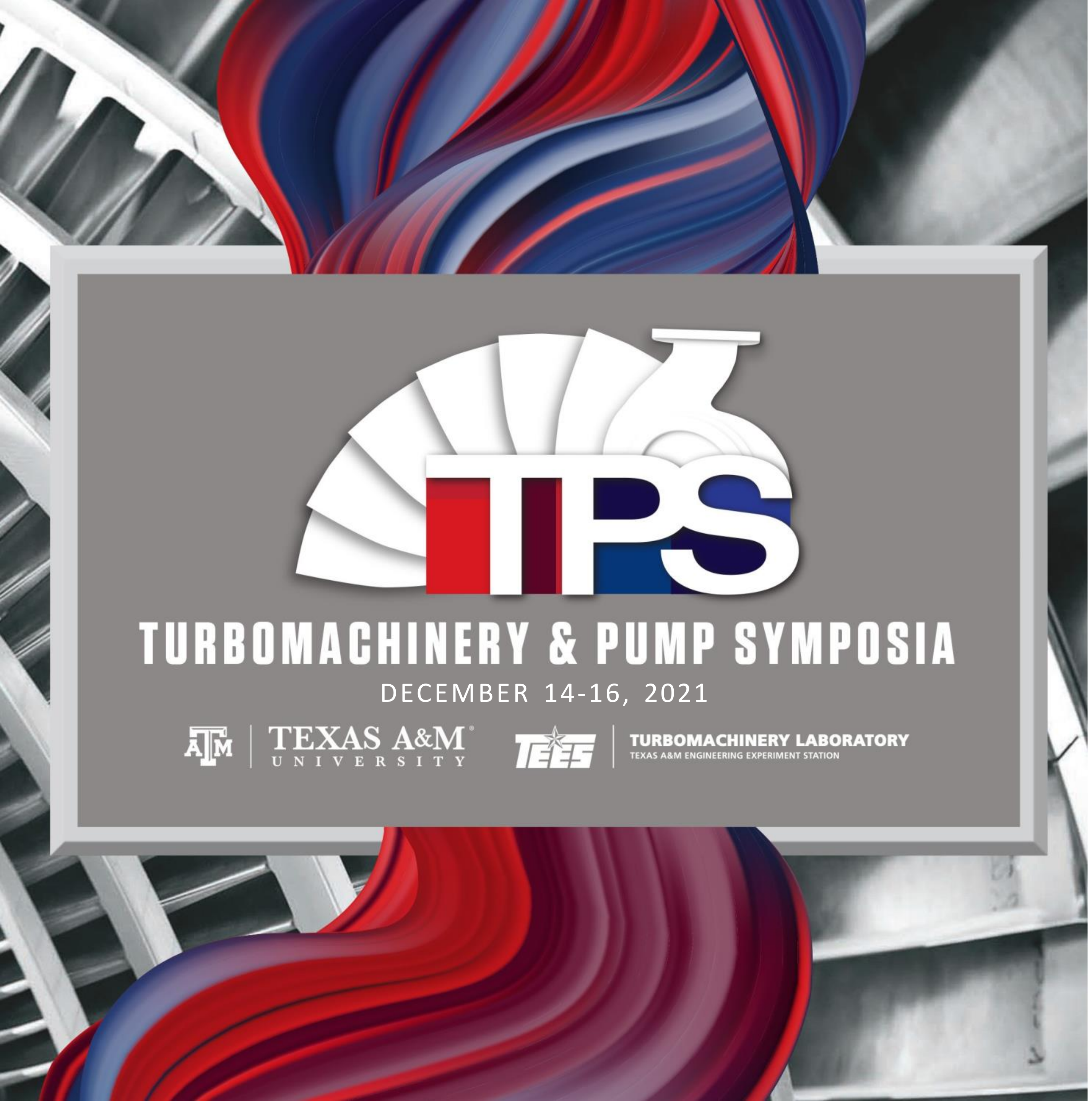

High Vibration of a Centrifugal Compressor Casing Caused by the Gas Leakage across a Damaged Gasket

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AHMAD ALAMER is a Group leader at the Consulting Services Department in Saudi Aramco. He joined Saudi Aramco in 2008 and worked in multiple roles as a rotating equipment expert and leader. Mr. Alamer holds an MS degree in Mechanical & Aeronautical Engineering from the University of California, Davis.

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Abstract

A sudden increase of casing vibration along with loud noise and shaking of casing and process piping were observed on a large sales gas centrifugal compressor, while the rotor vibration was below alarm level.

When the unit was shut down and restarted several times, the issue occurred again after few hours of operation.

A detailed investigation of site data combined with aerodynamic analysis (CFD) and mechanical analysis (FEA) allowed identification of the unexpected root cause of this severe casing vibration issue as the rupture of an elastomer gasket inside the compressor casing.

TURBOMACHINERY & PUMP SYMPOSIA

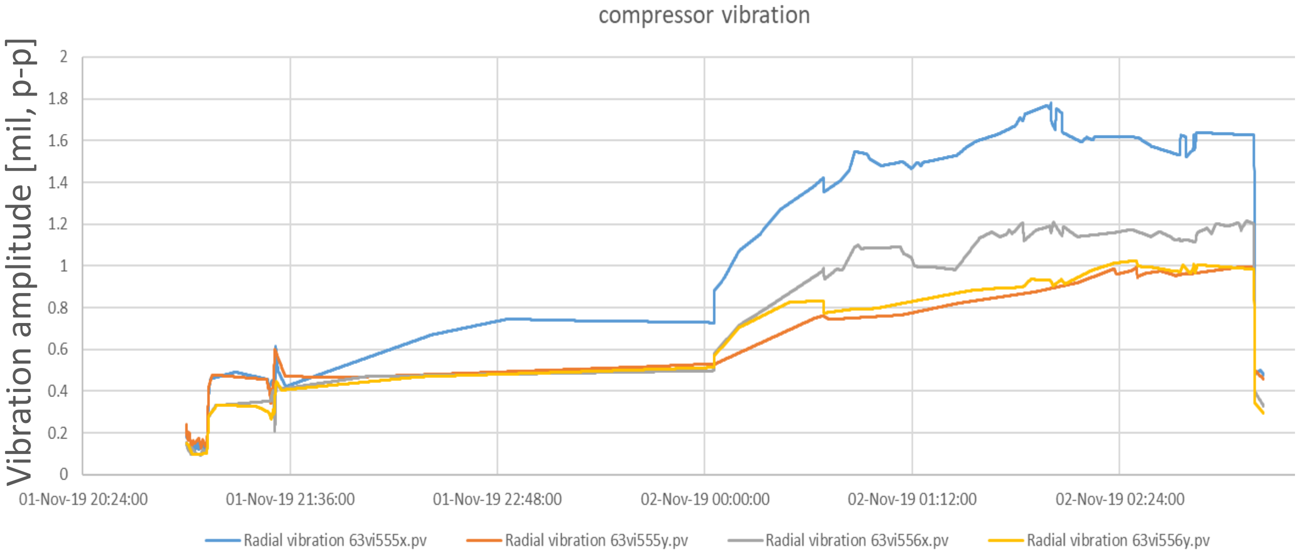
Problem Statement

What:

During stable operation, casing vibration increased suddenly to very high level. Rotor vibration also increased, but still within the alarm level. Loud noise was heard, as well as shaking of compressor and process piping. Unit was manually stopped. Subsequent restart attempts showed initially low vibrations, but after few hours vibration increased again to the same high level.

When:

on Oct. 31st 2019, during normal operation (unit had been running for 10 years). Then again after several restarts, in November 2019.



Where:

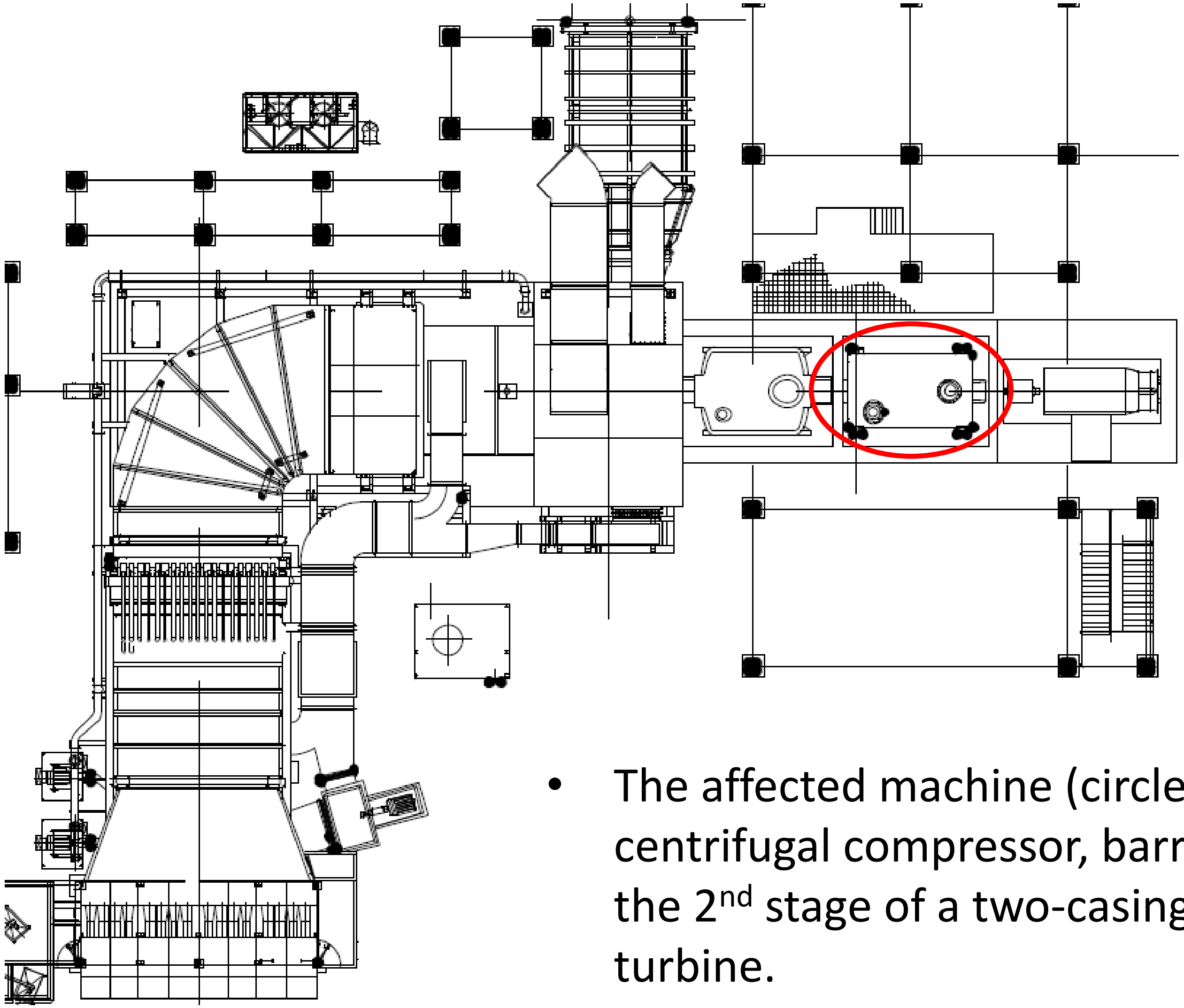
At an Aramco NGL plant, Saudi Arabia.

Significance:

Compressor casing vibration was unacceptable, preventing unit operation. Production is interrupted.

TURBOMACHINERY & PUMP SYMPOSIA

Presentation of Plant and Affected Unit



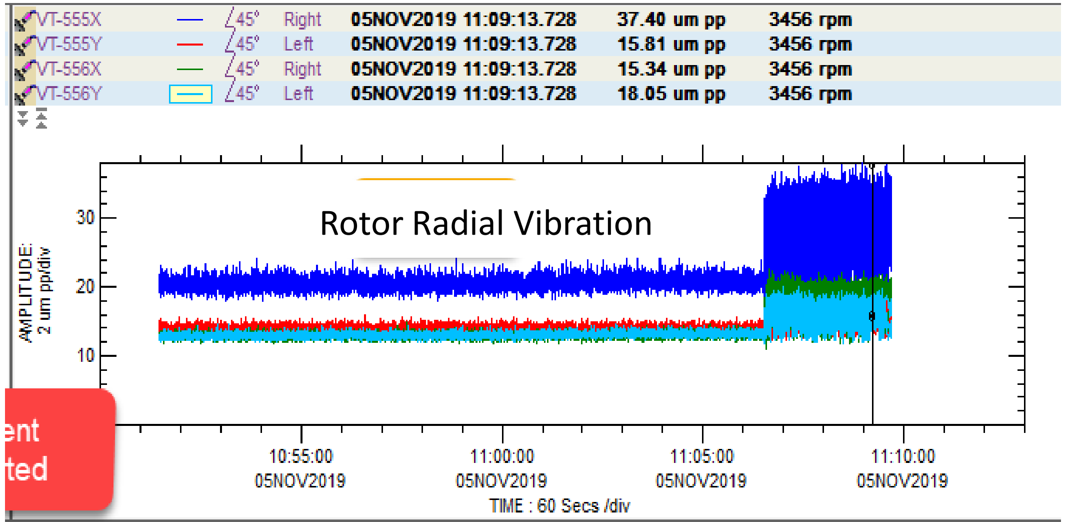
- The affected machine (circled in red) is a natural gas centrifugal compressor, barrel-type with 8 stages in line. It is the 2nd stage of a two-casing compression train, driven by gas turbine.
- The plant includes a total of 4 identical compression trains, running in parallel.

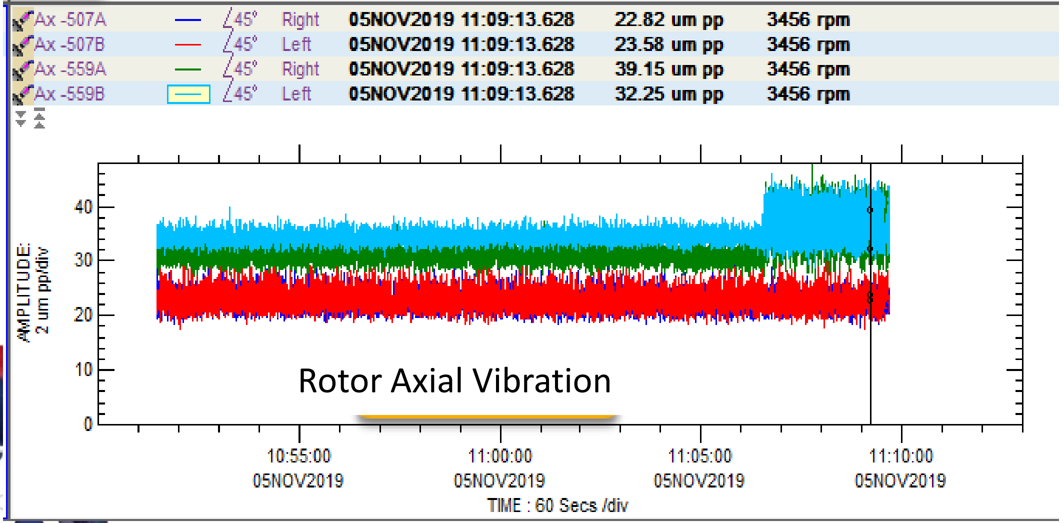
Investigation Methodology

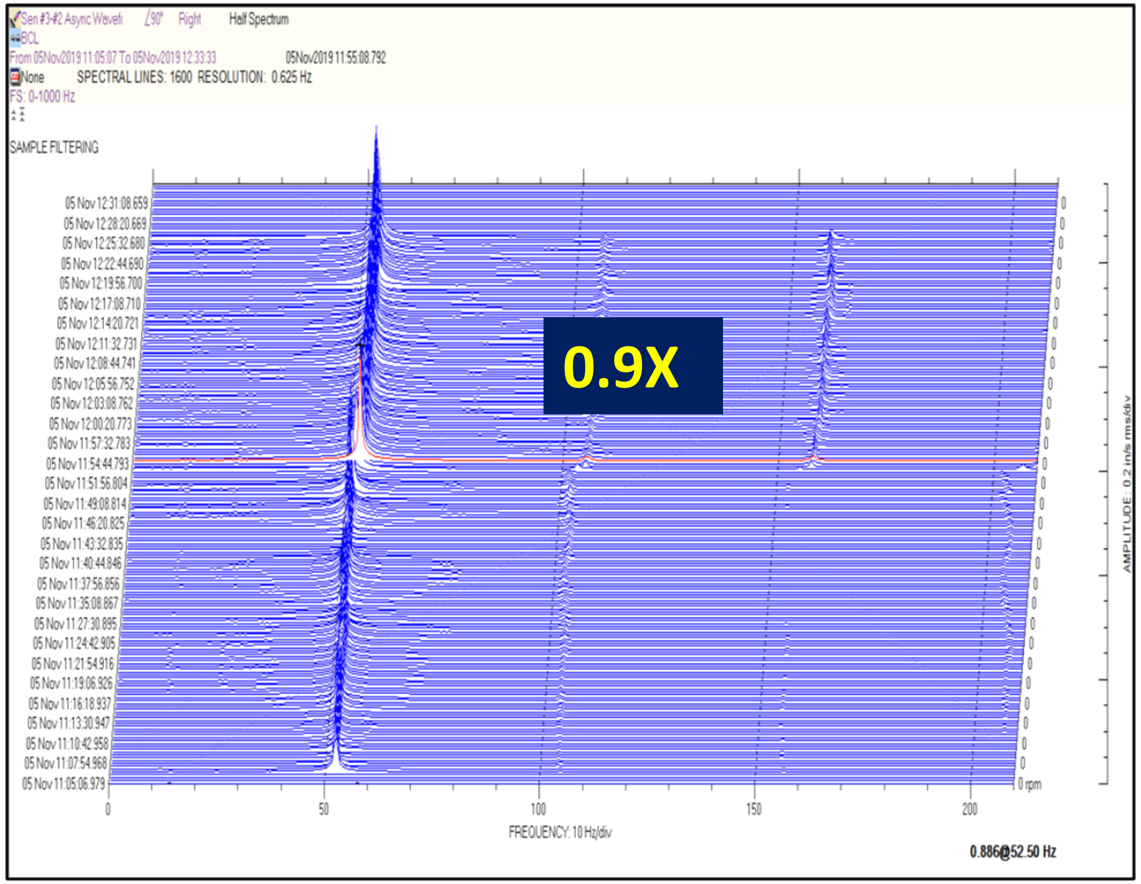
Start the investigation where the problem appeared: abnormal noise and high vibration

- The result of conventional rotor vibration analysis was not conclusive.
  - Resonance was suspected; but didn’t have enough evidence to identify the source
- The decision to open the compressor for inspection has consequences
  - Must remove the motor and clutch to reach the compressors
  - Complex piping layout on top of the compressors
  - Special tools required for dismantling the compressor
  - Hence, isolating the problem was critical
- Utilize state-of-the-art technologies to identify the source of resonance
- Correlate between vibration and process

Investigation Methodology - Vibration

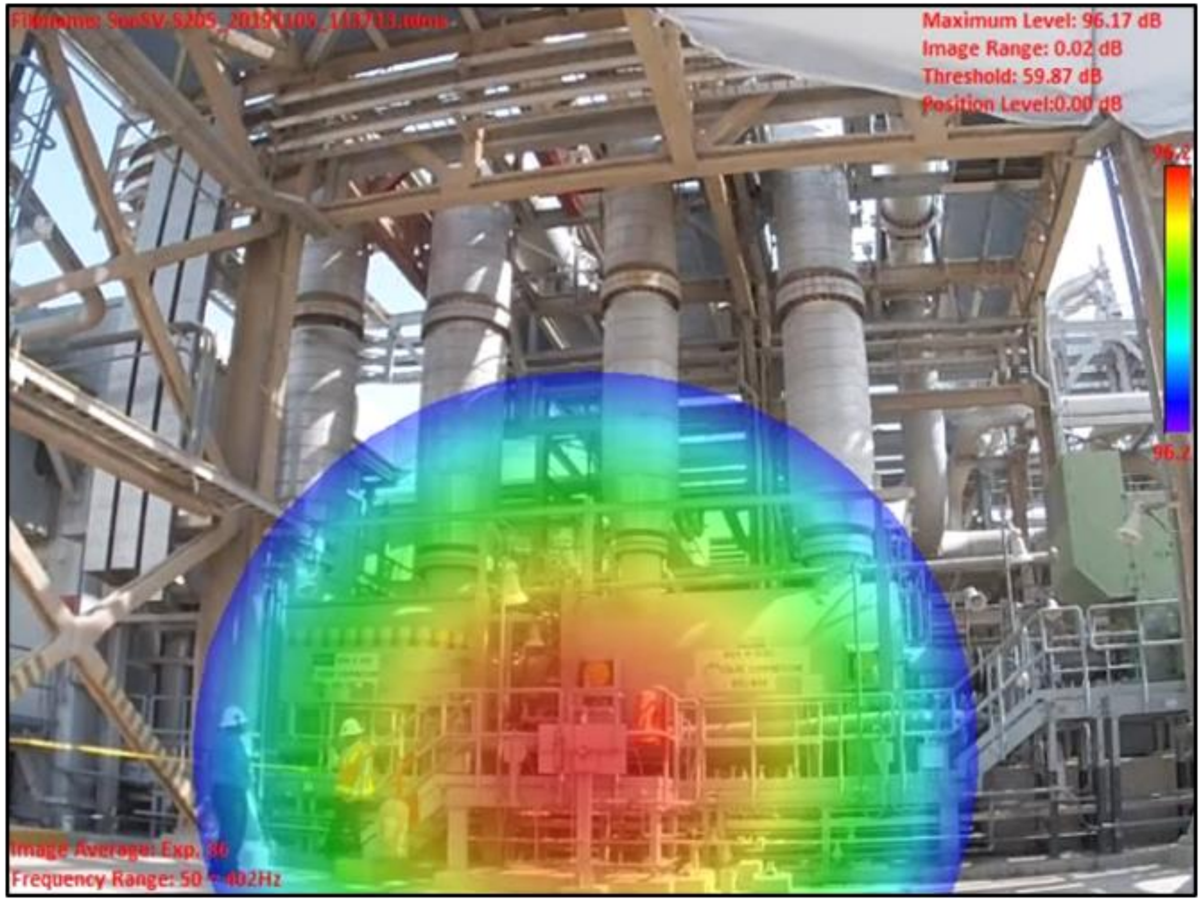






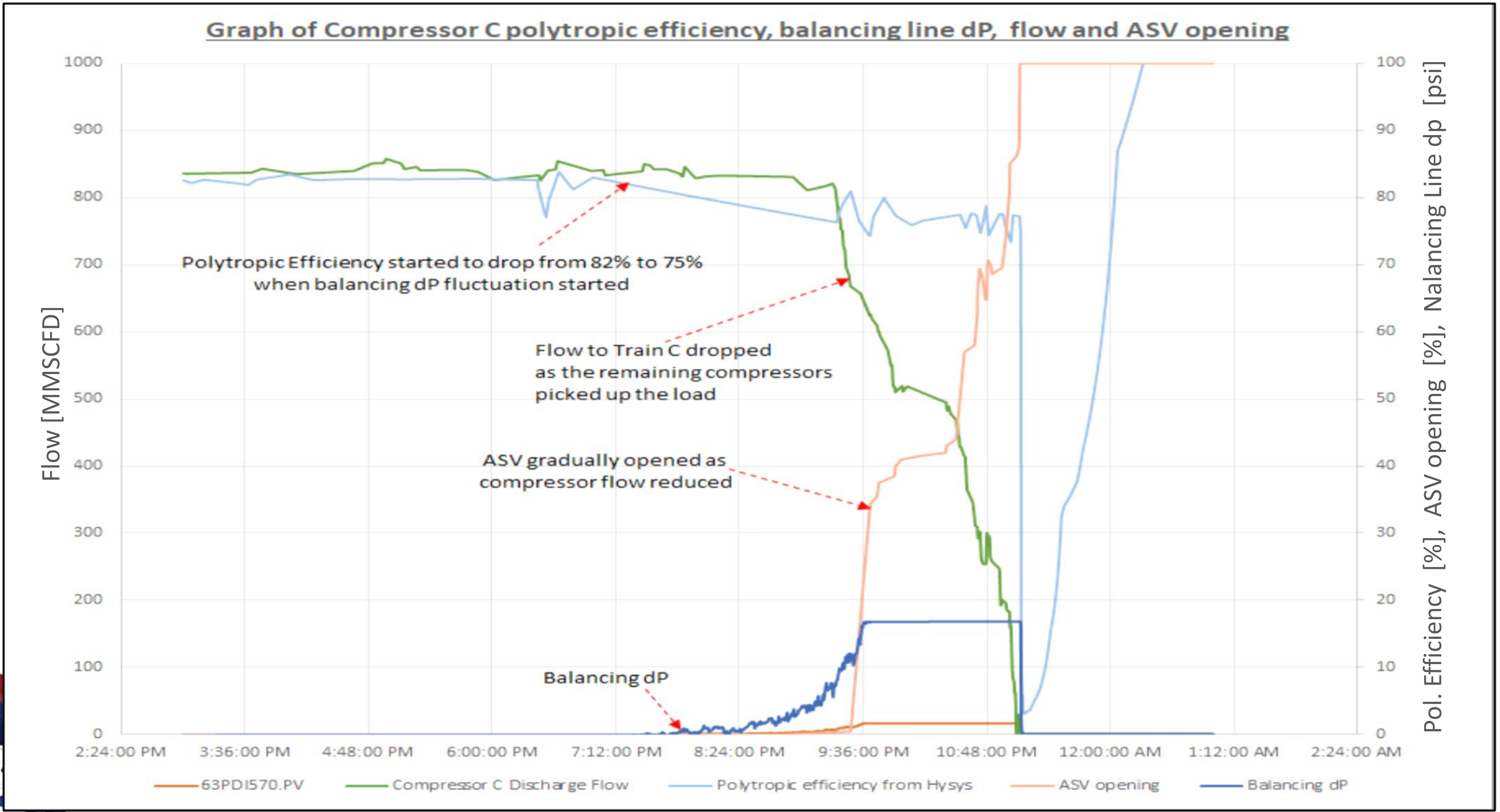
Investigation Methodology - Noise

Sound camera confirmed the noise source to be from compressor discharge side

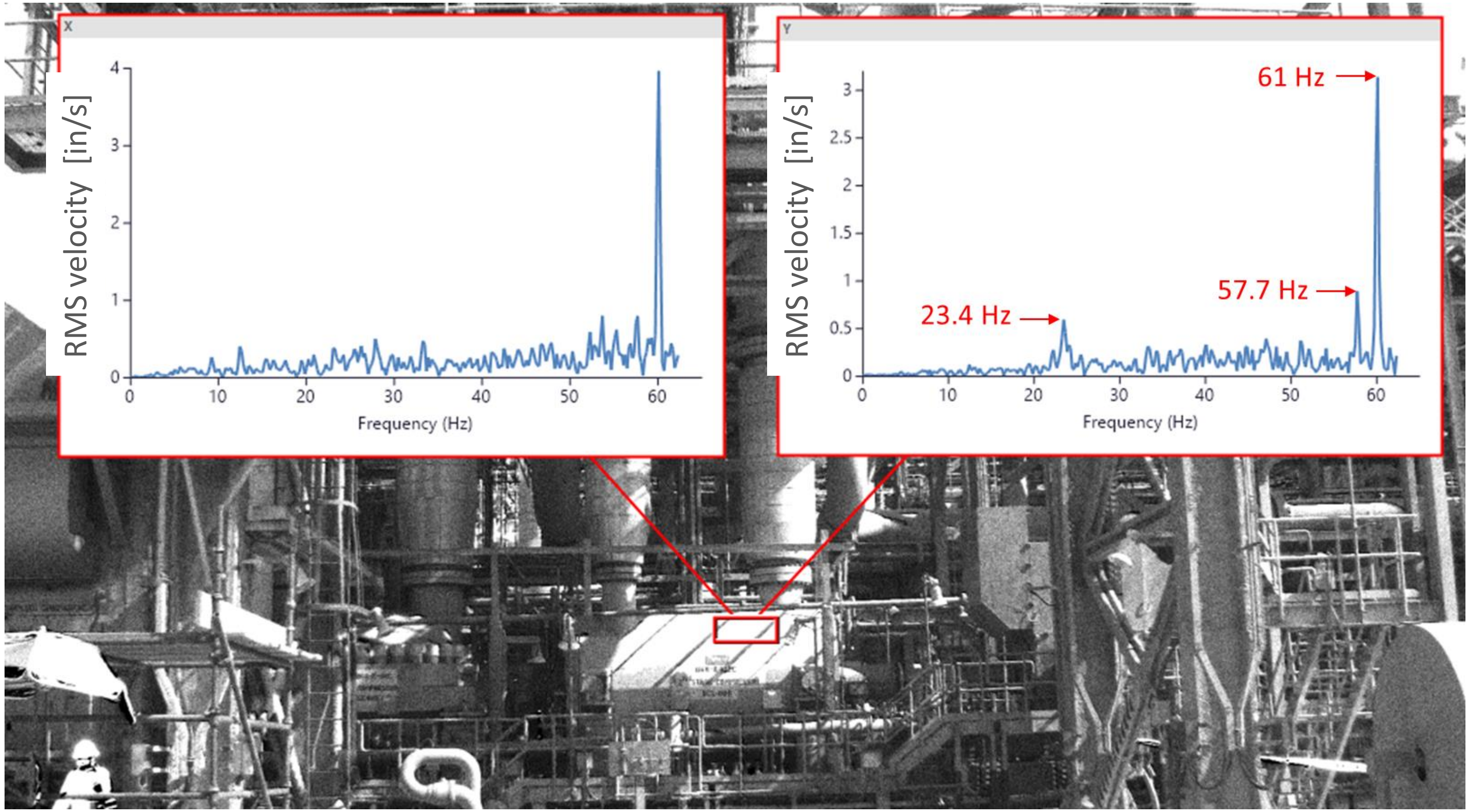


TURBOMACHINERY & PUMP SYMPOSIA

Investigation Methodology - Process



Investigation Methodology - Structure

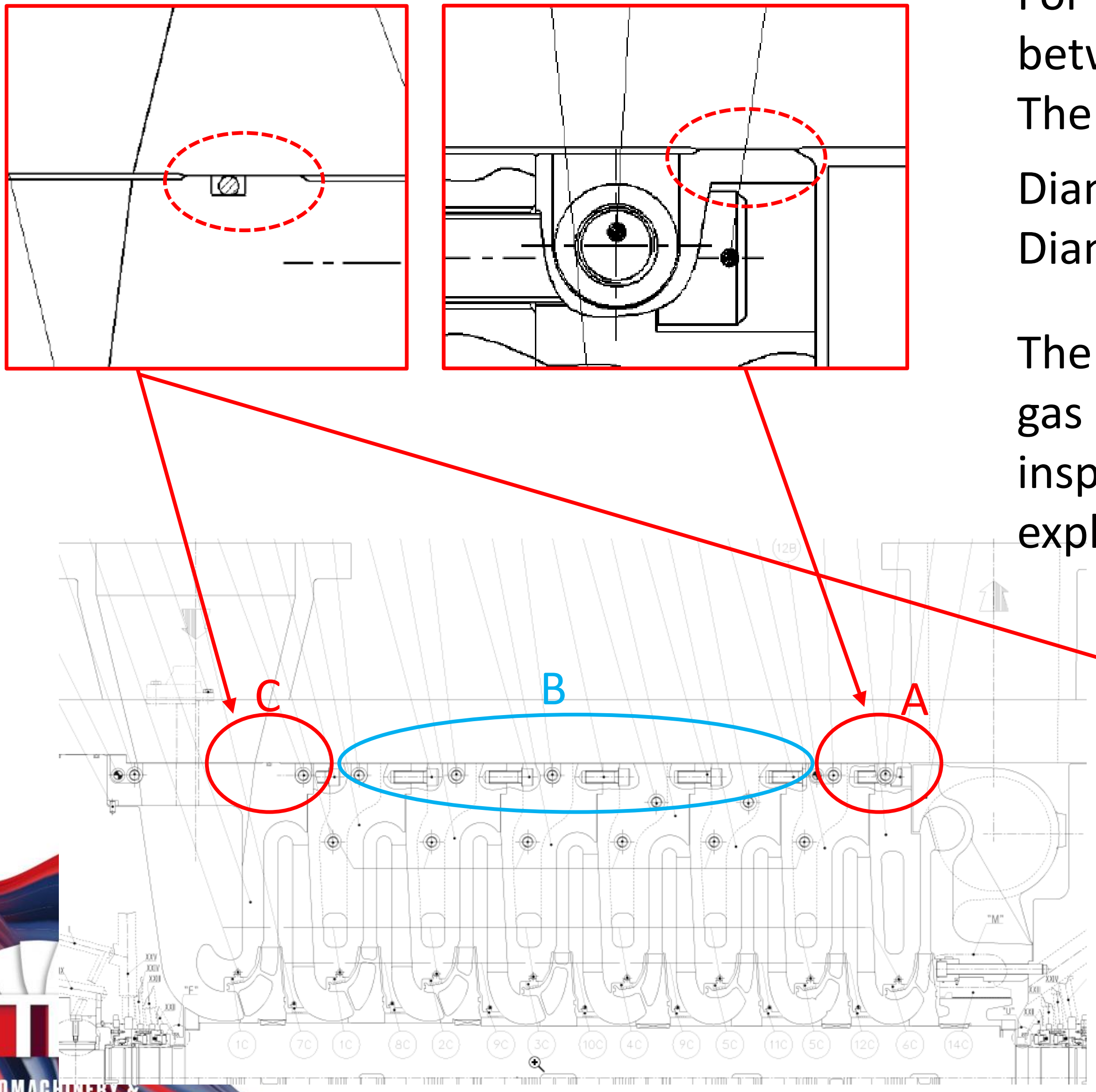


Motion amplification technology was utilized to confirm that the abnormal vibration stems from the casing itself and not from the surrounding structure.

- The dominant frequency of 1x (61 Hz) seen in the frequency spectrum originates from the machine running speed.

- The abnormal frequency component of 0.9x (57.7 Hz) observed on radial and casing vibration was also observed with the motion amplification camera.
- An abnormal frequency of 23.4 Hz was identified in the frequency spectrum which was sensitive to overall vibration levels. The source of this component was not clear.

Data from site

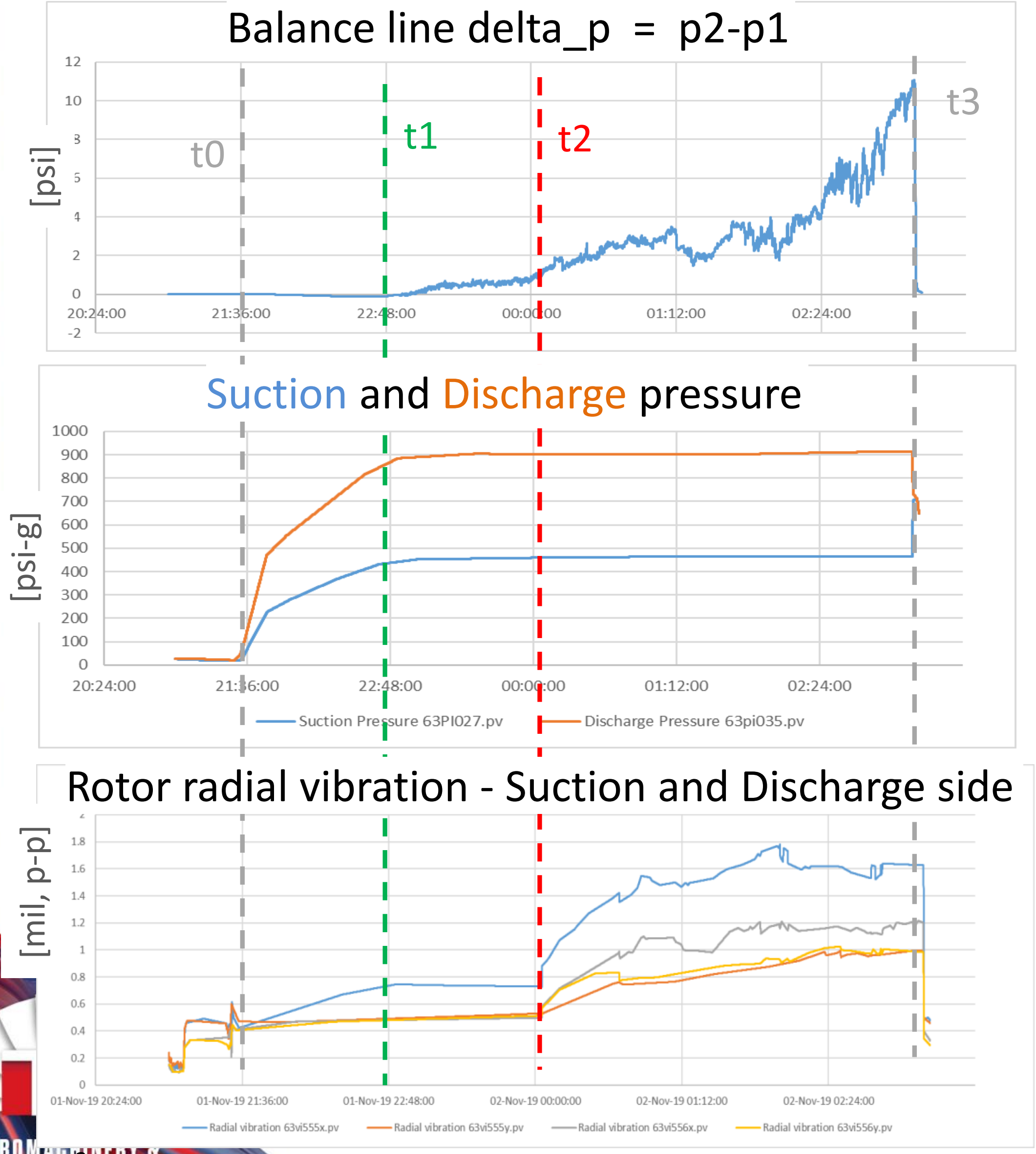


For fitting & sealing purpose there are two close-gap points between bundle and casing: A and C.
The clearance in zone B (large-gap) is much wider.

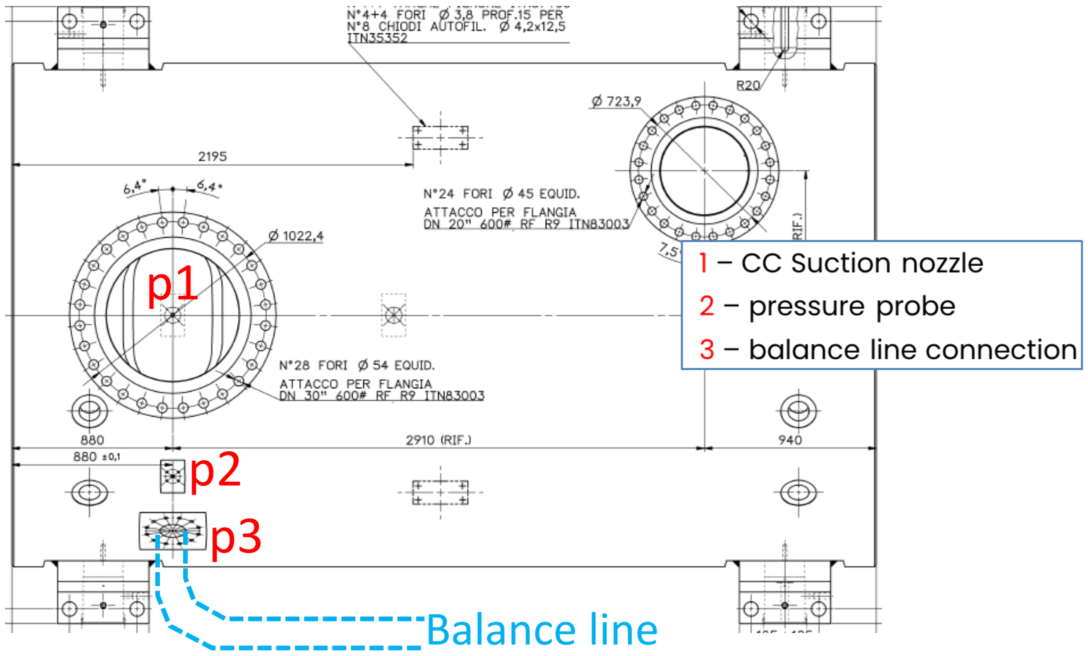
Diametral clearance @ location A & C: a = 0.23 mm
Diametral clearance @ location B: b = 6.25 mm

The O-ring at point C, separating gas @ p_discharge from gas @ p_suction, was found severely damaged during inspection. The appearance of the failed O-ring is typical of explosive decompression.

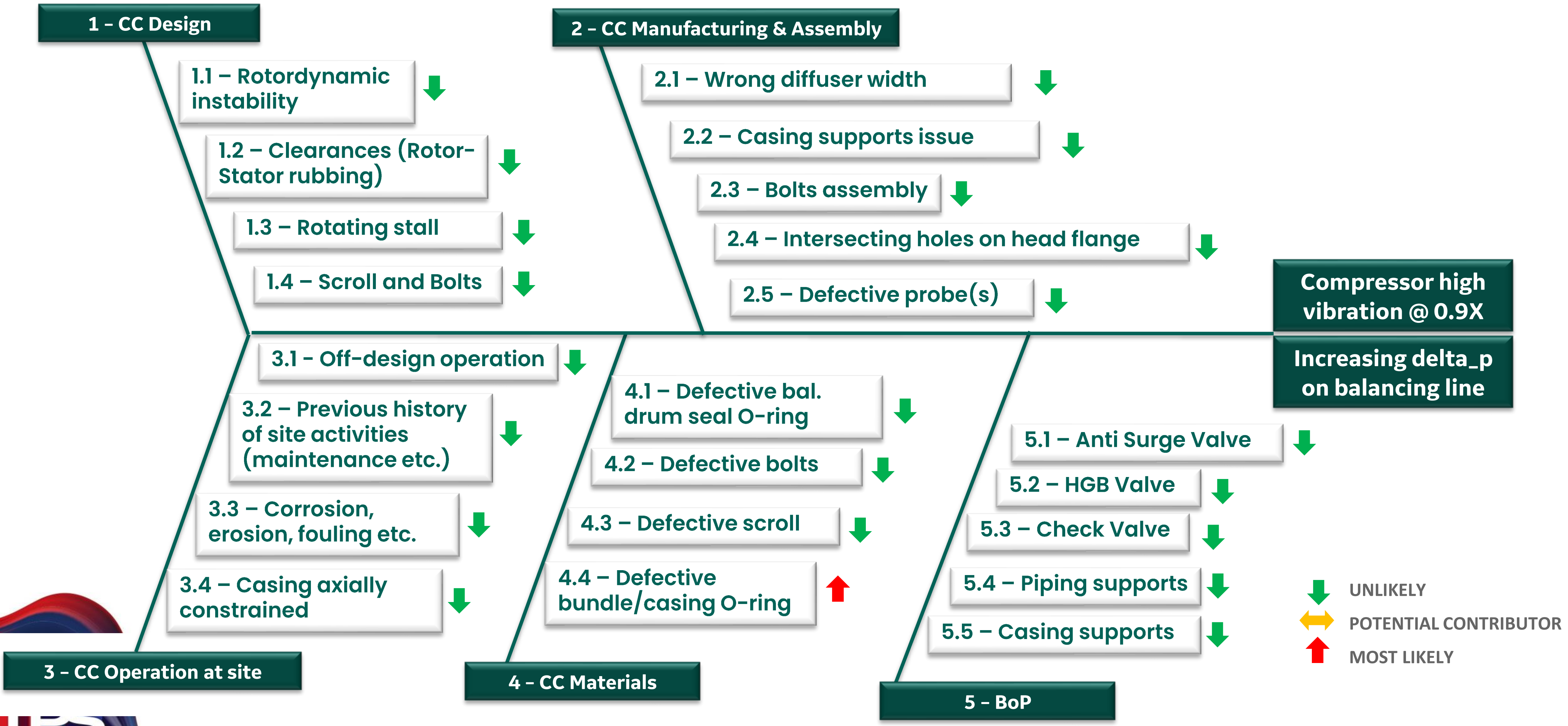
Data from site



- Unit is started up at time t0. During normal operation, at time t1 (when pressures are stabilized) the balancing line delta_p starts increasing.
- At time t2 rotor and casing vibrations suddenly increase
- Unit is tripped at t3.
- Running speed is constant from t0 to t3.



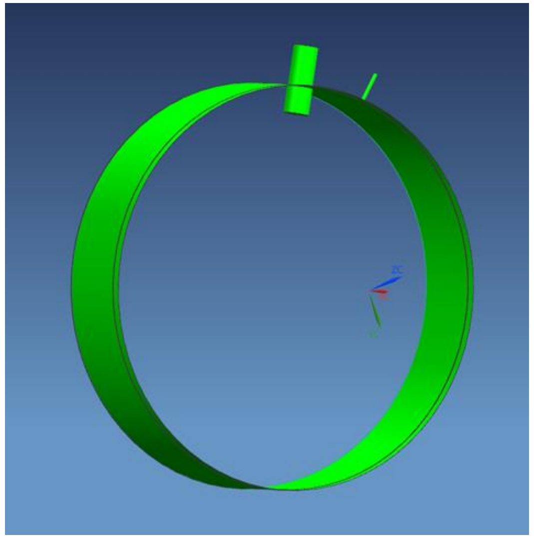
Analysis of the problem



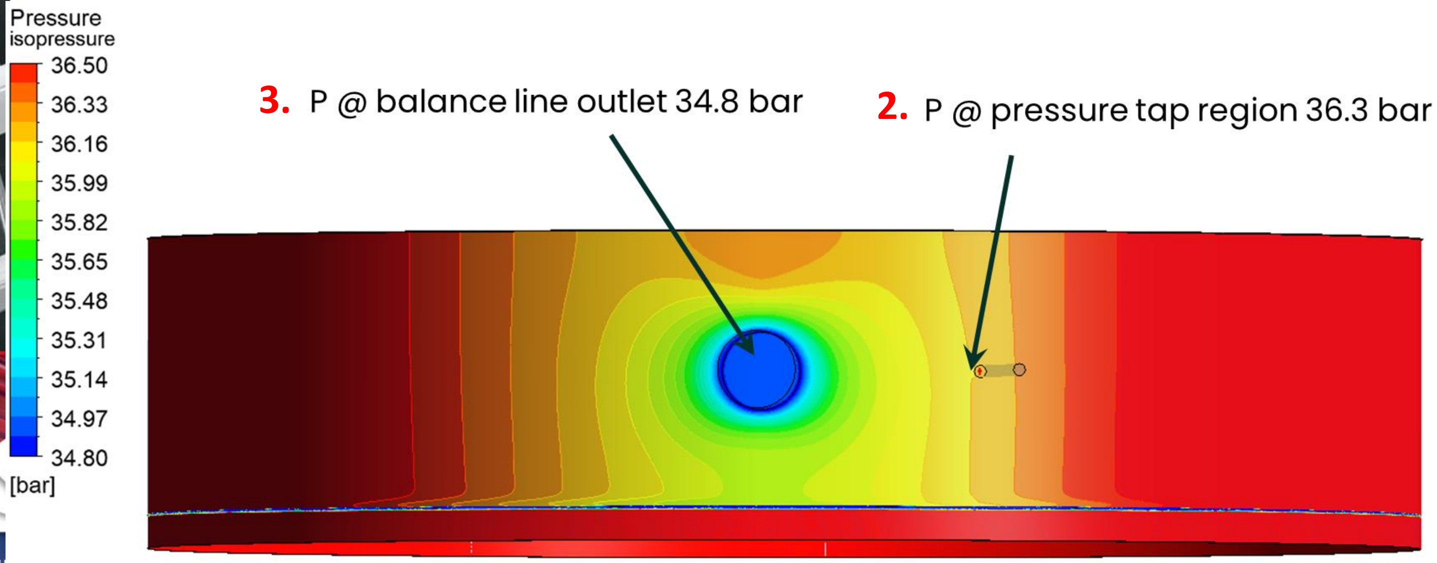
Analysis of the problem

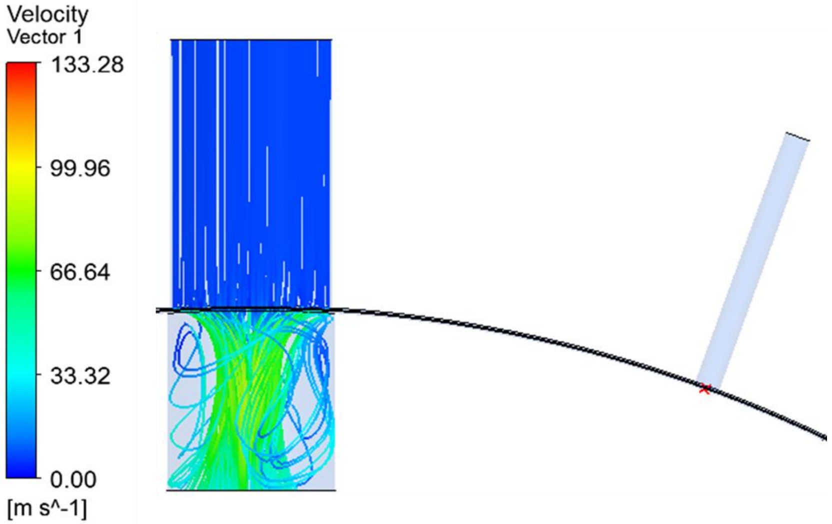
A CFD analysis was run to simulate the pressure differential between balancing line outlet **p3** and pressure tap **p2**, when the O-ring at location C is failed and therefore there is a calculated 7.1 kg/s leakage across the gap, limited by the sonic flow across location C gap (area S_C=828 mm^2).

Results show a differential pressure up to 1.4 bar, fully compatible with the DP value measured at site.



Starting from these data, a physical model was developed to evaluate the pressure distribution over bundle surface, and verify the possible link with vibration issue.

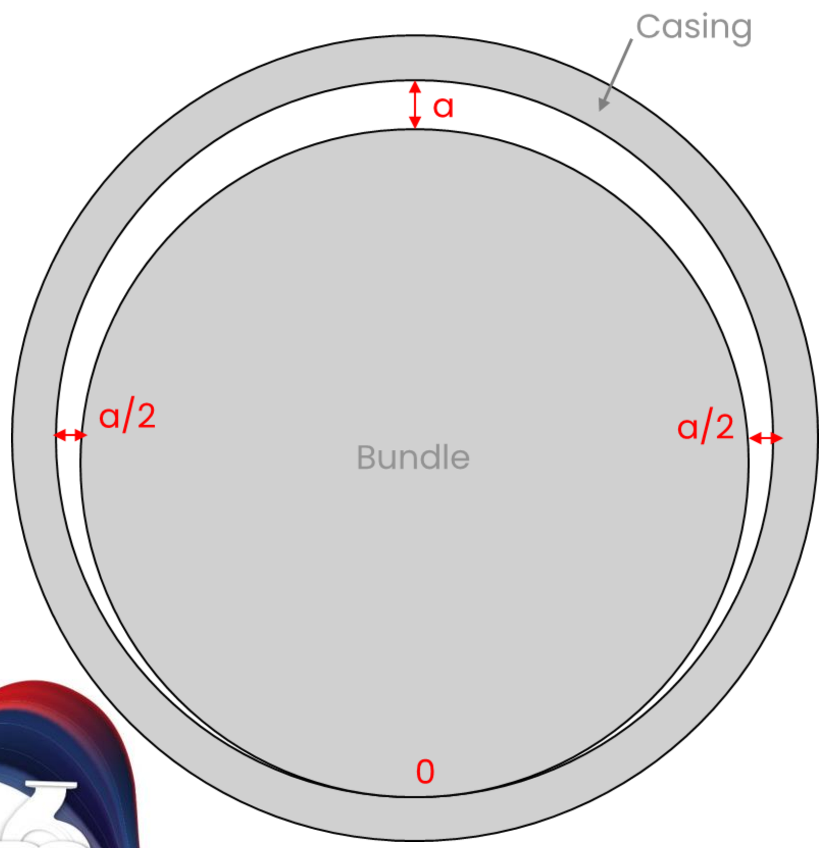




Analysis of the problem

When the O-ring is working the bundle is resting on the bottom of the casing. The casing-bundle clearance at close-gap locations is a=**0.23**mm at top, **0**mm at bottom and a/2=**0.115**mm at sides.

Gas leakage velocity is ≅ 0, due to the presence of the O-ring.



When the O-ring fails, the gas leakage is G=7.1 kg/s, limited by the sonic flow across close-gap location (area S_C=828 mm^2).

The velocity of the gas leakage is not uniform around the bundle, since it is proportional to the local gap width:

- at bundle top, = 17.2 m/s
- at bundle bottom, = 0 m/s
- at bundle sides, = 8.6 m/s

Therefore the mean velocity along the bundle surface is **v_{TOP} = 12.9 m/s** for the top half, while it is **v_{BOTTOM} = 4.3 m/s** for the bottom half.

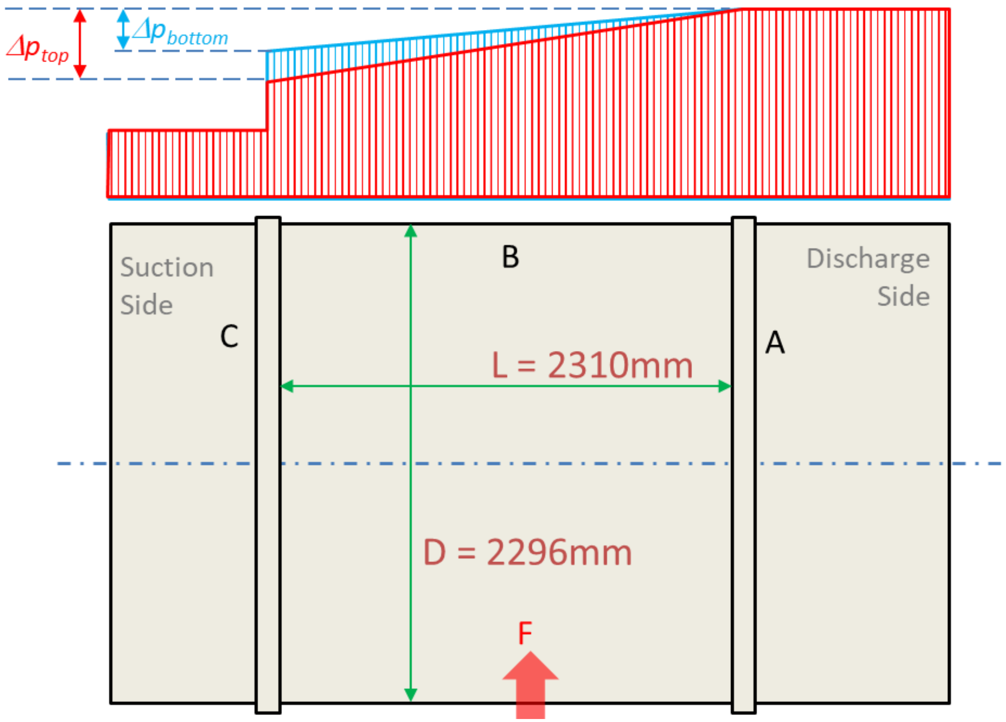
Analysis of the problem

The pressure drop Δp along L is given by:

$$\Delta p = \rho * f * (L / d) * v^2$$

Δp_{TOP} = 2.1 bar
(avg for bundle top half)

Δp_{BOTTOM} = 0.2 bar
(avg for bundle bottom half).



The resulting pressure distribution yields a net upward force $F = ½(\Delta p_{TOP} - \Delta p_{BOTTOM})LD$.
Considering a ±10% uncertainty (mainly due to *f*):

F = 46,200 to 56,400 kgf

F in the same range of bundle weight (~54,000 kg): the O-ring failure may cause the lifting of the bundle.
The lifted bundle may then vibrate according to its natural frequency, when excited by the inlet process gas. Vibration may be further increased by matching of acoustic natural frequencies of the gas volumes.

d = gap width, *f* = friction factor (0.1±0.01), *v* = gas velocity.

Analysis of the problem

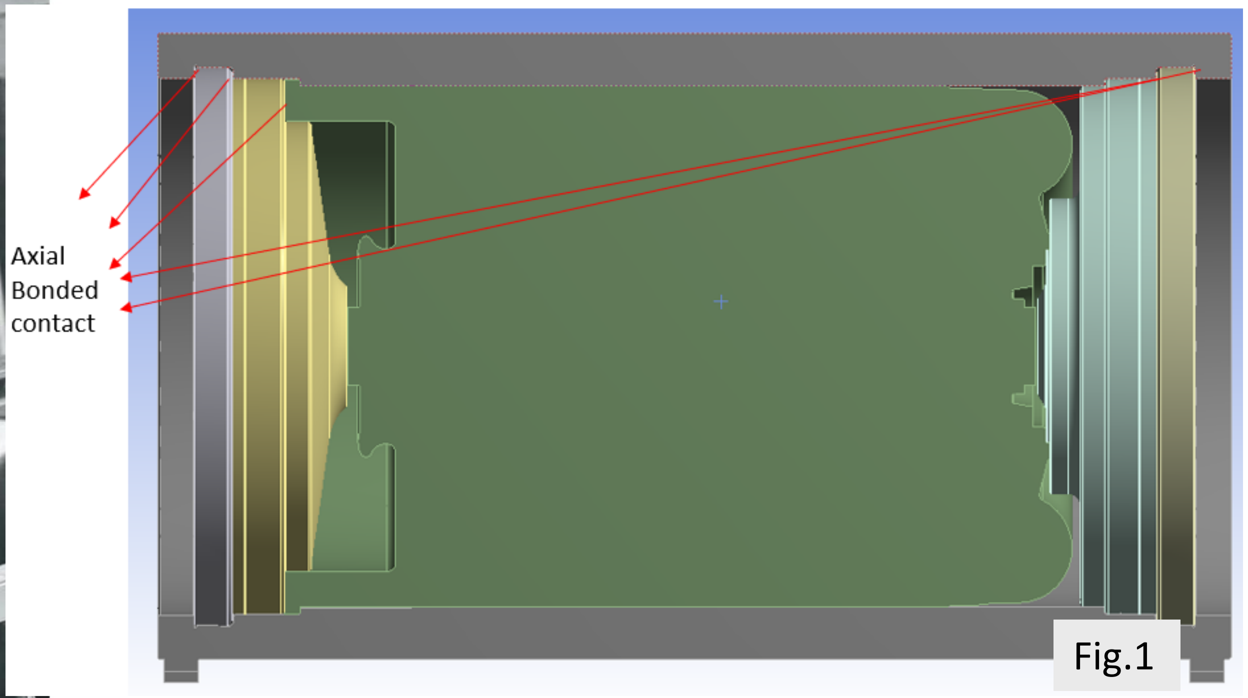


Fig.1

A FEA was run constraining the bundle, casing and head flange as in Fig.1. Constraints were also placed on casing feet, to simulate the locking on supports.

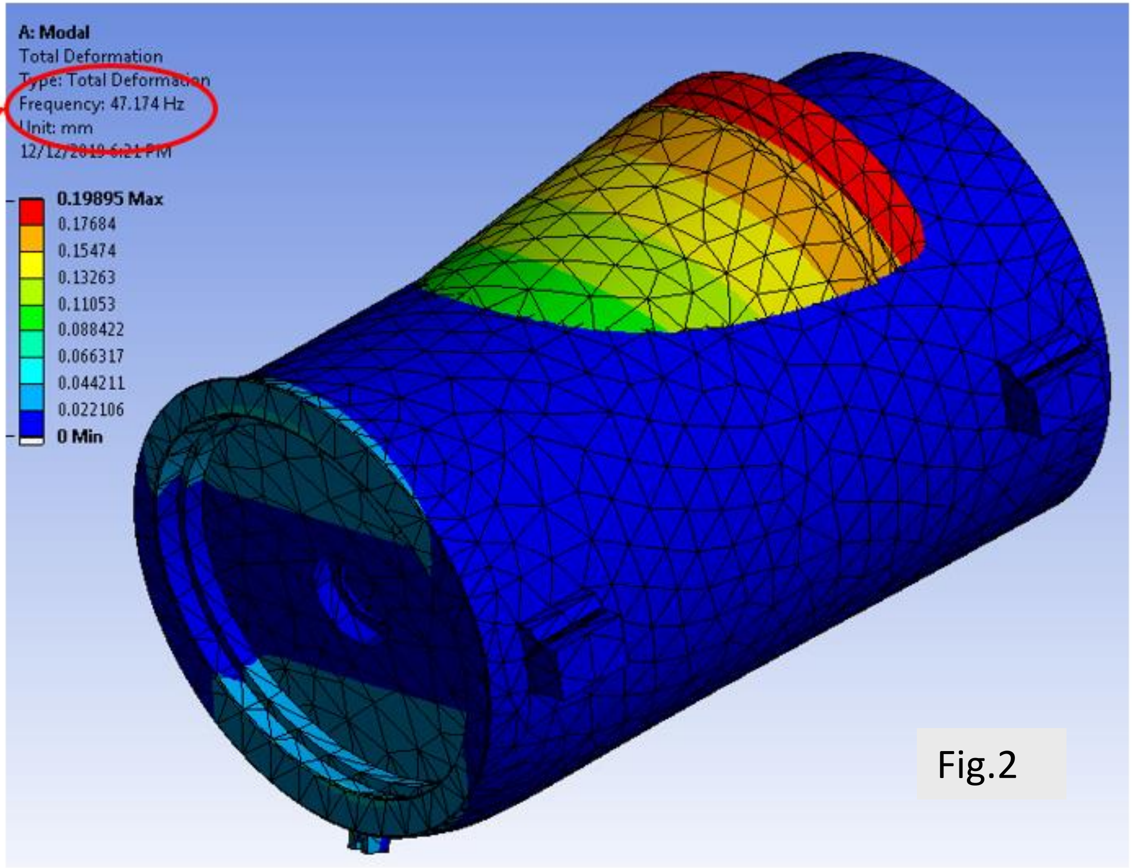
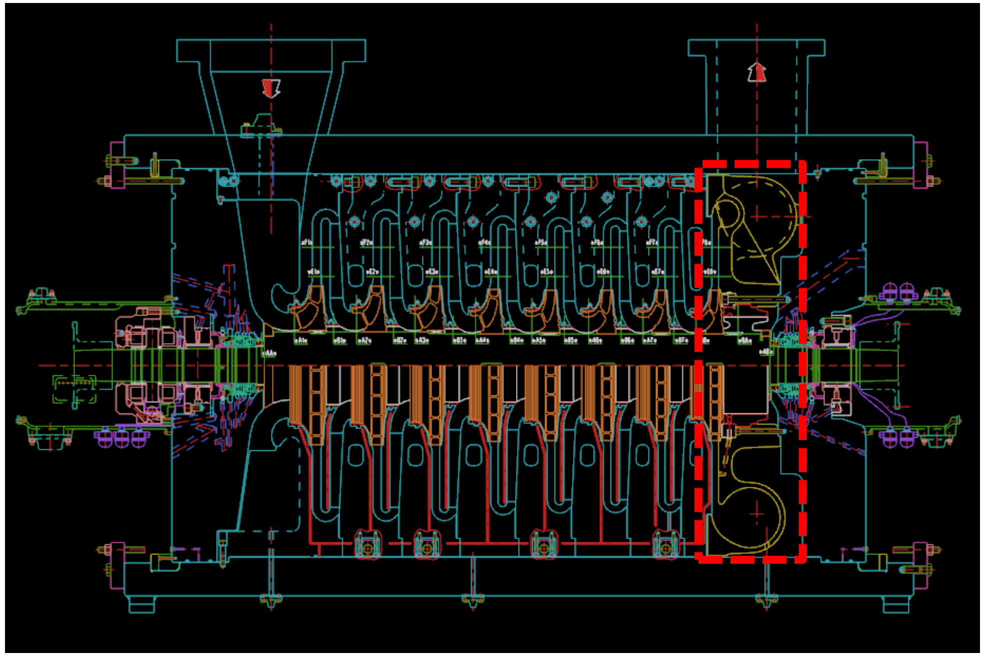


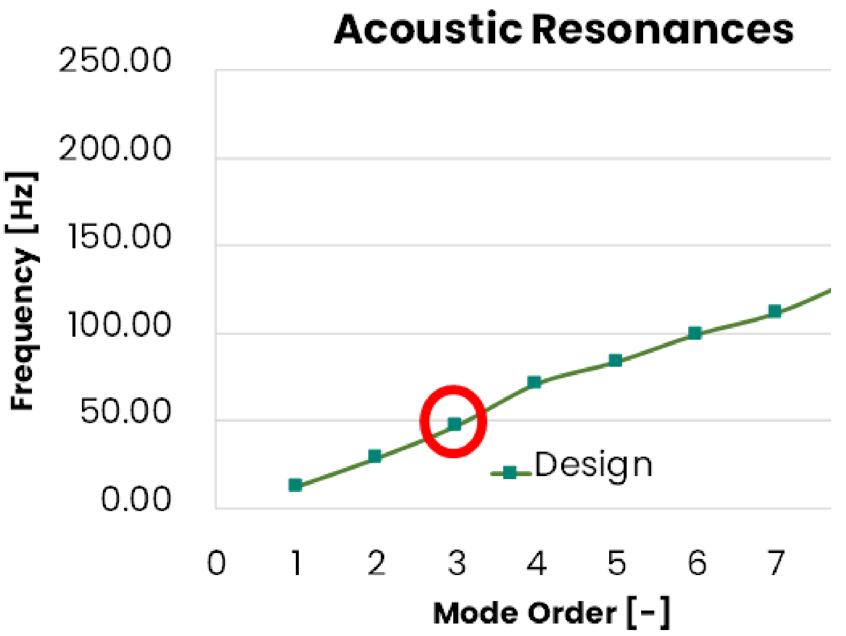
Fig.2

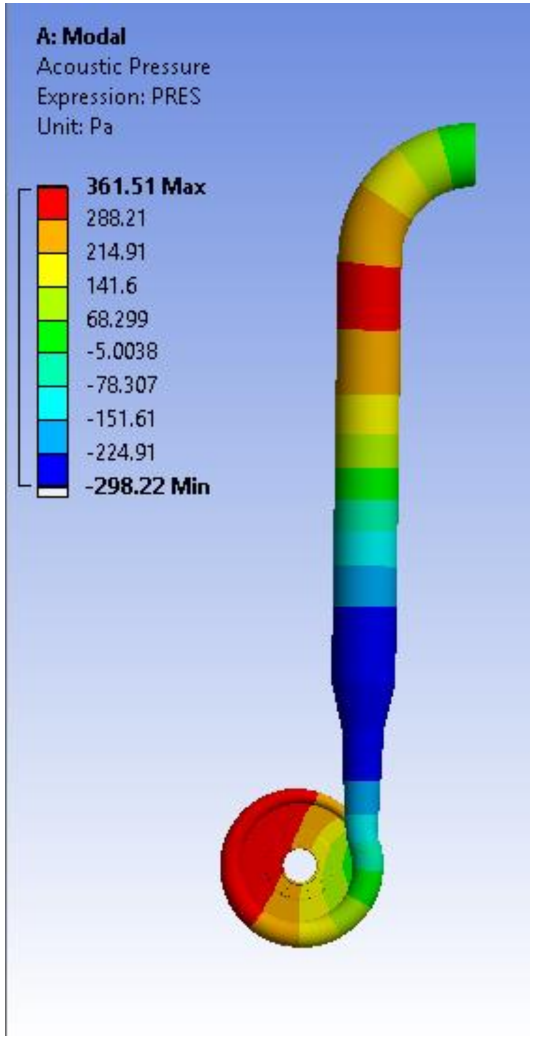
The first natural mode of the system (Fig.2) is ~50 Hz, corresponding to 0.9X of rotating speed and matching the observed frequency.

This mode is normally suppressed by the own weight of the bundle. But the uneven pressure distribution associated to the failed O-ring can lift the bundle, removing this constraint.

Analysis of the problem







- Acoustic cavities at compressor discharge were modeled by CFD: 1) discharge scroll + piping, 2) cavity btw. last impeller and diaphragm, 3) cavity btw. discharge scroll and casing, 4) balancing line.
- Acoustic analysis results show that the 1st natural mode of cavity #3 is 47 Hz, matching the vibration frequency.
  - This coincidence btw. acoustic & vibration frequencies yields an amplification of the excitation, and is the most likely reason for the loud noise detected @ compressor discharge.

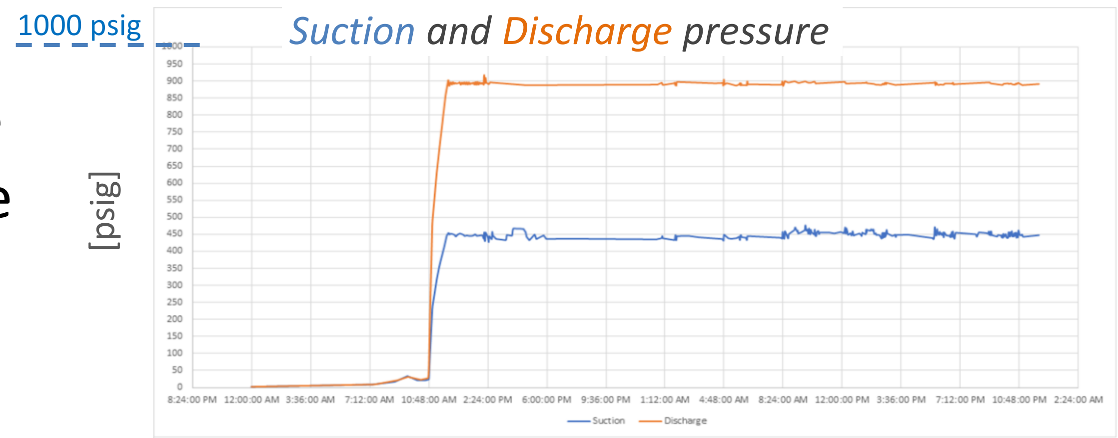
Conclusions and Recommendations

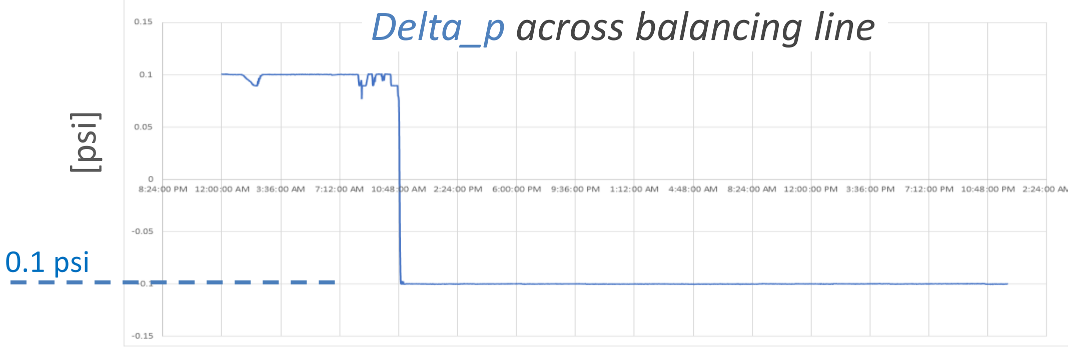
- Typical consequences of a bundle O-ring failure are 1) gas recirculation from discharge to suction, 2) minor preformance drop. The consequence observed in this case study - extremely high stator vibration - is not well documented in literature and deserves attention due to its severity.
- The physical mechanism causing the vibration is quite complex and required an extensive analysis to be fully understood. Nevertheless the final model can explain all the observed features (vibration trend, balance line dp trend, vibration frequency, behavior at subsequent startups, compressor inspection outcome...)
- The solution included the replacement of the original casing O-rings (in FKM 75) with new ones in FKM 90 Special, resistant to explosive decompression and validated through dedicated laboratory tests.

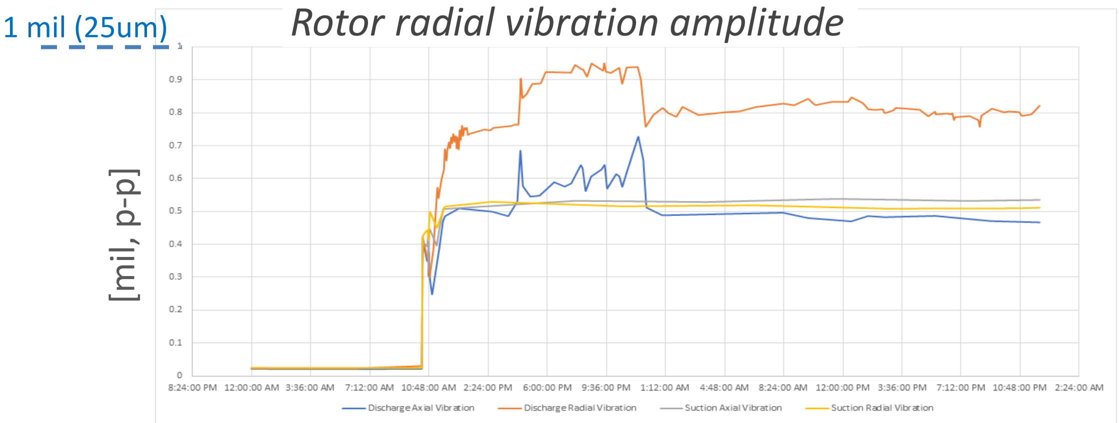
Results from Field Implementation

The failed O-ring was replaced with a new one (resistant to explosive decompression) and the unit was reassembled and restarted in Feb. 2020.

The vibration level of compressor rotor and stator was stable and did not highlight any anomaly since then.





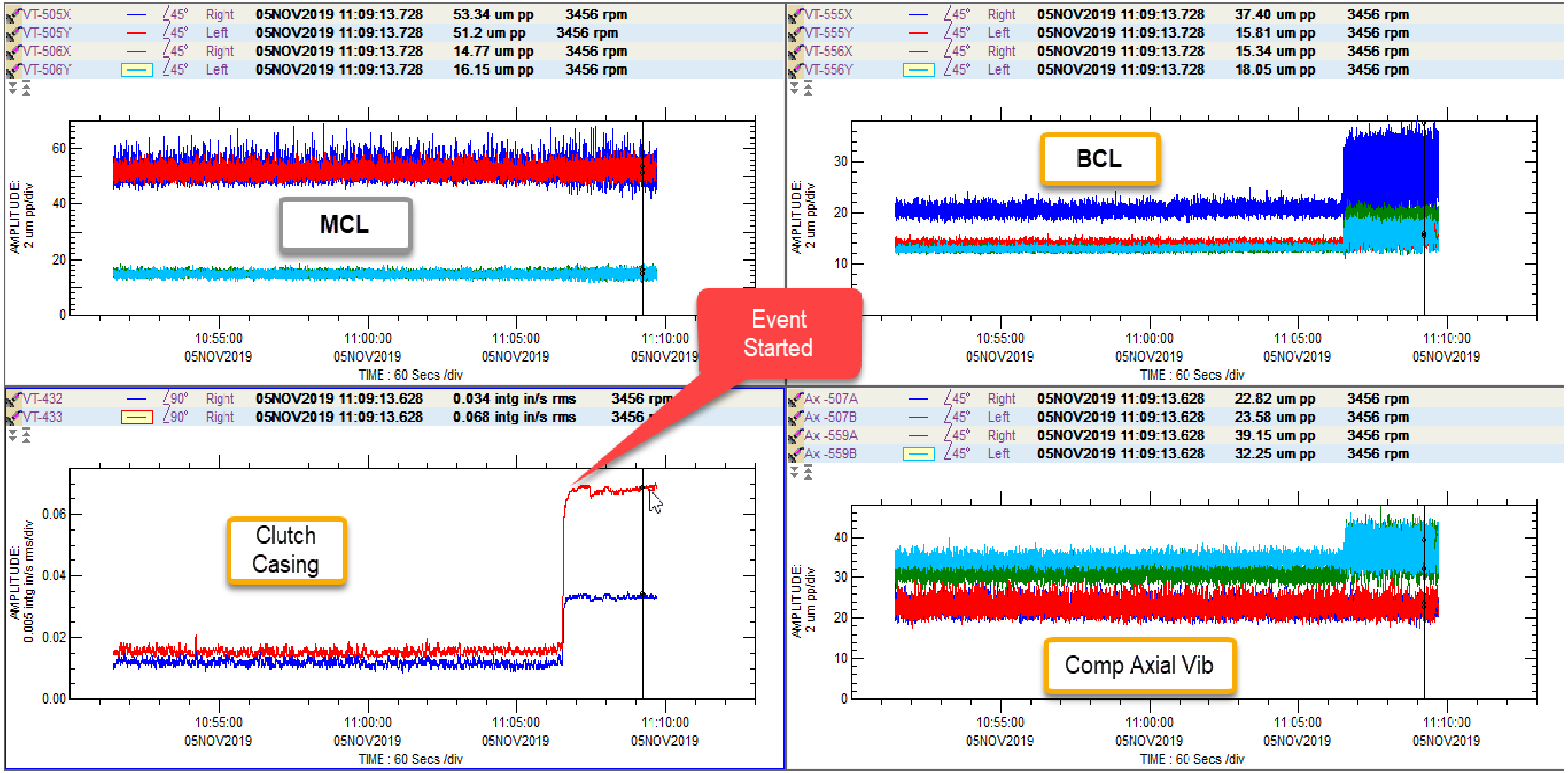


Lesson Learned

- Compressor casing and end wall gaskets are critical items, that must be carefully designed, selected and tested to ensure their integrity over the required operating life. In particular the risk of explosive decompression shall be mitigated. The max depressurization rate of the compressor shall be carefully assessed during the design phase.
- Evaluate the installation of seismic vibration transducers on compressor casing, as they can detect vibration issues that are not adequately monitored by rotor radial & axial vibration probes. Additional casing vibration monitoring could be particularly useful on unmanned/remote plants.
- Whenever a sudden increase of stator vibration is observed on a centrifugal compressor and is unrelated to process parameter variation, this specific flow-induced vibration issue shall be verified as potential cause in a Root Cause Analysis.

Back Up

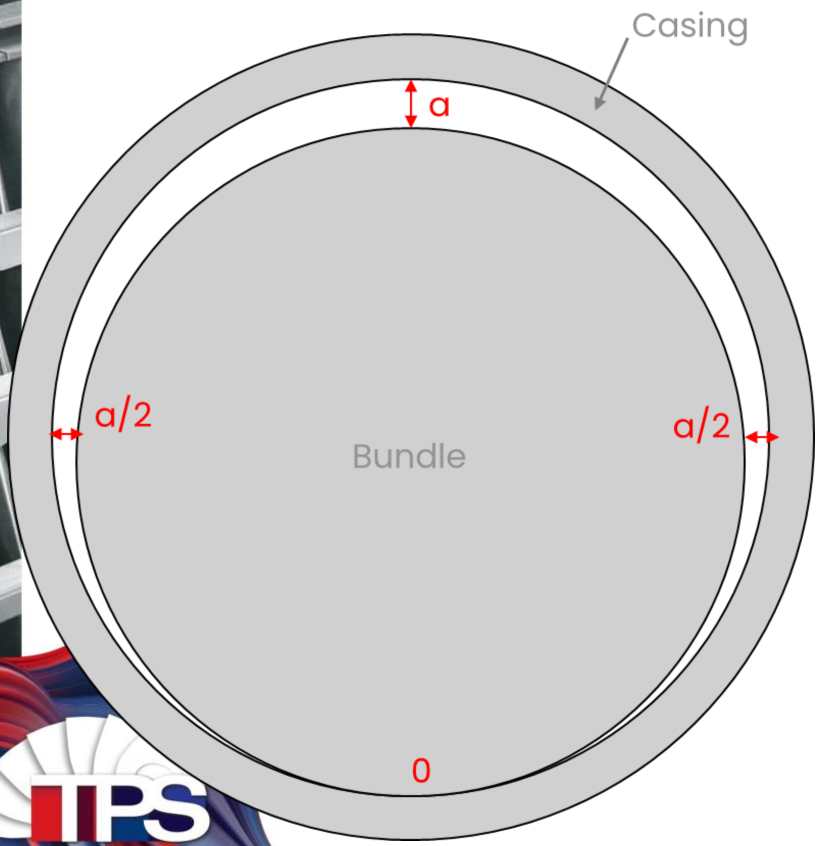
Investigation Methodology - Vibration



Analysis of the problem

At startup the bundle is resting on the bottom of the casing due to gravity. Therefore the casing-bundle gap at locations A & C is a=**0.23**mm at top, **0**mm at bottom and a/2=**0.115**mm at sides.

The gap at location B is almost constant around the bundle: from 3.01 to 3.25mm.



When the O-ring is failed, the gas leakage is limited by the sonic flow across location C gap (area S_C=828 mm^2). The gas flowrate across C gap is G=7.1 kg/s.

Across B gap (area S_A=22577 mm^2) this leakage corresponds to is Q=0.194m^3/s and therefore to an average velocity of Q/S_A=8.6 m/s.

The local gas velocity across A gap is variable with angular position. It is roughly proportional to location A gap width at the same angle:

- at bundle top, = 8.6 * 0.23 / 0.115 = 17.2 m/s
- at bundle bottom, 8.6 * 0 / 0.115 = 0 m/s
- at bundle sides, 8.6 * 0.115 / 0.115 = 8.6 m/s

Therefore the average velocity along B for the top half of the bundle is v_{TOP} = (17.2+8.6)/2 = **12.9 m/s**, while for the bottom half of the bundle it is v_{BOTTOM} = (0+8.6)/2 = **4.3 m/s**.