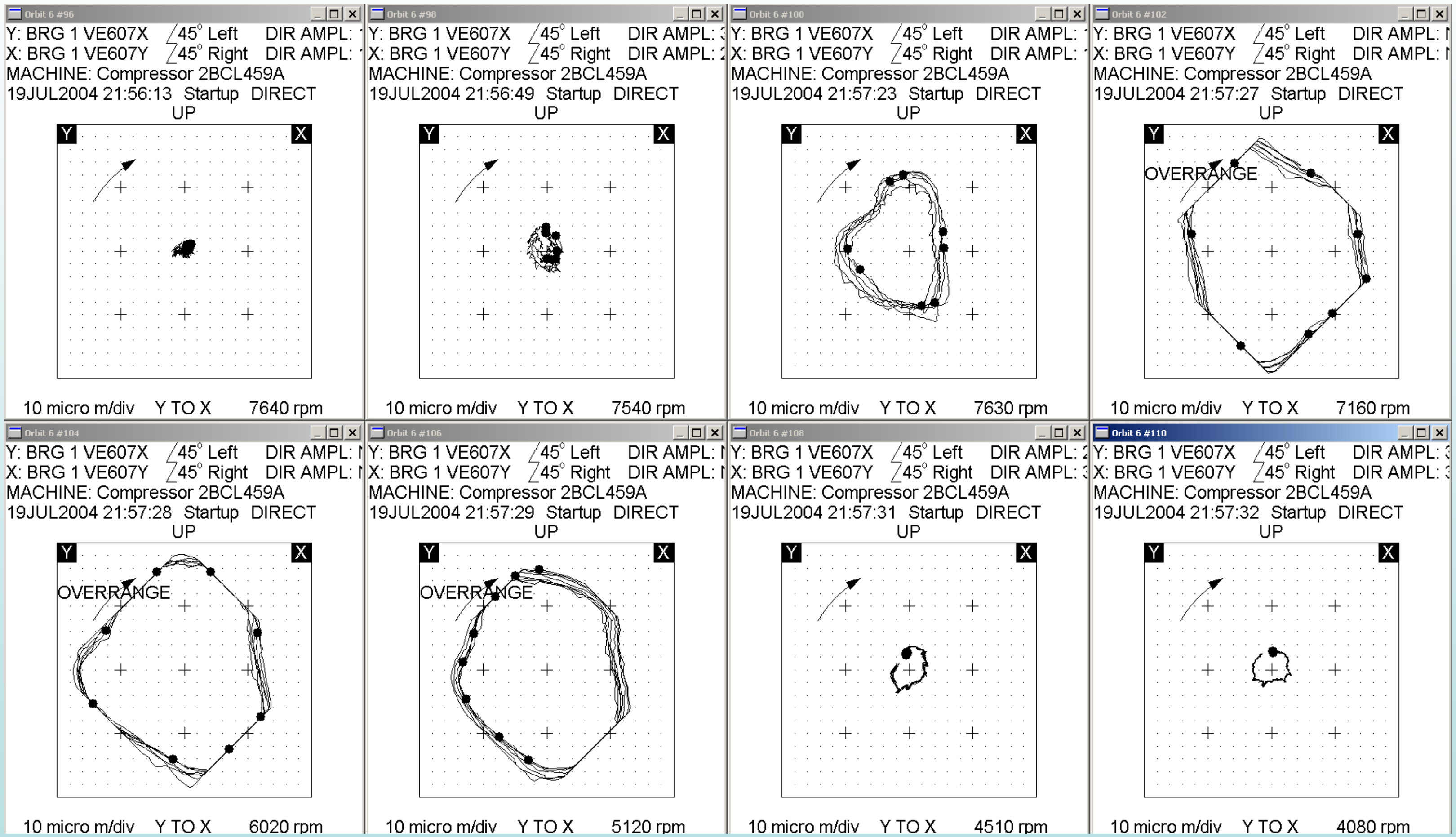




Brg 1 1X…………..Unfiltered
RPM…………..DC

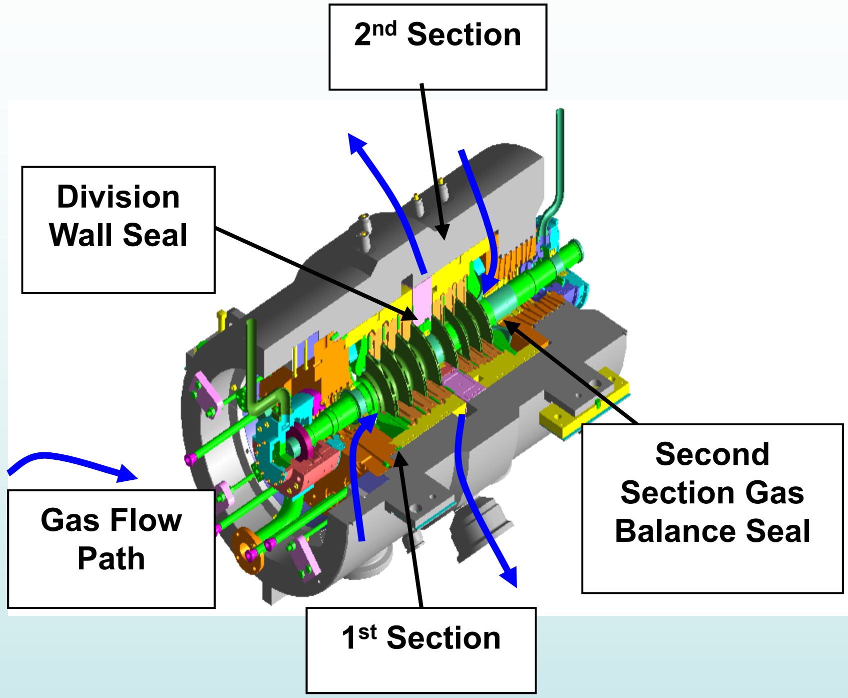


Brg 1 Orbits


Orbit 6 #96
Y: BRG 1 VE607X ∠45° Left DIR AMPL:
X: BRG 1 VE607Y ∠45° Right DIR AMPL:
MACHINE: Compressor 2BCL459A
19JUL2004 21:56:13 Startup DIRECT
UP
Y
X
10 micro m/div Y TO X 7640 rpm
Orbit 6 #98
Y: BRG 1 VE607X ∠45° Left DIR AMPL:
X: BRG 1 VE607Y ∠45° Right DIR AMPL:
MACHINE: Compressor 2BCL459A
19JUL2004 21:56:49 Startup DIRECT
UP
Y
X
10 micro m/div Y TO X 7540 rpm
Orbit 6 #100
Y: BRG 1 VE607X ∠45° Left DIR AMPL:
X: BRG 1 VE607Y ∠45° Right DIR AMPL:
MACHINE: Compressor 2BCL459A
19JUL2004 21:57:23 Startup DIRECT
UP
Y
X
10 micro m/div Y TO X 7630 rpm
Orbit 6 #102
Y: BRG 1 VE607X ∠45° Left DIR AMPL:
X: BRG 1 VE607Y ∠45° Right DIR AMPL:
MACHINE: Compressor 2BCL459A
19JUL2004 21:57:27 Startup DIRECT
UP
Y
X
OVERRANGE
10 micro m/div Y TO X 7160 rpm
Orbit 6 #104
Y: BRG 1 VE607X ∠45° Left DIR AMPL:
X: BRG 1 VE607Y ∠45° Right DIR AMPL:
MACHINE: Compressor 2BCL459A
19JUL2004 21:57:28 Startup DIRECT
UP
Y
X
OVERRANGE
10 micro m/div Y TO X 6020 rpm
Orbit 6 #106
Y: BRG 1 VE607X ∠45° Left DIR AMPL:
X: BRG 1 VE607Y ∠45° Right DIR AMPL:
MACHINE: Compressor 2BCL459A
19JUL2004 21:57:29 Startup DIRECT
UP
Y
X
OVERRANGE
10 micro m/div Y TO X 5120 rpm
Orbit 6 #108
Y: BRG 1 VE607X ∠45° Left DIR AMPL:
X: BRG 1 VE607Y ∠45° Right DIR AMPL:
MACHINE: Compressor 2BCL459A
19JUL2004 21:57:31 Startup DIRECT
UP
Y
X
10 micro m/div Y TO X 4510 rpm
Orbit 6 #110
Y: BRG 1 VE607X ∠45° Left DIR AMPL:
X: BRG 1 VE607Y ∠45° Right DIR AMPL:
MACHINE: Compressor 2BCL459A
19JUL2004 21:57:32 Startup DIRECT
UP
Y
X
10 micro m/div Y TO X 4080 rpm

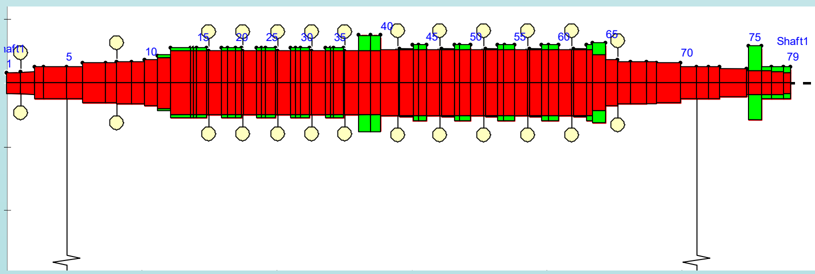
Rotordynamic Modeling

- Break the series of smaller segments at diameter steps
- Components like impellers, couplings, thrust disks do not add shaft stiffness are modeled as added mass
- Stations added at bearings centerlines



Typical High Pressure Centrifugal Compressor

Sample 10-Stage Compressor Model



Reference: Moore, J.J., Soulas, T.S., 2003, "Damper Seal Comparison in a High-Pressure Re-Injection Centrifugal Compressor During Full-Load, Full-Pressure Factory Testing Using Direct Rotordynamic Stability Measurement," Proceedings of the DETC '03 ASME 2003 Design Engineering Technical Conference, Chicago, IL, Sept. 2-6, 2003

Rotordynamic Theory

Modeling Turbomachinery

- Continuous system modeled by a system of springs and masses formulated using either finite element or transfer matrix methods
- Results in following system of equations:

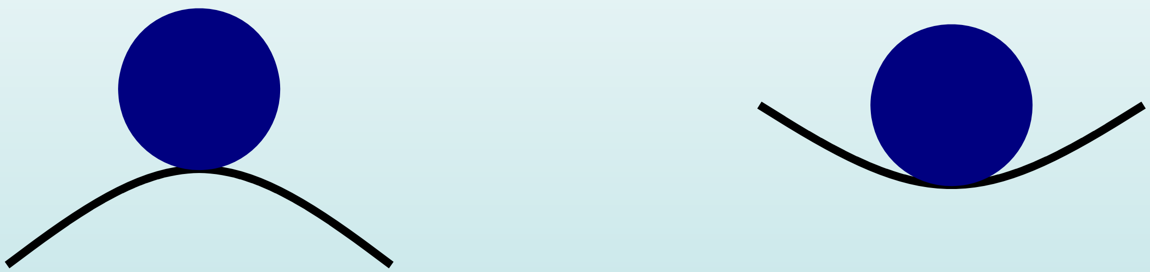
$$[M]\ddot{X} + [C]\dot{X} + [K]X = F(t)$$

- Similar form as the single degree of freedom
- Use Matrix solution techniques to solve for natural frequencies, unbalance response, and stability

Rotordynamic Theory

Stability Analysis

Unstable   Stable

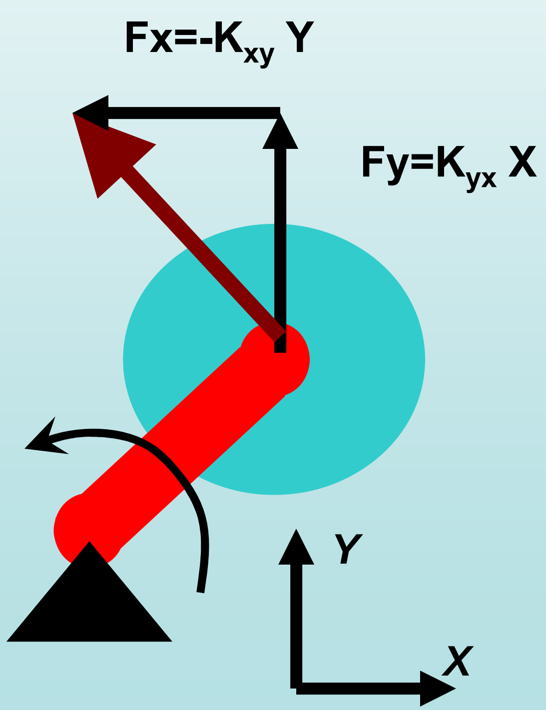


- A Rotor System Is Unstable When The Destabilizing Forces Exceed Stabilizing (Damping) Forces

Rotordynamic Theory

Stability Analysis

- Damping is a Stabilizing Influence
- Destabilizing Forces Arise from Cross-Coupling Effects that Generate Forces in the Direction of Whirl
- Cross-Coupled Stiffness Yields a force in the Y-direction for a displacement in the X
- Sources include: fixed arc bearings, floating ring oil seals, labyrinth seals, impeller/turbine stages



Rotordynamic Theory

- Stability Calculated by Solving the Eigenvalue Problem:

$$[M]\ddot{X} + [C]\dot{X} + [K]X = \{0\}$$

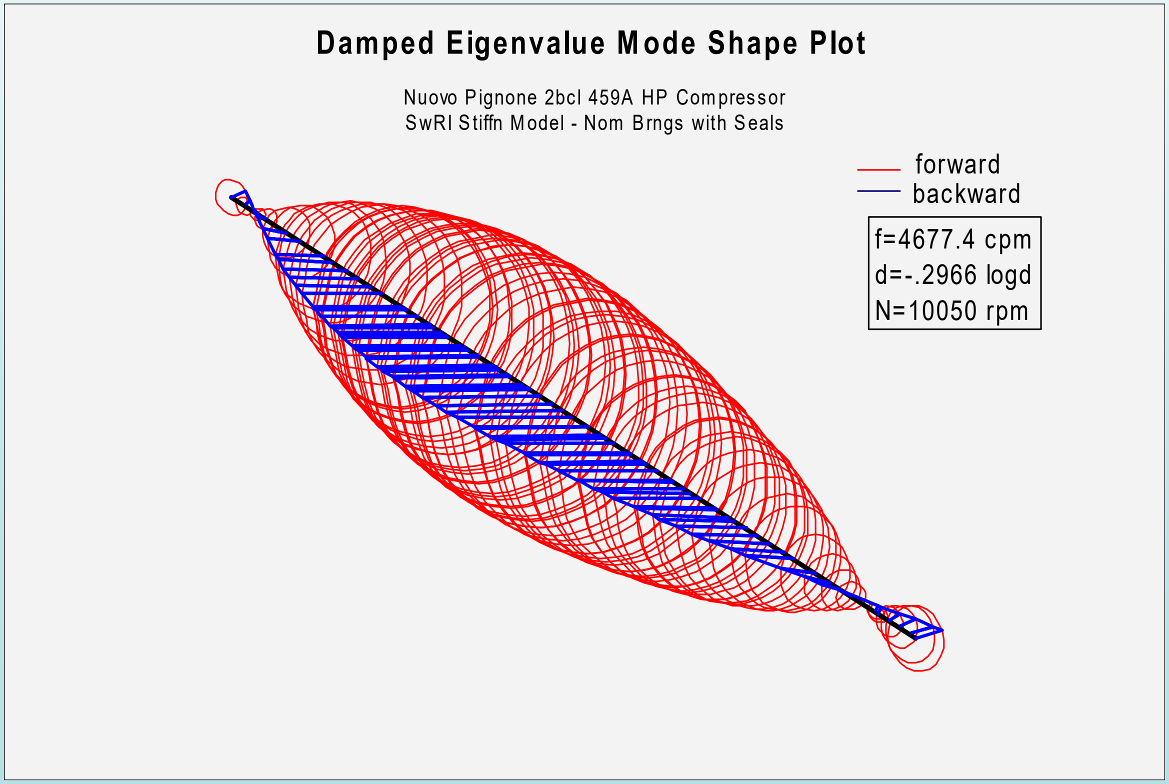
- Eigenvalues of the form: $s = -\zeta\,\omega_n + i\,\omega_d$
- Imaginary part gives the damped natural frequency
- Real part gives the damping ratio (ζ), or stability
- Logarithmic decrement (log dec) is related by:

$$\delta = \frac{2\pi\zeta}{\sqrt{1-\zeta^2}}$$

- Instability characterized by subsynchronous vibration near the first whirling frequency that rapidly grows to a large amplitude bounded only by rotor/stator rubbing
- Can be brought on by small changes in load, pressure, or speed.

Stability Analysis

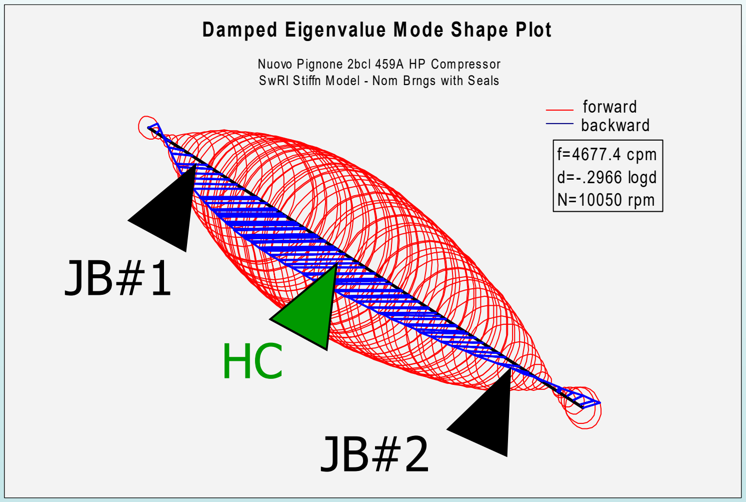
- **Mode Shape with API Impeller Excitation at MCS/Surge**

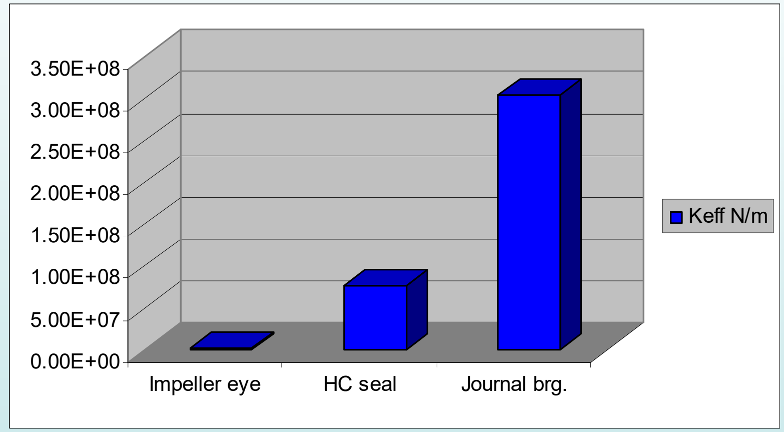


Effective Stiffness & Damping

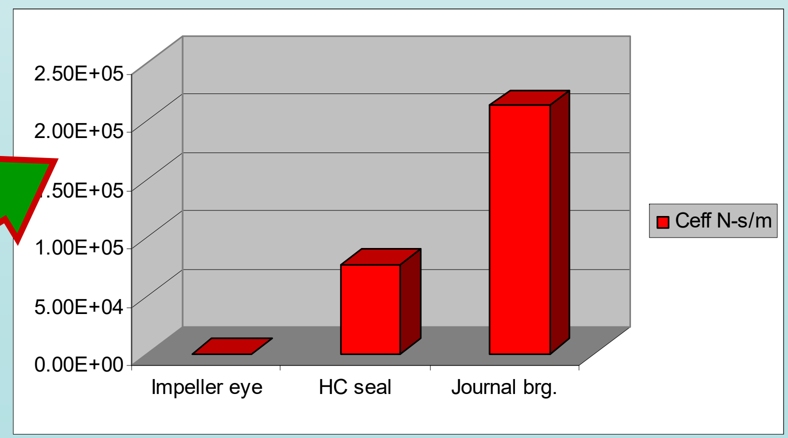
$$K_{effective} = K_{xx} + \omega C_{xy}$$

JB and HC seal shows same order effective stiffness and damping.





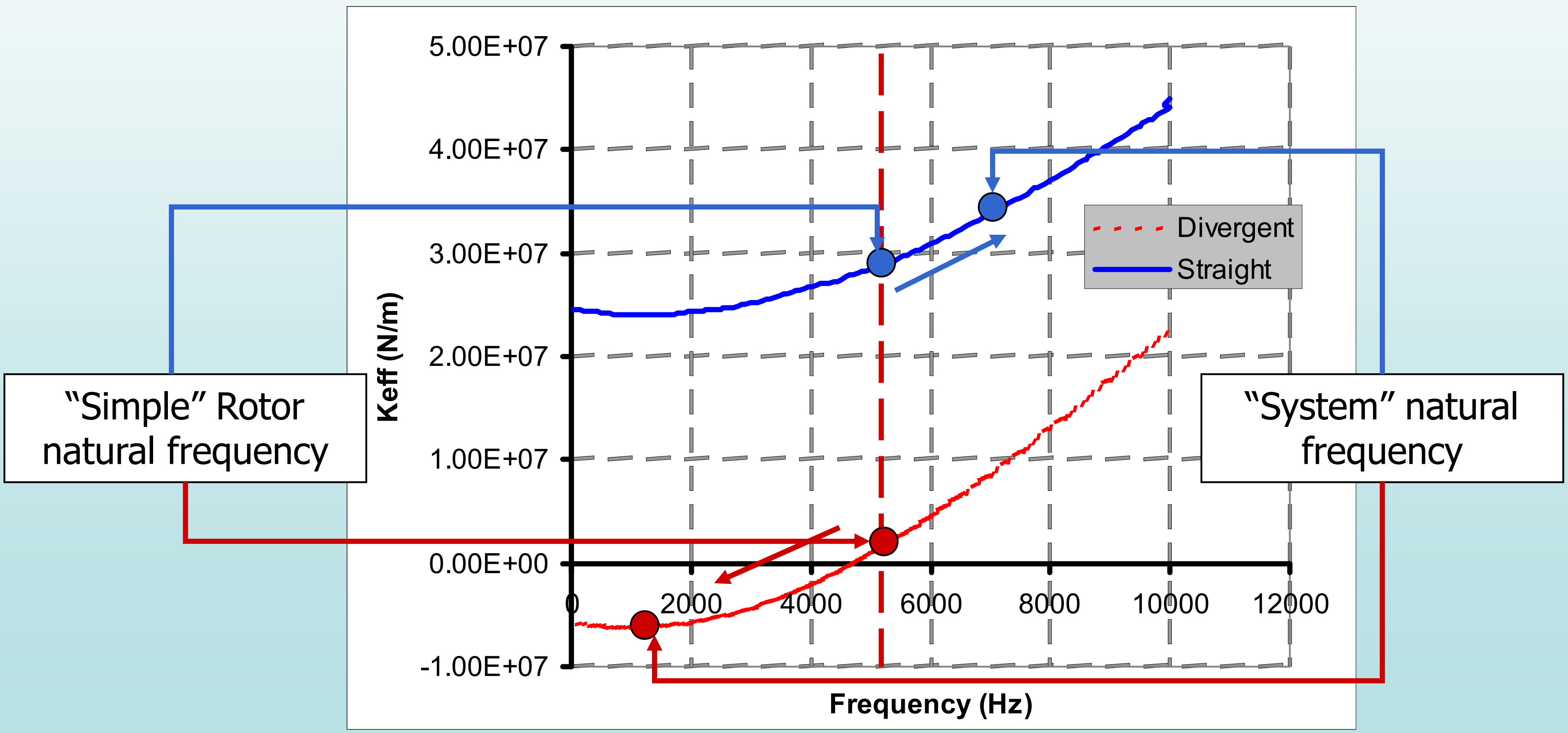
Due to midspan location HC seal plays a major role in rotor stability



$$C_{effective} = C_{xx} - K_{xy} / \omega$$

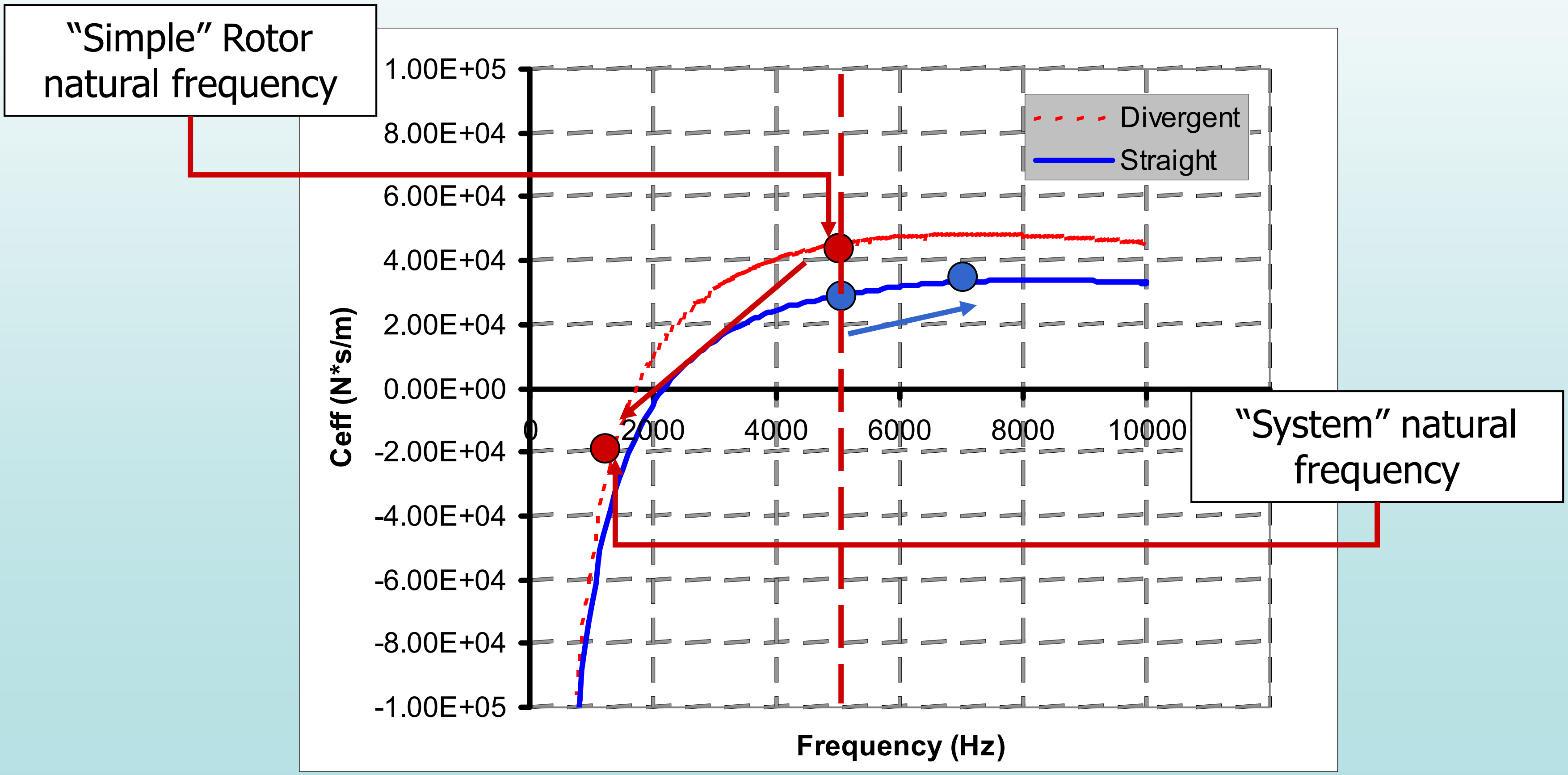
Effective Stiffness

- Low Frequency Stiffness can be strongly negative if HC is Divergent
- Honeycomb cross over frequency location with respect to “simple” rotor natural frequency is key factor for “system” natural frequency



Effective Damping

- $C_{effective}$ Varies w/Frequency
- If Natural Frequency in Region of Negative $C_{effective}$ => Rotor Unstable

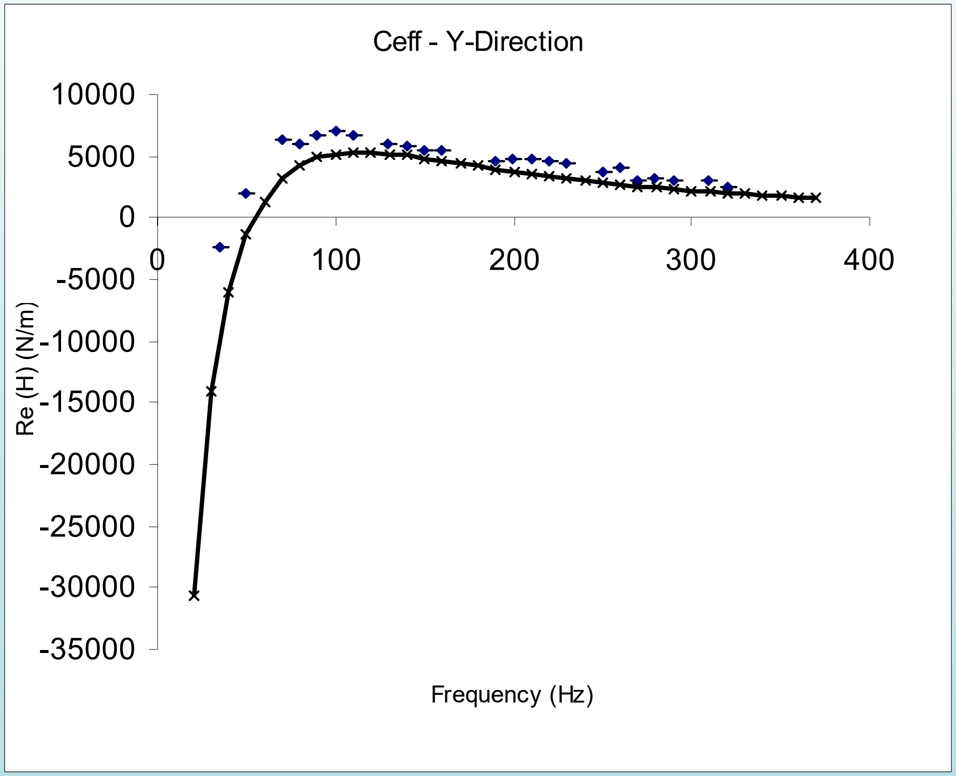


Damper Seal Testing

Honeycomb Seal Damping Test Data vs. Predictions

- Damper seals like honeycomb seals provide substantial damping
- Damping increases with increasing pressure differential

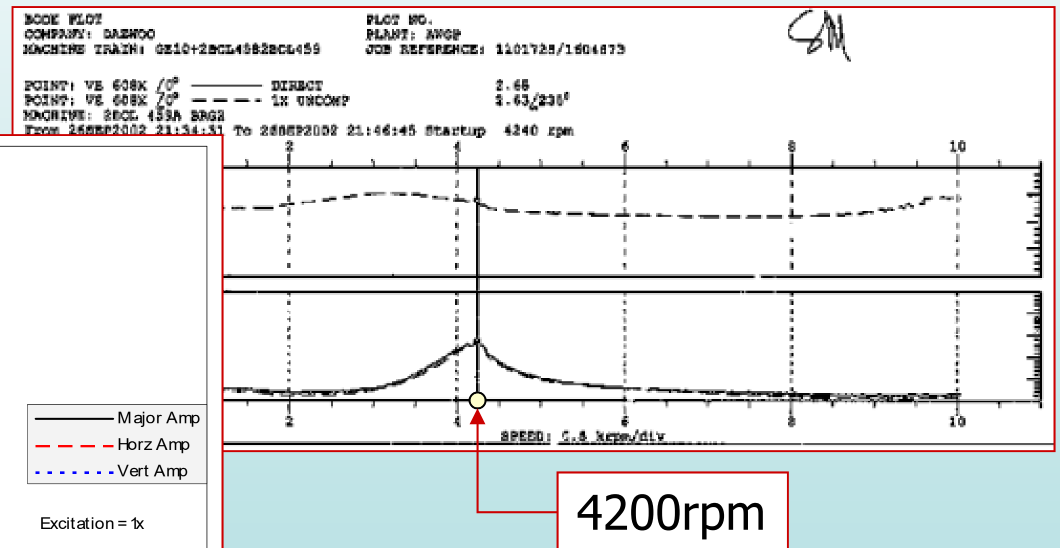
http://www.dresser-rand.com/insight/v9no1/art_6.asp

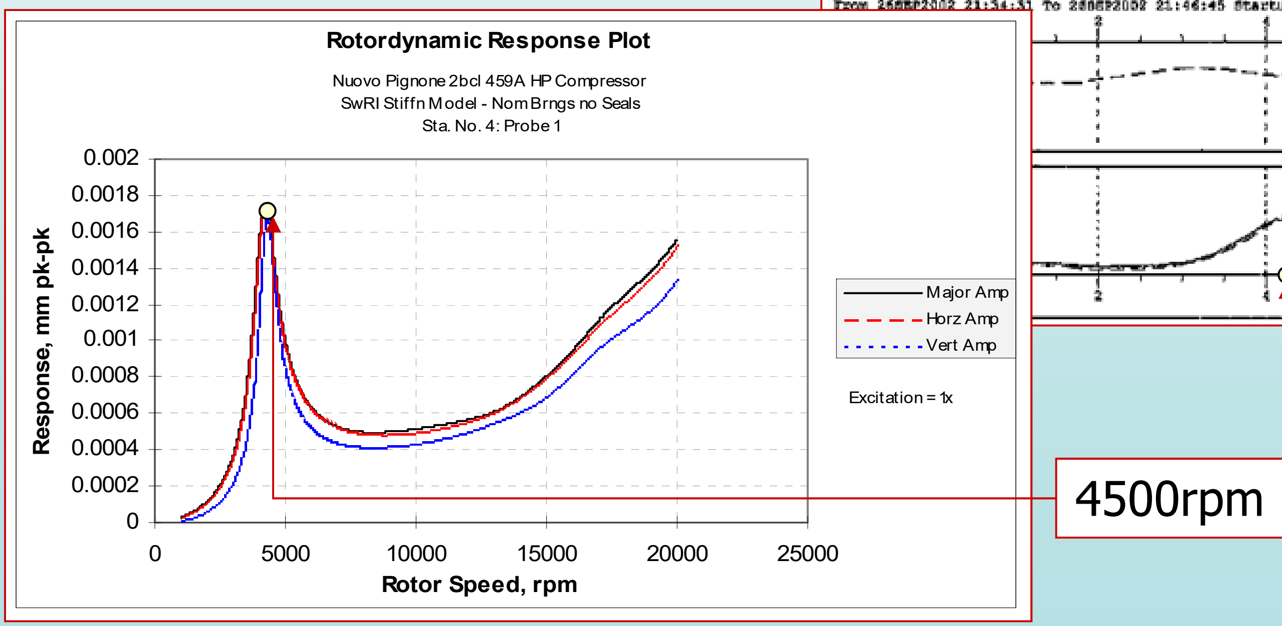


Reference: Smalley, A. J., Camatti, M., Childs, D. W., Hollingsworth, J. R., Vannini, G., Carter, J. J., "Dynamic Characteristics of the Diverging Taper Honeycomb-Stator Seal," *Proceedings of ASME Turbo Expo 2004*, June 14-17, 2004, Vienna, Austria.

Unbalance Response

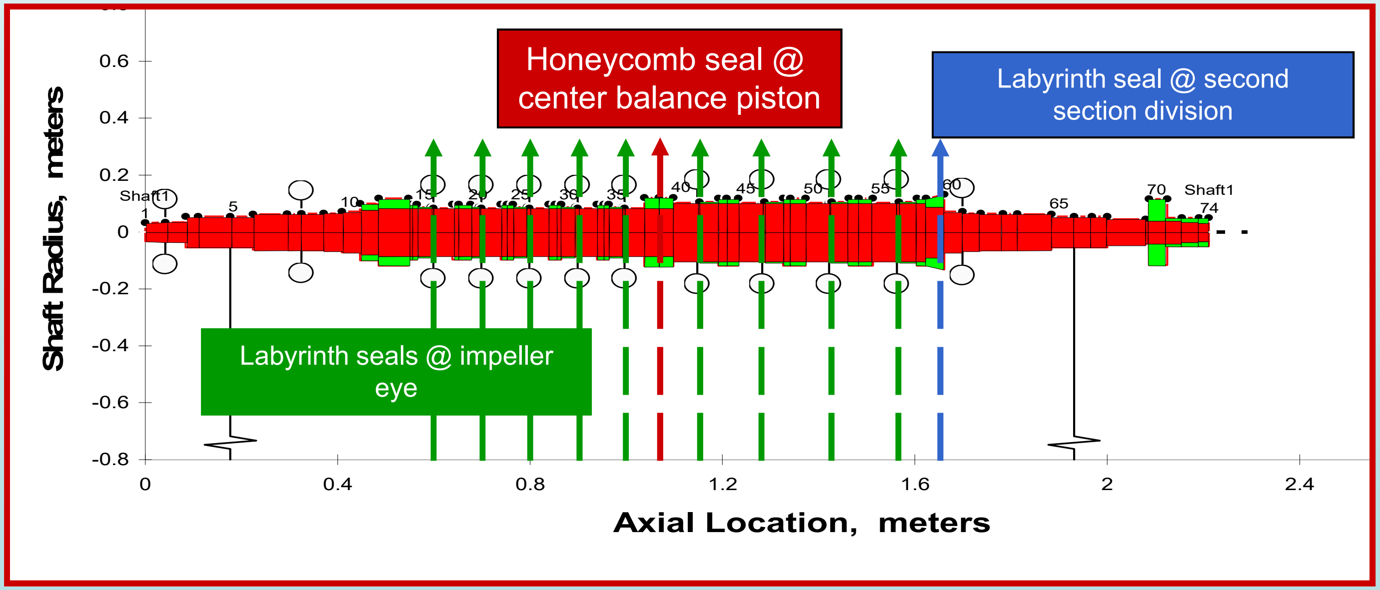
- First critical speed predicted about 4500 rpm (45% of running speed); good agreement with factory mech. test (no load) where first critical speed was 4200rpm
- What caused the frequency to drop to 12% running speed?





Rotordynamic Modeling

- Model includes rotor, bearings, impeller eye labyrinths, second section balance seal and center division wall honeycomb seal
- Used the XLTRC² suite from Texas A&M University
- XLTFPBrg for journal bearings (K, C) matrix
- XLLaby for each labyrinth seal (K, C) matrix
- ISOTSEAL for honeycomb seal (K, C) matrix



Rotordynamic Modeling

Assumptions used in Stability Analysis

- Swirl ratios into seal as given below:
- Impeller eyes = 0.68
- Calculated impeller exit swirl ratio
  = 0.25 (with swirl brakes)
- Honeycomb = 0.68 (original)
  = 0.15 (with Shunt holes)
- Lateral drum = 0.2

Rotordynamic Modeling

Aero Cross-Coupling

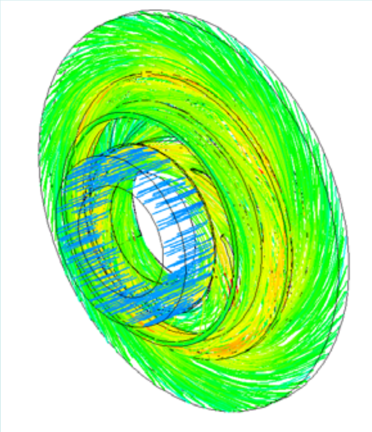
- Arises from Impellers of Centrifugal Compressors
- Most Common Method version of Wachel Equation

$$\left(K_{XY}\right)_i = 63{,}000 * \frac{Mole\ Weight}{10} \sum_{j=1}^{(N_S)_i} \frac{\left(Horsepower\right)_{i,j}}{RPM\ *\ D_i\ *\ h_i} \left(\frac{\rho_D}{\rho_S}\right)_j$$

- CFD Methods Have Been Developed
  - Show good correlation to experimental data for pump and compressor impellers

$$k_{xy} = \frac{C_{mr}\ \rho_d\ U^2\ L_{shr}}{Q / Q_{design}}$$

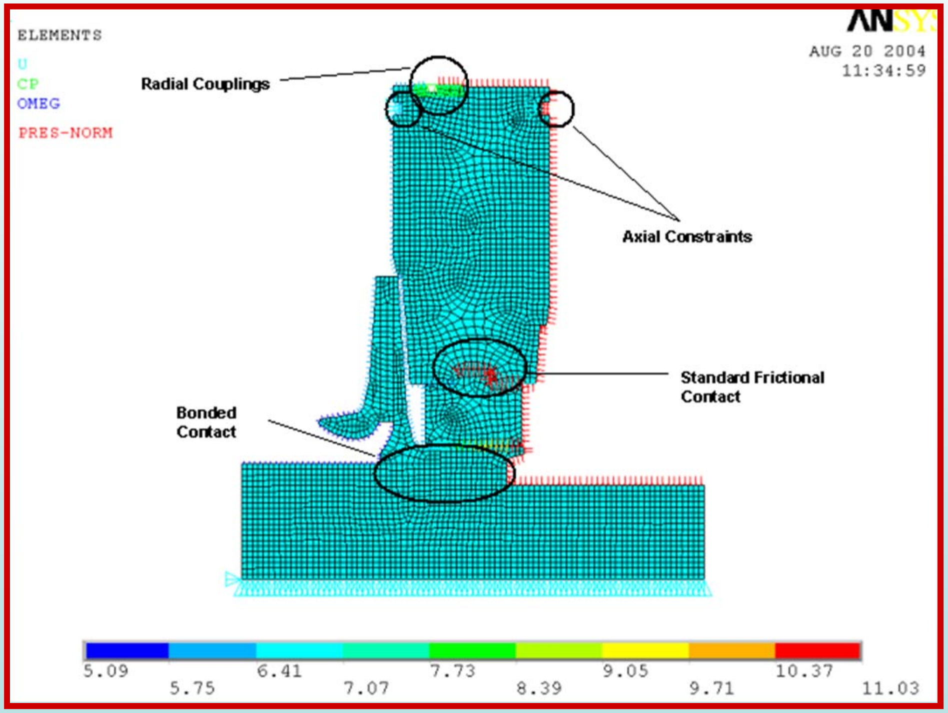
Refined equation based on CFD for Centrifugal Compressors



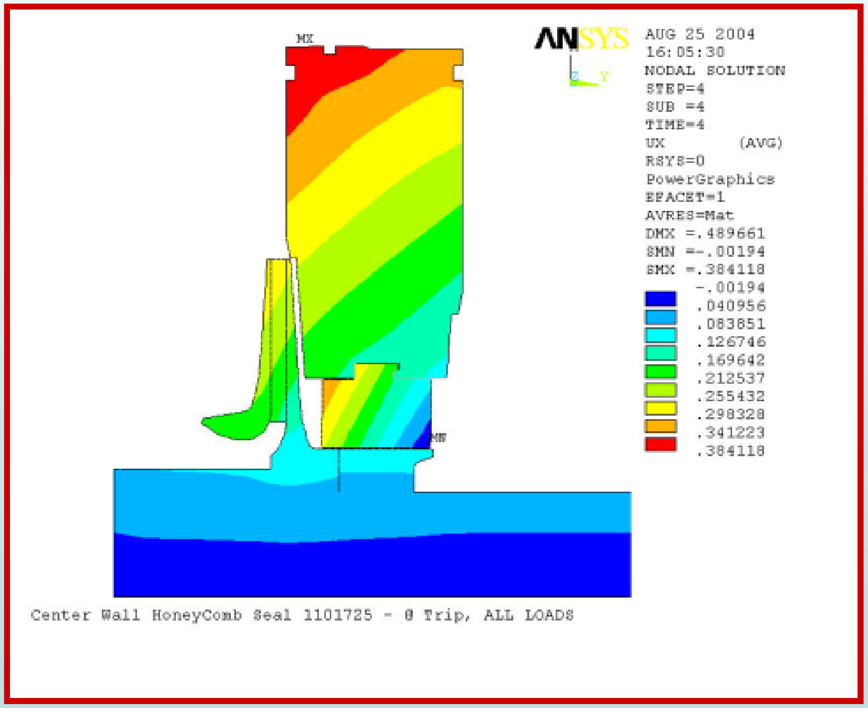
- Moore, J.J. and Ransom, D. "**Centrifugal Compressor Stability Prediction Using a New Physics Based Approach**," Proceedings of ASME Turbo Expo 2009: Power for Land, Sea and Air, June 8-12, 2009 in Orlando, Florida.

FEA Prediction of Seal Deformation

- Seal Deforms Due to Pressure Differential
- Resulting Clearance Becomes Divergent



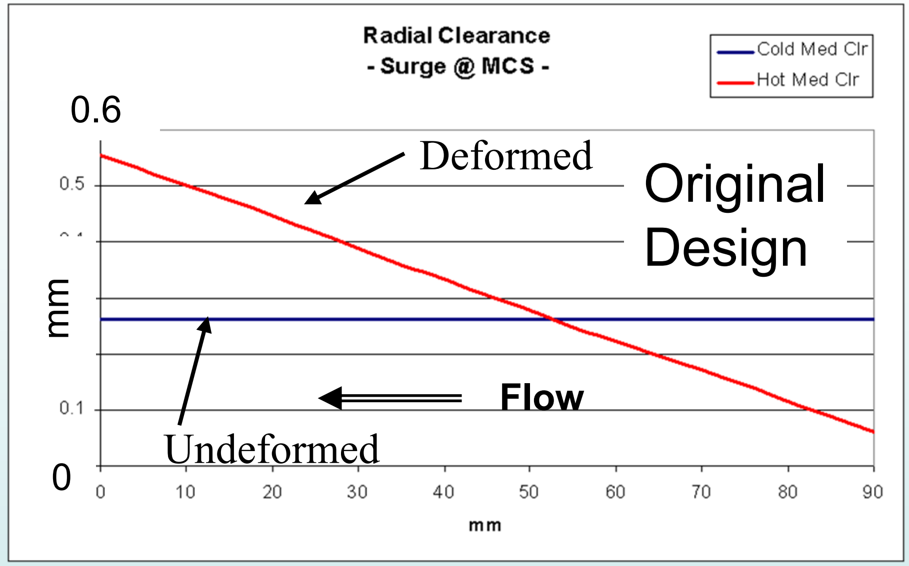
FEM model main input

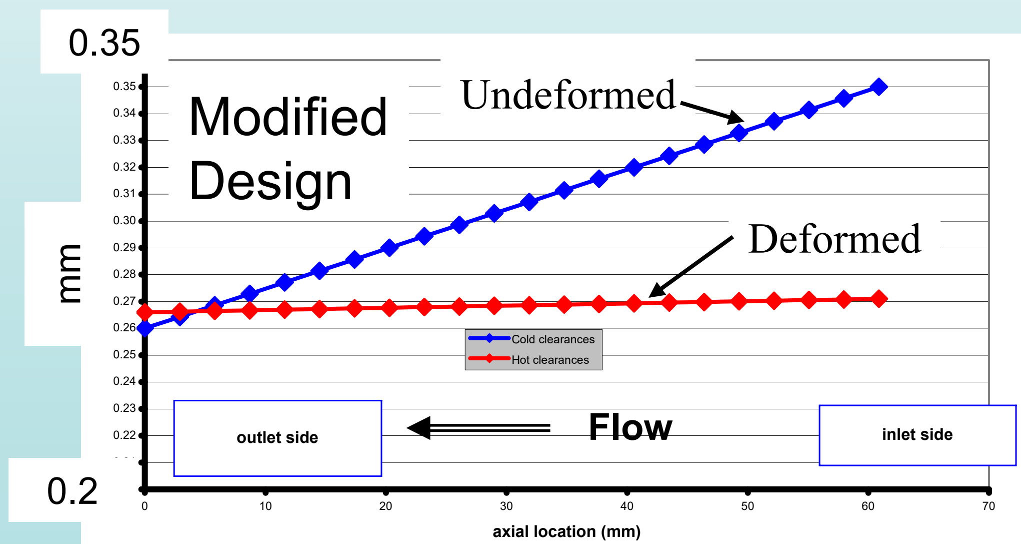


FEM model radial displacements

FEA Prediction of Seal Deformation

- Seal Deforms Due to Pressure Differential
- Resulting Clearance Becomes Divergent
- Design Modifications Include:
  - Mechanical changes to reduce deformation
  - Machining positive taper into the seal
- Modified design results in constant clearance under loaded conditions





Conditions Analyzed

- 5 Seal Geometric Conditions Considered:
  - Cold, Nominal Clearance
  - Hot and Deformed Clearance
  - Hot and Deformed with Worst-Case Tolerance Stackup
    - -0.12 mm additional taper
  - Hot and Deformed with 2X Clearance
  - No Seals

HP Compressor Modifications

- Rev 1 modified the seal mounting design to minimize deformation
  - Positive taper machined into seal bore to reduce divergence during operation
- Rev 2 increased amount of initial taper and increased average clearance
- Shunt injection added to center division wall seal
- Swirl brakes added to impeller eye seals

	Original	Rev. 1 Modification	Rev. 2 Modification
Honeycomb Seal	No Shunt (0.68 swirl) Zero Taper Cold clearance -0.494 mm Divergence in Operation	With Shunt (0.15 Swirl) 0.075 mm Cold Taper -0.05 mm Taper in Operation	With Shunt (0.15 Swirl) 0.09 mm Cold Taper 0 Taper in Operation 25% Larger Clearance
Eye Labyrinths	No De-swirl (0.68 swirl)	Swirl Brakes Added (0.25 swirl)	Swirl Brakes Added (0.25 swirl)

HP Stability Predictions – Trip Conditions

Original Configuration

HP Compressor

Run#	Deformation at HC?	Kxy-> (N/m)	API 6.15E+06	SWRI 8.01E+06
1	No	Log Dec->	0.192	0.138
		Freq->	4608	4616
2	Yes	Log Dec->	-11.47	-10.71
		Freq->	899	978

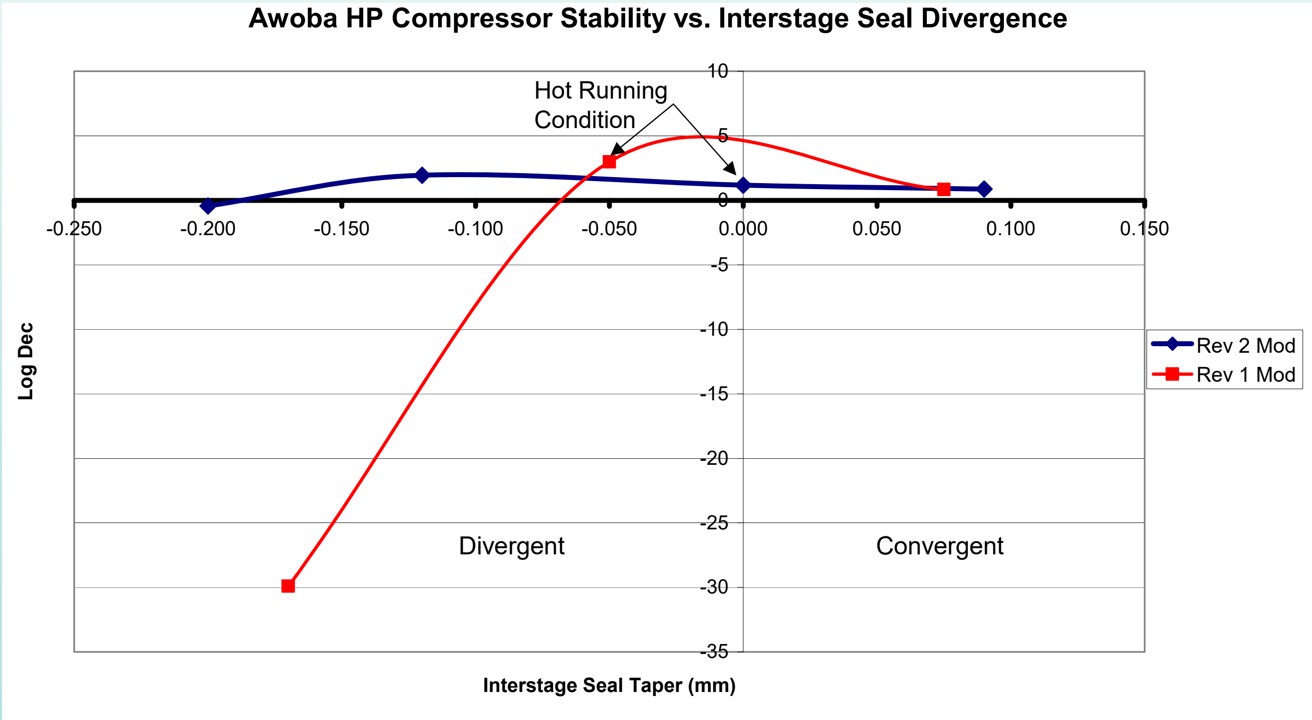
LP Compressor

Run#	Deformation at HC?	Seal Divergence	Kxy->	API 1.86E+06	SWRI 1.91E+06
1	No	0	Log Dec->	0.058	0.056
			Freq->	3523	3523

- Analysis predicts HP Compressor instability when seal deformation accounted for using both formulations for aero cross-coupling
- Predicted frequency closely matches observed subsynchronous frequency in the field (~900 cpm)
- LP Compressor predicted to be stable (it was) but log dec is low

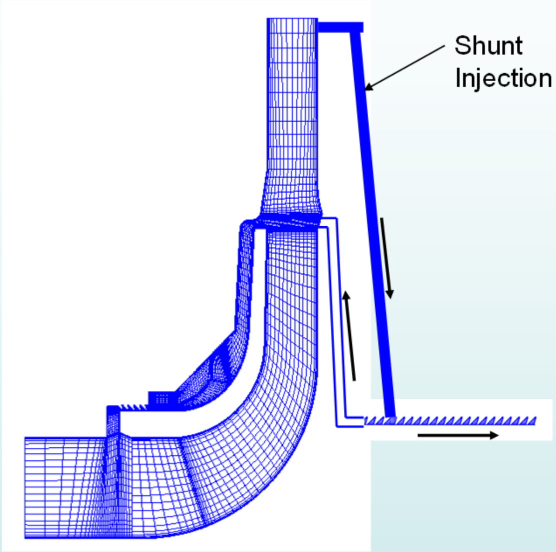
HP Stability Predictions – MCS/Surge

- Shows sensitivity of rotor system to seal divergence
- First modification was stable to but too close to "cliff"
- Second design increased clearance reducing sensitivity to divergence
- Rev 2 can accommodate effects of manufacturing tolerance



LP Compressor Modifications

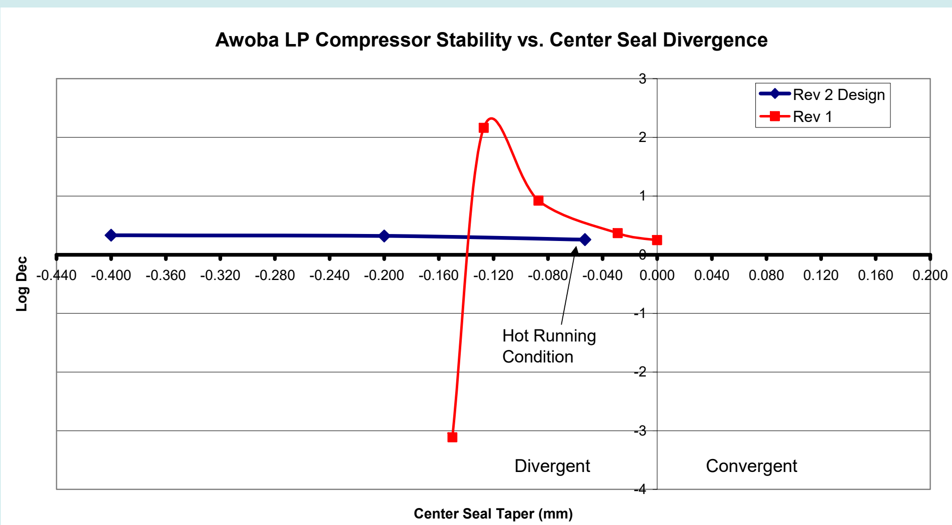
- Honeycomb seal with shunt injection added to center balance piston
- Swirl brakes added to impeller eye labyrinths
- Rev 2 design increased average clearance and introduced a divergent initial taper to increase damping
- This initial machined taper is opposite to that used on the HP compressor



	Original	Rev 1. Modification	Rev. 2 Modification
Interstage Diaphragm Seal	Tooth-on Stator Laby Seal No Shunt (0.68 swirl) Cyl. Cold Clearance	Honeycomb Seal With Shunt (0.15 Swirl) 0.0 mm Cold Taper -0.005 Taper in Operation	Honeycomb Seal With Shunt (0.15 Swirl) -0.05 mm Cold Taper -0.053 Taper in Operation 50% Larger Clearance
Eye Labyrinths	No De-swirl (0.68 swirl)	Swirl Brakes Added (0.25 swirl)	Swirl Brakes Added (0.25 swirl)

LP Stability Predictions – MCS/Surge

- Original design predicted to be unstable at worst-case operating condition
- Rev 1 design showed similar characteristics as the HP compressor
- Rev 2 increased clearance and machined a divergent taper into the seal and showed low sensitivity to divergence
- Log decrement substantially improved



Original Design

Run#	Deformation at HC?	Seal Divergence	Kxy->	API 2.09E+06	SWRI 3.30E+06
1	No	0	Log Dec->	-0.182	-0.233
			Freq->	4677	4696

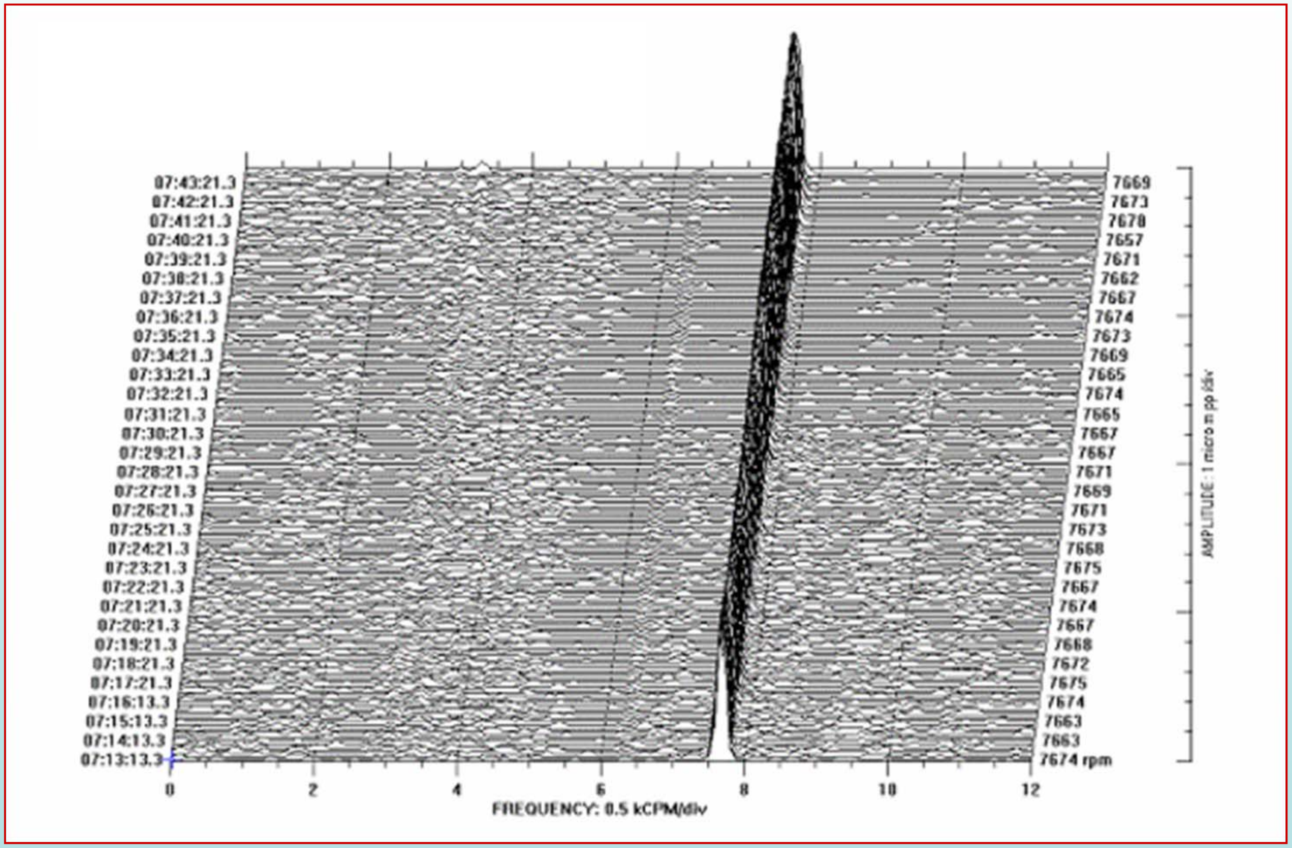
Rev 2 Modified Design

Run#	Deformation at HC?	Seal Taper	Kxy->	API 2.09E+06	SWRI 3.30E+06
1	No	-0.05	Log Dec->	0.231	0.184
			Freq->	3524	3525
2	Yes	-0.053	Log Dec->	0.259	0.211
			Freq->	3495	3496
3	Yes+Toler.	-0.173	Log Dec->	0.367	0.308
			Freq->	3173	3175
4	Yes, 2X Clear	-0.106	Log Dec->	0.176	0.13
			Freq->	3543	3543
5	No Seals		Log Dec->	0.055	0.01
			Freq->	3620	3620

Field Re-Start

Re-Start After Modifications

- Subsequent Re-start showed no signs of subsynchronous activity on either HP or LP compressor even at fully loaded conditions



Summary

- Modified compressor demonstrated good stability on subsequent start-up.
- Instability was predicted when seal deformation is taken into account.
- Divergence of the damper seal reduced first natural frequency of the rotor causing the seal to become destabilizing.
- Modifications on HP compressor were made to seal to prevent divergent condition.
- A tighter clearance seal is more sensitive to divergence.
- Damper seals must be designed like bearings rather than seals.
    - i.e.. Tight control on clearance
- Damper seal clearance can be designed differently depending on the operating pressure.
- Shutdown resulted in 3 months downtime with approx $40 million lost production (based on $7 per MMBtu gas)

Questions ???

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