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A High Bearing Temperature Troubleshooting of Centrifugal Heat-Pump Compressor

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0. Bio of Presenter

Woonsil Lee is currently a Senior Rotating Machinery Engineer at Hyundai Oilbank in Daesan, South Korea. He worked for Hyundai Engineering starting in 2009 as a Rotating Machinery Engineer and provided engineering service for Pumps, Compressors and Package items in Oil & Gas Projects. Then, he joined the Engineering Team of Hyundai Oilbank in 2013 and worked in CDU and HOU Revamping Projects and Retrofits of Rotating Equipments.

Woonsil Lee received a B.S. degree in Mechanical Engineering in 2009 from Youngnam University. He is presently a visiting fellow to KIMM(Korea Institute of Machinery & Material) and pursuing a M.S. degree in Mechanical engineering at Hanyang University.

0. Abstract

On September 2013, a centrifugal heat-pump compressor began showing an abnormal temperature rise in its tilting pad journal bearing (DE-side). The phenomenon appeared when increasing the rotor speed following a process of load-up.

This case study presents the chronology of the investigation conducted to identify the root cause of the unacceptable temperature rise: ***Troubleshooting* the cause of *High Bearing Temperature***

The troubleshooting process took several months while there were severe economic losses as the compressor could not deliver its full load

1. Executive Summary

Hyundai Oilbank Co., Ltd. is an oil processing company located in Daesan, South Korea with a refining capacity of 390,000 barrels per day. The company has 102 operating compressors in its factory where diverse O&G products and derivatives are produced.

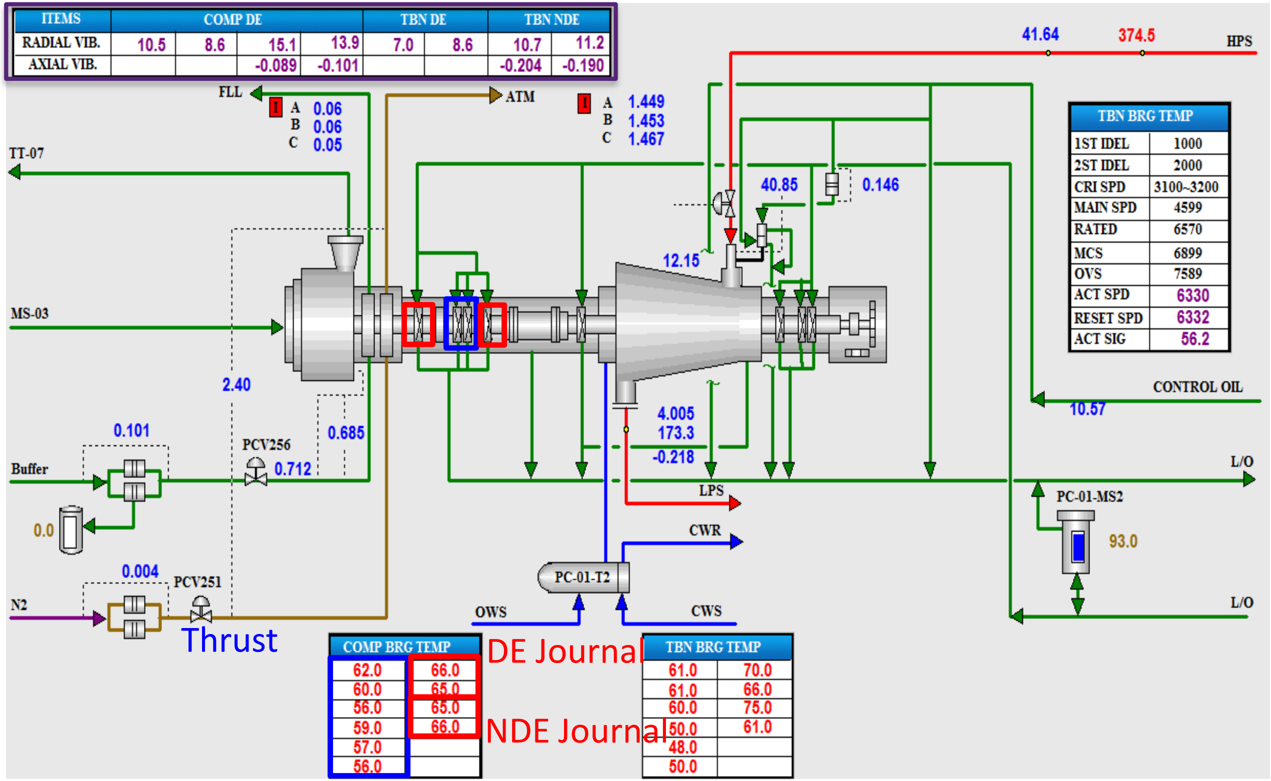
1. Executive Summary

The machine in question is an one-stage overhung centrifugal heat-pump compressor that supplies heat to the Reboiler in the propylene (PRP) process. A 6.8 MW steam turbine drives the compressor.

On September 2013, The temperature in the compressor DE bearing rose to around 100 ^{o}C in a short time when increasing the shaft speed while following a process load-up. The investigation to identify the root cause of the unacceptable temperature rise and the methodology for solution follow.

2. Background

➢ Overview of Machine Monitoring



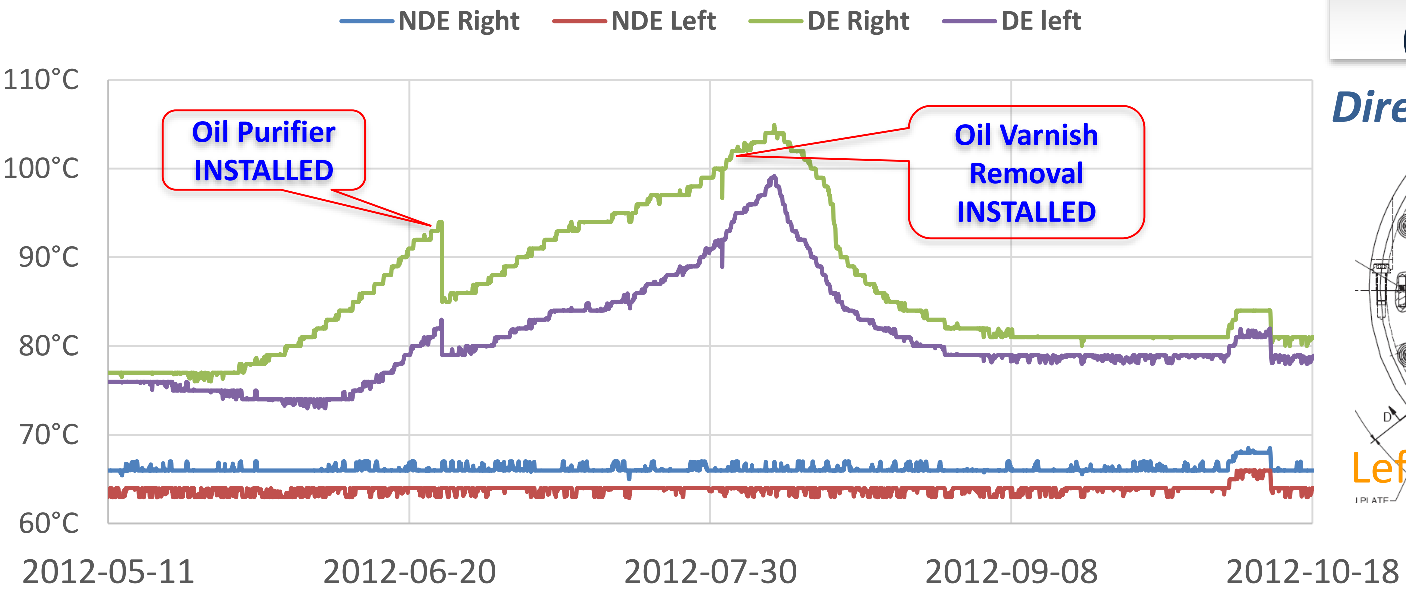
- ✓ Speed: 6100 ~ 6500 rpm
- ✓ Three (3) RTD for each Thrust bearing
- ✓ Two (2) RTD for DE radial bearing
- ✓ Two(2) RTD for NDE radial bearing
- ✓ Vibration sensors for each radial bearing

3. Problem History and Some Actions

✓ In fact, a similar rapid temperature rise had occurred in the same bearing in May 2012.

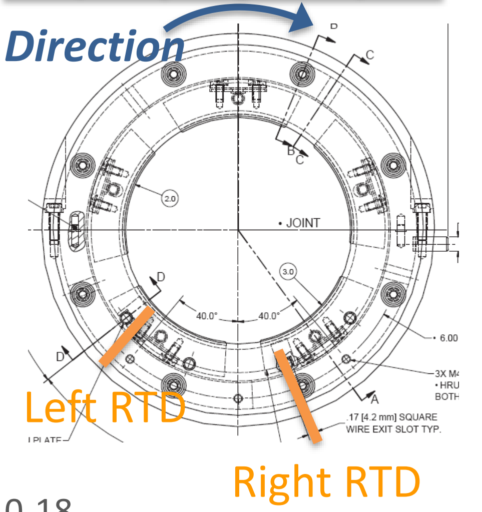
✓ Then the cause of the problem was judged to be **Oil Varnish**; hence, an **Oil Purifier and Varnish Removal Equipment** were installed in the Oil Delivery System. After this action, the DE-BRG temperature decreased and stabilized at around 80°C.

✓ **But a similar problem occurred one year later!**



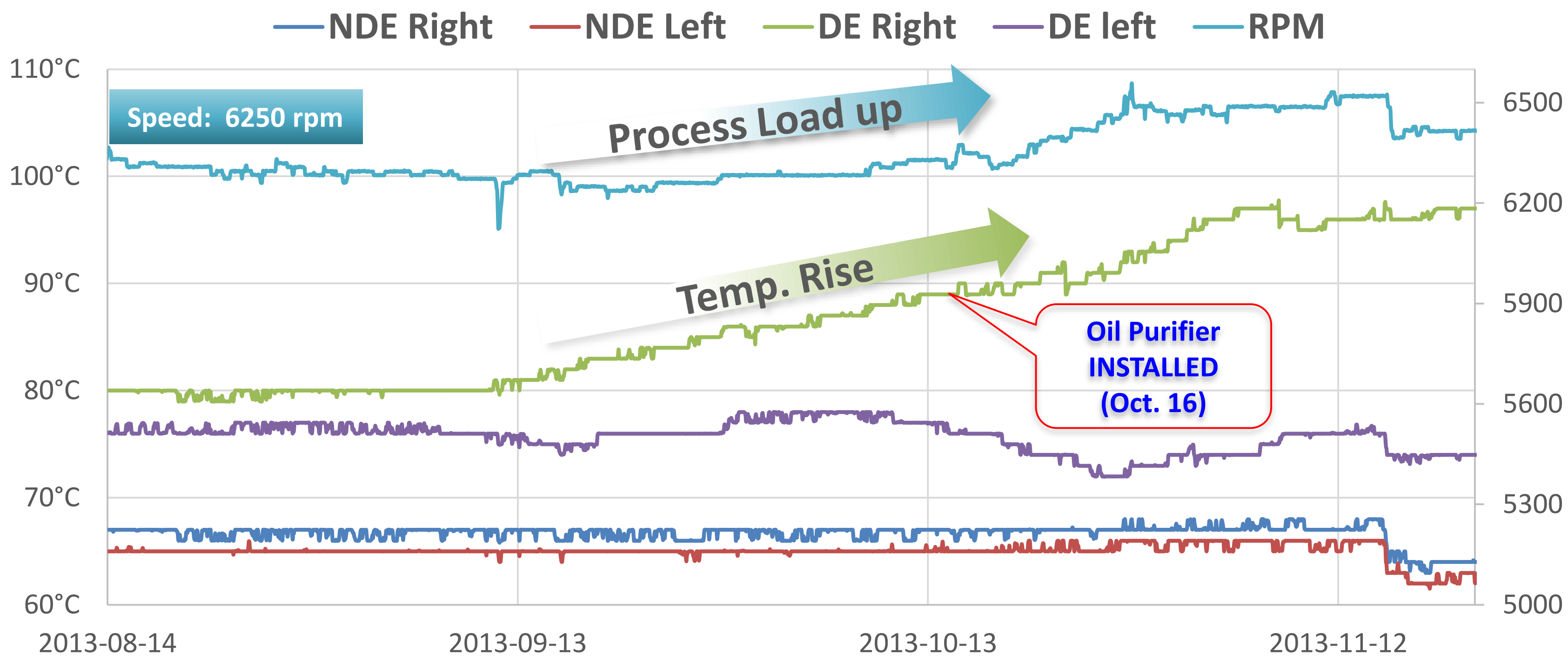
TE Position

(Coupling view)



3. Problem History and Some Actions

In Sept. 2013, the phenomenon appeared again when increasing the rotor speed following a process of load-up. After installing the **Oil Purifier** and **Varnish Removal Equipment**, the DE-BRG temperature was stabilized at around 100℃.



3. Problem History and Some Actions

Although further temperature rises stopped, since the compressor rerating (to increase the rated speed from 6899 rpm to 7109 rpm) would be planned in 2015 in combination with the entire process revamping, this high temperature problem at the DE-side bearing should have been resolved in advance in order to remove any potential risks.

Thus, some actions were conducted to verify integrity of operating conditions.

➢ **INSPECTION & TEST**

I. Lube oil inspection (samples tested) to check physical & chemical integrity (Lube Oil type: ISO VG 46) → **Result : Adequate**

II. Lube Oil **Supply pressure** increased from 1.25kg/cm2 to 1.40kg/cm2 (18 → 20 psi) to increase flow rate. → **Result : 1℃ reduction in oil temperature.**

III. Lube Oil **Inlet Temperature** increased to reduce its viscosity (39℃ → 42℃)

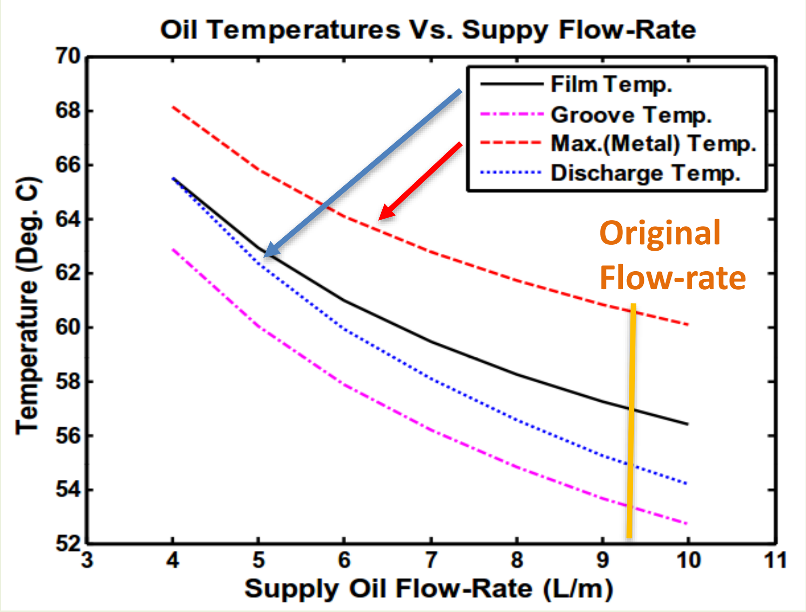
→ **Result: All bearing temperatures increased as much as the supply oil temperature increased.**

4. Problem Grasping-1. Analysis to Reduce BRG Film Temperature

➢Design differences between DE and NDE bearings were checked, and **predictive analysis** carried out and a technical solution taken into account to reduce temperature.

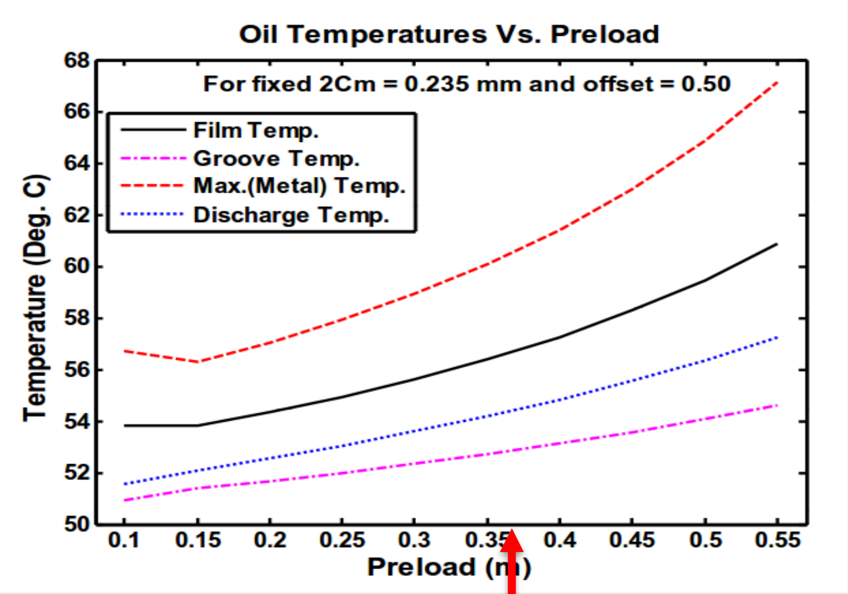
Specification	NDE JNL BRG	DE JNL BRG
Bearing configuration	5-Pad	5-Pad
Journal dia. (mm)	152.25	101.6
Bearing length (mm)	66.8	44.5
Pivot offset	0.50	0.50
Preload	**0.365**	**0.351**
Rated speed (rpm)	6899	
W/LD (kPa)	**227.2**	**174.5**
Supply Flow Rate (liter/min)	**27.4**	**9.3**
Oil type	ISO VG 46	
Oil supply temp.(℃)	40	

Solution (1) Effect of Supply Oil flow rate checked. Significant improvement (reduced T) when increasing flowrate to the bearings.



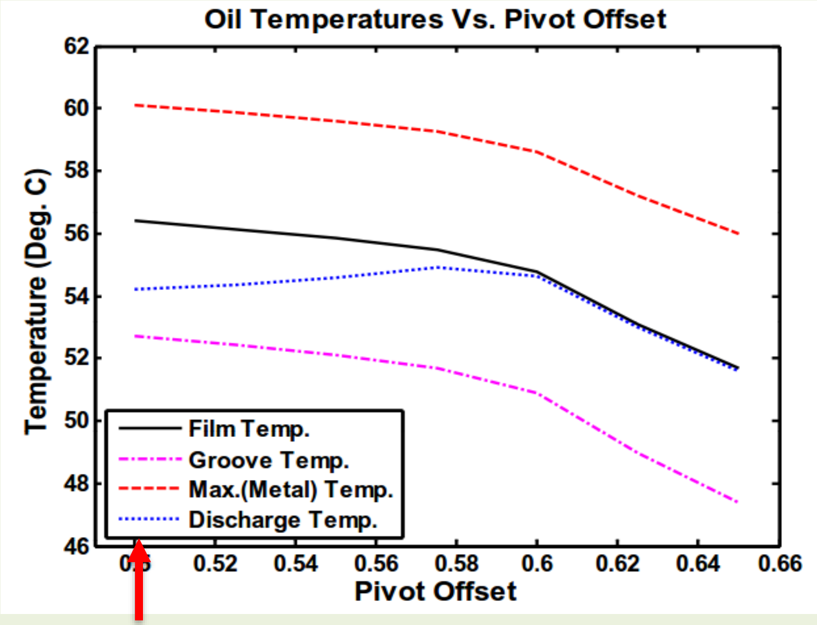
4. Problem Grasping-1. Analysis to Reduce BRG Film Temperature

Solution (2) **Effect of pad preload** with fixed machined clearance checked.



Original Preload(0.351)

Solution (3) **Effect of Pivot Offset** checked.



Original Offset(0.5)

4. Problem Grasping-1. Analysis to Reduce BRG Film Temperature

Solution (4) **Effect of Oil viscosity** checked.
Comparison of predicted oil temperatures for LBP and LOP configurations shows no difference.

Oil Type	Oil Temperature prediction Result(℃)		Analysis Parameters
	Film	Max (Metal)	
LBP with VG46	56.40	60.09	Offset=0.50 Preload=0.351 2Cm=0.235mm Oil flow=10 l/m
LOP with VG46	56.40	60.09	
LBP with VG32	53.36	56.37	

The results of the predictive analysis indicate that if the bearing design could be changed or adjusted by combining the above four design variables, the oil film temperature could be reduced considerably

4. Problem Grasping – 2. Clarification w/ Manufacturer

➢ During face-to-face meetings, differences were discovered or identified between the bearing designs in actual operation and those submitted.

Received in 2009, During **Eng. Stage**	
Supply Flow Rate (l/m)	**9.3**
Pivot offset	**0.50**
Preload	**0.351**

Discovered in July 2014	
Supply Flow Rate (l/m)	**5**
Pivot offset	**0.55**
Preload	**0.574**

➢ During the Detail Engineering Stage, **the compressor manufacturer received & followed a recommendation from the bearing supplier to deliver oil flow-rate at 5 l/m.** This oil flow-rate was much lower than the original calculated value of 9.4 l/m.

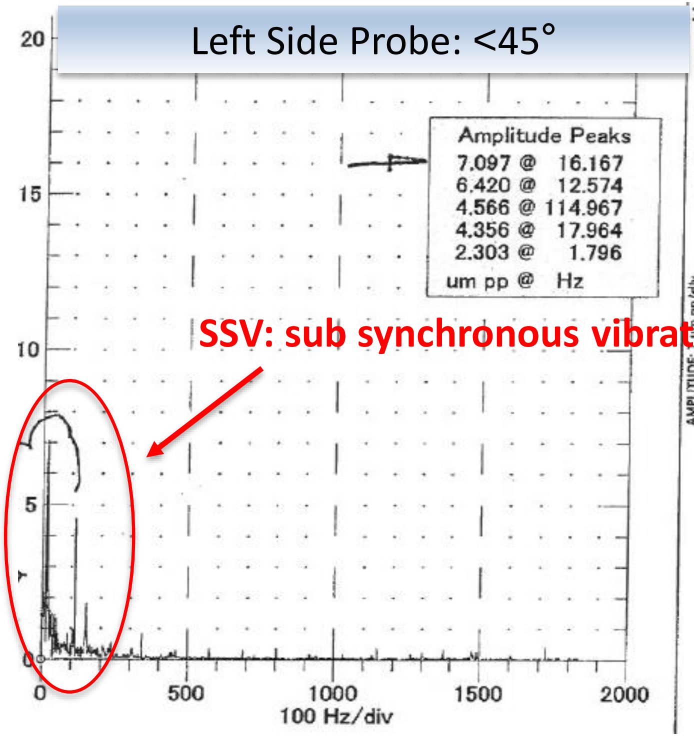
Specification		Comp. NDE BRG	Comp. DE BRG
Size		6” x 2.63”	4” x 1.75”
Oil flow	Bearing Vendor	22.0 l/min	**5.0 l/min**
	Comp. Vendor	27.4 l/min	**9.3 l/min (or 10)**
Orifice Hole	Bearing Vendor	Φ 5.8	Φ 2.8
	Comp. Vendor	Φ 6.5	Φ 3.8

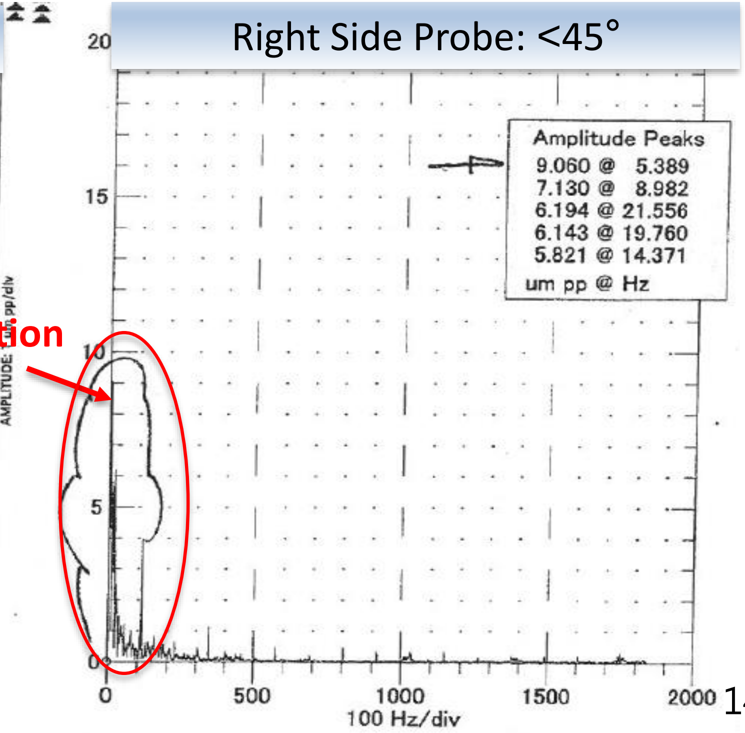
4. Problem Grasping – 2. Clarification w/ Manufacturer

➢ In addition, since SSV Harshness was observed during a Mechanical Running Test of unit, the designed **Pad Preload of the bearing was changed from 0.351 to 0.574** (with Pad Shim) to eliminate the low frequency vibration.

TEST CONDITION

- AT DE side Journal Bearing
- Rpm:6898 (MCS)
- Temp: 38.9℃
- Pressure : 117kpa(16.9psi)
- Flowrate: 5 l/m
- Preload: 0.351





4. Problem Grasping – 3. Root Cause Analysis

➢ Using the actual Bearing Design Data., the analysis was run again. The result indicated a **Side Leakage**, the minimum oil consumption of the bearing, estimated at 5.94 l/m. Hence, the DE BRG operated under a **"Starved flow condition."**

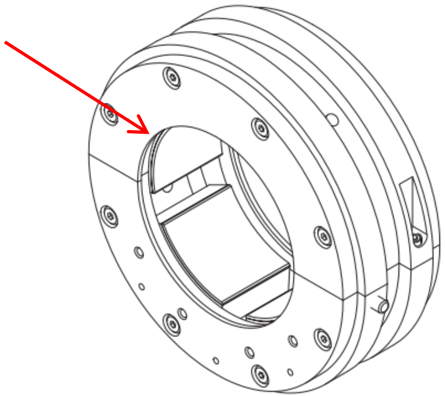
Oil type	ISO VG 46
Supply-Oil Temp. (°C)	49
Supply Flow Rate (l/m)	**5.0**
Pivot offset	0.55
Preload	0.574
Max. Temp. (°C)	75.7
Outlet Temp. (°C)	70.9
Side Leakage (l/m)	**5.94**
Min. Calculated/Given Flow Rate (l/m)	**5.94**

Starved Flow Condition

4. Problem Grasping – 3. Root Cause Analysis

➢ In addition, since the **End Seals' Clearance** on the bearing = 1.52 mm >> than Normal Clearance (0.4 ~ 0.66 mm), the bearing would have actually operated under a **"much more starved flow condition."**

Bearing Maker	End seals Clearance(mm, Dia.)	Remark
A	**1.52**	**Existing Bearing**
B	0.4~0.65	
C	0.64	
D	0.66	



➢Meanwhile, the compressor manufacturer and the bearing supplier increased the **Preload at the 0.351 given Pad Clearance** to 0.574 thereby deteriorating further the operating temperature of the bearing. This was done in despite that **SSV Harshness could be removed by increasing the oil flow-rate**.

5. Trouble Shooting

➢ Case Studies were conducted to search appropriate values of design variables to reduce bearing temperatures. Based on the predictions, a new bearing design was resulted.

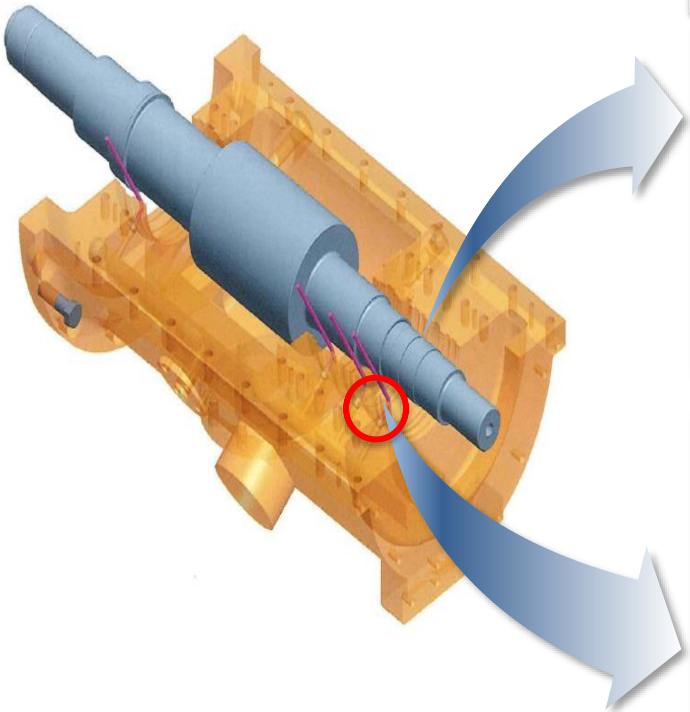
ISO VG 46 at 49°C	Case (1) , **Existing Design**	Case (2), Oil flow increase	Case (3), Preload change	Case (4), Oil flow increase	Case (5), Rerate Design
Rpm (MCS)	6899	6899	6899	6899	7109
Supply FlowRate (l/m)	5	10	10	15	15
Preload	0.574	0.574	0.351	0.351	0.351
PREDICTION Max. Metal Temp. (°C)	75.7	72.0	64.7	62.6	63.0
Side Leakage (l/m)	5.94	5.94	6.69	6.69	6.89
Min. Calculated/Given Flow Rate (l/m)	5.94	10.0	10.0	15.0	15.0
Running Condition	**Starved**	**Flooded**	**Flooded**	**Flooded**	**Flooded**

* Bearing Seal Clearance is 0.42mm as per Bearing Manufacturer Guideline

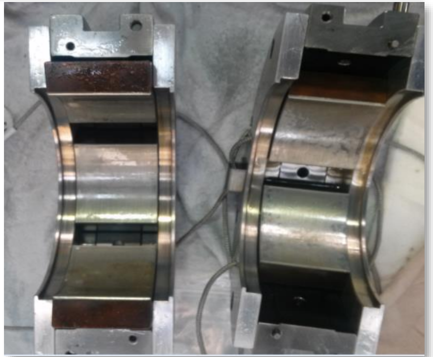
* Changing Lube Oil type was not considered due to convenience for maintenance work.

5. Trouble Shooting

➢ The purchase of new DE side Bearing, reflecting the final Retrofit Design, was followed immediately. The bearing was replaced during the maintenance and rerate work in **May 2015**.



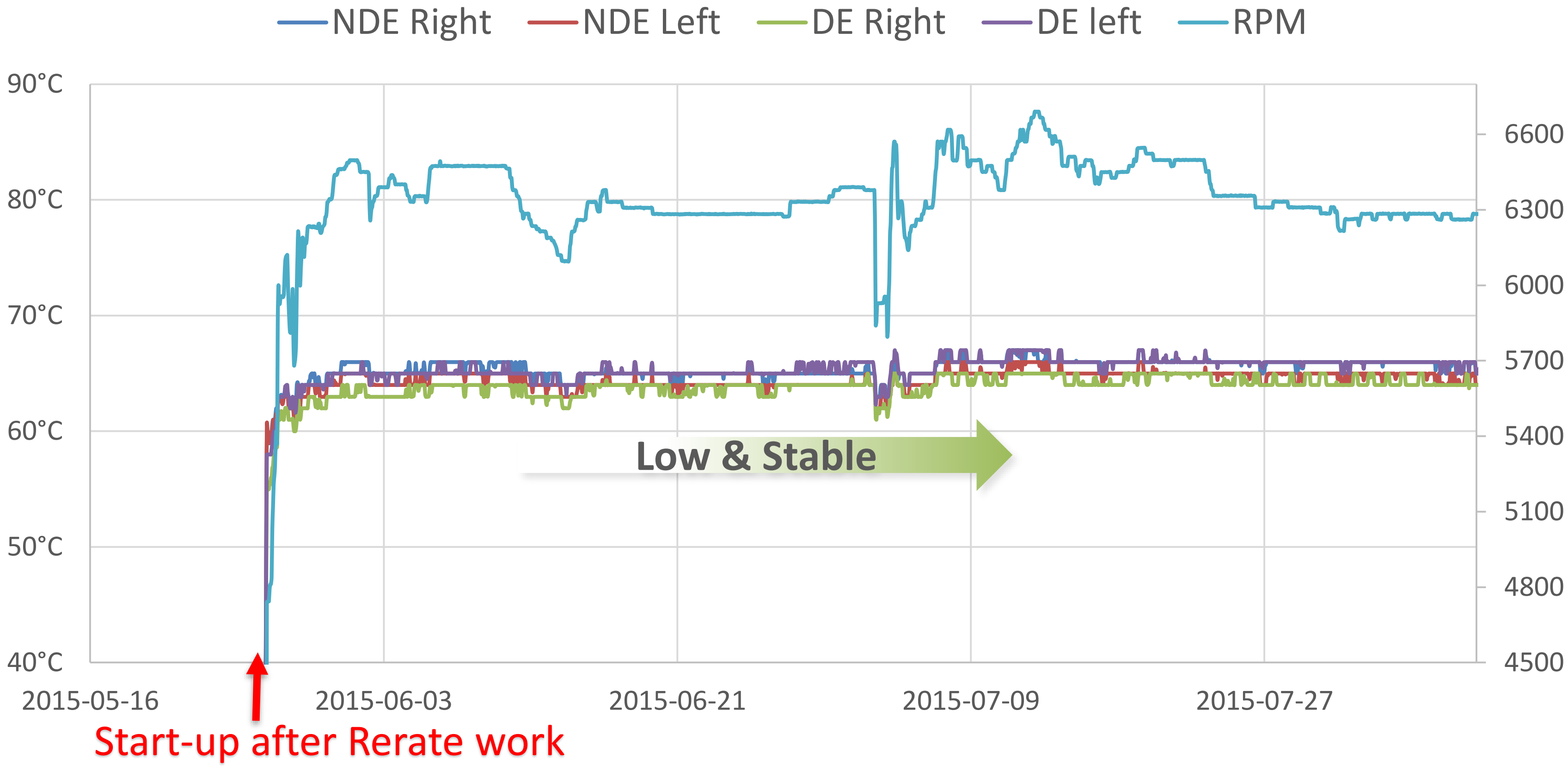
Replacement of DE Journal Bearing



Replacement of Orifice to increase Oil flow rate

6. Verification of Performance after Bearing Replacement

➢ DE bearing Temperature Vs PRM trend is also steady, in spite of speed variation.



7. Lessons Learned

- In the process of solving the problem, several lessons were learned.

Design & Engineering

- Minimizing misunderstandings in the process of clarifying with Sub-Suppliers is important
- **Design changes in major components such as bearings should be always notified to the ordering body, jointly reviewed and appropriately judged during the project engineering**.

Machine problems occurring during operation

- The root-cause should be found and remedial actions taken to prevent reoccurrence of the same problems.
- If the fundamental cause had been analyzed and appropriate actions taken in 2012 according to the root-cause analysis, **when the problem had occurred for the first time,**

→ Operation losses of about several million USD in the following year (2013) due to the **failure of the compressor to follow the process load-up would have been prevented.**

Thanks and Questions

ACKNOWKEDGEMENTS

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