FLOATING SEAL RING ACTING AS A THIRD BEARING

Case History

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Background

- The synthesis Gas Compressor was recently overhauled.
- Replaced the IP Compressor rotor, interstage seals and floating seal rings.
- Replaced the Inboard radial & thrust bearings of IP Compressor.
- When the machine started, high vibration was experienced on the Inboard bearing while Outboard bearing indicated low vibration levels.
Machine Train Diagram

Balance resonance speed 5700 rpm

Comp, IP Case  Steam Turbine  Comp, LP Case  Comp, HP/Recycle

High vibration
Data collected during steady state
Direct Orbit/Timebase Plot

ROTATION: Y TO X ICW
8361 nm

Y Comp/J/B V-201
X Comp/J/B V-202

MACHINE: P Compressor
31 AUG 2001 06:30:47.2 Steady State DIRECT
Full Spectrum Plot

POINT: Comp V/B V-201 45° Left  DIR AMPL: 2.55 mil pp
POINT: Comp V/B V-202 45° Right  DIR AMPL: 2.50 mil pp
MACHINE: IP Compressor  MACHINE SPEED: 9361 rpm
01 AUG 2001 06:30:47.5 Steady State
WINDOW: None  SPECTRAL LINES: 400  RESOLUTION: 0.032X
Data collected during shutdown
Shutdown Polar Plot

POINT: Comp VB-V-281 \(45^\circ\) Left 1X COMP SR: 0.437/131\(^\circ\) 2.72/280\(^\circ\) @9361 rpm
MACHINE: IP Compressor
From 01AUG2001 06:30:47.5 To 01AUG2001 08:00:03.6 Shutdown

What is the problem?
Shutdown Bode Plot

No phase change
Symptoms of the Problem
Symptoms:

- Vibration amplitude did not peak at the designed balance resonance speed of 5700 rpm
- The amplitude kept changing as the square of the speed changed from 5000 rpm to 9200 rpm.
Symptoms:

- The phase angle also did not change while passing the designed balance resonance speed.
All these symptoms prove that the balance resonance speed has increased to beyond operating speed.
Analysis

- In general, the rotor’s balance resonance speed is a function of the rotor mass and spring stiffness.
- It will remain unchanged unless the rotor mass or spring stiffness changes. A simple formula that describes this relationship is:

$$\omega_{res} = \sqrt{\frac{K}{M}}$$

Where:
- $\omega$ = rotor natural resonance frequency
- $K$ = system spring stiffness
- $M$ = rotor mass
The rotor mass probably did not change. It is now obvious that the system spring stiffness had increased.

What could have increased the spring stiffness?
Severe Misalignment!

- The coupling is flexible type
- The orbit shape doesn’t suggest misalignment
Rub!

What can cause Rub?
Interstage seal rub!

- Shaft will bow
- Vibration will increase on I/B and O/B bearings
- Machine can trip in few minutes
- Bearing rub!

- Tilt Pad Bearing
- Normal bearing temperatures
- Bearing clearance is usually bigger than the seal ring clearances. So, it will rub in the seal area before a bearing rub can occur.
- *Lubricated seal rub is suspected*
CONCLUSION:

Locked-up seal ring acting as an additional bearing
Verification of the Problem
Modal Analysis
POINT: 2YD  45° Right  1X UNCOMP  22.8/15°  10589 RPM

From 20Mar02 19:21:04 To 20Mar02 19:21:04 TRANSIENT

50.0 um pp  FULL SCALE  CW ROTATION
Corrective action:

- Increased the seal rings clearance, typically from 0.001 to 0.003 inch (OEM Recommendation)
- Adjusted the anti-rotation pins to allow maximum floating
Data collected during startup
Startup Polar Plot

POINT: IP Comp [R V-201] 45° Left 1X COMP SR: 1.13/168° 2.10/118° @ 8849 rpm
MACHINE: IP Compressor
From 06AUG2001 07:06:07.4 To 06AUG2001 11:27:04.2 Startup

1st Balance Resonance
Vibration still high

2 mil pp FULL SCALE CW ROTATION
Balance weight (8 grams)
Data acquired after balancing
Startup Polar Plot

POINT: IP Comp V-B 201 45° Left   TX COMP  SRF: 0.704/174°  0.910/59° @8820 rpm
MACHINE: IP Compressor
From 07AUG2001 04:40:46.3 To 07AUG2001 08:29:25.8 Startup

2 mil pp  FULL SCALE  CW ROTATION
CONCLUSIONS:

- The system stiffness increased, due to a locked-up seal ring acting as an additional bearing
- Transient data helped determine the root cause of the problem
- This problem could be misdiagnosed as an unbalance problem
THANKS,
ANY QUESTIONS?