ASIA TURBOMACHINERY & PUMP SYMPOSIUM
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Investigation of Process Gas compressor shaft vibration phenomena

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Slide 3: Short text of an Abstract

This paper presents details of investigation results of issues observed during plant start-up on a centrifugal compressor. Compressor was operated with air/ nitrogen during start-up and high shaft vibration (approx. 75 um) were observed on DE side of compressor accompanied by high levels of coast down vibration levels (exceeding alarm levels). This paper presents subsequent detailed rotor dynamics analysis to understand root cause of the high vibrations.

Investigation for Process Gas HP compressor shaft vibration phenomena

- This paper presents details of investigation results of issues observed during plant start-up on a centrifugal compressor and subsequent detailed rotor dynamics analysis to understand root cause of the high vibrations.
- This is the main process gas compressor (PGC) in the cracker and is a 5 stage 3 body machine driven by a steam turbine. The compressor train is rated for 55 MW and operates at 3800 rpm. The high vibration problem was observed on HP (high pressure) case of the machine which is rated for 22 MW approximately.
- The problem was encountered during nitrogen operation before commercial operation with cracked gas at site. PGC compressors was operated with air and higher shaft vibration (approx. 45 um) were observed on DE side of HP compressor, these were higher than factory tested values (approximately 10μm during mechanical run). After a few months, PGC compressors were operated with nitrogen and higher shaft vibration (approx. 75 um) was observed on DE side of HP compressor however the vibration levels exceeded alarm levels during coast down of the machine. For both (air and nitrogen) operations, strong correlation with operating conditions (operating pressure and temperature) and shaft vibration level was observed.
- Compressor casing was opened for inspection and some rusts on the rotor and casing was found. Rotor residual unbalance check was performed at local workshop. The residual unbalance was found higher but it was not adequately high to cause higher vibration level and also did not explain reason for high vibrations during coast down of the machine. Root cause of rusts was established as exposure to hydrotest water during piping hydro test at site.
- Detailed inspection of the rotor was carried out at MHI Hiroshima facility and various inspections were carried out (visual, dimension, low speed and high speed balance check). Analytical work for rotor dynamics analysis, FEM analysis for impeller gripping force and FEM analysis for rotor thermal bending etc were performed.
- The rotor dynamics analysis including thermal bending of the rotor caused by the non-uniform heat transfer could simulate the shaft vibration phenomena qualitatively. This paper presents details of analysis, observations and techniques used for establishing effect of rust on a rotating component.

History

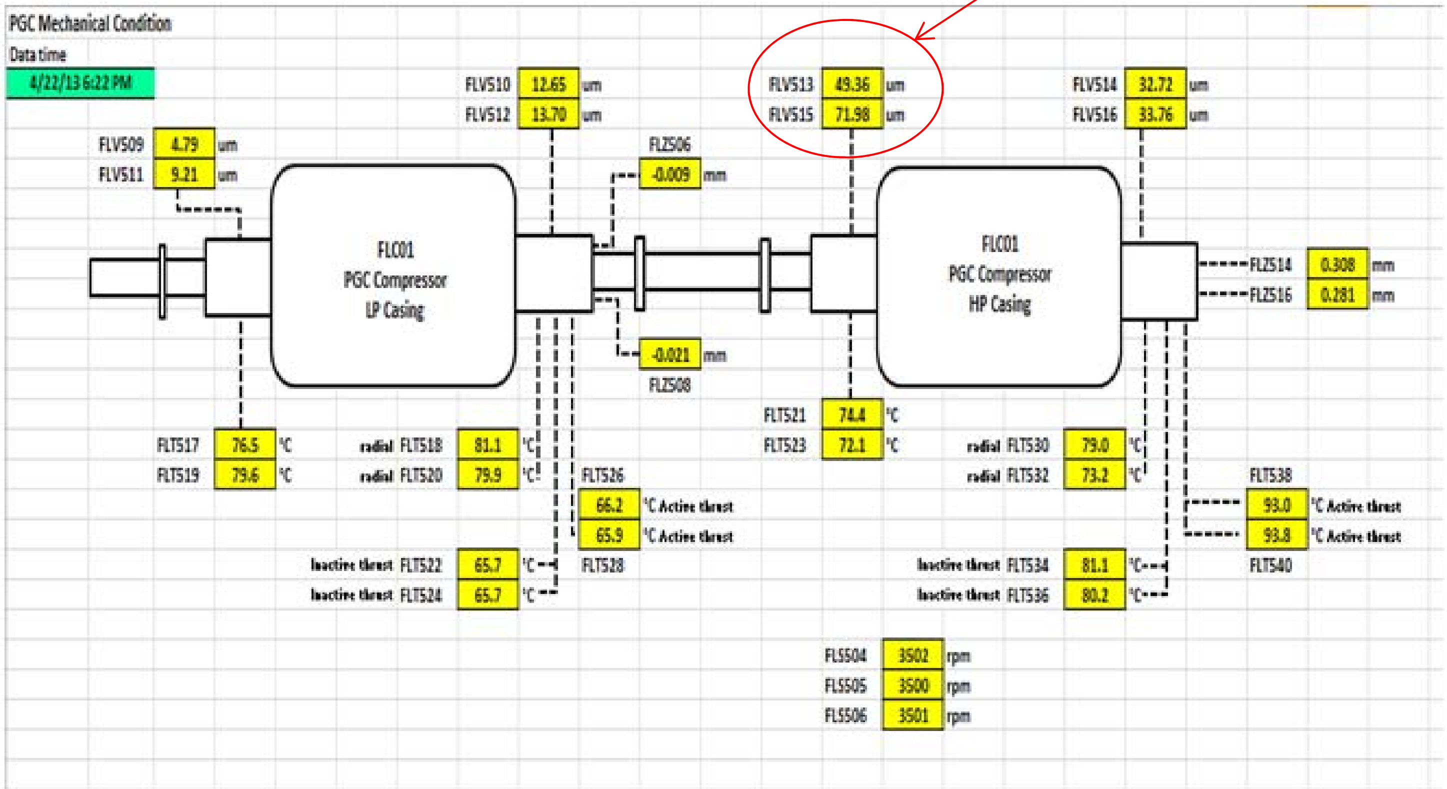
Run 1 (Working)	November , 2012:	PGC Compressors were operated with air, and shaft vibration of 45 um was observed on DE side of HP compressor.
	December , 2012:	Detailed operation data for PGC reviewed. This was not normal operation and compressor operated at high discharge temperature and different gas composition. High vibration was assumed to be caused by operating temperature differences than design.
Run 2 (Working)	February , 2013:	PGC Compressors were operated with air, and shaft vibration was same level as Nov. 2012.
Run 3 (Working)	April , 2013:	PGC compressors were operated with nitrogen, and shaft vibration of 75 um was observed on DE side of HP compressor, Phase change noticed during shutdown.
	April 29th , 2013:	Rotor replacement carried out. - no abnormal rubbing was identified. - water line was observed.

Run 4 (Spare)	May 5th , 2013:	PGC compressors re-started.
	May 9th , 2013:	The vibration level was 19μm
	May 10th , 2013:	The vibration level was 13μm (close to design operating conditions).

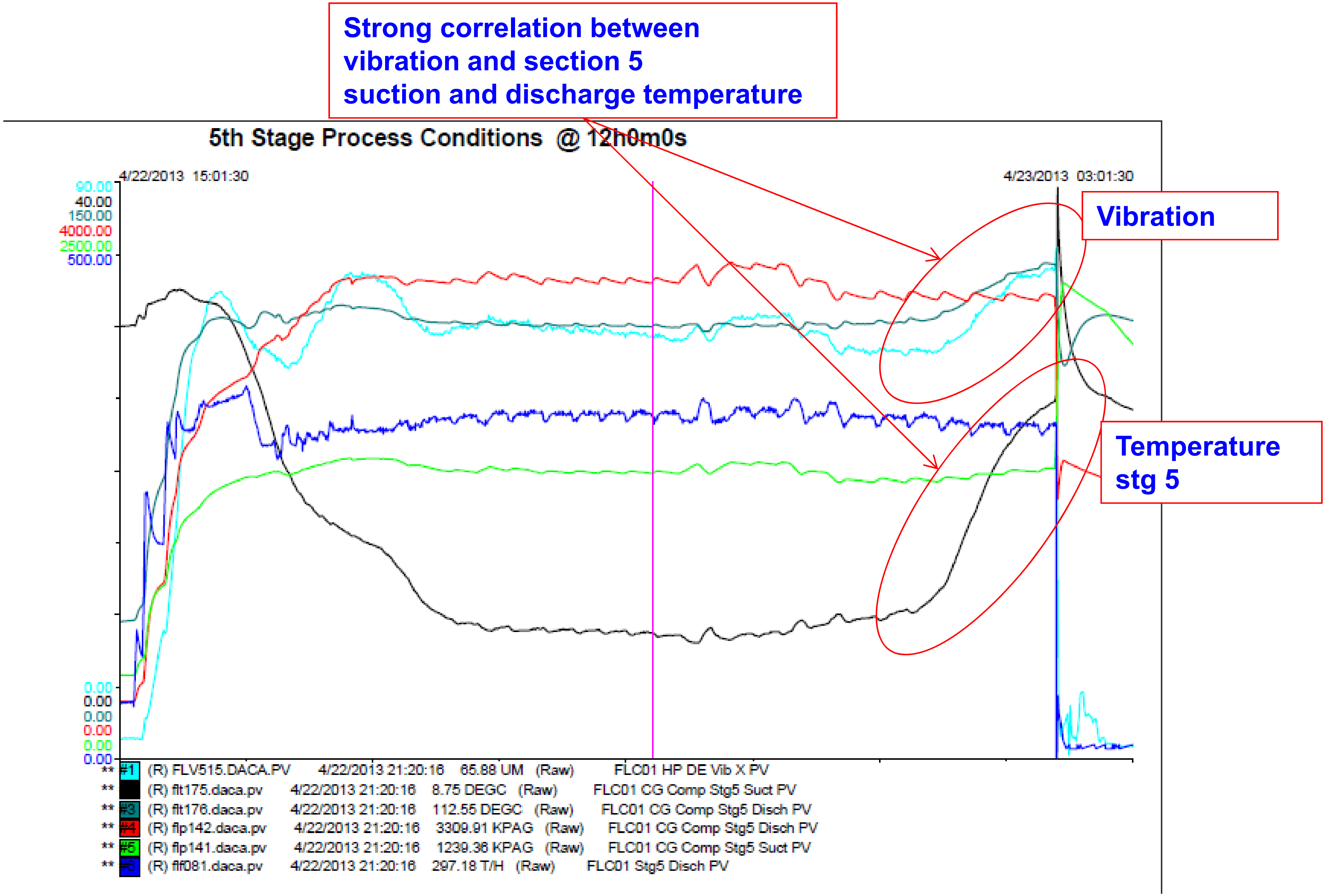
Operating data of working rotor – N2 run

April 22nd. 2013:
Working rotor

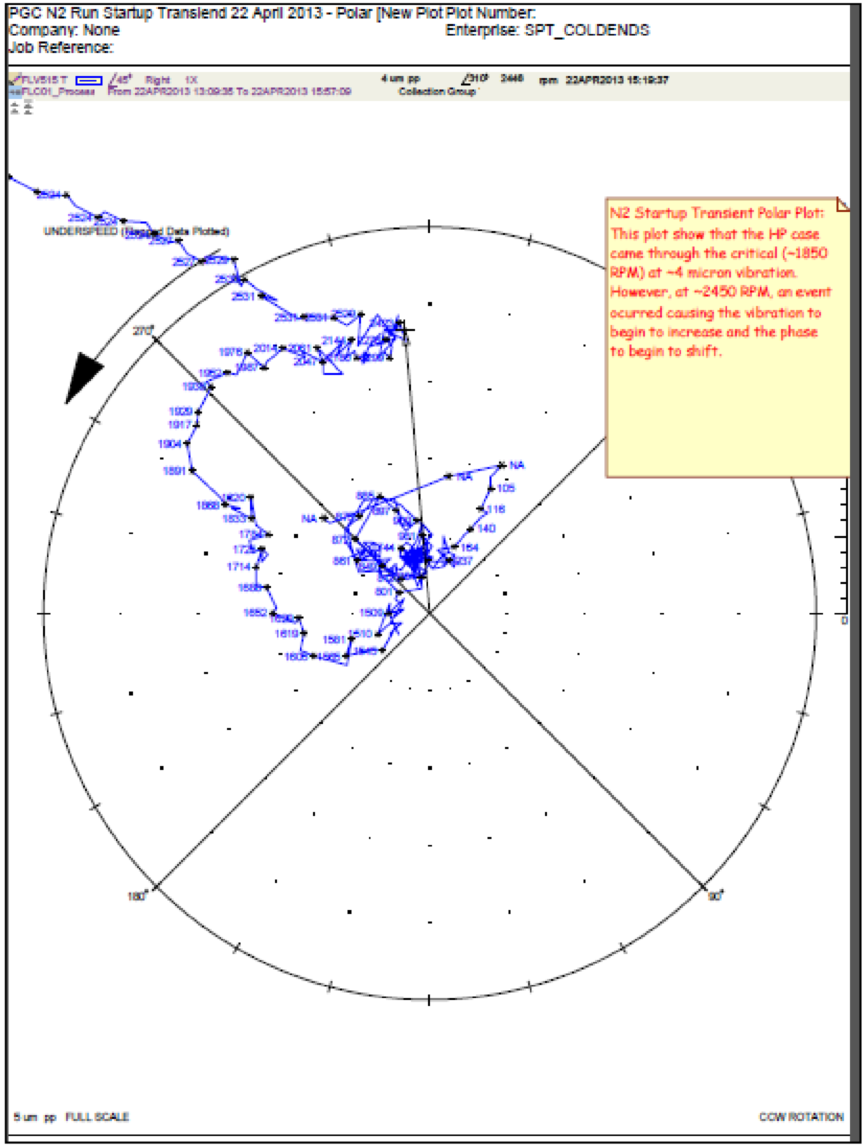
DE side vibration is higher than NDE side vibration for both rotors (working and spare rotors).

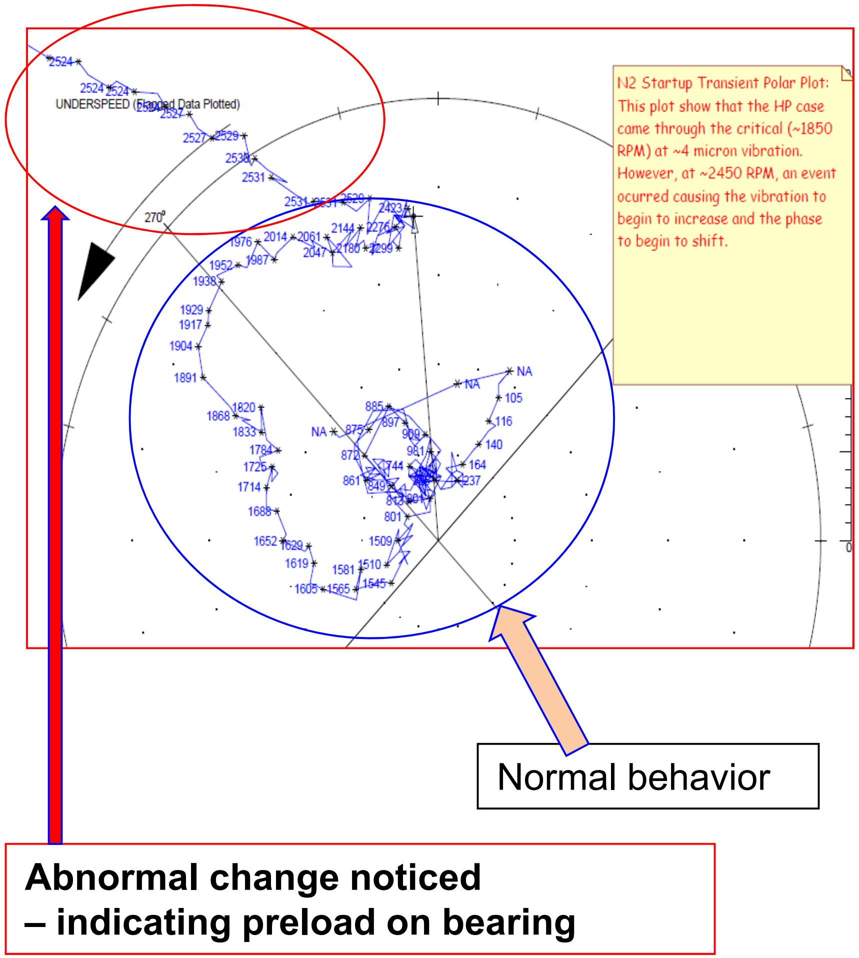


Trend data of working rotor – N2 run

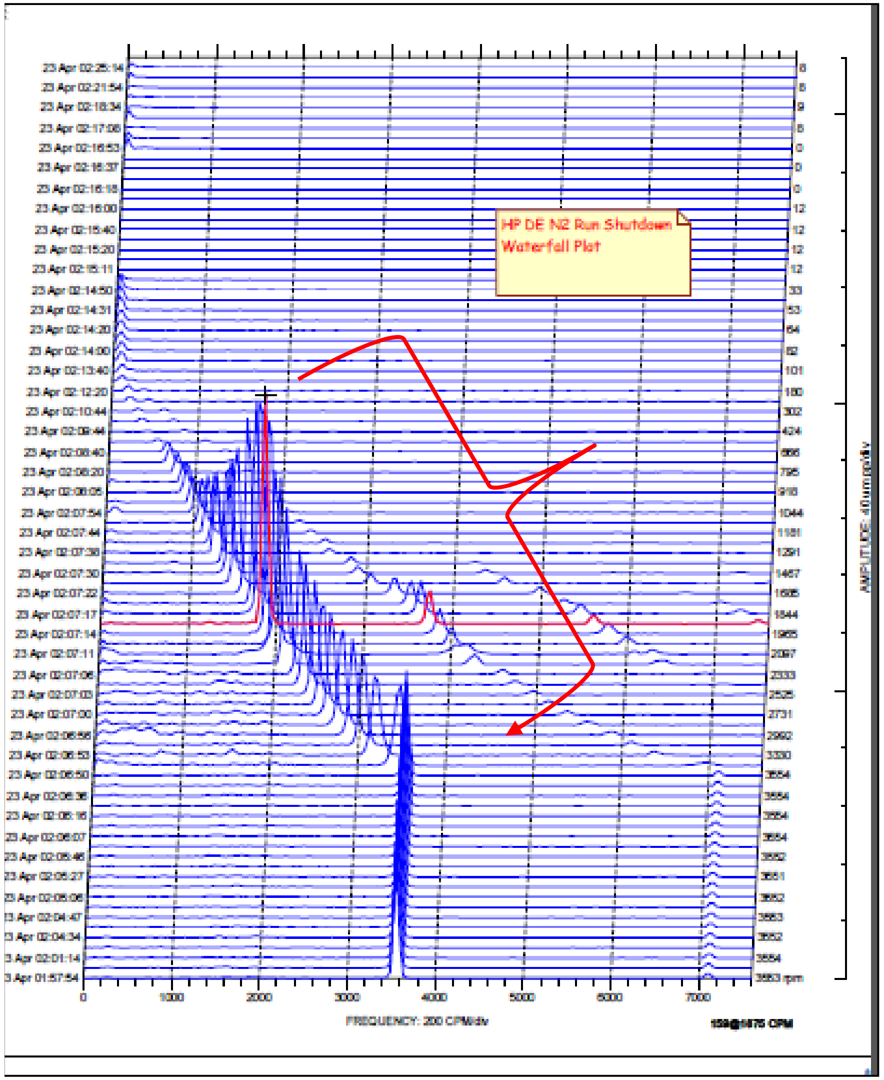


Transient polar plots of working rotor – N2 run





Shutdown plot of working rotor – N2 run



Abnormal waterfall plot for shutdown – Vibrations up to 140 microns observed during shutdown

Rotor replacement initiated

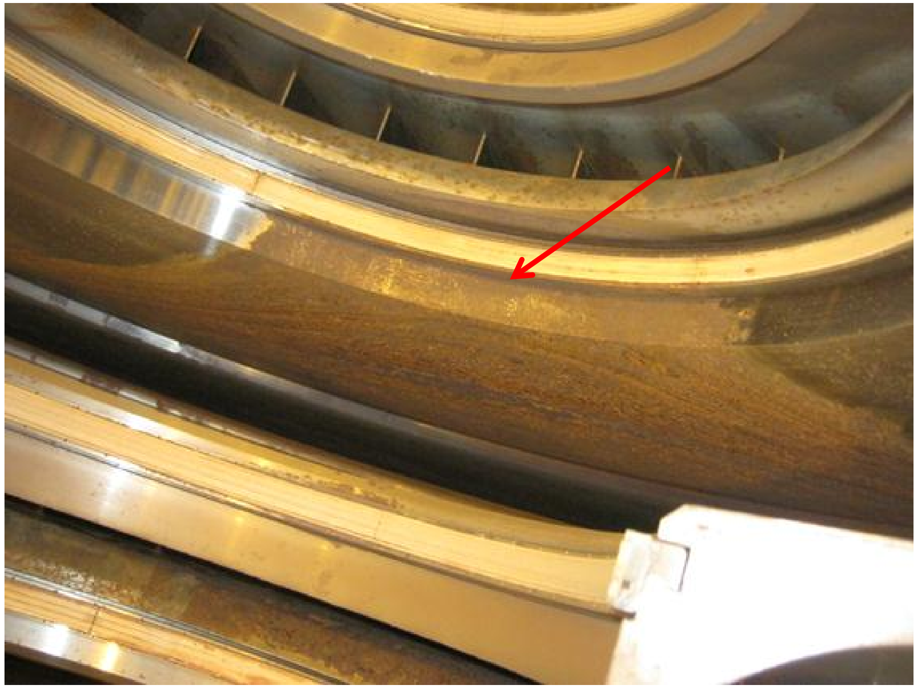
Rotor is replaced - Inspections and Observations

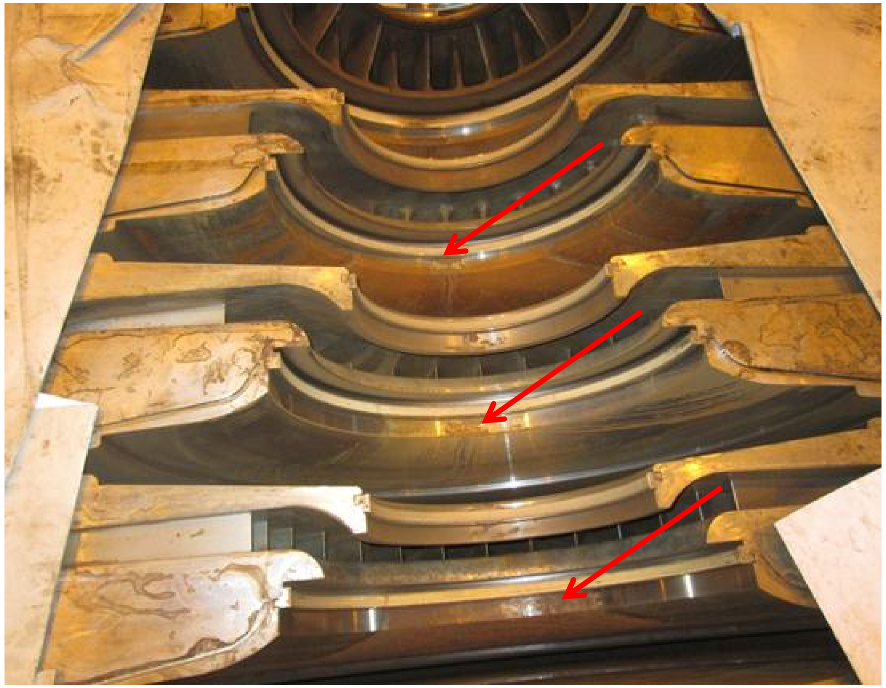
Onsite as-found conditions

- Slight rust observed on rotor; otherwise rotor still looked good from outside

Rotor Inspections and Observations

Onsite as-found conditions





- Significant rust observed on stationary diaphragms
- Water mark visible and indicated water accumulation to shaft centerline at 5th stage section

Rotor Inspections and Observations

The removed rotor was sent back to MHI for detailed inspection and repair

Workshop Inspection Scope:

•Both high and low speed balancing check was conducted

•Rotor was visually inspected and stack-up dimensions checked

•Impellers were de-stacked for detailed inspection

•Dimensional checks on all components were performed

•EM witnessed as-found conditions of shaft and impellers and conducted joint RCFA with MCO engineers

Rotor Inspections and Observations

Results of Detailed Rotor Inspection

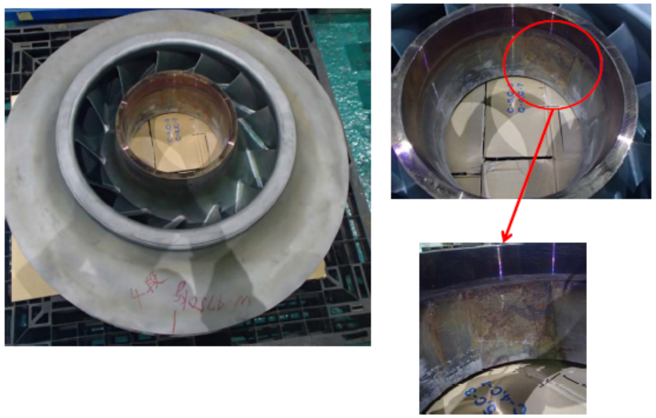
- Low speed balancing check
  - Residual unbalance exceeded value of API 617 at low speed balance check (highest was 39.3gm Vs 1.35gm during original shop test)
- High speed balancing check
  - Shaft vibration was less than 25um during high speed balance check at Max Continuous Speed. However, it was higher than the original recorded value during manufacturing (20um Vs 14um during original shop test)
- Visual inspection and NDT
  - No major abnormalities found
- Stack-up dimensions
  - 1^{st}, 2^{nd} and 4^{th} impellers' positions was slightly different from the design value.
- Impellers were de-stacked for detailed inspection
  - Rust sediments were observed accumulated within the clearances under the 4^{th} to 8^{th} impellers.
  - 4^{th} and 8^{th} impeller had most severe rust accumulation
  - Process stage 4 (1^{st} to 3^{rd} impellers on the rotor) impellers were found to be in good condition
- Dimensional checks
  - 2^{nd} impeller, balance piston and 2 shaft sleeves were found to be oversized in the bores, but MCO evaluated the interference fit and gripping force to be still acceptable even with the oversize.

Rotor Inspections and Observations

✓ Heavy rust was observed around HP section (4th to 8th impeller) mainly. (Stage 1-3 is 17-4PH. Stage 4-8 is SNCM 431.)

Compressor shaft

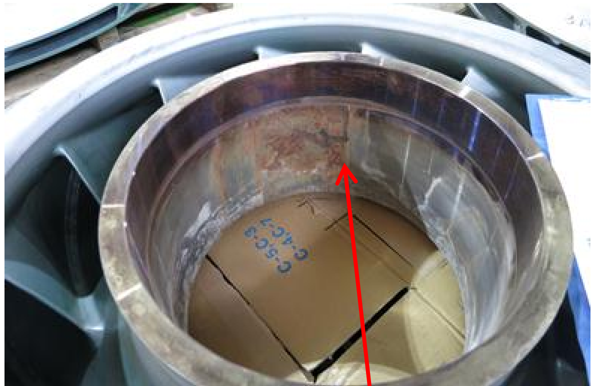
4th impeller



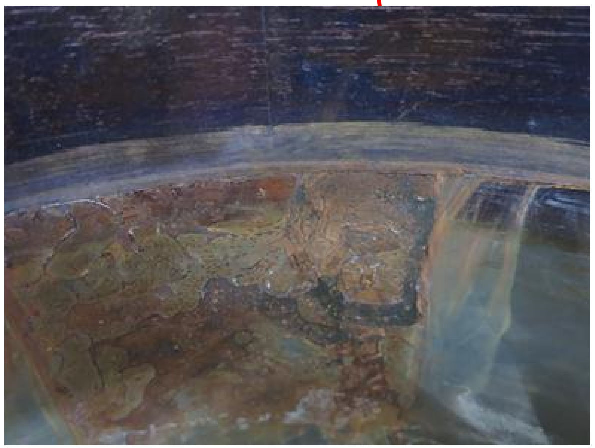
8th impeller

Rotor Inspections and Observations

Rotor as-found condition after disassemble



Under 4th impeller



4th Impeller and shaft

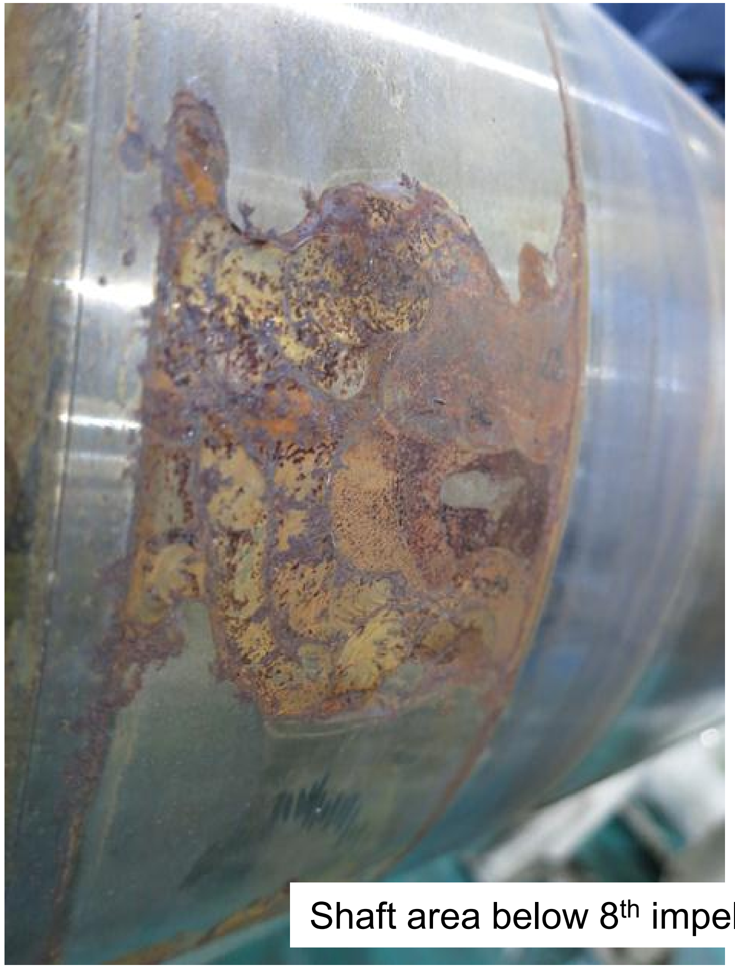
Deep scratch marks across the width of the impeller created during impeller removal. This indicated the presence of hard particles within the clearances between the shaft and impeller

Rotor Inspections and Observations

Rotor as-found condition after disassemble

8th Impeller and shaft

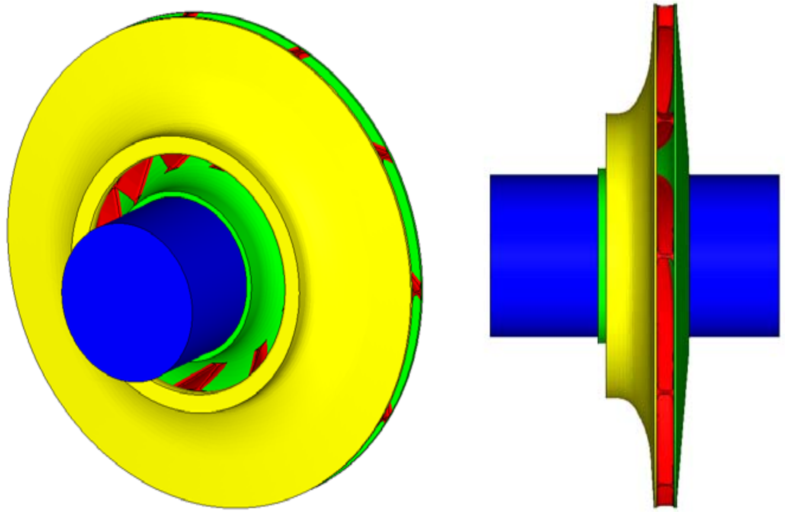
Under 8th impeller



Shaft area below 8th impeller

FEM Analysis for impeller grip condition

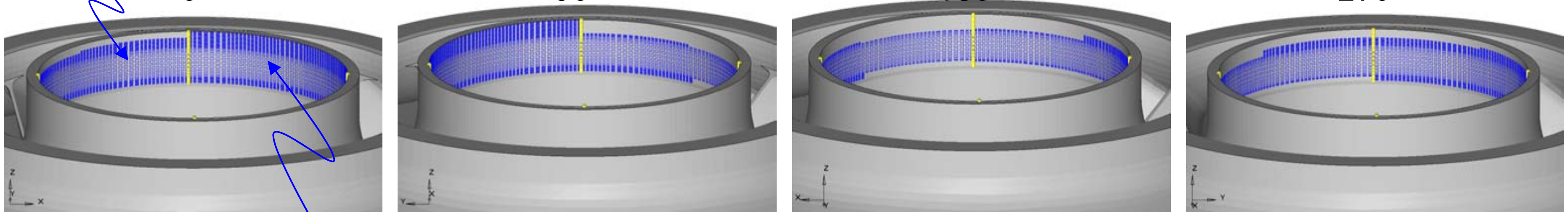
Item	Analysis condition
Objects	Shaft and 5th Impeller
Speed	3982rpm(MCR)
Load	Centrifugal force + Shrink fit pressure
Assumptions	・Recorded dimension was used. ・Color distribution was considered as contact area. (Area is 76.6% of design.) ・Thermal expansion was not considered.

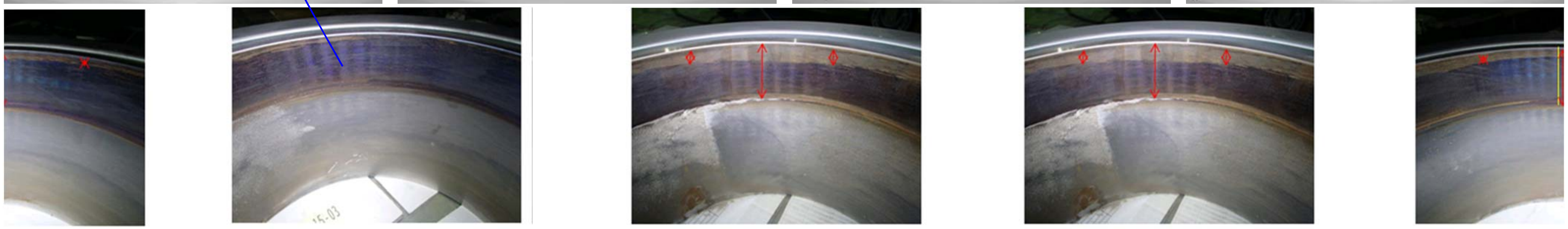


Analysis model

Modelled Contact area

0° 90° 180° 270°





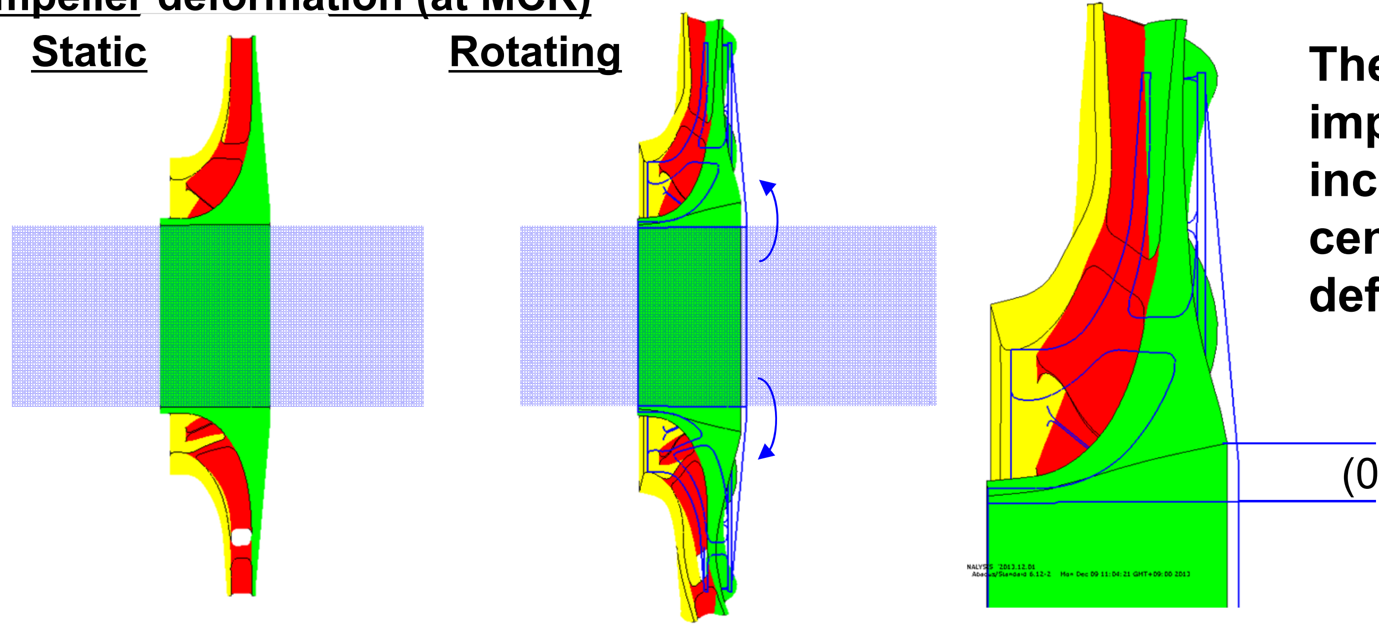
As found condition of contact area

FEM Analysis for impeller grip condition

FEA performed by MHI indicating the estimated deformation of a single impeller at Max Continuous Speed

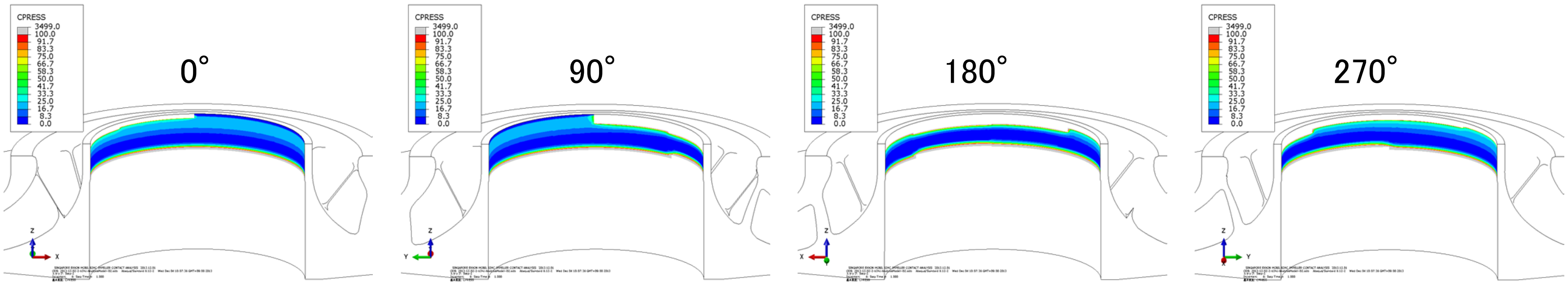
<u>Impeller deformation (at MCR)</u>

<u>Static</u> **<u>Rotating</u>**



The gap between impeller and shaft is increased by impeller centrifugal deformation.

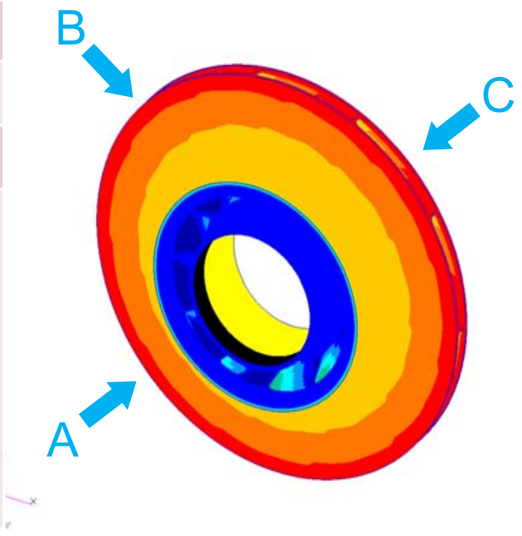
<u>Contact pressure distribution (at MCR)</u>



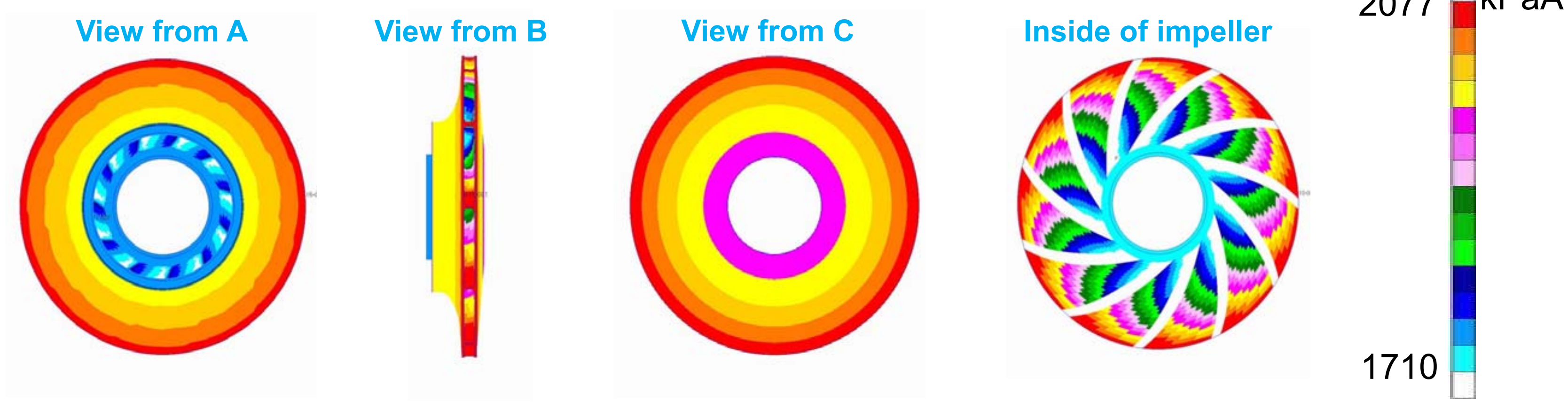
FEM Analysis for impeller grip condition

Second FEA model to estimate impeller deformation under operating pressure and speed

Item	Analysis condition
Objects	Shaft and 5^{th} Impeller
Speed	3982rpm(MCR)
Load	Centrifugal force + Shrink fit pressure + Gas pressure
Assumptions	・Recorded dimension was used. ・Color distribution was considered as contact area. (Area is 76.6% of design.) ・Thermal expansion was not considered. ・Suction pressure of 5^{th} impeller : 1710kPaA(*) Discharge pressure of 5^{th} impeller : 2077kPaA(*) (*) Estimated as per site operation data

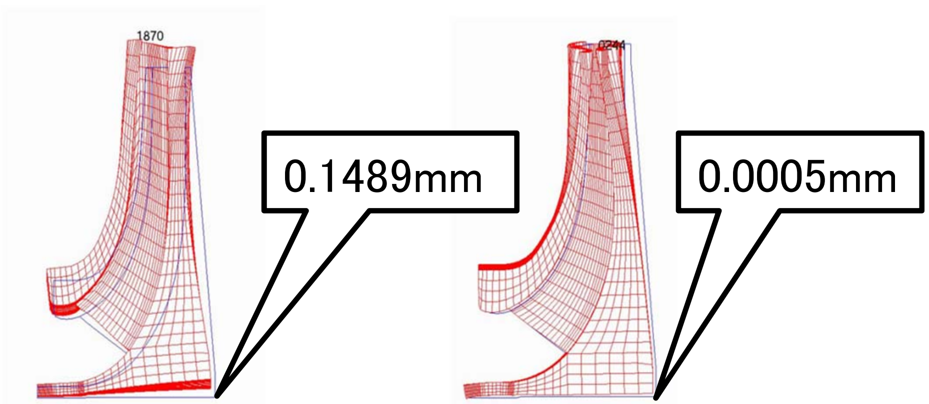


Boundary condition for pressure distribution



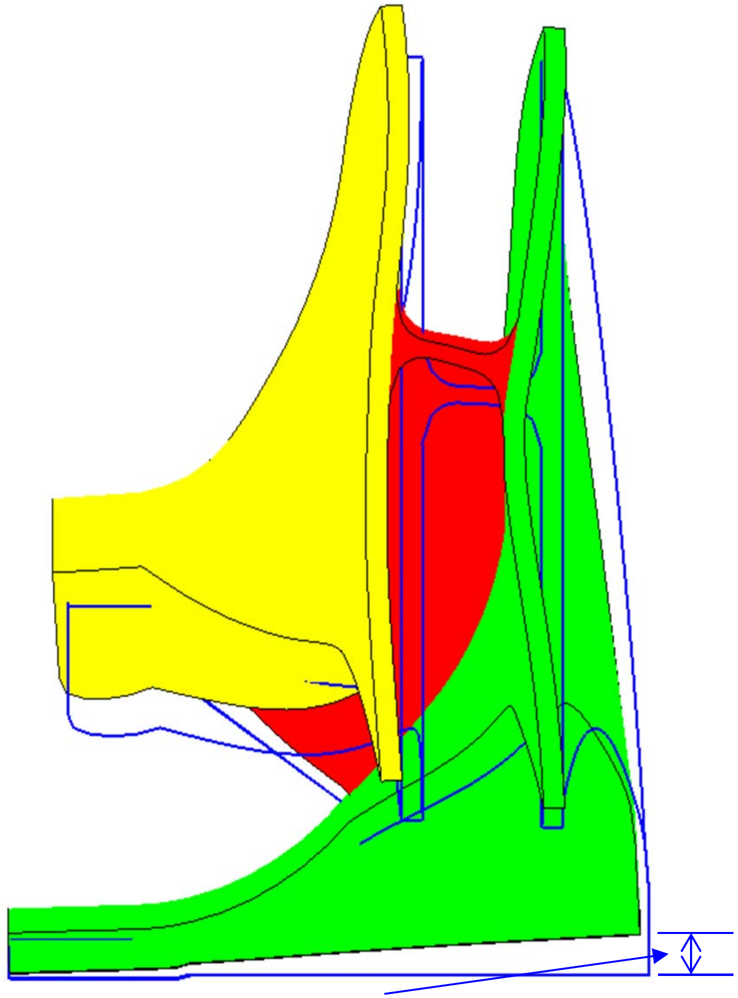
FEM Analysis for impeller grip condition

- 0.0005mm of deformation at impeller inside edge was caused by gas pressure.
- The impeller deformation by gas pressure was very small comparing to the deformation by centrifugal force.



Centrifugal force   Gas pressure

Impeller deformation (at MCR)



0.1494mm / on radius
(Centrifugal force＋Gas pressure)

FEM Analysis for impeller grip condition

Gripping force evaluation result

	Contact area [mm2]	Ratio [-]	Gripping Force [kgf]	S.F. [-]
Actual (at MCR)	3.79×10^4	76.6%	18810	2.82

Note 1 : Gas thrust force is 6659kgf.
Note 2 : Static friction coefficient is 0.15.

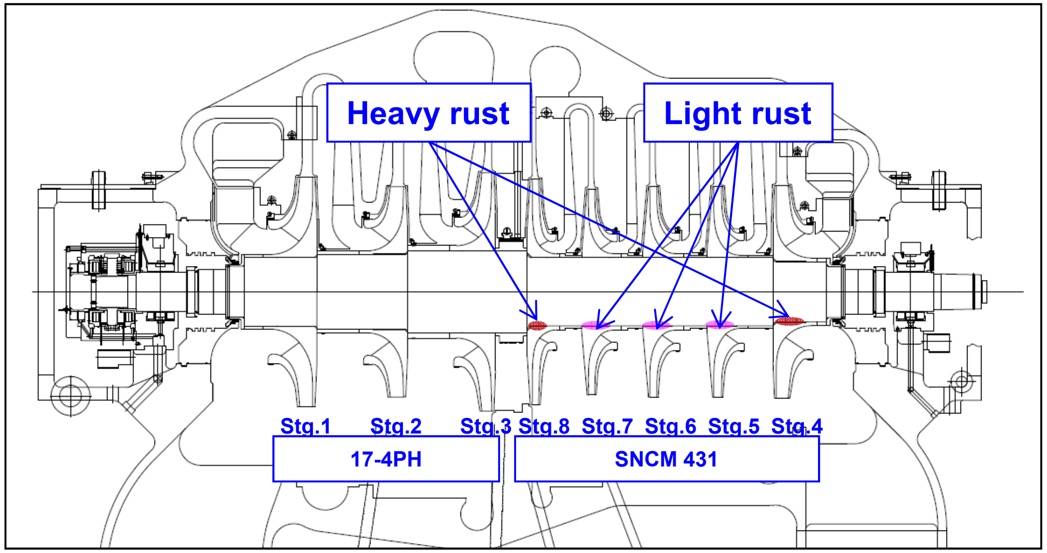
- **Contact area was not changed by impeller centrifugal deformation.**
- **Contact area was not changed by impeller deformation due to gas pressure too.**
- **Enough gripping force against gas thrust force was confirmed by FEM analysis.**
- **The impeller did not shift during operation.**

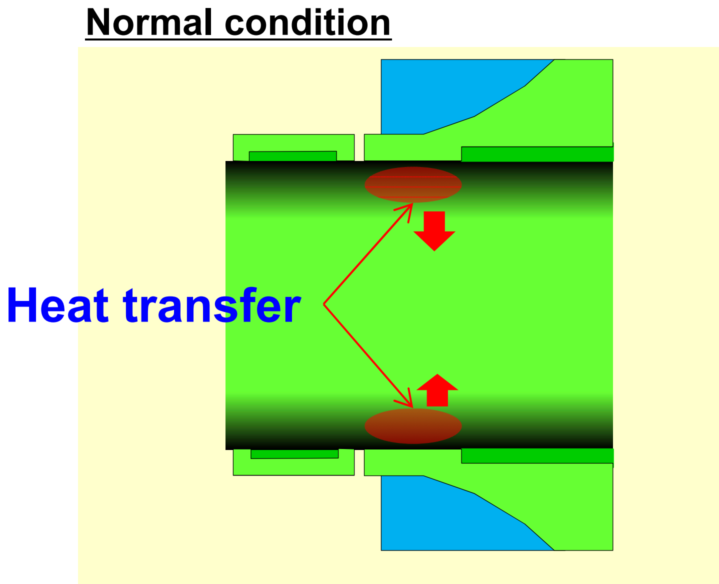
Root Cause Analysis

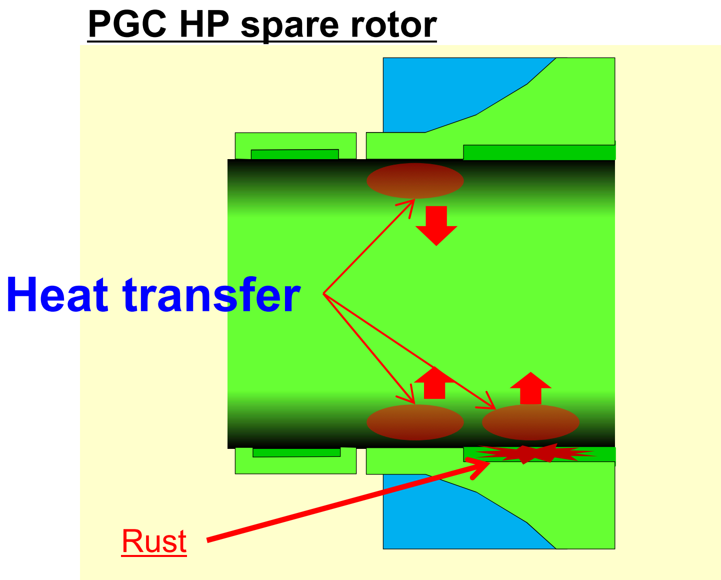
Phenomena	Possible Cause-1	Possible Cause-2	Descriptions	Possibility
High Shaft vibration ・Main DE side bearing ・Main component was 1X ・Vibration amplitude change during constant speed	Rotor Unbalance	Unbalance change due to impeller movement	After receiving inspection; •Larger gap between impellers and sleeves were observed. •Color distribution was observed at shrink fit area. (Shrink fit contact pressure might be not uniform) •Expansion of impeller bore size was observed.(1st and 2nd impeller.) LSB and HSB check result were not so much changed from previous MCO test result.	Not Possible.
		Impeller restrained by the rust or sediments	After receiving inspection; •Scratch on the rotor was observed at 4th impeller (Some sediments may be located.) ⇒The impellers might be restrained by the rust or sediments and then rotor robustness to vibration was decreased, because distribution of impeller displacement is not symmetry. FEM analysis shows the gap between impeller and shaft is increased by impeller centrifugal deformation.	Not Possible.
		Thermal bending of shaft	• The heavy rust was observed around HP section (4th to 8th impeller) ⇒ Thermal bending of the rotor might be caused by the rust, due to the no uniform heat transfer.	Possible. FEM analysis shows the possibility. (P.21-24)

FEM Analysis for rotor thermal bending

Thermal bending of the rotor could be caused by the rust, due to the uneven heat transfer from impeller.





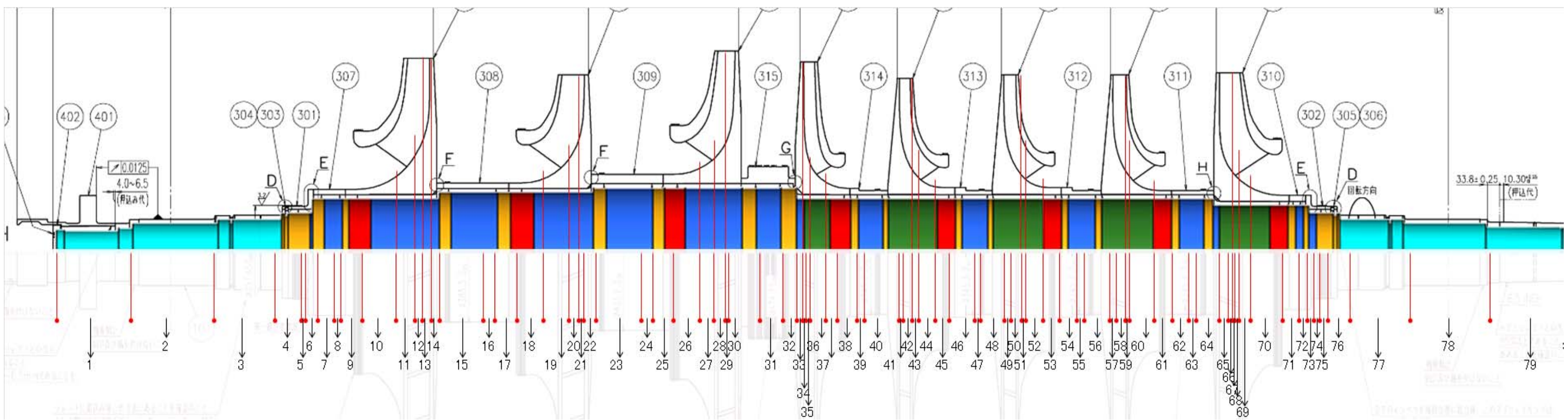


FEM Analysis for rotor thermal bending

Item	Analysis conditoin
Objects	Shaft
Analysis type	Steady thermal analysis
Assumptions	・N2 operating condition of each stage was considered (Pressure, temperature, velocity). ・Thermal transfer coefficients for each parts are shown on right table.

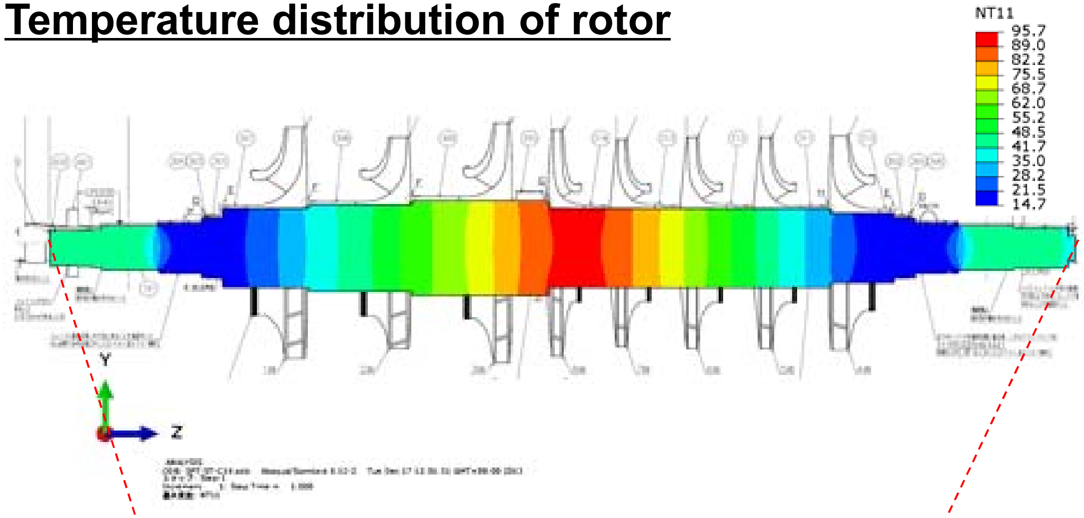
Parts	Thermal transfer coefficient.	Unit
SNCM431	0.036	W/mm℃
SUS 401	0.025	W/mm℃
Air	2.28E-05	W/mm℃
Rust	0.001-0.015	W/mm℃

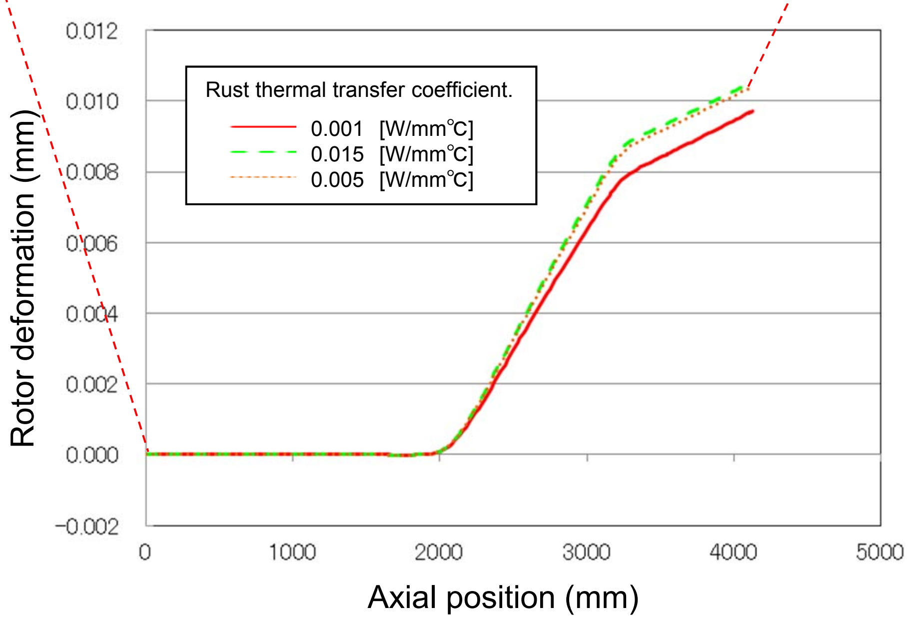
Thermal boundary condition



FEM Analysis for rotor thermal bending

Temperature distribution of rotor



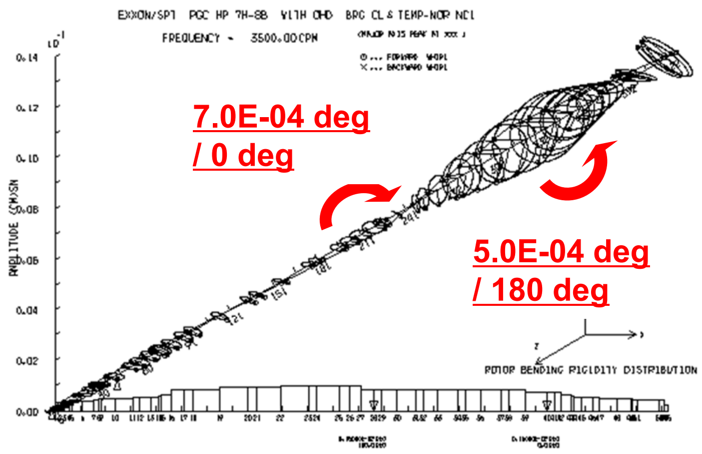


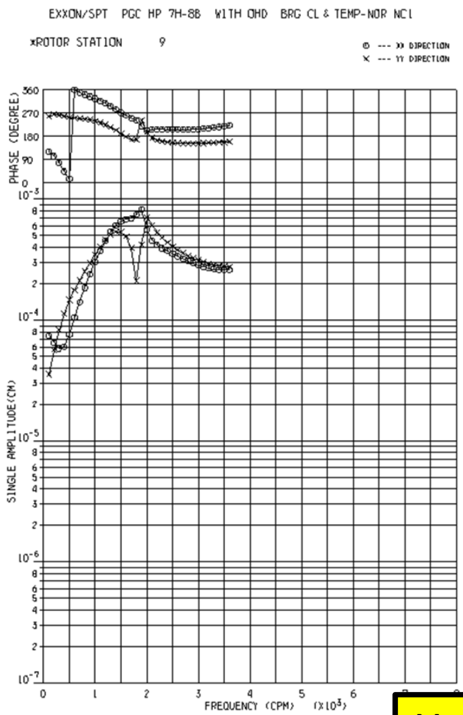
Rotor thermal bending analysis was performed based on N2 operating condition.

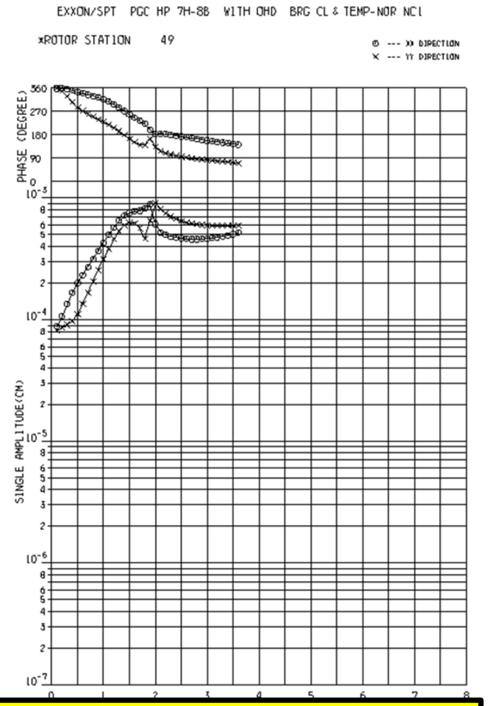
Small effect of rust thermal transfer coefficient difference was confirmed by the analysis.

Rotor lateral analysis for rotor thermal bending

Case study







Only rotor thermal bending is considered in this analysis.

Magnification factor of almost 2 between DE and NDE, almost matched with site observation

Vibration amplitude (at 3500 rpm)

	TH side H [μmp-p]	TH side V [μmp-p]	N-TH side H [μmp-p]	N-TH side V [μmp-p]
Site operation	34	33	72	49
Analysis considering rotor bending due to the rust	6	6	12	10

Conclusion

✓FEM analysis for impeller gripping force was performed and it was confirmed that the gripping force is adequate.

✓Low Speed Balance and High Speed Balance check result were not conclusive and did not point to any specific abnormality.

✓Rust observed around HP section. (4th to 8th impeller). Lateral analysis to include effect of uneven heat transfer conducted. The results matched actual rotor observations.

✓Magnification factor of almost 2 between DE and NDE, almost matched with site observation. Thermal bending of the rotor in dynamic condition (due to rust between impeller and shaft) resulted in high amplification of vibrations during shutdown.