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GEORGE R. BROWN CONVENTION CENTER

UNDERSTANDING DESIGN PARAMETERS THAT AFFECT THERMAL STABILITY OF HIGH-SPEED TURBO MACHINERY (ALSO KNOWN AS THE MORTON EFFECT)

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TEXAS A&M ENGINEERING EXPERIMENT STATION

Presenter/Author bios

Mr. Benton graduated in 1989 from Carnegie-Mellon University with a B.S. in Mechanical Engineering. He has spent the last 29 years in Air Products within the CryoMachinery Department, designing and commissioning cryogenic expanders and compressors for use in air separation and hydrocarbon service. Focuses included equipment for oxygen enriched, flammable and toxic services, rotordynamics and finite element analysis. From 1989 to 1995, he was responsible for Engineering, Testing, and Project Execution for various expander projects. In 1999, Mr. Benton became the lead engineer in the mechanical engineering group within the CryoMachinery department of Air Products and in 2003 assumed the role of Head of the Engineering Group. In 2012 Mr. Benton entered into the role of Global Expander Technology Manager. He is an active member of API-617 and API-614 task forces.

Presenter/Author bios

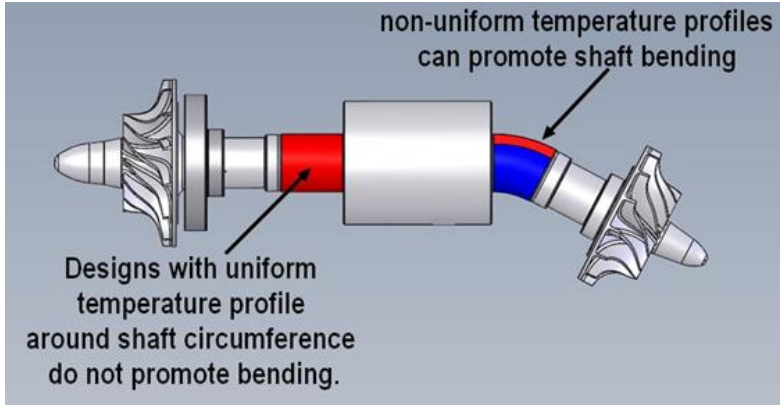
Mr. Eiswerth graduated in 2002 from Bucknell University with a B.S. in Mechanical Engineering. He began his career as a maintenance and reliability engineer in Air Products' Pace, FL plant. In this role, he conducted functional performance evaluations of plant equipment, performed root cause analysis for pump mechanical seal failures, and directed capital expense projects to improve equipment reliability and mechanical integrity. Mr. Eiswerth joined the CryoMachinery Department in 2003, where he successively served as a manufacturing engineer, a test facility engineer, and a tooling engineer with significant involvement in the manufacturing and testing of the companders for the land based AP-X® LNG Process Trains. In 2011, he became a machinery design engineer with a primary focus on the design, manufacturing, testing, commissioning, and startup of the companders for the offshore AP-N™ LNG Process Trains.

Abstract

At present, there are no commercially available codes in industry that have been found to be accurate, reliable and consistent enough to predict both the onset of the Morton Effect and a rotor's response to the Morton Effect to be used as a design tool with hard acceptance criteria in the upfront design of turbomachinery. The available tools have however shown the ability to predict the proper trends associated with many changes made to help enhance machine stability when the Morton Effect is recognized. The Morton Effect refers to synchronous rotor instability due to non-uniform heating of shaft journals. The industry's inability to reliably and consistently predict this phenomenon has caused both plant start-up delays and shutdowns due to machinery vibration. The multiple case studies that will be presented assess this problem and summarize the solutions that were developed, tested and ultimately implemented to address the Morton Effect.

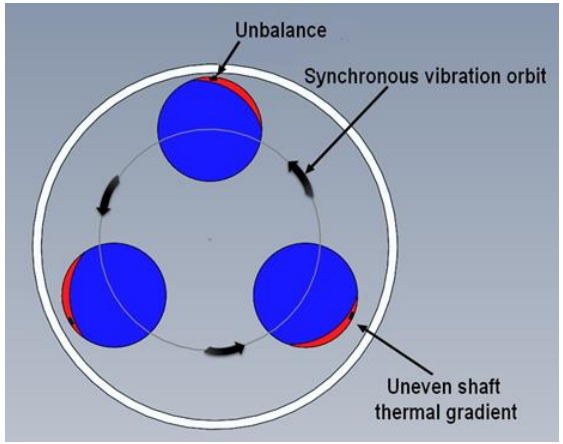
Morton Effect

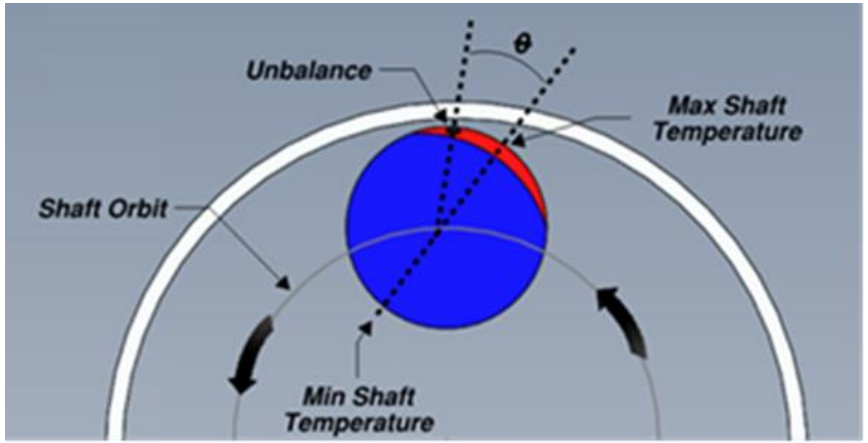
A non-uniform temperature distribution along the journal circumference which can lead to rotor bending, which, in combination with an overhung mass such as overhung impellers, can significantly increase rotor imbalance and thus synchronous rotor vibration. The Morton effect is also known as the hot spot phenomena.



Morton Effect

The non-uniform temperature distribution has a peak temperature (hot spot) that lags the phase angle of the unbalance (high spot) and as such results in a forward procession of the high spot on the shaft due to shaft distortion.





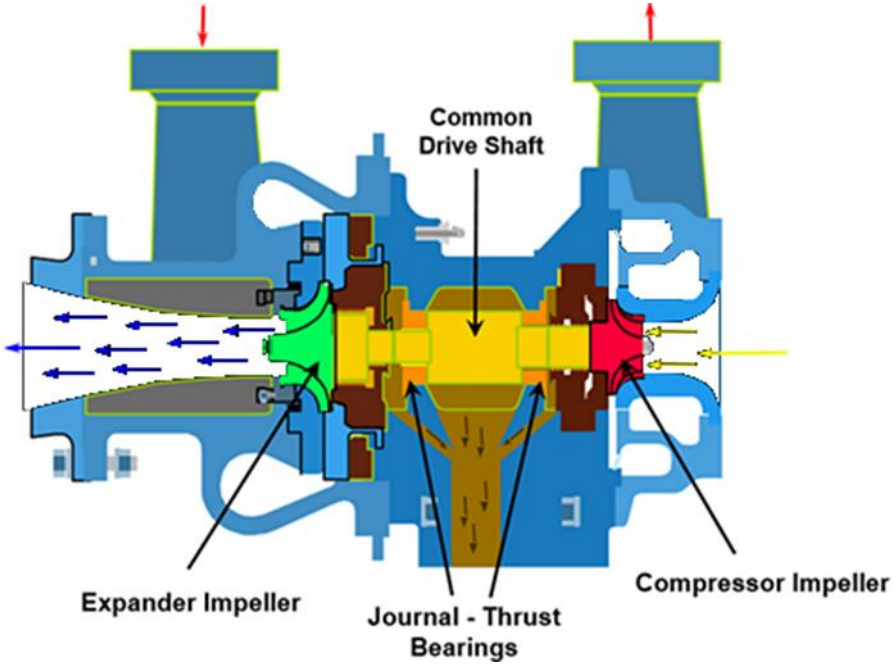
Morton Effect

Numerous predictive codes (VT-Map, TRC-TAMU, XLVCEL, MADYN 2000, etc.) exist to help identify and deal with the Morton Effect. Our design and analysis experience is limited to the use of VT-map, which generally shows good agreement with trends on susceptibility to Morton Effect but onset prediction accuracy has not been suitable for upfront design purposes.

CASE STUDY #1

- This case study involves a 1,500 HP cryogenic expander-driven compressor operating in a nitrogen loop. The unit design consisted of a 7" expander impeller driving a 7.5" compressor impeller at a maximum continuous speed of 31,400 rpm. The rotor had dry gas seals on each stage.
- Rotor dynamic analysis predicted a stiff shaft design with the first critical speed at 46,500 rpm, (48% above MCS). The rotor bearing system met all API-617 acceptance criteria required for a stable rotor bearing system.

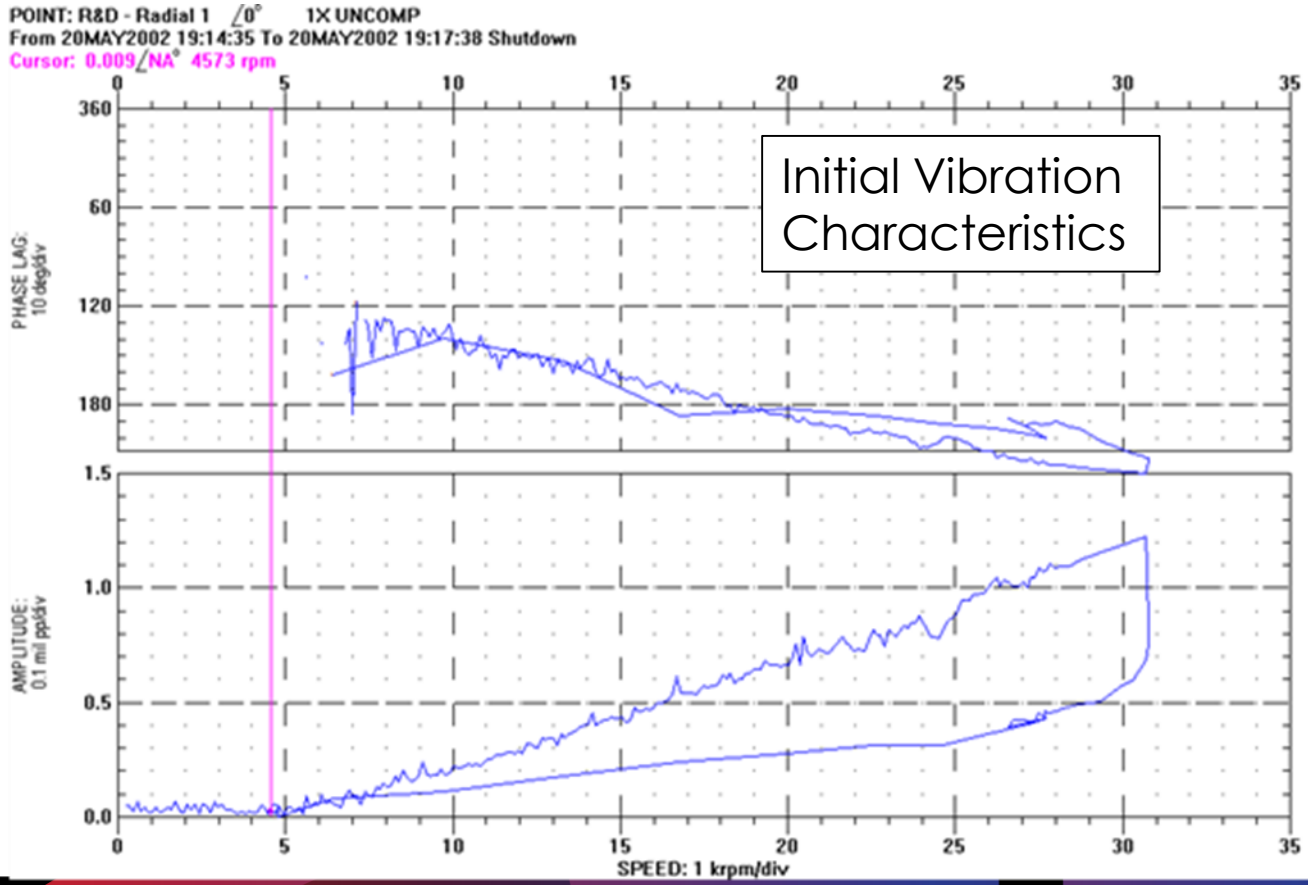
CASE STUDY #1


Common
Drive Shaft
Expander Impeller
Journal - Thrust
Bearings
Compressor Impeller

CASE STUDY #1

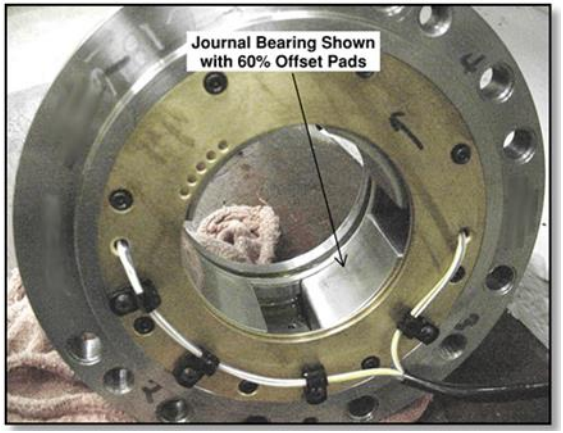
During the shop test the unit exhibited unstable vibration at speeds above 30,000 rpm.
A Bode plot of the 1x synchronous vibration and phase vs speed clearly shows the classic hysteresis loop associated with the Morton Effect. Operating above 30,000 rpm the vibration is shown to increase at constant speed. In addition, coastdown vibration follows a different track from that recorded during run up.

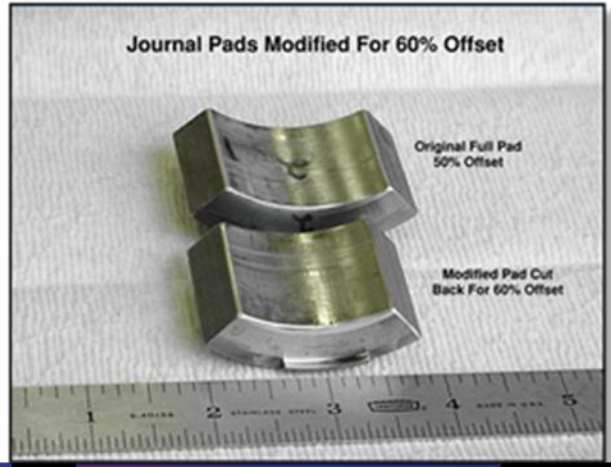
CASE STUDY #1


POINT: R&D - Radial 1 ∠0° 1X UNCOMP
From 20MAY2002 19:14:35 To 20MAY2002 19:17:38 Shutdown
Cursor: 0.009∠NA° 4573 rpm
PHASE LAG: 10 deg/div
AMPLITUDE: 0.1 mil pp/div
SPEED: 1 krpm/div
Initial Vibration Characteristics

CASE STUDY #1

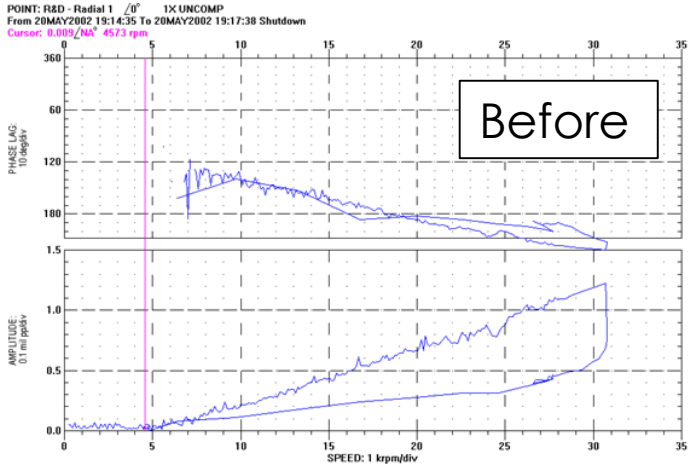
Corrective Actions: For each of the five bearing pads, the trailing edge was machined back to achieve a 60% offset pivot design. Analysis of the bearing models predicted the reduced pad lengths would reduce HP loss and pad film temperatures. The model also predicted an increase in stiffness and damping that would reduce synchronous orbits and lower differential journal temperatures.

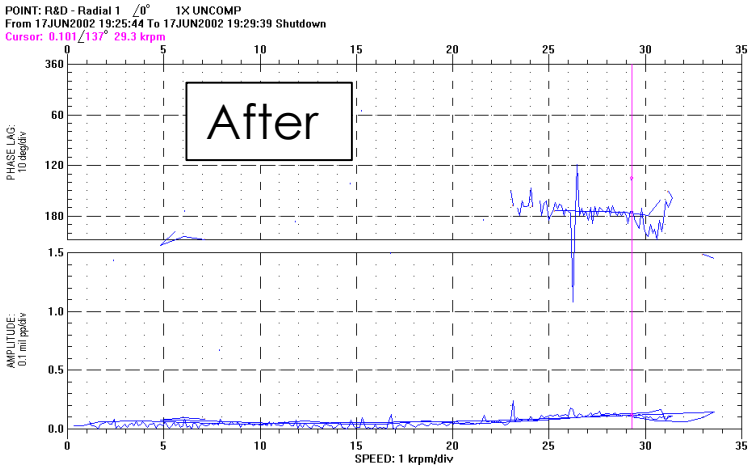




CASE STUDY #1

Test Results with Modified Bearings: The 60% offset pad modifications eliminated the Morton Effect. In addition, the fact that hysteresis has been eliminated indicates that the rotor is not bending and balance is maintained at all operating speeds.





Case Study 2

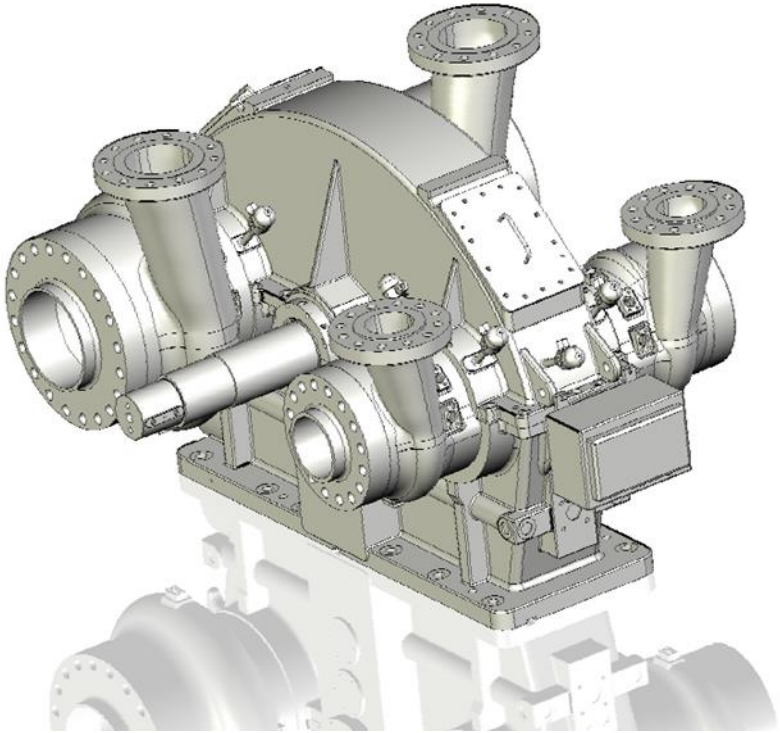
4,000 HP twin-pinion, four-stage, integrally geared, motor driven cryogenic nitrogen compressor.

LS pinion: 18,482 rpm and 1,750 HP

HS pinion 28,955 rpm and 2,250 HP

A detailed rotor dynamic analysis showed that both pinions met all API-617 acceptance criteria for a stable rotor bearing system. The LS pinion was predicted to be a stiff shaft design with its first critical speed 35% above design speed

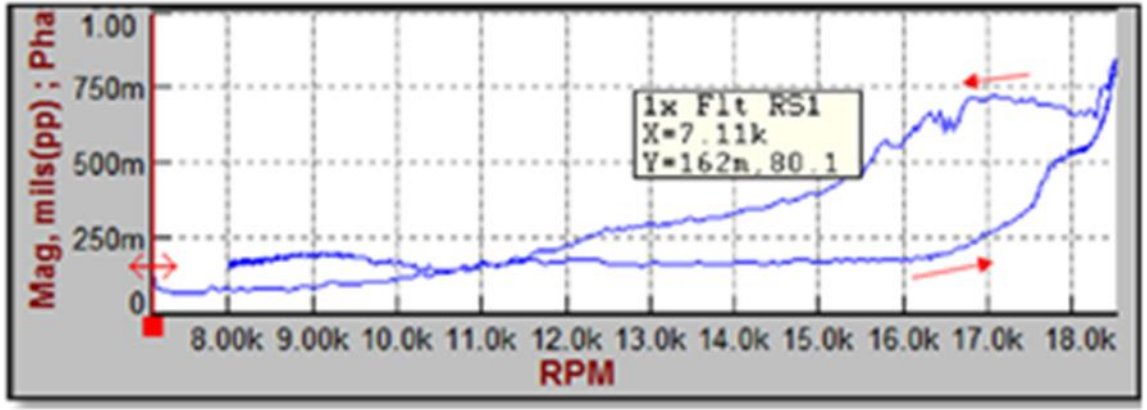
Case Study 2



Case Study 2

During the shop test, the low-speed pinion exhibited unacceptably high vibration that was sensitive to oil supply temperatures. This sensitivity to oil temperature is a known characteristic of the Morton Effect. Cool oil will increase oil viscosity and shaft differential temperatures, which acts to increase the thermal instability of the rotor. In addition, a Bode plot of the 1x synchronous vibration vs speed showed a hysteresis loop typically associated with the Morton Effect.

Case Study 2



	Supply Oil Temperature		
	Cool	Design	Hot
Pinion rpm	18,482	18,482	18,482
Oil temperature	100°F	120°F	140°F
Total vibration (mils)	1.6	1.3	1.2

Case Study 2

Shop test at ~10% rated power. This resulted in lightly loaded journal bearings where residual imbalance dominates the shaft orbit. In the field, however, the compressor would be driven by a 4,000 HP induction motor where gear loads would force the shaft toward one side of the bearing. This increase in shaft eccentricity exposes the entire journal surface to close clearance oil shear resulting in even heat distribution of the journal circumference

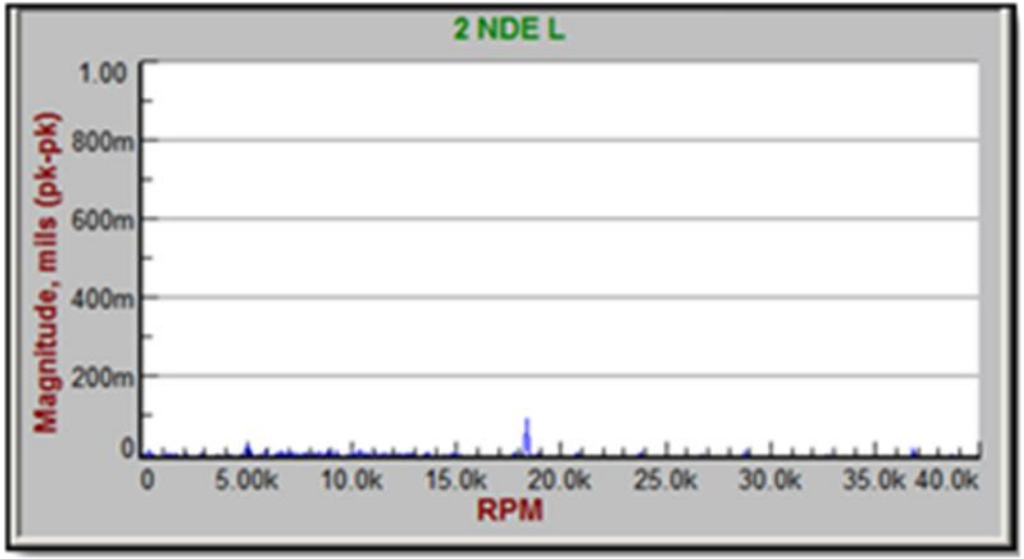
Case Study 2

Vibration > API acceptance criteria for shop test, but viewed as stable. Because the vibration could be reduced with increased oil temperatures and/or reduced oil viscosity, it was thought that full-load operation in the field would likely eliminate any thermal instabilities. If instability was observed, the oil supply temperature could be increased and the oil viscosity would be reduced from an ISO VG-46 to an ISO VG-32.

Case Study 2

This hypothesis was supported when field operation with increased rotor load (and eccentricity) did, in fact, eliminate the Morton Effect.

Compressor vibration never exceeded 0.4 mils while operating at 60% to 100% rated power and with oil supply temperatures at 90°F to 130°F



Case Study #3

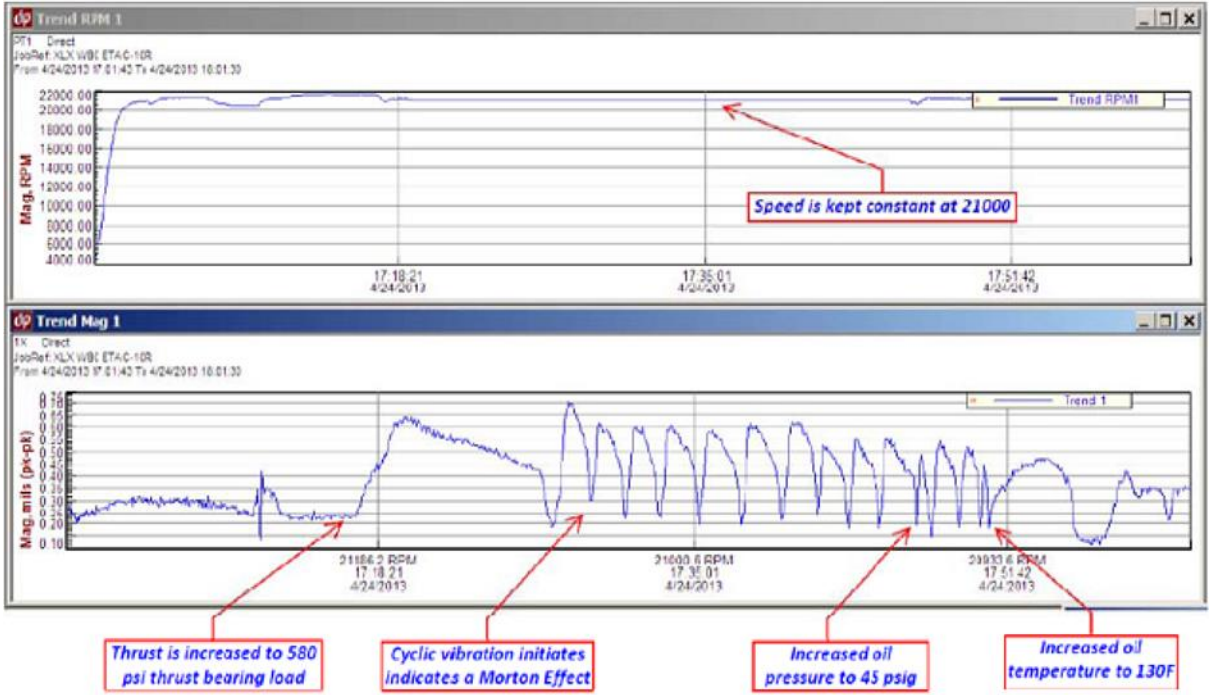
This case study involves a 3,400 HP cryogenic expander-compressor with a 10" expander and 8" compressor impeller (both aluminum) at a maximum continuous speed of 21,000 rpm.

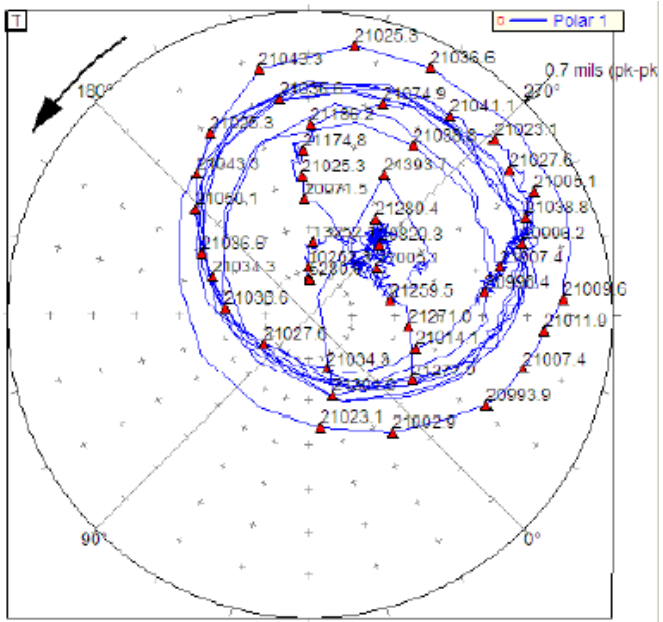
The rotor bearing analysis met all API-617 acceptance criteria for a stable rotor bearing system, with the first critical speed at 37,000 rpm, or 76% above its design speed.

Case Study #3

During the shop test, operation at a constant speed of 21,000 rpm, at thrust loads of 500+ psi, the vibration would cycle between 0.7 mils and 0.1 mils, with the cycle time ranging from 2 to 3 minutes. Increasing oil pressure from 30 to 45 psig resulted in a 10 to 20% reduction in both the vibration amplitude and cycle time. Increasing the oil temperature from 120°F to 130°F restored thermal stability and resulted in a steady vibration of 0.35 mils. The phenomena did not appear for bearing loads < 150 psi.

Case Study #3


Trend RPM 1
Trend RPM1
Mag, RPM
Speed is kept constant at 21000
17:18:21
4/24/2013
17:35:01
4/24/2013
17:51:42
4/24/2013
Trend Mag 1
Trend 1
Mag, mils (pk-pk)
Thrust is increased to 580 psi thrust bearing load
Cyclic vibration initiates indicates a Morton Effect
Increased oil pressure to 45 psig
Increased oil temperature to 130F


Polar 1
0.7 mils (pk-pk)
180°
270°
90°
0°

Case Study #3

An alternate bearing design was chosen which was able to maintain vibration levels below 0.3 mils.

It is worth noting that even this alternate design exhibited Morton effect characteristics although the overall vibration levels were low enough to not be a concern.

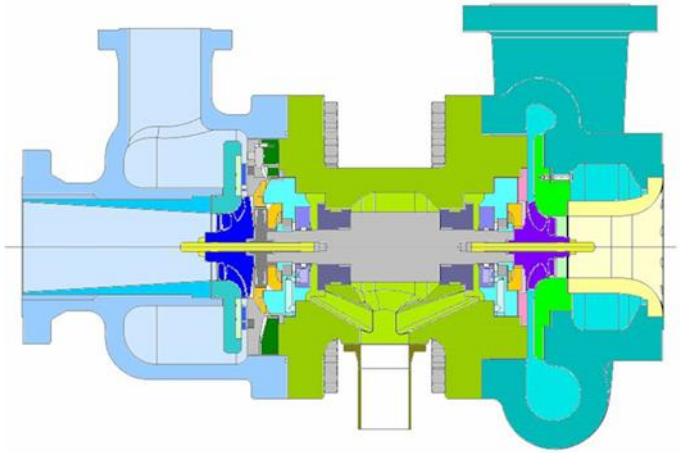
VT-map had predicted the alternate bearing design to be well below the onset of instability due to the Morton effect.

Case Study #3

In a later application, a stainless steel expander impeller was utilized and was enough of a change to induce unacceptable levels of vibration due to the Morton effect in operation, even with the alternate bearing design. This trend of reduced RPM to cross the stability threshold was also replicated in an analysis with VT-map, although the program predicted the passing of the stability threshold at 24,867 RPM which would have been well above operating speed.

Case Study #4

A 12,000 HP cryogenic expander-driven compressor operating on nitrogen for a shipboard LNG application. The design consisted of a 16" expander impeller and a 20" compressor impeller (both SST) at a maximum continuous speed of 11,000 rpm



Case Study #4

Rotor dynamic analysis predicted this would be a stiff shaft design with the first critical speed at 16,000 rpm, or 33% above its maximum continuous speed. The rotor bearing system met API-617 acceptance criteria for a stable rotor bearing system.

However, during the shop test, the Morton Effect was observed at speeds above 11,000 rpm when operating on cool 100°F oil but was stable at this speed at higher temperatures.

Case Study #4

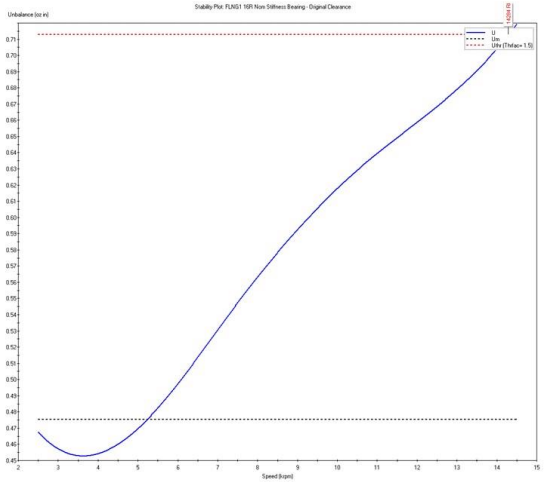
The rotor overhung weight could not be reduced because the high power density of this application demanded a SST impeller. Cutting back the trailing edge of the pads – the solution in Case Study #1 – was not an option, as the pads were designed with distinct, directed lube grooves and SSV grooves.

Case Study #4

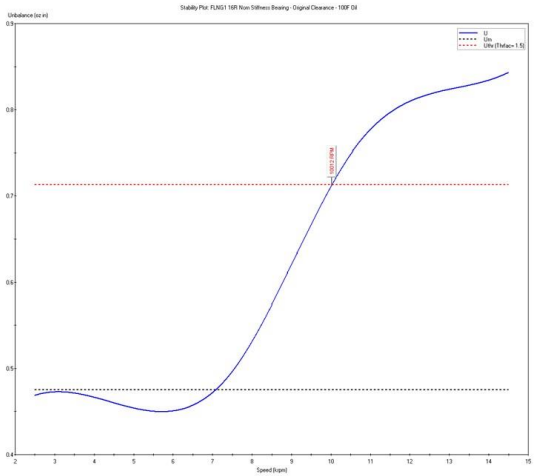
The bearing clearance was increased to reduce viscous shear and differential shaft journal temperatures. VT-map predicted this to have a beneficial effect. This modification did in fact eliminate the Morton Effect during the factory shop test.

Case Study #4

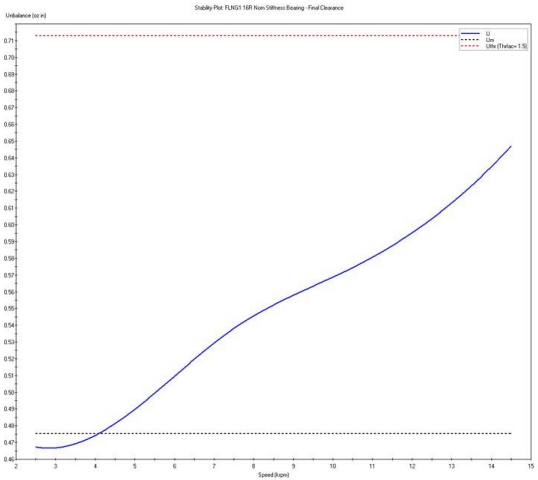
Original Nominal Stiffness



Original Max Stiffness



Final Nominal Stiffness



Case Study #4

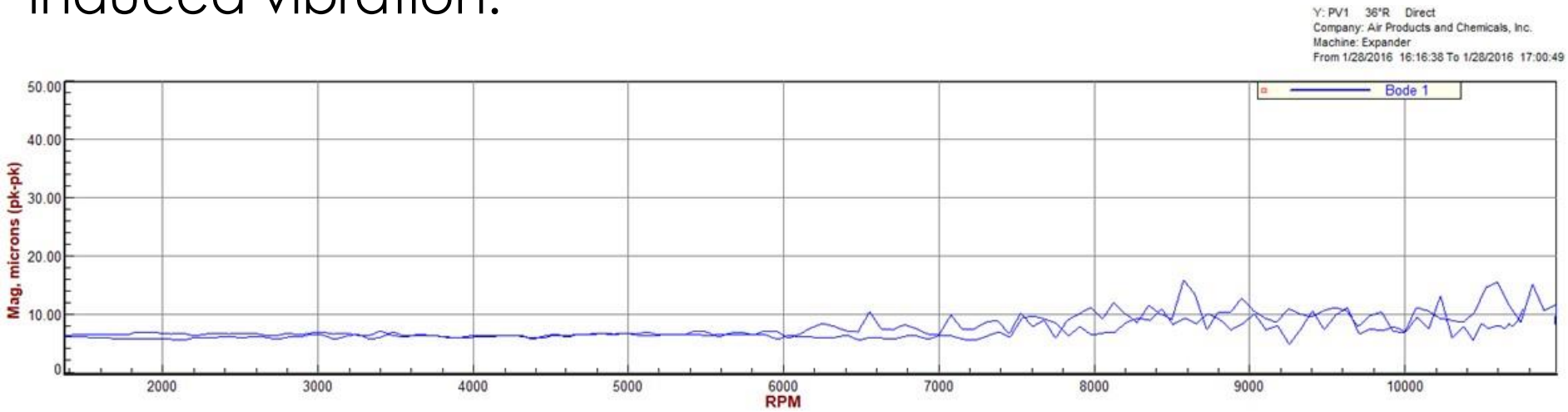
Given the new frame design, it was unclear if thermal stability would be maintained when operating in the field at full power and cryogenic temperatures. Therefore, a alternate bearing was designed with features that were predicted to reduce thermal instabilities. Spray bars were designed with five oil feed ports. Three ports channeled cool oil onto each pad, while two additional ports on the side of each pad sprayed cool oil directly onto the shaft to limit thermal expansion.

Case Study #4

The top two pads were machined with stepped pockets for increased pressure profiles; the bottom three pads were machined without pockets. The pad L/D ratio was reduced from 0.5 to 0.4. Bearing models predicted these design features would increase eccentricity and reduce the differential journal temperatures.
Ultimately this bearing design was not utilized as the units in the field have operated successfully with the as-tested bearing design.

Case Study #4

The units have operated in the field under typical operating conditions with the original bearing design with increased clearances and have shown no signs of Morton effect induced vibration.



Conclusion

The Morton Effect can manifest itself in numerous ways in various types of turbomachinery. Several examples with various observed characteristics have been presented as have several methods for improving stability.

Software does exist to attempt to predict the onset/severity of the Morton Effect. It has been shown that the trends of at least one of the programs (VT-map) has aligned well with actual experiences but was unable to provide accurate enough predictions of actual onset to be used as an upfront design tool with hard acceptance criteria.