The Dirty Dozen

The 12 Most difficult problems I have worked on during the last 45 years of troubleshooting vibration related problems

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INTRODUCTION:

There are several aspects of a troubleshooting assignment that contribute to the difficulty of identifying the root cause of a problem and in coming up with a solution.
In my opinion, the following are the main issues that can contribute to the difficulty of finding a clear path that proceeds from determining what is causing a problem to its solution:
1: <u>Technical difficulty in obtaining the proper data to identify the source</u>. Getting good torsional vibration is an example of where it is difficult to get the needed information. Getting a good reliable tach signal is another problem that is often encountered when attempting to get useful data.

2: <u>The complexity of the problem itself</u>. This can be compounded if there is more than one contributing factor. For instance, a combination of acoustical and structural resonances can significantly complicate the situation.

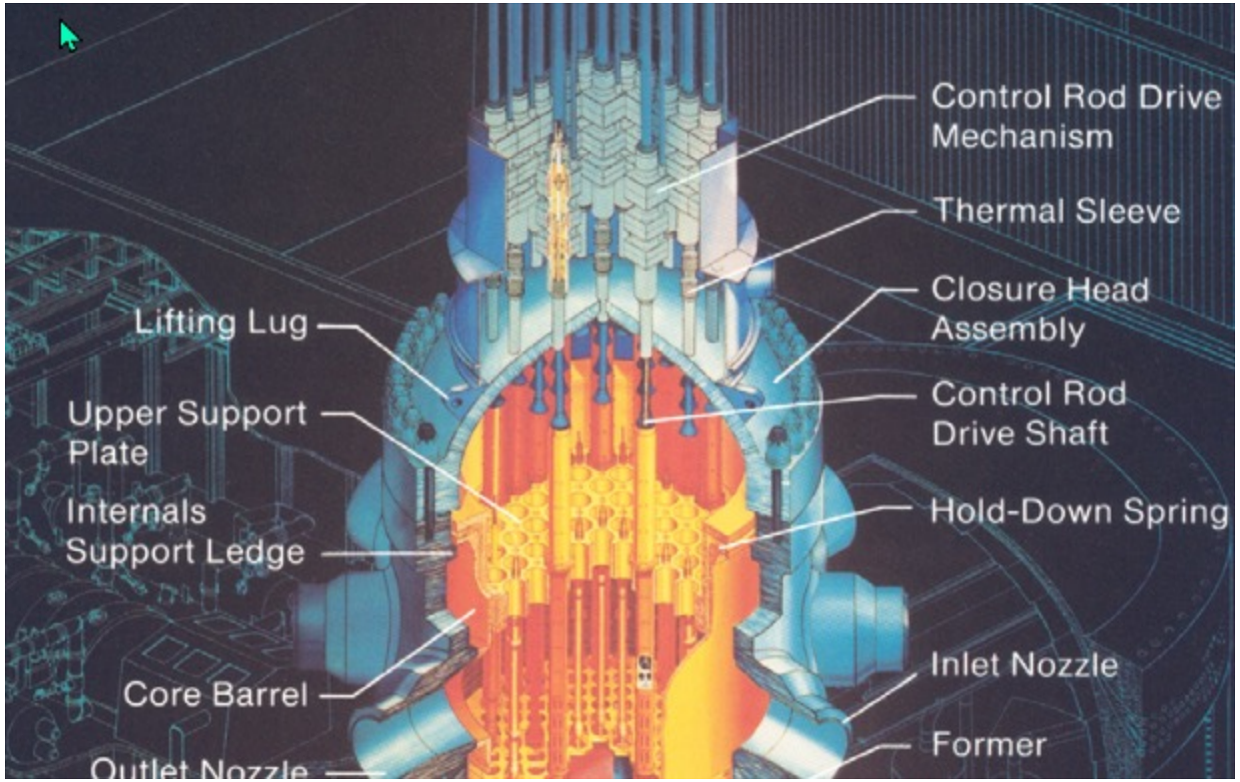
3: <u>The physical environment</u>. This can cover anything from the temperature, humidity, deafening sound levels, cleanliness, noise, smoke, rain, snow to even electrical interference. A challenging physical environment can make it very difficult to get data and significantly hamper communication between those doing the field work.

4: **Lack of machine information**. Bearing numbers, gear tooth counts, number of vanes on a pump, number of rotor bars on a motor, pump suction and discharge pressures and electrical current readings can be vital in determining the root cause of a vibration problem. In some instances, this technical data can be difficult to find. A facility may not have any manuals of the manufacturer of the equipment may have gone out of business.

5: Time Pressure- Some jobs are under severe time pressure. This often is the case when working on large steam turbine generators. The costs associated with when they are down can run into the hundreds of thousands of dollars per day. In certain nuclear power plant related problems, if there is a tech spec violation on a machine that is deemed necessary for the safe operation or shut down of the reactor, then there is a limited time in which the problem must be resolved, otherwise they must shut the plant down. The cost can quickly get into the millions.

6: Political Pressure- If there is a lot of pressure coming from above that can be very detrimental to finding the most efficient path forward. A harsh political environment or one in which management is looking for someone to blame can cause the flow of information to come to a standstill. In many instances operations or maintenance may have vital information. If the flow of that information is interrupted, then finding the source of the problem may become very difficult or even impossible to determine.

12th Most Difficult Case



CRDM FAN AT NUCLEAR POWER PLANT

- Question:
  - What does nuclear fission and organic chemistry have to do with vibration analysis?

- A nuclear power plant called with a problem. When the plant was down for an outage they tested one of their Control Rod Drive Mechanism Cooling Fans. The fans move very clean air to cool the Control Rod Drive Mechanisms that lower the control rods into the nuclear reactor.

Question

- Why had vibration on the fan tripled?
  - The fan wheel was clean
  - No weights had been thrown of the wheel
  - The wheel did not have any cracks

Getting To The Answer

- The fan is exposed to high energy gamma rays from the fission reaction. The oxygen in the water when exposed to neutron radiation forms N16 which has a 7.1 second half life and gives off a very high energy 6 Mev gamma ray.

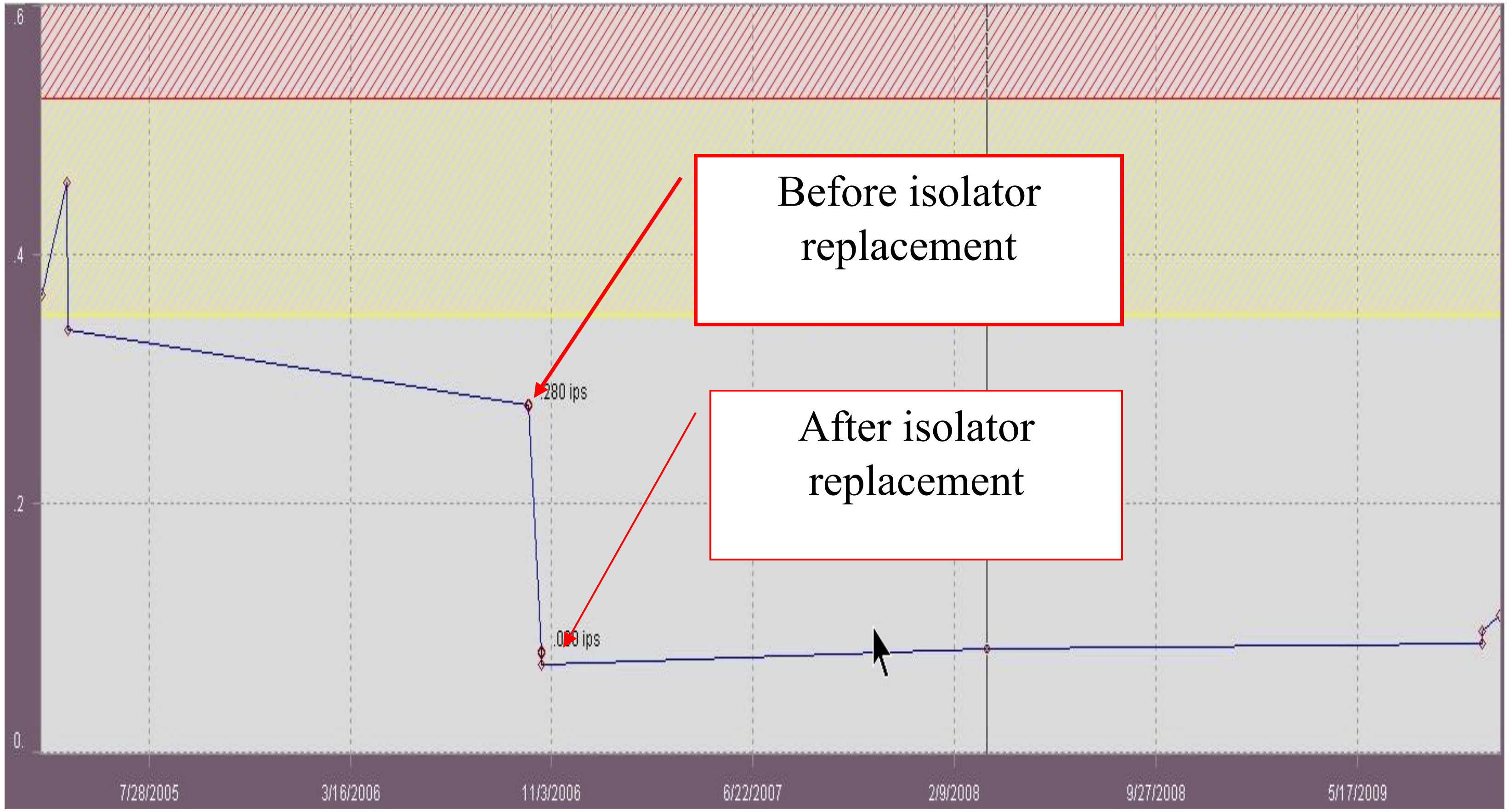
Organic Chemistry

- Rubber is an organic compound. Being such, it means that the chemical bonds are covalent bonds which are weaker than ionic bonds meaning that they are more easily broken. Humans are organic based that is why our DNA is damaged by gamma rays.

Putting the pieces together

- The fan was mounted on rubber isolators.
- The gamma rays broke the organic bonds making the rubber hard and brittle.
- The isolators stiffened up raising the natural frequency of the isolated system to 28 HZ which was very near the 30 HZ operating speed.
- That caused the tripling of the vibration
- This problem was put on the list because it required knowledge in three separate areas. Gamma radiation from fission process, organic chemistry and the amplification factor.

Before and after Isolator Replacement



11^{th} Most Difficult Case
Turbine Driven Boiler Feed Pump that was failing seals on the pump's inboard side as well as failing couplings

Boiler Feed Pump and Turbine Driver

This turbine driven pump was located in a new power plant. After a short amount of operating time, the inboard pump bearing failed along with the inboard pump seals and the coupling. An alignment problem was suspected. In order to study how the alignment moved with load, Dodd bars were installed. DODD Bars consist of brackets with targets mounted on one machines and bars with proximity probes looking at the targets mounted on the other machine.

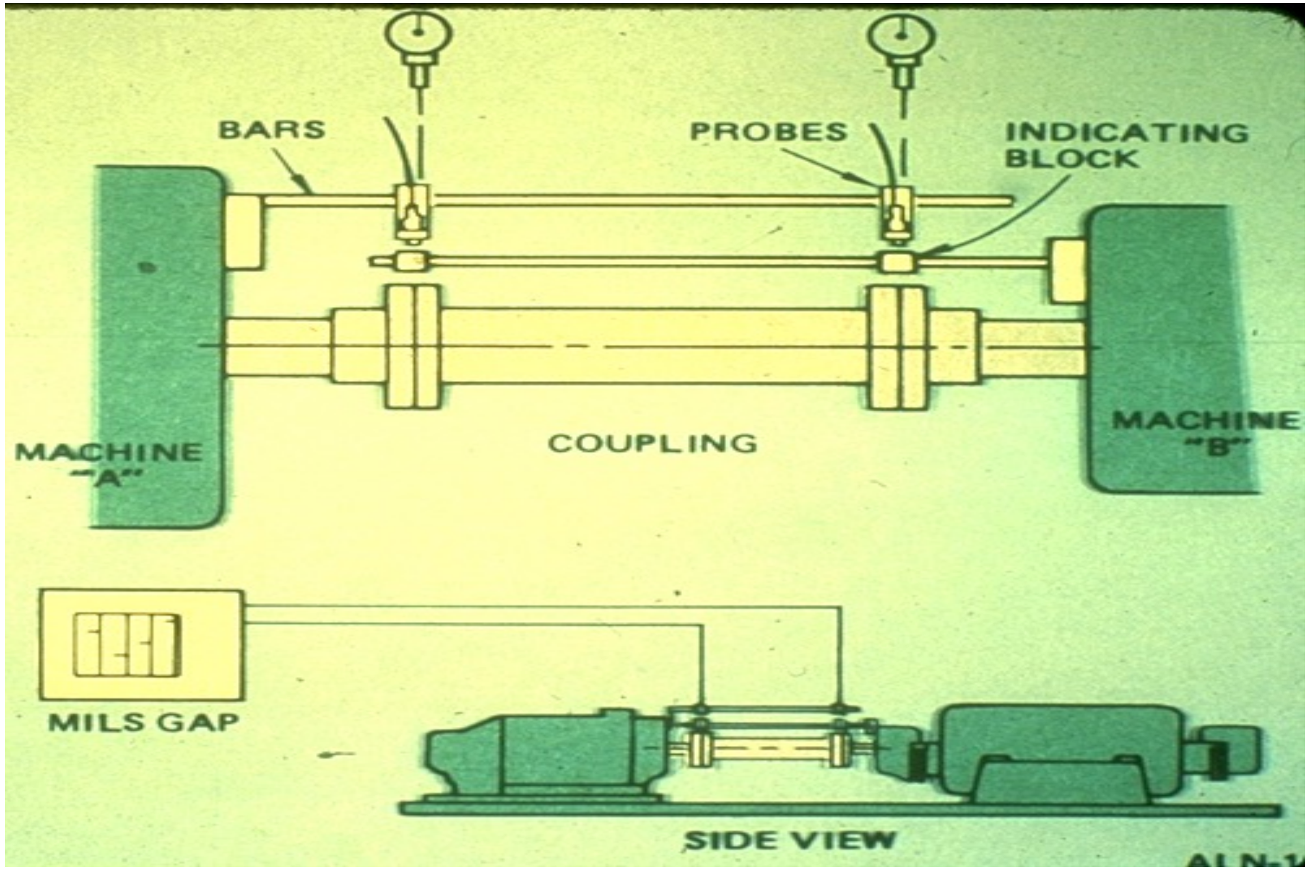
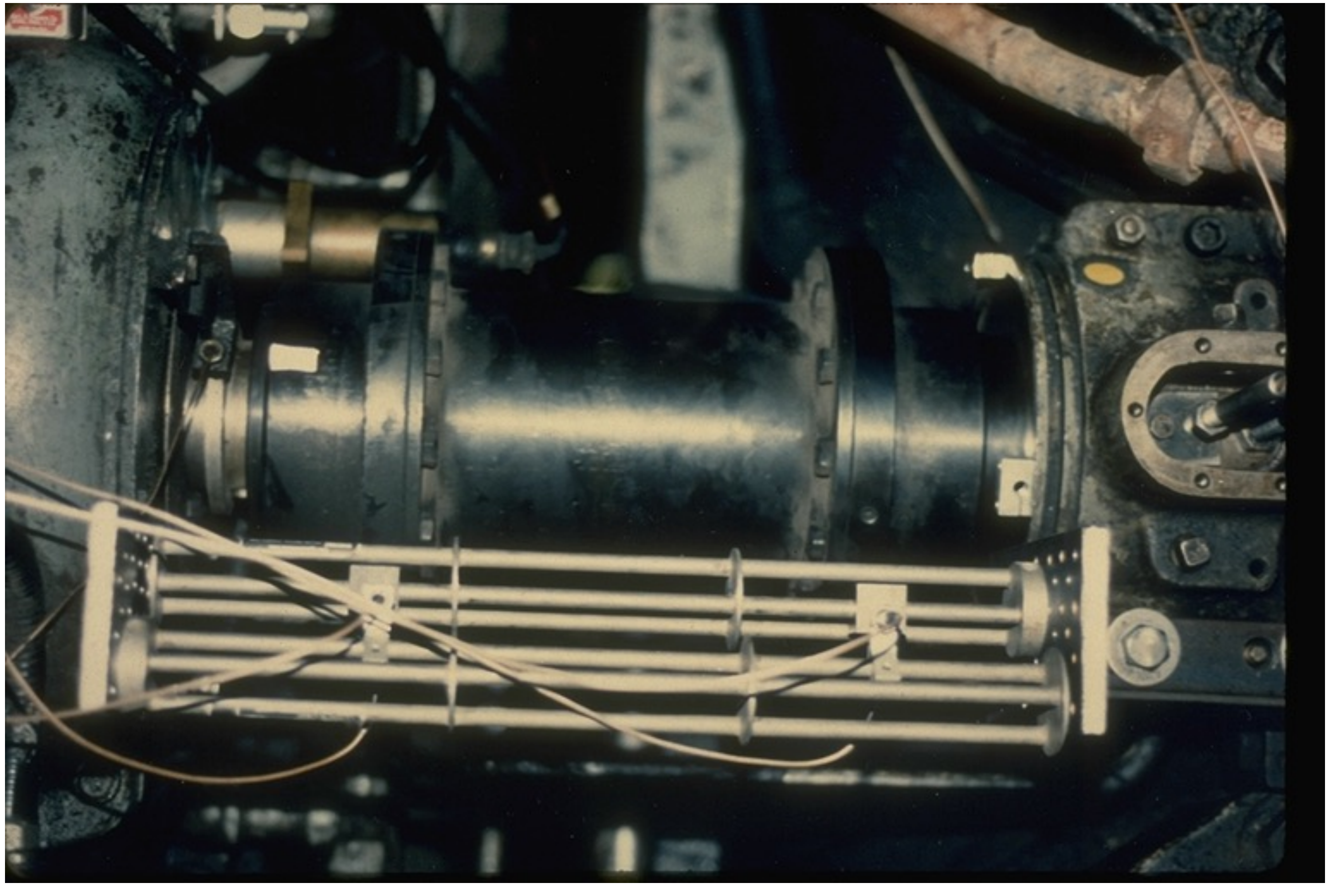


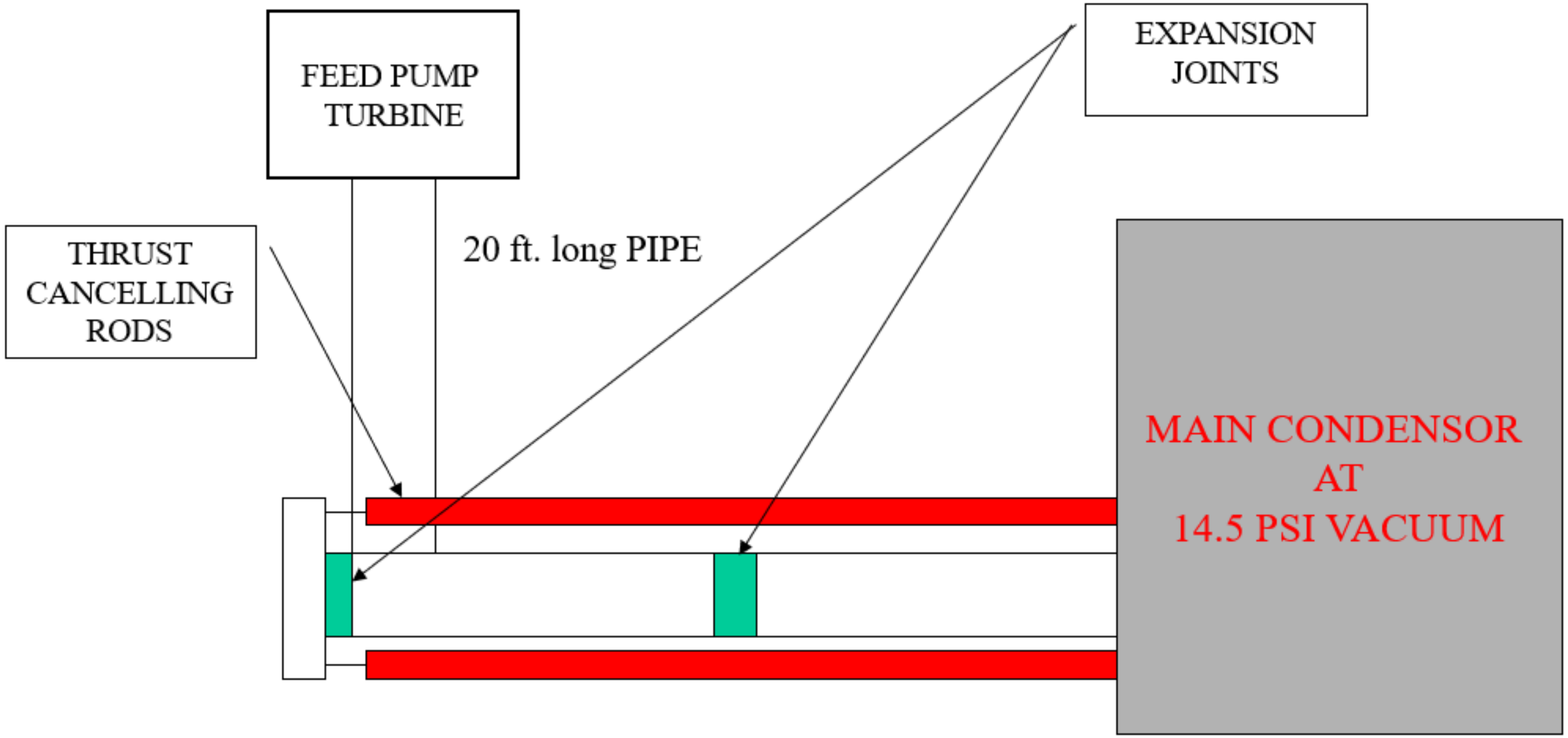
Photo of Dodd Bar Installation

By monitoring the changes in the DC gap voltages and graphing the results it was possible to measure the amount of vertical and horizontal movement of the turbine relative to the pump.

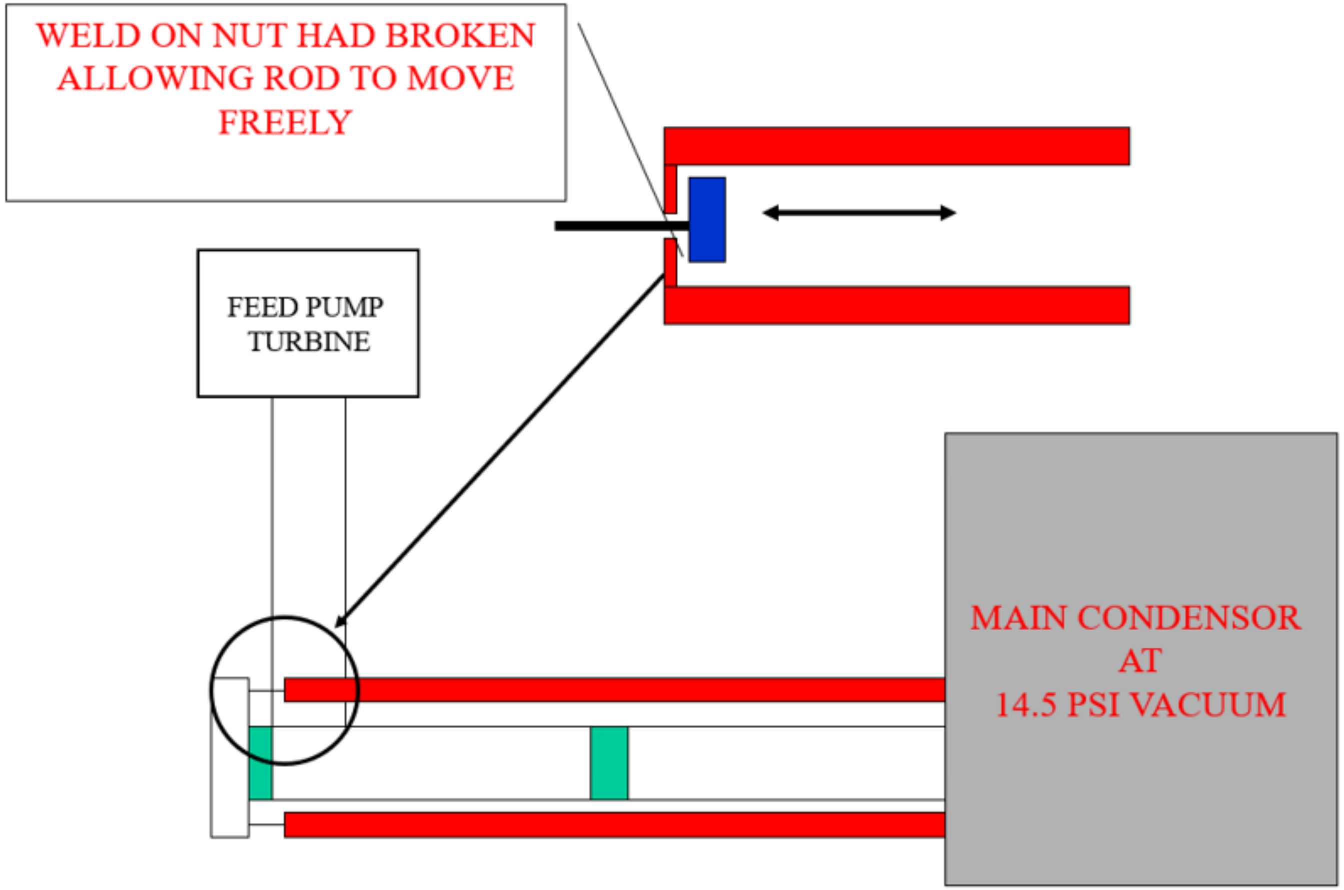


When Vacuum was broken, the alignment moved over 100 mils.

Diagram of feed pump and condenser piping

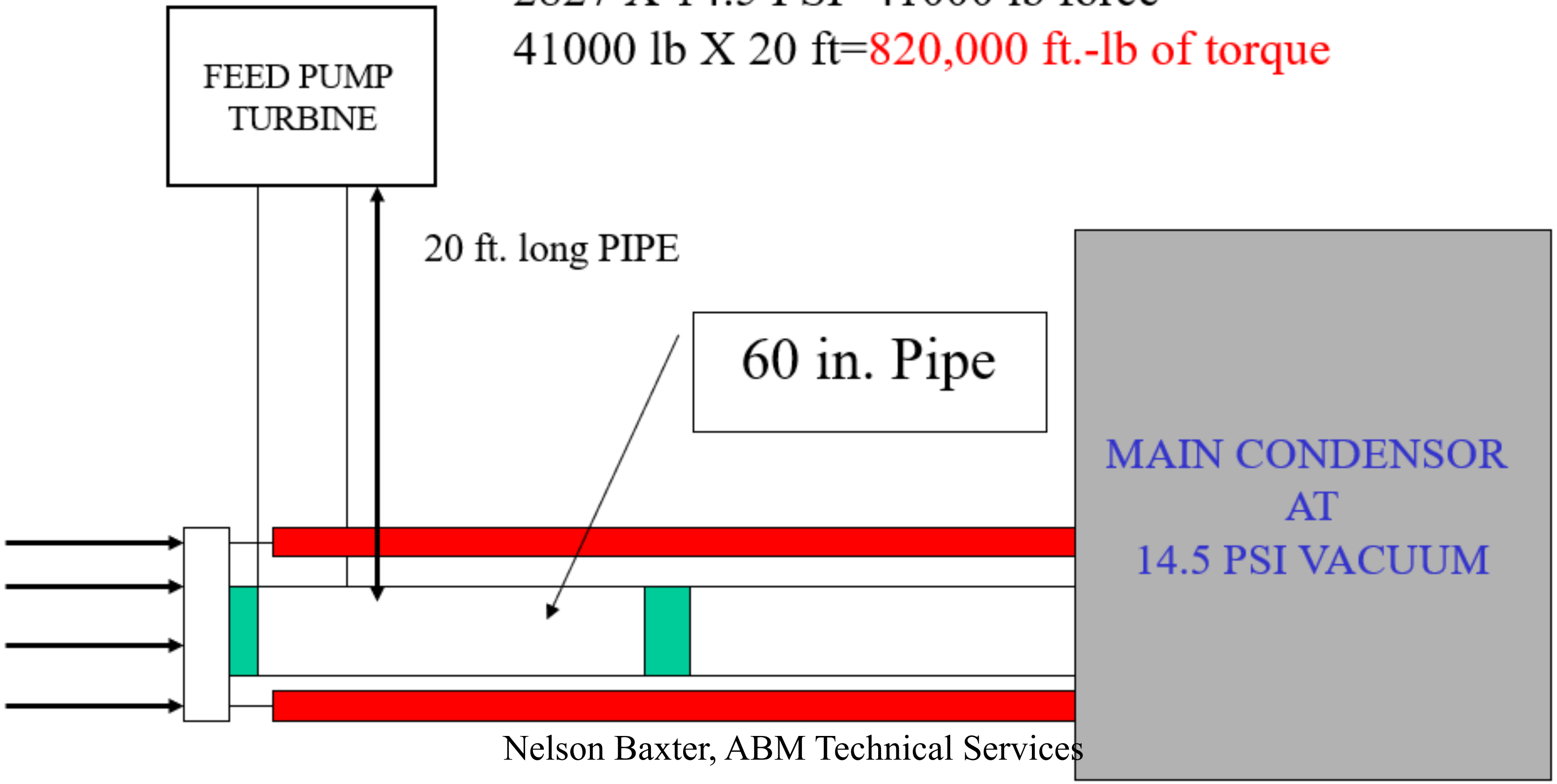


Examination of thrust cancelling rods showed movement of thrust cancelling rod.



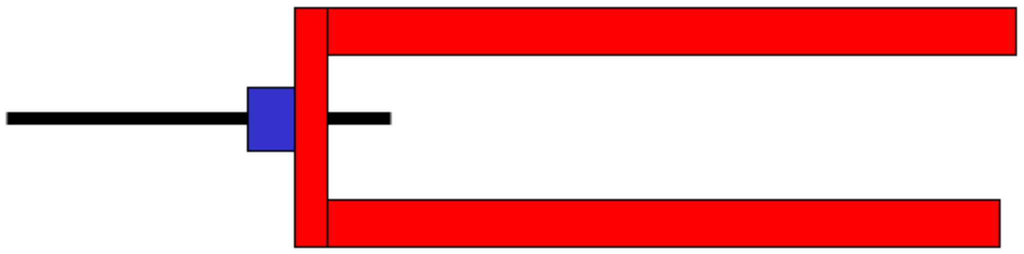
Force Calculations

60 " PIPE
AREA = PI*30*30=2827 square inches
2827 X 14.5 PSI=41000 lb force
41000 lb X 20 ft=820,000 ft.-lb of torque



Solution

- PUT NUT ON OUTSIDE OF END CAP



FORCE PUSHES AGAINST END CAP INSTEAD OF PULLING ON WELD

Final Comment

- This case illustrates how a simple weld failure had a large effect. When the weld on the nut of the original design failed, that prevented the force cancelling rods from doing their job. This resulted in a torque of 820,000 lb-ft being applied to the turbine which caused a very large change in its alignment.

10th Most Difficult Case
Failing Turbine Bearings on Chiller Unit at a University

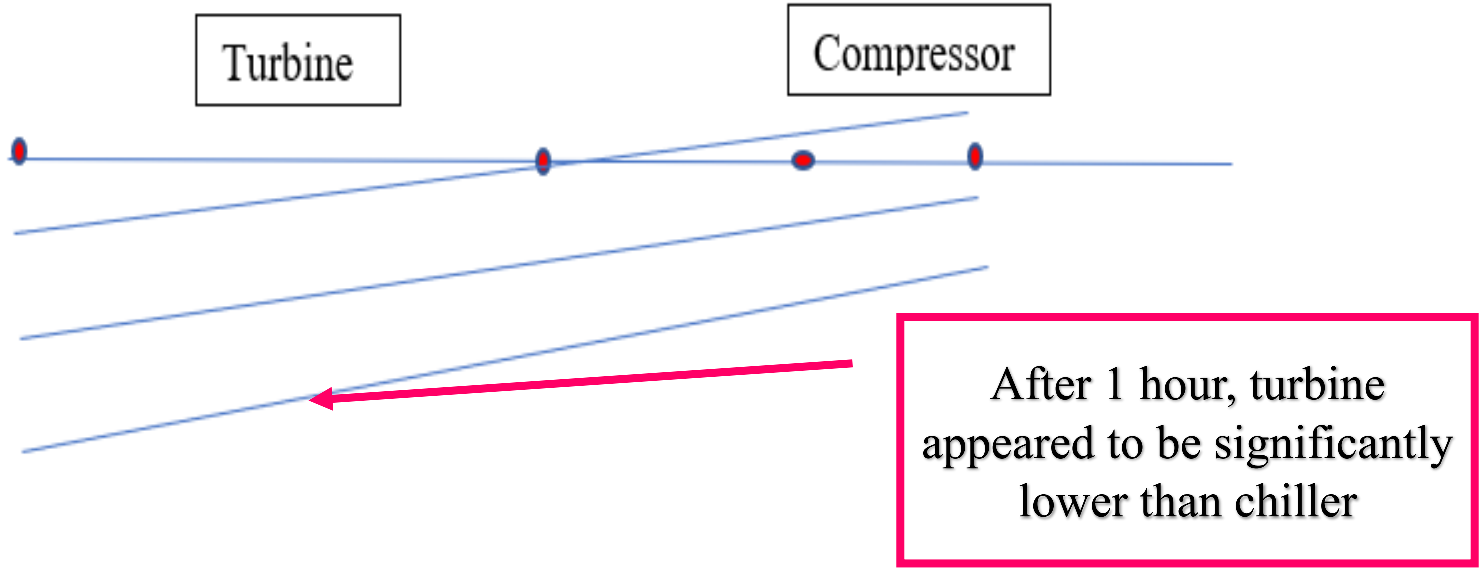
MACHINE SETUP

CHILLER TURBINE WAS DESTROYING BEARINGS

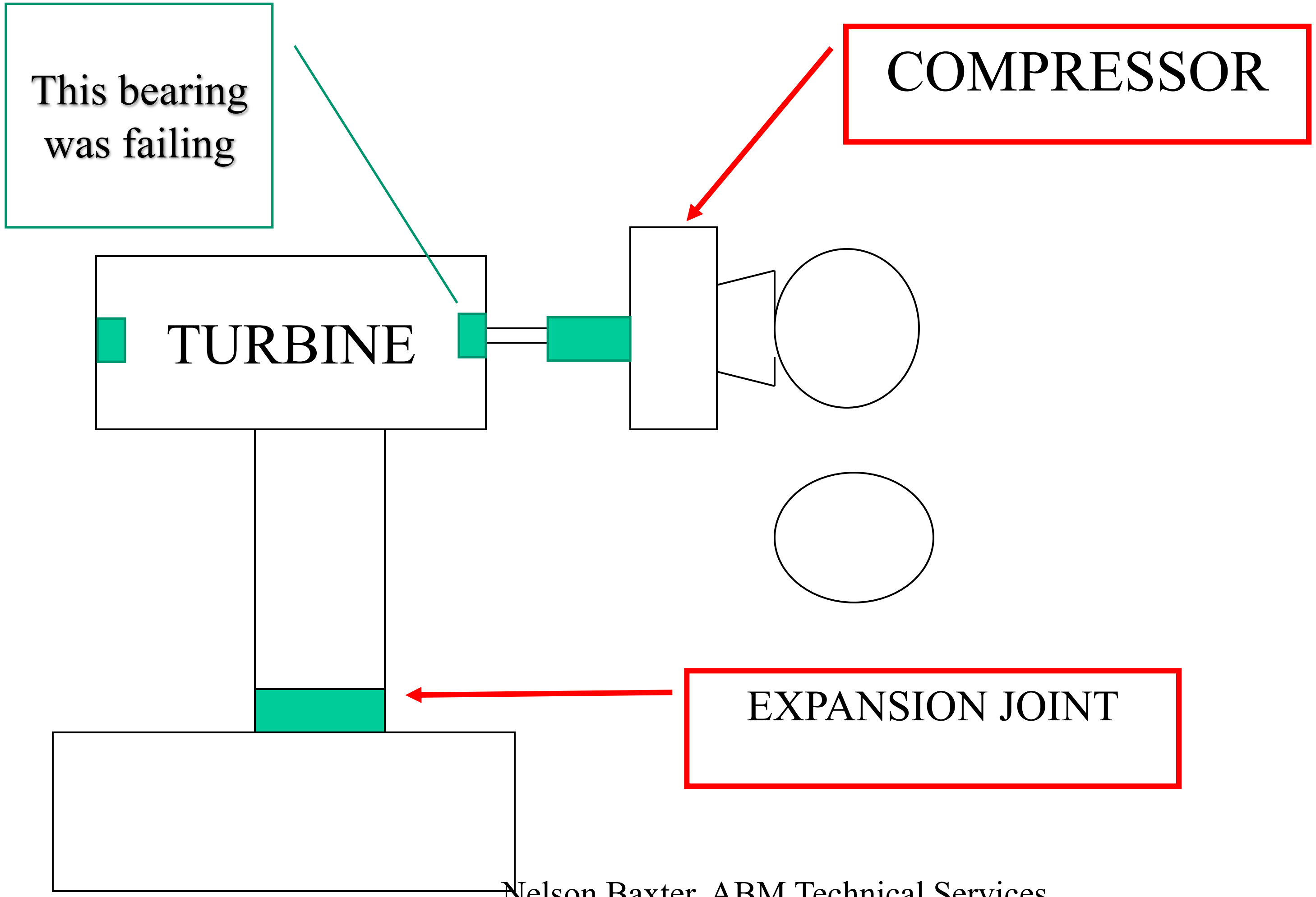
UPPER HALF OF BEARINGS WERE FAILING BY FATIGUE

MONITORING OF ALIGNMENT

The alignment would vary with time. The plot below shows the movement over a two-hour period following a typical startup. The graph shows that the turbine would appear to drop relative to the compressor. The movement below occurred over the first hour of operation.



MACHINE LAYOUT

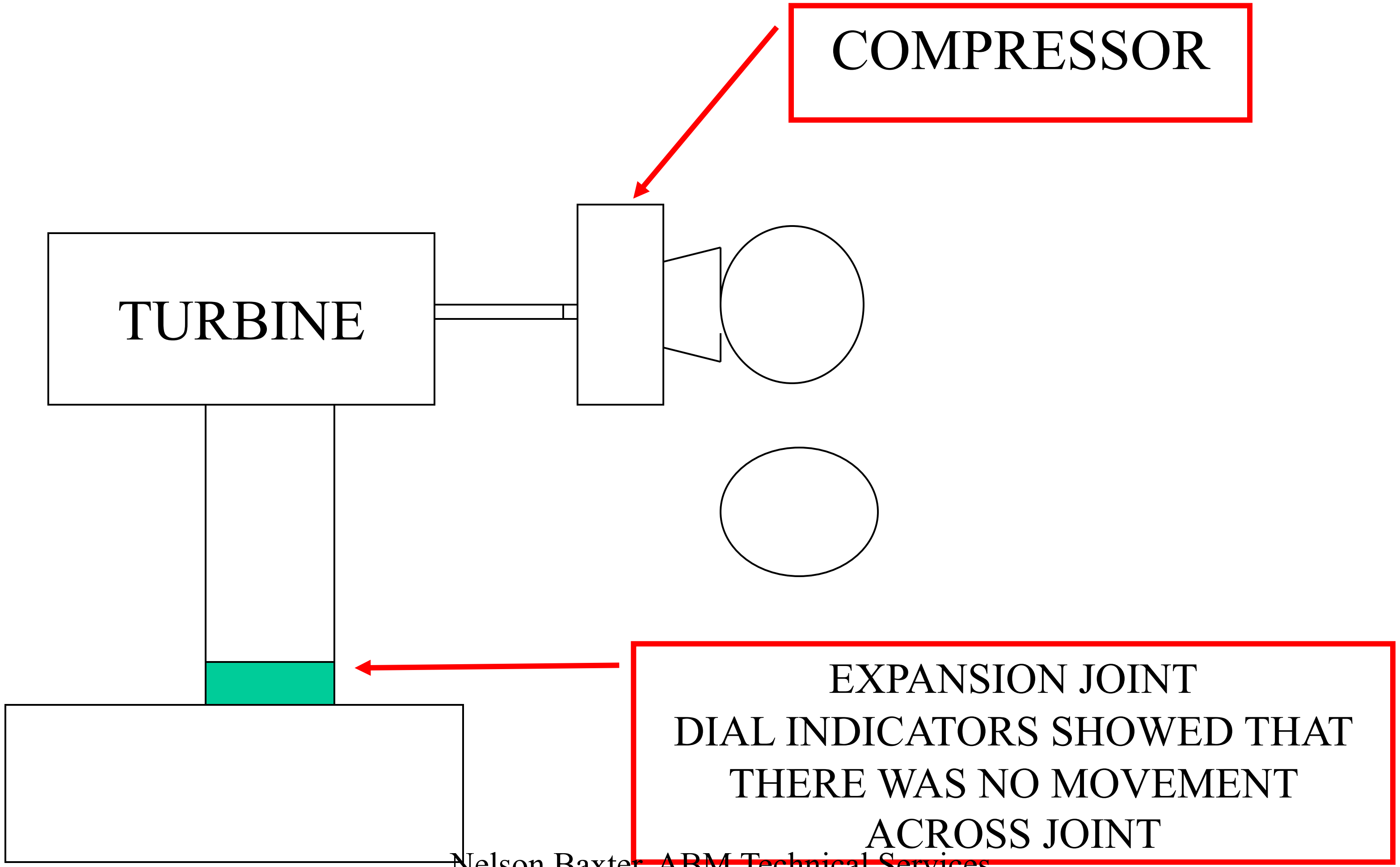


Why was the alignment changing so much during the 1st hour of operation.

The change in the alignment was measured with Dodd bars. The movement was very repeatable. The first thought was that the condenser draw down forces on the turbine were pulling it down. In order to test this theory, dial indicators were installed to measure any movement across the expansion joint located between the turbine and the condenser.

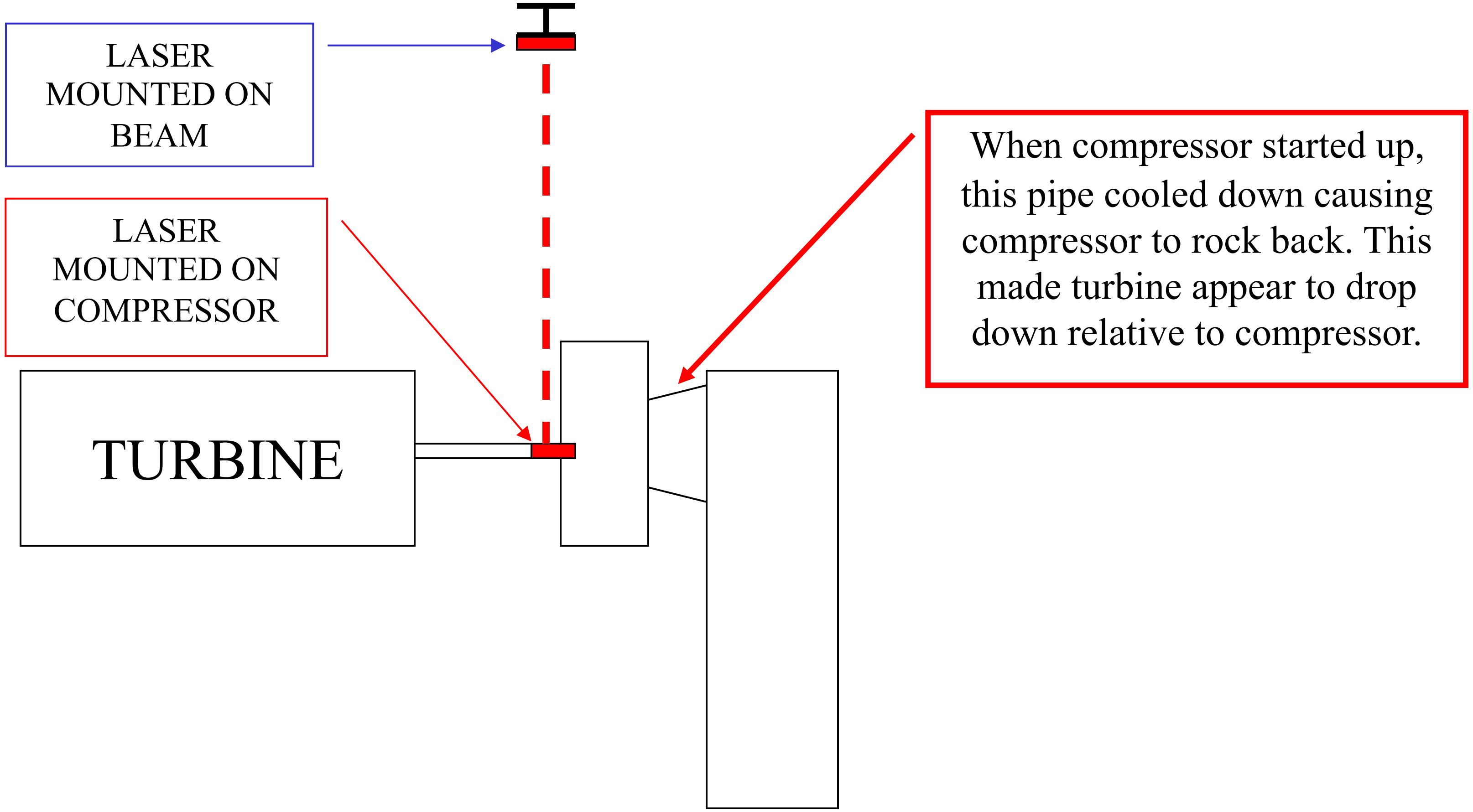
During the time while the alignment changed, there was no movement detected between the turbine and the condenser.

TEST RESULTS



Since the turbine was not moving down, attention was placed on the compressor. A laser system was installed with the transmitter on the compressor bearing housing and the receiver on an I-beam a few feet away.

ACTUAL PROBLEM



The laser showed the compressor was rocking backwards towards the heat exchangers. The readings were taken three times and the results were the same each startup. During the startup period, the large pipe between the compressor and the heat exchangers cooled down and got shorter thus pulling on the compressor making it rock backwards.

Solution: Change to tilt pad bearing to minimize any oil whirl issues. Set the turbine high to compensate for movement of compressor. This was done every overhaul. This machine ran without further problems until it was retired 30 years later.

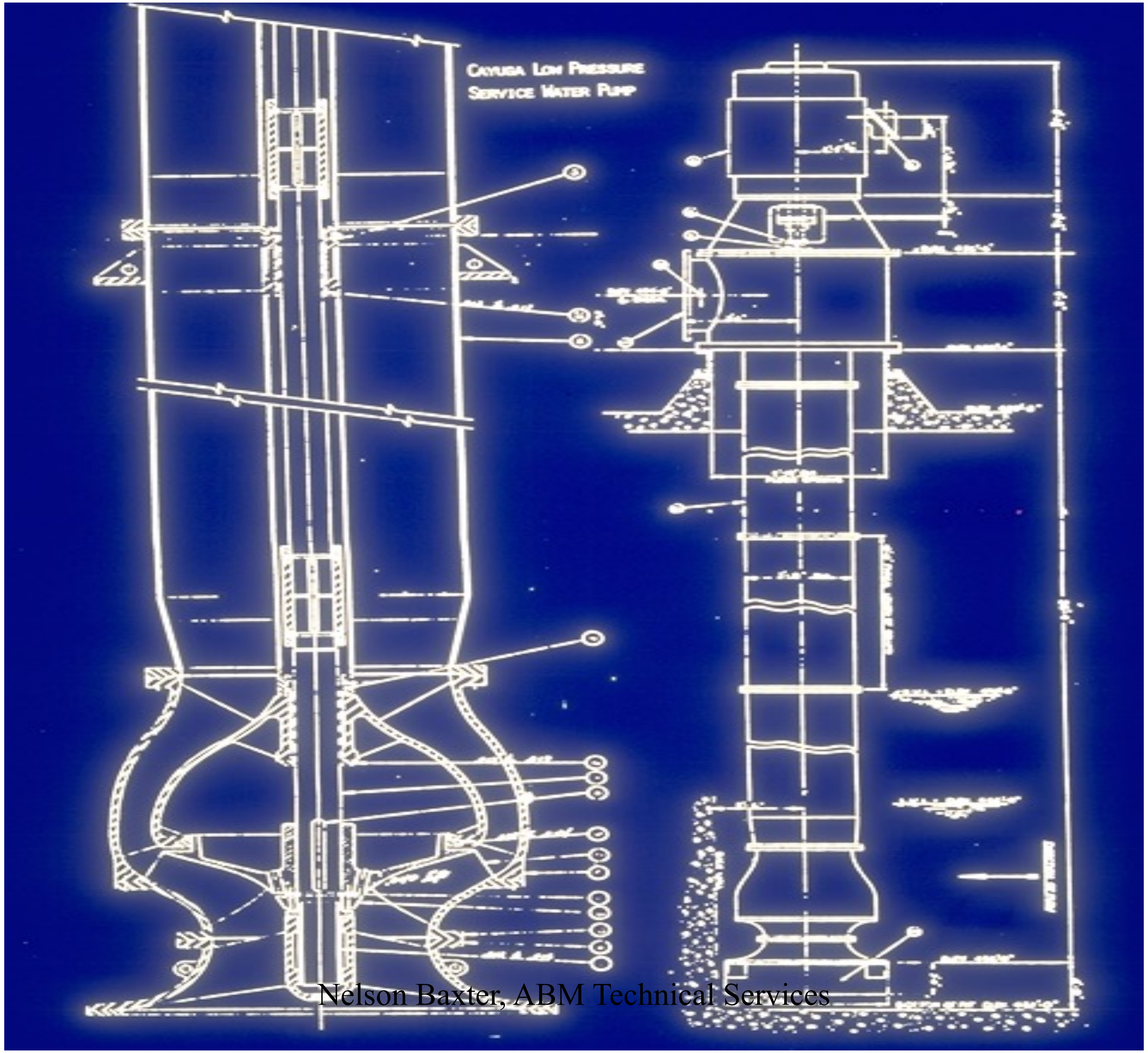
9th Most Difficult Case

Six shaft failures on large service water pumps that occurred shortly after overhauls.

BROKEN SHAFTS

AFTER SIX PUMP SHAFTS HAD BROKEN, TESTS WERE RAN TO DETERMINE CAUSE. THE PUMPS HAD OPERATED FOR MANY YEARS WITH NO PROBLEMS THEN THE SHAFTS STARTED TO BREAK.

CROSS SECTION OF PUMP



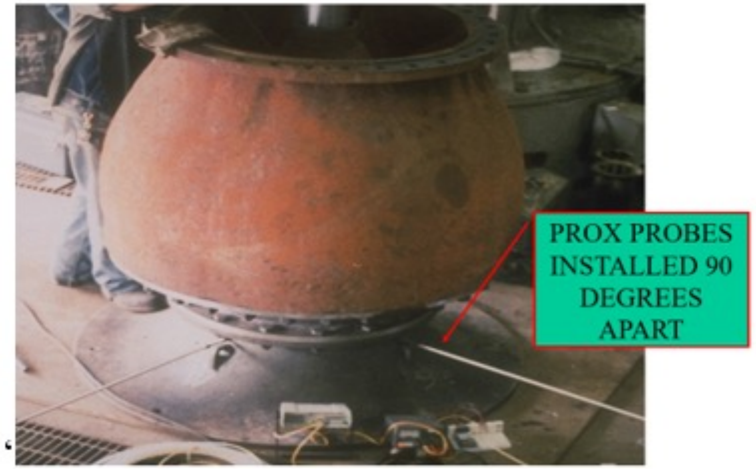
PUMP IMPELLER

IMPELLER ON SHAFT

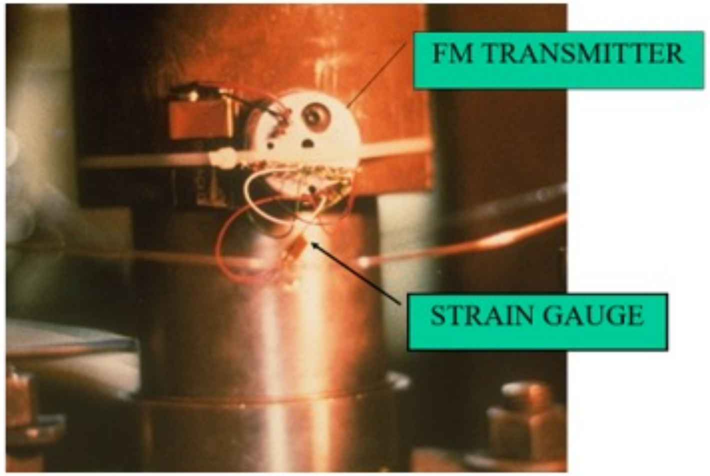
IMPELLER IN WATER FOR RESONANCE TEST

Because of the multiple failures, several tests were performed on the pumps.

Proximity probes were installed on lower bearing



Torsional Measurements were made

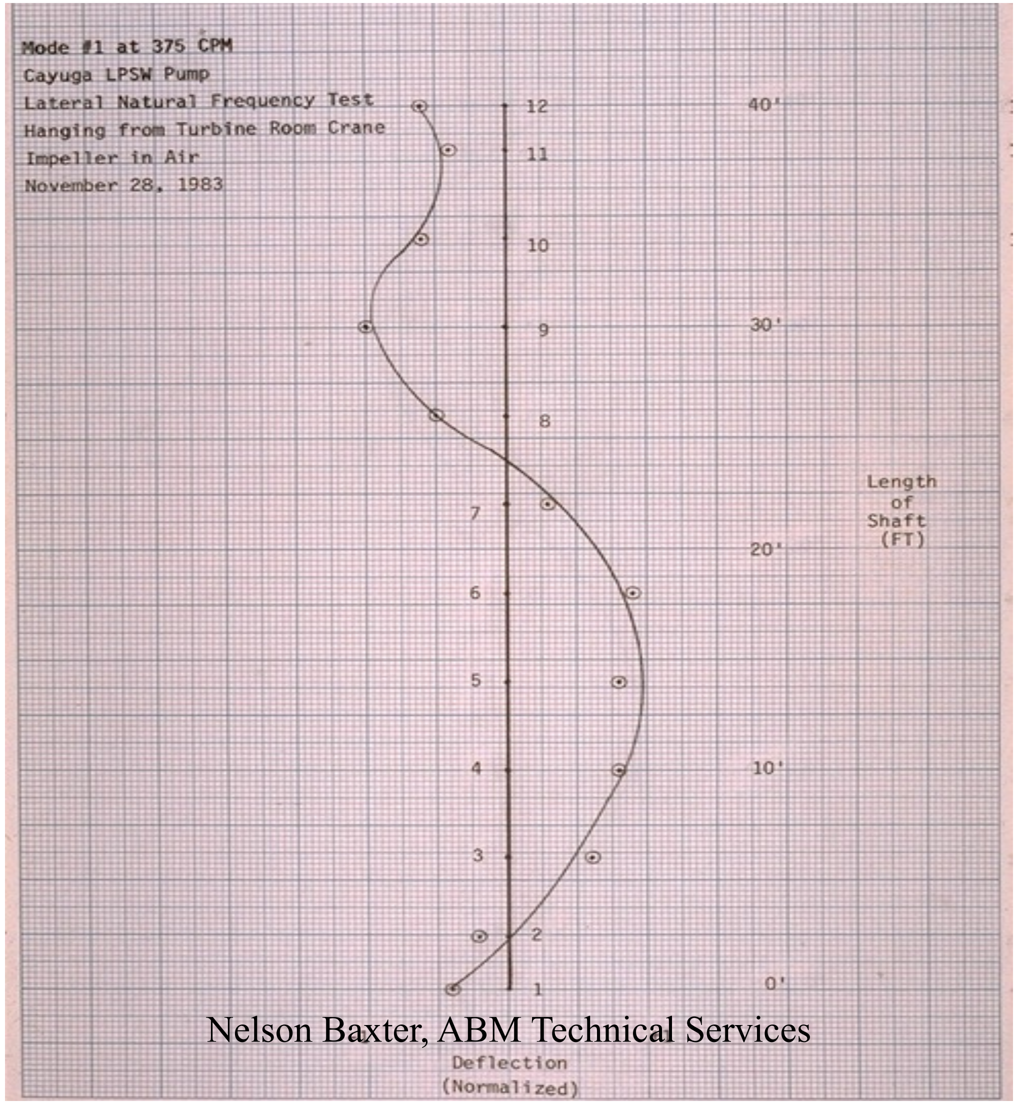


Impact Tests were performed

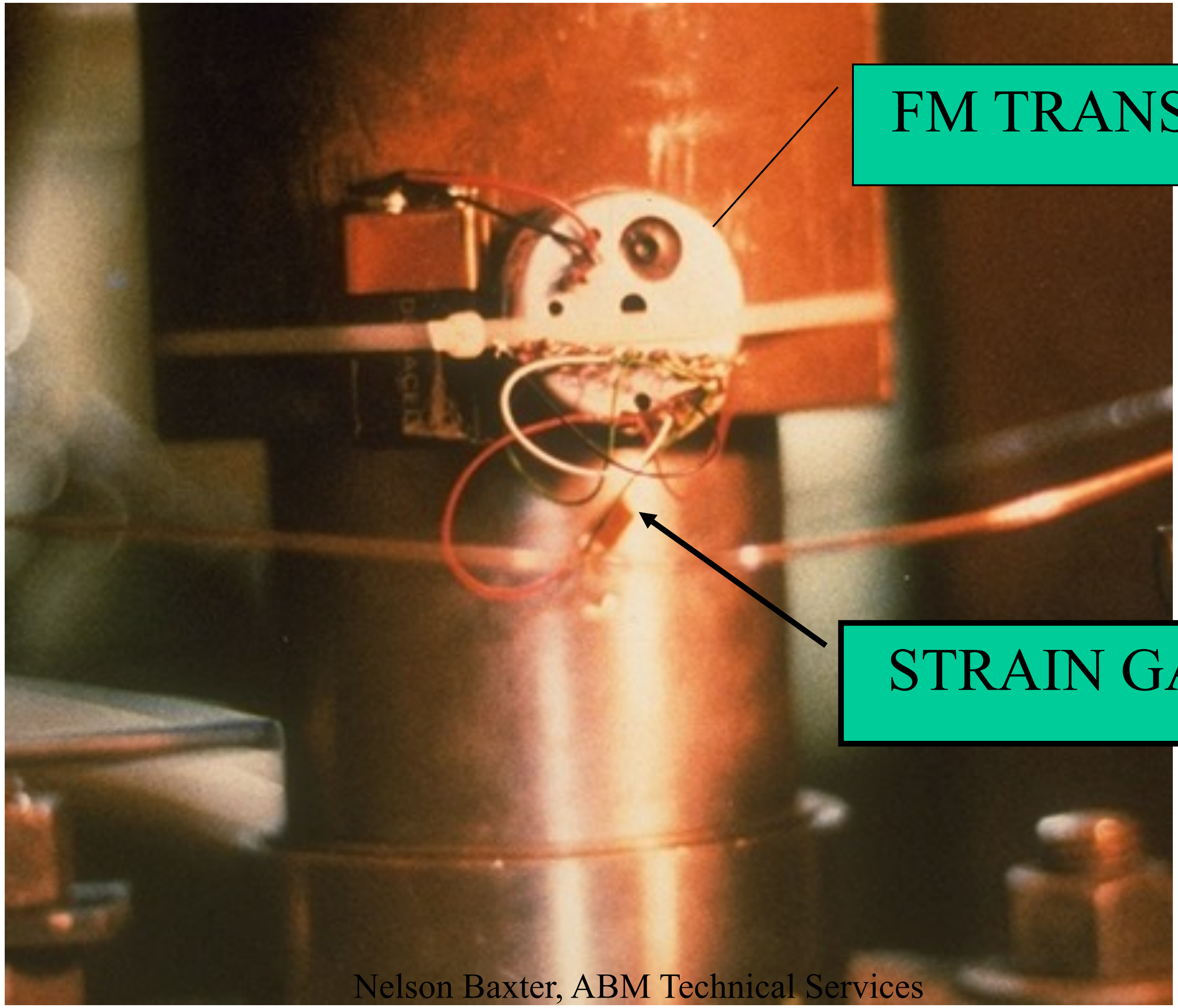
The impeller on its shaft was put under water for tests

MODE SHAPE TEST

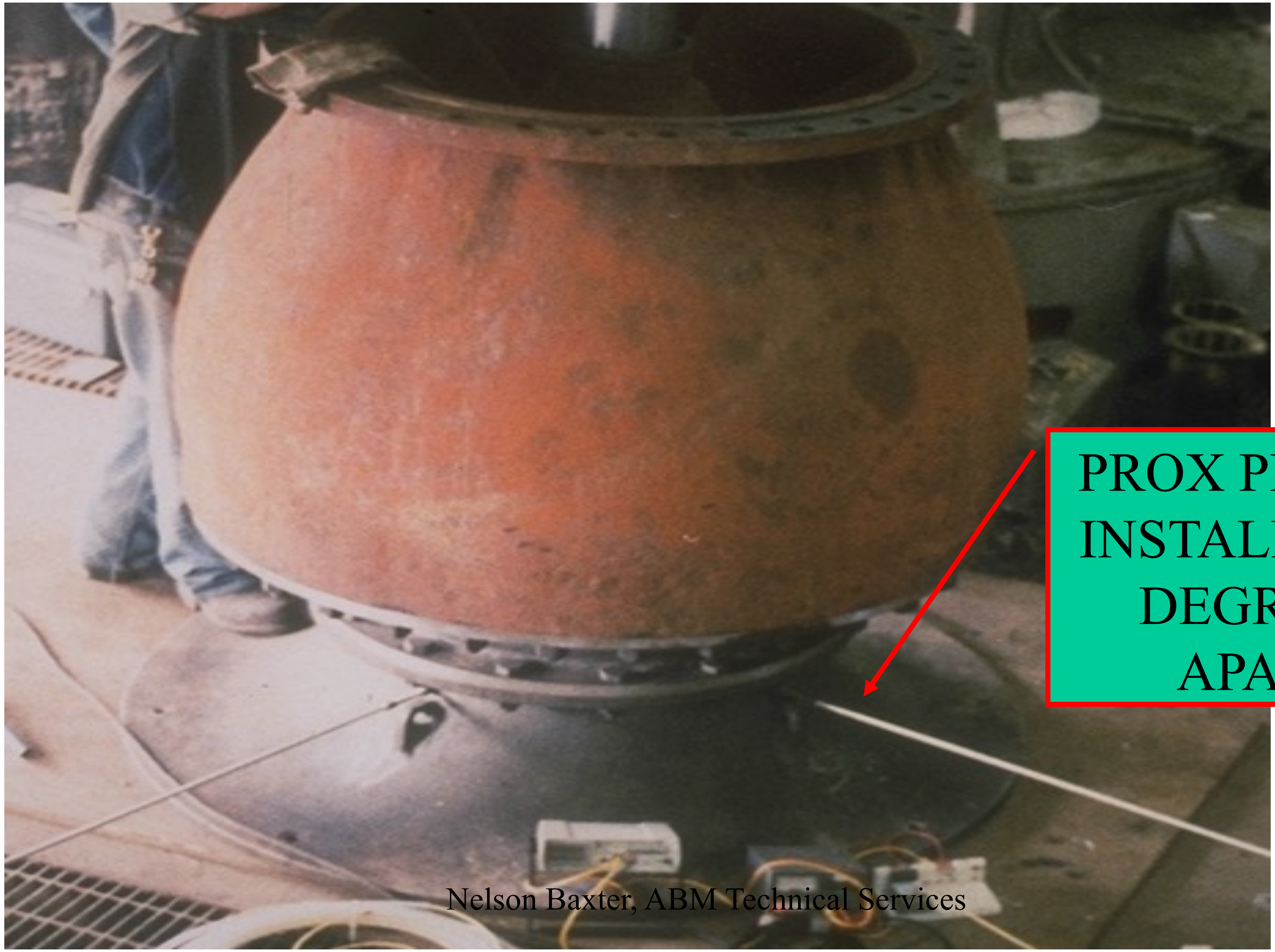
1 St MODE 375 CPM



TORSIONAL TEST SETUP

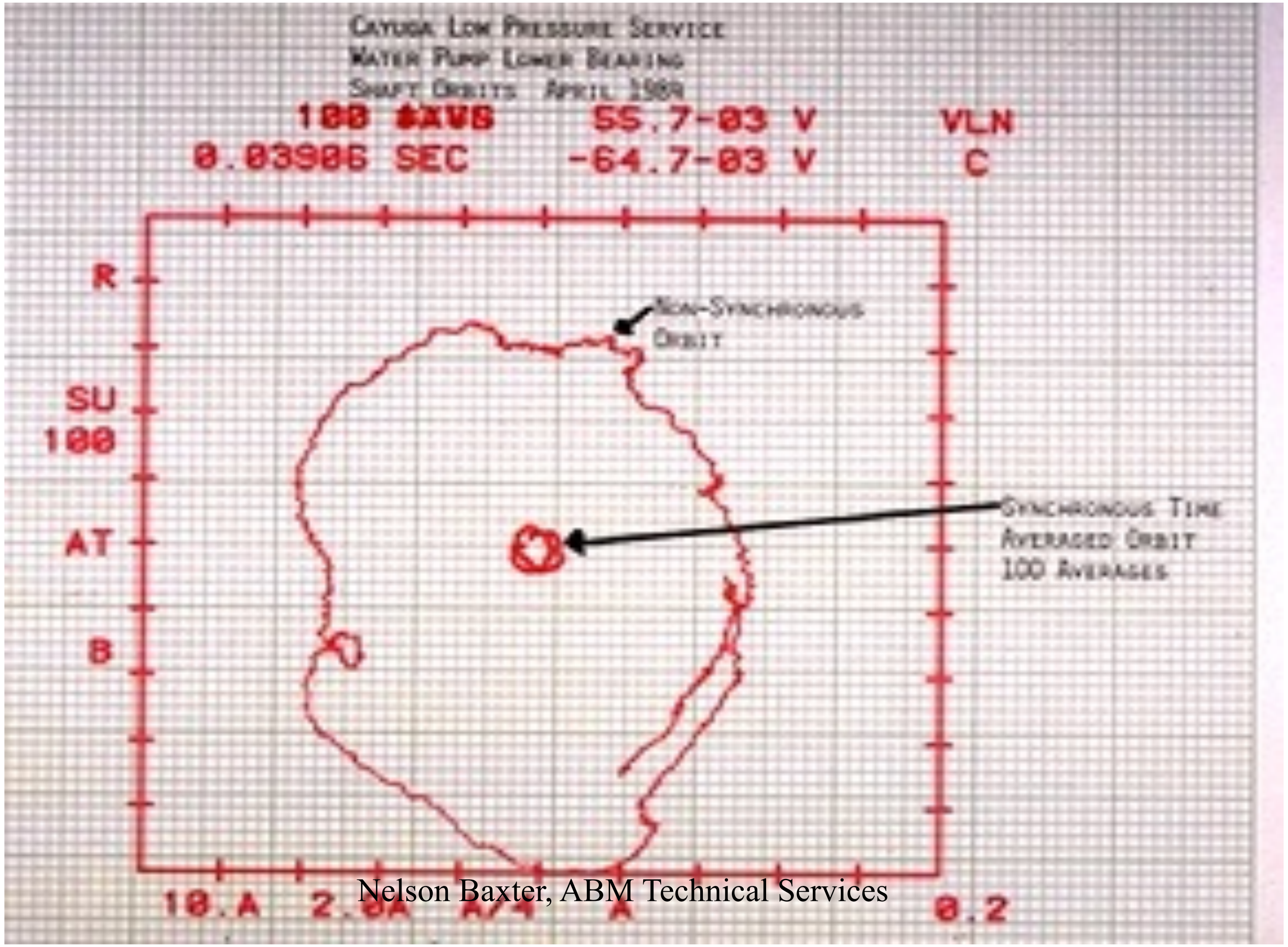


FIELD TEST SETUP



PROX PROBES INSTALLED 90 DEGREES APART

ORBIT


CAYUGA LOW PRESSURE SERVICE
WATER PUMP LOWER BEARING
SHAFT ORBITS APRIL 1988
100 #AVG
55.7-03 V
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10.A
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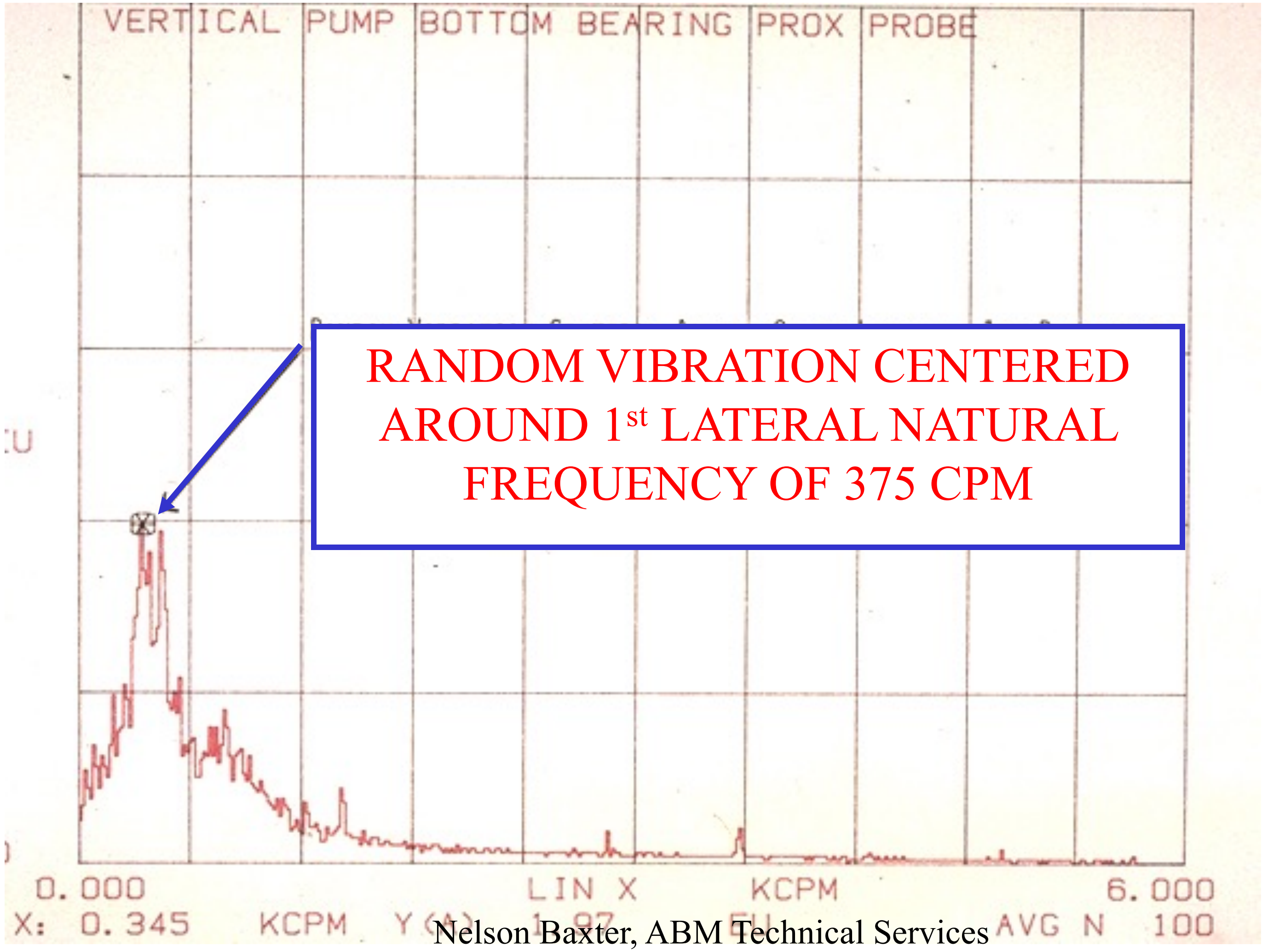
VIBRATION SPECTRUM



Test Results

When the vibration spectra were examined, there was no dominant frequency. What was found was the vibration was broad band in nature and centered around 375 cpm. This frequency matched the first bending mode of the shaft. When synchronous time averaging was applied there was almost no running speed vibration. There was no measurable torsional vibration present. This agreed with the type of breaks that were occurring which did not show any signs of torsional fatigue.

Conclusion

Broad band flow energy was exciting the vibration on the shaft at sub-synchronous frequencies. This resulted in alternating bending at approximately every other revolution. Cracks then developed at the notched high stress points where the shaft couplings were installed.

Question

Why did these pumps run for many years, then only after they were overhauled did the breaks start to occur? The pumps had run over 10 years with no failures then after the overhauls the shafts were breaking within a year of operation. This brought up the obvious question of what was changed during the overhauls?

Investigation

Maintenance personnel were interviewed to determine if there had been any changes to the pumps when they were overhauled. Because of a pricing issue, plant personnel had elected to go with a different supplier of pump impellers. When the replacement impellers were closely examined, the angles and the thickness of the impeller vanes were found to be significantly different than the original design. This resulted in very high turbulent flow exciting the shaft's first natural frequency. When the original manufacturer impellers were put back in the pumps, the shaft failures immediately stopped. There have been no shaft failures over the past 20 years.

This case history was difficult because of the following:

1: The pumps had operated for 10 years then the problem suddenly arose. There had been no change in flow or suction and discharge pressure.

2: With a vertical pump, it is difficult to measure vibration on the pump itself. Getting proximity probes installed on the pump and getting the wiring above the foundation was a difficult task.

3: A famous organization used by utilities to study problems kept wanting to do a computer model of the pump. They were no help whatsoever in going after the root cause.

4: Since shaft breakage was occurring, it seemed necessary to do torsional measurements so that added another layer to the approach to instrumentation.

5: The shaft was very long so obtaining mode shape information was difficult.

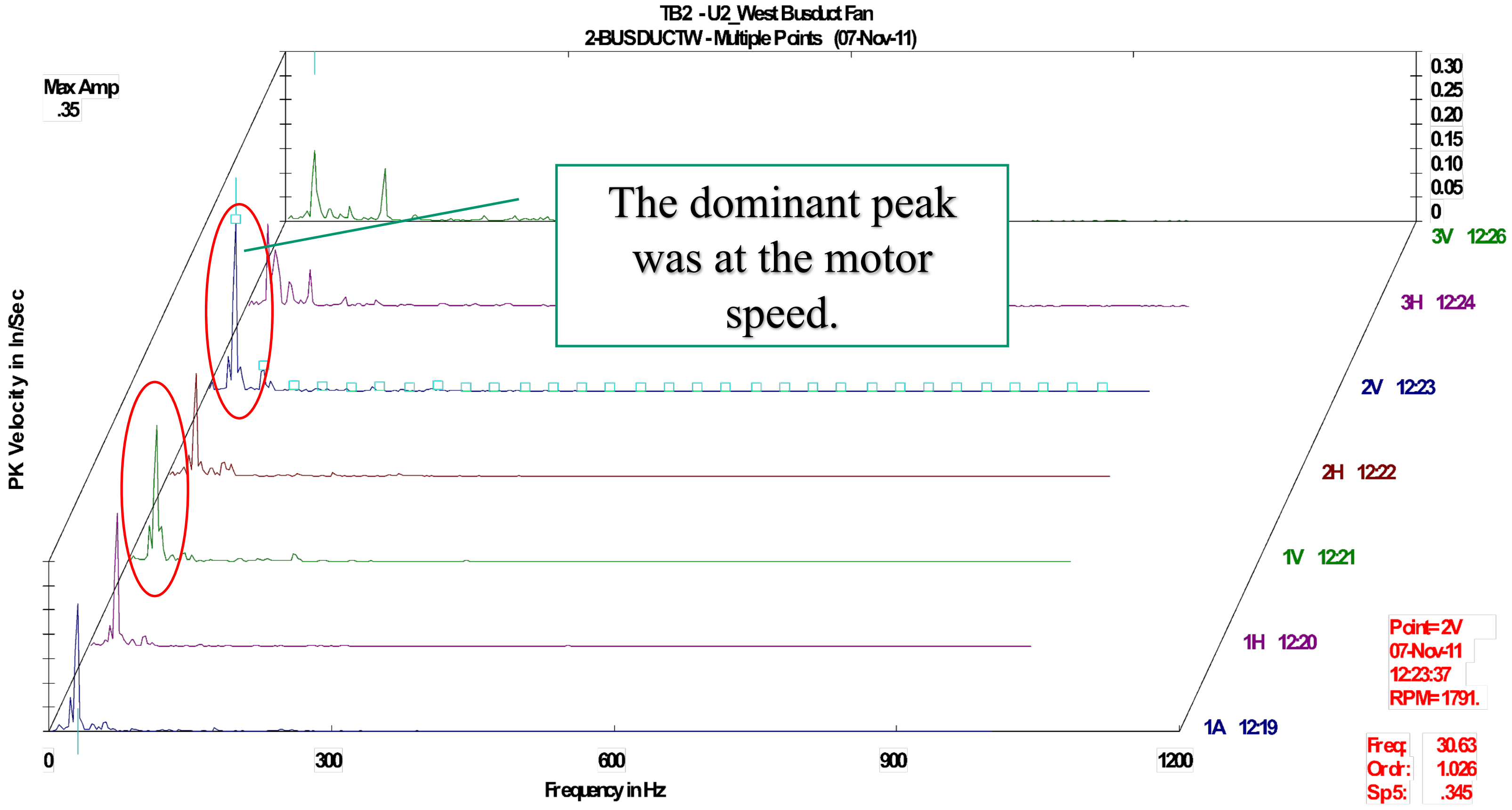
6: Plant personnel did not understand how critical it was to let us know they had changed the impeller design.

8th Most Difficult Case
High Motor Vibration On Bust Duct Fan at Nuclear Power Plant

Fan was located on a stand

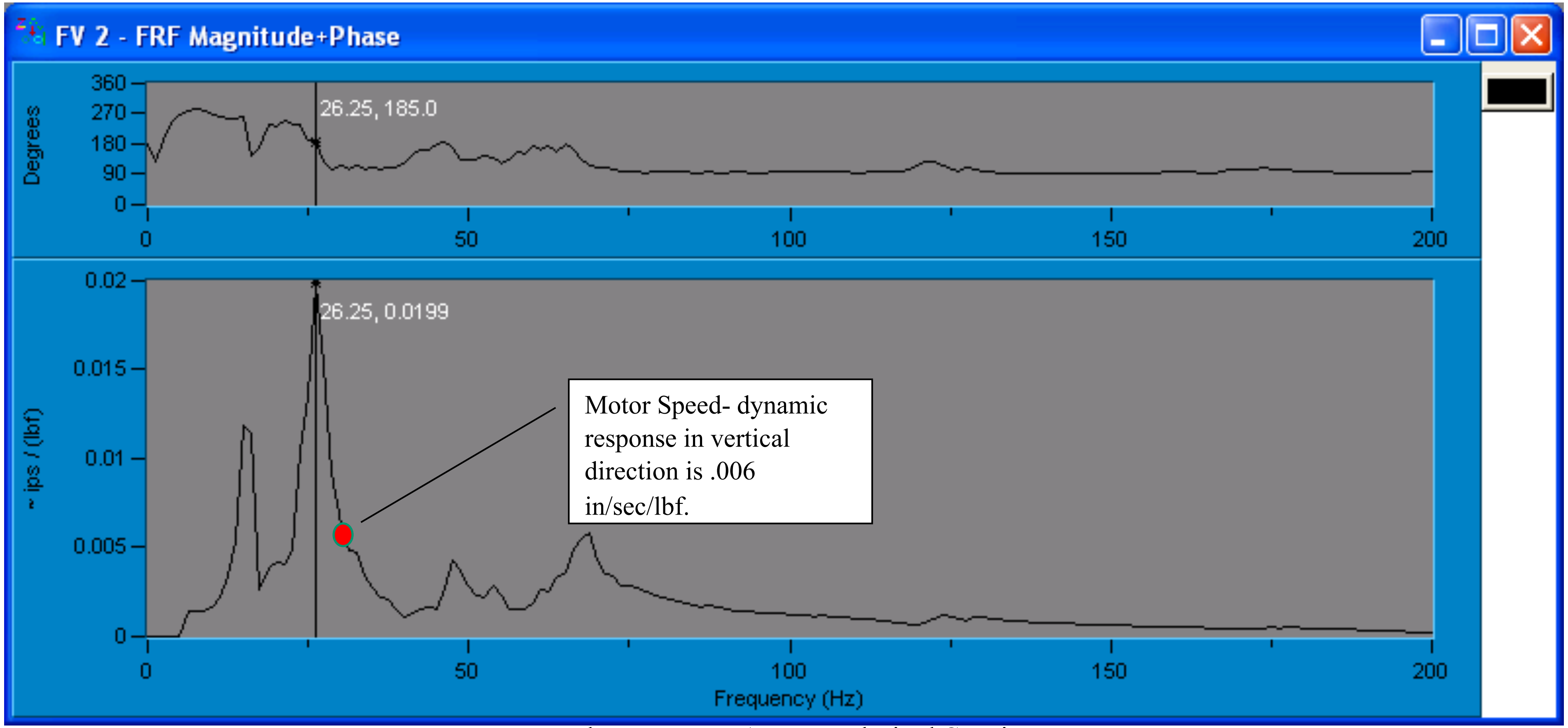
Motor and fan were mounted on a metal floor

Motor vibration levels were above alarm point. All vibration was at motor speed.

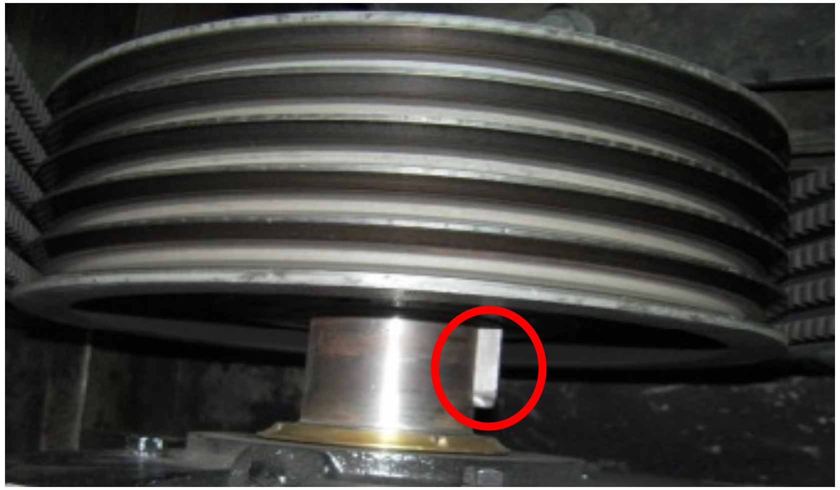


This was a nuclear power plant and because of past resonance problems, the customer insisted that an ODS and resonance tests be performed. The ODS, as would be expected, showed the metal base was deforming.

Resonance Test Results showed there was some amplification due to a resonance located around 15% below the operating speed.



Further Examination showed presence of excess key.



Excess key between sheave and motor

Excess key on outboard end of shaft

Question: Was excess key enough to cause the vibration levels to be in alarm?

Comparison to ISO Standard balance grade 2.5

The motor weighted 517 lb. ISO grade 2.5 for a 517 lb motor turning 1800 RPM:

For a rigid frame, the allowable unbalance is 4.2 oz-in.

For a flexible frame the allowable unbalance is 1.7 oz-in.

This fan was mounted in the air and the vertical natural frequency on the metal base was below running speed, so the 1.7 oz-in flexible frame limit applied. There were 4 inches of excess key and the key was ¾” wide and ½” thick. The volume of the excess key was therefore 1.5 cubic inches. When taken times the density of stee; (.283 lb/in cube) the weight of the key was calculated to be .42 lb or 6.8 oz. The distance of the key from the shaft center was 1.5 inches. The unbalance was therefore 10.2 oz-in or 6 times the ISO flexible frame standard.

Calculation of response based upon FRF dynamic stiffness values

The calculated force from the 10.2 OZ-in of unbalance at 1800 RPM was 58 lbf. The dynamic stiffness from the FRF was .006 in/sec/lbf. The response to this unbalance force therefore calculates out to be .35 in/sec which is very close to the .379 in/sec measured on the motor.

Conclusions: The excess vibration was due to a combination of the excess key and to a lesser degree the motor operating on the response curve of a structural natural frequency. The solution was to install a step key. This was actually a very simple problem to solve. What made it onto the most difficult list?

Why was this problem difficult?

1: This was a nuclear power plant. As shown in Figure 14, a ladder had to be climbed to get to the fan for testing. There were three people watching how we climbed the ladder pointing out if we did not have a constant three points of contact. This was very distracting.

2: Rather than do simple tests, plant personnel wanted an ODS. Because of restrictions to the space, all the test points had to be taken at once. This meant it was necessary to install 32 accelerometers and their cables.

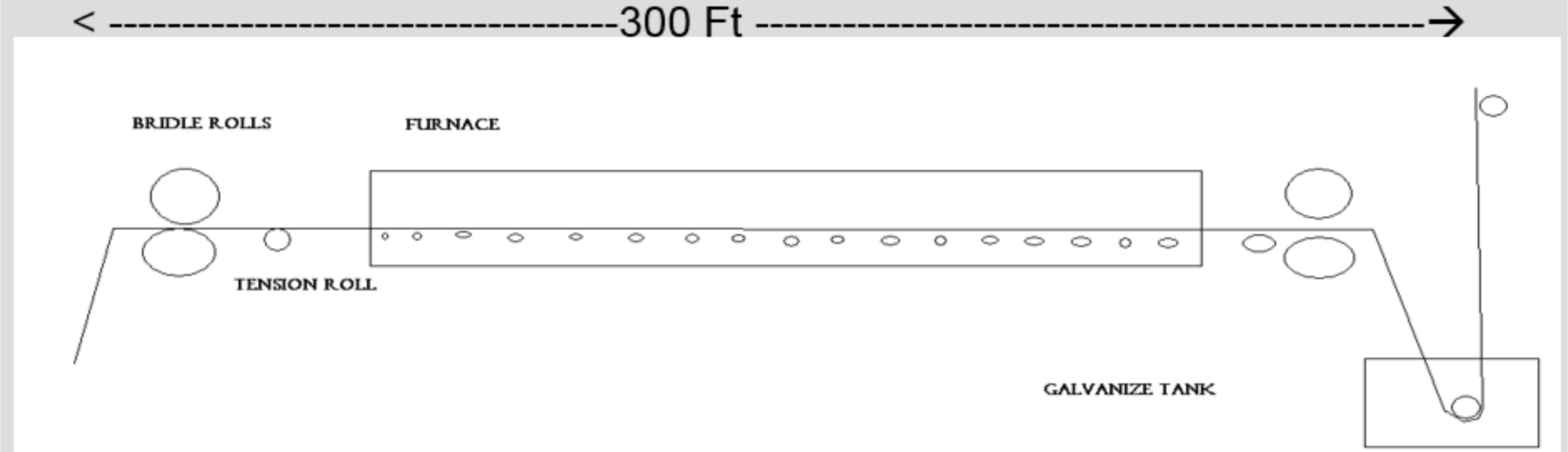
This task combined with people watching every move and policing the ladder for anytime three points of contact were not made getting the data for the ODS a difficult thing to do. As it turned out the ODS was not that helpful.

3: In order to convince everyone the excess key was the primary issue, it was necessary to look up standards and to frequency response calculations.

This was actually a very simple problem to solve, but the added pressure of it being a nuclear power plant and also the past history on this fan greatly complicated getting to an easy solution.

7^{TH} MOST DIFFICULT CASE HISTORY
A GALVANIZE LINE AT A STEEL MILL

- A steel mill in the Midwestern part of the US had a 7 Hz vibration on its galvanize line. The galvanize line was over 300 ft. in length and this entire large structure would vibrate noticeably at the 7 Hz frequency. Noticeable levels of speed oscillation wrre seen on the bridle rolls.

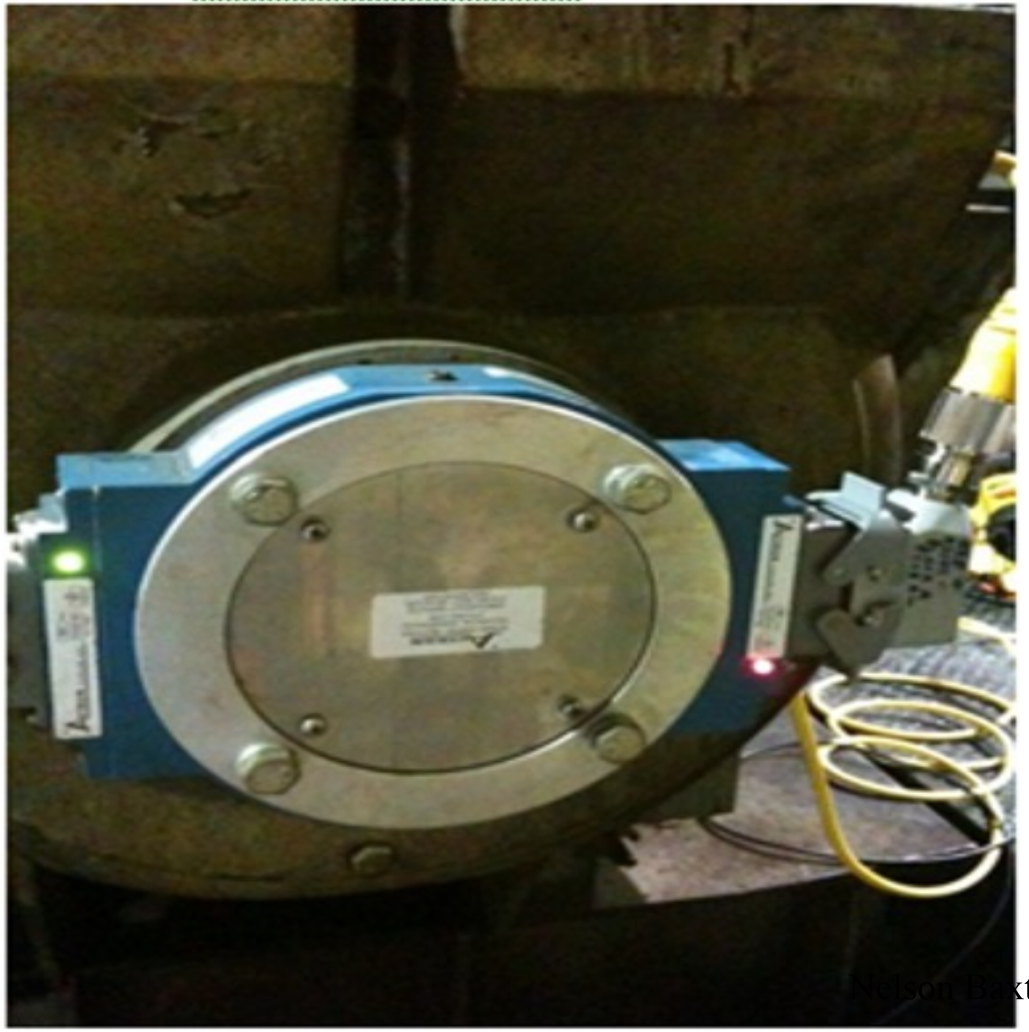


History

- A rather well known consulting organization had been brought in and at the end of a large thick report all they came up with was the 7 Hz frequency was close to a cage frequency on one of the bearings. Since a bearing cage was not a likely source of vibration which could stimulated movement of a 300 ft. long structure, it was elected to due further investigation. Initial measurements indicated the vibration was nine times the large bridle roll speeds. A photo tachometer was installed on the bridle rolls and using synchronous timer averaging and high resolution spectral the data showed the vibration <u>was not an exact multiple of the bridle roll</u> <u>speed</u>. In order to dig deeper into the problem both torsional and additional radial vibration measurements were taken.

Torsional Measurements

Shaft Encoder on main rolls

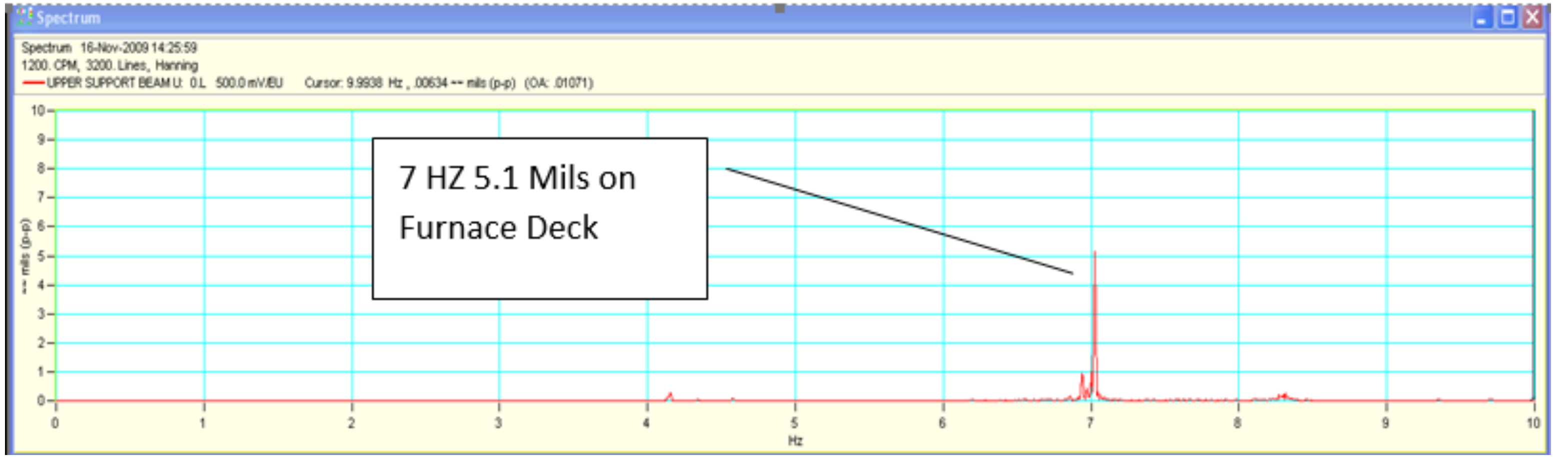


Portable encoder to test other rolls

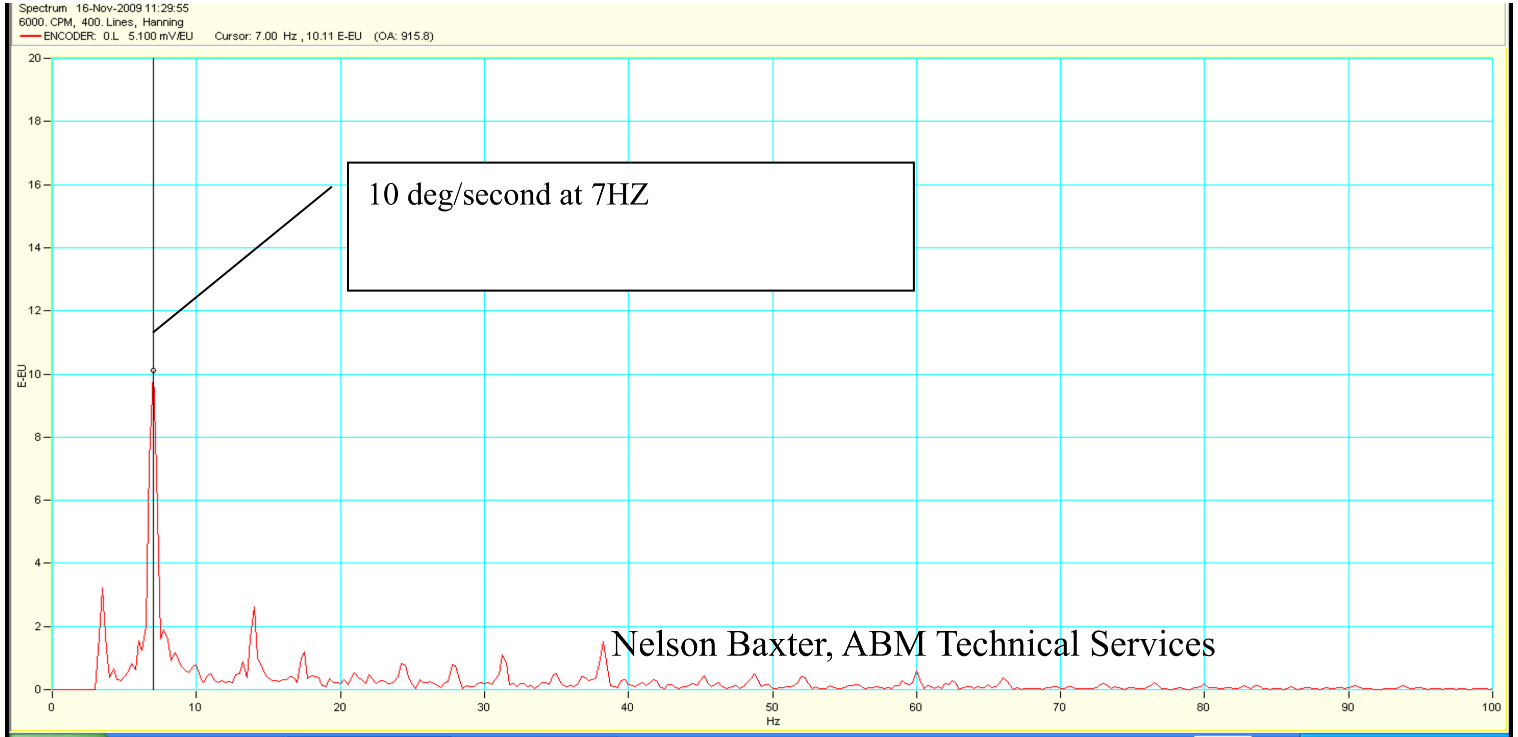
Torsional Demodulator for encoder

Optical encoders power supply and demodulator

Lateral vibration seen on furnace deck



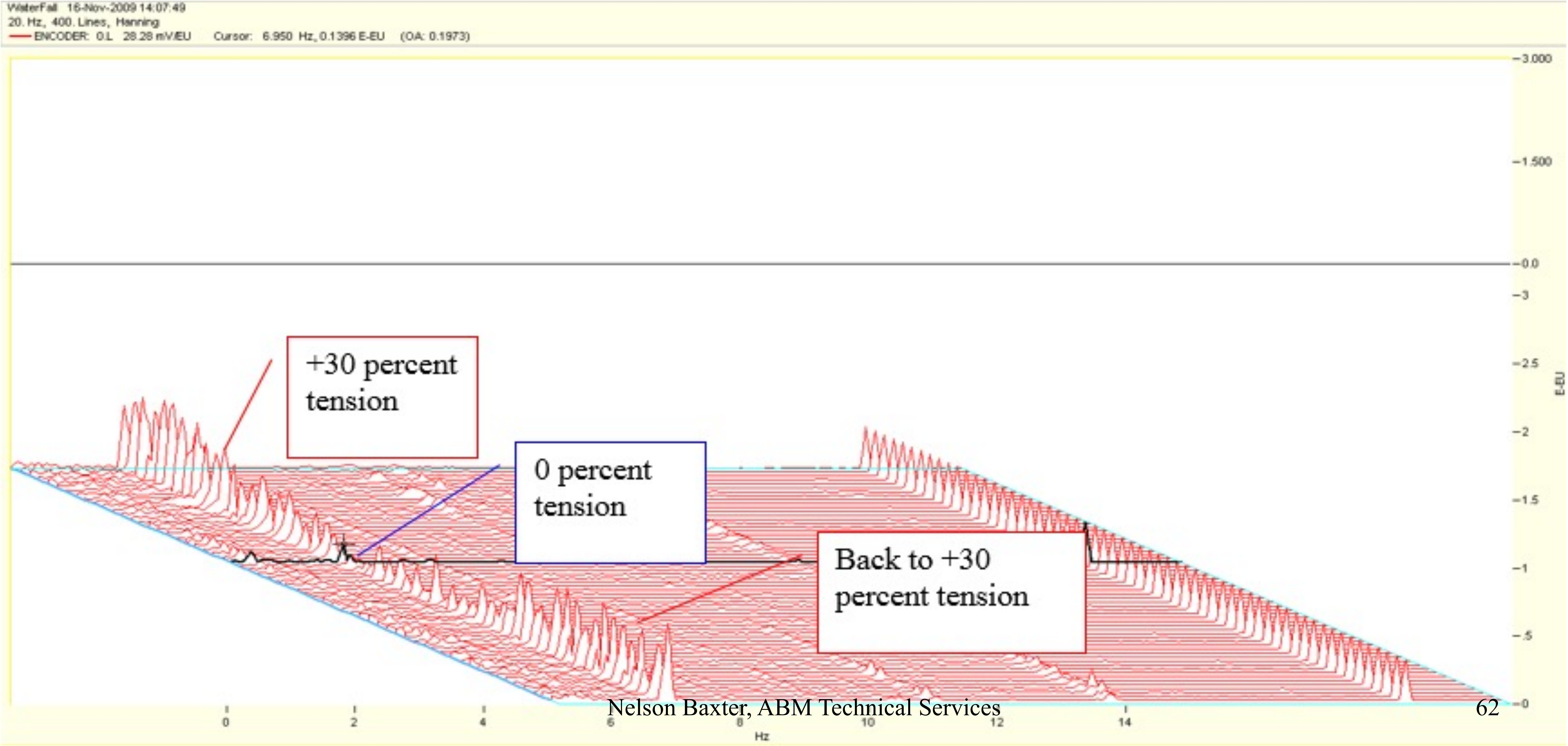
Torsional vibration measured on rolls



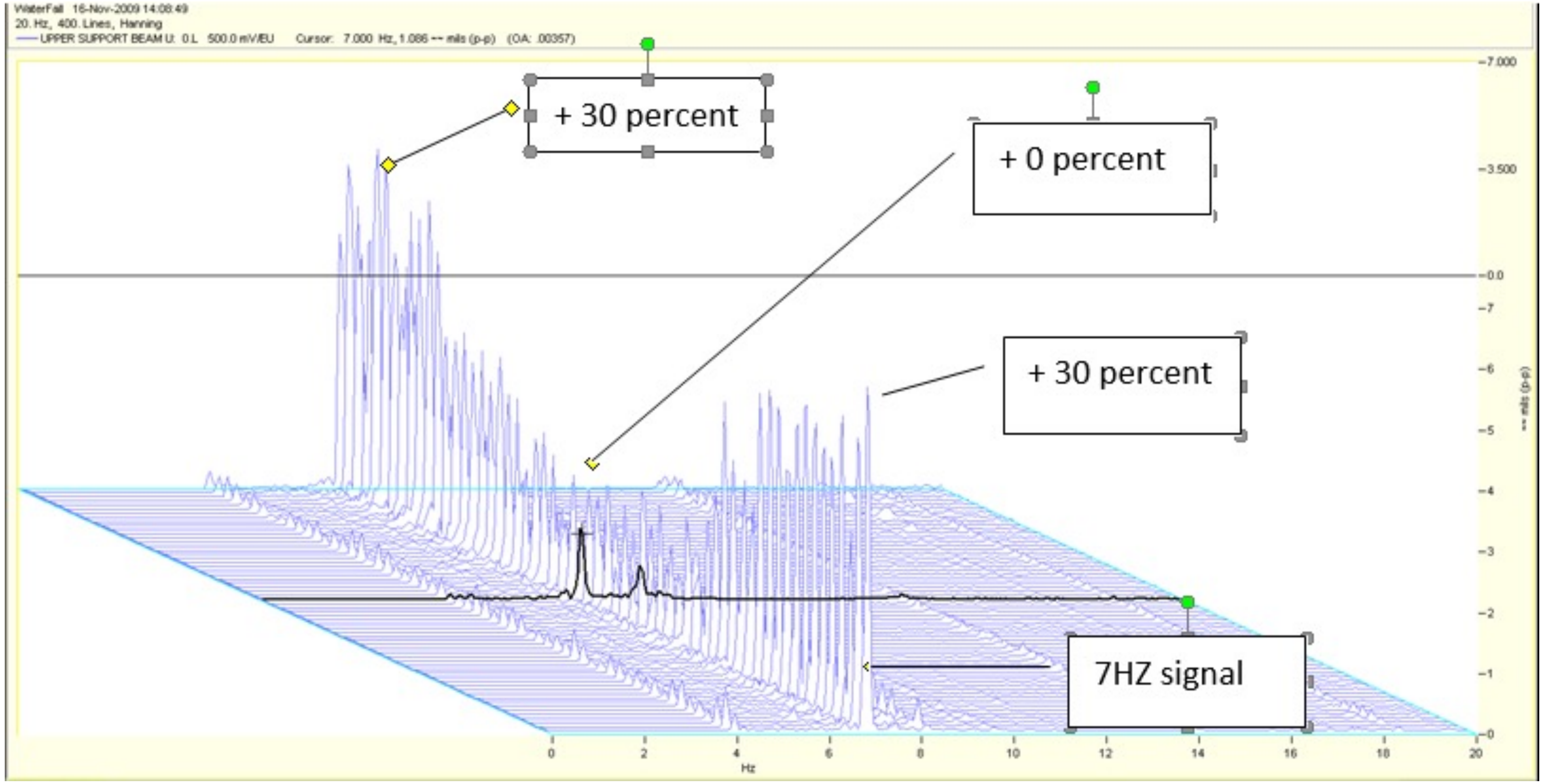
The speed on the steel strip was changed and the frequency of the torsional vibration and the lateral vibration exactly tracked the steel strip speed.

- With the knowledge the torsional and lateral vibration exactly tracked along with the speed of the steel strip, further discussions were made with operations personnel.
- Operations shared that they had noticed the vibration amplitude seemed to go along with how much tension was on the steel strip.
- Based upon this new knowledge, lateral and torsional vibration measurements were obtained while the steel strip tension was changed.

Map Plot of torsional vibration as tension was varied

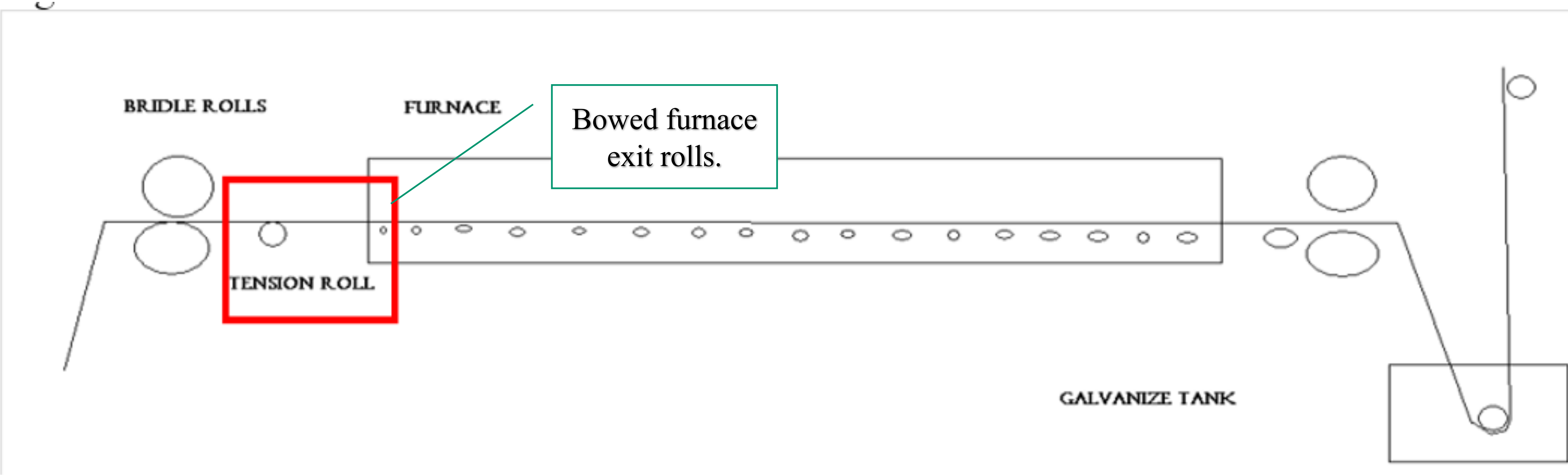


Map plot of lateral vibration as steel strip tension was adjusted.



Both the torsional vibration on the rolls and the structural vibration showed the amplitude of the 7 Hz frequency was directly related to the tension on the steel strip. <u>However, the source of the stimulus that was causing the entire structure to vibrate was still unknown.</u>

- **<u>Source of 7 HZ discovered-</u>** Drawings were examined and the 7 HZ problem was discussed with plant personnel. It was discovered that the small 4 ½" diameter furnace exit rolls turned at exactly 7Hz.
- **Question:** How could small 4 ½" rolls cause the entire 300 ft. structure to vibrate and all the other rolls to respond in a torsional manner ?



Answer- The control system is programmed to keep a constant tension on the sheet. The bowed rolls located near the tension detector caused the tension to increase, then decrease at once per revolution. When the control system sensed a decrease, several hundred horsepower of bridle roll motors would apply more torque trying to increase the tension. The reverse was true when the tension decreased. This occurred at 7 HZ. The relatively minor perturbation caused by the small bowed rolls was being amplified by the control system.

- Sometimes it is necessary to dig deep and think outside of the box.
- Common sense lets you know that the force generated by a small component is unlikely to cause a major structure to vibrate so something else would have to be involved.
- In this case, this was a problem caused by the controls. Perturbations in the signals supplying the controls brought several hundred horsepower of bridle roll motors into play. The solution was to reduce sensitivity of the controller and watch for future bowed exit rolls.

- This case history was difficult because:

1: Another organization had already worked on it. This tends to muddy up the water a bit.

2: Both lateral and torsional measurement had to be made. This always increases the difficulty of the data acquisition.

3: This problem involved both mechanical stimulation and feedback from the control system. Adding in control system response complicates the analysis.

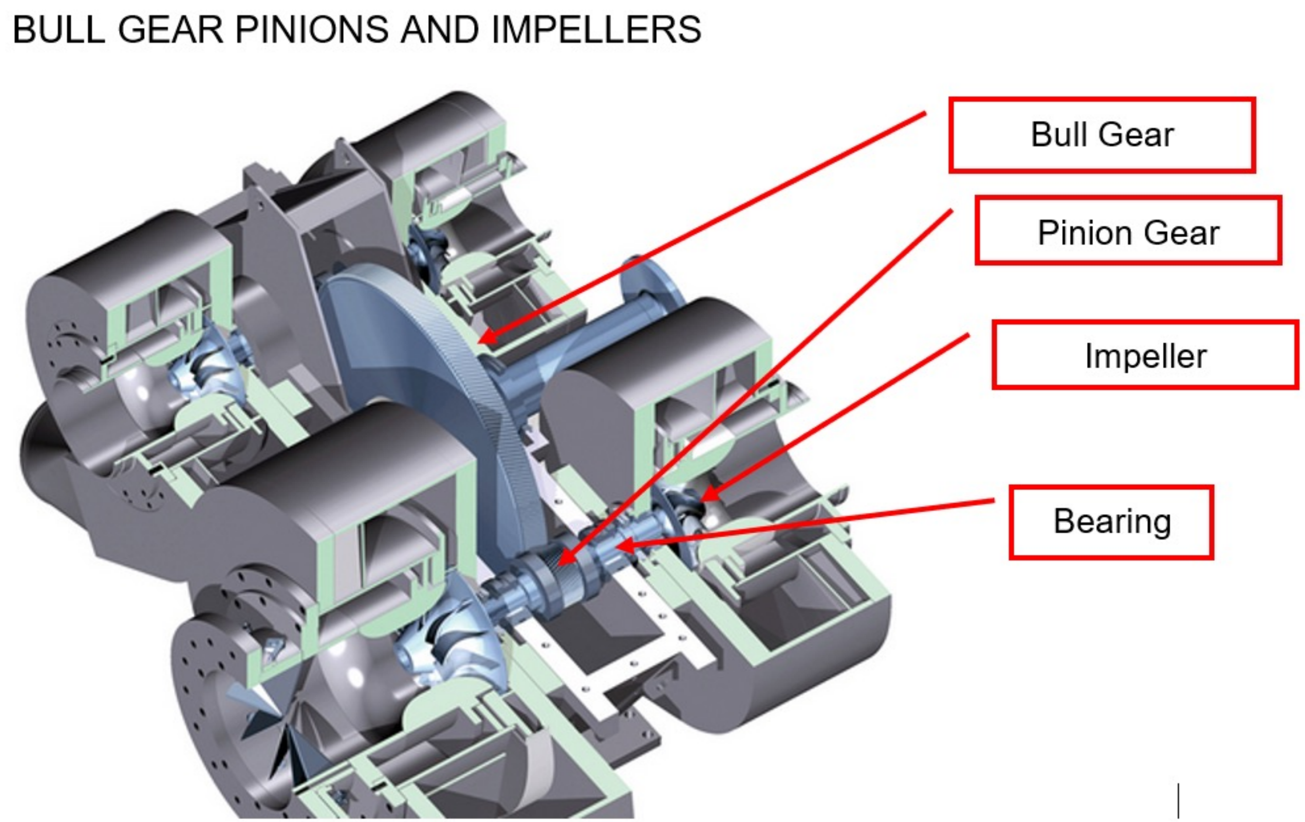
4: The analysis required significant input from operations.

5: It was difficult to track down the components generating the problem frequency.

6^{th} Most Difficult Case 40000 HP Compressor with oscillating vibration that resulted in vibration trips.

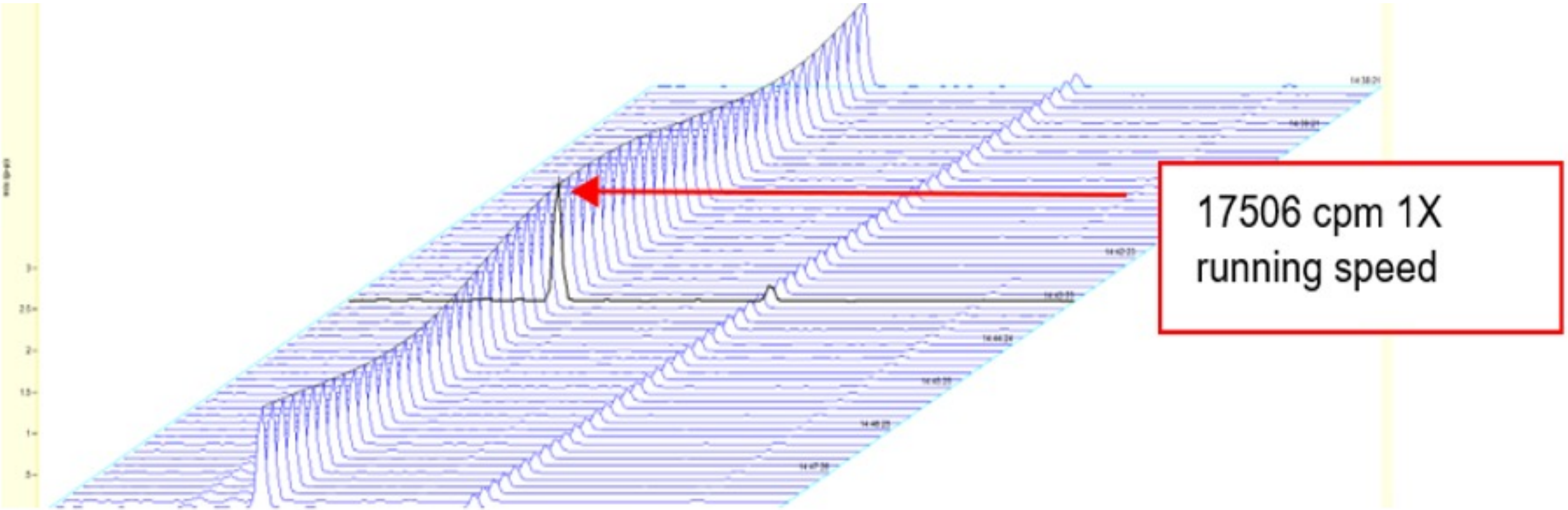
40,000 HP Compressor

Components within compressor



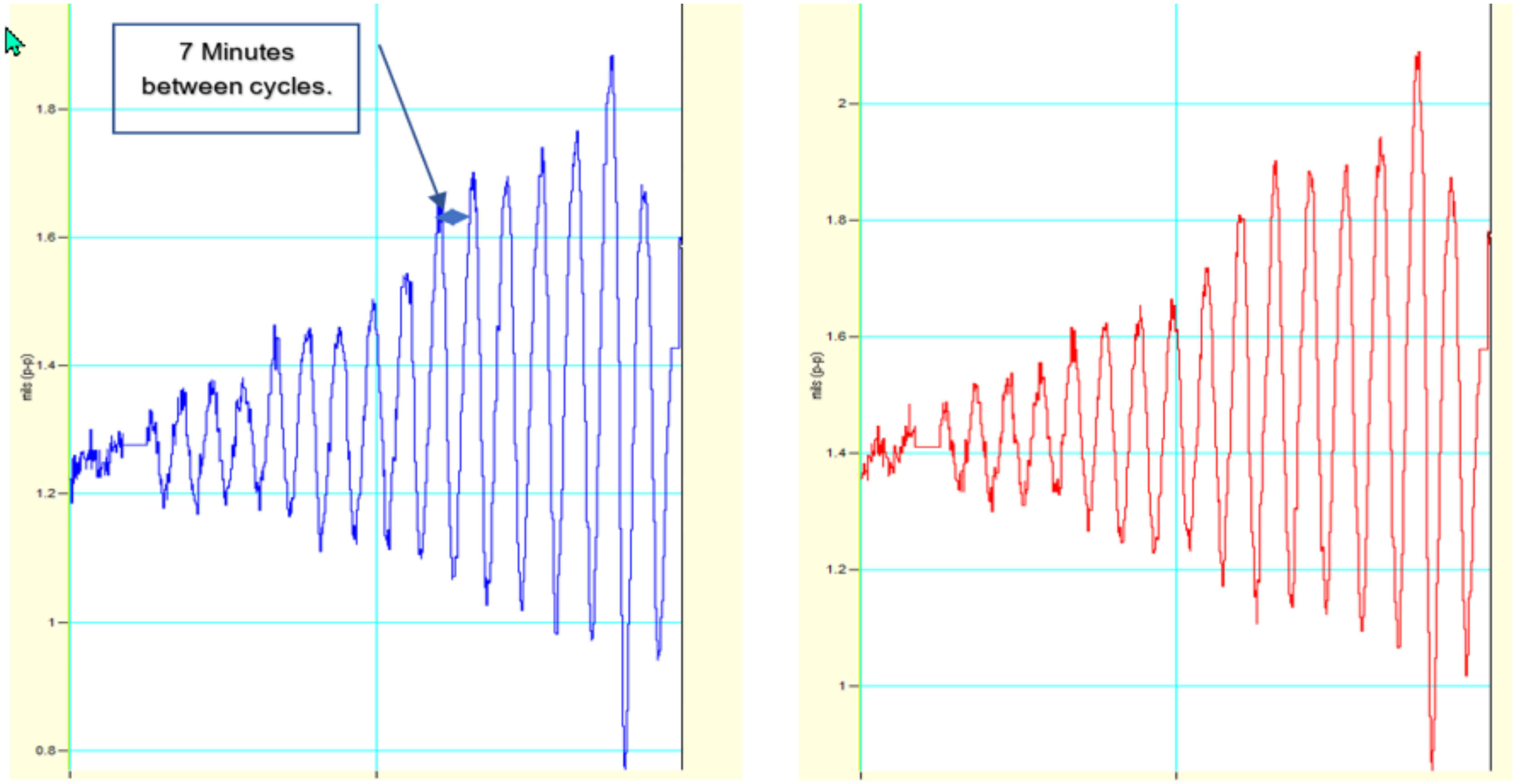
The impellers are overhung. In this instance, the impeller speed was 17506 RPM

<u>Symptoms of problem:</u> The vibration on both stage 3 proximity probes would slowly increase in amplitude followed by an equally slow decrease. <u>The period of one oscillation was approximately 7 minutes.</u> The peak in the amplitude would get higher each cycle until the compressor finally tripped on high vibration.

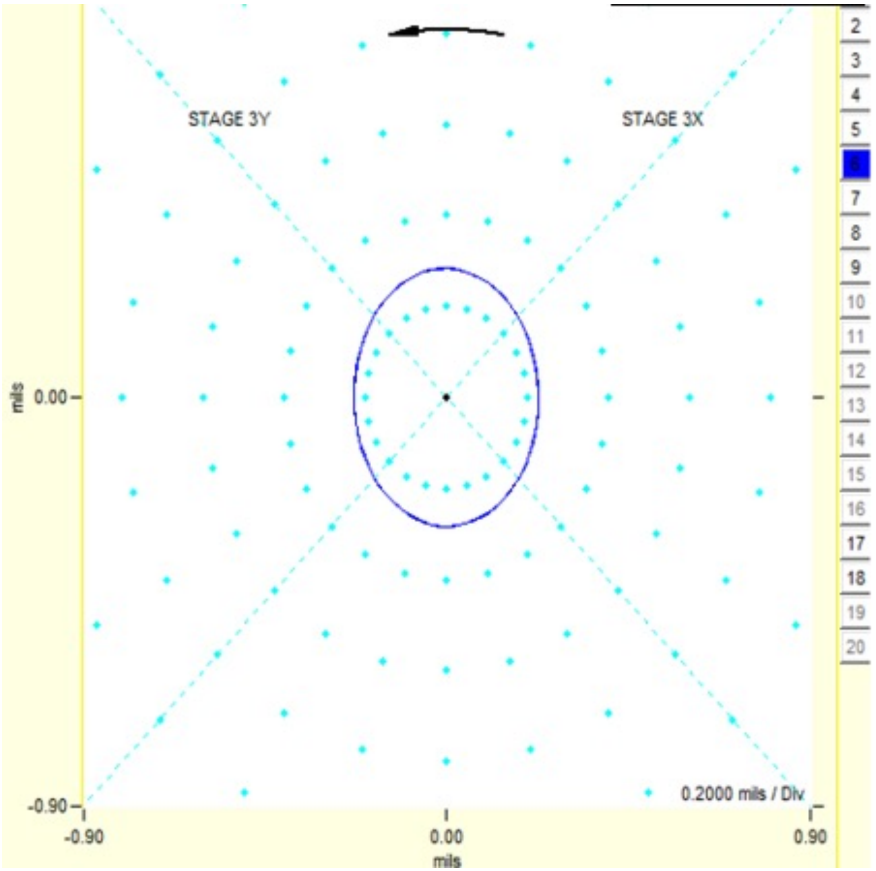


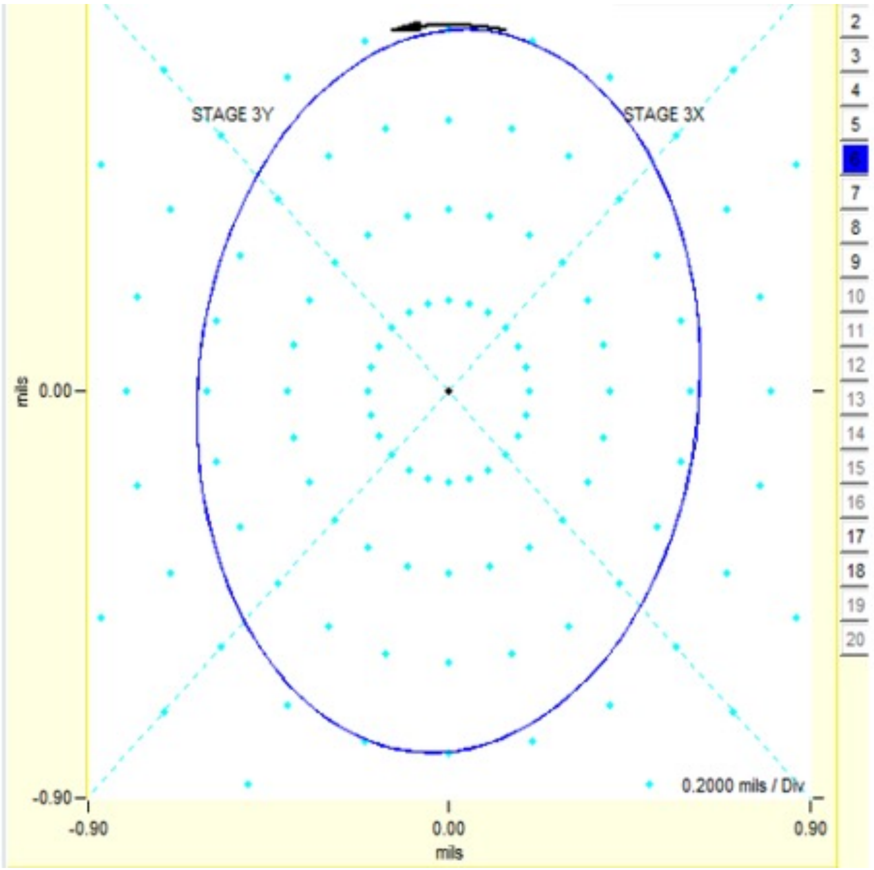
Waterfall plot of spectra from 3d stage proximity probe. Note how 1X running speed increases and decreases.

Two-hour time span showing trend of vibration on stage 3X and 3Y proximity probe amplitudes. Note how each cycle lasts approximately 7 minutes. Every cycle, the vibration gets progressively higher. This was disturbing to the operators in that they would see the vibration was getting worse over time.

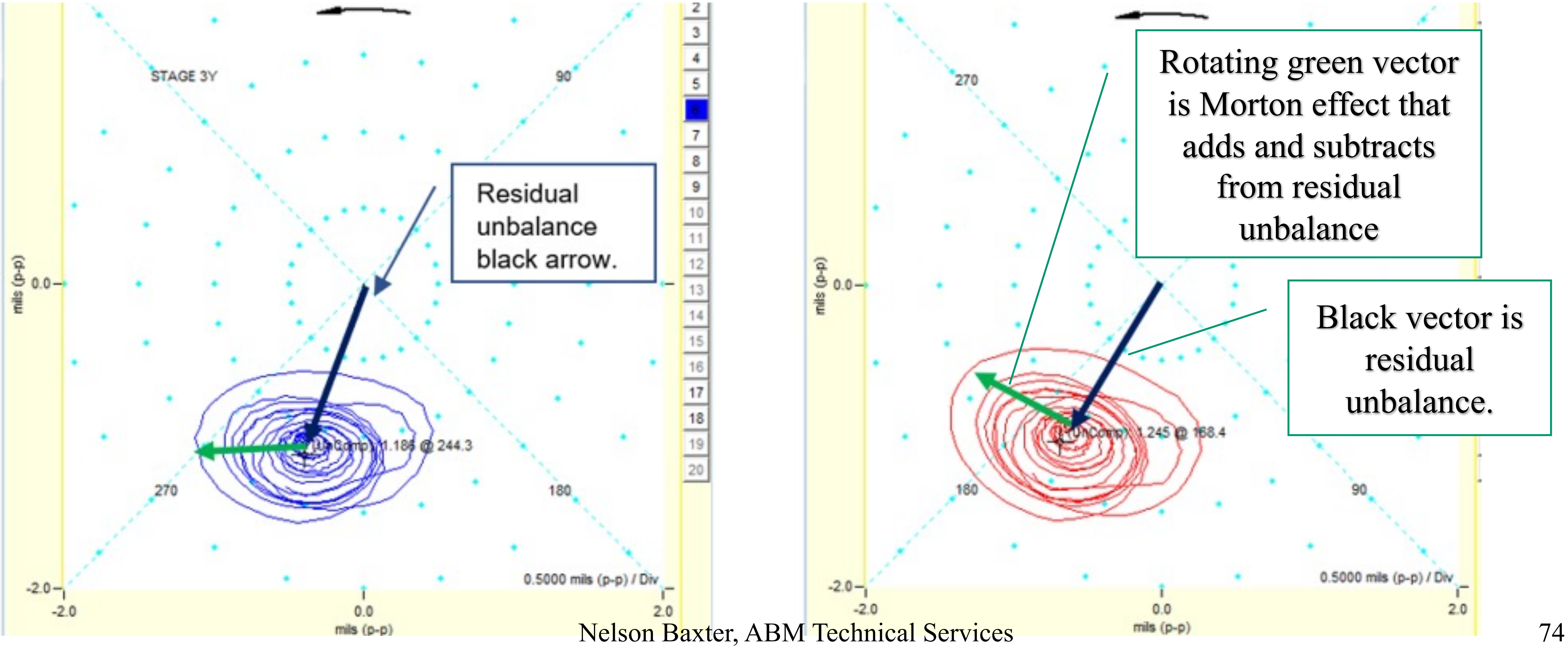


Orbits from low to high point in one 7 minute cycle





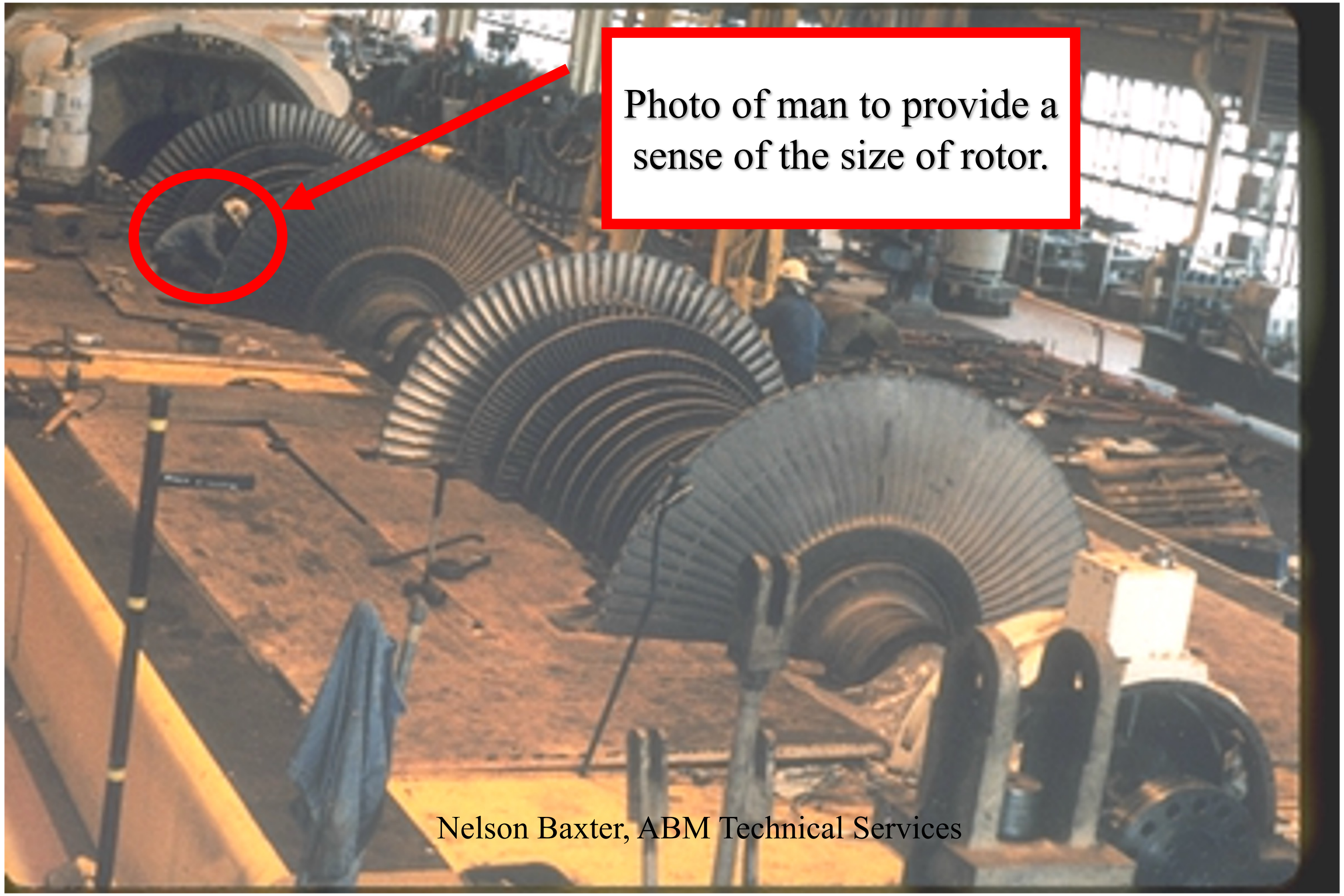
Polar plots from X and Y probes show spiraling outward vectors.



- The Morton effect occurs as a result of uneven heating of the shaft at the high spot caused by the residual unbalance of the impeller. Because of the high-speed nature of these overhung impellers, the point of the slightly elevated shaft temperature slightly lags the unbalance. This means the unbalance slowly rotates around the shaft casing to come in and out of phase with the unbalance. The heating also tends to slowly increase with each cycle. The solution was to change the bearing design to reduce the amount of shaft heating. This case is a good example of using various test techniques.

- This case is a good example of using various test techniques. The overall values shown on the strip chart gave insight as to the amount of time between the cycles. The orbit displays showed how much bearing clearance was being used. The polar plots which showed the cyclical relationship between the unbalance vector and the slowly rotating Morton effect vector were very important in gaining an understanding into the nature of this problem.
- This case was difficult because the Morton effect is extremely rare. It also required very good instrumentation to monitor several parameters simultaneously during non-steady state conditions.

5th Most Difficutl Case:
Low Pressure Rotors of Large Steam Turbine

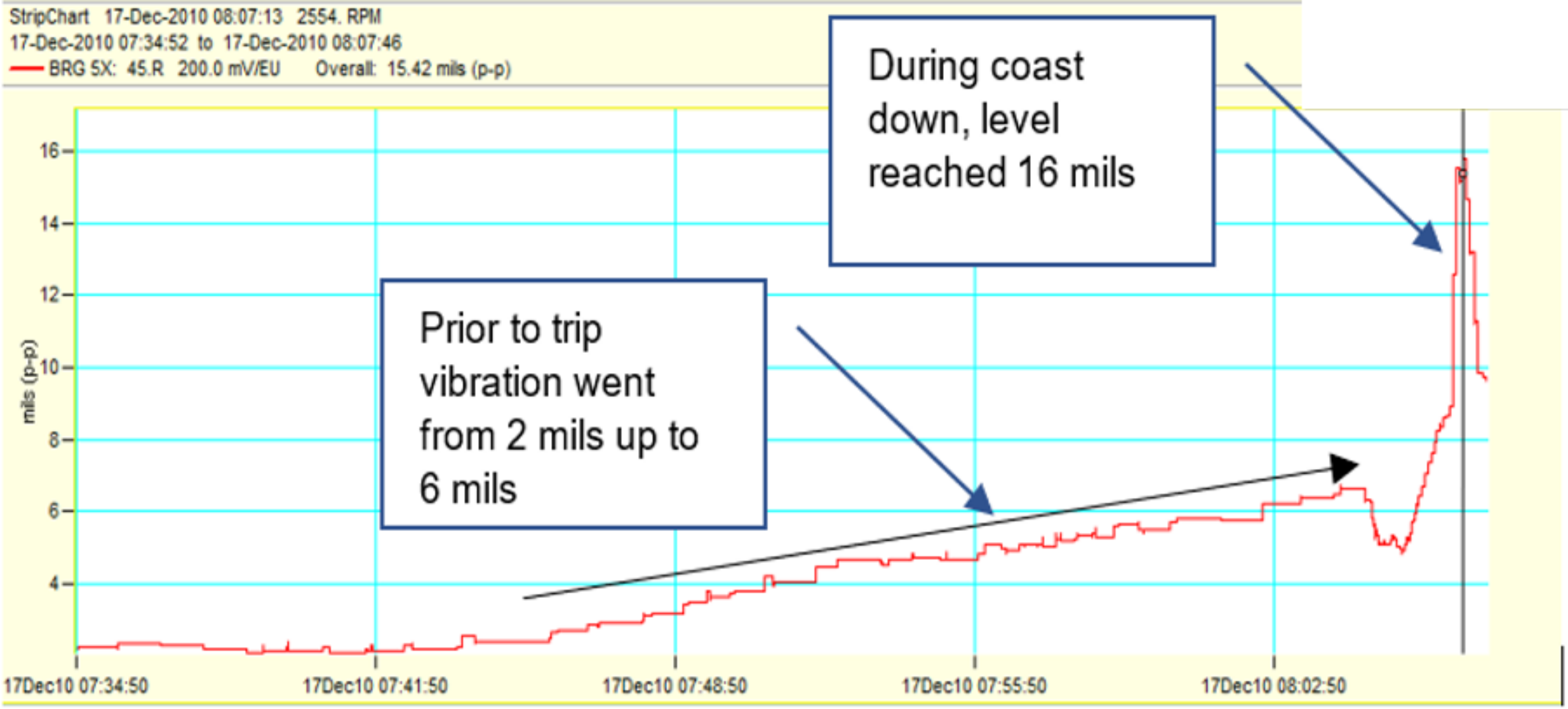


Test Setup

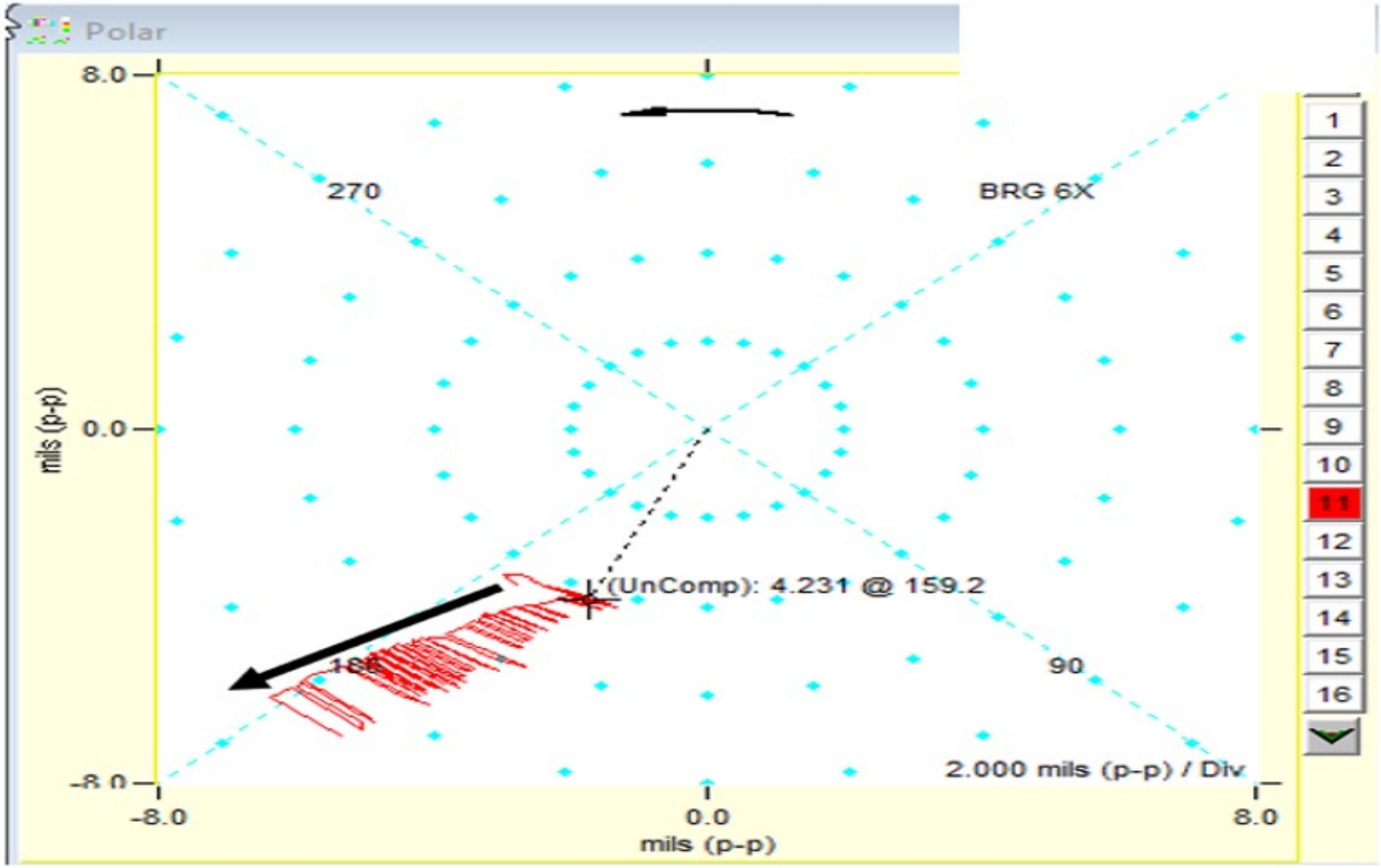
- A large turbine was monitored with an advanced monitoring system following an overhaul. The following X-Y proximity probe data was recorded:

  - Overall amplitude plots
  - Time waveform plots
  - Spectra
  - Waterfall Plots
  - Bod’e Plots
  - Shaft Orbits

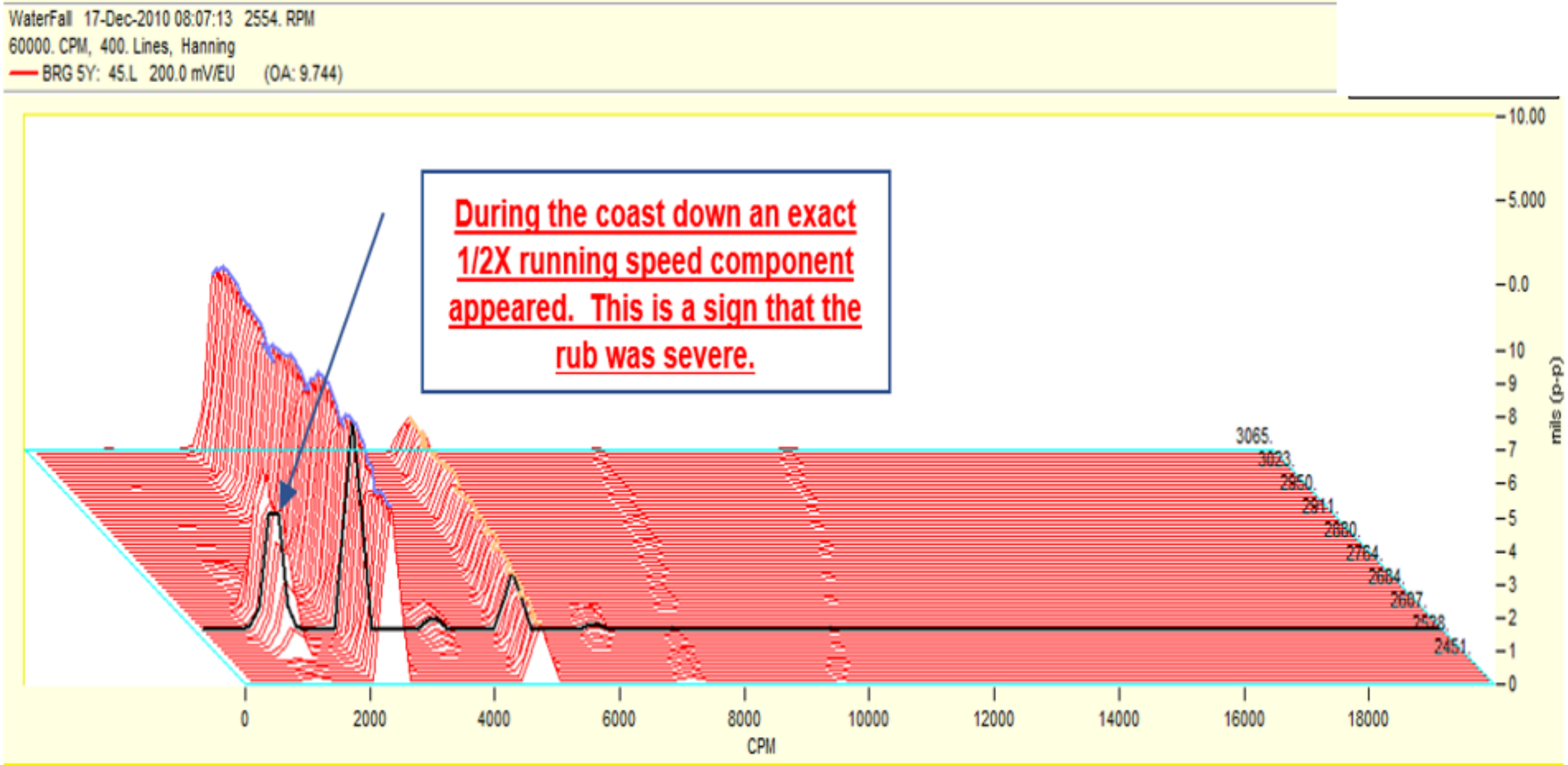
Strip Chart of B LP Rotor overall vibration level



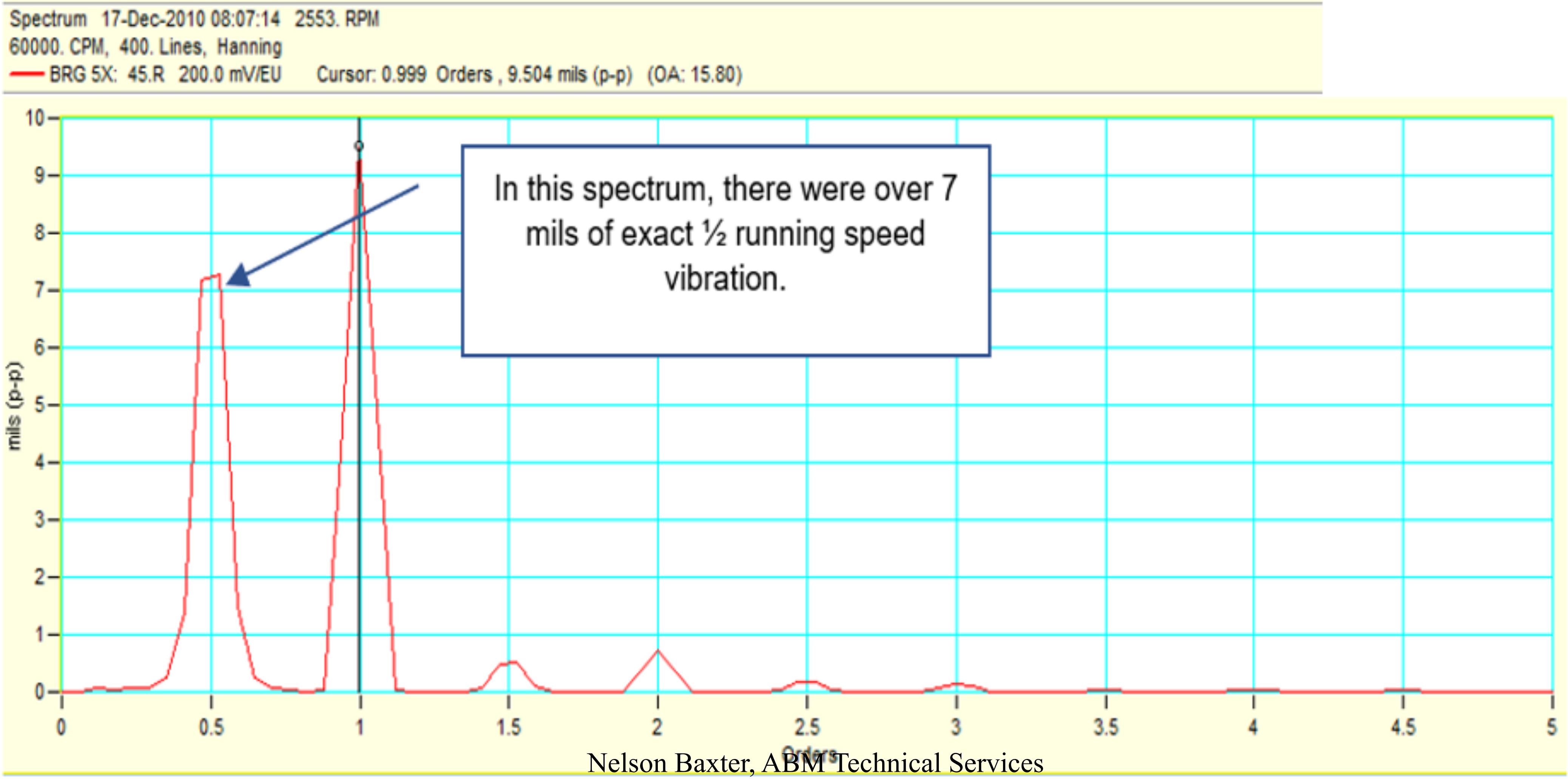
LP Turbine Polar Plot with speed held constant. Rub is increasing vibration and changing phase.



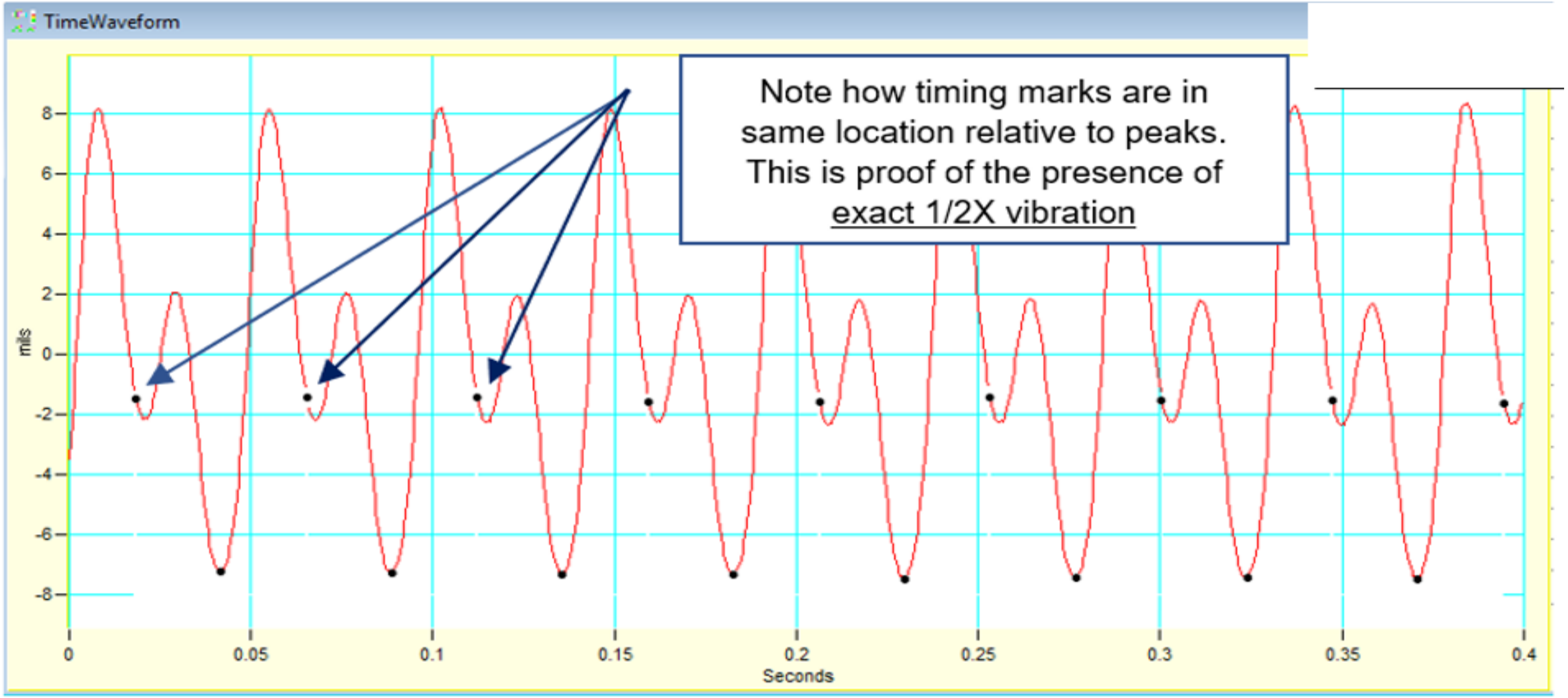
Map Plot During Coast Down



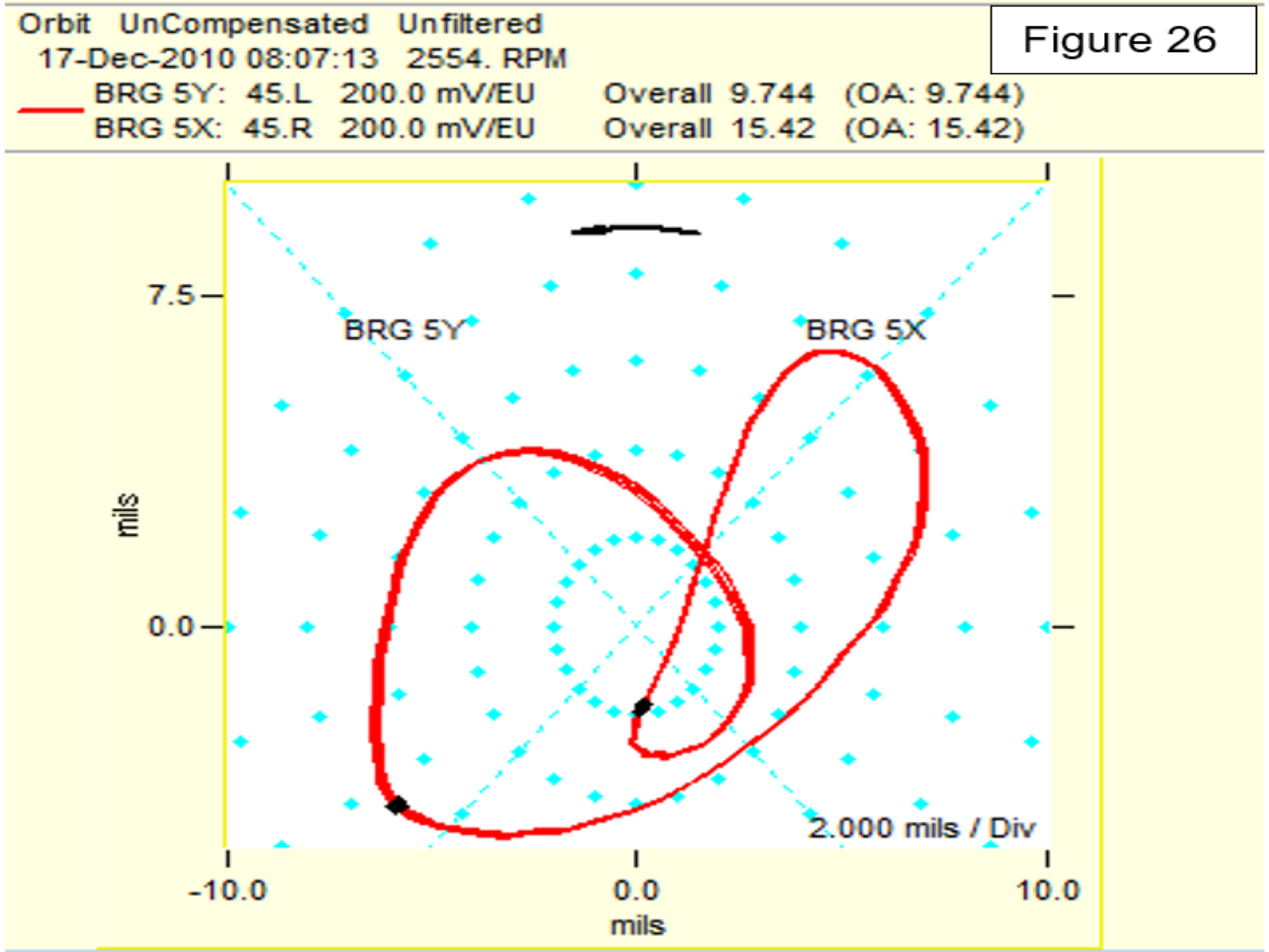
Spectrum with exact 1/2X Running Speed



Use of time plot to determine vibration is exact 1/2X Running speed.



Orbit when exact 1/2X running speed was present



Important Points

- Polar Plot showed that at constant speed the amplitude was slowly increasing and the phase was changing.
- The map plot clearly showed the introduction of the 1/2X as the critical was traversed.
- The time plot with key phaser mark showed that the vibration was indeed exact 1/2X
- It was discovered the LP rotor spray system was wired wrong causing the spray to be activated causing thermal distortion and the rub. Once it was fixed, the turbine started without incident.

Difficulty of problem

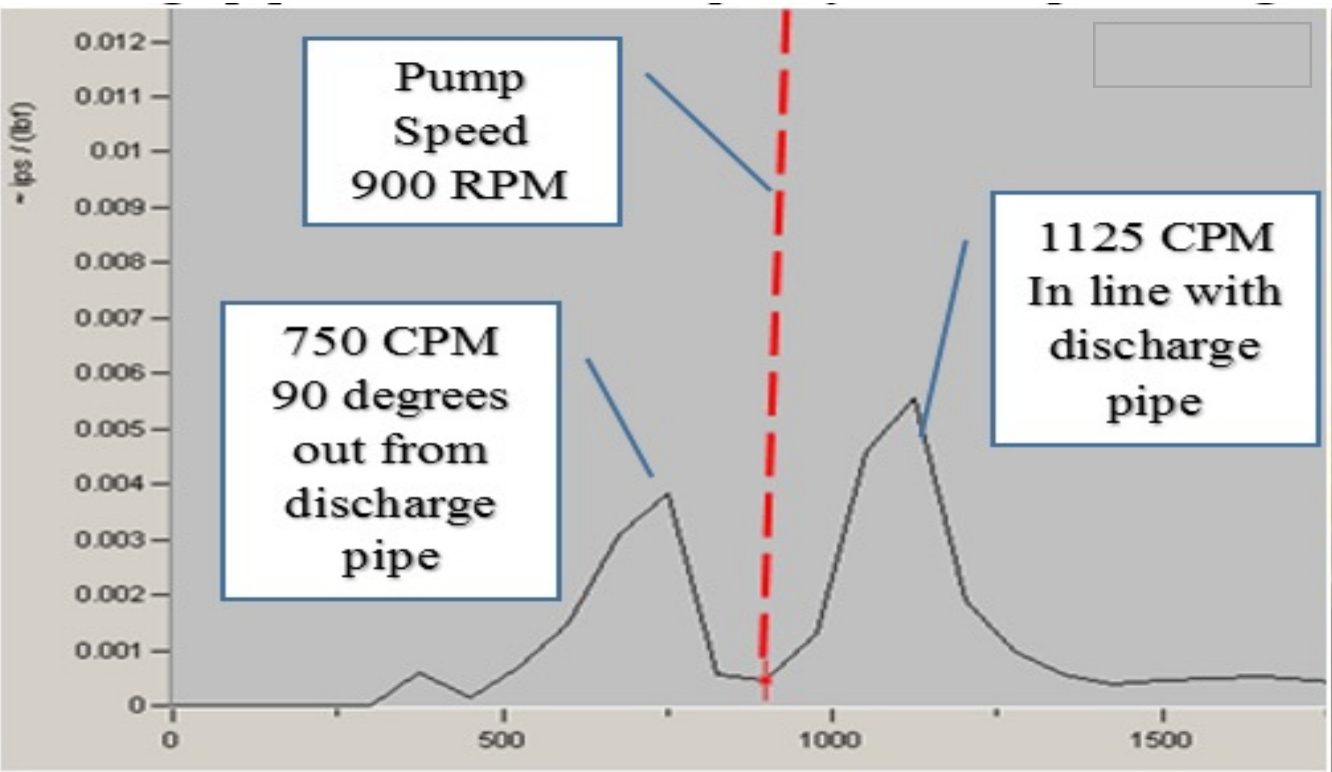
- This problem was difficult because this was a 500 Megawatt turbine that had just been overhauled and it needed to get back into operation to supply power to a major city.
- It required good test equipment that was able to record the transient vibration that occurred.
- The analysis required very closely looking at the data. This problem could have been mistaken for oil whirl. Looking at the time plot and the key phaser mark showed the high vibration that occurred during the coast down was exactly at ½ running speed. All the data together indicated there had been a very hard rub.
- The knowledge there was a hard rub, most likely from thermal distortion, led to discovering the LP rotor spay system had malfunctioned and sprayed the hot rotor.

4^{th} Most Difficult Case: Vertical Pump put on VFD for speed control

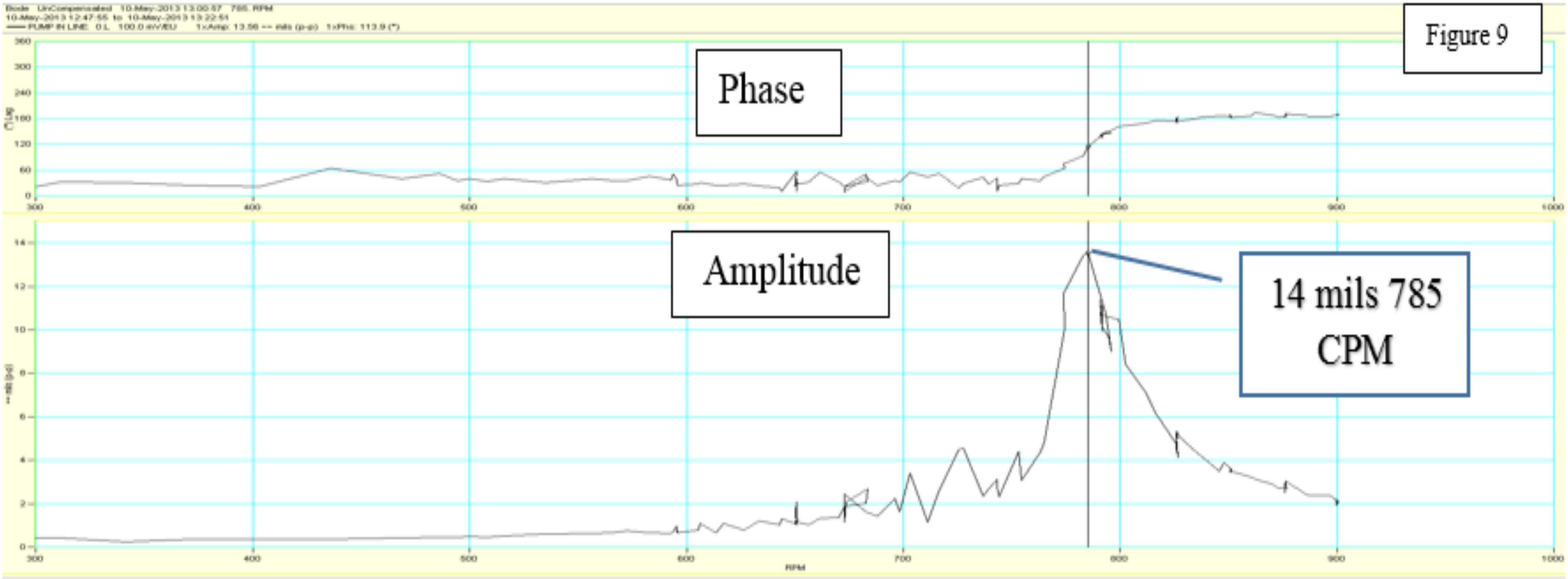
Case history of Vertical Pump change to VFD operation

- A water company installed a VFD on one of its vertical pumps to allow it to more closely match the flow to the demand.
- While on site to do other work, an impact test was performed on the pump that was to have the VFD installed.

Impact Test Result



Vibration vs speed with VFD installed



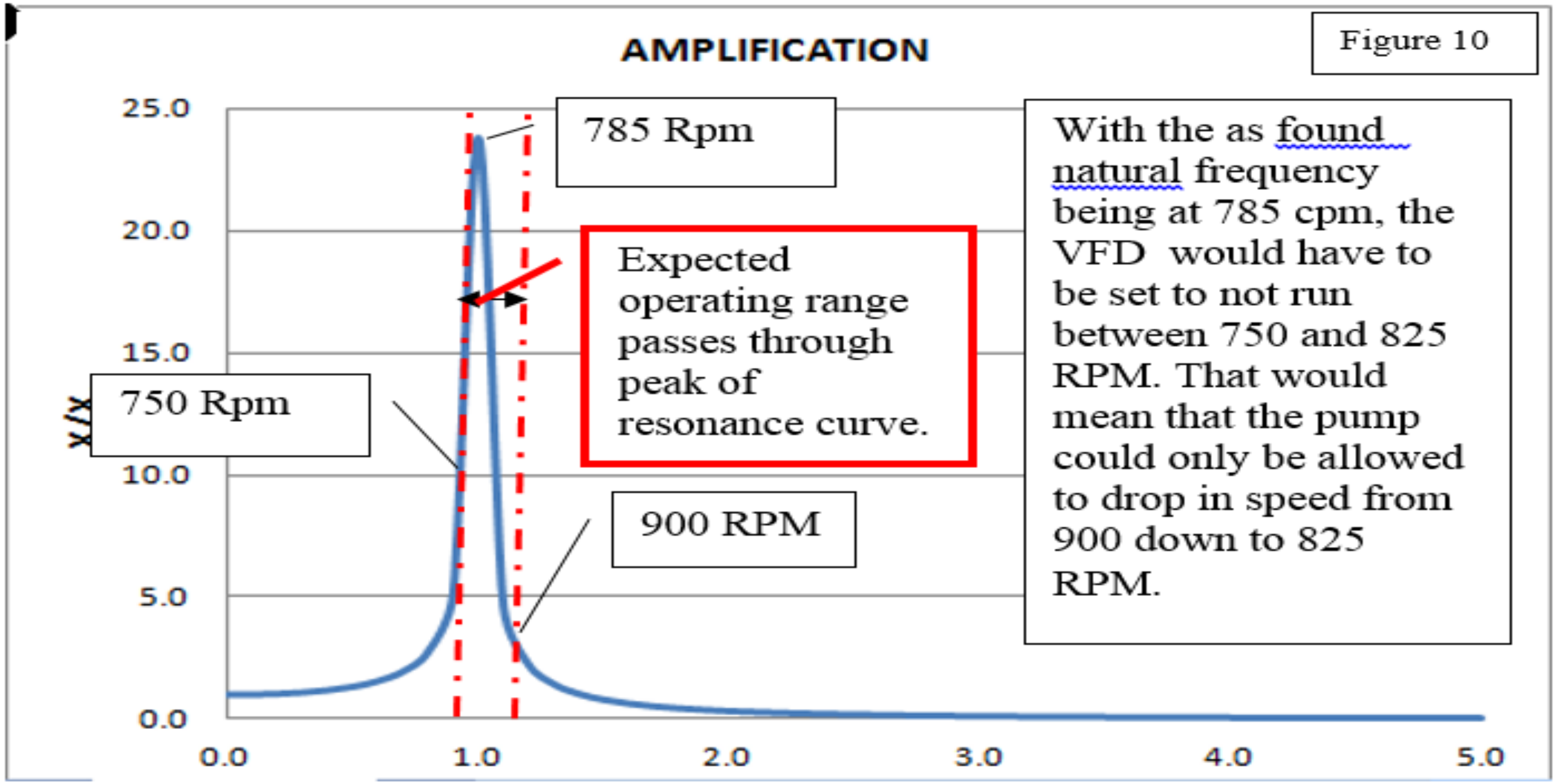
As predicted by the impact test, there was an amplified response at 750-800 RPM

Solution

- Step 1- Determine damping Q= $\frac{\pi N_C \Delta\theta}{360\ X\ \Delta F}$
- Step 2 Calculate amplification at resonance Q= 23.5
- $\xi = \frac{1}{2Q}$ This results in a damping ratio = .021
- The damping ratio can be used in the amplification factor equation to determine response as a function of where the pump operated at relative to natural frequency

Solve response equation $X/X_0 = 1/\sqrt{[1-(\frac{\omega}{\omega_n})^2]^2 + (2\xi\frac{\omega}{\omega_n})^2}$

CALCULATED RESPONSE RATIOS WITH NO MASS ADDED



Shift Natural Frequency 10% by adding 21% more mass.

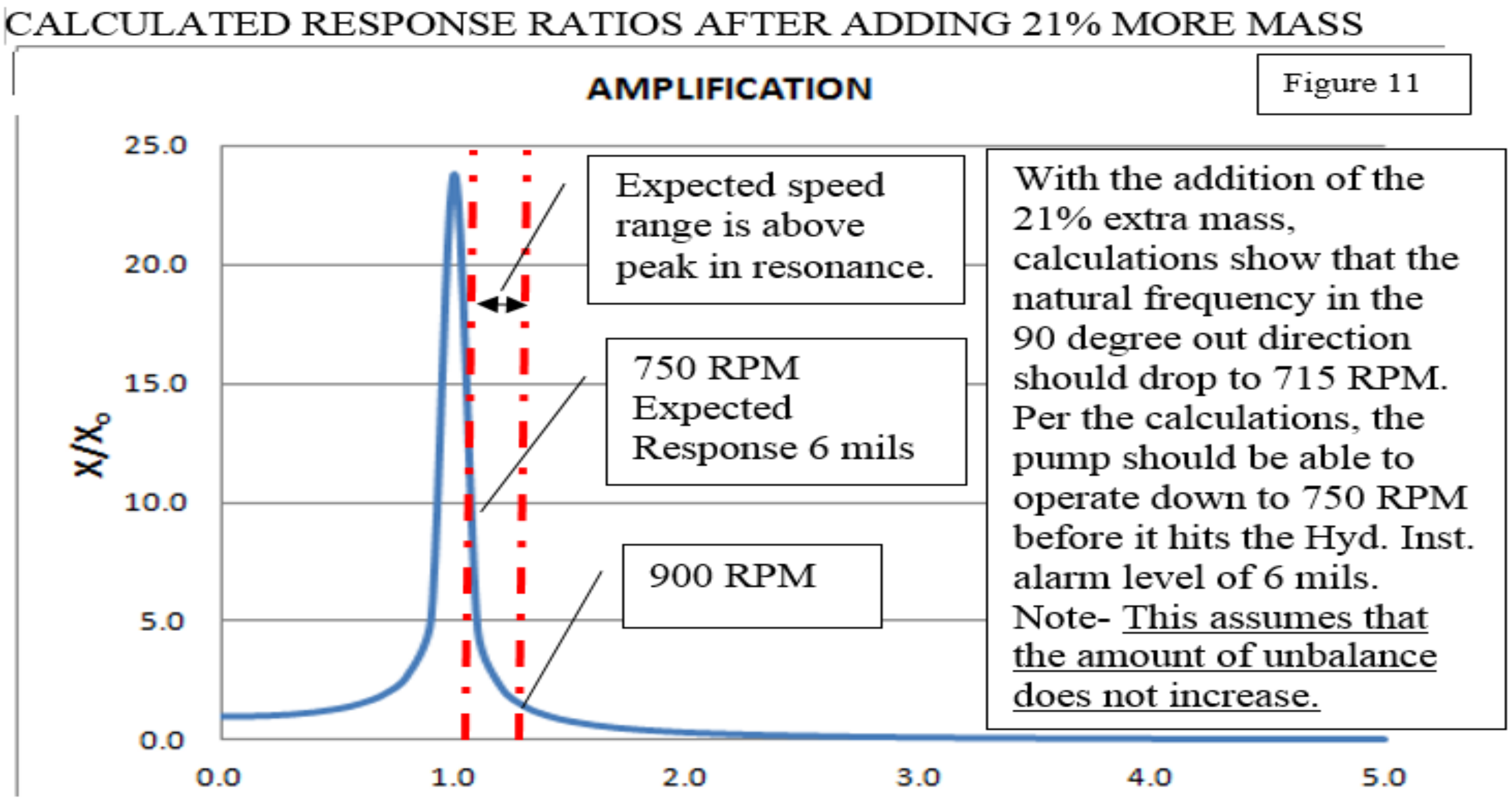
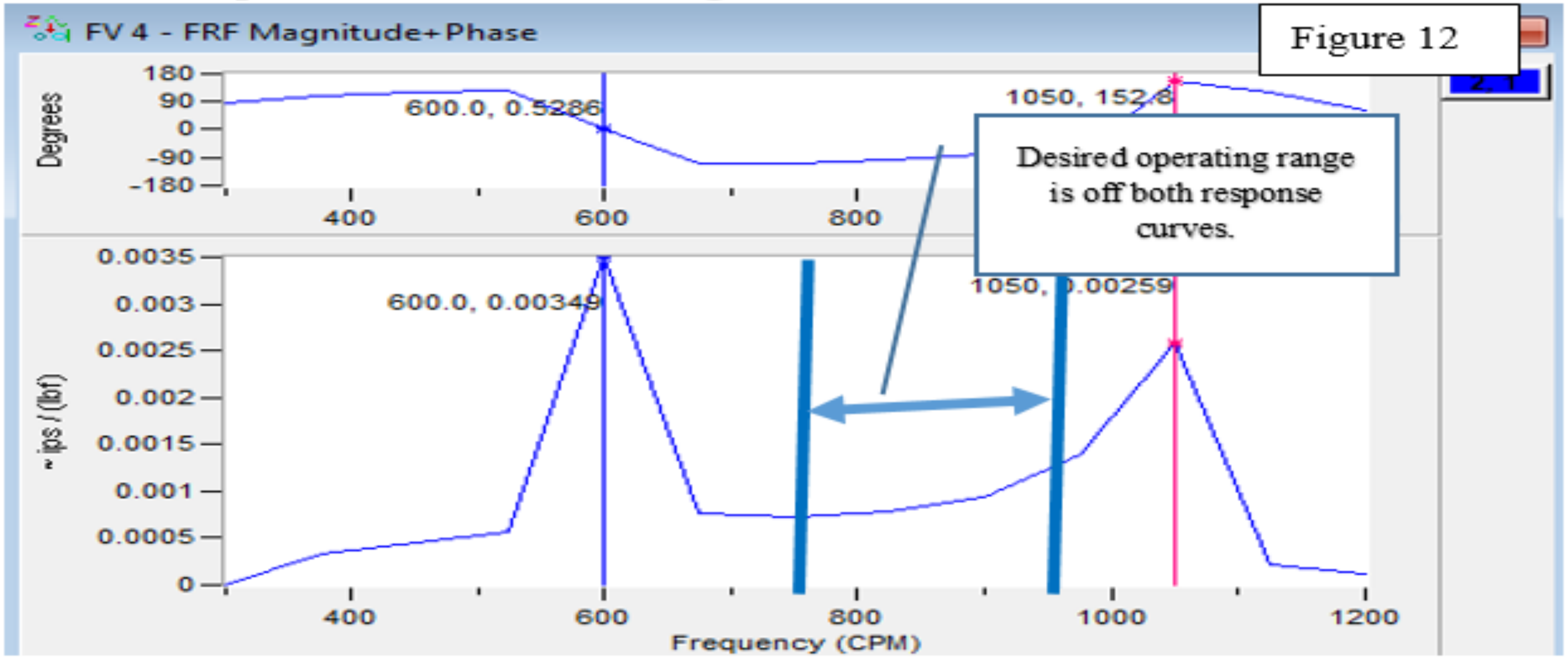


Photo of Pump with weight installed

Vertical Pump with Mass added to reduce natural frequency

After installing 3100 lbs at top of motor, impact test was repeated.

Actual Response Curve Following 21% Mass Increase



- Final Results:

	Before Mass	After Mass	
In line Natural Frequency	1125	1050	
90 Degree Out Natural Frequency	785	600	
Maximum Response in 750-900 Range	14 mils	3.7 mils	74% Reduction

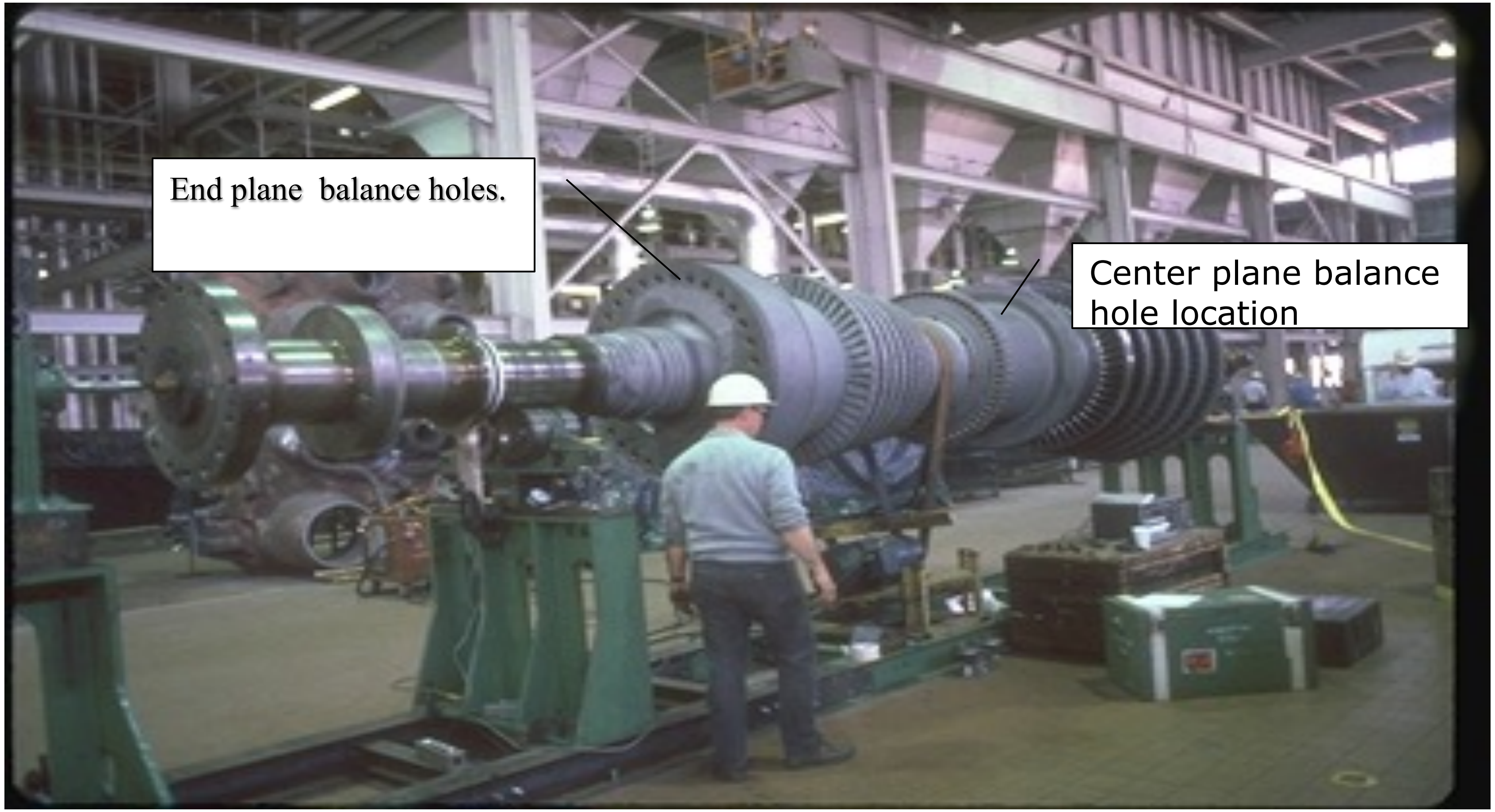
Conclusions: When making the decision to convert a vertical pump to variable speed

- Impact test the pump in both the in line and 90 degree out directions.
- Make sure the drive system has proper filters to eliminate line harmonics.
- Locate the drive as close to the pump as possible
- Make sure that if non drive pumps are in the same header that pump will not be pushed too far back on its curve.
- Check piping lengths to determine if there could be acoustical resonance problems.

3d Most difficult Case: 500 Megawatt Turbine HP Rotor modal balance 2 modes. Required both mid-plane and end plane balance shots.

- Two Problems
  - HP Rotor of 500 Megawatt Turbine had 15 mils of vibration as it traversed its 1420 RPM critical speed.
  - After the turbine got to full speed, the vibration was also high at over 7 mils at 3600 RPM.

500 Megawatt Turbine Rotor



1st Critical Speed Response both bearings in phase 13 & 15 mils

BEARING 1 RUNOUT COMPENSATE NYQUIST BEARING 2 RUNOUT COMPENSATED NYQUIST

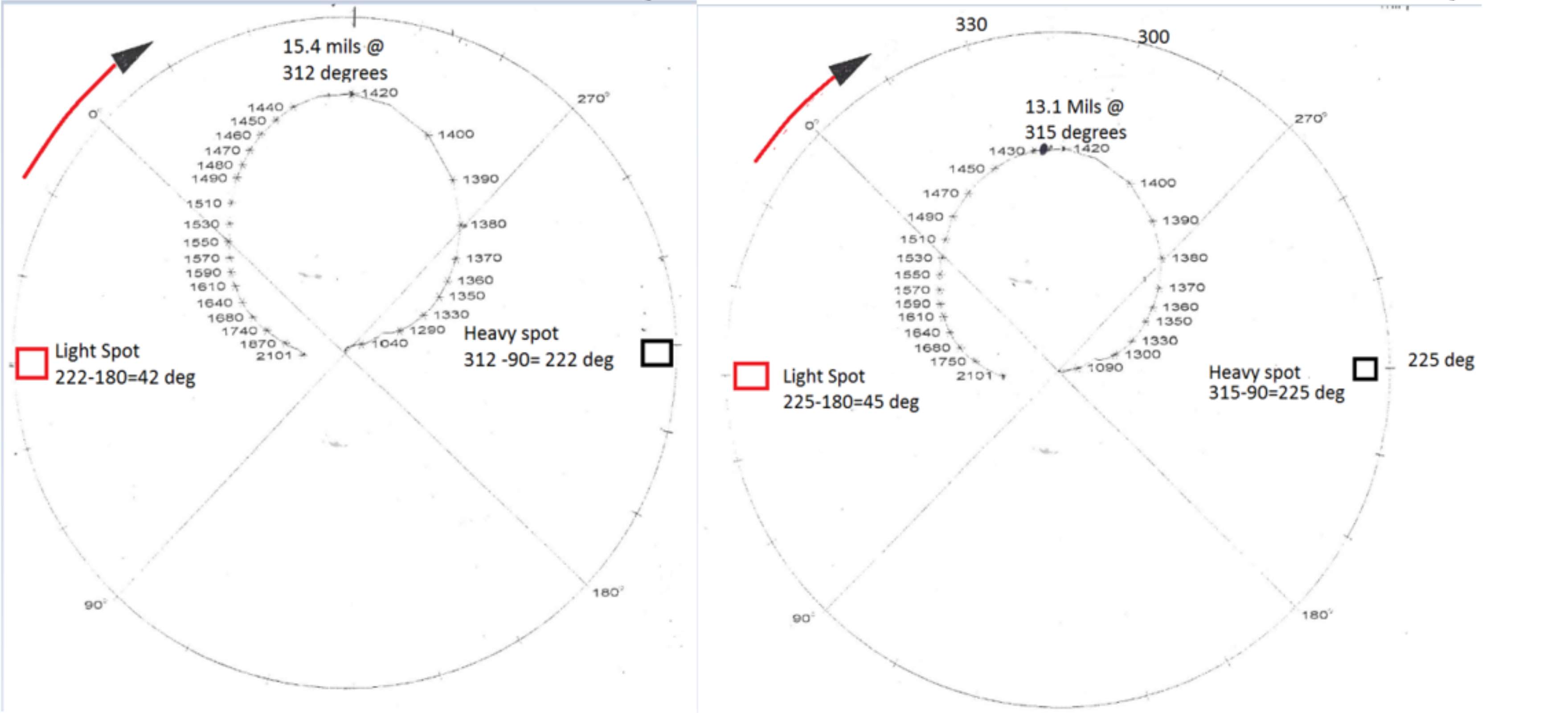


Figure 20 Nyquist plots showing 1st critical speed response

Solution to first mode critical speed response

- Vibration response was static in nature(phase angles were nearly identical) as would be expected at the 1^{st} critical.
- Past experience with this rotor design showed that the cause was a bow in the rotor. Since the rotor was bowed, the weight had to be installed in the mid rotor balance plane rather than in the end planes.
- A 90 degree lag at the critical speed was used to estimate the angle.

Solution Continued

- The inner and outer cylinder plugs were removed and 3 balance plugs weighing 9 oz each were installed. Note the temperature in that location was 1000 deg F. To insure that the plugs did not expand and get stuck halfway in the balance holes, they were preheated.
- Following installation of the weights, the critical speed response dropped from 15 mils down to 5 mils which was acceptable.

Full speed 2nd mode shape balancing.

- Following the mid plane balance shot, the unit was brought to speed and the following readings were obtained:
- Bearing 1 Left Probe 6.3 mils @ 260 degrees
- Bearing 2 Left Probe 6.9 mils @ 80 degrees
- Note that phase angles are 180 degrees apart indicating excitation of second mode which would require a couple shot (equal amounts of weight in each end installed 18 degrees apart).

Solution

- Past experience showed that the couple sensitivity was 1.5 oz/mil and that the couple lag angle was 68 degrees.
- Based upon the full speed data and using the above values, 9.8 oz was added at 28 degrees in plane 1 and 9.8 oz/mil was added at 200 degrees in plane 2.
- Final results
- Bearing 1 left probe 1.4 mils at 165 degrees
- Bearing 2 left probe 2.4 mils at 0 dereees

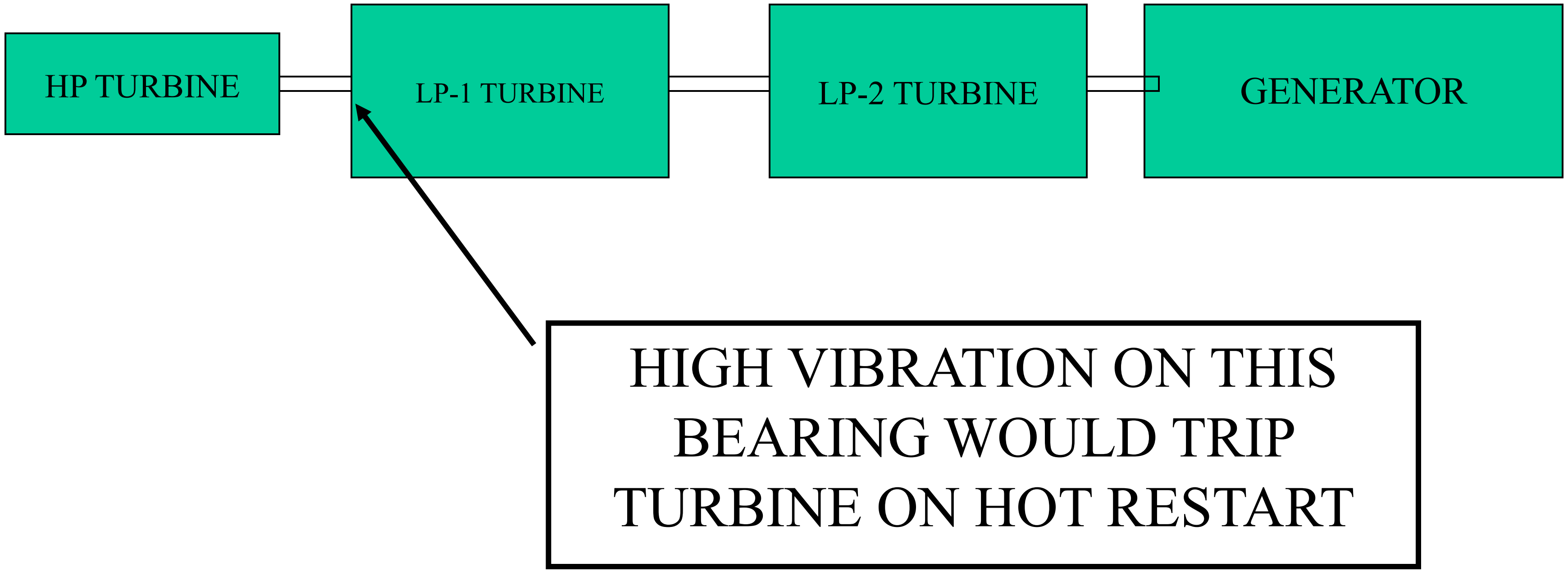
Conclusions

- This case history is an excellent example of balancing separate modes.
- The first step was to attack the first mode with a shot to the center of the rotor to reduce the critical speed.
- The second step was to install a couple shot to reduce the running speed motion that was the result of the excitation of the 2^{nd} mode shape. The amount of 2^{nd} mode influence at running speed is determined by using the static couple derivation method to graphically separate the at speed influence of the 2^{nd} mode.

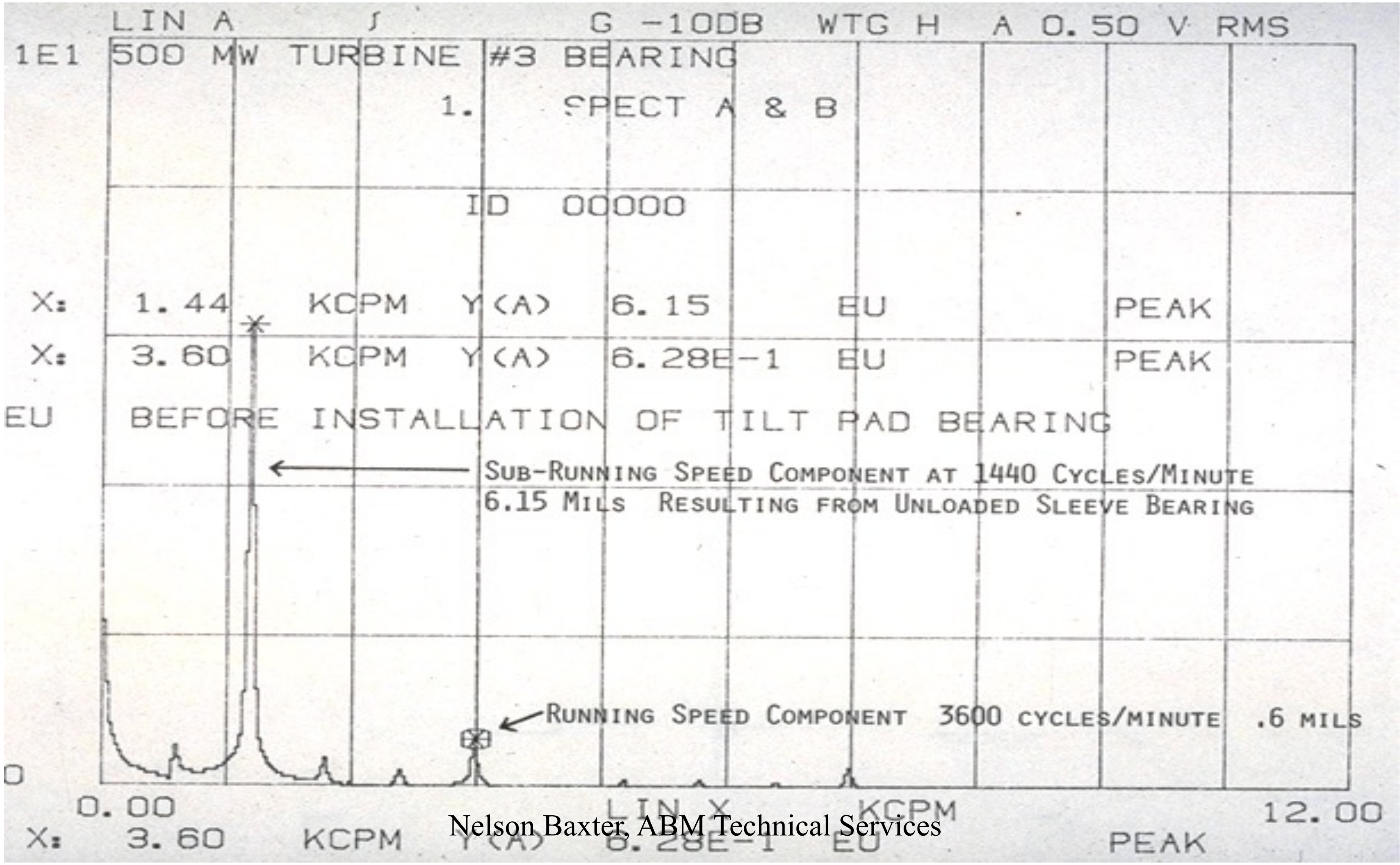
2nd Most Difficult Case: 500 MW TURBINE OIL WHIRL-WHIP

- When turbine would be shut down, if it was not started up within 2-3 hours, it could not be started up for two days because of excessive vibration.
- Sounds like Xenon poisoning on a nuclear reactor. Xenon poisoning in a nuclear reactor is what occurs when a reactor is shut down and it cannot be brought back in operation until the Xenon produced by the fission product decays.
- In this case the turbine could not be started for a couple days after it was shut down because of oil whip.

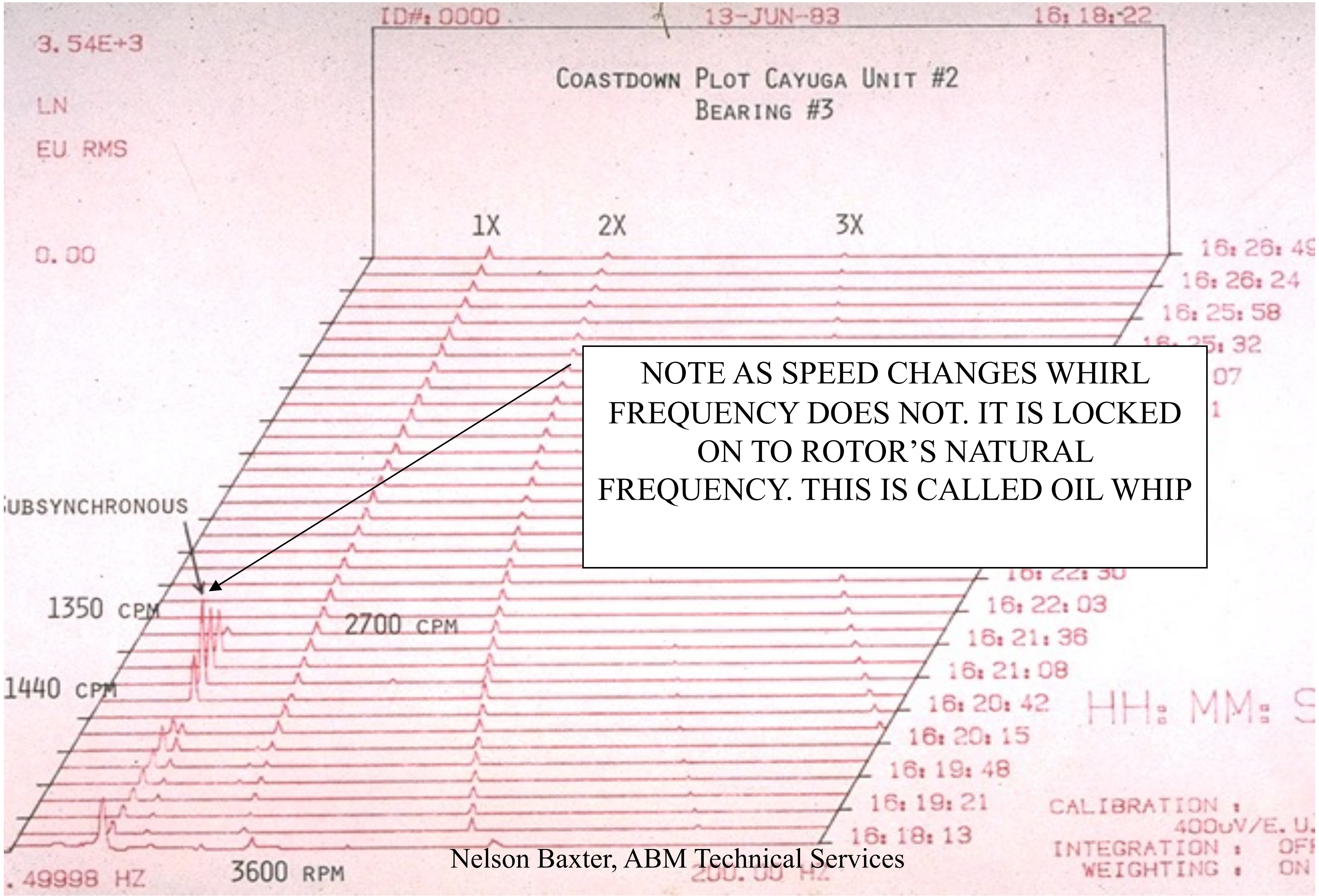
MACHINE LAY OUT



VIBRATION SPECTRUM



MAP PLOT



NOTE AS SPEED CHANGES WHIRL FREQUENCY DOES NOT. IT IS LOCKED ON TO ROTOR'S NATURAL FREQUENCY. THIS IS CALLED OIL WHIP

As the speed increased, the vibration was ok until the speed reached 2700-2800 RPM. At that point, the vibration would suddenly appear. As the speed increased, the oil whirl spectral component did not increase in frequency with the speed. The frequency of the vibration matched the 1st critical speed of the High-Pressure rotor. Once twice the critical speed was reached, the vibration would appear then as the rotor speed increased, the vibration stayed locked onto the rotor's natural frequency. What was occurring was oil whip. This is a condition where the oil whirl locks onto the rotor's natural frequency.

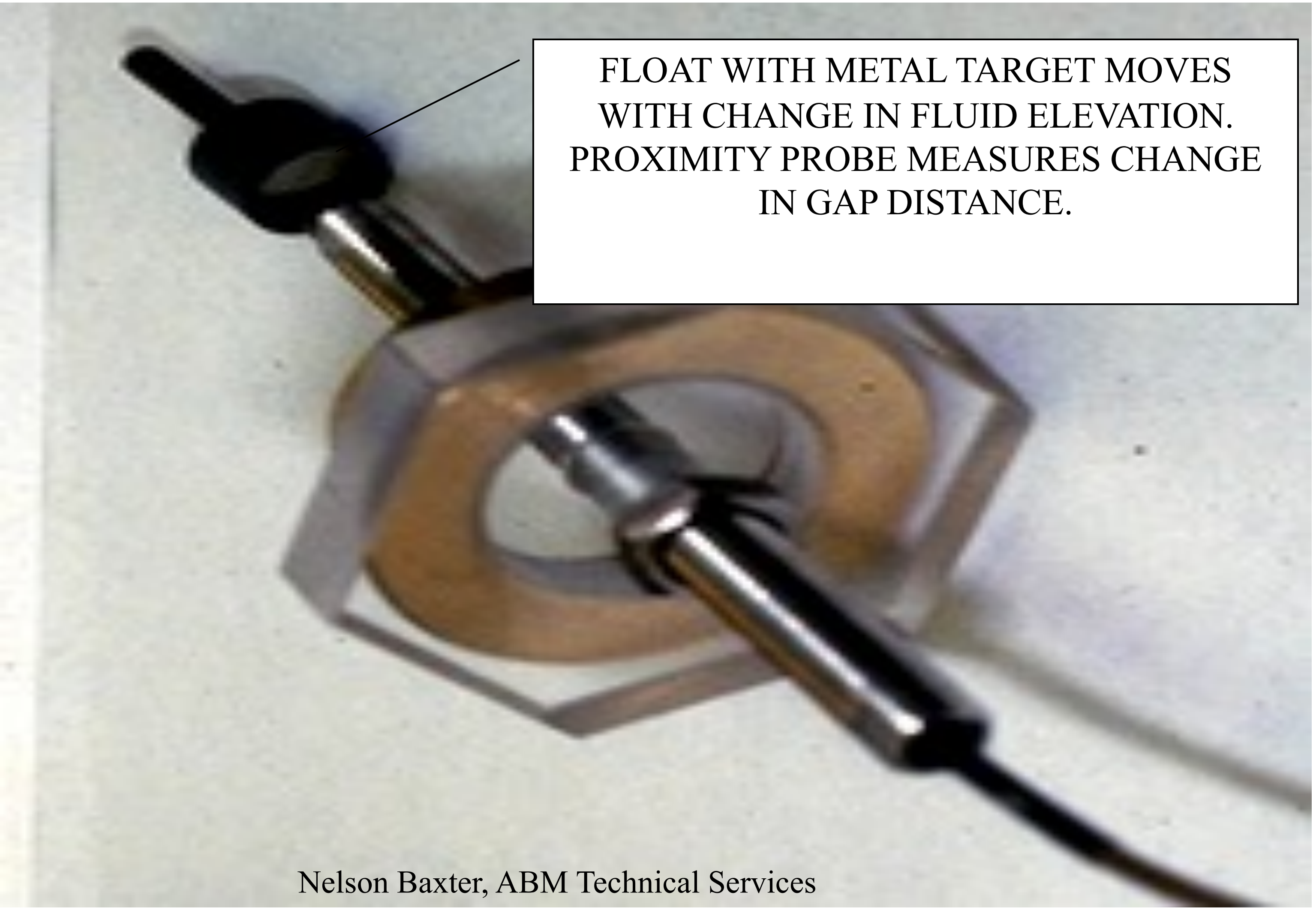
The question was then why did this problem only occur on a hot startup and not a cold startup. The most likely source of this problem was unloading of the bearing which would therein point towards a change in the alignment.

Alignment was the suspected cause of the oil whip.

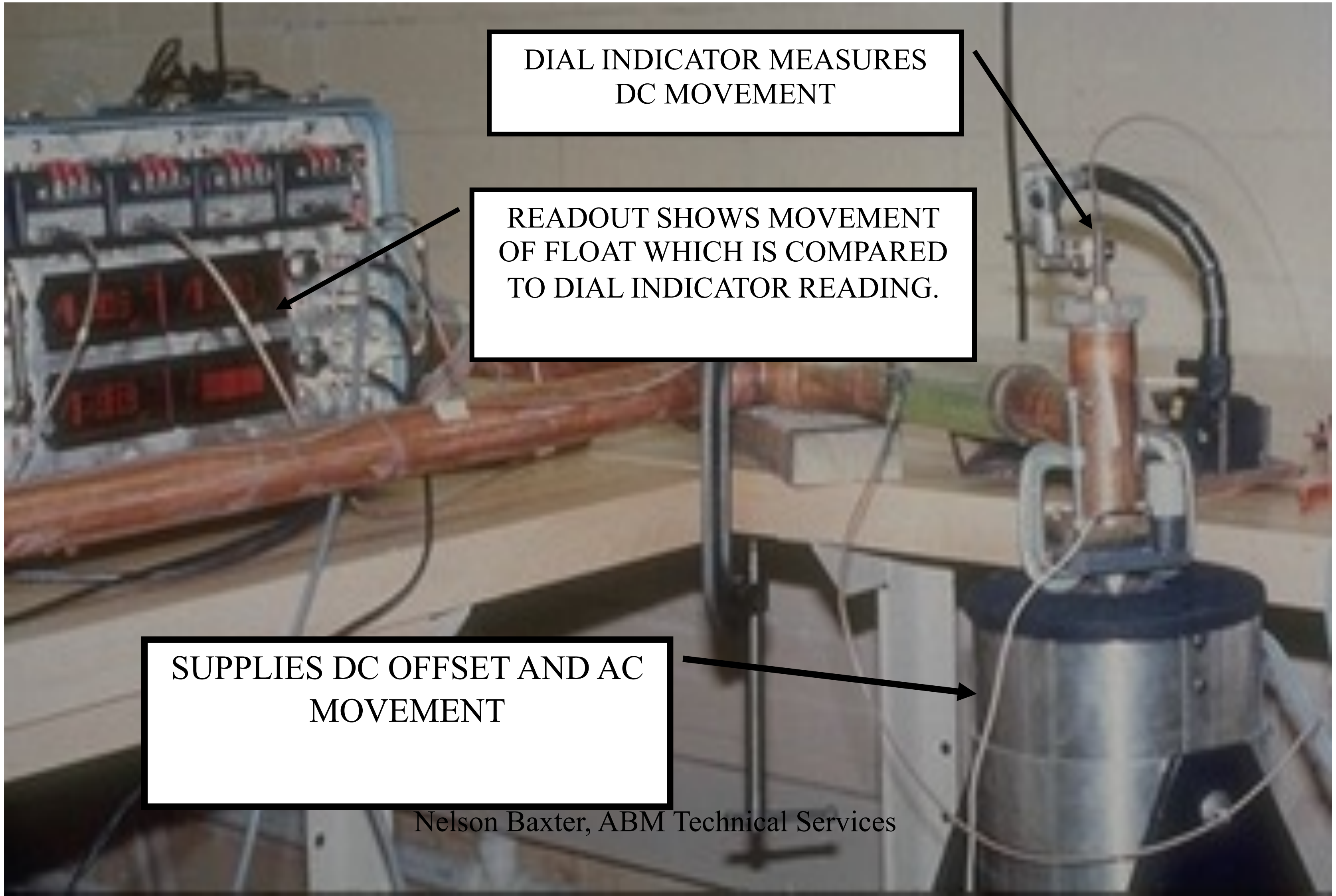
There was no readily available system to measure thermal growth in the bearings of a large steam turbine so technical papers were searched to see if it had ever been done and the method of doing so publicized.

What was found was the Canadians had used a glycol float system to successfully monitor the amount of vertical movement of the turbine bearings. The Italians had used a system that was filled with mercury. The Canadian system was chosen and all the components necessary were put together then the system was tested in a lab.

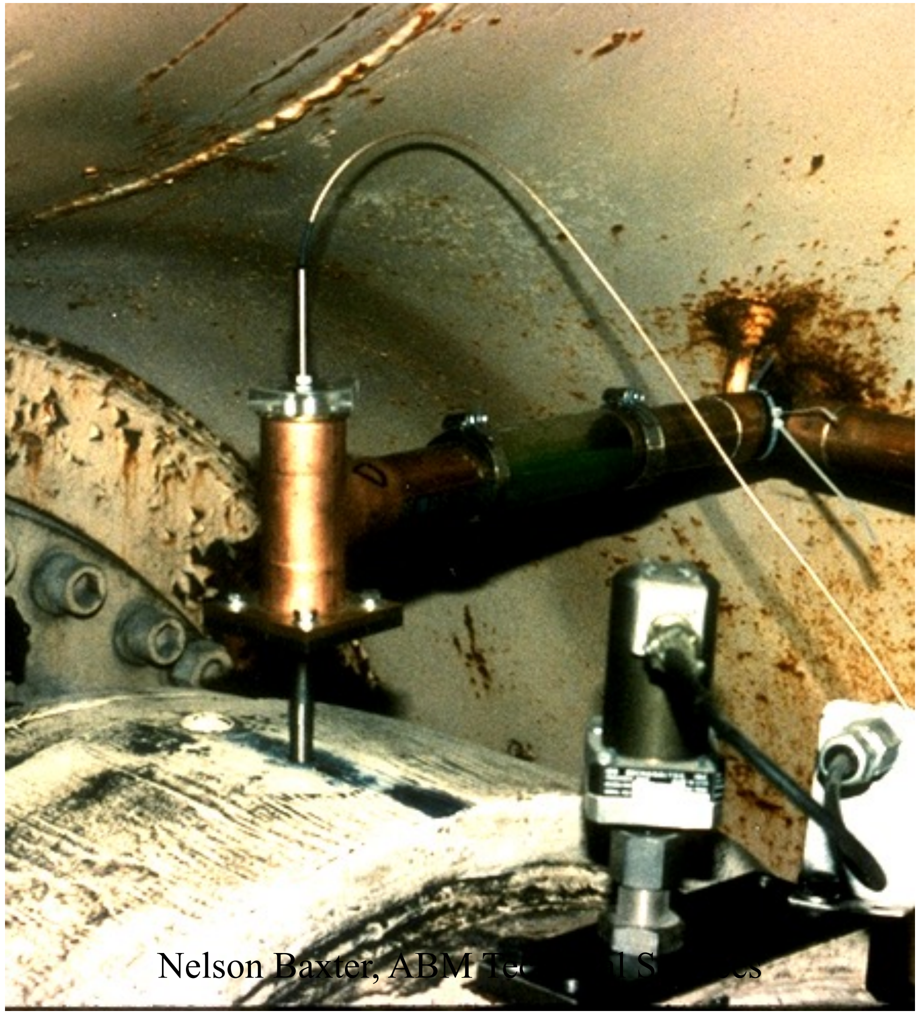
HEART OF SYSTEM



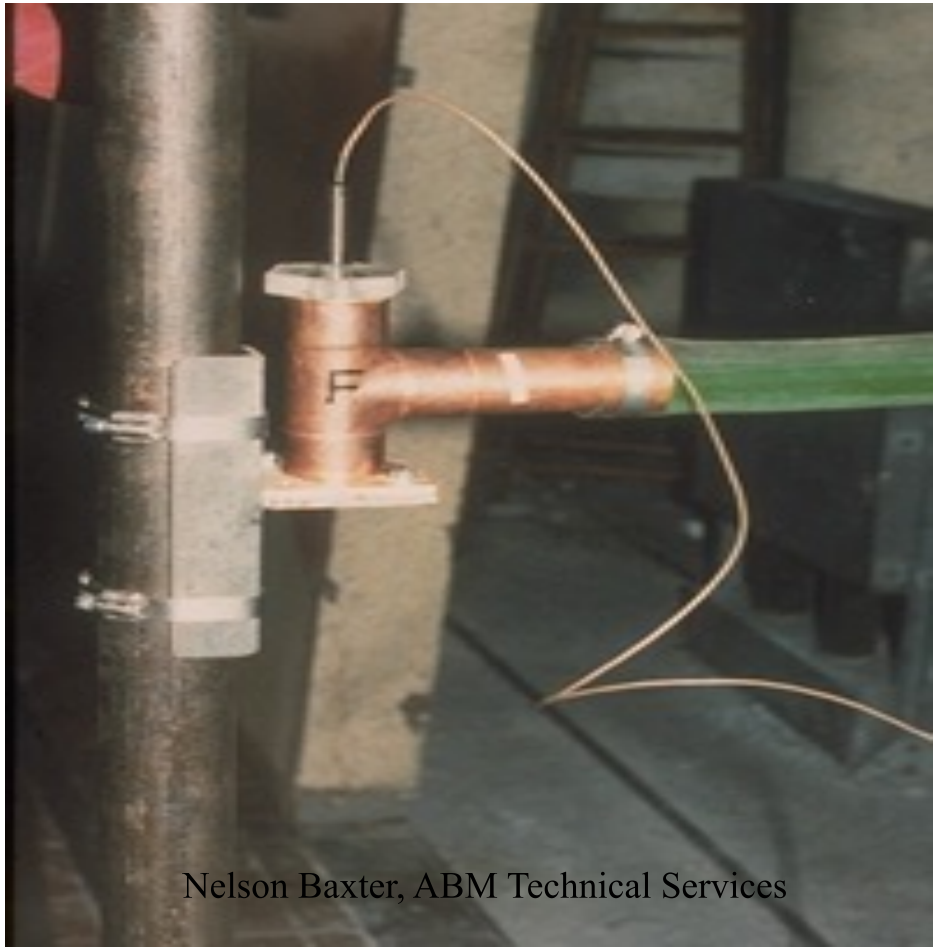
ALIGNMENT TEST SYSTEM



SETUP ON TURBINE



REFERENCE PICKUP TO ACCOUNT FOR FLUID EXPANSION OR LOSS. ANY CHANGES NOTED ON THE REFERENCE PROBE FROM GLYCOL EXPANSION OR LEAKAGE WERE SUBTRACTED THUS NULLIFYING ANY CHANGES IN GLYCOL LEVEL DUE TO THOSE CAUSES.



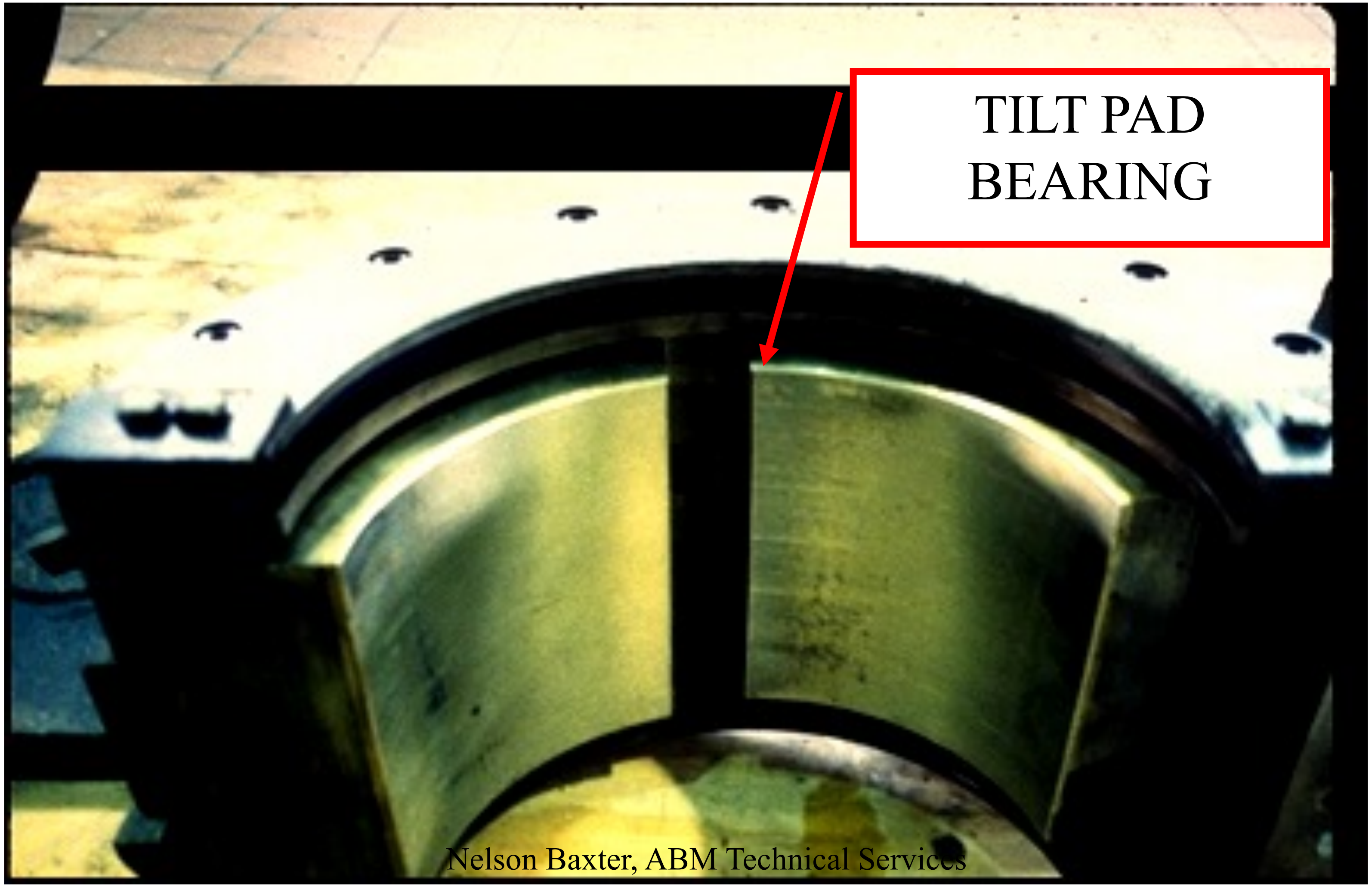
TEST RESULTS
MILS MOVEMENT VERSUS VACUUM

DISCUSSION

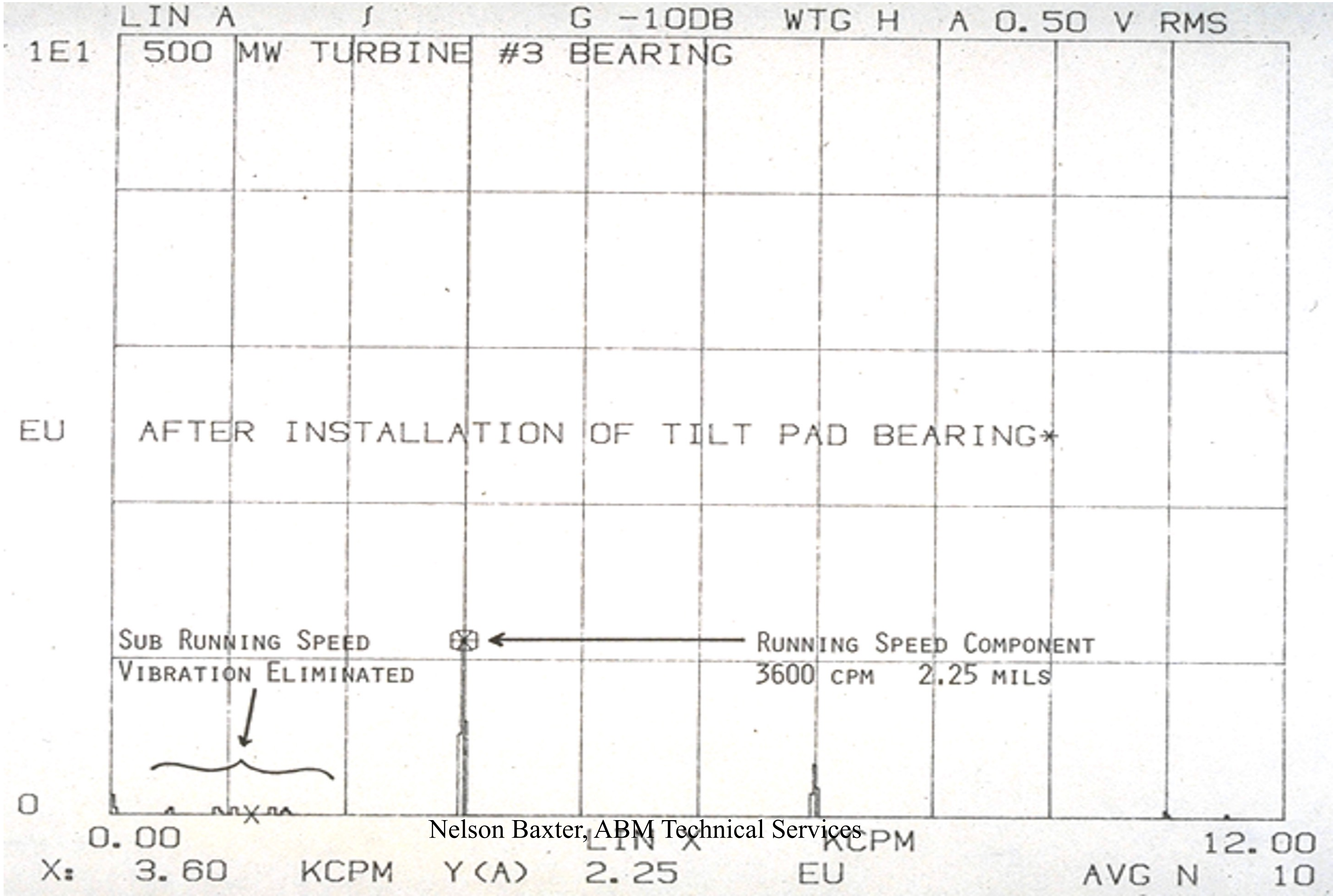
VACUUM DRAW DOWN COMBINED WITH THERMAL DIFFERENTIAL GROWTH UNLOADED BEARING 2 CAUSING IT TO GO UNSTABLE. WHEN BOTH WERE COLD, BEARING WOULD BE STABLE. WHEN BOTH WERE HOT, BEARING WOULD BE STABLE. THE PROBLEM OCCURRED AFTER A TRIP. THE THINNER LP SECTION WOULD COOL DOWN QUICKER THAN THE THICK HP SECTION. THIS DIFFERENTIAL ADDED TO THE VACUUM DRAW DOWN UNLOADED BEARING CAUSING OIL WHIRL THAT PREGRESSED INTO OIL WHIP.

BEARING METAL TEMPERATURE READINGS CONFIRMED THIS FINDING.

SOLUTION



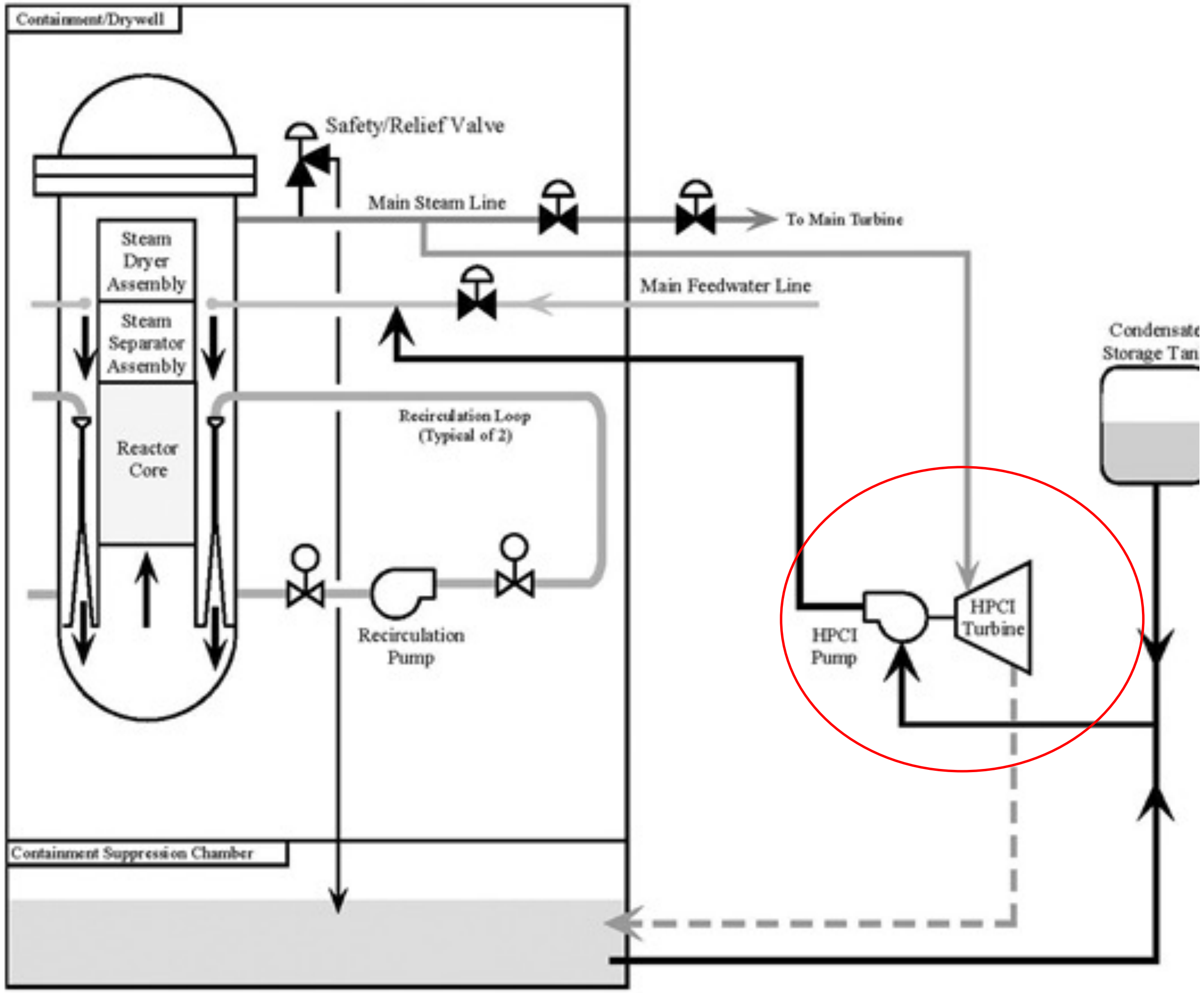
FINAL RESULTS



This problem was high on the difficulty list because:
1: There were two causes. Vacuum draw down and uneven thermal cooling.
2: Getting measurements was very difficult and required building a special system to make the measurements.
3: Oil whirl is fairly common, but oil whip which resulted from the fact <u>the rotor operated above twice the first critical speed</u> is not too common.
4: The utility had lost a lot of revenue because of the losses in power production that resulted when this large unit could not be operated so this was a high profile case.
5: It was difficult to install the test equipment and I got wrote up by union personnel during the installation thus adding to the already difficult nature of the work being done.
All these factors combined to make this a difficult troubleshooting case.

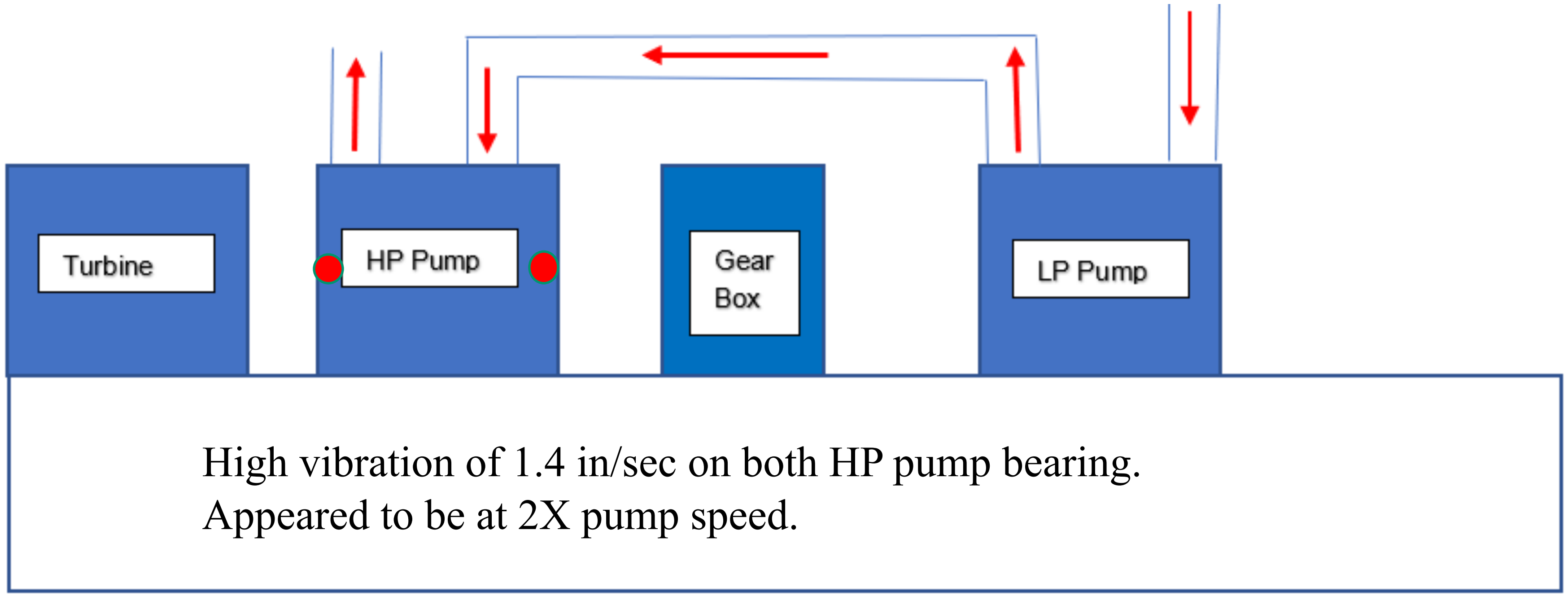
Note: The conversion to a tilt pad bearing completely solved the problem.

1^{st} MOST DIFFICULT CASE HIGH PRESSURE CORE INJECTION PUMP AT BWR NUCLEAR POWER PLANT



Containment/Drywell
Safety/Relief Valve
Main Steam Line
To Main Turbine
Steam Dryer Assembly
Main Feedwater Line
Steam Separator Assembly
Condensate Storage Tan
Recirculation Loop (Typical of 2)
Reactor Core
HPCI Turbine
HPCI Pump
Recirculation Pump
Containment Suppression Chamber

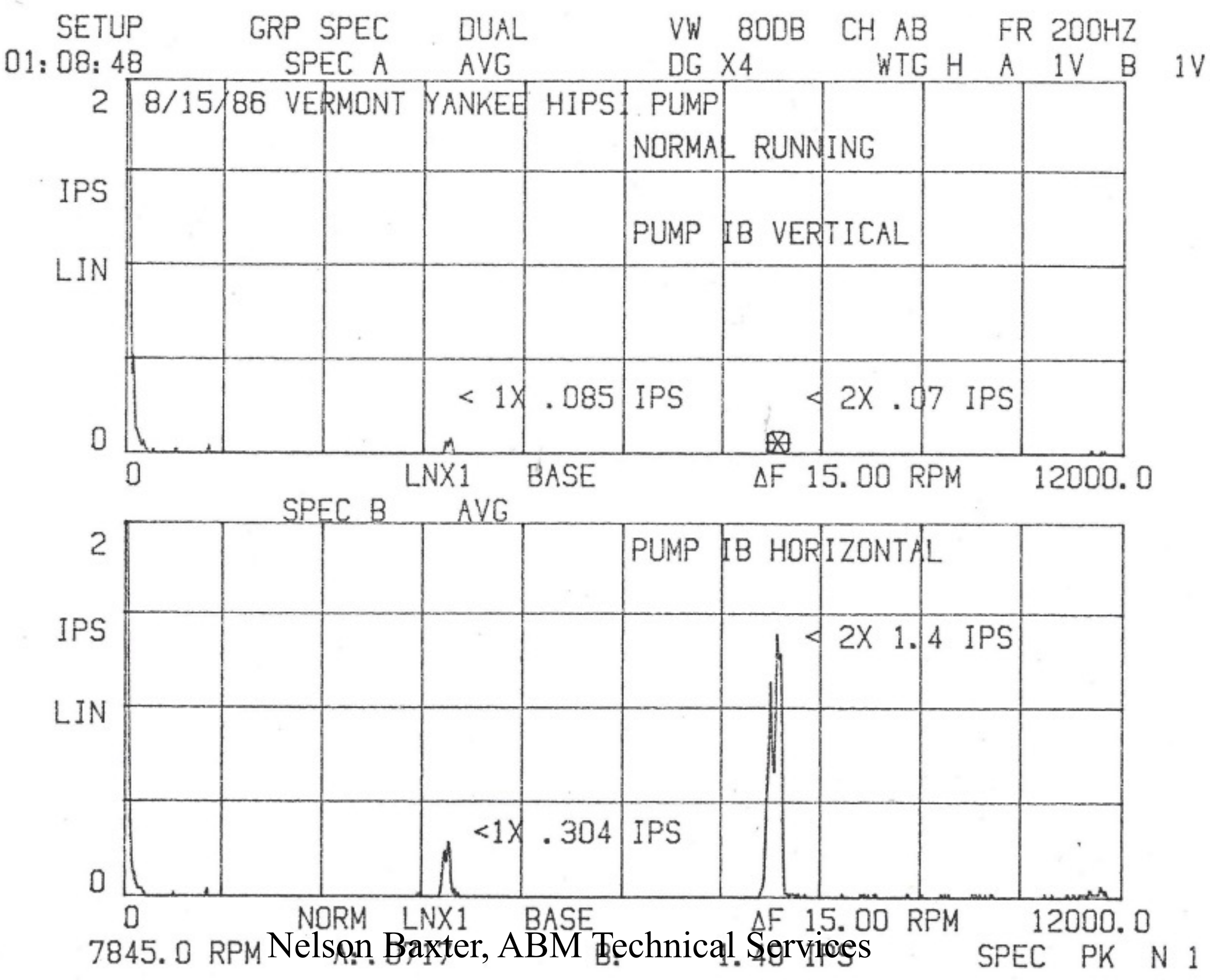
HIGH PRESSURE CORE INJECTION PUMP AT NUCLEAR PLANT



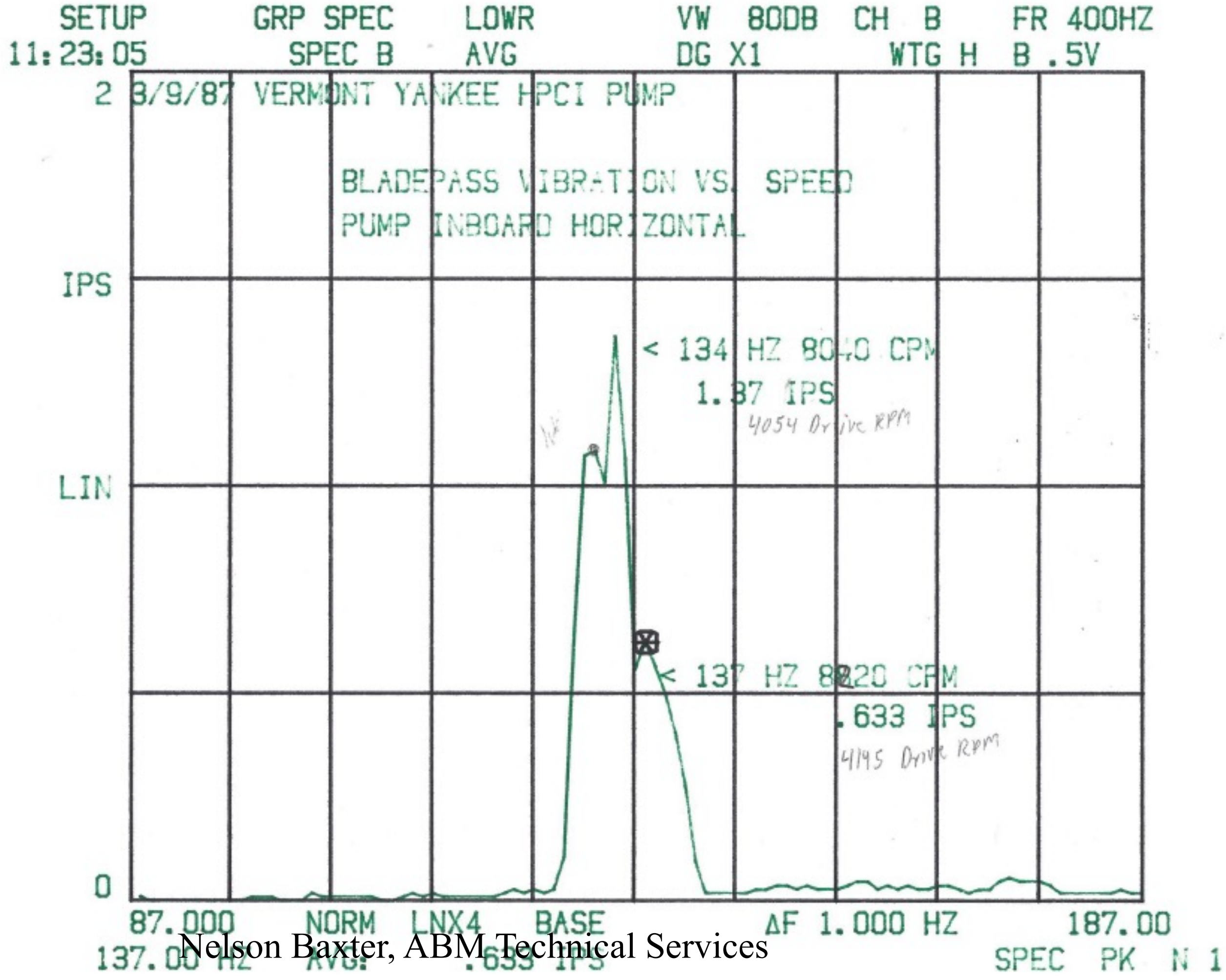
High vibration of 1.4 in/sec on both HP pump bearing.
Appeared to be at 2X pump speed.

A HIPCI pump supplies cooling water to a BWR style nuclear reactor in the event of an emergency. It pulls water from the containment suppression chamber and pumps it into the nuclear reactor to cool the fuel rods. Even though the nuclear fission process is no longer present, the fission by products still produce a significant amount of heat. This is therefore a very important pump that ensures the safe shutdown of a nuclear reactor. This pump is driven by a steam turbine. Previous testing in mils displacement had been the norm for this pump. When the plant started testing in velocity, this pump was found to be exhibiting very high vibration levels. The cause of the high amplitudes appeared to be at twice the speed of the pump. The amplitude varied significantly from test to test.

Spectrum From HP Pump 2X Vertical .07 in/sec 2X Horizontal 1.4 in/sec

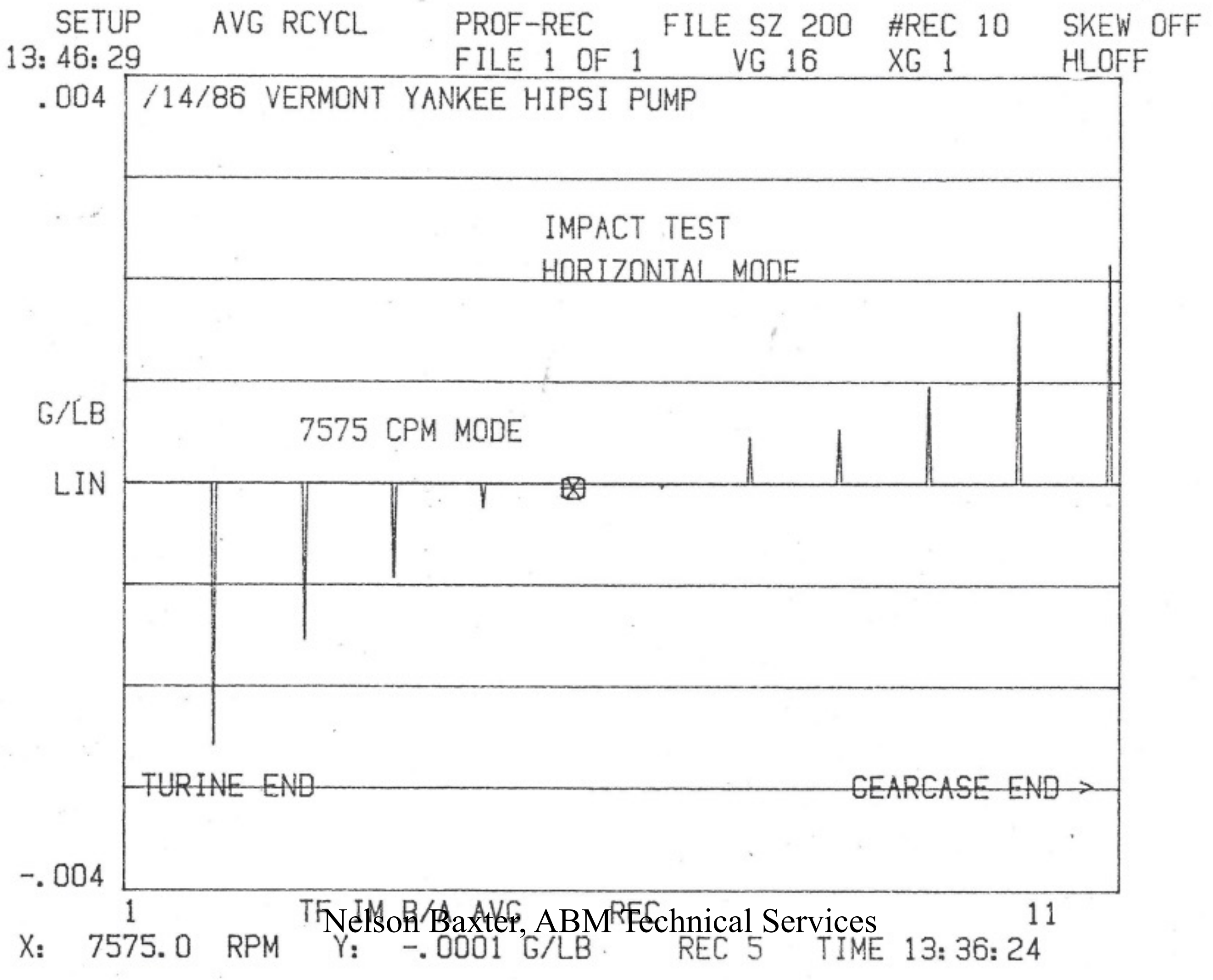


Speed 4145 RPM- Apparent 2X .63 in/sec
Speed 4020 RPM-Apparent 2X 1.4 in/sec
Vibration was highly speed dependent



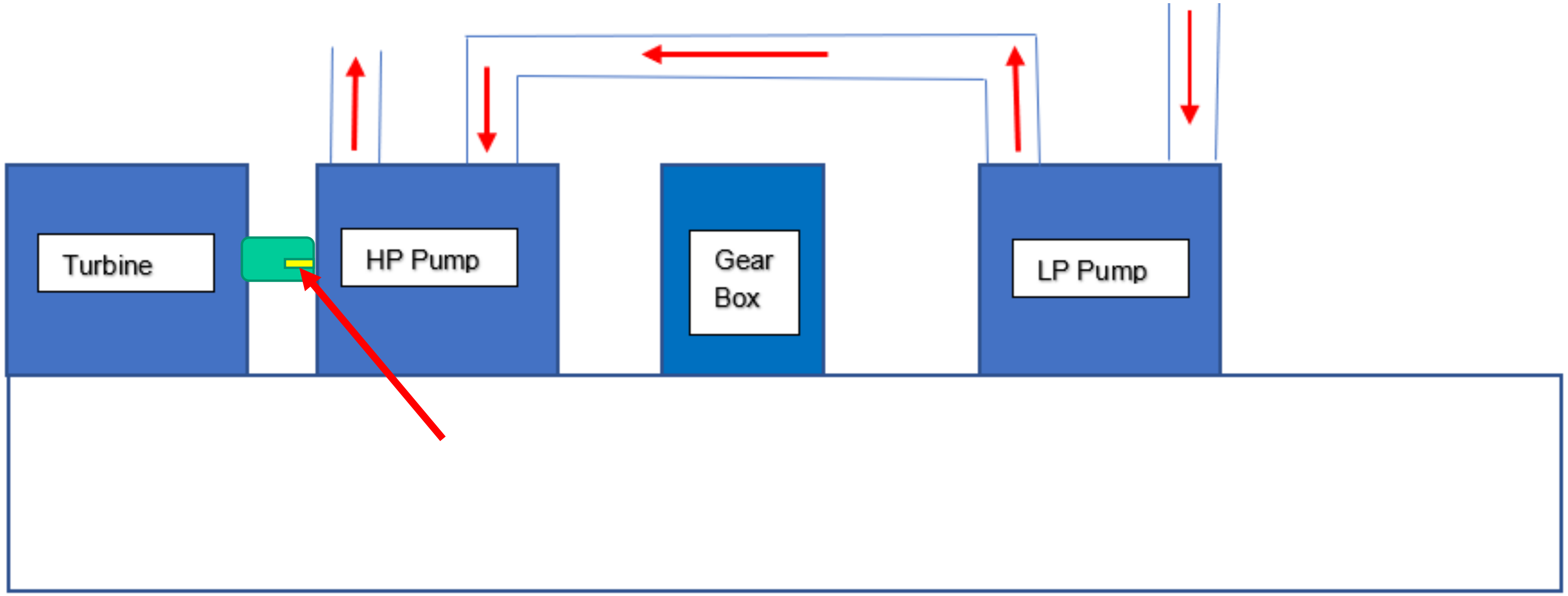
The very large horizontal to vertical amplitude ratio plus the sensitivity to speed changes pointed towards a natural frequency problem in the horizontal direction. An impact test was performed and there was a horizontal natural frequency in the 7500-8000 cpm range. Then next figure shows the mode pivoted around the center of the pump case. When operational phase measurements were made the inboard and outboard phase readings were very nearly 180 degrees apart. This confirmed the presence of the pivotal structural mode.

HORIZONTAL MODE SHAPE



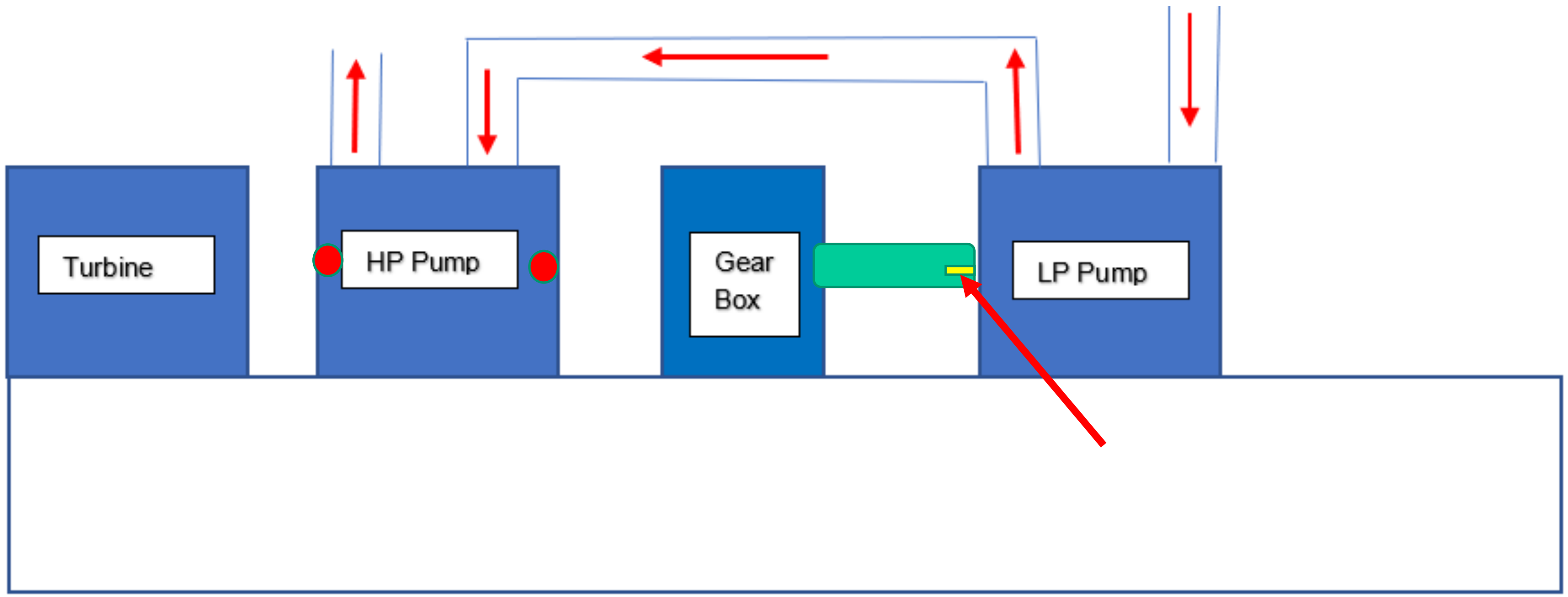
SYNCHRONOUS TIME AVERAGING TO GET THE PHASE READINGS TO CONFIRM THE OPERATIONAL MOTION PHASING WAS THE KEY TO GETTING TO THE ROOT CAUSE OF THE PROBLEM.

HIGH PRESSURE CORE INJECTION PUMP AT NUCLEAR PLANT



When the tach tape for the synchronous time averaging was placed on the High Pressure pump, the what had been supposed to be 2X vibration disappeared.

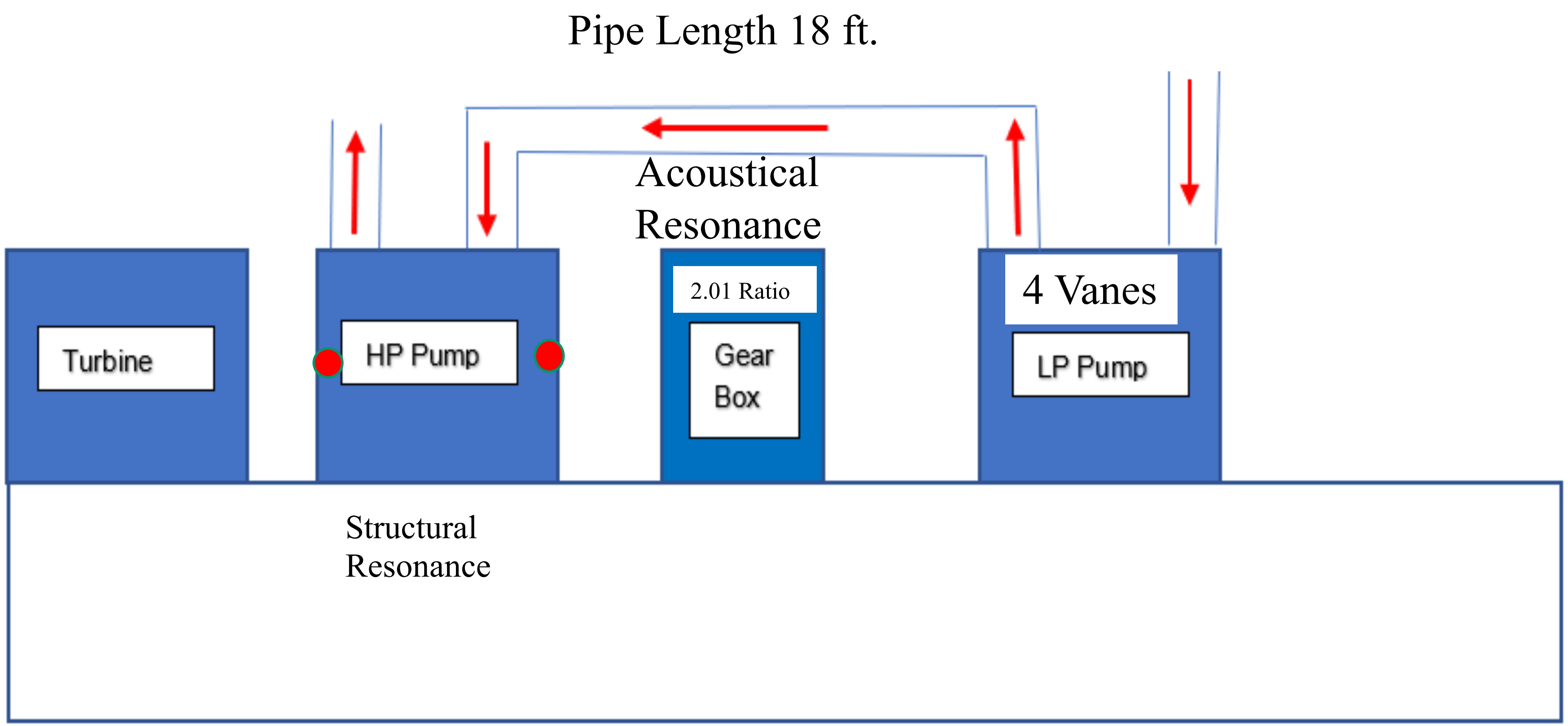
HIGH PRESSURE CORE INJECTION PUMP AT NUCLEAR PLANT



When the tach tape for the synchronous time averaging was placed on the Low Pressure pump, the what had been supposed to be 2X High Speed Pump vibration <u>remained</u> in the spectrum of the vibration on the High Pressure Pump.

The fact that the vibration on the High Pressure Pump was phase locked to the speed of the Low Pressure Pump was a complete revelation in tracking down the source of the vibration.

The causes of the high vibration were due to more than one problem.
1: The first was that the Low Pressure Pump had a 4 vane impeller.
And the gear box had a 2.01 reduction. <u>When the speed reduction was combined with the number of vanes that made the vane pass vibration appear to be at almost exactly 2 times the Turbine and HP pump speed.</u> Since the turbine speed had some speed variation, that smeared the spectrum so it was not possible by looking at the spectrum to see that the high vibration was not actually at 2X running speed.
2: As noted before, there was a horizontal structural natural frequency that fell within the 2X running speed range of the High Pressure Turbine.
3: What made it possible for the pressure pulses in the Low Pressure Pump to excite the High Pressure pump's structural natural frequency was an acoustical natural frequency that matched the blade frequency of the Low Pressure Pump.



½ wave length of vane pass frequency from 4 vane impeller in LP Pump matched length of pipe connecting it to HP Pump. The vanepass stimulus amplified by acoustical resonance then excited the horizontal structural natural frequency of the HP Pump.

Solution: The Low-Pressure Pump was changed to a 5 vane impeller. Odd numbered higher vane impellers tend to produce lower pressure pulsations. In this case the increased vane pass frequency moved the excitation away from both the acoustical natural frequency and the structural natural frequency of the High-Pressure Pump.

Things that made this a very difficult case.

1) It was a nuclear power plant. There is always a lot of pressure when working at one of these facilities. Numerous individuals were packed in the room observing the testing.

2) The room was seven stories under the ground and the room was very hot.

3) You only got one run in a month when they tested the pump.

4) When alignment measurements were made, the turbine to the pump alignment was found to be out .058” so it led to a false conclusion on the first trip that alignment was the source of the problem. The alignment issue was resolved and no improvement was observed, so another trip across the country was required.

5) The 2.01 reductio ratio of the gearbox combined with the 4 pump vanes was very close to the 2x running speed of the High-pressure pump.

6) The turbine varied in speed. That combined with item 5 made it impossible to see the source of the problem buy looking at the spectra.

7) There was a structural resonance problem and an acoustical problem.

The key to identifying the problem was the synchronous time averaging test showed the vibration was phase locked to the Low-Pressure pump instead of the High-Pressure Pump.

The End