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Tel (513) 231-1748 / (513) 633-5429 | Fax (513) 231-2104

Case History

Project Scope

This project involved emergency onsite strain and vibration measurements concerning a 50 inch outside diameter disk/arbor assembly at a customer's repair facility in the US. The CT disk was made of Inconel and the temporary arbor was steel. The purpose of these measurements was to verify and document centrifugal expansion and yielding of the disk, particularly in the tangential (hoop) direction at the disk bore.

Throughout this project, DyRoMa conventions were used, particularly regarding vibration sensor naming and orientation. In other words, the disk/arbor assembly and balance stand unit is "viewed" as being from driver (first electric drive motor) toward driven (disk/arbor). Therefore, the coupling side bearing supporting the disk/arbor would be bearing #1 and the non-drive end (outboard) disk/arbor assembly bearing would be bearing #2. This balance stand has permanently installed dual XY orthogonal shaft relative proximity probe sensors in the "vertical" or "Y" direction (45 degrees left, "45L", of top dead center, TDC) and "horizontal" or "X" direction (45 degrees right, "45R", of top dead center, TDC) at each of the two bearings. This disk/arbor assembly also had two proximity probes "observing" the outside diameter of the disk in the radial direction. These probes were mounted on a fixture beneath the disk. A tangential and axial strain gage were also bonded to the inside diameter of the disk bore (discussed below).

This disk/arbor assembly was also equipped with a proximity probe sensor that functioned as the phase trigger (keyphasor®). The orientation of this phase trigger reference or keyphasor® and the direction of rotation (CCW, counterclockwise) follow this same DyRoMa convention. The phase trigger was oriented at 90 degrees right (90R) of TDC and located adjacent to the balance stand drive shaft. This phase trigger is a proximity probe which senses the passage of a protruding key on the balance stand drive shaft once per revolution, thus yielding a speed and phase reference signal. This phase reference is critical in the process of balancing rotor in balance stand.

For these measurements, a Data Physics 32 channel Abacus® Turbo vibration data acquisition system was field wired using BNC patch cables and tee connectors into the Bently Nevada ADRE® 208 inputs to acquire the XY shaft relative proximity probe signals at bearings #1 and #2. Signals from the disk OD proximity probes were also acquired using BNC patch cables and tee connectors at the BNC terminal strip adjacent to the ADRE® system in the back of the control panel. The strain gage signals were obtained via BNC cables at the output BNC connection on the two wireless telemetry systems. The Abacus was used in this instance for strain and vibration measurements; however it can also be used for dynamic balancing of rotating machinery.

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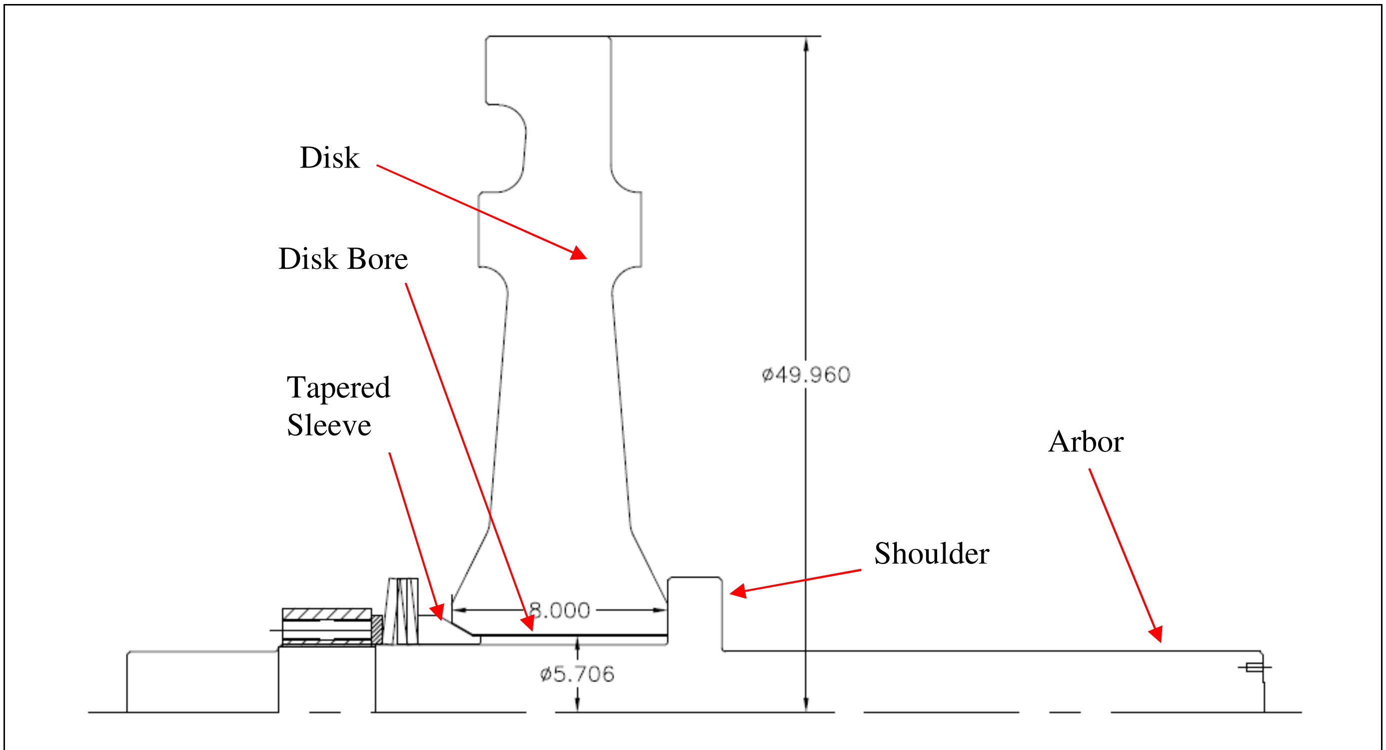
Onsite strain and vibration measurements took place on November 22nd. The strain and vibration data was acquired during transient and steady state operation of the disk/arbor assembly, while the disk/arbor was in the balance stand bunker under vacuum. A Data Physics Abacus® Turbo 32 channel data acquisition system was used to capture strain, disk radial expansion, speed, and arbor shaft relative vibration signals. Objectives for this project were outlined as follows:

- Record and document the strain and vibration response characteristics of the disk/arbor assembly during transient and steady state operating conditions.
- Review and evaluate the strain and vibration response characteristics and prepare a summary with recommendations.
- Provide a final documentation covering significant aspects of the project.

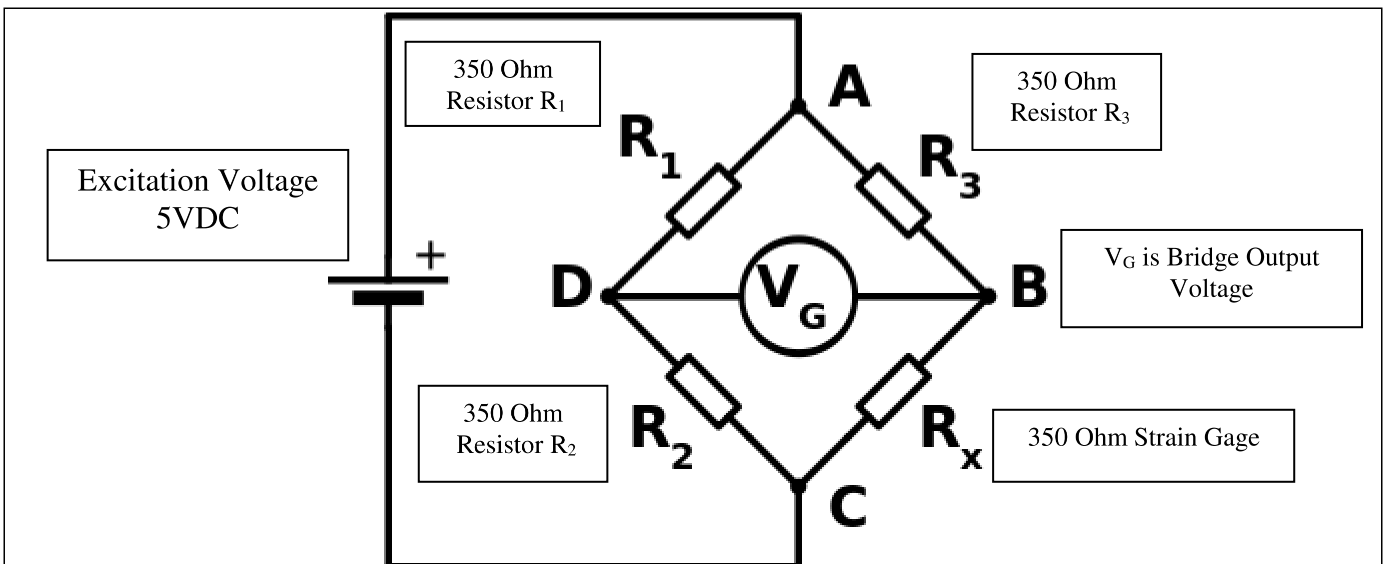
Instrumentation

To measure the disk inside diameter tangential and axial strains, two uniaxial encapsulated strain gages (Micro-Measurements J2A-06-S047K-350, 0.060 inch gage length) were bonded to the bore surface using a quick curing epoxy adhesive, see drawing 1. A third strain gage was also bonded to the bore in the tangential direction as a backup gage; however it was not used during the measurements. The tangential strain gage was bonded to the bore surface at 3.35 inches from the arbor shoulder, and the backup strain gage was at 2.875 inches. The axial strain gage was bonded at 1.58 inches, see photo 1. These lead wires were bonded to the inside diameter of the disk bore using a 5 minute epoxy. The leads were feed through an arbor radial hole to an arbor center drilled axial hole, then to a telemetry system housing mounted on the non-drive end of the arbor, see photos 2 and 3.

Within this housing, these strain gages were connected to two Wheatstone bridge circuits using strain gage lead wires; one bridge circuit for each bonded strain gage, see drawing 2. Excitation voltage for the bridge circuits was provided by the ATi wireless telemetry transmitters; see drawing 3 and photo 4. Bridge excitation voltage was 5 volts DC. Each transmitter also “senses” the bridge output voltage of the attached bridge circuit, which it then broadcasts this voltage to the ATi wireless telemetry receiver. With no strain in the disk bore, each Wheatstone bridge circuit is “balanced” because the resistance of the strain gage is 350 ohms, which matches the other three resistors in the bridge circuit so the bridge output voltage (V_G) is zero. If a tensile strain occurs at the strain gage, the gage resistance increase, which cause the bridge circuit to be “unbalanced” and a positive voltage proportional to the magnitude of the strain occurs at the bridge output. Conversely if a compressive strain occurs at the strain gage, the gage resistance decrease, which cause the bridge circuit to be “unbalanced” and a negative voltage proportional to the magnitude of the strain occurs at the bridge output.



Drawing 1: Disk and arbor assembly



Drawing 2: Wheatstone bridge circuit

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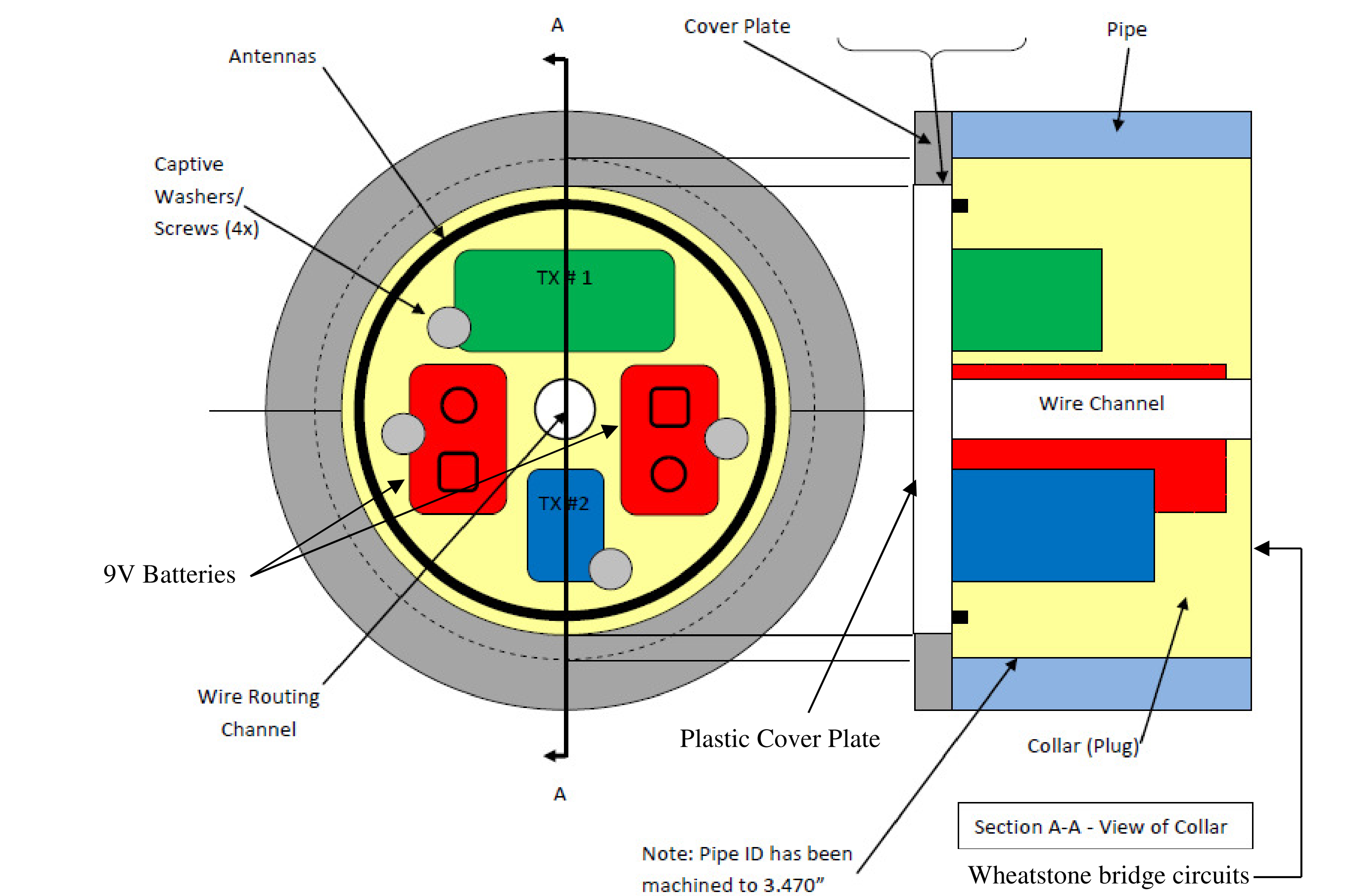

Drawing 3: Wireless telemetry system arrangement, TX#1 is transmitter 1, TX#2 is transmitters 2

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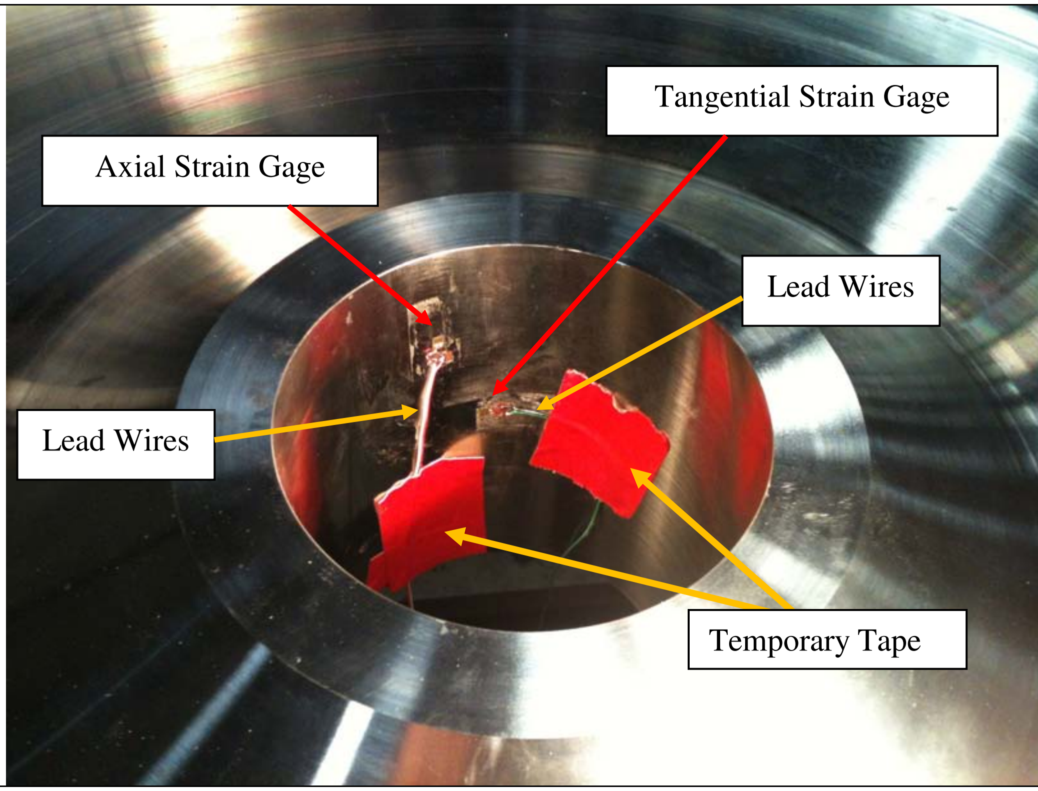

Photo 1: Tangential and axial strain gages

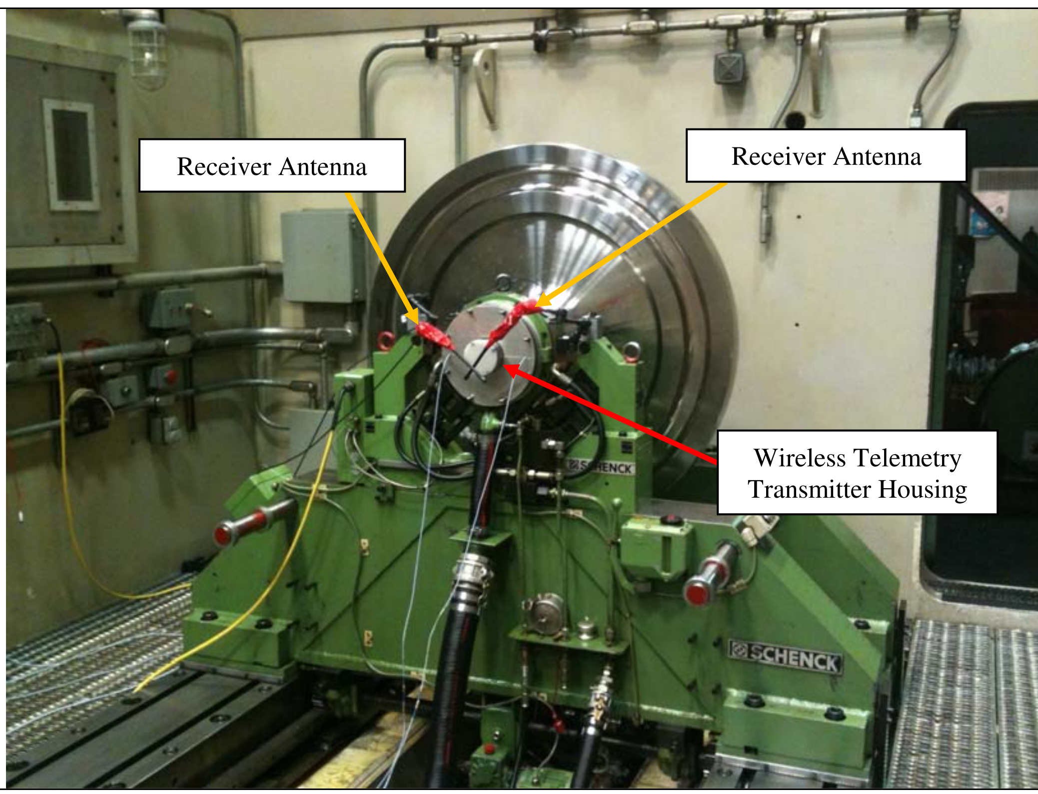

Photo 2: Disk/arbor with wireless telemetry system installed

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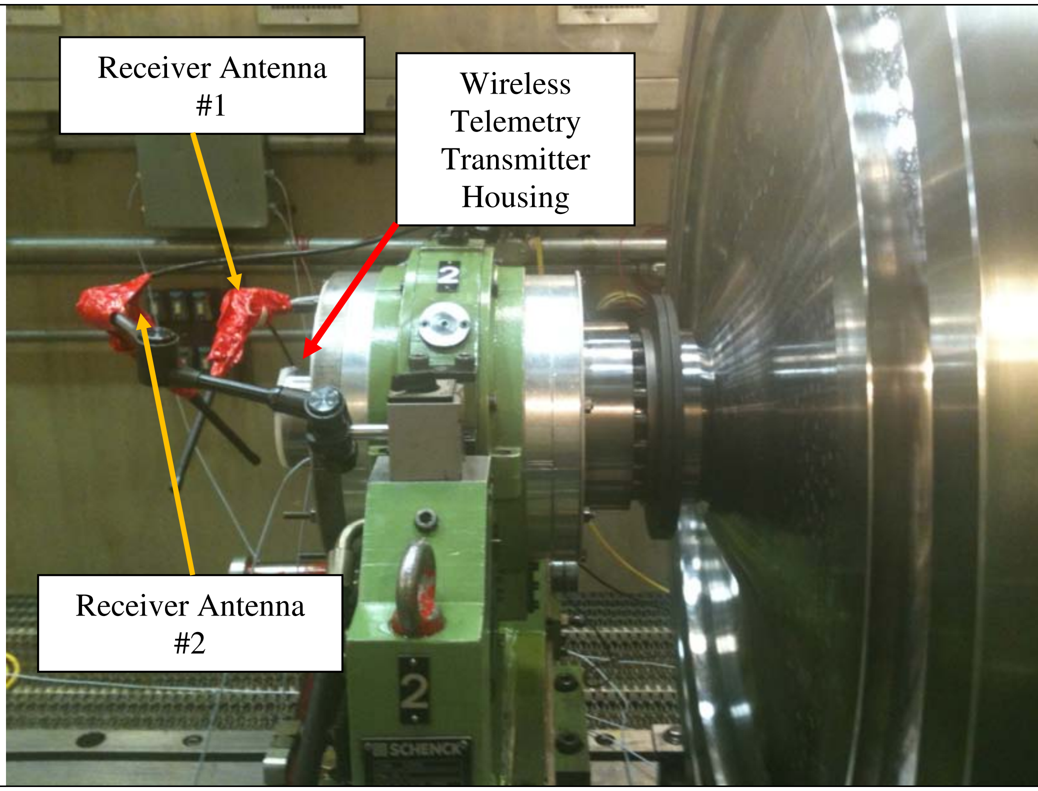

Photo 3: Disk/arbor with wireless telemetry system installed

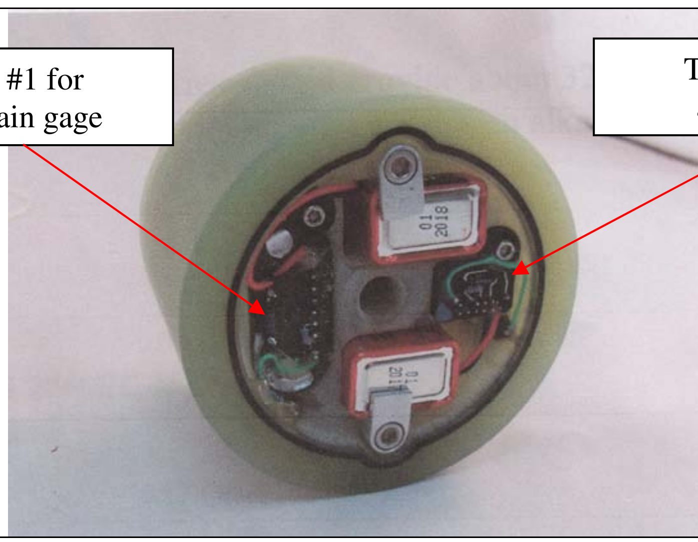

Photo 4: Telemetry transmitters and 9V batteries installed in "plug"

At the start of the measurements with the disk/arbor at zero rpm, each Wheatstone bridge is balanced to zero volts output at the telemetry receiver and then a 10,000 ohm shunt calibration resistor is momentarily connected in parallel to one of the 350 ohm resistors of the bridge circuit to adjust the telemetry systems span. This shunt resistor simulates a strain of 16,577 microstrain (1 microstrain = 10^{-6} in/in, or 10,000 microstrain = 1% strain) resulting in an output of 4.300 volts DC at the receiver. Therefore, 10,000 microstrain at the strain gages causes a receiver output voltage of 2.594 volts DC. Therefore, the strain gage signal channels have sensitivity of 0.2594 millivolts per microstrain.

With the tangential and axial strain gages measuring the deformation of the disk bore, two proximity probes were mounted beneath the disk to measure radial expansion of the disk as it rotated. These probes were also calibrated prior to performing any measurements. The disk itself was used as the target material (Inconel). Proximity probe sensitivities for the first radial probe (Probe1) were 0.246 volts per mil, and the second radial probe (Probe2) was 0.262 volts per mil. These probes were gapped to -19.0 volts (Probe1) and -19.8 volts (Probe2), so that the radial expansion would cover the linear range of the two probes.

Vibration of the arbor at bearings #1 and #2 was measured using XY orthogonal shaft relative proximity probes. These shaft relative proximity probes were orientated in the "vertical" or "Y" direction (45 degrees left, "45L", of **t**op **d**ead **c**enter, TDC) and "horizontal" or "X" direction (45 degrees right, "45R", of **t**op **d**ead **c**enter, TDC) at each of the two arbor bearings. The disk/arbor rotational speed or rpm was also being measured with an additional proximity probe orientated at 90R at the input drive shaft, and was detecting the once per revolution passage of a protruding key. This phase trigger probe (keyphasor®) provided speed and phase information throughout these measurements.

Strain gage and proximity probe signals were acquired with a 32 channel Data Physics Abacus® Turbo data acquisition system, see photo 5. This system was setup to acquire synchronously and asynchronously sampled data with vector samples obtained every 3 seconds and/or 30 rpm change. Spectral bandwidth was limited to 30,000 cpm with 800 spectral lines of resolution.

Photo 5: Data Physics Abacus Data Acquisition System

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Summary

Three preliminary runs of the disk/arbor were conducted prior to the final run discussed below. During these first three runs, rotational speed of the disk/arbor was kept at or below 6000 rpm. During these initial three runs, the tangential and axial strain gages were cycled to reduce residual strain from bonding and curing of adhesives during installation. This is a normal strain gage practice and is called a “zero return” or “stability” test (see Vishay Micro measurements VMM-8 application document).

The speed profile for the fourth and final run is shown in figure 1. The maximum rotational speed was 8236 rpm at 12:44:49 EST. Overall (unfiltered) and synchronous (1X) vibration amplitudes and phase information for vibration at arbor bearings #1 and #2 are shown in figures 2 and 3. Vibration amplitudes at 8000 rpm were acceptable. These shaft relative vibration measurements also revealed a first critical speed (first balance resonance) of the disk/arbor assembly at about 3700 rpm, see figure 4.

The change in disk radius was also measured during this final run, see figure 5. These trends do not include a correction for the arbor centerline position change due to arbor journal lifting due to the bearing oil wedge. These shaft centerline position changes or movements are shown in figures 6. When arbor centerline motion is included, disk radial expansion is increased, see figure 7. The maximum expansion occurred at 8236 rpm, and it was 62.4 mils and 62.2 mils for radial probes #1 and #2 respectively. Figure 8 shows a bode plot of the disk radial expansion during startup and shutdown. Clearly, the disk radius appears greater during shutdown. The final bore radius changes were 3.2 mils and 3.5 mils for probes #1 and #2. This effect could be due to yielding of the disk or the tapered sleeve being stuck and not fully retracting during the shutdown. Disk bore diametric measurements would be required to confirm whether the disk had yielded and to what degree.

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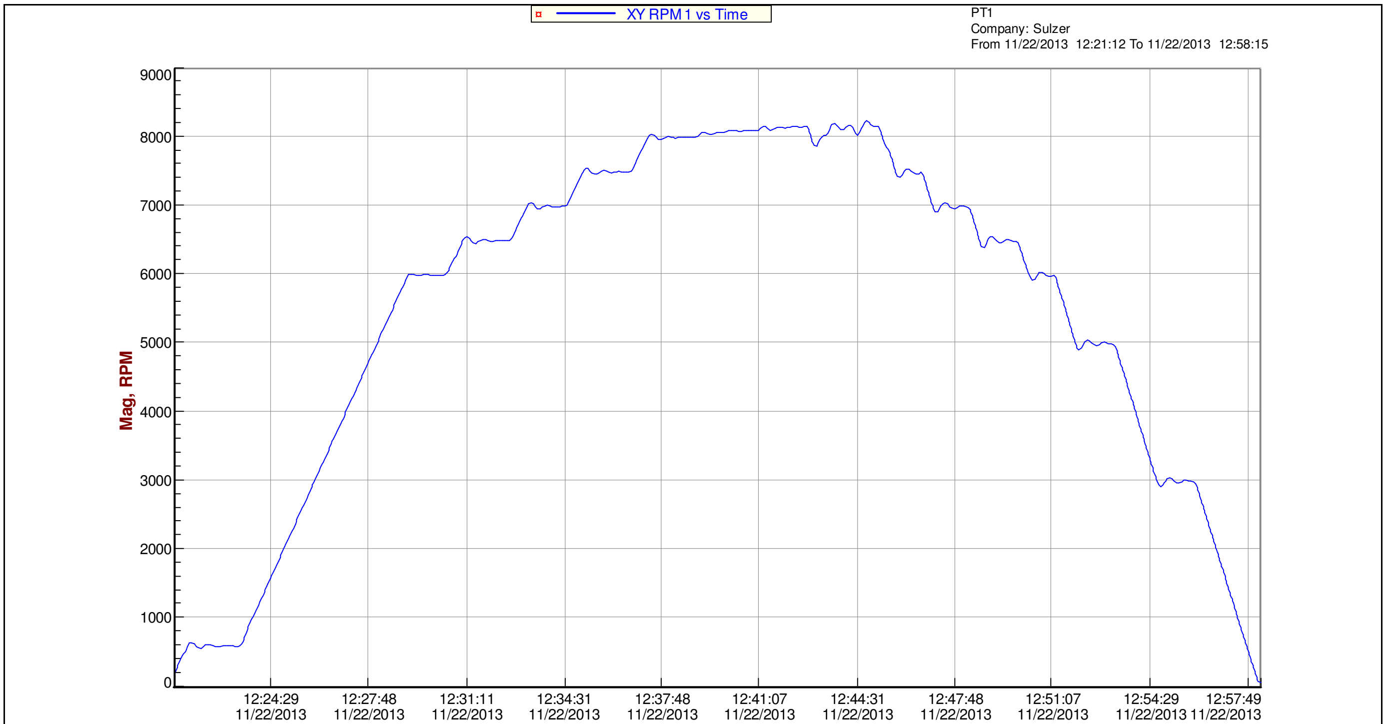


Figure 1: Rotational speed profile, Disk/arbor, Fourth Run

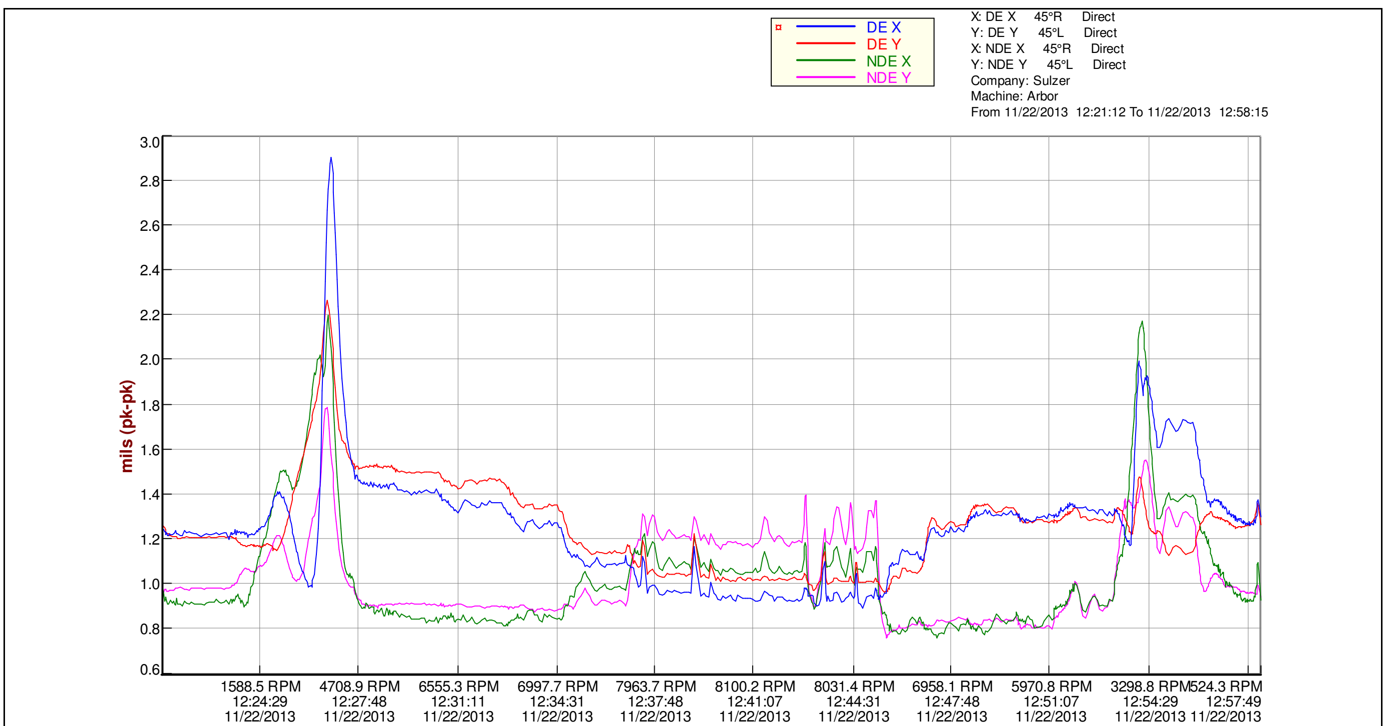


Figure 2: Trend Plot, Shaft Relative Vibration, Overall (unfiltered) Amplitude, Bearing #1 X Probe (blue trace) and Y Probe (red trace), Bearing #2 X Probe (green trace) Y Probe (magenta trace)

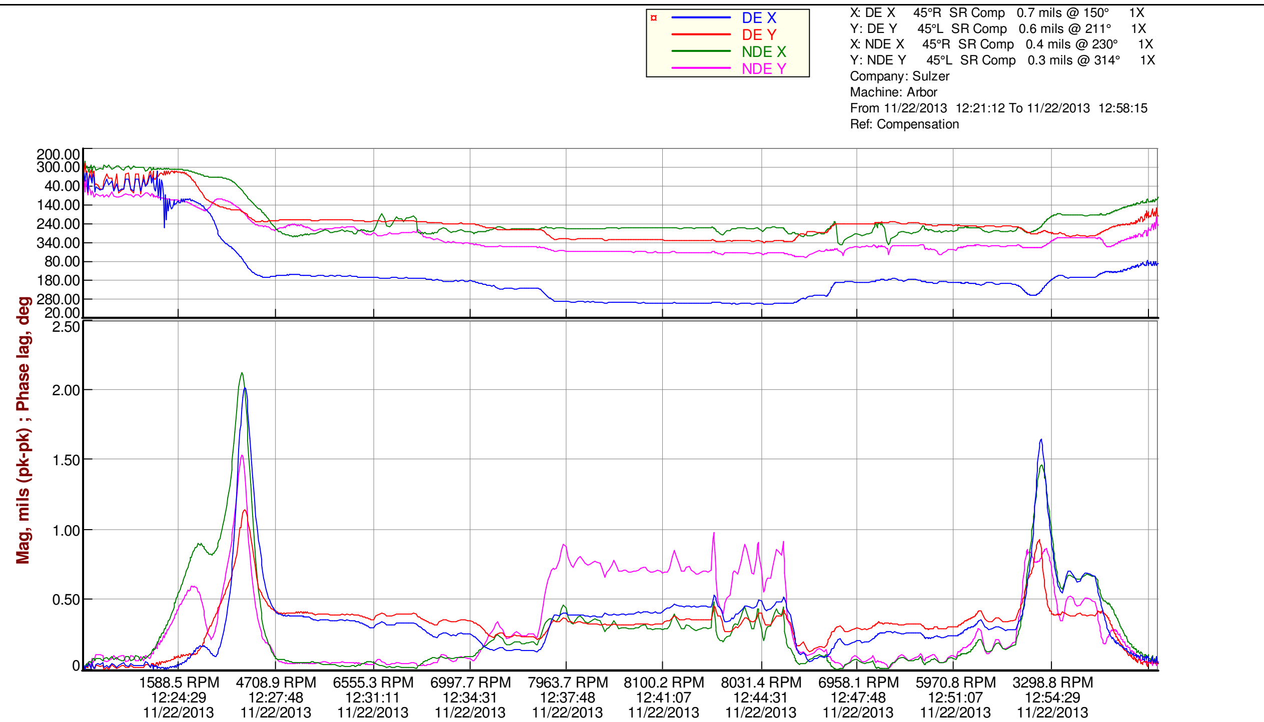


Figure 3: Trend Plot, Shaft Relative Vibration, Synchronous (1X) Filtered Amplitude and Phase, Bearing #1 X Probe (blue trace) and Y Probe (red trace), Bearing #2 X Probe (green trace) Y Probe (magenta trace)

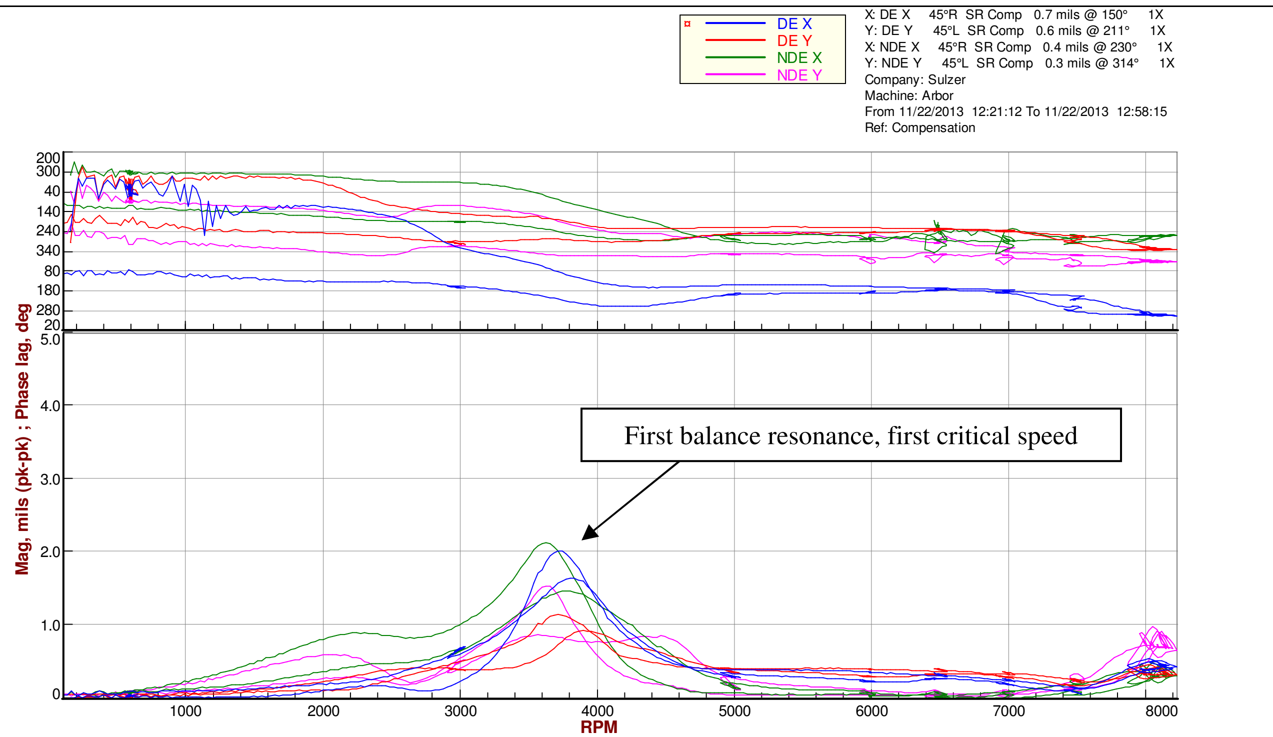


Figure 4: Bode Plot, Shaft Relative Vibration, Synchronous (1X) Filtered Amplitude and Phase, Bearing #1 X Probe (blue trace) and Y Probe (red trace), Bearing #2 X Probe (green trace) Y Probe (magenta trace)

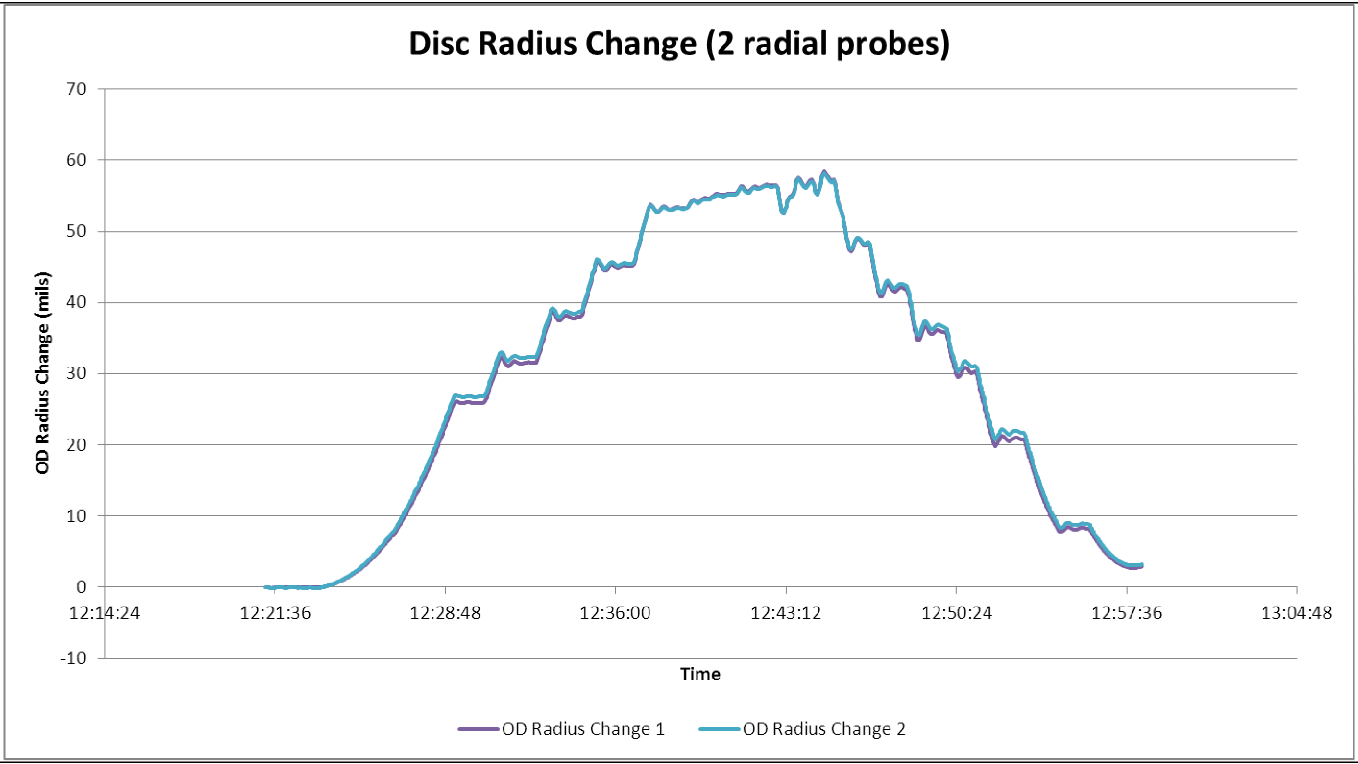
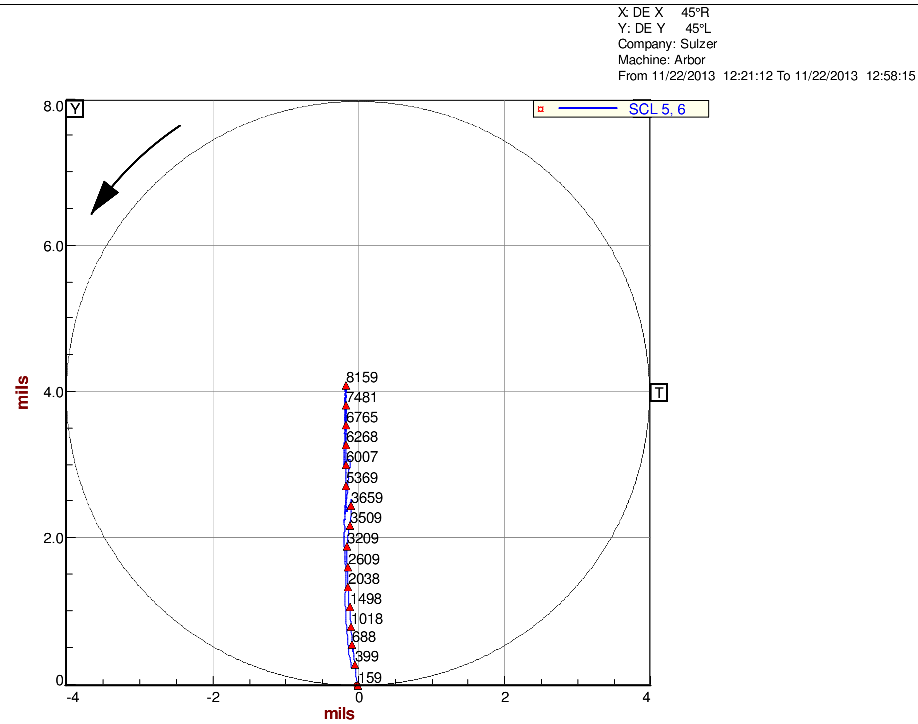


Figure 5: Trend Plot, Disk Radial Probes #1 and #2, Disk Radial Expansion



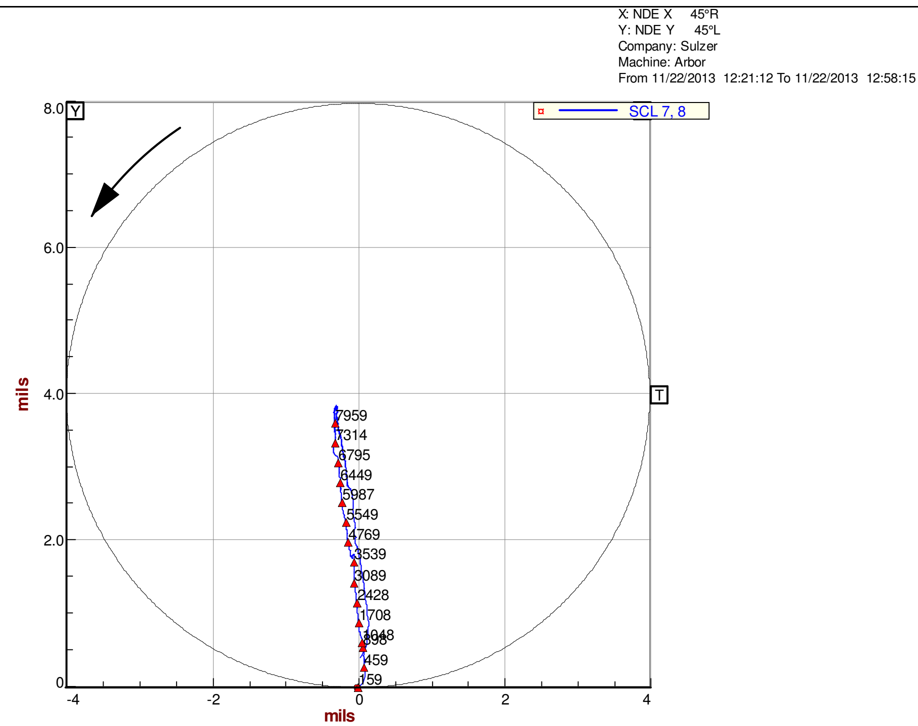


Figure 6: Shaft Centerline Plots, Bearing #1 (top), Bearing #2 (bottom)

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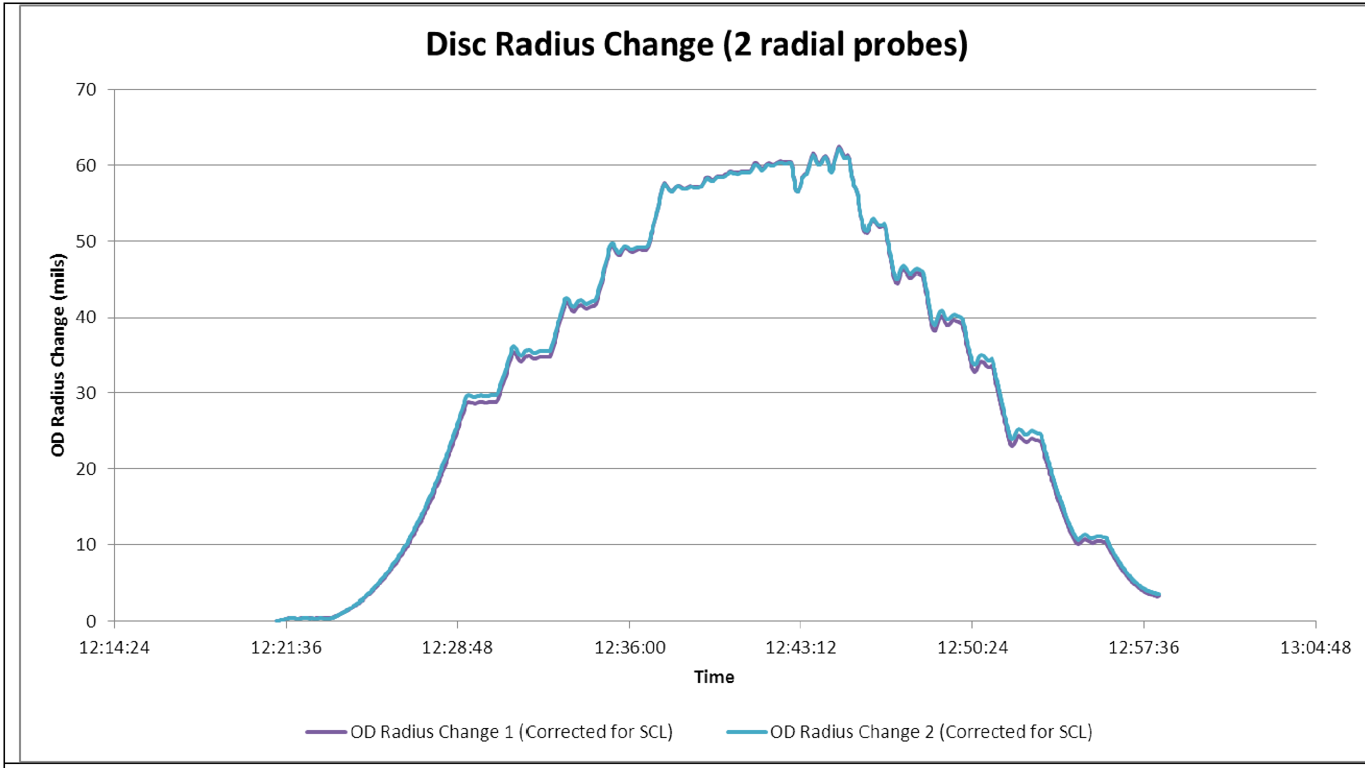

Figure 7: Trend Plot, Disk Radial Probes #1 and #2, Disk Radial Expansion, Arbor shaft centerline movement included

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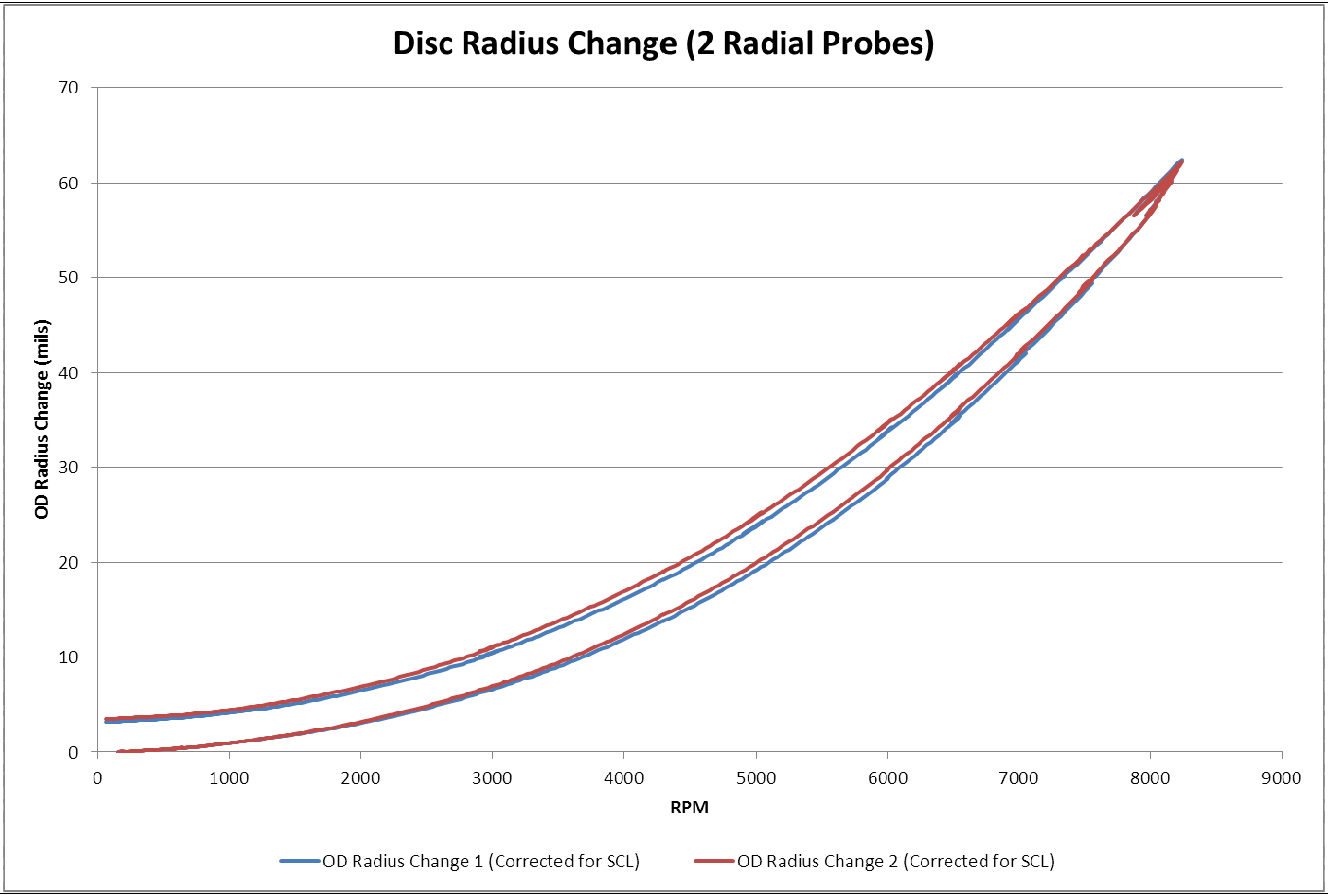

Figure 8: Bode Plot, Disk Radial Probes #1 and #2, Disk Radial Expansion, Arbor shaft centerline movement included

The tangential and axial strain trend/histories for the final run are shown in figure 9. As expected, the tangential strain gage showed tensile (positive) strain, which reached a maximum of +9785 microstrain at 8236 rpm. The axial strain gage showed compressive strain as expected, and it too reach a maximum at 8236 rpm. Maximum compressive (negative) axial strain was -4445 microstrain. A bode plot of the strain histories clearly show residual tensile strain in the bore tangential direction and compressive strain in the bore axial direction, see figure 10. The final residual strains were +2356 microstrain and -1955 microstrain for the tangential and axial directions respectively. Again, this effect could be due to yielding of the disk or the tapered sleeve being stuck and not fully retracting during the shutdown. As mentioned previously, disk bore diametric measurements would be required to confirm whether the disk had yielded and to what degree.

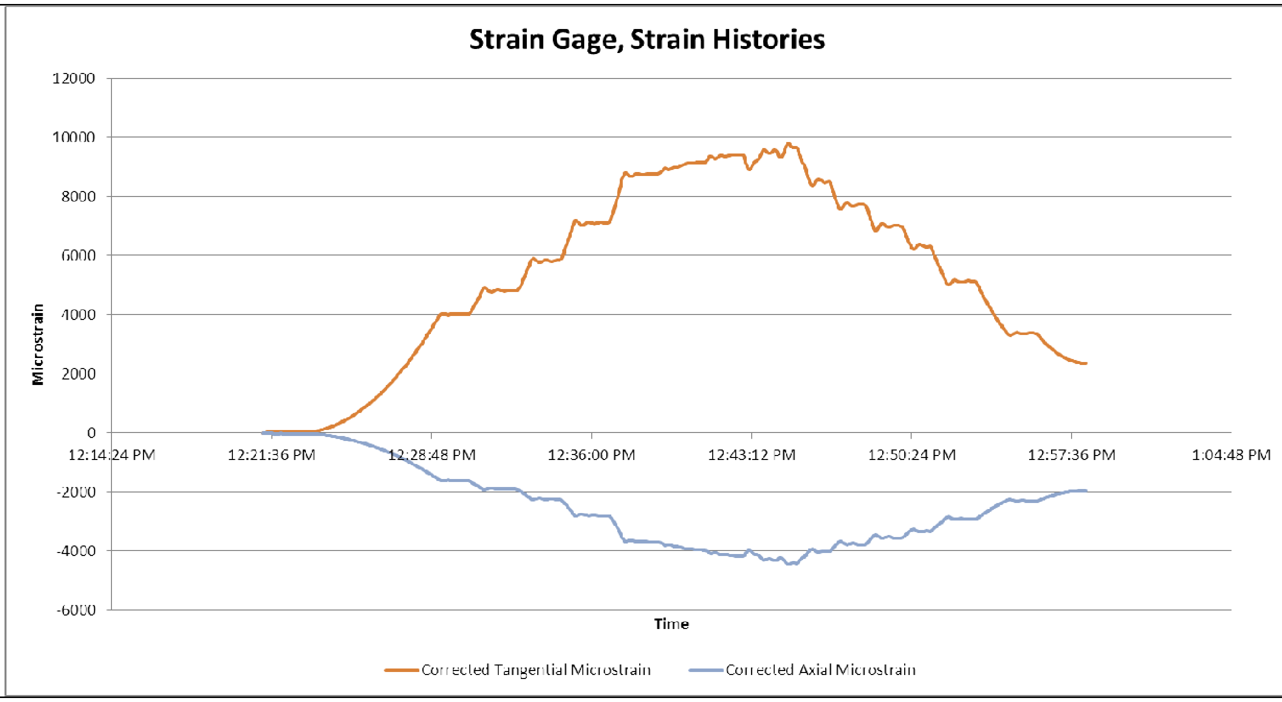

Figure 9: Trend Plot, Tangential and Axial Strain Gage, Strain Histories, Initial Residual Bridge Zeroing Voltage Removed/Corrected (zero initial strain @ zero rpm)

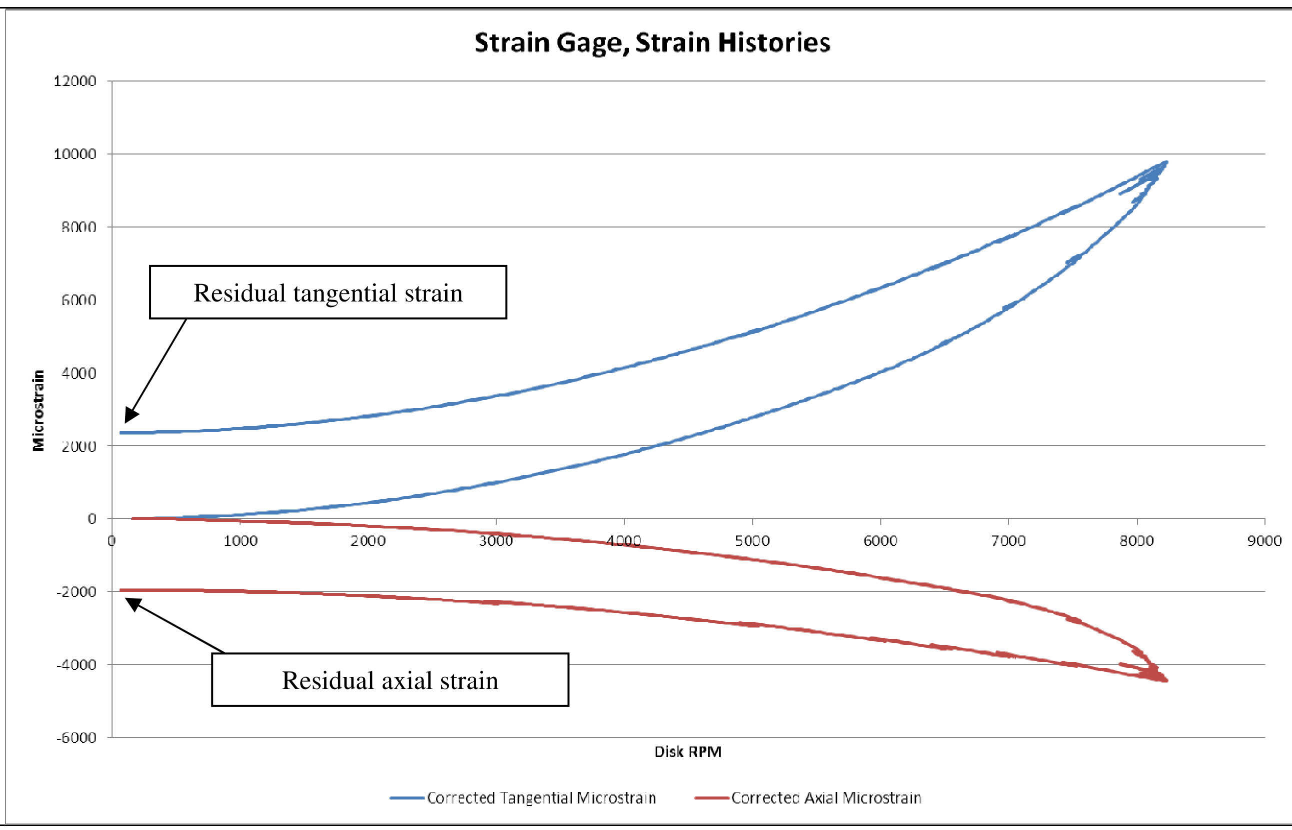

Figure 10: Trend Plot, Tangential and Axial Strain Gage, Strain Histories, Initial Residual Bridge Zeroing Voltage Removed/Corrected (zero initial strain @ zero rpm)

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Conclusions

- Disk bore diametric measurements were subsequently performed and confirmed that the disk had yielded to its target strain level during the final run.

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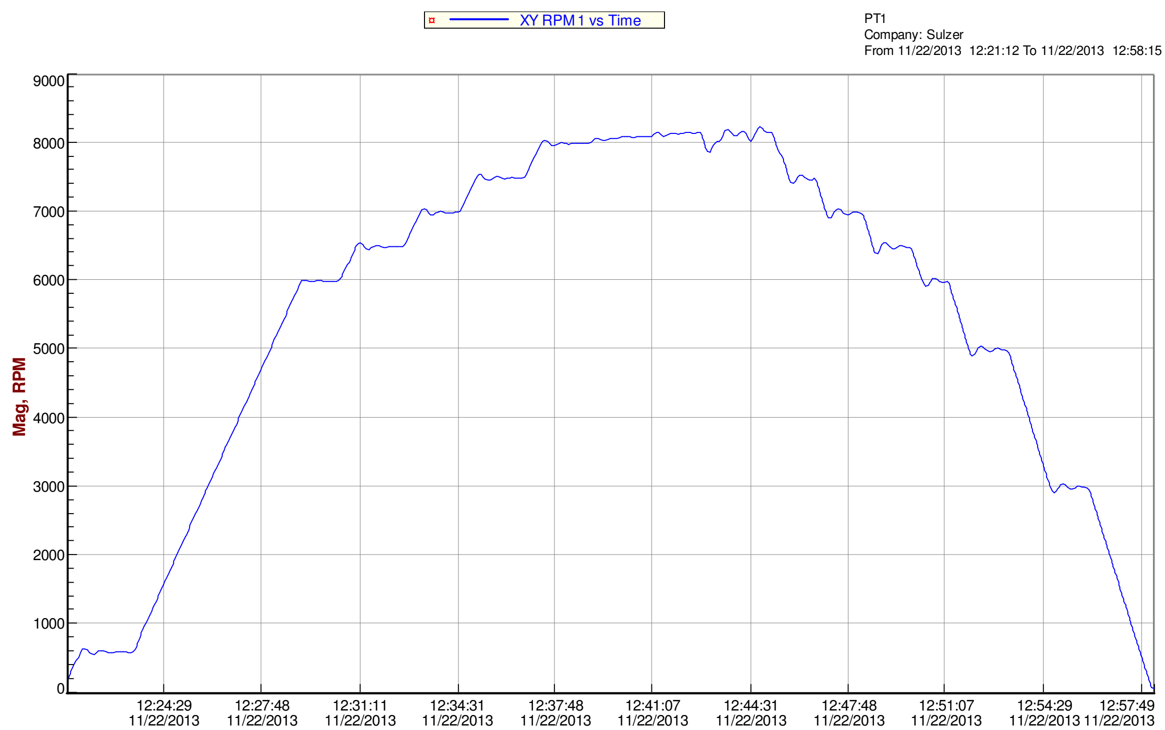
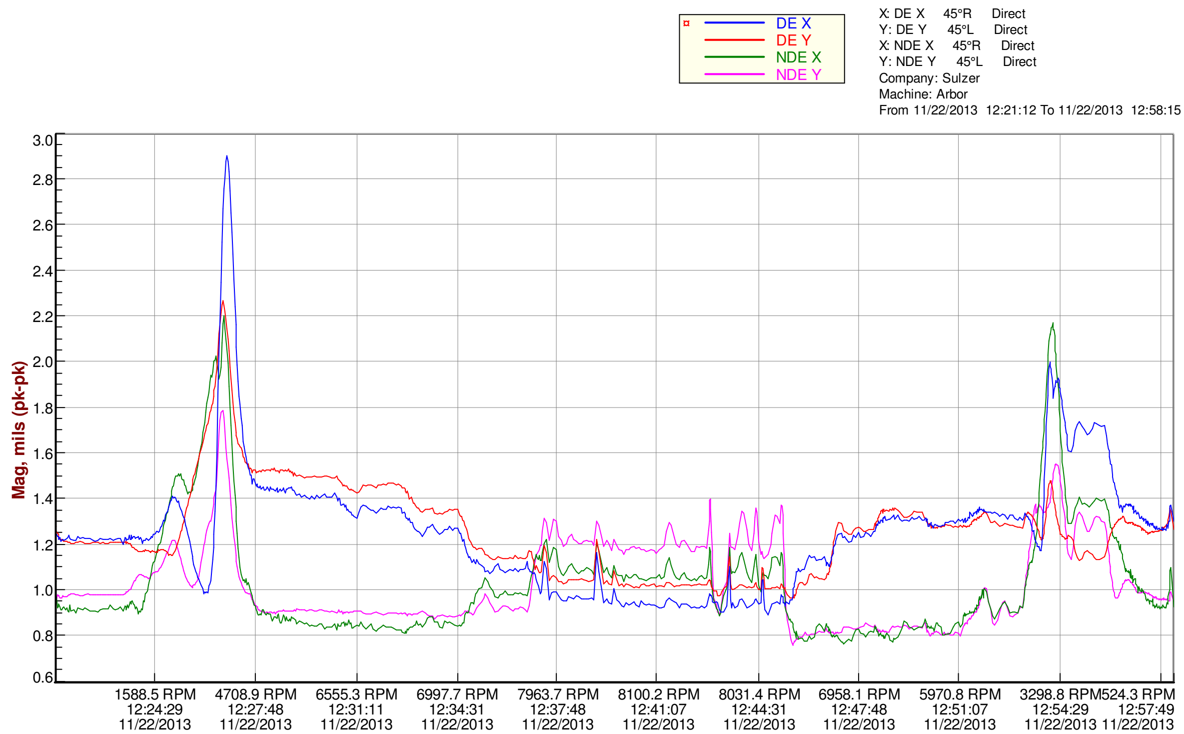
Appendix A
Data Plots





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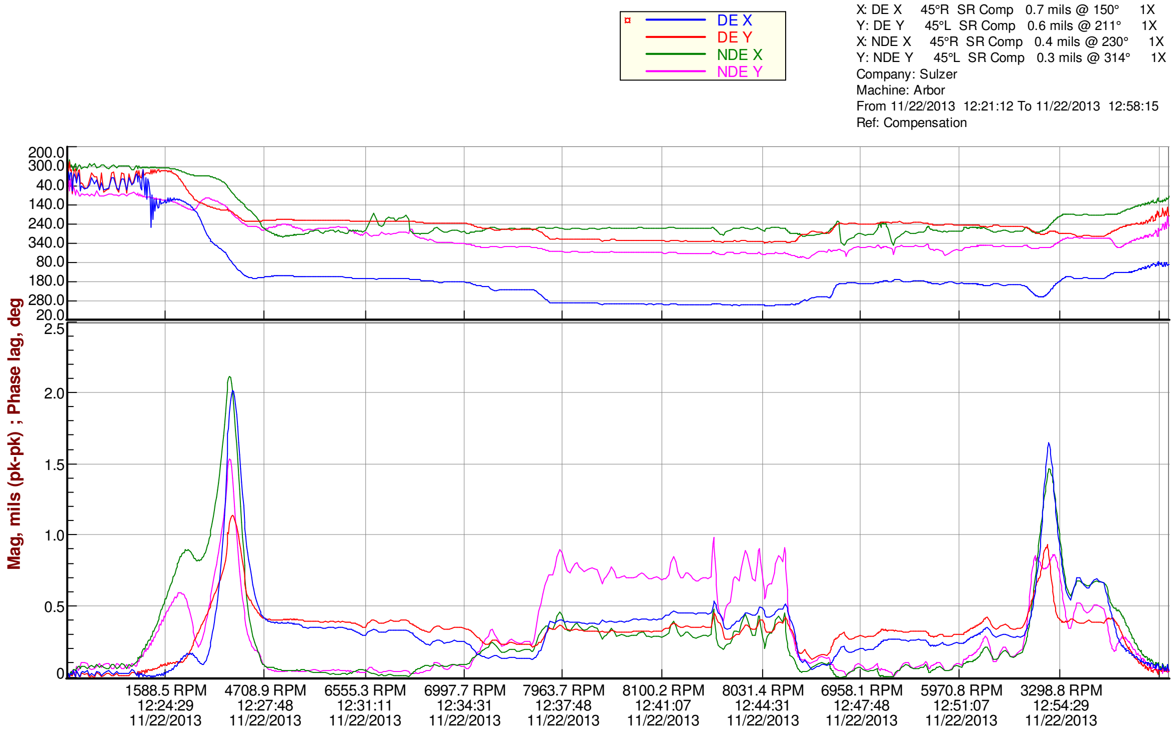

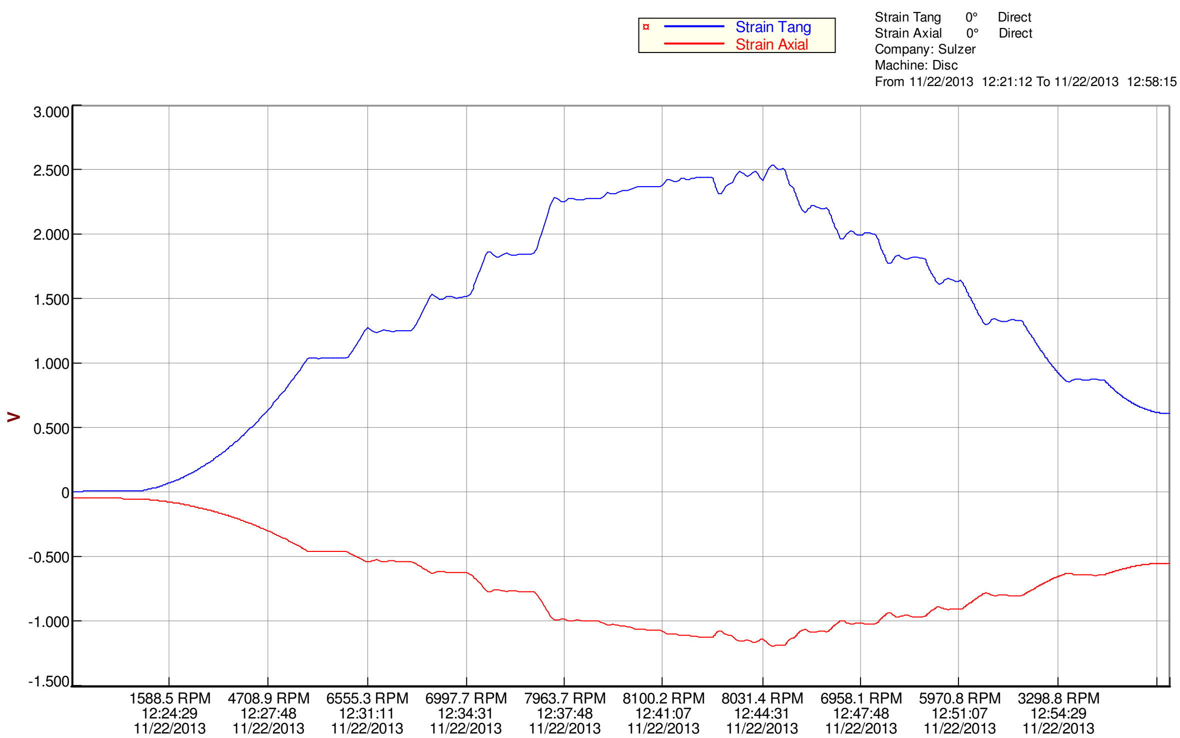

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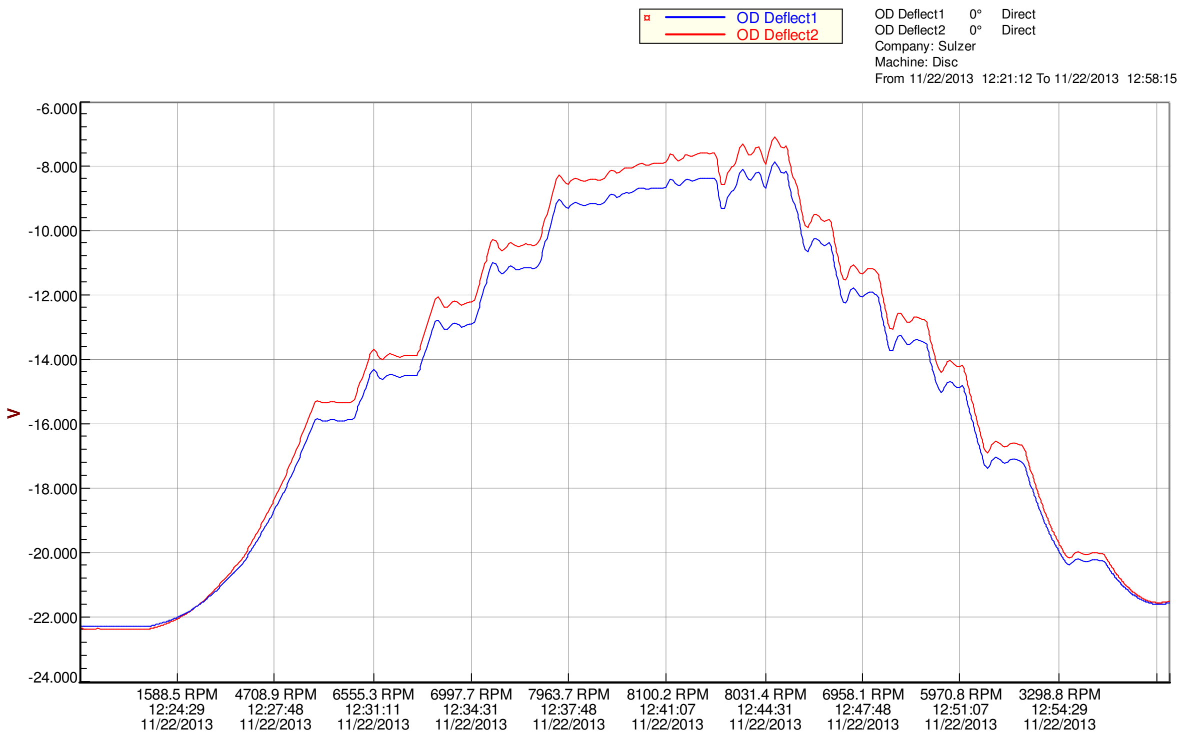
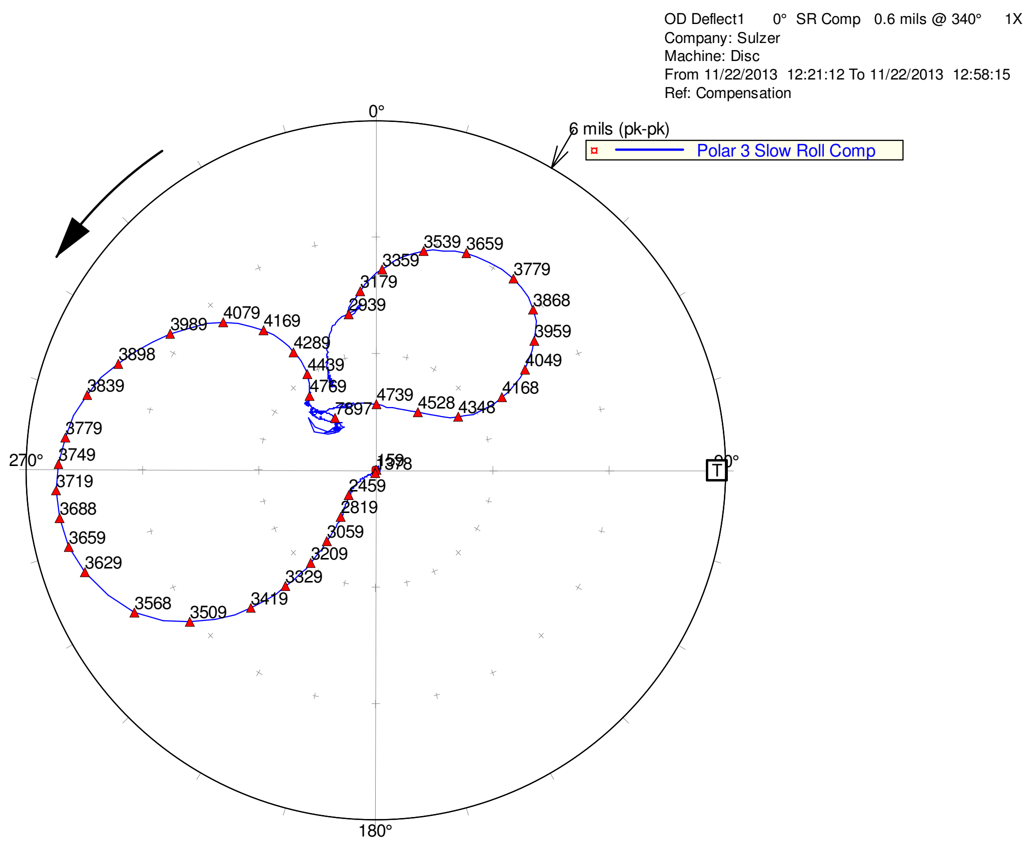
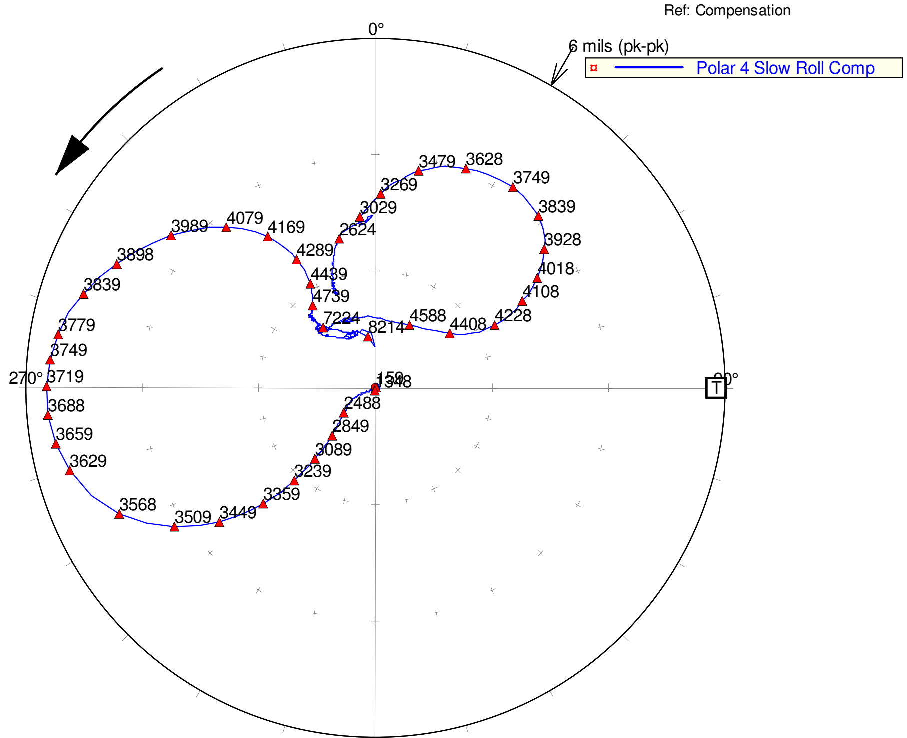
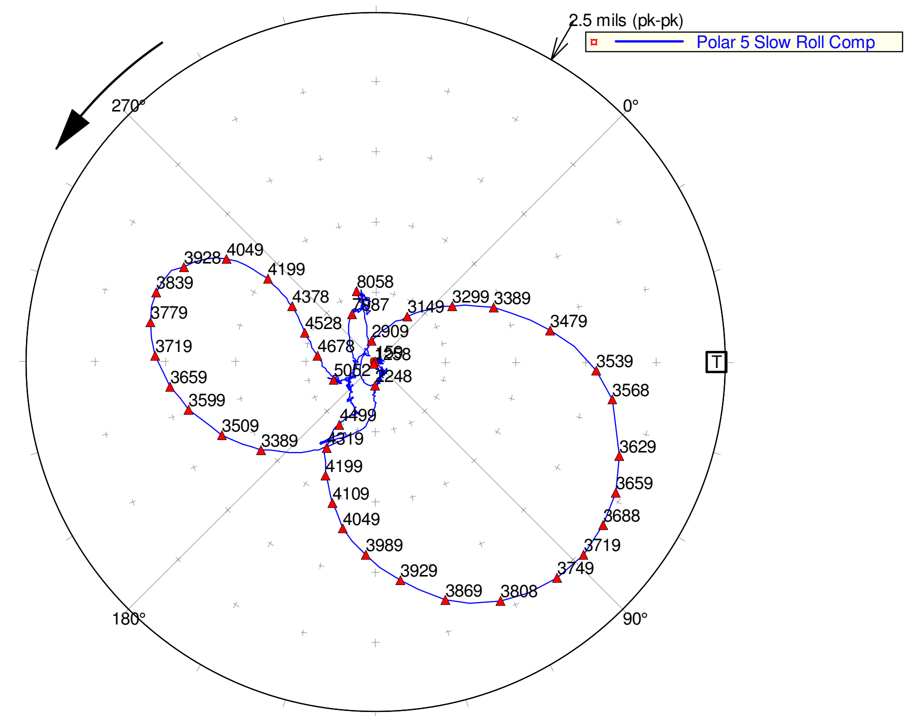




OD Deflect2 0° SR Comp 0.6 mils @ 345° 1X
Company: Sulzer
Machine: Disc
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation



X: DE X 45°R SR Comp 0.7 mils @ 150° 1X
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation



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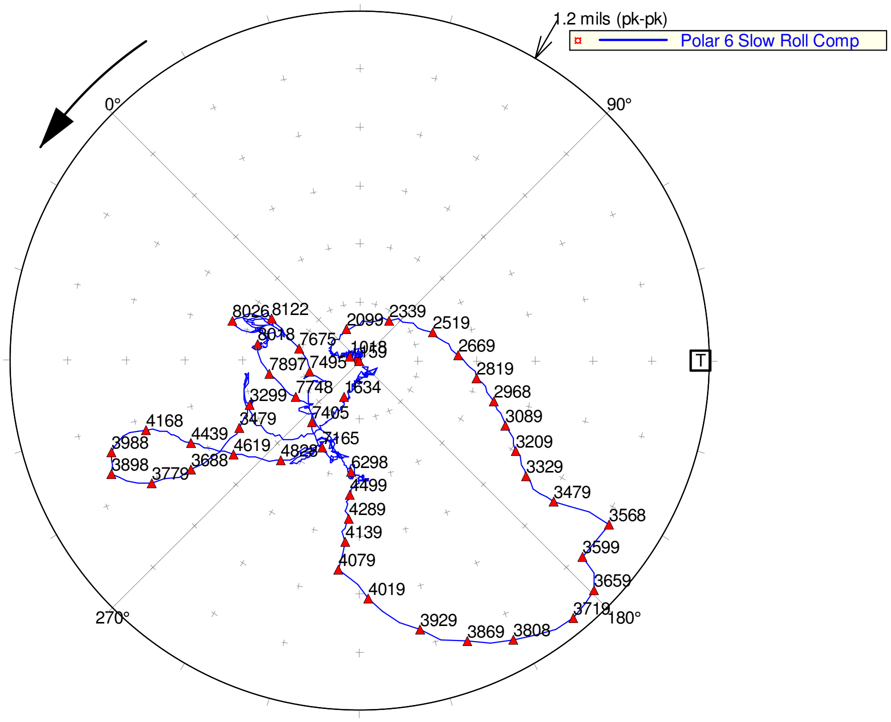
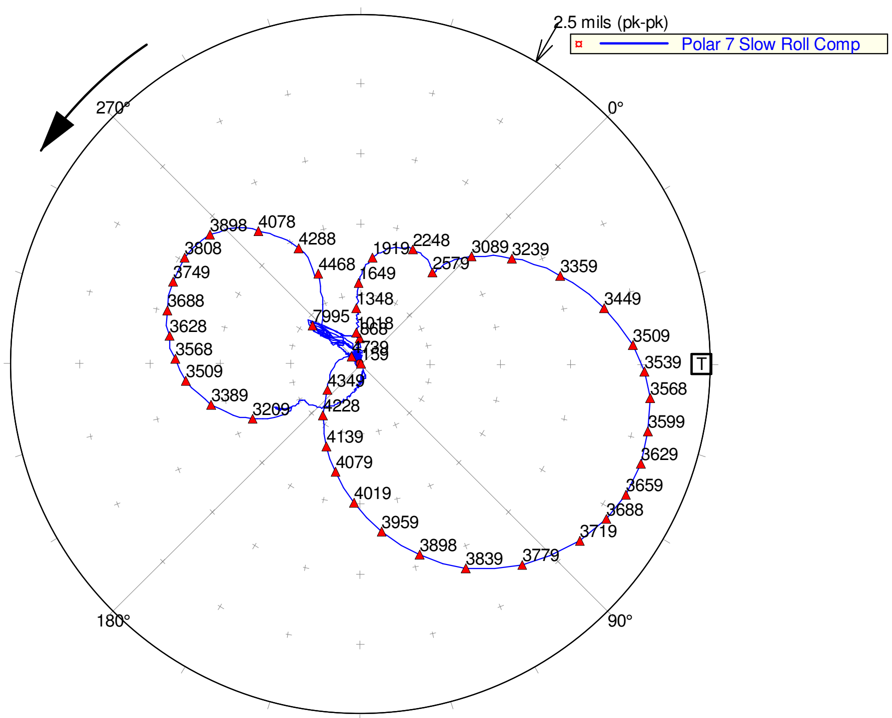
Y: DE Y 45°L SR Comp 0.6 mils @ 211° 1X
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation



X: NDE X 45°R SR Comp 0.4 mils @ 230° 1X
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation



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Dynamics Of Rotating Machinery
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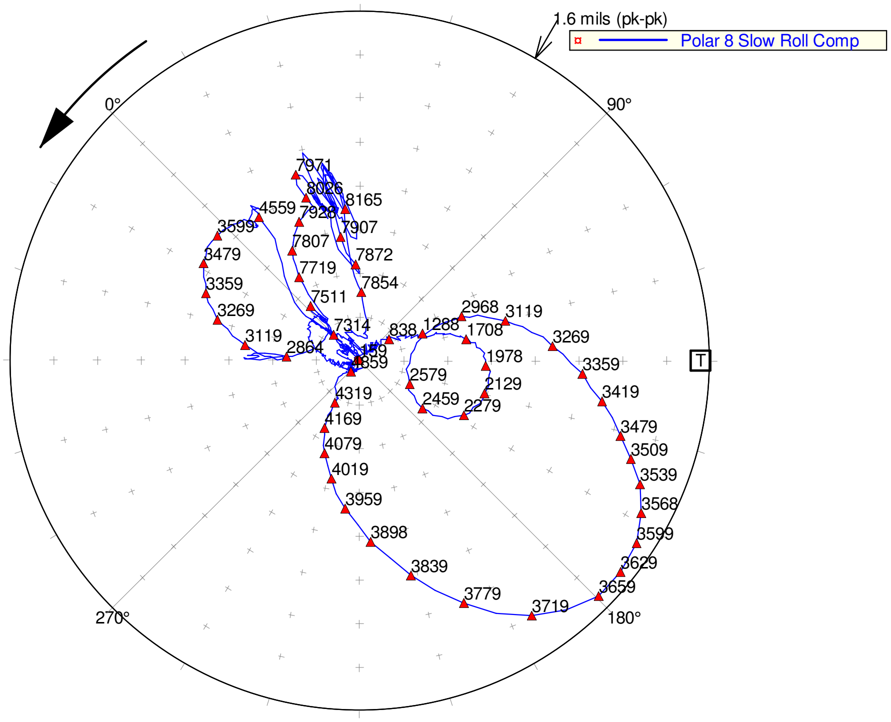
Tel (513) 231-1748 / (513) 633-5429 | Fax (513) 231-2104

Y: NDE Y 45°L SR Comp 0.3 mils @ 314° 1X
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation



OD Deflect1 0° SR Comp 0.6 mils @ 340° 1X
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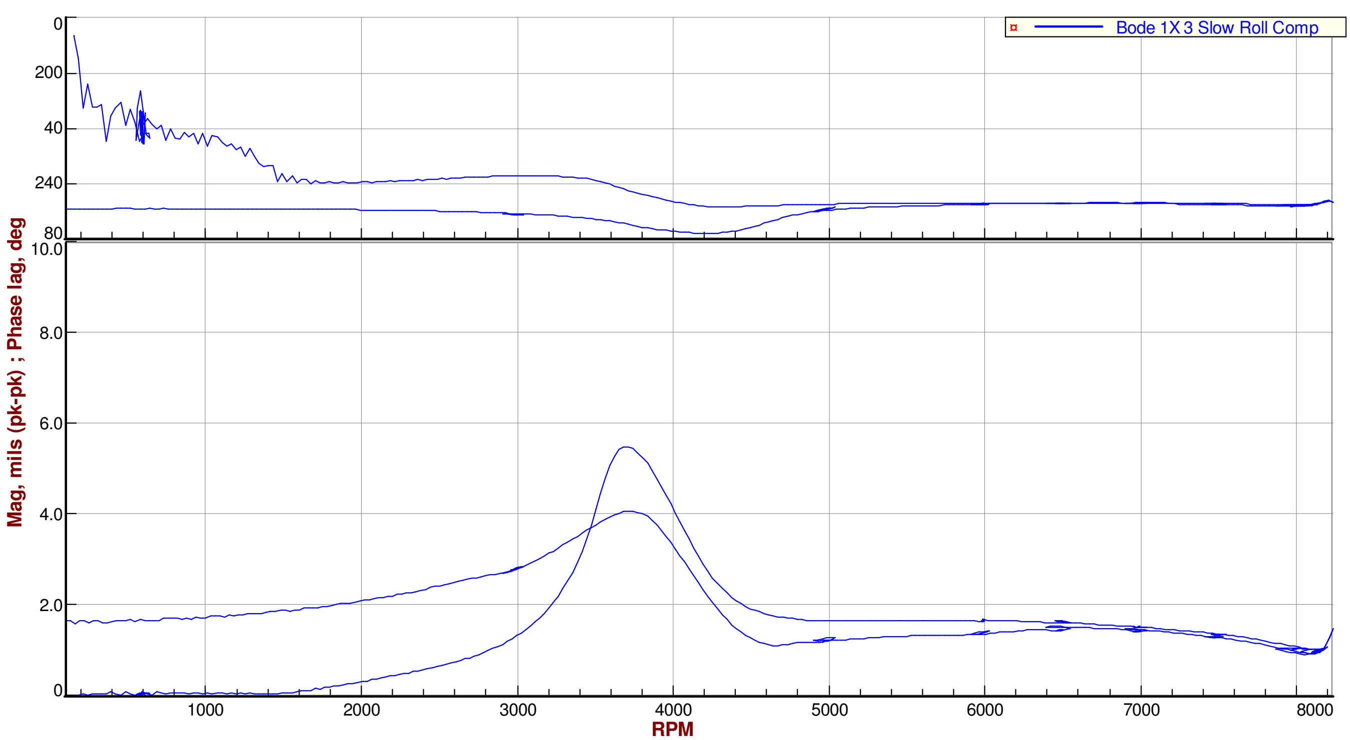
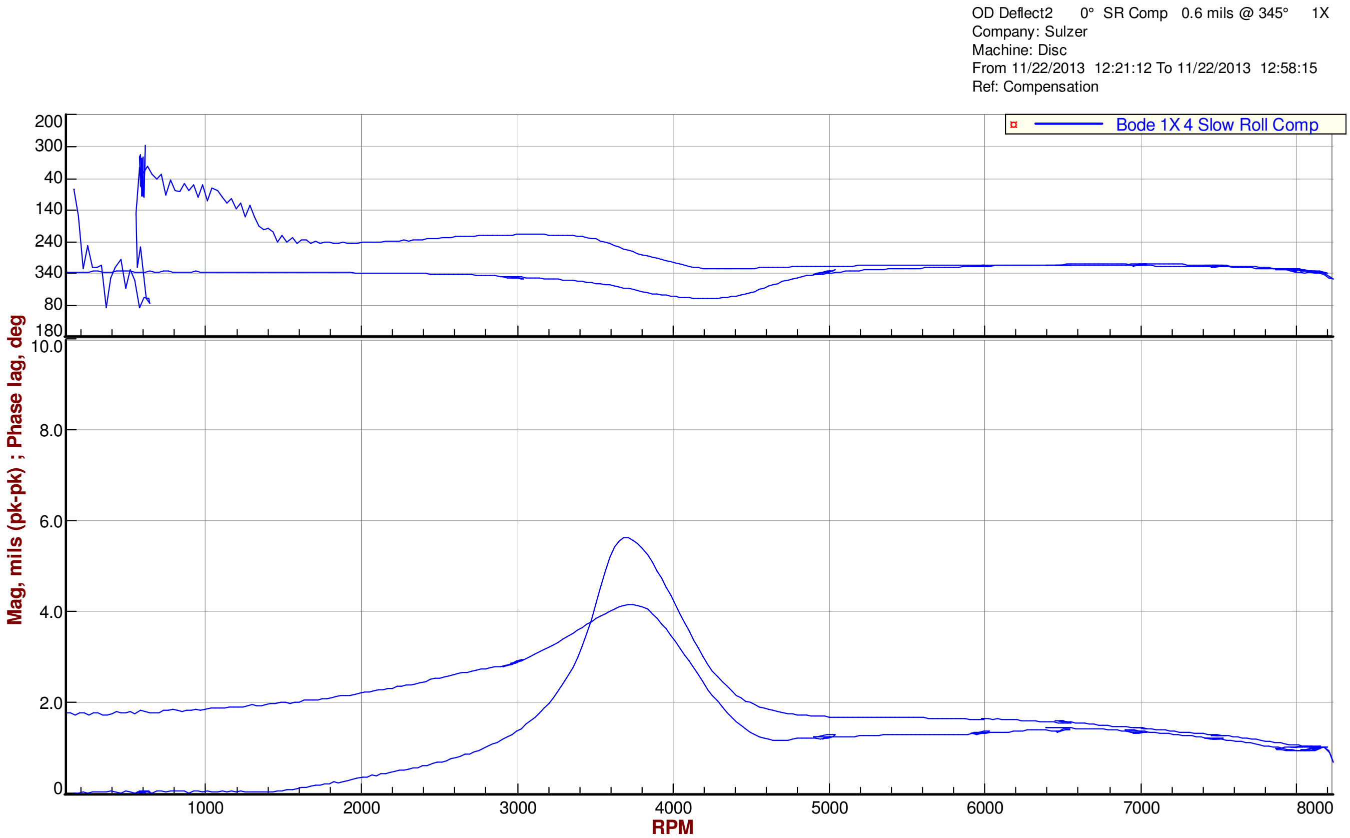
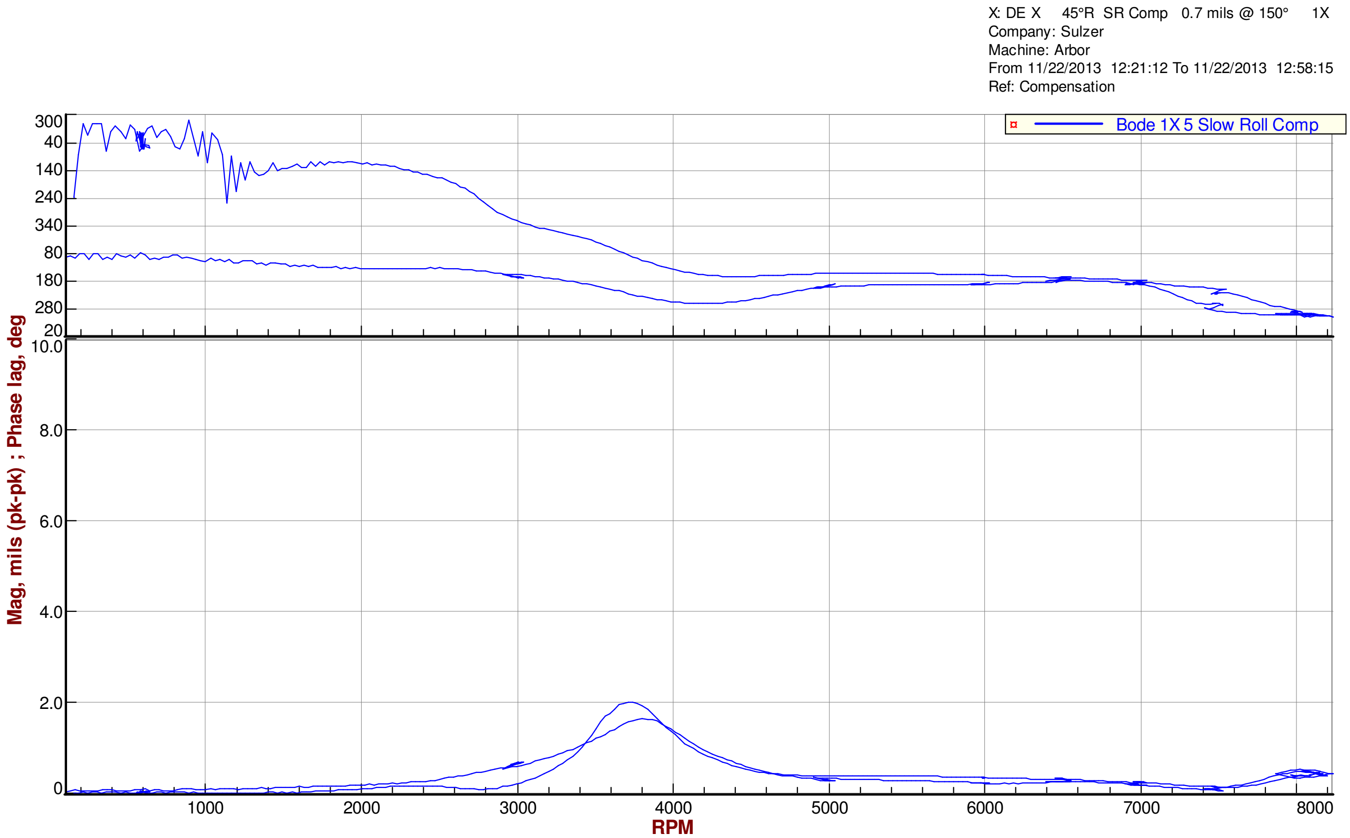
Machine: Disc
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation




OD Deflect2 0° SR Comp 0.6 mils @ 345° 1X
Company: Sulzer
Machine: Disc
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation
Bode 1X 4 Slow Roll Comp
Mag, mils (pk-pk) ; Phase lag, deg
RPM


X: DE X 45°R SR Comp 0.7 mils @ 150° 1X
Company: Sulzer
Machine: Arbor
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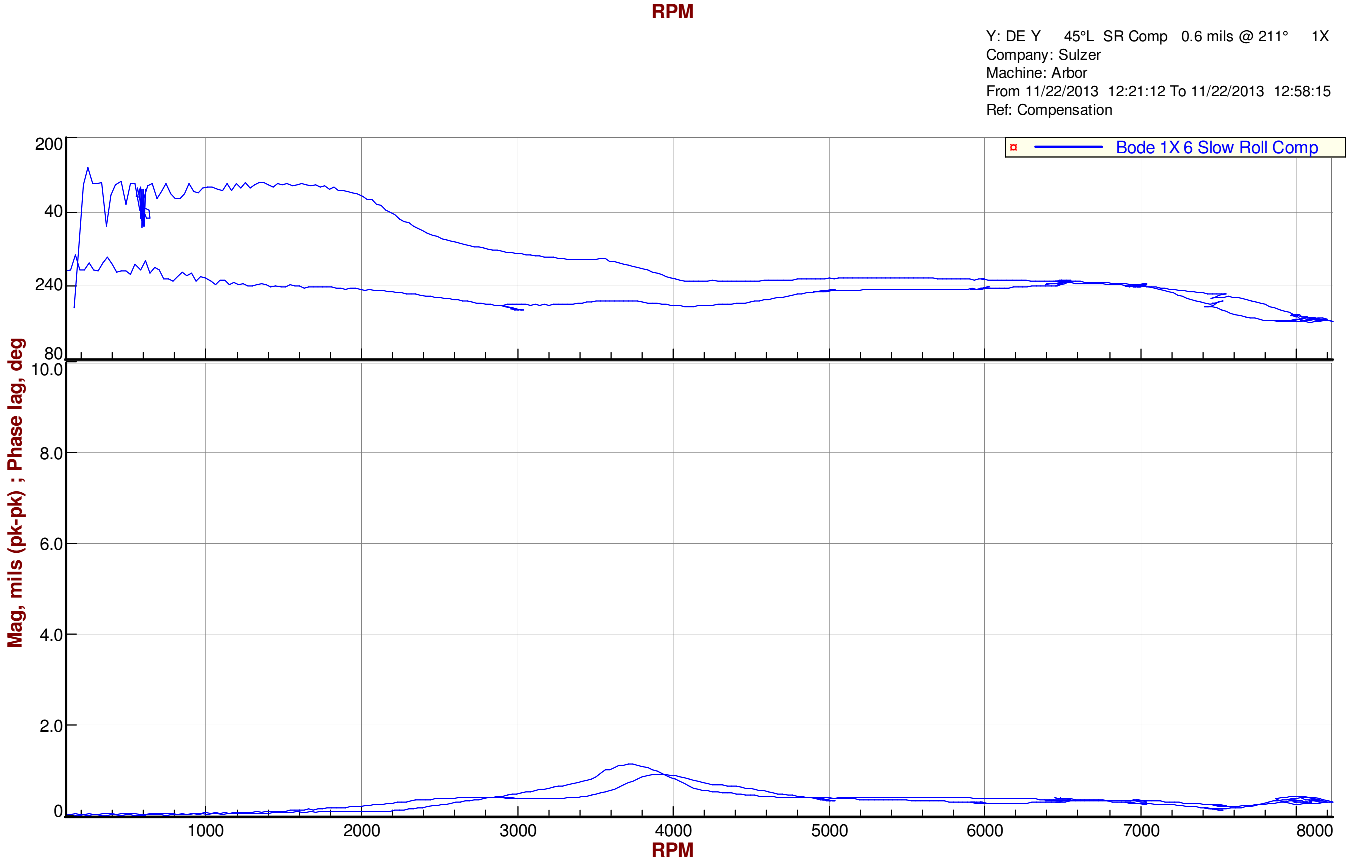
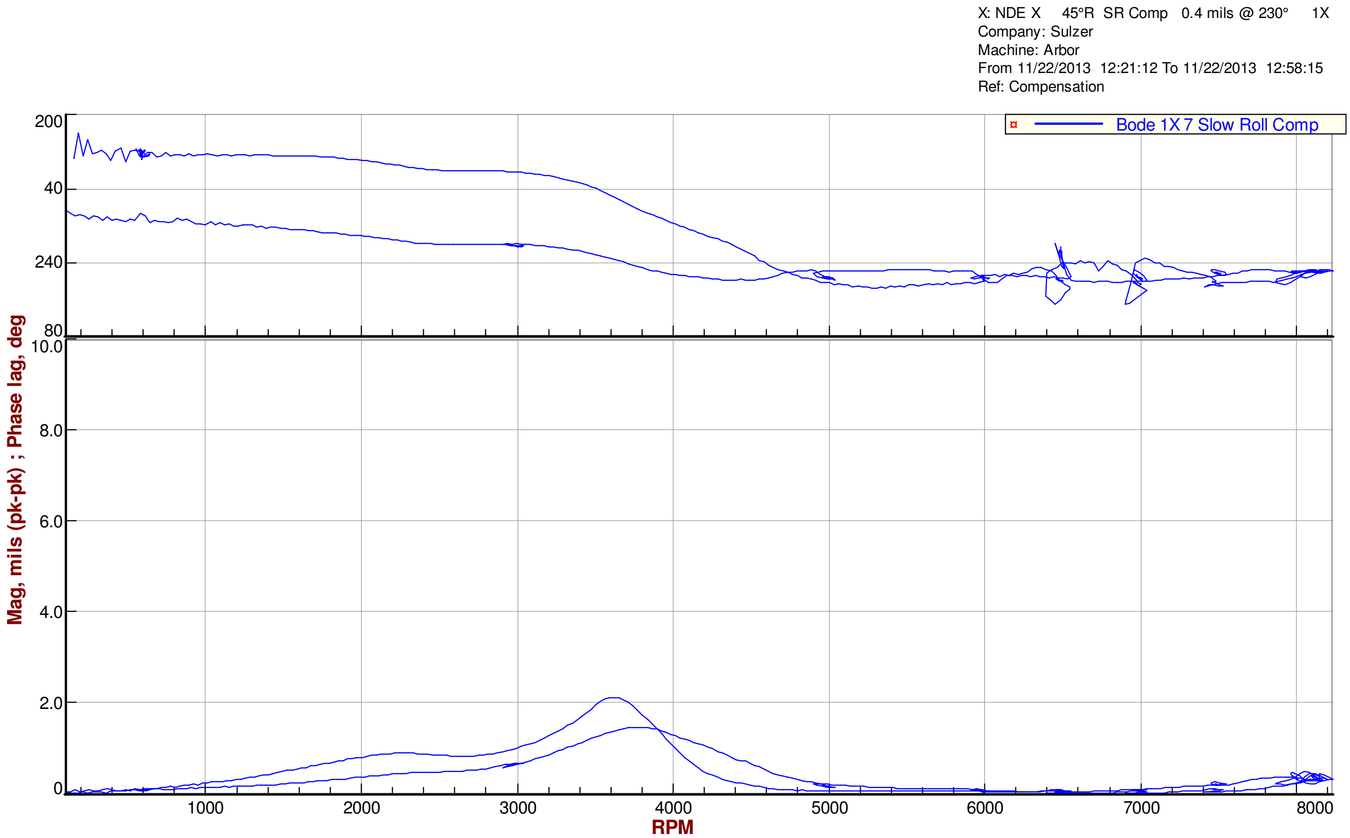
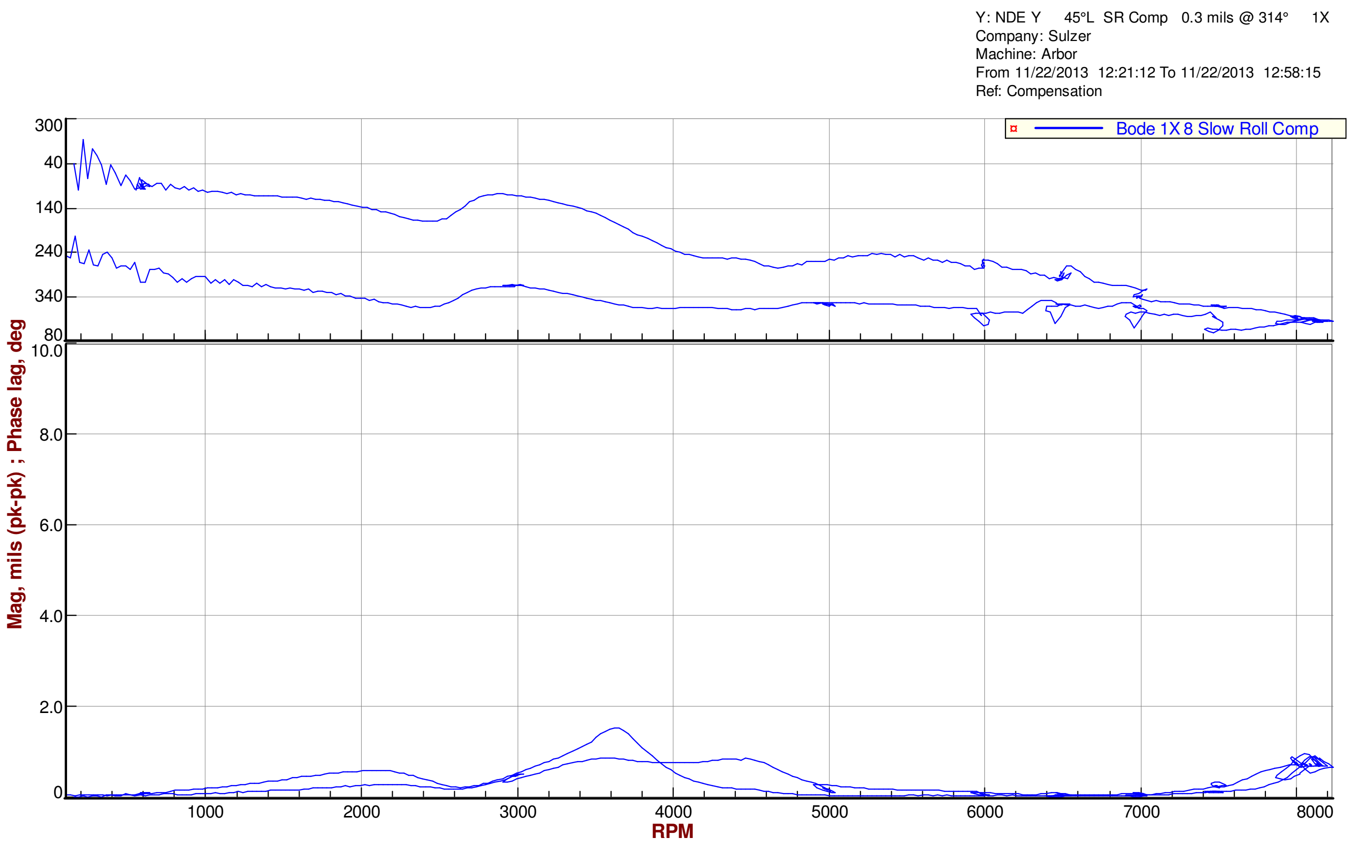
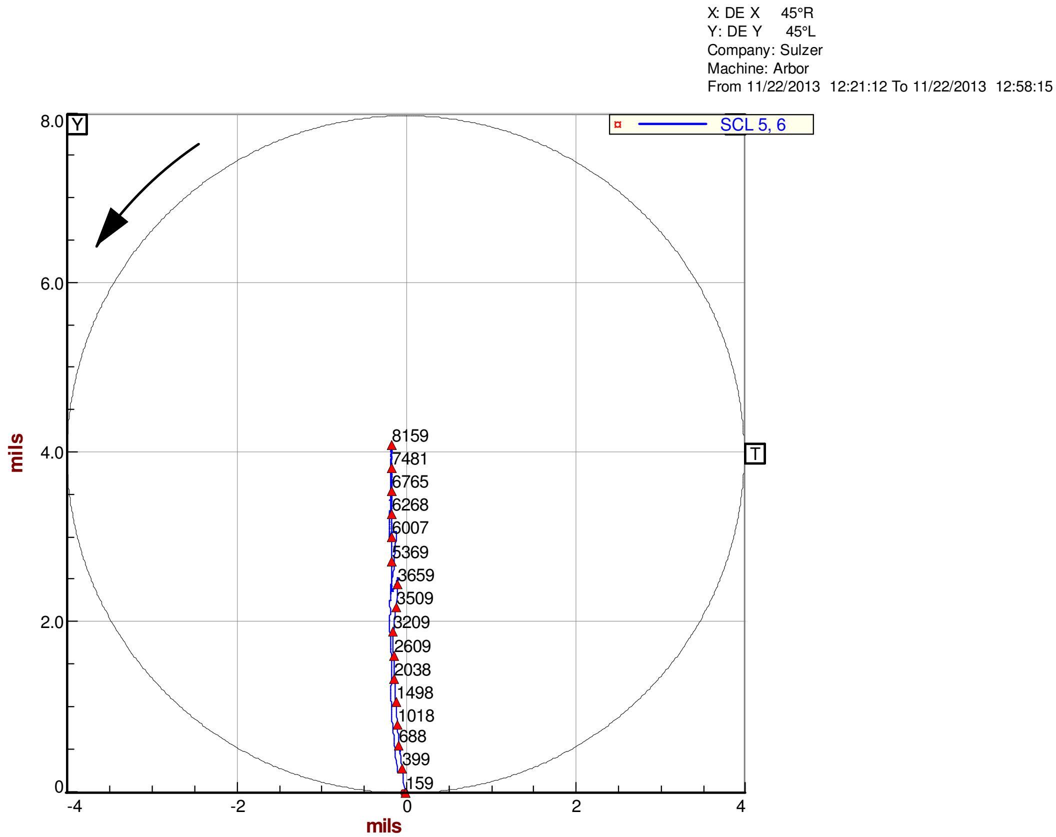
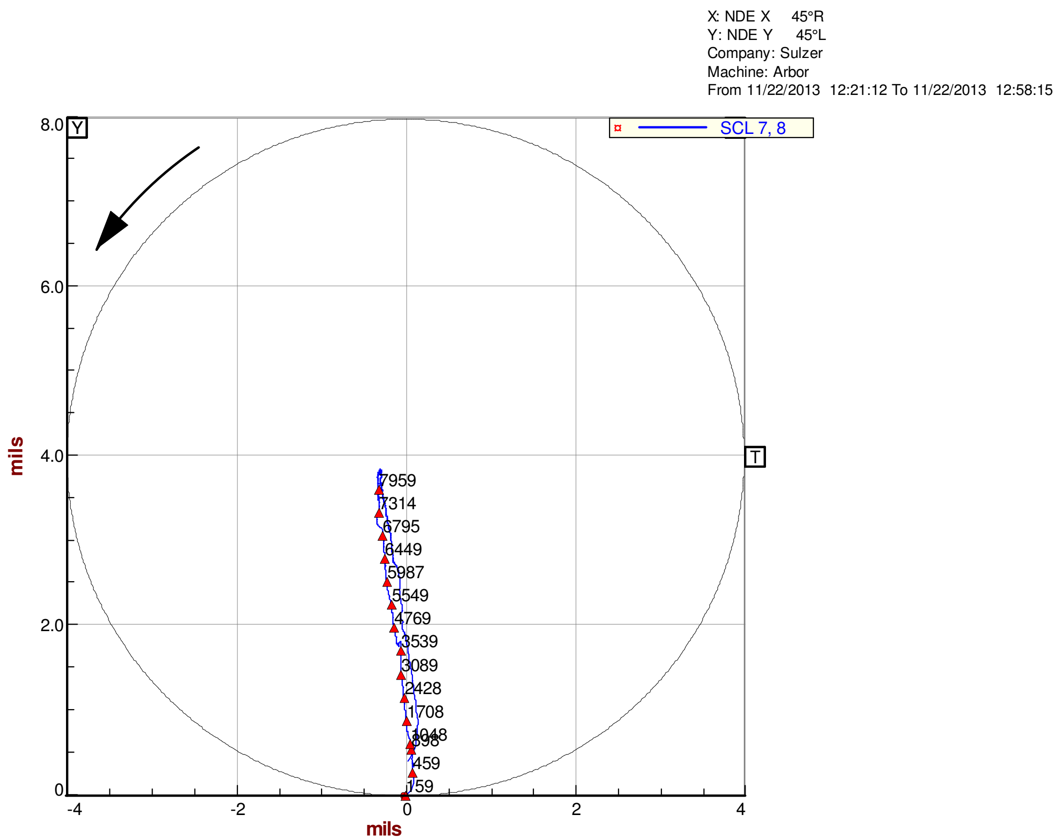
Ref: Compensation
Bode 1X 5 Slow Roll Comp
Mag, mils (pk-pk) ; Phase lag, deg
RPM


Y: DE Y 45°L SR Comp 0.6 mils @ 211° 1X
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation
Bode 1X 6 Slow Roll Comp
Mag, mils (pk-pk) ; Phase lag, deg
RPM


X: NDE X 45°R SR Comp 0.4 mils @ 230° 1X
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation
Bode 1X 7 Slow Roll Comp
Mag, mils (pk-pk) ; Phase lag, deg
RPM


Y: NDE Y 45°L SR Comp 0.3 mils @ 314° 1X
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
Ref: Compensation
Bode 1X 8 Slow Roll Comp
Mag, mils (pk-pk) ; Phase lag, deg
RPM


X: DE X 45°R
Y: DE Y 45°L
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
SCL 5, 6
Y
T
8.0
6.0
4.0
2.0
0
-4
-2
0
2
4
mils
mils
8159
7481
6765
6268
6007
5369
3659
3509
3209
2609
2038
1498
1018
688
399
159


X: NDE X 45°R
Y: NDE Y 45°L
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:12 To 11/22/2013 12:58:15
SCL 7, 8
Y
T
8.0
6.0
4.0
2.0
0
-4
-2
0
2
4
mils
mils
7959
7314
6795
6449
5987
5549
4769
3539
3089
2428
1708
459
159

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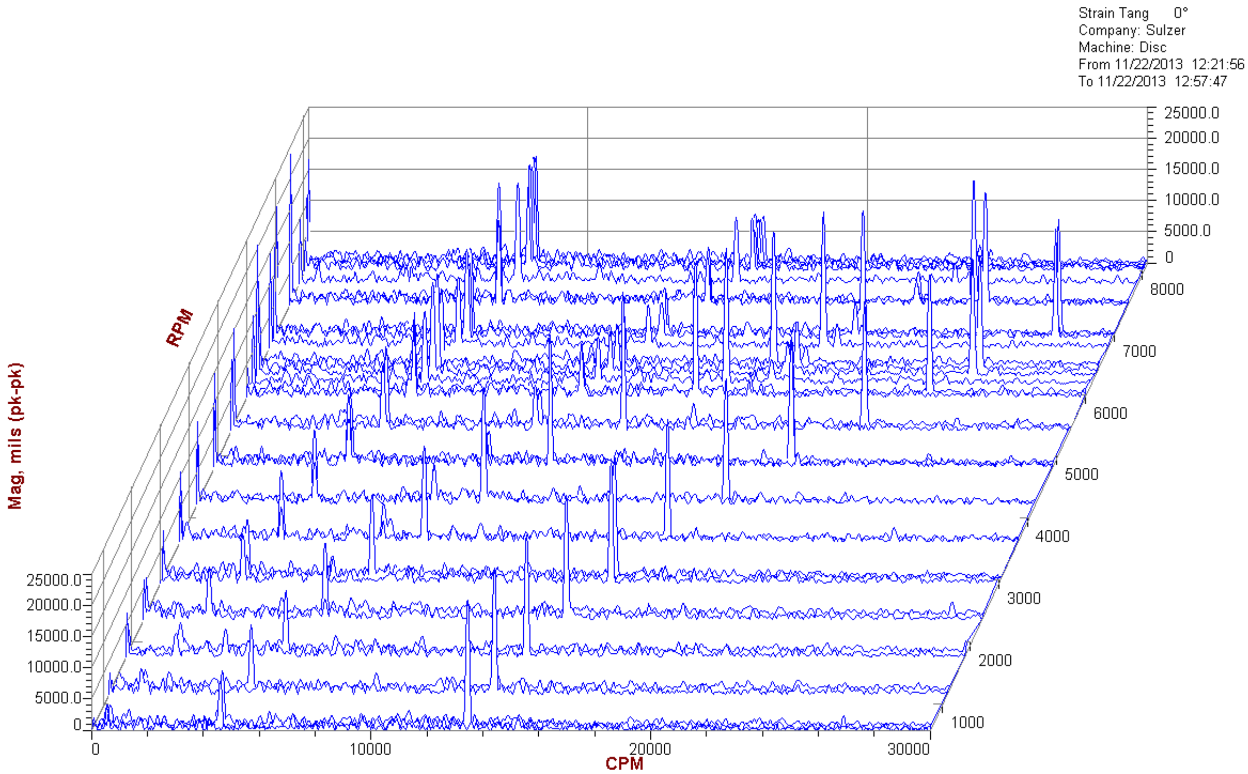
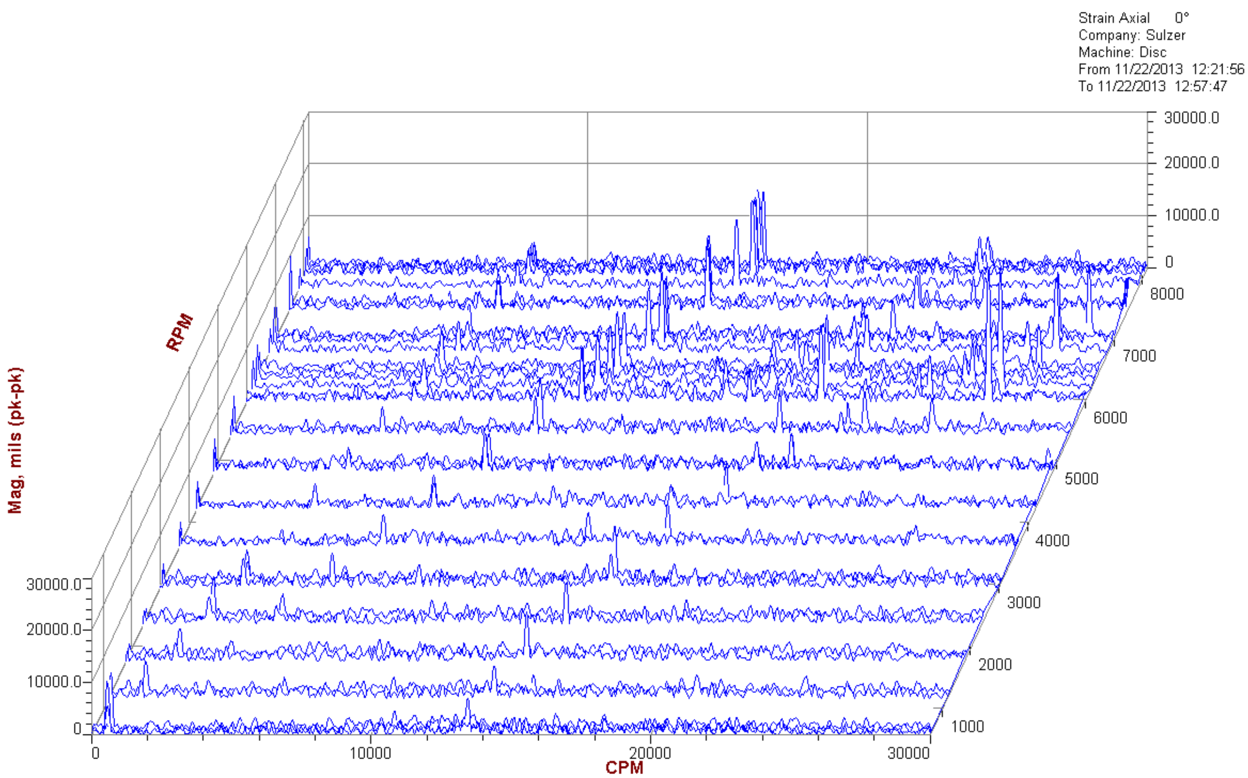
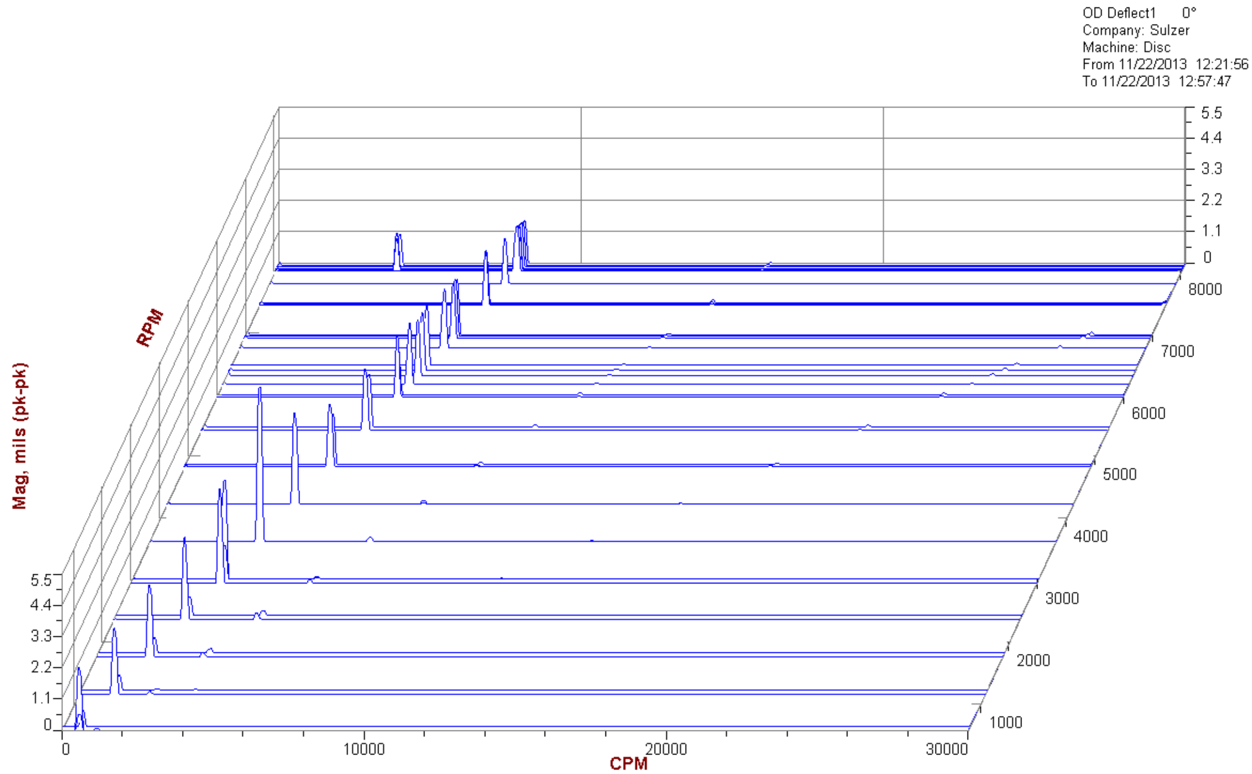
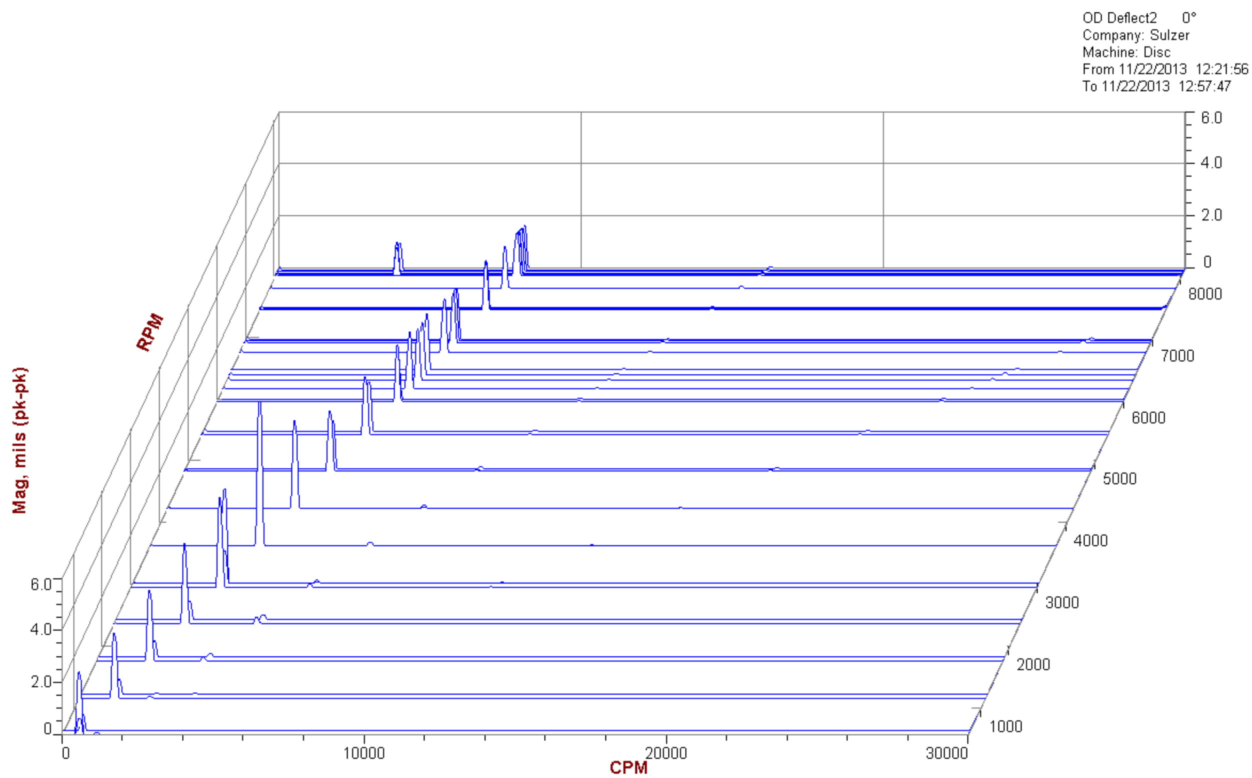





OD Deflect1 0°
Company: Sulzer
Machine: Disc
From 11/22/2013 12:21:56
To 11/22/2013 12:57:47
Mag, mils (pk-pk)
RPM
CPM
0
10000
20000
30000
1000
2000
3000
4000
5000
6000
7000
8000
0
1.1
2.2
3.3
4.4
5.5


OD Deflect2 0°
Company: Sulzer
Machine: Disc
From 11/22/2013 12:21:56
To 11/22/2013 12:57:47
Mag, mils (pk-pk)
RPM
CPM
0
10000
20000
30000
1000
2000
3000
4000
5000
6000
7000
8000
0
2.0
4.0
6.0

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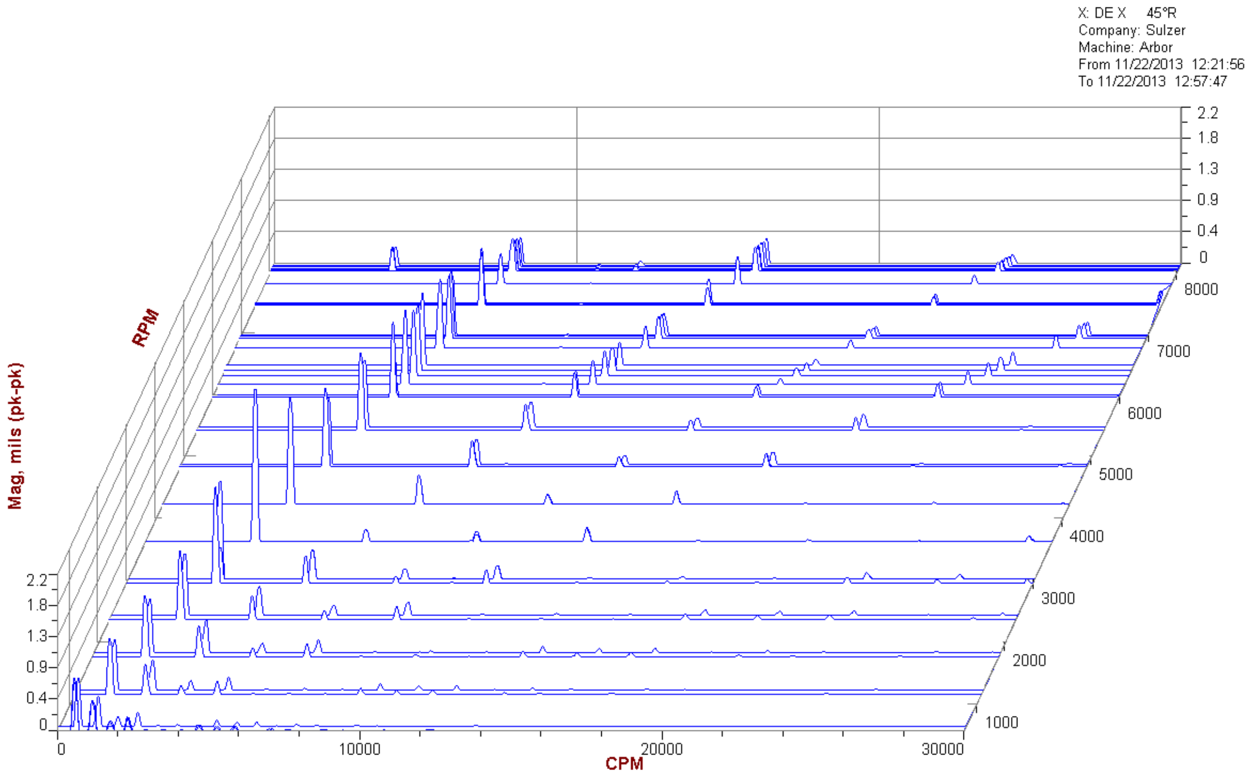

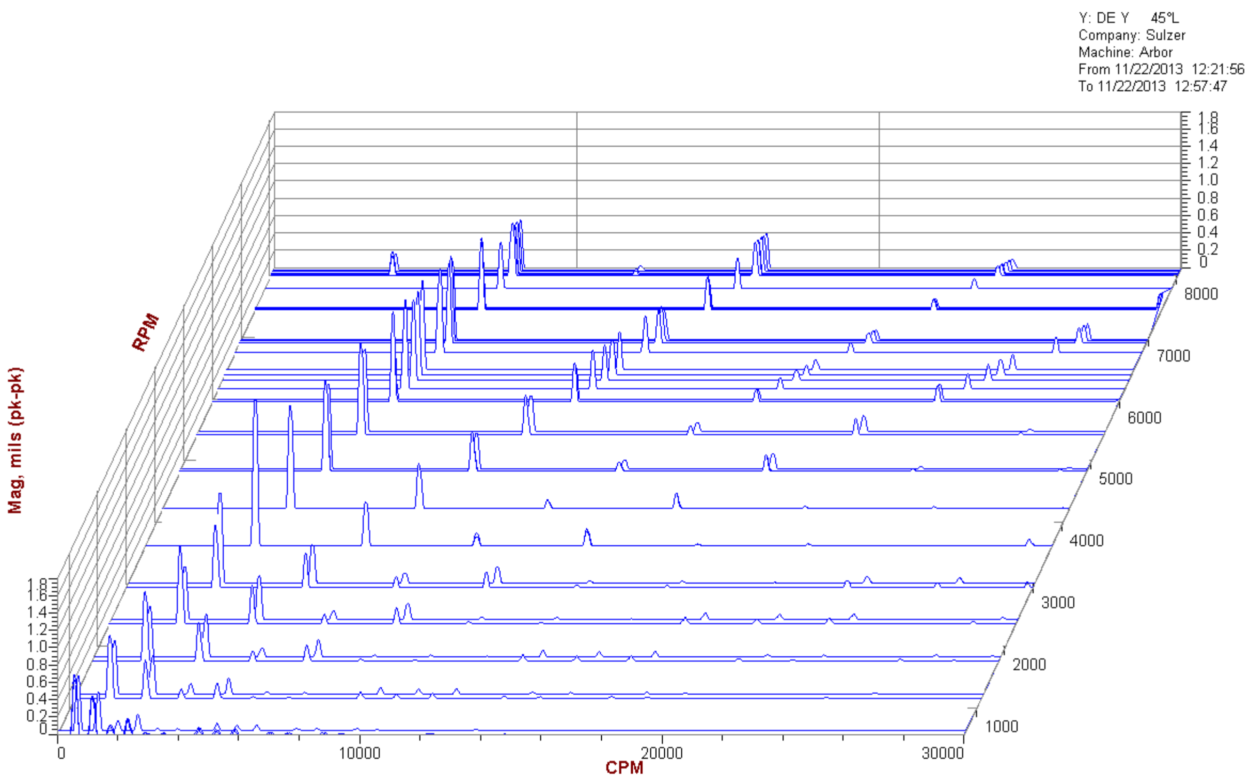

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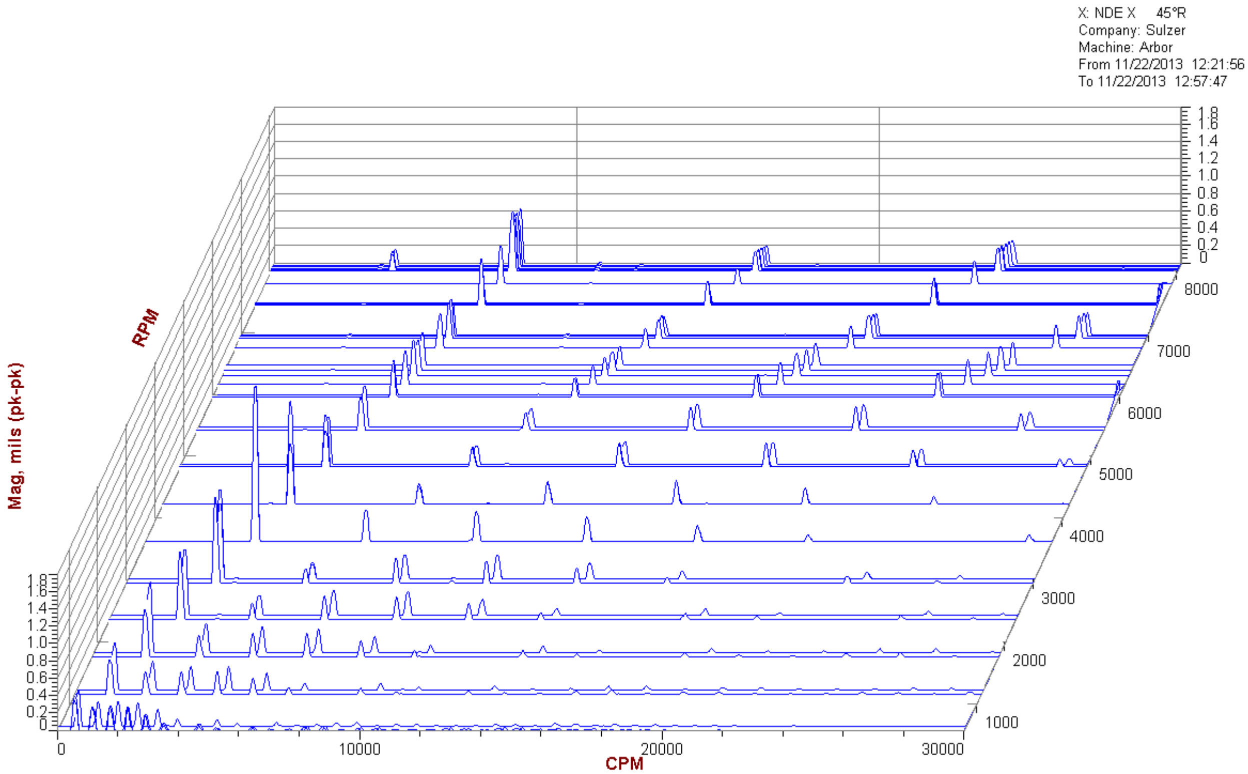

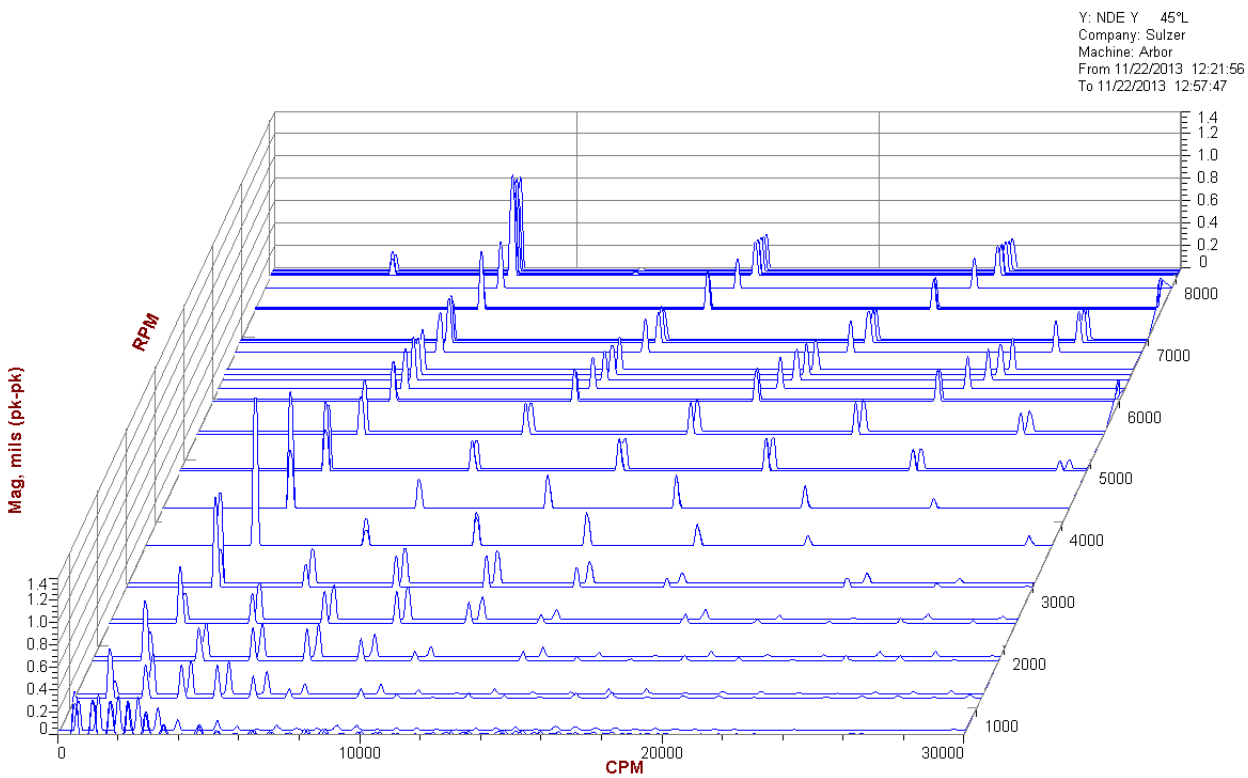

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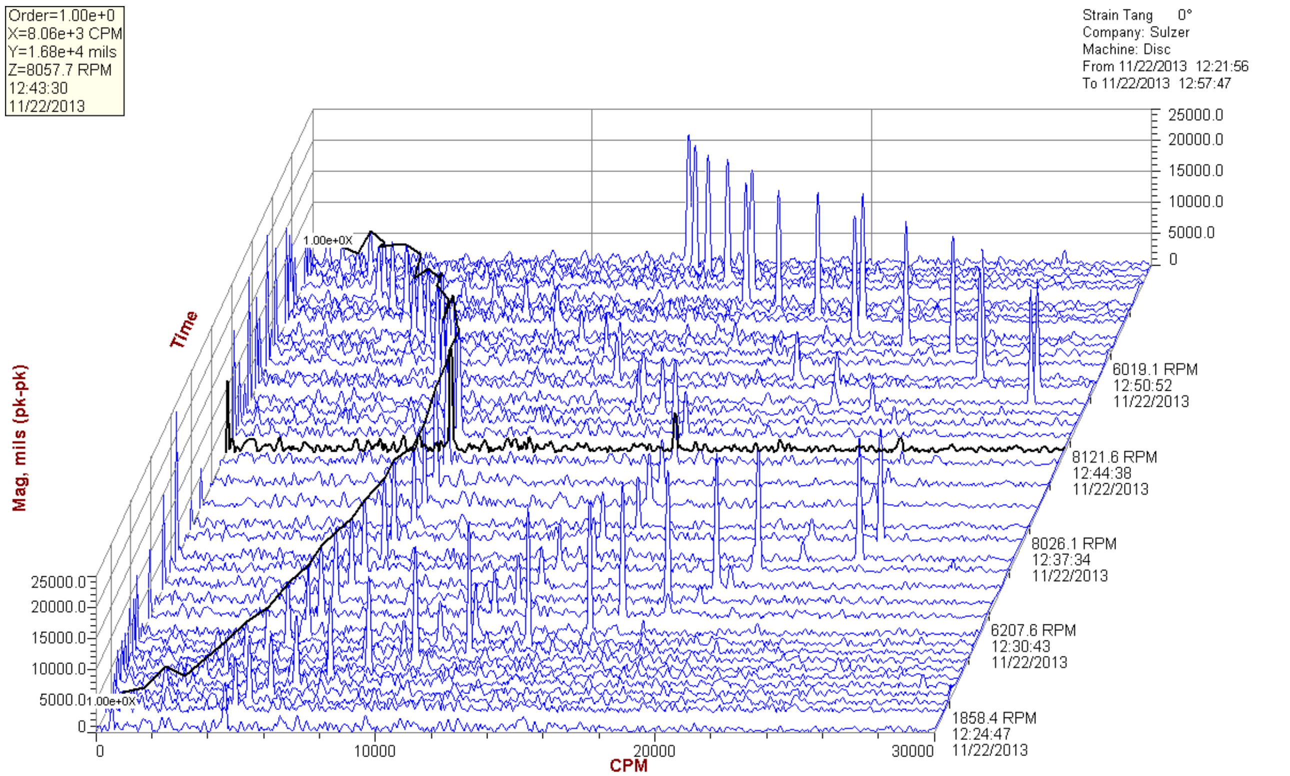

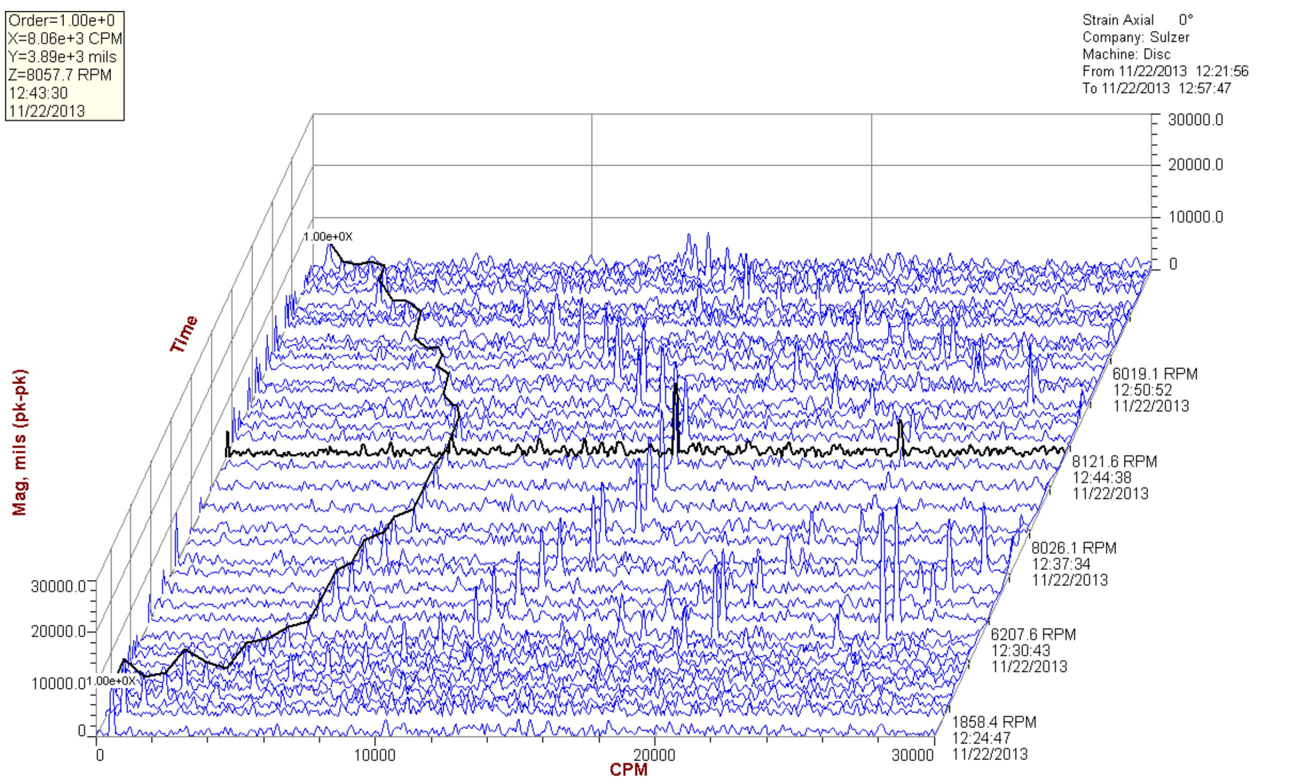

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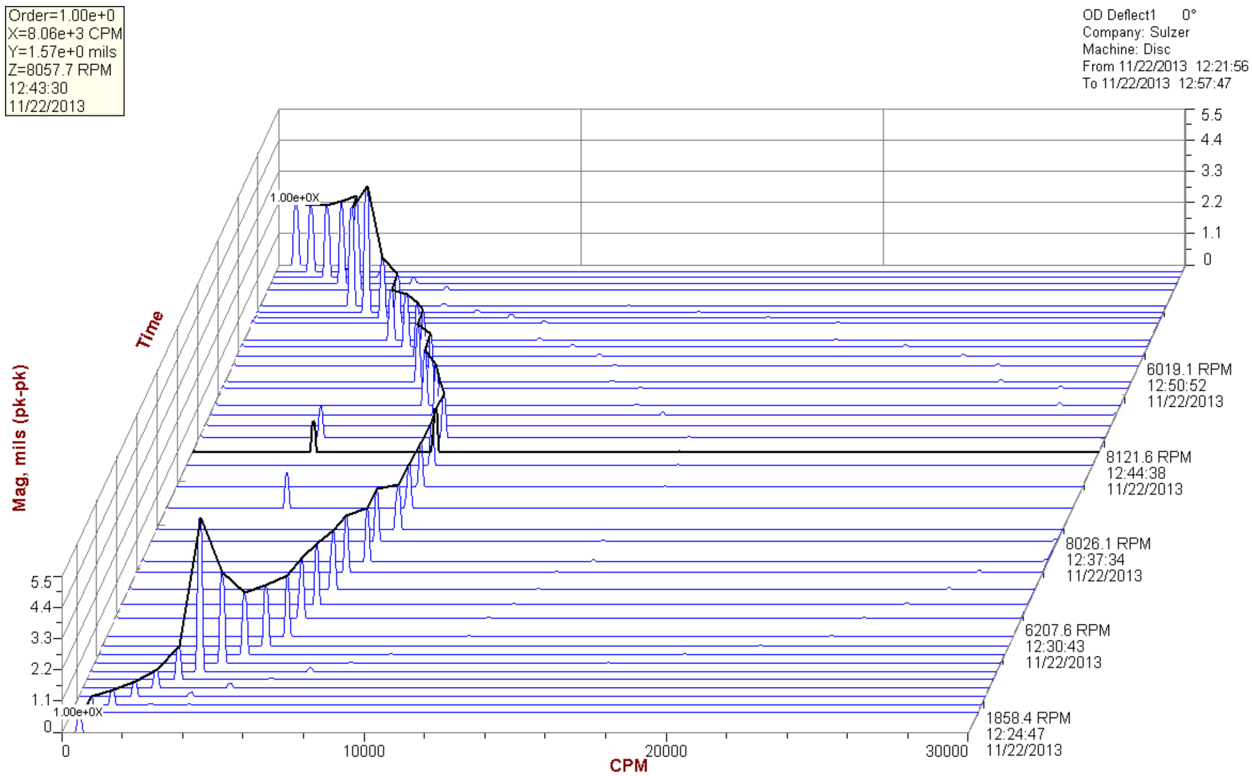
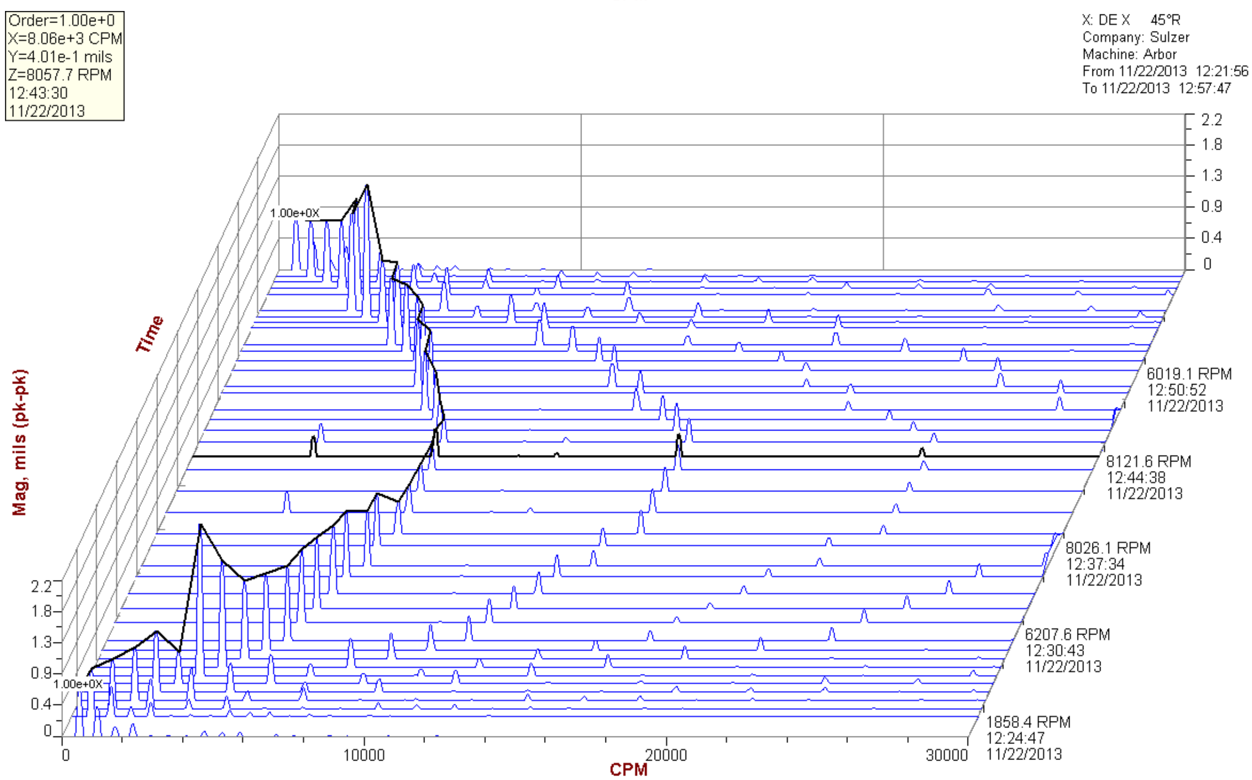
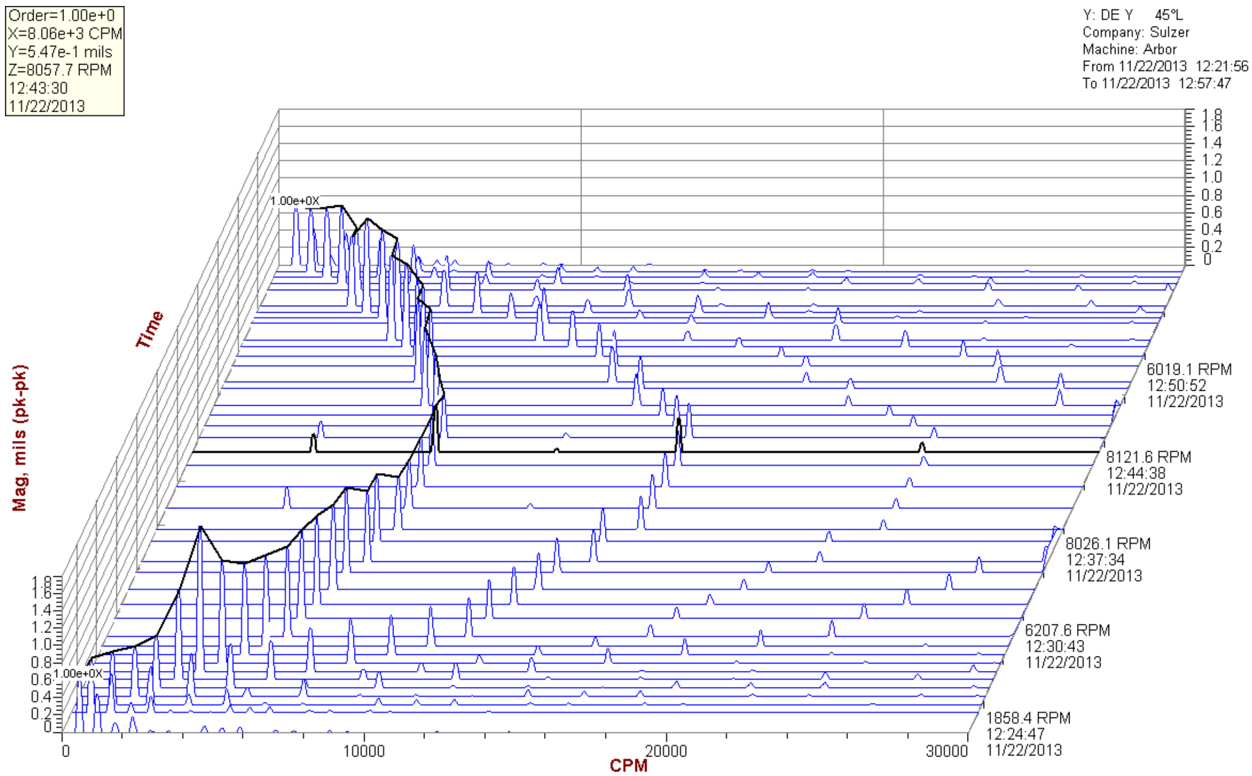
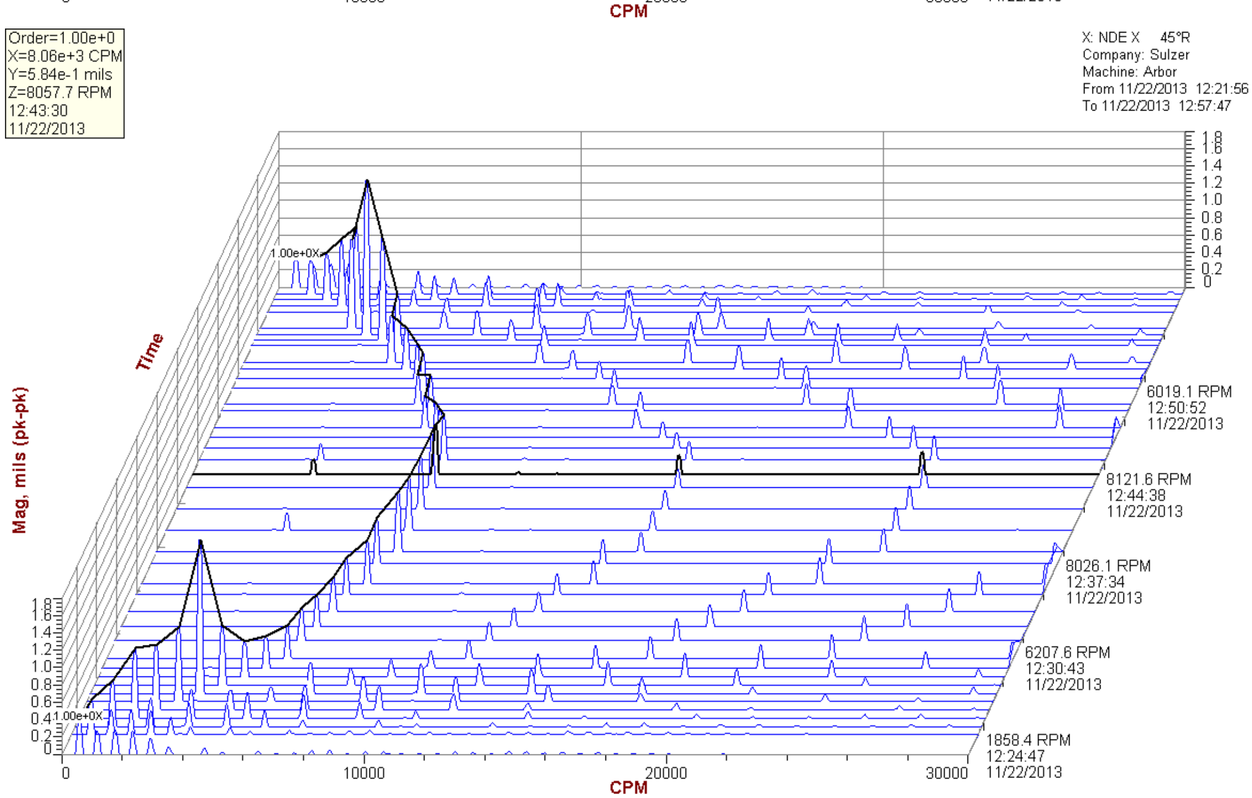





Order=1.00e+0
X=8.06e+3 CPM
Y=5.47e-1 mils
Z=8057.7 RPM
12:43:30
11/22/2013
Y: DE Y 45°L
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:56
To 11/22/2013 12:57:47
Time
Mag, mils (pk-pk)
1.00e+0X
6019.1 RPM 12:50:52 11/22/2013
8121.6 RPM 12:44:38 11/22/2013
8026.1 RPM 12:37:34 11/22/2013
6207.6 RPM 12:30:43 11/22/2013
1858.4 RPM 12:24:47 11/22/2013
0
10000
20000
30000
CPM


Order=1.00e+0
X=8.06e+3 CPM
Y=5.84e-1 mils
Z=8057.7 RPM
12:43:30
11/22/2013
X: NDE X 45°R
Company: Sulzer
Machine: Arbor
From 11/22/2013 12:21:56
To 11/22/2013 12:57:47
Time
Mag, mils (pk-pk)
1.00e+0X
6019.1 RPM 12:50:52 11/22/2013
8121.6 RPM 12:44:38 11/22/2013
8026.1 RPM 12:37:34 11/22/2013
6207.6 RPM 12:30:43 11/22/2013
1858.4 RPM 12:24:47 11/22/2013
0
10000
20000
30000
CPM

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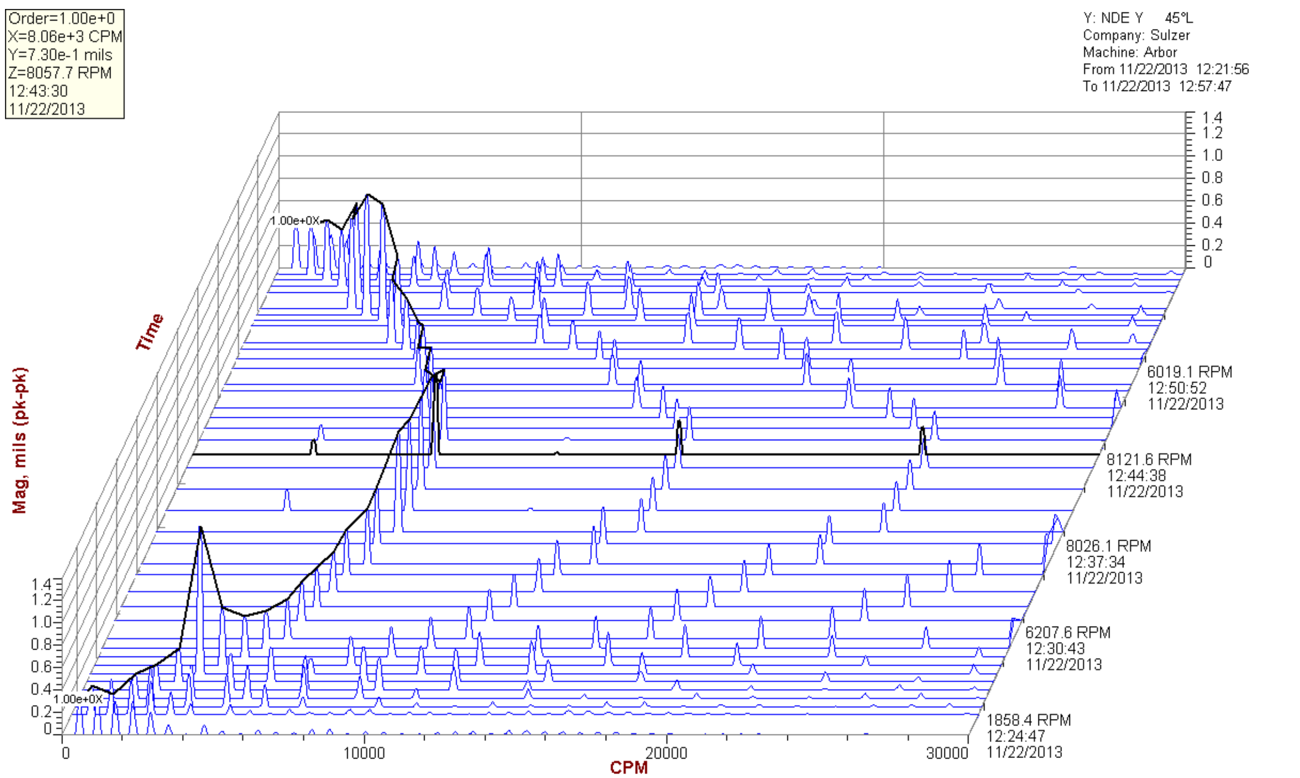
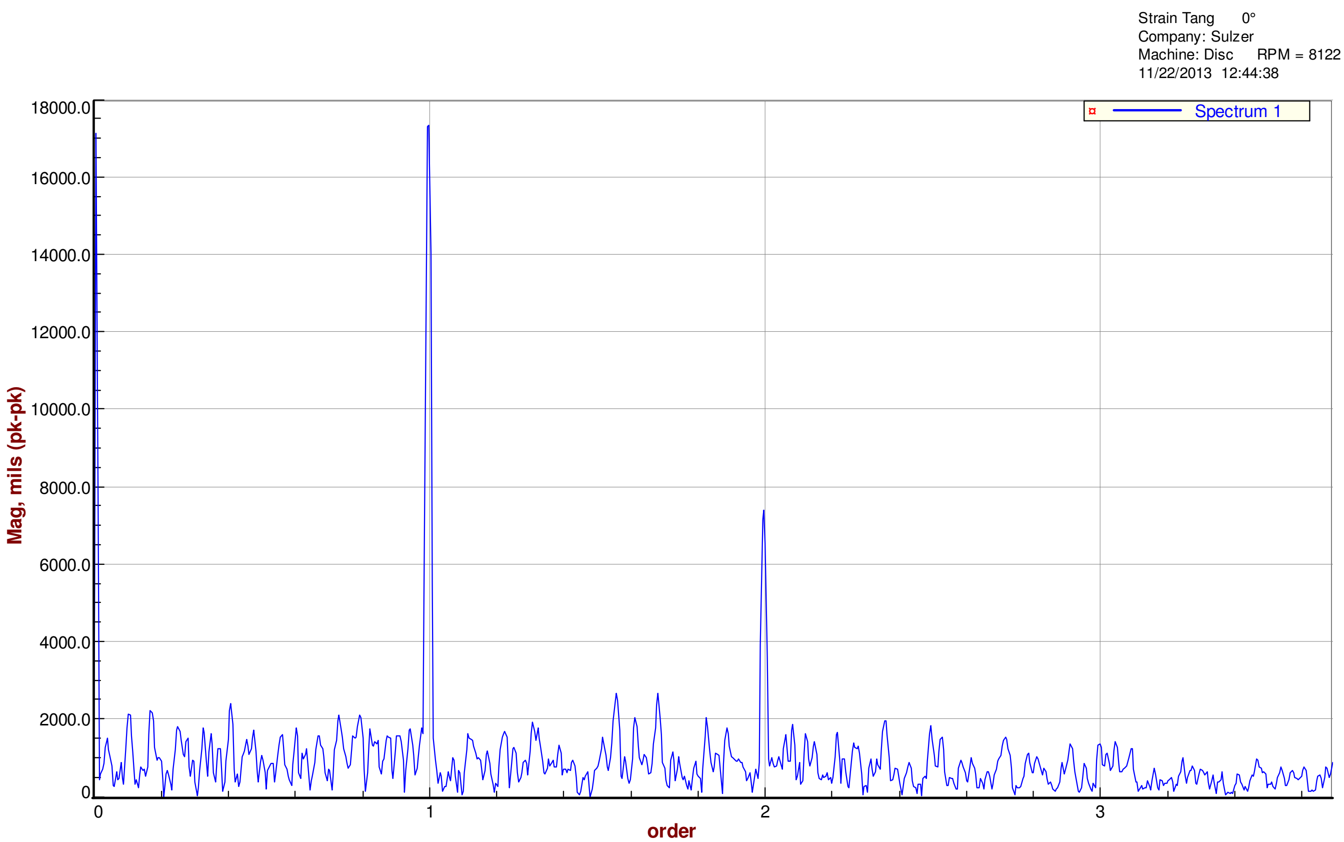
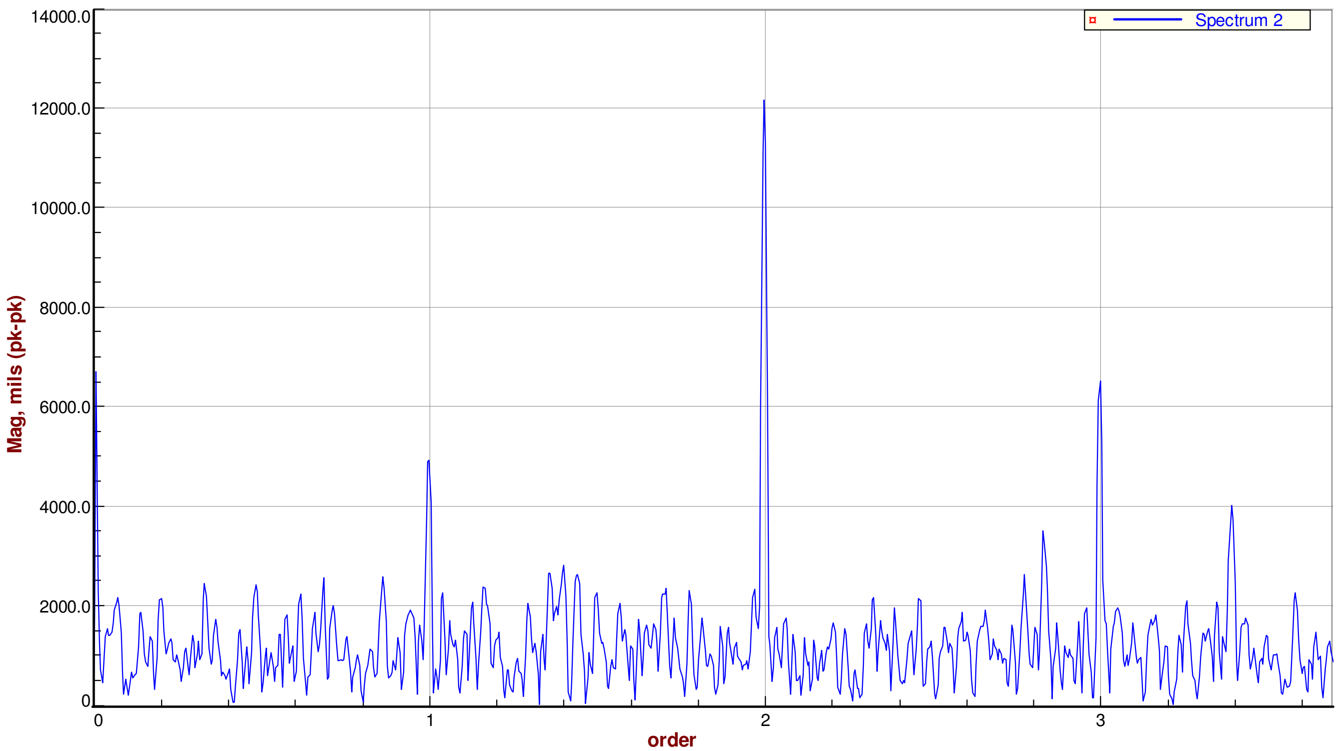
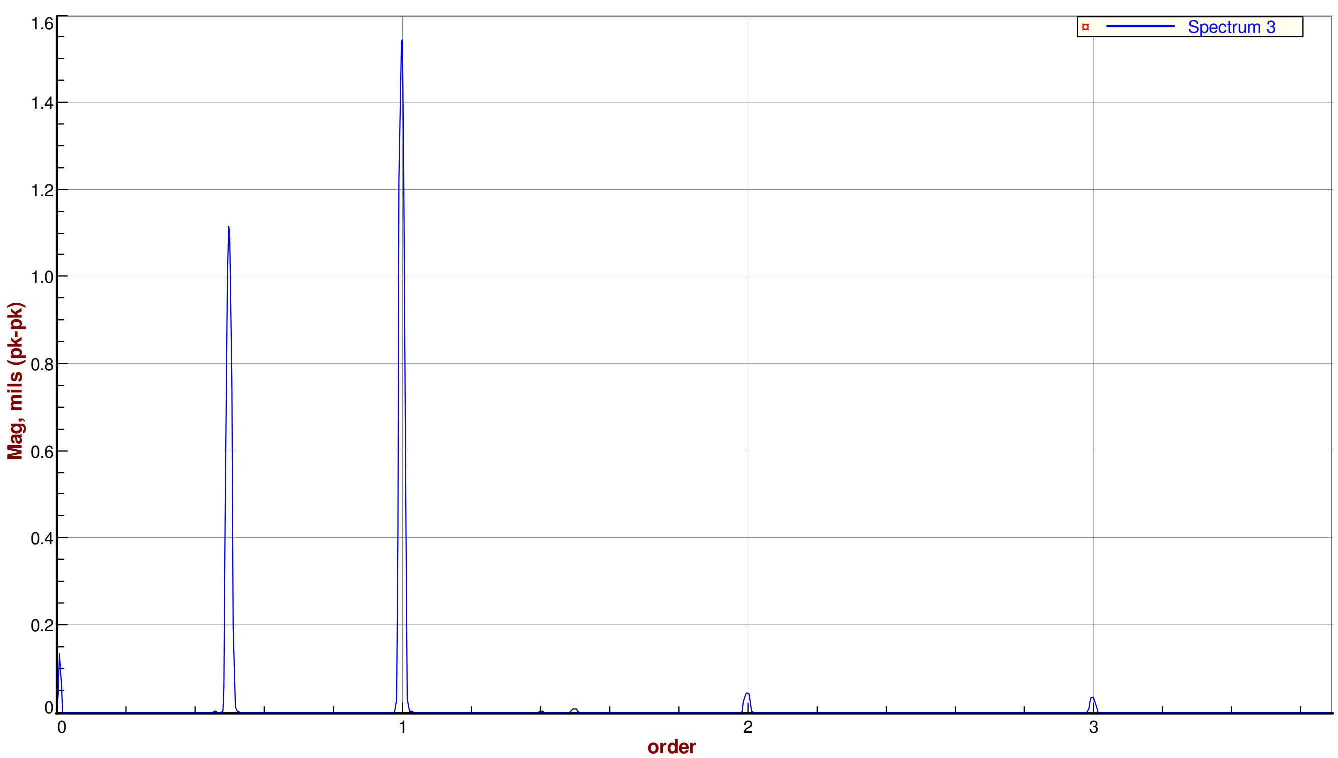


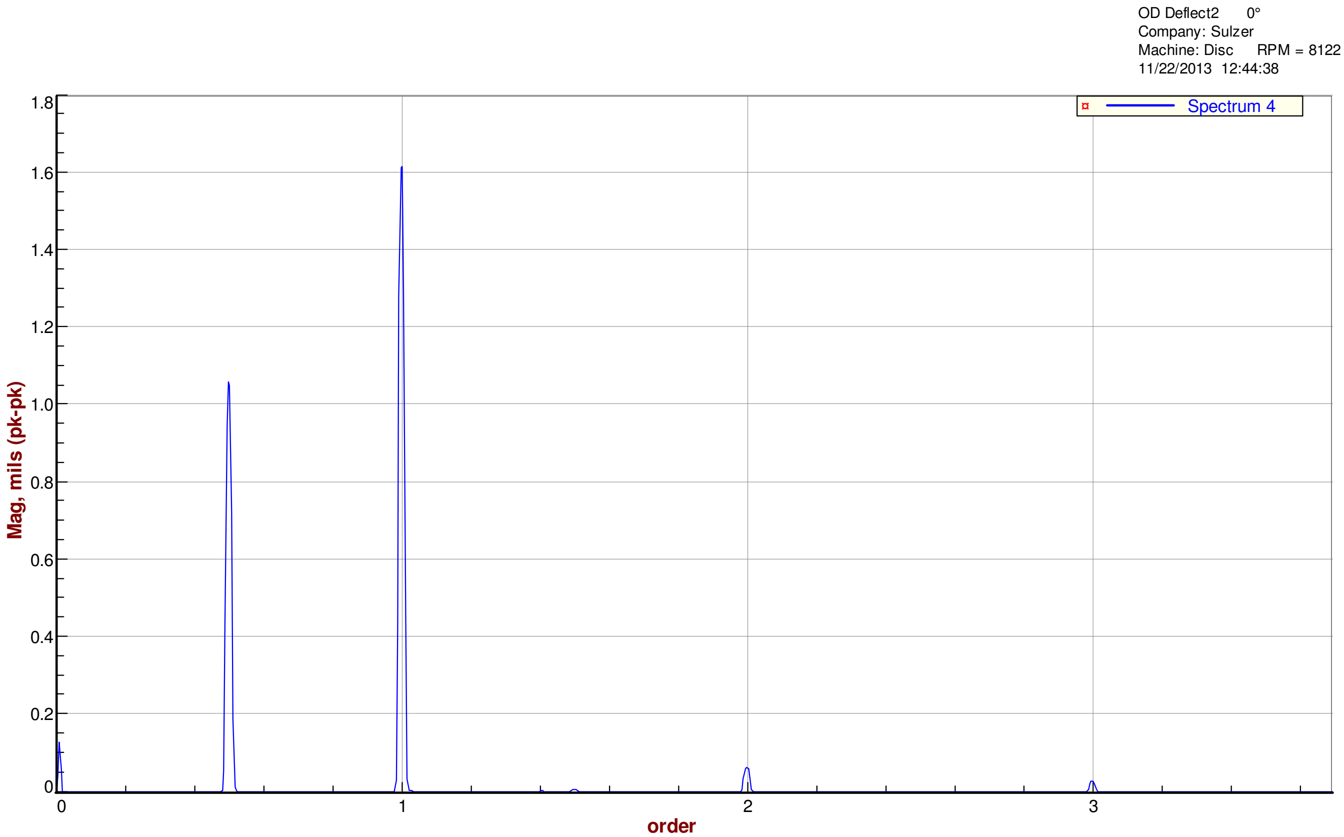


Strain Axial 0°
Company: Sulzer
Machine: Disc RPM = 8122
11/22/2013 12:44:38



OD Deflect1 0°
Company: Sulzer
Machine: Disc RPM = 8122
11/22/2013 12:44:38




OD Deflect2 0°
Company: Sulzer
Machine: Disc RPM = 8122
11/22/2013 12:44:38
Spectrum 4
Mag, mils (pk-pk)
order
1.8
1.6
1.4
1.2
1.0
0.8
0.6
0.4
0.2
0
0
1
2
3

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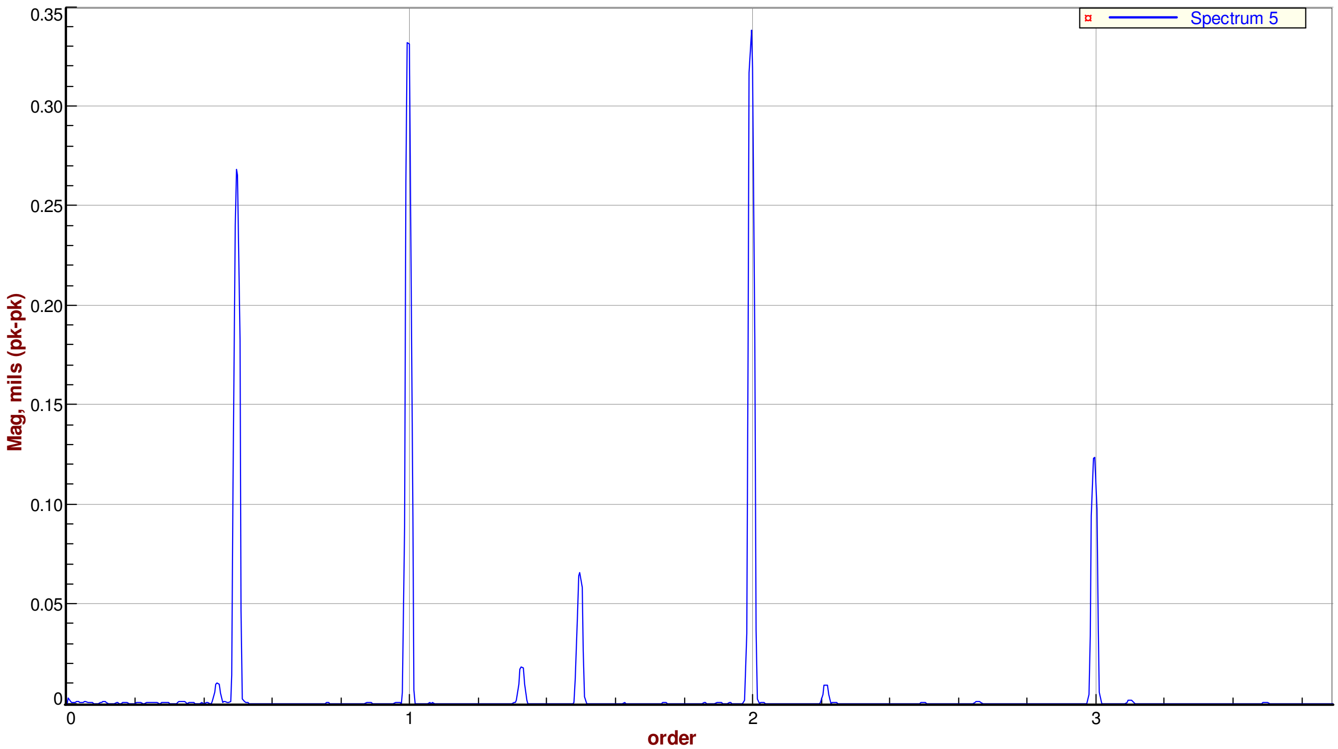
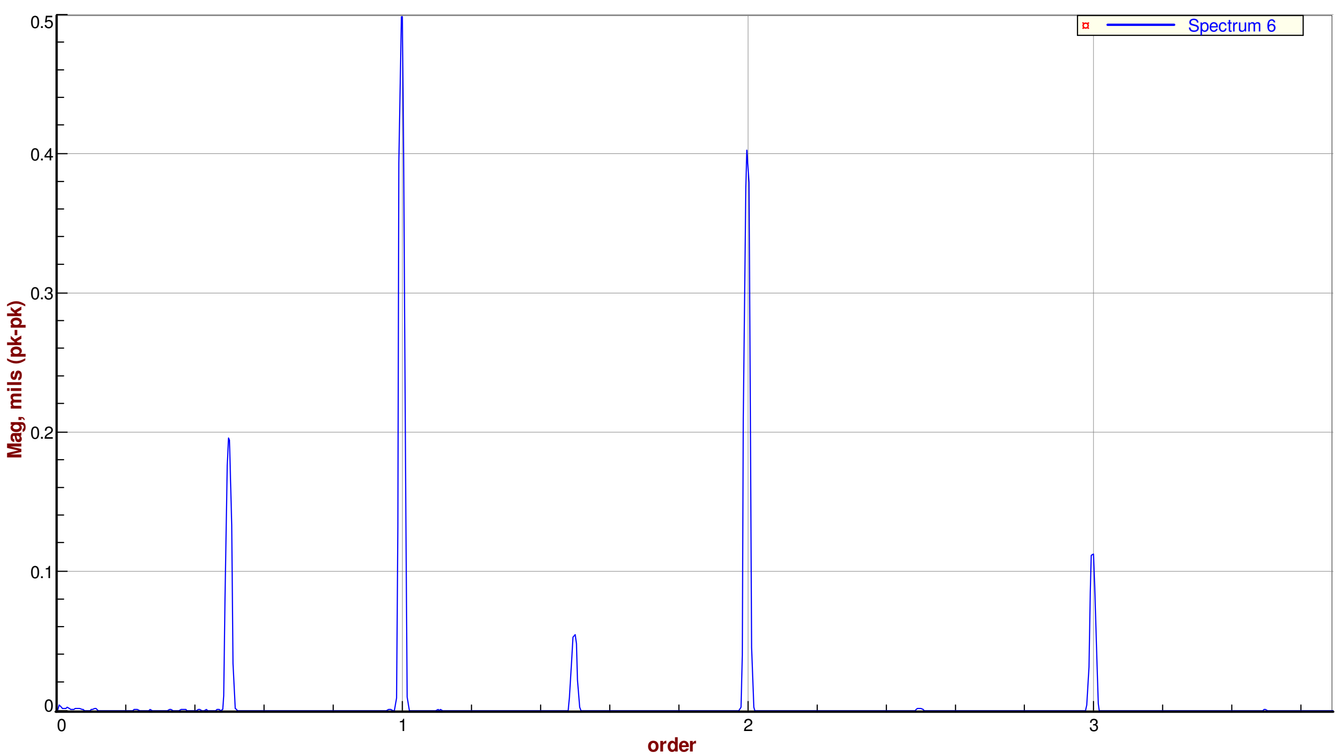
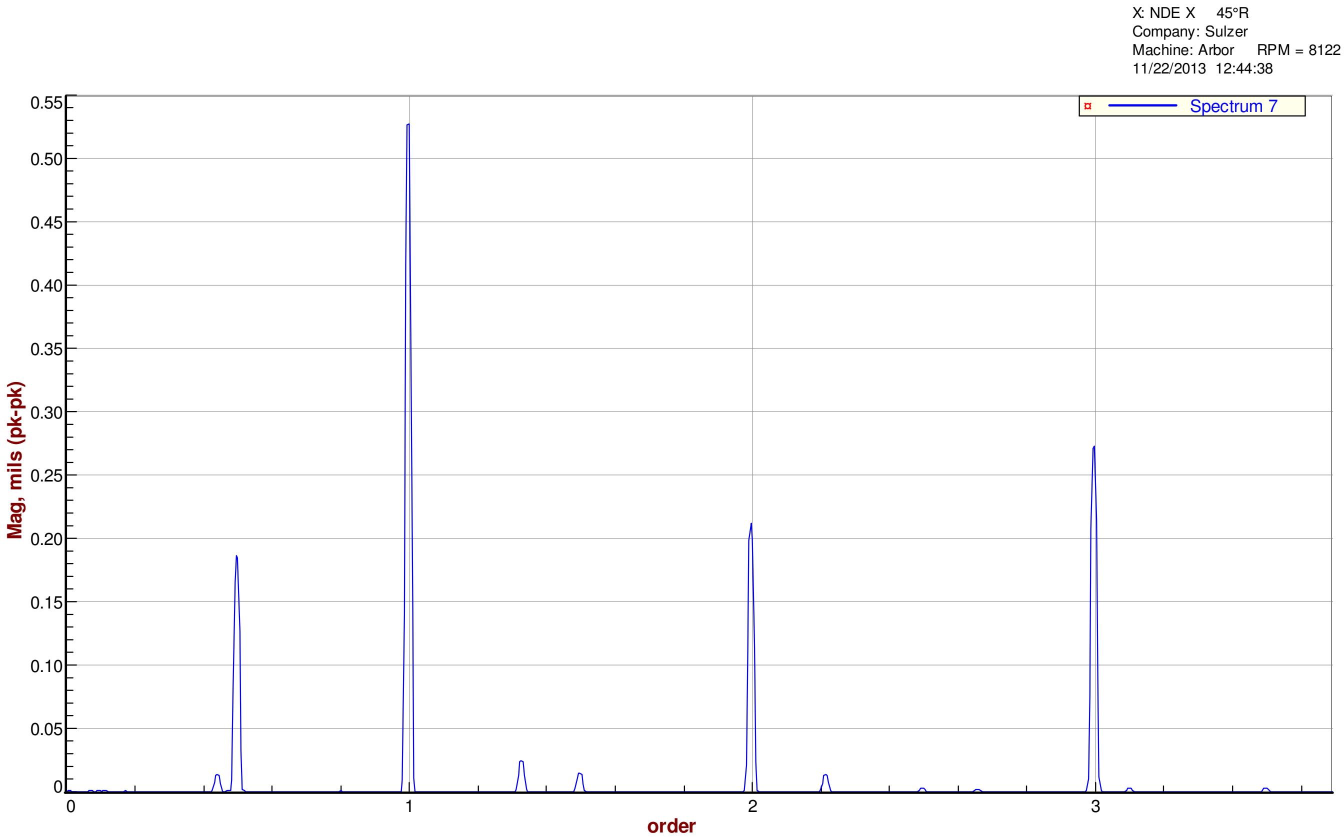
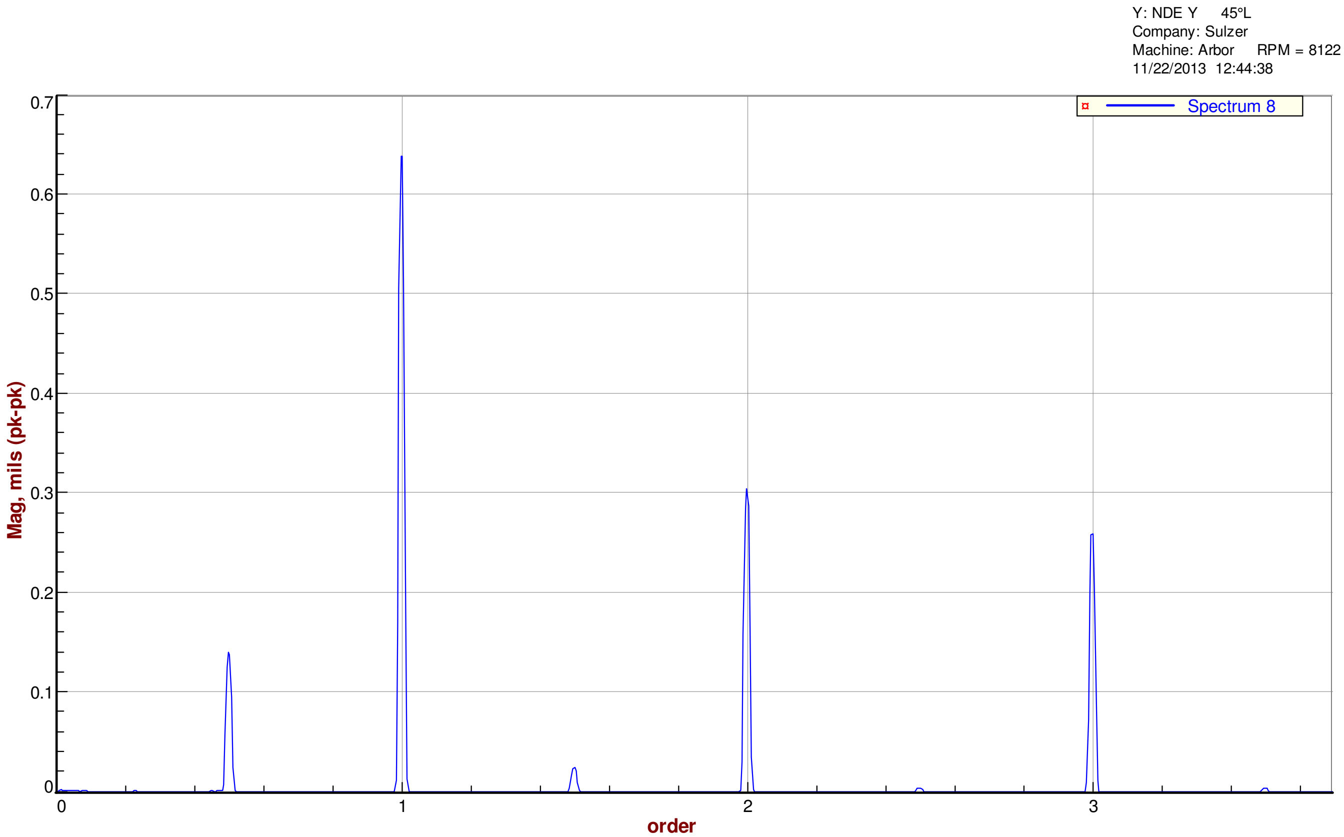
X: DE X 45°R
Company: Sulzer
Machine: Arbor RPM = 8122
11/22/2013 12:44:38

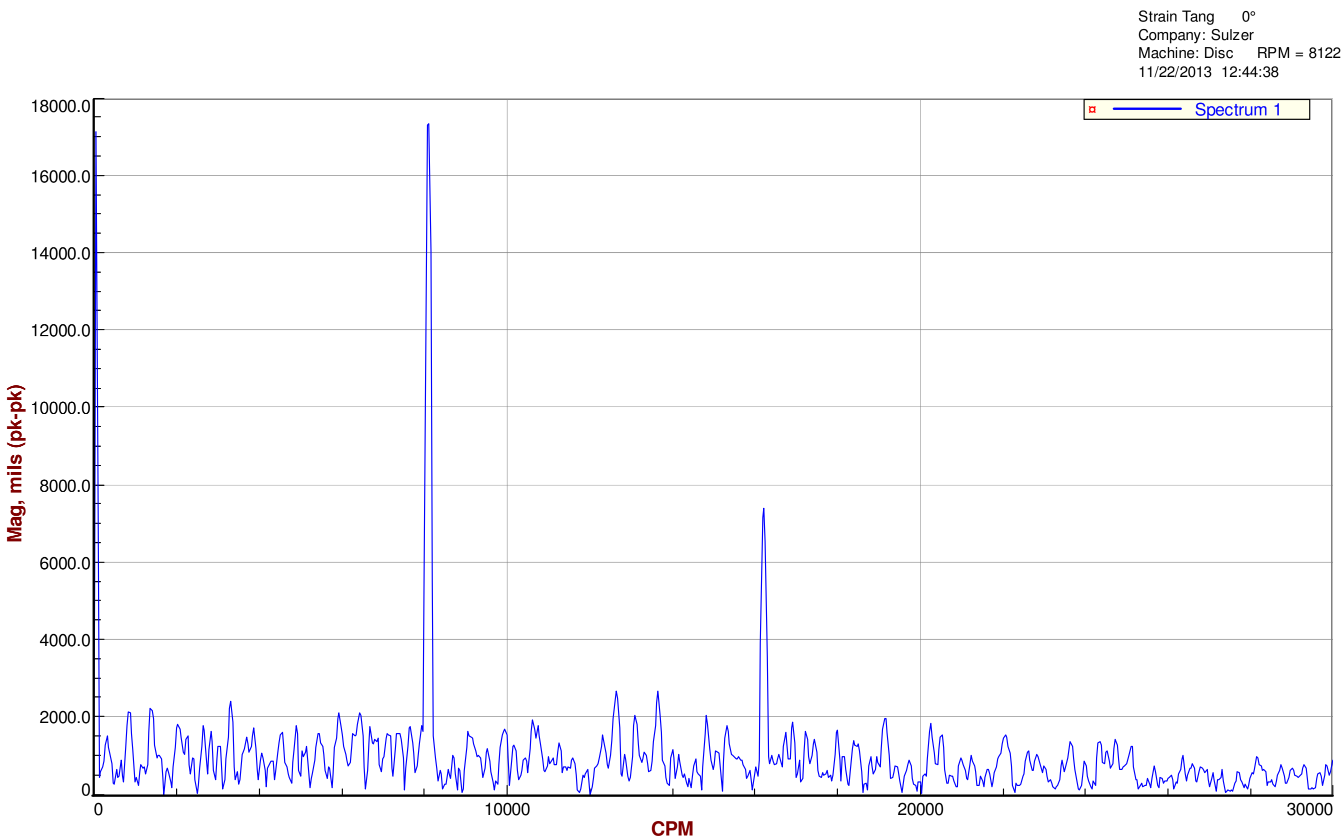


Y: DE Y 45°L
Company: Sulzer
Machine: Arbor RPM = 8122
11/22/2013 12:44:38

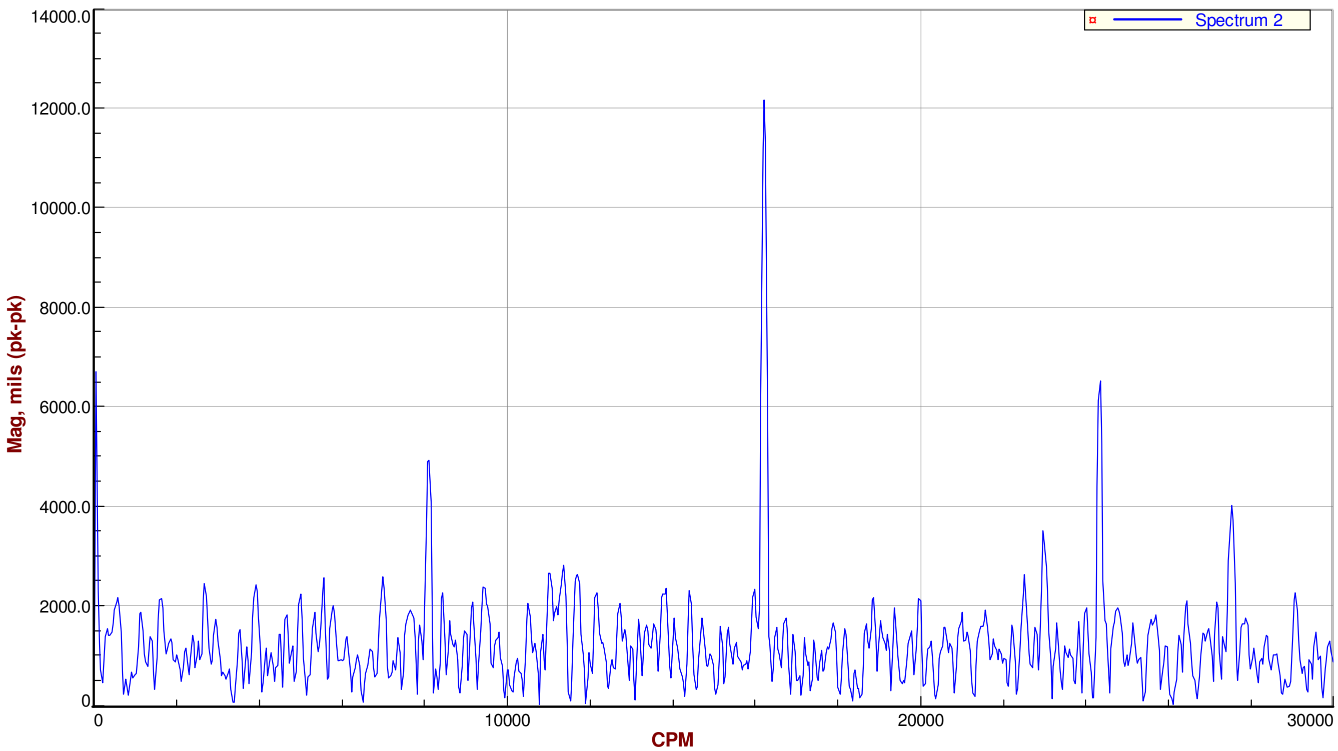



X: NDE X 45°R
Company: Sulzer
Machine: Arbor RPM = 8122
11/22/2013 12:44:38
Spectrum 7
Mag, mils (pk-pk)
0.55
0.50
0.45
0.40
0.35
0.30
0.25
0.20
0.15
0.10
0.05
0
1
2
3
order

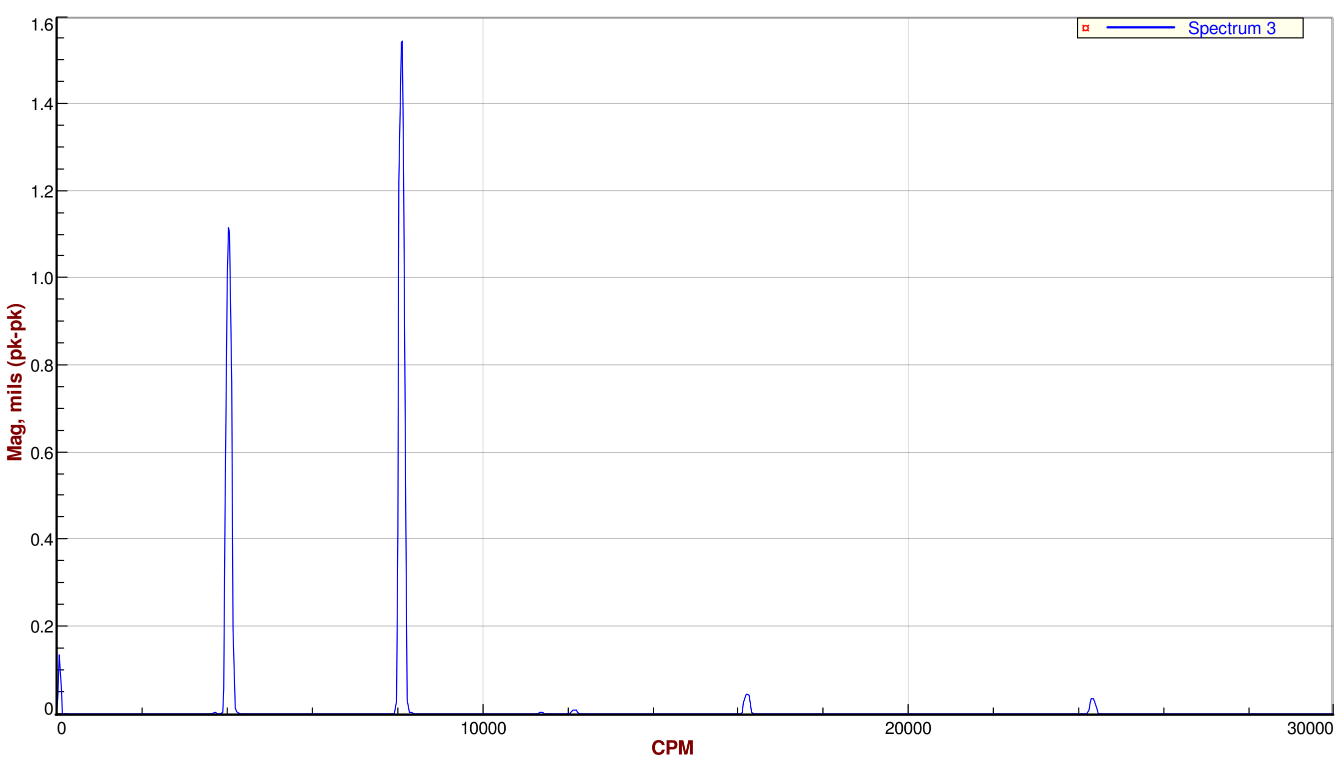

Y: NDE Y 45°L
Company: Sulzer
Machine: Arbor RPM = 8122
11/22/2013 12:44:38
Spectrum 8
Mag, mils (pk-pk)
order
0.7
0.6
0.5
0.4
0.3
0.2
0.1
0
0
1
2
3


Strain Tang 0°
Company: Sulzer
Machine: Disc RPM = 8122
11/22/2013 12:44:38
Spectrum 1
Mag, mils (pk-pk)
CPM
18000.0
16000.0
14000.0
12000.0
10000.0
8000.0
6000.0
4000.0
2000.0
0
0
10000
20000
30000

Strain Axial 0°
Company: Sulzer
Machine: Disc RPM = 8122
11/22/2013 12:44:38



OD Deflect1 0°
Company: Sulzer
Machine: Disc RPM = 8122
11/22/2013 12:44:38



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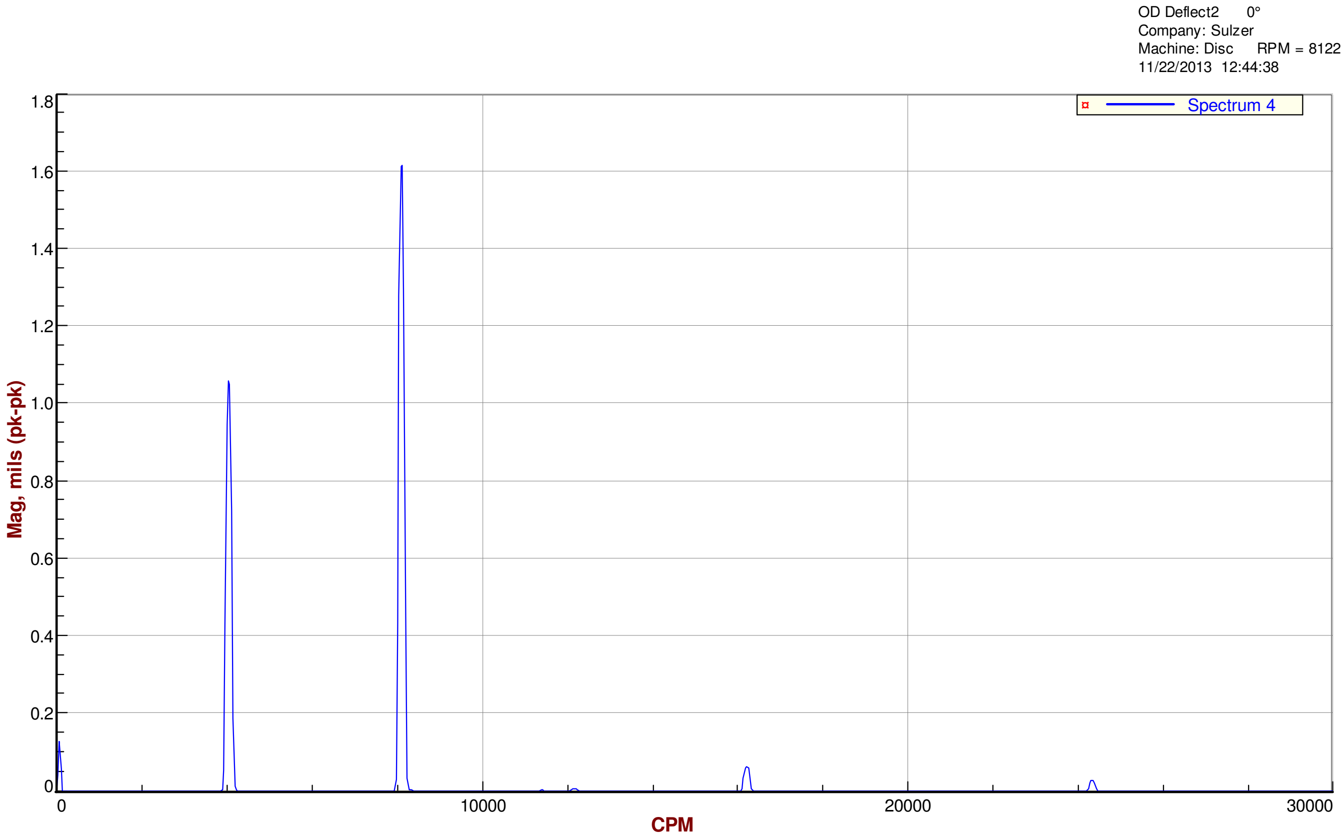

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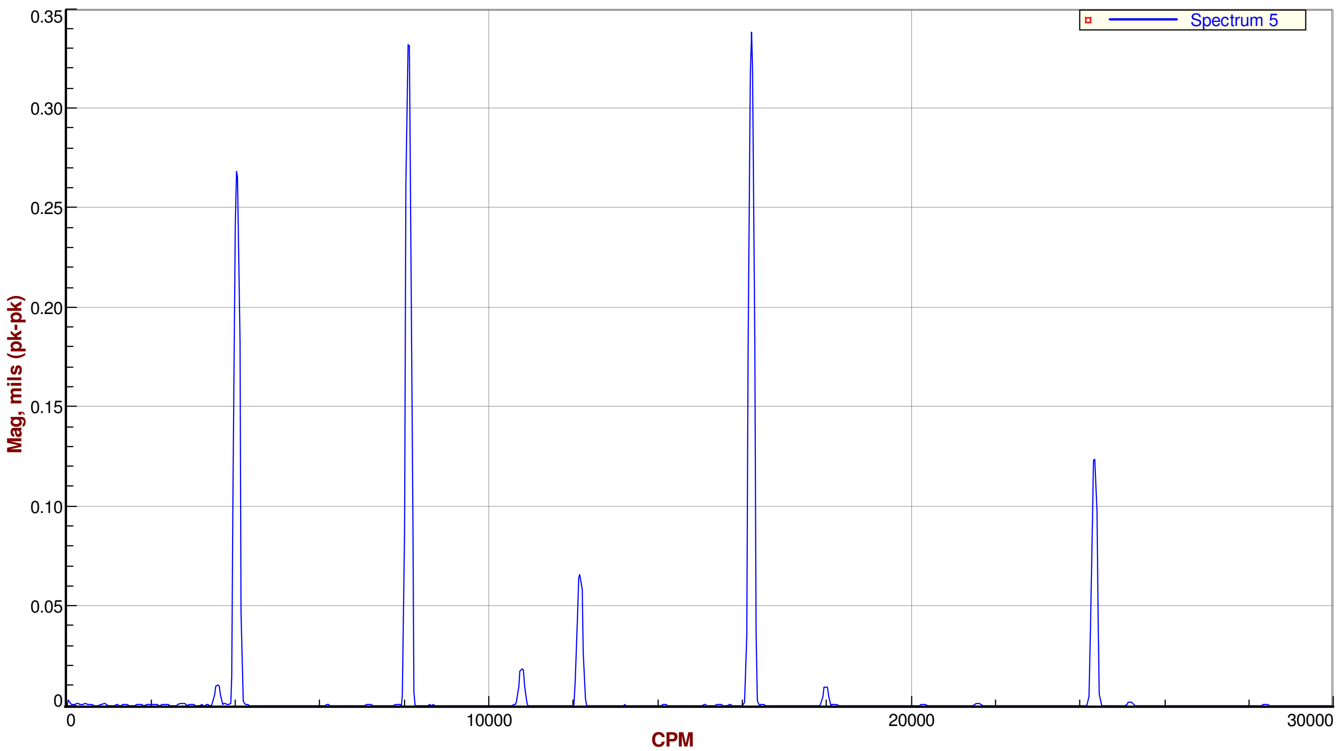
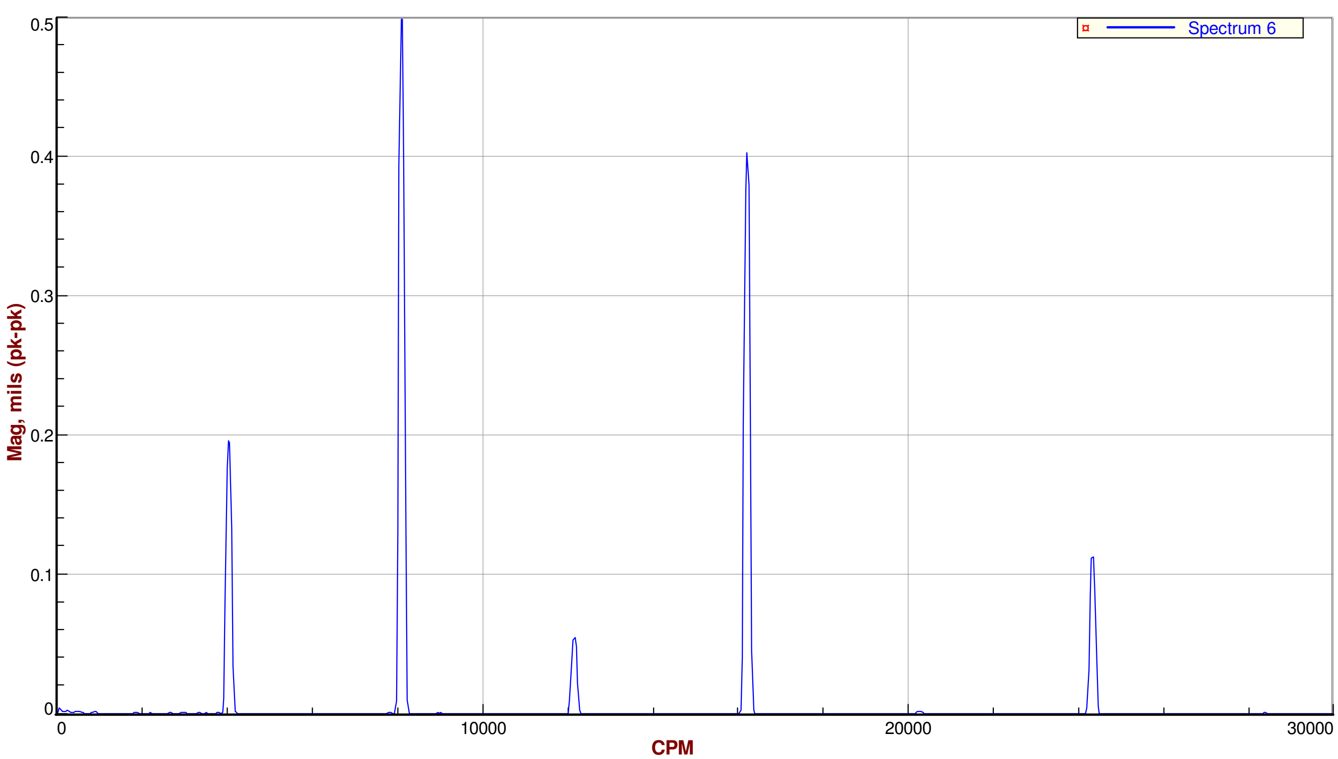
X: DE X 45°R
Company: Sulzer
Machine: Arbor RPM = 8122
11/22/2013 12:44:38



Y: DE Y 45°L
Company: Sulzer
Machine: Arbor RPM = 8122
11/22/2013 12:44:38



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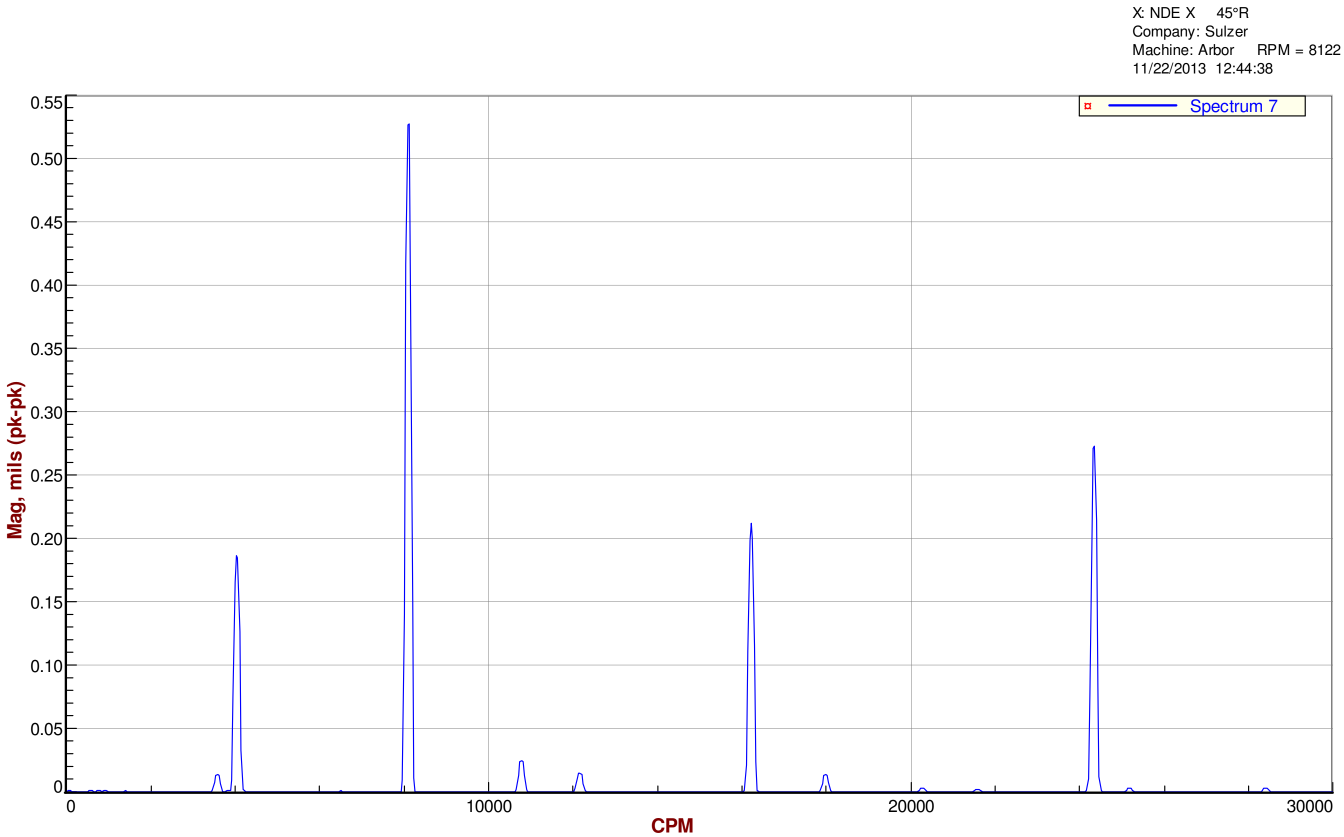

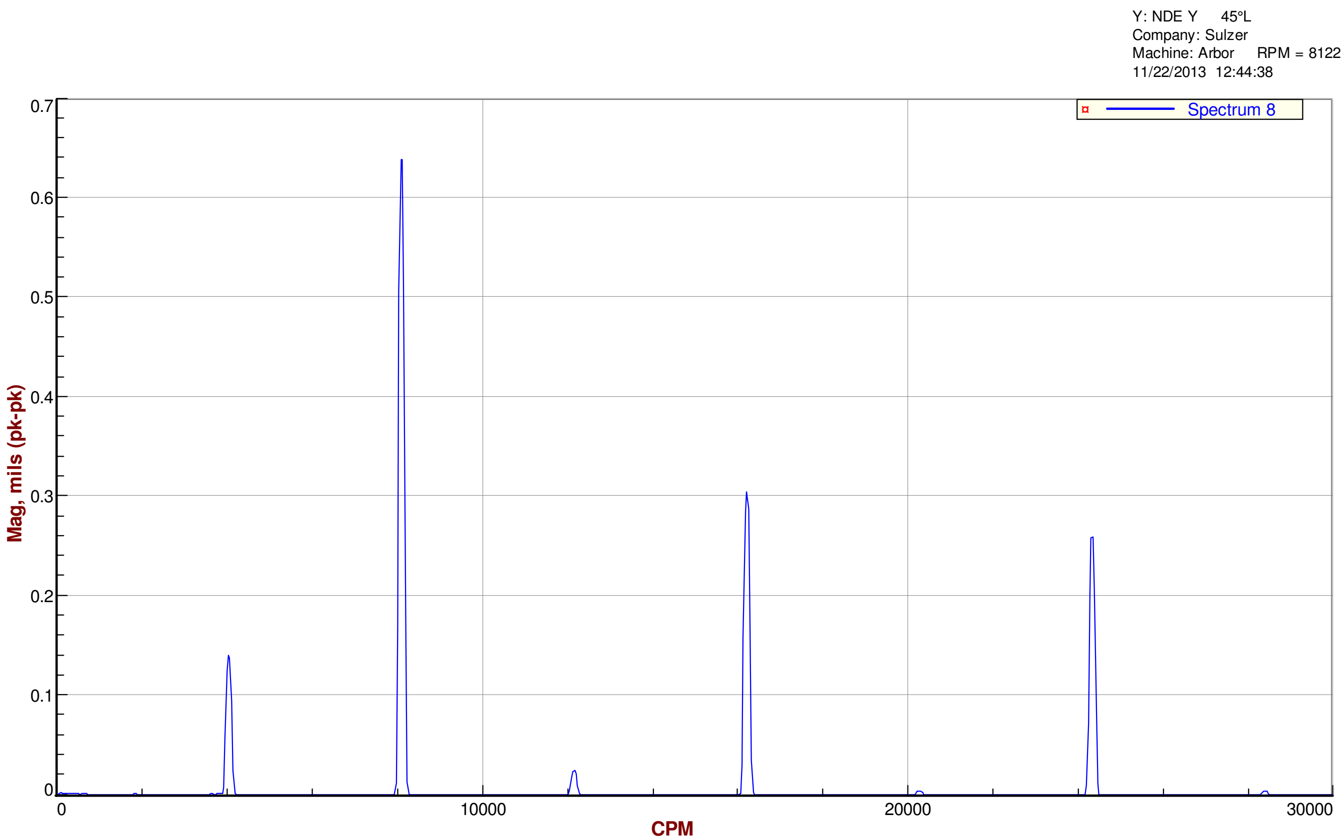

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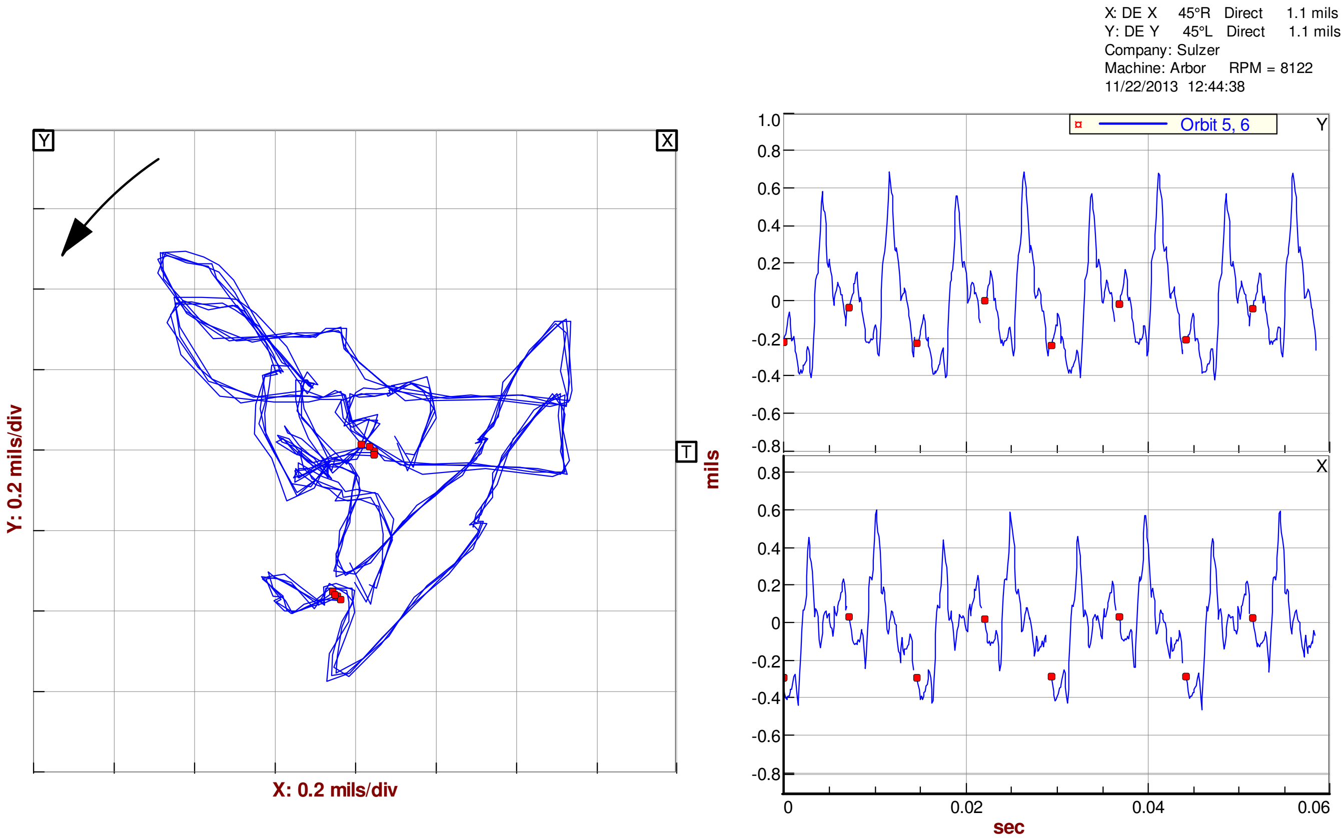

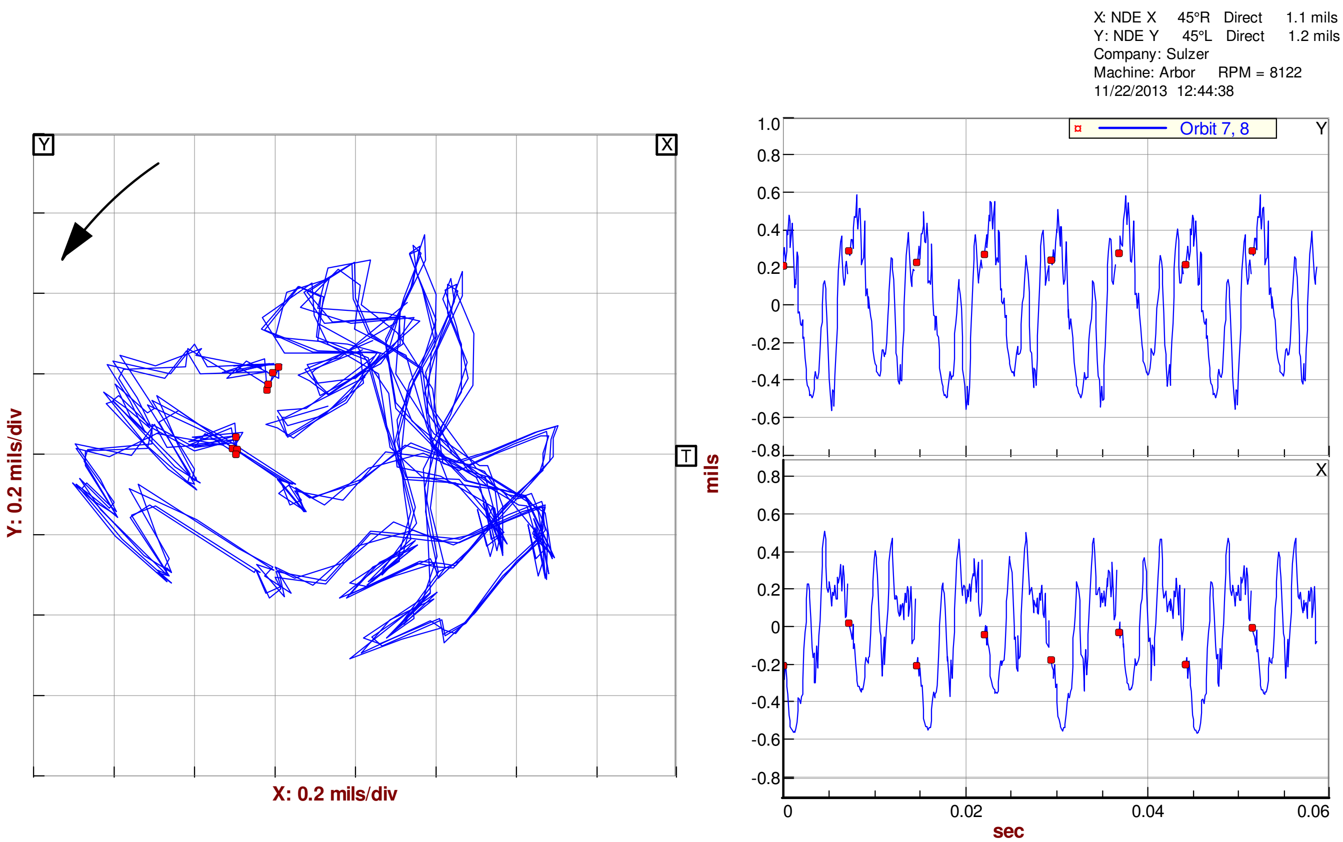

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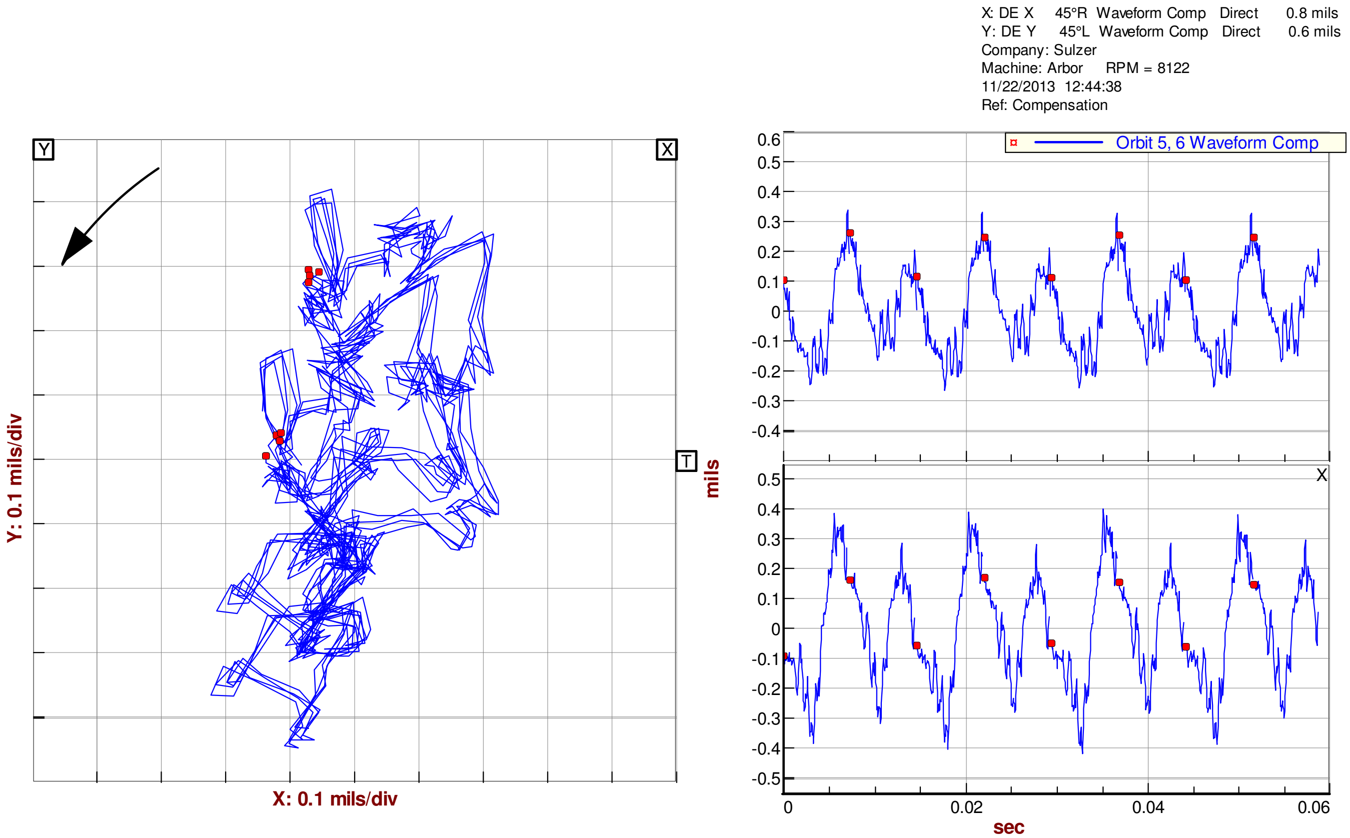
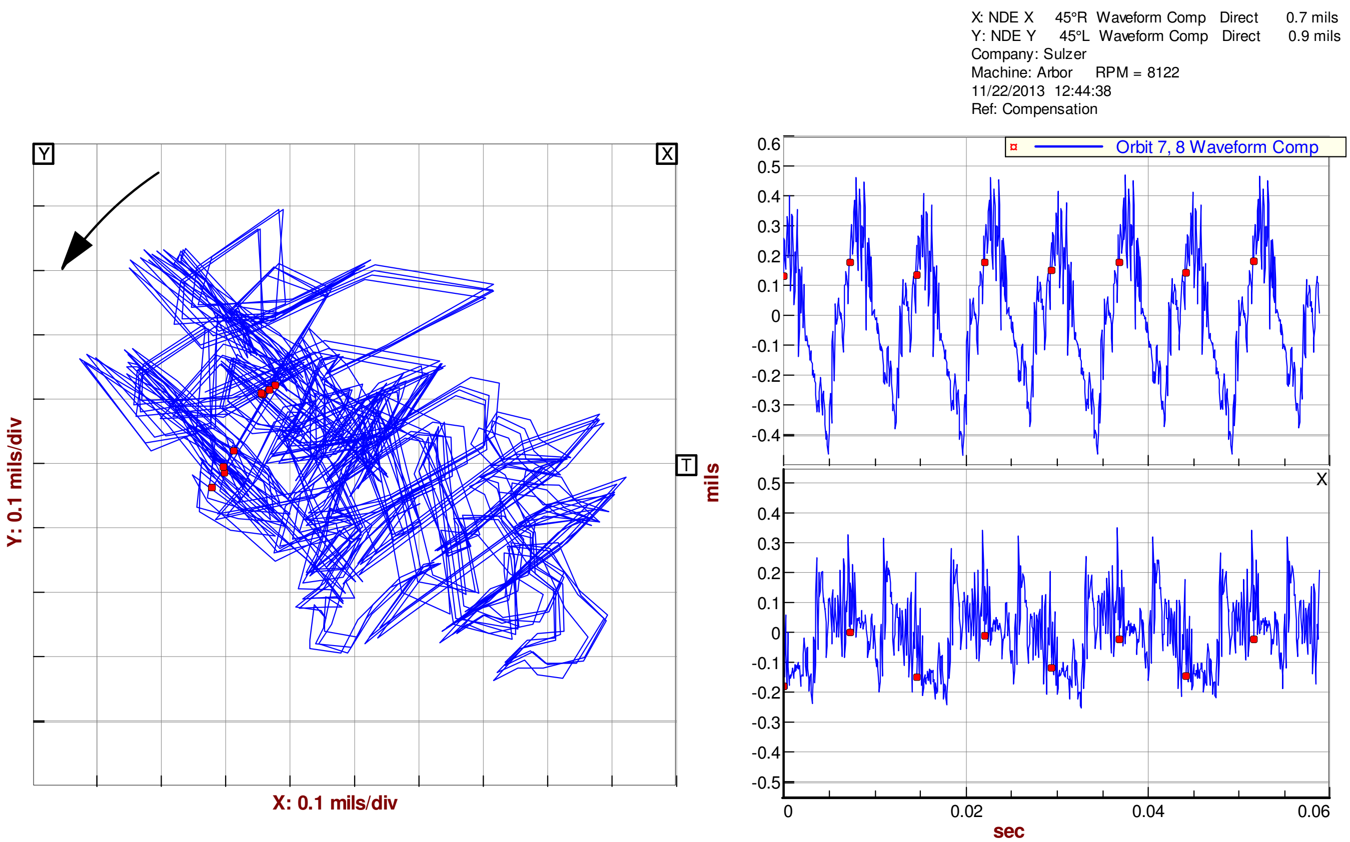
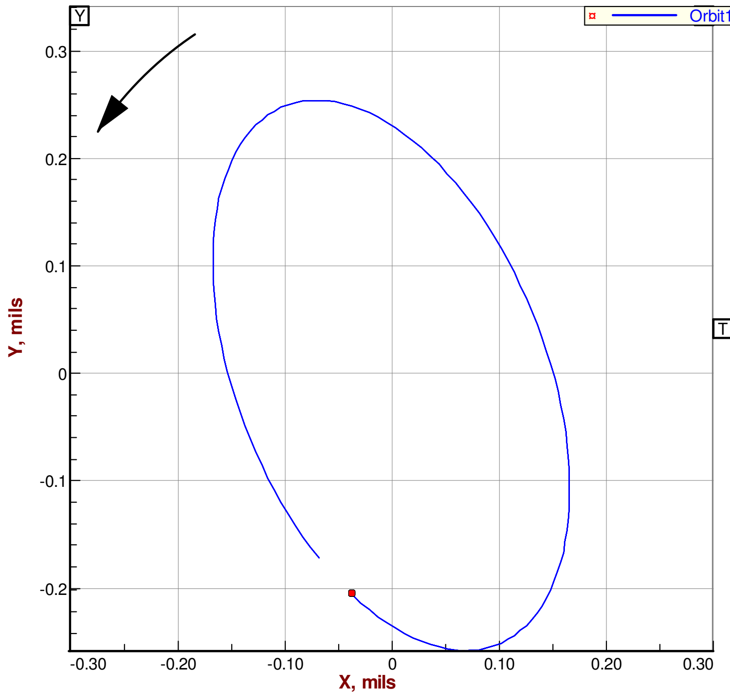
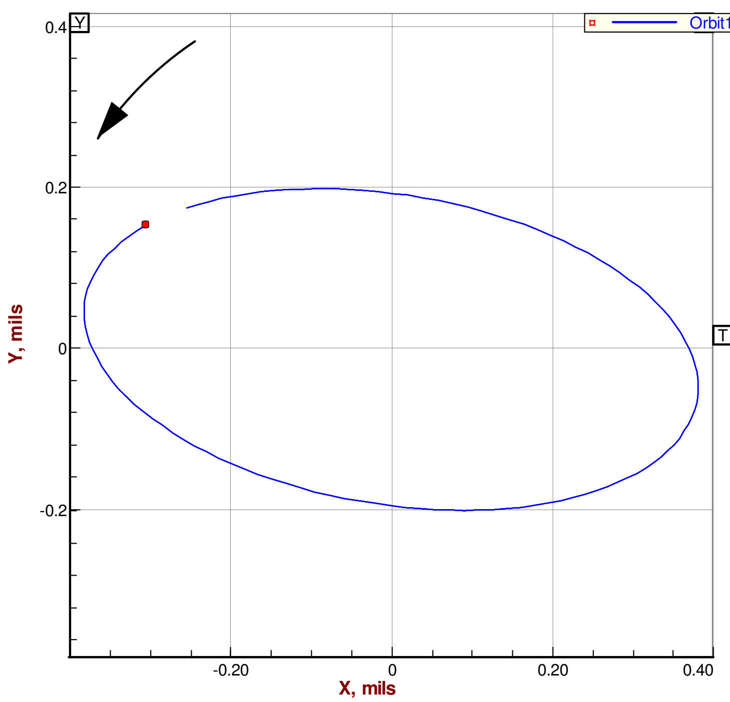




X: DE X 45°R 1X 0.3 mils; 178°
Y: DE Y 45°L 1X 0.5 mils; 242°
Company: Sulzer
Machine: Arbor RPM = 8122
11/22/2013 12:44:38



X: NDE X 45°R 1X 0.5 mils; 247°
Y: NDE Y 45°L 1X 0.7 mils; 12°
Company: Sulzer
Machine: Arbor RPM = 8122
11/22/2013 12:44:38



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Appendix B

ROTATION vs. PRECESSION

- Rotation is the angular motion of the rotor about its geometric center, or shaft centerline. In the absence of any external forces, a perfectly balanced rotor will spin in place around the geometric center, of the shaft without any change in position (vibration) of that center.

- Precession is the vibration of the rotor's geometric center in the xy plane that is perpendicular to the axis of rotation. The vibration of the rotor's geometric center in the xy plane results in the shaft orbit. The direction of precession (vibration) can be expressed as counterclockwise (CCW) or as clockwise (CW).

- When the direction of precession (vibration) is the same as the direction of rotation, then the vibration is defined as being forward precession.

- When the direction of precession (vibration) is opposite to the direction of rotation, then the vibration is defined as being reverse precession.

- These concepts of forward and reverse precession have powerful application in full spectrum and in the diagnosis of certain types of malfunctions.

- If the only force acting on the rotor is unbalance; i.e. by definition a circular, forward, sinusoidal force, and if the restraining stiffnesses are equal in all directions, then the resulting orbit will also be perfectly circular, and forward in precession.

- Uni-directional forces that push on the rotor; i.e. misalignment, stiffness asymmetry, rubs, aerodynamic loading, etc. will restrain the vibration in one plane, resulting in the orbit becoming, to some degree, elliptical in nature. By definition, an elliptical orbit is comprised of both forward and reverse vibration components.

- As the orbit shape changes the forces and restraints acting on the system are becoming more complex.

- The orbit of the rotor can range from purely circular to a very complicated shape containing many frequencies of vibration.

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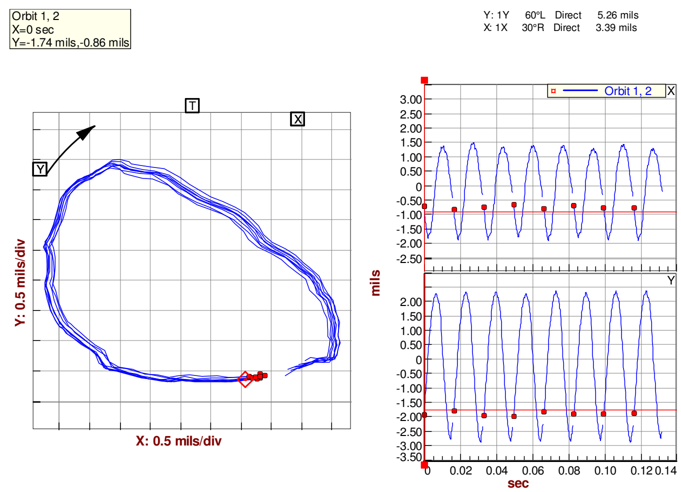
DATA PLOT FORMATS

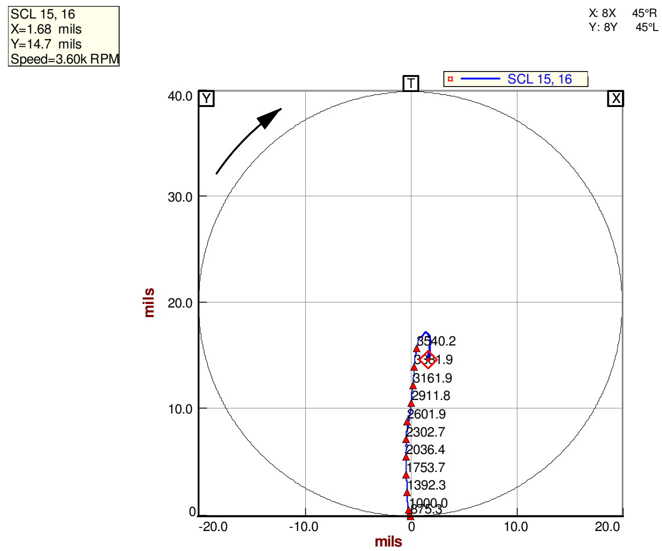
Orbit/ Timebase plots are used to primarily examine the shape or form of the vibration; i.e. the two dimensional centerline motion of the shaft as it vibrates in the xy plane perpendicular to the axis of rotation.



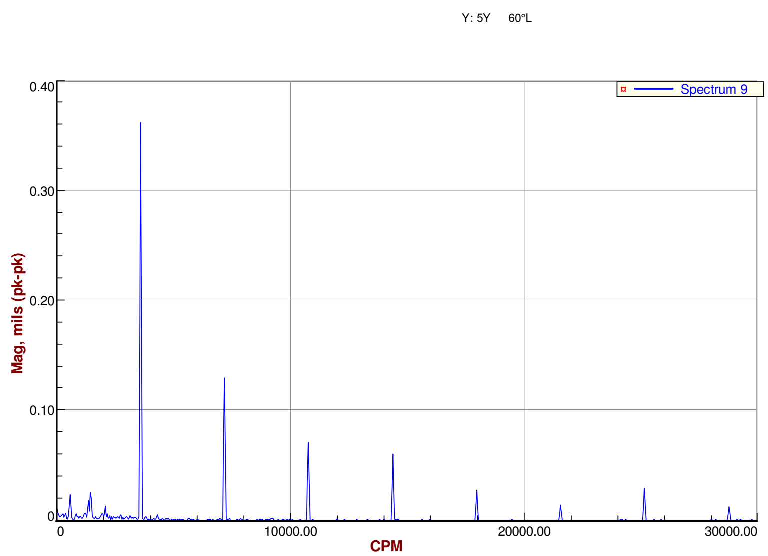
Amplitude information is simultaneously plotted from a pair of orthogonal (90°) transducers. The amplitude information, coupled with a phase angle reference, is essential for analyzing the dynamic motion of the rotor. Interpretation of Orbit/ Timebase plots provides insight into the nature of the vibration, i.e. the presence of steady state preloads as well as system asymmetry.

The leading edge of the phase trigger appears on both the orbit and timebase plots as a blank, with the trailing edge displayed as a dot. Vibration precession around the orbit is from blank to dot and is forward if it is in the direction of rotation and reverse if it is opposite the direction of rotation.

Shaft average centerline plots are used to observe changes in the average position of a rotor within its radial bearing clearance. Changes in shaft centerline position can provide a warning for changes in the alignment state of a machine and may indicate bearing wear. Average shaft radial position indicates the presence of preloads as well as overall bearing stability.



Spectrum plots are used to examine the frequency components useful for identifying certain machine problems as some machine malfunctions produce vibrations at characteristic frequencies. **Since many machines exhibit similar vibration frequency characteristics, spectral analysis must be utilized with the time domain to fully appreciate the dynamic characteristics of the rotor / bearing system.**



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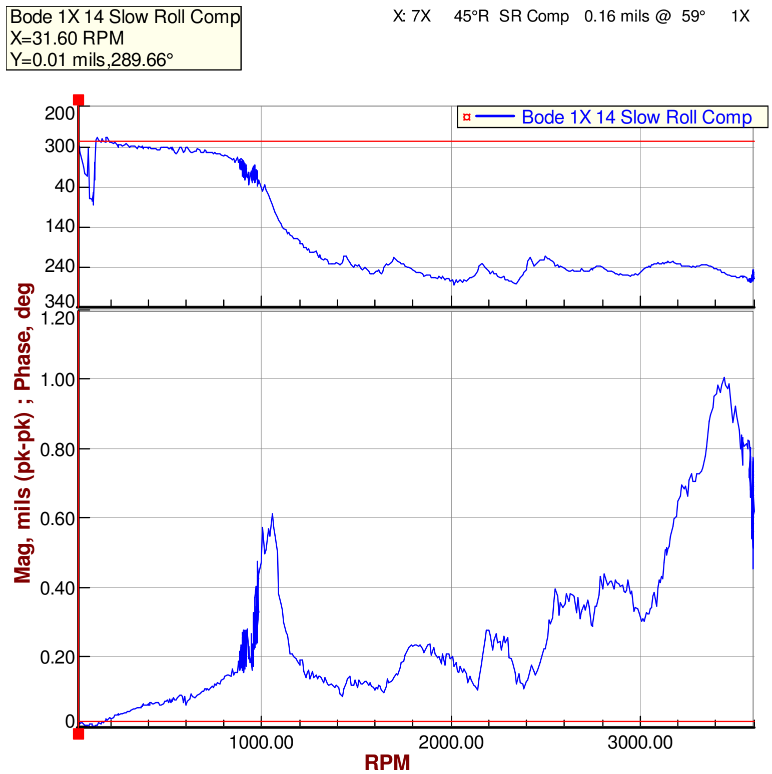
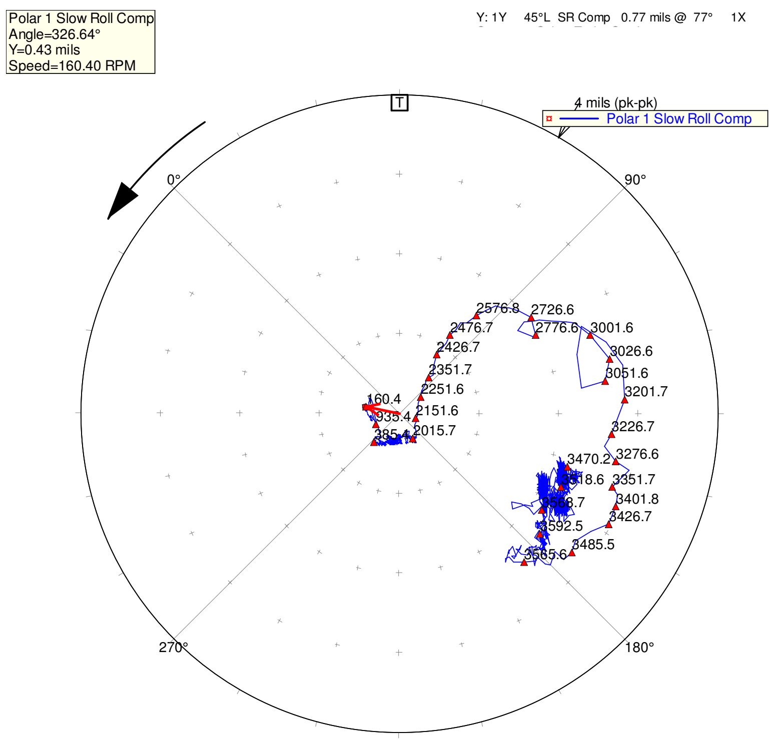
Bode' and Polar plots show the synchronously filtered change of amplitude and phase angle as a function of shaft rotative speed. The bode plots enhances the amplitude response and provides the information to calculate other rotor / bearing system parameters such as the Synchronous Amplification Factor (SAF). In the Bode format, the amplitude and phase information is plotted as a function of rotative speed, whereas in the Polar format it is the vibration vector, plotted in polar coordinates, as a function of rotative speed.





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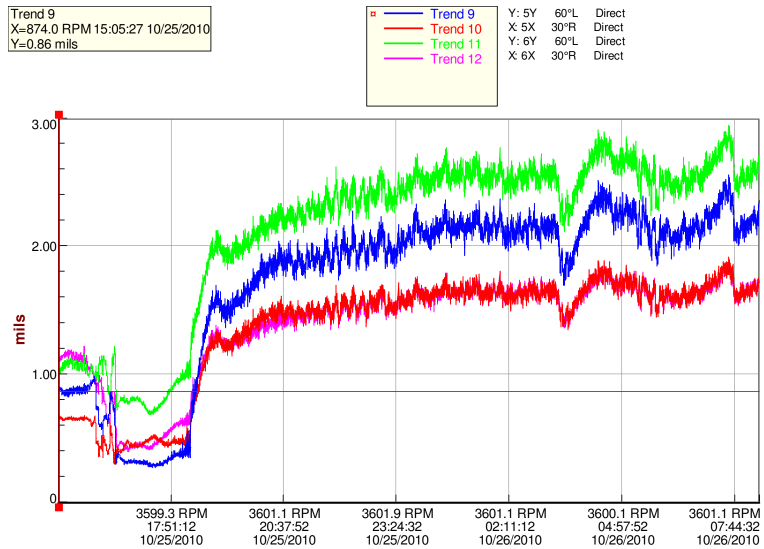
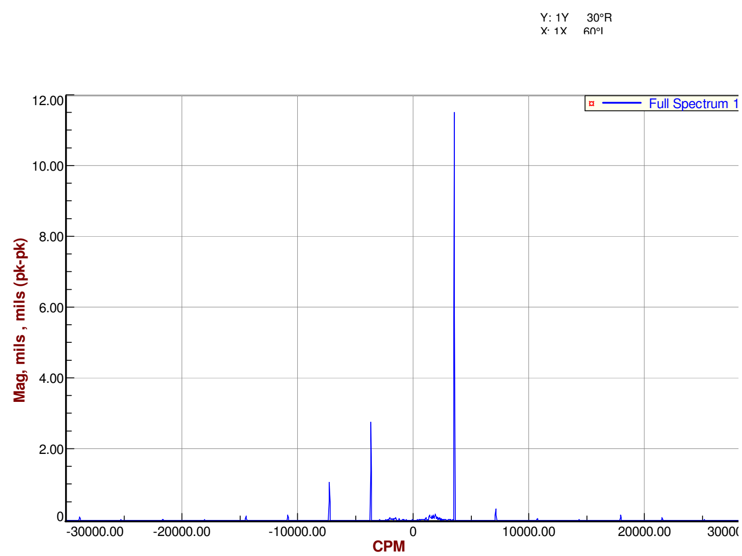
Trend plots are used to detect changes of any measured variable from any channel over a period of time. A change in a parameter, increasing or decreasing, is often an early indication of possible problems.



Full Spectrum plots (The Spectrum of an Orbit) are enhanced spectrum plots produced by using the dynamic data from orthogonal (90°) XY transducers to calculate the amplitudes of the forward and reverse (backward) vibration frequency components. The full spectrum presents shaft vibration orbital data in a complex filtered data presentation. **Any shaft orbit that is not purely circular by definition is comprised of both forward and reverse vibration components.** In the orbit timebase plot, this is appreciated by directly viewing the ellipticity of the orbit. In the full spectrum data plot, this same evaluation is made by evaluating the relative amplitudes of both the forward and reverse components at a given frequency.



Notes on Full Spectrum Data Plots:

In much the same way that a traditional spectrum plot breaks down a complex vibration waveform into its individual filtered frequency components that can be summed back to reproduce the original complex waveform, the full spectrum data plot breaks down a complex orbit shape into its individual filtered frequency orbits that can be summed back to reproduce the original complex orbit shape. Mathematically it can easily be shown that any filtered orbit can be represented by two vectors, one which rotates in the direction of rotation (forward) and one which rotates opposite of rotation (reverse). The ratio of the forward to the reverse vector indicates the ellipticity of the filtered orbit at that frequency. For example, consider the full spectrum shown above. The forward vibration component at 3600 cpm is approximately

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4.5 times as large as the reverse component at -3600 cpm. The resulting filtered orbit at 3600 cpm is elliptical with a minor axis that is approximately 77% the length of the major axis.

"Complicated orbits will have forward and reverse [vibration] components at many frequencies. Each pair of components represents a set of vectors that rotate in forward and reverse directions at a specific frequency. The most complex orbit can always be described by a set of such vectors and full spectrum lines. The lines in the full spectrum represent the precessional structure of the orbit.....The entire orbit can be expressed as the sum of its forward and reverse components in the same way that a timebase waveform can be expressed as the sum of its sine wave components.

At first glance, the full spectrum might seem abstract. What is significant about forward and reverse precession? It lets us easily identify key orbit characteristics that might otherwise be obscured. Precession direction and ellipticity provide insight into the state of health of a machine. More importantly, some rotor system malfunctions can have characteristic signatures on a full spectrum plot that are not available on traditional spectrum plots. These characteristics can be used to discriminate between different malfunctions that produce vibration with similar frequencies."[1]

The 1st Law of Machinery Diagnostics

$$\overrightarrow{\boldsymbol{Displacement}} = \frac{\overrightarrow{\sum \boldsymbol{Forces}}}{\overrightarrow{\sum \boldsymbol{Dynamic\ Stiffness}}} \qquad [1]$$

Equation 1 is often referred to as the *"1st Law of Machinery Diagnostics"*. The 1st Law reminds us that vibration is not a scalar quantity; it is a vector quantity in that it has both a magnitude and a direction. Additionally, the 1st Law acknowledges that if the vibration displacement changes in amplitude, either increasing or decreasing, it is due to either a change in the applied forces to the rotor bearing system, a change in the dynamic stiffness of the rotor bearing system, or both.

Although beyond the scope of this report, it can be readily shown that:

$$K_{Dynamic\ Stiffness} = \left(\boldsymbol{K} - \boldsymbol{M}\Omega^2\right) + \boldsymbol{j}\left(\boldsymbol{D}\boldsymbol{\omega} - \boldsymbol{\lambda}\boldsymbol{D}\Omega\right) \qquad [2]$$

In Equation 2, K is the spring stiffness, M is the rotor modal mass, D is the rotor/bearing viscous modal damping, and ω is a term that represents the frequency of the vibration precession (or orbiting), which may be different from the rotation speed, Ω. λ is the Fluid Circumferential Average Velocity Ratio of the fluid in the bearing or seal. The first two terms to the left of the equal sign, contained in the first set of parentheses of Equation 2, are the spring stiffness K (shaft, bearing, support pedestal, etc.) and the mass-inertial stiffness, $-M\Omega^2$. The spring and mass-inertia stiffnesses form the *Direct Dynamic Stiffness*. It is called Direct because the *Direct Dynamic Stiffness* acts directly along the line of action of the summation of forces $\overrightarrow{F}$ that are applied to the rotor system. Also, the Mass-Inertia Stiffness, $-M\Omega^2$, is a speed squared dependent term. When the magnitude of the Mass-Inertia Stiffness, $-M\Omega^2$, is equal to the Spring Stiffness K, the *Direct Stiffness* becomes zero; i.e. the undamped natural frequency or critical speed of the rotor / bearing system.

[1] *Fundamentals of Rotating Machinery Diagnostics*; Bently, Donald. E.; Chapter 8, pages 120-122

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The next two terms, contained in the second set of parentheses, of Equation 2, are the *Quadrature Dynamic Stiffness Terms*, i.e. the Damping Stiffness, $+jD\omega$, and Tangential Stiffnesses, $-jD\lambda\Omega$[2]. It is called *Quadrature* because, as indicated by the +j (i.e., $\sqrt{-1}$), acts at 90° to the line of action of the summation of the applied forces, $\vec{F}$.

When a fluid, either liquid or gas, is constrained within the space between two concentric cylinders, one rotating and one stationary, the fluid in the clearance between the two cylinders will be set into circumferential motion. This can happen in rotating machinery within the fluid-lubricated bearings, seals, around pump impellers, or in any fluid filled gap between the rotor and the stator.

In a hydrodynamic, fluid film bearing, the fluid velocity at the surface of the bearing is zero, while the fluid velocity at the surface of the rotor is equal to the rotor surface speed. The fluid near the rotor surface moves at a slightly slower velocity that continues to decrease with the distance from the rotor and reaches zero at the bearing surface. It is easy to see that the fluid must have some overall average velocity which is less than the rotor speed, and that the faster the rotor turns, the faster the average fluid velocity must be. In fact, previous work has shown that the fluid circumferential average angular velocity in the bearing can be expressed as $\lambda\Omega$, where λ is the Fluid Circumferential Average Velocity Ratio, and Ω is the rotor rotative speed, with the value of λ normally less than ½X; i.e., 0.43X to 0.48X.

If a radial load is applied to a rotor operating in the center of a fluid-lubricated bearing, the rotor will be displaced from the center of the bearing. The reduced clearance on one side will restrict the flow of fluid around the journal bearing clearance. Because of this restriction, the fluid has to slow down as the available flow area gets smaller. As the fluid slows down, the pressure in this region increases. The pressure exerts a force on the rotor. This force can be separated into a radial part, which points back toward the center of the bearing, and a tangential part, which acts at 90° to the radial force, in the direction of fluid flow; i.e. shaft rotation. Both the radial and tangential forces are proportional to the displacement of the rotor from the center of the bearing. Thus, the lubricating fluid acts like a spring with a complex stiffness. The radial part of the fluid wedge stiffness is referred to as the *Direct Dynamic Stiffness* and the tangential part of the fluid wedge stiffness is referred to as the *Quadrature Dynamic Stiffness* as defined by Equation 2 above.

It is this complicated spring effect that provides the primary support for the rotor. The rotor and the fluid pressure "wedge" move until the fluid spring forces in the bearing exactly balance the summation of radial loads applied to the rotor and the rotor settles into an equilibrium position.
If the rotor is disturbed from the equilibrium position, the direct spring stiffness acts to return the rotor to the equilibrium position. However, the net force produced by the damping stiffness acting in the direction opposite of rotation and the tangential stiffness acting in the direction of rotation

[2] Cross coupled stiffness

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prevents that from happening and the rotor orbits (vibrates) around the equilibrium position at some displacement amplitude.[3]

If the forces that perturb the rotor could be removed, the orbit would spiral back to the equilibrium position. It is the complicated effect of these forces that maintains rotor stability.

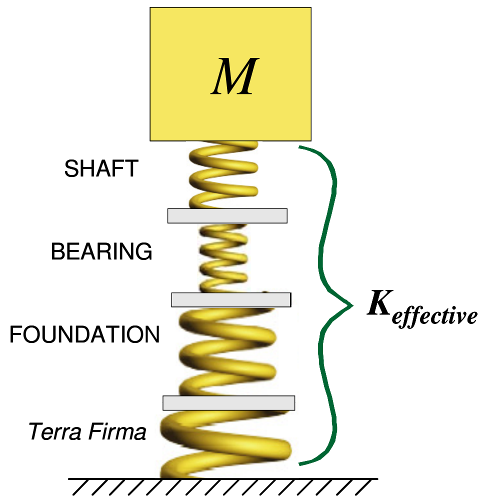

Figure A1

Figure A1 is a graphical representation showing that in reality the overall system *Dynamic Stiffness* is comprised of a series of springs from the rotor to ground with the weakest (smallest) spring being the fluid film stiffness of the bearing itself. This fact is logical in that it is the weakest spring that will control vibration. Furthermore the resulting *Dynamic Stiffness* resulting from these springs in series will always be less than the stiffest spring in the series.

In the case of the bearing fluid film, vibration energy is absorbed and dissipated via the direct damping term, $D\omega$, which is the rotor pushing on the fluid – the shock absorber effect. However, for adequate damping to occur within the bearing, the bearing must be rigidly mounted to the foundation spring and the foundation spring rigidly connected to ground. Any looseness between the bearing and the housing, the housing to the foundation, the foundation to ground will result in unwanted seismic relative motion which will adversely affect damping within the bearing. Any looseness between the bearing and its housing, the housing and the foundation, or the foundation and ground will reduce overall damping which can lead to elevated vibration amplitudes at one or more frequencies.

[3] ***A Short Course in the Practical Application of Rotordynamics as a Tool for Machinery Diagnostics*; Thomas, G. Richard, Paper presented at the Canadian Machinery Vibration Association Annual Meeting / Seminar; 29 October 2009**