AN ELASTOMERIC COUPLING CASE HISTORY

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A new elastomeric coupling installed on four large forced draft fan caused high motor shaft vibration as the motor speed (load) increased. The vibration exceeded the motor shaft trip set point before full speed and limited power output of the 500 MW unit. Numerous potential causes were investigated including unbalance, misalignment, thermal growth and coupling concentricity while suppliers argued if there was a problem and if so who was responsible. The elastomeric coupling inserts were found to be the actual cause and improved quality control eliminated this vibration problem.

Two 500 MW gas fired generating units replaced their four original 8000 HP AC induction motor driven Forced Draft fans with four 7500 HP, 1200 RPM motors driven by VFDs. This modification was done to save energy, allow greater load following flexibility, and in theory, should require less maintenance. The installation included a Bently 3500 monitoring system with X-Y proximity probes on all fan and motor bearings.

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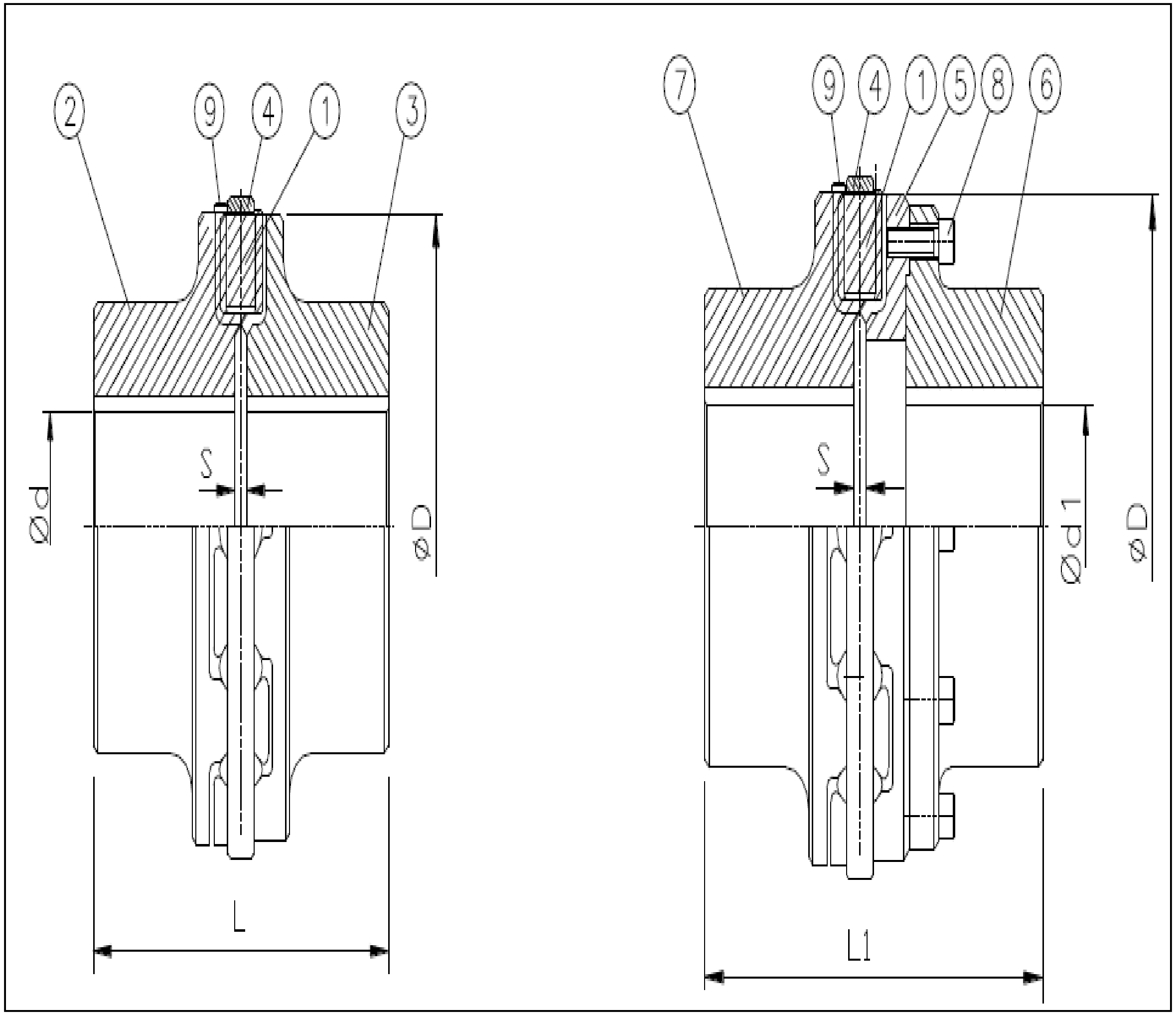

Coupling cross section view, Item #1 is the Elastomeric Element.

INTRODUCTION & INITIAL DATA

High drive end motor shaft vibration of the newly installed VFD driven 1B Forced Draft Fan prevented the 500 MW unit it served from reaching full output. Motor bearing seismic vibration was only 0.11 in/s, well within industry limits but the shaft vibration reached the trip set point of 6 mils pk-pk overall before full speed/load conditions were reached. Fan housing/seismic vibration was acceptable per AMCA 204-05 Class BV-4 criteria. Motor solo shaft vibration was consistently acceptable at only 1.5 mils pk-pk overall.

Initial motor solo vibration of the first motor was excellent with casing amplitudes below 0.1 in/s and a maximum of 1.2 mils pk-pk shaft displacement. The motor was then coupled and plant startup was commenced. When the fan reached ~ 450 RPM the motor inboard shaft vibration climbed to 4.5 mils with an Alarm of 3.5 mils. By 900 rpm, the Trip set point of 6 mils was reached (bearing clearance of 11 mils). Motor bearing seismic vibration was acceptable during the entire speed range per the motor vendor specification as well as NEMA MG1.

Coupling with Elastomeric elements removed. Note the flange on the left side for separating the coupling halves

Coupling elastomeric elements removed from coupling and marked with their diameters measured.

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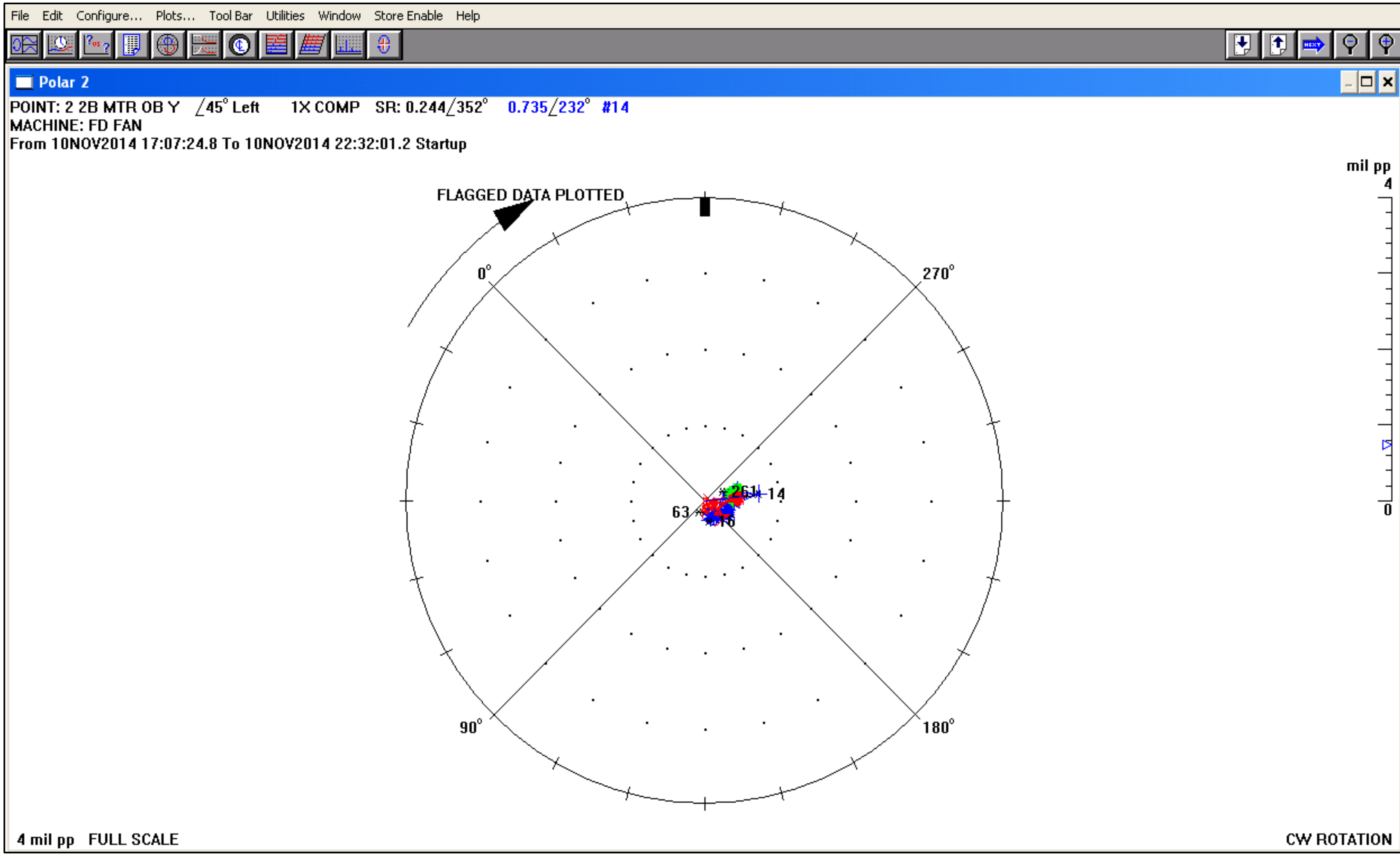

Initial Coupled Run - Motor OB Channel Y Shaft Relative

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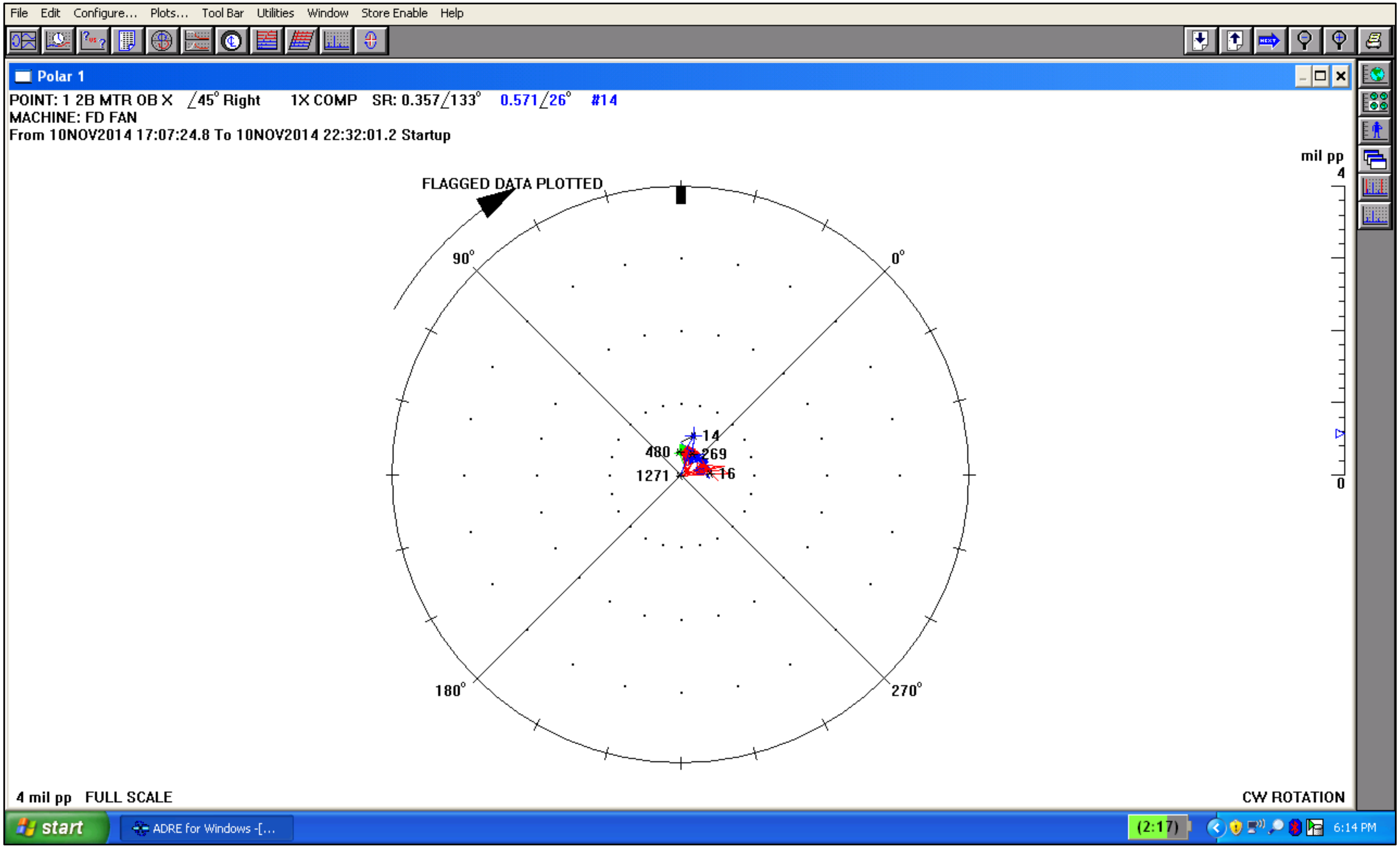

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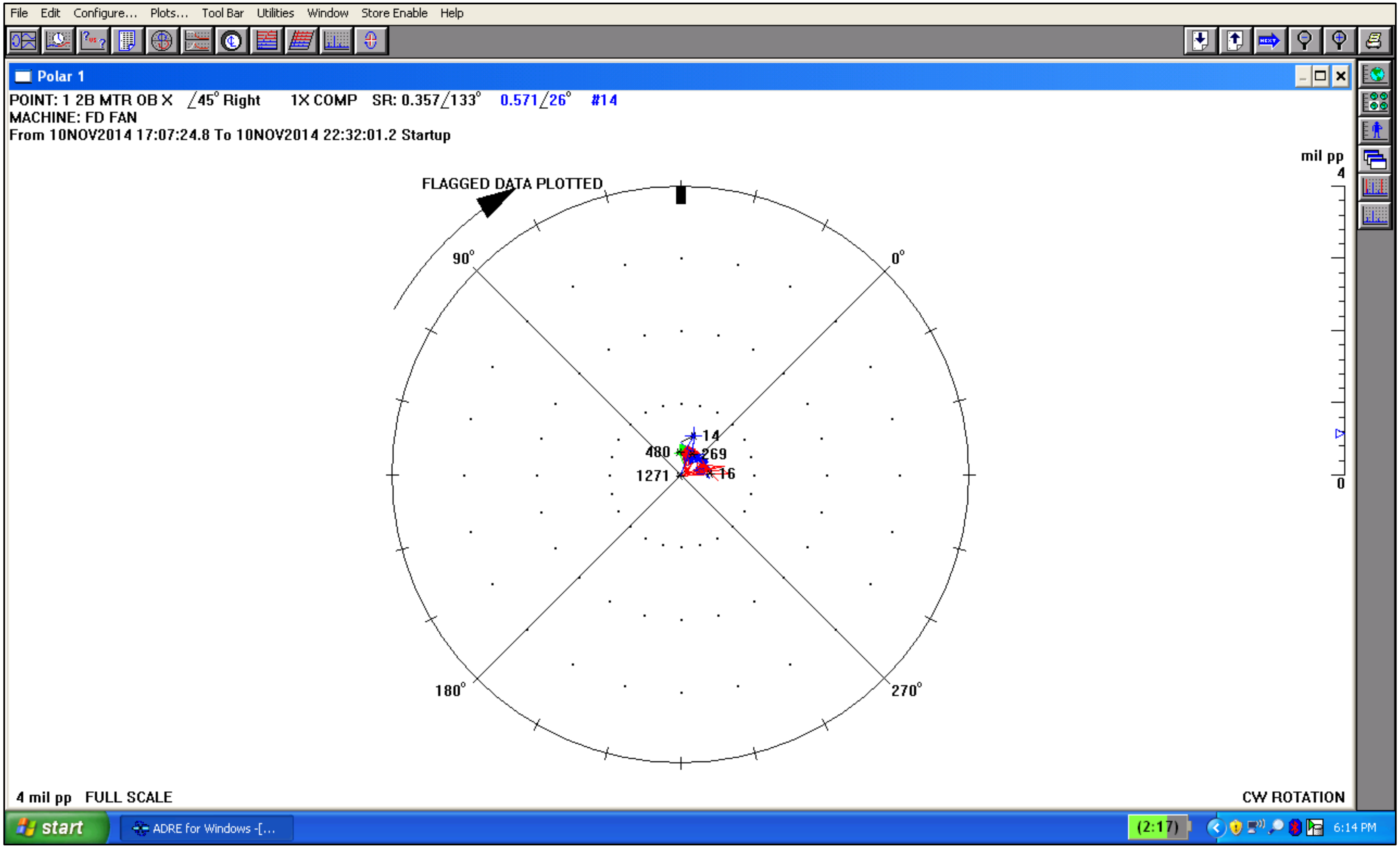

Initial Coupled Run - Motor OB Channel X Shaft Relative

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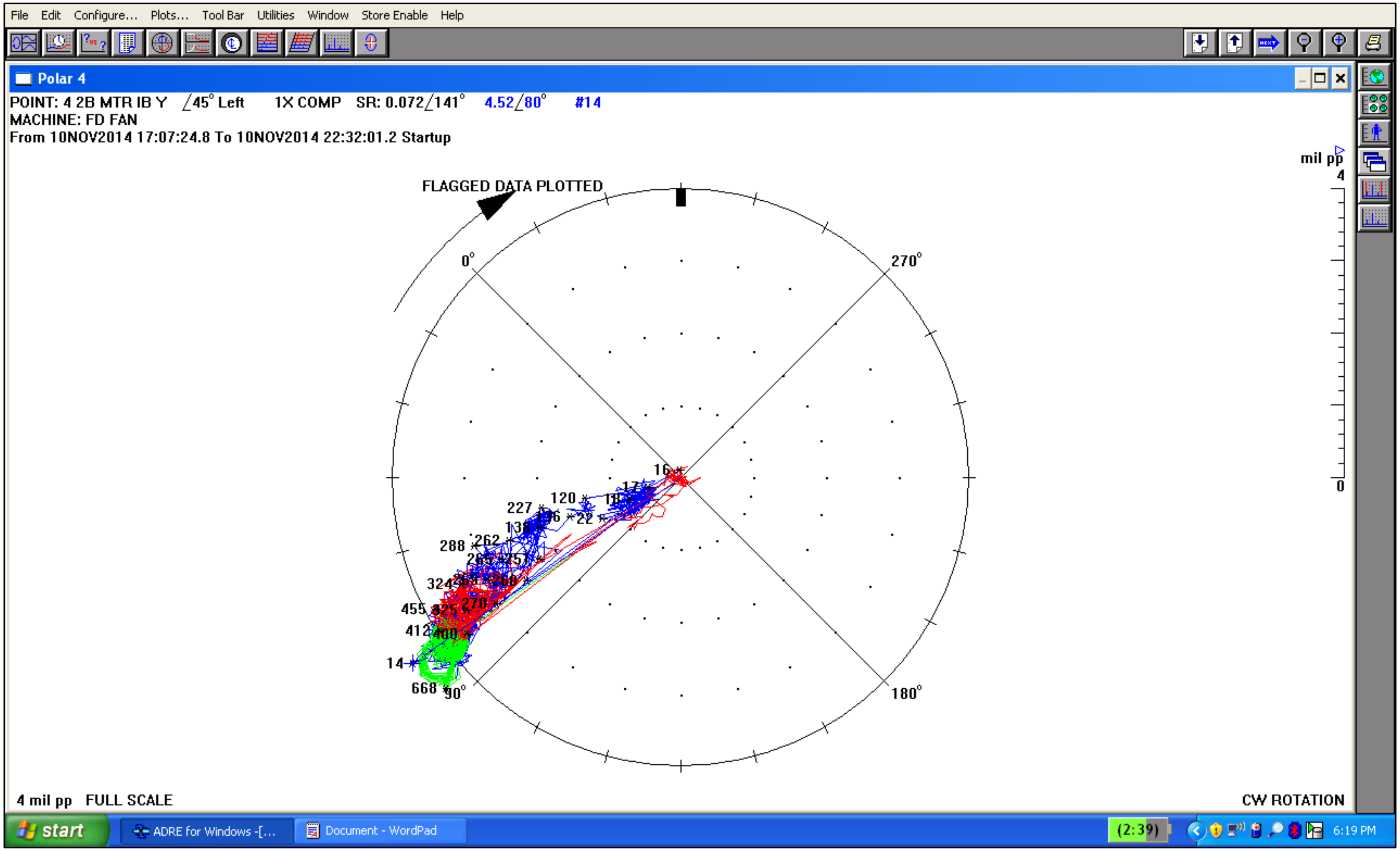
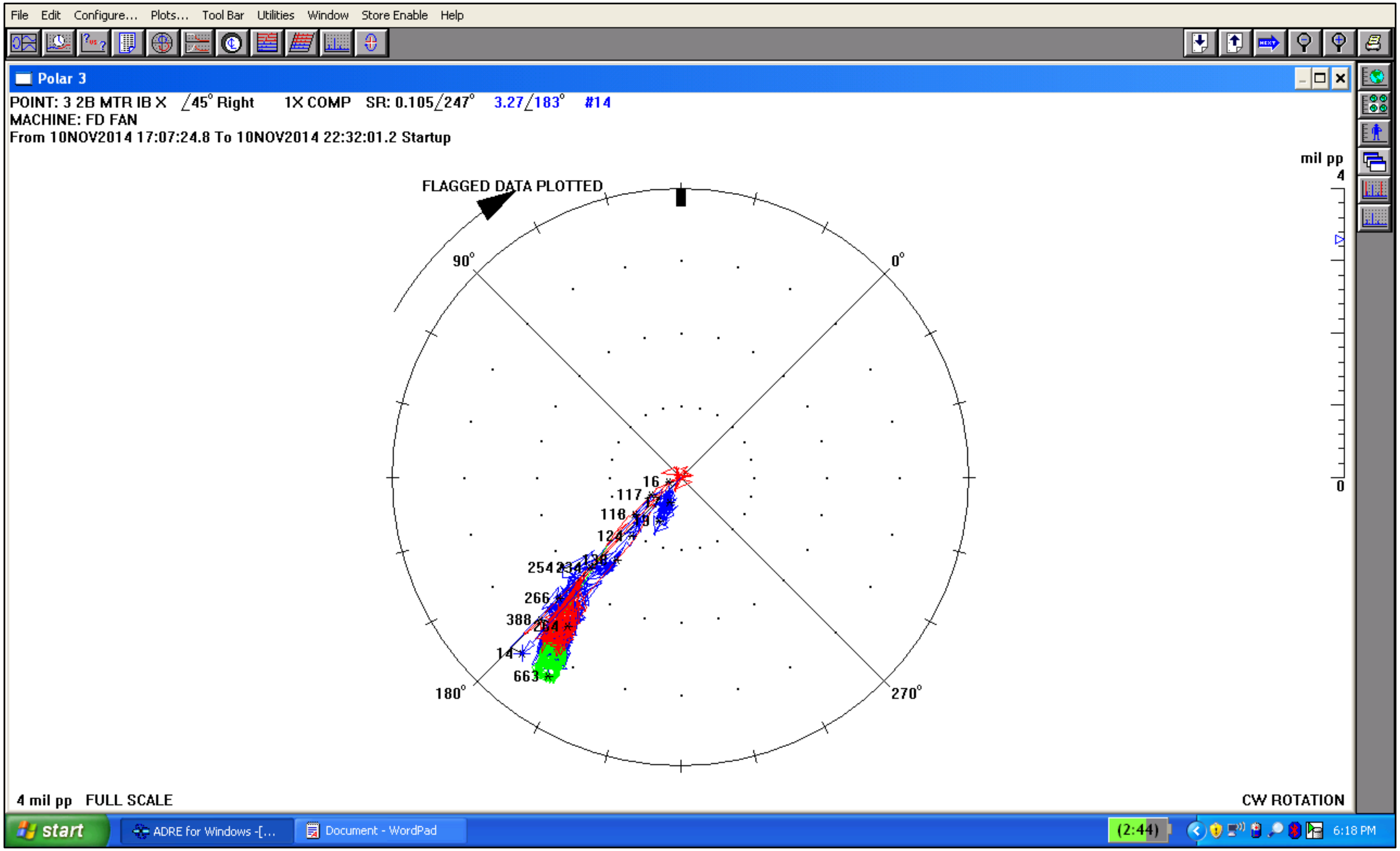


Initial Coupled Run - Motor IB Channel Y Shaft Relative

Note how motor IB-Y vibration levels increase at relatively constant phase (~ 80 deg) with increasing speed.



Initial Coupled Run - Motor IB Channel X Shaft Relative

Note how motor IB-X shaft vibration levels increase at relatively constant phase (~ 183 deg) with increasing speed.

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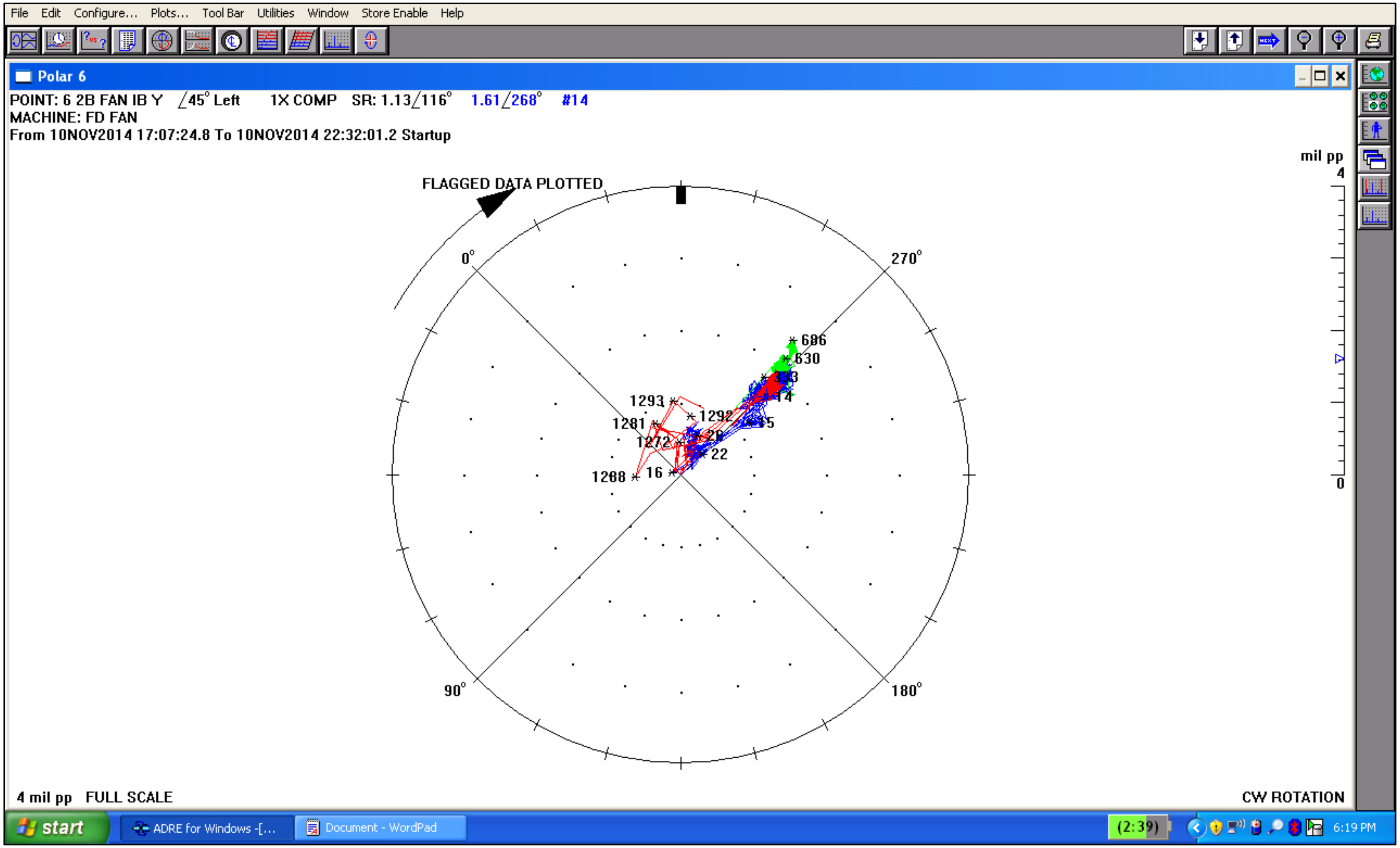

Initial Coupled Run - Fan IB Channel Y Shaft Relative

Note how fan IB-Y shaft vibration levels increase at relatively constant phase (~ 268 deg) with increasing speed. Note also how an ~ 180 deg phase difference exists between the motor & fan across the coupling (ie: compare 268 to 80 deg).

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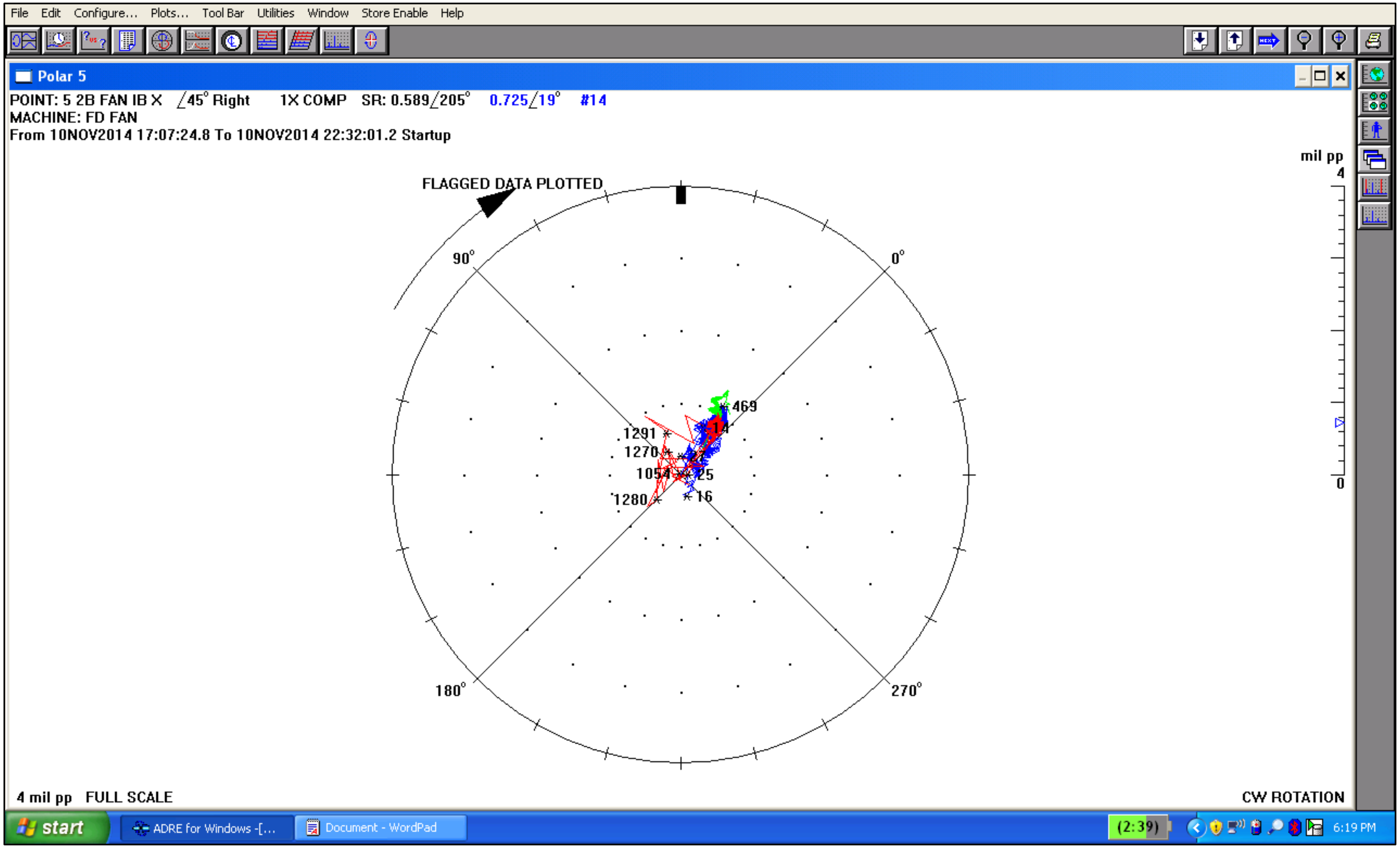

Initial Coupled Run - Fan IB Channel X Shaft Relative

Note how fan IB-X shaft vibration levels increase at relatively constant phase (~ 19 deg) with increasing speed. Note also how an ~ 180 deg phase difference exists between the motor & fan across the coupling (ie: compare 183 to 19 deg).

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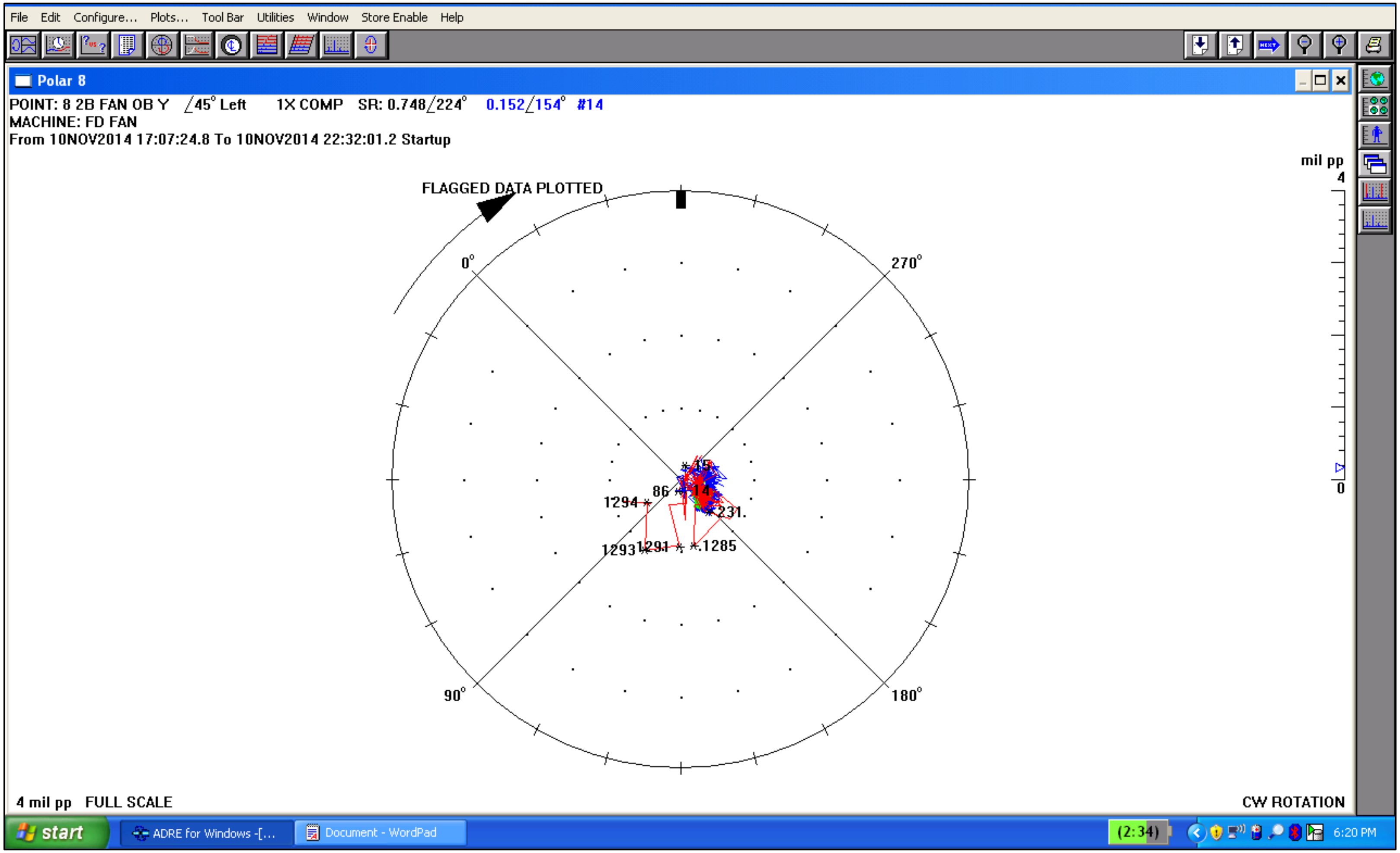

- **Initial Coupled Run - Fan OB Channel Y Shaft Relative**

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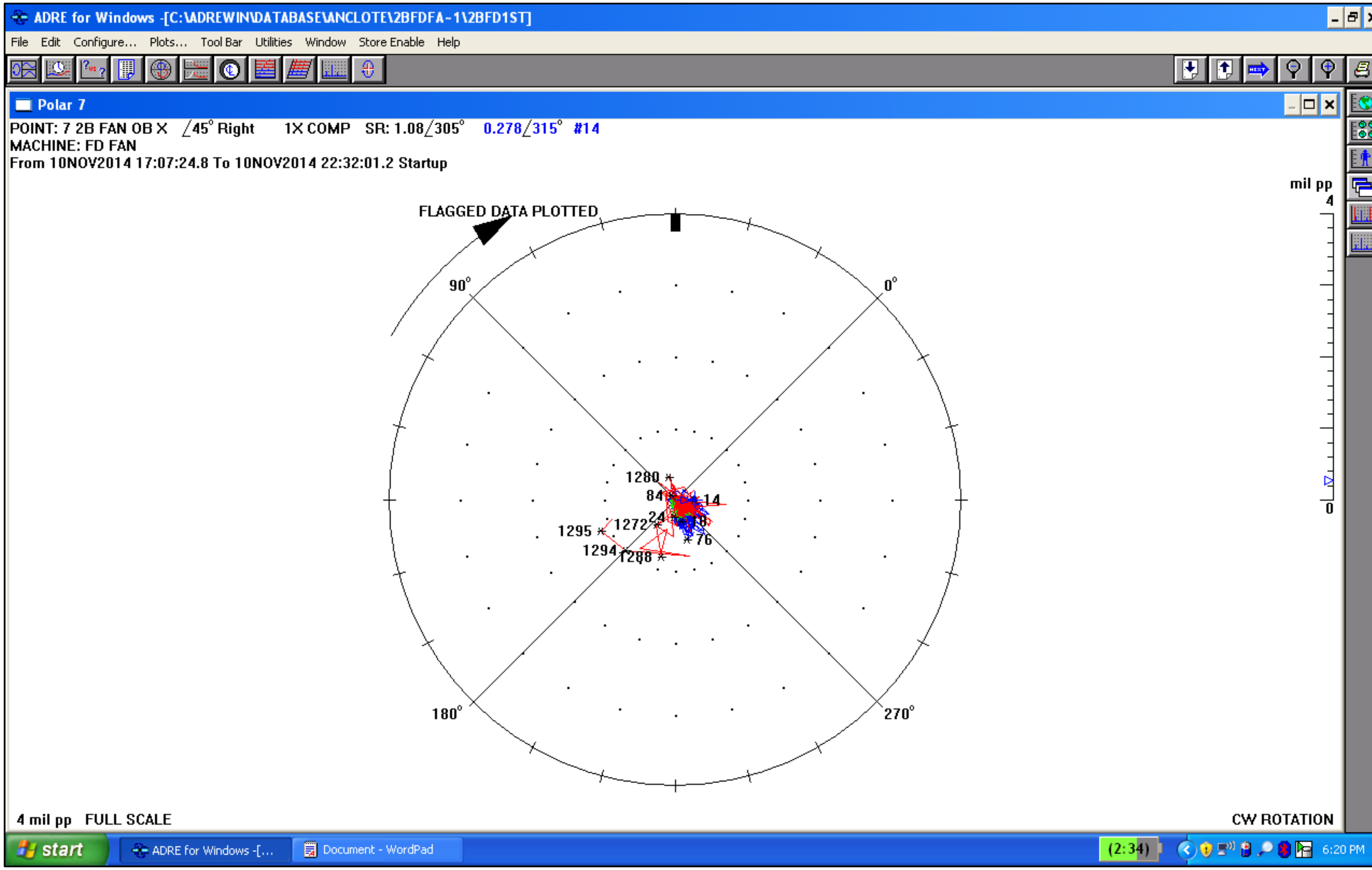

- **Initial Coupled Run - Fan OB Channel X Shaft Relative**

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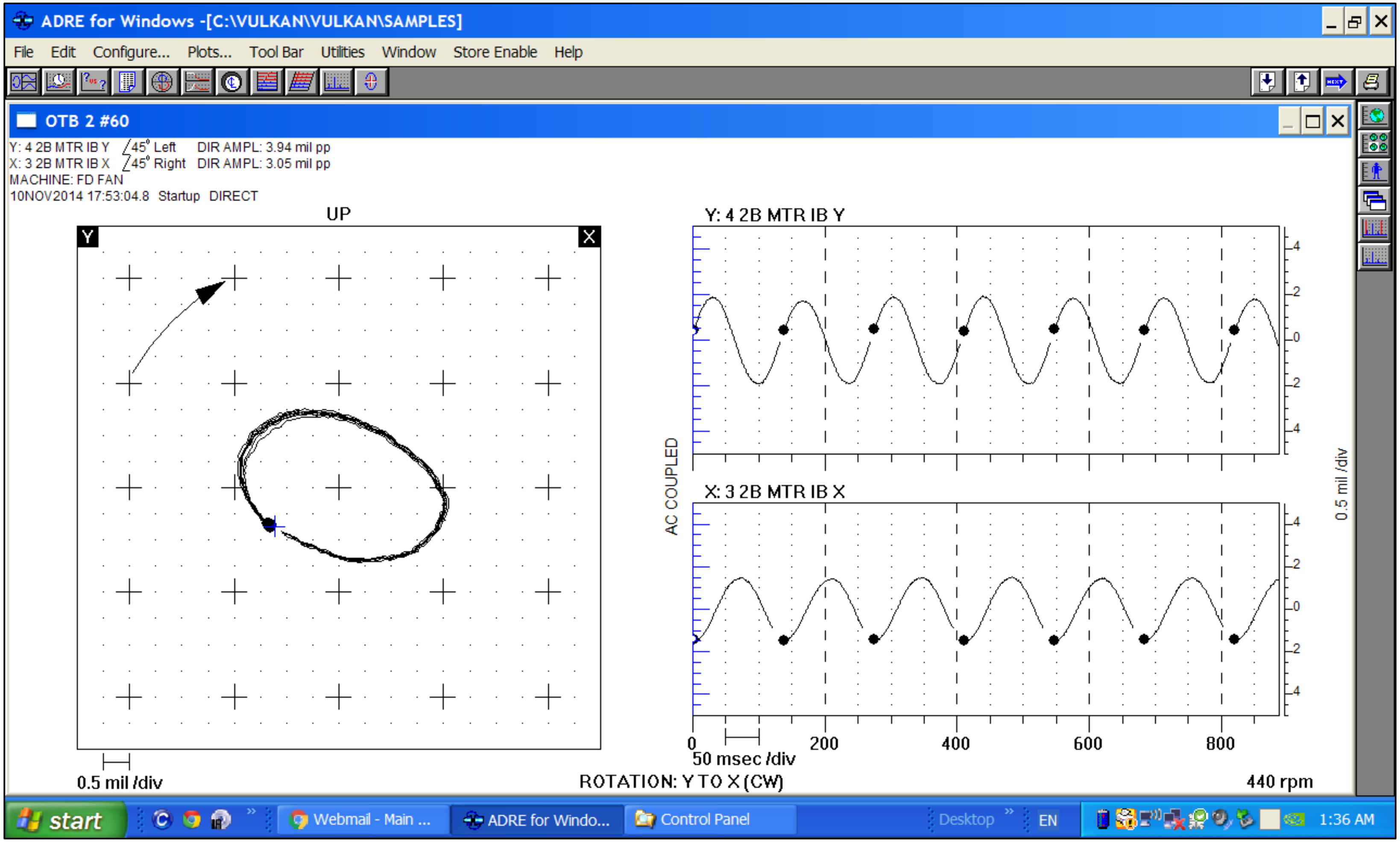

Initial Coupled Run - Motor Inboard Orbit & Time Waveforms
Note normal looking orbit (elliptical) and sinusoidal waveforms
Note dominant vibration at 1x rpm (440 cpm)

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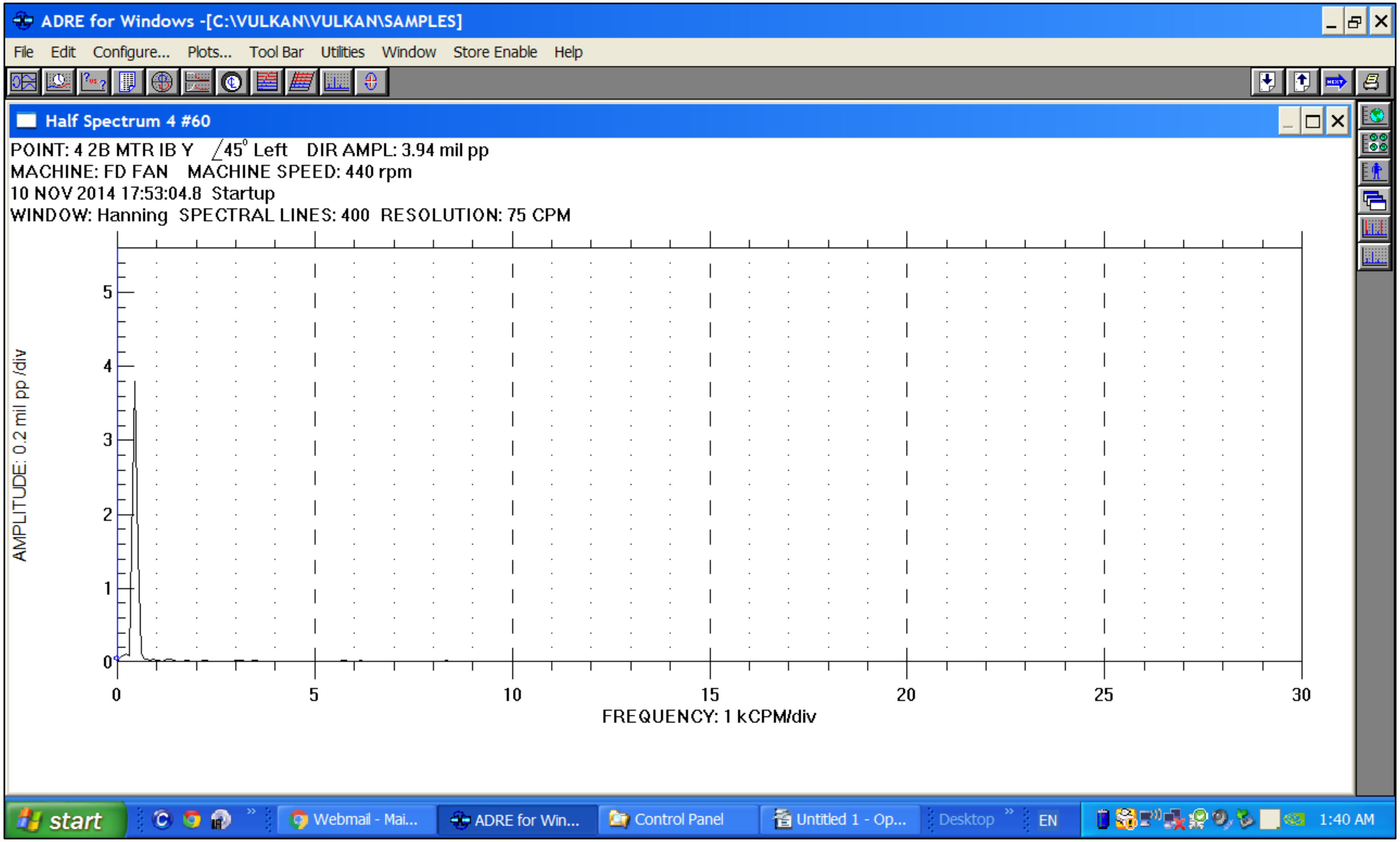

Initial Coupled Run - Motor Inboard Y Spectrum

Note dominant vibration at 1x rpm (440 cpm)

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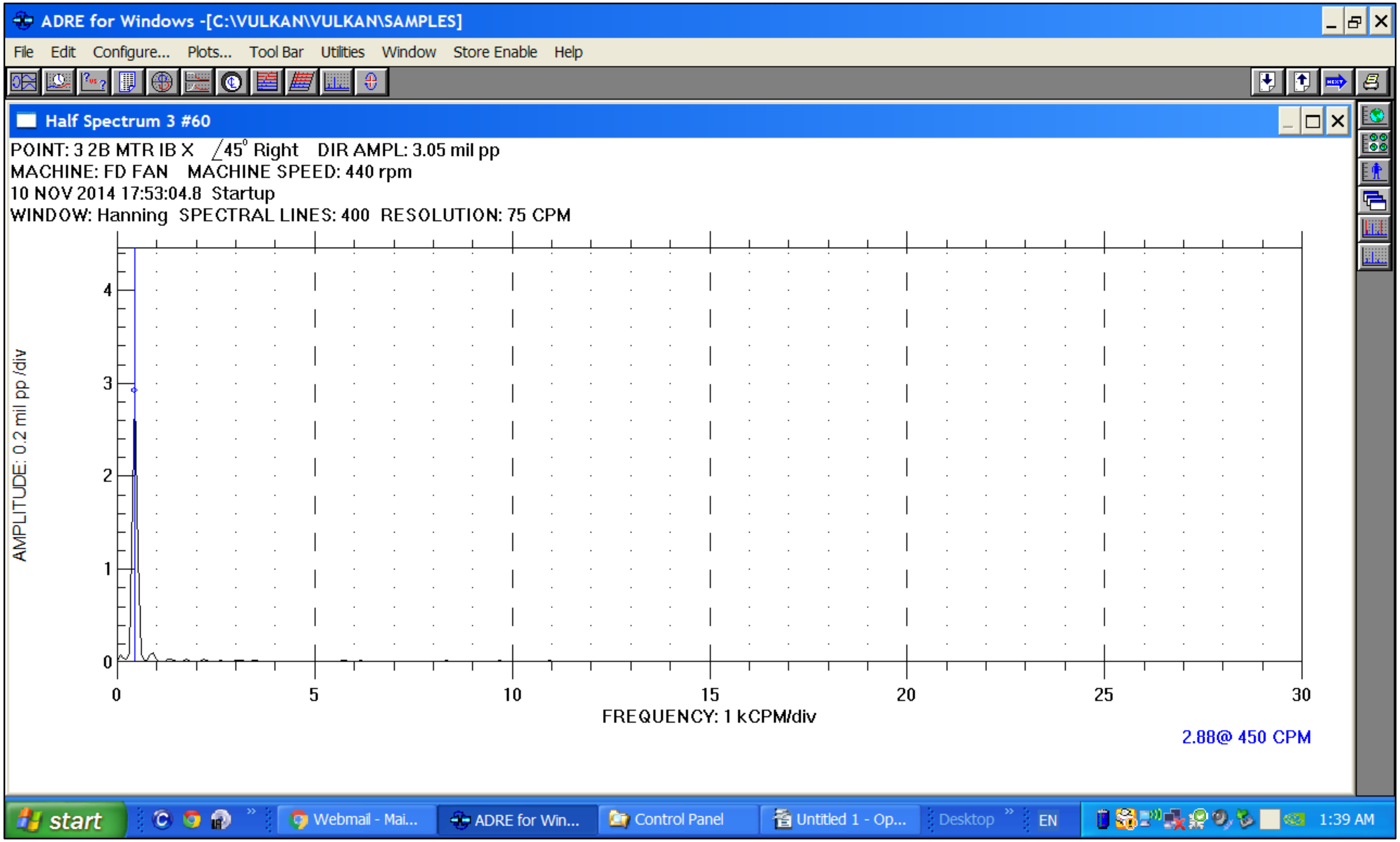

- **Initial Coupled Run - Motor Inboard Spectrum X**

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- **<u>ANALYSIS</u>**
  The initial concern was with the new Bently 3500 system and questions about the installation and calibration began. Installation documents showed that the Bently system was installed correctly and properly calibrated. None-the-less probes were pulled and new calibration curves were run, again with acceptable results. Very little "glitch" was observed indicating the target areas were void of defects that would appear as high vibration

  The fan was restarted and the same vibration amplitudes occurred; unit output was again limited by motor shaft vibration. A Bently ADRE system was used to collect shaft vibration data. Again the motor shaft vibration trended up steadily with fan speed and we saw that almost all of the energy was at motor 1X. Unbalance or misalignment were now considered possible causes. The 1X phase data of the corresponding inboard motor and inboard fan shaft channels showed significant change of the 1X vector. Motor inboard channels move 180 degrees (or away) from the corresponding Fan IB channels. Misalignment was investigated first.

- The alignment was checked using a laser system and found to be in specification using the OEM recommended thermal targets. In addition, the shaft and coupling hub runouts were checked and were also within specification. (Laser alignment systems are great but too often we forget that a dial indicator check of the shaft and coupling hub are still necessary). Some suggested that misalignment from thermal growth was possible and recommended a study to validate the thermal targets used; this seemed a reasonable position given the circumstances. From the ADRE polar plot data, we saw that the motor 1X vector returned quickly and consistently with lower speed operation suggesting that this was a mechanical and not a thermal problem. But what about this crazy phase shift?

  The ADRE Shaft Center Line plots showed no anomalies nor did we have any significant slow roll levels (AKA Glitch), again suggesting that the alignment was acceptable. But what about unbalance?

- More discussion transpired on the topic of motor, fan and coupling unbalance between the vendors and plant staff. Our investigation found acceptable balance records for the motor and fan however it was discovered that the coupling was not dynamically balanced. The vendor claimed that balancing of the coupling was not necessary per industry standards. The coupling is machined from 3 cast iron parts, two of which have a registered fit. The coupling transfers power through 14 elastomeric elements, seven in the forward direction and the other seven for reverse loads. Inspection of the registered fits was performed and found to be concentric and within specs. The fan and coupling vendor pressed the motor vendor for acceptance of the high shaft vibration based on acceptable motor housing/seismic vibration using industry standards, however the motor vendor held firm and pushed for more diagnostics. But what was left to investigate?

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- To determine if the fan or motor was the vibration source I recommended we “index” the coupling by 90 degrees, basically rotating one half of the coupling at the flange. If the source of the problem was on the fan side of the coupling, the phase angle would remain the same since the Key Phasor is mounted on the Fan outboard end. If the motor side of the coupling was the source, which includes the elastomeric elements, then the phase angle would change. Result – the phase angle changed by 100 degrees after indexing the coupling. This indicated that the motor half of the coupling with the elastomeric elements were the source of our vibration problem.

- At this point we had eliminated motor unbalance, fan unbalance, misalignment, and misalignment due to thermal growth as causes of the speed dependent high vibration. Rotating the coupling 90 degrees told us that the motor side needed more investigation. The next leading theory was that the coupling was not transferring the torque symmetrically but instead it was producing some net radial force. How would that happen?

- **<u>ELASTOMERIC ELEMENTS</u>**
  Our attention turned to the elastomeric elements since all of the coupling checks found no issues. Basic inspection of the elements began and we documented the length and diameter of each element and compared this data to the vendor drawings. One major deviation came to light: the diameter of the elements were found significantly out of spec with a variation of 0.039" while the drawing limit was 0.011". With little time or spare parts we decided to place the seven closest sized elements in the seven forward drive slots and the remaining seven elements were placed in the reverse drive slots.

- Success, the fan reached 1,195 RPM with shaft vibration levels well below the 3.5 mils Alarm set point. But wait, there is more. The next week we performed the same checks and element assignments with an installed variance of only 0.004" to the second fan with no success; the inboard motor shaft vibration reached the trip set point of 6 mils before 1,000 RPM. We must have overlooked something.

Coupling with elastomeric elements installed and the retaining ring not yet in place.

- **COMPREHENSIVE ELEMENT INSPECTIONS**
  Since we planned to install four new fans the site received four full sets of elements, so we began comprehensive inspection & testing of the three sets that were not yet installed. Each set was assigned a letter and each element was given a unique number. Mechanical inspections included length, diameter, weight, and hardness using the Shore A scale and the results were compared to the vendor specs. One additional deviation came to light: the hardness should be a 95 on the Shore A scale and within 2, however what we found was a range of 92 to 95.5. Maybe this was the source of our coupling radial force? I decided to do a “modified" compression test. We needed to know the hardness of the complete elastomeric element cross section and not just the hardness of the surface. Remember that the seven forward drive elements are transferring up to 7,500 HP at 1,200 RPM (or about 33,000 ft. lbs. of torque).

- **MODIFIED COMPRESSION TEST OF ELEMENTS**
  Since the Shore A durometer test only checks about 0.070" of depth below the surface I had to find a test method that would approximate the change of the elements when in service. I decided to use a standard automotive valve spring tester and apply the maximum force of 1,000 lbs. While this would be only about half of the service loading, it seemed a reasonable method. Valve spring testers are not typically used on everyday automotive engines but are reserved for racing engines where high rpm and high valve lift cause potentially damaging stresses. The process also required that we quickly documented the reading as urethane continues to compress over time with load and this could skew the results. Every element was tested, marked and cataloged. The loaded diametric range found was 2.022” to 2.064”. A table of the data was made to more easily find the seven closest matching elements when under simulated load conditions.

Automotive Valve Spring Tester with one element being tested. Note the large dial is the pressure reading and the small dial is the element diameter.

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ID	1KPSI	SHORE	WT	DIA	LONG
A1	2.06	95.5	191	2.155	2.825
A2	2.042	95.5	190	2.154	2.827
A3	2.038	93	190	2.154	2.828
A4	2.041	92	191	2.154	2.826
A5	2.049	94	191	2.052	2.829
A6	2.048	95	190	2.154	2.822
A7	2.031	93	190	2.166	2.822
A8	2.045	94	191	2.153	2.829
A9	2.064	95	190	2.153	2.824
A10	2.046	94	190	2.153	2.824
A11	2.059	95	191	2.154	2.823
A12	2.045	93.5	191	2.154	2.82
A13	2.059	95	191	2.155	2.823
A14	2.038	93.5	191	2.155	2.825

Results For Group "A"

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ID	1KPSI	SHORE	WT	DIA	LONG
B1	2.033	93	191	2.156	2.827
B2	2.057	95.5	190	2.155	2.823
B3	2.038	94	190	2.154	2.827
B4	2.031	93.5	191	2.154	2.83
B5	2.044	94.5	189	2.156	2.806
B6	2.05	93.5	191	2.155	2.828
B7	2.039	94	191	2.256	2.824
B8	2.035	93.5	190	2.156	2.829
B9	2.022	93	191	2.153	2.825
B10	2.058	96	190	2.153	2.82
B11	2.03	93	190	2.156	2.823
B12	2.051	94	191	2.153	2.83
B13	2.059	95.5	191	2.155	2.808
B14	2.035	93.5	190	2.156	2.826

Results For Group "B"

The 1,000 PSI values were used on groups "A" and "B" to successfully match the 7 drive elements. For group "C" the elements were only compression tested as the other parameters were deemed not significant .

FINAL RESULTS & CONCLUSIONS

The compression matched elements were installed in the second fan and have performed well. This process was repeated for both of the unit 2 fans and they have operated at full load/speed well below the shaft vibration alarm limit of 3.5 mills.

The asymmetric stiffness of the seven forward drive elements caused the coupling halves to “pivot” and become effectively misaligned. This crank shaft type effect is believed to be larger on the motor because the motor rotor is lighter than the fan rotor. Any elastomeric coupling using multiple and separately produced elements could suffer the same problem if the stiffness/compression quality varies significantly. Couplings using a single elastomeric element formed by a single molding process should not cause vibration problem of this type. The compression method of matching elements has become our standard quality test to match elements and maintain symmetric coupling loading.

Proximity probes are the industry standard for machines operating in sleeve bearings, however our vibration industry has too frequently viewed seismic data as the most representative of machine condition - this is usually an error.

Bio

Talmadge Ward is a Senior Engineering Technologist with Duke Energy Florida. He is also Chairman of the Central Florida Chapter of the Vibration Institute and owns a small business that buys, sells and rents vibration analysis equipment - VibRental LLC. Talmadge has worked in the vibration field since 1986, been a Vibration Institute member since 1989 and has a US Patent for a balancing method.

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