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

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Benefiting government, industry and the public through innovative science and technology
Machinery Department

- 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
Machinery Department Test Facilities

- Operating Fluids: Air, CO₂, N₂
- Multiphase with Air/H₂O
- 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

Diagram:
- Gas Turbine: PGT10B
- Gearbox
- LP Casing: CE/CO LP Casing 2BCL458
- HP Casing: CE/CO HP Casing 2BCL459/A
- High Speed: 9198 rpm
- Low Speed: 7585 rpm
- Tripped on site
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
  - No swirl brake or shunt injection on any seal

- Design of compressor predated more recent experience with adverse effects of damper seals

- Design Pressure
  - $P_1 = 22$ bar (319 psi)  $P_2 = 133$ bar (1930 psi)

- Maximum Discharge Pressure
  - 189 bar (2740 psi)  (Maximum Continuous Speed & Near Surge)
- 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 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)

Linear Vibration

\[ \delta = L_n \left[ \frac{X_{n-1}}{X_n} \right] = 0 \]

- Neutrally Stable
- Unstable
- Stable

Rotor Vibration

\[ \delta < 0 \]

Undesirable

\[ \delta > 0 \]

Desirable
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
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

Sample 10-Stage Compressor Model

Typical High Pressure Centrifugal Compressor

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
Stability Analysis

A Rotor System Is Unstable When The Destabilizing Forces Exceed Stabilizing (Damping) Forces
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 (\( \zeta \)), 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

Damped Eigenvalue Mode Shape Plot

Nuovo Pignone 2bcl 459A HP Compressor
SwRI Stiffn Model - Nom Brngs with Seals

- forward
- backward

\( f = 4677.4 \text{ cpm} \)
\( d = -0.2966 \text{ logd} \)
\( N = 10050 \text{ rpm} \)
Effective Stiffness & Damping

\[ K_{\text{effective}} = K_{xx} + \omega C_{xy} \]

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

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

\[ C_{\text{effective}} = C_{xx} - K_{xy} / \omega \]
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.
- $C_{\text{effective}}$ Varies with Frequency
- If Natural Frequency in Region of Negative $C_{\text{effective}}$ ⇒ Rotor Unstable
Honeycomb Seal Damping Test Data vs. Predictions

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


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?

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**Rotordynamic Response Plot**

Nuovo Pignone 2blc 459A HP Compressor
SwRI Stiffn Model - NomBrngs no Seals
Sta. No. 4: Probe 1

- **Excitation = 1x**
- **Response, mm pk-pk**
- **Rotor Speed, rpm**
  - 0 5000 10000 15000 20000 25000
  - 0.0000 0.0002 0.0004 0.0006 0.0008 0.001 0.0012 0.0014 0.0016 0.0018 0.002

- **Major Amp**
- **Horz Amp**
- **Vert Amp**

**Points:**
- **1 x**
- **2 x**
- **3 x**

**Excitation Levels:**
- **2.45x**
- **3 x**
- **4 x**

**SPEED:**
- **5000 rpm**
- **4200 rpm**
- **4500 rpm**
Rotordynamic Modeling

- Model includes rotor, bearings, impeller eye labyrinths, second section balance seal and center division wall honeycomb seal
- Used the XLTRC$^2$ 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
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
**Aero Cross-Coupling**

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

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

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

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

Refined equation based on CFD for Centrifugal Compressors

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

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**Diagram:**
- Radial Clearance
  - *Surge @ MCS*
  - Cold Med Cir
  - Hot Med Cir
  - Deformed
  - Original Design
  - Undefomed
  - Flow

**Graphs:**
- Modified Design
  - Undefomed
  - Deformed
- outlet side
- inlet side

**Flow:**
- mm
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

<table>
<thead>
<tr>
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<th>Original</th>
<th>Rev. 1 Modification</th>
<th>Rev. 2 Modification</th>
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</thead>
<tbody>
<tr>
<td><strong>Honeycomb Seal</strong></td>
<td>No Shunt (0.68 swirl)</td>
<td>With Shunt (0.15 Swirl)</td>
<td>With Shunt (0.15 Swirl)</td>
</tr>
<tr>
<td></td>
<td>Zero Taper Cold clearance</td>
<td>0.075 mm Cold Taper</td>
<td>0.09 mm Cold Taper</td>
</tr>
<tr>
<td></td>
<td>-0.494 mm Divergence in Operation</td>
<td>-0.05 mm Taper in Operation</td>
<td>0 Taper in Operation</td>
</tr>
<tr>
<td></td>
<td></td>
<td>25% Larger Clearance</td>
<td>25% Larger Clearance</td>
</tr>
<tr>
<td><strong>Eye Labyrinths</strong></td>
<td>No De-swirl (0.68 swirl)</td>
<td>Swirl Brakes Added (0.25 swirl)</td>
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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.
- Shows sensitivity of rotor system to seal divergence
- First modification was stable 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

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<tr>
<td><strong>Interstage Diaphragm Seal</strong></td>
<td>Tooth-on Stator Laby Seal No Shunt (0.68 swirl) Cyl. Cold Clearance</td>
<td>Honeycomb Seal With Shunt (0.15 Swirl) 0.0 mm Cold Taper -0.005 Taper in Operation</td>
<td>Honeycomb Seal With Shunt (0.15 Swirl) -0.05 mm Cold Taper -0.053 Taper in Operation 50% Larger Clearance</td>
</tr>
<tr>
<td><strong>Eye Labyrinths</strong></td>
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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
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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