Solving Super-Synchronous Vibration on a Double Suction Pump

William D. Marscher, P.E.:

– Mechanical Solutions, Inc. - Technical Director
– B.S.M.E. & M.S.M.E. Cornell U., M.S. Applied Mechanics RPI, Fellow STLE
– Past President, STLE, & Machinery Failure Prevention Technology Society
– Member of US Delegation, ISO108 S2 Machinery Vibration Standard Committee
– Author, 7 Handbook Chapters on Vibration & Predictive Maintenance
& co-author "Centrifugal Pump Design & Performance, Oxford Univ. Press, 1997

Maki M. Onari:

– Mechanical Solutions, Inc. – Principal Engineer
– B.S.M.E., Zulia Universiry, Venezuela
– Staff Engineer, PDVSA Machinery Maintenance
– Responsible for all MSI Turbomachinery Testing
– Co-Author Pump Vibration Chapter, McGraw-Hill Pump Handbook

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<u>Eugene P. (Gene) Sabini:</u>

– Director of Research at ITT Goulds Pumps (Retired in 2012).

– Responsible for applied research and hydraulic design of all new products and field re-rates.

–43 years of experience in the pumping industry including design and development of many centrifugal pumps for the chemical, API, power utilities, and municipal industries.

– Authored of numerous papers and holds seventeen patents.

– Recipient of The 2007 ITT Industries' Engineered for Life Lifetime Achievement award for Hydraulic Design Expertise and Innovation.

– BSME and M.S. degree from Stevens Institute of Technology in Hoboken NJ and has been member of the International Pump Users Symposium Advisory Committee since 1999.

<u>Kris Olasin:</u>

– Rotating machinery engineer at Motiva Enterprises Convent Louisiana Refinery. He

– Currently is the lead rotating equipment specialist assigned to projects.

– Duties include application of rotating equipment specifications to new purchases, equipment fabrication drawing review, and installation quality assurance.

– His 30 years at this facility have included assignments for run and maintain support of rotating equipment and several new unit projects including upgrade of existing units.

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By:

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

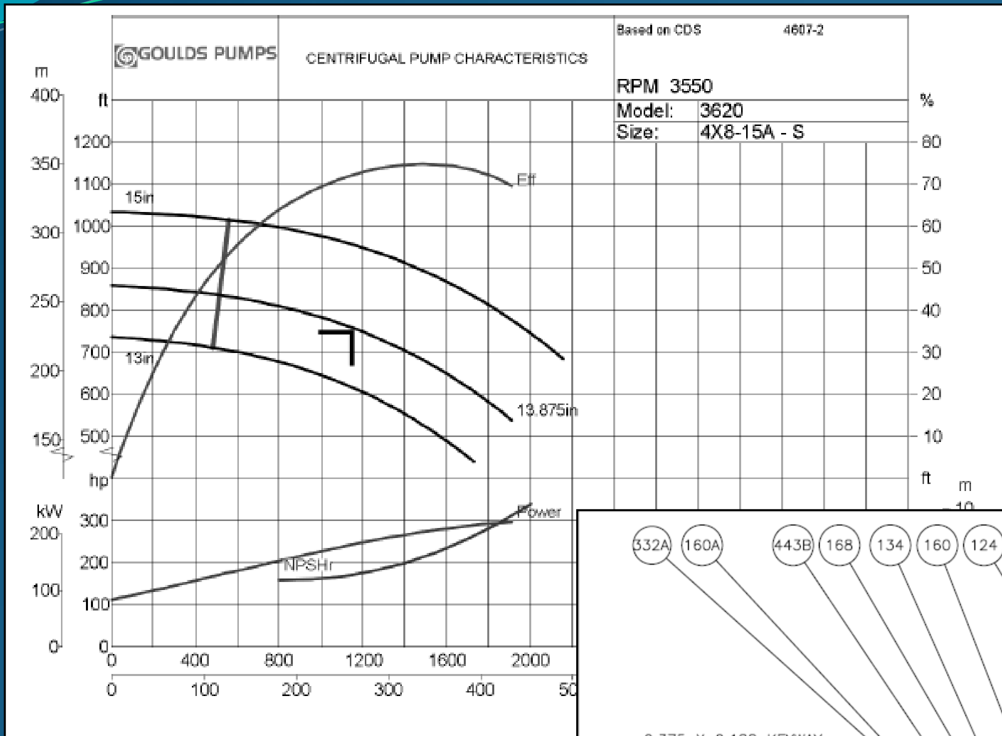
Gene Sabini
ITT Goulds – Seneca Falls, NY

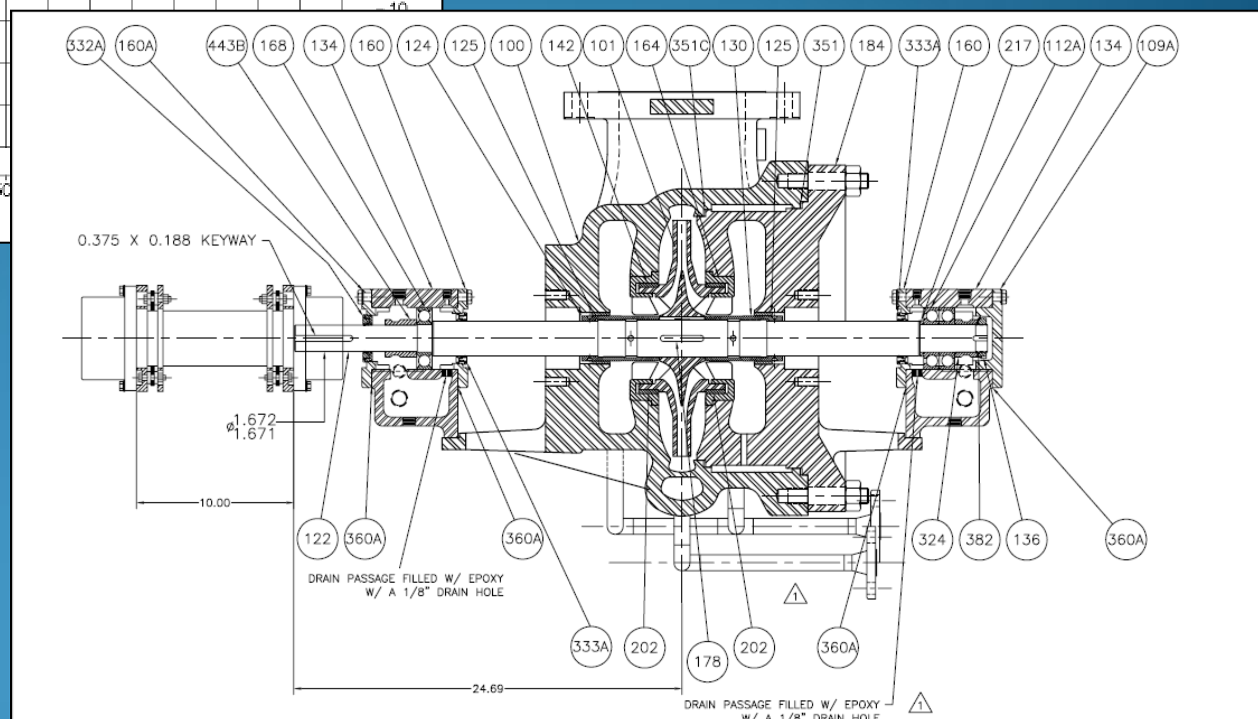
Kris Olasin
Motiva Enterprises – Convent, LA

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
- For testing purposed, 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
Based on CDS
4607-2
RPM 3550
Model: 3620
Size: 4X8-15A - S
15in
13in
13.875in
Eff
NPSHr
Power
m
ft
%
hp
kW

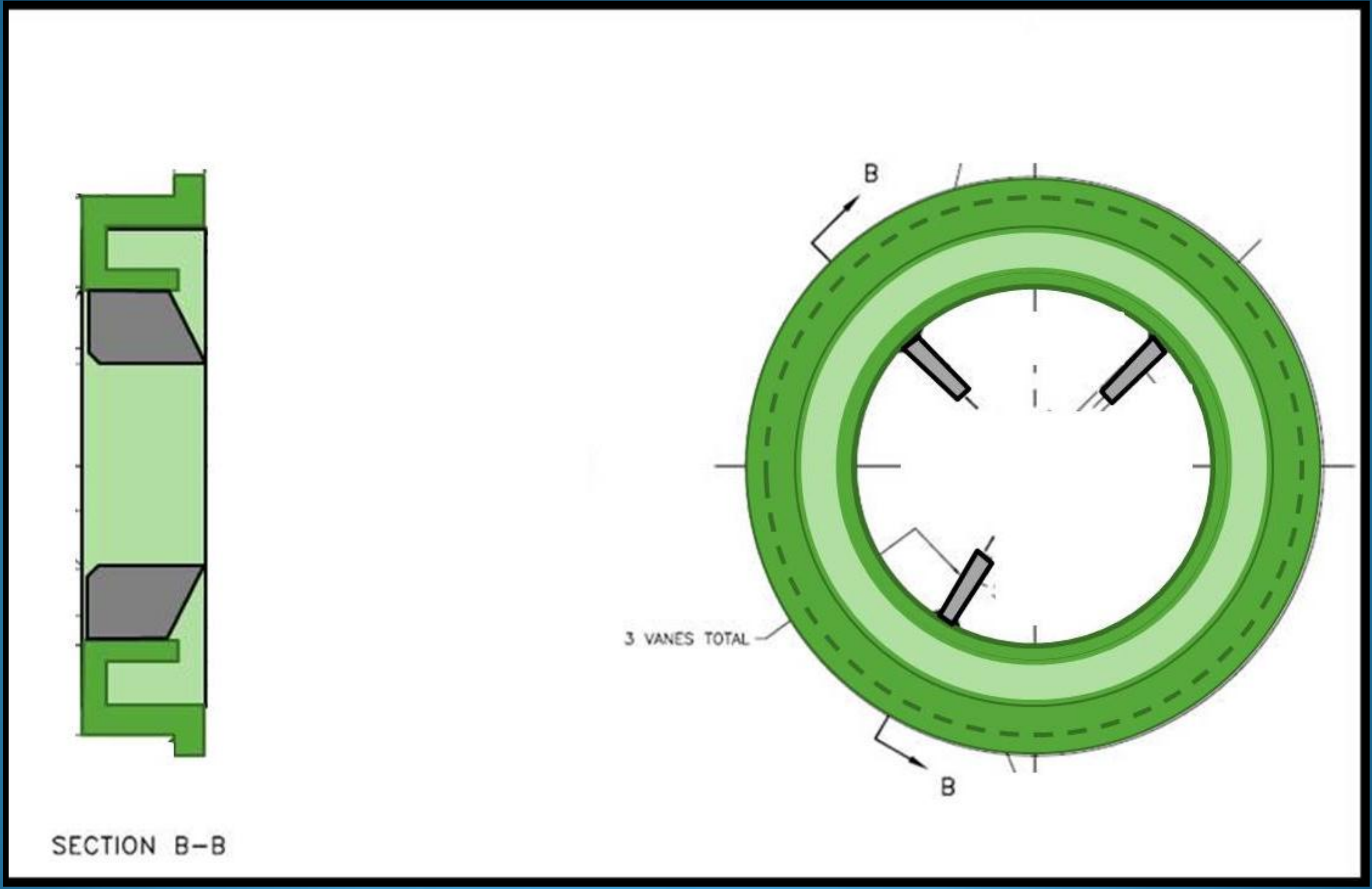

332A
160A
443B
168
134
160
124
125
100
142
101
164
351C
130
125
351
184
333A
160
217
112A
134
109A
0.375 X 0.188 KEYWAY
ø1.672
1.671
10.00
24.69
122
360A
360A
324
382
136
360A
DRAIN PASSAGE FILLED W/ EPOXY
W/ A 1/8" DRAIN HOLE
333A
202
178
202
360A
DRAIN PASSAGE FILLED W/ EPOXY
W/ A 1/8" DRAIN HOLE

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 10th 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 at 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

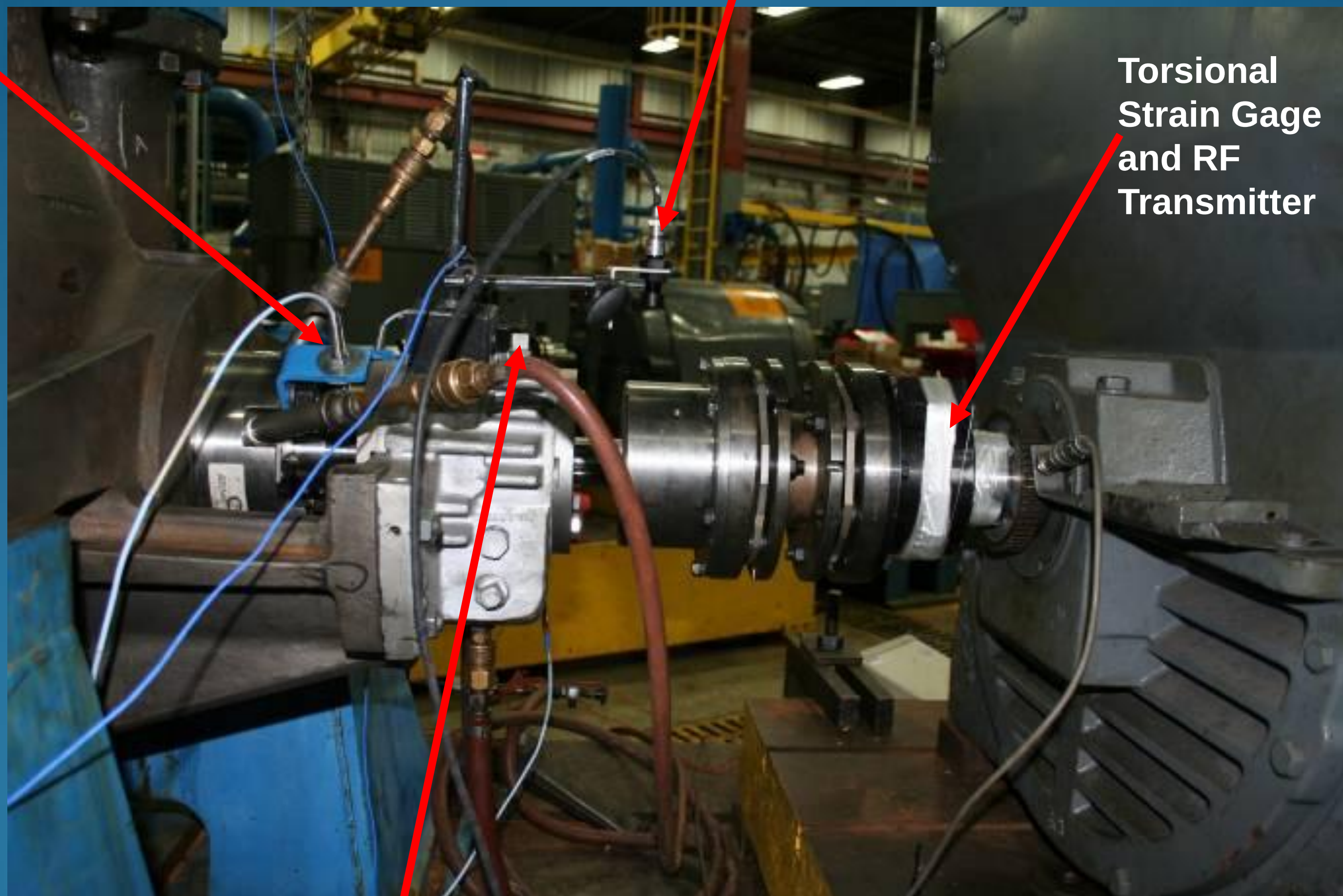
- Continuous Monitoring testing during transient and steady operation to monitor the shaft and bearing vibration amplitude, structural natural frequencies, pressure pulsations, torsional natural frequencies, etc.
- Operating Deflection Shape (ODS) testing during steady operation.
- Experimental Modal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

Vibration Monitoring

Optical Tachometer

Radial Proximity Probes

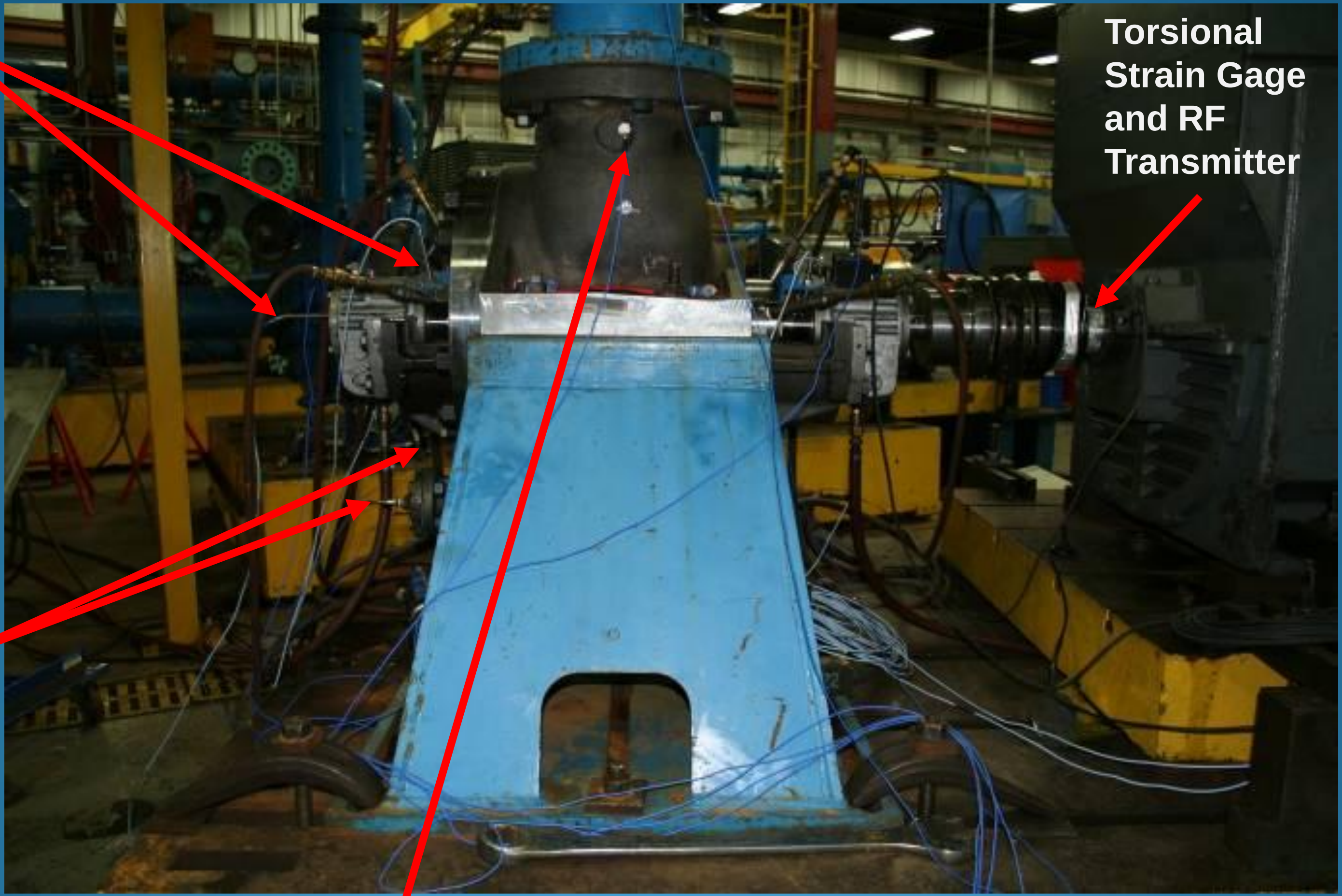
Torsional Strain Gage and RF Transmitter



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

Condition				Overall Vibration (in/s RMS)											
				Pump OBB			Pump IBB			Suct. Nozzle			Disch. Nozzle		
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

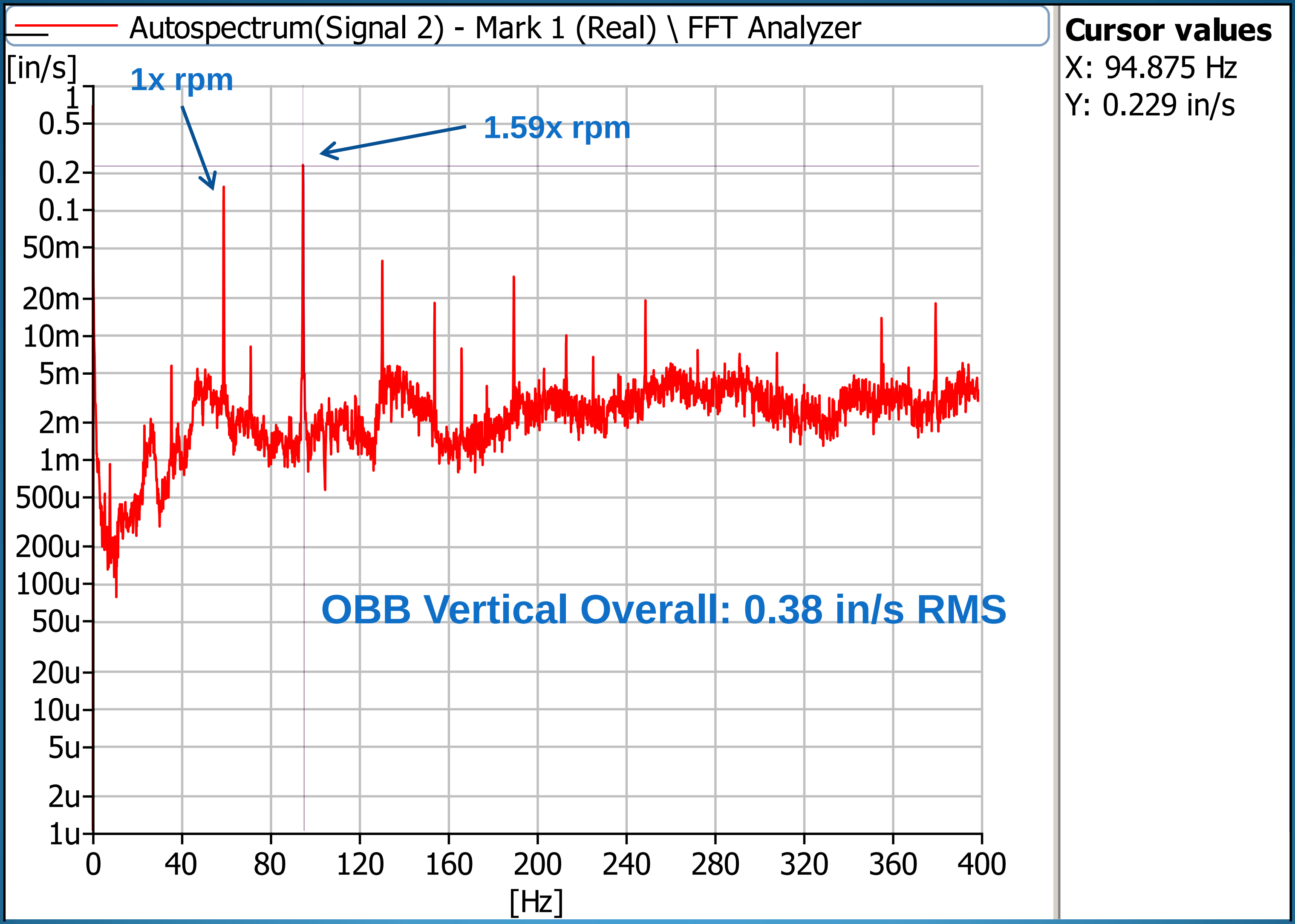
Worse vibration case for each speed condition

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

Continuous Monitoring Test

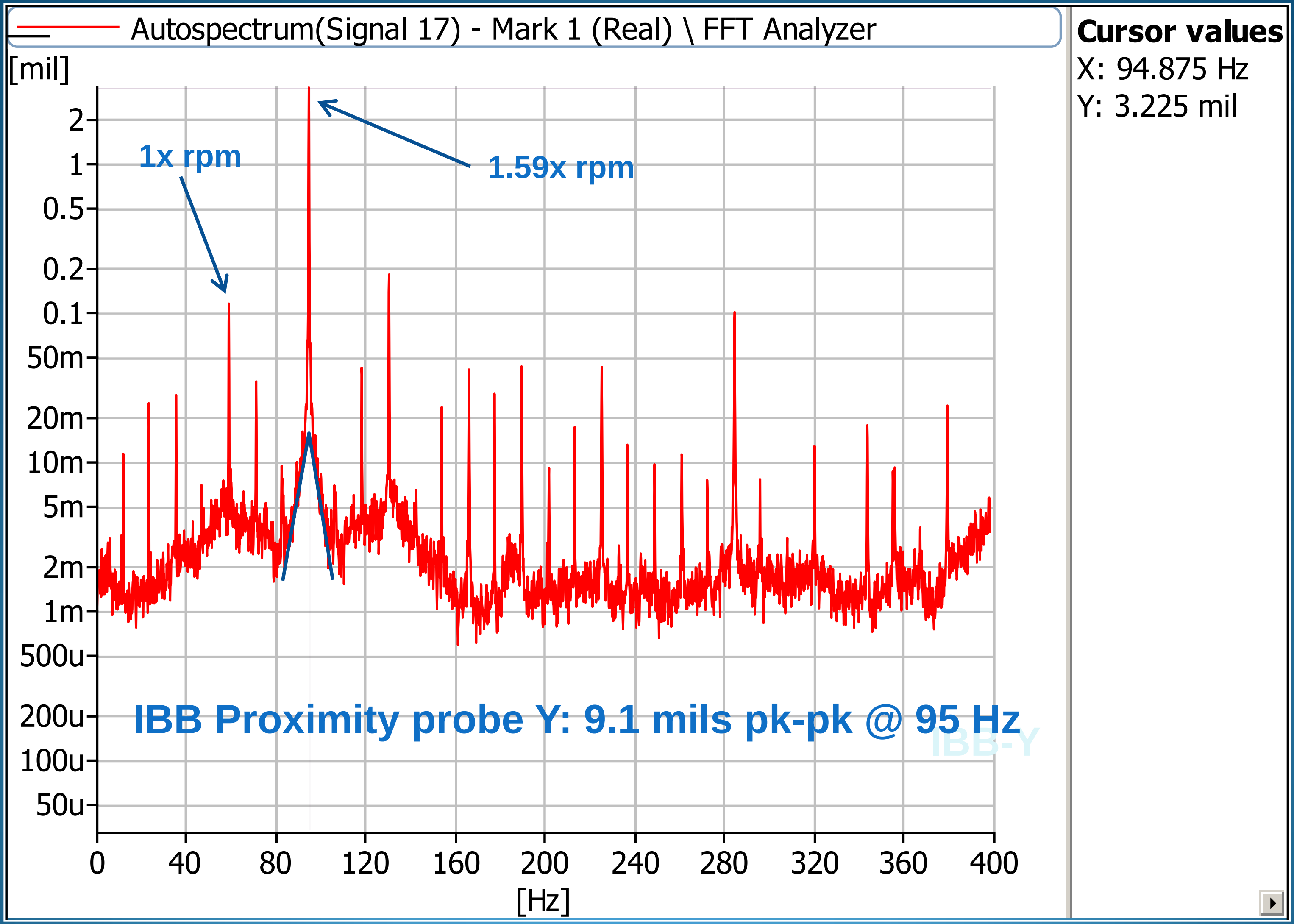
Running Speed 3570 rpm – 1163 GPM – 764 ft

Vibration Monitoring



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

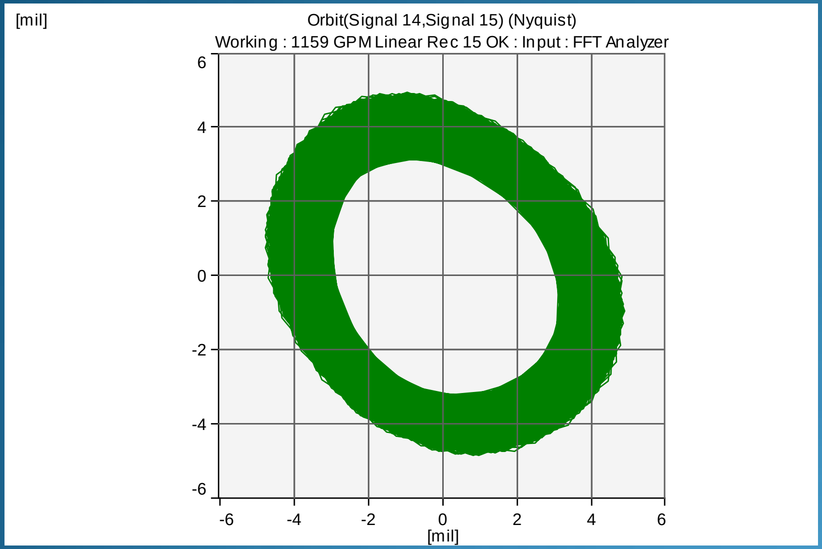
Vibration Monitoring



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

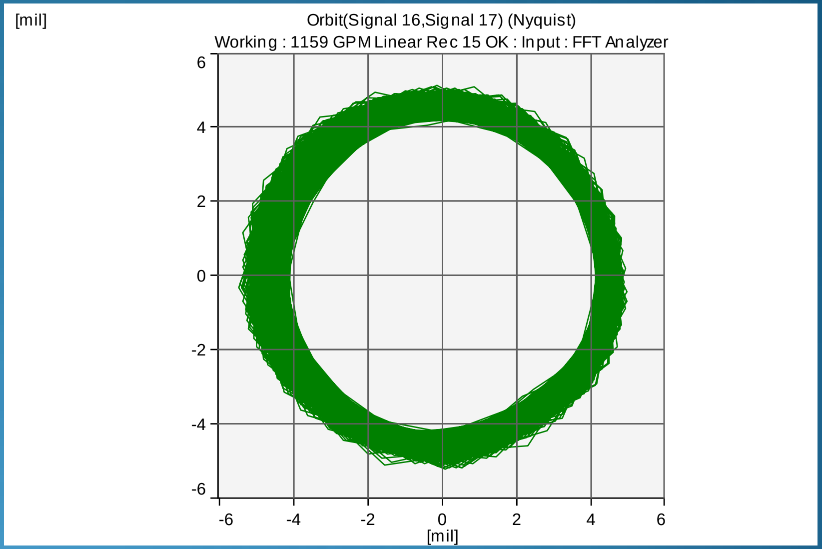
Vibration Monitoring

~9.0 mils pk-pk



OBB

~9.5 mils pk-pk



IBB

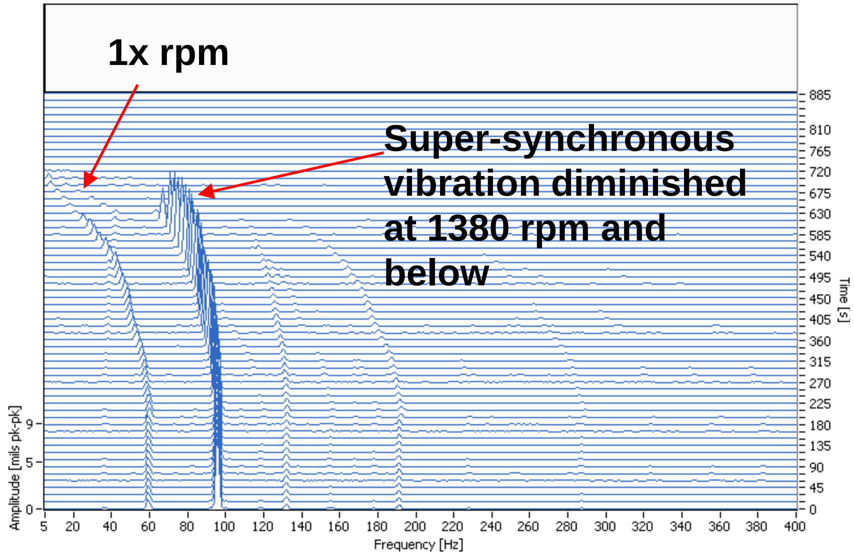
Radial Proximity Probes

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

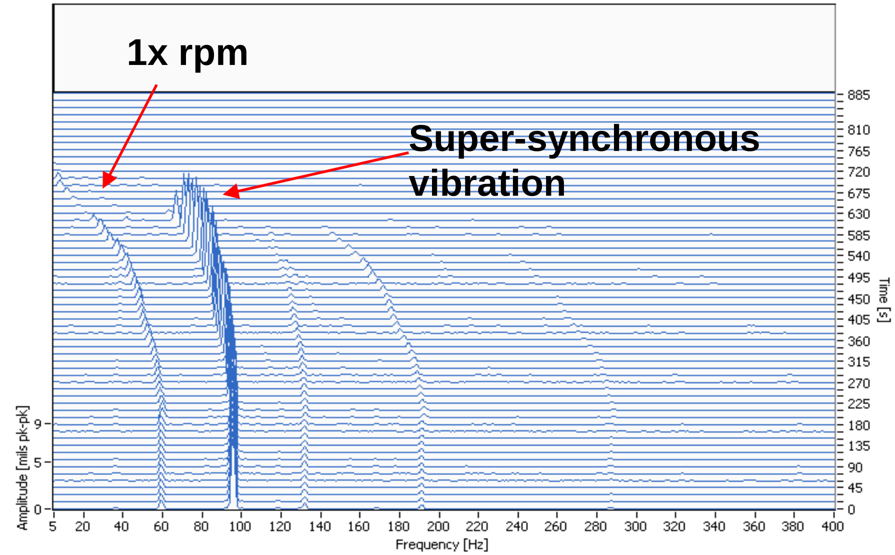
Transient Continuous Monitoring During Coast-Down from 3570 rpm to 0 rpm

Vibration Monitoring

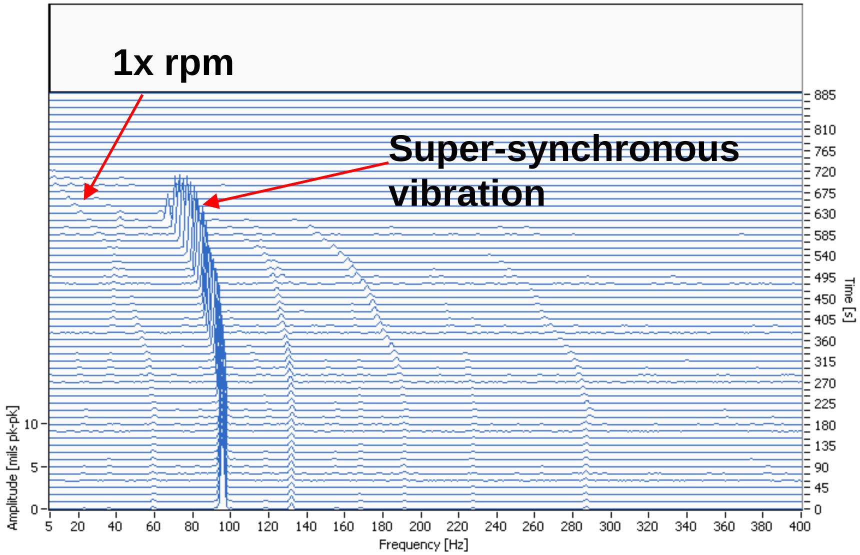
Pump OBB X



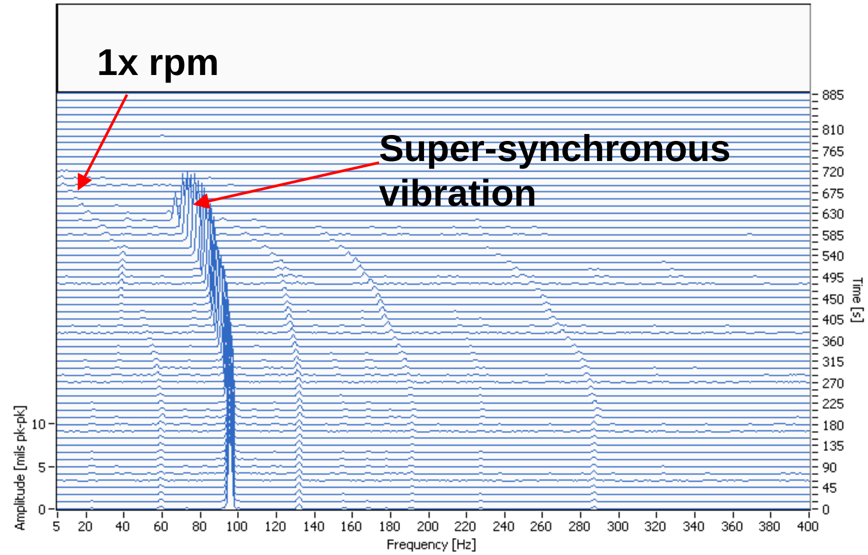
Pump OBB Y



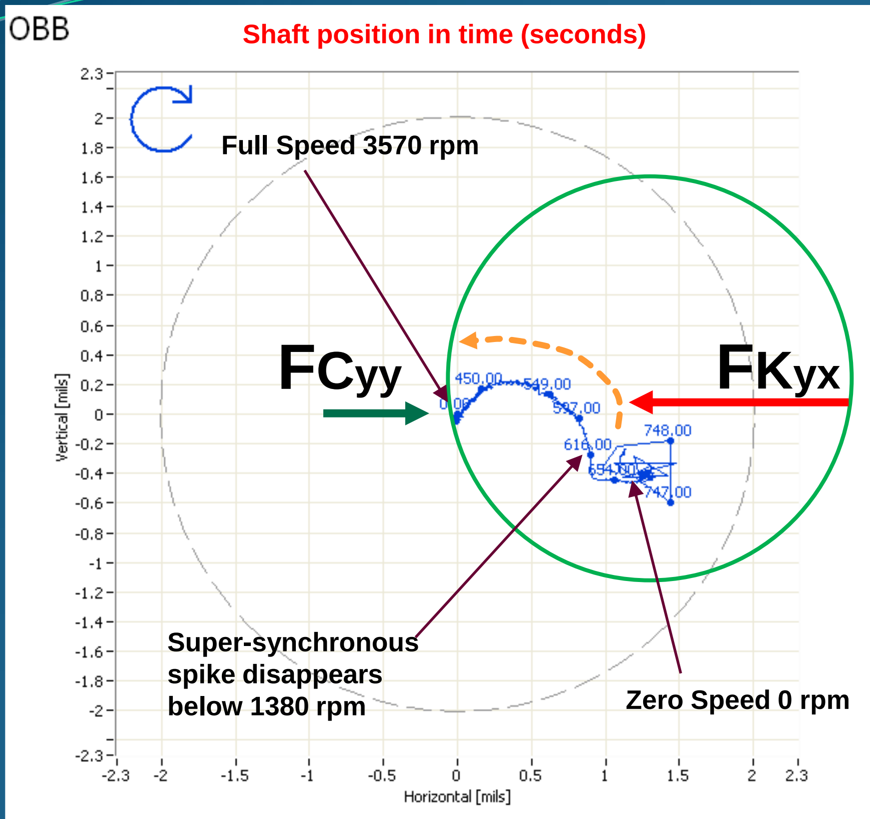
Pump IBB X

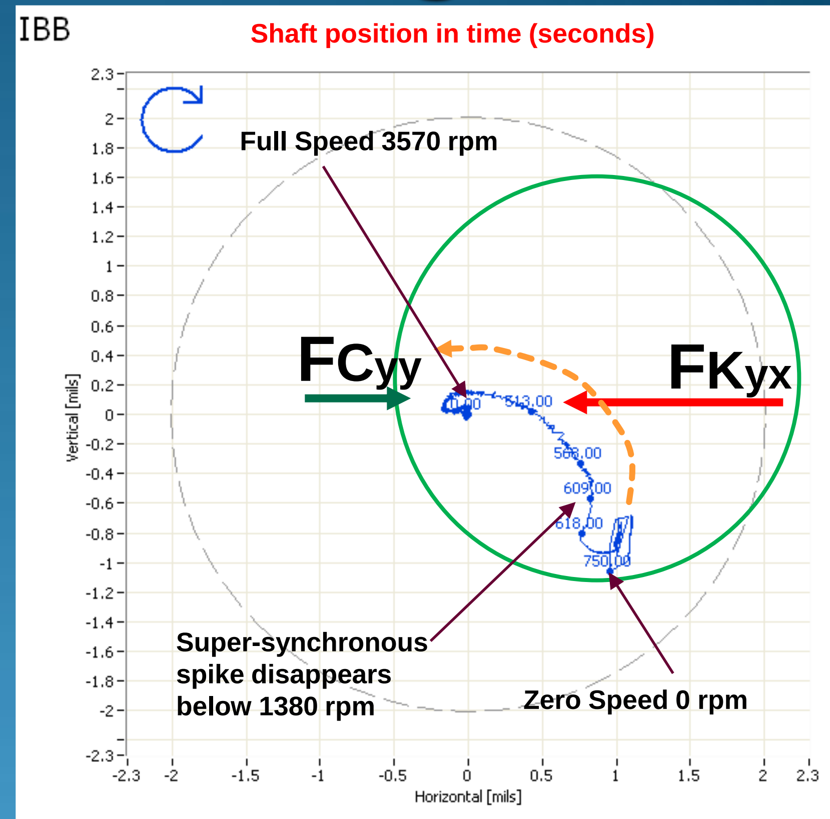


Pump IBB Y



Vibration Monitoring

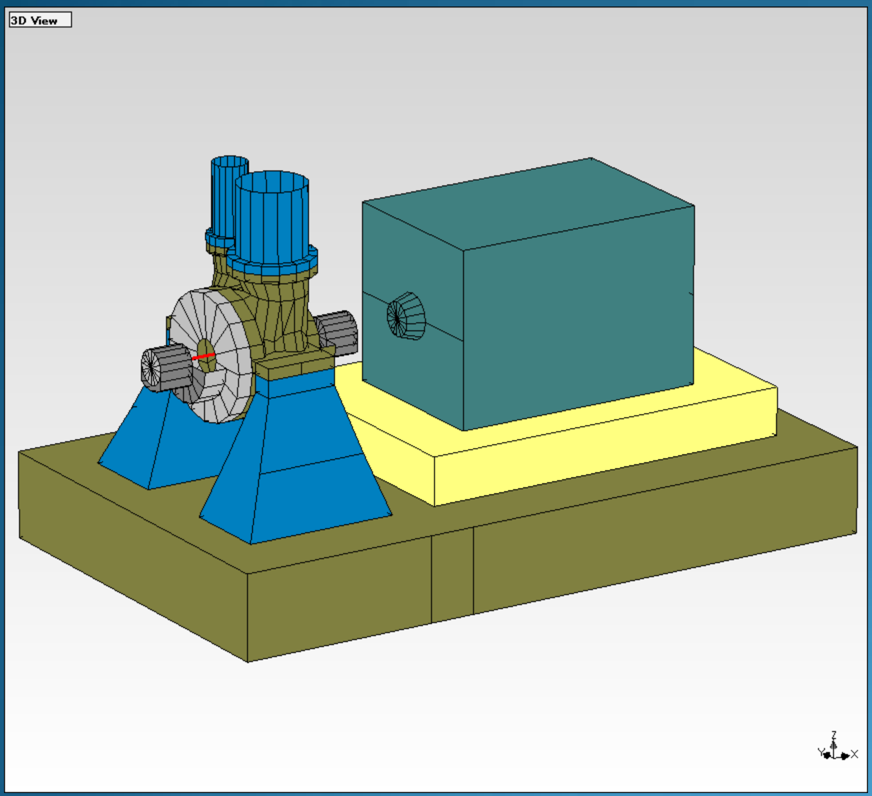




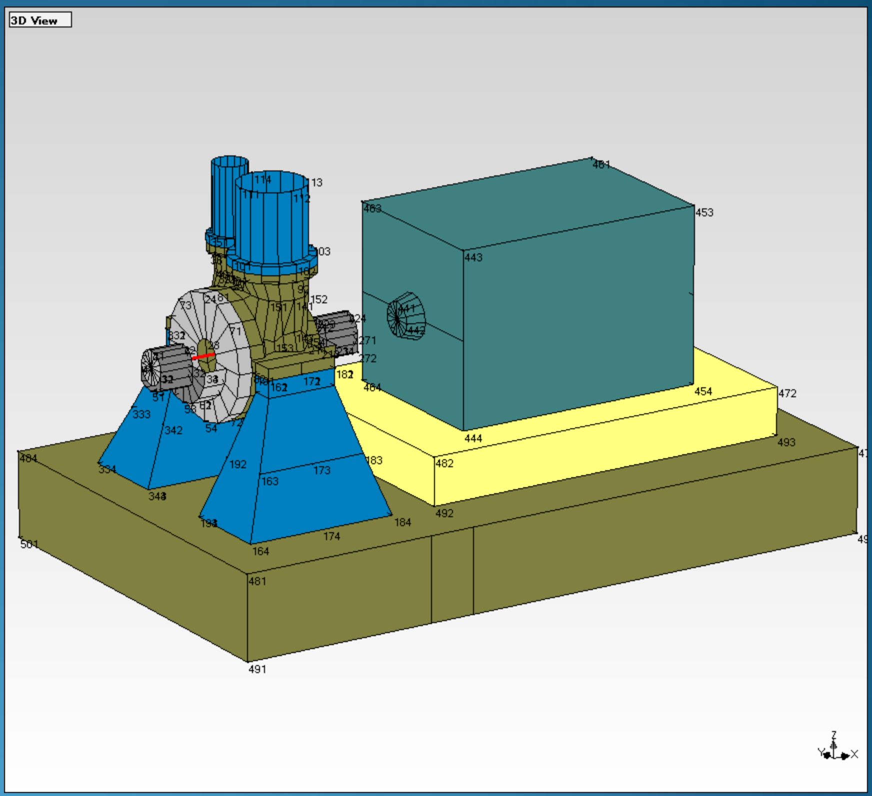
- Shaft center-line plots viewed from the NDE
- Shaft position in seconds during the coast-down from 3570 rpm (0 sec) to 0 rpm 750 sec.
- The shaft moves towards the upper left position after the pump starts (towards 9 O'clock position).
- Green circles represents the shaft centerline plots simulating 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

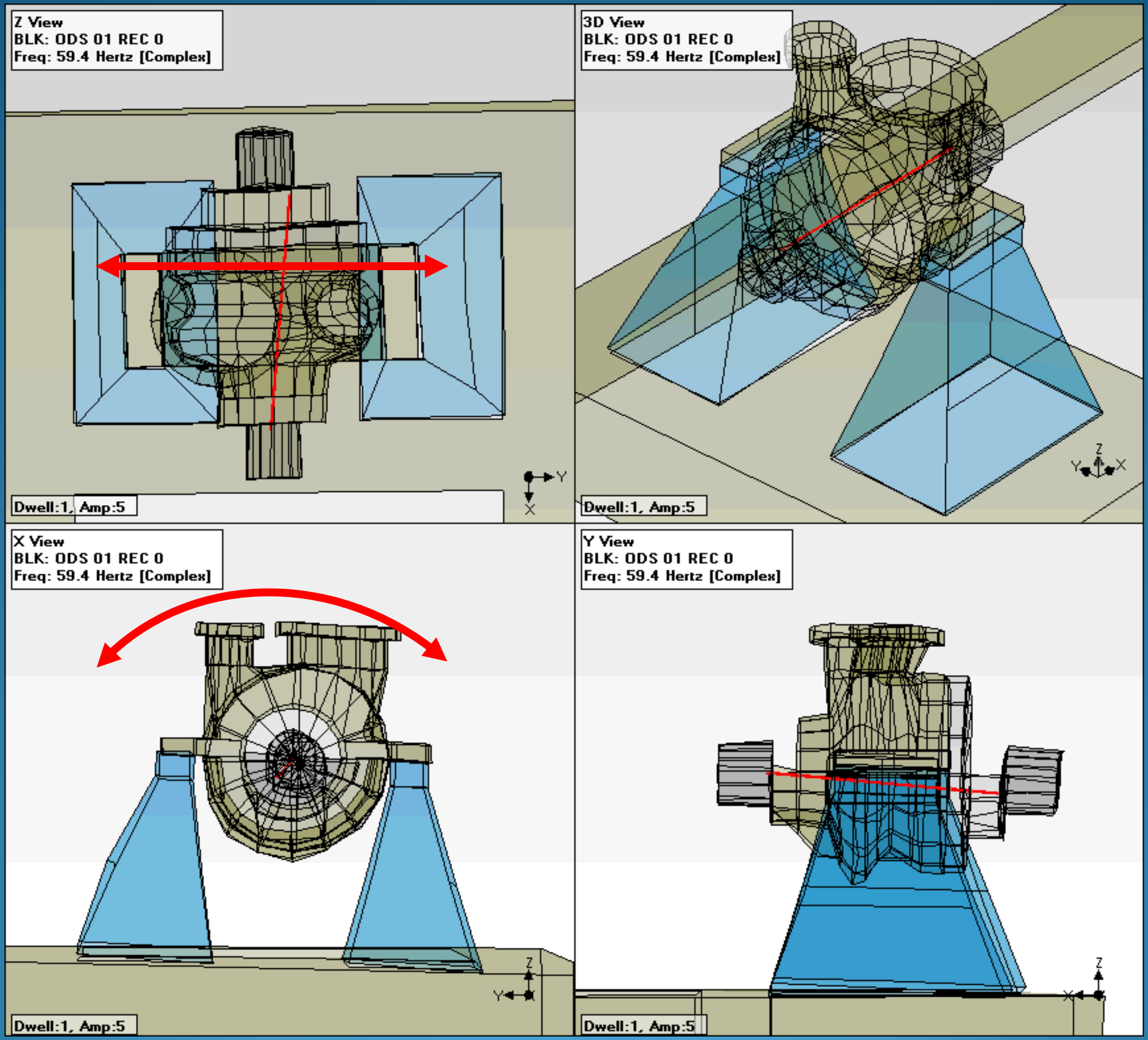


ODS Computer Model



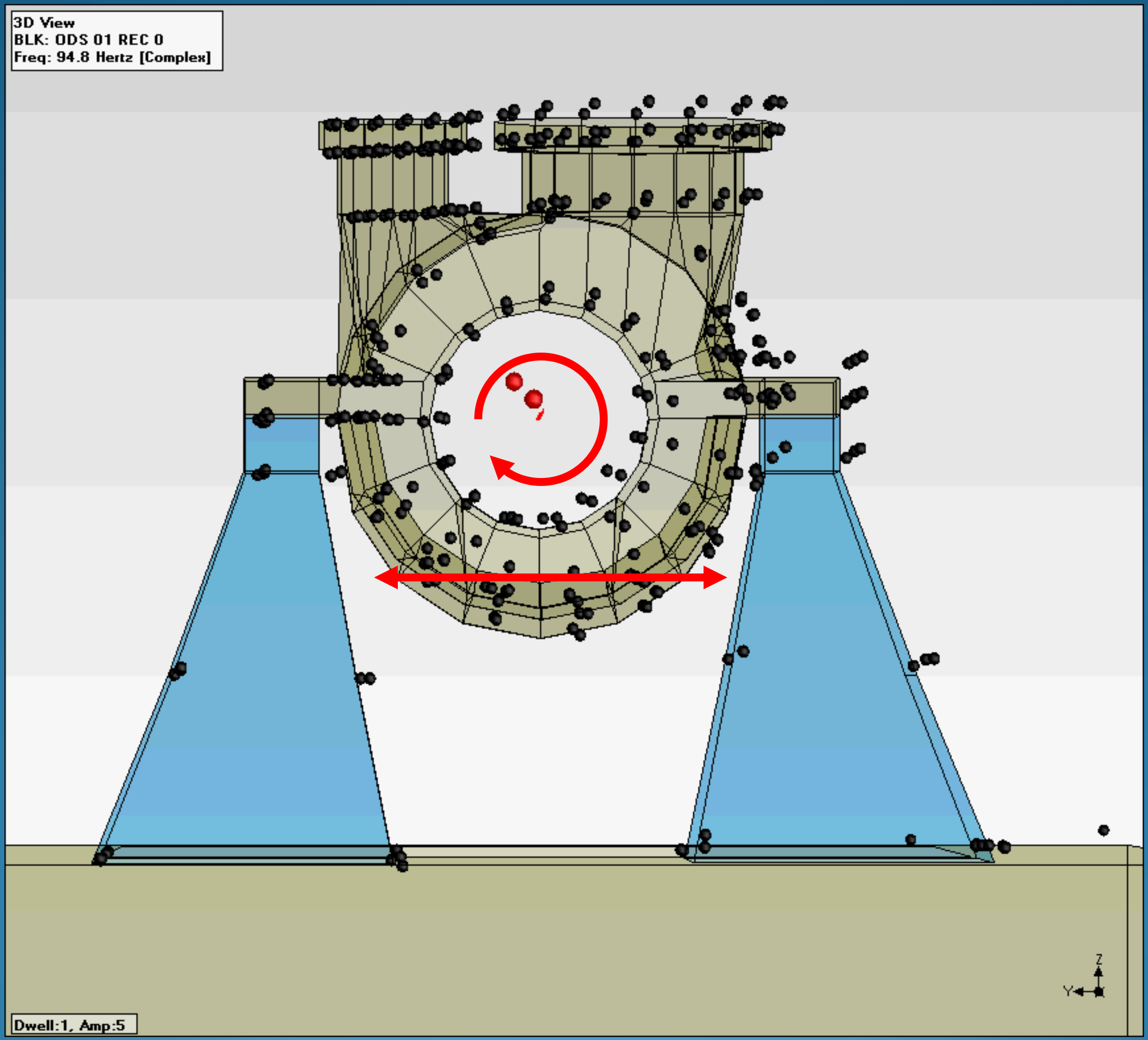
Over 600 vibration measurement/ directions

Operating Deflection Shape



ODS Animation @ 1x rpm (59.4 Hz)

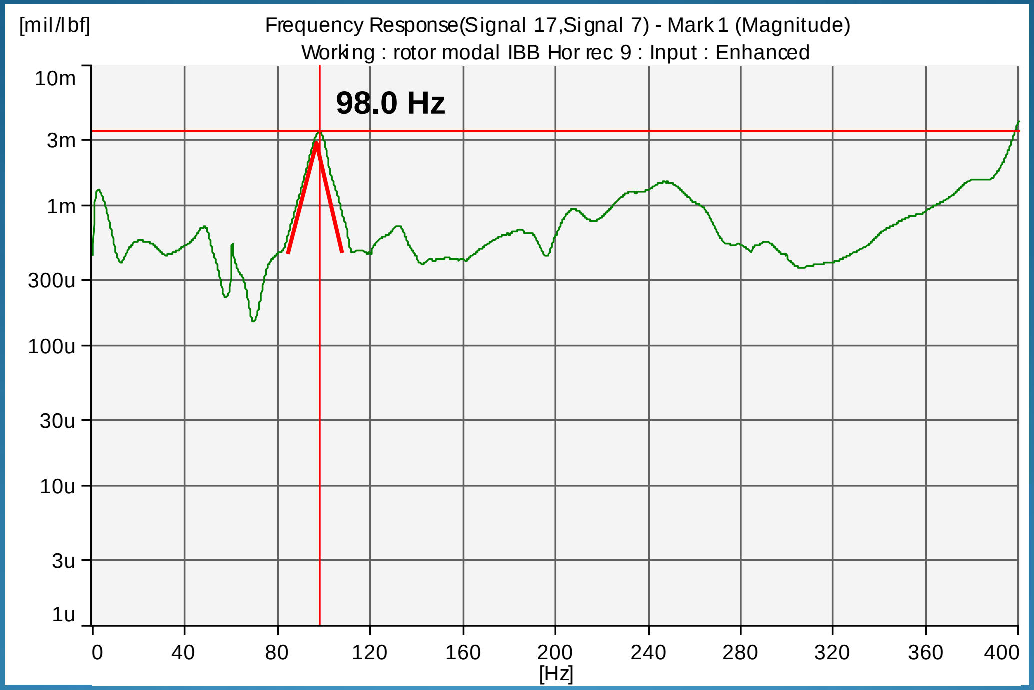
Operating Deflection Shape



ODS Animation @ 94.8 Hz

Experimental Modal Analysis (EMA) Testing
Frequency Response Test Results

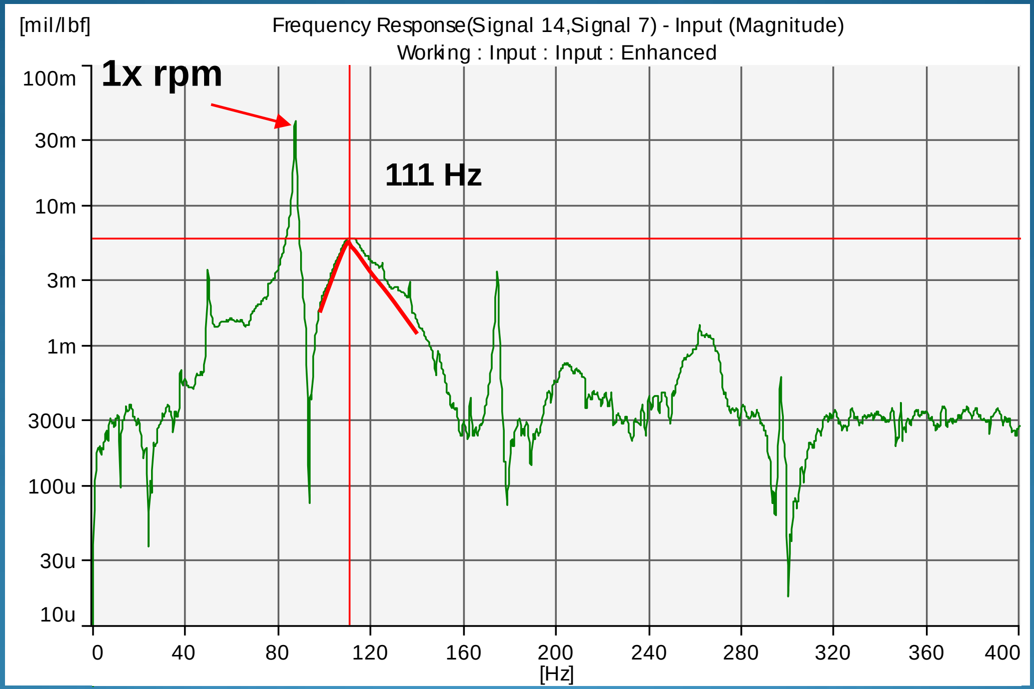
Experimental Modal Analysis



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

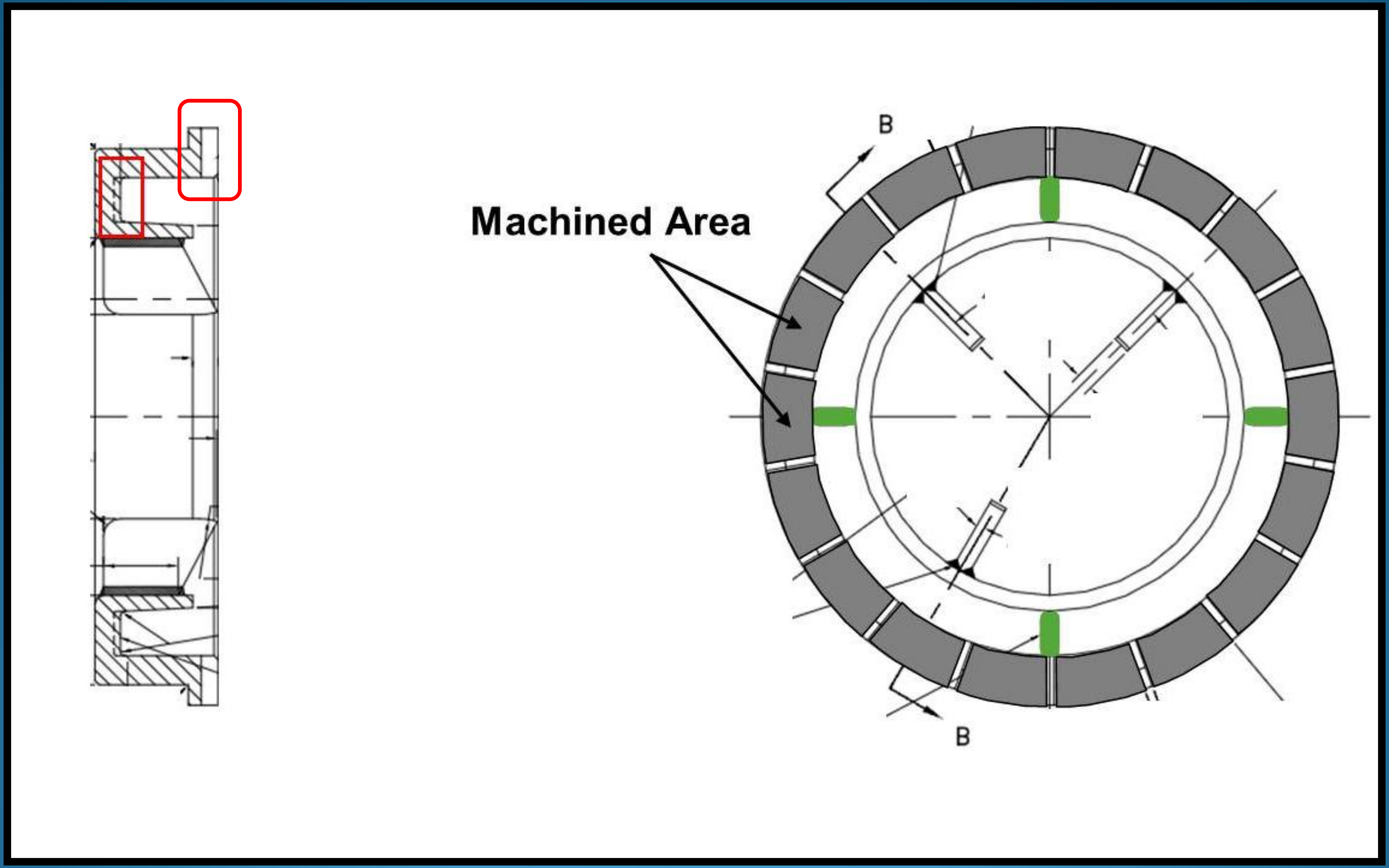
Preliminary Conclusions

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 (1.6x to 1.66x).
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

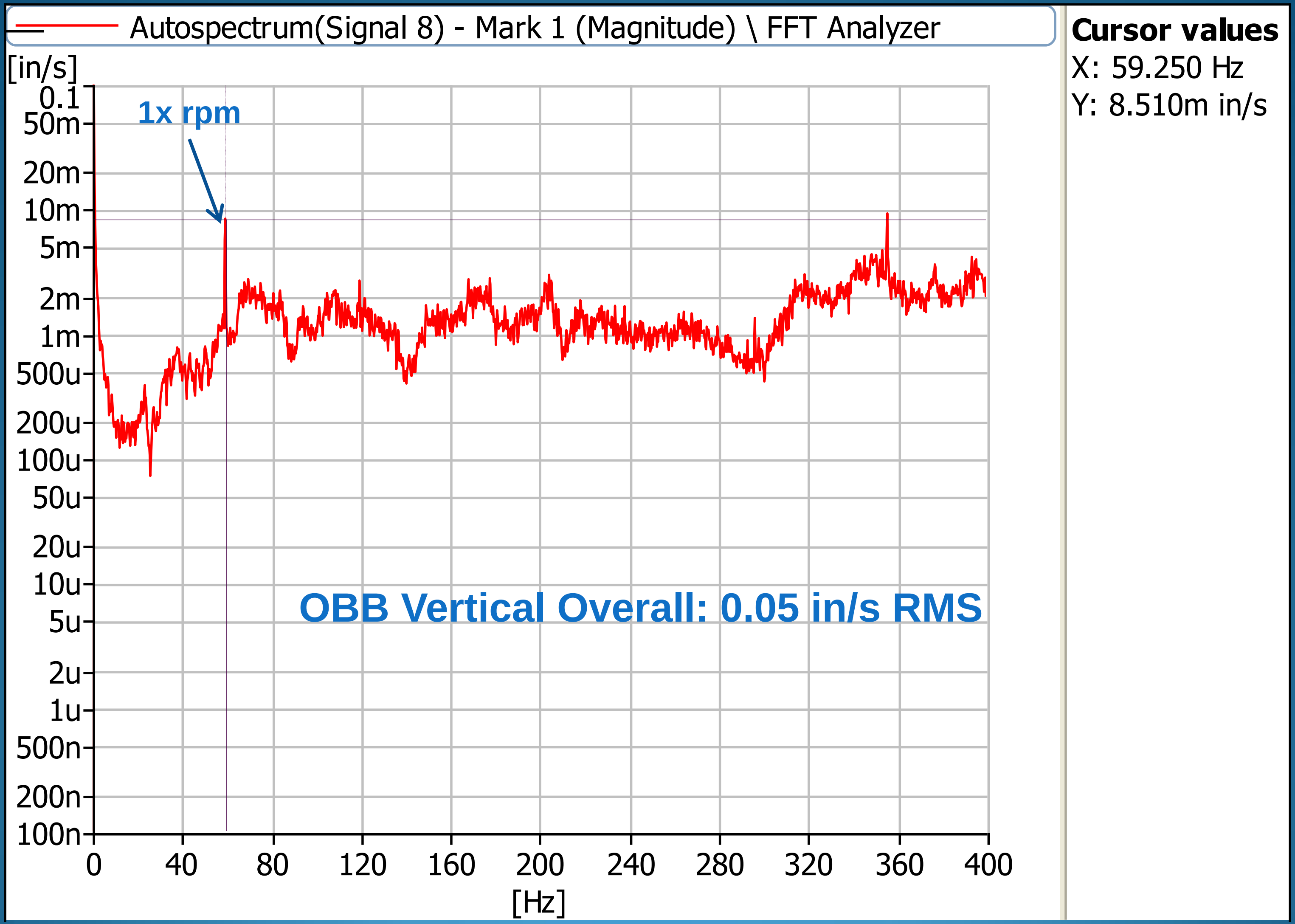
1. Based on a collaborative discussion with the OEM, one fix option for this instability was the addition of swirl-breaks by milling radial vanelets (slots) on the case wear ring. These 18 slots were equally spaced leaving vanelets of 1/8” of width and ¼” of axial depth.
2. The wrap-around wear ring 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.
3. Four slots were recommended to be machined in four clock positions at the bottom of the “pocket”.

Proposed Recommendations



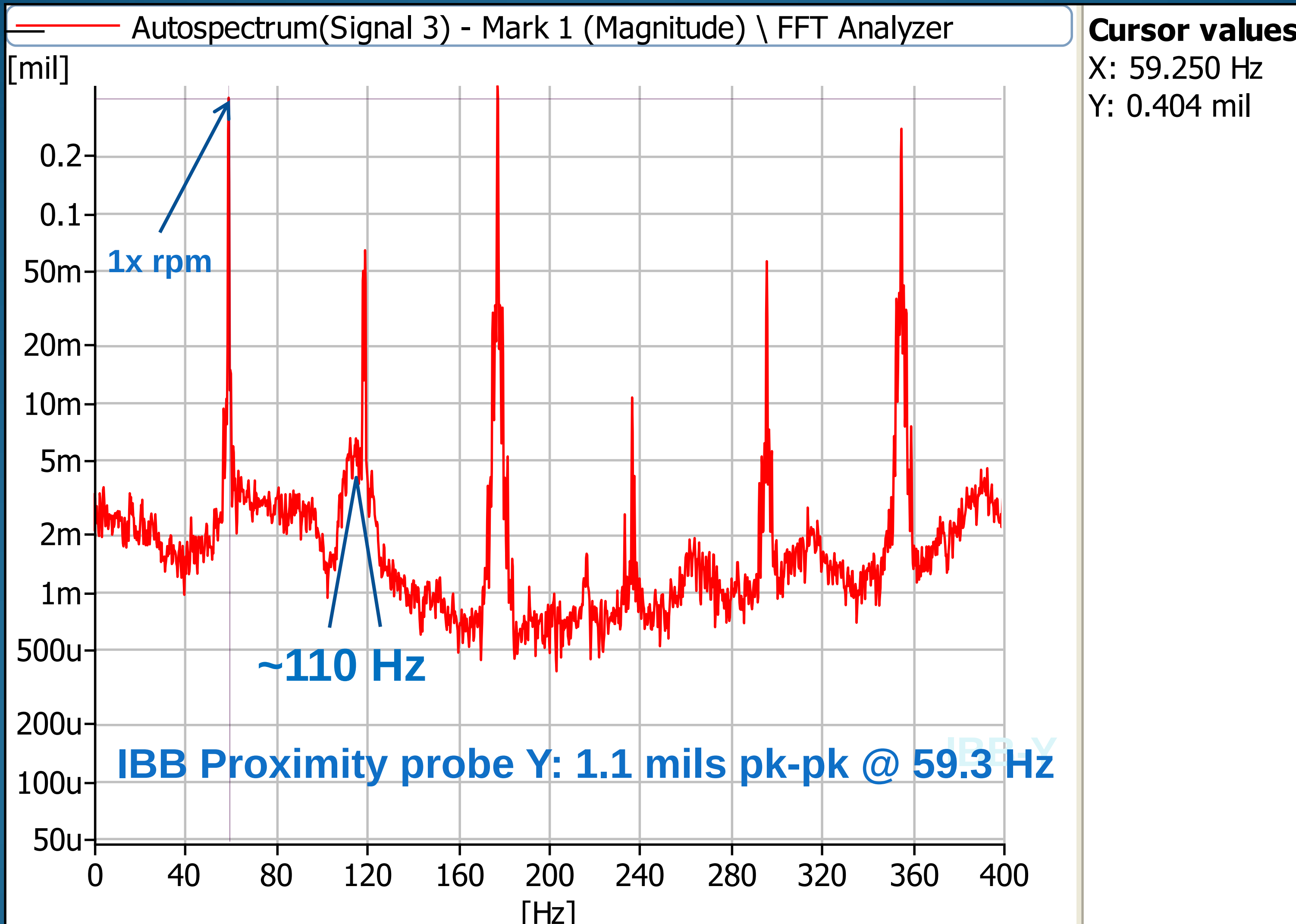
Follow-Up Testing

Vibration Monitoring



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

Vibration Monitoring

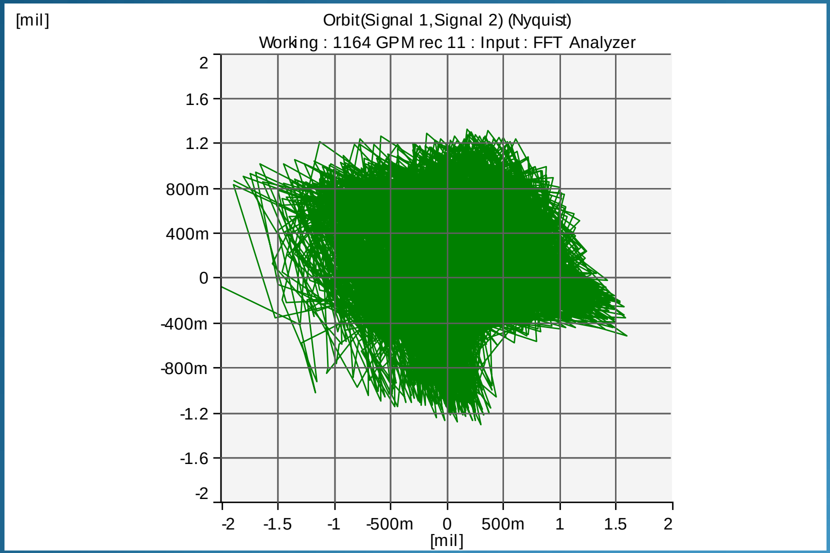


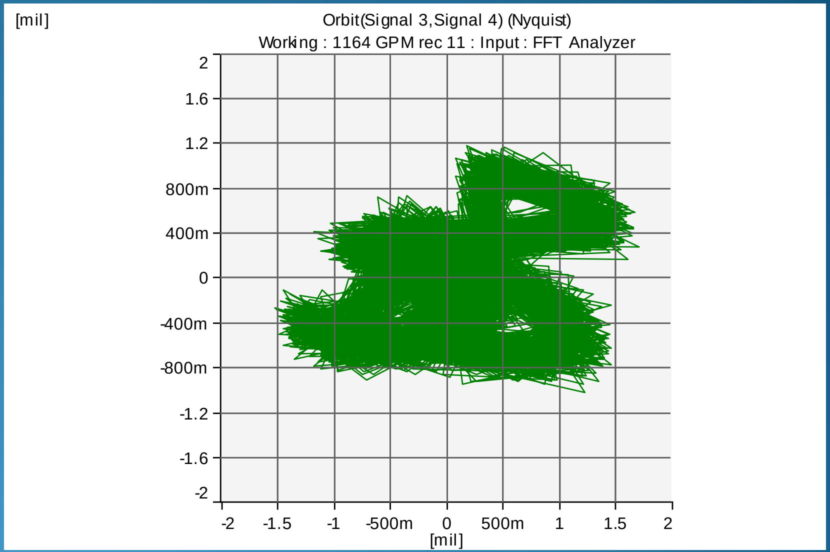
Running Speed 3555 rpm (59.3 Hz) – 1163 GPM – 741 ft

Vibration Monitoring

~2 mils pk-pk
Mostly Run-Out

~2.1 mils pk-pk
Mostly Run-Out





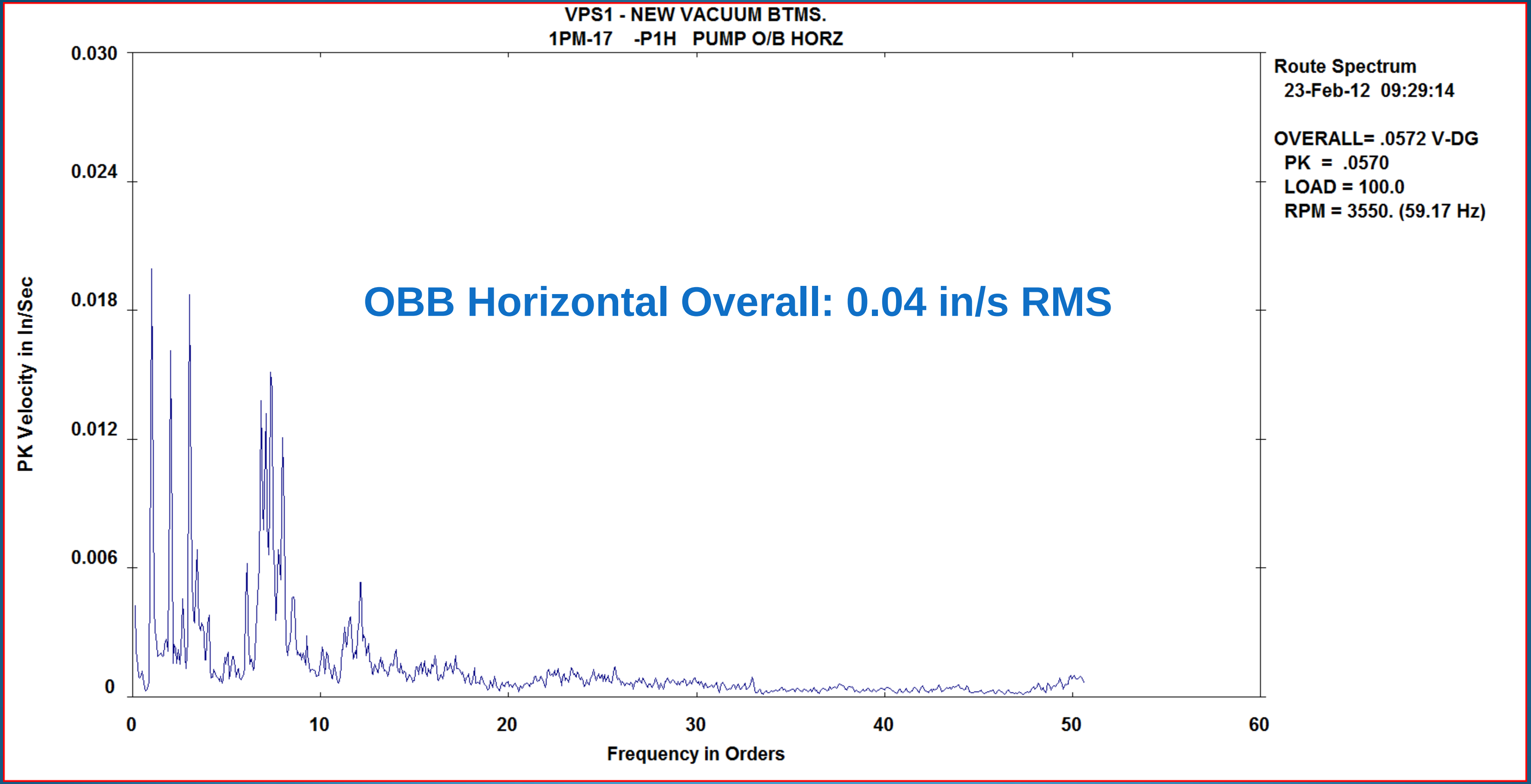
OBB

IBB

Radial Proximity Probes

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

Vibration Monitoring



Running Speed 3555 rpm (59.3 Hz)

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.

References

- Smith, D., Price, S., and Kunz, F., (1996), "Centrifugal Pump Vibration caused By Supersynchronous Shaft Instability," Proceedings, 13th Pump Users Symposium, pp. 47-60.
- Corley, J., (1978), "Subsynchronous Vibration in a Large Water Flood Pump," Proceedings, 7th Pump Users Symposium, pp. 103-110.

Thank you

Any Questions…?