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Eliminating a Rotordynamic Instability of a 12 MW Overhung, Radial Inflow Expander

Chris D. Kulhanek
J. Jeffrey Moore, Ph.D.

Southwest Research Institute®
Post Office Drawer 28510
San Antonio, TX 78228-0510
Phone: 210-522-5812
Fax: 210-681-9661
chris.kulhanek@swri.org

Joseph Lillard
Grant Nordwall

Atlas Copco Gas and Process
Atlas Copco Mafi-Trench Company
3037 Industrial Parkway
Santa Maria, CA 93455
Phone: 805-928-5757
Joseph.Lillard@us.atlascopco.com

Greg Elliott
Tom Shoup

GE Oil & Gas
Lufkin
PO Box 849
Lufkin, TX 75902
Phone: 936-637-5471
Fax: 936-633-3916
greg.elliott@ge.com

Authors

***Dr. Jeffrey Moore** is an Institute Engineer in the Machinery Section at Southwest Research Institute in San Antonio, TX. He holds a B.S., M.S., and Ph.D. in Mechanical Engineering from Texas A&M University. His professional experience over the last 25 years includes engineering and management responsibilities related to centrifugal compressors and gas turbines at Solar Turbines Inc., Dresser-Rand, and Southwest Research Institute. His interests include advanced power cycles and compression methods, rotordynamics, seals and bearings, computational fluid dynamics, finite element analysis, machine design, controls and aerodynamics.*

***Joseph K Lillard** graduated with a Bachelor Degree in Mechanical Engineering from California Polytechnic State University in 1982. After graduation, he began work at Mafi-Trench Corporation as a Thermodynamics Engineer. Over the course of his career at Mafi-Trench and now at Atlas Copco Mafi-Trench he has held several positions including Chief Engineer, Operations Manager, R&D Manager, Product Marketing Manager, and Technology Manager. Joe has been happily married to his wife Mara for 37 years, and is proud of their three adult children and four grandchildren.*

***Gregory Elliott** worked for Lufkin Industries, now a part of GE Oil & Gas, since 1990 in a variety of mechanical engineering roles related to gear drives and other machinery. He currently holds a Consulting Engineer role providing technical assistance and consultation both within and outside Lufkin. He holds BS and MS degrees in Agricultural Engineering from Texas A&M University in College Station, Texas.*

***Chris D. Kulhanek** is a Research Engineer in the Machinery Section at Southwest Research Institute in San Antonio, TX. He holds a B.S. and M.S. in Mechanical Engineering from Texas A&M University. He leads rotordynamic design efforts on many types of rotating equipment. His interests include lateral and torsional rotordynamics, fluid-film bearing analysis and design, structural finite-element analysis (FEA), test rig design, and root cause failure analysis (RCFA).*

***Grant Nordwall** is a Senior Machinery Engineer for Mafi-Trench Company, LLC, in Santa Maria, California. He has been with Mafi-Trench since 1990 working in the areas of aerodynamics, rotordynamics, acoustics, and field troubleshooting. Grant received his B.S. degree in Mechanical Engineering from California Polytechnic State University. Grant is a registered Professional Engineer in the State of California.*

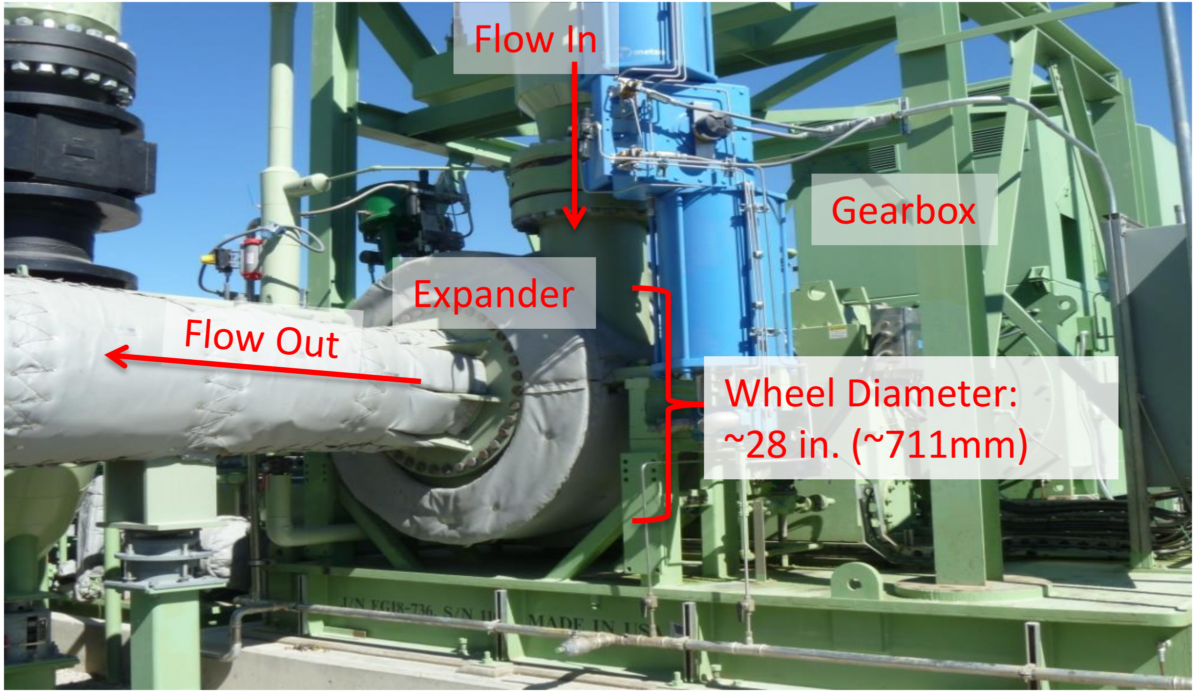
***Thomas P. Shoup** is a Consulting Engineer for Bearings in GE Oil and Gas Turbomachinery Solutions. He was formerly Engineering Manager of Lufkin-RMT, when this work was done. He has worked in the rotating machinery industry for 30 years in rotordynamics and bearing design. Before Lufkin-RMT, he worked for Dresser-Rand, Sverdrup Technology, Inc., and Siemens Demag Delaval Turbomachinery, Inc. He holds a B.S. degree in Engineering Science and Mechanics (VA Tech) and a M.E. degree in Mechanical Engineering (University of Alabama in Huntsville). Mr. Shoup is a member of ASME*

Abstract

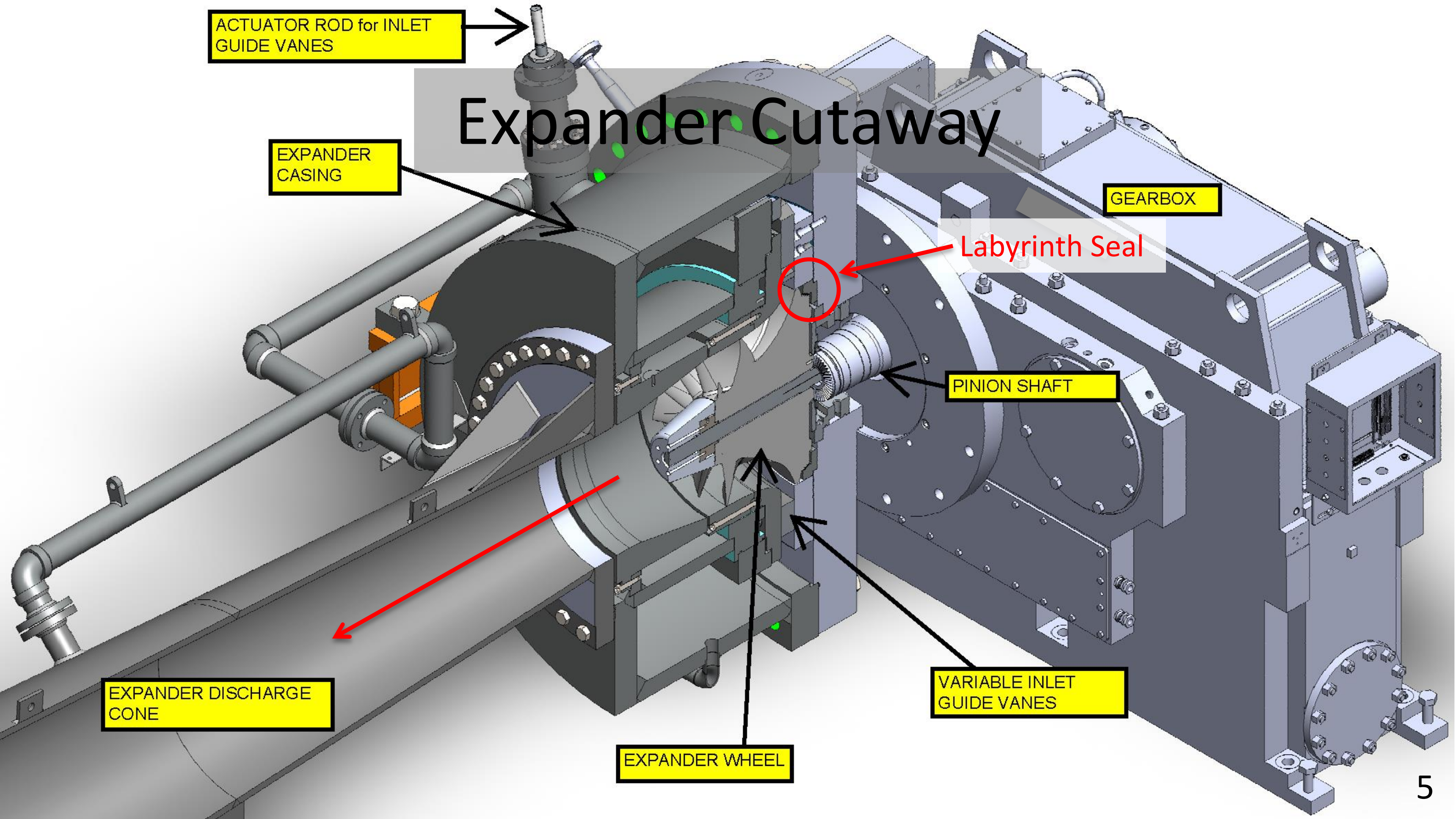
A rotordynamic instability was observed for an overhung, unshrouded, radial inflow expander in a geared generator train. A rotordynamic model was developed to assess stability as well as evaluate the potential design modifications. A transient CFD analysis was found to predict an amount of cross-coupling greater than that predicted by the Alford force equation. Implementation of a swirl brake for the labyrinth seal and modified journal bearings on the expander rotor produced acceptable vibration levels. A unique implementation of a squeeze film damper bearing was employed which reduced the vibration at low loads and enabled reliable operation at all required conditions.

Background: San Emidio Geothermal Expander

- Integrally-geared single-stage turboexpander unit
- R134a working fluid (molecular weight = 102)
- 15 MW rating (normal operating range 10-12 MW)
- Pinion speed 4,515 rpm Output speed 1,800 rpm (driving synchronous generator)

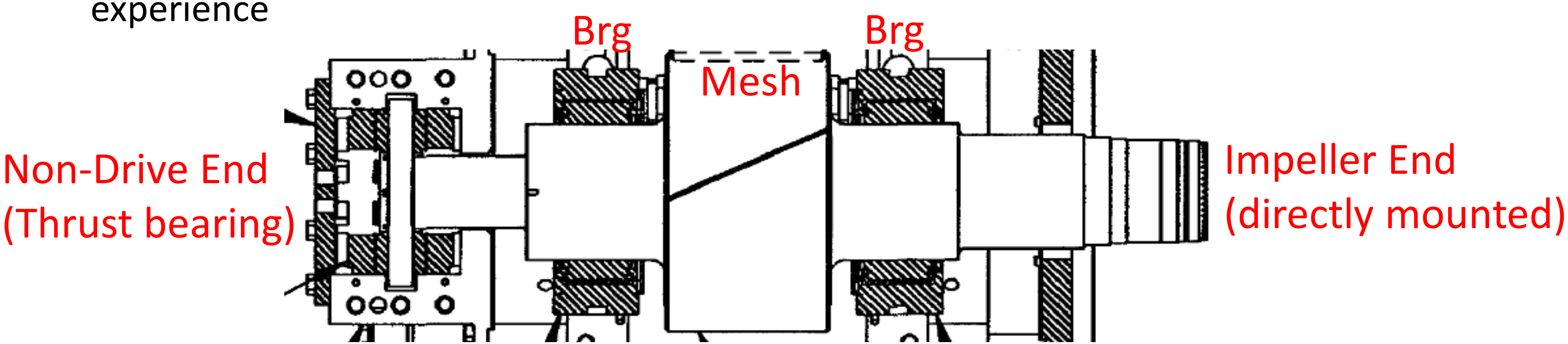


Expander Cutaway



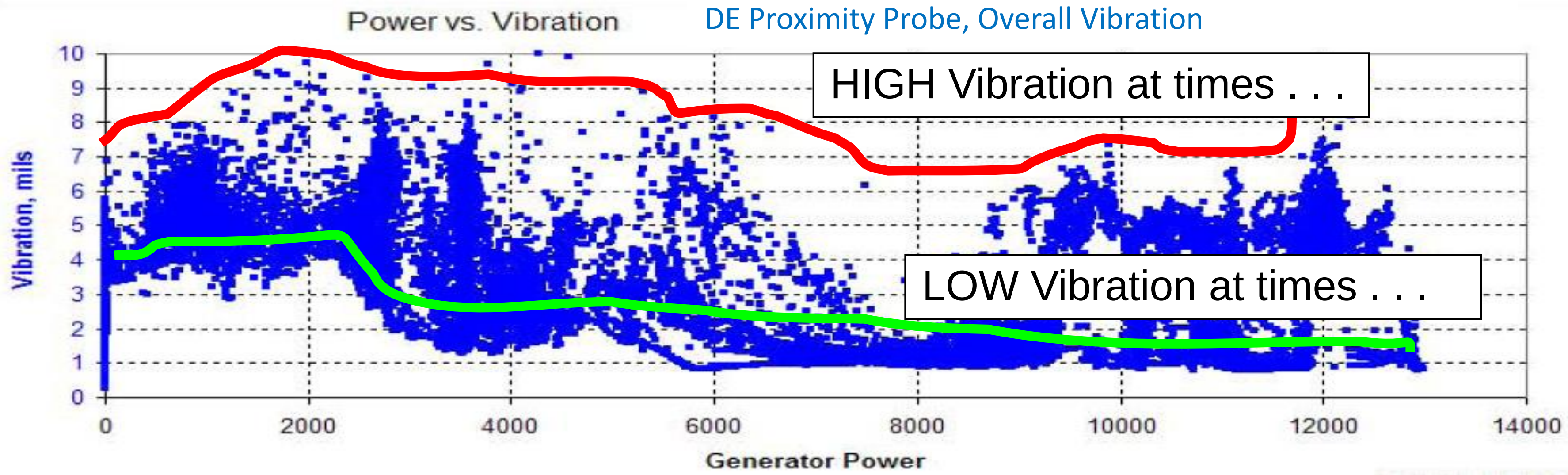
Original Configuration Operating Characteristics

- **PROBLEM: High vibration on pinion shaft drive-end bearing**
  - Operating Speed = 4,515 rpm; Alarm setting = 2.5 mils; Trip setting = 4.0 mils
  - Actual vibration levels: Up to 10 mils p-p at pinion Drive End (DE)
  - Vibration at Non-drive End (NDE) is notably less (on the order of half of DE)
- Onset of vibration at high power (>6 MW) highly sensitive to process variables.
  - Small changes in process conditions would produce large increase in vibration
  - Operator could maintain low vibration levels only by operating at specific points based on experience

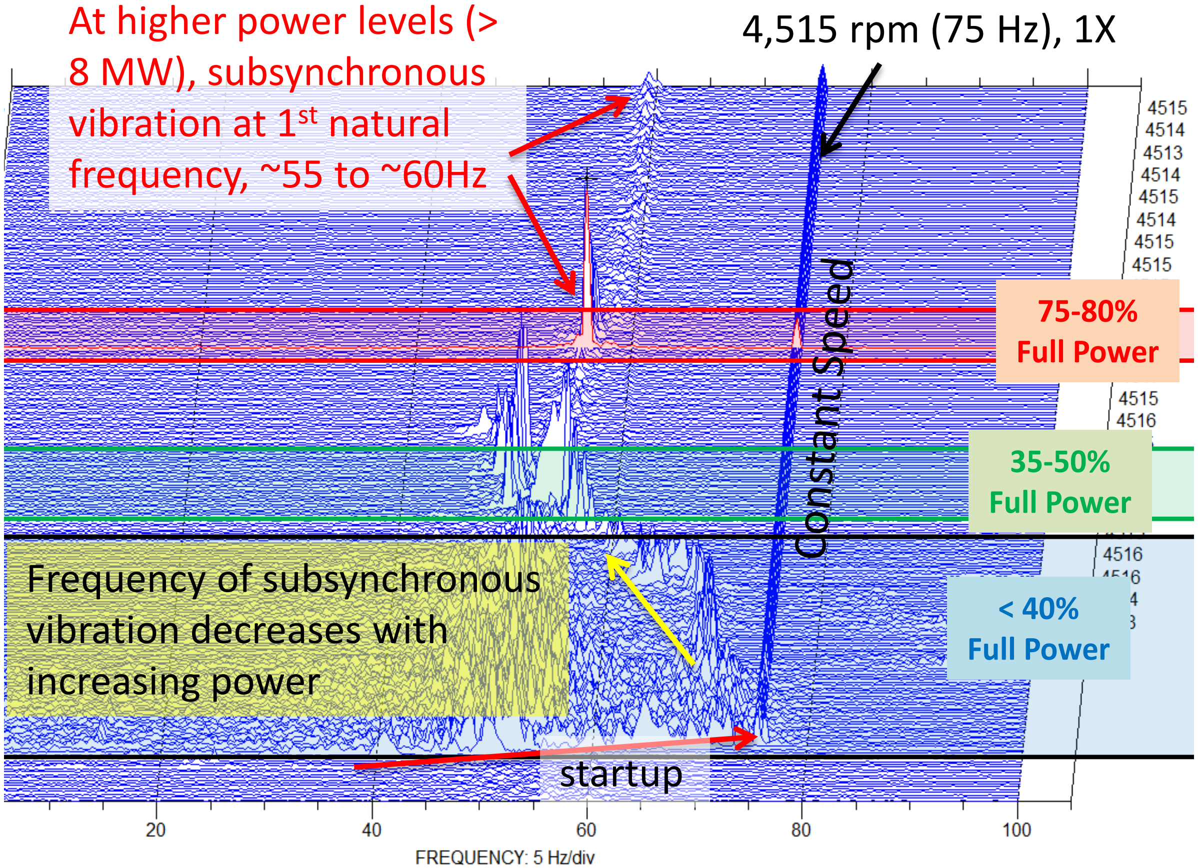


Characteristics of Issue

- Non-repeatable cause-effect relationship was observed between power and vibration, inlet guide vane position and vibration, inlet pressure and vibration, outlet pressure and vibration.
- When plant was completely steady, vibration was also steady. When plant conditions varied (even slightly), the vibration would begin to change, usually rising. This sensitivity to process conditions occurred at high and low powers.

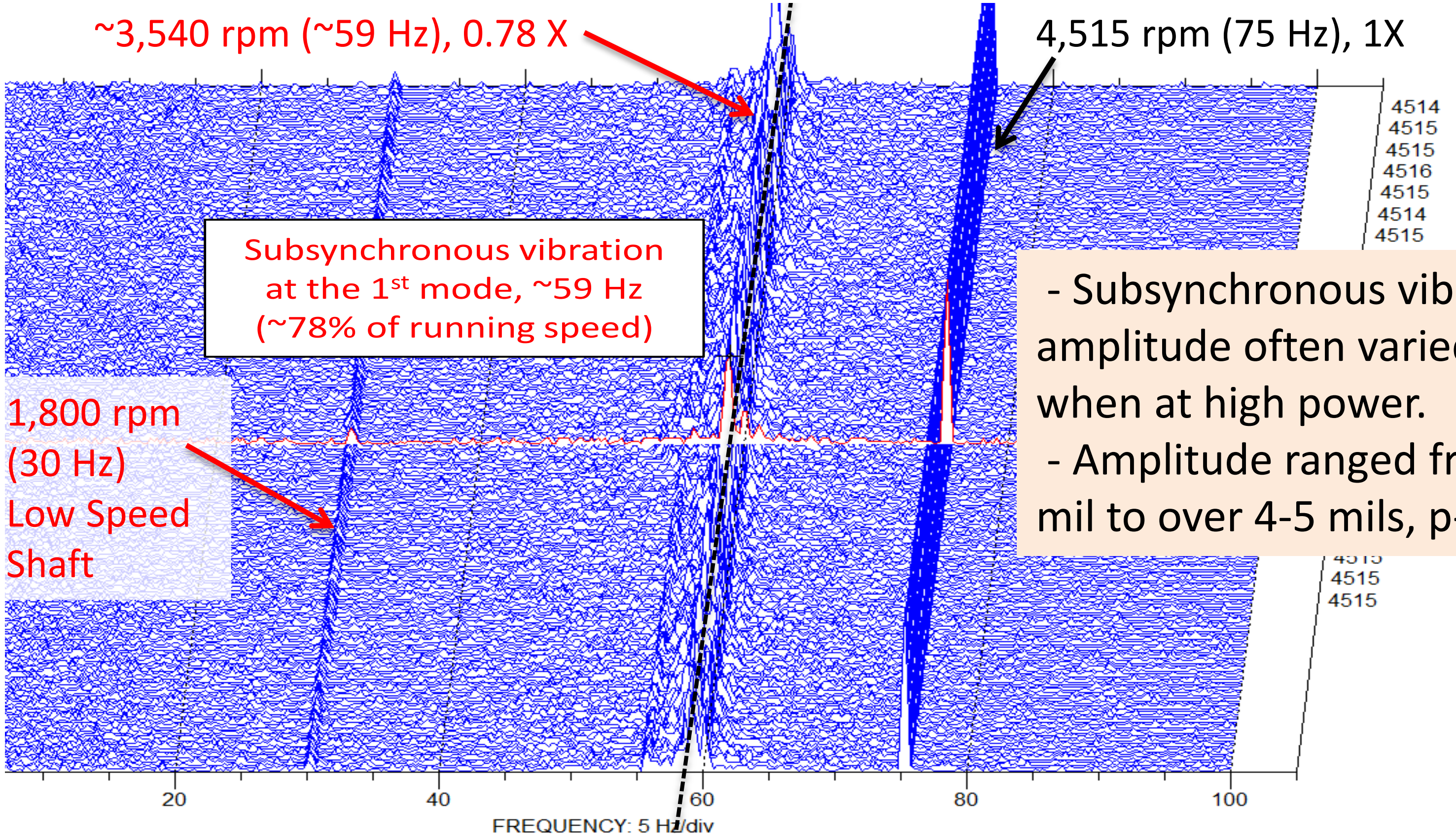


Characteristics of Issue: Waterfall Plot



- <u>Sequence of events:</u>
  - Unit started at low inlet pressure and low power.
  - Generator synchronized at 4515/1800 rpm.
  - Power increased by raising inlet pressure and flow.
  - Broadband subsynchronous vibration at low power levels.
  - Narrow band subsynchronous at high power levels (indicative of rotordynamic instability).

Narrow-band Subsynchronous Vibration at 12 MW: consistent with rotordynamic instability



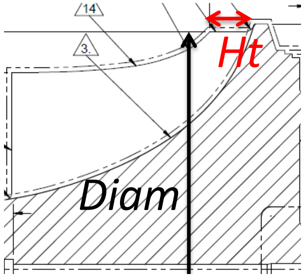
- Subsynchronous vibration amplitude often varied, even when at high power.
- Amplitude ranged from 1 mil to over 4-5 mils, p-p

OEM Engineering Analysis

- OEM worked with gearbox supplier and bearing supplier to investigate the rotordynamic design
  - Theorized that aerodynamic forces acting on the impeller blading and on the impeller-back labyrinth produced shaft motion at the first natural frequency of the pinion of approximately 3,500 rpm (59 Hz), or about 78% of nominal speed
  - The Wachel equation was used to estimate the magnitude of the total aerodynamic cross-coupled stiffness
  - High molecular weight of R134a (MW = 102), is more than 3 times that of typical API gas compressor (MW on the order of 30), indicating higher aerodynamic cross-coupling

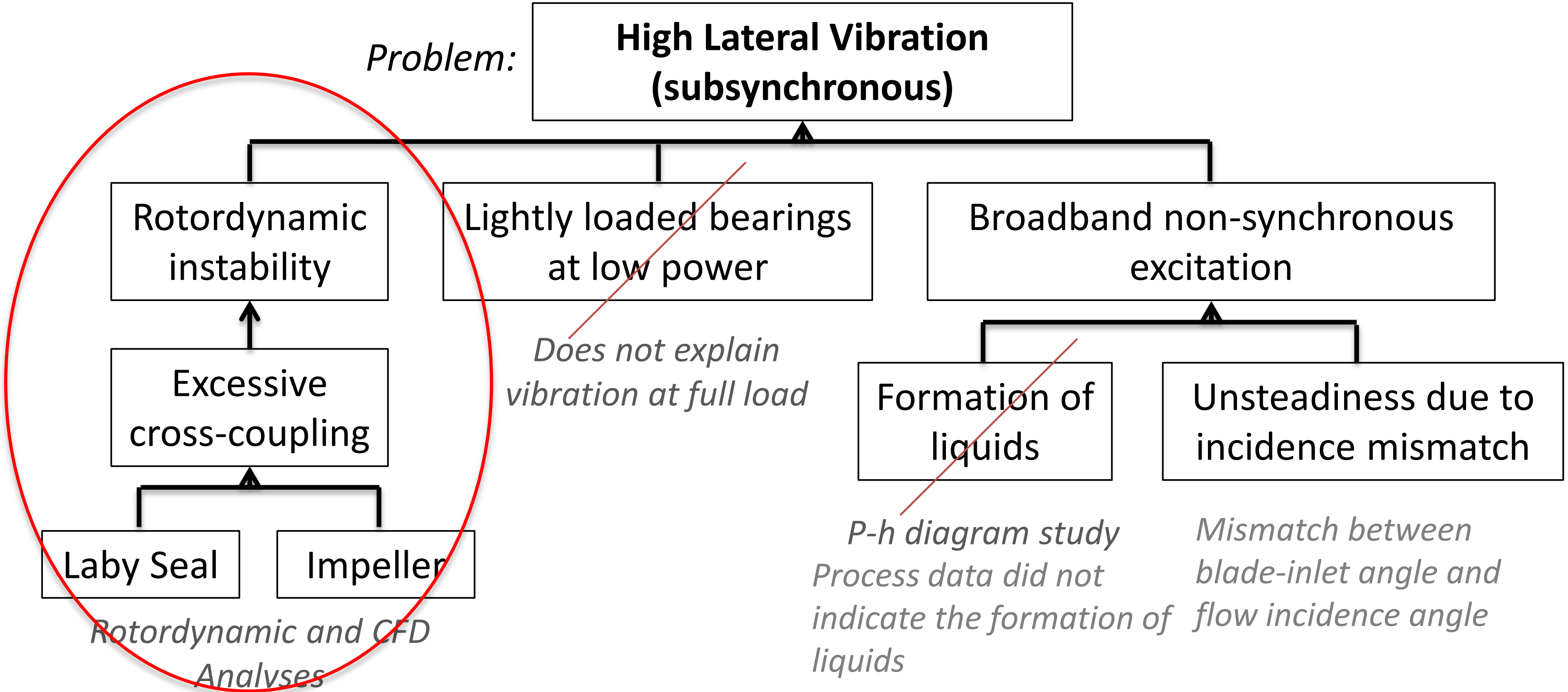
Wachel:

$$Kxy = -Kyx = \frac{6300 * HP * molewt}{Diam * passage\,Ht * RPM} * \frac{\rho_{wheel\,OD}}{\rho_{disch}}$$



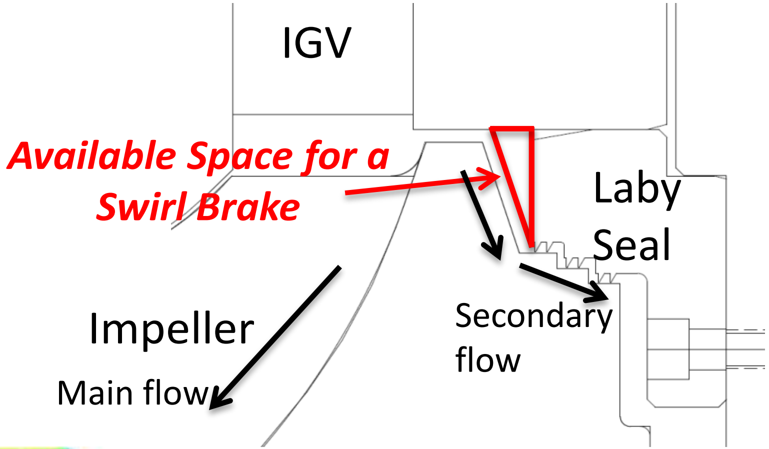
Root Cause Analysis: Fault Tree

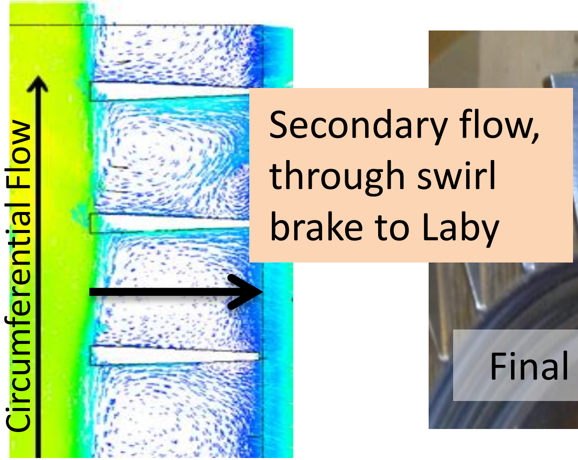
- Investigate means to reduce subsynchronous lateral vibration
  - Decrease cross-coupled stiffness (aerodynamic excitation)
  - Increase damping in system

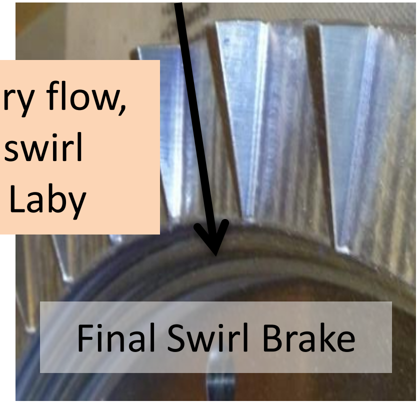


Cross-Coupling Due to Labyrinth Seal

- Estimated value for cross-coupling due to swirl entering labyrinth varies from 9000 to 15,000 lb/in (approximately 1/4 of total magnitude required to induce instability in rotordynamic model)
- Use of "swirl brakes" to eliminate swirl at inlet of labyrinth are typically used to reduce or eliminate associated aerodynamic instability
- CFD (Computational Fluid Dynamic) analysis used to determine optimum number of straightening vanes and overall reduction in inlet swirl.
- Initial swirl (w/out brake) is 1.0 or higher. Swirl brake predicted to reduce swirl to about 0.15.
- Rotordynamic analysis predicted an increase in log dec of about 0.05 (net increase) when adding swirl brake

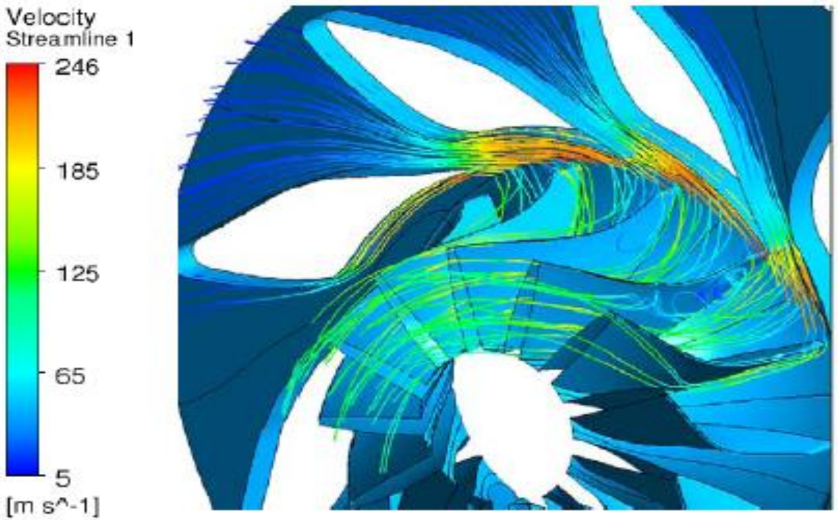






Cross-coupling due to Wheel

- Used CFD to determine magnitude of aerodynamic cross-coupling that can be attributed to the expander wheel
- The transient response model was solved with a displacement of the shroud surface (moving wall transient analysis)
- Assumed a whirl orbit radius of 10% clearance. The aerodynamic cross-coupled stiffness is calculated based on the predicted aerodynamic forces and the orbit radius (dynamic eccentricity)



Streamlines Through the Flow Path are Used to Illustrate the Separation Region at the Impeller Blade Inlet (0.048185 sec)

Learn more: **GT2013-95137**

ROTORDYNAMIC FORCE PREDICTION OF AN UN-SHROUDED RADIAL INFLOW TURBINE USING COMPUTATIONAL FLUID DYNAMICS

Andrew H. Lerche
Grant O. Musgrove
J. Jeffrey Moore
Chris D. Kulhanek
Southwest Research Institute®
San Antonio, Texas, USA

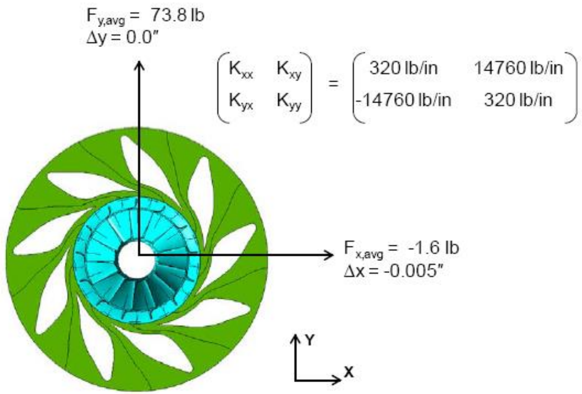
Grant Nordwall
Atlas-Copco Mafi-Trench Company LLC
Santa Maria, California, USA
grant.nordwall@us.atlascopco.com

Impeller CFD Results

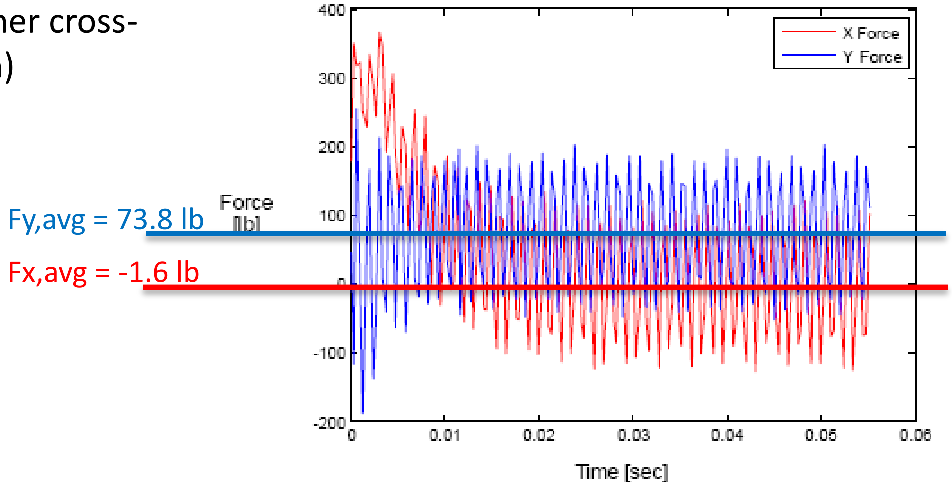
Original Wachel

- Impeller cross-coupled stiffness value of approximately 15,000 lb/in is estimated (accounting for 1/4 of total cross-coupling needed to induce instability in analysis)
  - CFD analysis predicted cross-coupled stiffness approximately 2 times that of original Alford force equation
  - Wachel equation predicts significantly higher cross-coupled stiffness (> 4 times CFD prediction)

Cross-Coupling Prediction Method	k_{xy} (lbf/in)
Alford	6,355
Original Wachel	70,763
API-617 (Wachel based)	20,806
Compressor CFD Equation (Steady State)	10,864
CFD Expander (Unsteady)	14,760



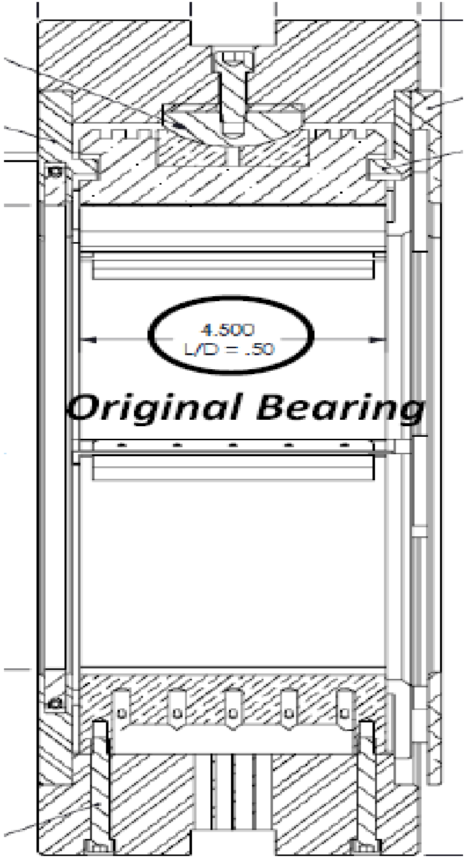
Average impeller forces and impeller displacements used to estimate aerodynamic stiffness



Forces acting on impeller have a blade-pass frequency component acting on top of a near-constant average value

Improved Bearings

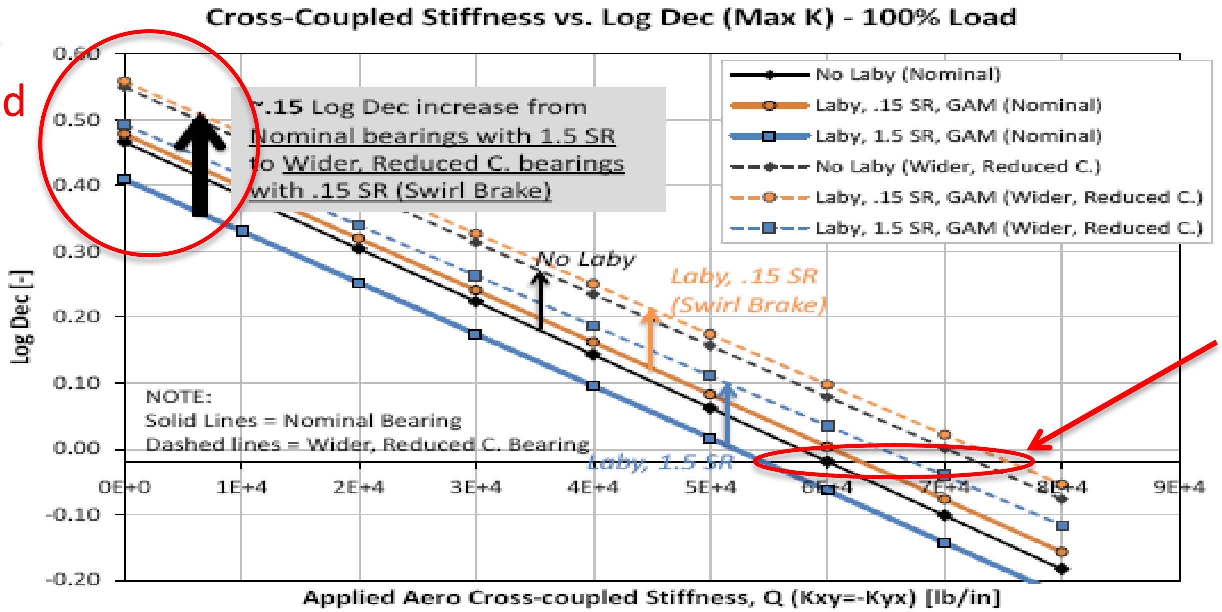
- Determined that optimum rotordynamic stability improvement occurs when:
  - L/D increases from 0.50 to 0.67
  - Offset changes from 65% to 60%
  - Clearance range originally 12-15 mils, (tightened to 9-10 mils by shimming). Now 10-13 mils
  - Preload changed from .21-.56 up to .32-.64
- Sensitivity study of above bearing parameters in the rotordynamic analysis
- New bearings estimated to improve bearing stability. Calculated log dec improvement of 0.10 (plus prior)
- No change to pinion shaft or gearbox casing required.



Summary of Improvements Made

- Qo is the kxy given for a system to produce a log dec = 0
  - "Before": Qo = 55,000-63,000 lb/in
  - "After": Qo= 72,000-75,000 lb/in
- **Log dec increase of 0.15 total (with bearing optimization plus swirl brake)**
  - Improvement of 14-36% (depending on actual bearing clearance)

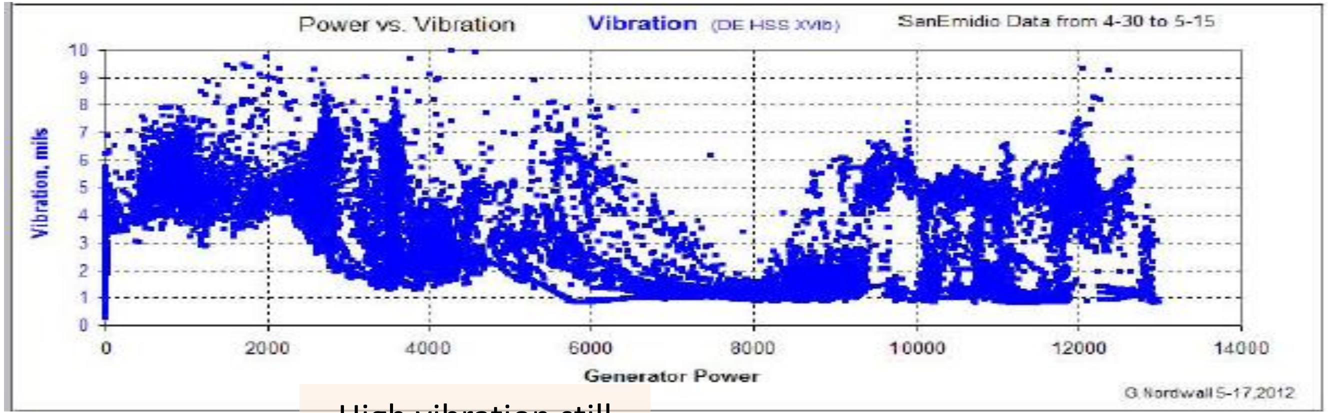
Total increase from improved bearing and swirl brake of ~0.15



Qo

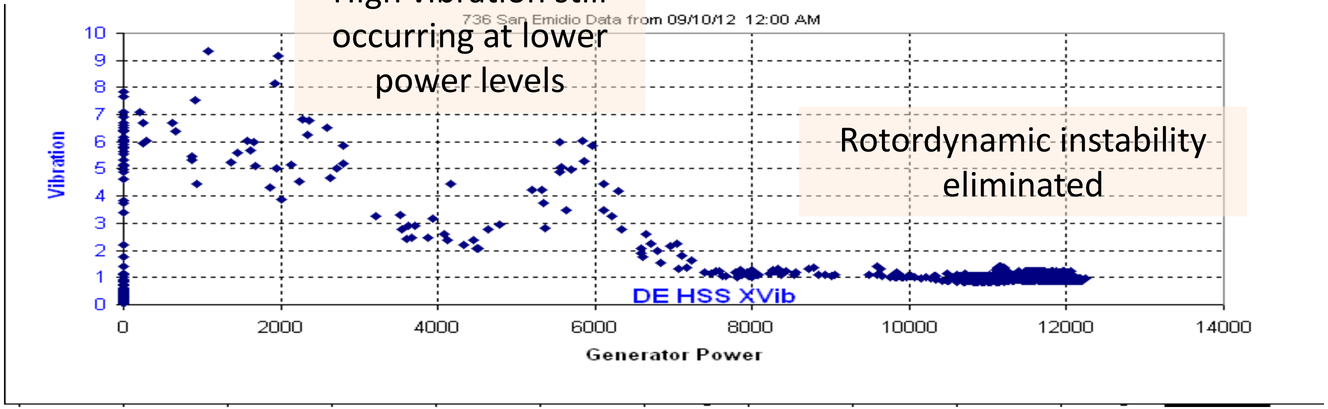
Results: Bearing Vibration

DE Proximity Probe, Overall Vibration



BEFORE

Note: Higher density of data is due to many measurements



High vibration still occurring at lower power levels

Rotordynamic instability eliminated

AFTER

- swirl brake
- improved bearings

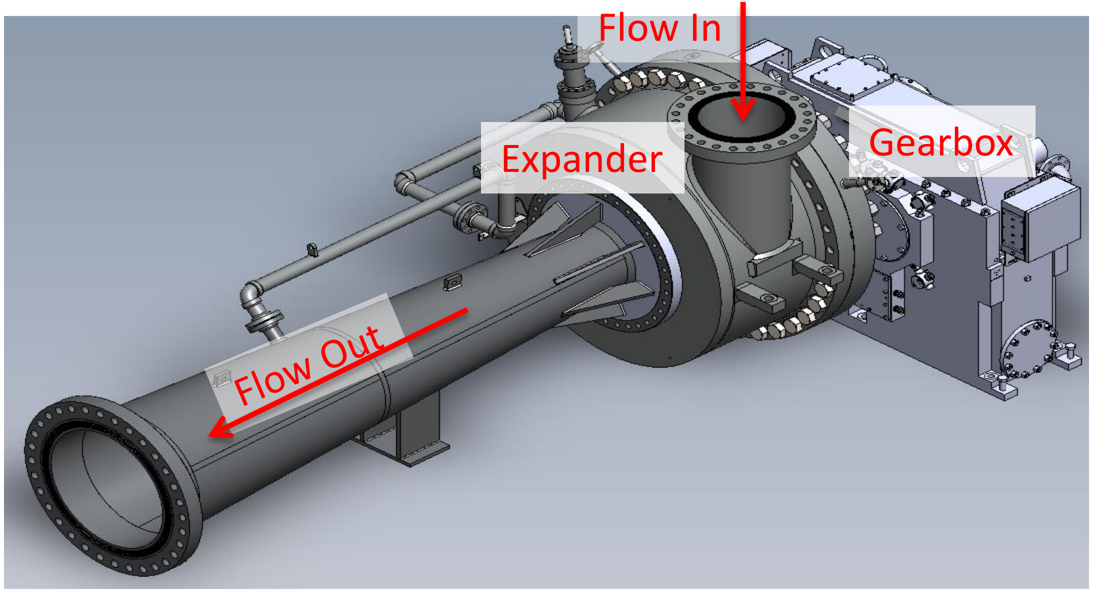
Note: Lower density of data is due to few hours of operation

Conclusion: Phase 1

- Addition of **swirl brake** and introduction of **improved bearings** eliminated the rotordynamic instability observed at high power levels and bearing loads.
- Recommended two-zone high vibration protection levels will allow start-up and temporary operation at low power levels
  - Avoids unnecessary machinery trips at low power levels where pinion bearings are lightly loaded while still protecting machinery at higher loads
- The same improvements (improved bearings plus swirl brake) applied at Neal Hot Springs
  - As of 23 Oct. 2012, two trains at NHS have operated up to 11 MW and show no sign of rotordynamic instability

A New Situation:

- Three expander units installed at Neal Hot Springs (NHS) geothermal plant in Eastern Oregon (similar design to previous shown expander)
- Rated at 15 MWe each, operating range 5 MWe up to 13 MWe
- Air-cooled Organic Rankine Cycle (ORC) plant. Condenser pressure varies from 66 psig (winter) up to 187 psig (hottest summer day)

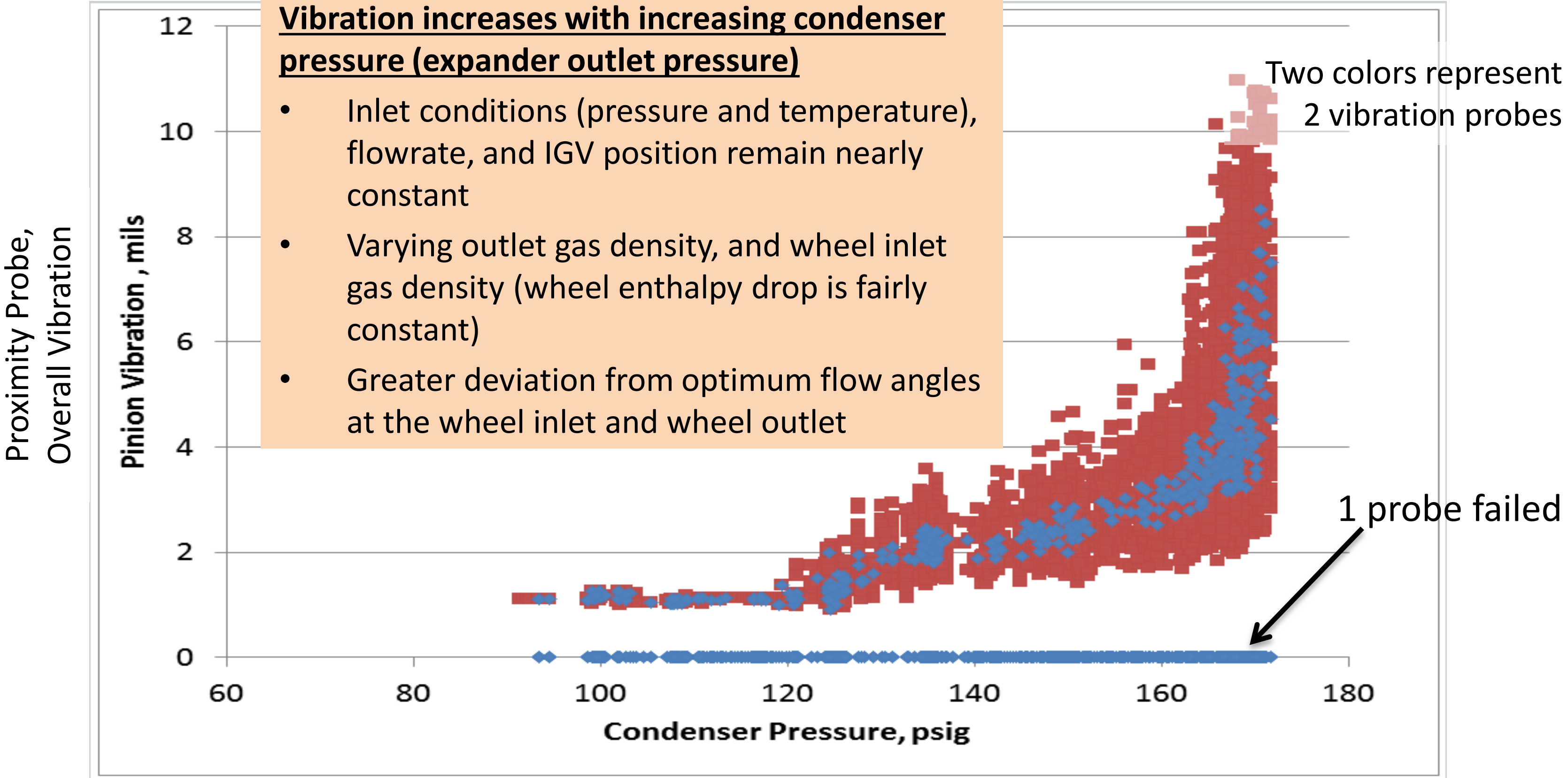


The Problem:

- High pinion vibration during test of simulated summer operating conditions

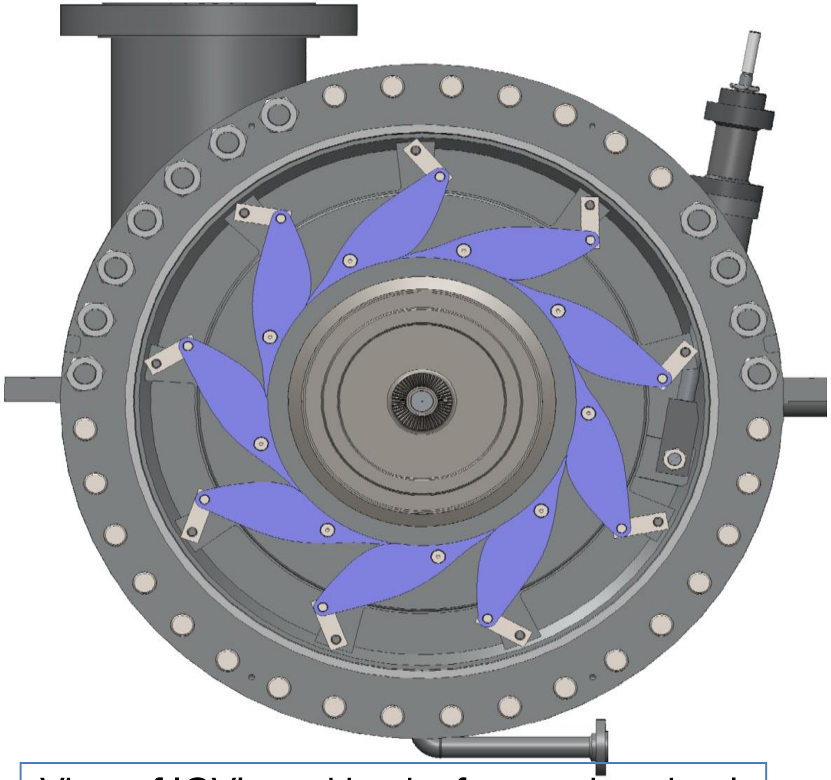
Vibration increases with increasing condenser pressure (expander outlet pressure)

- Inlet conditions (pressure and temperature), flowrate, and IGV position remain nearly constant
- Varying outlet gas density, and wheel inlet gas density (wheel enthalpy drop is fairly constant)
- Greater deviation from optimum flow angles at the wheel inlet and wheel outlet



Probable Cause of High Vibration

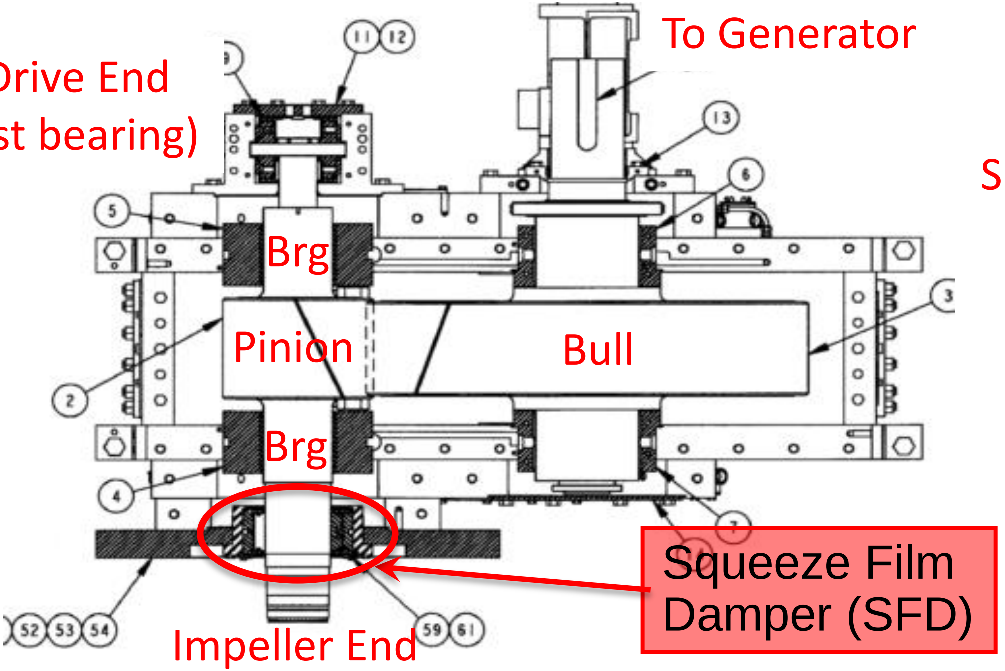
- Variable inlet guide vanes (IGV's)
  - Direct flow into impeller
  - Create swirl at the inlet of the expander wheel
- More inlet swirl, increased density (due to increased pressure) both tend to increase destabilizing forces (cross-coupled stiffness)
- Flow conditions and IGV position effect the incidence angle of the flow entering impeller
- Possible causes of high vibration:
  - Rotordynamic instability
  - Broad-band excitation due off design flow angles
- Unfortunately, vibration data at this site is limited to overall vibration (no frequency content); Difficult to determine root cause of vibration
- Design approach: **<u>Add damping</u>** to rotor system

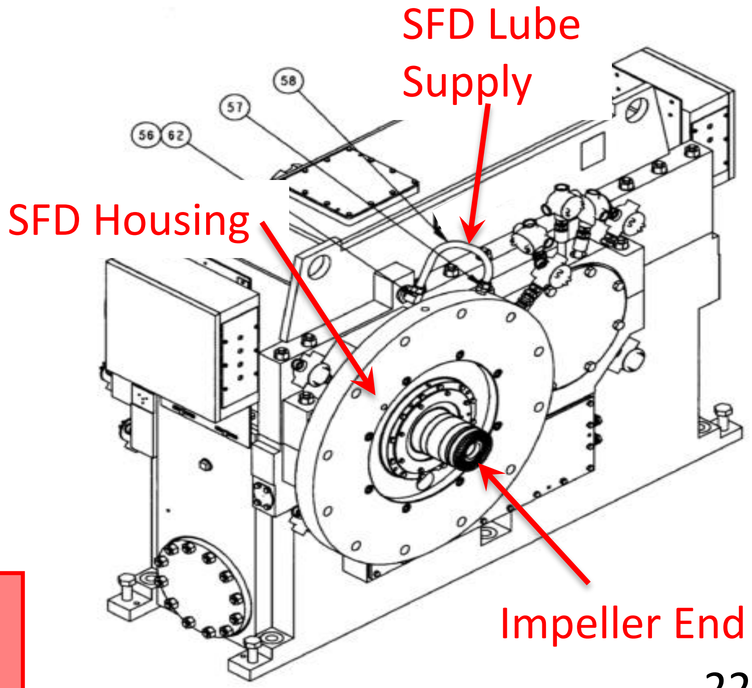


View of IGV's and back of expander wheel after removing gearbox and housing cover plate. IGV's shown in fully closed position.

Solution: Add Squeeze Film Damper (SFD)

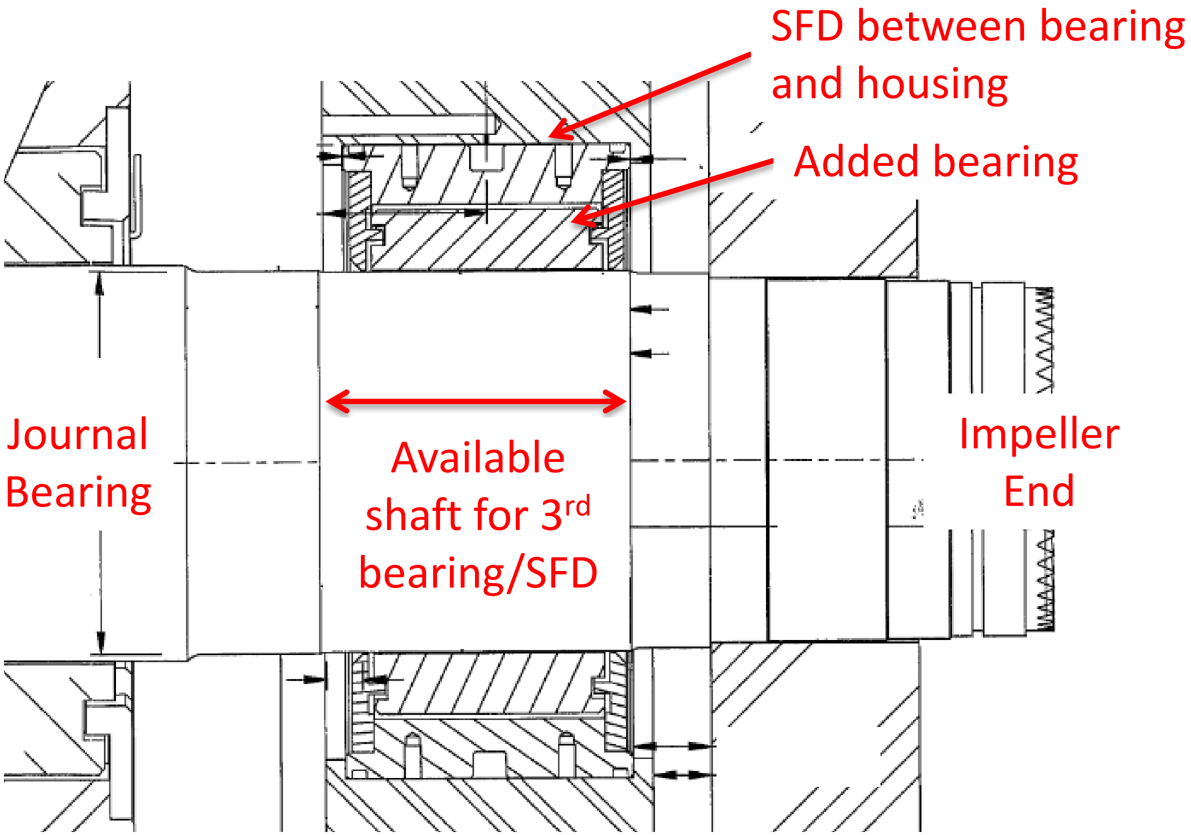
- SFD is located between pinion drive-end bearing and pinion mechanical seal
  - External tubing used for oil supply to SFD (see figure on right below)
  - Shaft modification limited to machining/grinding at SFD location
- SFD adds damping closer to source of vibration (i.e. closer to expander wheel)
  - More effective than increased damping at bearing





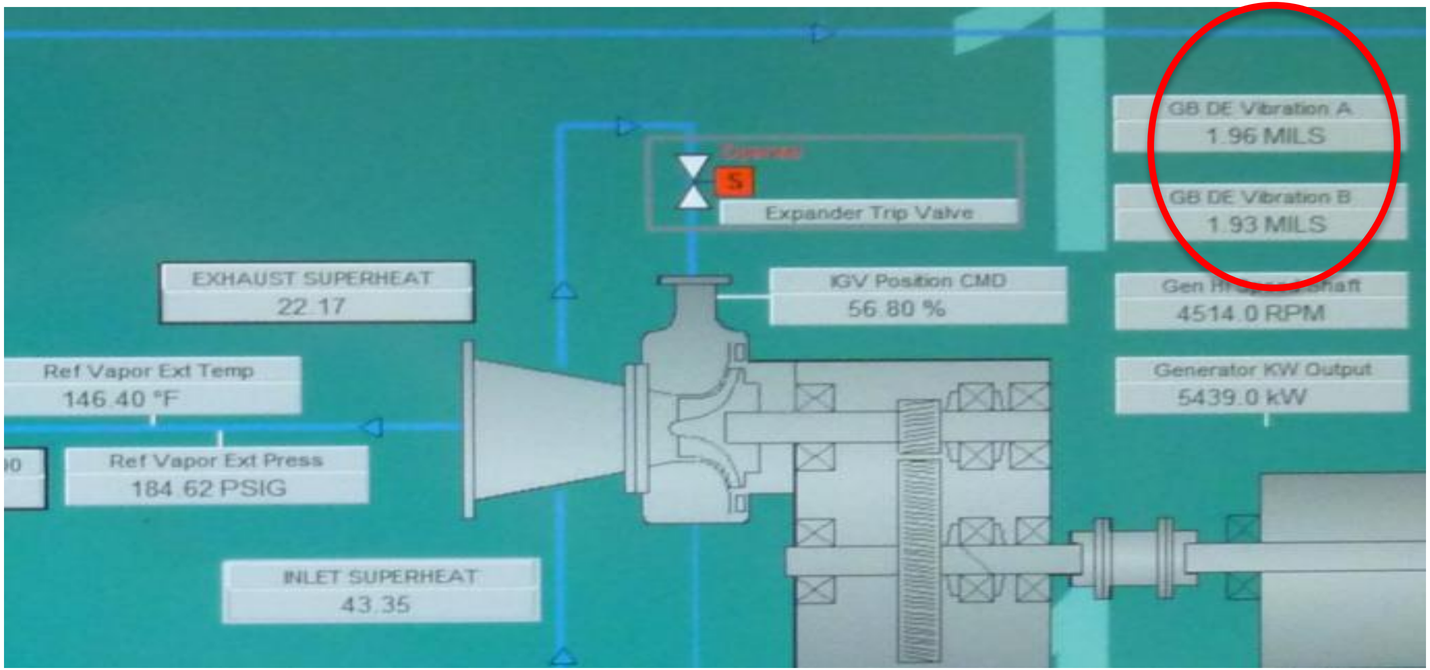
Squeeze Film Damper (SFD) Details

- SFD adds damping to the rotor system by displacing (squeezing) fluid within an annulus as a result of lateral motion/displacement
- The loading on the two original bearings was too high to permit a SFD to function
- For this application, the design included a third bearing, operating unloaded
- The two traditional journal bearings continue to support the loading, while the SFD bearing runs centered allowing effective motion of the damper annulus



Results: Acceptable vibration at high outlet pressure and low power

- Was >10 mils at 185 psig outlet pressure
- Now <2 mils at 185 psig outlet pressure



Conclusion: Phase 2

- Addition of **swirl brake** and introduction of **improved bearings** eliminated the rotordynamic instability previously observed at high power levels and bearing loads
- Addition of **squeeze film damper** was required to achieve acceptable vibration levels at lower power (less than 8 MW) and higher condenser pressures (greater than 120 psig)

Thank you

- A special thank you to all parties involved:
  - Atlas Copco Mafi-Trench Company (ACMT)
  - Lufkin, part of GE Oil & Gas
  - Southwest Research Institute (SwRI)