

**EXPERIMENTAL INVESTIGATION OF CYCLIC
VIBRATION MORTON EFFECT IN THE BEARINGS OF
A DOUBLE OVERHUNG COMPRESSOR**

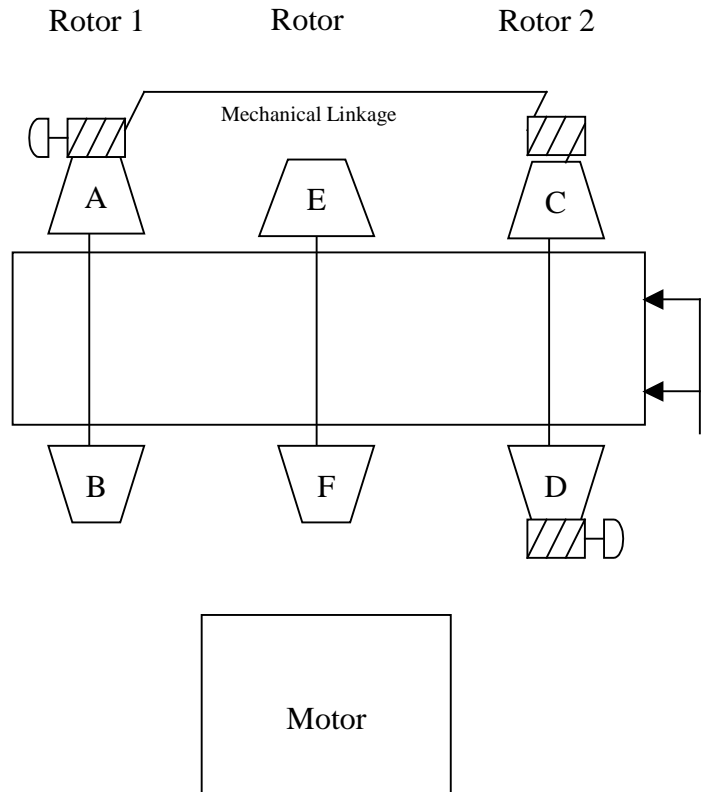
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Presentation Overview

- Installation and Compressor Description
- Operational History
- Field Testing Procedures
- Rotordynamic Test Results
- Design Changes
- Conclusion

Installation and Compressor Description



Service	Stage	Bearing	Rotor	Speed RPM
CP53 Nitrogen	1	A	1	29,150
	2	B		
	3	C	2	28,210
CP14 Dry Air	1	D	3	23,635
	2	E		
	3	F		

Service	Stage (Brg Pos)	Inlet Press Psia	Inlet Temp °F	Power KW
CP53 Nitrogen	3 Stage (C)	330	95	660
CP14 Dry Air	1 Stage (D)	280	95	1200

Overhead View of Integrally Geared Compressor



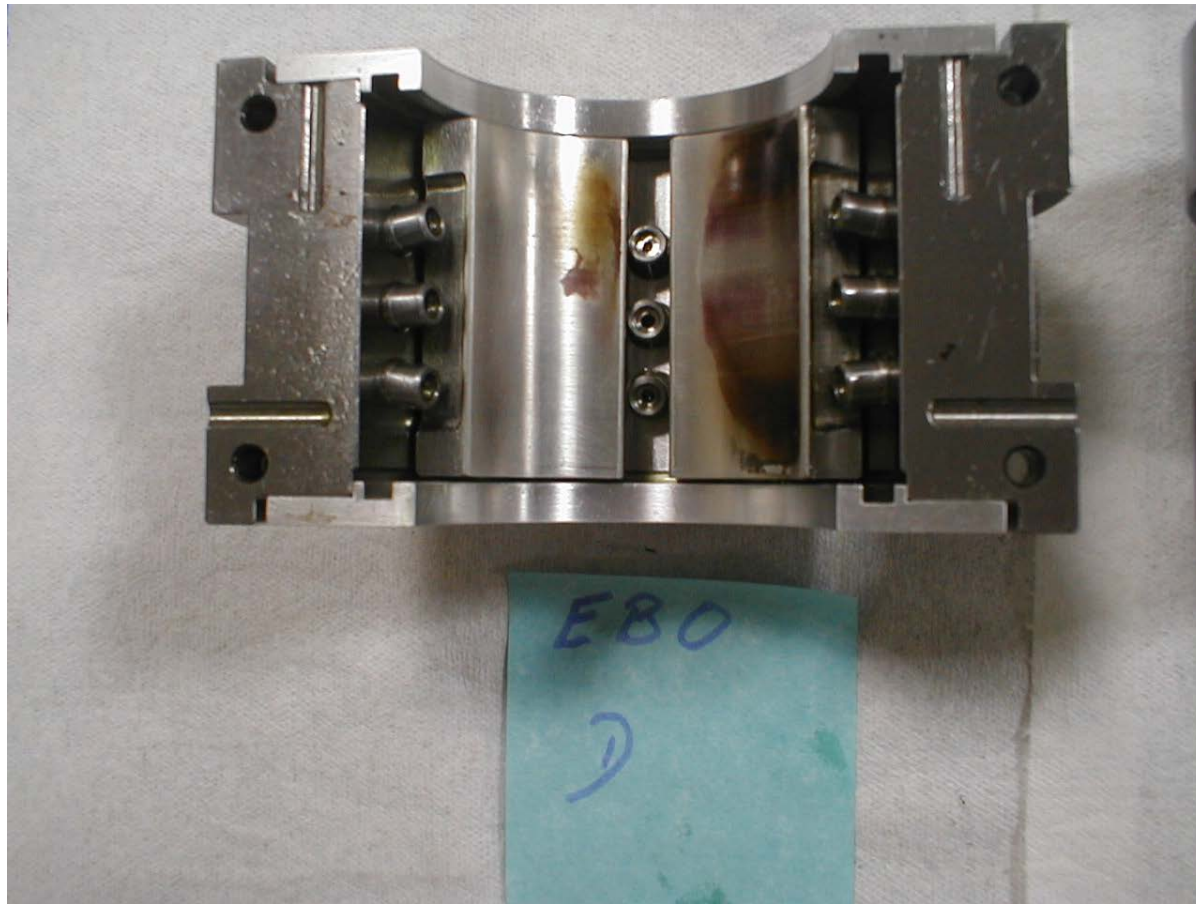
Side View of Gear Box



Operational History

- 1980 Initial Commissioning
- Year 2000- Sensitivity of Trips to Ambient Temperature
- July, 2003 – Field Testing

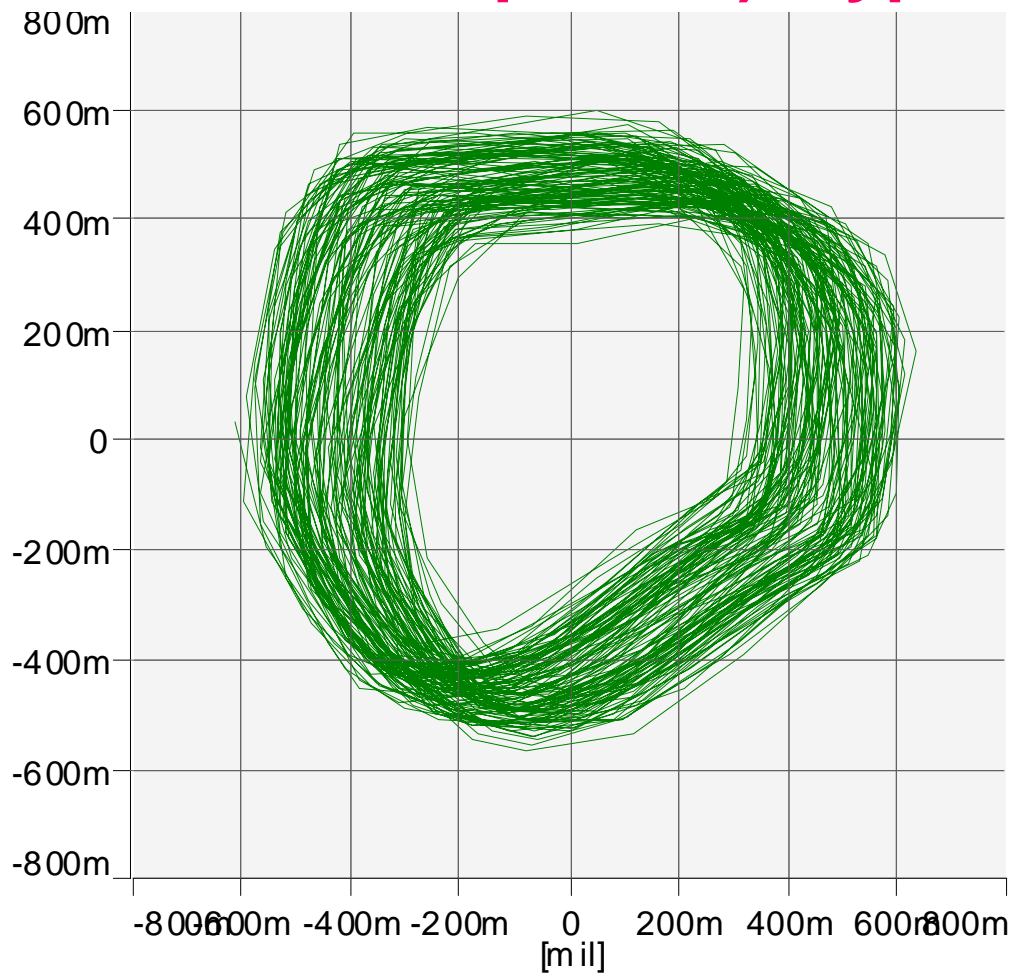
Bearing Style (from a Similar Application)



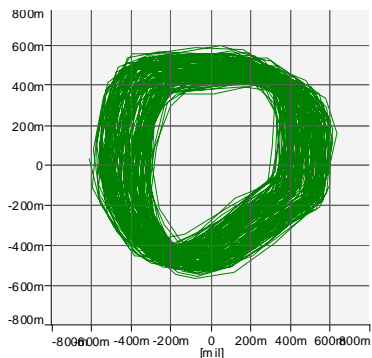
Field Testing Procedures

- Basic Instrumentation - Proximity Probes, Accelerometers, Dynamic Pressure
- Process Variation Tests
 - IGV variation
 - Nitrogen stage depressurization
 - Unloading air stages
- Oil Temperature Variation Tests

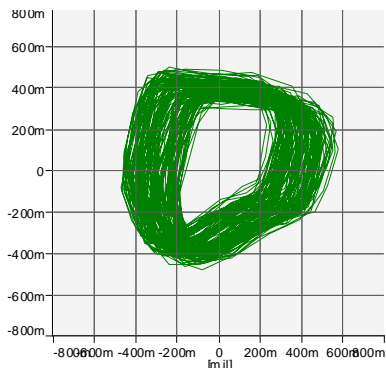
Left Bearing Orbit (Near Peak Amplitude), Typical



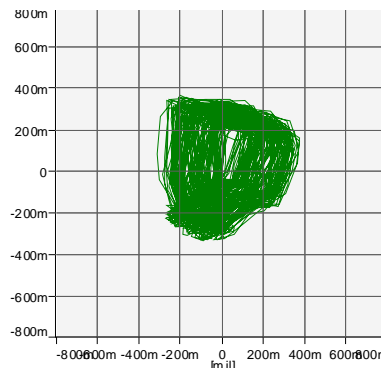
Left Bearing Orbit at Various Moments into Test



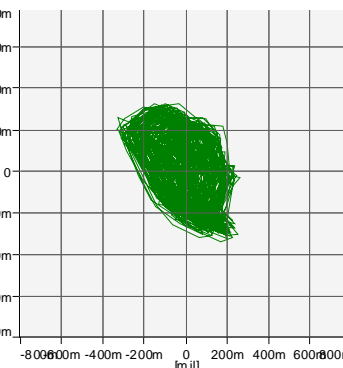
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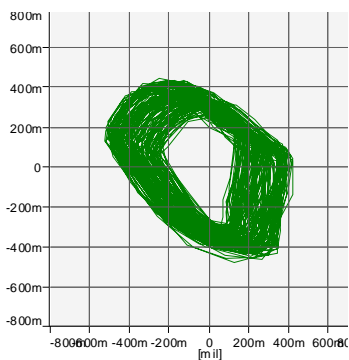
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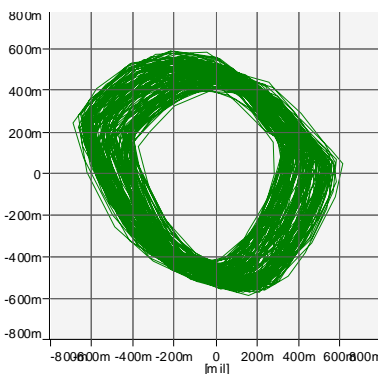
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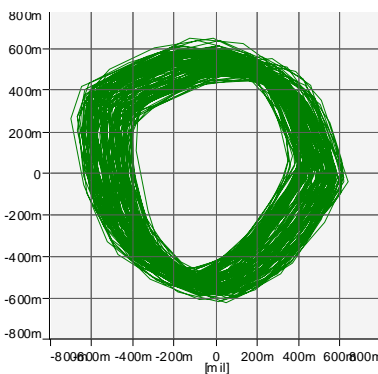
135 sec



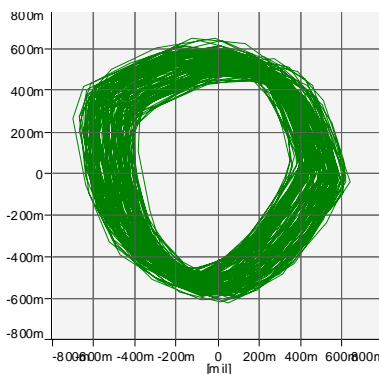
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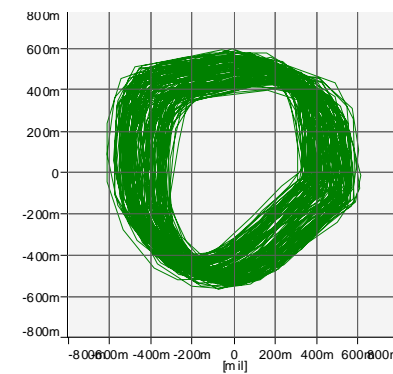
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270 sec



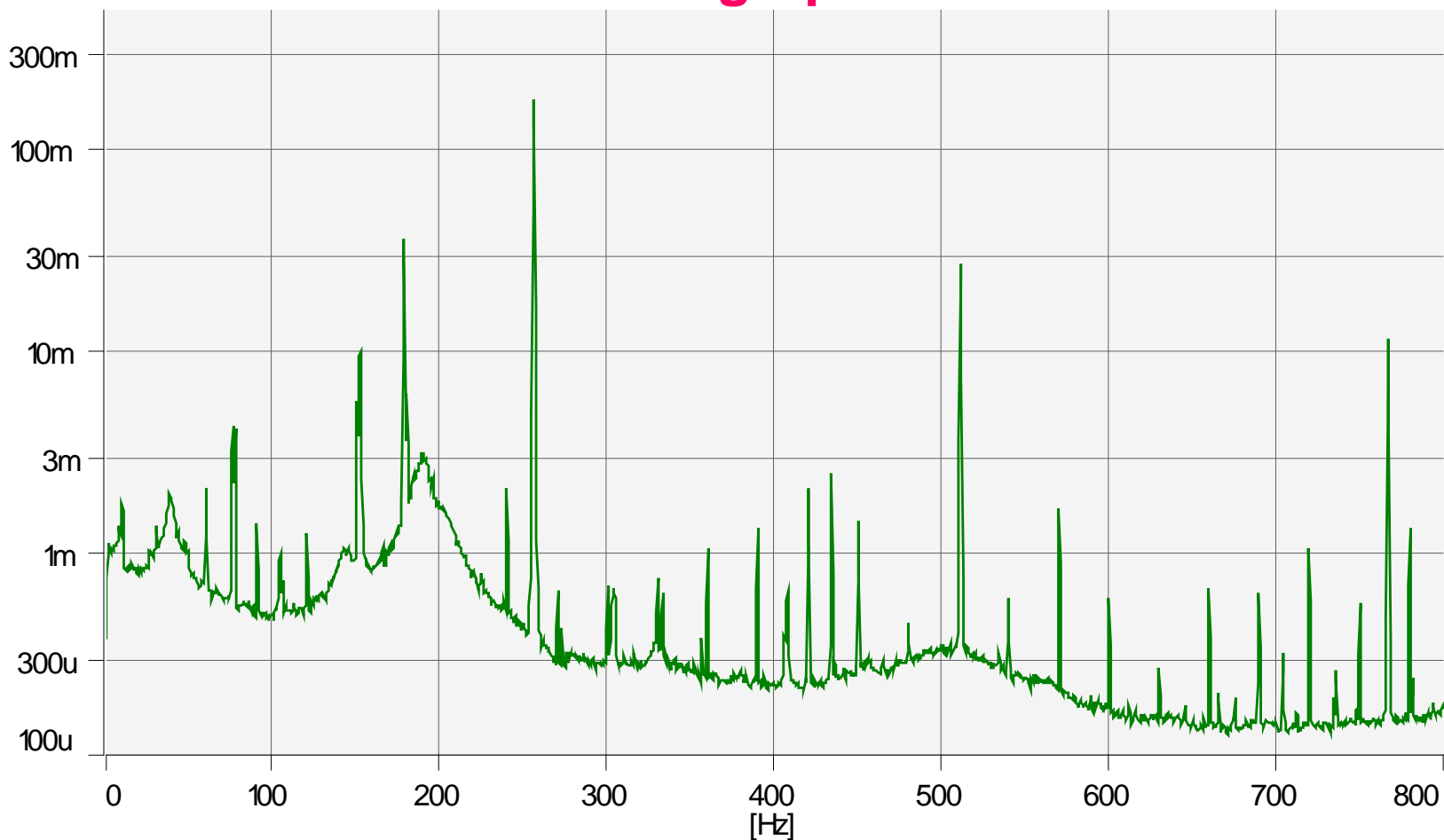
315 sec



360 sec

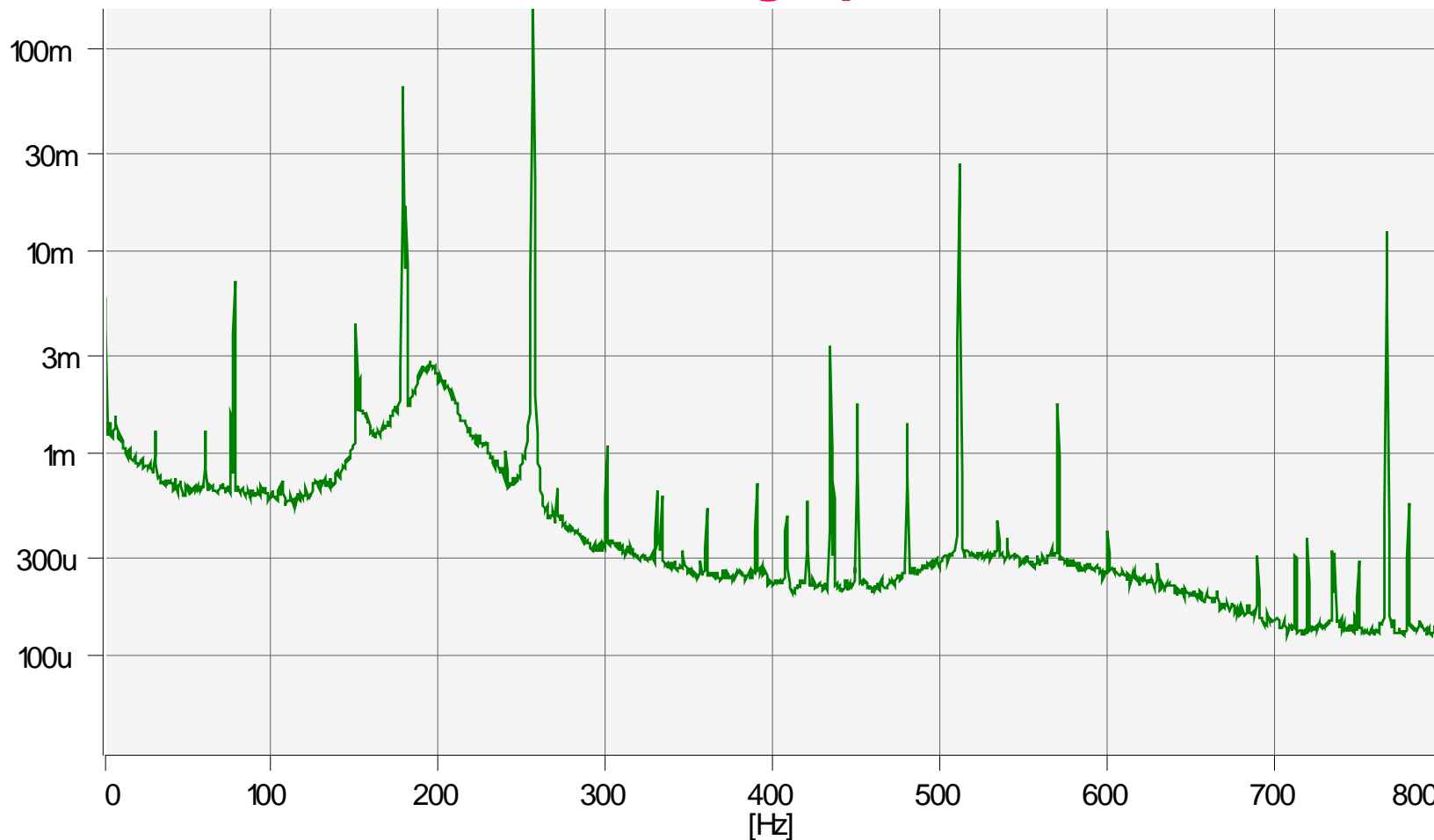
Vibration Spectrum of Left Bearing Proximity Probe (100 Linear Averages)

NOTE: Running Speed is 255 Hz.

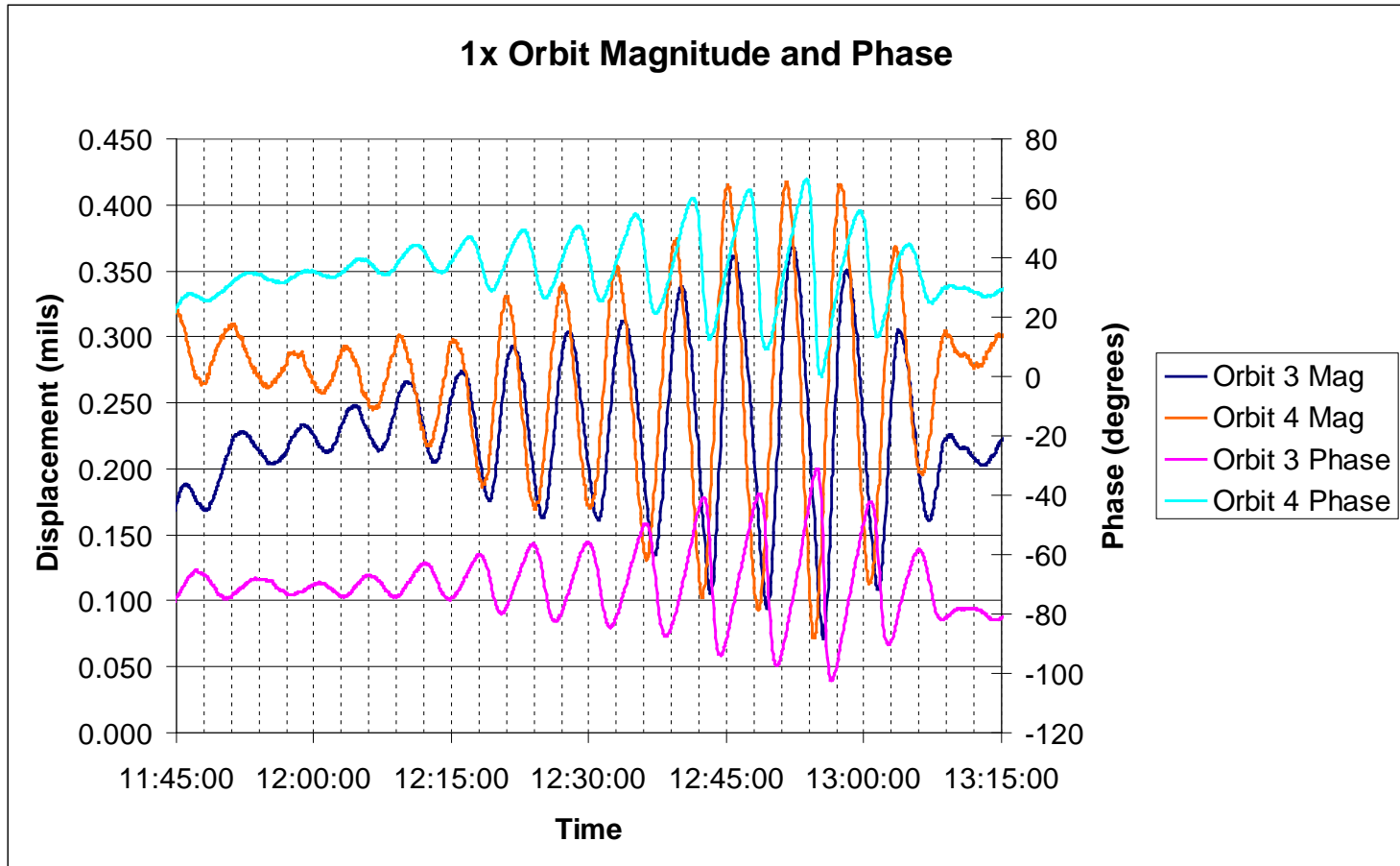


Vibration Spectrum of Right Bearing Proximity Probe (100 Linear Averages)

NOTE: Running Speed is 255Hz.



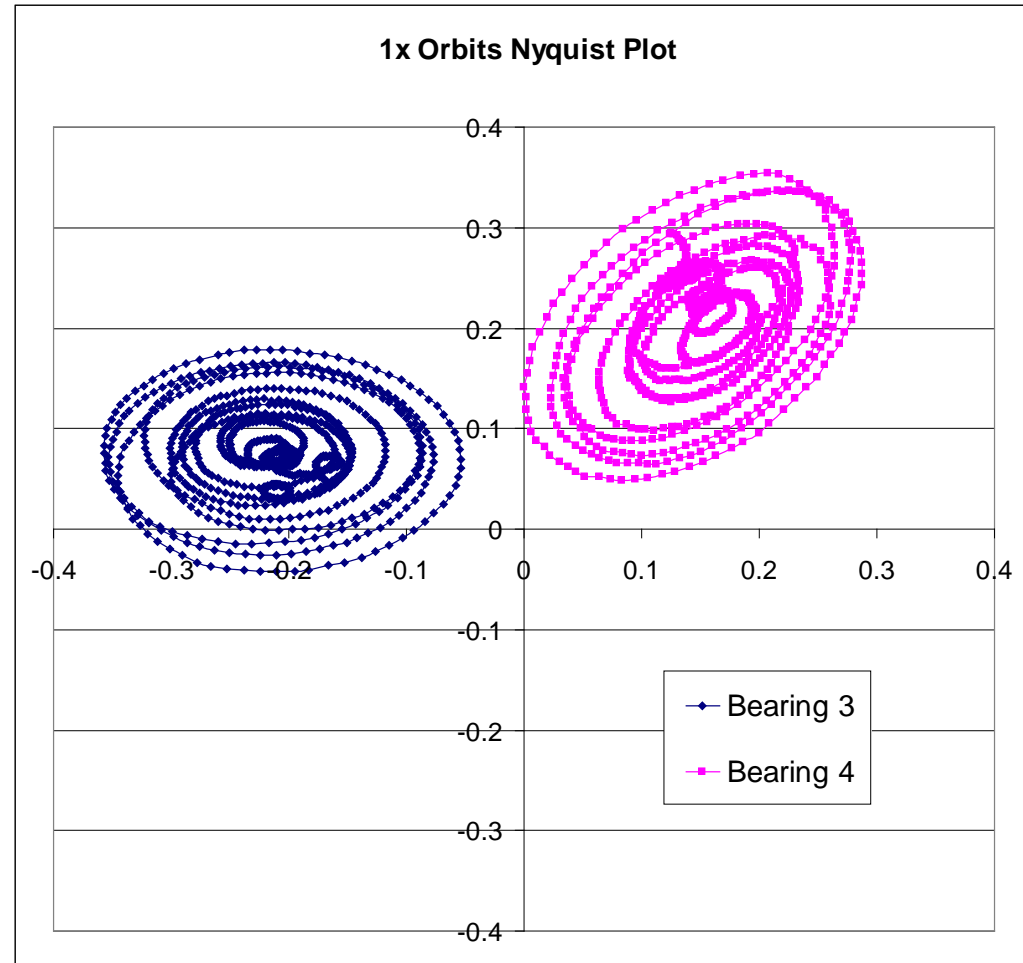
Synchronous Vibration Levels of Bearings vs. Time (Hour:Min:Sec)



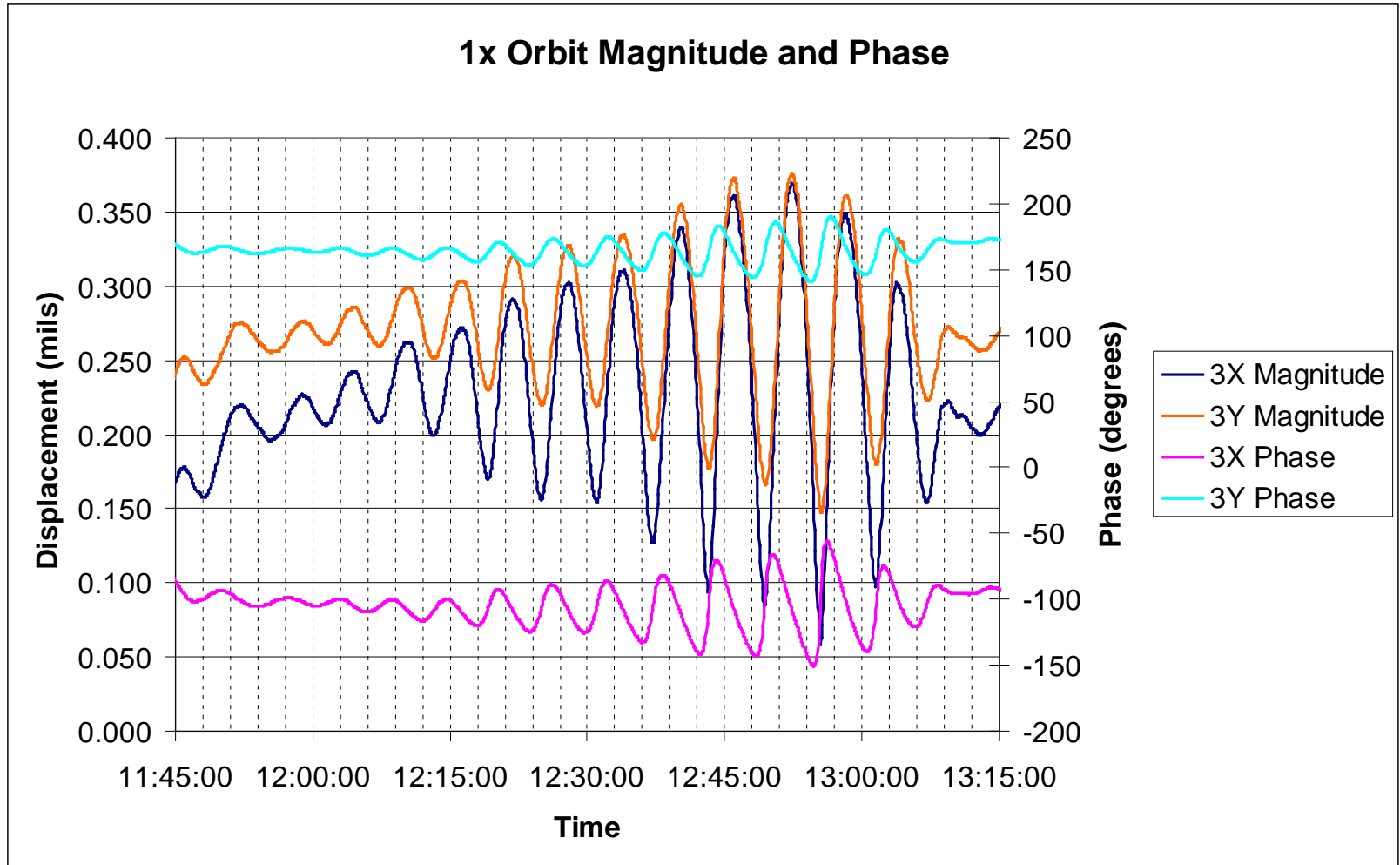
Nyquist Plotting Shows Net Imbalance Change Over Time

- These are **polar plots** of **amplitude** (vector length from graph center) and **phase angle** (angle the vector makes with the x-axis).
- As time progresses, if imbalance is changing, the tip of the vector plots out as an arc.
- If the imbalance change is cyclic, the vector tip plots as an ellipse or circle.

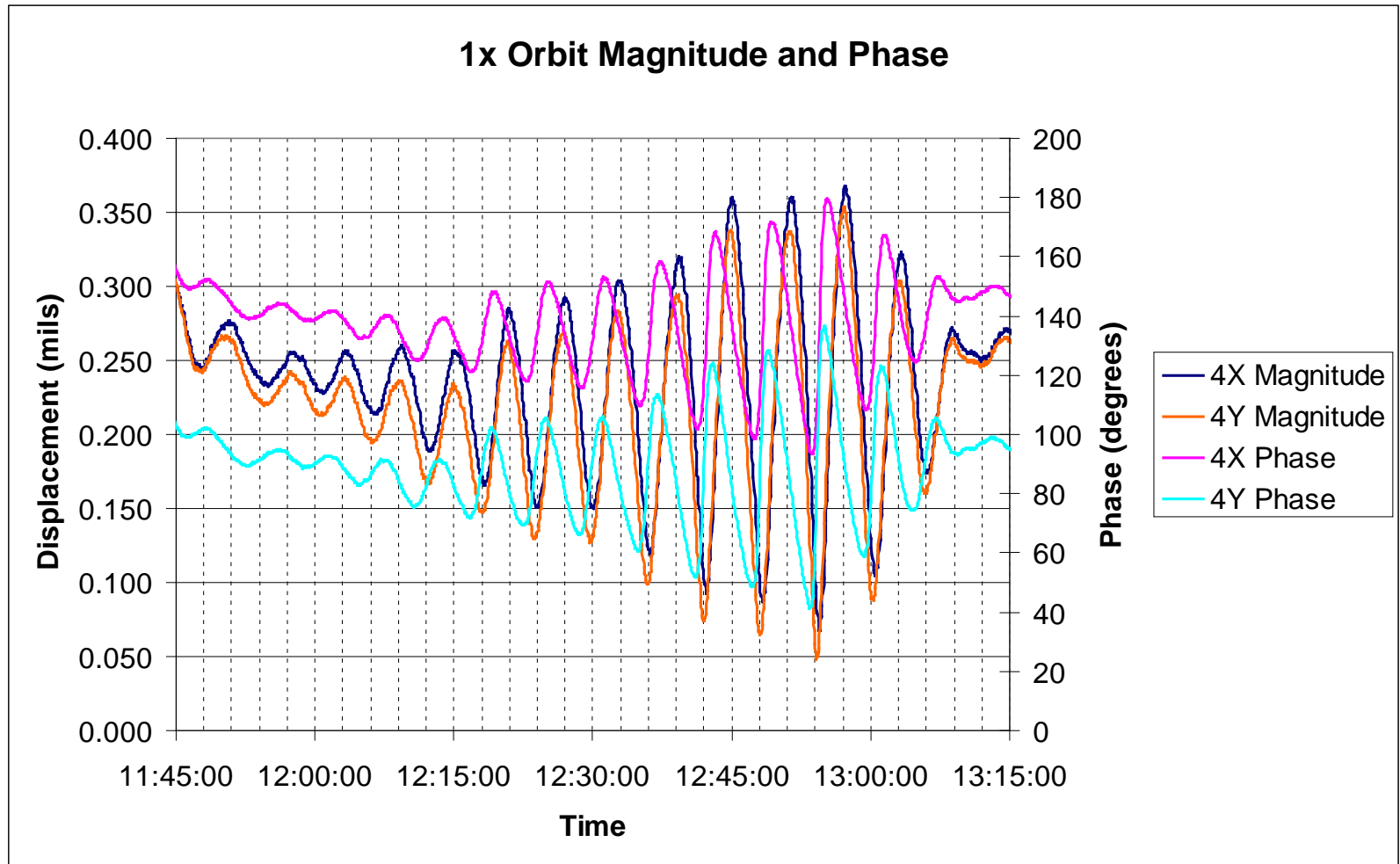
Nyquist Plots of Orbit Size & Phase for Left (“Bearing 3”) and Right (“Bearing 4”) Orbits



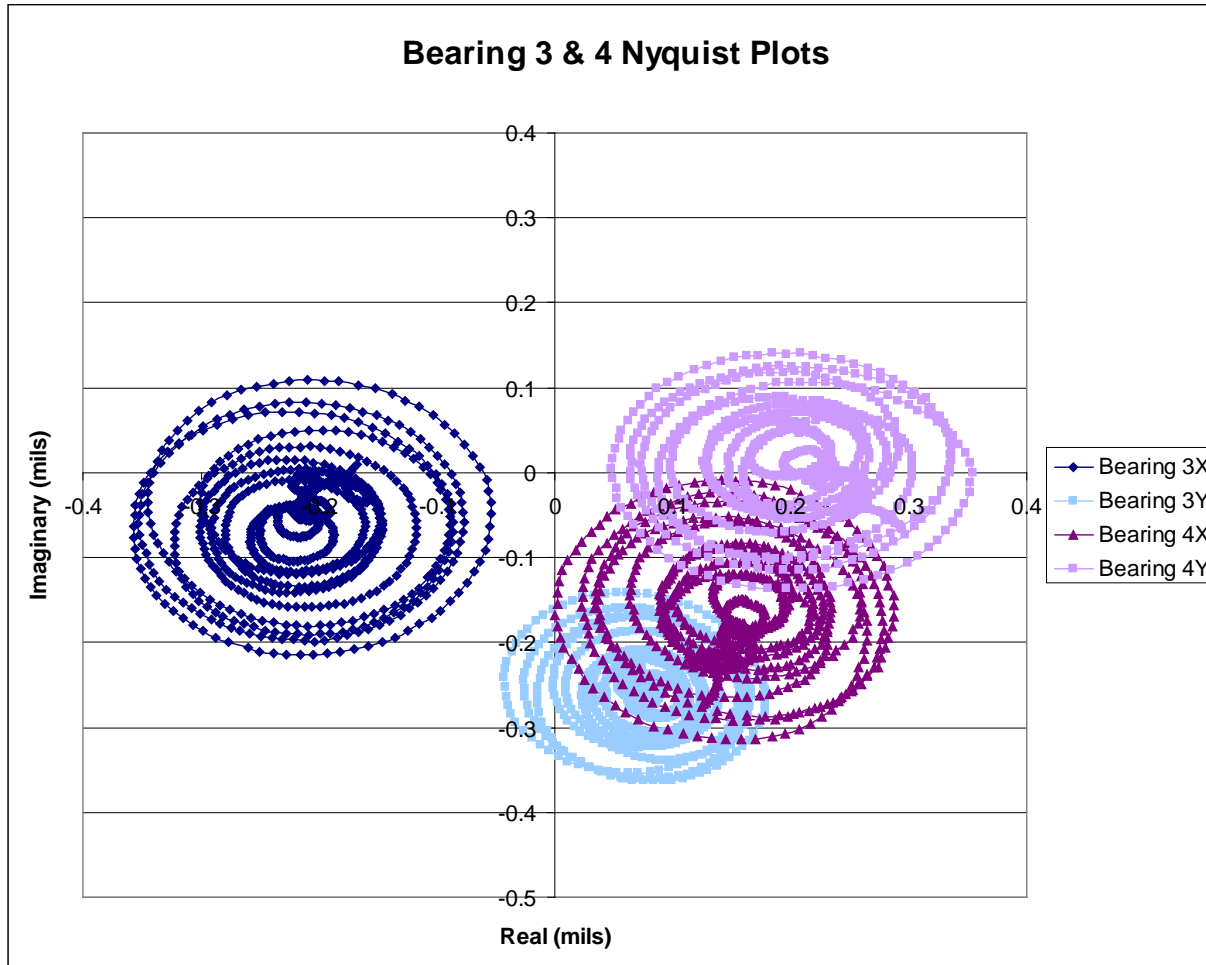
Synchronous Vibration Levels of Probes 3X and 3Y During Testing



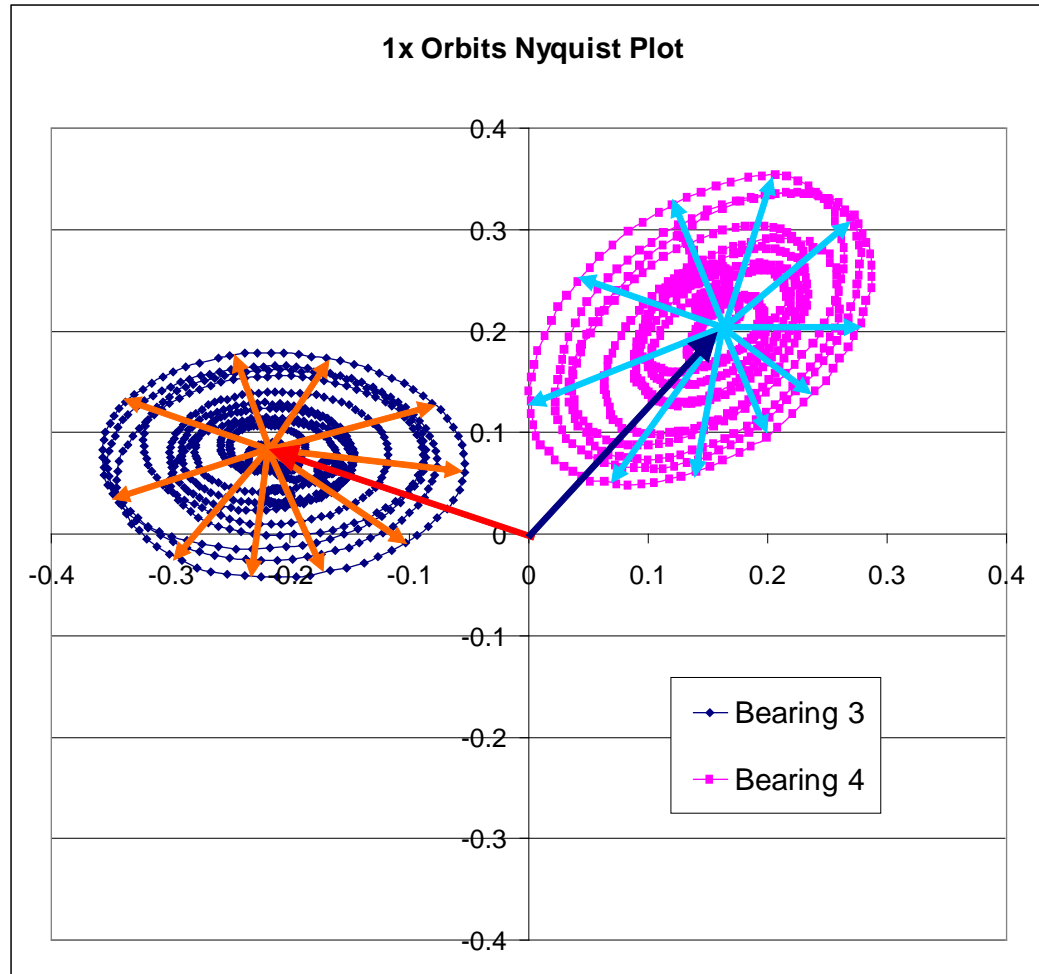
Synchronous Vibration Levels of Probes 4X and 4Y During Testing



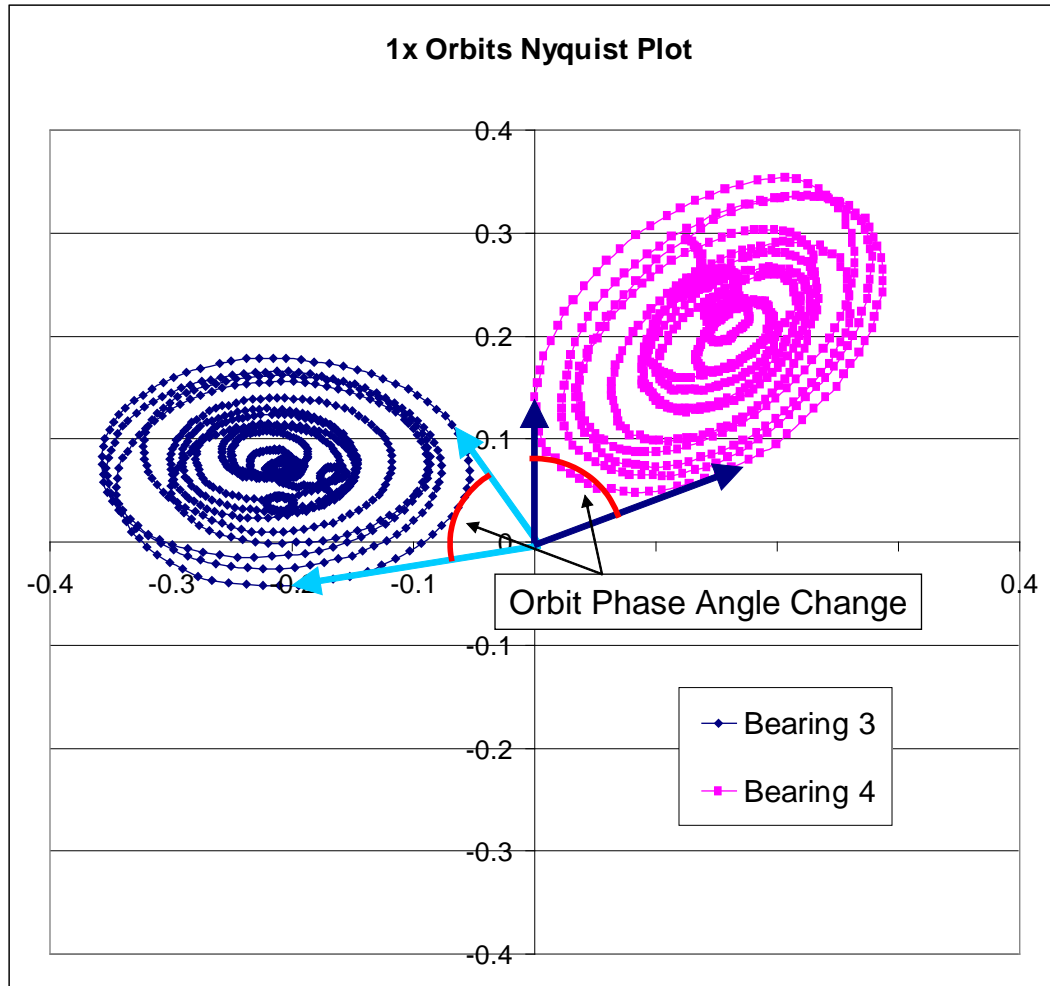
Nyquist Plots of Probes 3X, 3Y, 4X, and 4Y During Testing



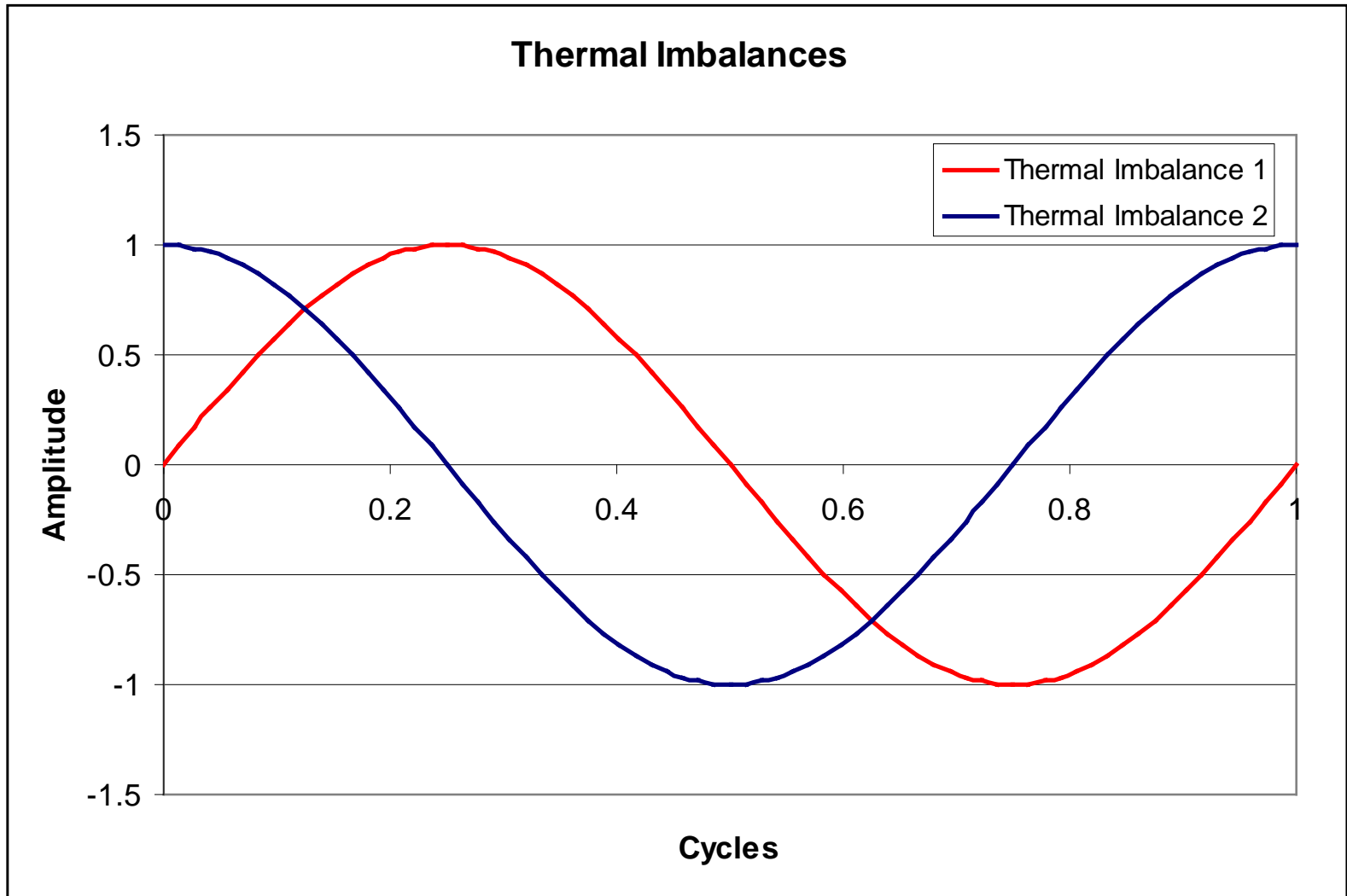
Nyquist Plots of Orbits with Residual Mechanical Imbalance & Cyclic Thermal Imbalance Vectors Overlaid



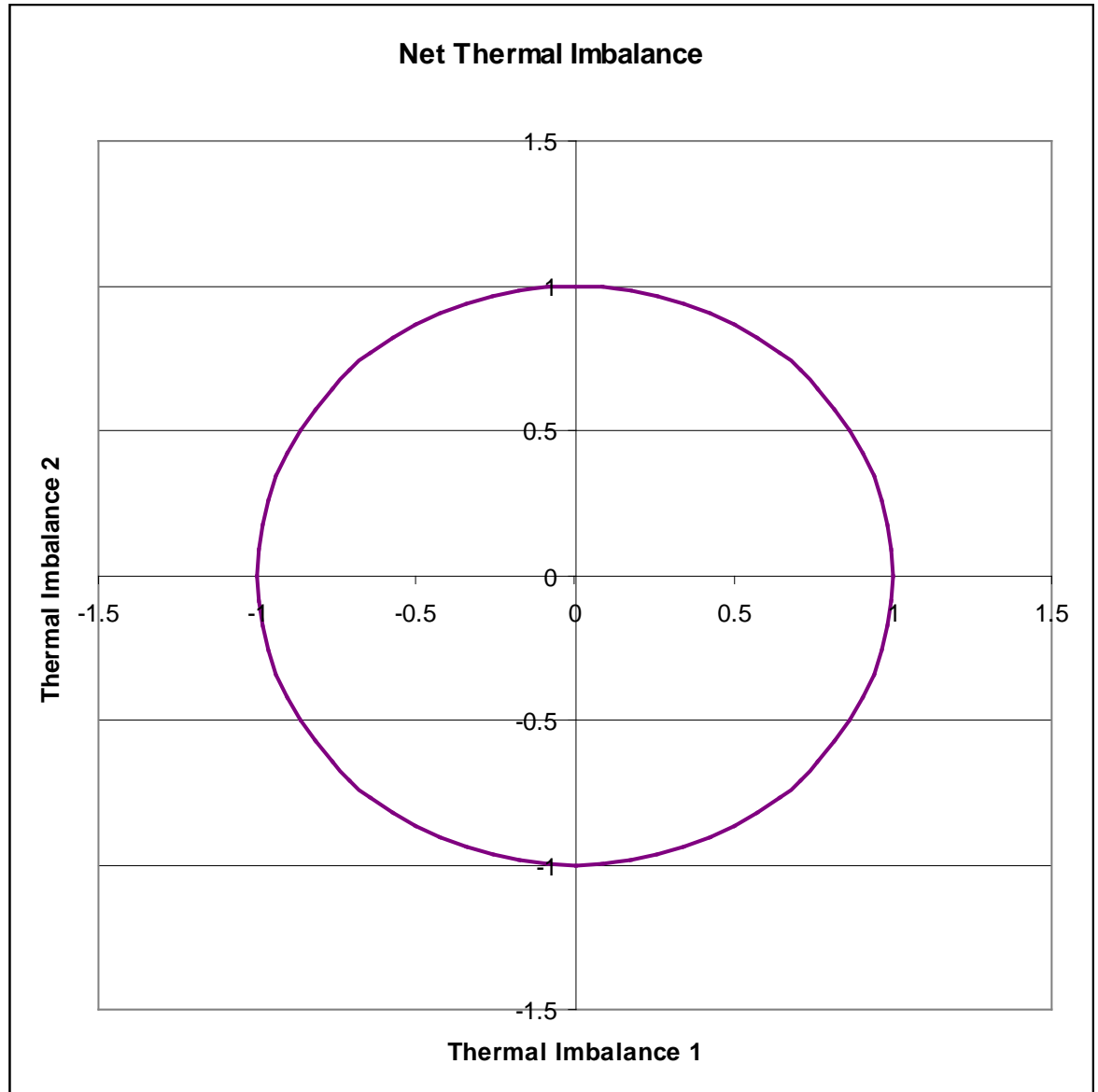
Nyquist Plots of Orbits 3 and 4 with Net Imbalance Force Vectors Overlaid



Example Thermal Imbalances



Net Thermal Imbalance From Addition of Example Thermal Imbalances



Observations

- The problem consisted of a slowly cycling amplitude of 1x vibration.
- The envelope of vibration of the rotor was observed to cycle to excessive levels according to a consistent period of about 6 minutes.
- The left stage versus right stage ends cycled roughly 60 degrees out of phase with each other.
- There was no evidence of unstable subsynchronous or resonant 1x vibration.
- The problem only occurred when oil inlet temperature was below a threshold level of 125F.

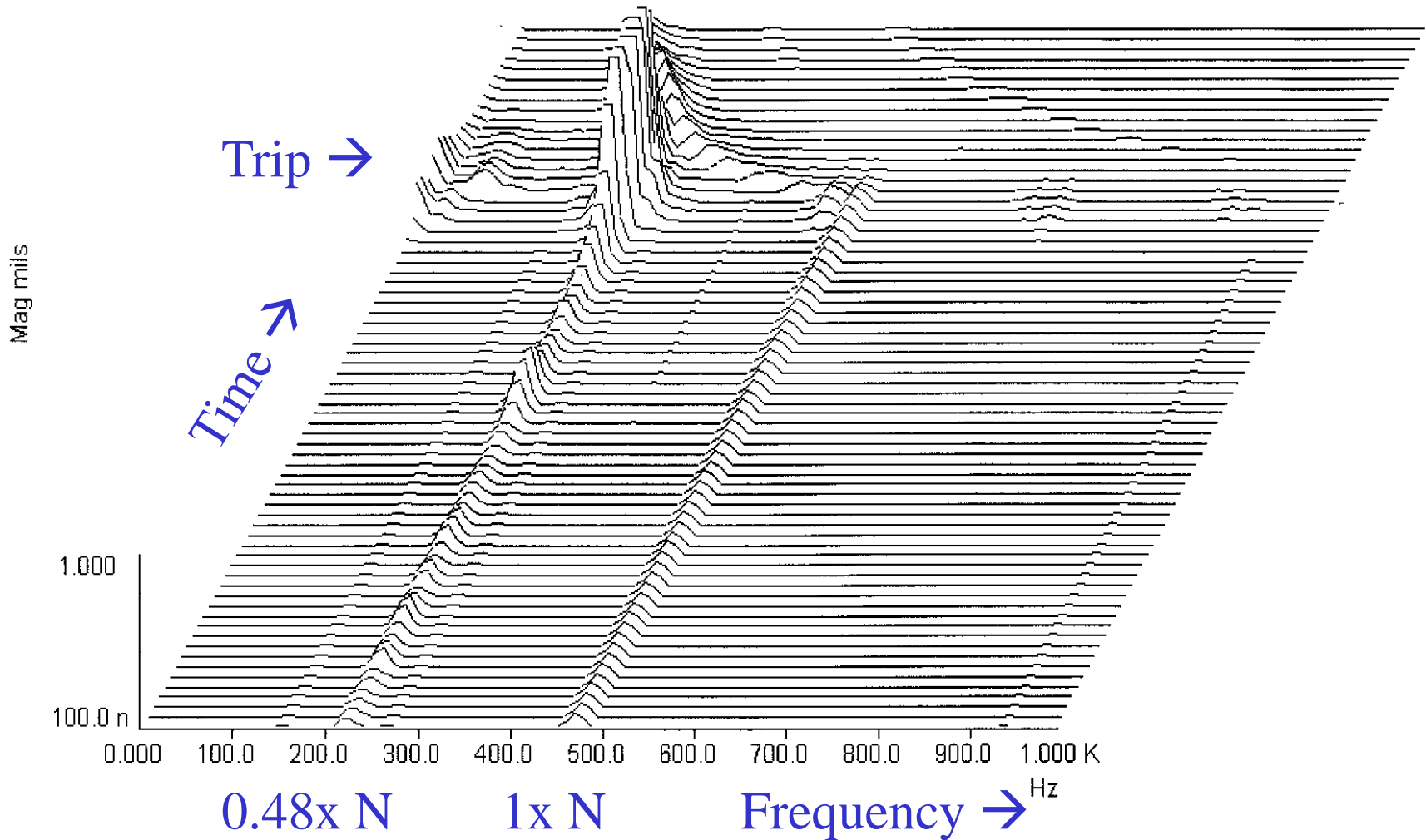
Evaluation Step 1

- Typically 1x running speed is symptomatic of resonance, imbalance, shaft bow, or insufficient bearing stiffness.
- If the problem was due solely to an imbalance, it would not be expected to slowly cycle, and the vibration of each stage of the rotor would be expected to increase or decrease in phase, end-to-end, due to changing conditions of operation.
- If the problem was due to mechanical shaft bow, symptoms similar to imbalance would be anticipated.

Evaluation Step 2

- If the problem was insufficient bearing stiffness, then a large 1x (synchronous) orbit would result, as was observed. Such low stiffness could represent a flaw in bearing design or manufacture, for example by the clearance being too large. Typically, this would not lead to rotor vibration amplitude cycling, however.
- In the case of resonant 1x vibration, typically a “skirt” occurs at the bottom of the 1x vibration narrowband peak, and such a skirt was not present.
- In the case of a rotordynamic instability, the vibration frequency would typically be in the range of 43% to 48% of running speed, which was not the case.

Typical Rotordynamic Instability



Conclusions

- In this case, slow oscillation occurred, in a manner that was not in phase end-to-end. Newkirk Effect (rotor rubbing, heating, wearing, cooling, and thereby reversing imbalance location on a cyclic basis) could possibly explain this, but if so the rotor would have been expected to quickly wear itself out, and this did not occur.
- “Morton Effect” is a form of imbalance or bow which changes its magnitude and phasing on a cyclic basis. Depending on the phasing of the thermal unbalance and the original residual mechanical unbalance of the two ends, a cyclic vibration pattern similar to the test data presented becomes possible.
- The circular relationship of cyclic vibration amplitude with phase change at each end is best explained by Morton Effect.

Bottom Line Conclusions

- Inlet oil temperature increase was effective in solving the problem, as was anticipated.
- A bearing design less prone to development of local hot spots would also have been an effective solution.