49TH TURBOMACHINERY & 36TH PUMP SYMPOSIA

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TEXAS A&M UNIVERSITY | TEES | TURBOMACHINERY LABORATORY
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Multi-stage pump vibration caused by excessive wear ring clearance, Improper balancing, rotating stall

GS Caltex

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Presenter / Authors bios

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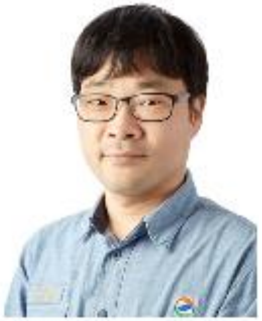
Rotating equipment reliability(RER) engineer of GS Caltex Refinery since 2006.

RER engineer in Chevron Richmond Refinery 2016~2017 as a joint venture company.

Key member of KRMEA(Korea Rotating Machinery Engineers Association).



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A team leader of Rotating Equipment Reliability team in GS Caltex. He has over 18 years of experience in rotating equipment, and he is a member of chevron rotating equipment reliability BIN(Business Improvement Network) group.

* GS Caltex : One of the largest(CDU Capa. 800,000bbl/day) unit refinery in the world located in South Korea

Contents

Abstract

Generally, the centrifugal pump vibration caused by mis-alignment, unbalance or bearing failure. But, I have seen two abnormal vibration phenomena in a multi-stage high pressure pump which has 13th stage impeller.

First issue is high vibration problem due to excessive clearance of wear ring and balancing issue. Second one is sub-synchronous vibration due to rotating stall at the inlet of diffuser.

The root cause was figured out based on rotor dynamic, CFD(computational fluid dynamics) analysis and shop test of pump manufacturer. This paper will show the detail vibration phenomena, root cause analysis result and the countermeasure.

Equipment Data

Process : VRHCR(Vacuum Residue Hydro-Cracker) Quench Oil(S.G 0.7) Pump

Pump : BB5, 13^{th} stage, Forced Lubrication, Motor Driven

Power : 1,272HP(954kW)

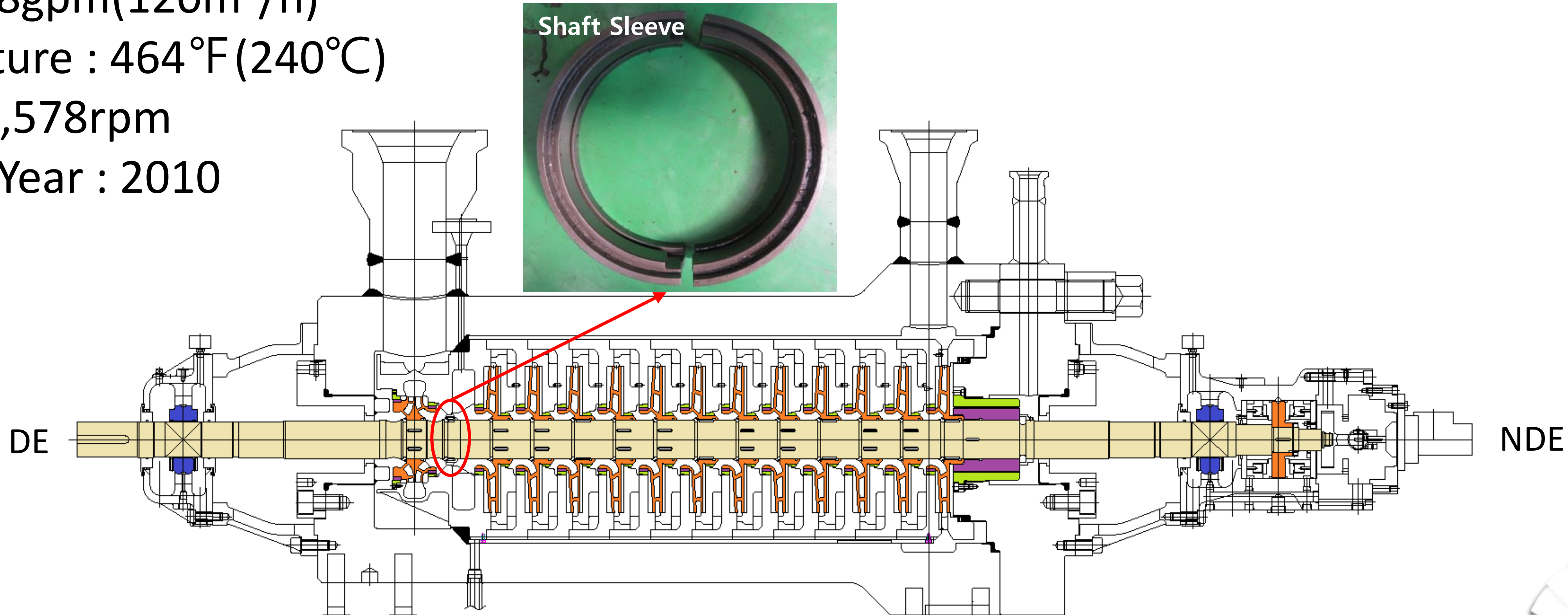
Head : 8,372ft(2,552m, Suction 25kgf/cm^2, Discharge 202kgf/cm^2)

Flow : 528gpm(120m^3/h)

Temperature : 464℉(240℃)

Speed : 3,578rpm

Installed Year : 2010



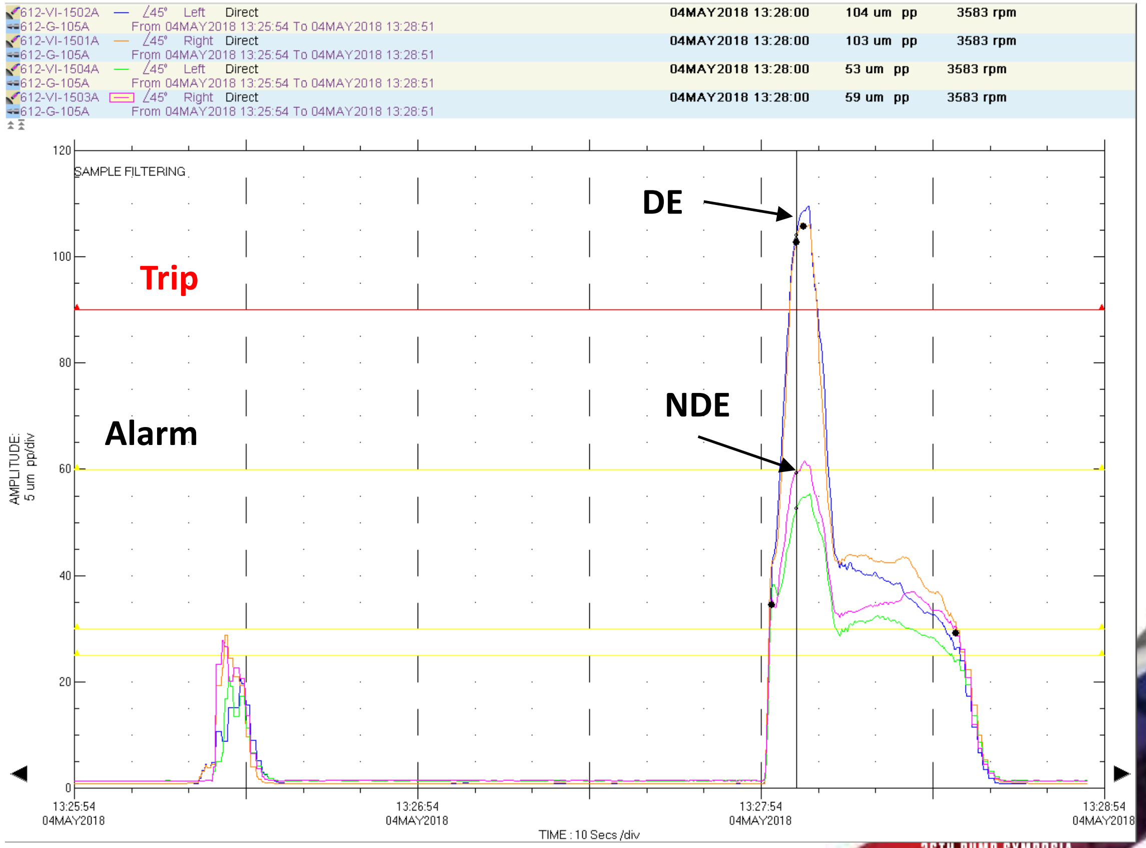
Pump Sectional DWG

Problem Statement

- **Pump overhauled due to DE bearing high vibration, 2.36mils(60µm)**
  - Broken shaft sleeve between 1st~2nd stage replaced
  - Impeller wear rings were not replaced even though it exceeded the allowable limit, because there was no performance issue.
  - Rotor assembly was 2-plane dynamic balanced without individual balancing.
- **During start up after above overhaul, pump tripped by DE vibration HH**
  - DE Radial vibration increased to : 4.33mils (110µm) with 1X(3,578rpm)frequency
    - Normal(before repair) < 0.78mils(20µm)
    - Alarm/Trip : 2.36/3.54mils(60/90µm)

Wear Ring Clearance	Front	Rear
Original Design	0.018~0.36inch (0.45mm~0.90mm)	0.013~0.025inch (0.32~0.64mm)
Measured Max.	0.032inch(0.83mm)	0.031inch(0.81mm)

Vibration Trend



Vibration Issue #1 Action

- **Pump overhauled again, DE radial vibration decreased to 12µm after repair as below.**
  - All wear rings of each impeller were replaced and clearance adjusted within design value.
  - All impellers were individually balanced before low speed assembly balancing.
  - Rotor balanced at operating speed(3,578rpm)

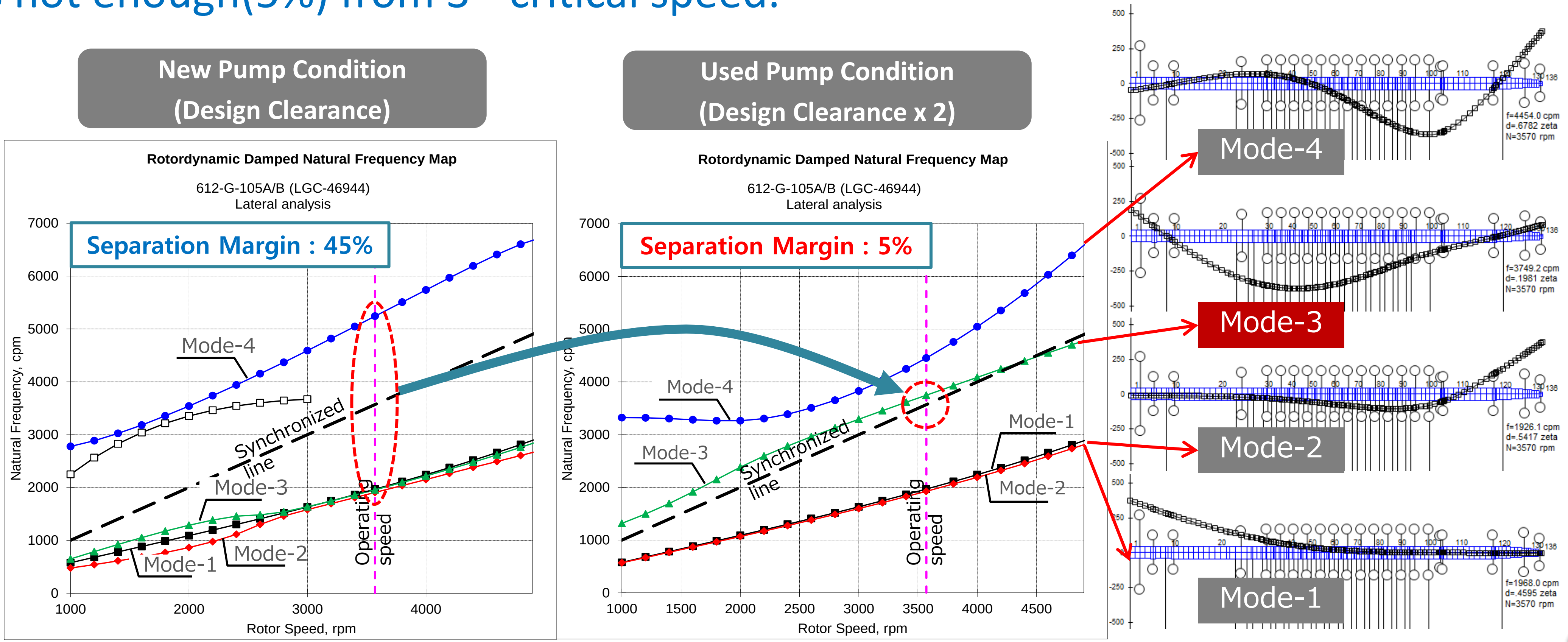
    - Generally, operating speed balancing is not required for pumps.
    - But, operating speed balancing conducted considering the previous high vibration issue and the slender rotor design(1st critical speed 1,837rpm)

Balancing Method	Residual Unbalance		Equivalent ISO Grade
	oz.-in	g-mm	
Before LSB	8.10	5,824	G 6.3
After LSB	0.17	122	G 0.13
After HSB	3.18	2,288	G 2.5

* LSB : Low Speed Balancing(two-plane)
* HSB : High(Operating) Speed Balancing(multi-plane)

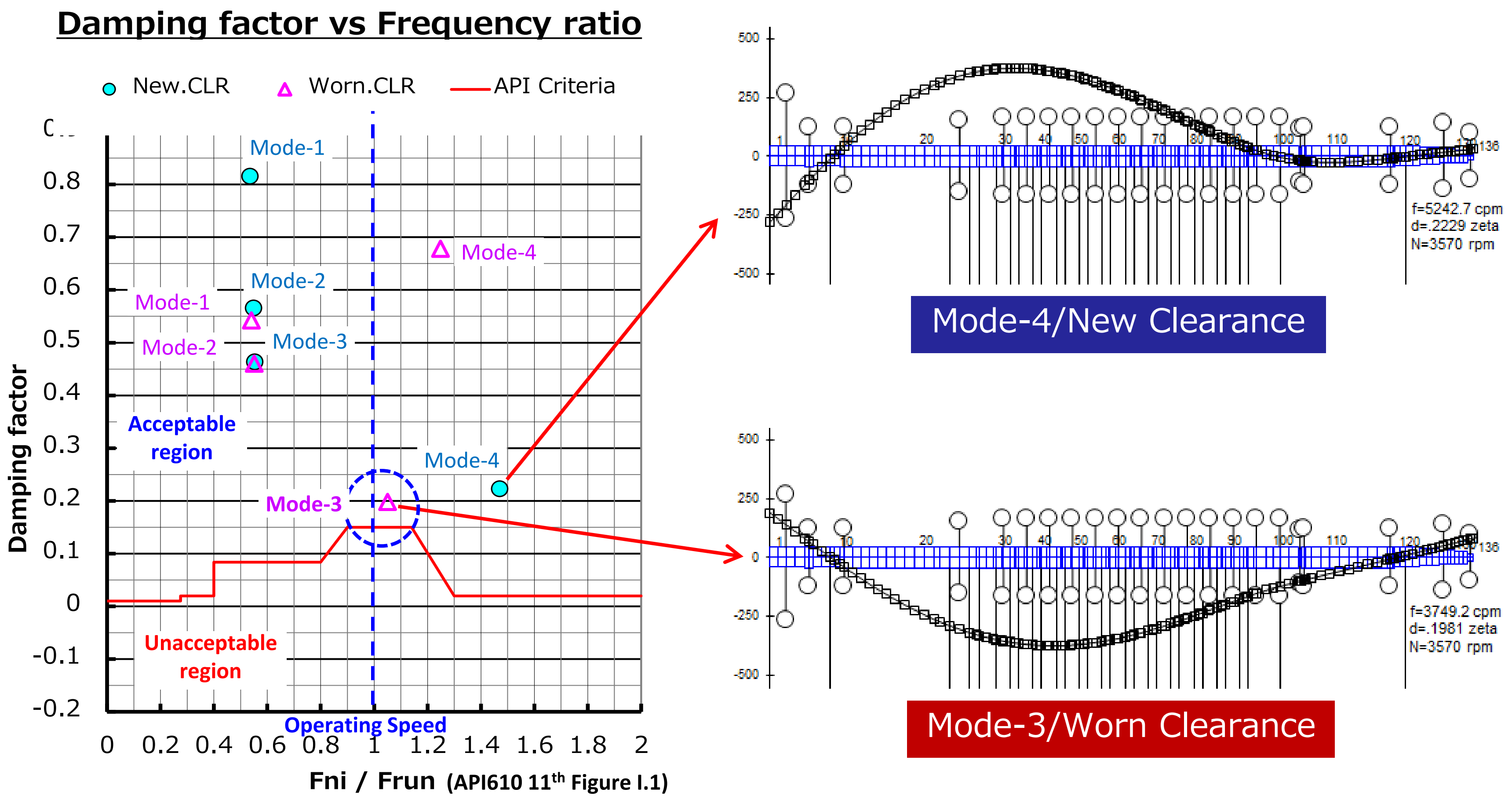
Troubleshooting

- **Rotordynamic damped natural frequency map analysis(Campbell Diagram)**
  - Rotordynamic analysis conducted after repair of the pump to find out the root cause.
  - In case of impeller wear ring has excessive clearance(2x of design), the separation margin is not enough(5%) from 3rd critical speed.



Troubleshooting

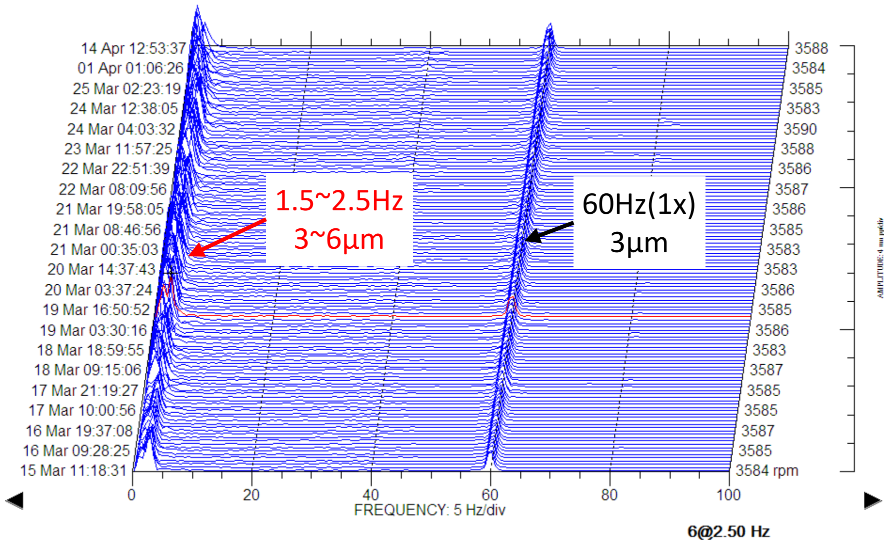
- **Damping Factor vs Frequency Ratio**
  - In case of impeller wear ring has excessive clearance(2x of design), damping factor is close to the limit(0.1) of API610.



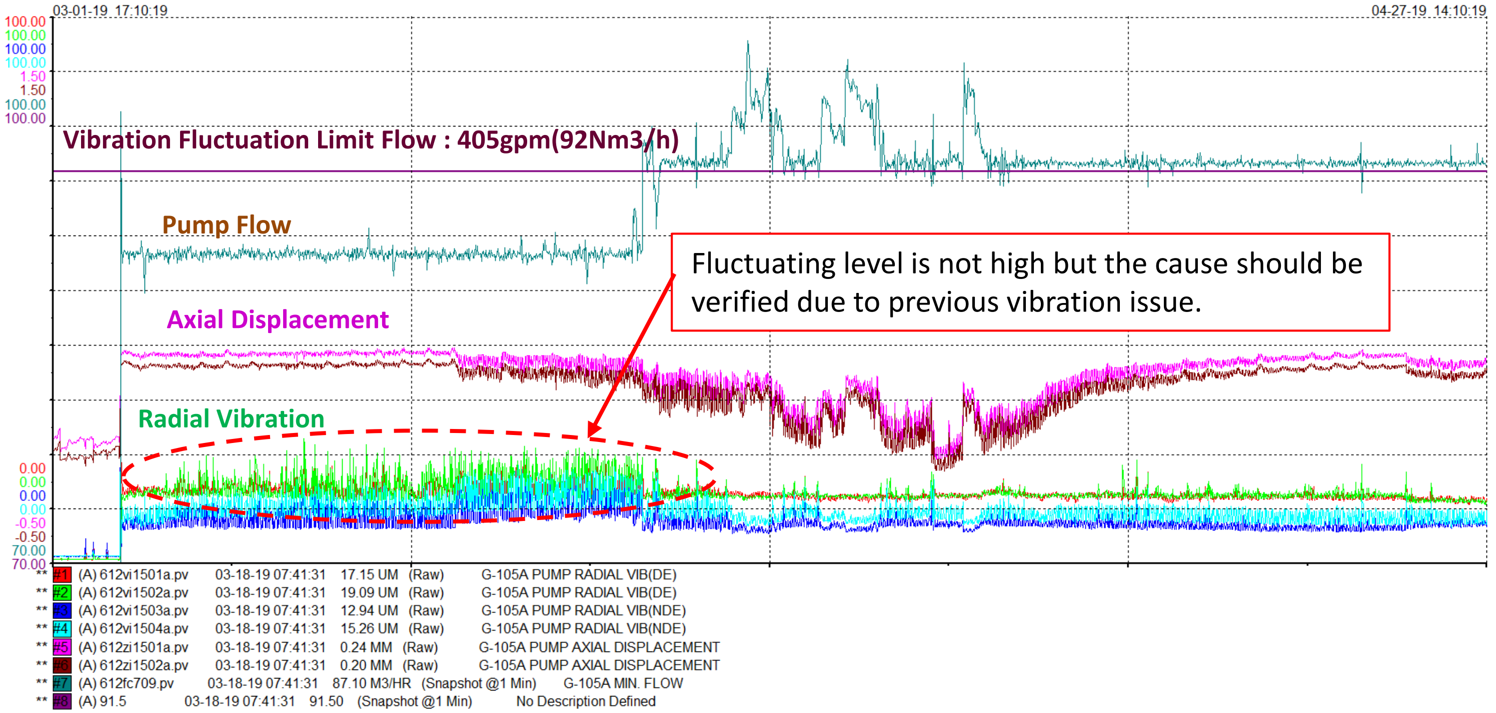
Vibration Issue #2 Problem Statement

- **Radial vibration fluctuation with sub-synchronous frequency**
  - The operating condition seems good after previous repair.
  - But, DE/NDE radial vibration started fluctuating between 0.47~0.78mils(12~20μm) when the pumping flow decreased below 405gpm(92Nm^3/h) by process requirement.
  - Vibration frequency : 1.5~2.5Hz
- Vibration fluctuation disappeared with increased flow

Waterfall Spectrum



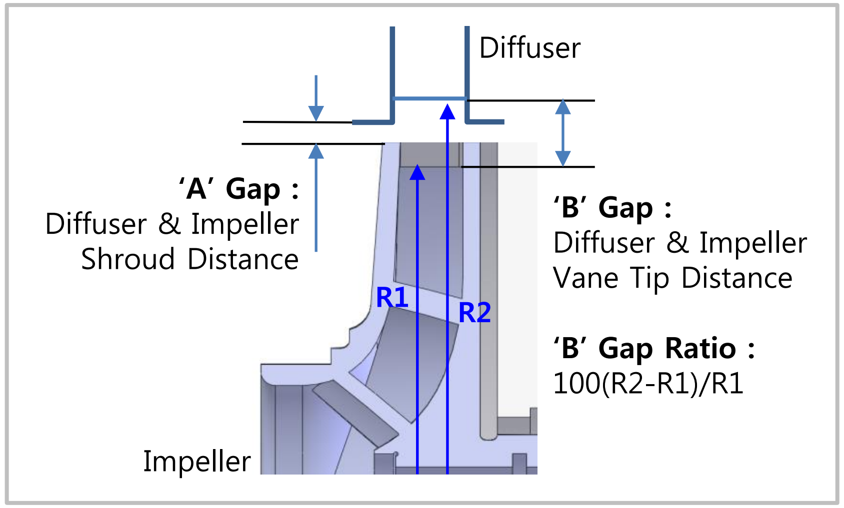
Vibration Trend

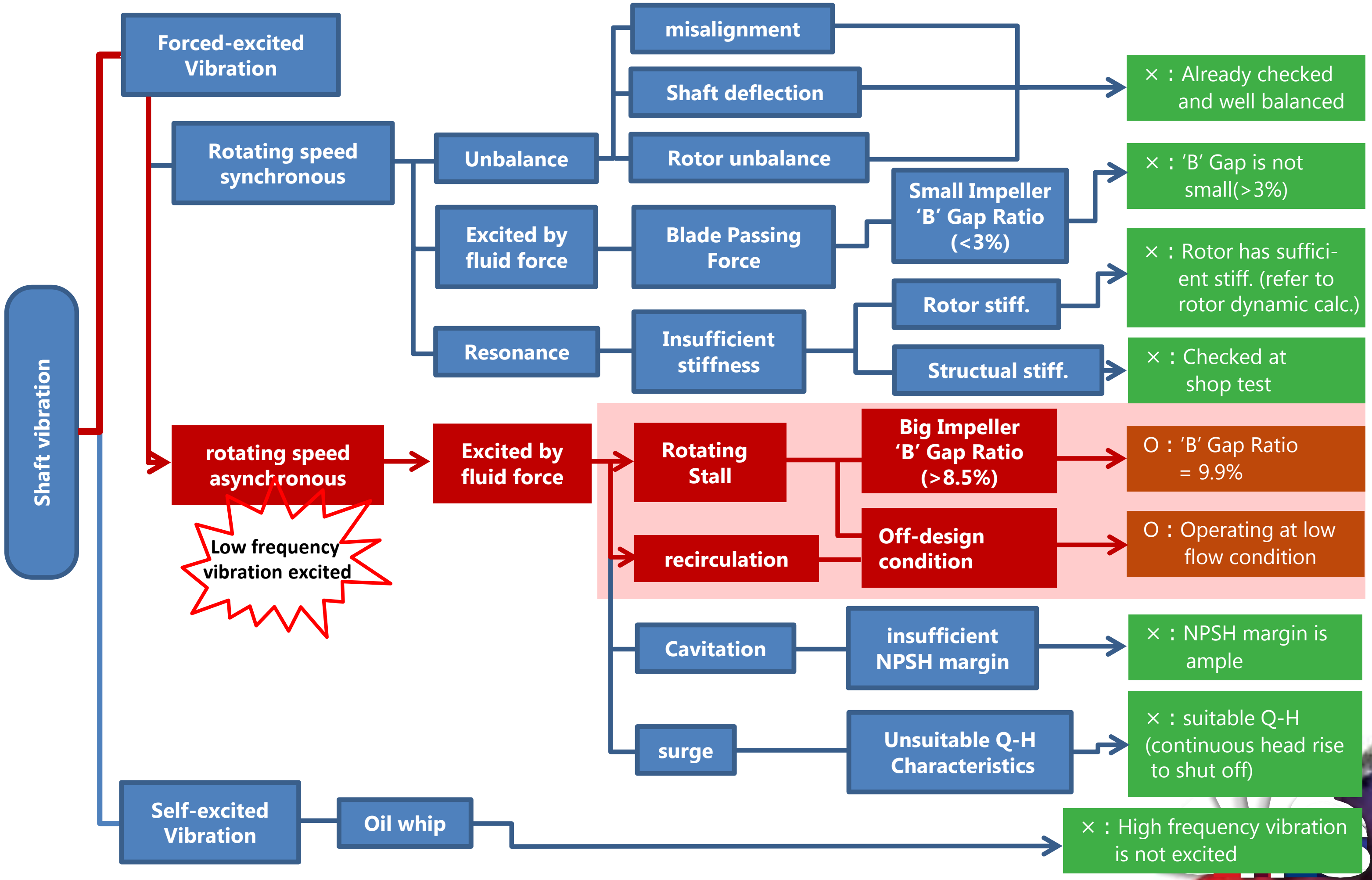


Vibration Issue #2 Troubleshooting

- **The cause of sub-synchronous vibration was rotating stall between impeller and diffuser**
- **Rotating stall can be generated in below cases**
  - Big impeller 'B' gap
  - Low operating flow
  - Low Ns(Specific Speed), high pressure design

Impeller 'A', 'B' Gap Definition



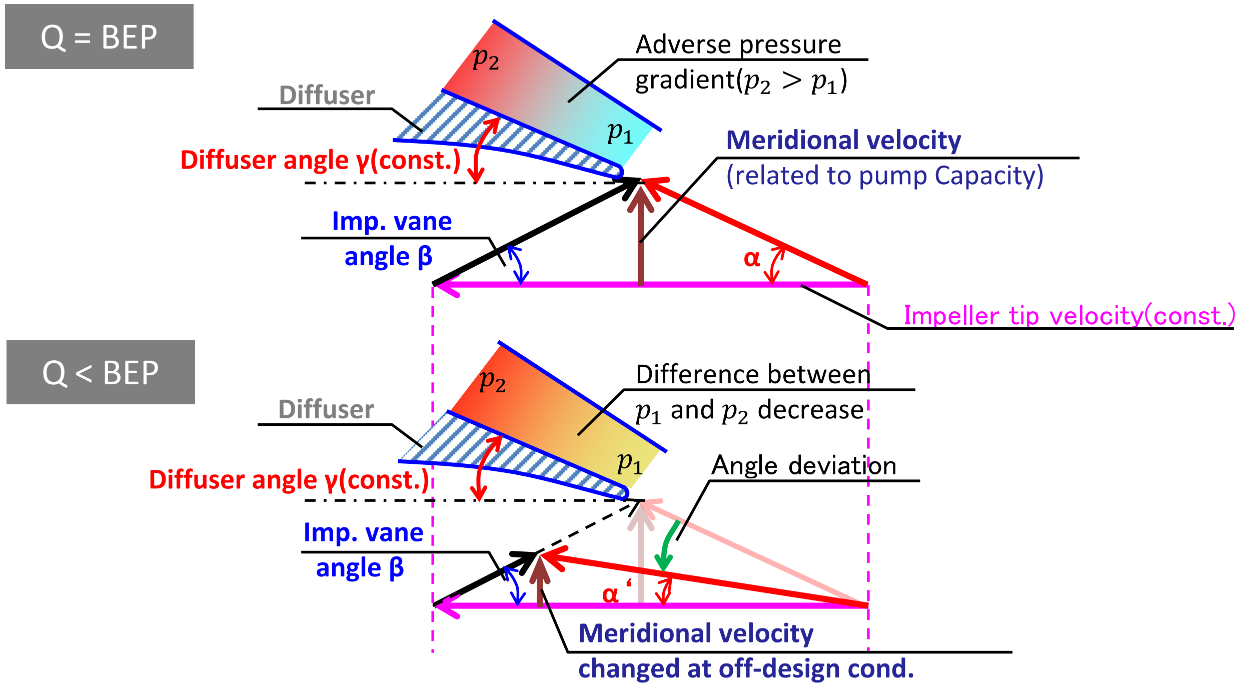


Troubleshooting

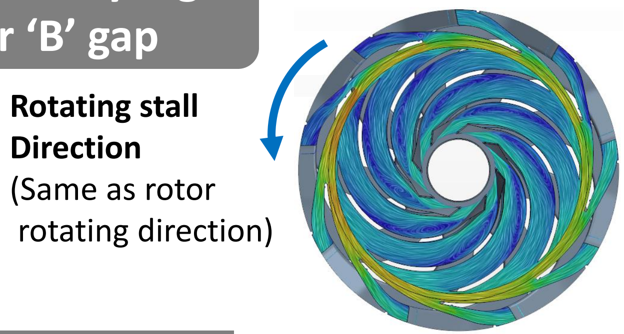
- **Rotating Stall Mechanism**

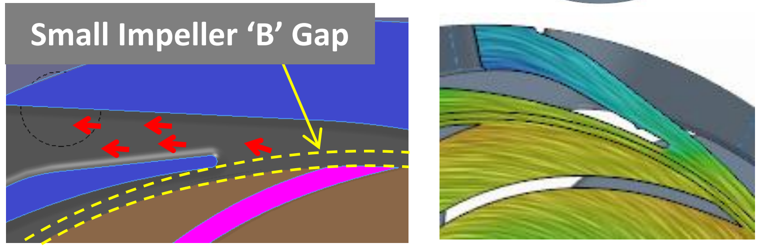
: Flow decrease → Meridional velocity decrease → Flow angle deviation from designed diffuser angle(this happen easily with big impeller ‘B’ gap) → Stall generation, then stall cell moves to subsequent diffuser and rotates(stall rotating frequency < 10% of rpm)

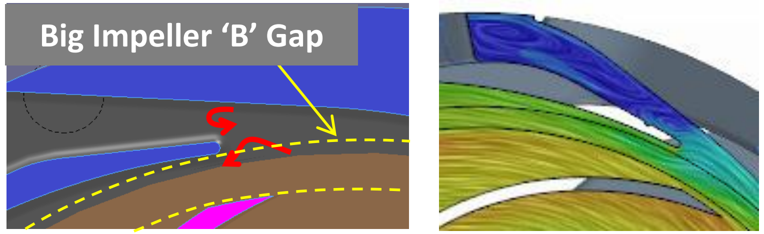
Rotating stall by low flow



Rotating stall by big impeller ‘B’ gap



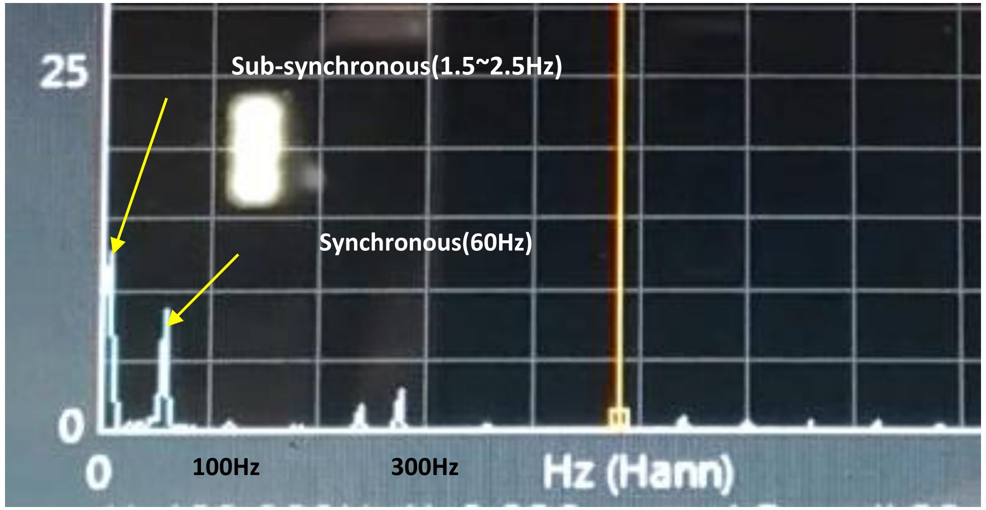




Troubleshooting

- **Vendor Shop Experimental Test**
  - Test Pump : Two Stage, Diffuser type, Ns(US Unit) 4,902(≒95 in SI Unit)
  - Manufacturer conducted test with available high pressure pump to figure out how impeller ‘B’ gap affect the vibration
  - Vibration was measured in below condition
    - Impeller ‘B’ gap Ratio : 2.5%, 5.3%, 12%
    - Pump Flow : 20~130% of design capacity

Vendor Shop Test Pump



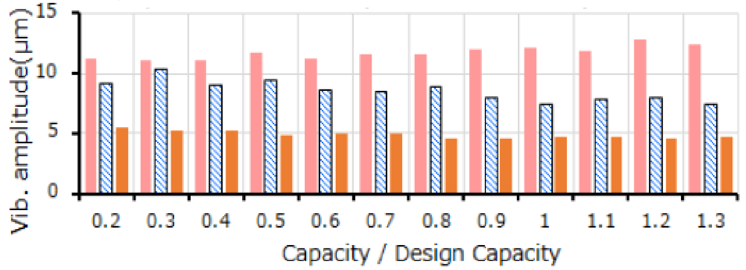
Measured Vibration FFT

Troubleshooting

- **Vendor Shop Test Result**

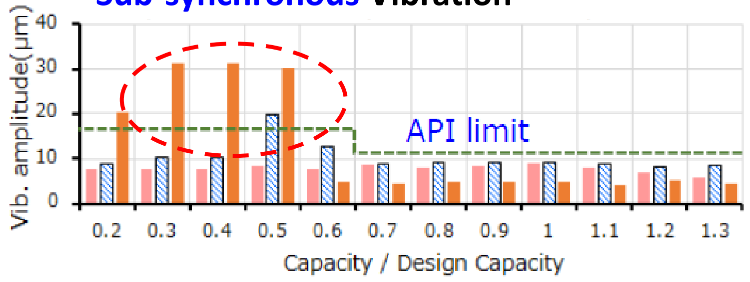
'B' Gap Ratio 2.5% 5.3% 12%

Synchronous Vibration



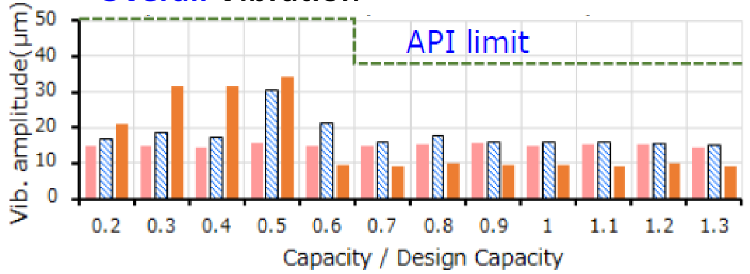
◦ Synchronous vibration decrease with bigger 'B' gap, because of lower vane pass vibration

Sub-synchronous Vibration



◦ Sub-synchronous vibration increase with bigger 'B' gap at low flow, because of stall

Overall Vibration



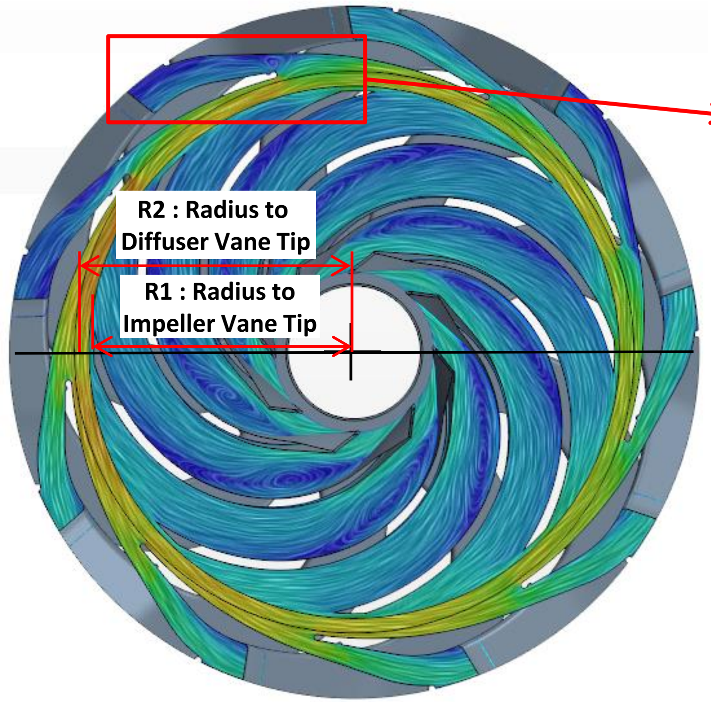
◦ Overall vibration increase with bigger 'B' gap at low flow, because sub-synchronous is dominant frequency

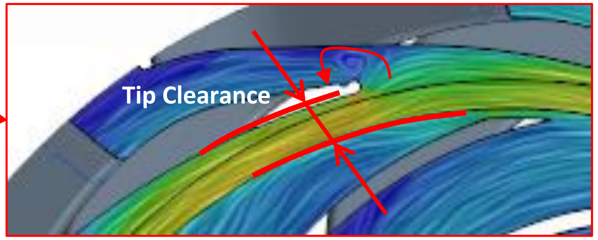
* API610(11th, Table 8) Vibration Limit(μm peak-to-peak) : Overall $(2.5 \times 10^6/\text{rpm})^{0.5}$, Discrete frequency < 30% of overall, Allowable increase of outside preferred flow < 30%

Troubleshooting

- **Pump manufacturer verified this rotating stall phenomena based on CFD and shop test.**
- **Rotating stall can be generated with the combination of below conditions**

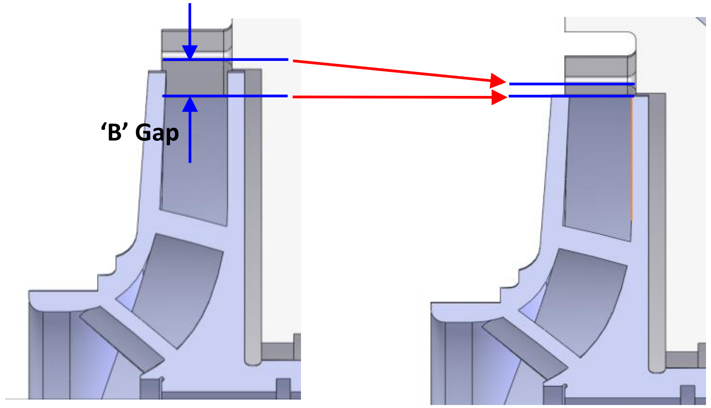
Rotating Stall Condition(Manufacturer's Analysis)		Issued Field Pump	Countermeasure	Field Improvement
High pressure design	Specific Speed(Ns) ≤ 5,160(100 in SI Unit)	**5,160**(100 in SI Unit)	N/A	N/A
Low Flow	Operating Flow < 50~60%	**57%**	Higher Flow	**Accepted**
Large Impeller 'B' Gap Ratio	'B' Gap Ratio[100(R2-R1)/R1] > 8.5%	**9.9%**	Smaller 'B' gap	Not accepted due to high cost





Diffuser Rotating Stall

CFD Condition
- Fluid: Water
- Inlet Pressure : 0 bar
- Speed : 3570rpm
- Solution : Simcenter Star CCM+
- Turbulence model : Realizable k-ε



New diffuser required to reduce 'B' gap → High Cost

Caution(API610 11th 6.1.15)
Minimum 'B' Gap Ratio is required to prevent vane passing frequency vibration
: 3% for diffuser type, 6% for volute type

Improvement

- **Multi-stage Pump Balancing procedure** : Impeller should be individually balanced before 2-plane assembled dynamic balancing.

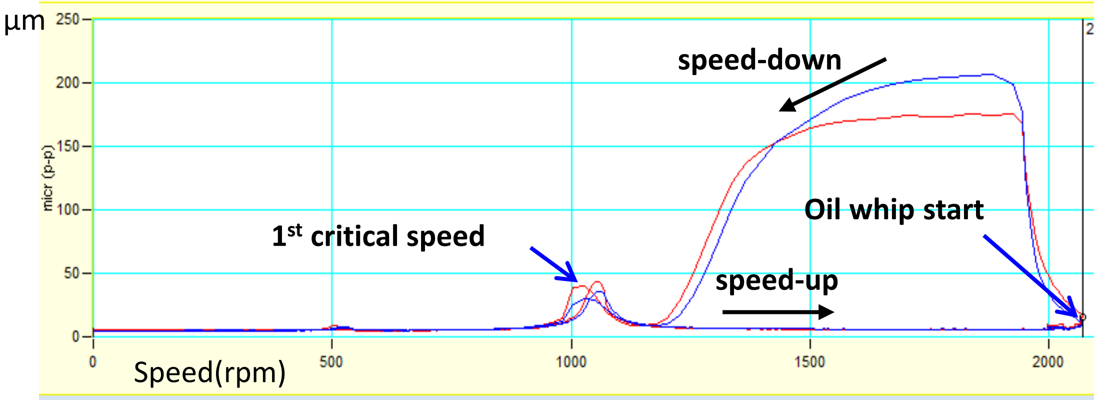
- **Allowable limit of wear ring clearance** : Manufacturer's original maximum clearance guide was 2 times of design minimum clearance. This allowable limit revised to 1.7 times of design minimum clearance for enough separation margin from critical speed based on the rotordynamic analysis. Wear ring in the multi-stage pump is not just a part to reduce recirculation flow. It works as a kind of inter-stage bearing, important component for rotor stability.

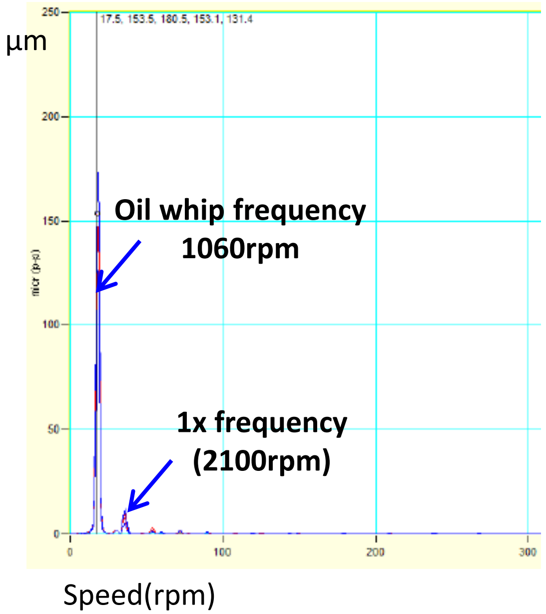
- **Operation Guide** : Minimum recommended flow(>60% of BEP) guided to operation team. The modification of impeller and diffuser to reduce impeller 'B' Gap was not implemented due to high cost.

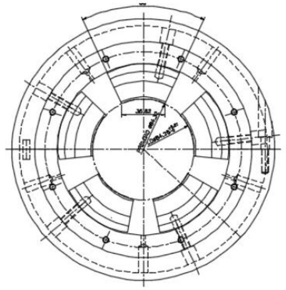
Lessons learned

- High speed balancing for multi-stage pump is not recommended. Unstable rotor behavior and DE/NDE high vibration(5mils, 200µm) caused by oil whip at 2100rpm detected while speed up to 3578rpm.
- During high speed balancing, rotor does not have sufficient stiffness and damping because only DE/NDE bearing support the rotor without wear ring. Newly designed tilting pad bearing in lieu of original pressure dam bearing was required to remove oil whip for the high speed balancing of this pump.

Oil Whip Vibration Trend







Newly designed tilting pad bearing

Thank you
&
Questions?

ACKNOWLEDGEMENTS

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