46TH TURBOMACHINERY & 33RD PUMP SYMPOSIA
HOUSTON, TEXAS | DECEMBER 11-14, 2017
GEORGE R. BROWN CONVENTION CENTER

Prediction and mitigation of high radial vibrations during hot restart of centrifugal compressors

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imagination at work

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Michele Fontana is Engineering Manager for Centrifugal Compressor Upstream and Pipeline applications at GE Oil & Gas. He is responsible for centrifugal compressor design activities, with main focus on rotordynamics and thermodynamics. Mr. Fontana received a B.S. degree in Mechanical Engineering from University of Genova in 2001. He joined GE in 2004.

Gianluca Conte is a Lead mechanical design engineer with GE Oil&Gas. His responsibilities include the development of centrifugal compressor design, support to test and manufacturing and RCA for field issues. Gianluca started to work in Oil&Gas as Field test and diagnostic engineer. He also contributed to develop new diagnostic system for rotating machinery (TVMS, Blade health Monitoring, etc.).

Mirko Libraschi works as Rotordynamics Senior Engineer for GE Oil&Gas. He works in the New Product Introduction department since 2010, specially involved in RCAs and several research activities on bearing. Mirko Libraschi received his Mechanical Engineering degree in 2002 at the University of Pisa and before becoming GE employee he worked as Member of Engineers register..

Abstract

Rotor thermal bowing is a common cause of high radial vibrations for turbomachinery. It may occur during hot restart, i.e. when a rotor is restarted before reaching thermal equilibrium.

The case of a centrifugal compressor showing high vibrations due to thermal bowing is presented. The train is composed by a natural gas centrifugal compressor driven by a fixed speed electric motor, through a variable speed fluid coupling.

The solution selected for the case study (replacement of the original journal bearings with new ones, featuring integrated squeeze-film dampers) is discussed in detail together with other possible alternatives. A comparison of vibration data before/after the intervention is presented to illustrate the resolution of this site issue.

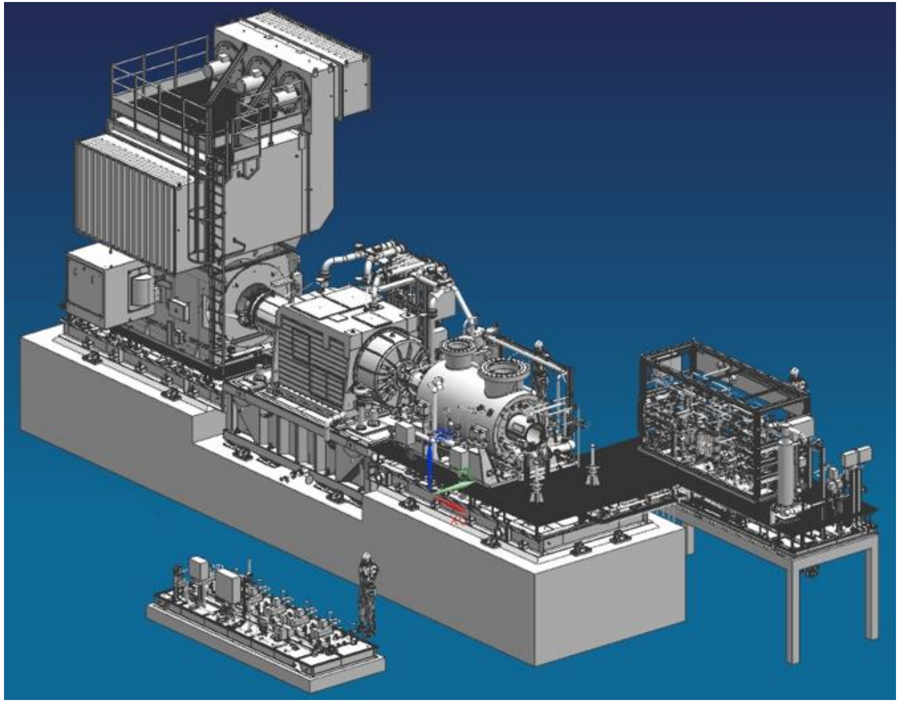
TPS

Summary

1. Problem Statement
2. Physical Model
3. Data analysis and Root Cause Identification
4. Solution applied for the case study and possible alternatives
5. Conclusions

1. Problem Statement

Object: Three compressor trains installed in MENAT region (onshore). Each compression train is composed by a fixed-speed electric motor, a variable-speed fluid coupling and a single phase, in-line centrifugal compressor (BCL605 model). The motor is not provided with a soft-start system; during startup it reaches the minimum operating speed in ~10 s.



Issue description: During several hot restart attempts, the centrifugal compressor tripped for high vibration on NDE side radial probes. The vibration level showed some correlation with parameters such as Settling out pressure (SOP) and standstill duration.

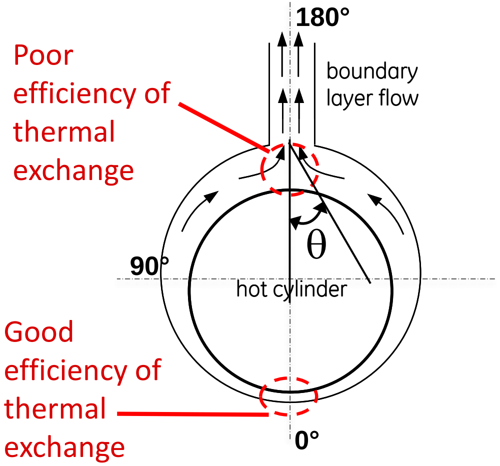
In order to investigate the issue and define the root cause analysis a survey in site has been carried out including a set of tests.

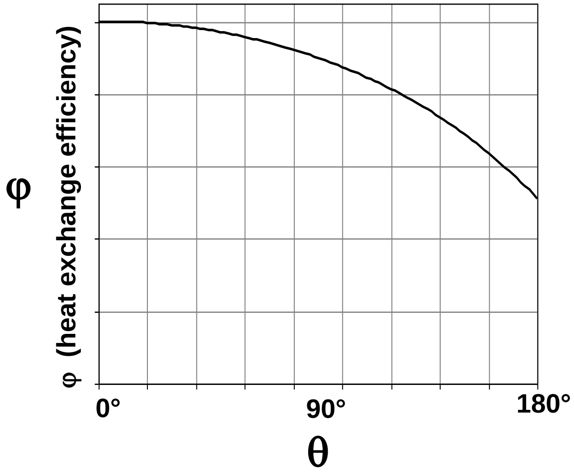
2. Physical Model of the Phenomenon

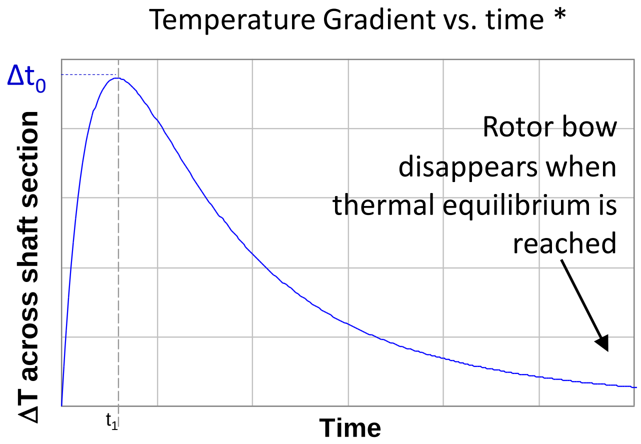
Rotor thermal bowing is due to asymmetrical cooling in standstill conditions, after compressor shutdown.

Natural convection of gas on a horizontal cylinder: The heat transfer coefficient *h* is function of the angular position around the cylinder's axis.

At the beginning of the cooling transient the gradient across the shaft section is zero (uniform T), then it increases up to a maximum and later it slowly decreases up to thermal equilibrium.



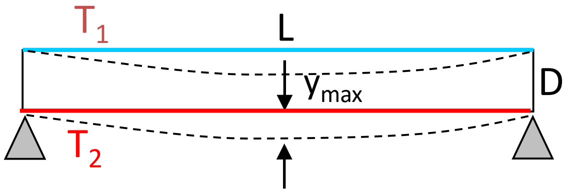




* the trend is qualitatively similar for any real case, while the values of max gradient ΔT_0 and time constant t_0 depend on operating parameters and rotor geometry.

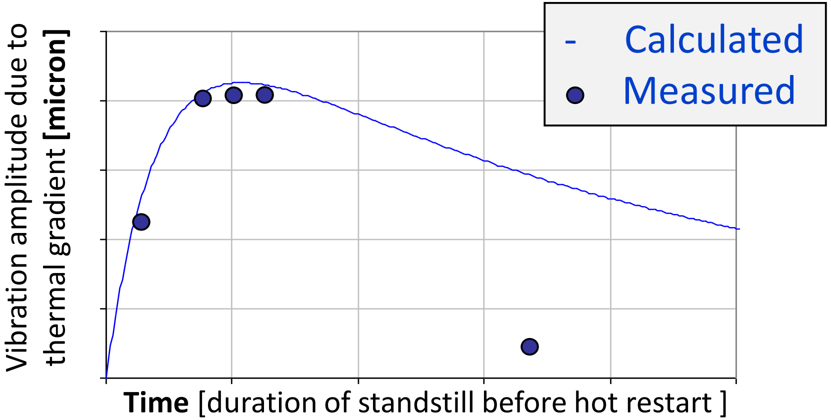
2. Physical Model of the Phenomenon

The thermal gradient across the shaft causes the rotor to bow, due to differential thermal deformation. For an uniform beam on supports, loaded with uniform thermal gradient, the maximum deflection is $y_{max} = (\alpha/8D)(T_2\text{-}T_1)L^2$



This deflection acts as an exciting force when the rotor is started up. This may lead to high radial vibrations at 1st critical speed crossing.

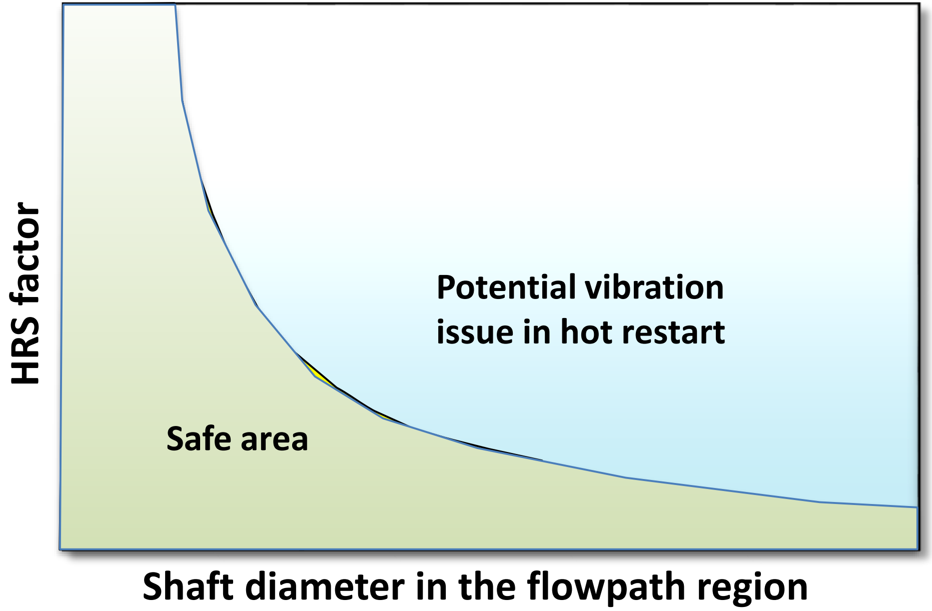
A calculation model based on fluid-dynamic theory was developed to simulate the evolution of the thermal gradient and its effect on vibrations. Its predictability was confirmed by comparison with experimental results (see diagrams).



2. Physical Model of the Phenomenon

$$HRS = N\left[L(T_{disch} - T_{suct})\right]^2 \sqrt{(p_{disch} + p_{suct})}$$

L = Bearing Span
N = Max Cont. Speed
T = temperature
p = pressure



The formulation of HRS factor was derived from the full analytical model of hot restart phenomenon.

NOTE: The above formula and diagram do not include the start up speed rate.

Shaft rotation is very effective in eliminating rotor bow. If the startup ramp is slow, more rotation cycles occur before 1st critical speed crossing.

The slower is the startup rate, the lower is the risk of high vibrations due to hot restart.

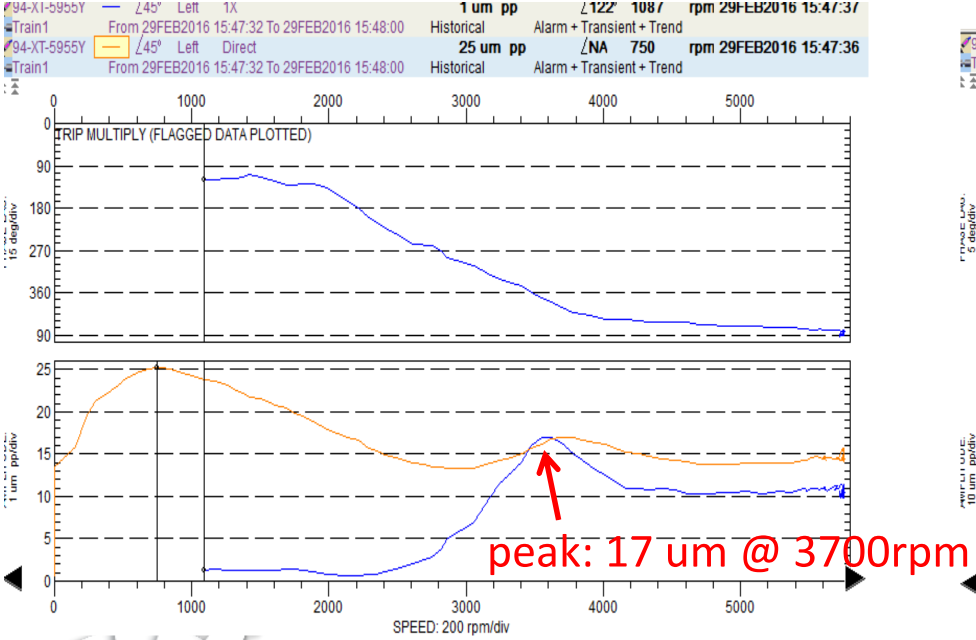
3. Case Study Description

High vibrations during hot restart occurred at site on a natural gas centrifugal compressor, whose main parameters are summarized in this table.

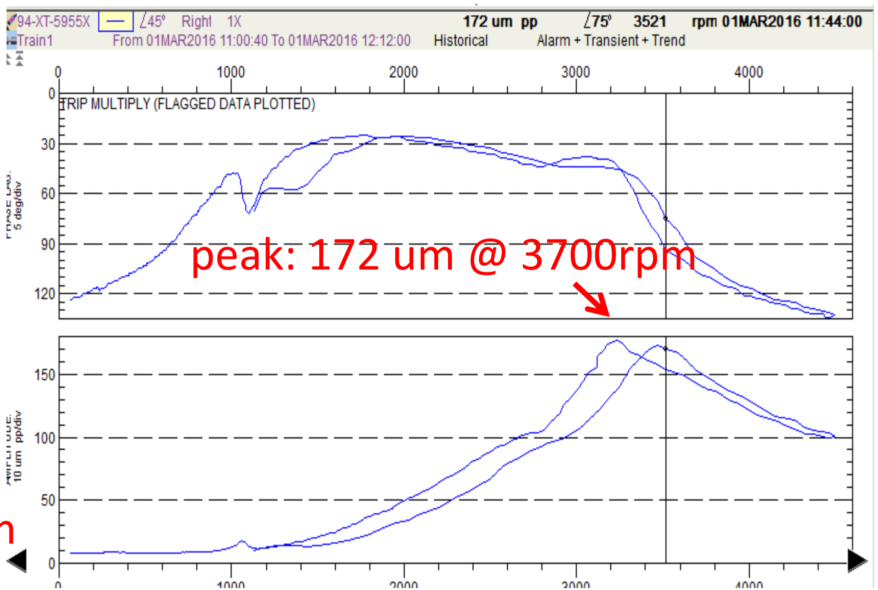
		Unit A
Casing Type	[-]	Barrel
Number of Stages	[-]	5
Impeller Max External Diameter	[mm]	655
Rotor Weight	[kg]	1543
Bearing Span	[mm]	2358
NC1	[rpm]	3476
Rated Speed	[rpm]	7470
Rated Suction Pressure	[bar-A]	8
Rated Discharge Pressure	[bar-A]	21
Rated Mass Flow	[kg/sec]	55.96
Gas Molecular Weight	[kg/mol]	19.09
Journal Bearings Size	[mm]	130
Journal Bearing Clearance (normalized)	[-]	1.41‰
Journal Bearings Type		Flooded, Tilting Pad

3. Data Analysis - CC Vibrations in Cold vs. Hot Startup

Cold startup (max vib 25µm -1X component 17µm)



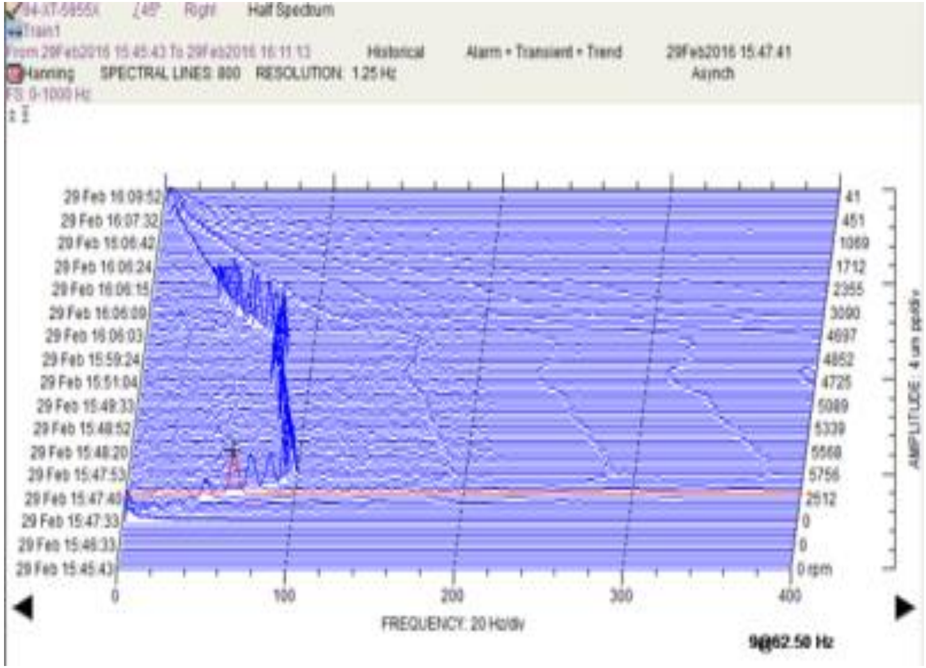
Hot restart (max vib 210µm – 1X component 172 µm)



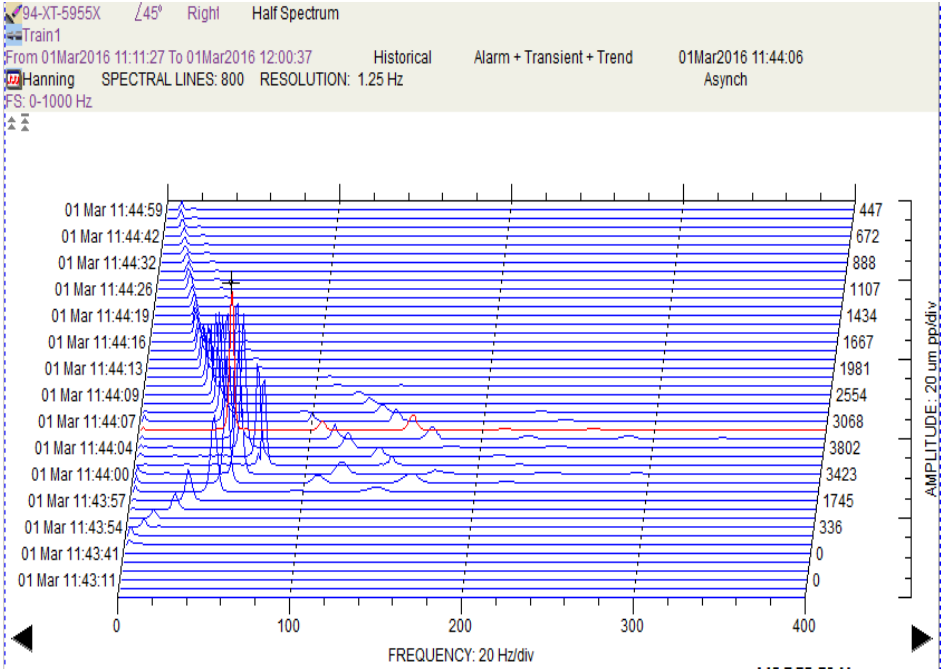
During hot restart the peak amplitude was up to 10X the cold startup value.

3. Data Analysis - CC Vibrations in Cold vs. Hot Startup

Cold startup (MAIN COMPONENT 1X)



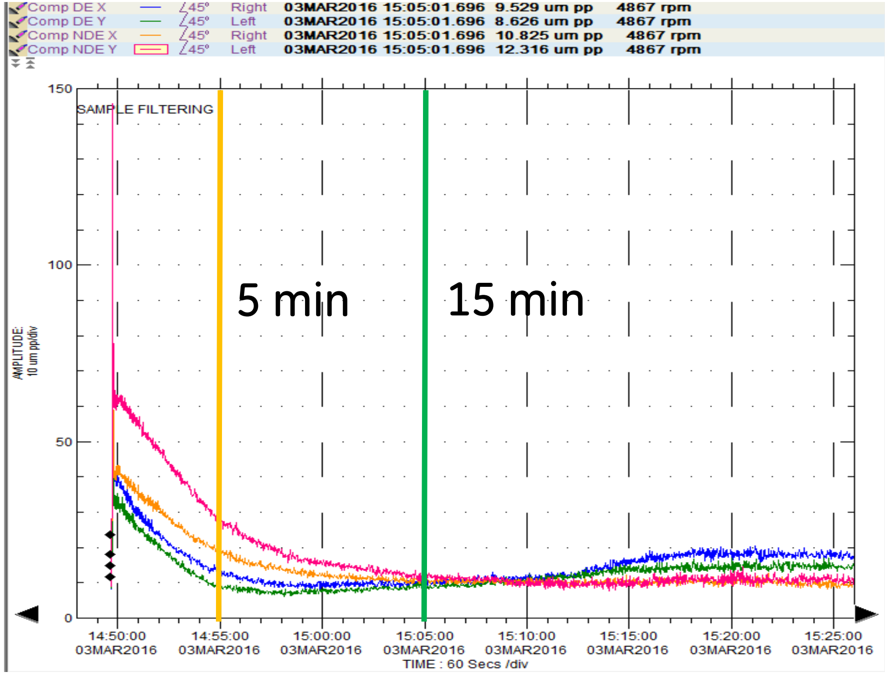
Hot restart (MAIN COMPONENT 1X)



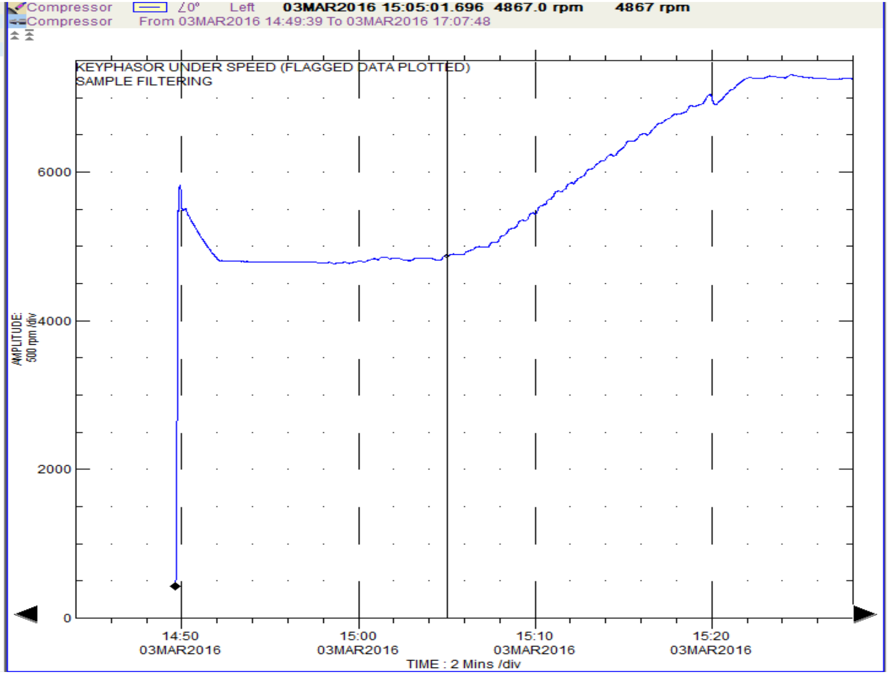
Waterfall diagrams show that the vibration is almost completely synchronous. This is a typical feature of vibrations due to rotor bow.

3. Data Analysis - CC Vibrations in Cold vs. Hot Startup

CC vibrations



Speed trend



After 5 mins running: vibration amplitude is reduced by ~50%.

After 15 mins running: vibration amplitude is stabilized

4. Overview of Possible Mitigations / Solutions

A set of potential corrective actions was identified:

Solution	Description	Type	Notes
''Multiple start'' automatic sequence implementation	Implement a startup sequence inside EM Software/MCC to perform a short startup cycle (3s), reaching a speed lower than 1st critical speed (approx. 1500 RPM), in order to reduce rotor bow, and then the normal startup.	Short term	Tested but not accepted by customer
Seal gas temperature reduction	After the updated worst gas composition a condensation analysis has been performed and seal gas temperature has been reduced down to 40°C.	Long term	Mitigation action
Slow roll by turning gear/barring gear	Slow roll before machine startup and/or after machine stop prevent the thermal bowing phenomenon occurrence	Long term	Standard solution
Journal bearings design	Replacement of standard-type journal bearings with journal bearings equipped with squeeze film dampers	Long term	Selected solution
Compressor flushing	Flushing of compressor after stop/before startup helps to uniform temperature along the rotor	Long term	Not tested
Trip Multiplier 1.25X*	Trip multiplier 1.25x (Delay 1s) has been set inside Vibration control panel in order to help the startup the compressors and be able to analyze the issue.	Short term	Mitigation action

* Setting a Trip Multiplier during startup is not an actual solution but allows to carry out a hot restart avoiding trip for high vibrations. A clearance check shall be performed, to properly define a multiplying factor that prevents rotor-stator rubbing at seal locations.

4.1. Electric motor multiple start sequence

- The compression train is started for 3 sec and coastdown time is 3min.
- This short run is sufficient to significantly reduce rotor bow, straightening the shaft.
- Immediately after this start-stop cycle, a normal startup of the compressor is carried out.
- Tests confirmed the effectiveness of this startup sequence in solving the issue.

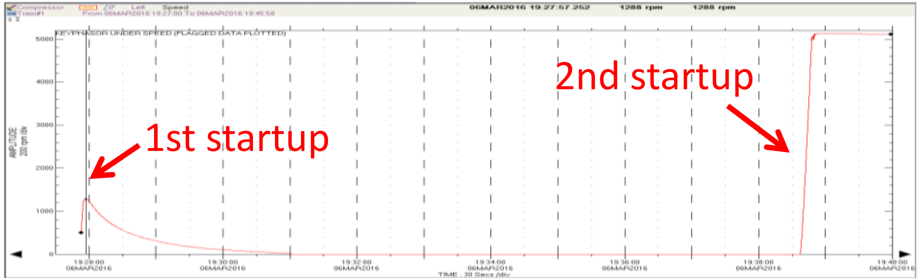
Pro's

- No additional cost, only software modification
- No additional components to install
- No impact on electric motor life

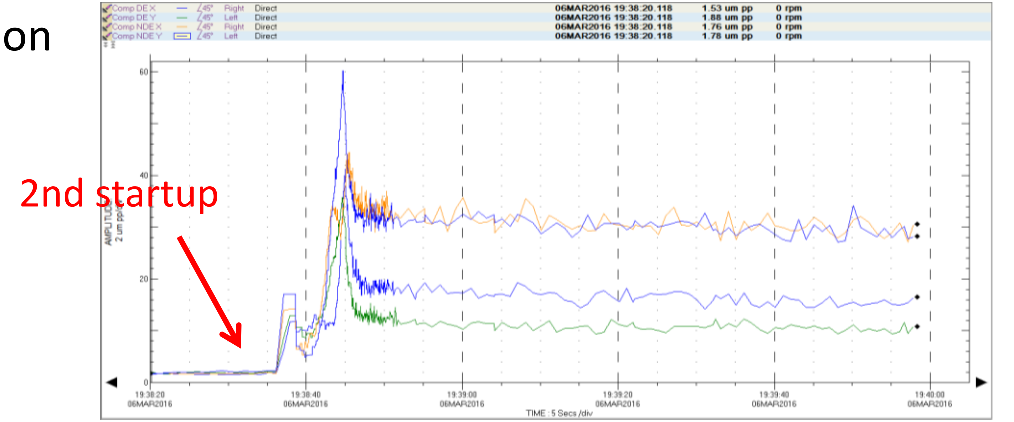
Con's

- Additional time at startup (few minutes)

Multiple start in 3 sec – Speed trend

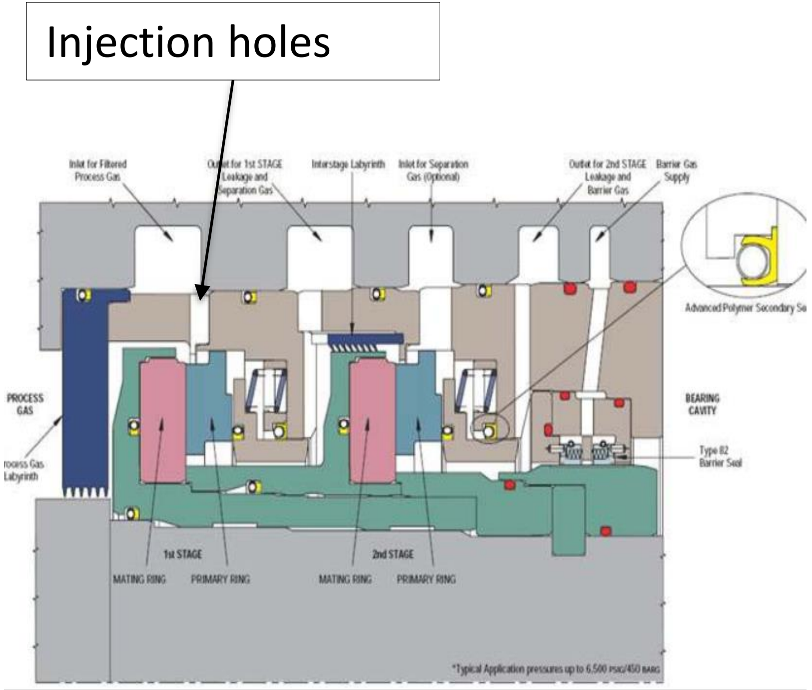


CC JB vibration trends in hot conditions with multiple startup



4.2. Seal Gas Temperature Reduction

Seal gas temperature can be reduced and optimized on the basis of seal gas dew point temperature and reducing diffused and localized thermal asymmetry in the rotor. This can be achieved by an annular chamber for seal gas and a multipoint injection, to make the thermal distribution as uniform as possible.

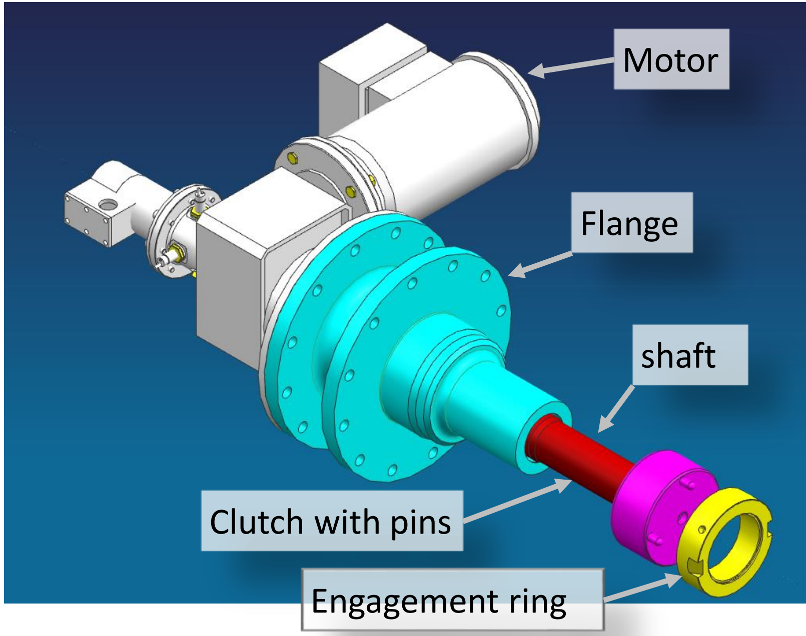


- **Pro's**
- Easy to implement
- **Con's**
- It is mitigation action but cannot solve completely hot restart issue

4.3. Slow roll by Turning gear (continuous or intermittence)

- Train is maintained in slow roll between stop and startup in order to avoid rotor bowing

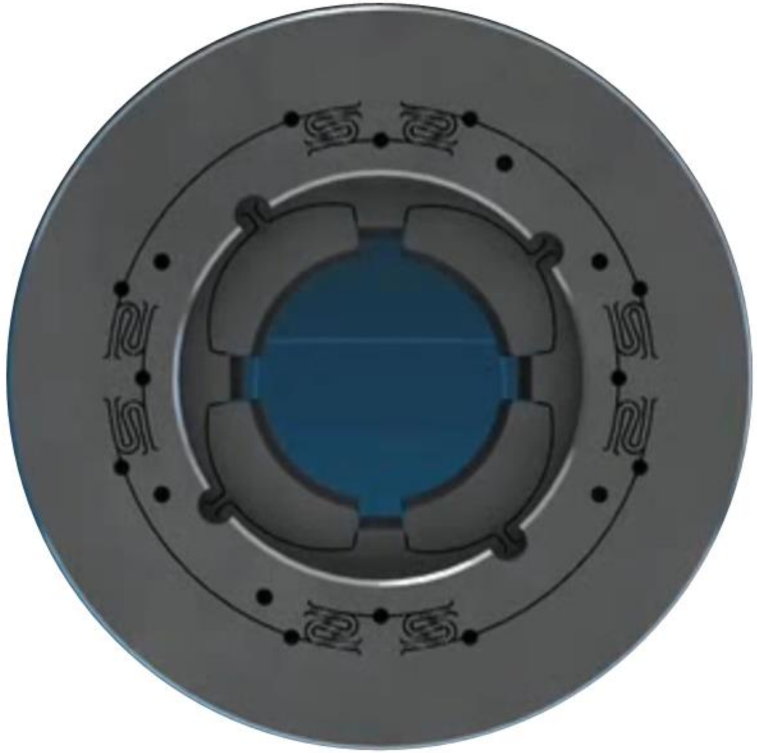
- **Pro's**
- Well known and proven technology
- Preventive solution (rotor bow is not just eliminated, but prevented)
- **Con's**
- Additional component to maintain
- Additional cost (main item, spare parts)
- Potential impact on rotordynamics (longer overhung)
- Additional component to size during design phase (Motor, clutch, break pins)



4.4. Journal Bearing Design

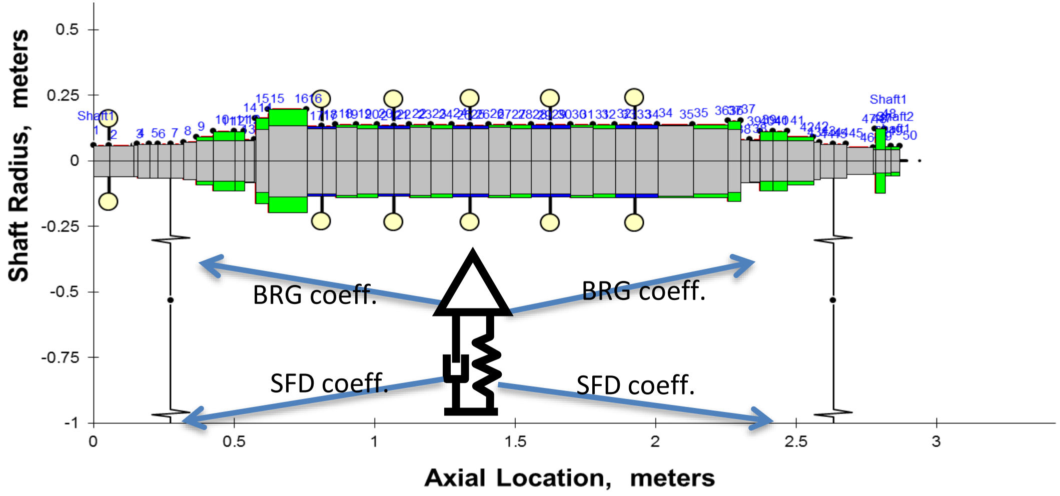
Standard tilting-pad journal bearings can be replaced with bearings featuring integrated squeeze film dampers, in order to maximize the damping coefficients and to decrease the amplification factor of 1st critical speed peak.

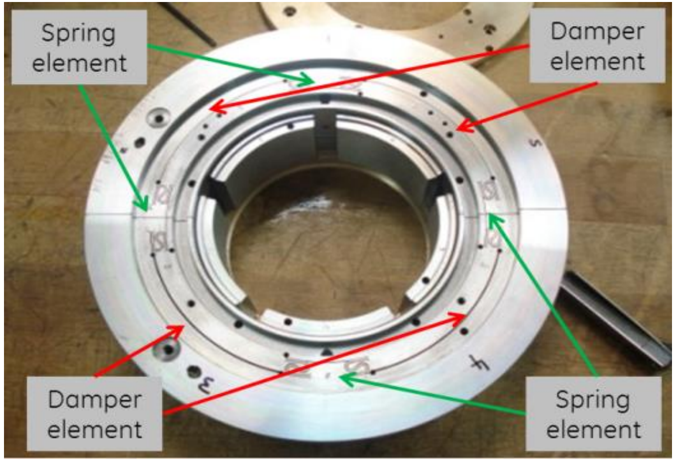
- Pro's
- No modification required to auxiliary systems, control logic, procedures…
- Referenced technology
- Con's
- Additional cost of SFD-equipped bearings
- Rotor bow is not reduced or eliminated; only the corresponding vibration peak is reduced.



Selected as solution for this case study

4.4. SFD Journal Bearing Design





SFD are described by stiffness and damping coefficients acting in series with the standard bearing coefficients.
The SFD geometry shall be tuned to obtain the optimum stiffness and damping for peak vibration reduction, as predicted by rotordynamic calculations.

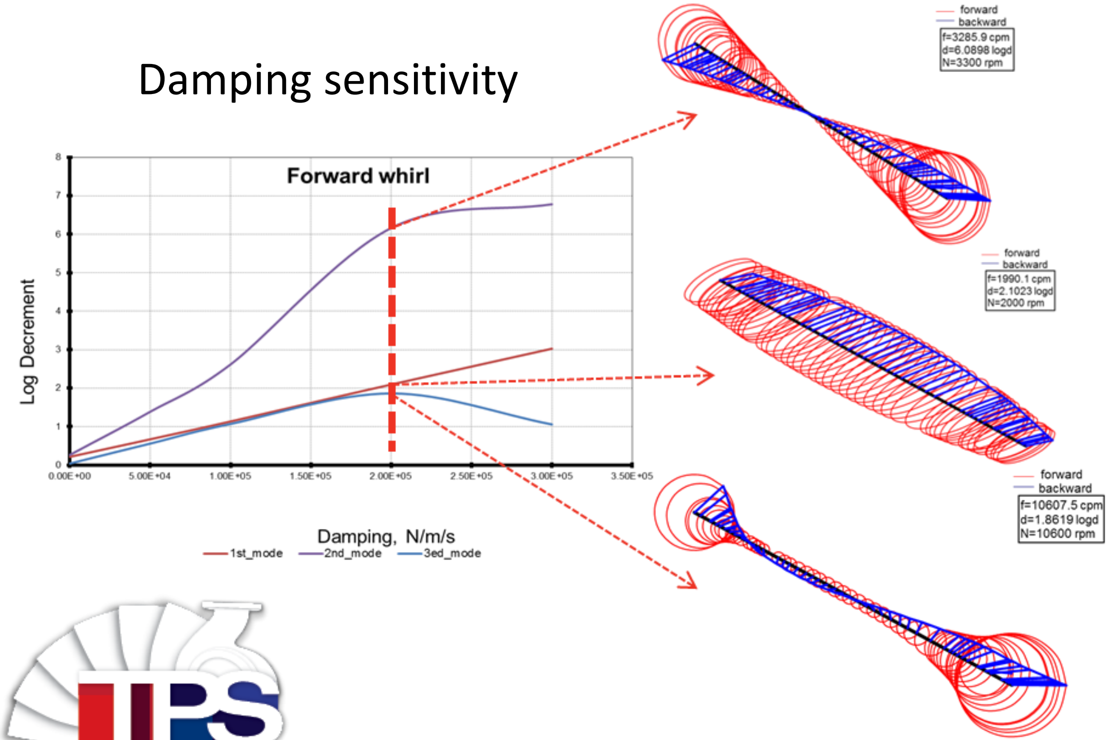
MCS = 7844 rpm
Shaft diameter = 130 mm
Bearing L/D ratio = 0.44
Load = **833** kg (DE side) / **688** kg (NDE side)
Min Clearance = 1.4 ‰
Max clearance = 1.8 ‰
Avg clearance = 1.6 ‰
oil type= ISOVG46
Inlet Temp=50°C
C_{sfd}=0.355 mm

4.4. SFD Journal Bearing Design - Sensitivity Analysis

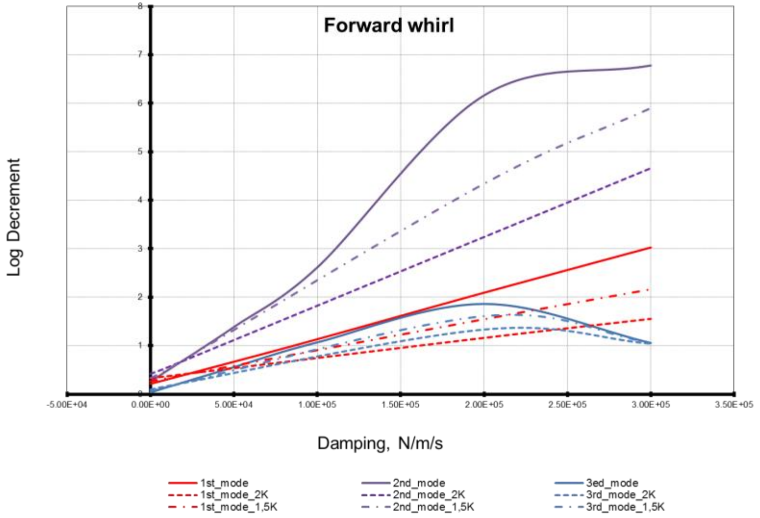
SFD design parameter optimization (stiffness & damping) to maximize Rotor+BRG+SFD system modal damping and mode shape. On the right the consistency's verification of the selected SFD stiffness.

SFD sensitivity analysis

Damping sensitivity

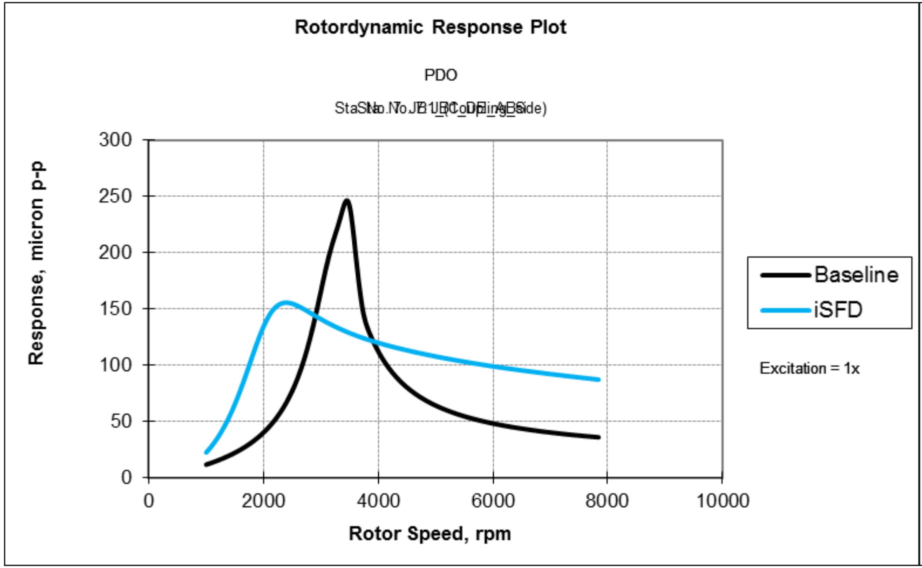


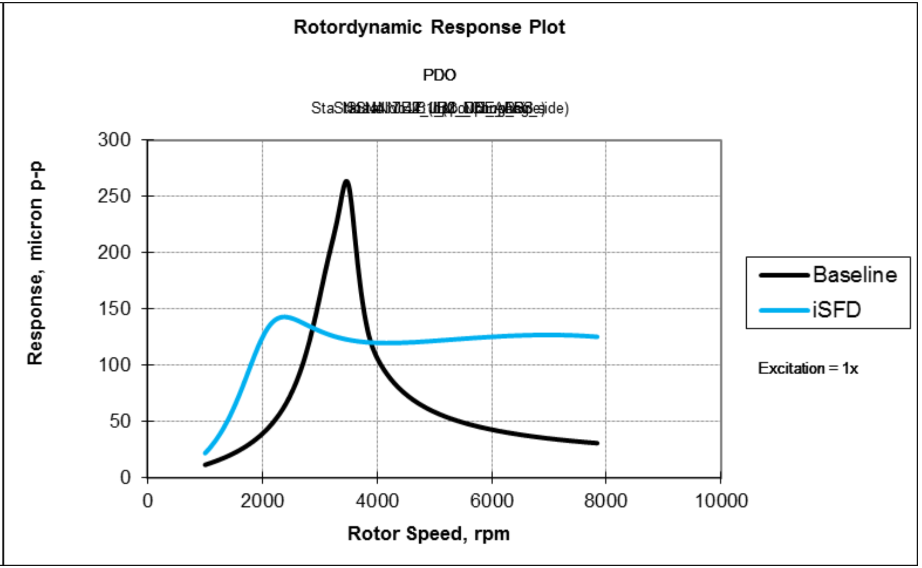
Stiffness sensitivity



4.4. SFD Journal Bearing Design - Predicted Vibrations

Absolute vibration comparison





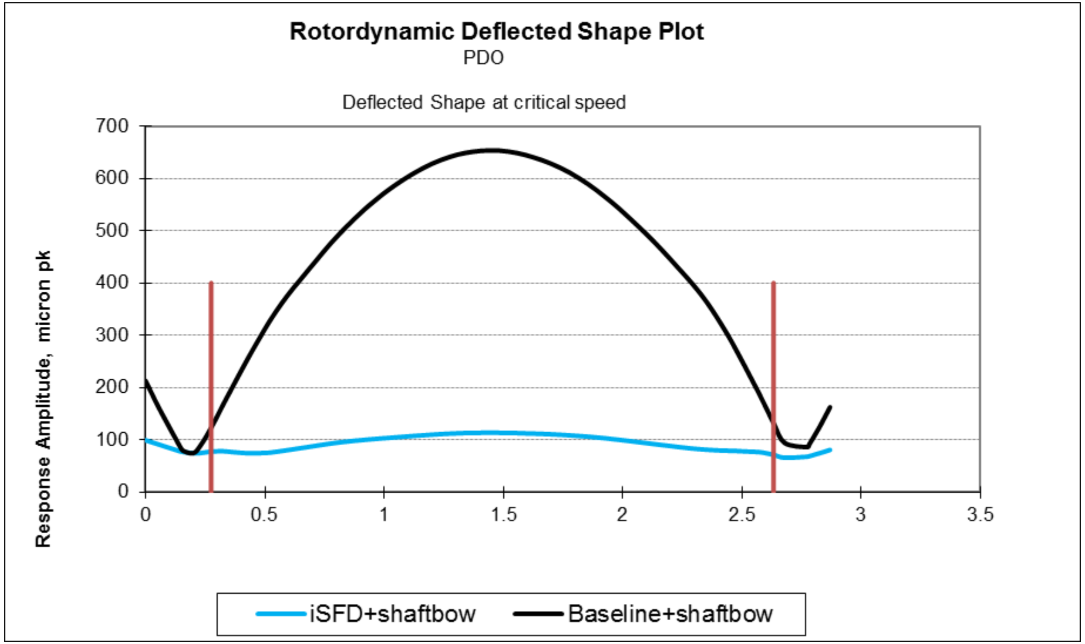
Black line: baseline rotor system's vibration response due to unbalance and shaft bow forces

Blue line: SFD-equipped rotor system's vibration response due to unbalance and shaft bow forces

This comparison allows to quantify the peak resonance decrease around 65%

4.4. Journal Bearing Design

Deflected shape comparison

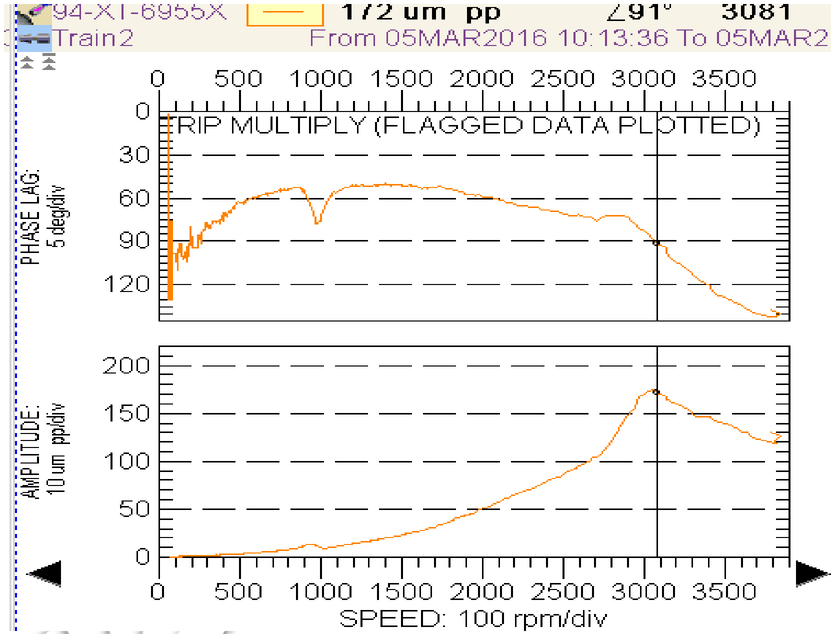


The mode shape comparison highlights the decrease of peak vibration inside the compressor bearing span (labyrinth seal rubbing safety)

4. Issue Resolution - Experimental Data

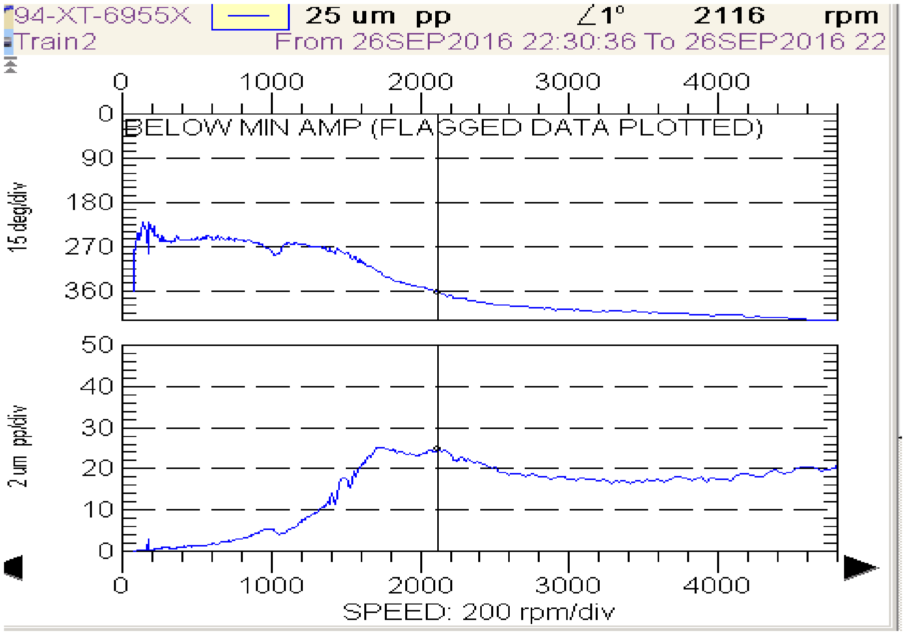
Vibration Before Upgrade

High 1X Amplitude response



Vibration After Upgrade - SFD Bearings

Highly damped vibration, 1X Amplitude response around 1st critical – 1900 to 2400 rpm as predicted in rotor dynamic response



The vibration peak @ 1st critical speed crossing was reduced by >80%

5. Conclusions

- A high vibration issue during centrifugal compressor hot restart was effectively solved by replacing traditional journal bearings with SFD-equipped bearings. The use of squeeze film dampers allowed to reduce the radial vibration peak by about 80%, well below the alarm value over the whole startup transient and the normal operating range.

- The phenomenon of high vibrations due to hot restart was discussed starting from its physical model, and a set of potential solutions was identified and analyzed in detail: multiple startup sequence, slow roll by turning gear, SFD introduction, seal gas injection optimization. The best solution shall be identified case by case according to design boundaries and to cost evaluation.