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IMPACT OF WIRELESS LASER BASED SHAFT ALIGNMENT ON VIBRATION AND STG COUPLING FAILURE

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ABSTRACT

Proper shaft-to-shaft alignment during normal machinery operation is essential for improved machinery reliability. Having the proper tools to perform an accurate coupling alignment are critical to maximized maintenance efficiency. Improvements in hardware and software technology have greatly simplified coupling alignment procedures.

A comprehensive machine alignment consists of two elements: measurement of the shaft-to-shaft alignment and measurement of the thermal growth and dynamic movement of each bearing housing. Accurate measurement of these elements should be used to position the cold alignment condition in order to properly align the shaft under loaded conditions.

Shaft-to-shaft alignment is typically measured using either laser or dial-indicator instrumentation. RoMaDyn used a laser based system utilizing bluetooth technology for shaft alignment readings. This system provides excellent accuracy, flexibility, and fast results. Often if unit casing temperatures change little immediately after a hot coast down, relative shaft to shaft thermal growth measurements can be acquired using laser coupling alignment readings. Hot and cold alignment data can then be incorporated into an alignment drawing to determine measured shaft relative thermal growth. The measured thermal growth readings or OEM calculated growth numbers are then incorporated into the cold unit alignment settings to provide the best hot alignment condition.

This paper will discuss a recent case history involving chronic coupling failures at a customer's facility. Acquired vibration and shaft alignment readings will be discussed. These measurements will illustrate the effects of alignment changes on machinery vibration levels and coupling reliability.

INTRODUCTION

Accurate shaft alignment is fundamental for coupling reliability and proper vibration performance. A critical step in performing an accurate shaft alignment is to have accurate thermal growth information. Typically this information is provided by the OEM, however actual thermal growth information can provide a more accurate means of obtaining an accurate hot alignment. Hot alignment readings are sometimes very difficult to obtain due to the change temperatures which occur immediately after a hot shutdown. Rapid measurement of coupling alignment is essential for obtaining accurate hot alignment readings.

For the project discussed below, alignment tolerances were critical due to the short span of the coupling, approximately 7 inches. Accurate knowledge of thermal expansion of the unit and hot alignment offsets was identified as an essential factor in obtaining an accurate hot alignment. Implementation of wireless laser based alignment tools resulted in being able to measure hot alignment offsets accurately due to minimal setup time of the equipment. Unfortunately, this unit had experienced chronic coupling failures since 2002. The OEM investigated the coupling failures and made several attempts to improve coupling alignment without success. The economic impact of the chronic coupling failures was approximately $200,000 in 2006. Resolving the coupling alignment issue lead to elimination of coupling failures, greater unit availability and reduced costs.

INSTRUMENTATION

Shaft to shaft alignment measurements were made with a Pruftechnik Wireless Rotalign Ultra laser system. This system provides excellent accuracy, flexibility, and fast results. This system was also equipped with bluetooth technology, which eliminated cabling, permitting rapid setup and measurements.

Proper vibration instrumentation is essential for obtaining accurate vibration data. This 20 MW steam turbine was driving a 4 pole 1800 rpm generator through a speed reducing gearbox. As shown on the Machinery Arrangement Sketch (figure 1), the unit was equipped with X-Y shaft-relative proximity probes on all bearings. The unit was also equipped with a thrust probe in the turbine. All probes were connected to a Bently Nevada 3300 monitoring rack located in an adjacent control cab.

All vibration monitors appeared to be functioning correctly throughout the measurements except for the differential expansion monitor that stayed in alarm during all machine conditions. It was suspected that this monitor/transducer system may not have been configured correctly.

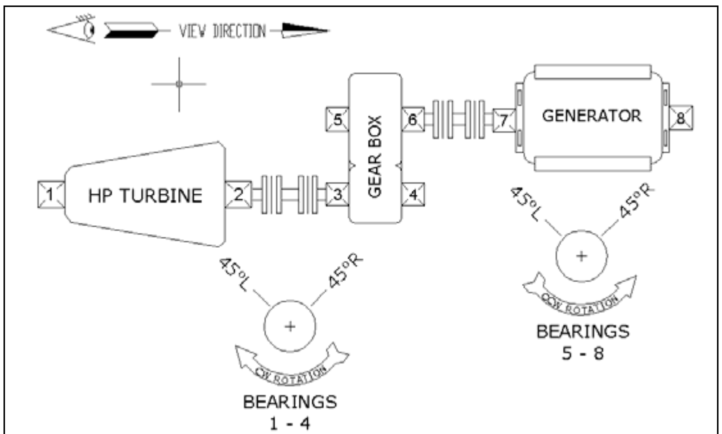

Figure 1: Machinery Arrangement Sketch.

In addition to the shaft relative X-Y proximity transducers, the unit was equipped with a Keyphasor® probe on the turbine shaft to provide a once-per-turn reference for diagnostic purposes. This signal was conditioned through a Keyphasor® multiplier/divider in order to also provide a once-per-turn Keyphasor® signal for the generator. All vibration probe signals were then field wired to a Bently Nevada ADRE® 208 DAIU to acquire diagnostic data.

ALIGNMENT PROGRAM DESCRIPTION

A comprehensive machine alignment consists of two elements: measurement of the shaft-to-shaft alignment and measurement of the thermal growth and dynamic movement of each bearing housing. Accurate measurement of these elements should be used to position the cold alignment condition in order to properly align the shaft under loaded conditions.

Shaft-to-shaft alignment is typically measured using either laser or dial-indicator instrumentation. RoMaDyn uses a Pruftechnik Wireless Rotalign Ultra laser system for shaft alignment readings. This system provides excellent accuracy, flexibility, and fast results. Casing temperatures on this unit change little immediately after a hot coast down allowing relative shaft to shaft thermal growth measurements to be acquired using laser coupling alignment readings. The hot and cold alignment data was then incorporated into an alignment drawing to determine measured shaft relative thermal growth. The measured thermal growth readings were then incorporated into the cold unit alignment settings to provide the best hot alignment condition.

Thermal growth and dynamic movement studies are designed to accurately measure the vertical and horizontal movement of each unit casing relative to the foundation as a unit progresses from ambient conditions to hot, full-load operation. Readings are typically acquired on or about bearing housings using optical scales or X-Y-Z plane proximity probes affixed to water cooled stand pipes. These casing measurements are relative to the unit foundation or machine frame and can be incorporated into the alignment drawing to determine total dynamic and thermal growth relative to the foundations.

By incorporating both the thermal growth data and the shaft alignment data, the cold shaft alignment offsets can be determined to provide an accurate alignment at full load conditions. Without actually measuring the bearing thermal growth values (either relative shaft to shaft or absolute), the cold alignment settings can only be estimated based on calculated thermal growth which in many cases can lead to poor alignment results.

Based on research conducted over the past 20 years and actual industrial machine evaluations, shaft alignment tolerances to mark the limit of an alignment dead band are evaluated for each individual drive train application based on machine speed and coupling span. If alignment is within the dead band, then further alignment work will not improve the running characteristics of the machine. These tolerances are not based on coupling manufacturer's allowances, as coupling tolerances may be excessive for the coupled machinery.

There are presently no specific machinery alignment tolerance standards published by ISO or ANSI, but typical tolerances for alignment are shown in Figure 2 below taken from *Piotrowski, John D., Shaft Alignment Handbook, 1995,* *pg. 145* . Tolerance tables established by Pruftechnik and built into the Rotalign Ultra alignment system are within the Piotrowski tolerances below and are used in most RoMaDyn projects unless specific tolerances are established or required by the equipment OEM or the customer.

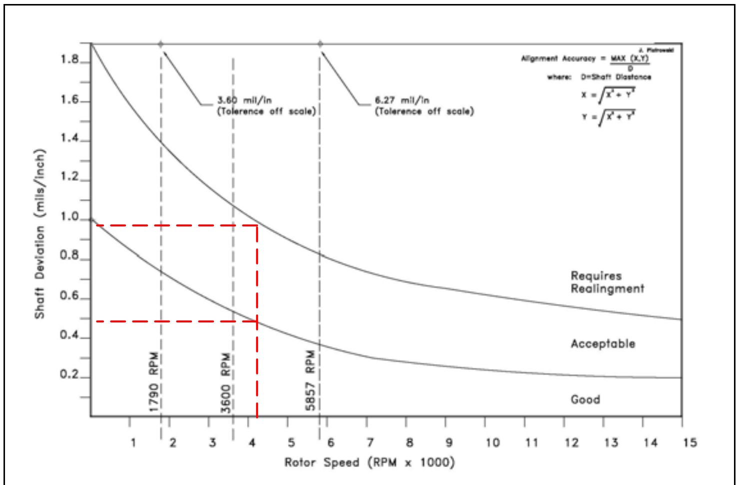

Figure 2: Coupling alignment tolerances.

Machine vertical alignment moves are generally accomplished using stainless steel slotted shims placed under the machine support feet. In this instance, the turbine inlet wobble plate vertical alignment can be accomplished by removing existing shims under the base of the wobble plate. Soft foot conditions should be minimized with additional shims under feet as required, holding the total number of shims under each foot to five or less if possible.

AUGUST 2006 ALIGNMENT ANALYSIS

Turbine/Pinion "As Found" Cold Alignment

The "as found" vertical and horizontal measured cold alignment for the turbine/gearbox shaft to shaft relationship is shown in figure 3. The turbine shaft is represented to the left, with the pinion shaft fixed on the right. Offsets at the coupling and feet corrections for co-linear alignment are shown in the figure.

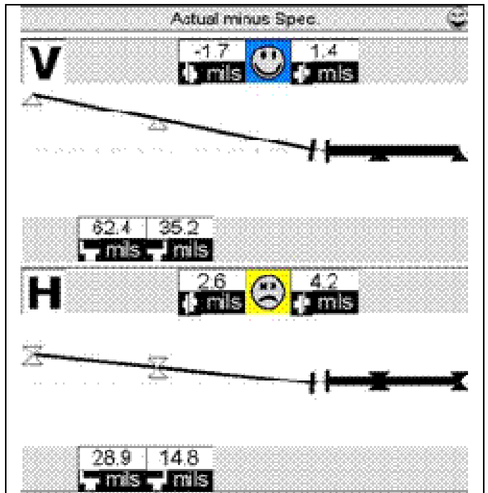

Figure 3: As found cold alignment

The offset numbers represent actual offsets minus any target offsets or thermal growth numbers programmed into the Rotalign system. For this machine, target offsets and thermal growth numbers were set to zero and displayed values represent actual measured coupling offsets. The dashed line projected left of the pinion shaft represents the co-linear alignment position that the machine feet moves on the turbine shaft are measured from this line. A positive foot number represents a shaft position that is high from a co-linear position and needs to move down to achieve a fair or co-linear alignment.

Turbine/Pinion As Found Hot Alignment

Figure 4 shows the as found hot alignment position between the turbine and pinion and is considered the root cause of the numerous coupling failures. The maximum industry acceptable coupling offset over a 7 inch coupling span at a machine speed of 4150 rpm is 3.4 mils at the 'A' and 'B' flange locations (0.48 mils/inch).

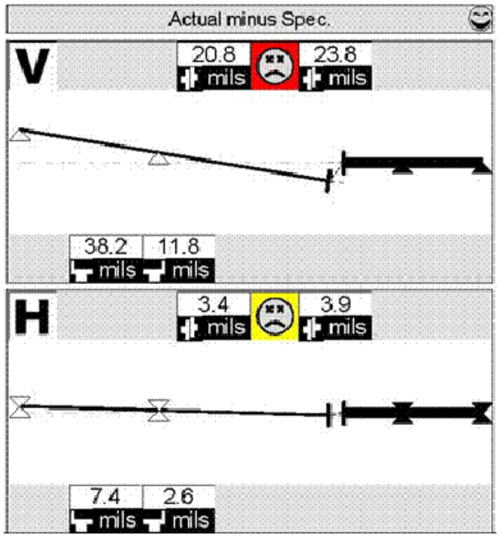

Figure 4: As found hot alignment condition

This poor hot alignment condition is the result of thermal growth on the turbine and/or gearbox when the cold shaft to shaft alignment is set close to fair. Although the coupling alignment readings indicate the relative amount of thermal movement between the two shafts, they do not resolve how much each shaft or casing is moving relative to the foundation or frame. In order to determine the amount of casing or shaft absolute thermal growth, the turbine and pinion were instrumented with proximity probes mounted on water cooled stands viewing the bearing housings relative to the unit frame. This data is shown below in Figure 5.

Turbine and Pinion Vertical Thermal Growth

In many cases the cold alignment offsets for a unit can be determined by measuring the cold and hot coupling alignment condition and producing an alignment drawing based on the shaft to shaft relative cold/hot positions. Several assumptions are made in this process: (1) that the unit has not cooled significantly when the measured hot alignment data is acquired immediately after shutdown; (2) that the shaft positions during loaded operation are not significantly affected by process loads; and (3) that the cold offsets determined from comparison of the hot/cold relative alignment data can be implemented in usually one machine case chosen to be the machine to be moved. If any of these assumptions are in doubt, measurement of the machine case thermal growth(s) relative to the unit foundation or frame is prudent to determine accurate thermal growth offsets for the cold alignment settings.

To ensure accurate thermal growth readings of the steam turbine, proximity probes mounted on water cooled stands were set up viewing the turbine inlet bearing, turbine exhaust bearing and pinion drive end bearing housings. The probes documented casing movement in the X and Y growth directions from the shaft centerline at the turbine exhaust and pinion drive ends and movement in the X, Y and Z (axial) directions at the turbine inlet end at the left support foot (viewed turbine to generator). Photos 1 and 2 below show the transducer arrangements. Photo 3 shows the alignment equipment setup. Figure 5 below displays the data acquired from these measurements transferred into the as found cold alignment position drawing.

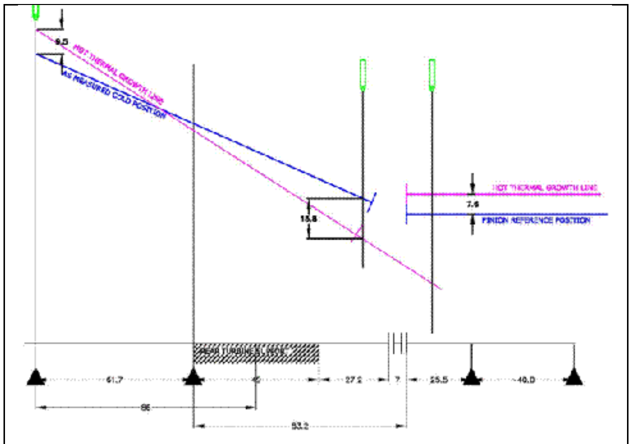

Figure 5: Turbine and pinion measured thermal growth.

The blue lines represent the cold relative vertical shaft positions from the cold alignment check. The magenta lines represent the turbine and pinion shaft position changes based on the thermal growth measurements of a 9.5 mils rise at the turbine inlet end, a 15.8 mil decrease at the turbine exhaust end and a 7.6 mil rise of the pinion drive end bearing.

Photo 1: Growth probes at coupling area

Photo 2: Growth probes at inlet end.

Photo3: Alignment equipment setup

Figure 6 below displays the relative vertical hot and cold shaft positions acquired from the hot and cold alignment checks. This data shows a relative (to the pinion) turbine shaft drop of approximately 24 mils from cold to hot operating conditions at both ends of the turbine. Remembering that these are relative shaft to shaft positions and some of the turbine drop is accounted for by gearbox rise.

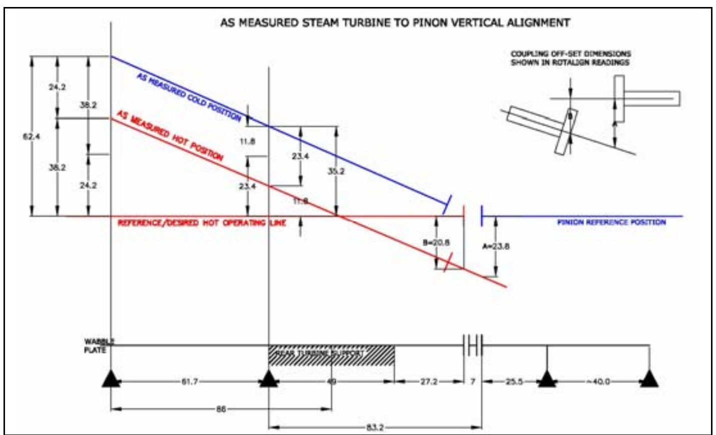

Figure 6: Hot and cold shaft relative alignment readings

This hot/cold alignment data is now combined with the thermal growth data in Figure 7 below to verify the hot laser alignment readings are close to actual and have not been affected by unit cooling or process load changes.

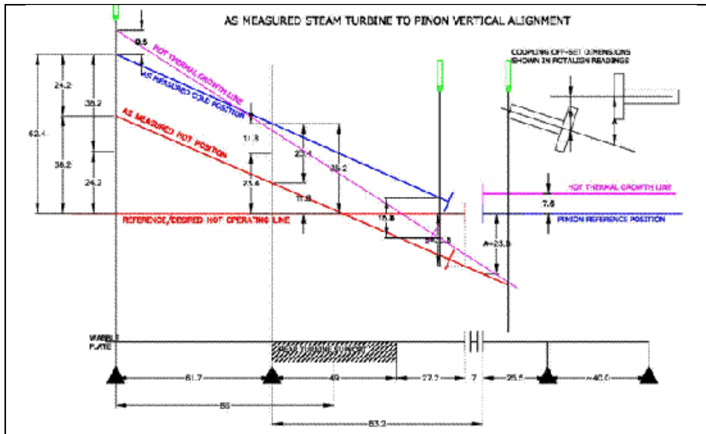

Figure 7: Hot and cold turbine/pinion alignment with thermal growth lines in magenta

This verification is accomplished by translating the magenta turbine and pinion hot thermal growth lines down relative to each other so that the magenta pinion line lies on the blue pinion reference line. The magenta turbine line is then compared to the red "*as measured hot position*" turbine line to ensure that they are close to the same. This is shown in Figure 8 below.

There is an obvious angular difference in the two lines; however, the coupling offset differences measure to within 4 mils indicating fairly good agreement between the cold/hot alignment readings and the thermal growth readings.

The angular difference in the turbine lines may be accounted for in some part by the fact that thermal growth readings of the pinion blind end bearing were not available to correct these readings. The assumption was uniform thermal growth on both ends of the gearbox. If there was any differential thermal growth in the pinion bearings, this would induce some angularity in the pinion thermal growth line, affecting the translated position of the turbine line. In order for the turbine line to be closer to the red as measured hot position, the pinion blind end would need to have slightly less thermal growth than the drive end, opening the angle between the turbine/pinion magenta lines.

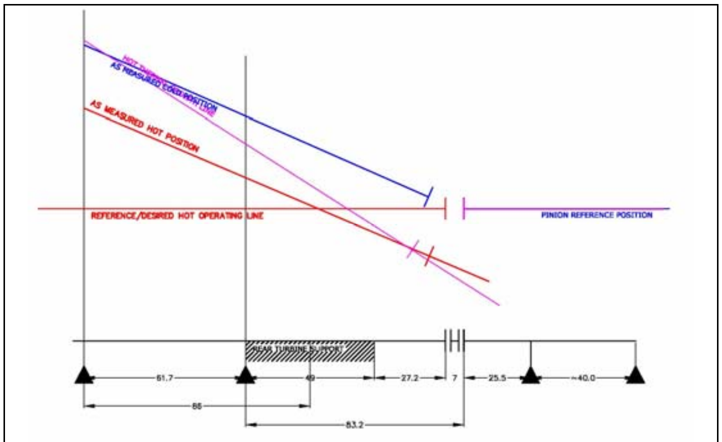

Figure 8: Comparison of cold/hot alignment readings and measured thermal growth readings. Magenta thermal growth lines translated so that pinion line lies on pinion reference line.

This thermal growth data was then combined with the cold/hot alignment readings and the shaft centerline vibration data to determine the proper cold offsets that the unit should be aligned with in order to achieve co-linear hot operating shaft alignment. This is discussed below in the Alignment Settings section of this paper.

Turbine and Pinion Horizontal Alignment and Thermal Growth

Although outside of the recommended alignment tolerances as shown above in Figure 2, the turbine to pinion cold and hot horizontal alignment was much closer than the vertical alignment. The thermal growth at both turbine bearings and the pinion drive end was recorded as 20 mils to the left at the turbine inlet, 1.4 mils to the right at the turbine exhaust and 7.6 mils to the left at the pinion drive end. Figure 9 documents the cold to hot thermal growth changes between the turbine and gear based on the laser alignment readings.

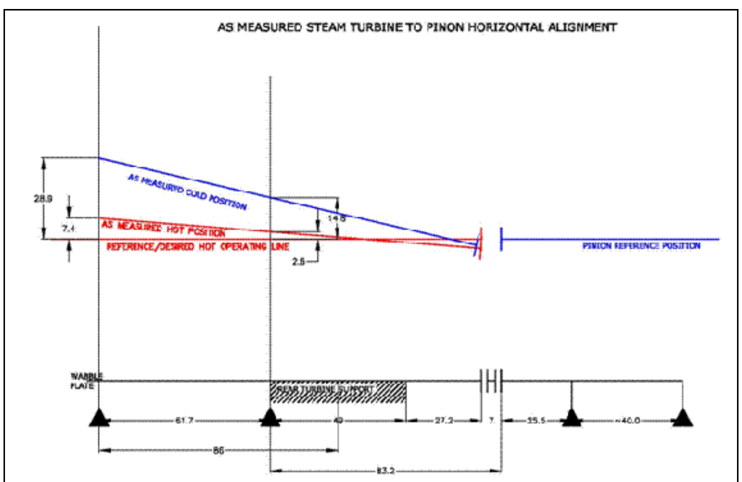

Figure 9: Horizontal cold and hot turbine to pinion alignment

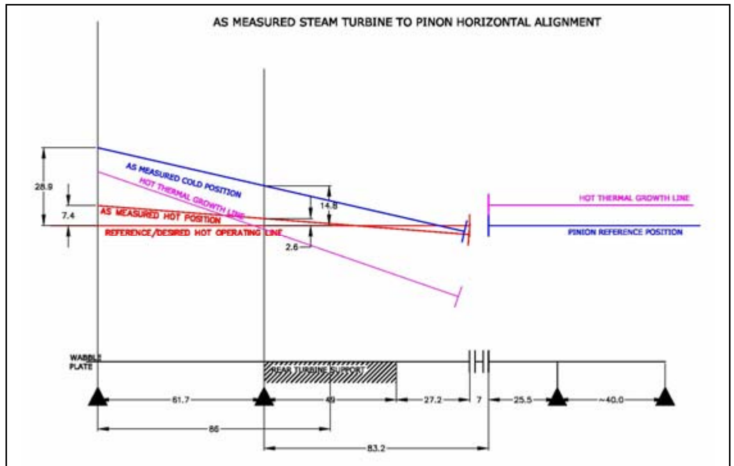

Figure 10: Horizontal cold and hot turbine to pinion alignment with hot thermal growth line.

Figure 10 above adds in the horizontal thermal growth line in magenta. As can be seen, the thermal growth line is further out of tolerance than the hot alignment readings. The bracket holding the thermal growth proximity transducers was water cooled in the main stand pipe that controlled vertical growth very closely; however, although the horizontal support arms on the coupling stand were water drip cooled as best as possible, it is believed that thermal expansion of this angle iron accounts for much of the difference in the laser alignment readings and the horizontal thermal growth data. The smaller differences in the readings at the turbine inlet are likely accounted for by the much shorter bracket lengths at this location. The hot and cold laser alignment readings are believed to better represent the horizontal alignment condition and will be used to determine the desired cold alignment settings.

Gearbox/Generator As Found Cold Alignment

The as found vertical and horizontal cold alignment condition between the gearbox and the generator is shown in Figure 11. The gearbox gear shaft is represented as the fixed reference shaft with the generator shaft to the right. The 10.375 inch coupling span on an 1800 rpm shaft accommodates larger offsets and the as found cold alignment was within the acceptable tolerance range.

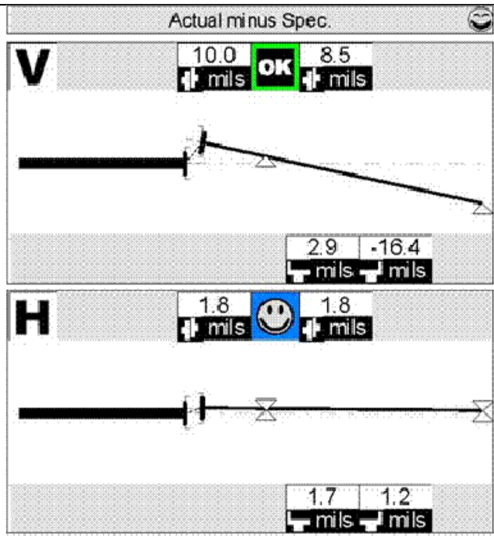

Figure 11: Gear/generator cold alignment condition

Gearbox/Generator As Found Hot Alignment

The as found hot alignment condition between the gearbox and the generator is shown in Figure 12 below. Gearbox and/or generator growth has improved the hot alignment condition; however, as will be shown in the Alignment Targets section below, both the gearbox and generator would likely need vertical shim moves in order to achieve the correct turbine to gearbox hot alignment.

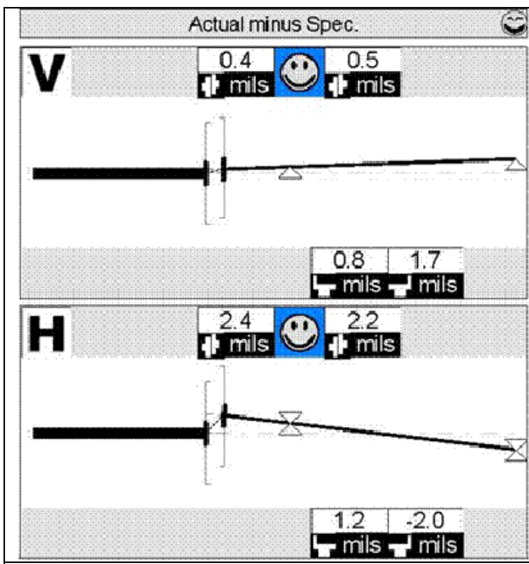

Figure 12: gearbox to generator hot alignment

August 2006 Alignment Targets

Incorporating the hot/cold shaft alignment readings with verification of the readings from the thermal growth data, and allowing for the shaft centerline rise from the vibration data helped determine the correct cold offsets necessary to achieve the desired hot co-linear unit alignment condition.

The data was plotted on to an alignment drawing to determine the correct amount of cold offset required to properly align the turbine/gearbox/generator.

The cold offset targets are referenced to one machine casing remaining fixed and the other as the moveable on each side of each coupling. For convention purposes, the gearbox in both the turbine to gearbox drawing and the gearbox to generator drawing is referenced as the fixed machine. To achieve the target offsets, all three machine cases were determined to require vertical shimming as the target offsets for the turbine require the casing to be lowered from the existing as found position and there were presently no shims under the turbine support feet. Raising the gearbox was also determined to require raising the generator to maintain proper alignment. In effect, the entire train would require lifting so that the turbine casing could be lowered the appropriate amount.

The target offset alignment drawings are broken up into 4 drawings: the turbine/gearbox vertical targets; the turbine/gearbox horizontal targets; the gearbox/generator vertical targets and the gearbox/generator horizontal targets. The drawings are shown below in Figures 13 to 16.

The target offsets in the drawings are shown as both feet positions on the moveable machine casing relative to a zero reference line foot position on the fixed machine and as ‘A’ and ‘B’ coupling flange offsets. If laser alignment equipment is used to re-align the unit often these coupling offset targets should be programmed into the system to help achieve the proper feet positions.

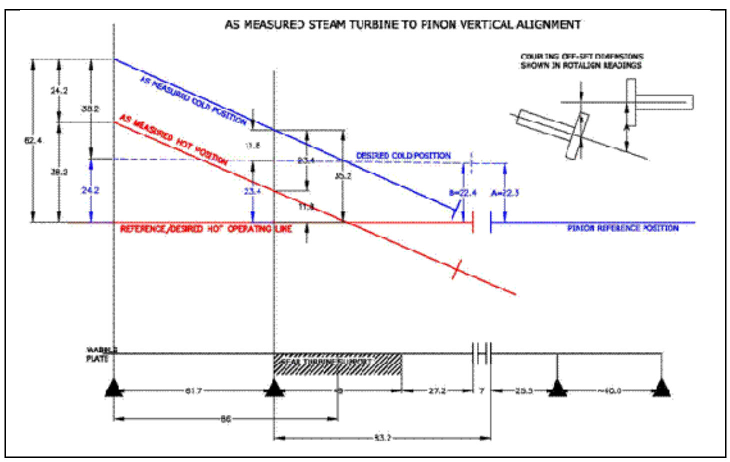

Figure 13. Desired vertical cold alignment offsets between turbine & gearbox.

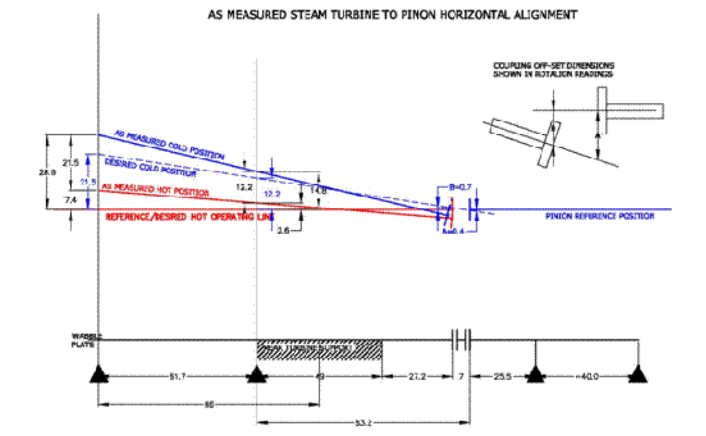

Figure 14. Desired horizontal cold alignment offsets between turbine & gearbox

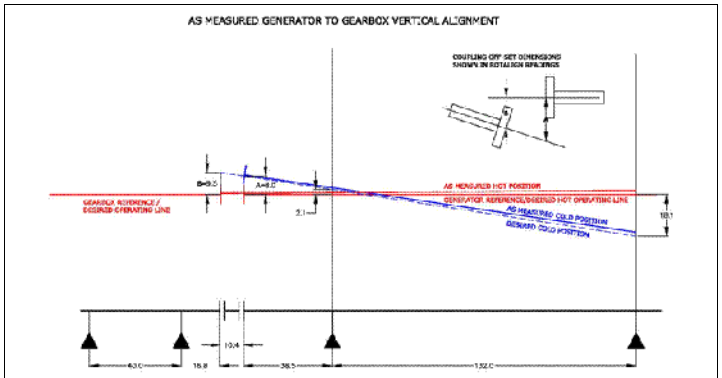

Figure 15. Gearbox to generator vertical alignment target offsets

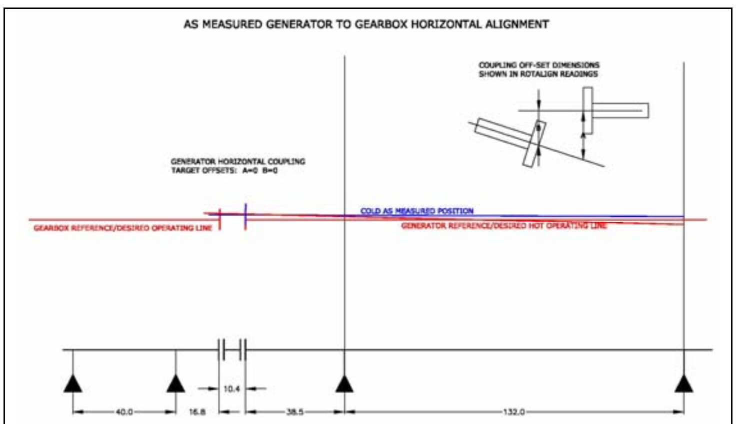

Figure 16. Gearbox to generator horizontal alignment target offsets.

The turbine to gearbox drawing vertical offset numbers did not incorporate the 3-5 mil turbine/pinion difference in shaft rise. Including this into the coupling offsets adjusts the 'A' and 'B' offset targets to 26 mils.

Proposed August 2006 Alignment Procedure

During August, the recommended procedure for the re-alignment of the unit was to first raise the gearbox and *precision level it* and dowel pin or re-dowel pin it to the foundation in one location under the pinion drive end prior to beginning alignment moves on the turbine then generator casings. In addition, the OEM recommendations on gearbox dowel pin procedures for proper pinning should be followed.

It was also recommended that additionally care should be taken to measure and remove all soft foot conditions from the gearbox prior to beginning turbine or generator alignment. The maximum allowable soft foot should be less than 1 mil.

AUGUST 2006 VIBRATION ANALYSIS

Analysis of the shaft centerline and orbit data supported the alignment findings of hot misalignment between the turbine and pinion bearings. Prior to alignment measurements, shaft centerline data shown below in Figure 17 indicated bearings #1 and #4 (turbine inlet & pinion blind end) were supporting the predominate load of the turbine/pinion coupled shaft.

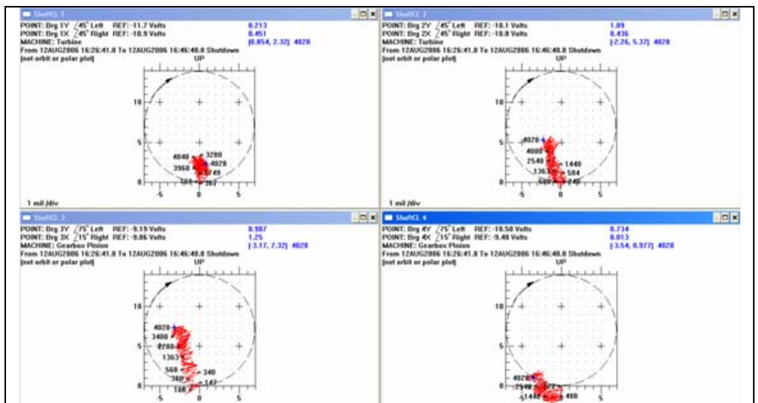

Figure 17. Shaft centerline data from turbine & pinion bearings.

The inlet turbine bearing operating position was abnormally low in the bearing clearance typical of heavy journal loading.

The differential rise position at steady state full load rpm of the pinion drive end and blind end by 4 – 6 mils was indicative of very poor alignment in this double helical set.

Shaft relative orbits showed indications that bearing load was not evenly distributed. The orbits of bearings 2, 3 and 4 were open and elliptical, see figure 18. They were also predominately forward precessed, see figure 19. Vibration amplitudes and orbit shapes at bearing 2 and 3 were not similar on either side of the coupling.

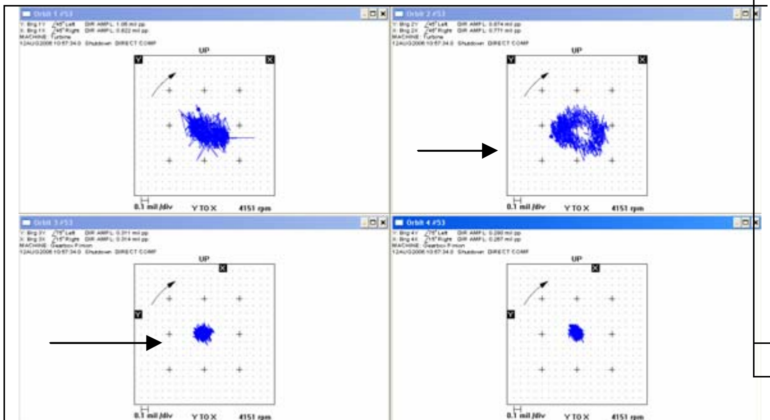

Figure 18: Direct (unfiltered) Orbits, Bearings 1,2,3 and 4

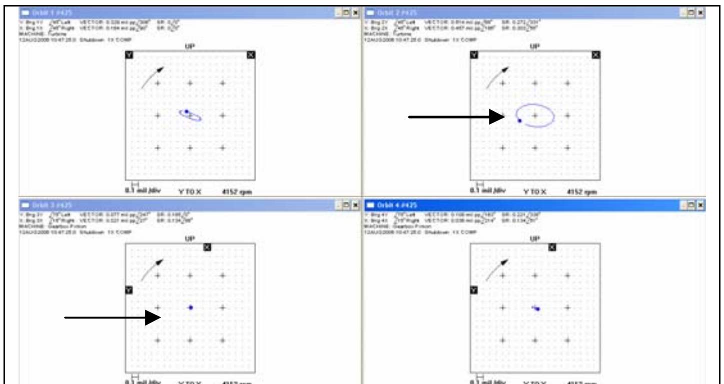

Figure 19: 1X Synchronous Filtered Orbits, Bearing 1,2,3 and 4

Waterfall data shown below in Figure 20 from the gear at bearings 5 and 6 showed a 6X gear frequency component that might be indicative of tooth wear that should be verified during a future outage. This vibration component drops out during coast down when generator initeria back drives the gear set and gear teeth are engaged on the opposite faces. A recommendation was made to open and inspect the gear and pinion for wear indications during the next outage. A blue check of the teeth was also recommended.

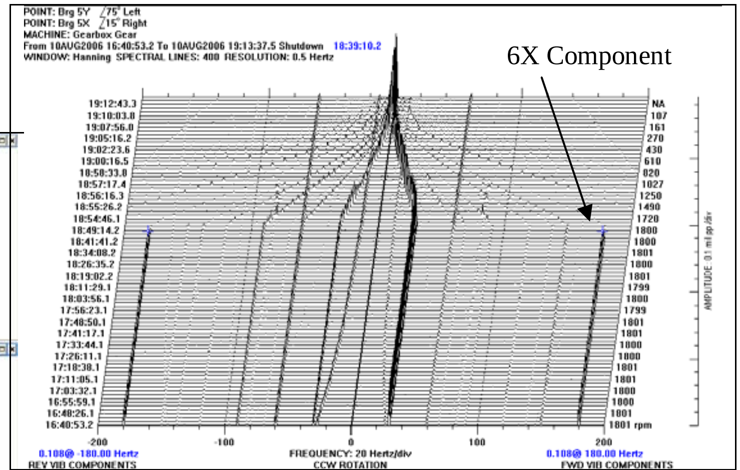

Figure 20. Waterfall plot of the #5 gear bearing.

Other August 2006 Vibration Data Observations

Resonances in the turbine, gearbox and generator were well damped with low (below 4) synchronous amplification factors through the observed resonances.

The rotor slow roll condition of all rotors was documented at less than 200 rpm on the shutdown and was considered free of dynamically induced vibration. It was therefore considered indicative of the mechanical and electrical runout condition of the rotors at the probe locations. The overall slow roll condition of the rotors was marginal to poor. The maximum recommended amount of slow roll permitted under the American Petroleum Institute's machinery standard API 670 is 0.25 mils pp or not more than 25 % of the maximum shaft relative vibration amplitude, which ever is greater. All slow roll values on this unit exceeded this recommendation. The highest values recorded are listed in Table 1.

Bearing	Direct Amplitude	1X Amplitude/Phase 'Y'
1	1.20 mils pp	0.09 mils pp∠ n/a
2	0.66 mils pp	0.54 mils pp ∠ 311°
3	0.53 mils pp	0.40 mils pp ∠ 78°
4	0.53 mils pp	0.46 mils pp ∠ 238°
5	0.63 mils pp	0.20 mils pp ∠ 243°
6	0.62 mils pp	0.31 mils pp ∠ 86°
7	0.58 mils pp	0.34 mils pp ∠ 78°
9	0.49 mils pp	0.23 mils pp ∠ 297°

Table 1: Slow Roll Values

Monitor vibration amplitudes are directly affected by slow roll values. Vibration amplitudes on monitor readings can appear artificially high by the slow roll value, *or may be dangerously suppressed by the same value* depending on the phase relationship between the slow roll signal and the vibration response. In all cases the slow roll value appeared to be adding to the monitor vibration amplitudes. A recommendation was made that during the next outage when rotors are *removed*, specifications should be included to return the transducer target areas to within API tolerances. Lathe diamond burnishing is the recommended method to surface finish transducer target areas.

At this time, the balance condition of all rotors appeared to be good. The highest relative 1X filtered amplitude was located on the turbine exhaust bearing at 1.44 mils 1X compensated.

The unit did not appear to be load sensitive and load vectors were well below 1 mill.

NOVEMBER 2006 ALIGNMENT ANALYSIS

Turbine/Pinion As Found Alignment Condition

From August until November 2006, a consensus formed that the previous shaft to shaft relative alignment readings should be verified before any corrective alignment moves were performed. Since there was an obvious angular difference in the "as measured hot position" line and "thermal growth line" in figure 8, a verification of the as found cold and as found hot alignment measurements was performed. Therefore, the general objectives of the November 2006 alignment measurements were as follows:

- **Re-measure the as found hot and cold shaft to shaft alignment condition between the turbine and gearbox.**
- **Provide alignment assistance to reposition the turbine relative to the pinion shaft to achieve a co-linear shaft to shaft alignment condition during hot loaded operating conditions.**
- **Record and evaluate the unit mechanical condition based on the lateral vibration data.**

The as found vertical and horizontal measured cold alignment for the turbine/gearbox shaft to shaft relationship is shown in Figure 21. The turbine shaft is represented to the right, with the pinion shaft fixed on the left. Offsets at the coupling and feet corrections for co-linear alignment are shown in the figure. This cold alignment position differed from the readings taken in August by a considerable amount, with these readings showing the turbine lower 42 mils at the front feet and 20 mils at the rear feet than those taken in August. The difference in the cold readings was determined to be the result of temperature differences in the turbine front wobble feet at the time of the readings with these readings colder than those in August and very near ambient (91°F at the turbine mount interface) after 48 hours. The August "cold" readings were taken 24 hours after shutdown.

The offset numbers represent actual offsets without any target offsets or thermal growth numbers programmed into the Rotalign® system. For this project targets offsets and thermal growth numbers were set to zero and displayed values represent actual measured coupling offsets.

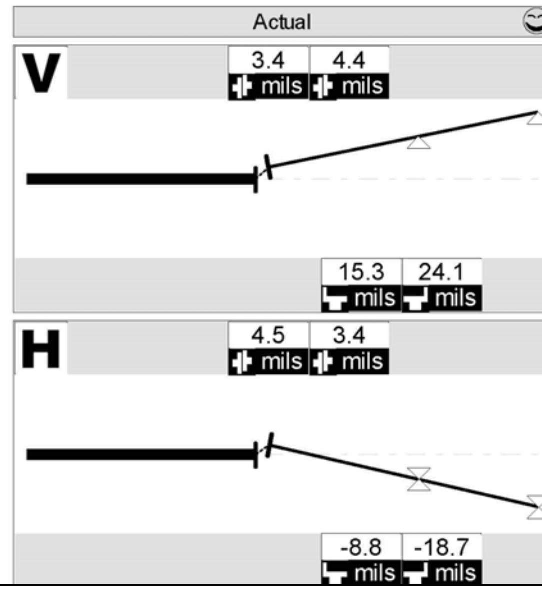

Figure 21. As found cold alignment readings.

The dashed line projected right of the pinion shaft represents the co-linear alignment position that the machine feet moves on the turbine shaft are measured from. A positive foot number represents a shaft position that is high from a co-linear position and needs to move down to achieve a fair or co-linear alignment.

Turbine/Pinion As Found Hot Alignment

Figure 22 shows the as found hot alignment position between the turbine and pinion. The maximum industry acceptable coupling offset over a 7 inch coupling span at a machine speed of 4150 rpm is 3.4 mils at the 'A' and 'B' flange locations (0.48 mils/inch) for a 'good' alignment condition.

This poor hot alignment condition is the result of thermal growth on the turbine and/or gearbox when the cold shaft to shaft alignment is set close to fair.

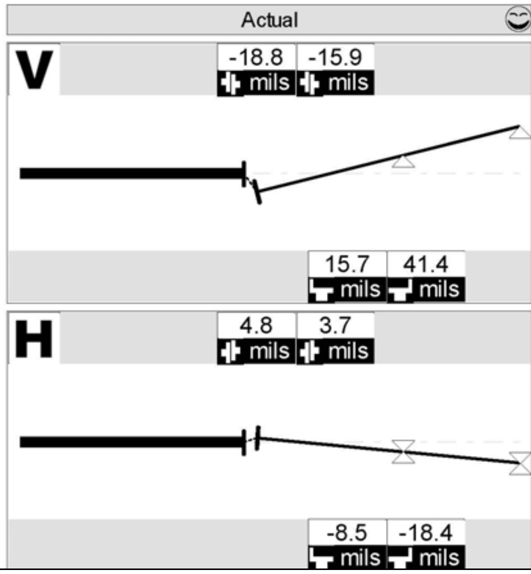

Figure 22. As found hot alignment condition

These ‘as found’ November cold and hot alignment positions were used to determine the final shim corrections needed to the turbine to achieve a hot co-linear alignment condition.

Gearbox to Generator Alignment

The cold/hot gearbox to generator alignment was not measured during November, but found to be acceptable in the hot condition during the readings taken in August.

Turbine to Pinion Alignment Targets and Moves

These November hot/cold shaft alignment readings were plotted on an alignment drawing to determine the correct amount of cold offset required to properly align the turbine to gearbox.

The cold offset targets are referenced to one machine casing remaining fixed and the other as the moveable on each side of each coupling. The gearbox is referenced as the fixed machine.

The target offset alignment drawings are broken up into 2 drawings: the turbine/gearbox vertical targets and the turbine/gearbox horizontal targets. The drawings are shown below in Figures 23 and 24.

The target offsets in the drawings are shown as both feet positions on the moveable machine casing relative to a zero reference line foot position on the fixed machine and as ‘A’ and ‘B’ coupling flange offsets. For the turbine rear foot alignment moves, rather than plot feet moves at the rear turbine casing, two front end shim moves (10 mils removed and 20 mils removed) indicated the shaft to be pivoting about the rear (exhaust) bearing coupling side edge. Shim moves for the turbine rear support were calculated from this lateral location.

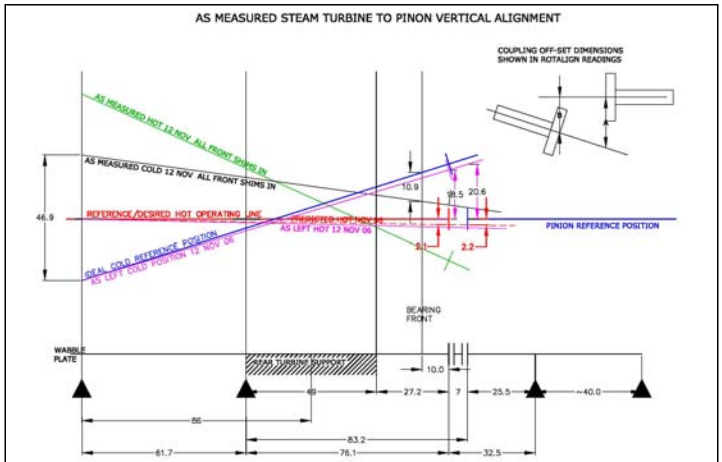

Figure 23. Vertical cold alignment targets and as left cold/hot vertical alignment positions.

Horizontal cold to hot movement on the turbine is not well understood as there was very little as found cold to hot horizontal movement as seen in Figures 19 and 20; however, there appeared to be a significant change between the cold/hot horizontal positions after the final vertical alignment moves as shown in Figure 22 below.

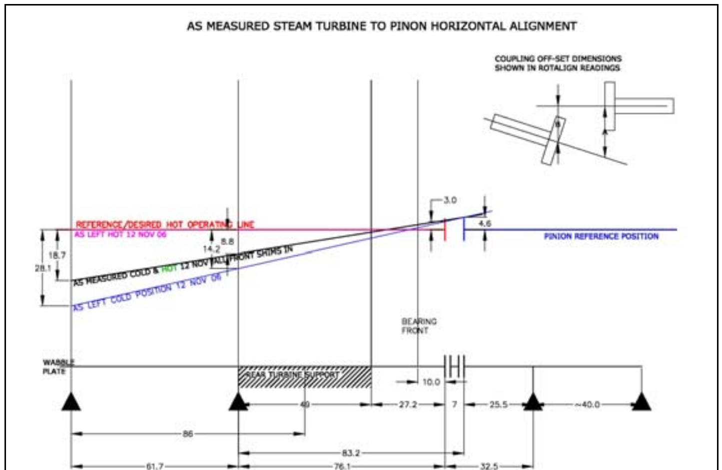

Figure 24. Horizontal cold alignment targets and as left cold/hot vertical alignment positions

The final hot horizontal alignment condition appeared very good with 1.3 mils off-set at the B coupling location the largest off-set. As the wobble feet are both pinned and the rear turbine supports are keyed in the horizontal and axial directions, there was no attempt to change the turbine horizontal alignment position.

The final vertical alignment moves on the turbine were as follows:

Remove 20 mils of existing shims from the turbine front wobble feet. Net total of 12 mils installed.

Add 10 mils to the turbine rear support. Net total of 10 mils installed.

The final hot alignment condition between the turbine and gearbox was considered good for the 7 inch span Form Flex spacer coupling. Off-sets at the coupling measured within 15 minutes of unit stop from a thermally stable 18 MW load run were 3.3 and 4.0 mils for the 'A' and 'B' measurements as shown in the hot rotor position drawing in Figure 25 below The turbine shaft is represented as the moveable machine to the right with the pinion to the left.

Industry recommendations for the upper limit of an acceptable alignment condition for a machine speed of 4150 rpm would be an offset *no larger* than 1 mil per inch of coupling span, or a total of 7 mils offset for this Form Flex coupling.

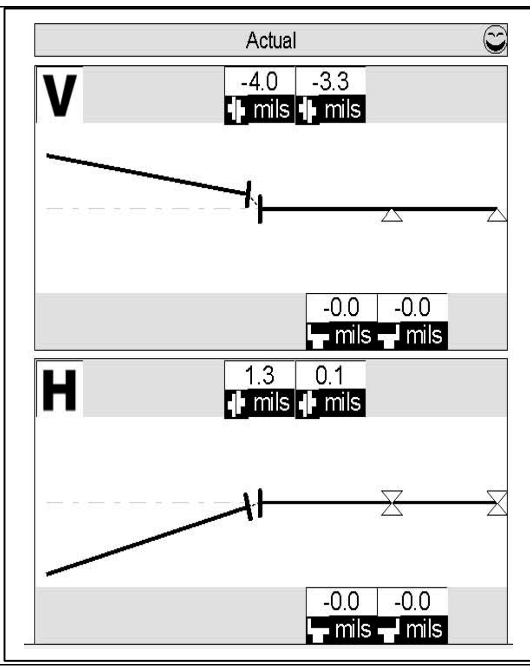

Figure 25: Final Turbine to Pinion measured hot alignment positions.

NOVEMBER 2006 VIBRATION ANALYSIS

Vibration amplitudes on the entire train at load are considered very good. The direct compensated vibration amplitudes on the turbine at 18 MW were 0.91 and 1.02 mils pp at inlet and exhaust ends respectively. 1X compensated amplitudes at the turbine inlet end were below 0.28 mils pp with the exhaust end at a maximum of 0.74 mils pp, although the orbit at the exhaust end was more flattened than desired.

Loaded operational vibration amplitudes on the turbine and pinion were lower since the unit realignment. Table 2 below shows August pre-alignment direct compensated (slow roll/runout removed) and 1X compensated amplitudes vs. November post alignment amplitudes. Data from the 'Y' probes is shown and all 'X' probe readings were below these amplitudes for the pre/post alignment.

Bearing	Pre alignment August 2006		Post alignment Nov 2006	
	Direct mils pp	1X mils pp	Direct mils pp	1X mils pp
1Y	1.03	0.17	0.92	0.19
2Y	1.95	1.44	1.06	0.17
3Y	0.82	0.45	0.74	0.29
4Y	0.65	0.38	0.46	0.19

Table 2: Loaded operation direct and 1X compensated vibration amplitudes

Oddly, shaft centerline and orbit data indicated little change between the August and November post alignment, see figure 26. Although amplitudes were lower, orbits on the turbine were still somewhat flattened and shaft centerline positions for both the turbine and pinion are very similar to the August data. Shaft centerline data still indicated the front turbine bearing to be heavily loaded and very low in the bearing clearance with the pinion blind end bearing lower than the drive end. Although the relative shaft to shaft alignment between the turbine and pinion now appears to be in good condition during operation conditions, the shaft centerline positions indicate the #1 and #4 bearing still to be carrying the predominate turbine/pinion rotor load. Turbine bearing loading may be more a function of the turbine rotor mass distribution than relative bearing alignment conditions.

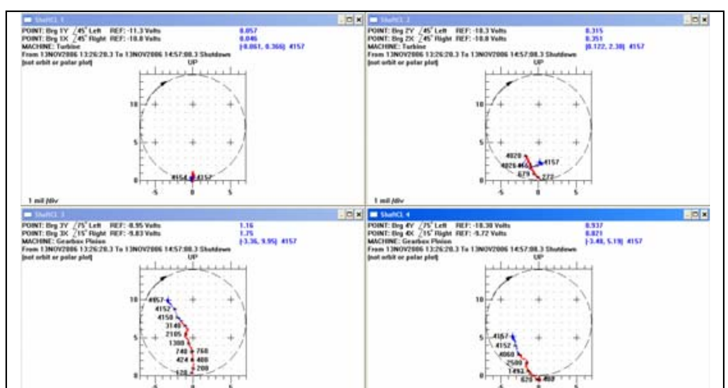

Figure 26. Shaft centerline data from turbine & pinion bearings.

Shaft relative orbits showed indications that bearing load was more evenly distributed. The orbits of bearings 2, 3 and 4 were open and elliptical, see figure 27. The orbits were also predominately forward precessed, see figure 28. Vibration amplitudes at bearing 2 and 3 were also very similar on either side of the coupling.

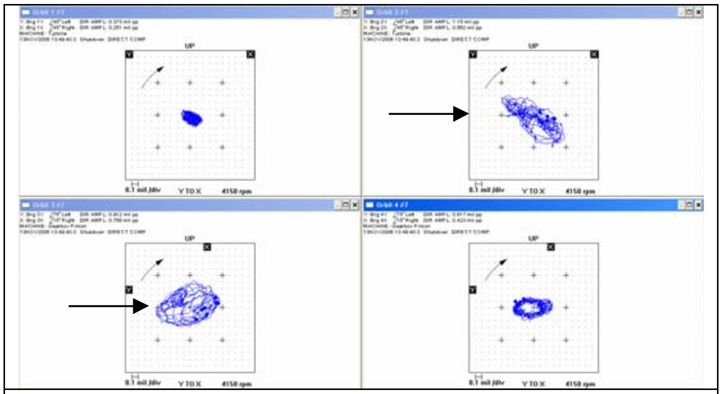

Figure 27: Direct (unfiltered) Orbits, Bearings 1,2,3 and 4

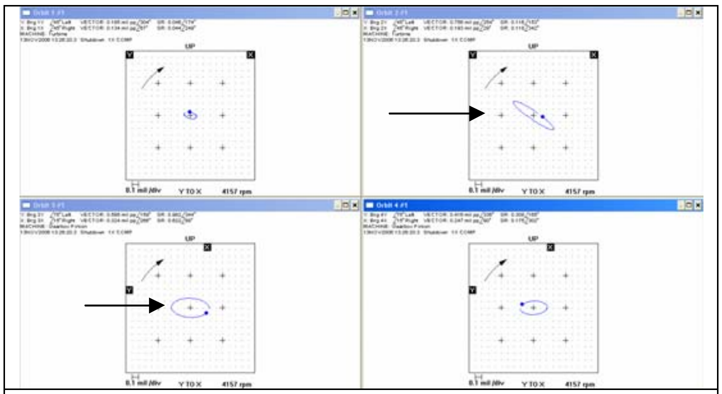

Figure 28: 1X Synchronous Filtered Orbits, Bearing 1,2,3 and 4

Waterfall data from the gear bearings still showed a 6X gear frequency component that may be indicative of tooth wear that should be verified during the next outage, see figure 29. The component drops out during coast down when generator inertia back drives the gear set and gear teeth are engaged on the opposite faces.

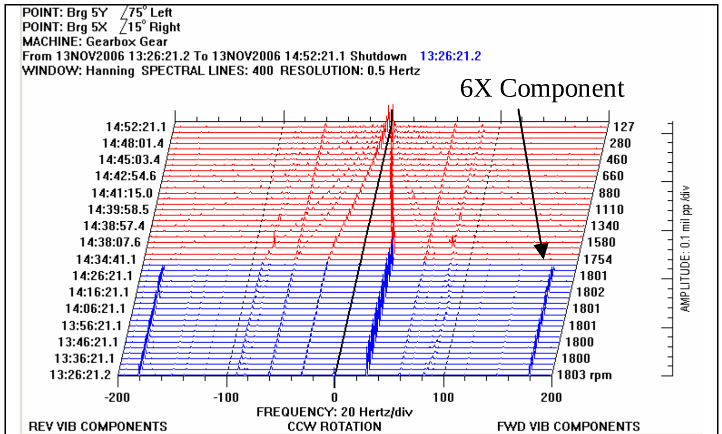

Figure 29. Waterfall plot of the #5 gear bearing.

In addition to the alignment corrections, recommendations were made to open and inspect the gear and pinion for wear indications during the next outage. A blue check of the teeth was recommended.

After the alignment change, resonances in the turbine, gearbox and generator were still well damped with low (below 4) synchronous amplification factors through the resonances.

During November, the rotor slow roll condition of all rotors were again documented at less than 200 rpm on the shutdown and was considered free of dynamically induced vibration. It is therefore considered indicative of the mechanical and electrical runout condition of the rotors at the probe locations. The overall slow roll condition of the rotors was again marginal to poor. The maximum recommended amount of slow roll permitted under the American Petroleum Institute's machinery standard API 670 is 0.25 mils pp (6.4µm pp) or not more than 25 % of the maximum shaft relative vibration amplitude, which ever is greater. All slow roll values on this unit still exceed this recommendation. The highest values recorded are listed in Table 3. These values did not appreciably change relative to the values record in August.

Bearing Number	Direct Amplitude	1X Amplitude/Phase 'Y'
1	0.94 mils pp	0.06 mils pp∠ n/a
2	0.37 mils pp	0.12 mils pp ∠ 345°
3	1.18 mils pp	0.99 mils pp ∠ 344°
4	0.47 mils pp	0.31 mils pp ∠ 163°
5	0.32 mils pp	0.39 mils pp ∠ 110°
6	0.80 mils pp	0.70 mils pp ∠ 260°
7	0.61 mils pp	0.40 mils pp ∠ 356°
8	0.46 mils pp	0.19 mils pp ∠ 215°

Table 3 Slow roll values

Monitor vibration amplitudes are directly affected by slow roll values. Vibration amplitudes on monitor readings can appear artificially high by the slow roll value, *or may be dangerously suppressed by the same value* depending on the phase relationship between the slow roll signal and the vibration response. In all cases the slow roll value appears to be adding to the monitor vibration amplitudes.

A recommendation was again given to correct the slow roll values of the probe target areas. At the next outage where the rotors are to be *removed*, specifications should be included in the repair schedule to return the transducer target areas to within API tolerances. Lathe diamond burnishing is the recommended method to surface finish transducer target areas.

The balance condition of all rotors still appeared to be very good. The highest relative 1X filtered amplitude was located on the generator inboard bearing at 24.8 µm 1X compensated.

The unit still did not appear to be load sensitive and load vectors were well below 1 mill.

SUMMARY

The above case history shows that coupling alignment and vibration conditions can be improved by using advanced wireless laser alignment equipment and thermal growth measurement techniques. The wireless laser alignment equipment allowed the hot alignment measurements to be made quickly and efficiently; resulting in more accurate hot alignment readings. Vibration measurements also showed that vibration conditions were improved after the alignment corrections. After the alignment corrections were implemented, the unit was returned to service, vibration levels remained low, and coupling failures have not reoccurred. The coupling was inspected in March 2007 and no coupling lamination damage was present.

REFERENCES

Piotrowski, John D., Shaft Alignment Handbook, 1995, pg. 1