44TH TURBOMACHINERY & 31ST PUMP SYMPOSIA
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GEORGE R. BROWN CONVENTION CENTER

Piping Load Effect on Shaft Vibration in a Multi-Stage Barrel Type Boiler Feed Pump

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Maki M. Onari:

- *Mechanical Solutions, Inc. – Manager of Turbomachinery Testing - Principal Engineer: responsible for all MSI Turbomachinery Testing*
- *B.S.M.E., Zulia University, Venezuela*
- *Rotating Equipment Engineer, PDVSA: responsible for the predictive maintenance of one of the largest petrochemical complex in Latin America*
- *Co-Author Pump Vibration Chapter, McGraw-Hill Pump Handbook*
- *Member of ASME and the ISO TC108/S2 Standards Committee for Machinery Vibration*

Keith Munn:

- *AEP Welsh Power Plant Maintenance Supervisor – 5 years: responsible for the mechanical maintenance of the HP & LP turbines, boiler feed pump turbines, boiler feed pumps, generators, valves, condensers, HP and LP heaters, water treatment facility, cooling towers and all other pumps in the electrical process.*
- *Welsh Maintenance Planner- 3 years: performed planning work for all mechanical maintenance.*
- *Welsh Station Machinist- 25 years: performed mechanical maintenance on all plant equipment.*

Gary Krafft :

- *HydroTex Dynamics, Inc. for 17 years as a technical and sales representative*
- *BSME Texas A&M University*
- *Registered Professional Engineer in the State of TX*
- *22 years in the electric power generation industry. He originated TXU's Equipment Repair Group formed in 1982 to improve reliability with rotating machinery. He has worked in fossil and nuclear plants as well as handled projects for gas pipeline and mining.*
- *He was a member of the Pump Symposium Advisory Committee for 5 years co-authoring one tutorial and leading discussion groups.*

General Information

American Electric Power (AEP) – Welsh 1 Power Plant in Pittsburg, TX
Base load: 1500 MW Coal Fired Power Plant (Welsh 1, 2, and 3 combined)

The Turbine Driven Boiler Feed Pump (TDBFP) was installed in 1974 - 1975
Continuous operation with one 100% BFP per Unit

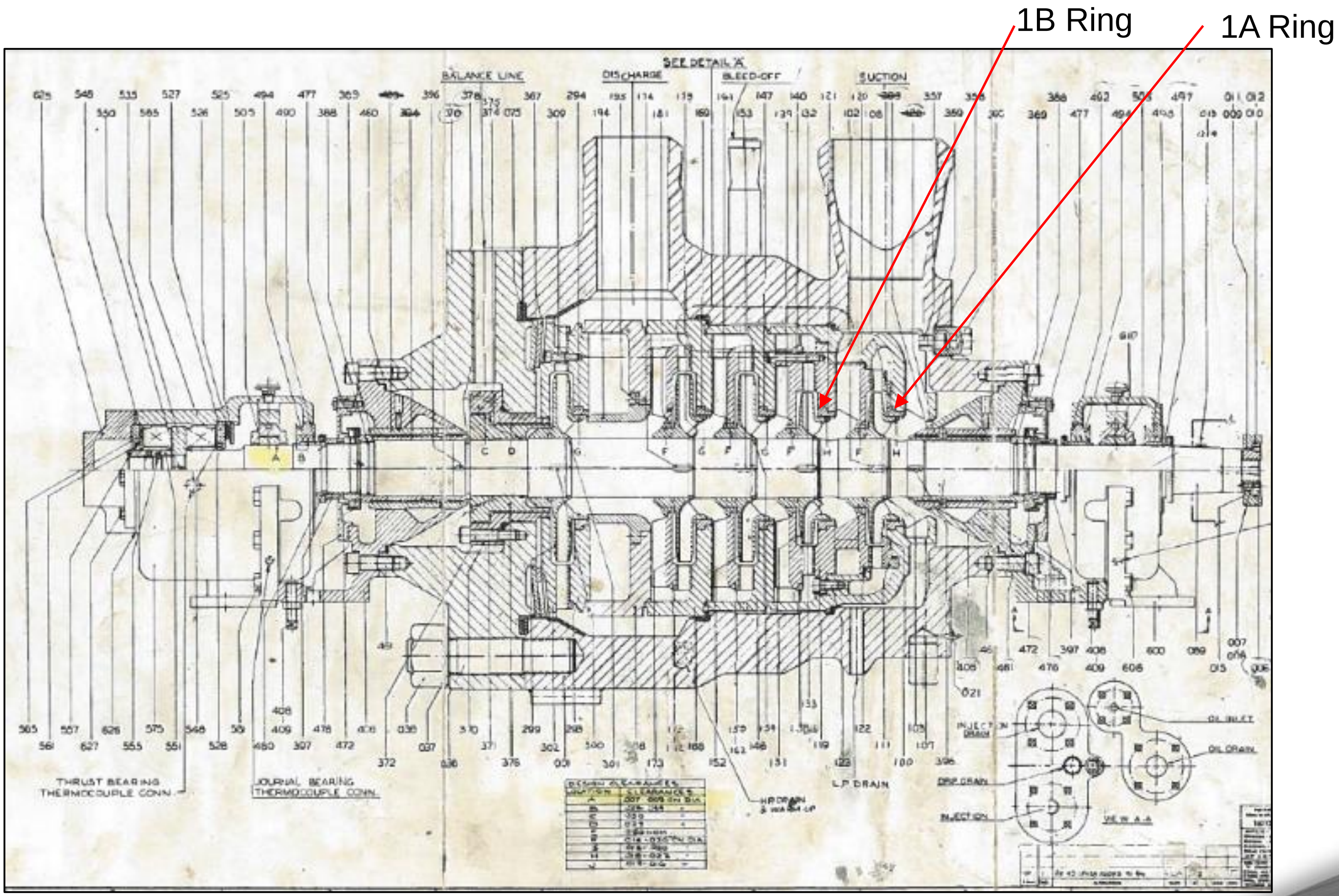
Type:	4 stage Double Case with Twin Suction and 2nd stage bleed-off
Rotation:	CCW Viewed from the Driver
Speed:	4,000 rpm to 4,860 rpm (66.7 Hz to 81 Hz)
Capacity:	9126 GPM
Suction Pressure:	216 psig
Discharge P.:	3016 psig
Total Dynamic H:	7,370 ft
Temperature:	375 ºF
Specific Gravity:	0.877
BHP:	17,530 HP (steam turbine)

5-pad tilting pad bearings with forced lubrication system
Bearing nominal clearance: 7 to 9 mils diametral

4 stage Boiler Feed Pump
Welsh 1 - TDBFP

Viewed from north

Welsh 1 – TDBFP Cross-Sectional Drawing



Viewed from south

Background

- Welsh 1 plant was built using a De Laval BFP (1974), while Welsh 2 and 3 were built with Pacific Pumps. All three plants have the same Westinghouse steam turbine.
- In May 2009, the pump case was weld repaired and machined on-site to correct potential concentricity and parallelism issues. The train ran without vibration concerns for a few months.
- In September 2009, elevated shaft vibration levels on the entire train were detected. It was concluded that the vibration was most likely due to internal rub and coupling alignment.
- In May 2011, HydroTex (HT) supervised a head in place balance assembly change out.
- In December 2011, a vibration issue in the coupling area was detected. Vibration apparently had started rising in late November 2011. HT inspected the pump and turbine bearings. The pump inboard X & Y amplitude difference increased after this inspection.

Background

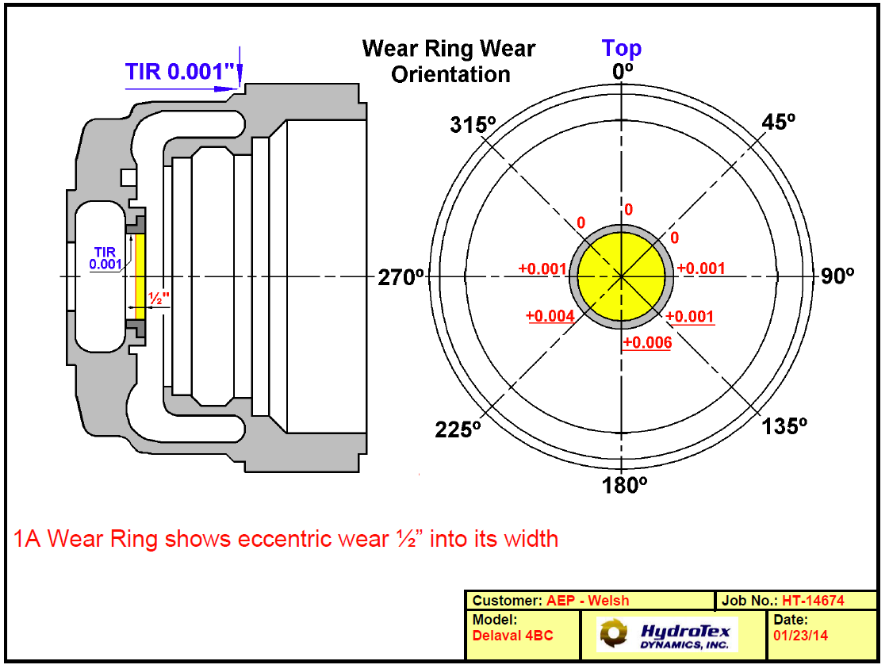
- In March 2012, HT took cold and hot alignment readings to determine a new coupling offset target. The new target was implemented in the vertical direction and a little off target in the horizontal direction. The entire machine train came up with low vibration levels and reduced sound from the turbine exhaust hood. After a few days and low load operation, the vibration gradually step increased, resulting in an X reading of above 3.0 mils pk-pk (pump inboard bearing or IBB), while the Y probe indicated 1.5 mils pk-pk. Vibration increased about 0.5 mils pk-pk every night until 3.0 mils pk-pk was reached at the IBB-X probe.
- In September 2012, the pump inboard bearing housing was machined by HydroTex Houston shop, without improvement on the IBB-X probe magnitude.
- From December 15, 2012 through January 5, 2013 the IBB-X probe was indicating 2.5 mils pk-pk (below alarm level of 3.5 mils pk-pk).
- It appeared that the alignment has been playing a significant role affecting the vibration readings (pump sensitive to the alignment). There was a time when slight adjustment of the seal water drain temperature would instantly change the IBB-X from 3.0 to 1.5 mils pk-pk.

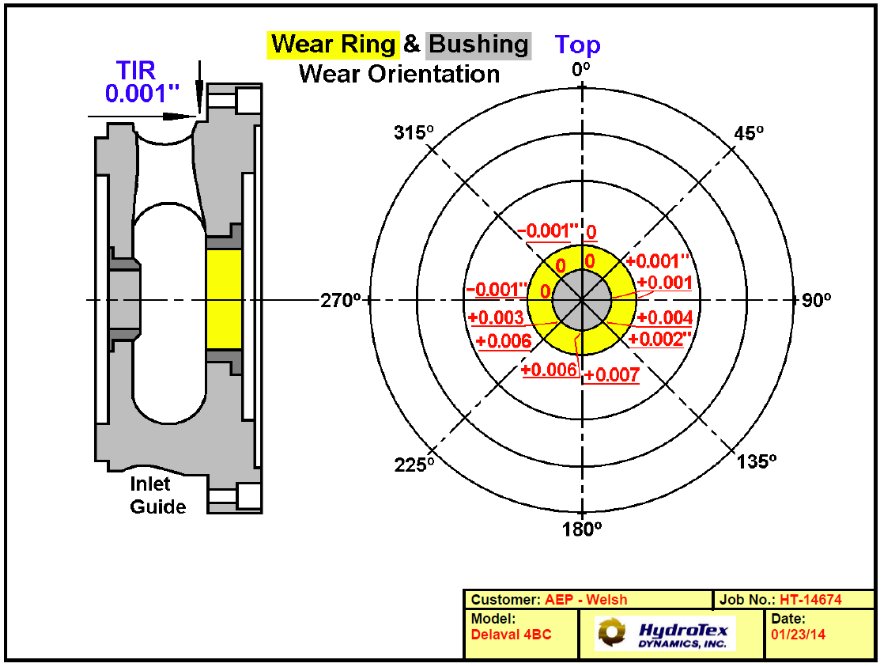
Background

- In March 2013, HT repositioned the pump IBB housing without improvement on IBB-X probe.
- In September 2013, HT swapped elements and adjusted the horizontal rim closer to target value. After start up, IBB-X & Y values were close in magnitude and low. Shaft vibration amplitude remained low and acceptable through December 19, 2013.
- On December 29, 2013 vibration levels ramped up.
- In late Dec '13 and early Jan '14 the IBB-X probe indicated over 4.0 mils pk-pk and the Y probe about 2.0 mils pk-pk. The levels were the highest at low load initially, then after several days, switched to being highest at the higher loads.
- The element that was removed in September 2013 showed “egged-shape” wear on top of the 1A and 1B wear rings, with about 2.0 to 3.0 mils of wear.

TDBFP Hydro Tex Inspection of Element Removed in September 2013

Welsh 1A Ring
Suction Diaphragm TIR
3.0 mils wear
at the top of the wear ring





Welsh 1B Ring
Inlet Guide TIR 3.5 mils wear
at the top of the wear ring

Background

- In January 2014, AEP raised the alarm level to 6.0 mills. Then, the vibration amplitude reached 8.0 mils pk-pk.
- On January 18, 2014, the plant reported 6.0 mils pk-pk at the IBB-X probe at low speed operation. Then this behavior changed and the high vibration occurred at high load.
- On February 1st, 2014, the plant was shut-down for a short outage and AEP decided to inspect and replace the IBB-X probe. Unfortunately, during the start-up, the main transformer from the main turbine-generator failed and the outage was extended for several weeks until March 6, 2014.
- During this extended forced outage MSI was engaged to set-up and perform thorough vibration and nozzle load testing.

Vibration Testing Approach

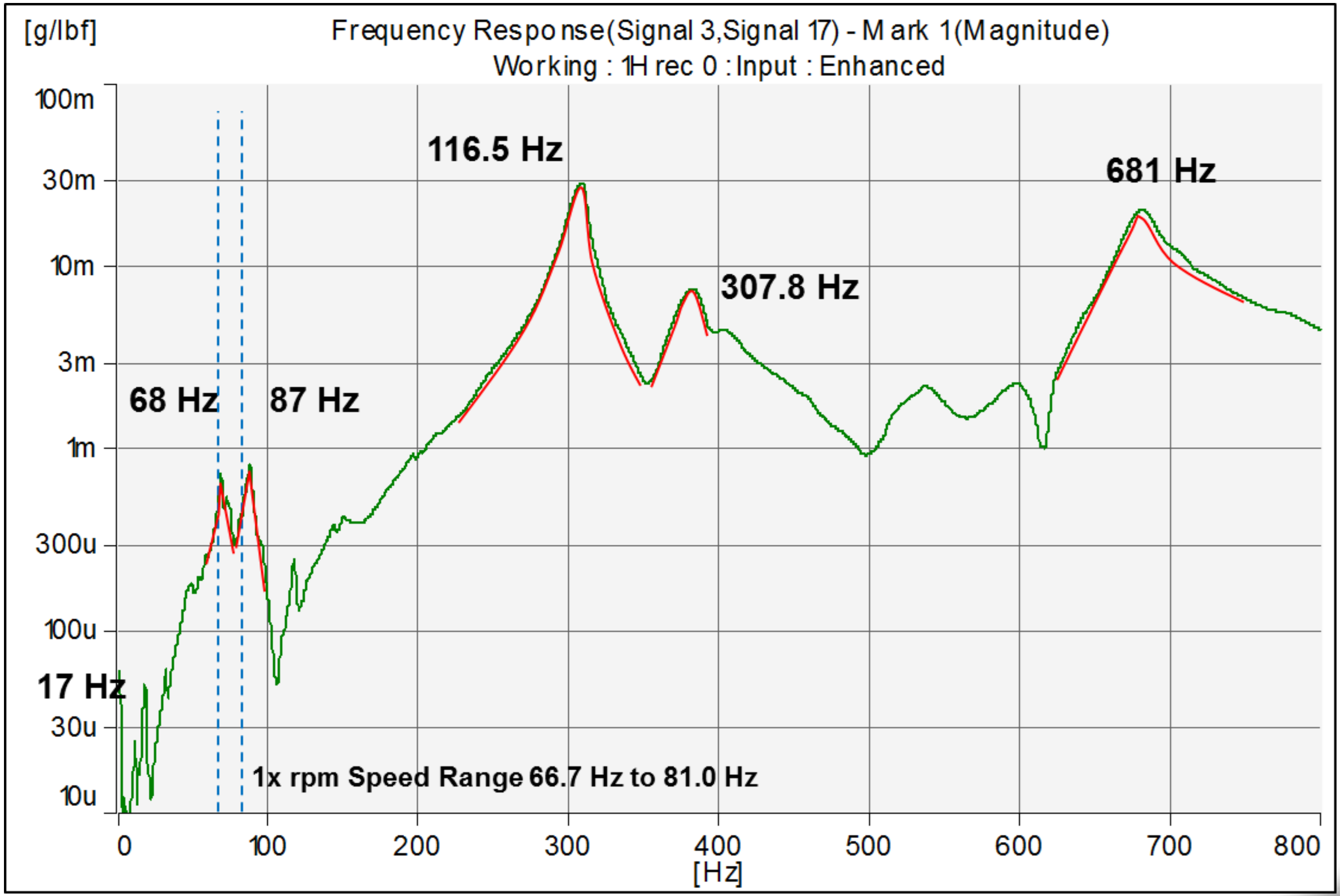
- Experimental Modal Analysis (EMA) test to determine the natural frequencies of the pump structure and the rotor system.

- Continuous Monitoring (CM) testing during transient and steady operation to monitor the shaft and bearing vibration amplitude, structural natural frequencies, and pressure pulsations. In addition, strain gauges were used to monitor forces and moments in three orthogonal directions for the suction and discharge nozzles of the TDBFP.

- Operating Deflection Shape (ODS) testing during steady operation at full load conditions.

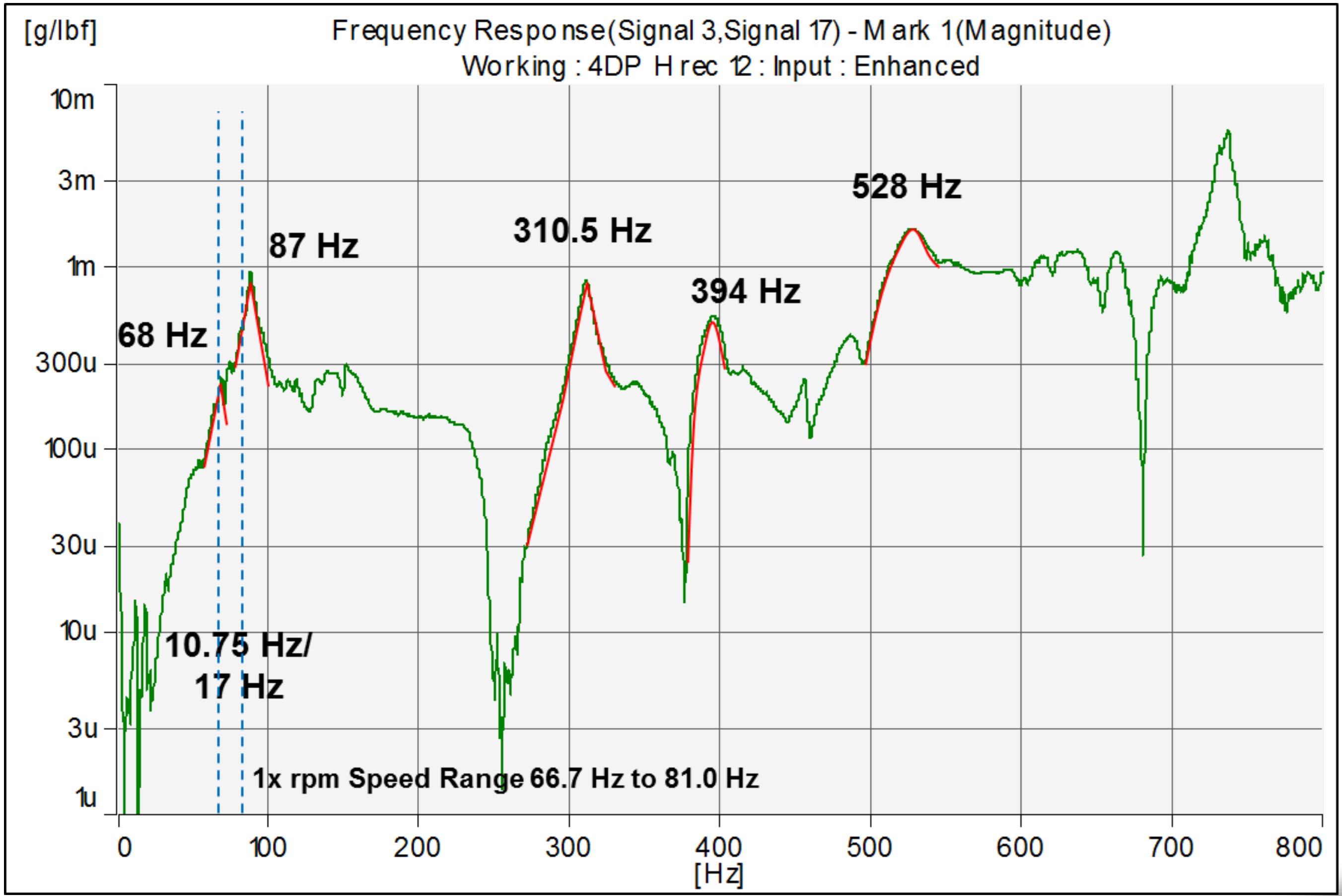
Experimental Modal Analysis (EMA) Test Results While TDBFP Was Not Operating

TDBFP OBB – Frequency Response Function (FRF) Plot Horizontal Direction



FRF – Pump Not Running

TDBFP IBB – Frequency Response Function (FRF) Plot Horizontal Direction



FRF – Pump Not Running

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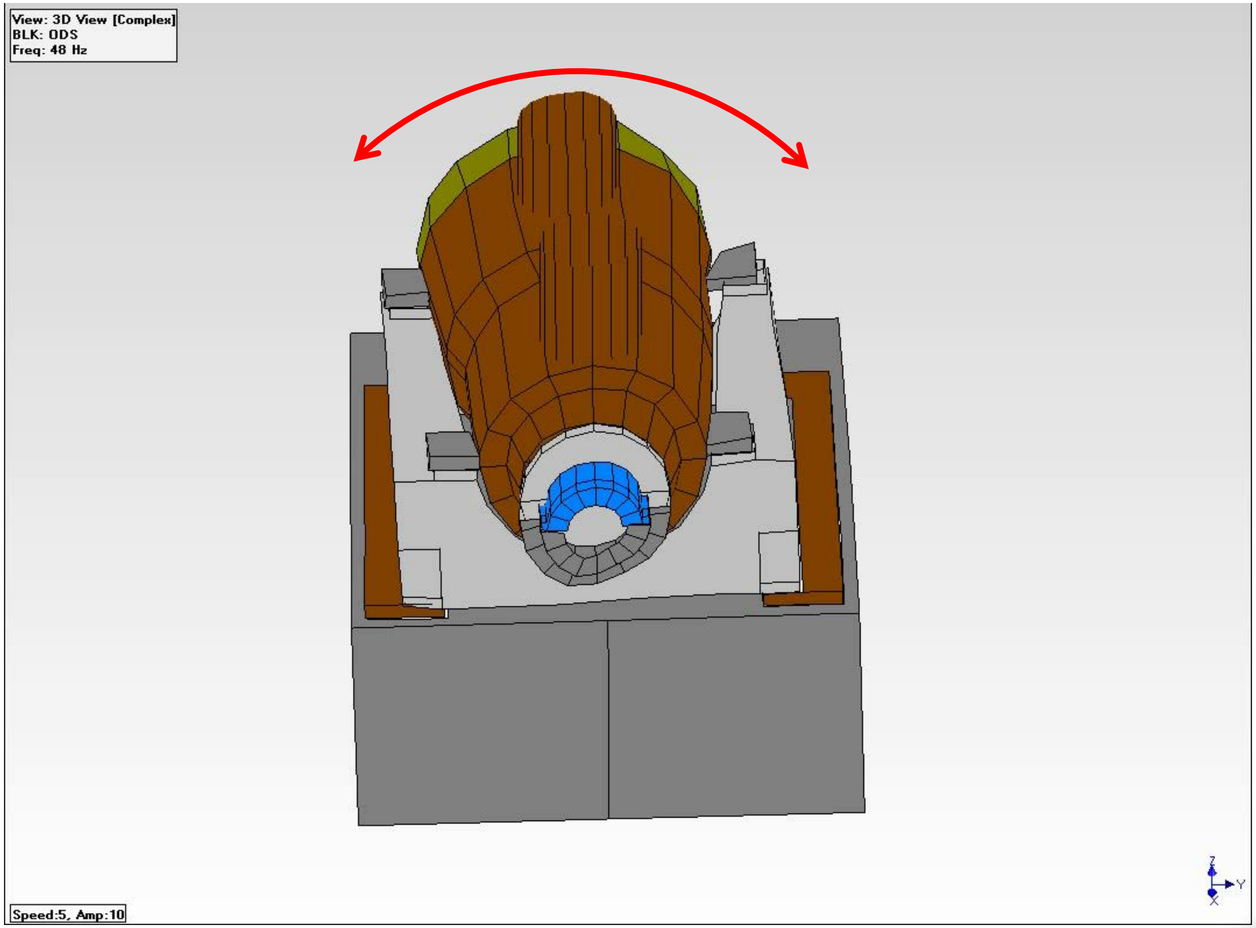
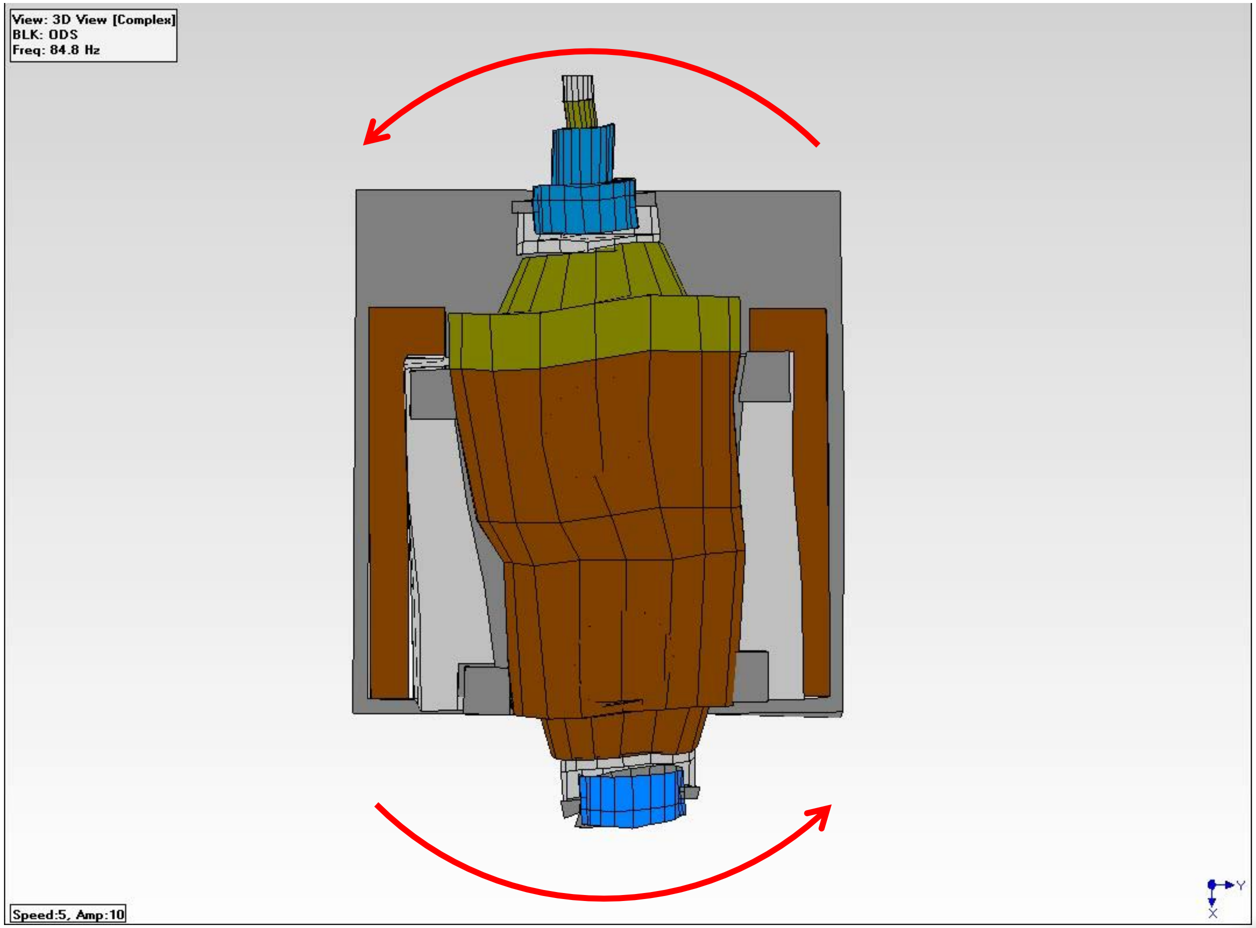
Experimental Modal Analysis (EMA) Test Results While TDBFP Was Operating @ 4100 rpm or 200 MW (Accumulative Time-Average Method)

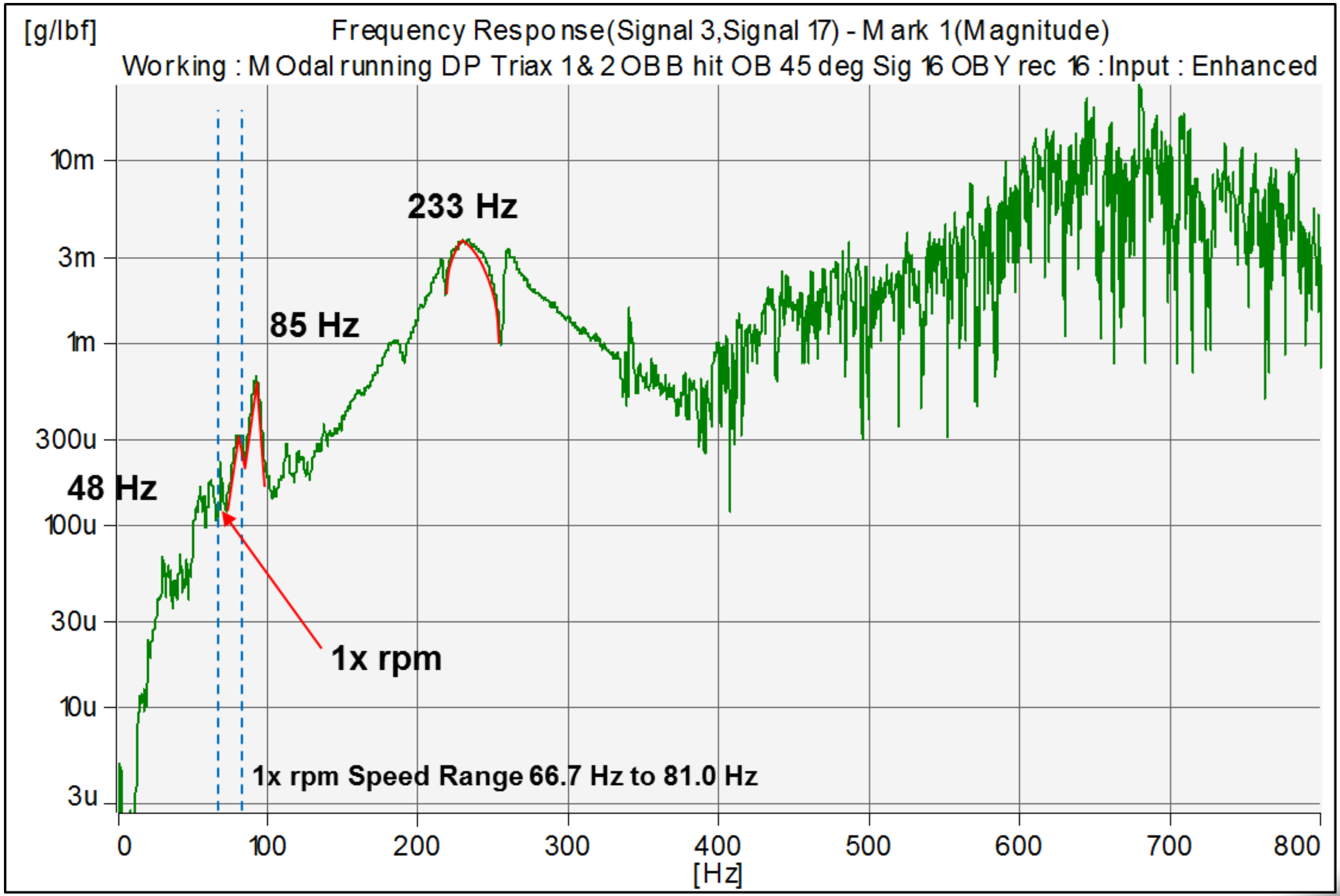
TDBFP Structural Natural Frequency at 48 Hz



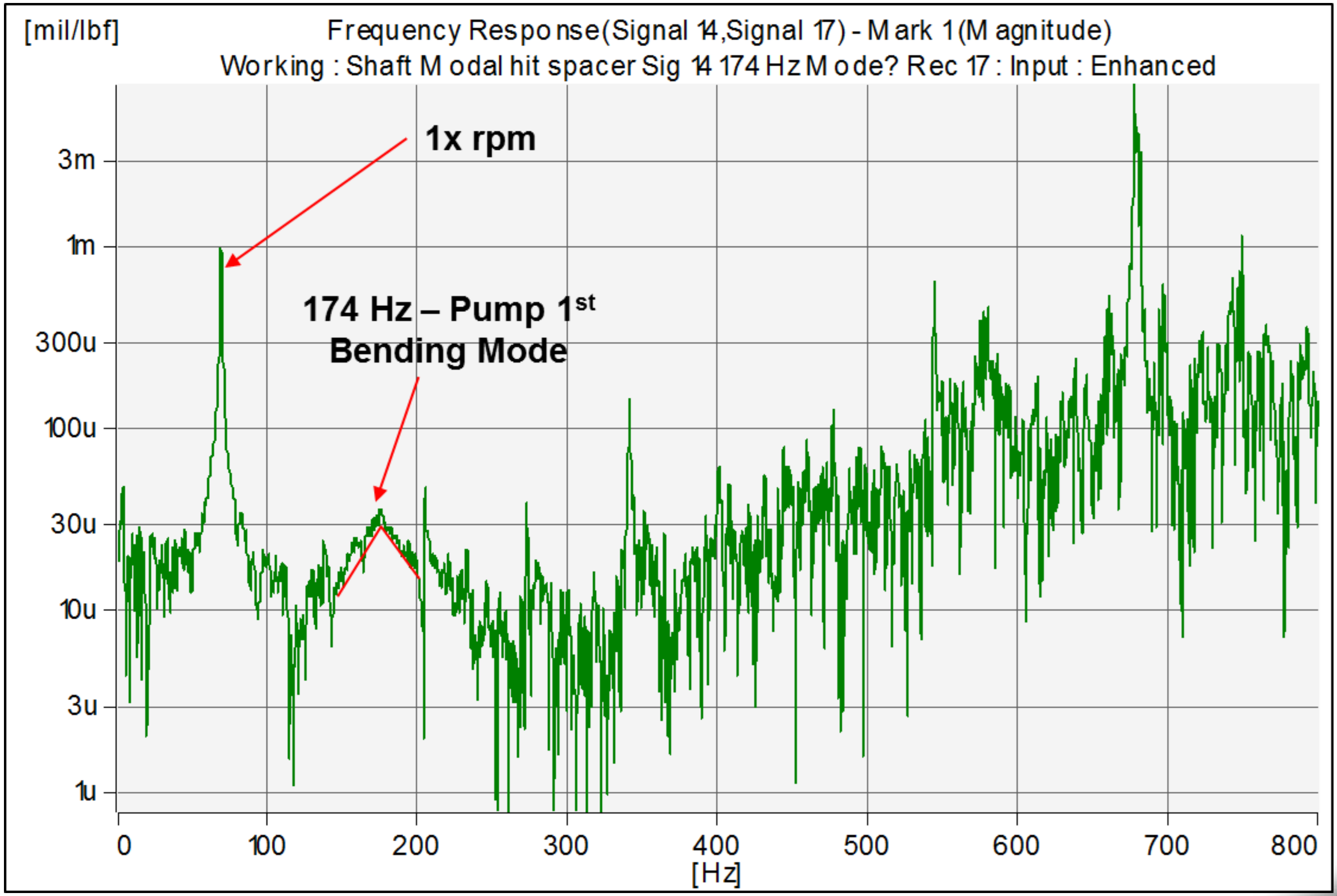
TDBFP Structural Natural Frequency at 84.8 Hz



TDBFP OBB – Frequency Response Function (FRF) Plot Horizontal Direction (Pump Running @ 4100 rpm)

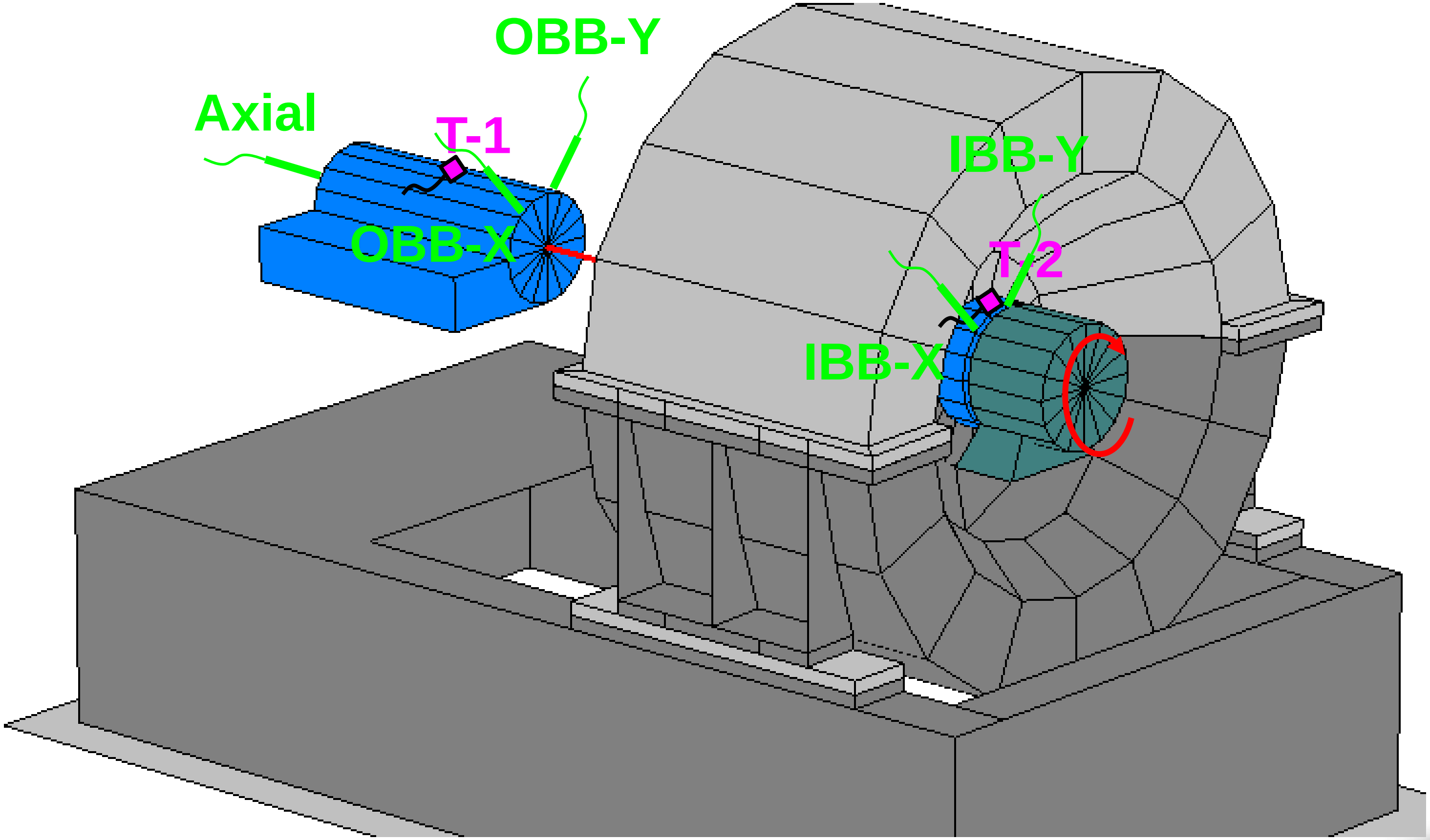


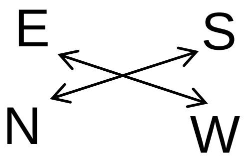
TDBFP OBB – Frequency Response Function (FRF) Plot IBB-Y Direction (Pump Running @ 4100 rpm)



Continuous Monitoring
Test Set Up and Results
from March 5 through 10, 2014

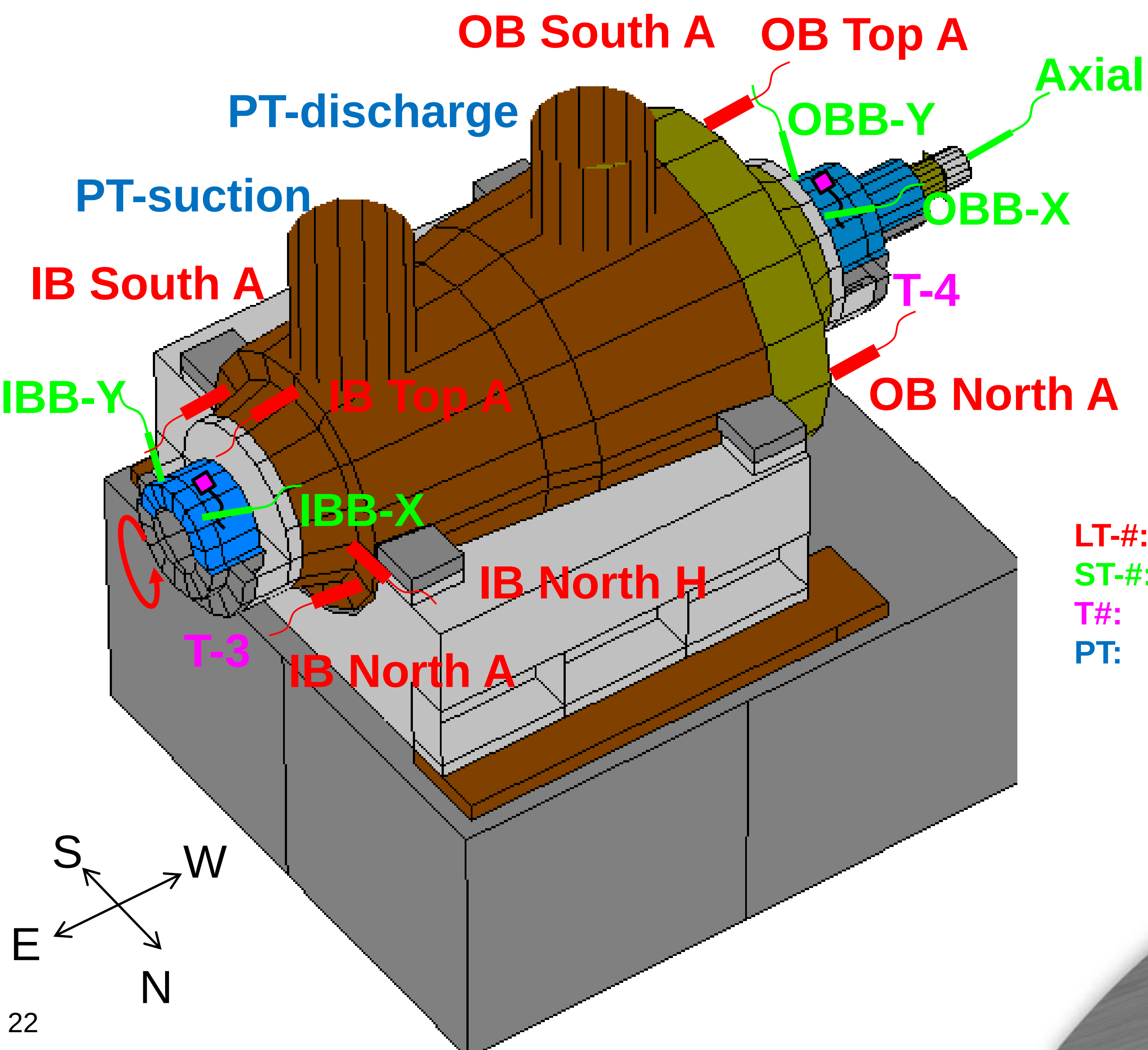
TDBFP Pump MSI's Instrumentation Distribution – 45 Channels of Data Acquisition





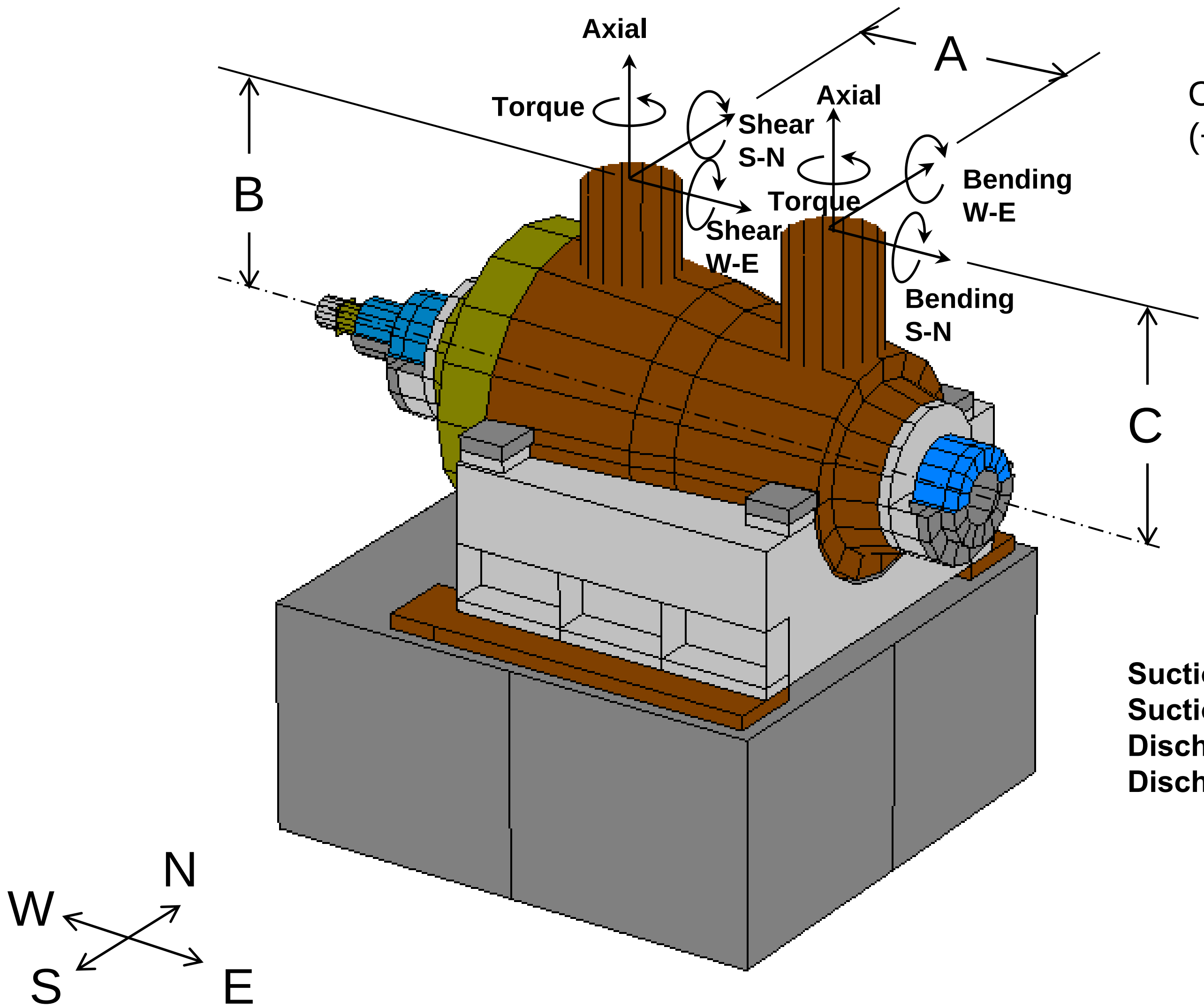
ST-#: Short Travel Proximity Probe
T#: Tri-Axial Accelerometer

TDBFP Pump MSI's Instrumentation Distribution – 45 Channels of Data Acquisition



LT-#: Long Travel Proximity Probe
ST-#: Short Travel Proximity Probe
T#: Tri-Axial Accelerometer
PT: Pressure Transducer

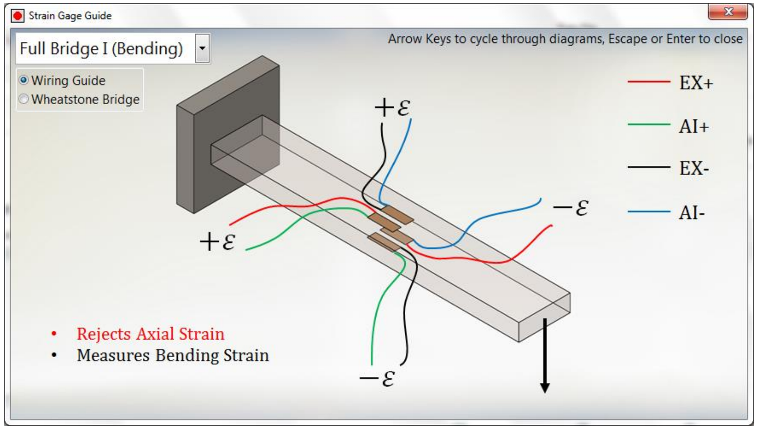
TDBFP Pipe Strain Gage Diagram



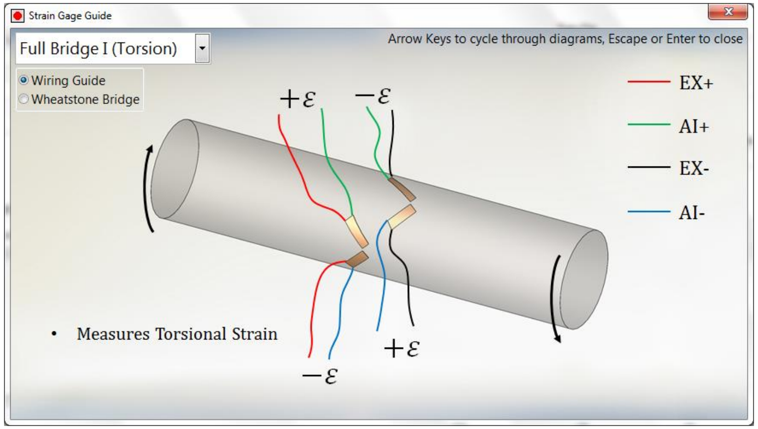
Orientation of Axis Positive (+) Directions Shown

Suction Pipe OD: 18.0"
Suction Pipe Thick: 3/8"
Discharge Pipe OD: 15.98"
Discharge Pipe Thick: 1.7"

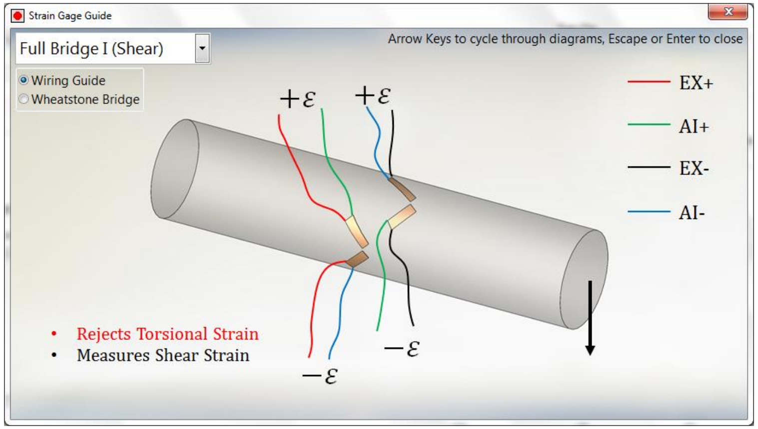
Strain Gage Rosettes Wiring Diagrams



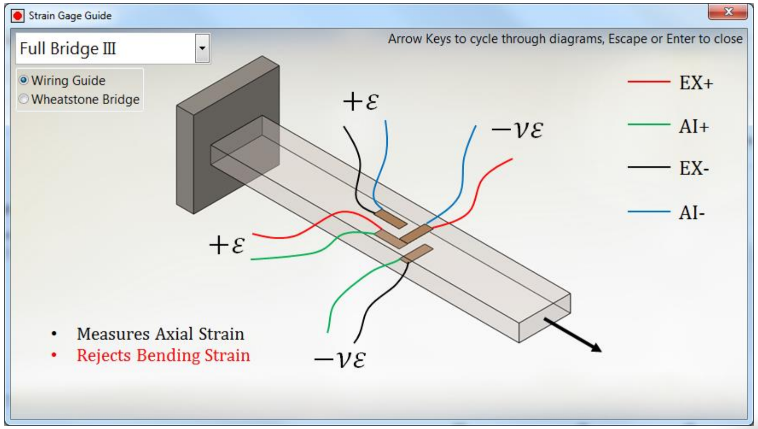
Bending



Torque



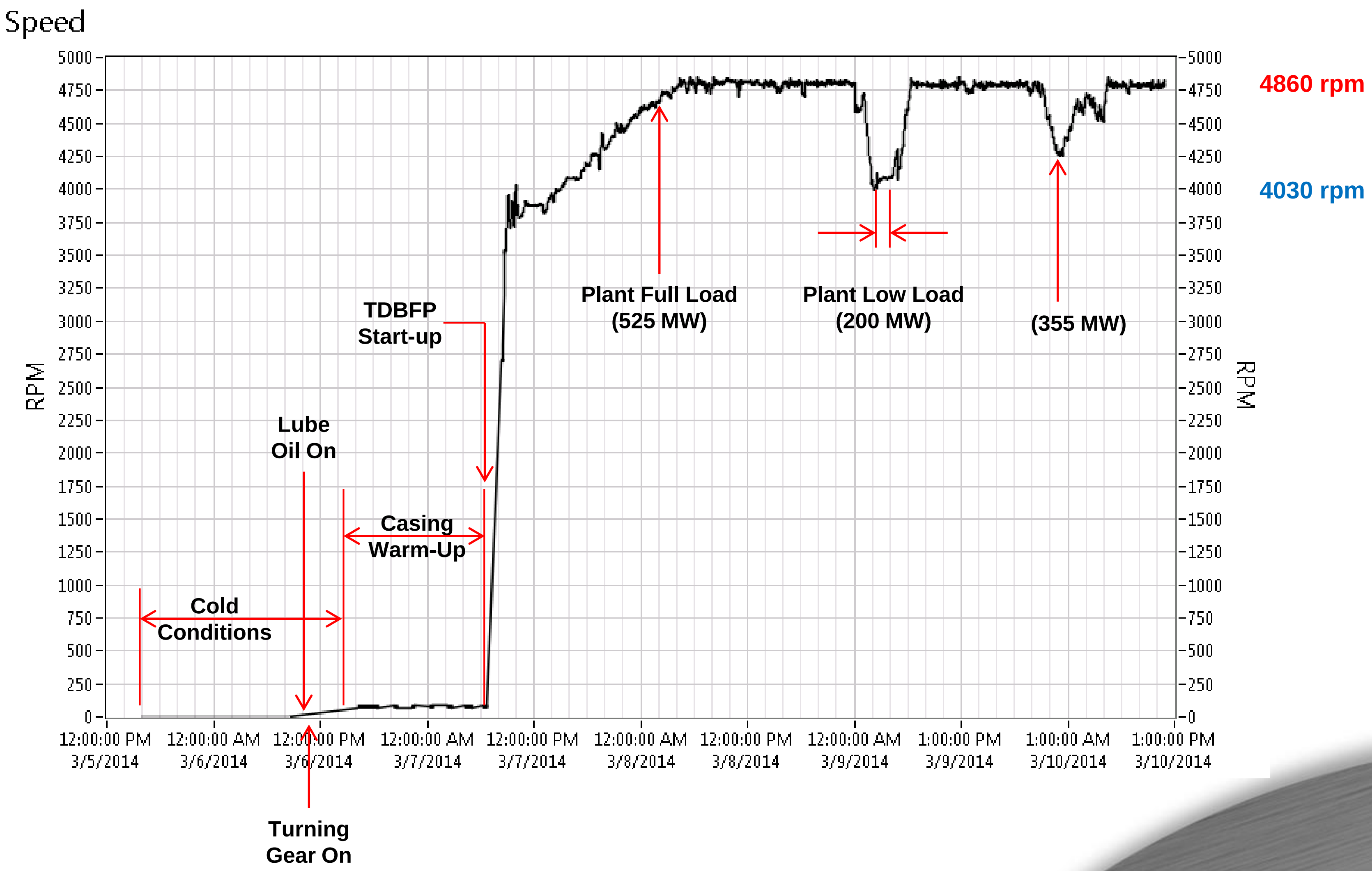
Shear



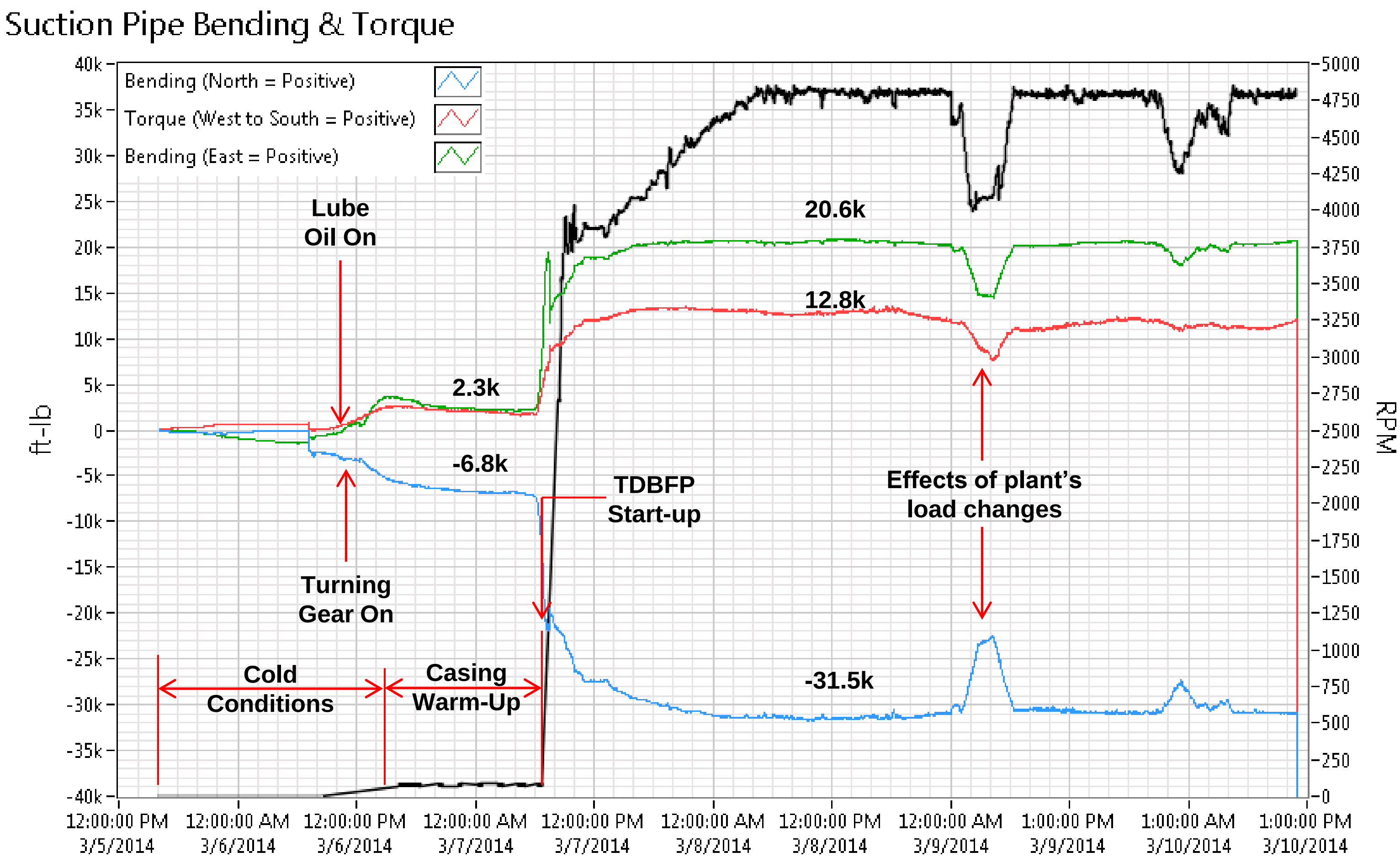
Axial

Strain Gages Tack-Welded on the Suction and Discharge Piping

Speed Trend Plot From 3/5/14 to 3/10/14

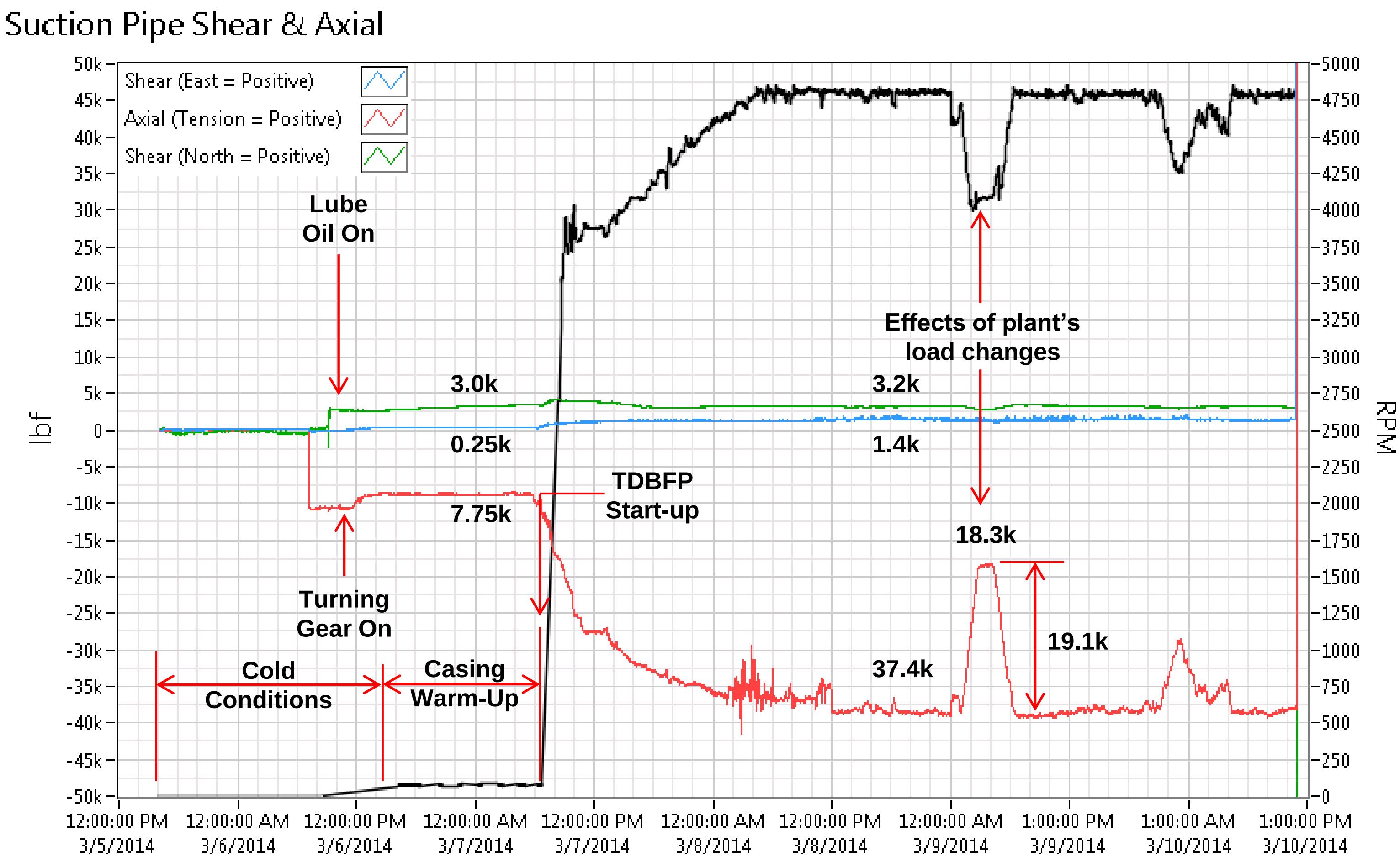


Suction Pipe Bending & Torque Trend



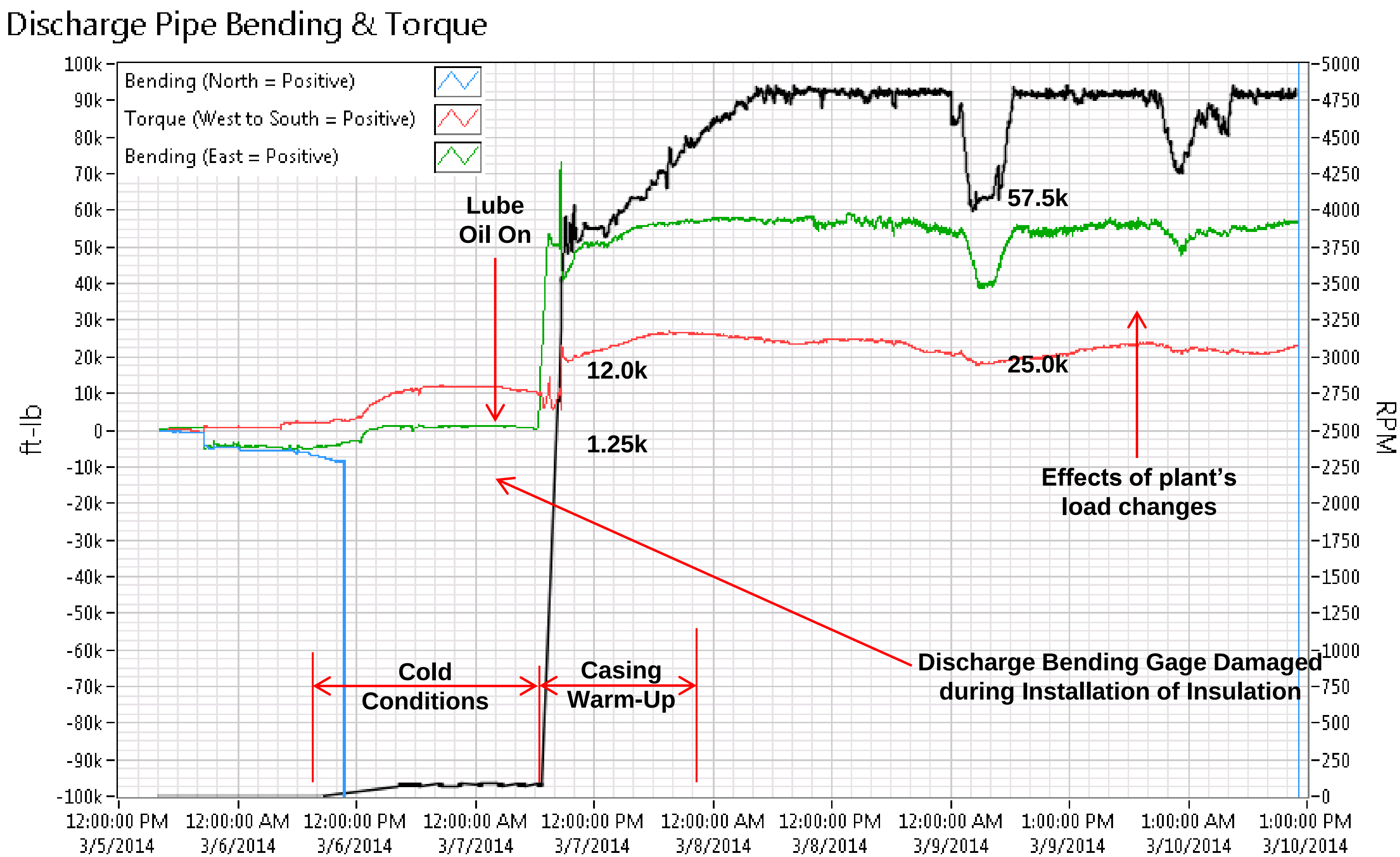
During warm-up, suction pipe is twisting from south to east and bending towards south-east. At full load (525 MW) maximum piping load was reached in the same direction. Plant load plays a role in changing the suction bending and especially the torque.

Suction Pipe Shear & Axial Trend Plot



During warm-up, suction pipe is in compression and lightly shearing towards north-east. At full load (525 MW) maximum piping load was reached, especially in the axial direction. The MW load of the plant applies significant downwards axial load to the pump casing.

Discharge Pipe Bending & Torque Trend



During warm-up, discharge pipe is twisting from south to east and bending towards east (north-south bending gage lost prior warm-up). At full load (525 MW) maximum piping load was reached in the same direction. Plant load plays a role in changing the discharge torque and especially the bending load.

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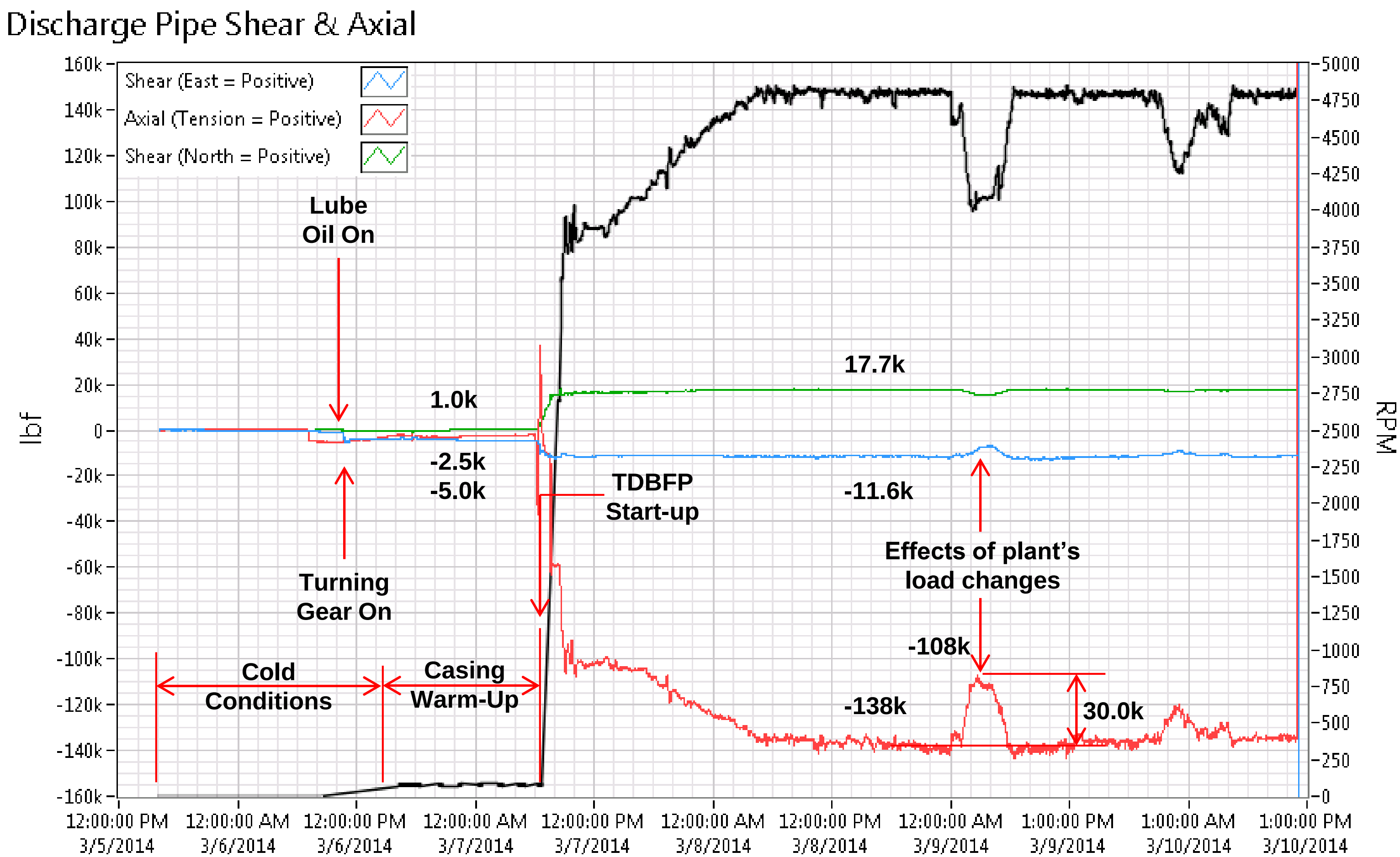
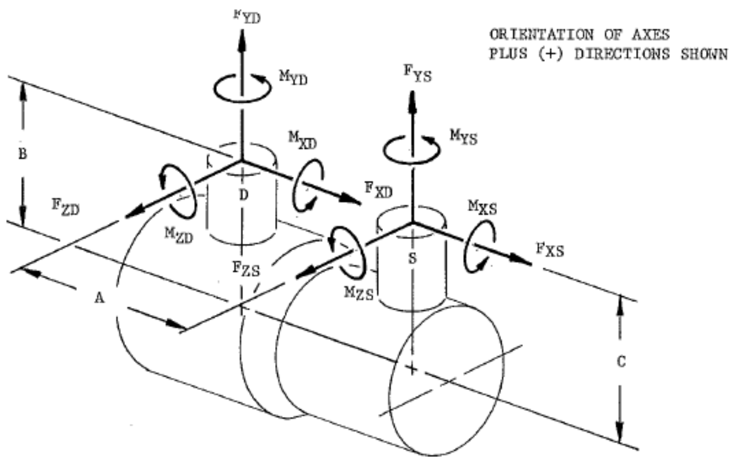
Discharge Pipe Shear & Axial Trend Plot



During warm-up, discharge pipe is lightly loaded (compared with full load operation). At full load (525 MW) maximum piping load was reached, especially in the axial direction (compression). The MW load of the plant applies significant downwards axial load to the pump casing. The discharge pipe is under shear in the north-west direction.

Welsh 1 TDBFP Pipe Strain
De Laval Maximum Allowable Force & Moments



RESOLUTION AND SUMMATION OF APPLIED FORCES AND MOMENTS ABOUT THE DISCHARGE NOZZLE

$$F_{XT} = F_{XS} + F_{XD} \qquad M_{XT} = M_{XS} + M_{XD} - \left[F_{ZS} \times (B-C)\right]$$

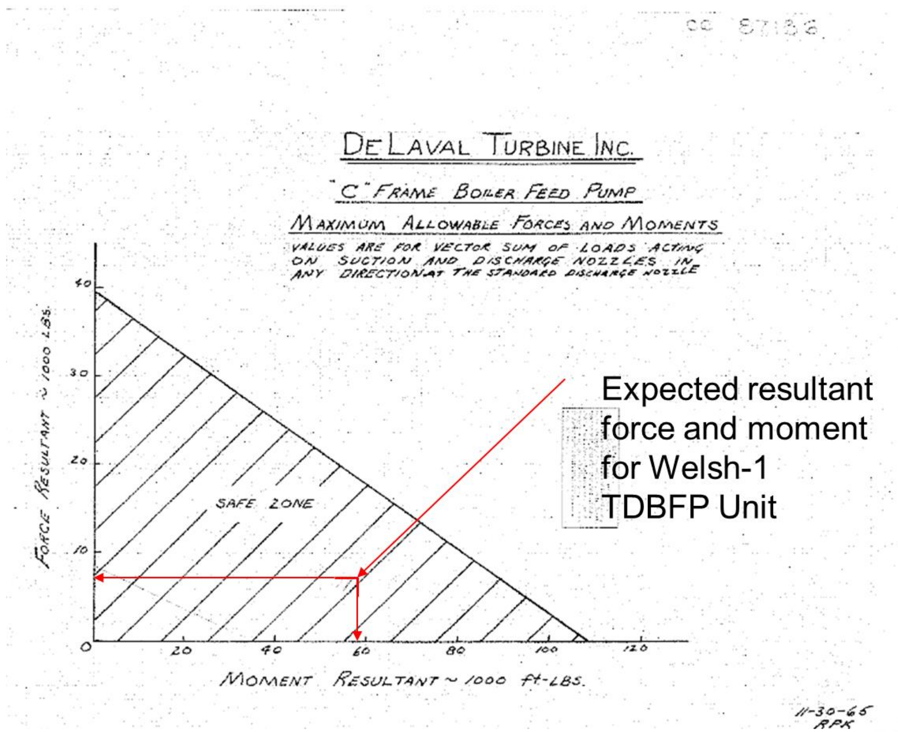
$$F_{YT} = F_{YS} + F_{YD} \qquad M_{YT} = M_{YS} + M_{YD} - (F_{ZS} \times A)$$

$$F_{ZT} = F_{ZS} + F_{ZD} \qquad M_{ZT} = M_{ZS} + M_{ZD} + (F_{YS} \times A) + \left[F_{XS} \times (B-C)\right]$$

CALCULATION OF FORCE AND MOMENT RESULTANTS

$$F_R = \sqrt{F_{XT}^2 + F_{YT}^2 + F_{ZT}^2}$$

$$M_R = \sqrt{M_{XT}^2 + M_{YT}^2 + M_{ZT}^2}$$



Welsh 1 TDBFP Pipe Strain
De Laval Maximum Allowable Force & Moments

		Suction	Discharge	Units
Shear E-W	**Fx (+E)**	1440	-11600	lbf
Shear N-S	**Fz (+S)**	3160	-17700	
Axial	**Fy (+Up)**	-37400	-136400	
Bending N-S	**Mx (+N to S)**	-31500	0	ft-lb
Bending E-W	**Mz (+E to W)**	-20600	-54700	
Torque	**My (+Up)**	12800	23700	

A	2.70 ft
B	3.75 ft
C	3.75 ft

Fxt	**-10160**
Fyt	**-173800**
Fzt	**-14540**

Fr	**174703**	**175**	x1000

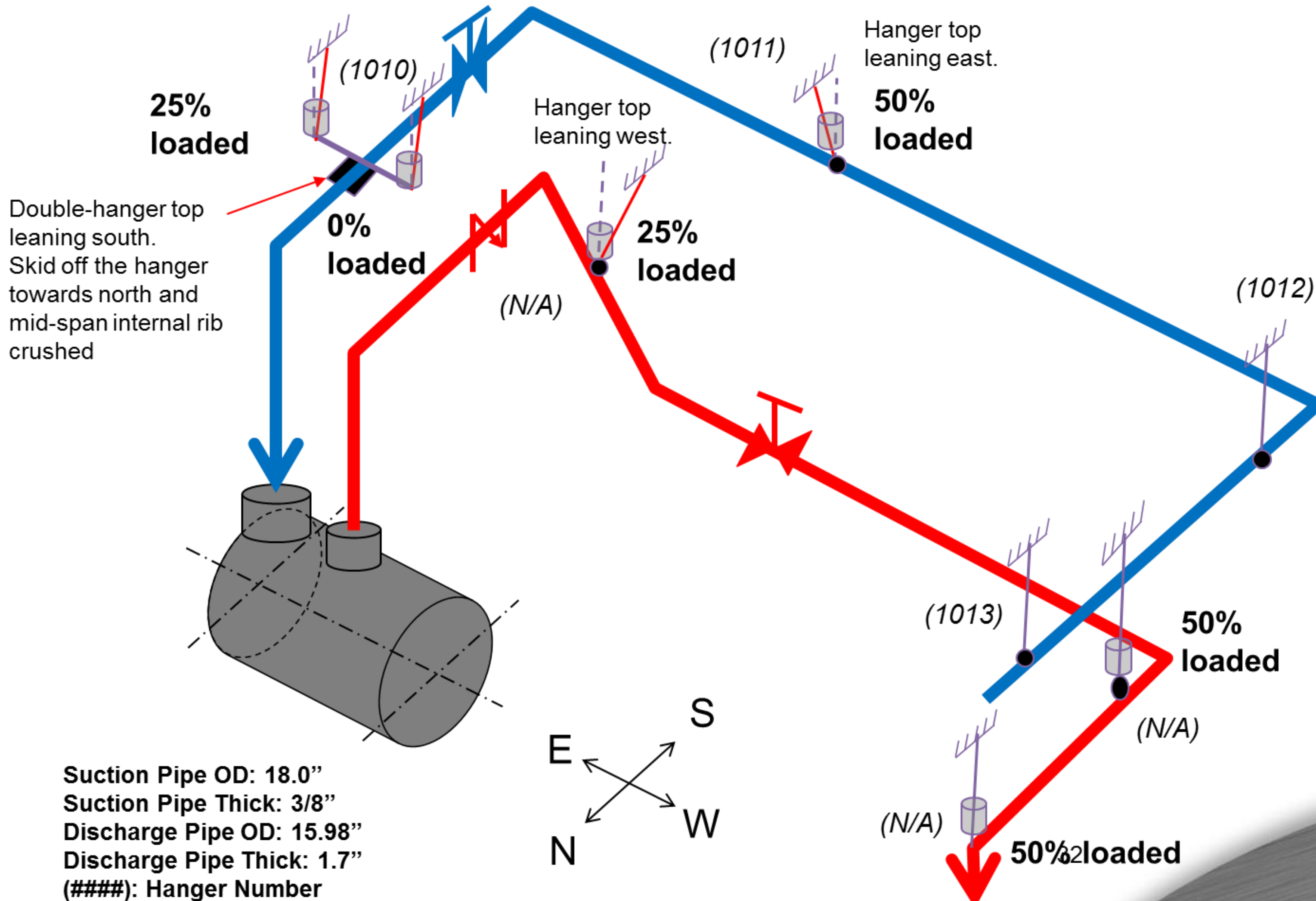
Resultant force exceeds 25 times De Laval Calculations

Mxt	**-31500**
Myt	**27975**
Mzt	**-176202**

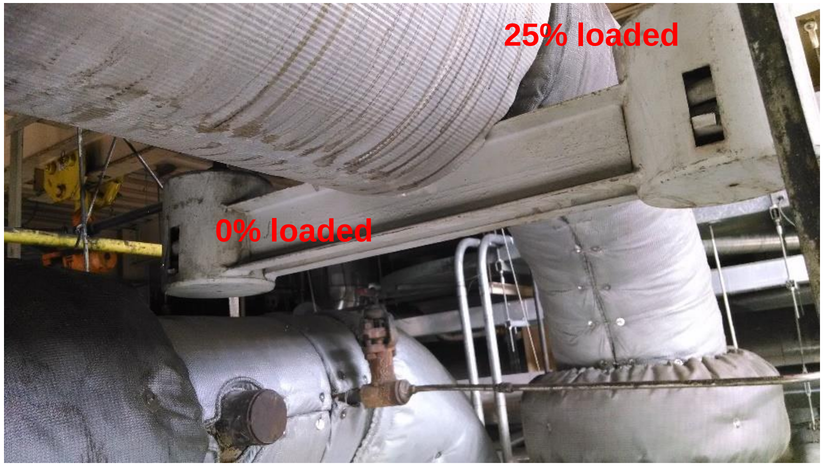
Mr	**181168**	**181**	x1000

Resultant moment exceeds 3.2 times De Laval Calculations

Welsh 1 TDBFP Pipe Diagram



Photos of Hangers for the Suction & Discharge Piping

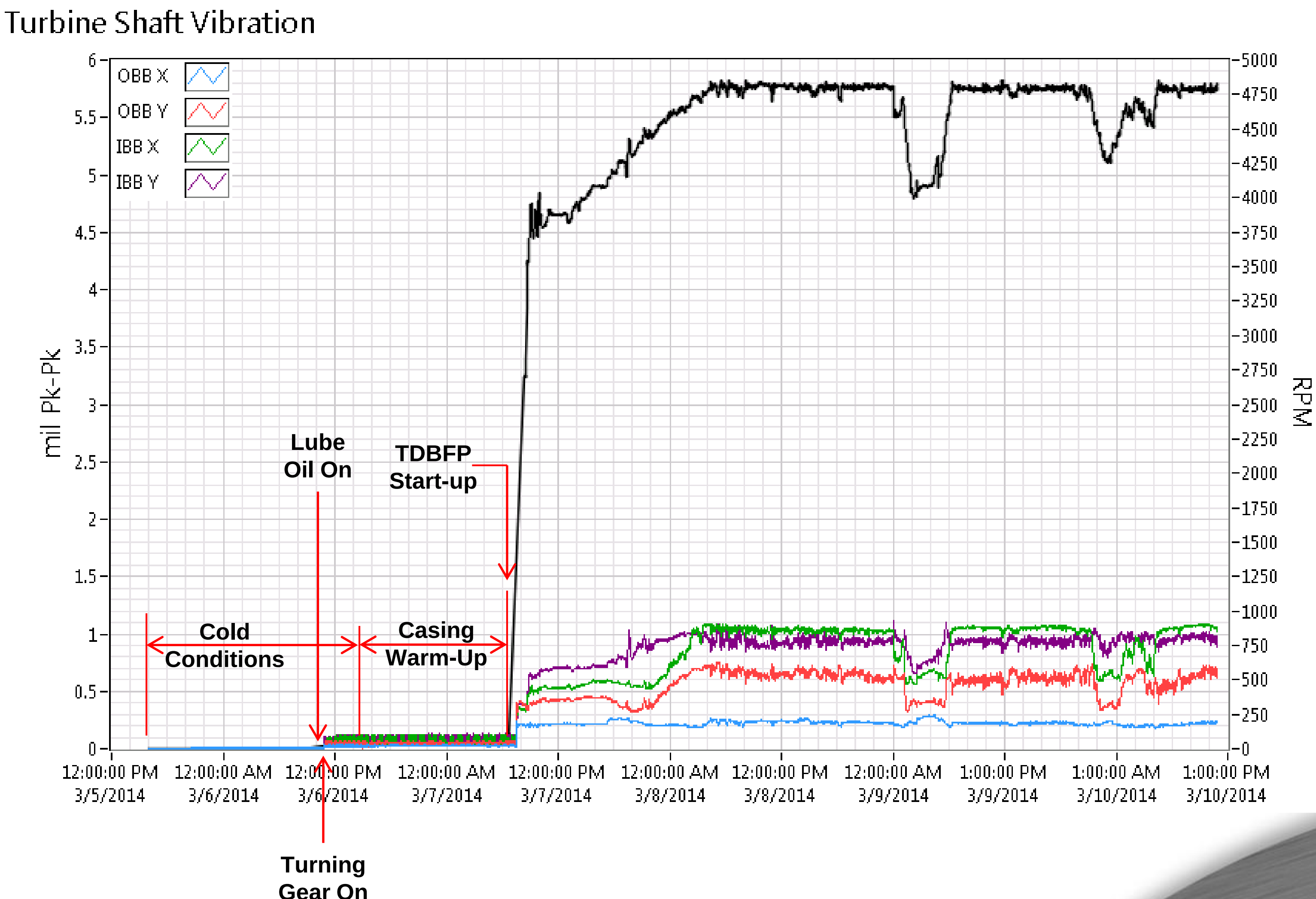


Suction Pipe

Discharge Pipe

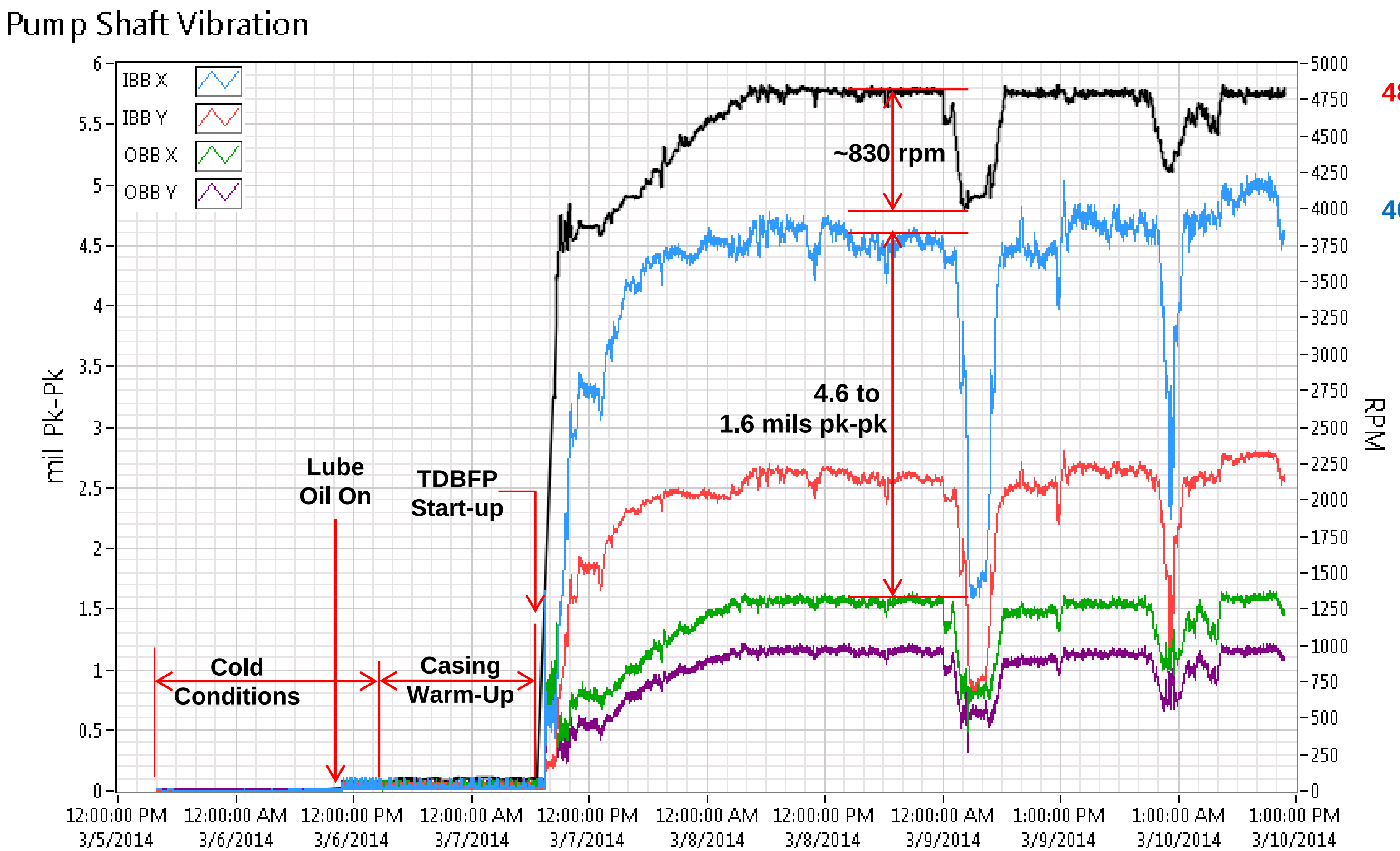
Discharge Pipe

Turbine Shaft Overall Vibration Trend Plot



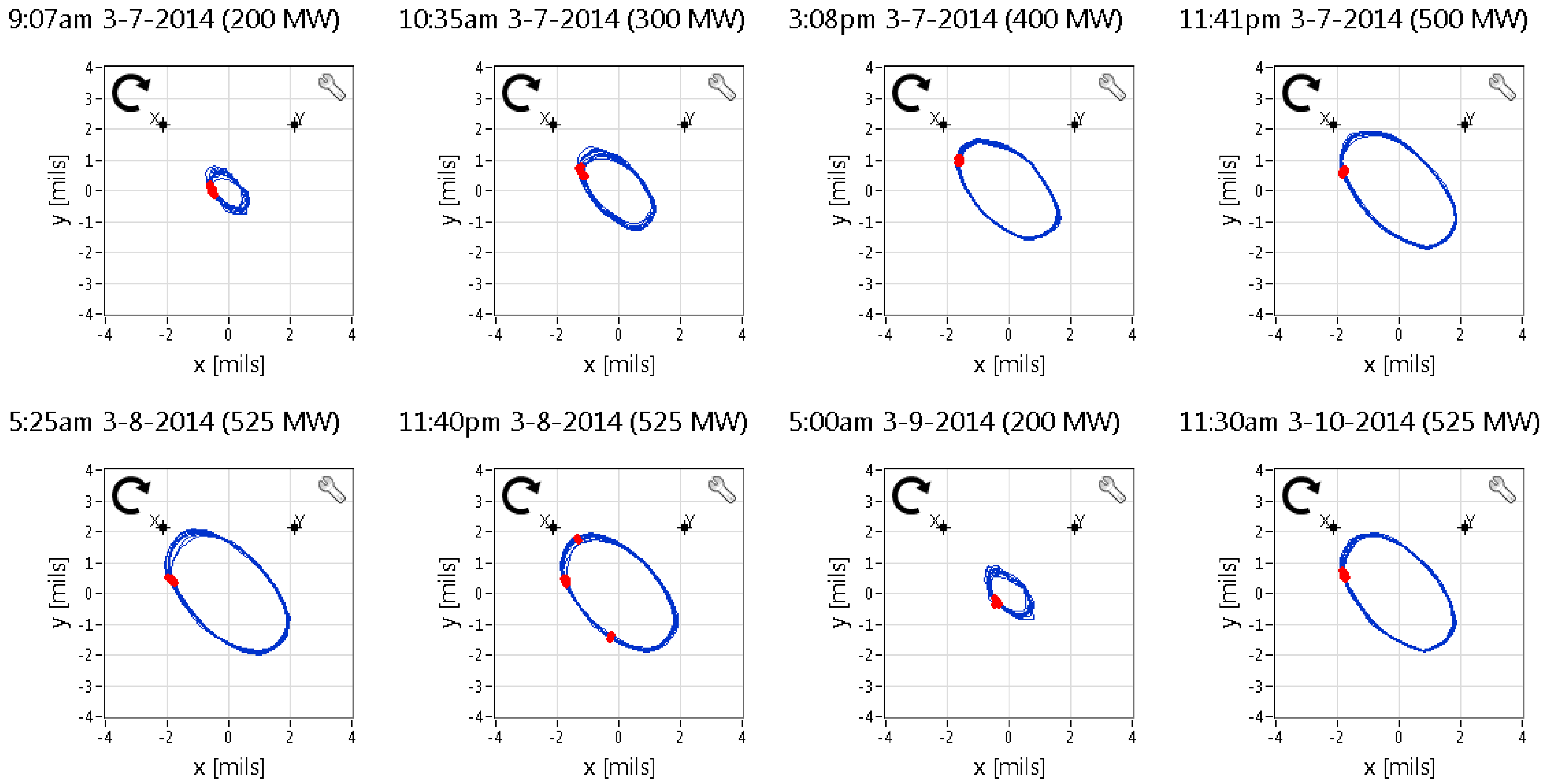
Turbine rotor overall vibration was measured to be low and acceptable (~1.0 mil pk-pk) for the entire start-up evolution of the test.

Pump Shaft Overall Vibration Trend Plot



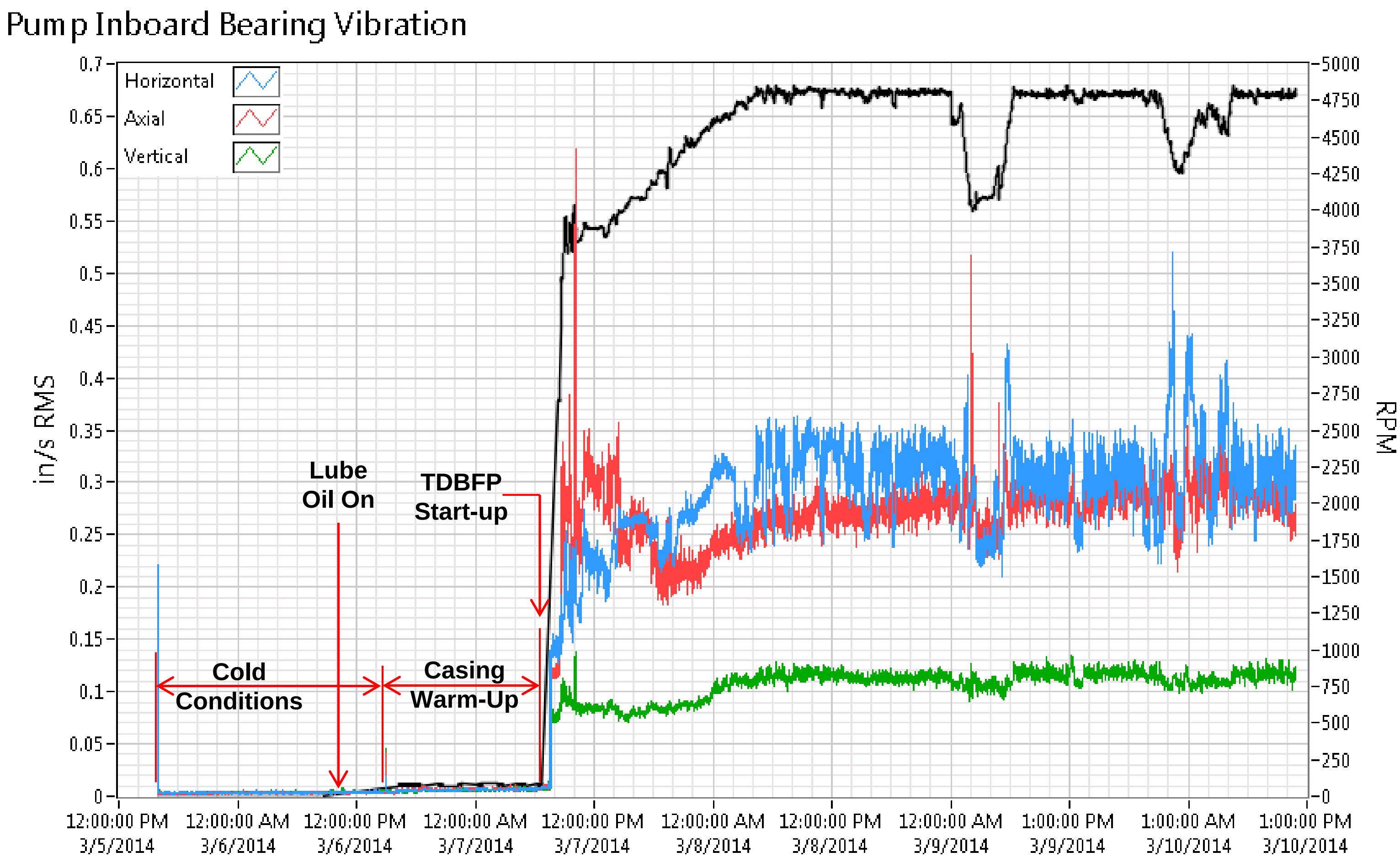
Pump rotor overall vibration at the IBB was measured to be elevated (~5.0 mil pk-pk) at full load operation (525 MW). Strong vibration change versus speed effect is observed in this trend plot. ~830 rpm reduction, the vibration at IBB-X was reduced from 4.6 to 1.6 mils pk-pk.

Pump Inboard Bearing Orbits



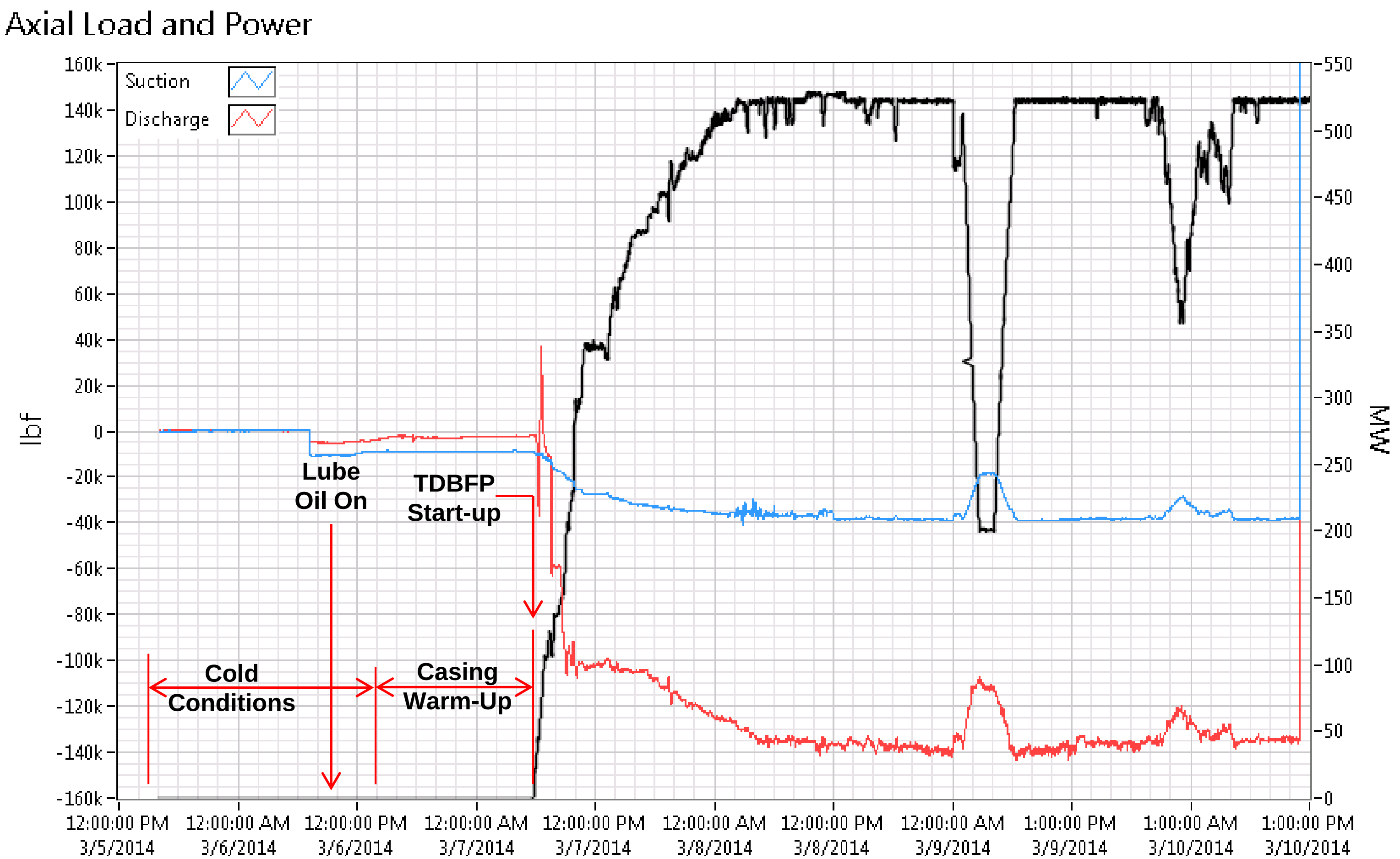
Sequence of the IBB orbit plots at different load conditions.

Pump Inboard Bearing Vibration



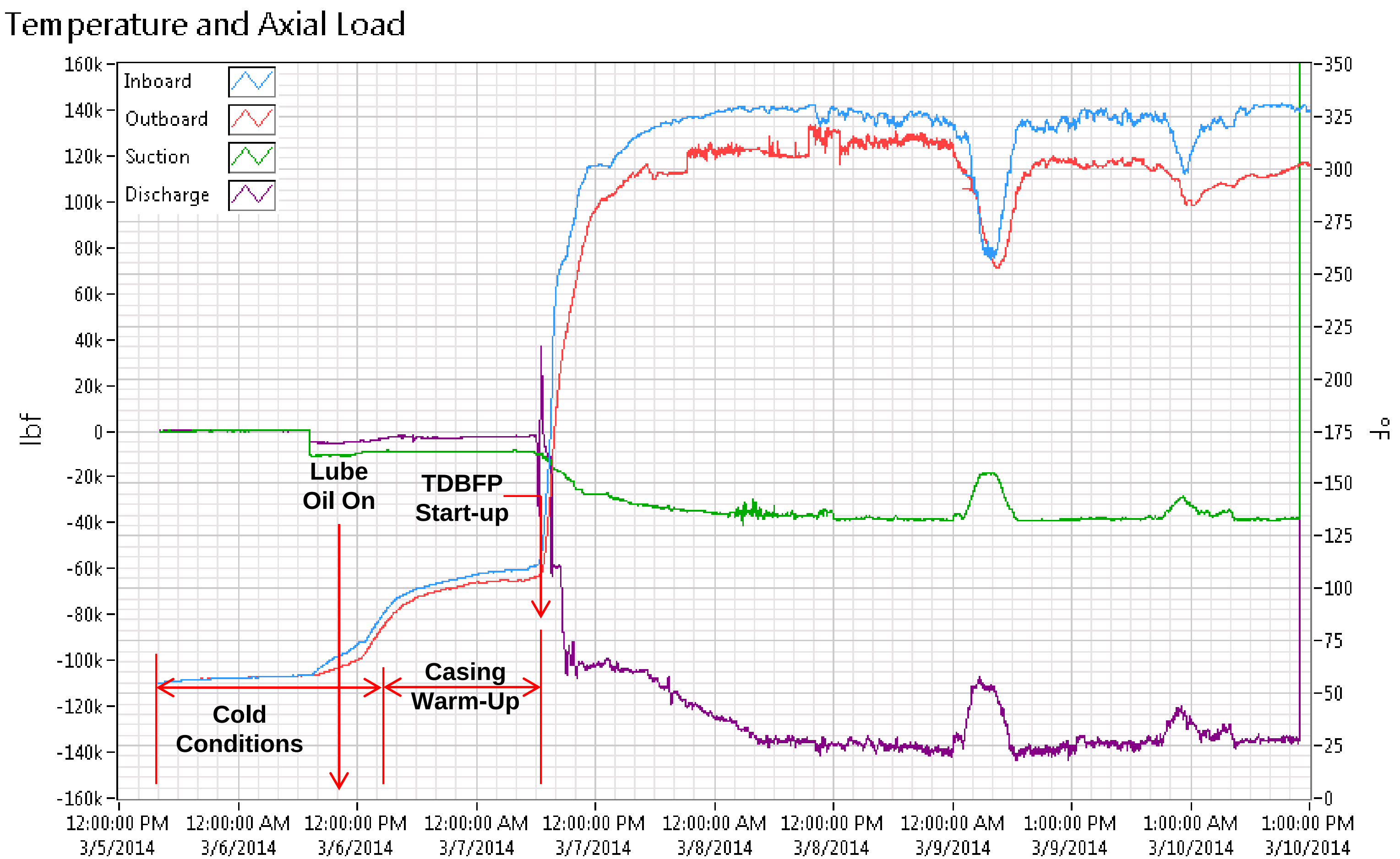
Pump IBB housing overall vibration was measured to be elevated mild (~0.35 in/s RMS in the horizontal direction) at full load operation.

Pipe Axial Load and Power



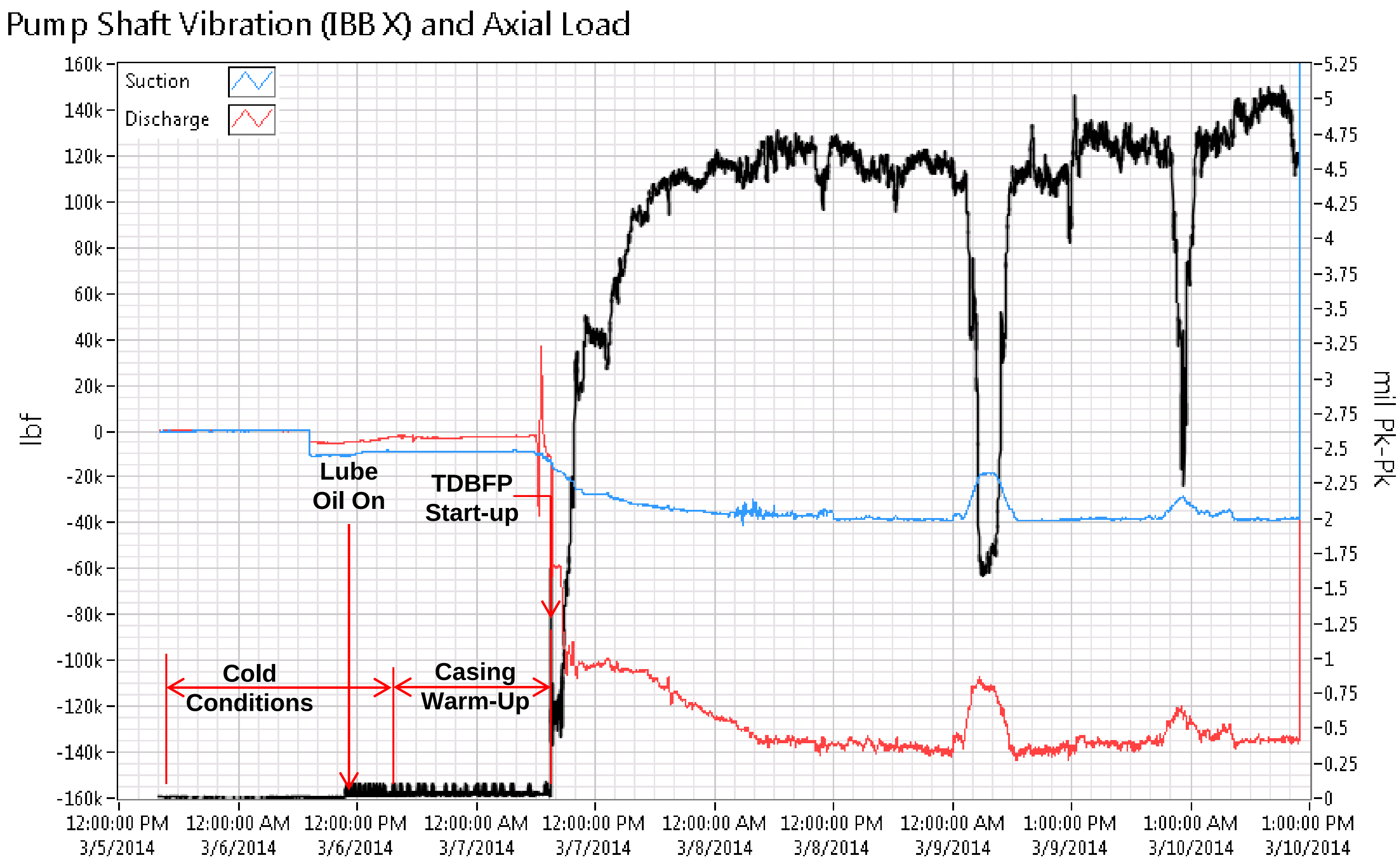
As the plant load was increasing the axial load of the piping against the casing was also increasing (compression load). The change in temperature of the condensate water was playing a significant change in load on the pump casing due to thermal expansion effect.

Temperature and Pipe Axial Load



As the plant load was increasing the axial load of the piping against the casing was also increasing (compression load). The change in temperature of the condensate water was playing a significant change in load on the pump casing due to thermal expansion effect.

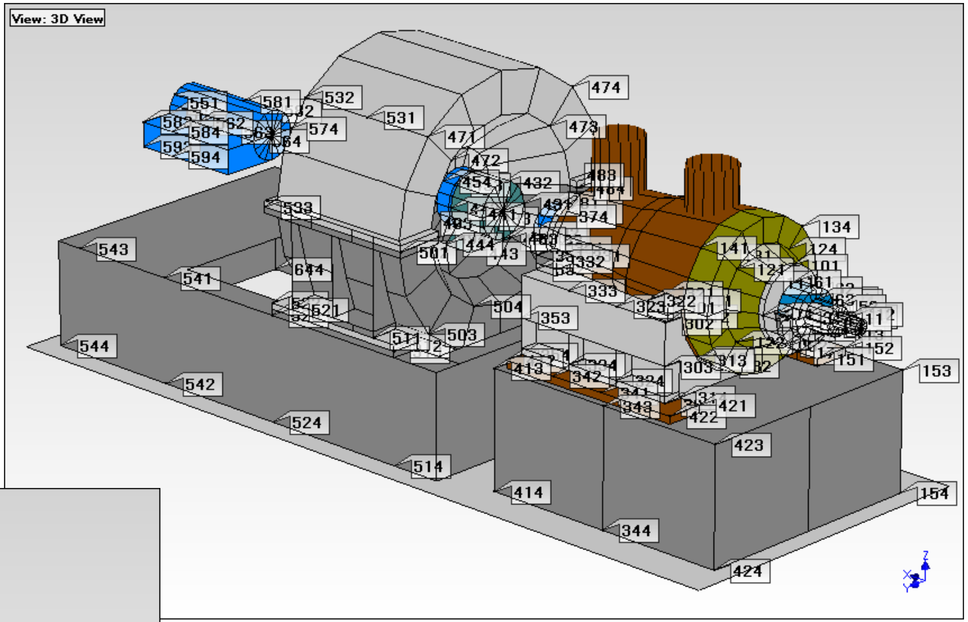
Pump Shaft Vibration (IBB X) and Pipe Axial Load

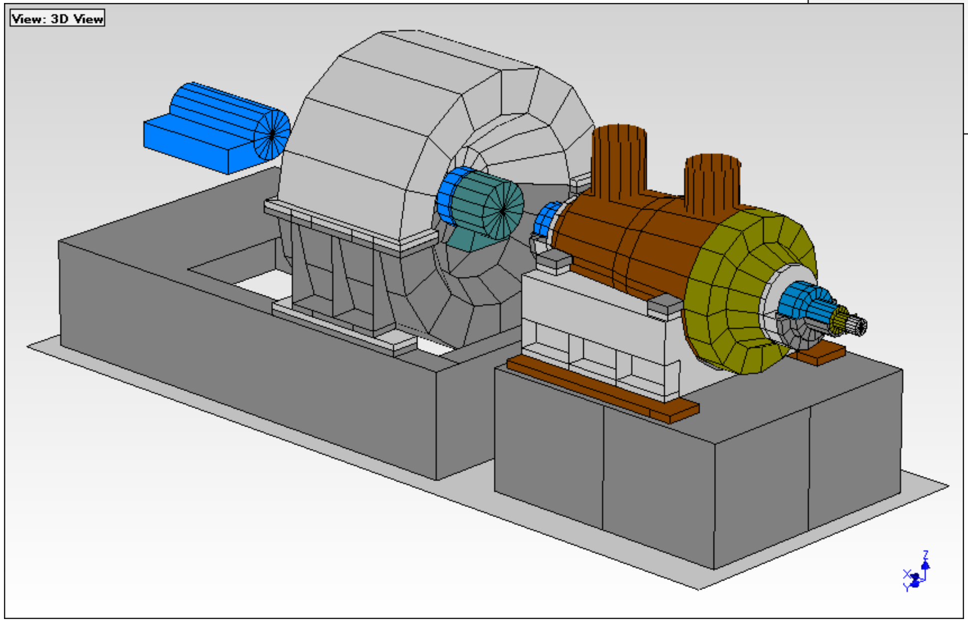


This plot shows the relationship between the IBB-X overall vibration and the casing temperature at the IB and OB ends.

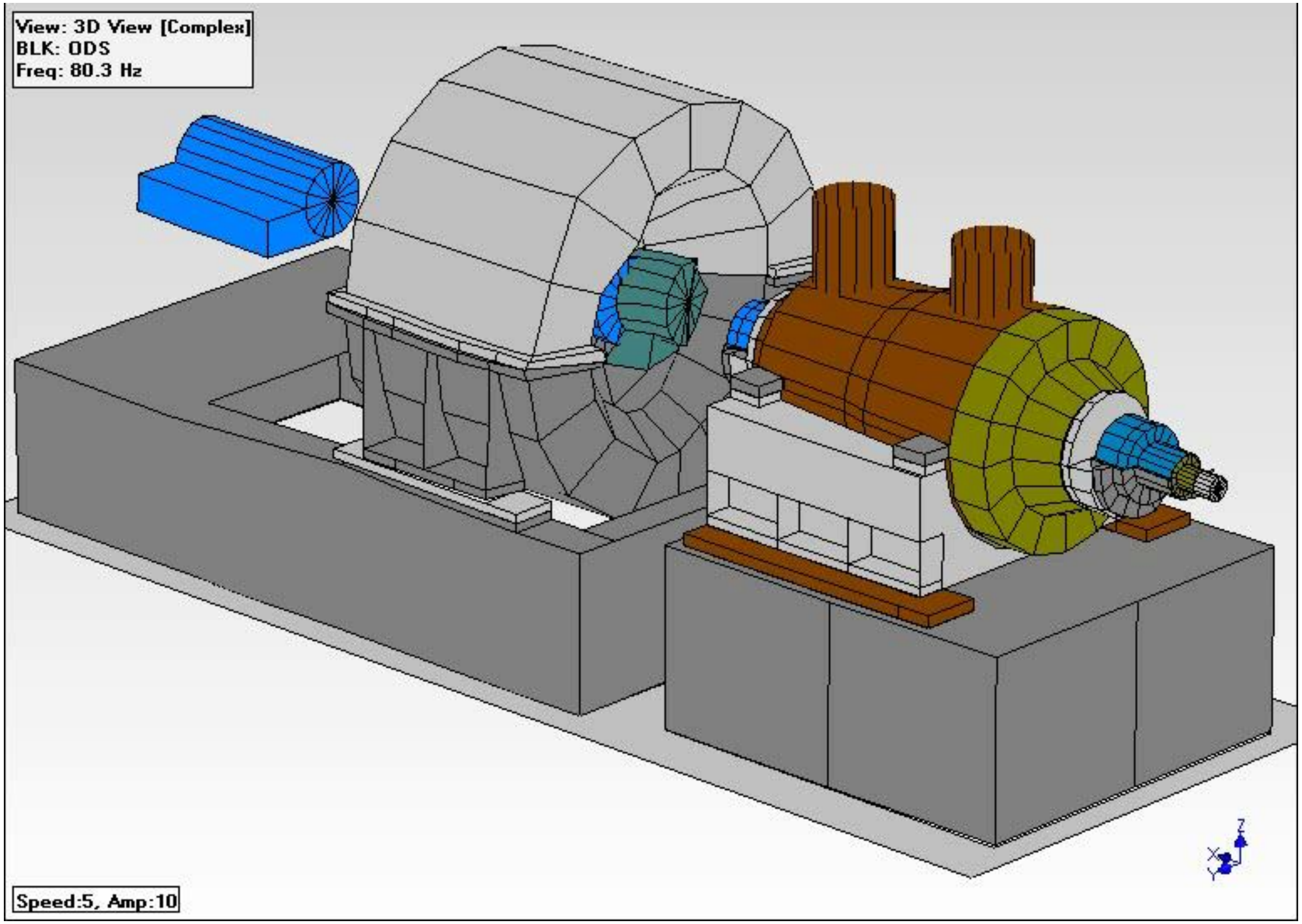
Operating Deflection Shape (ODS) Test Results while TDBFP was Operating at a Steady High Load (~525 MW)

Operating Deflection Shape (ODS) Testing
778 Vibration Locations/ Directions

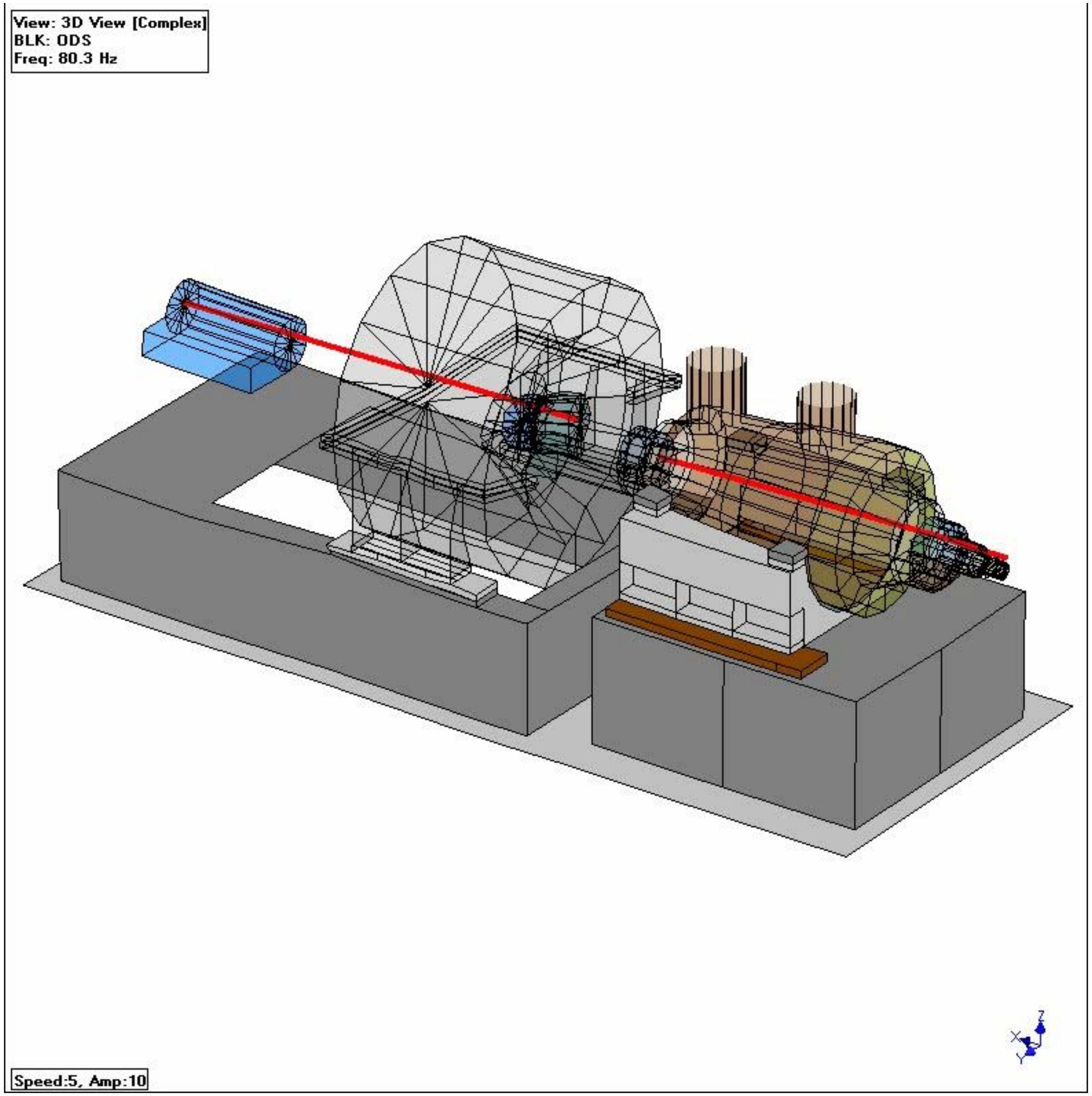




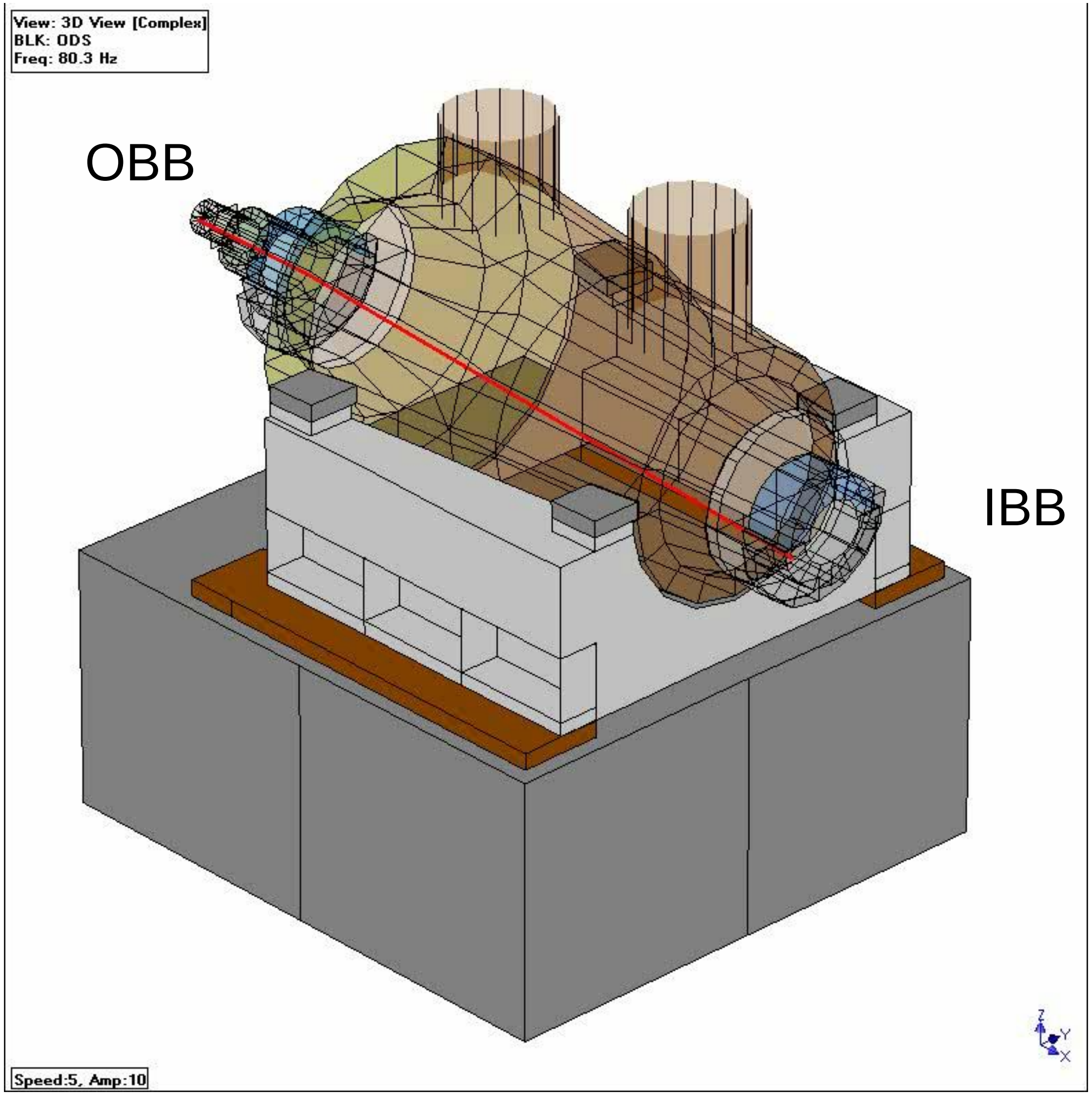
ODS Animation at 1x rpm (without Shaft)



ODS Animation at 1x rpm (with Shaft)



ODS Animation at 1x rpm (with Shaft)



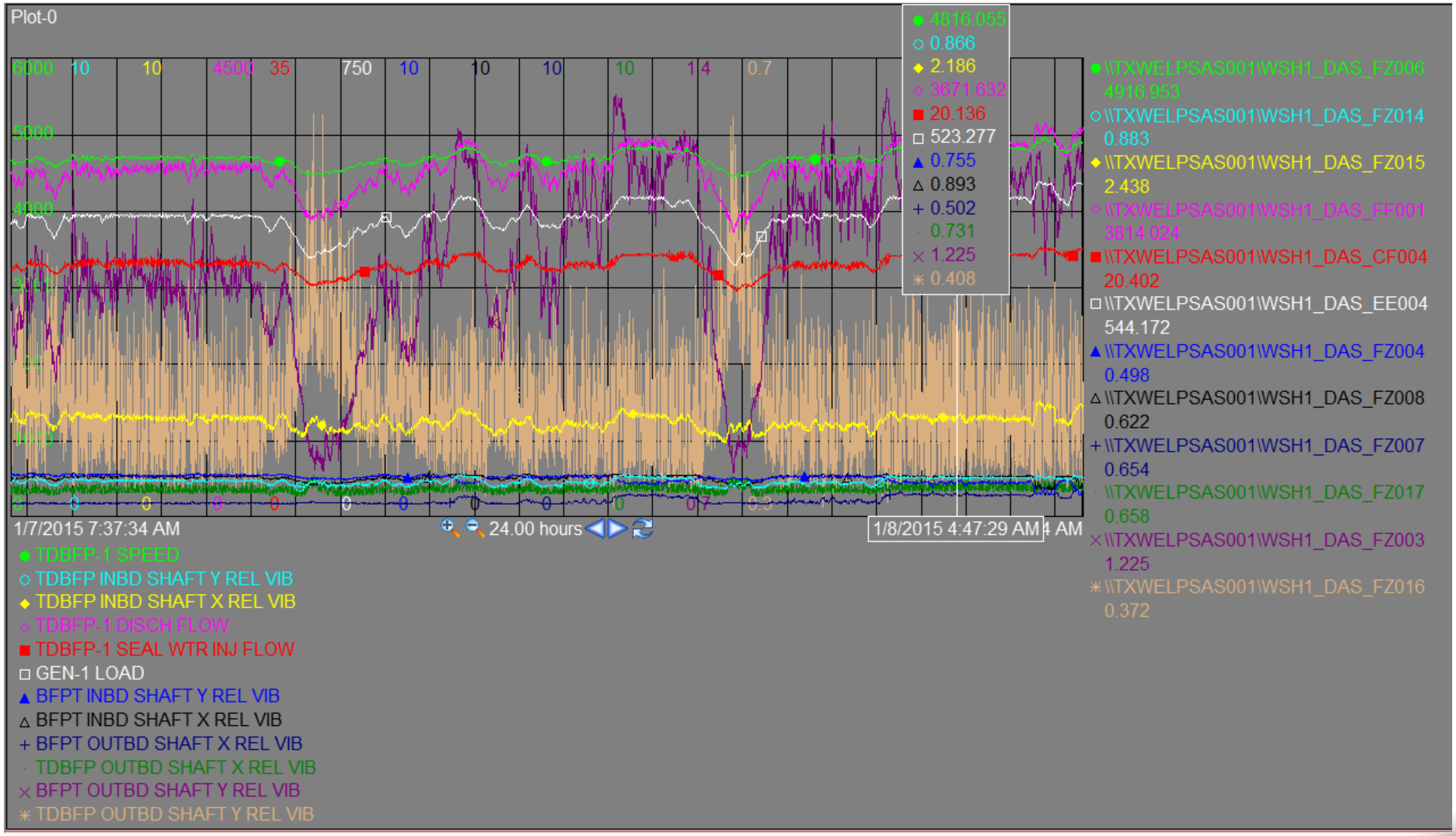
Conclusions

1. The root cause of the high vibration amplitude at the IBB of the pump is due to excessive preload acting on the bearing. Excessive piping strain was measured from the suction and discharge loading. The pump casing was acting as a pipe anchor or support.
2. Distortion of the pump casing was taking place. “Egged” shape of the wear ring from previous element confirmed that the casing was distorted at high load operation (i.e. high temperature).
3. The closest hanger to the pump on the suction piping (double hanger) was lightly or barely loaded (0% to 25% loaded) with the load indicators at the bottom end of the scale.
4. The closest hanger to the pump on the discharge piping was detected to be lightly loaded (only 25% loaded) and the upper end leaned towards west direction.
5. The suction and discharge nozzles were under severe axial or vertical loading (downwards - in compression). Both nozzles were also under torque in the CCW direction (view from the top).
6. The lateral rocking and the twist modes of the pump on its pedestal were not reacting with the running speed frequency. The first lateral mode of the pump rotor was detected above the 1x rpm (174 Hz).

Recommendations

1. The loads of the hangers for the suction and discharge piping must be corrected. The hangers must be set in their straight vertical condition. The excessive vertical load was considered the main contributor for the casing deformation and therefore, pre-loading of the IB bearing of the pump.
2. The loads for each hanger should be corrected at cold conditions, taking into account the longitudinal thermal growth of the piping. Once these issues are corrected, the new baseline alignment approach should be conducted.
3. Conduct a hot alignment for verification purposes.

Follow- Up Vibration Data After Correcting Pipe Hangers



Pump Speed:	**4816 rpm**
Load:	**523 MW**
Pump IBB-X:	**2.2 mils pk-pk**
Pump IBB-Y:	**0.9 mils pk-pk**