A SOLUTION TO YEARS OF HIGH VIBRATION PROBLEMS IN THREE REINJECTION COMPRESSOR TRAINS RUNNING AT 33 MPa DISCHARGE PRESSURE

Jong Kim, PhD/Presenter
Authors

**Jong Kim, PhD**
Senior Principal Engineer, Waukesha Bearings Corporation
Bachelor of Science, Mechanical Engineering, 1985 – Busan National University
Master of Science, Mechanical Engineering, 1987 – KAIST (Korea Advanced Institute of Science and Technology)
Doctorate of Philosophy, Mechanical Engineering, 1991 – KAIST (Korea Advanced Institute of Science and Technology)

**Marcio Felipe dos Santos**
Senior Maintenance Engineer, Major South American Oil Company
Bachelor of Science, Mechanical Engineering, 1979 – Universidade Federal of Rio de Janeiro

**Barry J. Blair**
Chief Engineer, Waukesha Bearings Corporation
Bachelor of Science, Mechanical Engineering, 1990 – University of Virginia
Master of Science, Mechanical Engineering, 1990 – University of Virginia
Master of Product, Design and Development, 2015 – Northwestern University
Abstract

Over a 13-year span, a major South American oil company’s maintenance department fought high vibrations in three gas reinjection compressor trains. To reduce the chances of machine trips, technicians field balanced the compressors every year and replaced worn point contact pivot tilt pad journal (TPJ) bearings and O-ring squeeze film dampers (SFDs) with new ones yearly. The downtime from implementing these preventative measures and from actual trips in the trains resulted in a loss of capacity of 1% a year and additional flaring of the gas.

After a thorough analysis of the compressors and inspection of damaged components, it was determined that the reoccurring problems would be solved by installing optimized Flexure Pivot tilt pad journal bearings with Integral Squeeze Film Damper (ISFD) technology into the compressors.

In 2013 the reinjection compressors were placed back into service with Flexure Pivot TPJ bearings with ISFD technology. Since then the reinjection compressors have exhibited lower vibration levels that do not grow over time, have had ZERO trips, and have not required field balancing for continuous operation. Overall efficiency has increased by approximately 1%.
Contents

• Background
• Problem Statement
• Modeling
• Upgraded Bearing and Damper
  – Design
  – Optimization
• Summary
• Lessons Learned
Unit Background Information

- 3 gas reinjection compressor trains operating since 2000
- Each compressor train has two casings
  - First casing (LP) experienced vibration issues
  - Second casing (HP) did not have vibration issues
- Discharge pressure: 33 MPa
- Rated speed: 11,456 rpm
- OEM bearing information
  - 114.3 mm bore x 50.8 mm long TPJ
  - 5-pad, load on pad
  - 60% offset
Problem Statement

• Rotor vibration levels increased over time
  – Downtime due to high vibrations resulted in about **1% loss in production** time and additional flaring gas
  – **Field balanced every year**
  – Installation of **new OEM bearings every year**

Vibration trend of LP compressor with OEM bearings over 5-month span

- **DE vibrations increasing**

Vibration with worn pivots

After replacement with new OEM bearings

Vibration with new OEM bearings of the original design
Root Cause

- Severe pivot wear in TPJ bearings
  - Bearing clearance increased by 63+ microns
- O-ring damper performance changed
  - Damper film eccentricity ratio change (bottoming out)

![Diagram of TPJ components and wear marks](image-url)
Rotordynamic Model
Baseline Mode Shapes with OEM Bearings

- **Small log dec** for baseline model (no SFD and no aero cross-coupling stiffness from seals and impellers)

\[
\begin{align*}
\text{Shaft Radius, mm} & & \text{Axial Location, mm} \\
0 & & 0 \\
100 & & 400 \\
200 & & 800 \\
300 & & 1200 \\
400 & & 1600 \\
500 & & 2000 \\
\end{align*}
\]

\[
\begin{align*}
\text{Cmin (minimum clearance)} & \\
-400 & \\
0 & \\
200 & \\
400 & \\
\end{align*}
\]

\[
\begin{align*}
f &= 4400.1 \text{ cpm} \\
d &= 0.2065 \log d \\
N &= 11456 \text{ rpm}
\end{align*}
\]
Stability with OEM Bearings

- Level I stability predicts that the rotor is unstable without SFD

\[ Q_A = \frac{(HP)B_c C}{D_c H_c N} \left( \frac{\rho_d}{\rho_s} \right) = \frac{(15000) \cdot (3) \cdot (63)}{(19.64) \cdot (0.78) \cdot (12142)} \cdot (8) = 121.932 \text{ klb in or } 2.13E07 \text{ N/m} \]
Division Wall Seal Contribution

- Insignificant improvement to stability with division wall hole pattern seal
- For stability, SFD required
O-ring Squeeze Film Damper (SFD)

- Damper radial clearance \( (c) = 0.110 \) mm
- Damper radius \( (R) = 95.25 \) mm
- Effective damper length \( (L) = 37.85 \) mm
- Stability is very sensitive to damper eccentricity ratio \( (\varepsilon) \)
- Added O-ring stiffness

\[
K = \frac{2\mu RL^3\varepsilon\omega}{c^3(1 - \varepsilon^2)^2} \quad C = \frac{\pi\mu RL^3}{c^3(1 - \varepsilon^2)^{3/2}}
\]

Damper stiffness and damping coefficients (without O-ring stiffness):
- \( K=1.94E+07 \) N/m, \( C=1.88E+05 \) Ns/m at \( \varepsilon=0.25 \)
- \( K=6.06E+07 \) N/m, \( C=2.62E+05 \) Ns/m at \( \varepsilon=0.50 \)
- \( K=1.70E+09 \) N/m, \( C=2.06E+06 \) Ns/m at \( \varepsilon=0.90 \)
Pivot Wear Effects on Synchronous Vibration

- Pivot wear increased operating bearing clearance and reduced preload, resulting in increased synchronous vibrations.
- A bearing with an SFD can make the rotor less sensitive to pivot wear than a bearing without an SFD.

![Graphs showing pivot wear effects with and without SFD.](image)
Bearing Upgrades (Journal Bearing)

1. Flexure Pivot Tilt Pad Journal Bearing
   - **No pivot wear**
     - Integral pivot
     - Maintains bearing clearance
   - High pivot stiffness
     - No pivot stiffness effect on bearing dynamic coefficients
   - Tight control of clearance and preload
     - Electrical Discharge Machining (EDM)
   - No pad flutter
Bearing Upgrades (Damper)

2. Integral Squeeze Film Damper (ISFD)
   - Accurate stiffness control by eliminating O-ring support
   - **No change in stiffness and damping over time**
   - Designed to counter static load
   - Optimized damping
   - Less cavitation

![ISFD Diagram]

- **ISFD Clearance**: 0.35 mm
- **Original Damper Clearance**: 0.11 mm
- **‘S-spring’**
- **O-ring**
- **Conventional SFD**
- **Damper Film**
Optimization of ISFD

- Optimized stiffness and damping are 4.375 $\times 10^7$ N/m (250,000 lb/in) and 2.625 $\times 10^5$ N-s/m (1500 lb-s/in)
- Additionally, the ISFD is designed to center the Flexure Pivot TPJ under gravity load by countering the static deflection
# Redesigned Bearing

## Original Design

<table>
<thead>
<tr>
<th>Feature</th>
<th>Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Conventional TPJ with SFD</td>
<td></td>
</tr>
<tr>
<td>5-pad, Load On Pad</td>
<td></td>
</tr>
<tr>
<td>Shaft diameter</td>
<td>114.300 ±0/-0.013 mm</td>
</tr>
<tr>
<td>Bearing bore</td>
<td>114.414 ±0.025/-0 mm</td>
</tr>
<tr>
<td>Clearance range</td>
<td>0.124/0.156 mm</td>
</tr>
<tr>
<td>Preload range</td>
<td>0.293/0.501</td>
</tr>
<tr>
<td>L/D</td>
<td>0.444</td>
</tr>
<tr>
<td>Pad arc</td>
<td>60°</td>
</tr>
<tr>
<td>Pivot Offset</td>
<td>60%</td>
</tr>
</tbody>
</table>

## Optimized Design

<table>
<thead>
<tr>
<th>Feature</th>
<th>Measurement</th>
</tr>
</thead>
<tbody>
<tr>
<td>Flexure Pivot TPJ with ISFD Technology</td>
<td></td>
</tr>
<tr>
<td>4-pad, Load Between Pad</td>
<td></td>
</tr>
<tr>
<td>Shaft diameter</td>
<td>114.300 ±0/-0.013 mm</td>
</tr>
<tr>
<td>Bearing bore</td>
<td>114.427 ±0.025/-0 mm</td>
</tr>
<tr>
<td>Clearance range</td>
<td>0.124/0.156 mm</td>
</tr>
<tr>
<td>Preload range</td>
<td>0.230/0.273</td>
</tr>
<tr>
<td>L/D</td>
<td>0.500</td>
</tr>
<tr>
<td>Pad arc</td>
<td>72°</td>
</tr>
<tr>
<td>Pivot offset</td>
<td>55%</td>
</tr>
</tbody>
</table>
1st Mode Shapes: Original and Upgraded Bearings

- Applied destabilizing cross-coupling stiffness of 2.13E+07 N/m to mid-span
- Without SFD, unstable
- Both O-ring SFD and ISFD can make the rotor stable
- No change in stiffness and damping of ISFD over time

![Graphs showing mode shapes and vibration characteristics for different bearings and SFD configurations.](image-url)
No Subsynchronous Vibration (SSV) with Upgrade

- Small SSV with O-ring SFD
- No SSV with ISFD
Vibration Improvement (Comp A & B)

- Compressor A vibration dropped to **less than half** and maintained that level over time.
- Compressor B vibration also down to **below 50 µm** from 90 µm (original) and kept the same level over time.
Compressor C was also upgraded with a Flexure Pivot TPJ with ISFD technology.

Again, the vibration level decreased to **below 30 µm** with the upgrade and maintained that same level due to no change in bearing clearance and SFD performance over time.

---

**Vibration Improvement (Comp C)**

![Compressor C vibration trend over 10 years](image)
Summary

- Three reinjection compressor trains suffered from **excessive vibration** over many years
  - Original configuration: point contact pivot tilt pad journal bearings with an O-ring SFD

- The root cause was **excessive pivot wear** and **degradation** of the O-ring SFD
  - Bearing bore increased
  - Stiffness and damping changed over time
Summary

• The compressors were retrofitted with optimized Flexure Pivot tilt pad journal bearings with ISFD technology

• Operating exceptionally well
  – Since 2013
  – **Low vibration levels**
    • 50% drop pk-pk compared to OEM bearings
    • Do not grow over time
  – **No field balancing required** so far (2 years)
  – **No trips** (continuous production)
  – **No expensive bearing replacements**
  – Overall **efficiency increased** by 1%
Lessons Learned

• Increase in synchronous vibrations may be an indication of bearing clearance increasing from pivot wear and/or change in O-ring damper performance

• Pivot wear may accelerate over time from increasing imbalance due to deposits on impellers

• Without eliminating pivot wear, just replacing the worn bearing with new build of the same design is NOT a long-term solution

• Proper bearing and damper selection and optimization can reduce or eliminate the likelihood of increasing vibrations and pivot wear

• Flexure Pivot technology is a proven design to eliminate pivot wear

• ISFD technology maintains performance over time
Feedback and Questions

Case Study: A Solution to Years of High Vibration Problems in Three Reinjection Compressor Trains Running at 33 MPa Discharge Pressure
Appendix: Damper Design Comparison

**Conventional SFD**

Squeeze Film (Outer Oil Film) – bearing whirls or orbits (not spins) in a precessional motion due to synchronous (unbalance) or non-synchronous excitation, squeezing the oil and thus generating an oil film pressure, and subsequently a damping force. Flow can be axial too, depending on sealing.

**Integral Squeeze Film Damper**

‘S-spring’

Damper Flow In/out of Orifices and Axial Gaps
Jong Kim, PhD, (+1 262.506.3055 jkim@waukbearing.com) is a Senior Principal Engineer at Waukesha Bearings Corporation, headquartered in Pewaukee, Wisconsin (USA) and a Senior Consulting Engineer at Bearings Plus, Houston, Texas (USA), which is a business unit of Waukesha Bearings. Dr. Kim has overall responsibilities for rotordynamic analysis, bearing upgrades and bearing/seal technologies including Flexure Pivot bearings, ISFD technology and brush seals. Dr. Kim joined Waukesha Bearings in 2011. Prior to joining Waukesha Bearings and since 2001, he worked for KMC and Bearings Plus. Dr. Kim received his Bachelor of Science (Mechanical Engineering, 1985) from Busan National University and both his Master of Science (Mechanical Engineering, 1987) and his PhD (Mechanical Engineering, 1991) from KAIST (Korea Advanced Institute of Science and Technology). He has authored several papers and has been granted multiple patents on bearing technologies.

Marcio Felipe dos Santos (+55 092.3616.6614 tombodafumaca@gmail.com) is a Senior Maintenance Engineer at a major oil company in South America, in the Amazon. Since 2000, Mr. Dos Santos has provided maintenance in LPG plants and on reinjection compressors. Prior to joining his current company, he maintained drilling rigs. He has his Mechanical Engineering Degree from Universidade Federal do Rio de Janeiro (1979). Throughout his career Mr. Dos Santos has had maintenance duties.

Barry J. Blair (+1 262.506.3043 bblair@waukbearing.com) is the Chief Engineer at Waukesha Bearings Corporation, headquartered in Pewaukee, Wisconsin (USA). Mr. Blair has overall responsibilities for research & development activities at Waukesha, including new products and overseeing the refinement of bearing design tools and methods. Mr. Blair joined Waukesha Bearings in 1993 and has served in increasingly responsible engineering and technology roles. Mr. Blair received both his Bachelor of Science (Mechanical Engineering, 1990) and Master of Science (Mechanical Engineering, 1990) from the University of Virginia, completing requirements of both degrees concurrently. He has authored and coauthored several papers on the development of both hydrodynamic and active magnetic bearing technologies.