Solving Super-Synchronous Vibration on a Double Suction Pump

27th International Pump Users Symposium
September 12 - 15, 2011

By:

Maki M. Onari & William D. Marscher
Mechanical Solutions, Inc. – Whippany, NJ

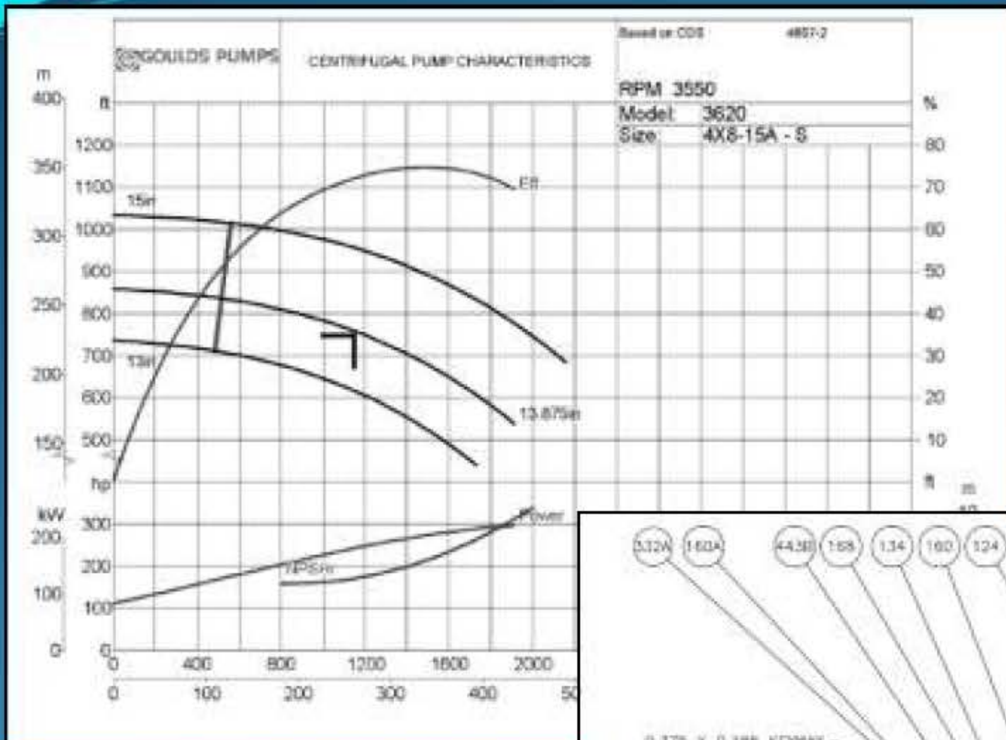
Gene Sabini
ITT Goulds – Seneca Falls, NY

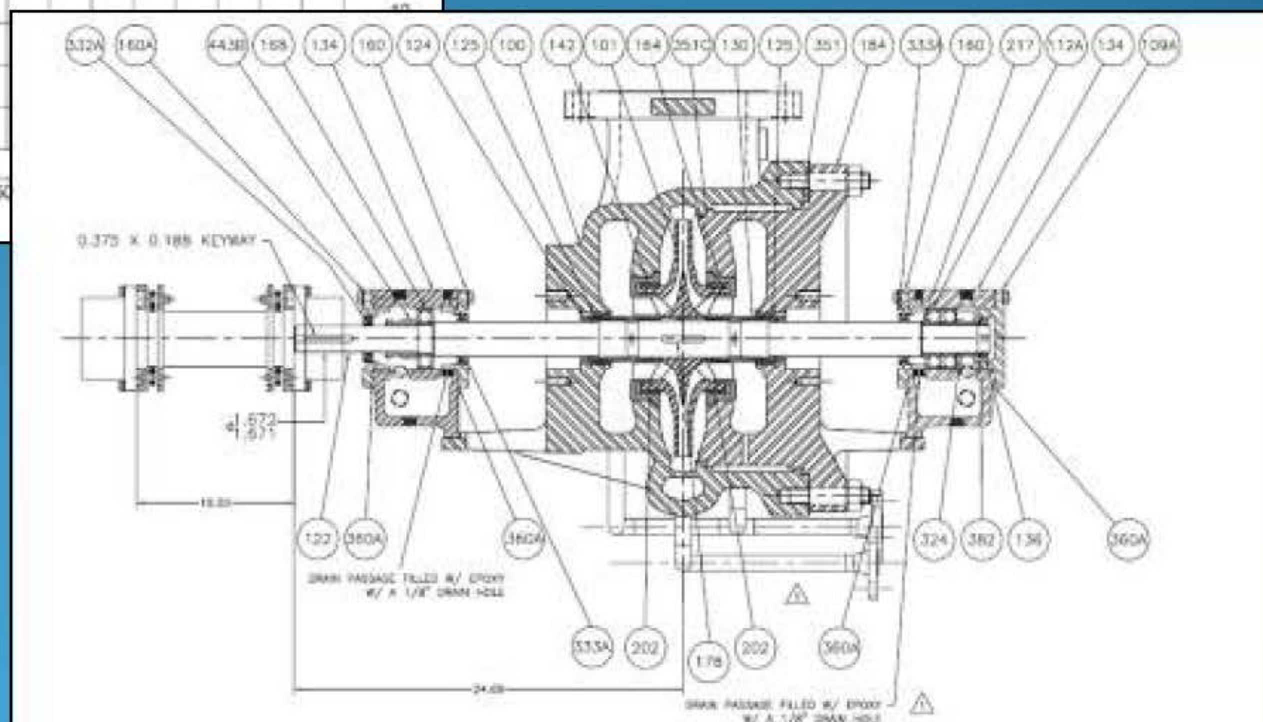
Kris Olasin
Motiva Enterprise – New Orleans, LA

Background

- Two single-stage double suction API pumps were designed by ITT Goulds to replace legacy 4x15 HVC Pumps at Motiva Convent Refinery for vacuum bottom service.
- Pump type / Model: 4x8-15A / 3620 modified to fit HVC
- Application: vacuum residuum (water, oil, and coke mix) at 705 degF
- The pump was directly coupled to an induction motor operating through a VFD to allow operation at two different speeds of 2970 rpm and 3555 rpm (rated speed) (49.5 Hz and 59.3 Hz).
- The rated capacity: 1158 GPM
- TDH: 754 ft and 244 HP

Background


GOULDS PUMPS
CENTRIFUGAL PUMP CHARACTERISTICS
RPM 3550
Model 3620
Size 4X8-15A - S
15in
13in
13.875in
Eff
Power
NPSHr

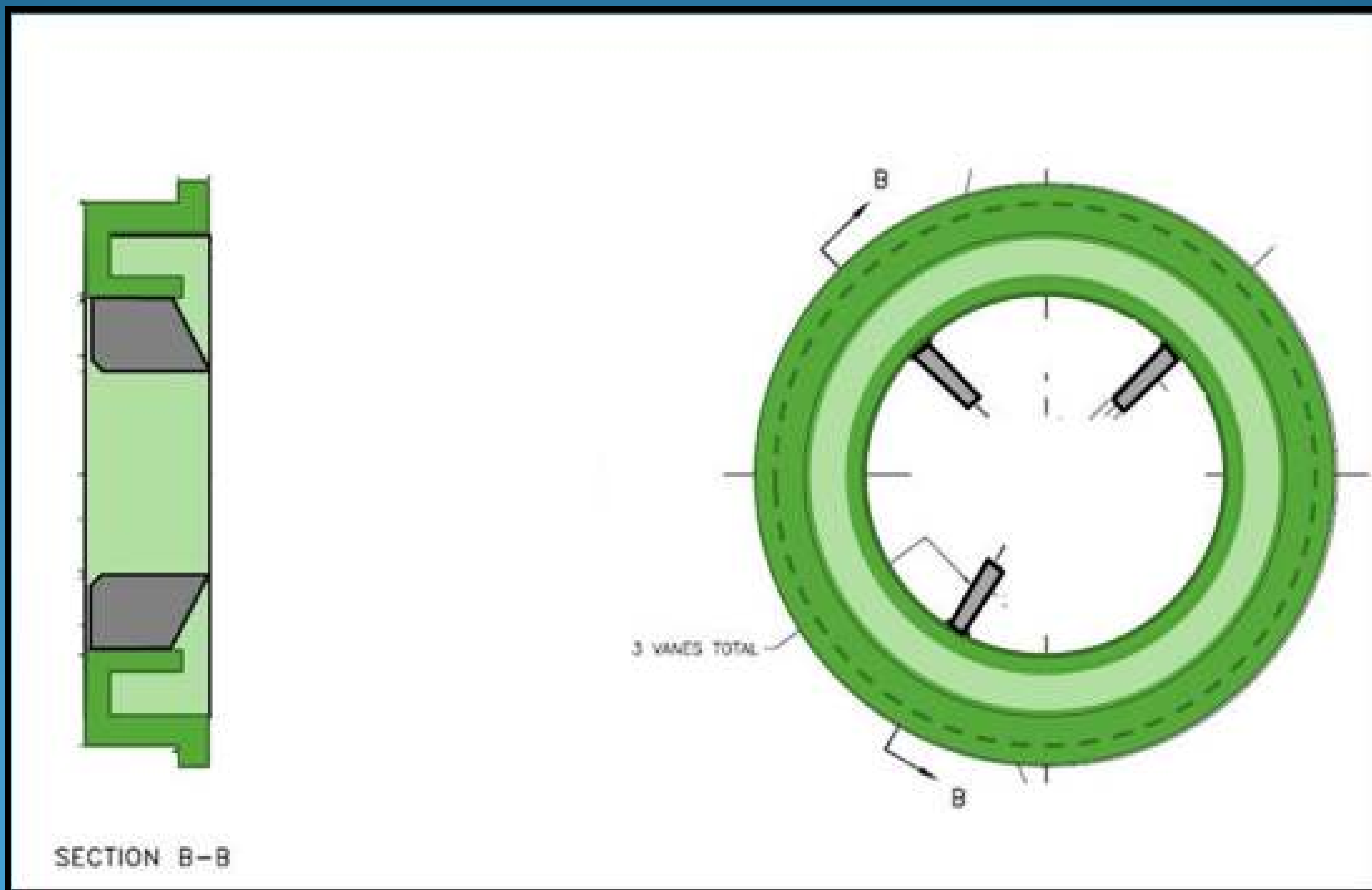

0.375 X 0.188 KEYWAY

Background

- In March 2010, during the factory performance test, high overall vibration (0.4 in/s RMS) was detected on the bearing housings at super-synchronous frequency (at approximately 100 Hz).
- API 610 10^{th} Edition spec.: 0.12 in/s RMS at BEP and 0.154 in/s RMS below 70% BEP.

Background

- Pump provided with "wrap-around" coke-crusher wear ring design with 3 struts (unevenly spaced) and a six-vane impeller.



Background

- Over 8 months period, the pump was tested in several test facilities with different drivers suggesting that the problem was internal within the pump and not related to a support structure natural frequency nor acoustic natural frequencies from the piping system. Later the pump was sent to the R&D Facilities for extensive testing, where the pump was tested at two different speeds without significant difference in the high vibration.
- Internal modifications were implemented by off-setting the coke-breaker struts and also removing them without success.
- MSI was requested to evaluate the basis for highly unusual and unexpected rotordynamic issues.

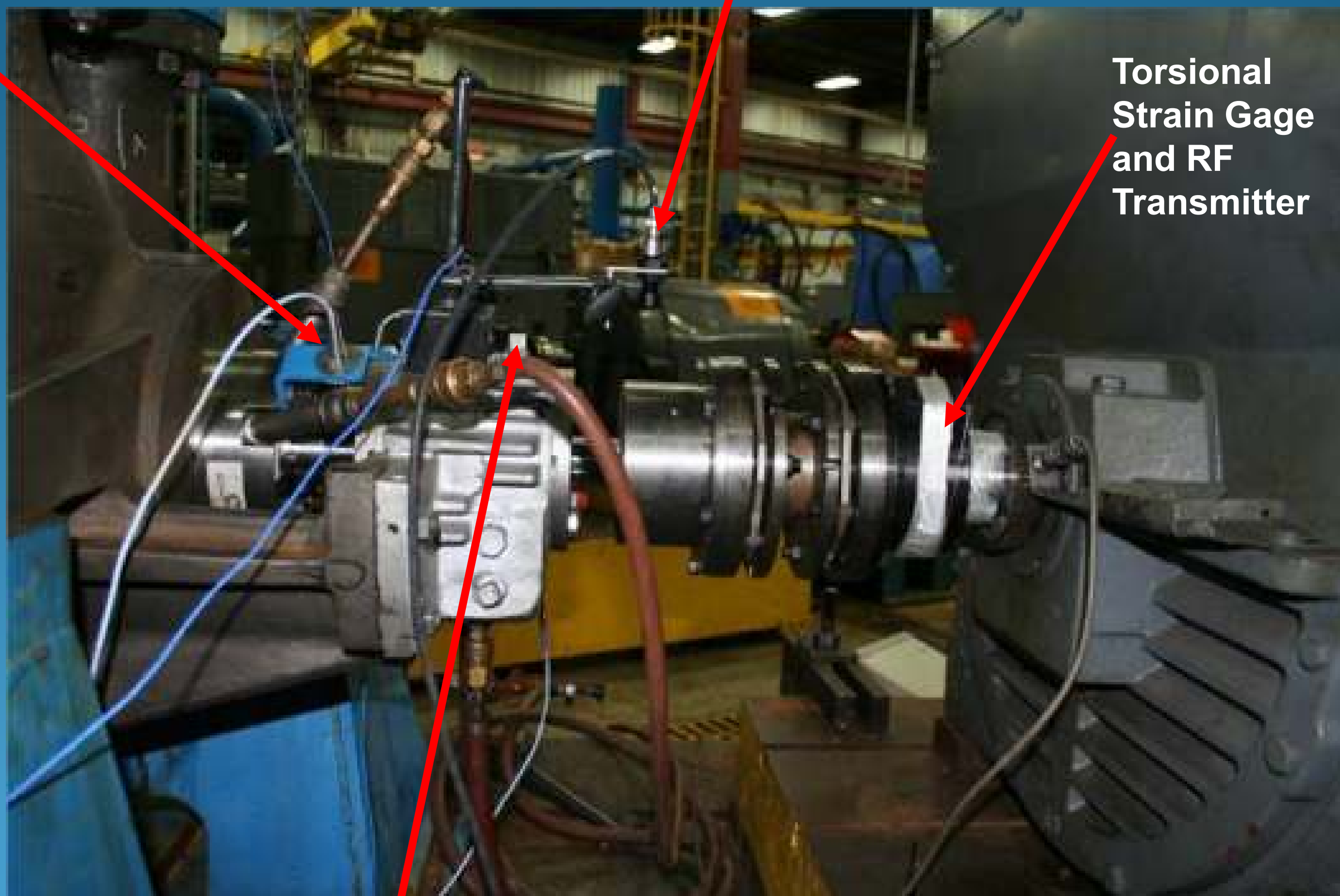
Vibration Testing

- Continuous Monitoring testing during transient and steady operation to monitor the shaft and bearing vibration amplitude, structural natural frequencies, pressure pulsations, torsional natural frequencies, etc.
- Operating Deflection Shape (ODS) testing during steady operation.
- Experimental Modal Analysis (EMA) test to determine the natural frequencies of the pump structure and the rotor system.

Vibration Monitoring

Radial Proximity Probes

Optical Tachometer



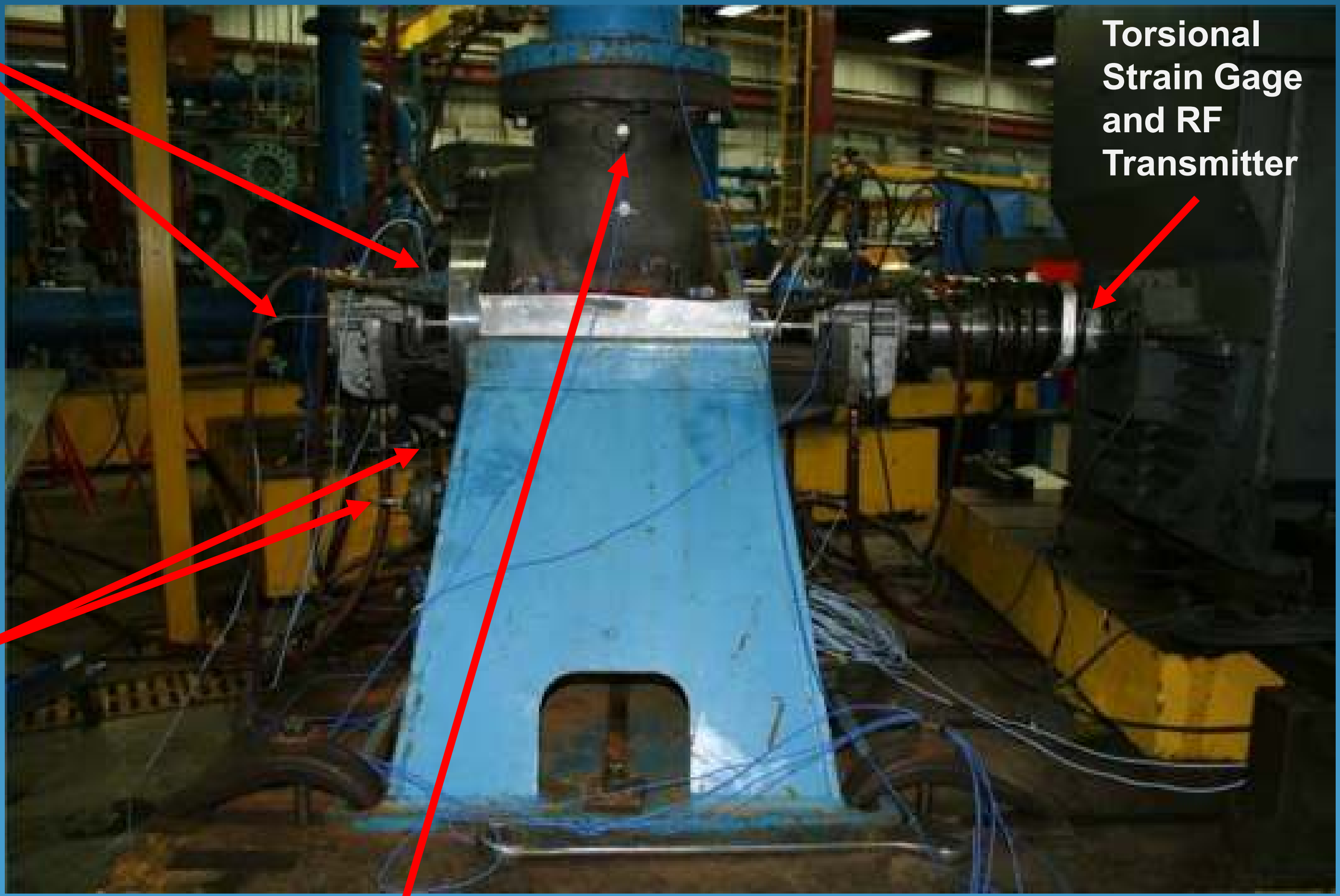
Torsional Strain Gage and RF Transmitter

Tri-axis Accelerometer

Vibration Monitoring

Radial & Axial Proximity Probes

Torsional Strain Gage and RF Transmitter



Dynamic Pressure Transducers

Tri-Axis Accelerometer

Vibration Monitoring

Summary Table Overall Vibration Amplitude at Two Different Speeds and Flow Rates

Pump OBB (Ch 1–3) | Pump IBB (Ch 4–6) | Suct. Nozzle (Ch 7–9) | Disch. Nozzle (Ch 10–12)

Condition				Overall Vibration (in/s RMS)											
Time	Speed (Hz)	Flow (GPM)	TDH (ft)	Ch 1 (Axial)	Ch 2 (Hor)	Ch 3 (Vert)	Ch 4 (Axial)	Ch 5 (Hor)	Ch 6 (Vert)	Ch 7 (Vert)	Ch 8 (Axial)	Ch 9 (Hor)	Ch 10 (Vert)	Ch 11 (Axial)	Ch 12 (Hor)
13:56	49.5	304	636	0.13	0.024	0.13	0.11	0.11	0.145	0.085	0.14	0.09	0.06	0.11	0.1
14:06	49.5	6.4	628	0.15	0.03	0.17	0.12	0.25	0.2	0.09	0.16	0.09	0.07	0.13	0.1
14:15	49.5	311	634	0.13	0.03	0.13	0.11	0.11	0.16	0.09	0.12	0.08	0.06	0.11	0.09
14:24	49.5	611	612	0.12	0.02	0.14	0.12	0.12	0.16	0.09	0.14	0.09	0.06	0.13	0.09
14:36	49.5	911	563	0.14	0.03	0.16	0.19	0.15	0.18	0.14	0.13	0.09	0.07	0.13	0.1
14:45	49.5	966	552	0.13	0.12	0.15	0.15	0.14	0.16	0.18	0.13	0.09	0.07	0.13	0.09
14:53	49.5	1226	484	0.15	0.14	0.19	0.21	0.18	0.22	0.22	0.15	0.1	0.09	0.17	0.1
15:00	49.5	1533	359	0.14	0.14	0.16	0.2	0.18	0.22	0.24	0.14	0.09	0.09	0.16	0.2
15:19	59.25	6	885	0.24	**0.3**	0.25	0.23	**0.47**	**0.34**	0.17	0.19	0.18	0.13	0.2	0.18
15:24	59.25	359	876	0.22	**0.31**	0.2	0.22	**0.37**	0.29	0.11	0.18	0.17	0.11	0.19	0.18
15:29	59.25	727	853	0.24	**0.31**	0.23	**0.3**	0.28	**0.31**	0.16	0.19	0.17	0.12	0.21	0.18
15:34	59.25	1092	781	0.25	**0.31**	**0.33**	0.28	0.26	0.28	0.22	0.19	0.17	0.15	0.22	0.18
15:39	59.25	1163	764	0.26	**0.32**	**0.38**	**0.31**	0.27	**0.31**	0.24	0.19	0.17	0.17	0.24	0.19
15:44	59.25	1463	680	0.25	0.29	0.28	**0.33**	0.23	0.27	0.19	0.17	0.15	0.13	0.18	0.17
15:49	59.25	1823	511	0.22	0.23	0.18	0.22	0.21	0.2	0.12	0.15	0.14	0.21	0.16	**N/A**
15:57	59.25	1159	767	0.24	**0.32**	0.26	0.27	0.26	0.27	0.16	0.19	0.17	0.14	0.21	0.18

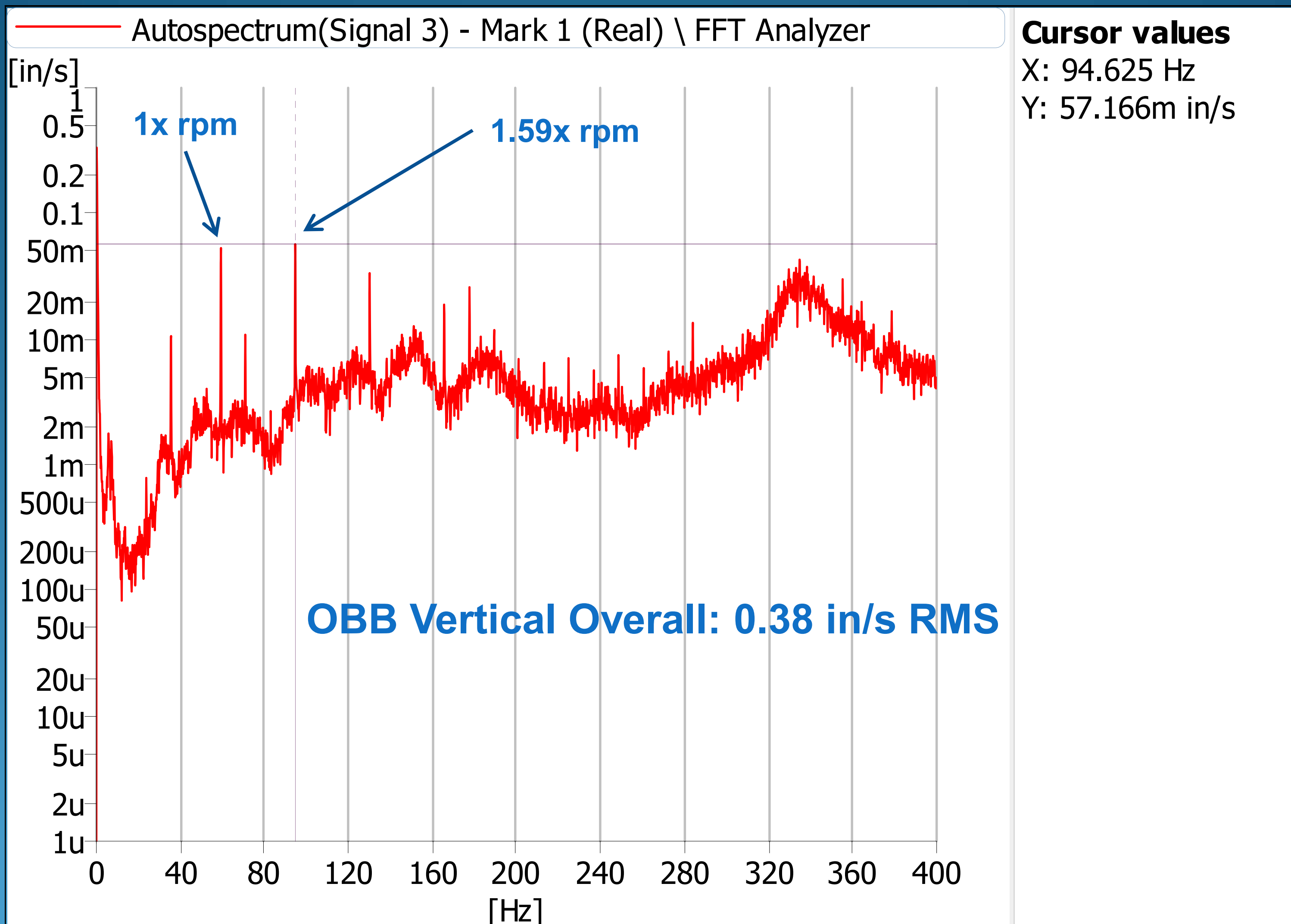
Worse vibration case for each speed condition

Note: Vibrations above 0.30 in/s RMS are red/bold

Continuous Monitoring Test

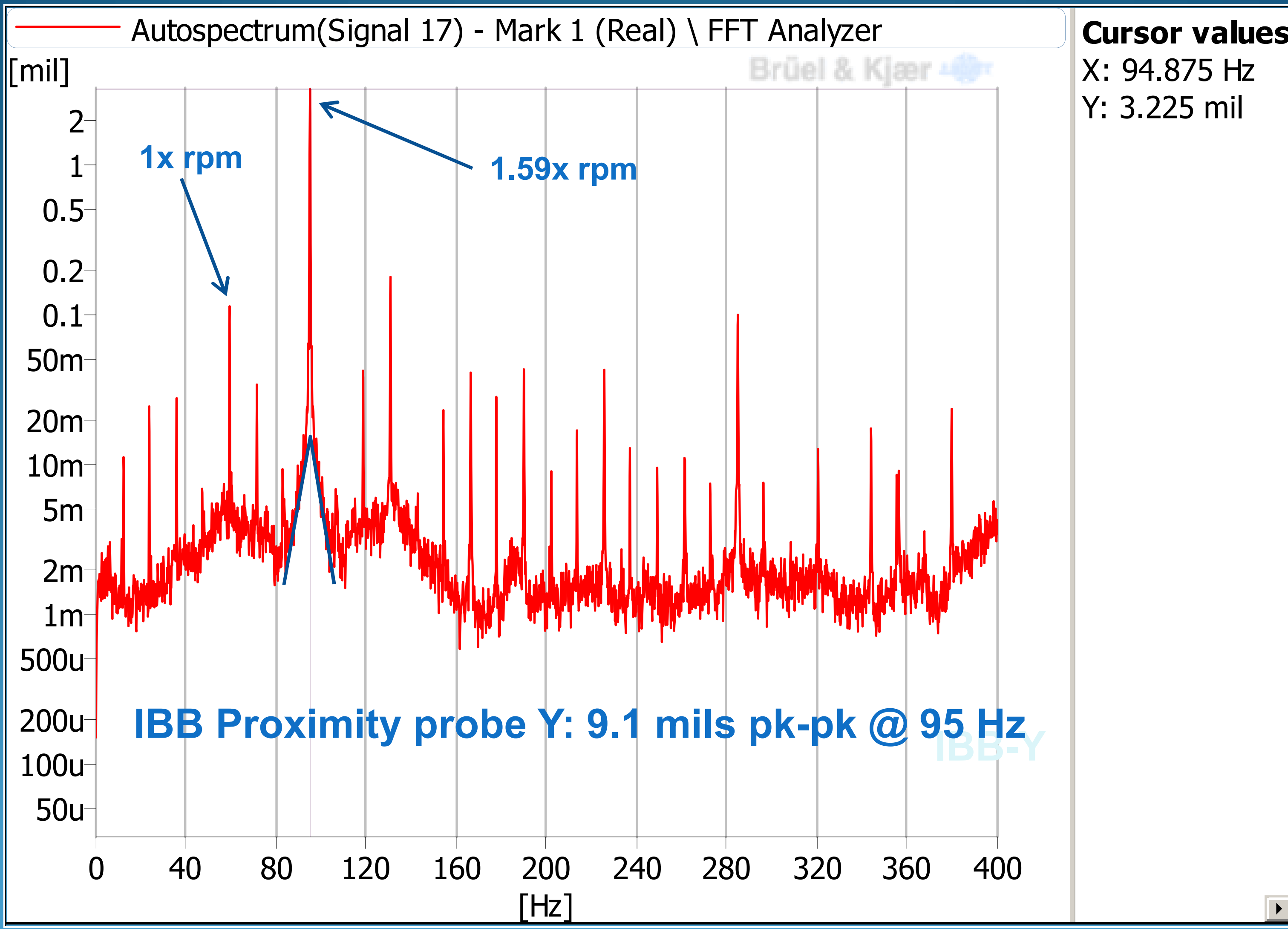
Running Speed 3570 rpm – 1163 GPM – 764 ft

Vibration Monitoring



Running Speed 3570 rpm (59.5 Hz) – 1163 GPM – 764 ft

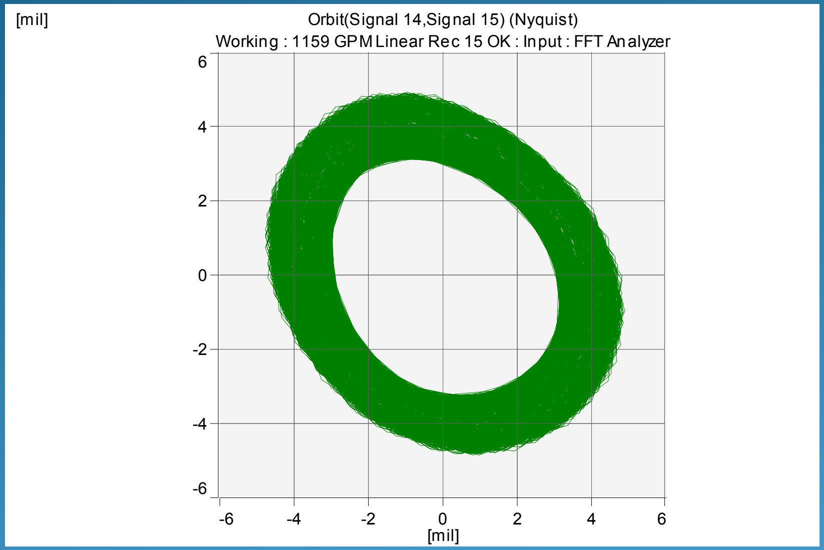
Vibration Monitoring



Running Speed 3570 rpm (59.5 Hz) – 1163 GPM – 764 ft

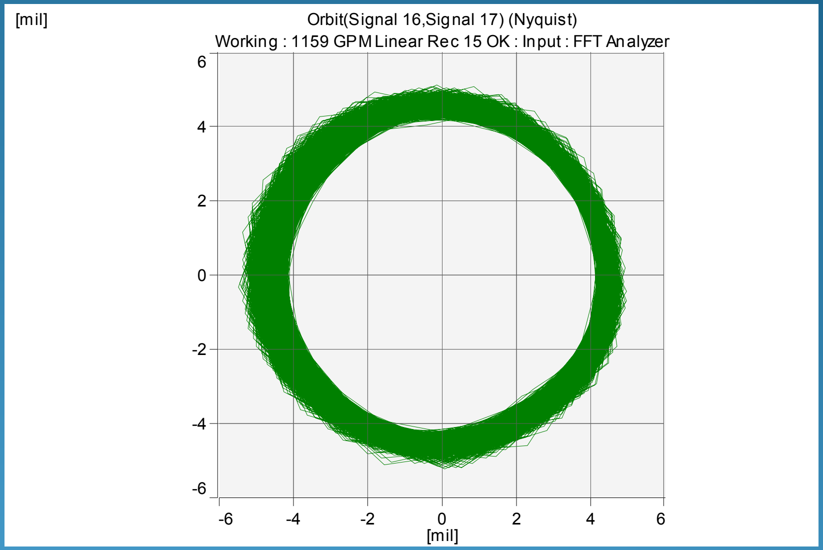
Vibration Monitoring

~9.0 mils pk-pk



OBB

~9.5 mils pk-pk



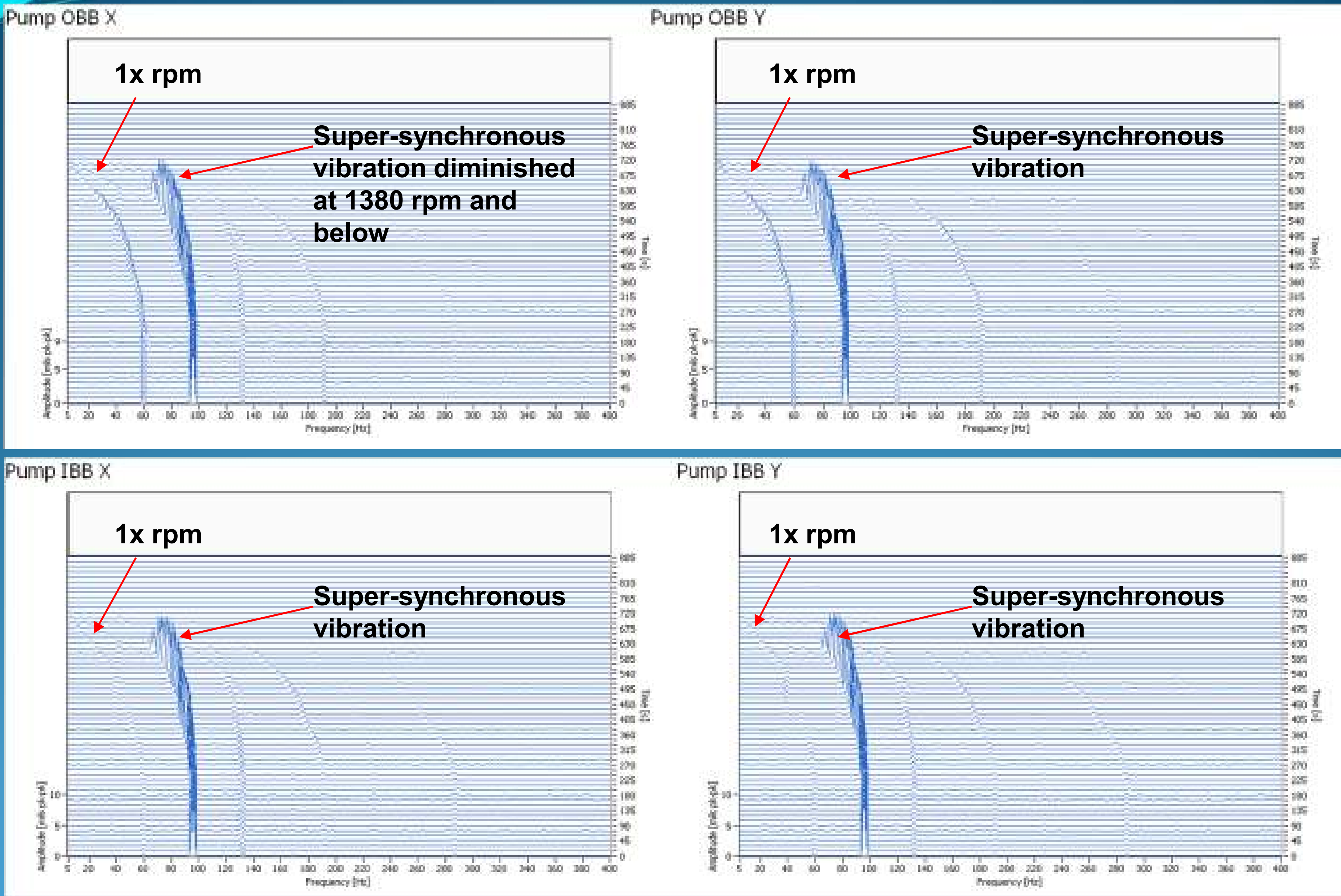
IBB

Radial Proximity Probes

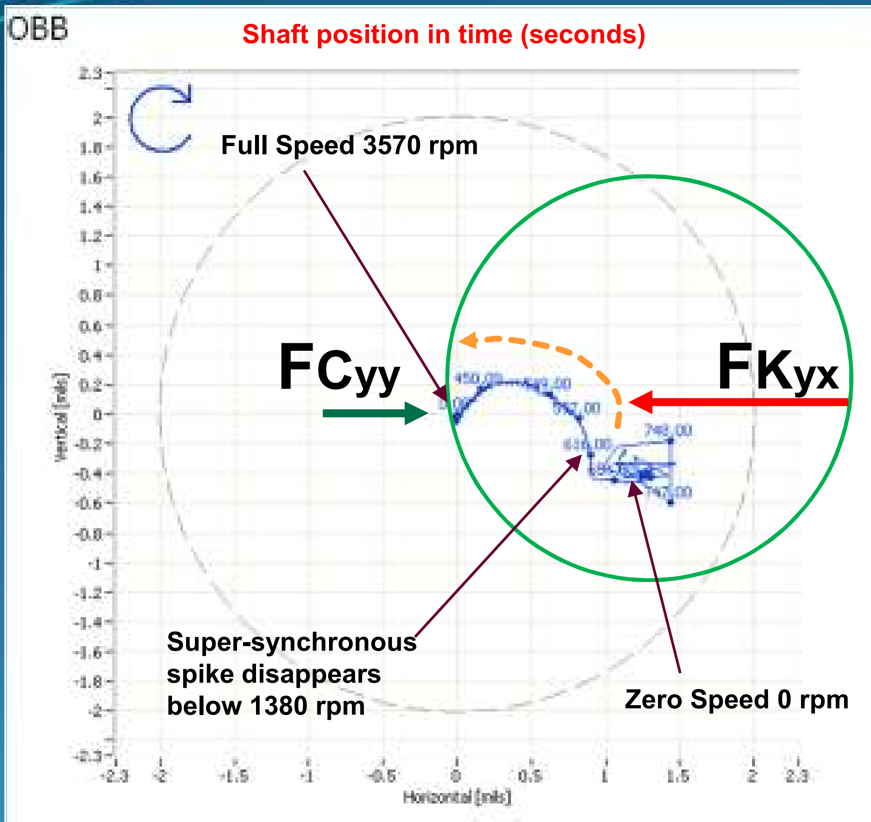
Running Speed 3570 rpm (59.5 Hz) – 1163 GPM – 764 ft

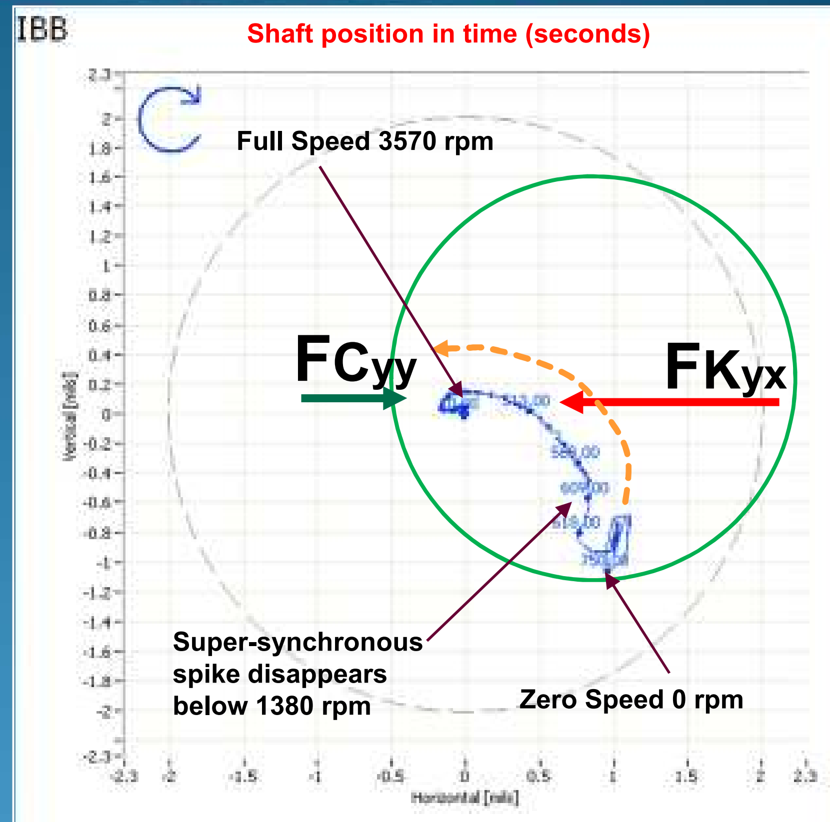
Transient Continuous Monitoring During Coast-Down from 3570 rpm to 0 rpm

Vibration Monitoring



Vibration Monitoring

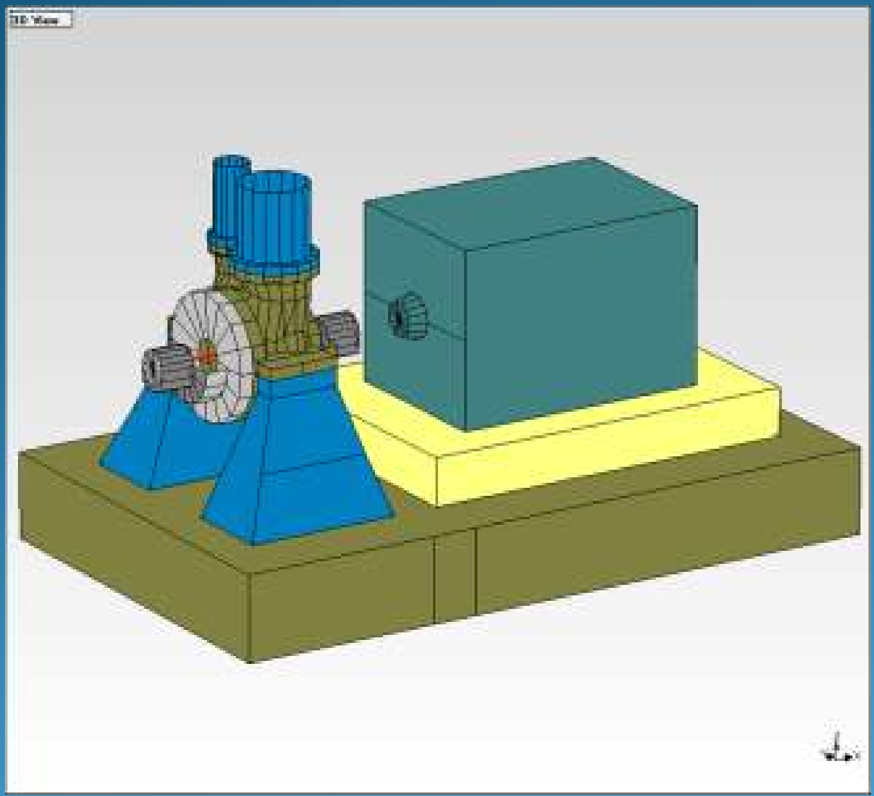




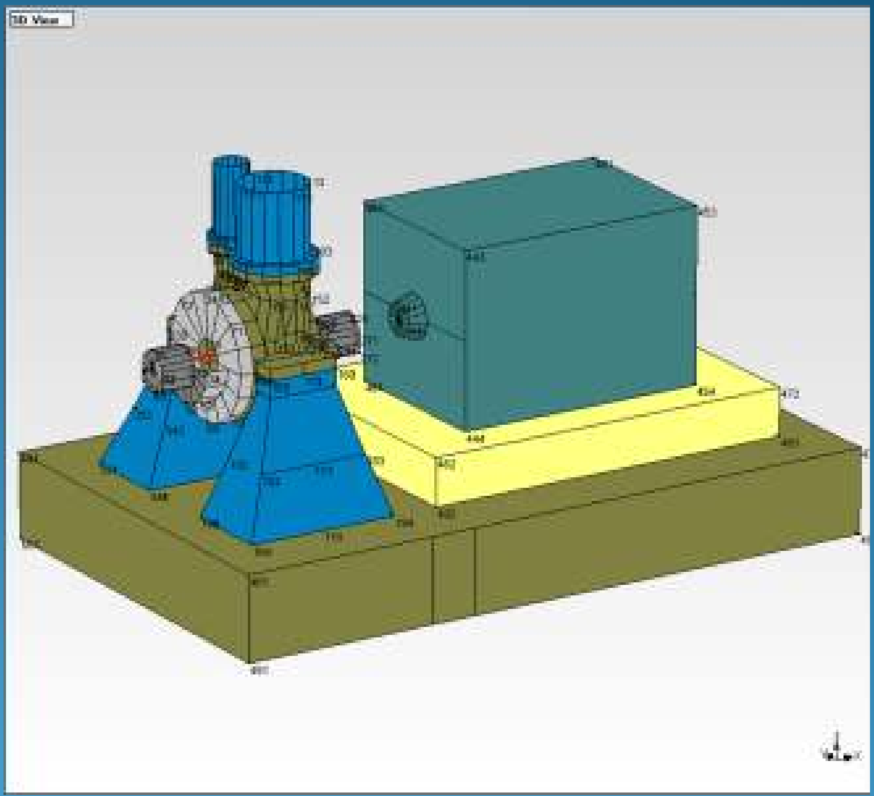
- Shaft center-line plots viewed from the NDE
- Shaft position in seconds during the coast-down from 3570 rpm (0 sec) to 0 rpm 750 sec.
- The shaft moves towards the upper left position after the pump starts (towards 9 O'clock position).
- Green circles represents the shaft centerline plots simulating the start-up process from the bottom (off-set plots). Note the static cross-coupled stiffness force (FKyx) is larger than the calculated synchronous damping force (FCxx) when the shaft speed is above 1380 rpm.

Operating Deflection Shape (ODS) Testing
Forced Response Test Results

Operating Deflection Shape

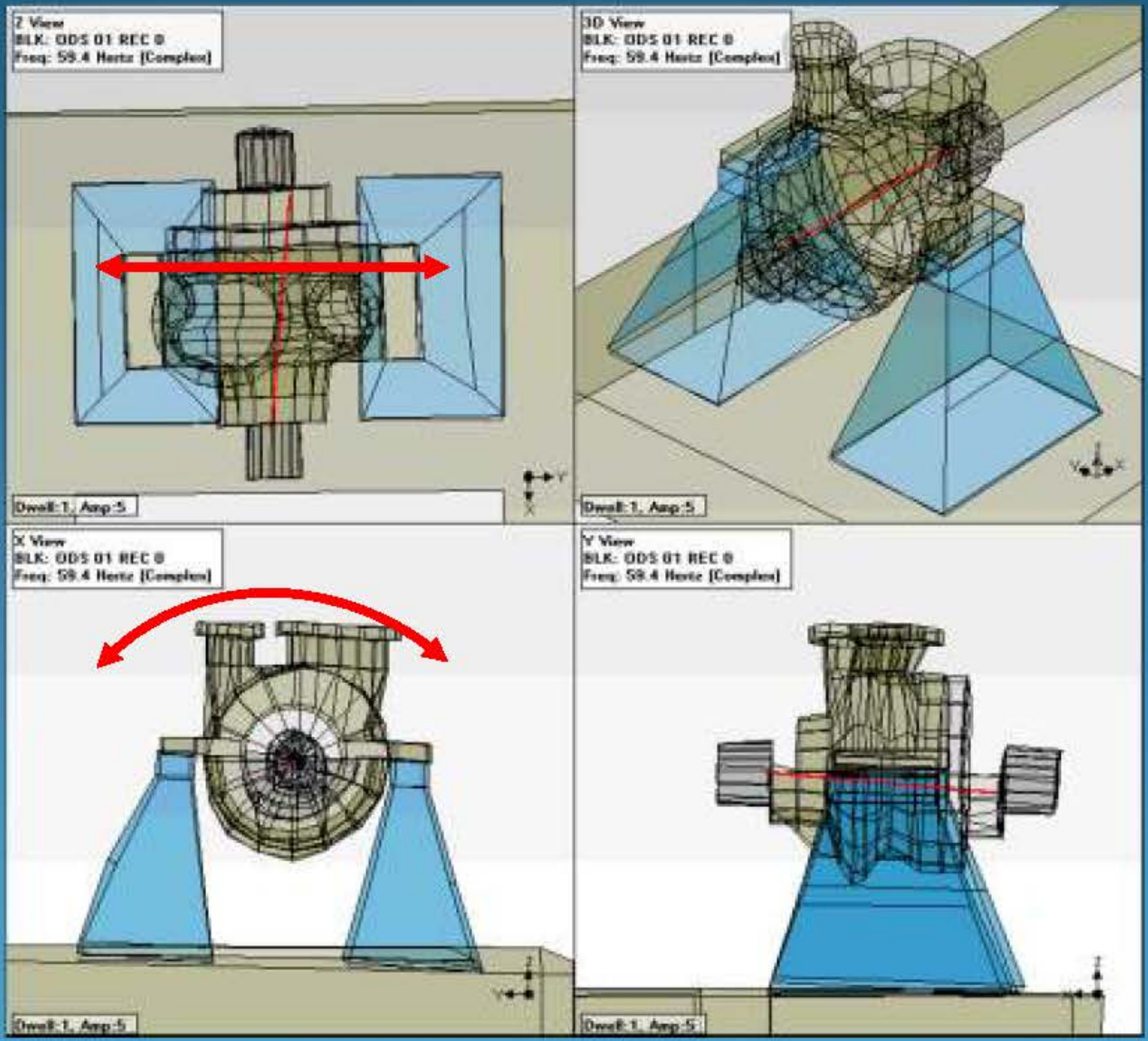


ODS Computer Model



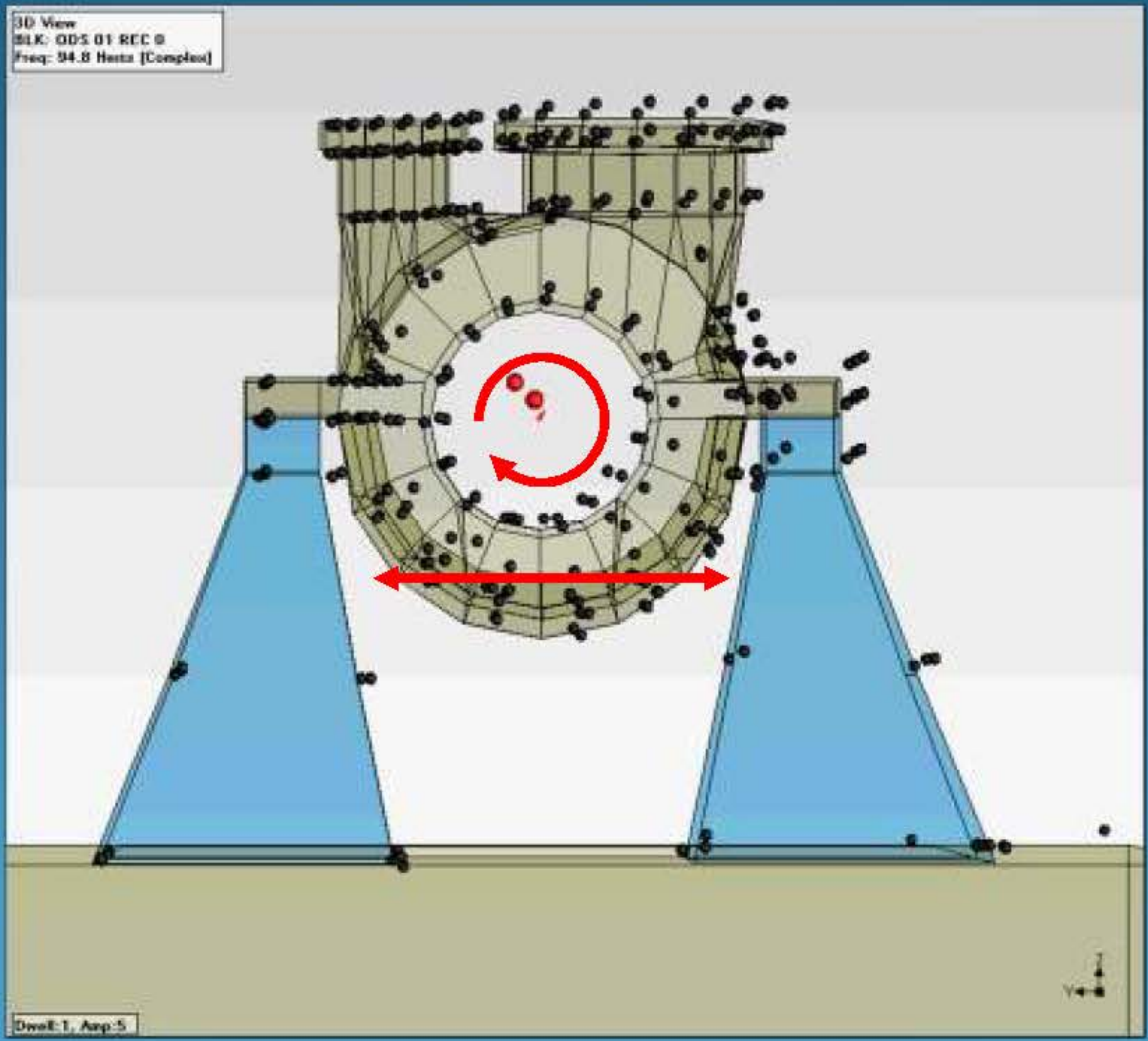
Over 600 vibration measurement/ directions

Operating Deflection Shape



ODS Animation @ 1x rpm (59.4 Hz)

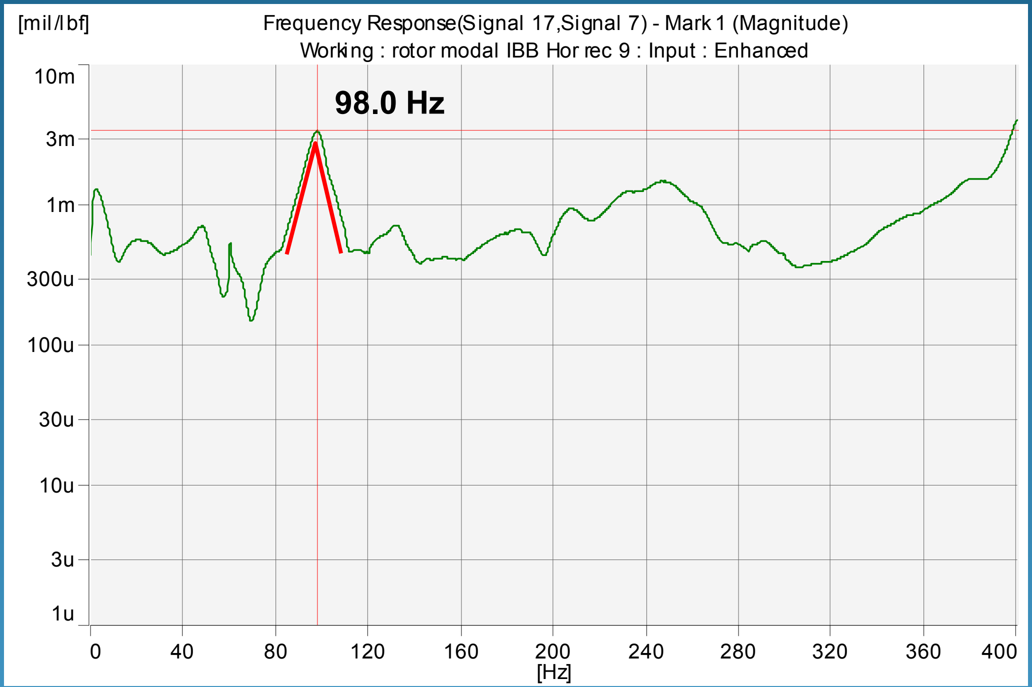
Operating Deflection Shape



ODS Animation @ 94.8 Hz

Experimental Modal Analysis (EMA) Testing
Frequency Response Test Results

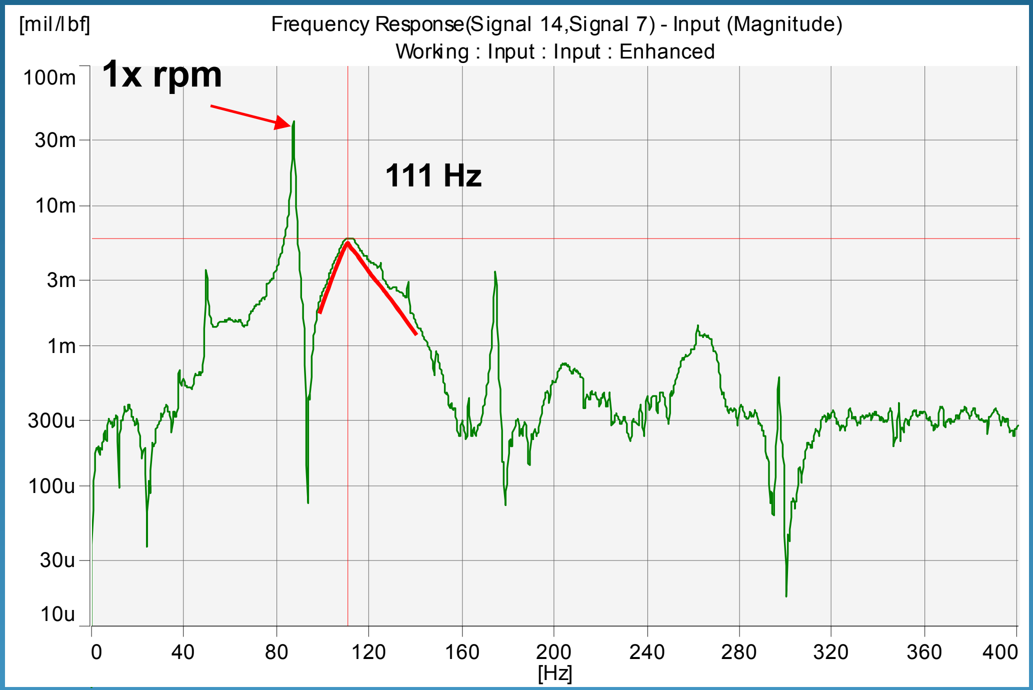
Experimental Modal Analysis



Pump Rotor Frequency Response Function (FRF) plot while the pump was not operating

Radial Proximity Probe IBB-Y

Experimental Modal Analysis



FRF plot while the pump was operating at 49.5 Hz – 615 GPM and 616 ft of TDH

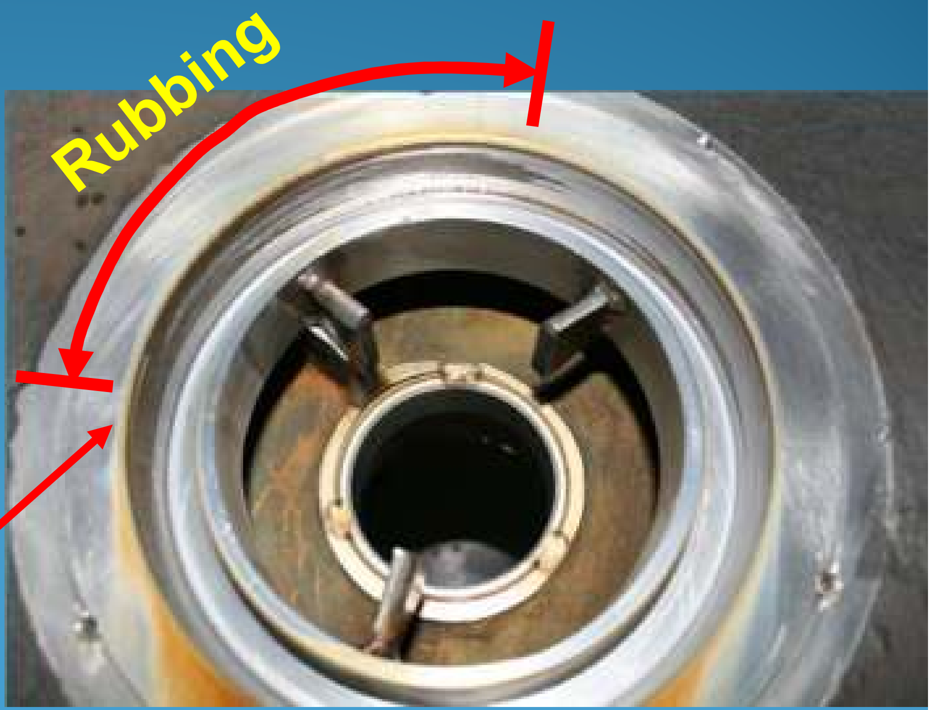
Reading from the radial proximity probe OBB-X

Note that the rotor natural frequency shifted upwards with the speed and the stiffness from the wear rings.

Visual Inspection

Visual Inspection

IB side impeller wear ring with evidence of rubbing



Case wear ring rub only between 9 and 12 O'clock as viewed from the NDE or OBB

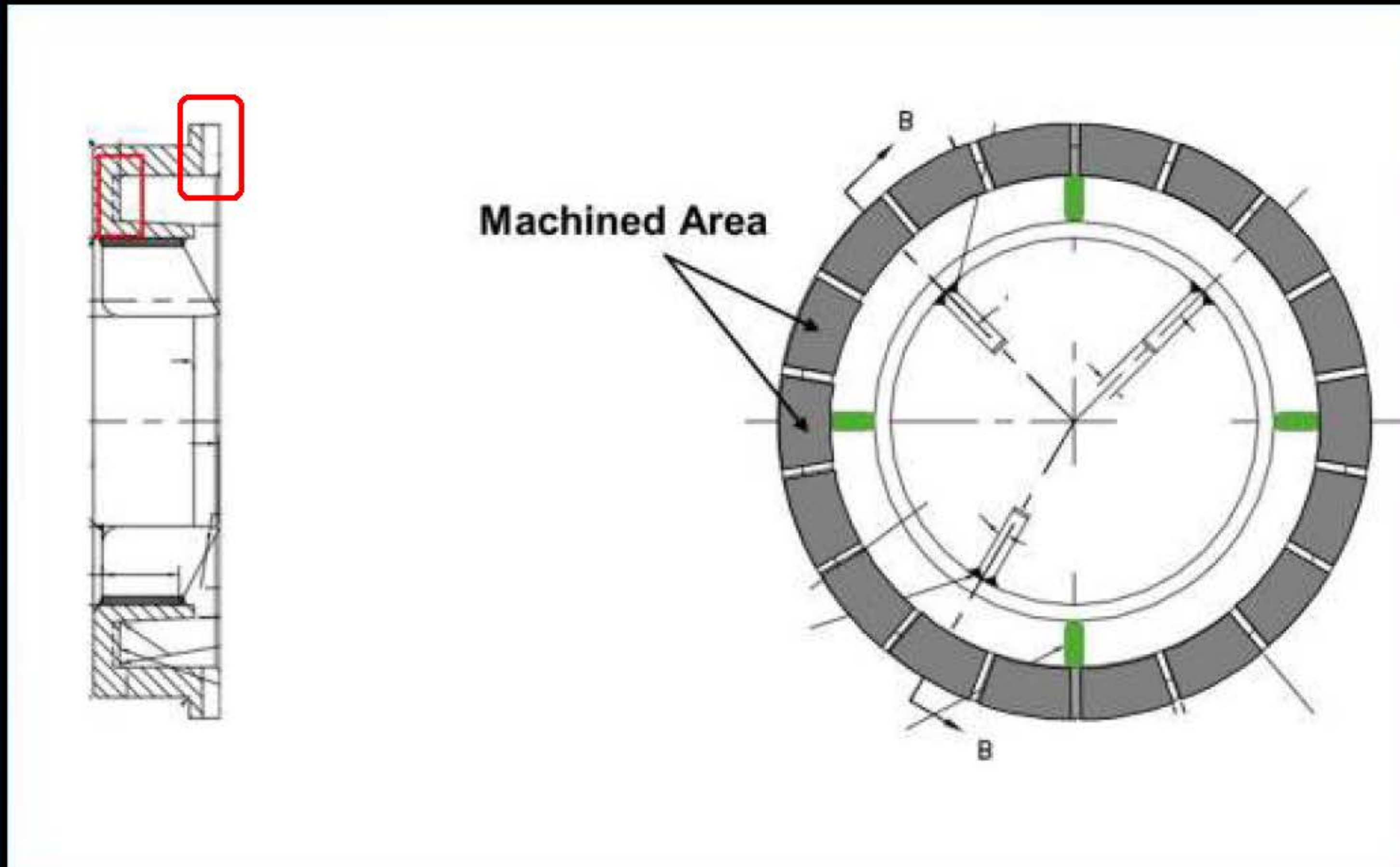
Preliminary Conclusions

1. The high vibration of the pump was due to a rotordynamic instability exciting the first bending mode of the pump shaft.
2. The excitation source was likely from fluid whirl and non axi-symmetric pressure within the "wrap-around" coke crusher wear rings that were acting as large sleeve bearings.
3. The ratio between the super-synchronous vibration frequency with respect to the running speed frequency was not constant, pending to sure out stall phenomena.
4. The first bending mode of the pump rotor at 98 Hz shifted to approximately 111 Hz while the pump was operating. The large excitation at 94.9 Hz, apparently from super-synchronous fluid whirl led to entrainment of the nearby rotor's lateral natural frequency, causing large amplification of the shaft vibration at the super-synchronous frequency.

Proposed Recommendations

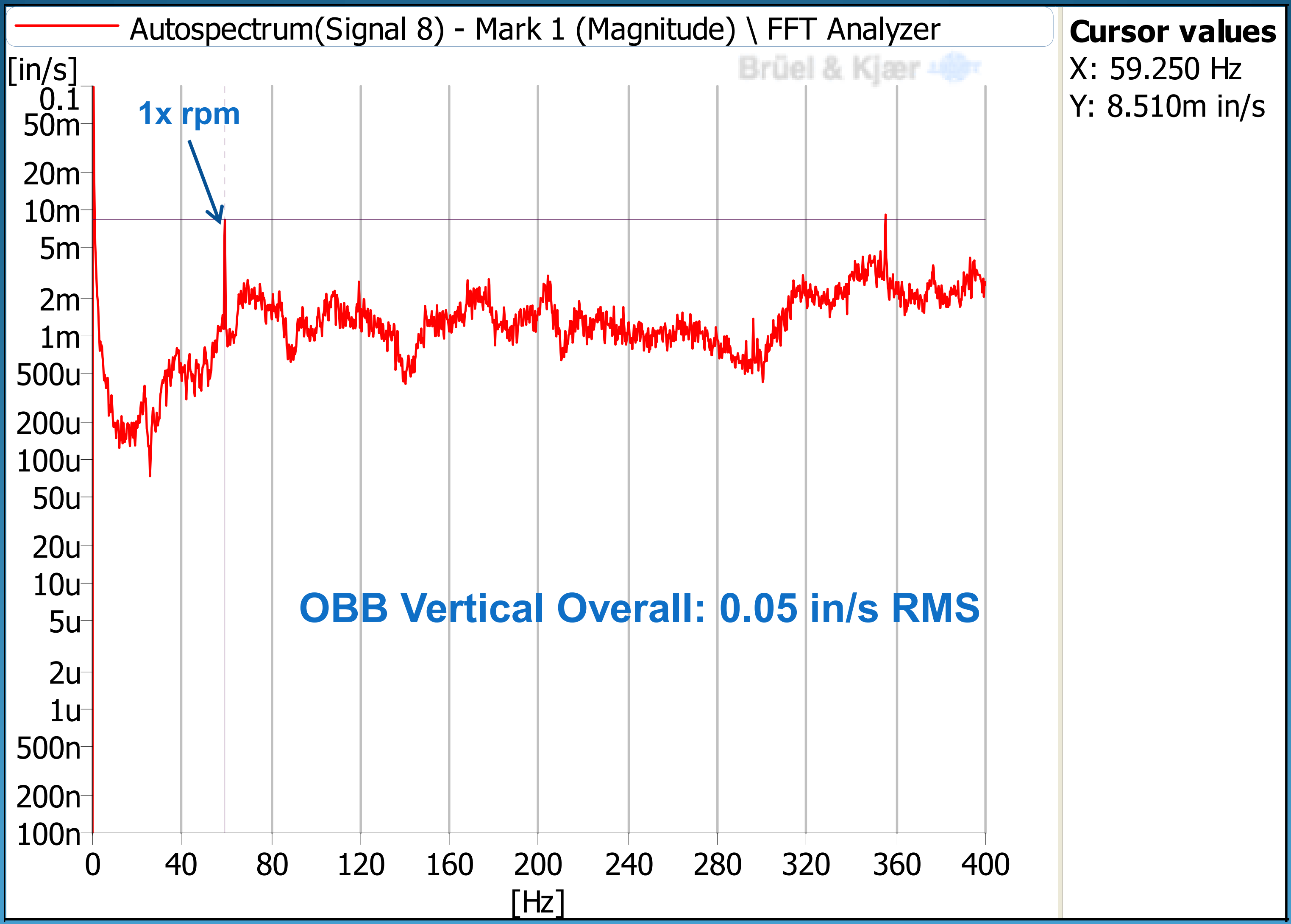
1. Based on a collaborative discussion with the OEM, one fix option for this instability was the addition of swirl-breakers by milling radial vanelets (slots) on the case wear ring. These 18 slots were equally spaced leaving vanelets of 1/8" of width and ¼" of axial depth.
2. The wrap-around wear ring was modified with a tapered design at the ID side of the ring with approximately 2 to 3 degrees with the widest clearance at the exit of the seal at the pump suction.

Proposed Recommendations



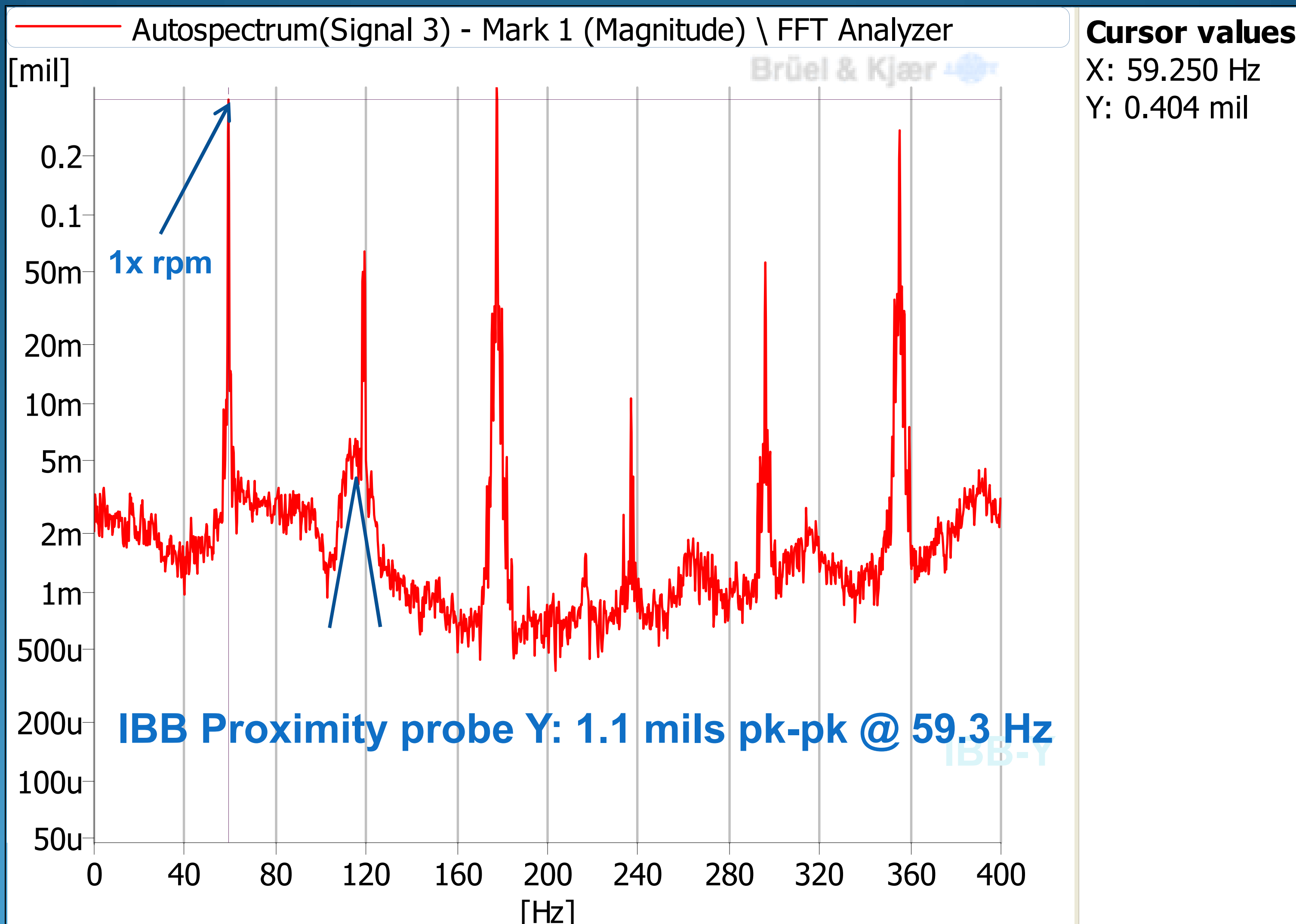
Follow-Up Testing

Vibration Monitoring



Running Speed 3555 rpm (59.3 Hz) – 1163 GPM – 741 ft

Vibration Monitoring

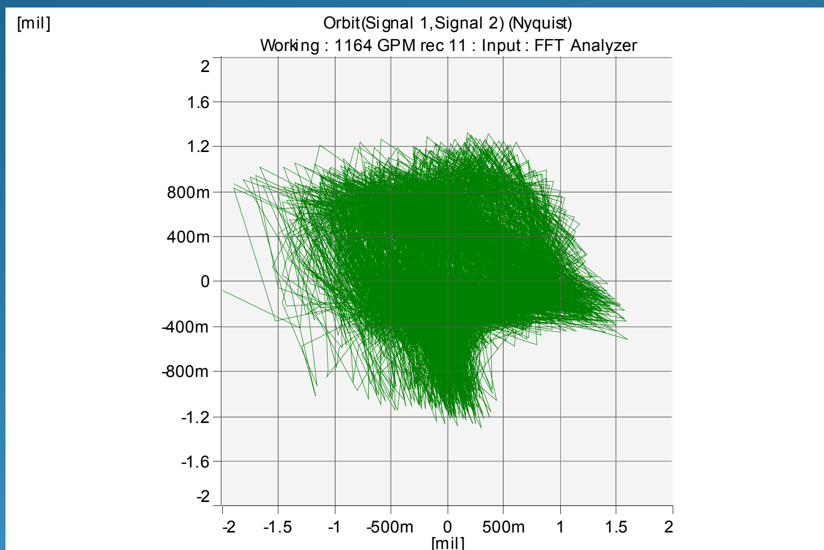


Running Speed 3555 rpm (59.3 Hz) – 1163 GPM – 741 ft

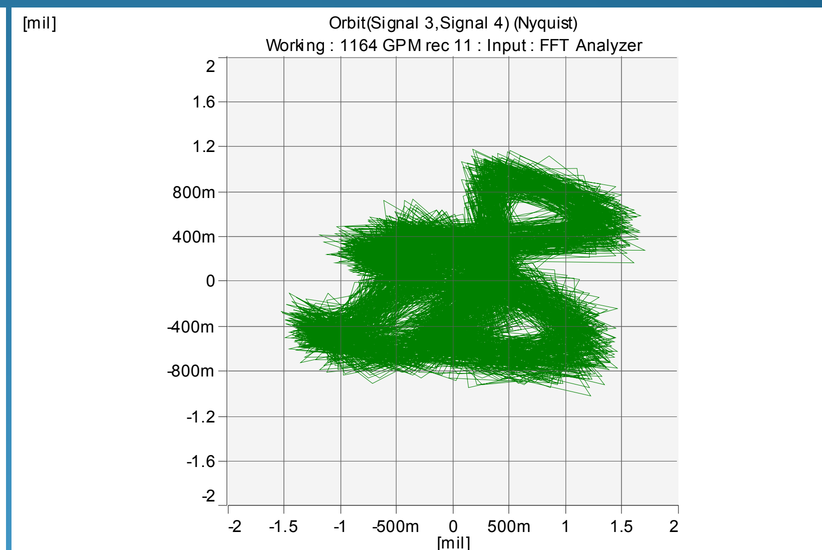
Vibration Monitoring

~2 mils pk-pk
Mostly Run-Out

~2.1 mils pk-pk
Mostly Run-Out



OBB



IBB

Radial Proximity Probes

Running Speed 3555 rpm (59.3 Hz) – 1163 GPM – 741 ft

Conclusions

- The root cause of the vibration on this pump was due to a rotordynamic instability exciting the first bending mode of the pump rotor.
- A typical rotor dynamic analysis would not be able to predict this type of excitation forcing function. In order to predict this type of excitation, a detailed CFD analysis would need to be performed, which is not a common practice.
- After modifications performed on the wrap-around wear ring, the super-synchronous vibration disappeared. The overall vibration from the bearing housing and the shaft were reduced by a factor of 4.5.
- EMA testing of the rotor, while in operation (Time-Averaged Pulse technique), is a powerful troubleshooting tool to determine rotor natural frequencies in any pumping system or turbomachine.

Thank you

Any Questions…?