51ST TURBOMACHINERY & 38TH PUMP SYMPOSIA
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Light Hydrocarbon Shipping Pumps Challenges with Repetitive Mechanical Seal Failure | Case Study

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About the Authors

Wesam Khalaf Allah is a rotating equipment engineer with the Consulting Services Department at Saudi Aramco, Dhahran, since 2012. During his career, he worked on the commissioning and startup of YASREF and Shaybah NGL projects. He also led the development of rotating equipment packages under Jafurah Program. Mr. Khalaf Allah has published more than 10 articles related to rotating equipment in different international magazines. Mr. Khalaf Allah received a B.S. degree (Mechanical Engineering, 2012) from Purdue School of Engineering and Technology, Indianapolis, and a master of engineering degree (Mechanical Engineering, 2020) from Texas A&M University, College Station.

Fernando Ballard is a seasoned mechanical engineer with extensive international experience in design, commissioning, operation and maintenance of fixed and rotating equipment. Mr. Ballard's unique career in Oil and Gas spans across different business lines including downstream, midstream and upstream. He started his career with ExxonMobil where he functioned as a fixed equipment engineer specializing in the Fluid Catalytic Cracking technology. In 2014, Mr. Ballard joined Saudi Aramco serving as a technical lead in the safe and successful commissioning and startup of the Shaybah NGL mega project. In his current capacity as a senior mechanical engineer at PEMEX Deer Park Refinery, Mr. Ballard supports the Hydro processing production team with an emphasis on rotating equipment reliability and maintenance cost optimization. Mr. Ballard is a graduate of the University of Texas at El Paso and is licensed as a Professional Engineer in the state of Texas.

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Abstract

NGL shipping pumps have experienced high number of mechanical seal failures during the first five years of operation. There are five pumps where each pump is equipped with two mechanical seals. In some instances, the mechanical seal failures occurred suddenly (noticeably immediately after the initial start-up of the facility in 2016).

The vast majority of the failures have occurred while the pumps have been continuously running for periods ranging between 6 and 12 months.

API standards require that a mechanical seal is designed for minimum life span of 25,000 hours or approximately three years.

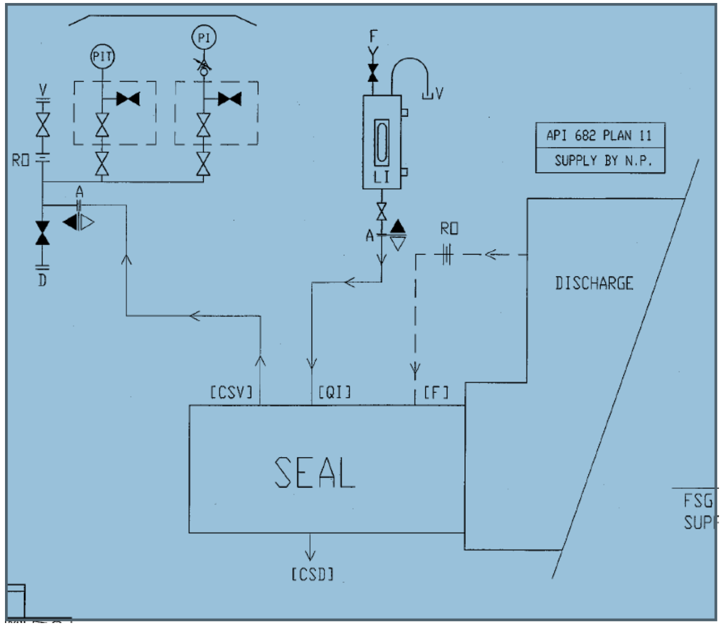
This case study will explain in details the common factors that caused these pumps to exhibit repetitive seal failures. In addition, the case study will explain the proposed API seal plan in order to extend the mechanical seal life.

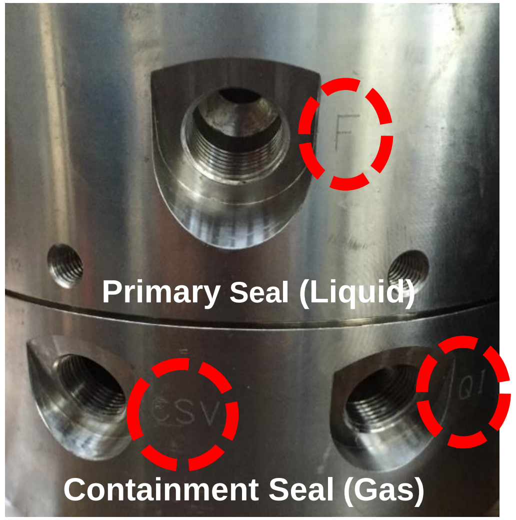
Operating Conditions

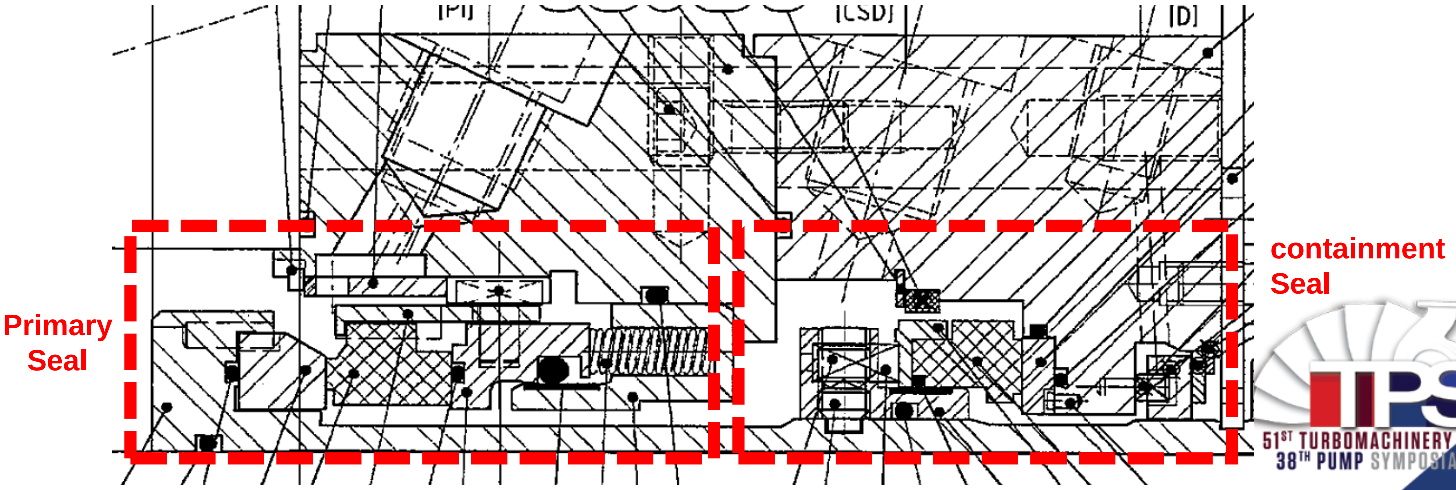
Pumped Fluid	NGL + C2
Normal Flow Temp.	124°F
Max. Flow Temp.	127°F
Specific Gravity	0.4319
Vapor Pressure	490 psia
Viscosity	0.0717 cP
Normal Capacity	2478 gpm
Rated Suction Pressure	549.8 psia
Rated Discharge Pressure	1249.8 psia
Maximum Suction Pressure	560.3 psia
Minimum Cont. Stable Flow	1363 gpm
Shaft Speed	3580 rpm

Mechanical Seals Arrangement

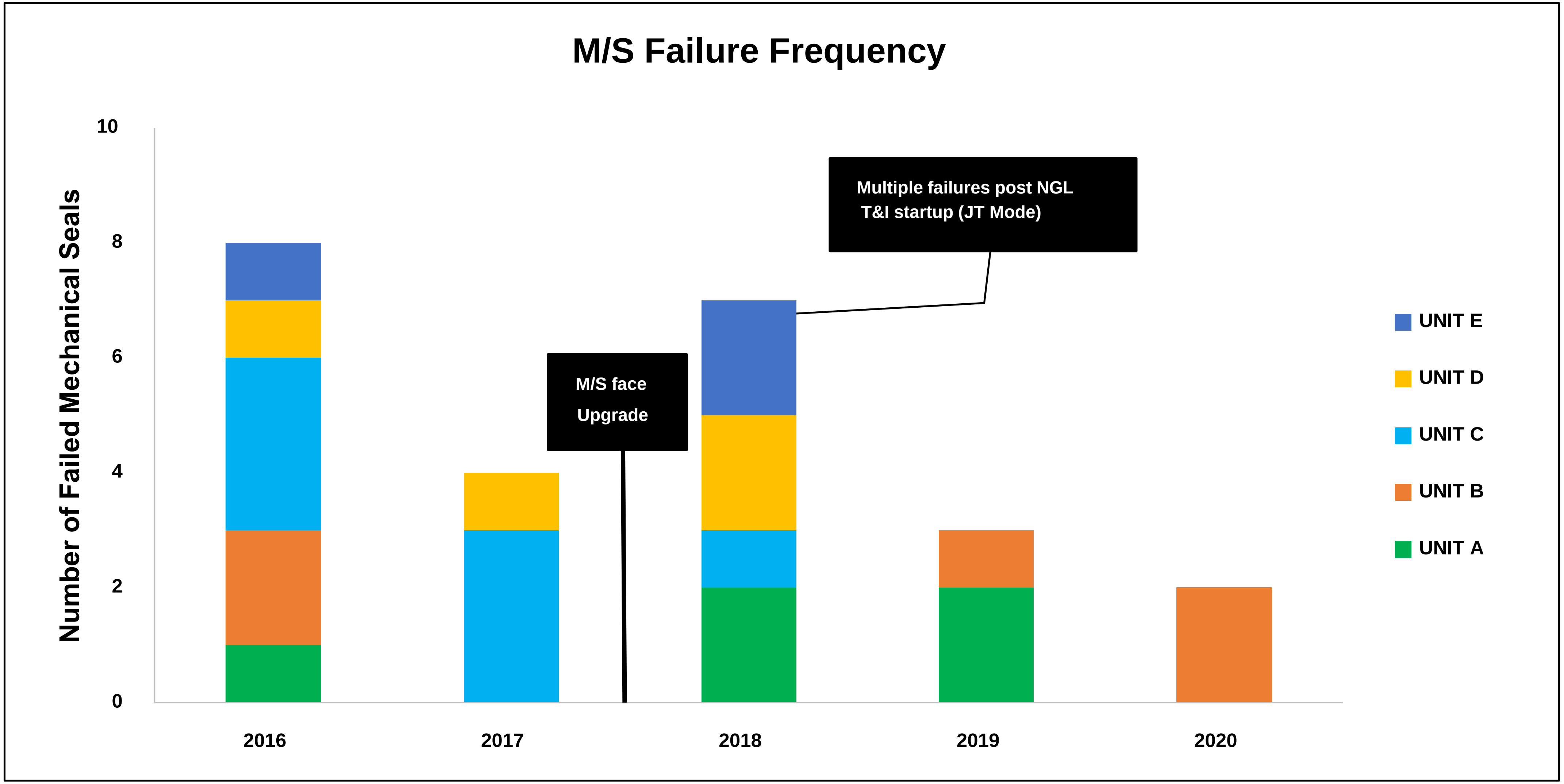
- The mechanical seals contain a primary seal (liquid seal) and a containment seal (gas seal).
- The Piping seal plans used for shipping pumps are plan 11, 51, 76 and 61.
- The Containment seal (gas) has three connections. Those are Quench (QI), Containment Seal Vent (CSV) and Containment Seal Drain (CSD) and is using the API plan 76 and 51.







History of Failure



Casual Factors

Methanol Drainage of Plan 51

Strainer Mesh Size

Intermittent Cryogenic Back-Pressure (Plan 76)

Exceeding Seal Chamber Pressure

Dry Running the Mechanical Seals

Exceeding Pumps Rated Capacity

TEX Mode vs. JT Mode

Methanol Drainage of Plan 51

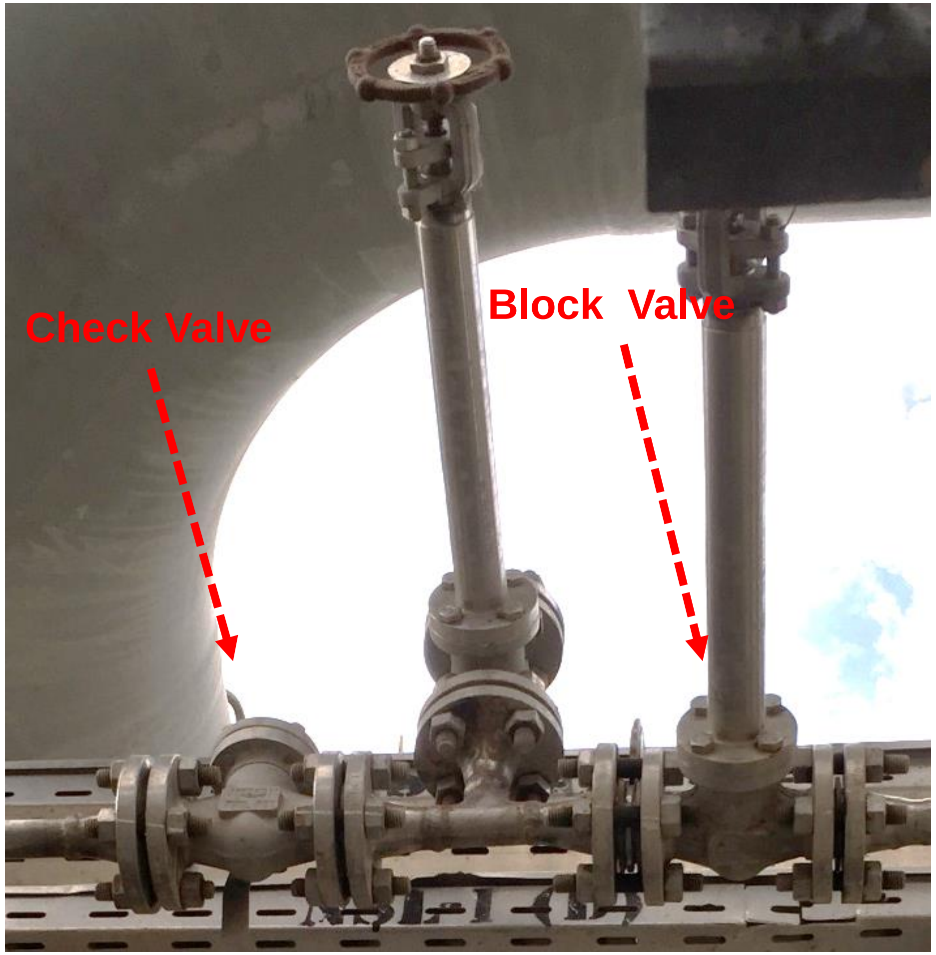
Plan 51 is designed to prevent ice formation around seal faces (inboard and containment), prior to pump start up to be operated as follow:

- The system valve between mechanical seal and reservoir should be open before start-up to remove ice, while containment seal drain (CSD) valve is closed.
- The valve downstream the reservoir can be closed after filling the mechanical seal cavity.
- Once seal gland temperature is at ambient and no signs of ice on piping or seals, methanol can be drained via CSD.
- The containment seal is designed to run on vapor.

Methanol was selected as nitrogen was not available at site.

Intermittent Cryogenic Back-Pressure (Plan 76)

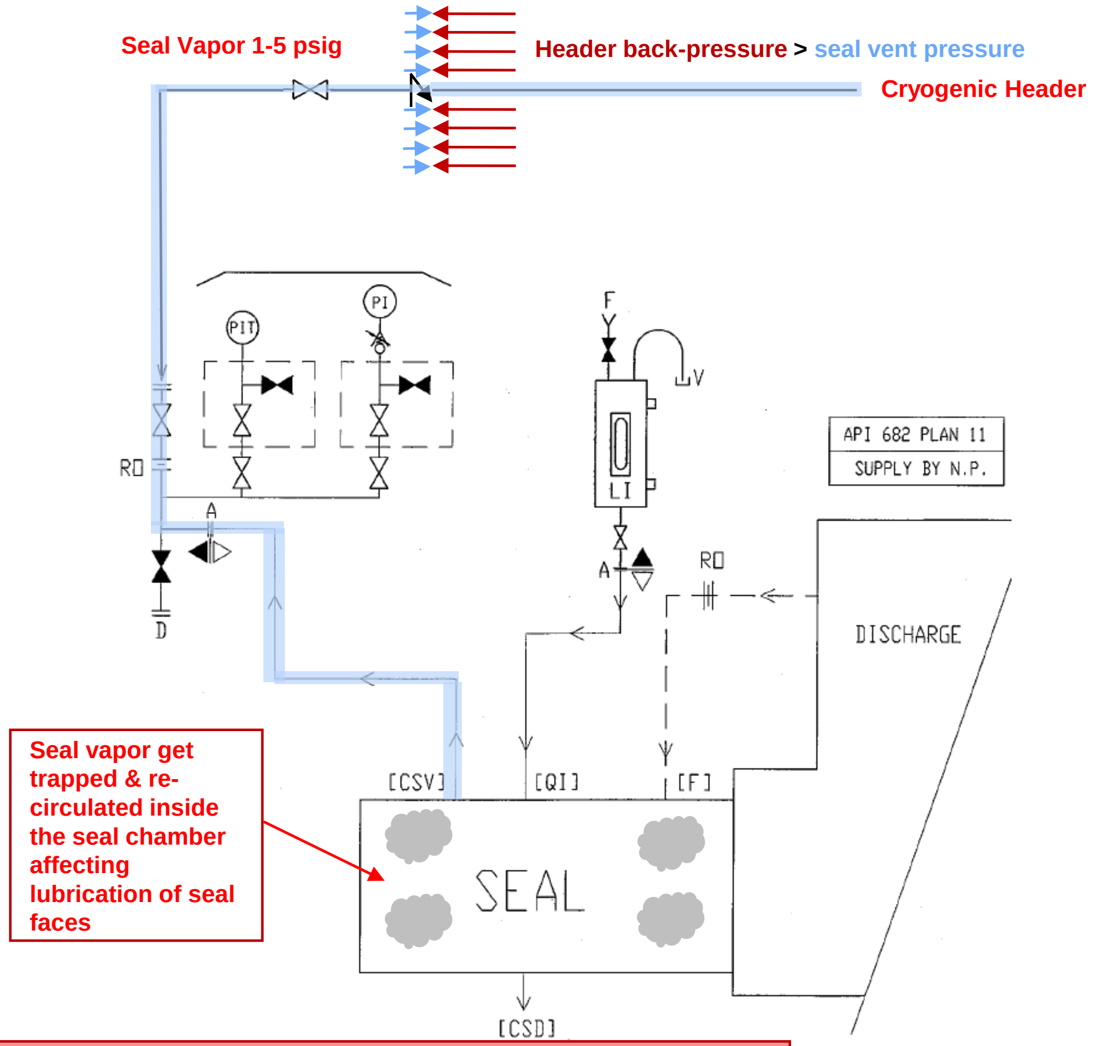
- Plan 76 is a leakage detection system connected to the cryogenic line.
- Pressure gauge reading is usually below 10 psig. If greater than 10 psig, the seal might be leaking unless there is a back-pressure from the cryogenic line where the check valve cannot release any pressure.
- If there is a back-pressure higher than 5 psig, the check valve will not be able to open until it reaches a higher pressure than the cryogenic line.



The CSV connection on plan 76 should be open and connected to a flare system, because the leakage from the inboard liquid seal will be gaseous in the gas seal cavity and will therefore be disposed via the CSV connection in upper part of the seal chamber.

Intermittent Cryogenic Back-Pressure (Plan 76)

- The mechanical seal is designed for continuous venting of the vapor generated from the seal chamber to the cryogenic header via seal plan 76.
- Normal seal venting pressure is between 1 and 5 psig.
- Back-pressure from the cryogenic header is higher than the normal seal vent pressure causing resistance in the vent.



The vent pressure continues to increase inside the seal chamber until it is sufficiently large to overcome the back-pressure and open the check valve to release trapped vapor. The back-pressure takes over again and the cycle repeats until the source of the back-pressure in the cryogenic line is removed.

Exceeding Seal Chamber Pressure

- Seal manufacturer confirmed the mechanical seal is designed for maximum pressure of 582.3 psia (567.6 psig).
- Seal faces were experiencing higher deflection and higher temperature generation than it was anticipated in the design.
- Based on the 18 months operating data for the 5 pumps, below is the actual pumps suction pressure values:
  - Maximum recoded suction pressure while the pump in operation = 660 psig
  - Minimum recorded suction pressure while the pump in operation = 270 psig
  - Average suction pressure over 18 months = 600 psig
  - Maximum recoded suction pressure while the pump is on standby = 837 psig

The high suction pressure while the pump is on standby is typically associated with passing closer of suction isolation valve while the pump is on standby. it is normal for check valve and discharge isolation valves to pass small amount of liquid but the closer of the suction isolation valve will result in increasing the pressure in the suction line and the stuffing box above the rated suction pressure which can result in mechanical seal failure.

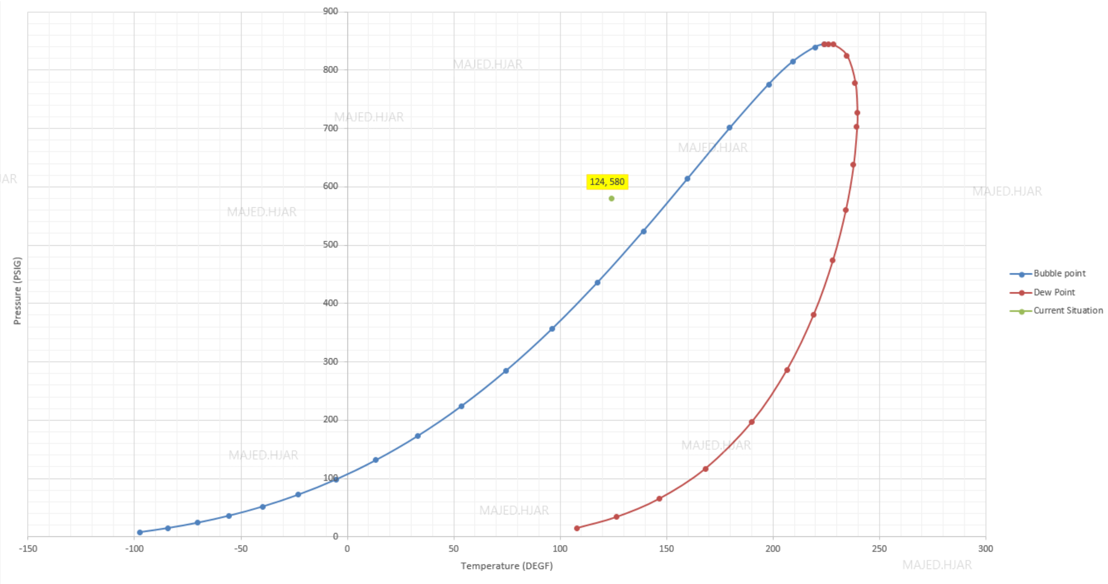
Exceeding Pumps Rated Capacity

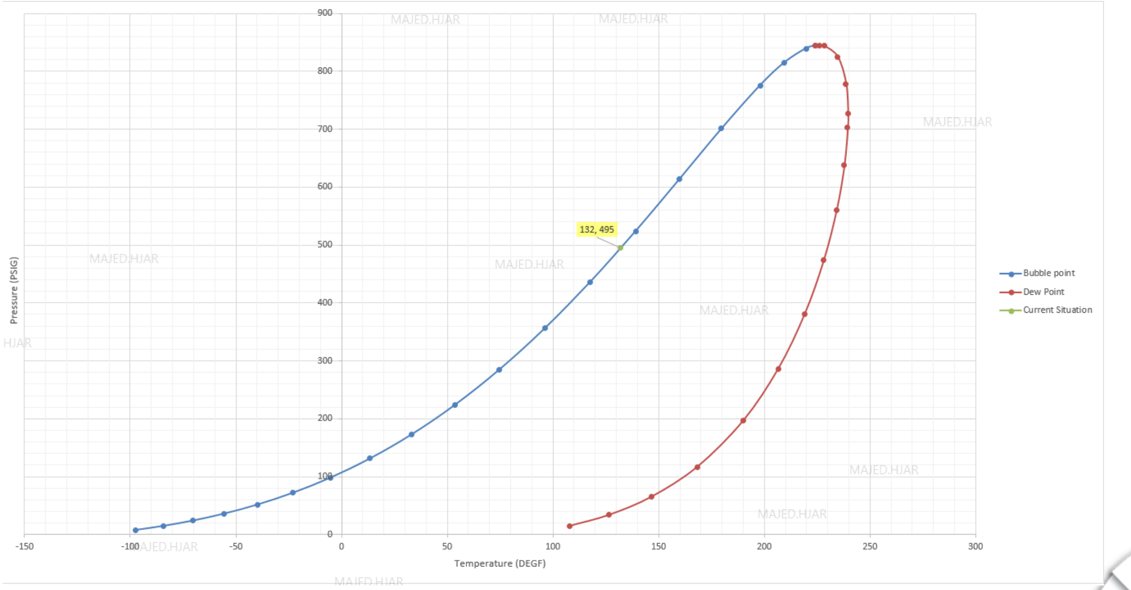
- Original mode of operation is 3 pumps running while 2 pumps are on stand-by mode (2016-2019).
- All three of the pumps were running on high recycle and hence the pumps were running more than 60% to the left of best efficiency point (BEP) on the curve (maximum allowable is 10%).
- Pumps were running under significant stress where flowrate used to drop as low as 1800 gpm. The drop in flowrate was identified as a causal factor of the reduced life span of the mechanical seals.

A comprehensive assessment was conducted to change the mode of operation from three pumps running to two pumps running. This was implemented, tested and proven successful during the last quarter of 2019

TEX Mode vs. JT Mode

- At Turbo-Expander (TEX) mode of operation (normal operation): the viscosity and specific gravity of the NGL product are 0.0676 cP and 0.4275, respectively. The NGL shipping pumps mechanical seals were designed considering TEX mode as the default mode of operation.
- The NGL temperature has the biggest factor on the primary cause of dry running the mechanical seals. Cooling water pumps are used to cool water by circulating the NGL in the heat exchanger. The product NGL is heated from 72°F to 124°F. Thus, NGL property will change as shown below.



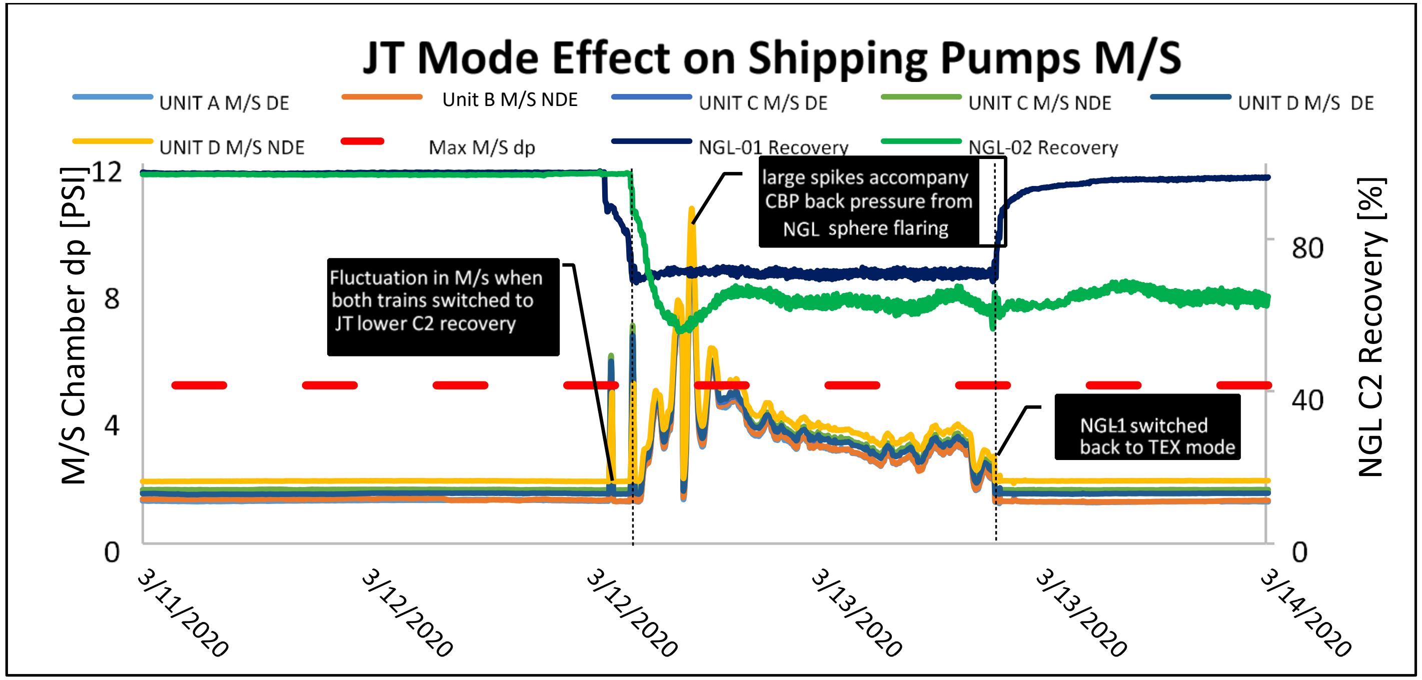


NGL bubble point curve showing NGL vapor pressure vs. NGL temperature. The curve on the right shows the effect of temperature increase which will change the NGL from liquid to gas.

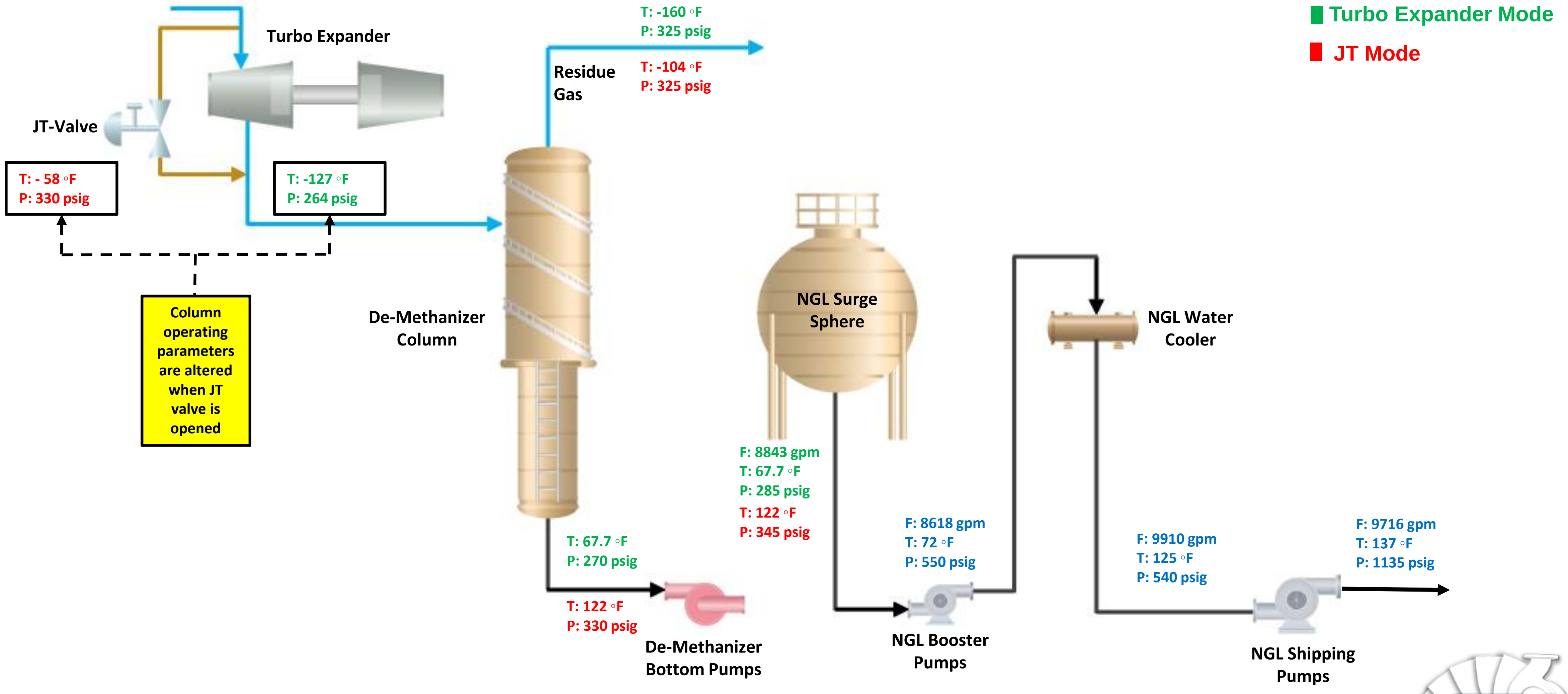
TEX Mode vs. JT Mode

- In Joule-Thomson (JT) mode, the NGL product becomes heavier and more viscous. The calculated NGL viscosity and specific gravity in JT mode were 0.0944 cP and 0.4922 cP, respectively. This translates to a 40% increase in viscosity and a 15% increase in specific gravity compared to TEX mode.

The higher viscosity of the process fluid reduces its ability to flow across the seal faces at the intended flushing flowrate. In this case, the flow of the process fluid across the seal faces is restricted, as the fluid has become heavier and more viscous, limiting its passage through the micro gap between the faces.



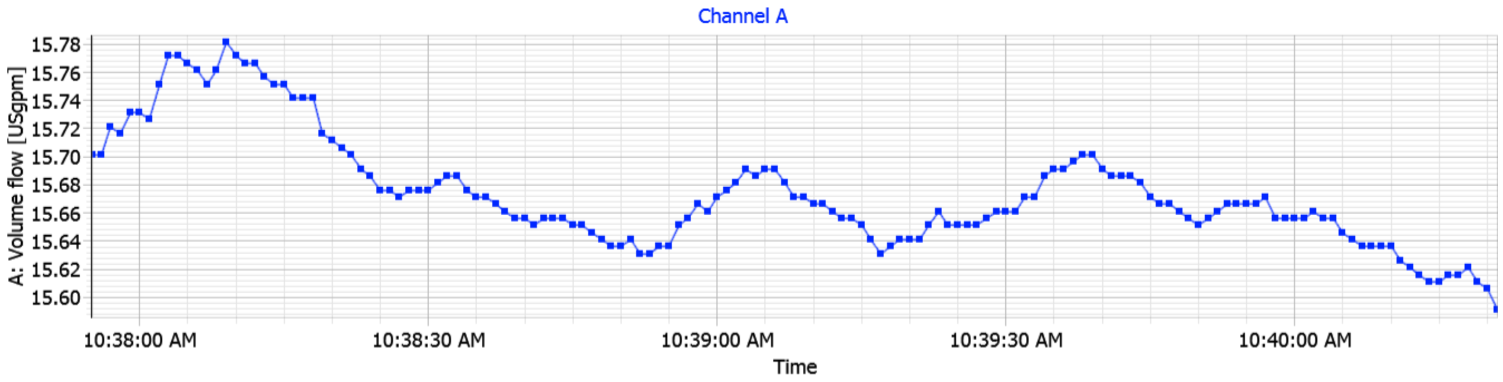
Process Description

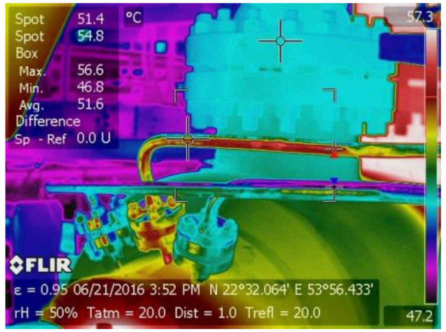


Ultrasonic Flowmeter Verification

Measurements were taken at the Drive End (DE) and Non-Drive End (NDE) for units C, D and B in the horizontal position. The readings showed that both units C and D had successful measurements indicating 14-16 gpm flow going to the mechanical seals.

Unit B, which had the bore orifice modification, did not show any signal and was not able to attain a volumetric flowrate. This is an indication of a possible phase change (vaporization) in the line. The bores were increased from 3.4 mm to 18.6 mm based on seal vendor recommendation. This modification was one of the reasons the pump continued to witness seal failures. The modification was only implemented to the branched orifices and not the main one as shown in the figure on the right.





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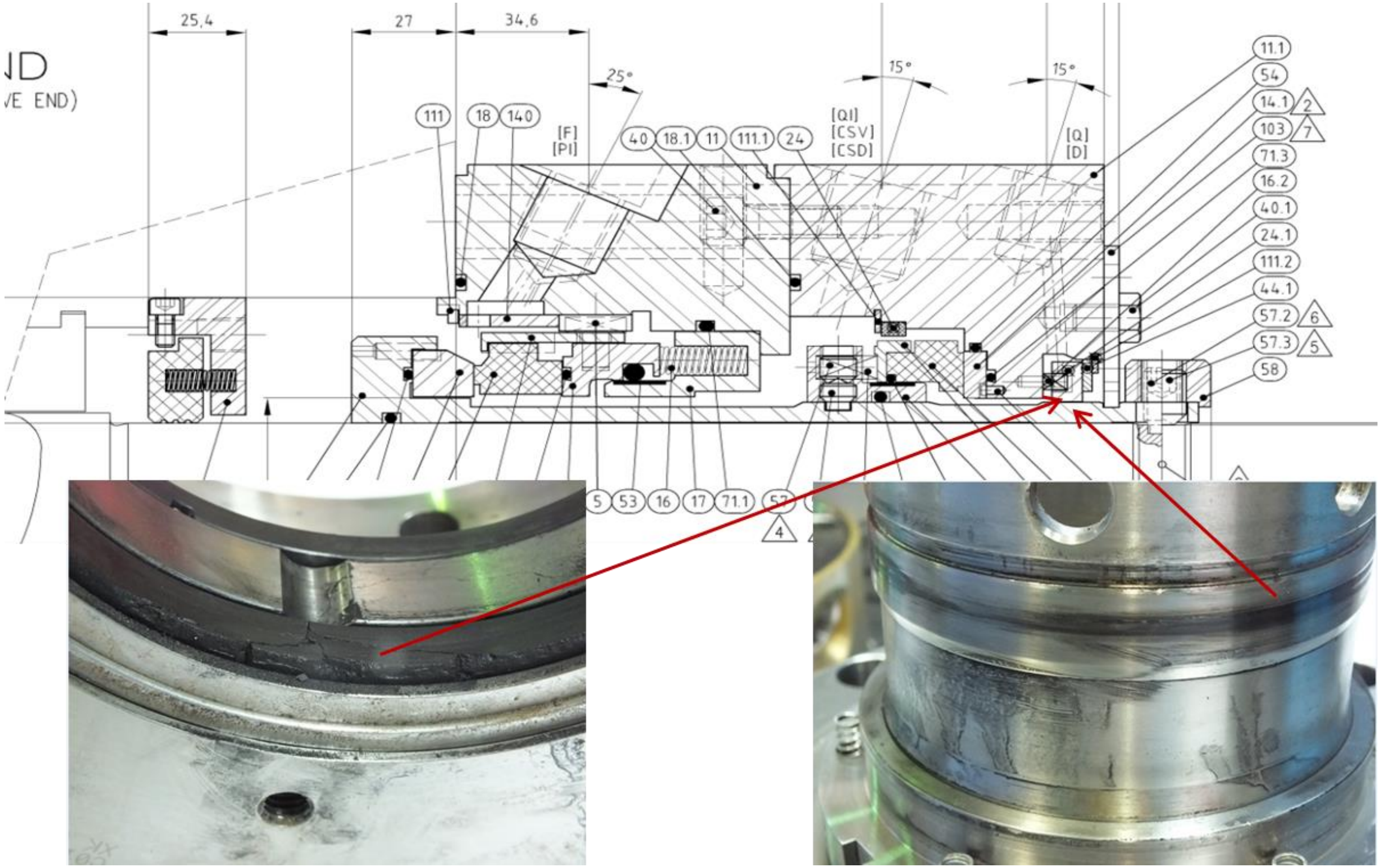
Mechanical Seal Investigation

Unit C:

- Gland bushing found damaged from contact with seal sleeve.



Gland Bushing

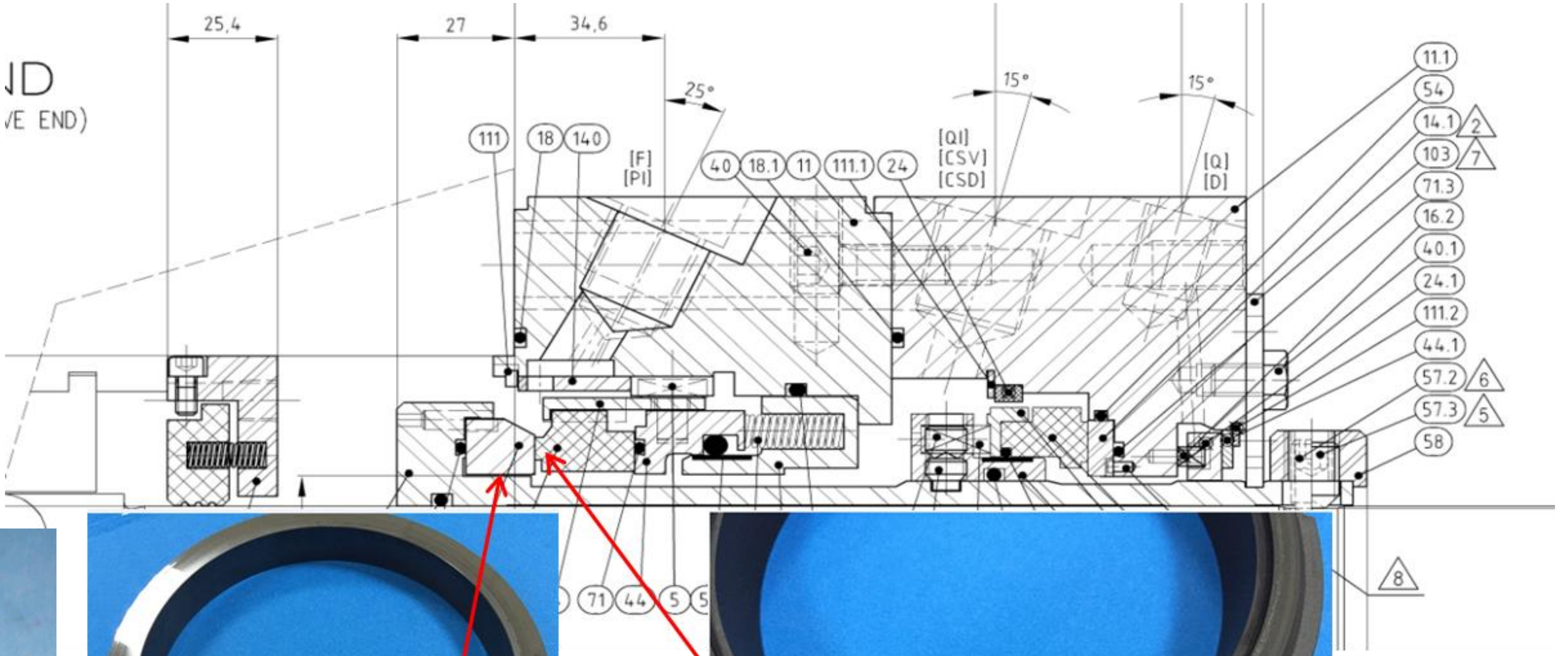
Seal Sleeve

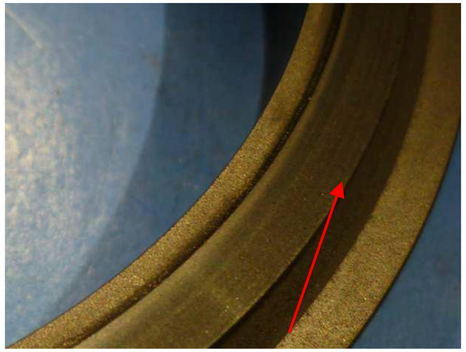
Mechanical Seal Investigation

Unit A:

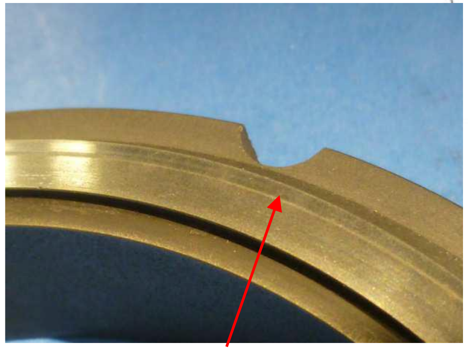
- Drive end seal showing Non-concentric wear tracks at contact surface on the stationary face. (right)

- Rotating face found polished signs of dry running & has an uneven wear. (left)

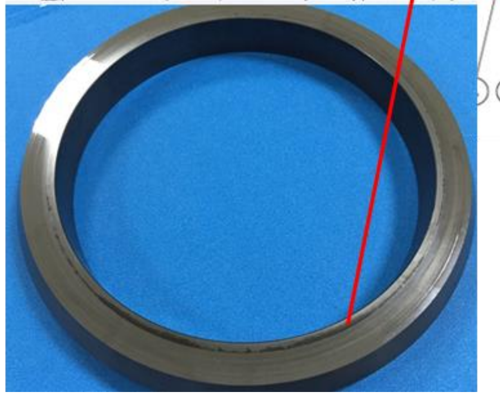




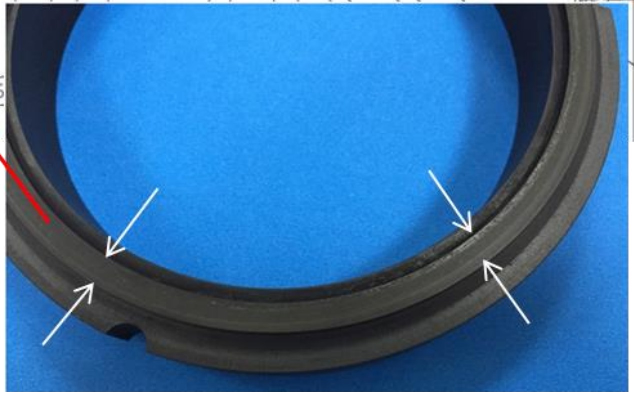
Top half line of contact



Bottom half line of contact



polished signs of dry running & uneven wear



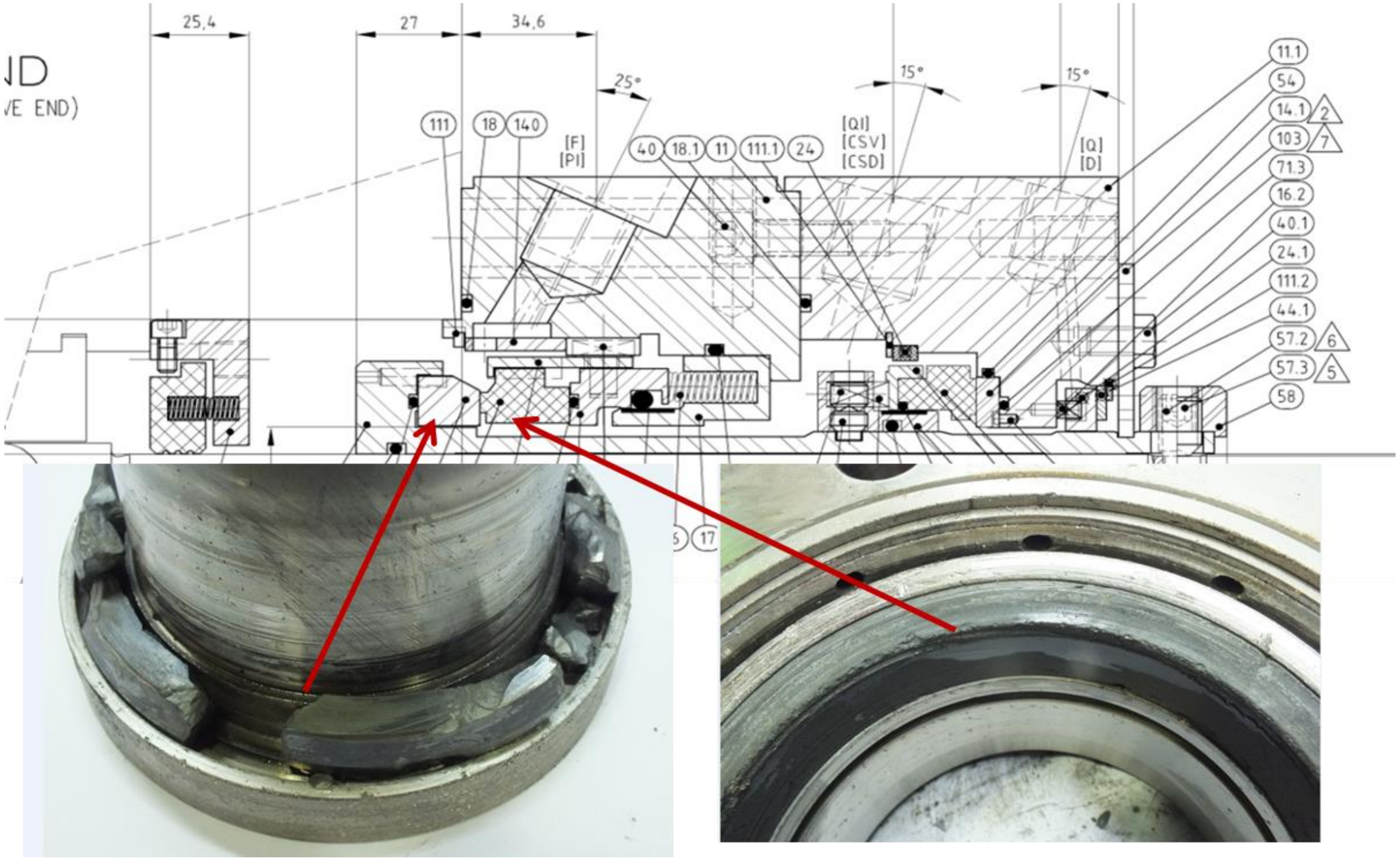
Non-concentric wear tracks

Mechanical Seal Investigation

Unit C:

- NDE seal showing severely damaged carbon Rotating face. (left)

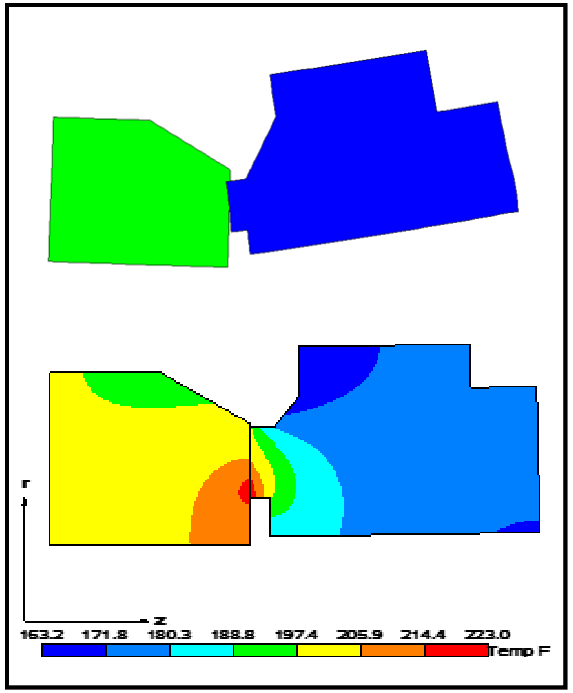
- Stationary face found completely damaged. (right)



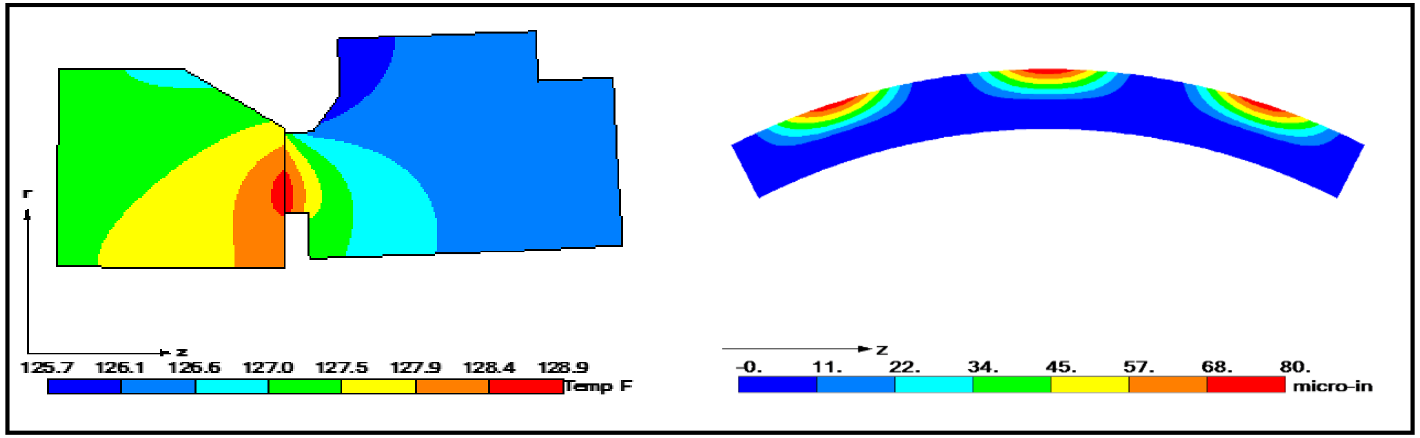
Rotating Face **Stationary Face**

Mechanical Seal Face Upgrade

- Mechanical seal installed on shipper pumps are designed for max 582 psig with 600 psig safety margin.
- pressure increase can increase seal face deflection causing a loss of lubrication.
- It was recommended by seal vendor to add face features, or waves, to improve lubrication.
- At 850 psig, the seal will generate heat around faces that can increase fluid temperature and therefore lower the vapor margin. In this case, the fluid inside the seal chamber will become vapor and seals that are designed to run on liquid will run on product vapor causing immediate damage.
- The upgraded seal face will generate much lower heat considering 850 psig. The leakage rate was reduced from 0.18 scfm to 0.021 scfm.



Old seal face design total deflection



New seal face design total deflection and face profile

Usually, the leakage increases with such new seal face design. However, leakage can remain very low for light fluid in C2 molecules.

Recommendation

The below recommendations were implemented during 2016 and 2017. These recommendations helped to identify the primary root failures of the seals:

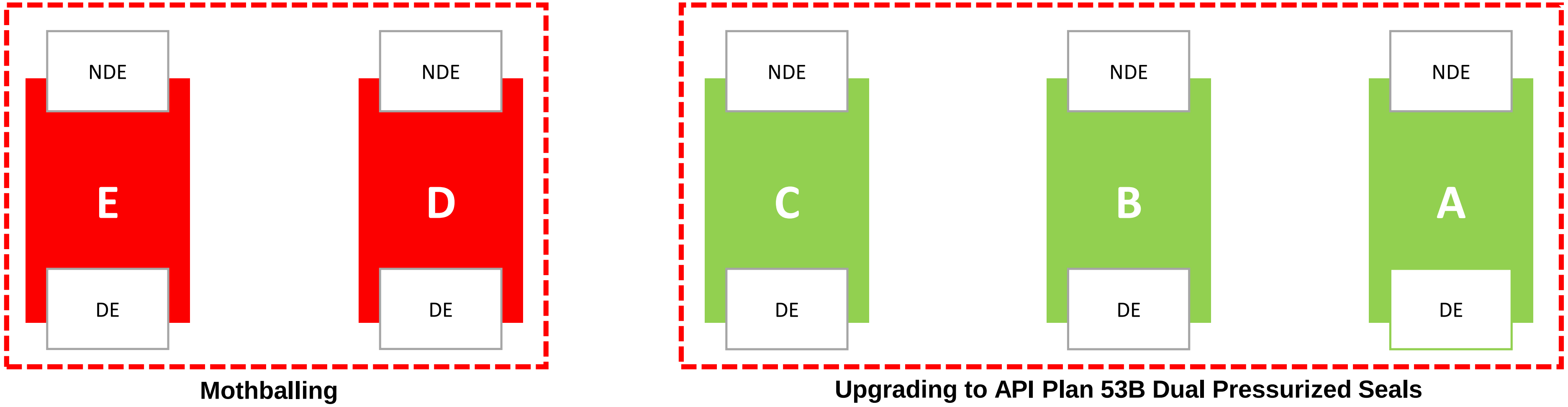
- Draining the methanol should be implemented each time before starting-up the pumps.
- Checking the valve for possible clogging or ice formation.
    - CSD should be closed during operation
    - Plan 51 reservoir should be closed during operation
    - Plan 76 vent should be open.
- Monitoring the differential pressure of the suction strainer. It should not exceed 1 bar (14.5 psig). If it reaches such limit, pump should be stopped immediately and strainer should be cleaned.
- Upgrading seal face design to improve lubrication. This recommendation was implemented in 2017. the MTBF has increased from 4-6 months to almost 12 months for most of the seals. The 6 months improvements helped the plant to secure more spare seals. The seal faces were modified to better suit the actual suction pressure which was found to be higher than the original process design conditions.

The Below recommendations were implemented in 2019. These had major impact in increasing MTBF of the seals:

- Changing the mode of operation from three pumps running to two pumps running. The seals MTBF was less than 12 months but it was improved to 16 months after reducing the number of running pumps as the pumps used to run far below rated conditions. This was implemented during the last quarter of 2019.
- As it was recoded that suction pressure exceeded 837 psig while the pump is on standby mode, it was recommended to either keep the suction isolation valve open or to keep the pump minimum flow recycle line open when the pump is on standby mode (but no both of them). This recommendation was implemented in 2019.

Long Term Solution

A detailed assessment took place in 2020. The assessment concluded that upgrading the mechanical seals piping plans to 53B dual pressurized seals is the most viable option. It was recommended to upgrade the seal piping plan for three pumps only (out of five). Such upgrade will eliminate the dry running issues witnessed on the failed mechanical seals. This upgrade will expand the MTBF of the mechanical seals. A similar pump experience was also cross-referenced with another end user and found satisfactory with MTBF of 6 years. It was also recommended to mothball two pumps since they will not be used.



Conclusion

The primary factor that increased the seals MTBF to almost 16 months is by changing the mode of operation from three pumps running to two pumps running as the pumps used to run far below rated conditions. Another primary factor observed after inspecting the seal faces is dry running. The seal faces were not designed properly to handle the high suction pressure, liquid specific gravity and liquid viscosity. Thus, upgrading the seal faces was implemented. Both solutions have resulted in increasing the MTBE from less than 6 months to almost 16 months.

It is very clear that these pumps have unusual high variation in the process conditions. This includes significant changes in suction pressure, liquid temperature, vapor margin, chemical composition, liquid viscosity and specific gravity. The site checks performed helped to focus more on the real issues.

For that reason, having a dual unpressurized seal design that fits all these conditions will mean that the seal life will definitely be reduced as witnessed for the existing seals.

For light hydrocarbon with specific gravity below 0.5, the international seal standard API-682 recommends the use of dual pressurized seals which utilize barrier liquid (typically seal oil) as the lubrication media for the seal faces. This will significantly reduce the effect of the variant process conditions on the seal life and can provide the targeted 3 year seal life. Thus, it was recommended to upgrade the mechanical seals to dual pressurized seals with plan 53B as it is the most viable technical option.