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Industrial Case Histories

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Abstract: This paper is a series of case histories encountered over twenty – two years of performing vibration analysis. While each case history is not necessarily outstanding in its own right, they do show the type of problems one sees in an industrial environment. The paper will cover cases that are fairly routine to in-depth problems that required rotor dynamic modeling, structural modeling or both. Equipment from paper machines to turbines and pumps (vertical and horizontal) are presented. Both sleeve bearing and rolling element bearings machines are discussed.

This paper is an attempt at providing a resource of information with examples for the inexperienced analyst. The goal of the paper is to help point the analyst in the right direction for the analysis or to jog their memory to look further into the problem. Each history will include a brief discussion on what equipment was required for the analysis. In today's proactive maintenance society, it is thought the data collector is the all encompassing equipment for analysis, in many of the cases, mention of equipment required for each analysis will helpfully dispel this theory.

Key Words: Imbalance; Blade Pass; Whirl; Whip; Structural Resonance; Variable Frequency Drives; Bearing Defect Frequency; Gearmesh

Case #1 – Balancing of 9.5 Mw Turbine after a Catastrophic Failure

Problem: This unit was repaired after suffering catastrophic shut down during a hurricane. The turbine is owned by a government utility on a small island nation in the West Indies. The utility had utilized diesel generators until a far eastern country built their first steam plant. The utility ran into problems during Hurricane Hugo. Operations personnel not being experience with steam turbine failed to maintain the DC battery system. When the storm struck the island; the electric distribution system was destroyed causing the turbines to trip. The turbine generator train (Figure #1) went from operating speed to a dead stop in 45 seconds without lubricating oil. The turbine journals and bearings were destroyed. The bearings in the gearbox and generator were also completely wiped; however, there was no damage to the shaft journals.

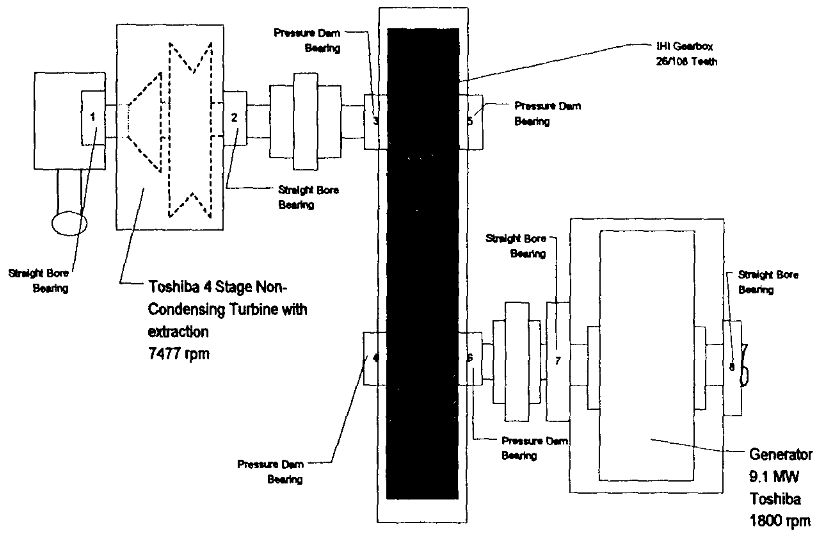


Figure #1

The unit was pre-warmed and run at slow speed. Once the slow rolling was completed and the oil system checked out, the turbine was to be brought up to speed. During this initial run the unit tripped on high vibration on the number one turbine bearing. The unit tripped below the first critical. The first critical is at 3400 rpm. The turbine supervisory system uses units of mm/100. All analysis and balancing data was collected utilizing casing mounted (magnet) velocity sensors. The turbine rotor sat in wood cradle for eighteen months after the journal repairs.

Symptoms: Vibration data indicated all the vibration was at running speed (1X). Since the unit was tripping at the first critical (first mode) and the rotor had sat for eighteen months in the cradles, there was a concern of a bow (first mode) in the rotor. The number one (turbine inlet) bearing balance ring was external to the turbine shell. This coupled with the turbine tripping on bearing #1 vibration it was decided to balance the unit with a single plane shot in the number one balance ring.

Test Data and Observations: Balance data was collected with a tracking filter and the analysis was performed with a Hewlett Packard Dual Channel FFT Analyzer. Velocity sensors were used throughout the balancing and analysis. Because no previous balance data was available; it was decided to perform a measured effect balance. This balance method was both ineffective and inclusive (figure #2).

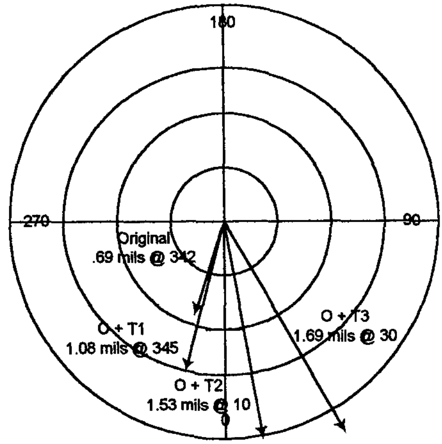


Figure #2

Since the effect was fairly decent, it was decided to again try a measured effect balance in conjunction with a four run no phase balance method. This proved to also be fruitless. Looking over the turbine schematic (figure #3) it was determined the #1 balance plane was too far from the center of the rotor (first mode) balancing on the #1 bearing balance ring.

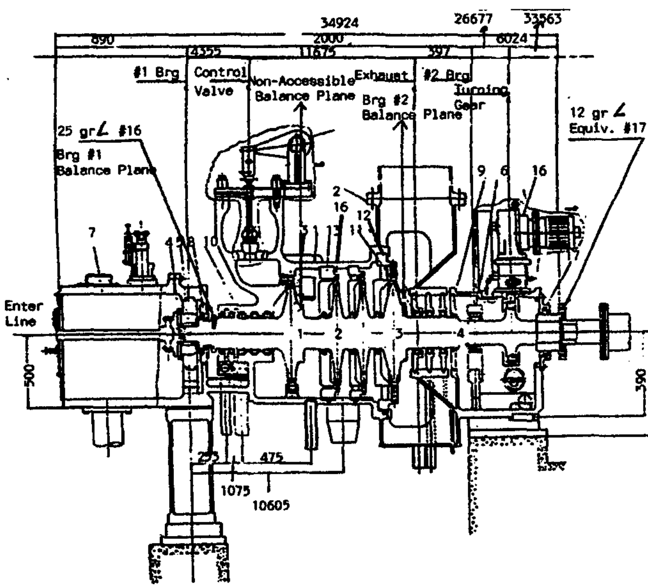


Figure #3

Corrective Action: Since the second balance plane (exhaust end) was closer to the mid span and we were trying to balance a first mode it was decided to place weights in this ring. Also, since this ring was closer to bearing #2 and the highest vibration was on bearing #1 a two plane balance program was entered. The unit was successfully balanced below the first critical; however, the unit again tripped at the second critical. A second two plane balance program was started below the second critical. During the balancing at the second critical two other problems developed. The unit suffered a rub and a bow was detected in the coastdown data (figures #4 & #5). The rub was not initially detected and several balance runs were lost to the rubbing.

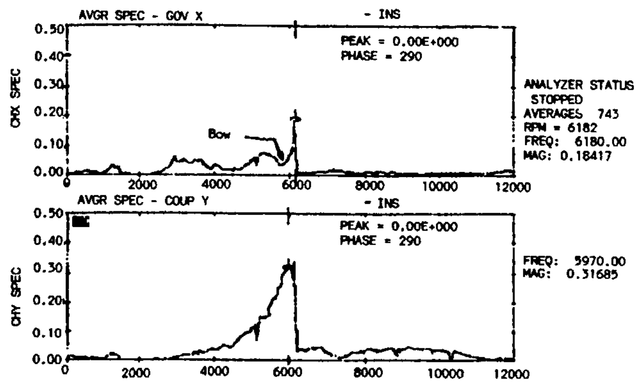


Figure #4

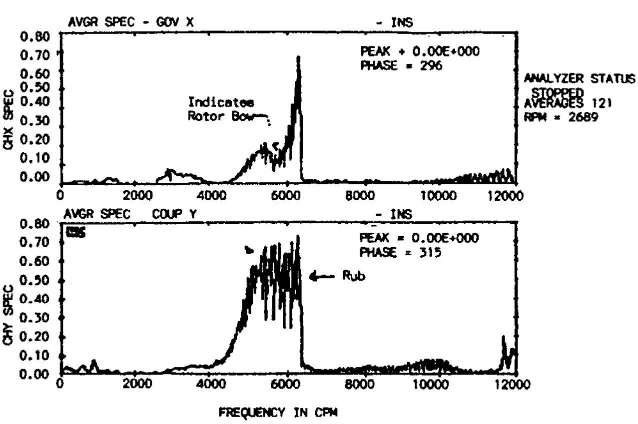


Figure #5

Results: Once the rub issue was eliminated balancing progressed without problem. The unit did have to be balanced at running speed with a two plane procedure even with balancing at the first and second critical. The ineffective results with the single plane balance on the #1 bearing really did not add much to the balance time.

The final data is listed in Table #1:

Bearing #	Control Room Data mm/100	Turbine Cap Data Mils (pk-pk)
1	1.6	.63
2	1.6	.63
3	.9	.35
4	.1	.04
5	.2	.08
6	.7	.28

Table #l

Conclusions: The balancing was hampered by several problems. The biggest problem; the bearings was designed with a clearance of three mils for every inch of journal diameter. This did not provide enough damping for the critical speed vibration. The clearances were designed over size because of not having spare parts on site and not wanting to wipe any bearings.

The rub problem was not detected early on because of data acquisition. The control system was not adequate in controlling speed to get good data during the balancing at the critical speeds. The speed would wander 200 to 400 rpm. Also, running the unit below rated speed caused the exhaust end to become hot. This in turn caused the bow in the rotor; therefore, the unit was run for very short times. While this would allow the collection of phase and amplitude data, it did not allow collection of analysis data. This caused those involved to miss the initial rubbing indications.

Case #2 – Induced Draft Fan ½ Order of Blade Pass

Problem: Vibration problems were suffered immediately upon running a new fan wheel. An Induced Draft Fan on the bag house of a cement plant was retrofitted to increase fan efficiency. This fan had a spare wheel and shaft. The new fan utilized the same shaft; however, the wheel design was changed. This fan was retrofitted with a new fan wheel using the existing shaft. The number of fan blades was increased from twelve (12) blades to eighteen (18) blades. The new shaft wheel system is approximately 250 pounds lighter then the old design. One other difference in this new fan was the addition of a turning vane to increase the efficiency of the fan.

Symptoms: Vibration problems were occurring when the fan is cooling down from 450 degrees F operating temperature to 220 degrees F operating temperature, during the plant start-up conditions. During the start-up process, the fan gas temperatures exceed 425 degrees due to some of the process duct work being by-passed. As these other parts of the process are brought in service the fan temperatures drop to a normal operating temperature of 220 degrees. Vibration amplitudes in the control room exceed 5.0 mils during the high temperature conditions.

Plant engineers balanced the fan from above 4.0 mils (pk-pk) to below 1.0 mil (pk-pk); however, within one day the amplitudes were back up to over 3.5 mils. Since this was essentially a new fan, the owners were concerned about the reliability of the new fan. One item noted in plant operation logs was how fast the vibration dropped when the fan was shut off.

Some initial data collected by the Cement Plant indicated there might be a natural frequency close to operating speed. The fan utilizes straight bore sleeve bearings and operates at 1196 rpm (19.94 Hz). The motor has been replaced several times and due to these replacements the motor and fan pedestals have been modified extensively.

During the initial field balancing by the fan manufacturer the turning van had to be cut back to get steady phase readings.

Test Data and Observations: The test plan was to look at the possibility of a natural frequency in the operating speed range of either the shaft or pedestal. Also; operating data would be collected from the casing using accelerometers and the shaft using prox probes. The accelerometers were powered by an external integrating power supply. A dual channel FFT analyzer was used for the impact tests. A three (3.0) pound force hammer was used for the impact tests. Start-up data, coastdown data and operational data was analyzed with either a dual channel data collector or a eight channel FFT analyzer.

Impact tests were performed first on both the shaft and fan wheel. Table #II contains the information from the impact tests. The concern was a natural frequency around operating speed and one close to blade pass (358.9 Hz). While the response at blade pass looked favorable for no problems, the frequencies at 23.0 Hz were a concern.

Once this testing was completed operational data was collected. Figure #6 contains a layout of the test points.

Point	Direction	Freq.	COH.	Critical Damping	"Q" Factor	Problem
Shaft	H	23.0	.970	.097	3.4	**Yes**
	V	30.5	.983	.087	5.7	
Shroud	A	333.75	.983	.014	35.2	?
	A	385.0	.974	.009	54	?
Blade	H	340	.893	.007	74	

Table #II

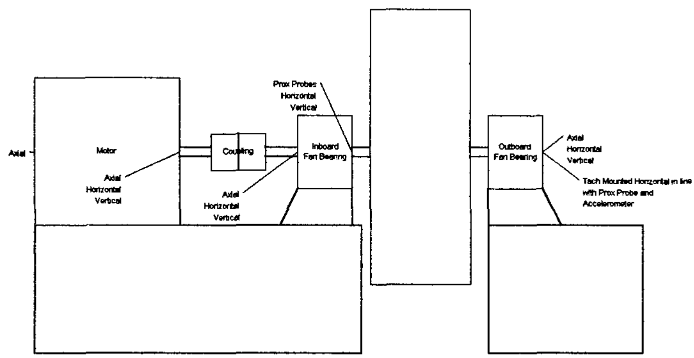


Figure #6

Phase and amplitude data showed thermal vectors as high as 6.8 mils (Figure #7). Coastdown data indicated the vibration dropped off almost immediately as the power was cut (Figure #8). This is a definite indication the fan is running very close to a natural frequency. In 52.5 cycles, the shaft vibration drops by over 76%

Analysis of the vibration data indicated there was an alignment issue on the fan bearings. Vibration data indicated the fan bearing to bearing alignment was off while the shaft to shaft alignment was correct. The axial vibration on the fan bearings was one-half the radial vibration; however, the axial vibration on the motor bearings was acceptable.

One other item that showed up in the vibration data collected during operation was the presence of a ½ X of blade pass (figure #9). When an orbit analysis was performed this 1/2x of blade pass also was present.

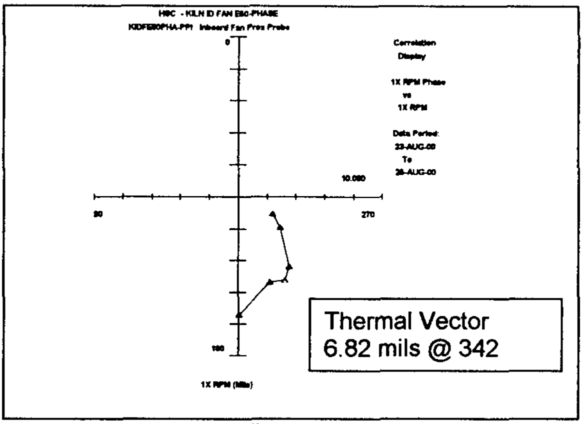


Figure #7

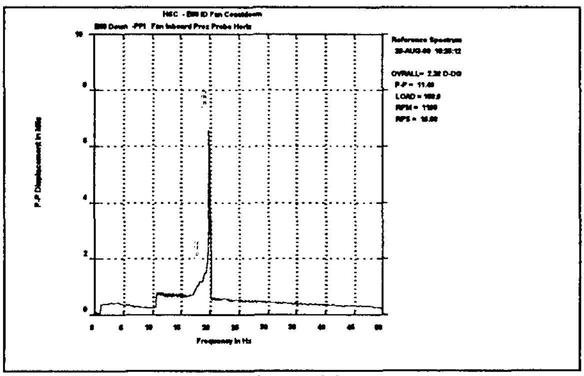


Figure #8

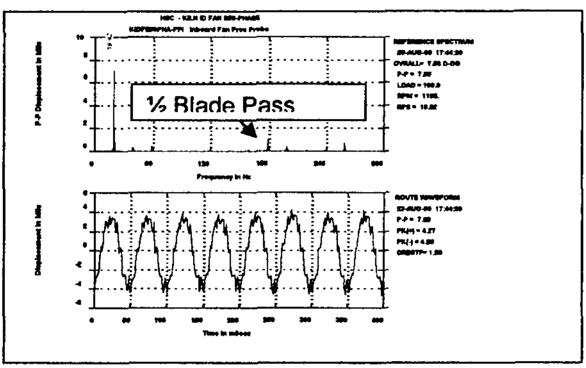


Figure #9

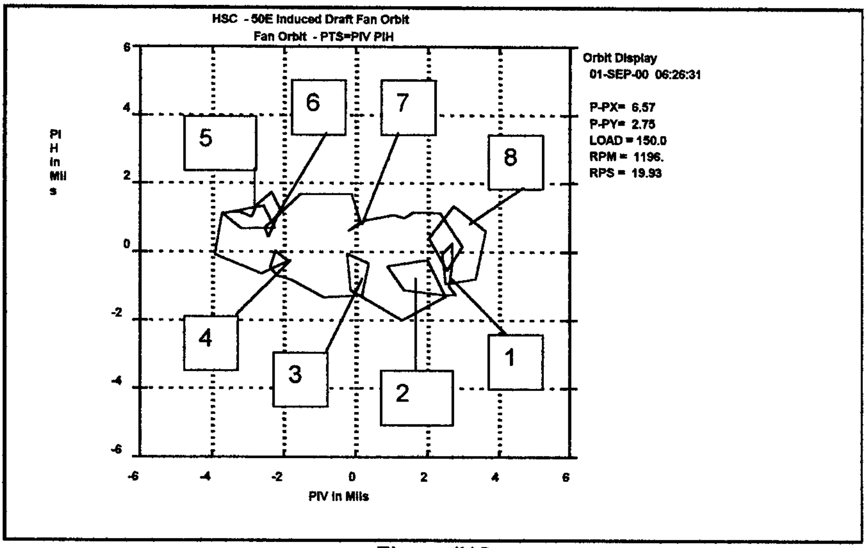


Figure #10

The internal loops indicate a forward whirl of the shaft. The frequency of the vibration is found from the following formula:

$$Frequency = \left[\frac{1}{n+1}\right]$$

n = number of internal loops

This means the period of the vibration is .111 (1/9). The frequency of the vibration is the whirl period divided by the shaft period or .111 divided by the shaft period (.0502). The period of the vibration is .0056 sec or 179.5 Hz. This frequency is one half blade pass.

Corrective Action: Due to the past history of balance problems with the turning vane, it was recommended to removed it. Also, recommended was to balance the fan with prox probes. The shaft vibration was 1.5X to 2X higher then the casing vibration. All previous balancing was only performed with seismic readings.

Improving the balance condition would lower the thermal effect due to the temperature extremes incurred from start-up to normal operating conditions.

It was also strongly recommended the fan bearing to bearing alignment be corrected as soon as possible.

Results: Vibration amplitudes dropped significantly when the balancing was completed. The alignment correction lowered the axial vibration. The plant eventually removed the turning vane. This removed the ½ blade pass vibration and cleaned up the orbit.

Conclusions: The fan was definitely running on a critical speed. Since all the operational data and transient data was recorded on a sixteen channel digital tape recorder, bode and nyquist plots were made of the coast downs. This data showed the fan was rolling off a critical speed.

Improving the balance condition and correcting the fan bearing to bearing alignment lowered the 1X forcing function reducing the vibration. This improved the vibration to the point, the thermal effects of the temperature excursions during start-up became a non issue. It should be noted that during the site data collection a production supervisor commented the thermal effects were present from the first day he started with the company and he was employed at this site for twenty – five years.

Also, this fan was always running close to a critical speed. The new shaft and fan wheel was 650 pounds lighter then the original rotor. This reduction in weight was not enough to change the shaft critical significantly; therefore, the fan has always operated close to a critical speed.

Finally the 1/2 Blade Pass was caused by a force on the fan wheel generated by the turning vane.

Case #3 – Cooling Tower Blade Pass Resonance

Problem: This unit had a history of running high vibration amplitudes in the 20 Hz frequency range. The operating speed of the "A" Cell Fan was lowered to 2.65 Hz from 3.33 Hz. This was done per recommendations from Cooling Tower Consultants who also changed the pitch of the fan blades to improve efficiency. The "A" Cell motor has an input speed of 1790 rpm (29.83 Hz) and the gearbox lowers this to 2.65 Hz. The blade pass frequency (6 Blades) is 15.9 Hz. All the vibration of the cooling tower is at the blade pass frequency. This has been the history of these fans.

Symptoms: Plant personnel had performed a shaker test on the "A" cell and found vibration response was high in the 15 - 17 Hz range (Figure #11). The dominant vibration frequency from the data plots (motor to gearbox) is blade pass. This is found in the axial, horizontal and vertical directions. Once the review of the past data was completed a test plan was developed.

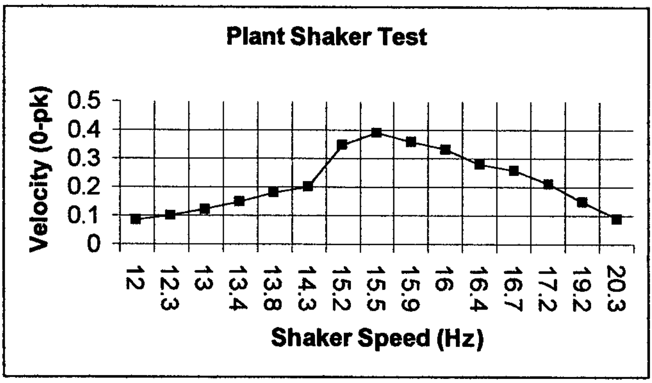


Figure #11

<u>Test Data and Observations:</u> A complete set of vibration data was collected with a dual channel data collector (Figure #12). Impact tests were performed with a dual channel FFT analyzer. A three (3.0) pound instrumented hammer was used for excitation. Along with the impact tests; another set of shaker tests were performed throughout the support structure of the motor, gearbox and fan system (Figure #13).

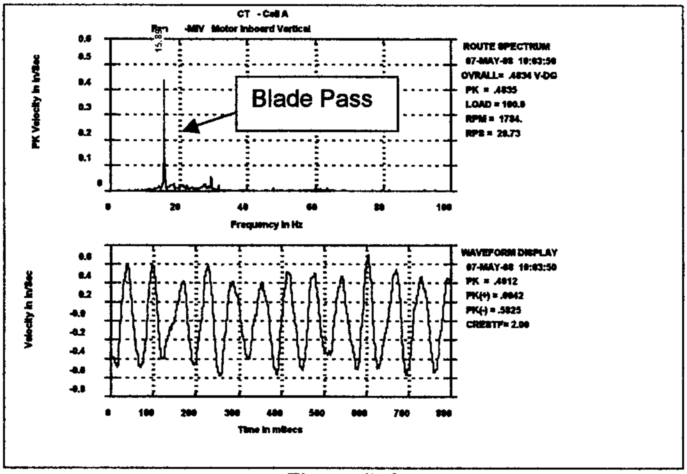


Figure #12

The Shaker tests indicated distinct natural frequencies around blade pass Figure #13). The amplification (Q Factor) from the tests were from 5.3 to 10.5 plus. Impact test (Figure # 14) and Shaker Test results can be found in Table III.

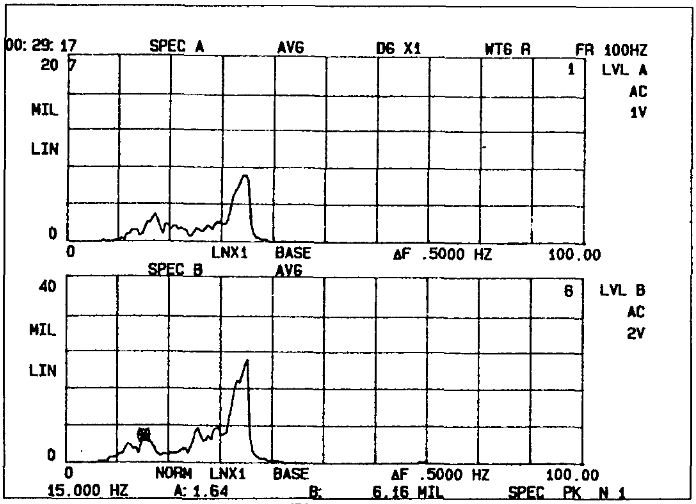


Figure #13

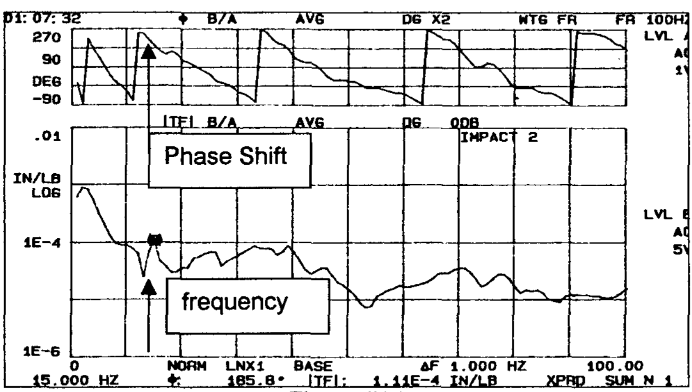


Figure #14

	South Beam (H)	South Beam (V)	North Beam (H)	North Beam (V)
Impact	12.0	15.0	12.0	15.0
	16.0	25.0	20.0	26.0
	19.0			
	26.0			
Shaker	15.0	30.0	12.5	15.0
	30.5		30.0	

Table III

<u>Corrective Action:</u> The data definitely indicates a natural frequency excited by the blade pass frequency of the fan. The vibration problem being experienced is caused by the six blades on the fan exciting the natural frequencies.

The natural frequency of a system is based on the mass and stiffness of the system. No one component is usually the fault; but, rather all the components of the system tied together cause the problem. Breaking the system up (detuning) by placing a neoprene material between the joints of connected parts may break the system up and eliminate the problem, however, this may or may not work. The final step would be a redesign of the system by replacing the six blade fan with a fan of five or seven blades.

Because this is a natural frequency problem the solution is to remove the forcing function which is the blade pass. Figure #15 shows an interference diagram for the different pieces of the system tested by impact hammer and shaker. The plot clearly shows that a seven blade fan operating at 2.65 Hz will have a blade pass frequency that will fall between natural frequencies and will have little or no excitation from the blades.

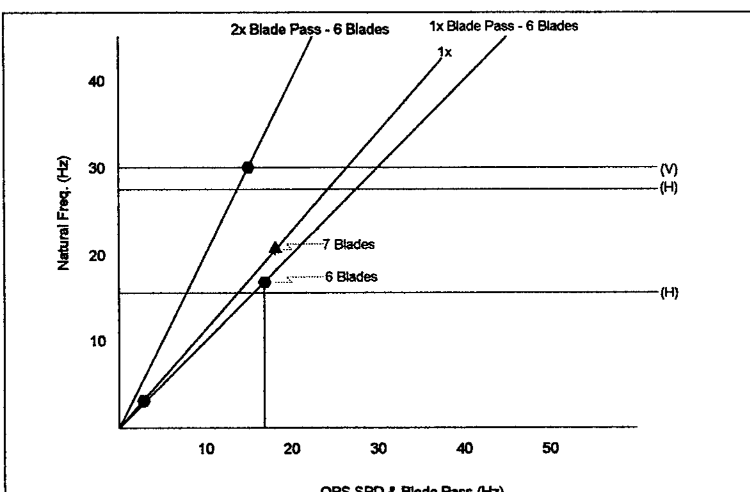


Figure #15

Since the fan has operated for so long (over eighteen months) with high vibration and no significant problems have been experienced the best option maybe to run as is and not change anything in the system. Based on the fact that no serious problems have occurred in the present condition the recommendation was made to run the cooling tower without any modification. If problems arise in the future; Febreeka should be placed under the gearbox and motor to detune the system, lowing the vibration. This detuning of the system should be done while a study on the effect of efficiency due to changing the fan from six blades to seven blades is performed. The change to seven blades is the long term fix.

Results: The unit was run for several months until the vibration amplitudes could no longer be tolerated. The system was detuned by installing Febreeka between the gearbox and mounting frame. This detuning provided enough damping so the fan could be run while a study on the effect of efficiency due to changing the fan from six blades to seven blades was performed. This study was completed and the fan was change to seven blades and now operates without problems.

Conclusions: The changing of the system operating speed; having been lowered to 2.65 Hz from 3.33 Hz, was the cause of the problem. This was done per recommendations from Cooling Tower Consultants who also changed the pitch of the fan blades to improve efficiency. However; no one looked at the possibility of excitation from the blade pass. Essentially the plant took a good running system and created their own problem. If the time had been spent performing resonance checks before making the change, this problem would have been identified and avoided. The ironic part of this case is plant engineering wanted to perform the resonance checks; however, they were deemed unnecessary by the corporate engineers.

Case #4 – Shaft Critical Primary Air Fan

Problem: This plant has had a history of recurrent vibration problems on their primary air fans. Looseness problems show up several times per year. The fan is an overhung design (figure #16). The replacement of the straight bore babbitt bearings is followed by a balancing of the fan using seismic readings. The fan is easily balanced below 1.0 mils (pk-pk). However; the vibration is always back into alarm with in a few weeks.

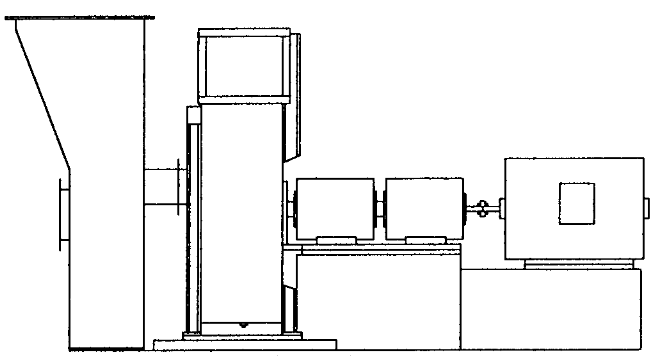


Figure #16

Symptoms: The symptoms remain consistent. The spectrum shows multiples of 1X and the timewave shows clipping (Figure #17).

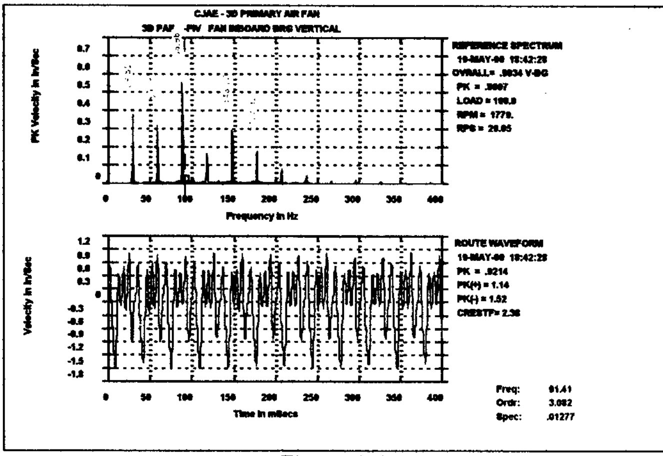


Figure #17

Test Data and Observations: This unit is on a monthly monitoring schedule; so it's easy to see when problems are developing. Once the looseness is apparent, the adjustment nut on top of the bearing has become loose. This nut is tightened down until the looseness is eliminated. This unit went through several cycles of the bearing nut becoming loose and having to be readjusted. The day after the last adjustment the bearing (outboard) closest to the fan failed. Failures of this nature have also occurred in the past. The failure required the bearing liner and bearing housing to be replaced. Data collected immediately after the fan was put back in service indicated the fan needed to be balanced (Figure #18).

The fan was balanced with seismic sensors. This has been the balancing procedure since the fan was put in operation in the early 1980's. The fan was balanced in one shot using past sensitivities and lag angles. Vibration levels were below .80 mils (pk-pk) at operating speed (Figure #19).

While the casing reading were more then acceptable, the cooling lines for the bearing were visibly moving. Vibration readings were taken on the cooling lines. Amplitudes were as high as 1.0 in/sec (0-pk). All the vibration was at the running speed of the fan. The first thought was the cooling lines could be resonant. The concern of the vibration analysts was the shaft vibration. Could the shaft vibration still be high even though the seismic readings were acceptable.

Prox probes were externally installed to read the shaft vibration between the bearings. Shaft vibration amplitudes (Figure #20) were still 15.0 mils (pk-pk) plus.

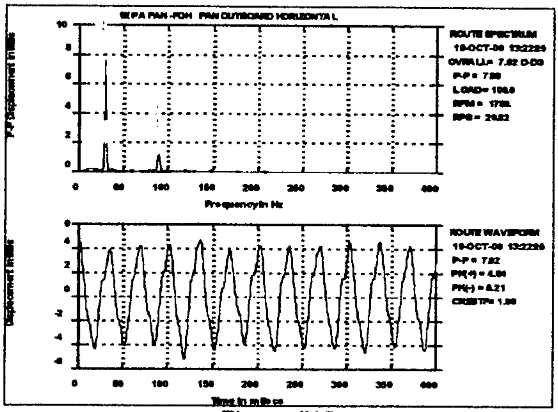


Figure #18

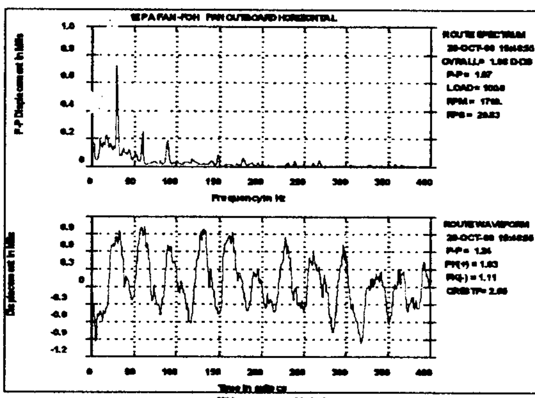


Figure #19

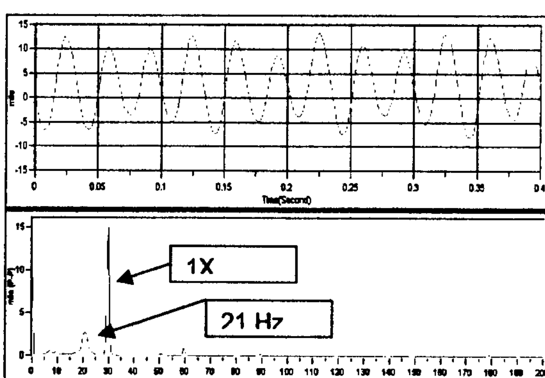


Figure #20

Also, the data showed a frequency at 21.0 Hz (1260 cpm). It was decided to collect a bode plot on a coast down (Figure #21). The bode plot shows a critical speed just below 1200 rpm. It should also be noted it appears the fan rolls off a critical as soon as the power is cut.

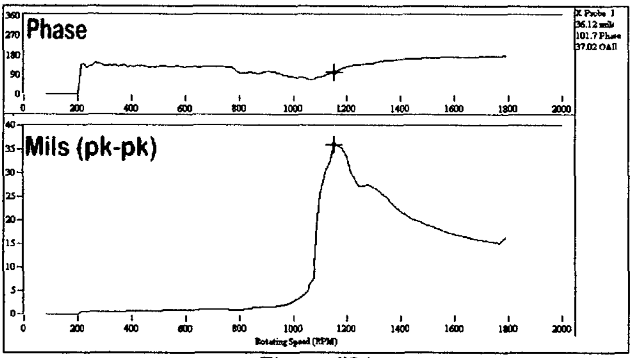


Figure #21

Corrective Action: Since this plant has six fans that behave this way, all the fans will be balanced using prox probe readings. It was recommended to perform an in-depth analysis of the fan bearing - pedestal system. A modal analysis of the fan movement in the axial, horizontal and vertical directions will be performed. Impact tests of the shaft and bearing caps will be done for validation of a rotor dynamic computer model of the shaft and bearing system.

Results: The modeling and shaft rotor dynamic study are presently in progress. Initial indications are the fan is running on a critical speed and the vibration is also effected by the critical speed just below 1200 rpm. Modifications to the bearing may provide the needed relief to allow for long term reliable operation.

Conclusions: The long term problems experienced with these fans are the result of bearing design. Balancing with seismic data was fruitless because the bearing liner is not in contact with the bearing housing. The bearing liner rests on a vertical support that has no horizontal support even though the bearings are held in place by the torque nut at the top of the bearing cap. The overhung design of the fan puts no load on the inboard bearing. This is evident the inboard bearing rarely needs changing and seldom shows any wear pattern.

Case #5 – Coupling Lockup Generating Gearmesh

Problems: Random occurrences of vibration started occurring throughout the load range of a turbine driven boiler feed pump. Vibration problems were experienced on both the turbine drive and the pump. The turbine drive is a six stage 16,700 horsepower turbine and the feed pump is a six stage double suction pump. The normal operating speed range of the system is 4000 – 5200 rpm.

Symptoms: The vibration occurrences had been monitored for a couple of weeks before they became steady. Once they became steady, the vibration occurred throughout the load range. Since the vibration amplitudes were high on both the pump and turbine it was felt there may be a problem with the coupling. Past documented information showed the coupling was very good at isolating the vibration into the pump or the turbine not allowing any cross effect.

Test Data and Observations: The initial request was to find if the unit could be relied upon for load sales the next day. Vibration data collected on the turbine indicated a ratty vibration spectrum with most of the vibration below running speed. Figure #22 shows the outboard pump bearing with over 1.40 in/sec of vibration at 3390 cpm. The inboard pump bearing had high vibration (>.75 in/sec) at 2900 cpm. This data indicate two possible problems. First the 3390 cpm frequency was a problem caused by rubbing of the rotating and stationary balance components of the pump. This was determined from past documented data on boiler feed pump vibration problems. Second, the spectrum from the inboard end of the pump indicated a frequency that was too low to be a rubbing problem in the pump.

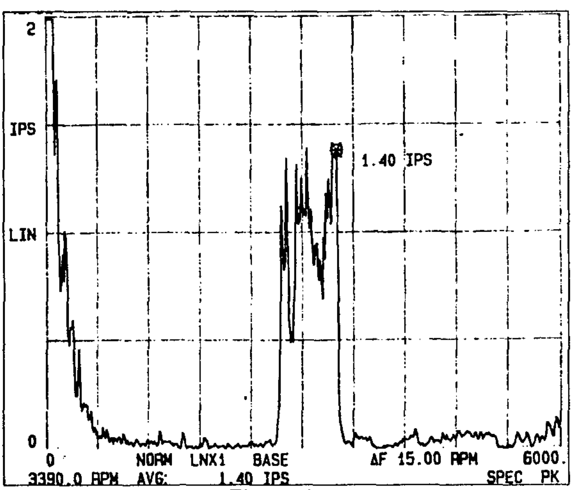


Figure #22

Next vibration data was analyzed from the turbine. Figure #23 is the outboard end of the turbine. The data is take off the shaft riders installed on the TSI system. The data shows ratty sub synchronous vibration. The peak amplitude at 2940 cpm has side bands at + and – 720 cpm. The 720 frequency is ten times the number of teeth on the coupling hubs. It was concluded from this data that the pump and coupling both had serious problems.

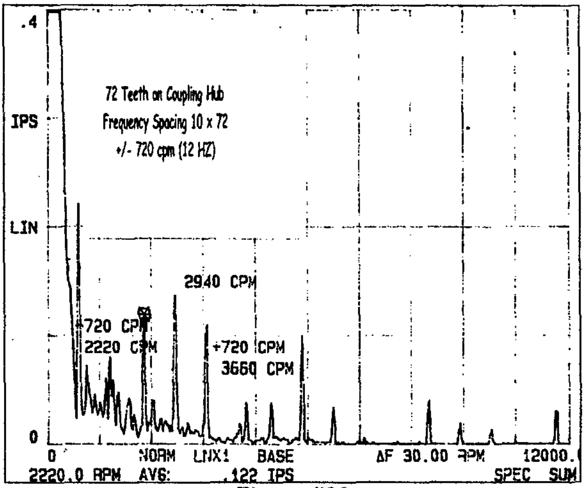


Figure #23

Corrective Action: It was recommended that the unit be shutdown and the coupling be inspected for wear. The total thrust of the pump was also to be measured. This would indicate if the pump and coupling were the problem or if further analysis would be required.

Results: The coupling inspection found severe wear and it had to be replaced. The thrust check found the pump had severe wear and would have to be rebuilt. This pump had been in operation over eleven years and was the last original pump in the plant. The coupling had been in service for at least seven years and may have been the original.

Conclusions: The documented data from past pump rubbing problems led to a fast analysis of the pump problem. The coupling problem was the result of a long operational life. If better maintenance had been done on the coupling, it may have been replaced at an earlier date. When the pump was put back in service it ran with vibration levels below 1.25 mils. The pump has operated without problems since this repair.

It is felt that the rubbing in the pump was the result of the coupling becoming locked in one position pulling the pump into the stationary balance components. If the coupling had not been worn, it may have allowed the pump to run another year; however, the pump repair showed we would have had to perform an overhaul within the next twelve months.

Case #6 – Oil Whip 150 MW Turbine Generator

Problem: This unit returned to service in November 1999 following a bearing inspection and realignment of the turbine generator set. The unit was subsequently balanced with all bearings below 2.7 mils (pk-pk). Vibration was very stable over a 48 hour period of operation; however, vibration started a steady climb while holding steady at 90 megawatts. Vibration amplitudes increased to over 7.0 mils (pk-pk) on both LP bearings. This symptom had been experienced in the past on this unit; however, no data was collected during these excursions.

This unit is a General Electric (Model D5R) 150 Megawatt Turbine Generator operates at 3600 rpm. The HP/IP turbine is opposed flow with a single LP rotor. A gearbox drives the exciter at 1200 rpm. This unit is instrumented with shaft riders for monitoring of bearing vibration. Velocity sensors are on the tops of the shaft riders. The subject unit has a history of vibration problems that center around running speed. Problems had been encountered in 1996 balancing this unit.

Symptoms: It was initially thought this unit might have developed a rub. In the past this type of problem had occurred. Temporary vibration instrumentation had been left on site to collect final data before clearing the unit for full operation. Before any testing was performed; trend data from the control room instrumentation was reviewed and analyzed.

Control room trend data indicated vibration amplitudes on the turbine LP section were very erratic. Amplitudes were instantaneously going form below 3.0 mils (pk-pk) to over 7.0 mils (pk-pk). If the problem were a rub the vibration amplitude should have been somewhat steady and not erratic. Since analysis equipment was still hooked up to all turbine bearings it was decided to perform a complete analysis. This problem occurred late on a Sunday afternoon and the utility was scheduled for a week long power sale Monday morning at 6:00 AM and the plant needed to know if they could meet the requirement.

Test Data and Observations: Plant personnel had dropped load when the vibration appeared. This did lower the amplitudes below the alarm settings; however, the amplitudes were not below the levels achieved with balancing. Test data was recorded with a sixteen channel digital tape recorder and analyzed with a dual channel data collector. Real time data was viewed on a Dual Channel FFT.

The data indicated the vibration was all subsynchronous (Figure #24). An oil whip problem was detected during the over speed testing of the unit (Figure #25); however, this was only present during the running of the over speed tests. During the over speed testing; the subsynchronous vibration was locked on 24.75 Hz

(1485 cpm) and appeared instantaneously. The onset of this whip during overspeed testing was at a shaft speed of 3510 rpm (58.5 Hz).

Normally; oil whirl or whip problems are caused by alignment problems or trying to operate the equipment with oil temperatures that are too cold.

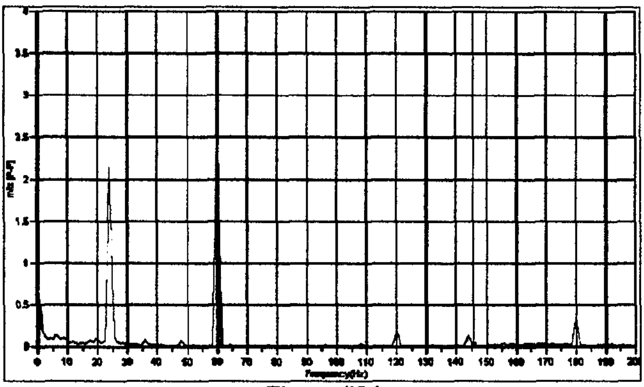


Figure #24

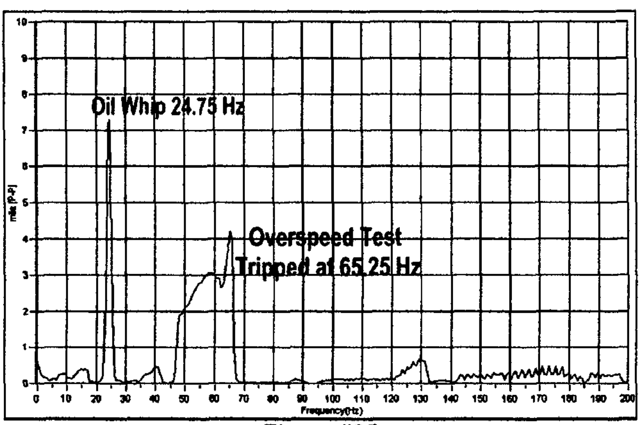


Figure #25

When this problem developed at 90 megawatts the oil temperature was at 105 degrees Fahrenheit. It was decided bring the unit back to 90 megawatts and raise the oil temperature from 105 degree F up to 110 degrees F. As the oil temperature was raised the subsynchronous vibration dropped. A three degree F temperature rise effected the subsynchronous vibration (Figure #26).

This unit was only on line for three days. During the outage the turbine was realigned. The outage lasted four weeks so the unit had not grown into alignment. The oil whip vibration amplitudes were highest on bearing #3.

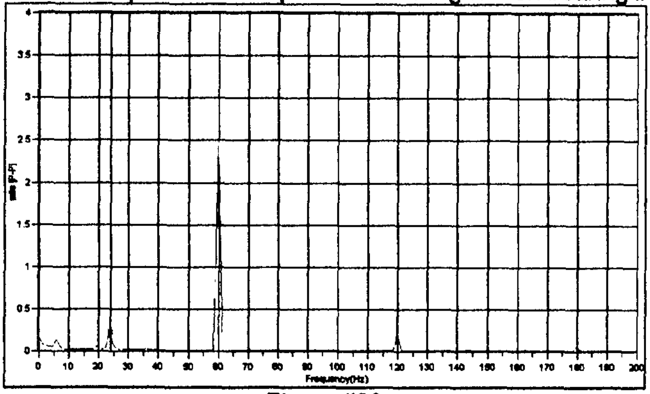


Figure #26

Corrective Action: Operations personnel at this plant had a history of running the oil temperature below 105 degrees F. Normal operating oil temperature main turbine generators is above 115 degrees F. Operating procedures were initialed to ensure the oil temperature was kept above 115.

Results: The unit has operated since this time without problems related to subsynchronous vibration. The operations personnel keep the oil temperature above 115 degrees F and will not roll the unit to speed with the oil temperature below 100 degrees F.

Conclusions: The oil whip problem was a combination of two factors. First the unit had not grown into alignment. This unit is aligned so that bearing #3 is loaded heavy during the start up. This is done because it is susceptible to whirl and whip conditions. The oil being cold and the unit not having grown into alignment allowed the whip to develop. Since this problem; questions have arisen about the alignment of this unit. Optical alignment data shows that bearing #3 may be unloaded. The alignment procedures used during the outage have also been brought into question.

Case #7 – Oil Whip 30 MW Turbine Generator – Alignment Generated

Problem: This unit is located in a power house at a paper mill. This unit is a General Electric Turbine Generator set. The turbine is a single rotor with extraction steam (Figure #27). Operating speed is 3600 rpm and the generator is rated at 30 megawatts. The unit had recently been overhauled. A spare rotor had been installed on the turbine. Within a few days of being put back in service the turbine bearings started to experience vibration excursions. The unit was not able to be run above 10 megawatts without vibration amplitudes above 7.0 mils (pk-pk).

During the rotor change out all bearings had been sent out for repair and the unit had been realigned. Realignment of the turbine took longer then planed. The generator had to be lowered to get the coupling alignment correct with the turbine.

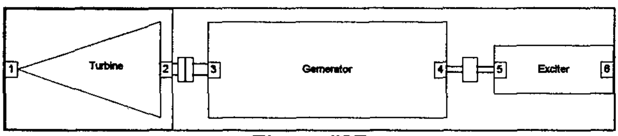


Figure #27

Symptoms: The vibration would cycle from low amplitudes to high amplitudes over a ninety minute cycle. This repeated consistently when the unit was operating above 20.0 megawatts. Below 20 megawatts this cycling would still happen; however, the pattern was not as repeatable. It was also noted that when the operators tried to bring the unit to full load when the turbine vibration cycle was at a low point the first generator bearing (bearing #3) would have a sudden increase in vibration that was erratic.

Test Data and Observations: Test data collected by the plant personnel indicated the vibration was at 1X (Figure #28).

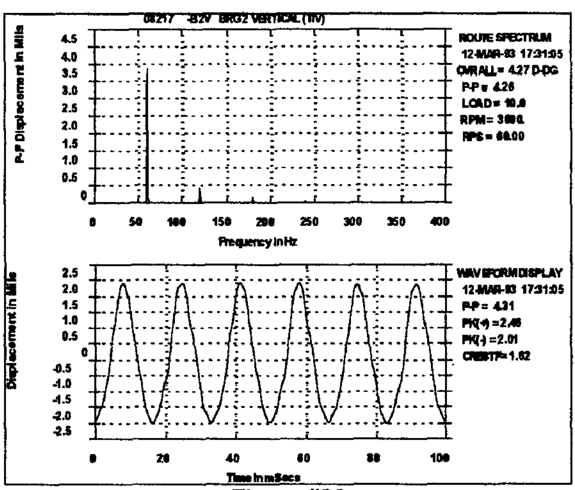
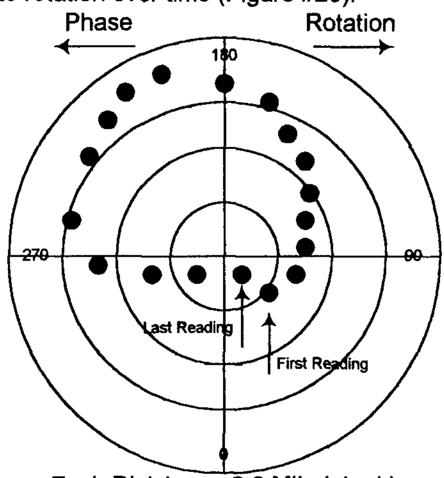


Figure #28

To get a better picture of the cycling of the vibration 1X amplitude and phase data was collected every five minutes for 90 minutes. This showed the vibration was moving opposite rotation over time (Figure #29).



Each Division = 2.0 Mils (pk-pk)

Figure #29

Phase movement with 1X vibration is an indication of a rub. When the unit was at a low cycle of vibration an attempt was made at running the unit at full load. Vibration spectra data was collected on bearing #3 to see what the cause of the erratic vibration was. When the unit reached 20 megawatts a subsynchronous vibration appeared (Figure #30).

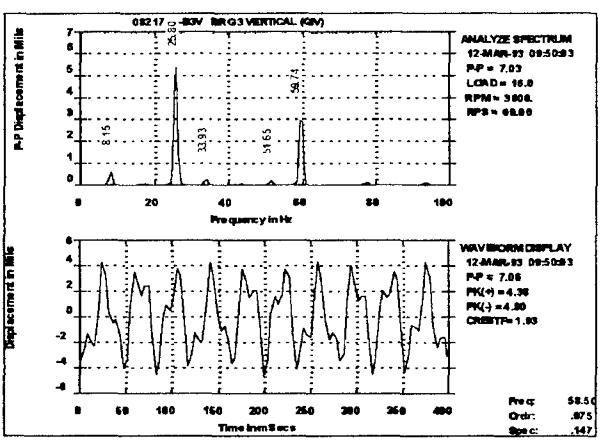


Figure #30

Once it was determined the turbine vibration was due to a rub and the generator problem was due to an oil whip, the cause of the problem had to be found. Looking over the operating parameters indicated the steam operating temperatures were all in the correct range. The oil temperature was also in the correct range.

The next thing to look at was the work done during the turbine outage. Since the unit was completely apart; realignment of the unit was a good place to investigate. If problems with the alignment developed, a bearing could become unloaded and develop a oil whip condition. Alignment could also cause a rub by closing up the clearances between a seal and shaft.

Corrective Action: It was recommended the alignment setting be verified and changed as necessary to relieve the situation. This needed to be done as soon as possible; because the plant generated its own power and could not run at full production without the turbine.

Results: It was found that during the calculation of alignment settings a dial indicator was read incorrectly. This resulted in the generator being set too low unloading the first generator bearing. In turn, bearing #2 was rubbing due to the seal clearances being closed up.

Conclusions: The problem could have been avoided with better quality control during the reassembly and realignment of the turbine generator set. Due to extenuating circumstances the repair work took longer then expected. In the rush to overcome the delays, mistakes were made during the alignment. If two people had been involved in the realignment calculations this problem may have been caught and problems avoided. The whip problems had damaged the first generator bearing to the point that it had to be replaced. Steam seals on the #2 turbine bearing also had to be replaced.

Case #8 – Oil Whip Bearing Starvation on Vertical Pump

Problem: Motors installed on flood control pumps suffer sudden increases in vibration and had an abnormal shaft orbit. This is a synchronous motor with a variable frequency drive (VFD). The design motor speed is 514 rpm (8.57 Hz). The motor and pump bearings are instrumented with dual prox probes (X & Y). . Also, dual seismic accelerometers are installed on each bearing. This motor is rated at 4865 horsepower and has tilting pad bearings in the top and bottom guide bearings. While this type bearing provides stability for whirl and whip conditions on horizontally mounted equipment; in vertically mounted equipment these bearings have problems controlling vibration due to the lower damping of the bearing. This is a six pad bearing (pads 3.0"W x 6.0'L) with zero preload.

The formula for determining excitation and rated speed is as follows:

$$Excitaion = \left(\frac{Line\ Frequency}{\textbf{Rated Speed}}\right) x\ \textbf{Shaft Speed} \qquad (1)$$

Representatives of the motor manufacturer felt the vibration problem was due to balance problems. Specific vibration specifications were to be met by both the pump and motor manufacturers before the owners would accept the equipment from construction.

<u>Symptoms</u>: Motor vibrations were above 5.0 mils (pk-pk) and as high as 10.0 mils plus. Vibration trips set on this motor pump train are set at 8mils (pk-pk) and have a ten (10) second trip delay. It had been noted on test runs the vibration suddenly increases when the excitation is at about 39 Hz. This equates to about 334 rpm shaft speed. The motor field service also identified an abnormal orbit with external loops.

<u>Test Data and Observations (1)</u>: All initial test data was collected with the motor uncoupled from the pump. Because of the sudden vibration trips when the excitation is at an excitation of 39 Hz to 40 Hz, data was recorded to determine the cause of the vibration. Data was collected from both the installed proximity probes and extra mounted seismic sensors. Trying to minimize the problem with tripping, vibration data was collected at 5 Hz to 10 Hz excitation intervals. Since the data was primarily 1X, Plant personnel and equipment vendors wanted the motor balanced (Figure #31).

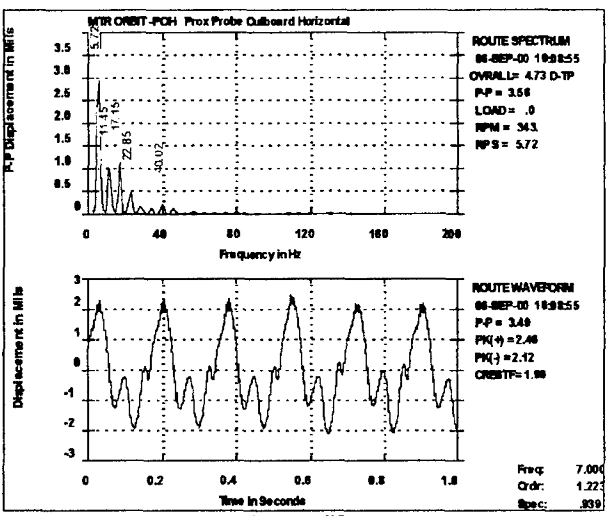


Figure #31

Balancing was able to somewhat lower the vibration in the uncoupled condition; however, when coupled up the vibration was still high at 40 Hz excitation and would trip the motor. There was also a thermal effect noted in the balancing process. During the step by step progression of collecting data at several points as the motor excitation was increased a poor mans bode plot was constructed (Figure #32). This data is from the outboard motor horizontal proximity probe.

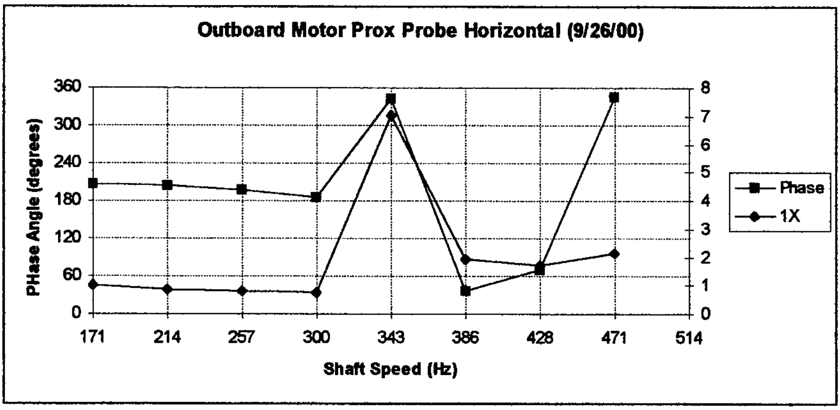


Figure #32

The increase in vibration at 343 rpm (39 Hz excitation) with the associated phase shift is showing the shaft is going through a critical speed.

<u>**Corrective Action (1):**</u> Recommendations were made to replace the tilting pad bearings with an elliptical or straight bore bearing. It was additionally stressed that a rotor dynamic study be performed to determine the best bearing for the situation. This recommendation was met with stiff resistance by the motor manufacturer. A second consultant was contracted to look over the analysis and balancing data along with the "poor man bode plot." This consultant agreed with the first conclusions and recommendations.

The motor manufacturer contacted the bearing supplier and asked for recommendations on how to make the installed bearing work. They recommended cutting off .75 inches from the length of the each tilting pas. This was to be removed from the trailing edge of the pad. The motor is running counterclockwise; therefore, the right side of the pad was removed (Figure #33).

Once this modification was performed the equipment owner wanted the motor rebalanced in the uncoupled condition. Immediately upon bringing the motor up to rated speed another problem developed that was not expected. A subsynchronous vibration appeared when the motor was running at 40 Hz excitation.

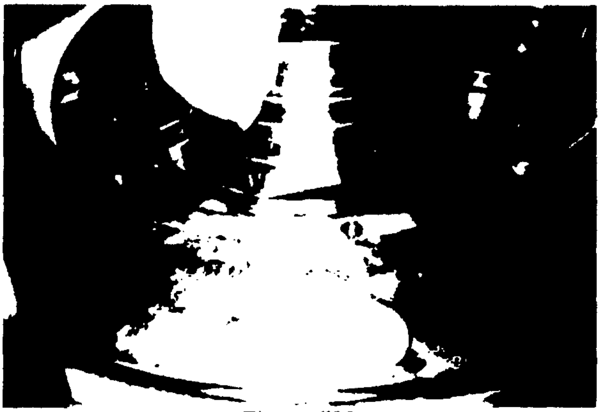


Figure #33

Test Data and Observations (2):

The subsynchronous vibration at 2.37 Hz is 47% of running speed (Figure #34). The subsynchronous vibration was only present on the outboard motor bearing. This vibration would only appear when the unit was at 40 Hz excitation and above. Below 40 Hz excitation this vibration was not present. It had been noted during the balance runs; a subsynchronous vibration was present; however, in amplitudes below .2 mils (pk-pk).

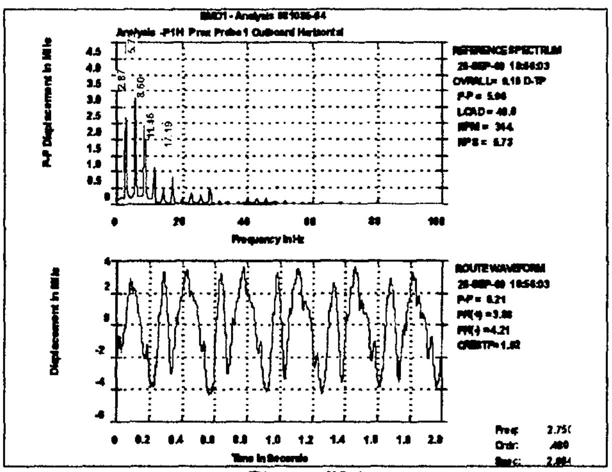


Figure #34

When moving extra seismic sensors at the top of the motor; it was noted the two oil site glasses on the top motor bearing had low levels of oil. The motor was shut down and oil was to be added. However; when the motor was shut down the oil levels were correct. Now the question became; was the oil level was correct when the unit was operating. Additional oil was added to the sump. However; when the motor was again run, the subsynchronous vibration was still present. The additional oil now had one site glass at the correct level and the second oil site glass with a low level. It was decided the oil sump oil baffle system was not distributing the oil evenly throughout the bearing.

Results: The oil baffle system was redesigned and the extra oil that was added was subsequently drained out. This corrected the situation. The bearing modification was made to all the motors at this installation and they have since operated with out problem.

Conclusions: The initial vibration problems were due to bearing design. Removing the trailing edge of the bearing loaded the bearing and added stability. The subsynchronous vibration was due to starving the bearing on one side. This allowed an oil wedge to develop causing the oil instability problem. If a rotor dynamic study had been performed during the motor design stage this problem may have been avoided. It is felt that even with the rotor dynamic study the oil starvation problem due to the sump baffles would have still been a problem.

Case #9 – Coupling Induced Externally Generated Whip

Problem: Excessive vibration was first reported by Plant Operations on the feed pump and then following pump repairs, vibration was reported on the feed pump turbine drive. The pump is a DeLaval six stage, double suction pump. The pump is direct coupled (gear tooth with spacer) to the drive turbine. Normal speed range is between 4000 and 5000 rpm depending on feed water requirements of the boiler. The first indication of a problem came when Operations reported high pump vibrations while bringing the pump up for morning load.

After the pump was overhauled, abnormal vibration appeared on the high pressure bearing (outboard turbine bearing) of the pump turbine drive. This vibration started around 4000 rpm on the initial run up after the pump overhaul. The vibration, at first, was random; but, after several hours, it was constantly present. The turbine is a General Electric six-stage, 16700 horsepower turbine with a maximum speed of 5200 rpm.

Symptoms (1): Operations reported that at approximately 4200 rpm the feed pump started to vibrate severely . The higher in the load range the pump was run, the more violent the vibration amplitudes. On-line vibration monitors showed 10+ mils (pk-pk) in radial direction (shaft riders mounted 45 from the horizontal). A single axial thrust probe (non-contact pick-up) went out of limits when the severe vibration started. Normal vibration levels were the observed on the turbine drive during this pump vibration (1.5 – 2.8 mils).

Test Data and Observations (1): The first decision that needed to be made was would it be safe to run this pump to gather test data for analysis. Normally; when the on-line axial position monitor goes out of limits, internal metal to metal contact has occurred in the pump. This would entail the rotating element coming in contact with the balance ring assembly. Since this had previously been documented on another pump, it was determined to immediately take the pump off as soon as the back-up pump could be placed in service. A tape recorder was used to record any data that could be acquired in the limited time the pump was being taken off the line.

With the operating speed on the pump at 4350 rpm a subsynchronous vibration is present at 3225 rpm (Figure #35); the running speed vibration is non-existent. Only the sub-running speed vibration is present. The subsynchronous vibration at 74-78% of running speed indicates hard, continuous contact between the rotating and stationary parts of the pump.

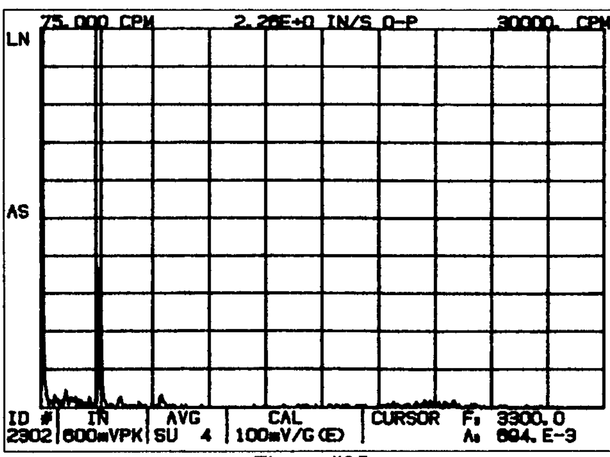


Figure #35

Corrective Action (1): Once Operations had shut the pump down, no emergency action was needed. Mechanical maintenance personnel were instructed to inspect for severe rubbing on the balance components. Mechanical maintenance mechanics found severe rubbing wear and abnormally large clearances on pump internals. It was decided to completely overhaul the pump at this time. A complete new pump barrel, balance components, and bearings were installed.

Symptoms (2): After the pump overhaul was completed, the pump and the turbine were warmed up through normal procedures for the initial roll-off to 1200 rpm. When this was completed, the unit was brought to 4000 rpm, abnormal vibration started to appear on the high pressure bearing (outboard inlet bearing) of the turbine. This vibration kept increasing as pump flow increased. Since this unit is direct coupled, pump flow is directly related to rpm of the turbine. During the first several hours, this vibration would come and go. Finally; after four hours, the vibration was constantly present. At 4200 rpm, vibration was 4.0 mils and steadily increased as speed was increased.

Test Data and Observations (2): Since the pump was just overhauled, our first thought was the turbine vibration was the result of some procedure done during the overhaul; even though the vibration was in the turbine and not the pump. Because this is a capability related piece of equipment, it was recommended that we go into a two part analysis program. First, a decision was needed to determine, if the pump could be used for operational needs; second, find the cause and solution to the problem. Before any decisions were made, a full set of data was collected to determine if the unit could be run without severe damage. Spectrum data shows approximately 7.9 mils of subsynchronous vibrations present at 1920 rpm and 3840 rpm (Figure #36).

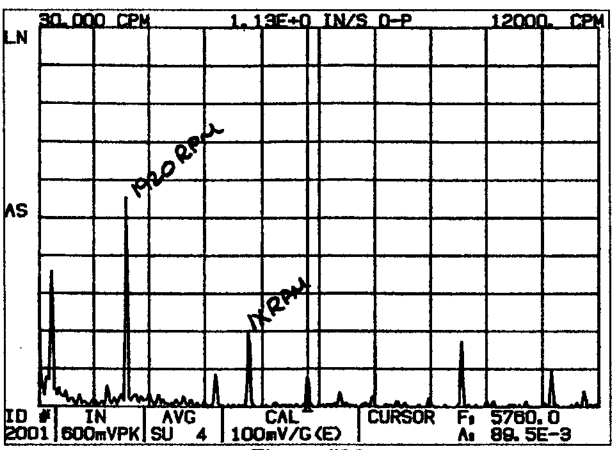


Figure #36

The predominate frequency is 1920 rpm (42.3% of running speed) with an amplitude of .627 in/sec (6.24 mils). Subsynchronous vibration should never be allowed to exceed 30% of the bearing clearance without immediate shutdown, 20% shutdown should be considered, and 10% would be allowable." Since we were over the 20% rule and being under the 30%, it was decided to run the turbine; but, limit its speed range except for testing purposes so it would fit into the afore mentioned preset rules.

The turbine speed was varied up and down the load range (speed range) to see if this subsynchronous vibration "tracked" running speed or "locked on" one frequency. This test indicated the vibration was locked on to 1920 rpm. Therefore; the cause of the vibration was some type of "whip".

Next, the turbine was brought up to 4700 rpm. Coast down curves (Figure #37) in peak average (Bottom Plot) were going to be plotted vs. instantaneous spectrums (Top Plot) to determine the speed this whirl develops. This data plots the vibration with the turbine running 4065 rpm. Note the subsynchronous vibration is present at 1920 rpm and at 3840 rpm.

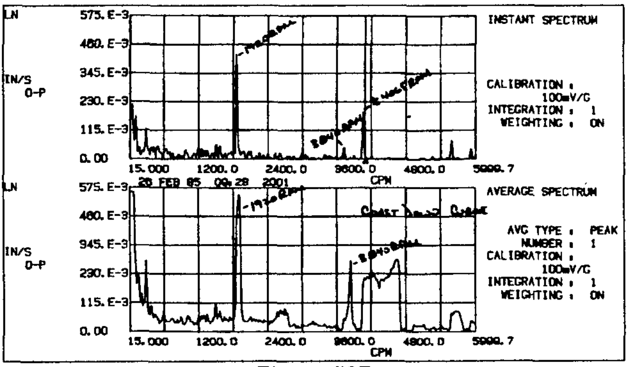


Figure #37

Figure #38 shows the vibration with the turbine running at 4010 rpm with no subsynchronous vibration. This indicates that the factor causing this vibration is not present until the turbine is in the last 20% of its speed range, the point at which steam flow is the highest.

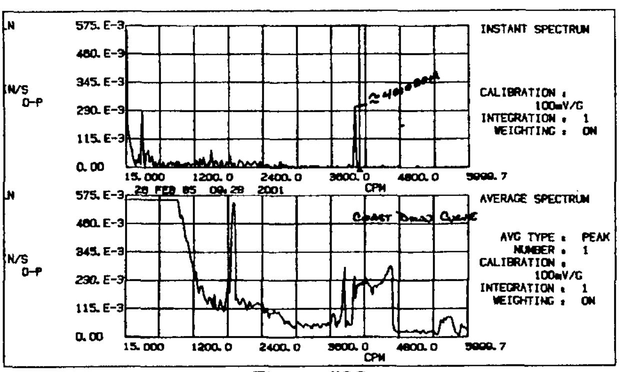


Figure #38

While the testing was progressing, discussions were taking place with Plant Operations and Mechanical Supervisors. The intent was to find if any abnormalities were encountered putting the turbine and pump back in service.

One statement stood out. A unit operator stated the critical around 1900 rpm was rougher than normal. The only critical should be at 2400 rpm. If the critical speed is around 1900 rpm, this would account for the subsynchronous vibration at 1920 rpm. The vibration being locked onto the first rotor critical at 1920 rpm means the first rotor critical would have been lowered from 2400 rpm.

Basic engineering tells us that critical speeds are based on mass, stiffness and geometric properties of the rotor. Vibration data did not show any high 1X, so it could be concluded no weight loss was suffered by the shaft. Also; if there were a lowering of the mass due to weight loss, the critical speed should have increased. The only way to lower the critical speed is to add mass or lower the bearing stiffness. Since the turbine bearings had not been changed it is unlikely the stiffness was lowered.

$$f_n = \sqrt{\frac{576EIg}{WL^3}} \quad (2)$$

Natural Frequency in Radians

E= Modulus of Elasticity (lbs/in^2)

I = Moment of Inertia (in^4)

G =, Gravitational Acceleration (ft/sec^2)

W = Total weight of Shaft (**lbs**)

L = Effective Length of Shaft (**in.**)

In this formula you have four (4) fixed parameters: the modulus of elasticity, the moment of inertia, gravitational acceleration, and the rotor weight. Using this formula and plugging in the 1920 rpm present critical speed and the 2400 rpm should-be critical speed; then solving for the shaft length, you determine the shaft needs to be eighteen inches longer to lower the critical speed to 1920 rpm.

Since the simple model showed the shaft had to be lengthened to lower the critical speed, it was decided to remove the coupling spool piece from the coupling and do a start-up and coast down curve looking for the critical speeds. When this was performed, only one critical speed was found at 2400 rpm proving the theory of a coupling problem was correct.

Corrective Action(2): Maintenance was instructed to inspect the complete coupling for abnormal wear or evidence of coupling torque lock.

Results: Operations Personnel correctly diagnosed a severe pump problem when on-line monitor indicated abnormal axial position and high radial vibration. Past case histories helped solve this problem. Previous data and reports can cut analysis time down when good, neat, documented data can be referenced. Another pump had the same problem eighteen months earlier. Maintenance personnel found that after the pump was overhauled, no grease had been packed into the turbine half of the coupling. The turbine coupling hub had cracked from heat generated by no lubrication. The complete coupling had to be replaced. When the turbine was put back in service, no subsynchronous vibration appeared, and the only critical speed was at 2400 rpm.

Conclusions: The low pressure turbine bearing (inboard turbine) did not experience this whirl vibration because, when steam flow lifted the shaft enough to lock the coupling; the shaft lifted off the bearing (4010 rpm). This was evidence because the bearing metal temperature dropped. Once steam flow dropped off, the coupling was able to unlock and slide, eliminating the whirling problem. If this problem had gone unchecked, a catastrophic failure of the coupling and turbine was a very real possibility.

A problem such as the friction whirl should never have occurred. The last step in an overhaul should be to grease the coupling. This type of externally excited friction whirl locks on the frequency of excitation. It is both harmonic and sub-harmonic. A problem like this does not occur without some cause; usually a maintenance oversight. The lack of grease in the turbine coupling half did not allow for spacer movement.

Case #10 – Structural Pedestal Resonance – New Installation

Problem: Immediate problems were suffered during the initial running of two new DC motors. The motors were a retrofit of older drive motors on an aluminum rolling mill. The new motors (two motors in series – Figure #39) were supplied by General Electric with a maximum operating speed of 1500 rpm. The motors are a complete retrofit including the motor frame and pedestal. It should be noted the steel pedestal for the system is hollow. A sister installation of this system is installed in another plant and operating without vibration problems throughout the speed range.

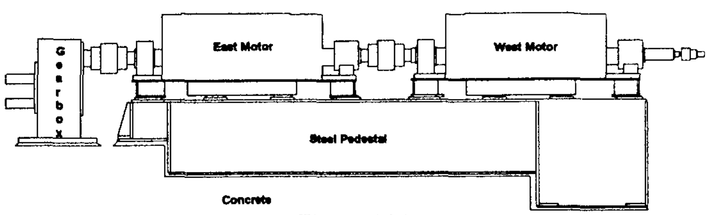


Figure #39

The only difference between the motors in this plant and the sister plant; is the problem motors utilize rolling element bearings. The sister plant is using straight bore babbitt sleeve bearings while the problem motors were retrofitted with rolling element bearings against the manufactures recommendations.

Symptoms: The motors were still not in operation; however, during trial runs high vibration amplitudes were found on the west motor while running around full rolling mill speed, 1260 rpm (21 Hz). While there was vibration on the east motor the amplitudes were acceptable. During the setup of equipment for testing the motor accidentally turned on and went up to just above operating speed (Figure #40). This data indicated the vibration increases suddenly when the motor speed is at the top end of the operating range.

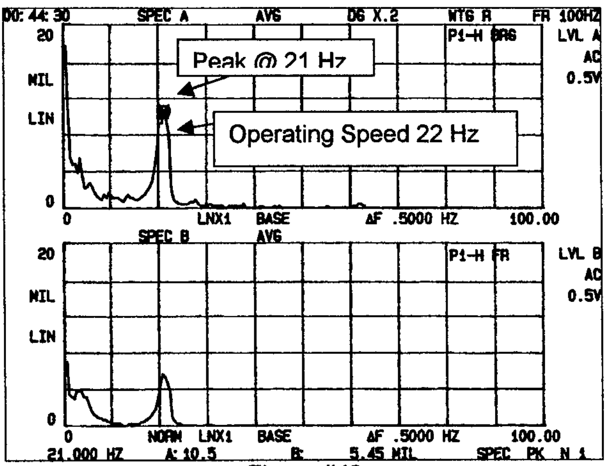


Figure #40

The operating speed reached 22.0 Hz with the peak vibration at 21.0 Hz.

Test Data and Observations: A test plan was developed to look for a natural frequency of the shaft and bearing pedestal system. Impact tests would be used to look for natural frequencies around 20 Hz. These tests would also show if there were other natural frequencies in the running speed range or if natural frequencies could be excited by multiples of operating speed. Impact tests would be conducted in the axial, horizontal and vertical directions. Tests would be conducted on the motor bearings, frame, steel pedestal and foundation. Included in the impact tests would be information on stiffness of each equipment component and information for amplification calculations.

Equipment used for the testing included: instrumented hammers for the impact tests, a dual channel FFT analyzer for the impact tests and the startup / coast down data, a sixteen channel digital tape recorder along with amplifying and integrating power supplies for the accelerometers used in the data collection.

Impact test data indicated natural frequencies at the upper end of the operating range (Table VI).

Test Position	**Horizontal**	**"Q" Factor**	**H- (K)(lbs./in)**	**V- (K)lbs./in)**
West Motor Outboard Shaft	**23.0 Hz** **40.0 Hz** **62.0 Hz**	**3.5** **7.0**	1,049,723	1,037,736
West Motor Outboard Bearing	**23.0 Hz** 31.0 Hz 62.0 Hz	**8.4** 13.3 13.8	787,735	1,432,836
West Motor Outboard Frame			3,957,115	264,957
West Motor Inboard Bearing	**23.0 Hz** 62.0 Hz	**6.9** 8.7		
West Motor Inboard Frame	40.0 Hz 64.0 Hz	16.7 10.4	4,134,420	113,600
East Motor Outboard Bearing	**23.0 Hz** 31.0 Hz 62.0 Hz	**4.0** 17.6 9.5		
East Motor Inboard Bearing	31.0 Hz	14.1 Hz		

Table #VI

Operating data showed the majority of the vibration to be in the horizontal direction and a drastic drop in vibration as the motor speed lowered form full speed operation (Figure #41). The vibration amplitude drops 3.0 mils in 2.0 Hz.

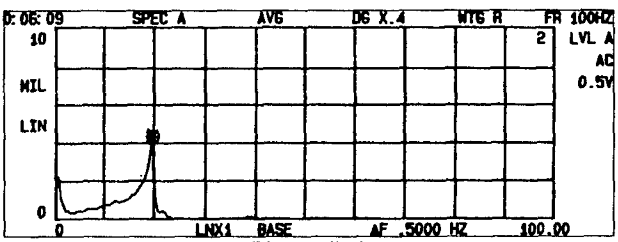


Figure #41

This equipment train is definitely operating at a natural frequency when the motor speed is at 1322 rpm (22.03 Hz). There are two ways to essentially move a natural frequency and they are to either change the mass or the stiffness of the system. Presently; there is approximately 2,425,000 lbs./in of stiffness in the system. Additional supports will probably not add to this total. Calculations were made to see if filling the steel pedestal with concrete would add enough mass to lower the natural frequency enough, so the equipment could be run at full speed operation. The addition of mass; likewise, will have little effect on the natural frequency. This leaves the options of adding additional support or installing a Dynamic Vibration Absorber.

Corrective Action: The idea behind a Dynamic Vibration Absorber (DVA) is to have the absorber vibrate instead of the equipment. The dynamic absorber is designed to have the same natural frequency as the offending machine. This equipment train would require a DVA to have a natural frequency around 22.0 Hz. This would then absorb the vibration energy and allow the machine to run normal vibration amplitudes. Since the west motor had the highest vibration amplitudes, the absorber would be designed for this motor.

The design of the DVA dictates that it must be at least 10% of the total weight of the machine. The west motor, frame and pedestal weighted roughly 25,000 pounds. This means the absorber must have at least 2500 pounds of weight. To design an absorber one has to determine the length of the vertical support based on the weight to be supported (Figure #42).

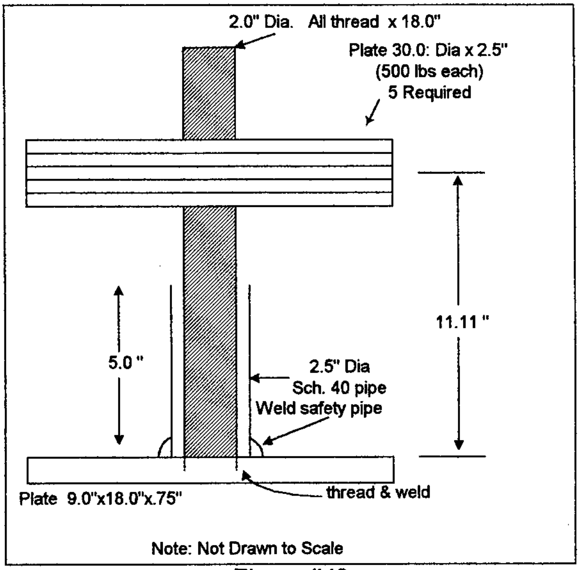


Figure #42

Results: This design was intended to be a temporary short term solution. The final recommendations were to retrofit the bearing to a sleeve design. In fact the motor manufacturer had the bearings and bearing pedestals in stock since the original design was for sleeve bearings. The intention was to install the absorber for a short time so the plant could get the new mill up and running. Once the mill was operating correctly the bearings would be changed out. The absorber dropped the vibration by 75% and has continued to operate without a bearing change back to the sleeve bearings.

Conclusions: The cause of the vibration was the type of bearings chosen by the plant. The sister plant chose to use babbitt bearing for their motors. Babbitt bearings were what the motor was designed to utilize. However; this plant decided to use rolling element bearings. This decision was made because plant personnel felt that the operating speed of 1260 rpm was too fast for the babbitt bearings. The rolling element bearing did not provide any damping; therefore, allowing the high vibration amplitudes.

Case #11 – Electrically Excited Structural Resonance

Problem: While collecting baseline data on a new rolling machine vibration was detected in one speed range on a variable speed machine. The motor drive is variable frequency rated at 1195 rpm (11.91 Hz), 600 volt and 350 horsepower. The motor drives a single reduction gearbox with a gear ratio of 15 to 1.

Symptoms: During the equipment baseline check out and acceptance process high vibration was experienced on the input shaft of the gearbox in the horizontal direction. Vibration amplitudes of over .60 in/sec 0-pk were documented. The plant had set overall alarm settings of .10 in/sec 0-pk. The vibration appeared to occur in only one area of the speed range. Figure #43 is a schematic of the gearbox layout.

Production personnel noticed floor vibration when the mill was operating at 2350 feet per minute (fpm). A plot of motor speed versus mill operating speed is in figure #44.

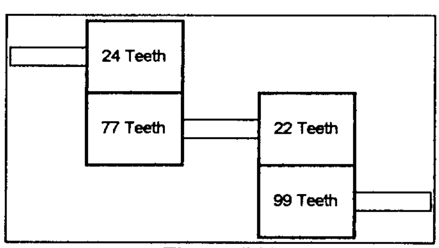


Figure #43

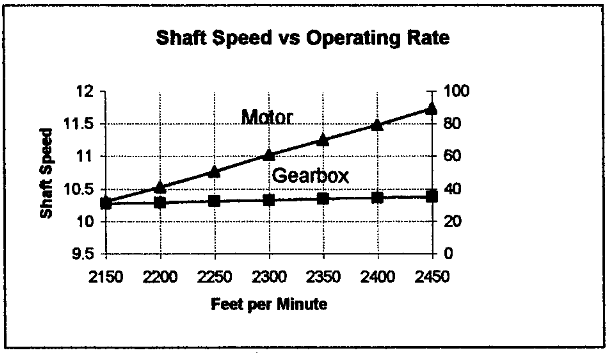


Figure #44

Test Data and Observations: The motor was operated through out its speed range and the overall vibration was recorded from 2100 fpm to 2450 fpm. Figure #2 shows us this is from 10.3 Hz to 11.73 Hz on the motor . The overall vibration data plotted against motor speed is in figure #45. Figure #45 illustrates how the vibration peaks when the motor is operating at 11.3 Hz. Vibration spectral plots were taken at various speed setting. Figure #46 shows the vibration at a motor speed of 9.88 Hz and Figure #46 shows the data from a motor speed of 11.25 Hz. The Data at 11.25 Hz shows a vibration at 3X running speed of the motor. The time data is essentially a pure sine wave.

This data points to a resonance problem because it is only in one area of the speed range. Generated frequencies of the bearing in the motor, gearbox and roll are all well above the operating speed of this frequency at 33.81 Hz. Since this is a variable speed motor with a maximum speed of 1195 rpm (19.91 Hz) the excitation frequency is equal to line frequency divided by rated speed times shaft speed.

$$\textit{Excitaion} = \left(\frac{\textit{Line Frequency}}{\text{Rated Speed}}\right) \text{x Shaft Speed} \qquad (3)$$

$$\textit{Excitaion} = \left(\frac{60}{19.91}\right) \text{x } 11.25 = 33.9 \text{ Hz}$$

The excitation frequency at 33.9 Hz is equal to the dominant frequency of 33.81 Hz in figure #47. This is a definite indication that the excitation frequency of the motor is exciting a natural frequency when the motor speed is 11.25 Hz.

An impact test was attempted on the gearbox to confirm the natural frequency; however, because of the system mass and close quarters it was very difficult to swing a hammer. Because this frequency was being generated by the excitation frequency of the motor, it was decided to shut the motor down and observe the vibration when the power was cut. Immediately upon the cut off of power, the vibration dropped out. This conclusively pointed toward a natural frequency excited by the excitation frequency of the variable speed motor.

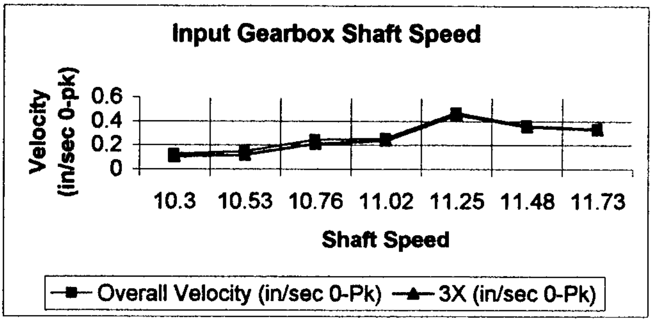


Figure #45

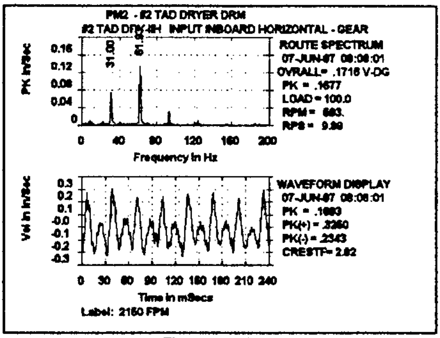


Figure #46

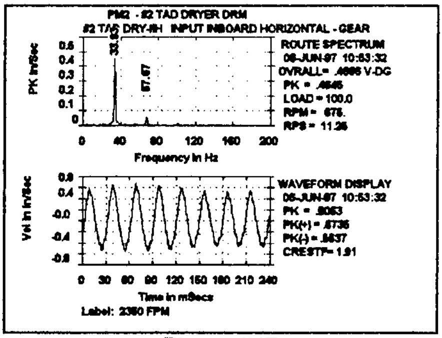


Figure #47

Corrective Action: The plant turned over the data to the equipment supplier and instructed them to redesign the equipment foundation.

Results: The equipment supplier designed isolators and snubbers on the gearbox to help relieve the vibration caused by the natural frequency. Figure #48 shows data after the isolators and snubbers were installed.

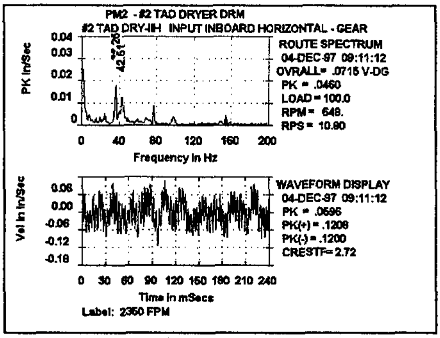


Figure #48

Case #15 – Rolling Element Bearing Fundamental Train Frequency

Problem: Trend vibration indicated the possibility of a bearing vibration on a Vertical Boiler Circulating Water Pump. The motor had been rebuilt in the last year and the pump radial bearings had been replaced. The plant suspected a bearing problem; however, they could not determine if the problem was in the pump or motor.

Symptoms: Vibration trend data showed a high frequency vibration at 3643.6 Hz (Figure #49). The overall amplitude of vibration had been trending up over the past several months.

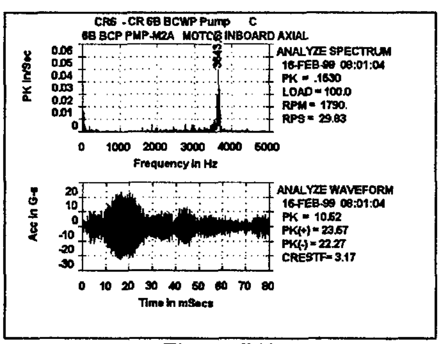


Figure #49

This data was collected with 400 lines of resolution with an Fmax of 5000 Hz. The resolution for this plot is 37.50 Hz per bin. Because of the bin width, frequency identification would be difficult. Timewave data shows peak acceleration at 23. 57 g's. Looking at the same data on a smaller frequency range shows the 1X vibration along with a frequency at 5.58 Hz (Figure #50). A walk down of the unit reveled a high pitch squeal around the pump. The motor pump train operates at 29.58 Hz with radial bearings in the motor, top, and bottom of the pump. Below the upper pump bearing are two thrust bearings.

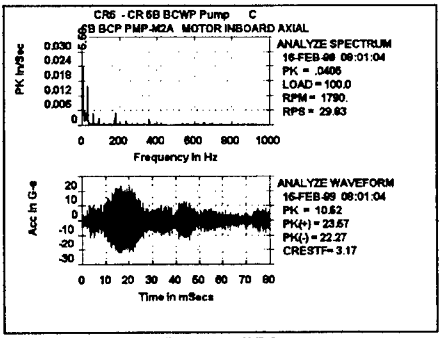


Figure #50

These thrust bearings were not replaced during the last pump overhaul. Utilizing a data collector to get an overview of the data, vibration on the pump casing around the location of the thrust bearings (Figure #51) was excessive (1.0 in/sec 0-pk +).

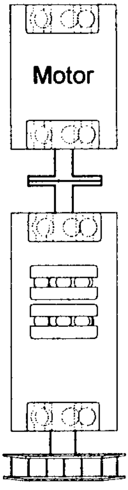


Figure #51

Based on these observations a test plan was developed.

Test Data and Observations: Since the previous data was collected with only 400 line of resolution, frequency identification was a problem. Also, the time plot was based on 1024 bits of data (400 line) and provided .08 seconds of data. This only showed 2.4 rotations of the shaft, not enough data to determine if the frequency of the energy bursts in the time data (Figure #49 or #50). The new data would be collected with 3200 lines of resolution and have a time plot of 4096 bits (1600 lines). This would provide a time plot of .32 seconds. Data was collected from all points available on the machine and on the pump housing in line with the thrust bearings in the horizontal and vertical directions.

Calculation of the bearing fault frequencies provided the data in Table #V.

	SKF 6226	SNR 7230	SKF 6324	SKF 29434	SKF 6326
FTF	12.18	12.82	11.58	**13.14**	11.58
BSF	81.24	83.56	64.87	79.63	65.0
BPFO	109.6	205.2	92.63	**210.3**	92.67
BPFI	156.6	268.2	144.0	263.1	143.9

Table #V

The new plots with the longer time data shows the bursts of energy are at 1X (Figure #52).Looking at the same point in Velocity (Figure #53) shows essentially the same information as figure #2; 1X and a subsynchronous vibration between 5.0 and 6.0 hertz. None of the velocity or Acceleration plots indicate any bearing frequencies. The time data is typical of bearing frequencies; however, no distinct frequencies can be identified. Since the goal of this analysis was to identify if the problem was a pump or motor bearing it was decided to try peak view analysis. The peak view data plots showed two distinct bearing problems on the thrust bearing in the pump (Figure #54).

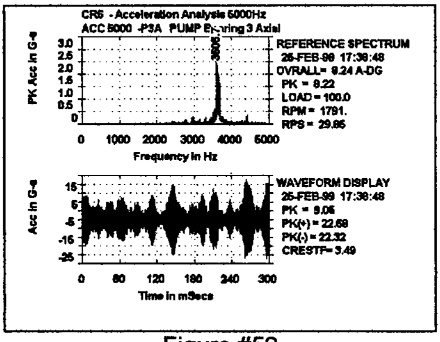


Figure #52

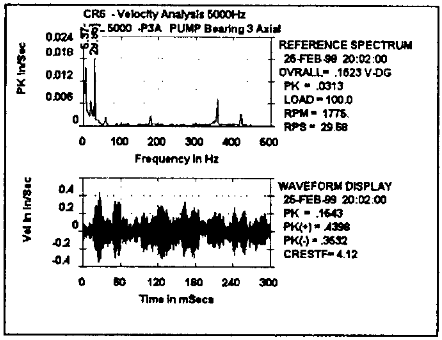


Figure #53

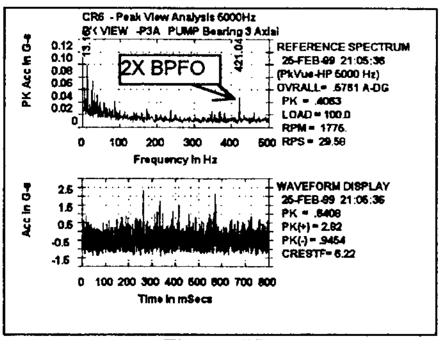


Figure #54

Corrective Action: The recommendation was to replace the thrust bearings.

Results: Both the cage and outer race of the bearings were cracked.

Conclusion: The decision to not replace the thrust bearing in the pump when the radial bearings were replaced was inappropriate. Replacing the thrust bearing required removing both radial bearings. In a two week time the radial bearings had to be replaced twice. This decision to not replace the bearing was made totally on the cost of the thrust bearing. In the end the thrust bearing replacement ended up costing replacement power in addition to labor and parts.