Benefits of Vibration Analysis for Deployment of Equipment in HLW Tanks – 12341

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ABSTRACT

Vibration analyses of equipment intended for use in the Savannah River Site (SRS) radioactive liquid waste storage tanks are performed during pre-deployment testing and has been demonstrated to be effective in reducing the life-cycle costs of the equipment. Benefits of using vibration analysis to identify rotating machinery problems prior to deployment in radioactive service will be presented in this paper. Problems encountered at SRS and actions to correct or lessen the severity of the problem are discussed. In short, multi-million dollar cost saving have been realized at SRS as a direct result of vibration analysis on existing equipment. Vibration analysis of equipment prior to installation can potentially reduce in-service failures, and increases reliability.

High-level radioactive waste is currently stored in underground carbon steel waste tanks at the United States Department of Energy (DOE) Savannah River Site and at the Hanford Site, WA. Various types of rotating machinery (pumps and separations equipment) are used to manage and retrieve the tank contents. Installation, maintenance, and repair of these pumps and other equipment are expensive. In fact, costs to remove and replace a single pump can be as high as a half million dollars due to requirements for radioactive containment. Problems that lead to in-service maintenance and/or equipment replacement can quickly exceed the initial investment, increase radiological exposure, generate additional waste, and risk contamination of personnel and the work environment. Several different types of equipment are considered in this paper, but pumps provide an initial example for the use of vibration analysis.

Long-shaft (45 foot long) and short-shaft (5-10 feet long) equipment arrangements are used for 25 –350 horsepower slurry mixing and transfer pumps in the SRS HLW tanks. Each pump has a unique design, operating characteristics and associated costs, sometimes exceeding a million dollars. Vibration data are routinely collected during pre-installation tests and screened for:

- Critical speeds or resonance,
- Imbalance of rotating parts,

- Shaft misalignment,
- Fluid whirl or lubrication break down,
- Bearing damages, and
- Other component abnormalities.

Examples of previous changes in operating parameters and fabrication tolerances and extension of equipment life resulting from the SRS vibration analysis program include:

- Limiting operational speeds for some pumps to extend service life without design or part changes.
- Modifying manufacturing methods (tightening tolerances) for impellers on slurry mixing pumps based on vibration data that indicated hydraulic imbalance.
- Identifying rolling element mounting defects and replacing those components in pump seals before installation.
- Identifying the need for bearing design modification for SRS long-shaft mixing pump designs to eliminate fluid whirl and critical speeds which significantly increased the equipment service life.

In addition, vibration analyses and related analyses have been used during new equipment scale-up tests to identify the need for design improvements for full-scale operation / deployment of the equipment in the full size tanks. For example, vibration analyses were recently included in the rotary micro-filtration scale-up test program at SRNL.

INTRODUCTION

Vibration is the back and forth motion (oscillation) of an object about an equilibrium point [1]. It can be periodic such as the motion of a pendulum or random such as the movement of a car tire on a bumpy road.

Vibration in machines, for the most part, is undesirable, wasting energy, creating unwanted sound (noise) and causing periodic dynamic stresses in machine parts, which can lead to fatigue failures or unwanted contact between machine parts, causing wear or damage. Because of this, the control of vibration is an important part of machinery management.

Machinery vibration analysis concerns simultaneous measurement and control of the periodic motion of rotors, casings, piping, and foundation systems. Some common causes for machinery vibration and failure include:

- Mass unbalance in rotating parts such as shafts, couplings, fans, impellers, pulleys or gears
- Hydraulic imbalance in impellers from unequal spacing of vanes or fabrication techniques
- Wear/clearance problems at journal bearings or seals

- Rolling element bearing defects/damage, which can sometimes occur prematurely (before the bearing design life) due to excessive static/dynamic loads transmitted to the bearing
- Misalignment of machine components from bent/bowed shafts, parallel/offset shafts, cocked bearings, belts/pulleys, eccentric gears
- Foundation problems (soft foot or sprung foot which results in twisting of the motor or pump casing when mounting feet are torqued to the mounting plate, machine support deterioration or inadequate design, or thermal warping)
- Resonance of rotors, casings, supports and/or foundations, piping, and belts
- Instabilities for example, shaft bow due to rubs/contact between rotating and stationary machine parts, loose rotating parts, shaft cracks, cavitation in pumps, flow turbulence in blowers, shaft whirling in journal bearings, and/or fluid whirl inside pump casings
- Uneven friction due to insufficient lubrication, contaminates in lubricate, wrong lubricant for application.

Description of Vibration Analysis and Common Applications

Vibration analysis is a form of non-destructive testing to provide information about machine components. Interpreting vibration signals can be a complex process that requires experience and specialized training. Important vibration parameters that can be measured and interpreted include:

- Frequency – Number of times the structure or machine vibrates per unit of time (Hz),
- Displacement – Total travel of the mass/object during a vibration cycle, peak-to-peak (mm),
- Velocity – Speed at which a mass moves or is vibrating during its oscillation (mm/s),
- Acceleration –The rate of change of velocity with time of the object measured in g's (mm/s^2),
- Phase – How one part of the machine vibrates in relation to another part or to a fixed reference such as a once per shaft revolution tachometer pulse.

The simplest technique for analyzing vibrations involves examining the individual frequencies and amplitudes of a signal [2]. State of the art equipment is usually used to transform a complex vibration waveform (time domain) into discrete frequencies (frequency domain) using a Fast Fourier Transform (FFT) process. These frequencies correlate to specific vibrations from different causes such as unbalance or misalignment, and to specific mechanical components (for example, parts that make up a rolling-element bearing). Every component in a machine vibrates at its own, identifiable frequency. By examining the frequencies, and by knowing information about the machine, the vibration analyst can often identify the cause of an excessive vibration and sometimes the root cause of that vibration. For example, a degrading rolling element bearing will often exhibit specific frequencies corresponding to the natural frequencies of bearing components, such as the balls or the race containing the balls.

During the early stage/incipient phase of a rolling element bearing defect, random, ultrasonic frequencies are also present, which may be detected using a stethoscope or other sound detection equipment.

A typical example of analyzing a high vibration is the case of a measured vibration at the operating speed (synchronous (1X) rotating speed of a machine) due to a mass unbalance of an eccentric rotor, an eccentric pulley, a bent shaft, or shaft misalignment. For misalignment, this can be further divided into angular misalignment or parallel misalignment. Angular misalignment is typically characterized by high 1X and 2X axial shaft vibration with 180° phase difference across a shaft coupling. Parallel misalignment has similar vibration symptoms to angular misalignment except the high vibration readings are in a direction perpendicular to the shaft and approach 180° out-of-phase across the coupling. This simplified example illustrates the value of trained personnel to assess vibration complexities, such as phase and vibration direction to help differentiate between several potential problem sources.

Frequency analysis is only one aspect of interpreting information in a vibration signal. More advanced vibration analysis uses additional diagnostic tools, such as examining the: 1) time domain signal, 2) phase relationship between vibrating components and a timing mark on a shaft, 3) trends of vibration levels and phases, 4) vibration mode shape, and 5) rotor position. All of these important aspects need consideration to provide an accurate diagnosis. To do so, engineers and mathematicians have developed a number of diagnostic vibration plots for identifying rotating machinery problems in equipment with fluid film bearings.

These graphical formats include:

- Bode Plots – A pair of graphs, one for amplitude versus shaft rotational speed and the other for phase of the 1X, 2X, 3X, etc. vibration vectors versus shaft rotational speed.
- Polar Plot – A graph consisting of a center reference surrounded by concentric circles for plotting the phases and amplitudes of a set of vibration vectors. Phase is represented by the angular position relative to the transducer orientation. The amplitude is represented by the distance from the center of the plot.
- Shaft Centerline Plot – The average shaft centerline position of a shaft/rotor plotted in rectangular/Cartesian format. Centerline plots are made from trend data or machine startup and shutdown data. Shaft centerline plot information is usually combined with information about machine clearances, rotational orbit dynamic behavior, and centerline plots from other measurement positions to obtain a more detail picture of the shaft motion, available clearances, and radial loads acting on the machine.
- Cascade Plot – A series of half spectrum plots or full spectrum plots generated during equipment startup or shutdown over a range of speeds. A half spectrum is a spectrum of a single time base waveform. The full spectrum is derived from the waveform obtained from two, orthogonal transducers combined with the knowledge of the direction of rotation. The full spectrum is typically the spectrum

of a shaft orbit. Full spectrum plots are usually examined for diagonal, vertical, and horizontal relationships. Vibration frequencies that change with, or track running speeds, are usually plotted on "order lines" drawn diagonally from the origin (zero speed, zero frequency) to assist in quickly identifying frequencies equal to or multiplies of running speed (i.e, 1X, 2X, 1/2X, etc.).

- Waterfall plot – A series of half spectrum plots or full spectrum plots generated during constant speed operation over a period of time. These plots are frequently used to examine changes in machine vibration as a function of operating parameters.
- Orbit Plot – The path of the shaft centerline relative to a pair of orthogonal eddy current probes. A Keyphasor™ dot caused by a once per shaft revolution event is usually added for phase and precession information. The probe locations and direction of shaft rotation are also included on the graph.
- Trend plot – A graph used to display any kind of data versus time. Multiple parameters are often plotted on the same time scale to correlate changes that occur during machine operation. Types of information frequently graphed include: unfiltered vibration, filtered vibration, and proximity gap voltages. Sudden changes in trend parameters can signal a problem in a machine.

Common vibration analysis applications include:

- Troubleshooting increased equipment vibrations and part failures.
- Condition monitoring of critical or essential machines. The use of condition monitoring allows maintenance to be scheduled, or other actions to avoid failure. Some machines in this category include steam and gas turbines in the power industry, and cooling water pumps in the nuclear industry.
- Acceptance tests of new equipment designs or modified equipment designs, especially in applications where maintenance is limited because of hazards such as radiation levels or where the equipment is not accessible after it is installed.
- Process control applications where the measurement of acceleration, motion and shock provide valuable input for making equipment or system adjustments.
- Balancing of rotor systems
- Input for "tuning" rotor-dynamic models used to guide the design of machine modifications for improving equipment reliability.

Description of Mechanical Damage Due to Vibration in SRS Pumps

Long-shaft pumps: Long-shaft pumps deployed in the SRS HLW tanks are vertical mixer and transfer pumps with cylindrical journal bearings provided by different manufacturers (Quad-Volute Slurry Mixing Pump (QVSP), Lawrence Slurry Mixing Pump, Standard Slurry Mixing Pump, and Hazelton Telescopic Transfer Pump). These pumps suffer from rotor-dynamic instability problems due to the inherent lightly-loaded condition that the vertical orientation places on the bearings. Ultimately, these pumps failed from contact wear between the sleeve bearings and shaft after as little as a 1000 hours of operation. Bearing wear leads to increased cyclic stresses and changes in the rotor natural frequencies. Both conditions result in increased equipment vibration levels

and ultimate failure to the lower mechanical seal. Failure is declared when the amount of lubrication bearing water leaking from the pump support column results in too much dilution of the tank contents, typically 0.13 to 0.19 liters/second (2 to 3 gpm).

Sulzer Pump: The Sulzer mixer pumps used at SRS are approximately 13.7 meter (45 foot) long and consist of a 40.6 centimeter (sixteen inch) diameter water filled pipe column surrounding a 5.08 centimeter (2 inch) diameter shaft. Ten carbon journal bearings along the shaft stabilize its motion. The bearings are 1.52 meters (five feet) apart and the upper and lower bearings are located near the mechanical seals. The mechanical seals at the top and bottom of the pump contain the water within the bearing column. A carbon sleeve bearing with 3 axial slots is located inside the lower bearing housing.

The damage due to vibration is evident in Figure 1(a) and (b)**.** Wear patterns indicate contact between the shaft and bearings, followed by abrasion between bearing particles and the rotating surfaces. The bearings are not flushed, and this stagnant condition prevents adequate removal of any particles from the fluid film, thus accelerating further wear. The primary damaging vibration (for this equipment) was caused by impeller imbalance, but both whirl and resonance are problems with this style pump. Consequently, the reliability of this pump can only be increased by lowering the pump speed or installing tilt pad bearings and machined impellers.

(a) (b)

Figure 1. Bearing and shaft damage from contact on a disassembled SRS standard slurry mixing pump.

QVSP: A damaged mechanical seal from a QVSP is shown in Figure 2(a). This pump was installed in a HLW tank. A new seal assembly is shown in Figure 2(b) for comparison. The QVSP is essentially the same design as the SRS standard slurry mixing pump but has a larger hydraulic casing, impeller and discharge nozzles (jets). The average life for this style pump was approximately 1000 hours following installation [3]. A leak through the seal started with a small leak rate of 379 to 757 liters per hour (100 to 200 gallons per hour). The pump continued to operate with this leak rate until a sudden increase in leakage occurred. The pump was then shut down.

Two seal failure mechanisms were determined to have occurred. The initial seal failure occurred due to loss of lubricant when the water between the seal faces evaporated due to excessive heat caused by vibration. Essentially the mechanical faces ran in a near dry condition and the seal surfaces spalled and cracked. The other mechanism was due to radial vibration which caused gross damage to the bellows on the seal. To correct the problem on new pumps, the new impellers were machined to replace the cast impellers. Other corrective actions to new pumps include modifications of the shaft diameter to raise the critical shaft speed, intentional deflections (friendly misalignment) of the shaft to reduce whirl, and modifications to the bearings.

(a) (b)

Figure 2. Damaged mechanical seal removed from a QVSP installed in a SRS waste tank.

SRS VIBRATION ANALYSIS APPLICATIONS

Short Shaft pumps

Deployment of short shaft pumps for mixing SRS HLW tanks began in 2005 [4]. Some reasons for changing to the short shaft pumps include: 1) they do not require a bearing water system, 2) mechanical shaft seals can be eliminated, 3) a foot could be fabricated for the bottom of the pump to supported the equipment weight on the tank floor, 4) likely hood of increasing the mixing capacity for a pump that could fit through a 24 inch tank riser, and 5) potential for developing a pump that could be reused for several bulk waste retrieval campaigns.

The procurement for designing and fabricating these submersible mixer pumps (SMPs) was awarded to Curtiss-Wright Electro-Mechanical Division, Cheswick, PA. Acceptance testing at the SRS Full Tank Test Facility included vibration analysis. See Figure 3. A concern was identified during the early phase of testing when vibration sensors mounted at the lower radial bearing of the motor were occasionally driven into saturation by high frequencies. Consequently, lower sensitivity sensors and sensors to filter out the high frequency signals were installed for subsequent testing. During testing, one pump had a bearing failure. The corrective action for the failure included redesign of the bearings and fluid flow passages to increase lubricant flow in the bearing regions.

Vibration analysis during acceptance tests resulted in a reliable pump design. Six pumps have been used in SRS waste removal campaigns and, with the exception of one failure; the modified pumps have performed reliably. The pump failure was attributed to operation at the extreme end of the design curve in terms of temperature, speed and tank level while performing chemical cleaning with oxalic acid.

Figure 3. Submersible Mixer Pump during acceptance testing at the SRS Full Tank Facility.

Unmodified Long Shaft Pumps

At SRS, vibration analysis is used to 1) evaluate each pump intended for HLW service against historical base line information for that specific pump design and 2) identify operating speeds for each individual pump. Detailed vibration analysis of these HLW

tank pumps is warranted because of the high equipment cost and high cost of testing, installation, maintenance, and replacement.

Vibration analyses are performed on the following types of HLW pumps: QVSP, Hazelton Telescopic Transfer pump, and Standard Slurry Mixing Pumps (Lawrence Pumps, Sulzer Pumps) to extend service life by optimizing operating speeds. These are vertical pumps with lightly loaded bearings that have known design issues (mentioned earlier) primarily due to the light loading configuration.

During initial testing of SRS long-shaft pumps, vibration monitoring was limited to measuring velocity at the upper end of the pump and motor. This type of analyses does not provide an adequate representation of the potential for mechanical problems that can be encountered under operating condition.

QVSP: Severe shaft damage was observed at the lower end of the QVSPs even though limited vibration analysis indicated no excess vibration. Therefore, more extensive vibration analysis program which included monitoring velocities at axial locations along the pump support column below the mounting point was incorporated into the pre-deployment testing. In addition shaft displacement at the lower end of the pump was recorded. An optical Keyphaser™ was also installed at the motor/pump coupling to provide a reference event for making phase comparisons.

The testing sequence of these pumps proceeds as follows:

- A push-pull test was performed at the impeller end to determine the nominal assembled clearance where a pair of proximity transducers will be mounted.
- Velomitors® and accelerometers or pads for attaching accelerometers were mounted along the support column for measuring radial vibrations. At a minimum, a pair of sensors were located at 4 elevations below the mounting point.
- Data collection equipment was configured to capture synchronous and asynchronous vibrations.
- A static impact test was performed before running the pump to estimate the natural frequencies in the system.
- Speed series were conducted to collect startup and shutdown data.
- A steady state run was performed to trend vibration amplitudes and phases.
- A partial disassembly inspection at the lower end of the pump was performed, if warranted by machine history.

Information collected during vibration testing is screened for:

- Fluid induced instability at the journal bearings. Historically, this occurs at a frequency of 0.46X to 0.48X in SRS pumps and the precession is forward.
- Increasing vibration amplitudes (direct and filtered) and phase changes (for the discrete frequencies, e.g., 1X) during trending.
- Changes in the average shaft centerline position

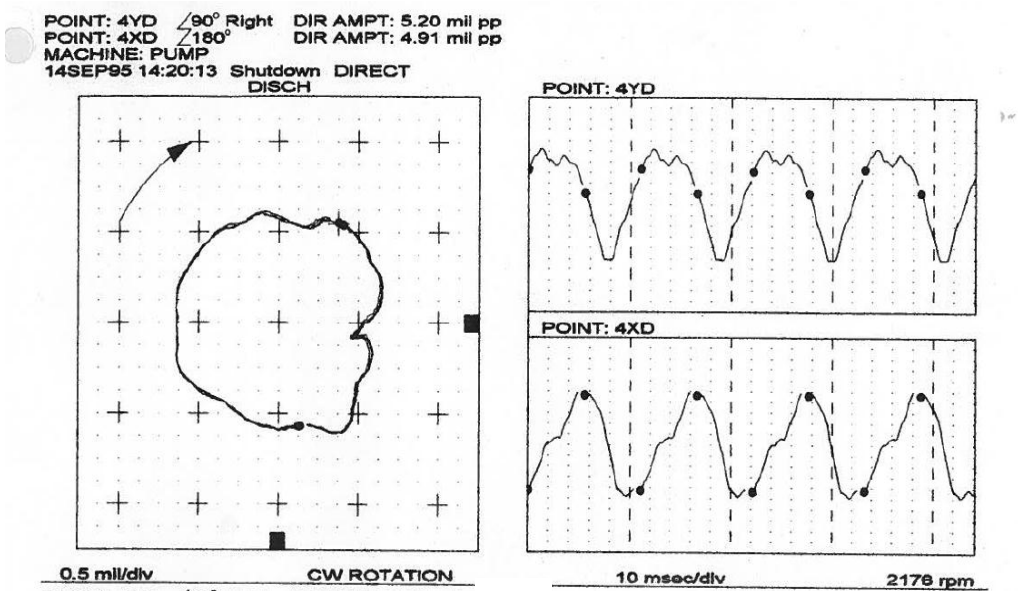
- Changes in orbit shapes and direction of precession
- Periodic 1X vibration changes (potential rub)
- Periodic or sudden change in 1X (loose part, potential rub)
- Significant 1X vibration at slow roll (rotor bow)
- Shaft whip
- High axial vibrations at pump casing (typical for impeller clearance issues)
- Impact pattern in the time waveforms
- Sudden changes in phase and vibration amplitudes during the speed series

A long-shaft pump, QVSP, is shown mounted on an overhead structural steel platform that spans a 25.9 meter (85 foot) diameter by 2.43 meter (eight foot) full scale one quarter depth HLW test tank. See Figure 4(a). The test facility has several platforms to allow equipment access for attaching vibration analysis sensors, Figures 4 (a) and (b). The locations of typical data collection points along the pump shaft are shown in Figure 4(c). The proximity probes near the bottom are installed in an X-Y configuration targeting the exposed shaft between the support column and pump casing, Figure 4(d).

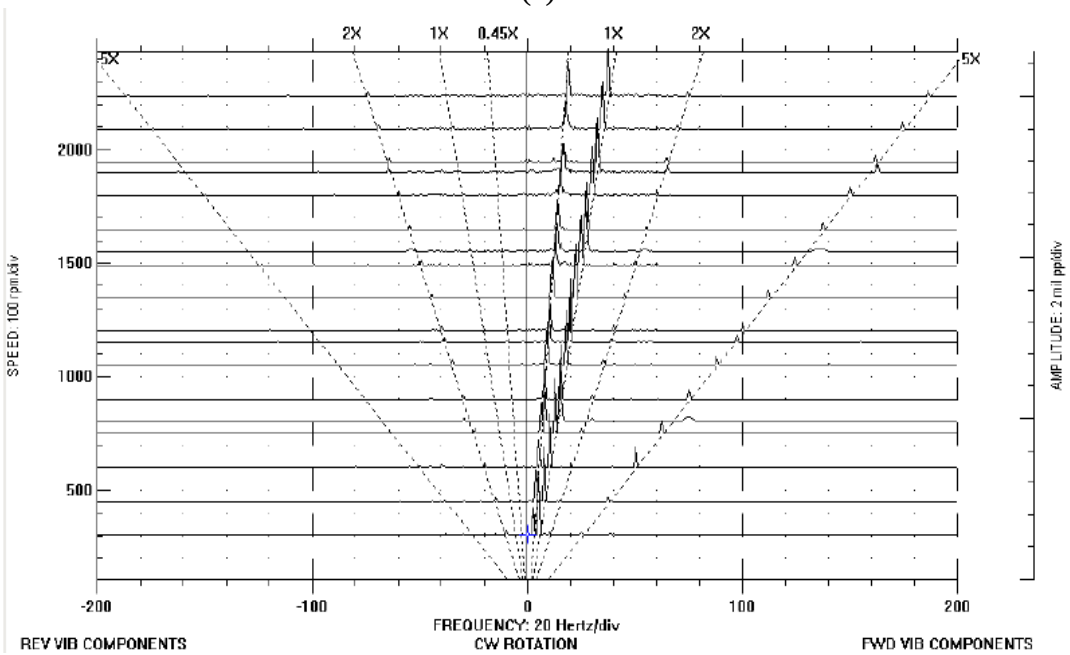
Vibration data from a QVSP is provided in Figure 5. The orbit time base plot and full spectrum cascade plot reveal a significant 1/2X component that is forwarding in precession. This vibration component will change to slightly less than 1/2X during the steady state pump run. The 1X vibration component was observed to be in forward precession during most of the test, except at the beginning when it was in reverse precession. The subsynchronous component in forward precession indicates fluid whirl. The subsynchronous component with reverse precession indicates a periodic rub mechanism or hydraulic influence from the submerged impeller.

The cascade plot in Figure 5 was created during the pump start up. The subsynchronous vibration as the equipment changed operating speed is visible to the left of the 1X order line. The vibration components tracking the 5X order line are due to the impeller and represent the vibrations from blade pass frequency.

Figure 4. Typical SRS long-shaft pump instrumented with vibration sensors during pre-deployment testing.



(a)



(b)

Figure 5. Long shaft pump with cylindrical journal bearings (QVSP) orbit time waveform plot indicating fluid whirl and potential rub (a), and full spectrum cascade plot (b) indicating fluid whirl .

Modified Long Shaft Pumps

Vibration analysis and rotor dynamic analysis have been used to modify existing SRS pumps to increase service life. Testing revealed that subsynchronous whirling, at approximately one-half the running speed, was the primary cause for long shaft pump failures. Examples are provided below:

Lawrence Slurry Mixing Pumps: Thirteen mixing pumps by Lawrence Pumps, Inc. (LPI) were delivered to SRS in 1996-97. Four of the pumps were installed in Tank 8-F and after about 1000 hours of operation began to leak excessive amounts of bearing water from the pump support column. The pumps installed were the best of eight tested (i.e., had the lowest subsynchronous whirl vibrations and lowest unfiltered vibrations).

In an effort to understand the cause of the observed problems, Tank Farm Engineering hired a consultant from No Bull Engineering Inc. to construct a rotor-dynamic model of the pump shafts and support column. Vibration data collected during pre-deployment testing of the pumps installed at SRS were used to tune the model. The results of the rotordynamic model were used to guide the design of new bearings to replace the original plain cylindrical journal bearings installed in the other 9 pumps, and thereby, resolve the rotordynamic instability problem [5].

The new bearing design for the LPI slurry mixing pump is illustrated in Figure 6. It is a four pad tilting pad bearing with pivot offset and preload. The new bearings were installed in the 9 remaining pumps which were deployed in Tanks 7-F and 11-H. Each pump was tested for 72 hours to collect vibration data in a similar manner as data collected for the mixing pumps installed in Tank 8-F (i.e., along the column and exposed shaft near the impeller).

Data analysis indicated that the modifications eliminated the subsynchronous 1/2X vibration along the shaft between the tilting pad bearings and nearly eliminated whirl vibrations in the region where the impeller is supported by one tilting-pad bearing and the process-lubricated journal bearing. This modification resulted in extending the service life of the pump by 10X (1000 to more than 10,000 hours).

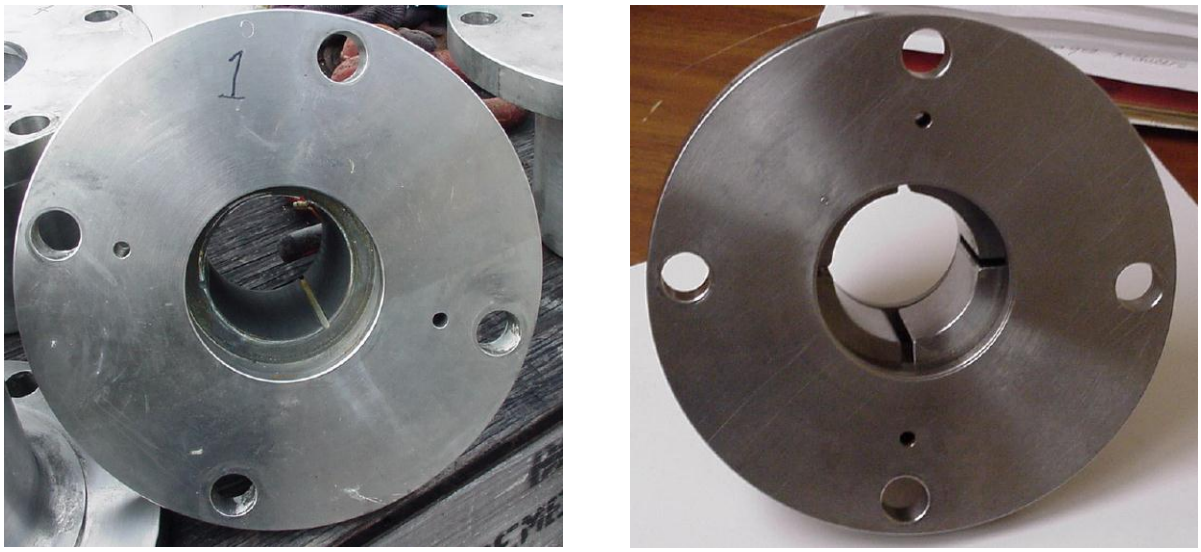


Figure 6. Unmodified Lawrence Pump bearing (a), and modified Lawrence Pump bearing (b).

Sulzer Quad Volute Slurry Mixing Pumps (QVSP): Because of the success in increasing the service life of the LPI long-shaft pumps, Tank Farm Engineering requested SRNL and No Bull Engineering personnel to create a rotordynamic model for the QVSPs and provide a new bearing design. Vibration from pre-deployment testing of the unmodified QVSP was used for model tuning. Modeling results guided the new bearing design.

The new style bearing for the QVSP is shown in Figure 7. The new design was a tilting pad bearing with four pads, pivot offset and preload. The new bearings have been installed in 3 QVSPs and at least one modified pump has been installed in a HLW tank.

Vibration data similar to those collected for the unmodified QVSPs were generated and analyzed. The data indicated that the modification virtually eliminated the near 1/2X vibration along the shaft between the tilting pad bearings and the vibration near the impeller. Unlike the LPI slurry mixing pumps, no process-lubricated journal bearing existed for this type of pump where the shaft penetrated the pump casing.

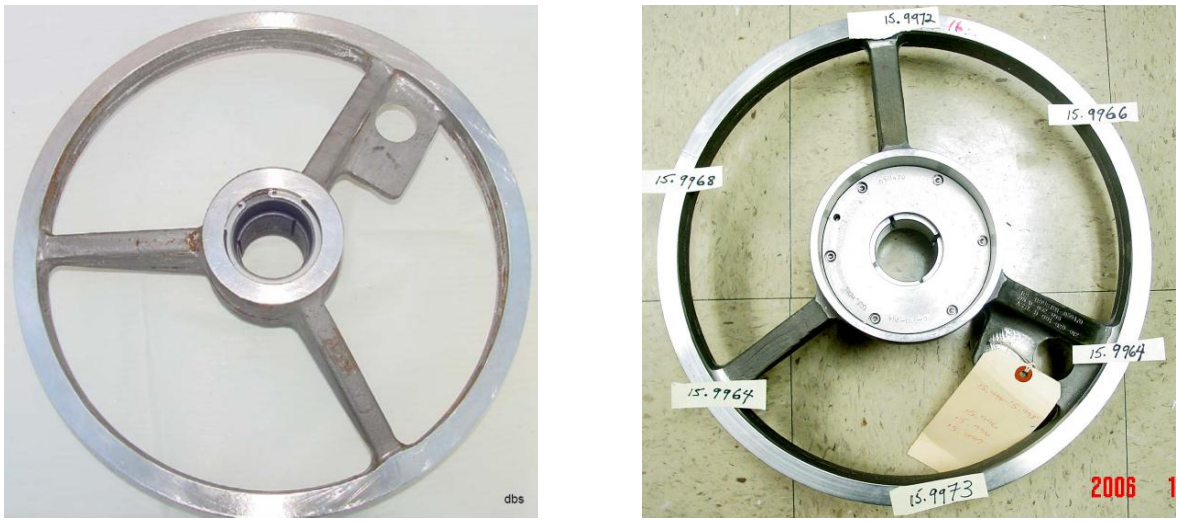


Figure 7. Unmodified QVSP spider bearing assembly which caused whirl vibration (a), and Modified QVSP spider bearing assembly which eliminated whirl vibration (b).

3H Evaporator Feed Pump

The feed pump for the SRS 3H evaporator is also a long-shaft pump with desired service life of 24,000 hours (3 years). The pump design consists of one rolling element bearing at the thrust location, 10 water lubricated journal bearing in the support column, and one process lubricated bearing at the impeller. Vibration testing on the replacement pump for the 3H evaporator indicated that the shaft relative vibration at the lower end of the pump matched the diameter of the process lubricated bearing. See Figure 8. This indicated that the process lubricated bearing at the bottom of the pump was ineffective at controlling the shaft motion. The problem was due to the large diametric clearance between the shaft and impeller bearing. The corrective action implemented for the problem was to tighten the clearance at the process lubricated bearing by adding a sleeve to the shaft.

Vibration analysis was an integral part of extending the predicted service life of this type of pump from 8,000 to greater than 18,000 hr. The pump is still in radioactive waste service and is expected to perform for at least the desired service of 24,000 hours.

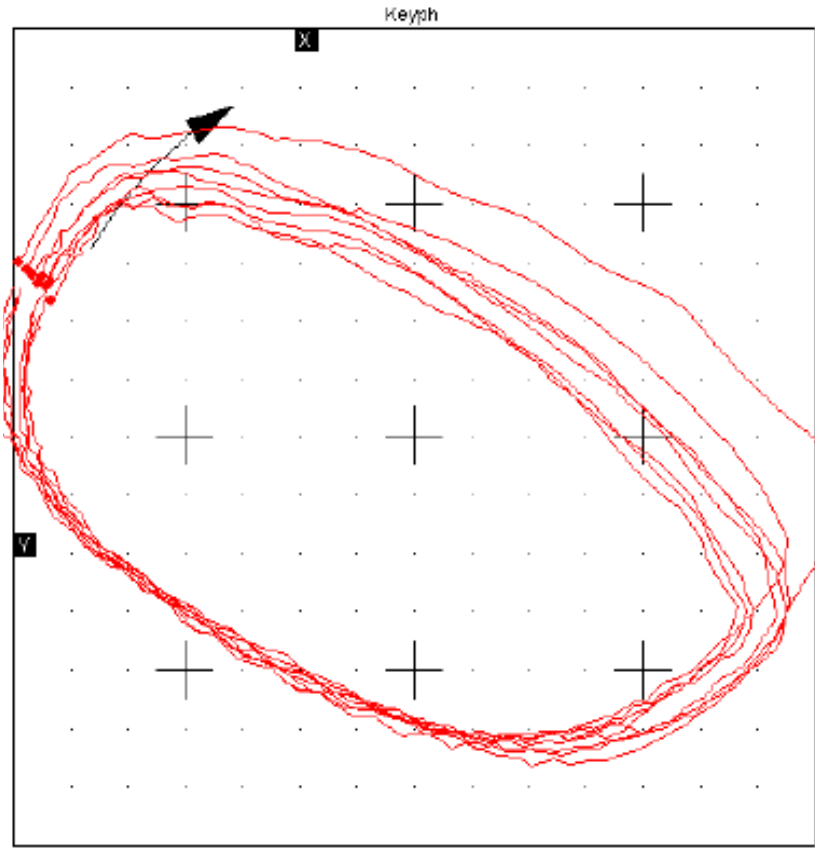


Figure 8. Lower Shaft Orbit for 3H Replacement Evaporator Feed Pump during Pre-deployment testing revealed clearance at process-lubricated bearing was too large. Two vibration probes (X and Y) are indicated on the plot.

Defective Pump Component Identification

Vibration analysis has been useful for identifying defective components. Two examples are provided below:

- Defective impellers in the QVSPs led to hydraulic imbalance for the replacement SRS Tank 51 slurry mixing pumps and were corrected before installation. These pumps have mixed flow impellers which were fabricated by a metal casting process. During the fabrication the molds shifted and caused unequal spacing of vanes. The vibration analysis on this pump was discussed earlier. The solution for this problem was to fabricate the new impellers by 5-axes machining.
- Defective upper mechanical seals in the LPI slurry mixing pumps. The defective seals were discovered as the result of mechanical testing of a modified LPI slurry mixing pump. The pump modification consisted of replacing the original equipment bearings with tilting pad bearings, and installing a new upper shaft seal. Vibration data were collected from the housing of a defective replacement upper mechanical seal. See Figure 9. The time waveform revealed cyclic/periodic impacting. The pump was shut down and upon disassembly of the mechanical seal, the clearance between the outer race of the rolling element bearing and seal housing was found to be out of tolerance. The corrective action for this malfunction was to machine a new seal housing.

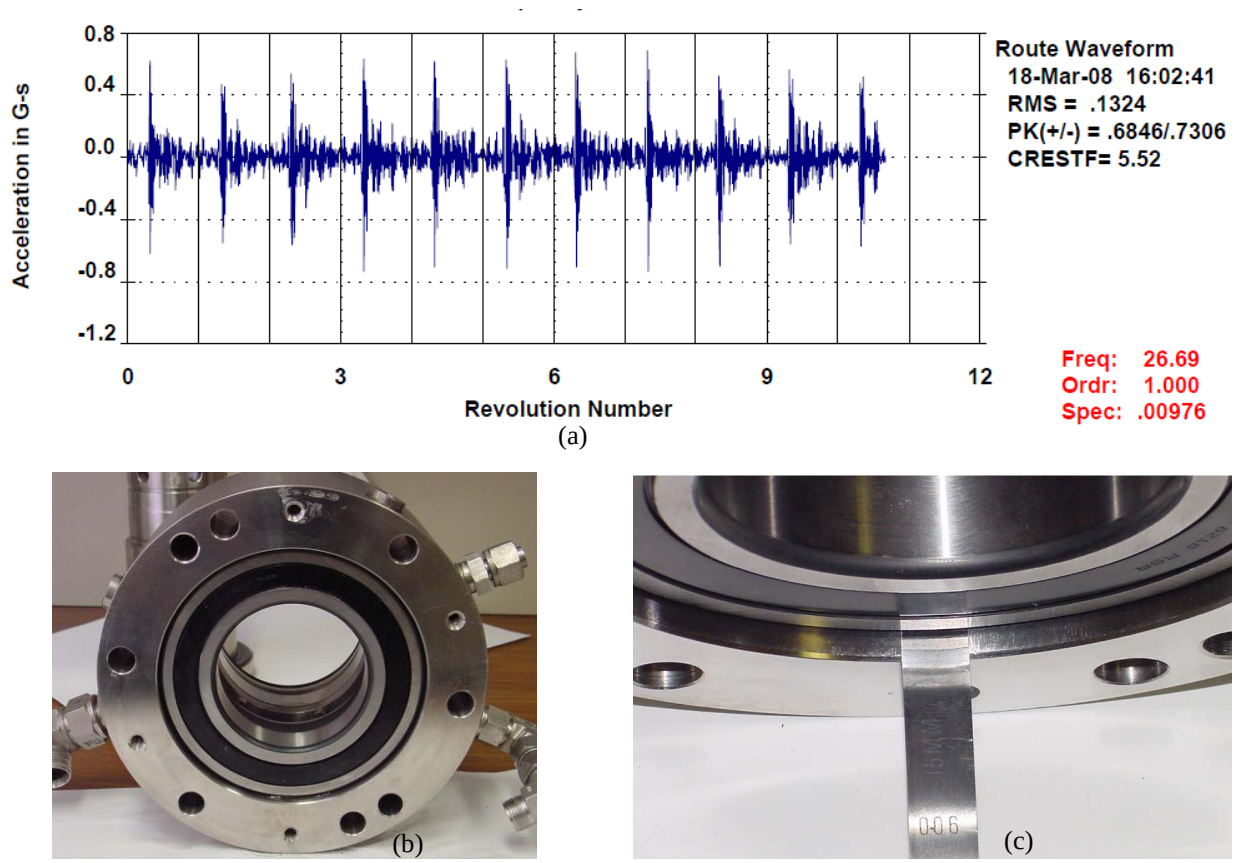


Figure 9. Time waveform for defective LPI pump which indicates impacting between bearing outer race and seal housing (a), Defective upper mechanical seal assembly (b), and tolerance problem (c)

Rotating Equipment Used for Radioactive Service

MCU Centrifugal Contactors: Initial vibration analysis of the CINC Model V-5 and Model V-10 centrifugal contactors fabricated for the SRS Modular Caustic-Side Solvent Extraction Unit (MCU) project were performed using casing mounted accelerators as part of initial factory acceptance tests. During early testing, a resonant vibration occurred in the V-10 unit at 2800 rpm which resulted in the unit failing the acceptance criteria [6, 7]. At the time, the failure was attributed to the long support legs in the test stand. Substitution of shorter, stiffer legs reduced the vibration to within the limits of the specification, allowing the tests to continue. The primary vibration always occurred at the equipment running speed, and the temperature of the V-10 bearing housing was noted as being hot, i.e., > 48.9°C.

During subsequent testing, the centrifuge was instrumented with displacement proximity probes and velocity seismo-probes. The proximity probes were installed in an X-Y configuration at five axial positions along the rotor; the velocity seismo-probes were installed at two axial positions. A Keyphasor™ probe was installed at the motor

coupling and provided a once-per-turn reference pulse. The predominant vibration component for the unit was at 1X running speed (synchronous) which indicated that the primary forcing function was due to unbalance. The maximum shaft deflection occurred at the axial position furthest from the motor (i.e., at the free end of the rotor).

The corrective action for the centrifugal contactors was to balance the rotors which resulted in reducing the shaft relative vibration from 50 mils to 14 mils peak-to-peak at the speed of maximum vibration (balance resonance at 2720 rpm). In addition, the casing vibration dropped below 1.5 mil peak-to-peak at the speed of maximum casing vibration (1750 rpm).

Rotary Microfilters: Vibrations were collected from the SpinTek Rotary Micro-filter (RMF) units during recent testing to develop control logic for multiple full-scale filter units and to characterize hydraulic behavior for deployment in the Small Column Ion Exchange (SCIX) system.

During testing, the RMF assembly developed considerable vibration as the temperature of the system increased during runs. This was attributed to the motors moving relative to the filters and changing the alignment between the two components. Unless addressed in the design, the gradual motor-filter misalignment will likely carry over to deployment, where operational, as well as ambient changes in temperature are expected to be greater than in the test configuration. SRNL personnel recommended that the motor be tied directly to the filter unit during deployment in HLW tanks to avoid the risk of equipment reliability problems [8, 9].

CONCLUSIONS AND RECOMMENDATIONS

Vibration analysis has been an effective tool for increasing service life of pumps and other rotating equipment at the SRS. Further use of vibration analysis can yield significant cost savings when installing new equipment. Vibration analysis is extremely useful for: 1) condition monitoring of critical or essential machines, 2) scheduling maintenance, 3) identifying design and fabrication flaws, and 4) evaluating new systems. Significant cost savings have been realized as the result of vibration analysis of HLW pumps and separations equipment at the SRS. For example, the use of tilt pad bearings on long shaft pumps resulted in a 15 million dollar cost savings that was documented using the Six Sigma process.

Vibration analysis should be performed on pumps and equipment prior to deployment in a HLW tank because of the risks of equipment failure and resultant high costs. Vibration analysis should also be performed on all other costly rotating equipment intended for deployment in a hazardous or high radiation environment. Detailed knowledge of each piece of equipment is essential for performing accurate vibration analysis. The proper placement of sensors, selection of transducers and data collection equipment, and understanding of parameters and expected operating conditions are required to maximize the benefits of vibration analysis.

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