COMPRESSOR ROTOR CRACK CASE

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ABSTRACT

The case study describes rotor crack detection using vibration measurements, documented on real case of Induction Motor driven Compressor Unit. The study highlights the complexity of rotor crack diagnostics as the primary problem as it can be often masked by other existing machine malfunctions.

In this case indication of the possible rotor crack, led to the Compressor bundle removal and rotor inspection. The rotor problem was confirmed, but not due to rotor crack but luckily only due to rotor thrust collar fit found to be worn and locating pin to be bent. Thrust collar had become eccentric (radial run out measured) causing a mass unbalance on the rotor in one direction, with similar symptoms as rotor crack.

The gradual radial vibration increase on the compressor NDE bearing was observed on an export gas Compressor and the increase continued for several weeks. The vibrations were purely synchronous (1X) with a constant unidirectional phase angle. Following transient data comparisons, Lube Oil supply temperature checks, compressor feet / bearing housing vibration measurements and a NDE bearing inspection (no findings) the BN team diagnosed a rotor issue and insisted the compressor rotor was inspected. BN remotely

witnessed the compressor inspection where it was discovered the thrust collar to rotor fit was worn and oversized, antirotating pin bended, causing the thrust collar to be eccentric resulting in an unbalance. This was proven on the high-speed balancing machine when the thrust collar eccentricity values could then be correlated with the rotor vibration response using an influence vector from installing a test weight at the thrust collar.
Following the initial diagnosis and continued monitoring by the BN SSA team the machine was able to be safely operated for 6 months allowing for a replacement compressor bundle to be prepared and installed minimizing the operational and business impact.

INTRODUCTION

The machine train (Figure 1) consists of 7,4MW Induction Motor (1790rpm), coupled via flexible disc coupling to the single stage Gearbox (gear ratio 7.76), with self-aligning tilting pad axial bearing on the Low Speed Shaft (LSS) and the shaft supported in the sleeve radial bearings. The High Speed Shaft (HSS) has an integrated quill shaft, supported in the sleeve radial bearings, and contains also fully-automatic SSS-overrunning clutch. The HSS is driving via high speed flexible disc coupling Centrifugal Compressor (6.5MW, 13890rpm), axially split (barrel type), equipped with tilting pad bearings. Whole machine train is very well equipped. Each of eight radial bearings have a pair of proximity probes, measuring shaft radial vibration and temperature sensor, measuring the bearing metal temperature. Gearbox has Accelerometer and Velomitor measuring the casing vibration too. Moreover, motor process parameters (as current, winding temperatures, cooling temperatures) as well as compressor process parameters are fully integrated in one management-diagnostic platform with a remote access, with 24/7 remote diagnostics support.

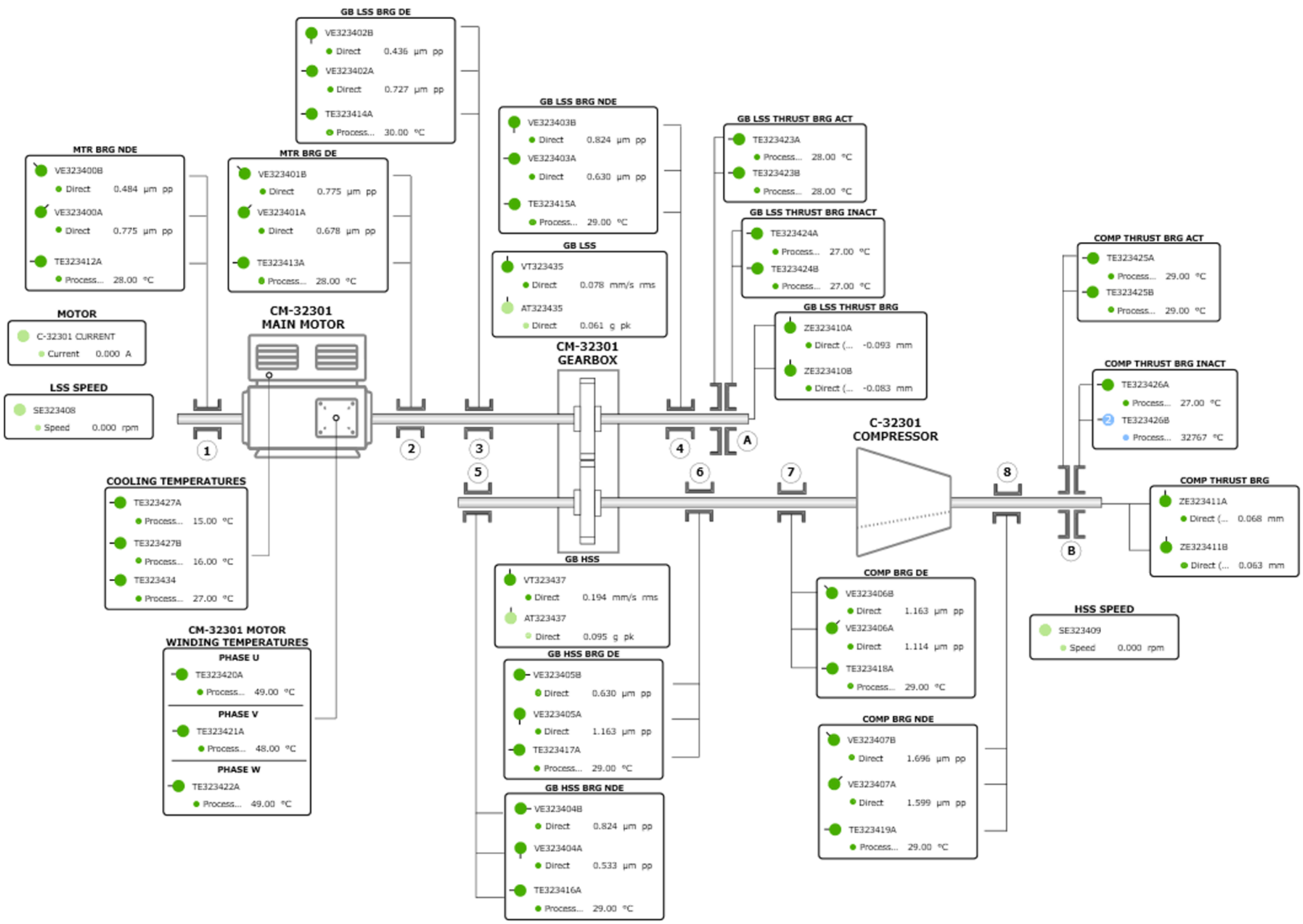

Figure 1: Machine Train Diagram

HISTORICAL BACKGROUND

The compressor was in operation without any problems for more than two years. In this period several starts and stops occurred.

On 8th August 2019 the plant was shut-down, this was not related to a compressor issue. A first restart on 9th August failed, there was a further shut-down after about one day. The second restart on 13th August 2019 succeeded. A gradual radial vibration increase on the NDE bearing was observed, the increase continued for several weeks. The vibrations were purely synchronous (1X), in one direction with constant phase.

On 7th October 2019 there was, shut down, restart, and shut down again for bearing inspection. Two loose bolts were found in the thrust bearing assembly. The bolts were tightened. On 21st October 2019 the machine was restarted on load and the 1X vibration continued to increase, unidirectionally.

On 8th November 2019 there was plant trip with controlled shutdown of the compressor. Restart on 9th November. On 21st November there was again plant trip with controlled shutdown of the compressor with the restart on 26th November.

On 27th December 2019 the compressor radial vibration started again to increase gradually on both bearings after a period of stable vibration levels.

On 17th January 2020, after a period of stable vibration level, the amplitudes began to increase at a faster rate. At the same time, the compressor NDE bearing temperature raised by 3-4 degrees and the lube oil (LO) filter D.P has increased sharply (~0.1bar). Further on, the compressor thrust bearing active pad temperature has increased. There is no significant axial position change correlating with the change in temperature. The faster rate of the vibration increases finally led to the decision to stop the unit and to remove the bundle for inspection.

ROTOR CRACK RULES, THEORY

As the vibration increase was predominantly 1X frequency component with unidirectional constant phase on both bearings, rotor problem was suspected from the beginning. The rotor crack was suspected as well as one of the possibilities and therefore additional rotor crack symptoms were investigated. Let’s summarize first the rotor crack symptoms/rules according the theory.

The shaft crack is a slowly growing fracture of the rotor. If undetected, it will grow until the remaining cross section of the rotor cannot withstand acting loads and the rotor fails in a brittle fracture mode. A rotating system holds a lot of stored energy, and when this energy releases within a short time, the result is usually catastrophic because of secondary damage due to liberated parts or rotor section components. The diagnostic methodology outlined here is based on [Bently, D., E., Hatch C., T., Grissom, B., 2002], unless stated otherwise, and it requires the long term monitoring of the machine, looking for changes in the 1X (component filtered to rotating speed frequency) and 2X (component filtered to twice rotating speed frequency) vibration vectors. Other machine malfunctions (unbalance, misalignment, loose rotating part, rub) also cause changes to the 1X and/or 2X vibration vectors. These other machine malfunctions must be reviewed and rejected to pursue the rules for shaft crack. [Mialkowski P., Péton N., Popálený P., 2020]

Rule #1: If rotor is cracked, it is almost certainly bowed.
The first rule is based on the fact, that when a cross section area of the shaft is reduced, the shaft bending stiffness lowers. So, under any acting unbalance force the stiffness change will cause a change in the peak to peak displacement, i.e. 1X vibration (or dynamic bow) of the rotor. A low value of the centrifugal force may not cause a detectable change of the 1X component due to the presence of a crack (or the crack is small). Depending on a crack position, location of the original unbalance and speed of the rotor, in relation to its resonance speed, the change in the 1X vector can be in virtually any direction. Thus, when a crack propagates both amplitude and phase changes can be expected.
Expected symptoms:

- Increase in rotor bow demonstrated by changes in slow roll vectors.
- Decrease in modal stiffness demonstrated by a shift in resonance.
- Decrease in effective damping demonstrated by higher Synchronous Amplification Factor (SAF)

Rule #2: Asymmetry in the rotor lateral stiffness, strong 2X
If a crack causes asymmetry in the rotor lateral stiffness, and if the rotor supports a radial static load, then strong 2X vibration will occur as 2X traverses a shaft critical speed (i.e when shaft speed equal to ½ of any resonance speed).
Expected symptoms:

- At half of any resonance speed 2X component is amplified

Rule #3: Thermal sensitivity of a cracked rotor

Additionally, to the long term 1X vector changes as described in Rule #1 the cracked rotors may show transient change of 1X vibration i.e. change in the dynamic bow when the temperature of the surrounding medium is changed fast enough. When temperature of the medium around the rotor is increased then, with some inertia, the rotor temperature starts to rise. But the temperature will change faster at the surface than at core of the rotor. In consequence the thermal expansion of the metal closer to surface is higher than at the core, so for some time the crack will “close”, and the stiffness of the rotor will be partially restored. When the temperature equalizes along the rotor section diameter, the crack “opens” again. Reverse process (temporary opening of the crack and reduction of the stiffness for time of thermal transient) can be expected when rotor cooled down.

Expected symptoms:

- During rotor heating, crack is closing in one direction
- After thermal equalization, crack is opening in the same direction
- Unidirectional rotor response change

Rule #4: Repeatability of abnormal behavior

The same repeatable symptoms for the same operating conditions and the repeatability of abnormal behavior is the one of the most important rule among all stated.

Expected symptoms:

- Abnormal unidirectional rotor response
- During rotor heating the crack is closing in one direction, then opening in the same direction
- Repeatability of abnormal unidirectional rotor response

Rule #5: New split resonance

The rotor transverse crack causes stiffness anisotropy in the rotating system. When the rotor support has also anisotropic stiffness, this may result in differences in the horizontal and vertical vibration responses of the rotor, especially noticeable at the resonance speeds. One peak for isotropic system response (no crack) will split into two adjacent peaks. The effect however is highly sensitive to the amount of damping in the system and may often be unnoticeable. Appearance of the split resonance in the rotor start up and shut down data and increase of their span may indicate a pending crack on the rotor [Muszynska, A., 2005].

Expected symptoms:

- If there is anisotropic stiffness in the rotor support and it’s paired with asymmetric stiffness in the rotor system, due to crack
- This may result in differences in the horizontal and vertical vibration responses of the rotor, especially noticeable at the resonance speeds.
- The one resonance peak will split into two adjacent peaks.

Rule #6: Rules #1-5 individually, do not indicate rotor crack

The crack as the main problem can be often masked by other existing machine malfunctions, such as unbalance, misalignment, bearing looseness or soft foot.

The case story root-cause analysis further shows the difficulty of detecting the rotor crack as primary problem as it can often be masked by other existing machine malfunctions and highlights the complexity of rotor crack diagnostics.

ROOT CAUSE ANALYSIS

The Compressor has seen a gradual increase in radial vibration on the NDE bearing no.8 following the restart on the 13th August 2019 (no maintenance was performed prior to restart). The radial vibration has been gradually trending upwards on the NDE and to a lesser extent on the DE. Gearbox HSS radial vibration and Compressor bearing temperatures remain stable. NDE radial vibration is approaching the alarm setpoint of 55um (trip 67um). The vibration increase is predominately at running frequency (1X) and is uni-directional (1X phase is constant). The orbit timebase remains circular in shape and increasing in amplitude. Increasing 1X radial vibration can be attributed to either a change in stiffness or a change in the balance condition, based on these three possible malfunctions were considered:

- Looseness – unlikely, non-isotropic change in stiffness resulting in non-circular orbit
- Fouling – change in balance condition, build up / removal of mass on the rotor
- Reduced shaft stiffness – damaged or cracked rotor / rotor assembly.

Initial Analysis on 13th August 2019

The second restart on 13th August 2019 succeeded. A gradual radial vibration increase on the NDE bearing was observed, the increase continued for several weeks. The vibrations were purely synchronous (1X) with constant phase, unidirectional.

The following observations have been made:

- Radial vibration gradually trended upwards on the Compressor NDE and to a lesser extent on the DE since the restart of the machine. The increase was predominantly 1X frequency component and no significant phase changes on both bearings (Figure 2).
- Compressor radial bearing temperatures remain stable
- Axial position and thrust bearing temperatures remain consistent
- Gearbox HSS NDE and DE bearing radial vibration remains stable and well below setpoints.
- Shaft Center Line (SCL) plots show no evidence in a change in external alignment (GB-Comp) or internal alignment (Comp)
- The compensated direct and 1X orbit timebase plots show and increasingly circular shape when compared to running data prior to the restart. (Figure 3)
- The 1X Polar plot for Compressor NDE bearing shows unusual unidirectional 1X vector change, with increasing amplitude but constant phase. This would suggest change in mass or stiffness in one direction, which is unusual and can be serious. (Figures 4)

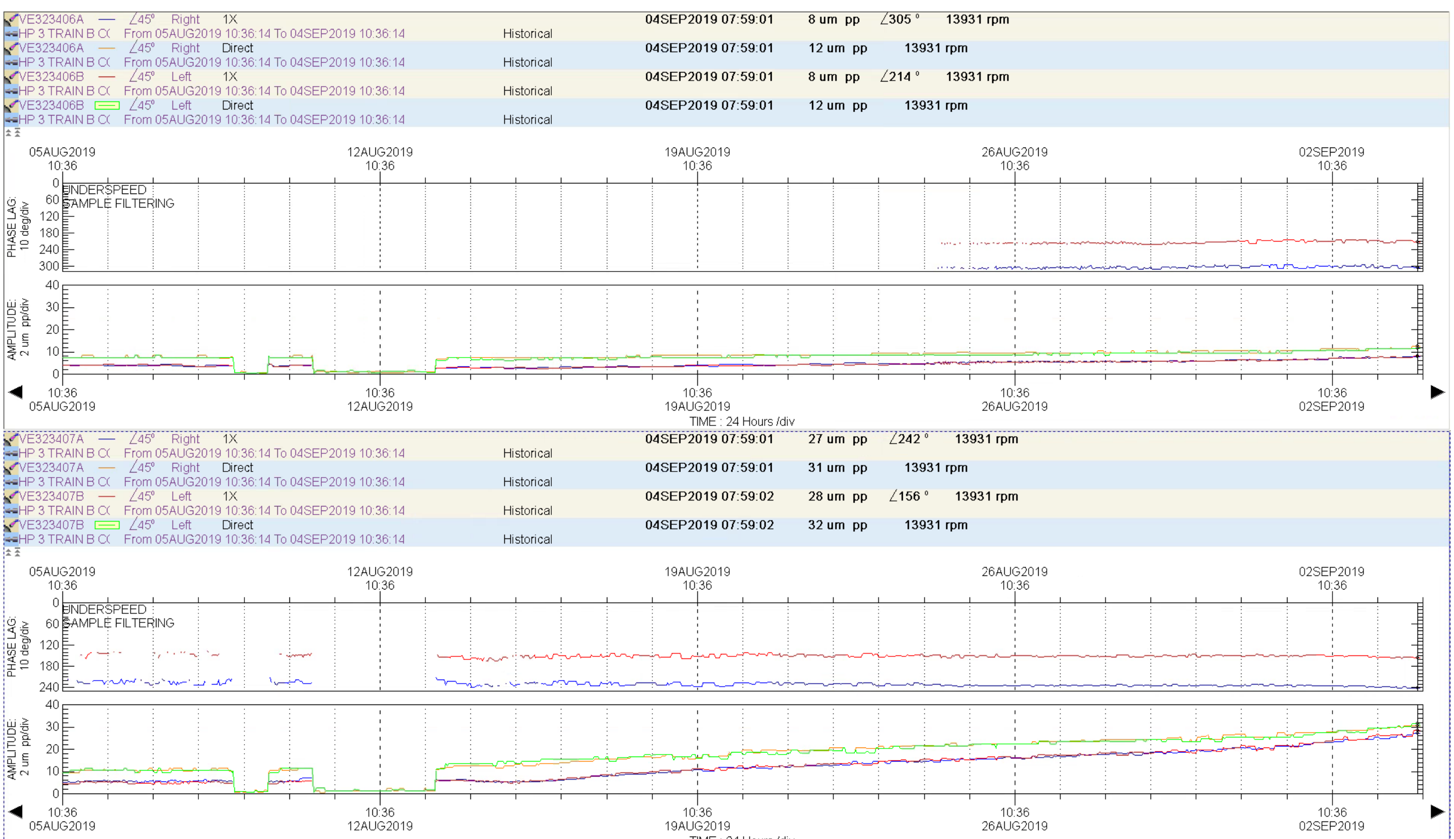

Figure 2: Compressor DE & NDE Radial Vibration Direct & 1X Trend

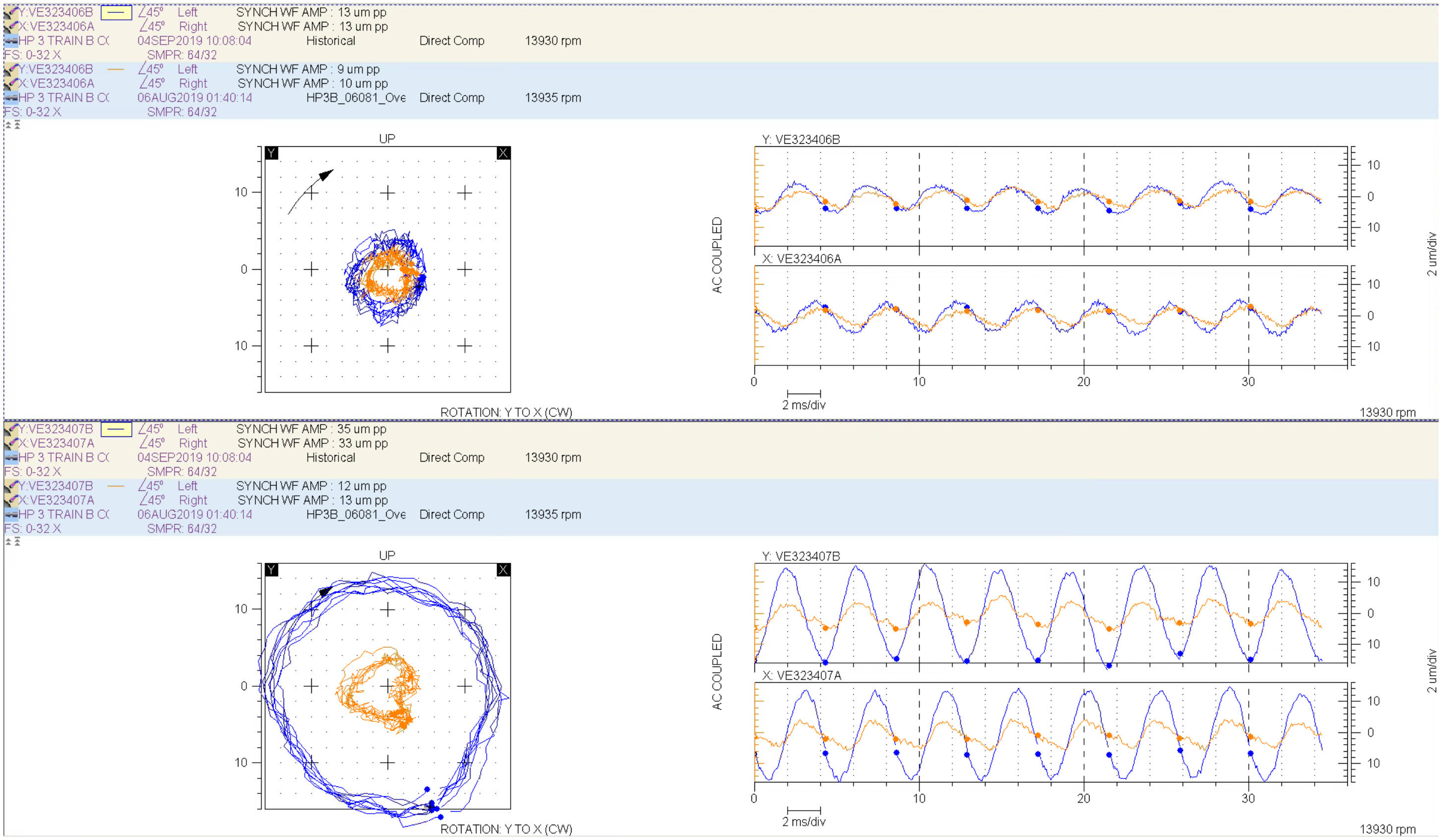

Figure 3: Compressor DE & NDE Compensated Direct Orbit Timebase plot comparison (Pre/ post re-start)

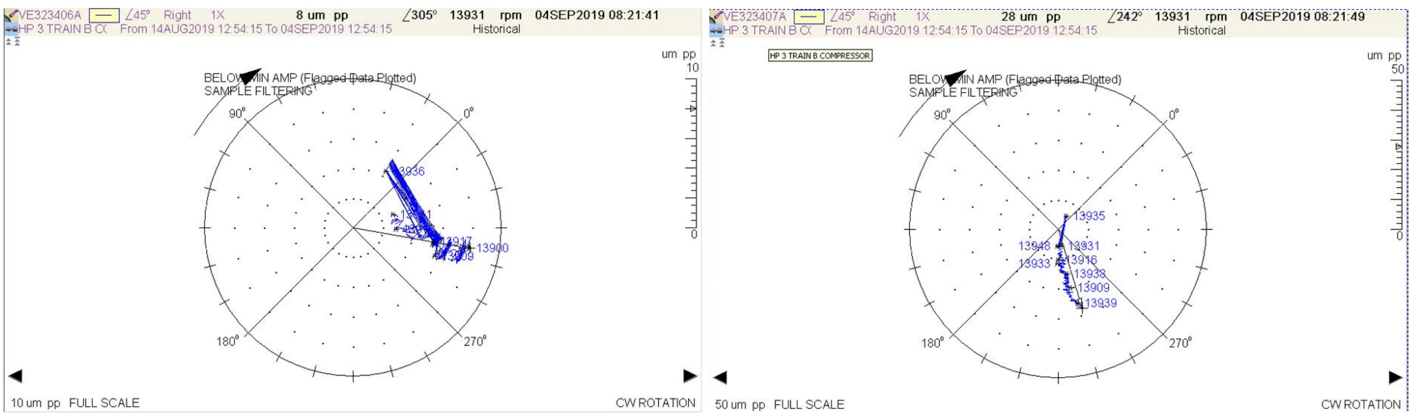

Figure 4: Compressor DE & NDE Polar plot (post re-start)

Conclusions:

- Increasing 1X radial vibration can be attributed to either a reduction in stiffness or a change in balance condition.
- There is no evidence of increased bearing temperatures or a change in alignment which would result in a change in stiffness. Additionally, any change in stiffness as a result of looseness would likely result in a non-isotropic change in stiffness and the orbit timebase would not be circular if this was the case.
- Based on there being little evidence of a change in stiffness the most likely cause of the increasing 1X vibration is a change in balance condition i.e. a reducing / increasing mass on the rotor. The gradual increasing nature indicates rather an increasing mass / deposit build up, than reduction in mass, which would most likely appear as single step changes in 1X.
- The unusual unidirectional 1X vector change, with increasing amplitude but constant phase needs to be closely observed as it may be because of reduced shaft stiffness.

Update Analysis on 30th September 2019

Following the initial exception report the compressor was running continuously at varied load conditions.

The following observations have been made:

- Compressor DE/NDE Radial Vibration has continued to trend upwards with the most significant increases visible on the NDE. The prominent frequency component remains 1X and there is very little corresponding phase change (Figure 5).
- There is no evidence of a direct correlation between the increased loading of the compressor and the vibration increase. The compressor vibration reacts abnormally to changing load conditions as it continues to increase at varied rates, with increased or decreased load. The load can be derived from the ASV position. Higher ASV value means higher load (Figure 6).
- Gearbox HSS radial vibration levels remain consistent.
- GB HSS and Compressor Radial Bearing Temperatures remain constant.
- Orbit timebase plots remain circular in shape and are increasing in amplitude.
- Compressor Axial vibration levels remain stable and low. The main frequency component is 1X with harmonics, there is no evidence of sub-synchronous behaviour which could indicate the presence of an instability.
- Compensated 1X polar plot plots show the continued unidirectional increase in 1X vibration on the Compressor NDE. This would suggest change in mass or stiffness in one direction (Figure 7).

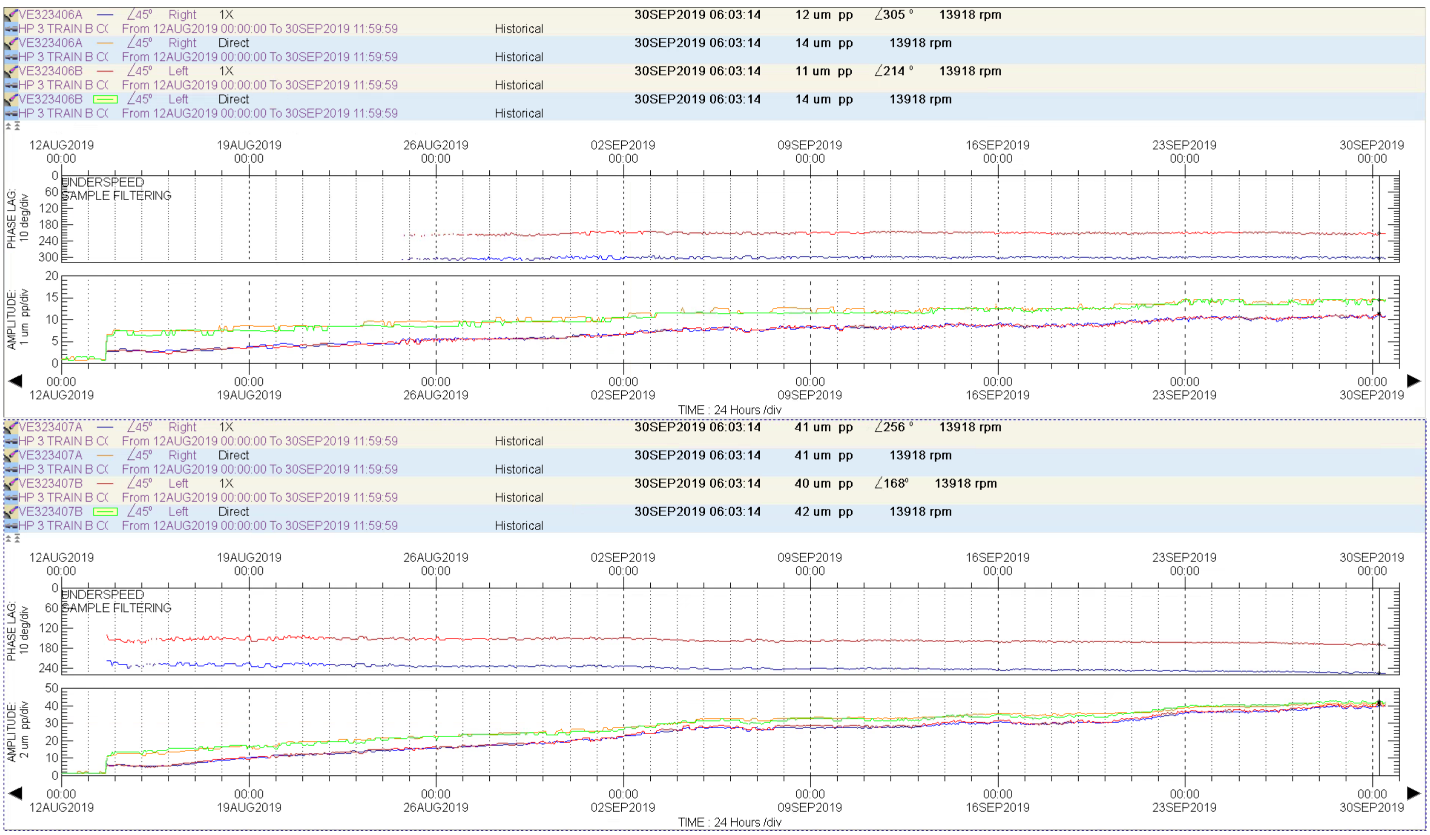

Figure 5: Compressor DE & NDE Radial Vibration Direct & 1X Trend

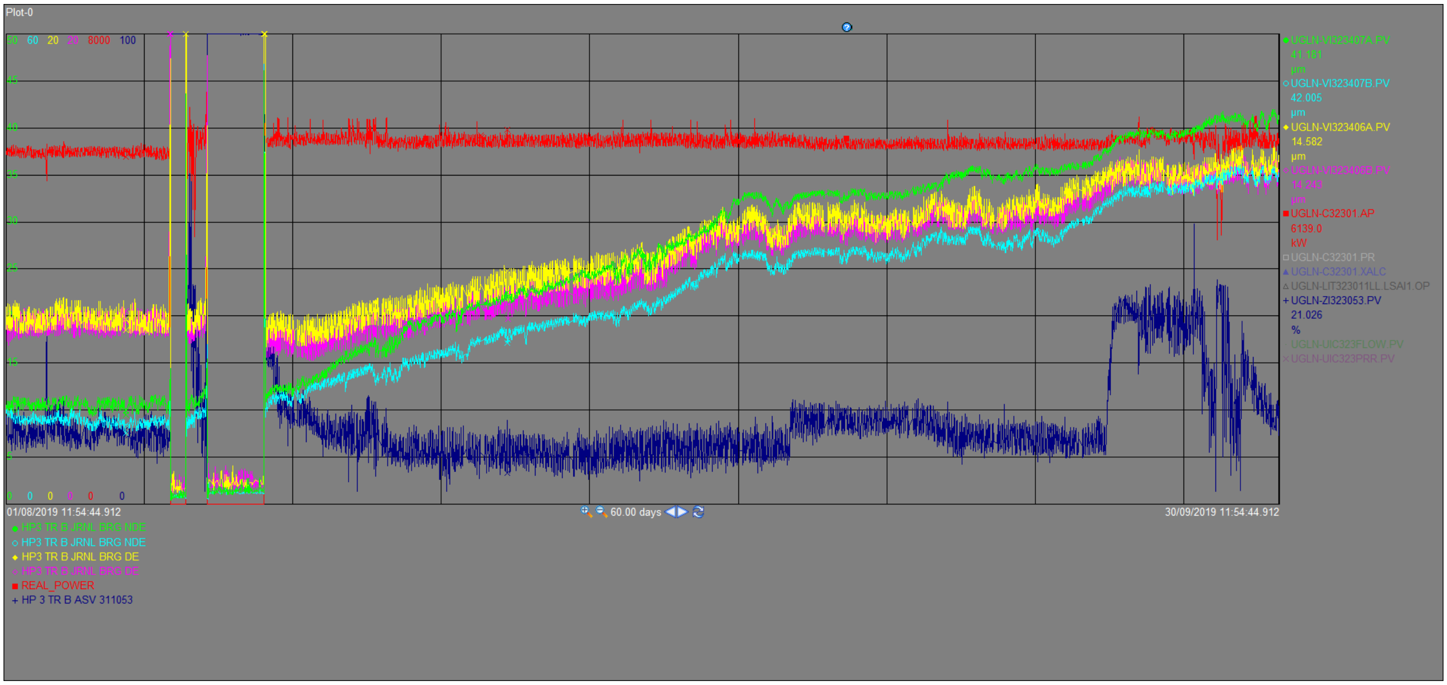

Figure 6: Compressor DE & NDE Radial Vibration Direct Vs Load (Anti-Surge Valve Position dark blue increased % = higher load)

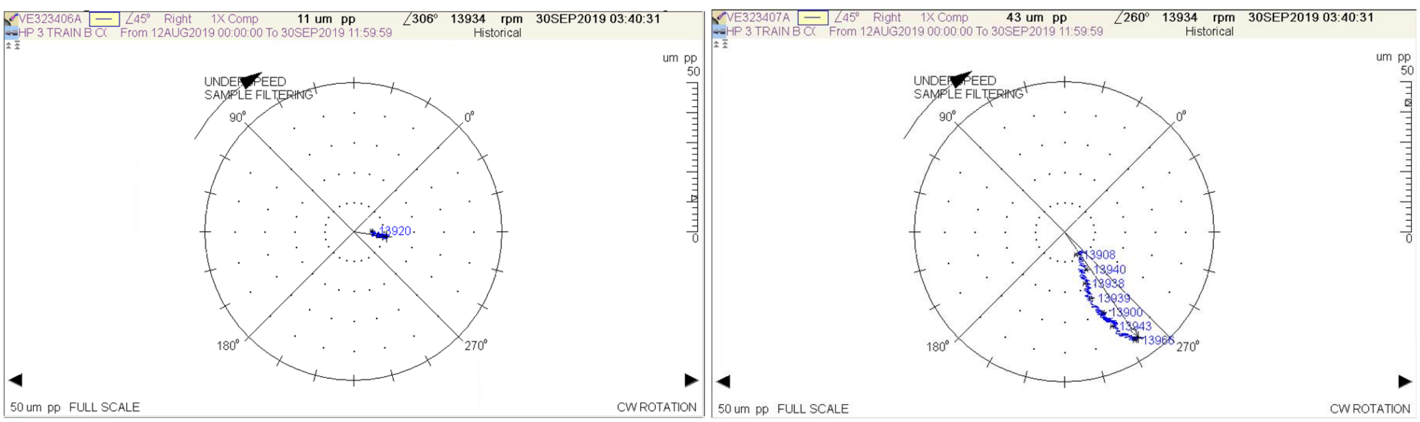

Figure 7: Compressor DE and NDE Compensated 1X Polar Plot

Conclusions:

- Crack (rotor or wheels): recently more likely malfunction, as vibration responds to load abnormally. A crack would respond abnormally to the load changes, temperature changes and speed changes. This abnormal behaviour should be repeatable.
- The same repeatable symptoms for the same operating conditions and the repeatability of abnormal behavior is the one of the most important rule.
- Repeatability of abnormal unidirectional rotor response, must be present and confirmed in case of rotor crack.

Update Analysis on 7th October 2019 on start up
Following the compressor was shut down, restarted, and shut down again. The transient data captured were analyzed.

Note:
- The cross channel phase measurements were taken on the feet of the compressor and no evidence of soft foot was observed.
- Additionally, casing measurements were taking on the bearing housing and show no evidence of looseness.

The following observations have been made:
- There was no evidence of a change in slow roll conditions, the amplitude and phase remained consistent when comparing with slow roll data from January (Figure 8). This indicates that there is no rotor bow. Rule #1 for rotor crack was not confirmed.

Startup data comparison (7th Oct vs 25th June):
- There is no evidence of a change in the first balance resonance when comparing the two starts (Figure 9). Rule #1 for rotor crack was not confirmed.
- There is a visible change in unbalance (increase) at operating speed which is not visible at resonance and this change is not permanent (Figure 9).
- The unbalance only begins to increase after some time at operating speed, the increase is unidirectional (Figure 10). This could indicate cracked rotor thermal sensitivity, demonstrated by unidirectional change as stated in Rule #3.

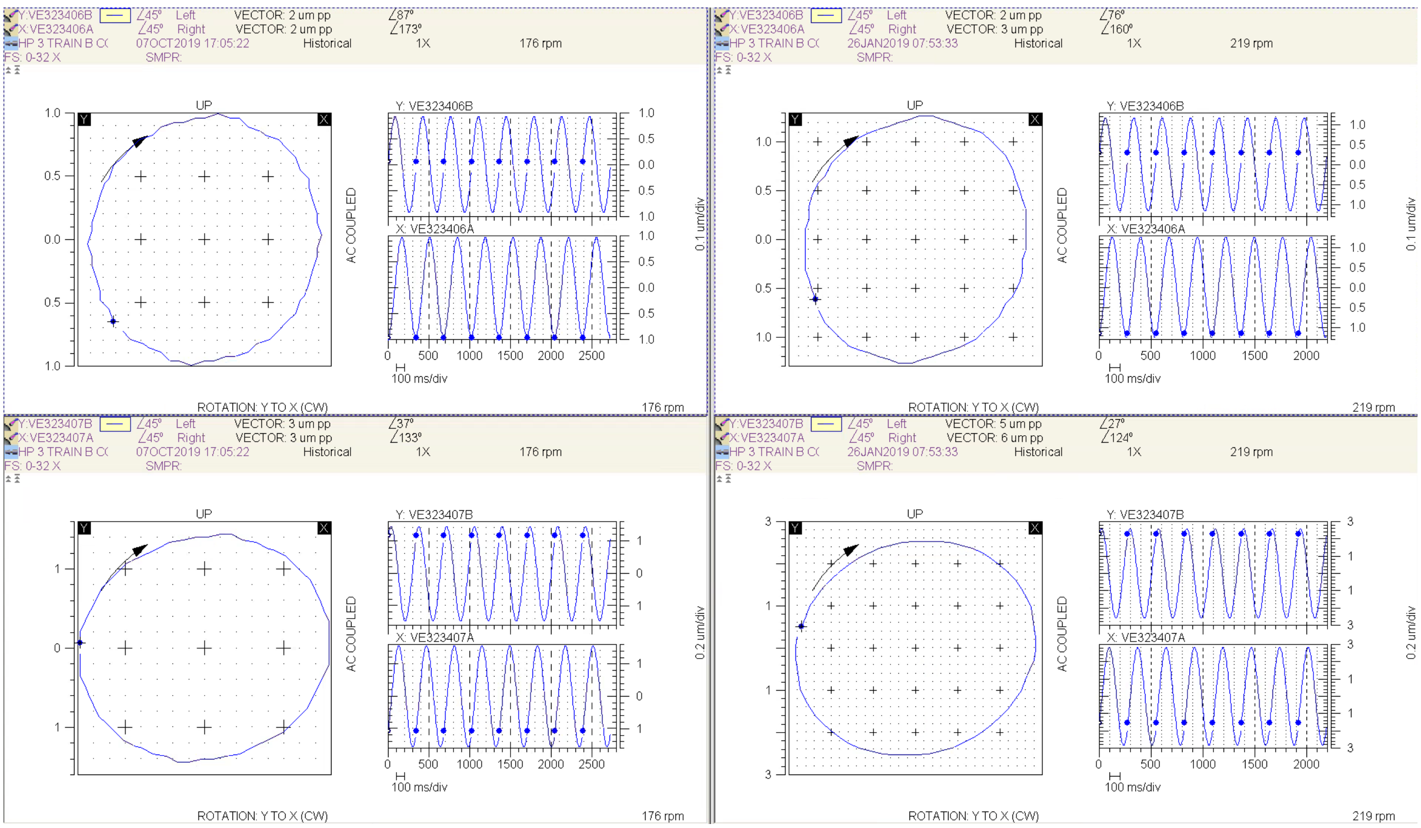

Figure 8: Compressor DE (top left) and NDE (bottom left) slow roll 1X orbit 7th Oct Vs Compressor DE (top right) and NDE (bottom right) slow roll 1X orbit 26th Jan

Conclusions:
- There is evidence of a unidirectional unbalance change and a rotor response which is repeatable (Rule #3 and Rule #4).
- The unbalance increase does not happen immediately after start-up indicating that there may be some temperature influence, possibly thermal sensitivity of a rotor due to crack as stated in Rule #3.

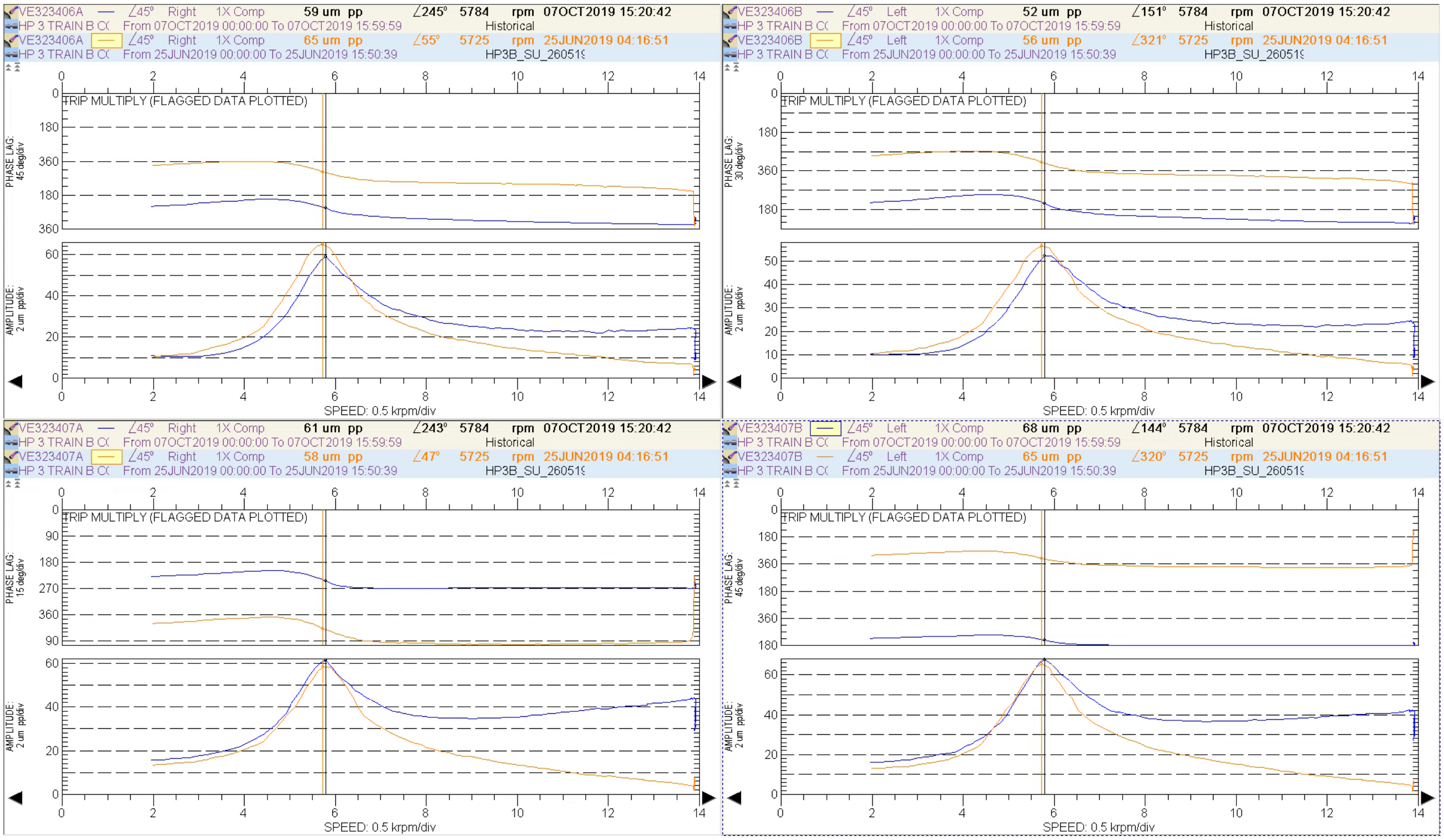

Figure 9: Start up Bode comparison 7th October (blue) and 25th June (orange)

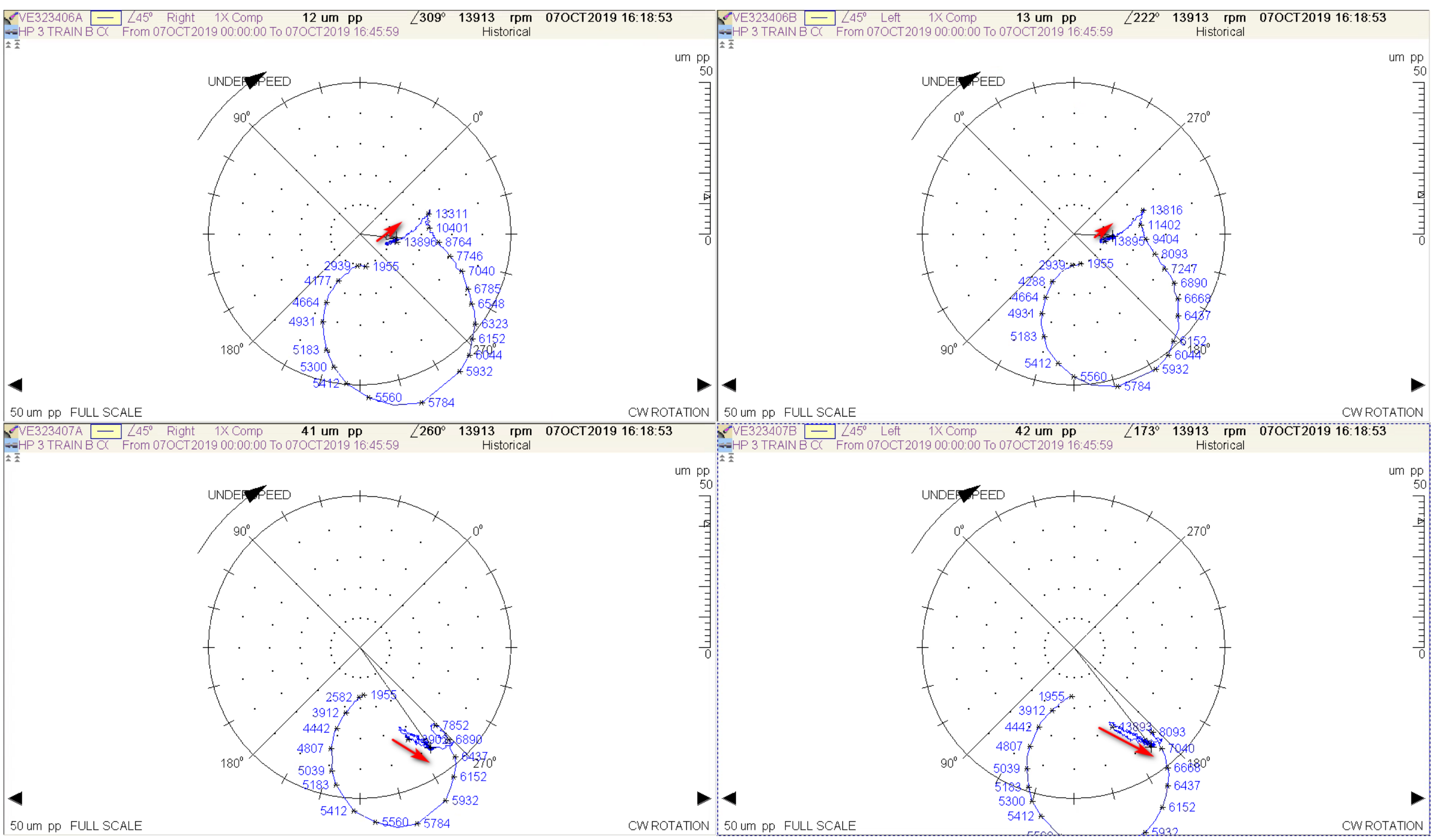

Figure 10: 7th October Start up Compensated 1X polar Plot (unidirectional increase following SU)

Update Analysis on 7th October 2019 on shut down

Shut down data comparison (7th Oct vs 26th Jan):

- No change in the first balance resonance is observed between shutdowns however there is a significant difference between the balance resonance during shut down and start up. The balance resonance on shut down is ~300rpm lower but this can be normal (Figure 11). The Rule #1 is still questionable.
- Significant increase in unbalance at operating speed and resonance on both bearings with the most significant seen on the NDE bearing (Figure 11).

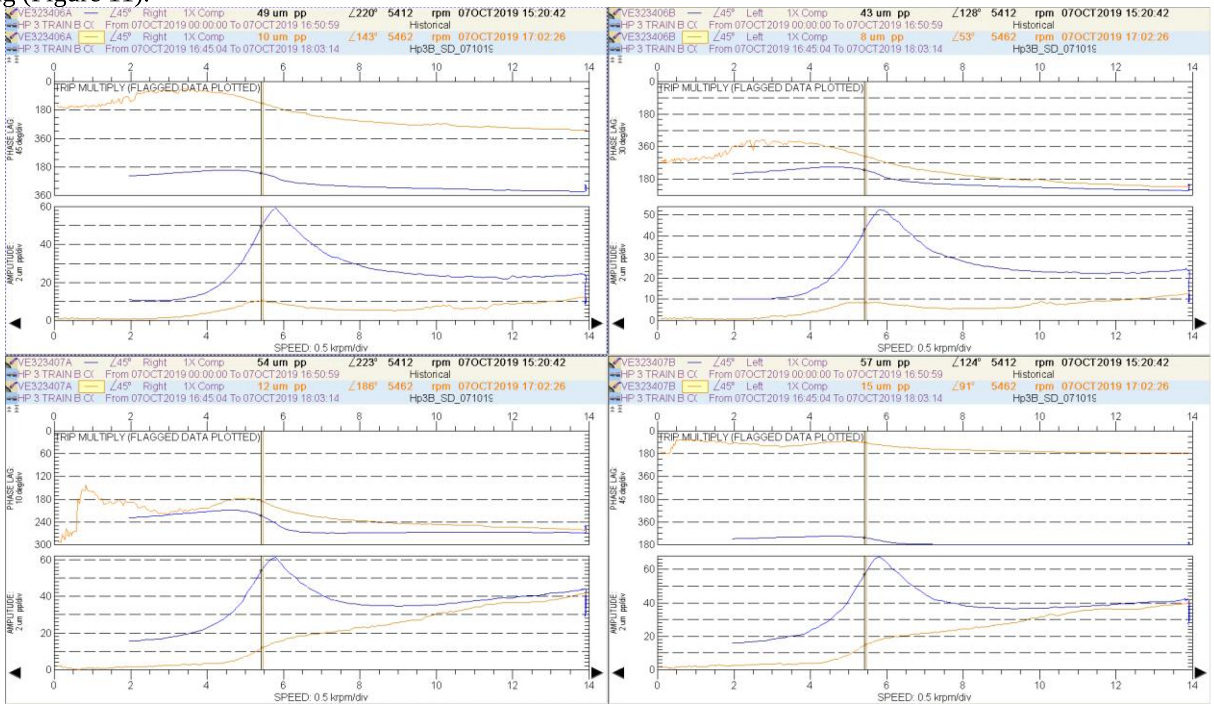

Figure 11: Bode plots comparison 7th October – start up (blue) vs shut down (orange)

Update Analysis on 7th October 2019 on steady state

Steady state data comparison (5th Oct vs 7th Oct):

- Vibration amplitudes and phase returned to levels like those prior to the shutdown. Amplitudes are slightly lower however the compressor was unloaded when restarted on the 7th October (Figure 12).

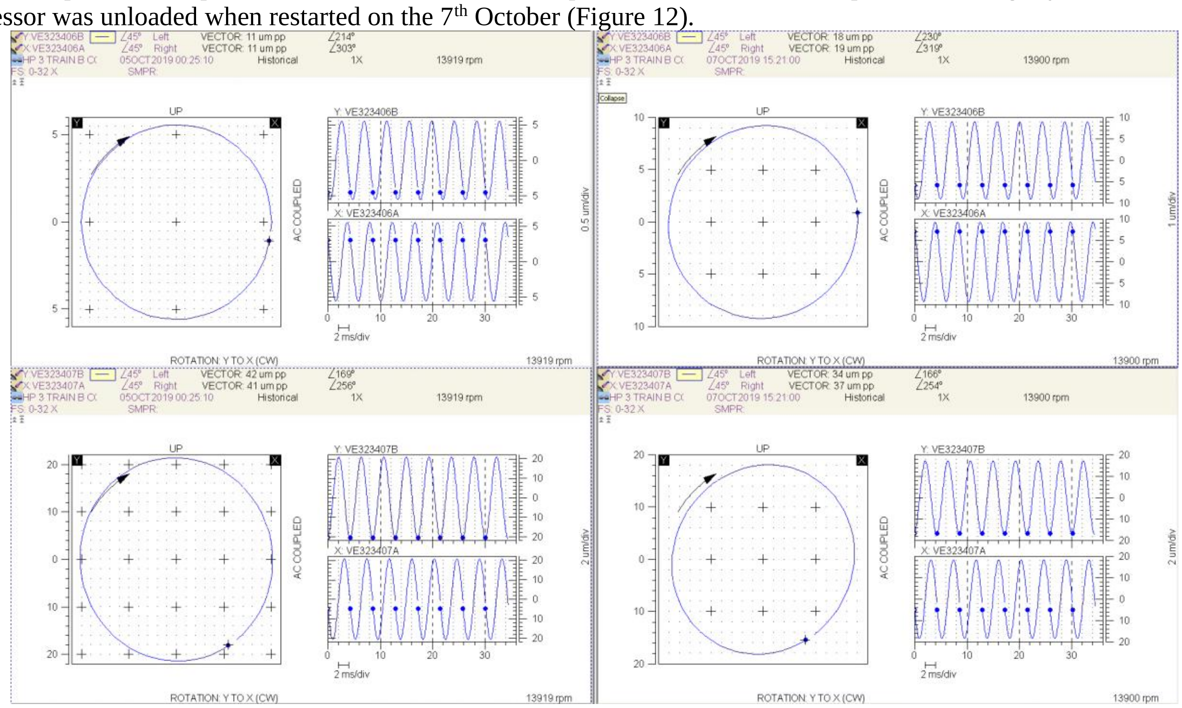

Figure 12: Compressor DE (top left) and NDE (bottom left) Steady State 1X orbit 5th Oct Vs Compressor DE (top right) and NDE (bottom right) Steady State 1X orbit 7th Oct

Update Analysis on 8-9th November 2019 on restart following a plant trip

Following a plant trip on the 8th November which resulted in a controlled shutdown of the compressor, the machine was restarted on the 9th November. The transient data captured during this event was reviewed. Comparing the latest start up with the start up from the 7th October, there is evidence of a significant change to the rotor response occurring from run to run (Figure 13) with the following conclusions:

- During the last start up changes to the rotor mass and shift of the rotor heavy spot have occurred, resulted in balancing the rotor at around 9000rpm, causing the amplitude almost to go to zero at this speed, but with significant phase change (120°). Above this speed the amplitude and phase return to the original values.
- The latest data shows a reduction in the overall amplitudes through the resonance, but the whole rotor transient response has shifted 90-120°, when compared to the previous start up. There has been a significant change in the heavy/high spot location.
- Once the machine reaches steady state there is a unidirectional increase in the 1X component, which if thermally sensitive, this rotor response is the same as the previous start up and indicates that the same issue is still present.
- Comparing the shutdown Polar on the 9th November with the shut down on the 7th October, there is no evidence that any changes to rotor occurred during the last shut down (Figure 14)

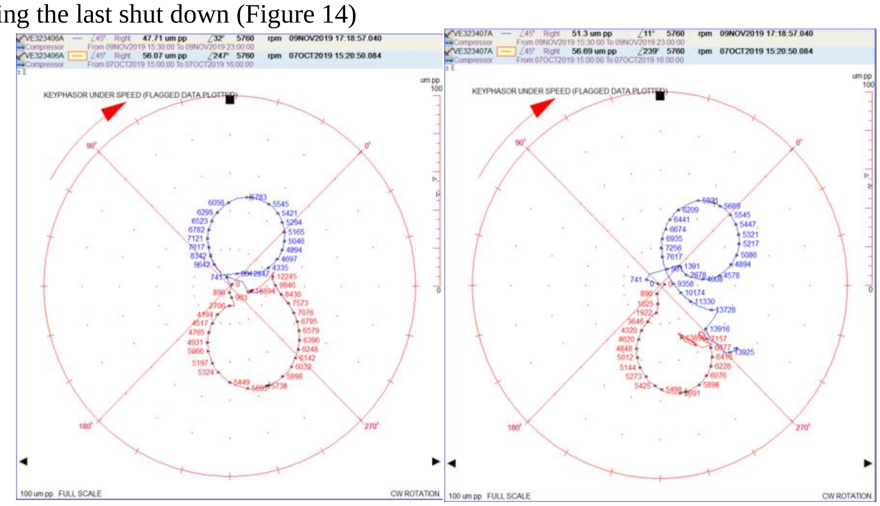

Figure 13: Compressor DE and NDE Polar plots comparison 7th October – start up (red) vs 9th November Start up (blue)

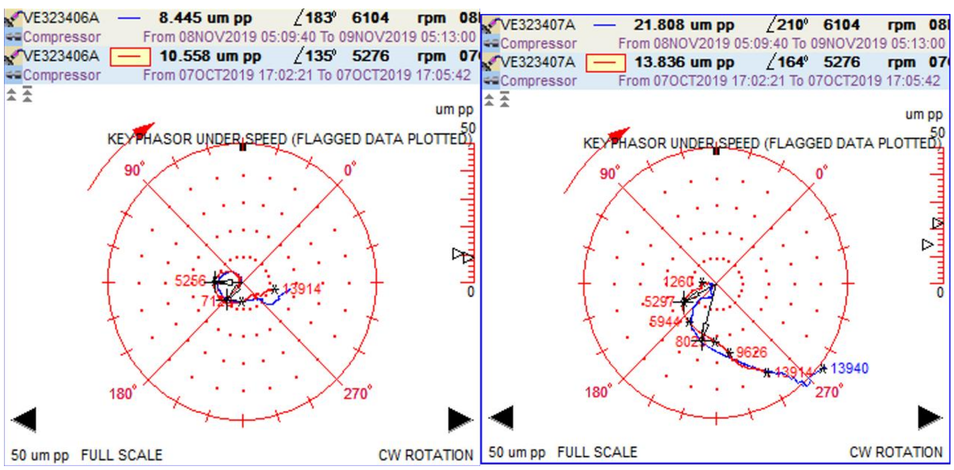

Figure 14: Compressor DE and NDE Polar plots comparison 7th October – Shut down (red) vs 8th November Shut down (blue)

Conclusions:

- It is evident from the latest restart that there has been a significant change in the rotating element, resulting in changes to the heavy/high spot location and rotor response passing the first balance resonance.
- The rotor response change and change in the heavy/high spot location is abnormal, unpredictable. This is not supporting the Rule #4 as the rotor crack has predictable and repeatable symptoms.
- There is still the evidence of an abnormal unidirectional unbalance change in rotor response and it's repeatable for different start-ups.
- Has there is no indication of a reduction in polytropic efficiency, it is therefore unlikely that fouling is occurring.
- These symptoms in conjunction with each other indicate that the integrity of the rotating element has been significantly compromised, resulting in changes to the first balance resonance and the 1X unidirectional response which is repeatable and increasing with temperature.
- It is recommended that the machine is stopped, and the rotating element is immediately inspected for damage.

Update Analysis on 26th November 2019 on restart following a plant trip

Following a gas plant trip on the 21st November which resulted in a controlled shutdown of the compressor, the machine was restarted on the 26th November. The transient data captured during this event was reviewed.

Comparing the latest start up with the previous three startups, there is evidence of a significant change to the rotor response occurring from run to run (Figure 15) with the following conclusions:

- During the last startups significant changes to the rotor mass and shift of the rotor heavy spot have occurred, which resulted in increase or decrease of the amplitude at the resonance depending where the mass was shifted in comparison of the original unbalance.
- The whole rotor transient response has shifted 90° with rotation, when compared startups (OCT07 and OCT21, 2019)
- There is further rotor transient response shift by 135° with rotation, when compared startups (OCT21and NOV09, 2019), what represents total shift of 225° with rotation, comparing the original unbalance heavy spot (OCT07, 2019).
- The rotor transient response has shifted back 135° against rotation, when compared startups (NOV09 and NOV26, 2019).
- The same changes are observed on the Compressor DE and NDE side.
- But what is very strange, despite the different startup's rotor transient response, once the machine reaches steady state there is always, the change to the exact same 1X rotor response (15.5umppp @ 222°for VE323406B reference and (50.5umppp @ 181°for VE323407B reference) and stabilizes exactly on the same value for each different start up.

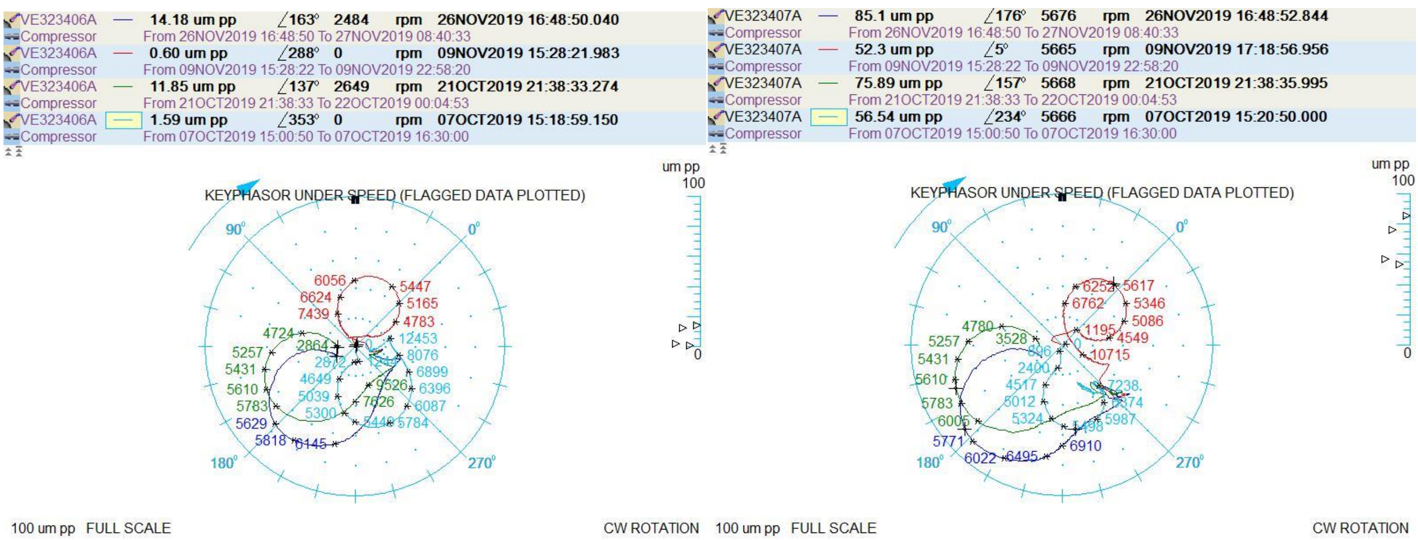

Figure 15: DE and NDE Startup Polar plot comparison from four most recent starts showing change in high / heavy spot relationship between starts.

Conclusions:

- There is further evidence from the latest restart that there has been a significant change in the rotating element, resulting in changes to the heavy/high spot location and rotor response passing the first balance resonance. The changes can be seen across the previous 4 startups which each start showing a different rotor response. The rotor response change and change in the heavy/high spot location is abnormal, unpredictable, and therefore the machine should be removed from the service, as secondary damage due to a failure of the rotor cannot be determined.
- There is still the evidence of an abnormal unidirectional unbalance change in rotor response and it's repeatable for different start-ups.
- These symptoms in conjunction with each other indicate that the integrity of the rotating element has been significantly compromised, resulting in changes to the first balance resonance and the 1X unidirectional response which is repeatable and increasing with temperature.

INSPECTION FINDIGS

Compressor bundle was removed and sent back to the OEM for inspection.

- Rotor to thrust collar interface fit was found to be worn and antirotating pin bent (Figure 17).
- The heavy-duty spring pin was deformed/bent/squashed (Figure 17, 18)
- The fitting diameter and the axial contact surface to the shaft was worn
- Thrust collar had become eccentric (radial run out measured) causing a mass unbalance on the rotor.
- High speed balancing test with various thrust collar eccentricities proved this was the cause of the vibration increases.
- The tension on the end screw was correct.

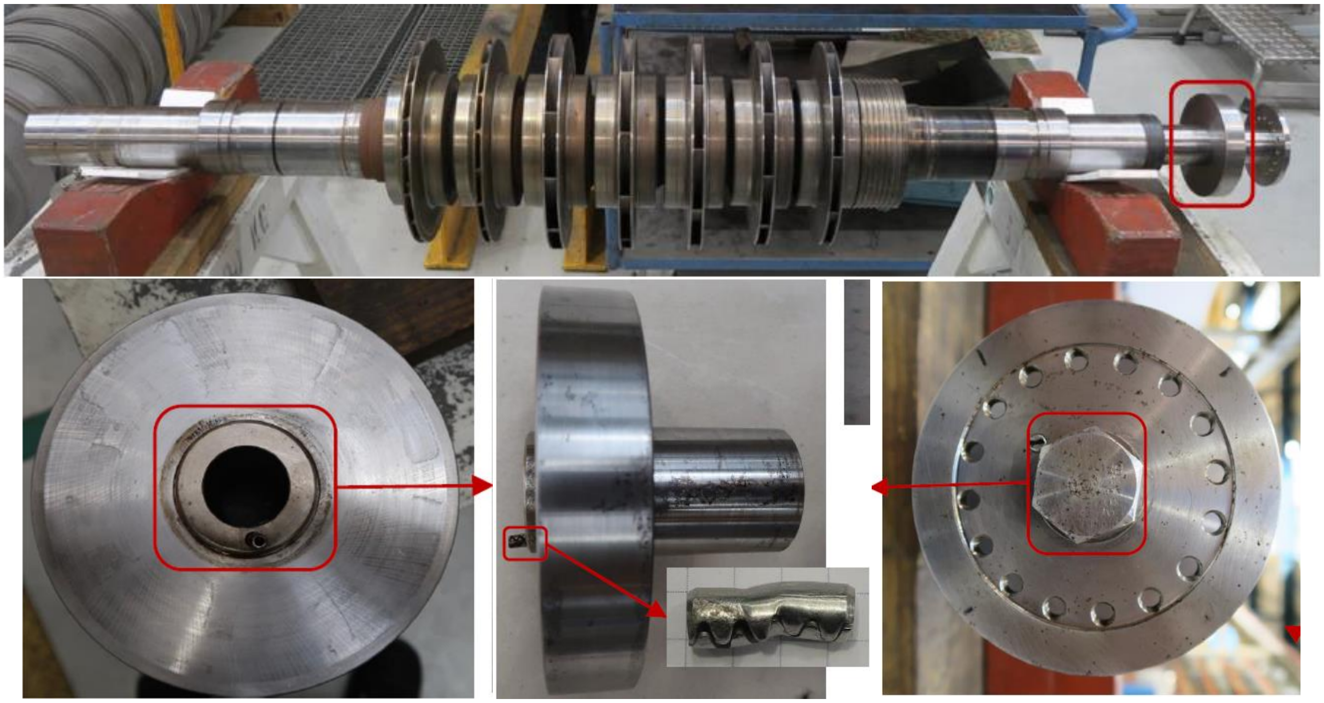

Figure 17: Compressor rotor and thrust collar assembly

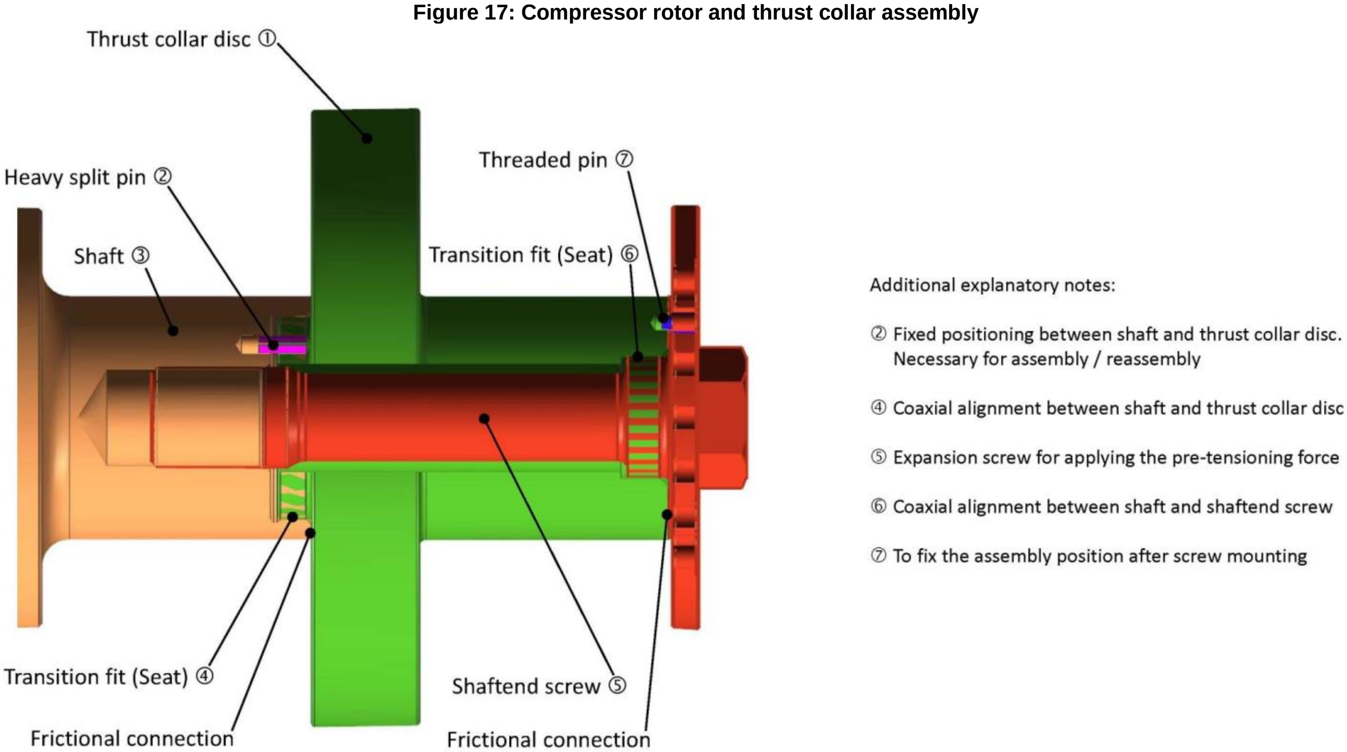

Figure 18: Compressor rotor thrust collar assembly

Findings continues (Figure 19):

- The thrust collar fitting diameter was unevenly worn
- The radial run out on the fitting diameter was out of centre due to the wear pattern
- The biggest wear (read color) was in the direction of Heavy split pin (antirotating pin)
- The sector on the bottom of the thrust collar fitting (green color) was in acceptable condition
- The axial contact surface to the thrust collar was worn
- The location hole for the anti-rotation pin was out of position and slightly too big
- The heavy-duty spring pin was deformed/bent/squashed

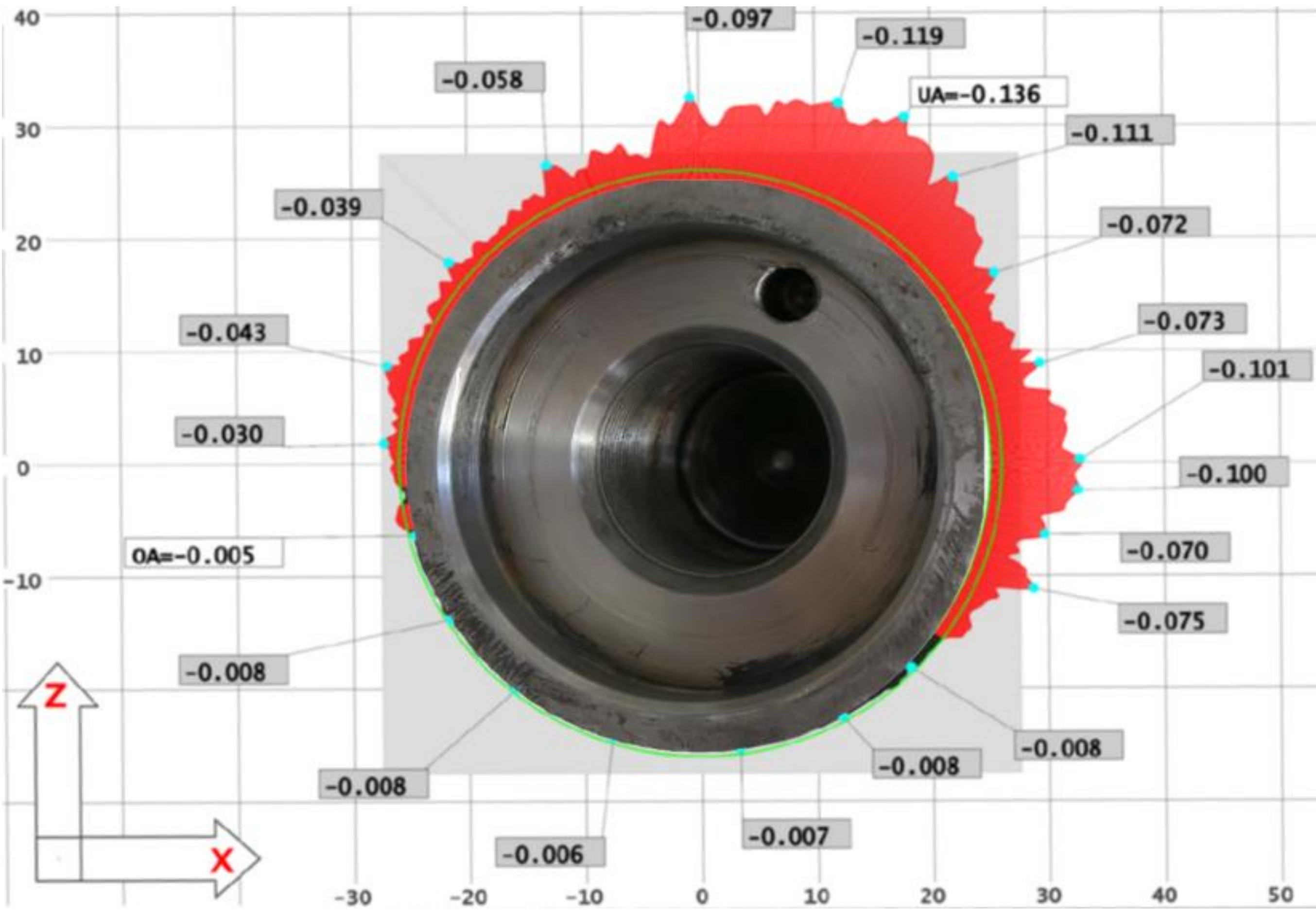

Figure 19: Compressor rotor thrust collar fitting, with worn runout pattern

Possible sequence of events:

- The axial thrust collar could originally relate to the shaft during the original assembly without noticeable signs. However, the positioning pin could already be deformed due to the misaligned holes between the shaft end and axial thrust collar.
- Thereafter the rotor was balanced at high speed with excellent values.
- The compressor has been operated without malfunctions for a period of more than two years 2017-2019.
- During this period, the transition fit could be macroscopically negatively influenced by the deformed pin.
- This influence on the connection, most probably led to steady deterioration under changing operational loads, but was not noticed.
- After another machine trip, presumably the "trigger", the machine was switched off in August 2019. During this incident, an eccentricity could start to build up, generating the progressive unbalance and vibration as measured.
- This further additional load was manifested in further operation until October 2019 by demonstrably measured vibration values that slowly increased steadily. This further load also increased the eccentric misalignment of the axial thrust continuously.
- Presumably due to a lower process load, a stable vibration level and calming at a higher level has been settled even with further shutdowns until the end of December 2019.
- A potentially increased load, however led to a further increase in vibrations and after a short period till alarm level and therefore to the following compressor shutdown in January 2020 and the consequently correct cartridge replacement.

CONCLUSIONS

The case study describes and highlights the complexity of rotor crack diagnostics as it can be often masked or mistaken by other existing machine malfunctions.

In this case the symptoms in conjunction with each other indicated that the integrity of the rotor element has been significantly compromised, resulting in changes to the first balance resonance and the 1X unidirectional response which is repeatable and increasing with temperature. On the other hand, the rotor response and change in the heavy/high spot location was abnormal, but random and not repeatable, therefore not indicating the rotor crack. The machine was removed from the service, as secondary damage due to a failure of the rotor could not be determined.

The Compressor bundle was removed, and the rotor was inspected. The rotor problem was confirmed, not due to rotor crack but luckily only due to rotor thrust collar worn fit and antirotating pin bent. Thrust collar had become eccentric (radial run out measured) causing a mass unbalance on the rotor in one direction, with similar symptoms as rotor crack. It was confirmed, that due to the mechanical wear on the compressor shaft end the only remediation was to have removed the bundle to address the condition. BN remotely witnessed the compressor inspection where it was discovered the thrust collar to rotor fit was worn and oversized, antirotating pin bended, causing the thrust collar to be eccentric resulting in an unbalance.

Following the initial diagnosis and continued monitoring by the BN SSA team the machine was able to be safely operated for 6 months allowing for a replacement compressor bundle to be prepared and installed minimizing the operational and business impact.

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