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Proceedings of the Eighth NRC/ASME Symposium on Valve and Pump Testing

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Washington, DC
July 12-14, 2004

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Abstract

The 2004 Symposium on Valve and Pump Testing, jointly sponsored by the Board of Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the U.S. Nuclear Regulatory Commission, provides a forum for exchanging information on technical and regulatory issues associated with the testing of valves and pumps used in nuclear power plants. The symposium provides an opportunity to discuss improvements in testing that help to ensure the continued reliable performance of valves and pumps. The participation of industry representatives, regulatory personnel, and consultants provides for a broad spectrum of ideas and perspectives to be discussed regarding the improvement of testing programs and methods for valves and pumps at nuclear power plants.

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Statements and opinions advanced in the papers presented at the Eighth NRC/ASME Symposium on Valve and Pump Testing are to be understood as individual expressions of the authors and not those of either the American Society of Mechanical Engineers or the U.S. Nuclear Regulatory Commission.

The papers have been copy edited and recast into a standard format. By consensus, English units have been used as an expression of current industry practice with metric units also indicated where possible.

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Session 1(a): Pumps

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Precursors to Cracked or Bent Shafts in High Pressure Centrifugal Charging Pumps

Shawn D. Comstock
Rick D. Raymer
Wolf Creek Nuclear Operating Corporation

Introduction

Certain high pressure centrifugal charging pumps supplied to the Nuclear Industry have a history of problems with shaft bending and cracking. These pumps are relied upon for High Pressure Safety Injection (HPSI) for small break loss of coolant accidents. The Pacific Pumps division of Dresser Industries originally supplied the pump that is the subject of this paper to the nuclear industry, a Pacific 2.5 RLIJ 11 stage-charging pump. The techniques used at Wolf Creek Nuclear Operating Corporation (WCNOC) to diagnose vibration results and identify the precursors to a cracked or bent shaft are the main focus of this paper.

Initial Problem

In October 2000, the outboard bearing housing horizontal vibration test point exceeded the ASME *Code for Operation and Maintenance of Nuclear Power Plants* (O&M) calculated alert range. A number of adjustments were made on the pipe hangers, u-bolt clamps and pump hold down bolts. As a result of these changes, vibration values fell below the alert range. In December 2001, vibration values began to trend upwards again. By August 2003, the outboard bearing housing horizontal vibration test point once again exceeded the ASME O&M calculated alert range. We made addition adjustments on the pump hold down bolts. Once again we were successful in reducing the vibration values below the alert range. However, the most recent step change in vibration performance on the pump outboard bearing in the horizontal direction caused WCNOC a great deal of concern. Considering we were preparing to begin our 13th refueling outage, we had to take another close look at the data that lead to the most recent change. A number of issues led us to make the recommendation to replace the pump rotating assembly.

Callaway is Wolf Creek's sister plant with nearly an identical design and plant layout. One of Callaway's HPSI pumps failed with a cracked shaft in 1992. Extensive troubleshooting was performed in an attempt to identify the cause of the vibration increase and to determine if shaft bending or cracking precursors were present.

Data Analysis

Evidence indicated that a resonance condition was responsible for some of the vibration problems we were experiencing. When equipment operates within 20% of its natural frequency, normal vibration can be magnified exponentially. The closer equipment operates in relation to its natural frequency, the greater the magnification of normal vibration. Refer to Figure 1 - "HPSI Pump Resonant Condition Vibration Magnification Factor" to see how changes in the natural frequency of the machine affect vibration at the pump's operating speed of 80 hertz (Hz). Figure 1 reflects a prediction of vibration performance with relation to this specific application using the formula obtained from reference 1.

Resonance Basics

Resonance is simply the natural frequency of a component or combination of components (assembly). All structures have a resonant frequency. If you impact the structure with enough force to make it move, it will vibrate briefly at its natural frequency. A structure will have a resonant frequency in each of its 3 directional planes (x, y and z, or as we call them, horizontal, vertical and axial). Resonance serves to *amplify* the vibration due to whatever vibration force is present at (or near) that resonant frequency. It is important to note that resonance does not cause vibration - it amplifies it.

Critical Speed Basics

A pump shaft has what is referred to as a "bending mode". The point at which the turning speed of a pump shaft matches its natural frequency is called its first critical speed. When a pump is at the first critical speed, the shaft bends in the shape indicated by the figure below. Most pumps operate above their first bending mode. Furthermore, the duration that a pump spends in this region of operation as its rotational speed accelerates is so short that the vibration at this point goes unnoticed. Good design practices ensure the operating speed is greater than 20% of the shaft rotational speed.

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Pump Shaft First Bending Mode

At the pump's second critical speed, the shaft bends into an s-shaped curve as indicated below.

Pump Shaft Second Bending Mode

Like a structural resonance that is being excited, the pump shaft vibration amplitude will be magnified as shown in Figure 1. The critical speed of a shaft is determined by its mass and stiffness. Normal wear results in increased looseness between parts, thus changing the critical speed of the shaft.

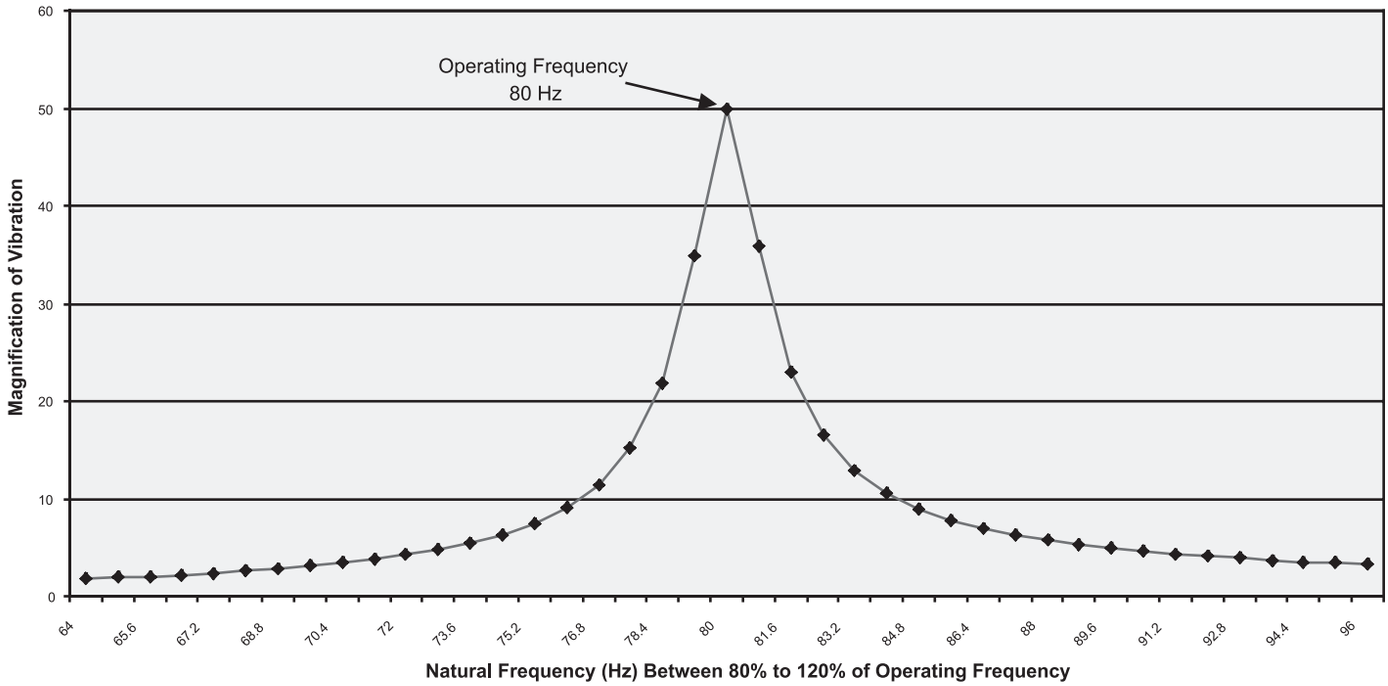
HPSI Pump Application

Based on information from Flowserve, the pump shaft 2nd critical speed is approximately 84 hertz. As a result of normal wear and subsequent loosening of mating parts, the critical speed of the shaft begins to decrease. In this particular component, as the critical speed decreases it becomes closer to the operating speed of the pump. Likewise, the vibration amplitude of the pump shaft increases as a result of amplitude magnification. We also discovered that the pump outboard bearing housing had a natural frequency of approximately 84 Hz. Based on historical spectra, impact and modal data, we suspect that normal wear allowed the pump shaft amplitude to increase enough to excite the natural frequency of the pump outboard bearing housing. Notice how the actual vibration results in Figure 2 closely mirror those predicted by the resonant amplification curve in Figure 1.

Corrective Action Strategy

If left unattended, resonance problems typically get worse until a catastrophic failure occurs. However, it is possible to change the stiffness or mass of a structure to shift the resonant frequency away from the operating frequency. Resonance is a magnifier of vibration, not a cause, so it is

Figure 1 - HPSI Pump Resonant Condition Vibration Magnification Factor



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also possible to reduce the vibration problem by improving vibration performance. Another factor in resonant vibration is the amount of dampening that is present. Increasing the dampening factor results in a reduction of the resonance magnification. Based upon the industry data at hand and the pump's original vibration performance prior to the step change, it was determined that de-tuning and dampening were the most effective means to improve the situation. Piping adjustments and pump outboard frame hold down bolting adjustments were performed which successfully changed the resonant frequency, resulting in vibration performance below the surveillance alert range. Figure 2 - "HPSI Vibration Performance" shows a trend of the vibration levels at various points.

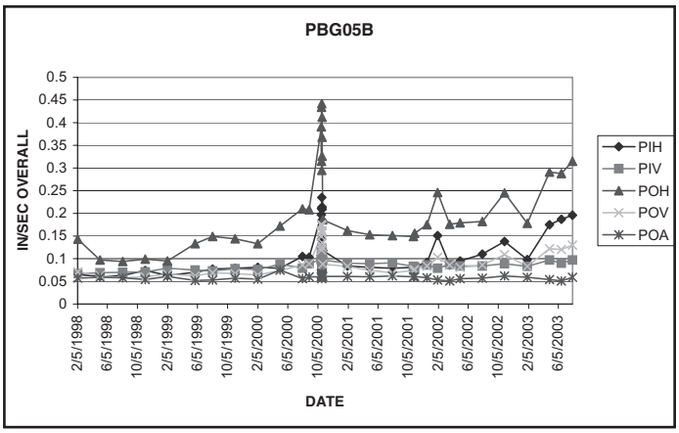

Figure 2 - HPSI Vibration Performance

The point descriptions in Figure 2 correspond to pump inboard horizontal (PIH), pump inboard vertical (PIV), pump outboard horizontal (POH), pump outboard vertical (POV) and pump outboard axial (POA).

Industry Data

In follow-up to the successful detuning and dampening of the outboard structure of the pump, an investigation was performed to determine the extent of the condition and to predict future performance for maintenance planning purposes. Data obtained from Callaway and the Institute of Nuclear Power Operations (INPO) Equipment Performance and Information Exchange (EPIX)[1] revealed a history of vibration and shaft failure problems with this model of pump at several other sites.

A search of the Nuclear Plant Reliability Data System (NPRDS) system using the keywords "Pacific Pumps" identified 573 hits. Other keywords were used to narrow the search and, after review, 12 records were identified that had descriptions of problems that closely matched those at Wolf Creek or shaft failures as indicated by Table 1.

A search of all EPIX records on the keywords "Pacific Pumps" revealed 670 hits. Other keywords were used to narrow the search and 3 informative records were found. Record number 558 from Byron 2 was very informative. It is dated 5/15/2003 and describes an event where both of their charging pumps failed over a relatively short period of time. Byron's B train pump failed with a broken shaft on 11/11/2002. Byron's A train pump subsequently failed due to high vibration on 2/25/03. This report also identified an industry trend with 34 pump failures out of 122 total pumps; 26 of which were failures due to cracked or broken shafts. Table 2 reflects the failure types described in the search.

***Table 1* – NPRDS Reports of HPSI Vibration or Shaft Problems**

Plant Unit	Discovery Date	Failure Cause Category	Problem
Salem 2	6/18/1988	Unknown	High Vibration
Salem 2	7/15/1988	Unknown	High Vibration
Beaver Valley 1	7/19/1977	Engineering/Design	Broken Shaft
Beaver Valley 1	7/29/1994	Age/Normal Usage	Shaft Failure
Connecticut Yankee 1	9/10/1992	Manufacturing Defect	High Vibration
Sequoyah 1	2/15/1991	Unknown	Cracked Shaft
Callaway 1	4/10/1991	Unknown	High Vibration
Callaway 1	2/3/1992	Unknown	Broken Shaft
North Anna 1	9/21/1983	Unknown	High Vibration
North Anna 1	8/29/1989	Unknown	High Vibration
North Anna 1	4/29/1991	Age/Normal Usage	Bent Shaft
North Anna 2	8/26/1983	Engineering/Design	Bent Shaft

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Table 2 – **Chemical and Volume Control System (CVCS) Industry Pump Failures**	
Failure Type	**Number of Failures**
Complete shaft failure	20
Cracked Shaft	6
Bent Shaft	3
Pump Seizure	5

Recent NRC Data

NRC Information Notice 2001-06, "Centrifugal Charging Pump Thrust Bearing Damage Not Detected Due To Inadequate Assessment of Oil Analysis Results and Selection of Pump Surveillance Points,"[2] describes a pump bearing damage event with this model of pump. A 40-fold increase in the oil particulate count was observed prior to the bearing failure. No change in vibration performance was identified. This incident highlights the importance of not relying upon a single predictive maintenance technology for the determination of pump condition.

Performance Monitoring

Based upon industry data relating to resonance problems, it was decided that condition monitoring of the vibration, phase angles (or direction of vibration) and structural resonance was an effective strategy for assuring the pump's operational capability. The purpose of this approach was to monitor and predict the rate of pump degradation for maintenance planning. This enabled a prediction of vibration performance to tell when in the future the pump would reach the point that its performance would become a concern in relationship to its safety function mission time.

Between October of 2000 and April of 2003, there were two other spikes in vibration as indicated by Figure 2. These were determined to be related to system flow conditions and temperature of the process fluid. Subsequent testing verified that this was a temporary condition not related to an increasing trend in vibration performance.

New Variable Introduced

During April 2003, a modification to replace the lubrication piping to the bearing housings was implemented. This modification replaced hard piping with high-pressure flexible hose to mitigate problems with leakage in the oil system thought to be related to the vibration of the pump. The post maintenance tests identified that this changed the outboard bearing housing's natural frequency to 83.45 Hz, nearer to the operating frequency of the pump at 80 Hz. This resulted in an increase in vibration that exceeded the ASME O&M Code Alert range. The test frequency was doubled in accordance with ASME O&M requirements while a new effort began to detune and dampen the structure.

Using the year 2000 maintenance history from this pump and the modal analysis model developed in 2003, a new action plan was developed. The new action plan would change the structure's response to excitation. Based on the modal model, relative movement was occurring between the pump in the area of the outboard end hold down bolts and mating support structure. Data collected on 8/1/03 included an impact test on the outboard bearing housing. The 8/1/03 impact test identified that the resonant frequency of the outboard bearing housing increased from 5007 cycles per minute (CPM) to 5040 CPM. This shifted the resonant frequency away from the operating frequency of the pump and should have resulted in lower vibration readings. Instead of the expected response, the vibration at both the outboard and the inboard bearing housings increased. This is the first time that the equipment has not responded as expected following changes in operational characteristics.

Wolf Creek subsequently increased the hold down bolts to 425 foot-pounds to eliminate the looseness that was thought to be contributing to the step change in vibration performance. Although vibration performance improved at PIH and POH on the 8/25/03 test after increasing hold down bolt torque, the levels did not return to the normal range. Additionally, no bolts were found loose and were at the as-left torque used to improve performance in the past. The pump hold down bolts were now at the maximum torque allowed by the vendor.

This change in performance closely represents the characteristics described by industry operating experience prior to the occurrence of a cracked or bent shaft. With a refueling outage only a month and half away, the decision was made to add work to the outage scope and replace the rotating assembly.

Insights

Industry data revealed that problems with these pumps have plagued the industry. These pumps are not operated at their best efficiency point (BEP) during normal operation or full flow check valve testing. At Wolf Creek, the normal flow for the safety-related charging pumps is at about 50% of the BEP. This undoubtedly has contributed to an increased wear rate. Minimization of run time is an effective strategy to ensure long-term reliability. Many plants have replaced their non-safety related positive displacement charging pumps with a more reliable centrifugal charging pump, including Wolf Creek and Callaway.

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With this particular model of pump, a step change in vibration performance is an early warning sign of degrading vibration performance that can quickly lead to a cracked or bent shaft. However, a step change is not always noticed before failure as described by EPIX/NPRDS reports. Under these circumstances, the crack or bend is initiated on the other side of the "heavy spot" of the shaft. A shift of vibration phase angles can identify the beginning of a crack or bending when this is the situation.

Bump testing enables the determination of structural resonance. This was an important test that enabled us to rapidly pinpoint the structural resonance problem. Taking these readings while the pump is known to be operating acceptably for comparison in the future can provide valuable insight about what may have triggered a step change in vibration.

High-speed data acquisition to measure the critical speeds of the shaft during pump start up is another approach that can identify problems with the rotating element leading to a pump failure. Vibration increases at the turning speed when each critical speed is reached as the pump speed increases. The vibration/revolutions per minute (RPM) data can be compared over time to see if the critical speed of the shaft is changing; thus indicating a problem with the internal rotating elements.

Modal analysis can be utilized to better understand how the structure is vibrating. Modal analysis of our B train pump showed looseness between two mating parts, even though the bolting in the area was at the maximum torque. Understanding how this structure vibrated enabled a more effective plan to de-tune this resonance and dampen vibration.

Oil Analysis has been proven as an effective means to monitor this type of pumps bearings for damage. NRC Information Notice 2001-06 describes the details of this event.

Wolf Creek has had numerous problems with the B train pump and very little problems with the A train pump. Callaway has experienced this phenomenon as well. The run time on Wolf Creek's pumps has been approximately the same over the life of the plant. Wolf Creek's B train rotating assembly was replaced in 1997 and lasted until 2003. The A train pump has never had its rotating assembly replaced.

These two pumps differ in physical piping design. For example, the B train recirculation line is a schedule 160 pipe, as opposed to the A train pump which has a schedule 80 pipe. Therefore, the B train pump operates at a slightly lower recirculation flow than the A train pump. Therefore, the B pump experiences a higher normal wear rate due to low flow operation than the A pump. Vibration measurements on discharge piping in the area during troubleshooting on the B train were compared to the A train. The B train piping vibration was significantly higher than the A train piping. This information suggests that the system piping design may play an important role in the resonant sensitivity of the pump. As stated before, any vibration is significantly magnified by structural resonance. Minor initiating events such as increased looseness due to normal wear, changes in process fluid temperatures, different flow configurations, small changes in rotational balance, minor shifts in alignment, and minor changes to stiffness of the structure can lead to a high cycle fatigue situation. Maintenance can be performed to improve performance and correct these problems, but each time this situation occurs the ability of the component to withstand this fatigue is lessened.

Flowserve Corporation, formerly Ingersoll-Dresser Pump Company, is the manufacturer of Pacific RLIJ, 11-stage, centrifugal pumps. Flowserve provides replacement parts for these pumps with upgraded materials and improved designs. Wolf Creek chose to replace the outboard bearing housing with an improved design provided by Flowserve. This design does not have the structural resonance problem of the original equipment. Flowserve also offers upgraded rotating elements that use stiffer materials to address operation near the shaft's second critical speed.

Conclusions

Predictive maintenance technology is very important for ensuring the long term reliability and performance capabilities for these pumps. These pumps are more sensitive to changes in vibration performance and structural integrity than most other safety-related pumps as evidenced by the number of industry reports of degradation and failure. An effective condition monitoring strategy, minimization of run-time, and upgrade of critical parts is recommended to ensure trouble-free operation.

References

1 – www.inpo.org - Institute of Nuclear Power Operations Web Site

2 - NRC Information Notice 2001-06, "Centrifugal Charging Pump Thrust Bearing Damage Not Detected Due To Inadequate Assessment of Oil Analysis Results and Selection of Pump Surveillance Points"

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WHAT HAPPENS WHEN A PUMP NO LONGER OPERATES AT OPTIMUM CONDITIONS?

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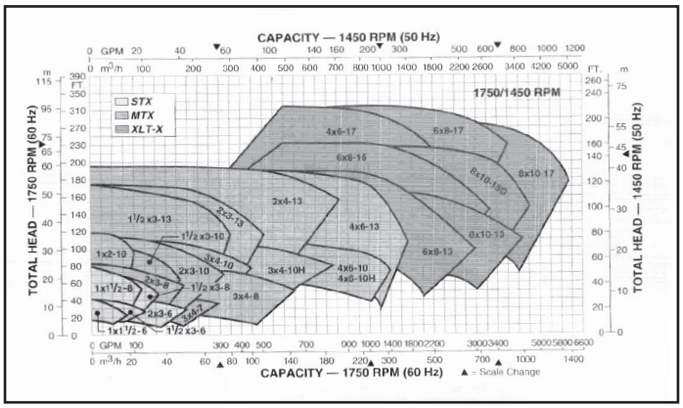
Abstract

With renewed attention to energy conservation, efficiency and equipment reliability, it is important to review implications of pumps operating to the left of the best efficiency point (BEP). Low efficiency, high radial loads, noise, vibration - become a real problem when that happens. Damage to the seal, shaft, couplings and poor reliability are a real and direct result of such operation. This paper examines the effect of pumps operating at off-design condition, with regard to efficiency and energy consumption.

Introduction

Let's explore the effect of such operation on *pump efficiency* and estimate wasted energy. There are also significant negative effects on radial load, cavitation damage, and other aspects, but, at this time, we will focus on the issue of energy only. The larger the pump, the more energy is wasted when a pump operates off-peak. A full range of ANSI pumps for chemical services, for example, as offered by pump manufacturers, may consist of many sizes, to span a wide range of flows – with larger sizes reaching over 4000 gallons per minute. Even larger sizes, such as encountered at power generation utilities (boiler feed pumps, circulating, screen wash, etc.) would have this issue even more emphasized.

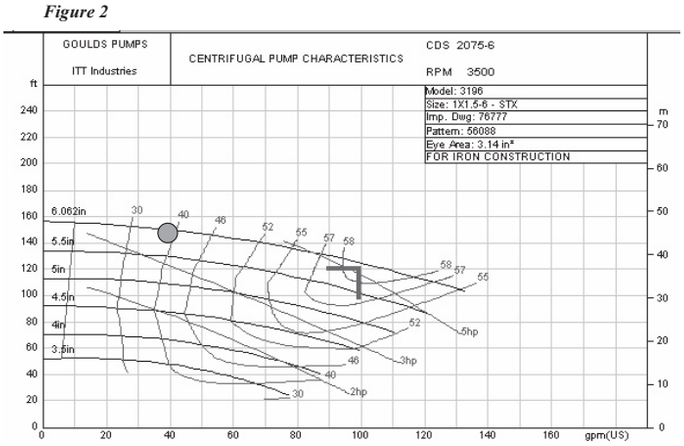
Hydraulic Coverage for a typical pump type range



The graph above is called an Overall Hydraulic Coverage Chart. For each size, a head-capacity curve at the maximum and minimum impeller diameter is plotted (sometimes, minimum diameter is not shown in order to make the chart less cluttered). This allows one to make an approximate selection of a pump size, and then to look up the individual detailed hydraulic performance curve for that size, to finalize the details.

Let's consider a case of a relatively small pump first. For example, if a pump user is looking for a pump to pump 40 gallons per minute (gpm) at 140 feet head, a 1x1.5-6 pump size (with approximately 6" impeller diameter) would be picked. The pump will work, but unfortunately will not be operating at its optimum design point.

Figure 2



As evident from the hydraulic curve for this pump size, this pump will have 40% efficiency (circle). Its design point (angle), however, is at 58% efficiency, i.e. the pump operates to the left of its BEP point for the impeller diameter required to achieve the desired head.

In this example, note that a horsepower (hp) line that passes near the operating point is approximately 4 hp, which is roughly 3 kilowatts (kW). How much does it cost to operate this pump if running continuously, 365 days per year, at, say $0.07 per kilowatt-hour?

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$$3\ (kW) \times 24\ (hr/day) \times 360\ (days/year) \times 0.07\ (\$/kW\text{-}HR) = \$1814$$

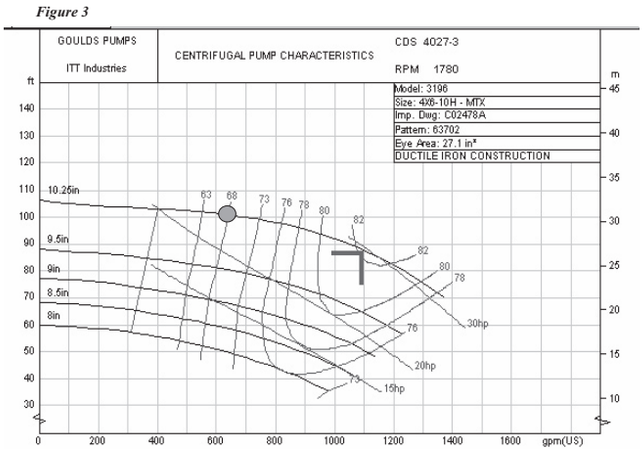
Now, what would it cost if the efficiency was somehow improved to the same 58% that this pump enjoys when operating at the design point? Obviously, if a pump runs more efficiently, it will take less power. In fact, the power (and thus cost) would be inversely proportional to efficiency:

$$1814 \times (40/58) = \$1251$$

The net savings would thus be 1814 – 1251 = $563, which is 31% less

Next, let's consider a somewhat larger pump. Say we have a 4x6-10H size operated at 600 gpm and producing 100 feet of head:

Figure 3



Again, the pump is off the efficiency peak. It operates at approximately 65%, whereas its peak efficiency at that diameter (10.25") could be 82%! Now, the energy dollars become more pronounced. Its power consumption is approximately 25 hp (19 KW), according to horsepower lines in the proximity to operating point:

$$19 \times 24 \times 360 \times 0.07 = \$11{,}491$$

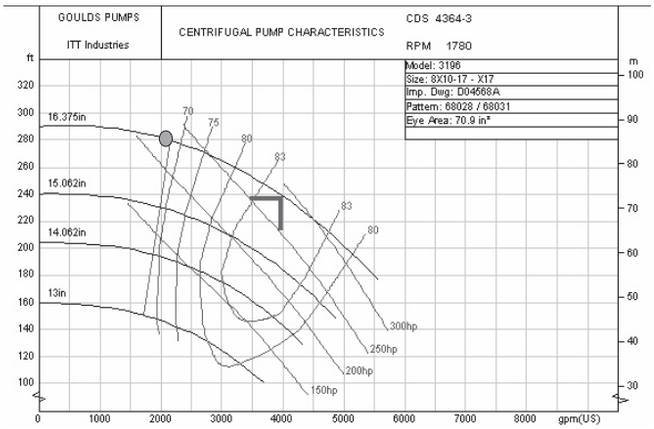
But it would be less if efficiency was restored to the designed 82%:

$$11491 \times (65/82) = \$9{,}108$$

The savings would be: 11,491 – 9,108 = $2,382, i.e. about 21% in this case

Let's next take even larger size, 8x10-17:

Figure 4



Let's assume this pump operates at 2000 gpm (280 feet head), instead of a peak point of 4000 gpm. The efficiency at the actual operating point is only 70% instead of the potentially achievable 83% by this pump.

The horsepower at the operating point is roughly 225 hp (168 KW), and the yearly energy bill is:

$$168 \times 24 \times 360 \times 0.07 = \$101{,}516$$

At restored efficiency, this would be:

$$101{,}516 \times (70/83) = \$85{,}616$$

The net savings would be:

$$101516 - 85616 = \$15{,}900$$

For larger sizes, the energy savings could be even greater.

As you can see, the net savings depend on how far back away from the Best Efficiency Point the pump operates. Unfortunately, this problem exists in all too many actual installations in the field. Many pumps, procured and installed years ago, often no longer operate at the originally intended hydraulic conditions. As operating conditions change, the pump is simply throttled further and further away from the BEP. The result – dollars literally "burned", - not to mention other problems (high loads, shaft breakage, etc.).

Obtaining a smaller pump is one approach. But, a smaller pump may still not (and usually does not) have the hydraulics sized to hit the operating point "dead on". It may help somewhat, but is expensive and not as efficient. The user choice is limited only to the pump sizes available, as standard, from the pump manufacturer's catalog, and even with a large number of sizes in the catalog, it is virtually

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impossible to cover each and every variation of the operating conditions. So, the user is forced to settle for the "second best", but not the optimum. More importantly, however, is the issue of economics and feasibility of *piping* change, to accommodate a proposed pump downsizing. Piping changes alone can often cost more then a pump.

Sometimes a better solution might be to have a new impeller, custom-designed and sized for your operating conditions. By doing that, a pump performance will essentially "shift" or "slide" to exactly where the Best Efficiency Point is, - and the net losses become zero. Such approach is effective, and the investment is minimal, with a payback of less than a year, and often just a few months.

Not only ANSI single stage overhung-impeller pump designs can benefit from this approach. Cooling water between-bearing pumps are known to have benefited greatly with improved impeller hydraulics. When a metal impeller is replaced with structural engineered composite material (80% lighter then metal), the combined effect of hydraulic fine-tuning with reduced weight (and thus load) can be dramatic. Rotordynamics benefits of such approach are obvious, and savings immediate. Other pump types, such as vertical multistage river intake pumps, condenser, circulating, etc. can have similar issues, and could be likewise retrofitted with improved hydraulics designs, - quickly, efficiently, and economically.

Conclusions

If you suspect that your pump is not operating at the optimum conditions, have it evaluated for the potential energy savings upgrade. Obtain your pump's hydraulic curve and indicate the desired operating condition. Have the potential energy savings evaluated, as a function of your operating conditions in relation to the actual pump BEP point. Then examine the evaluated costs, and impact of rotordynamics, and consider engineering recommendations, provided by your technical team, or an outside consultant. You may be surprised how much money you may potentially save.

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Consulting and Troubleshooting

Send your comments to:

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New Monitoring Technology for Pump Health

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Abstract

Power plant owners and operators are concerned with the "health" of their active components – particularly their pumps. Pump health at power plants is typically determined by the combined assessment of a number of system parameters. No single parameter provides an indicator of pump health, but engineers can incorporate multiple parameters into an overall health assessment.

Advancements in data processing and graphical presentation have significantly improved the ability of engineers to monitor system parameters. Quickly and easily an engineer may graph the pressure in a system over a given period and compare it to other parameters recorded over the same interval. The ability to graphically view parameters on the same time scale allows the engineer to correlate and assess component health.

Each of these activities requires time to apply the skill and judgment of a qualified engineer to the correlation of parameters, and each correlation is based on judgment. Therefore each conclusion is subjective. However, with the advent of the newest monitoring technology these correlations can be pre-programmed into a monitoring tool that constantly monitors these parameters and alerts the engineer only when the parameters indicate degradation. If this tool were also able to automatically "learn" data patterns from individual data streams in order to employ pattern recognition technology the accuracy of the tool would be significant. This would result in a highly accurate, objective, and continuous pump health assessment that effectively becomes "health by exclusion" – i.e. pump health is assumed unless the monitoring tool alerts the engineer.

This paper will describe the fundamental elements that make up such a monitoring system and describe the advantages that the system provides to the power plant professional performing pump health assessment. The paper will also note the potential future applications of such a system and some of the potential hurdles to implementation.

Introduction

Appropriately evaluating the parameters measured during component testing can be a challenge. However the combination of patented signal analysis algorithms and data processing technology can provide a solution for monitoring the health of components that operate routinely or continuously. This approach to monitoring the health of components starts with signal analysis algorithms that allow any chosen group of signals to be "modeled" by a methodology known as similarity-based modeling. A model is a recreation of the pattern of the group of signals that is compared to the signal outputs in real-time. The comparison is the difference between the model (called the estimate) and the actual signal. This difference is called the residual. As any signal (or signals) deviates from its normal pattern the residual shows certain patterns of behavior that provide early warning of degrading condition.

The component health analyst can program the system to provide alerts when the residual exceeds certain thresholds for certain levels of sensitivity. These alerts notify the analyst when a component or signal should be investigated for degradation. When a number of assets are monitored in this manner the health analyst allows the system to monitor the health of those components. Properly set up this system results in "health by exclusion" in that the analyst responds only to those components and signals that deviate from normal behavior and cause alerts.

Feedwater (FW) pump data from the Arizona Public Service Palo Verde Nuclear Generating Station are used to provide an illustration of this monitoring method as embodied in SmartSignal's Equipment Condition Monitoring (eCM) system.

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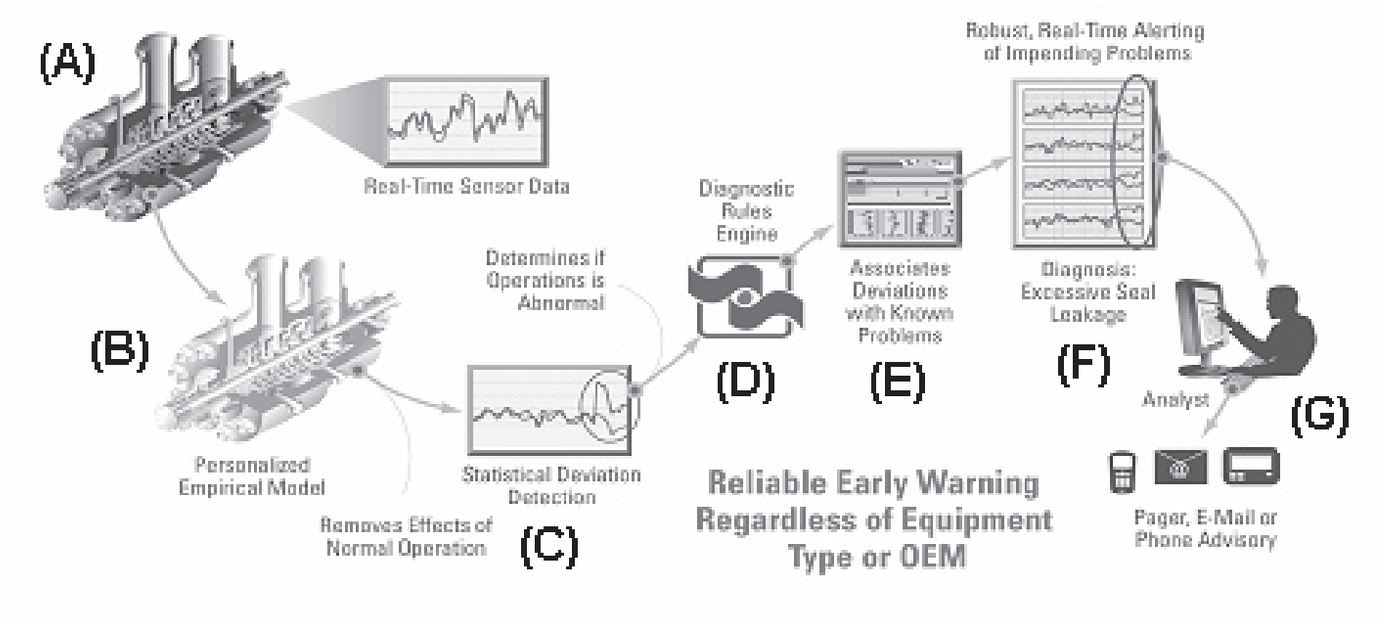
Monitoring Architecture

The monitoring approach described employs similarity-based modeling. Figure 1 illustrates its components.

Figure 1 – Monitoring Architecture



The monitoring system "flow" starts with real-time sensor data collected by a data historian (i.e. plant computer), **(A),** for a system or component. The sensors are chosen to represent the key correlating input and output values. The data is fed into a separate server where a personalized, empirical model captures the patterns and relationships of the group of signals **(B)**. A brief overview of this model is provided in the next section. The monitoring process compares the actual signal to an estimate of the normal signal behavior to generate a residual signal – i.e. the difference between the normal and the estimated behavior **(C)**. As any signal (or signals) within a group begins to deviate from its "normal" behavior the residual will demonstrate that a statistical deviation is occurring. The ability of this method to detect individual signal deviations in a group of signals provides early warning of degraded conditions.

Early warning is enhanced by the diagnostic rules engine of the monitoring process **(D)**. The diagnostic rules engine alerts the analyst when specific conditions (rules), programmed by the analyst **(E)**, are met. In the case of pumps, analysts can write rules that focus on failure mechanisms and rely on the sensor to provide early warning that a degraded condition exists. By reflection, when the system is in operation the analyst may determine that an active component is not degraded based on an absence of alerts. The diagnosis is directed to the analyst by means of a web-based application **(F)**. The analyst can then review and evaluate the equipment status **(G)** and contact the appropriate individuals.

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To briefly explain the monitoring architecture shown above, the major steps an analyst would follow using this process are described below.

Training

The heart of the monitoring system is the ability to model a group of signals by recreating the patterns found in a set of signals during normal operation. The analyst imports data and then designates "good" operating data to train the model. The diagram below illustrates the screen where the analyst chose "good" data for training. The dark columns indicate ranges of data the analyst has chosen for training the model and can be adjusted by simple "point-click-drag" operations.

The analyst is working with a group of sensors. In this case the sensors are the pump speed, flow, suction and discharge pressures. The green vertical lines indicate the data chosen for training.

The sensors are chosen to represent the key correlating input and output values of the system. For example, the discharge pressure remains nearly constant but the pump speed increases when the suction pressure lowers to compensate for the additional work necessary for the required head.

The behavior of each sensor shows some correlation to the behavior of the other sensors within that group. The analyst chooses which sensors to include within the model to capture the range of correlated behavior that indicates good or healthy operation. The program then analyzes the patterns of correlation found within the designated training regions to generate estimates of the behavior of the group. Once training data has been selected, as shown above, the analyst tests the model.

Testing (Good Data)

The diagram below demonstrates the results of a test of the model shown above. Due to variation in correlation and normal process deviation a model will not perfectly match actual signals. If the analyst determines that his model does not work well enough then another cycle of designating "good" data ensues and is repeated until the model is satisfactory. Once satisfied, the analyst saves the model and activates the real-time data feed. At that point the model is "on-line" and monitoring equipment. However, the analyst will need to create or modify the alerts that the system provides.

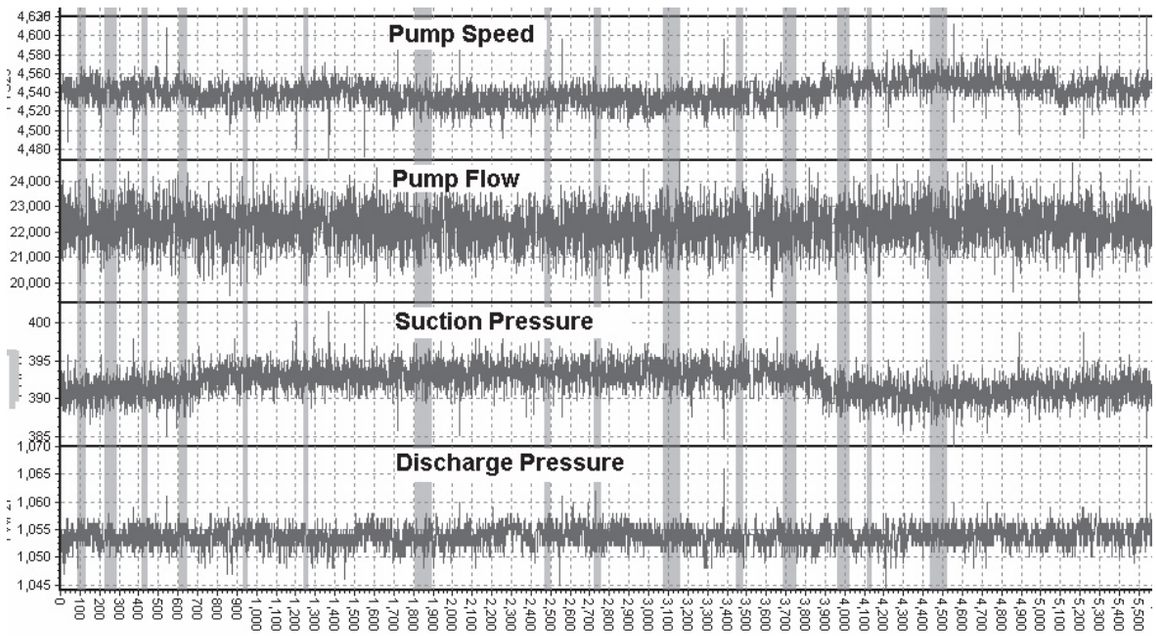

Figure 2 – Training Data Chosen for FW Pump Model
[X-axis = sample #, Y-axis = magnitude of signal
highlighted columns = data chosen as training data]

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In Figure 3, an actual signal (darker = blue) is overlaid by the estimated signal (lighter = green) for the same pump parameters. In the next Figure, 4, the residual (difference between the actual and estimate) is displayed for the same parameters. Note that the residual generally distributes normally about zero. When the residual shifts from this distribution the analyst can conclude that something has changed. In some cases a sensor or data feed has a problem. In other cases the signal is indicating degradation of a component. Notice also that the magnitude of the residual is significantly smaller than the magnitude of the signal itself. For these four signals the magnitude of the residual runs between 0.2% and 2% of the signal which gives much greater sensitivity to identification of changes in the signal.

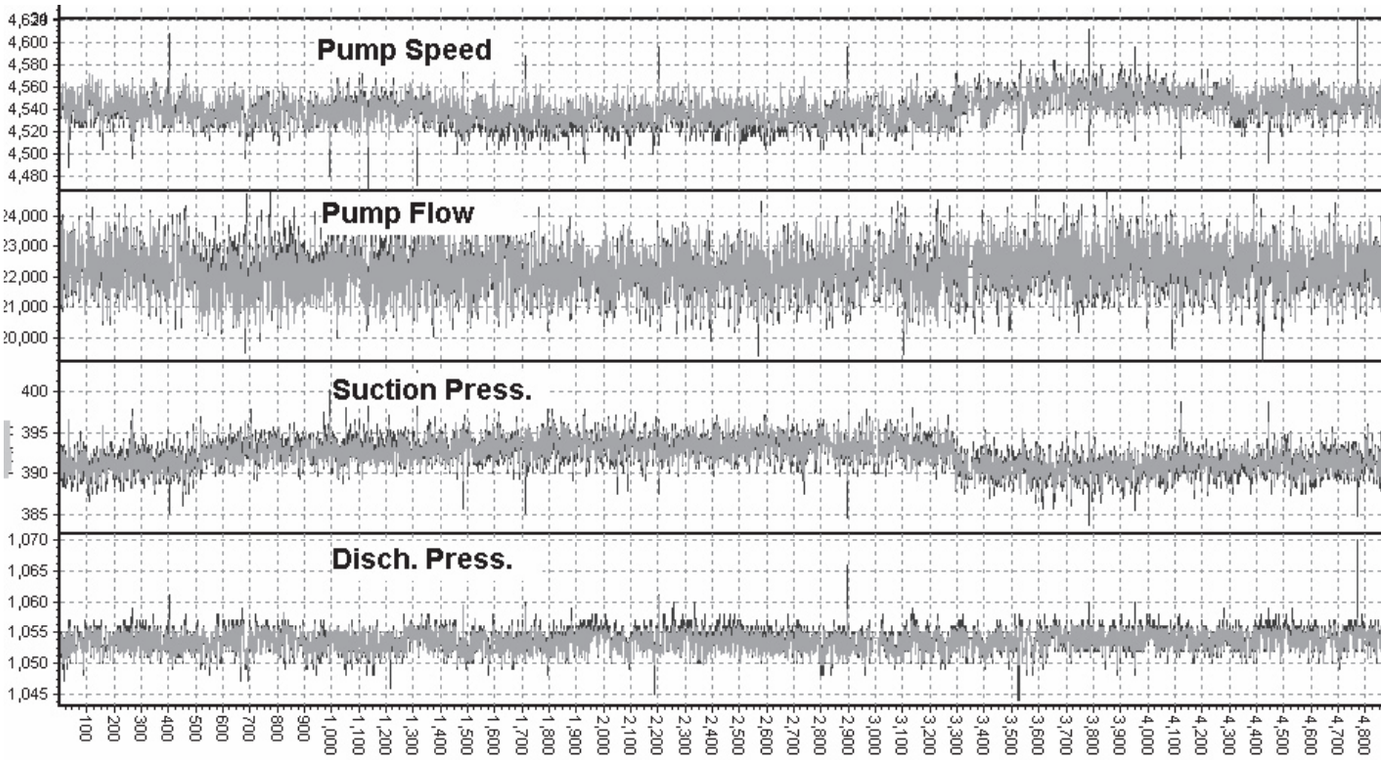

Figure 3 – Test Results for FW Pump Model (good data)
[X-axis = sample #, Y-axis = magnitude of signal,
dark line = actual signal, light line = estimated signal]

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Testing (Faulted Data)

In a simple test of the sensors shown in Figure 4 a total 2% increase was incrementally added to the pump speed over an eight-month period, see Figure 5. The dark line at the top of the Pump Speed chart indicates that the sensor exceeded the pre-programmed threshold and would have provided an alert to the analyst. The threshold is variable, and the analyst can choose a more sensitive level. The alert occurred at about a 1.35% speed increase. Note that this approach detects faulted behavior when pump speed has increased from approximately 4550 rpm to 4600 rpm – typically undetectable due to the signal magnitude. This is about a three-sigma deviation of the individual signal.

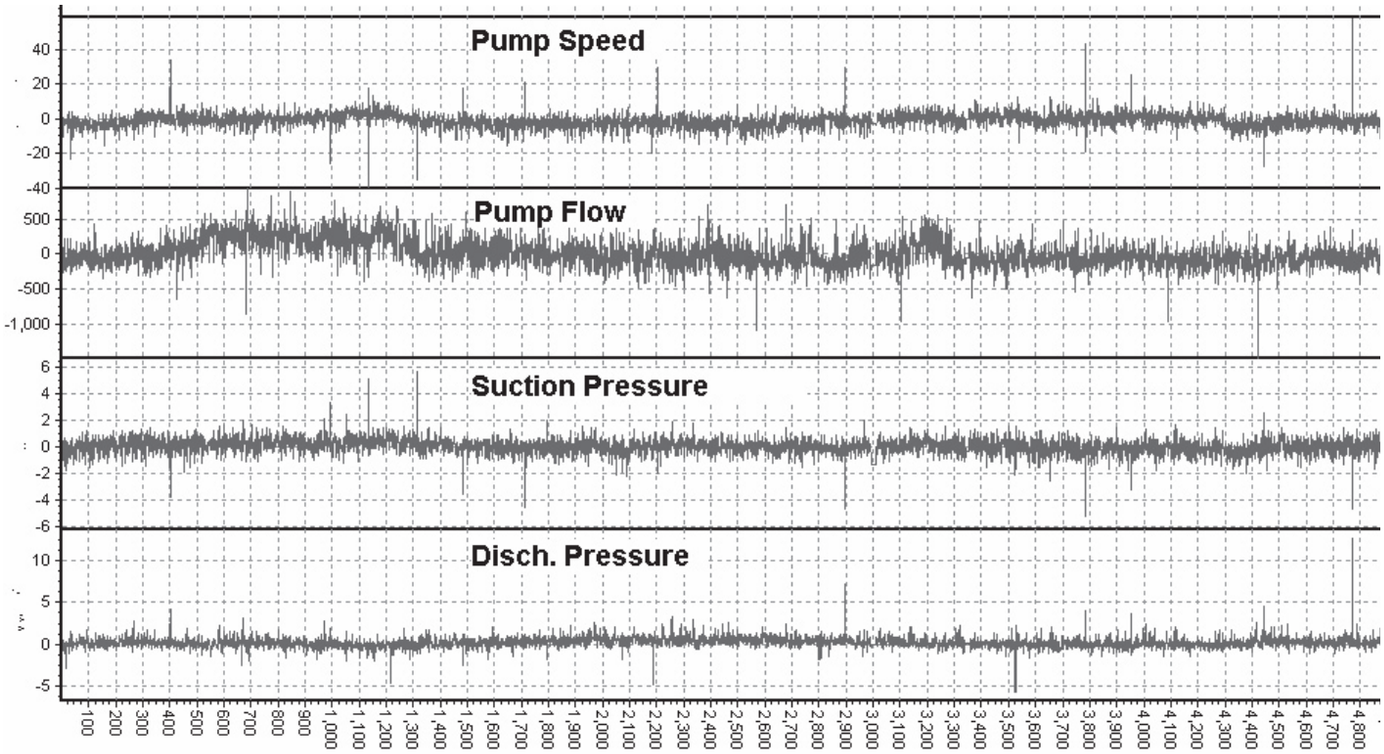

Figure 4 **– Test Results (residual only) for FW Pump (good data)**
[X-axis = sample #, Y-axis = magnitude of residual, dark line = residual]

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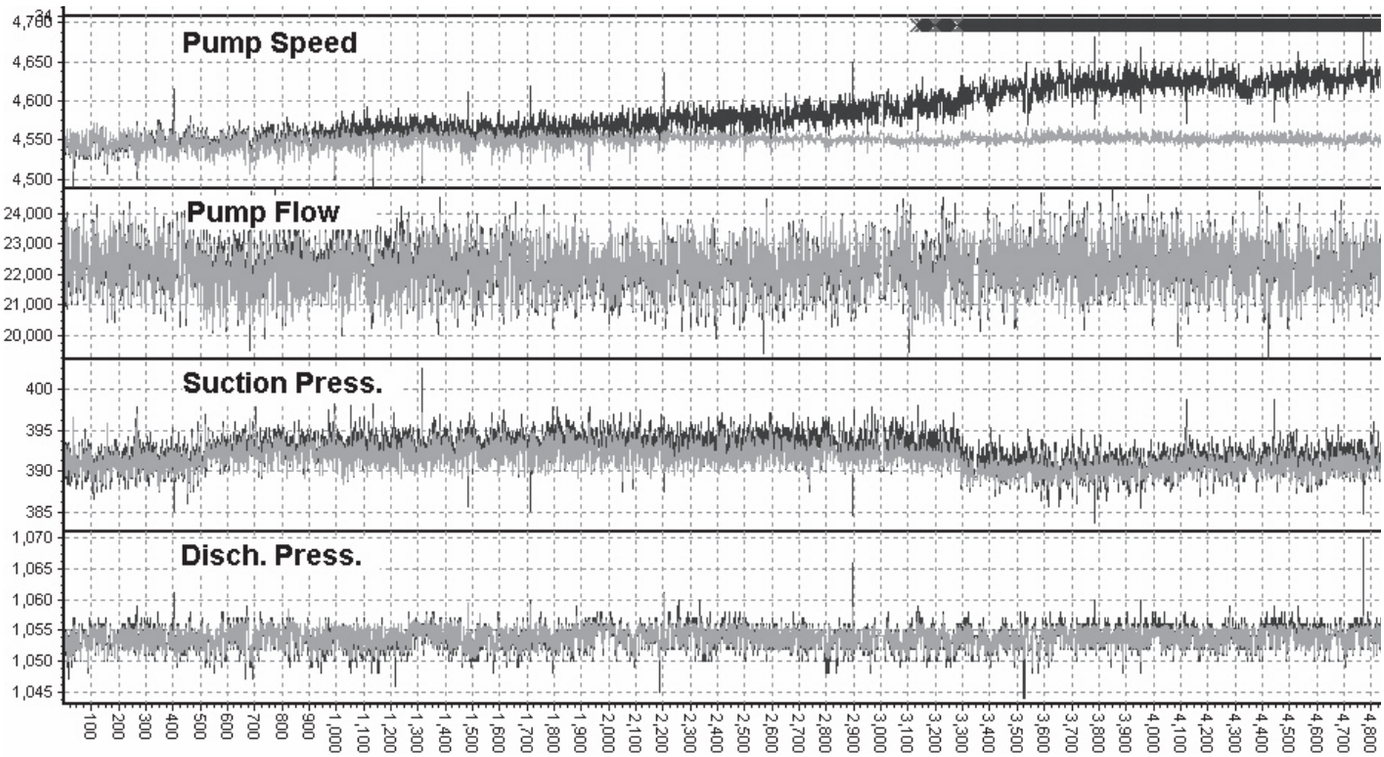

Figure 5 **– Test Results for FW Pump Test (speed increase)**
[X-axis = sample #, Y-axis = magnitude of signal,
dark line = actual signal, light line = estimated signal,
dark line at top-right of chart = alert]

For the same signals (same faulted test) the residual chart for the pump speed is shown below. The residual shows a steady rise with time and does not exhibit a normal distribution about zero.

This demonstration is simple and could be more complex. Only one signal is faulted, and the value rises at a steady rate. The alert level is fairly high and could be programmed to alert earlier. More complex models may include more sensors than shown for this model. Furthermore, the analyst may program the diagnostic rules engine to focus on multiple signals that highlight a specific failure mechanism.

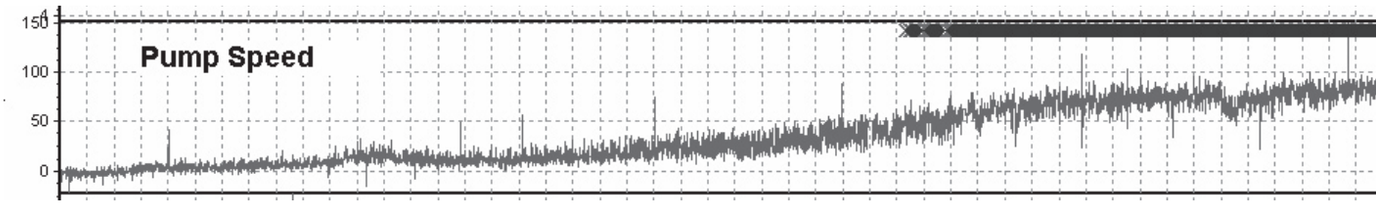

Figure 6 **– Test Result (residual only) for FW Pump Test**
[X-axis = sample # (not shown), Y-axis = magnitude of residual,
dark line = residual, dark line at top-right of chart = alert]

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Diagnostic Rules

The analyst creates and edits rules that alert the analyst or operator to conditions requiring attention. An example of rule logic is shown below

Rule #1 Name: Pump Motor Winding Hot

Rule is true when:

Pump Motor Winding Temp > Threshold

And

Rule #2 is false

Result: Post alert to watchlist

Rule #2 Name: Pump Motor Problem

Rule is true when:

Pump Motor Winding Temp > Threshold

And

Pump Motor Amps > Threshold

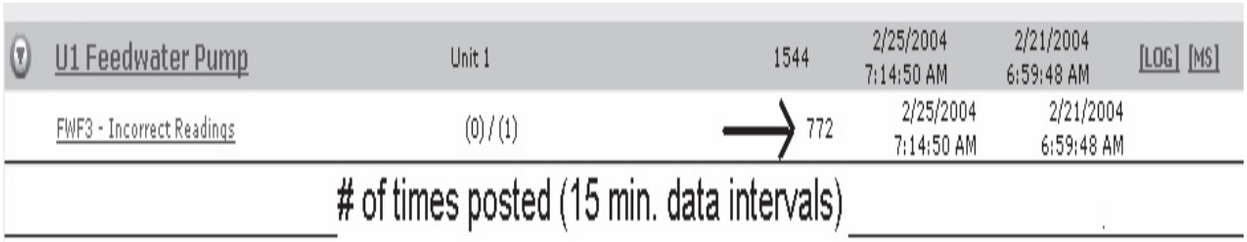
Result: Post Alert to watchlist

The result of any rule "posting" is a listing on a "watchlist", a web-browser-based viewer that provides access to charts of the sensors. In the example shown, rules have been combined to provide more meaningful information to the analyst and operator because a single sensor indicating behavior outside the range of normal behavior could have several causes. However, by not allowing that rule to be true by associating it with another rule presents a more meaningful alert to the analyst or operator.

Figure 7 illustrates a "watchlist" posting for a FW Pump. This example is provided to explain how the watchlist is used by the analyst, but it does not incorporate the rules identified above. A diagnostic rule which causes a posting is shown below the machine monitored (see Figure 7), and analysts can view both the number of posts and first/last date of the postings.

When a rule posts, the watchlist is automatically updated. When the analyst clicks on the rule, a page opens with the respective sensor graphs. Figure 8 shows the result of a posting to the watchlist for the rule shown above.

Figure 7 – Watchlist View for FW Pump



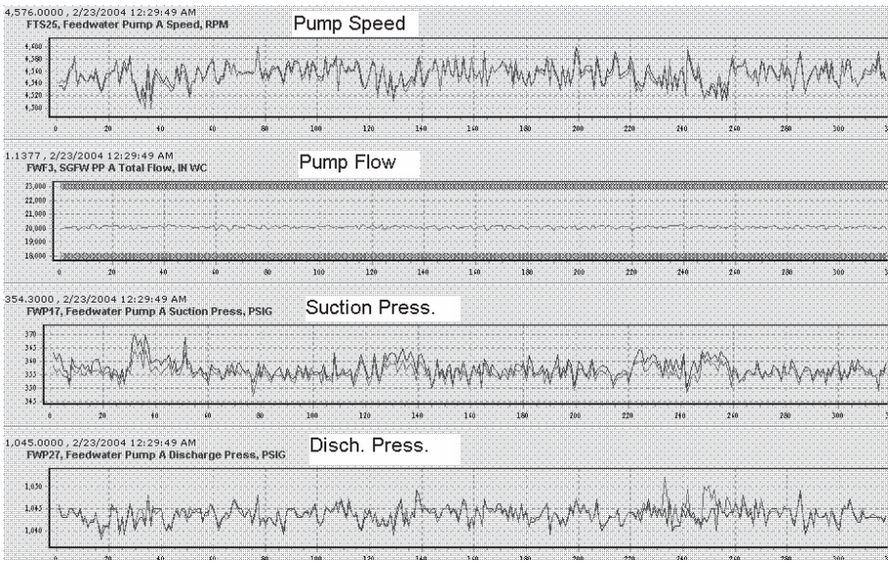


Figure 8 **– Watchlist Chart View for FW Pump**
[X-axis = sample #, Y-axis = magnitude of signal
dark line = actual signal, light line = estimate]

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Summary and Conclusions

A monitoring process that employs similarity-based modeling has been very briefly described. An analyst creates empirical models involving multiple sensors by the following steps:

1. Import data to train a model
2. Select the data that designates "good" operation
3. Test the model (reselecting if needed until the test is satisfactory)
4. Create diagnostic rules to generate alerts

This monitoring process provides certain distinct advantages for monitoring active components at power plants. The patented signal analysis algorithms provide accurate estimates of groups of signals. The real-time monitoring and diagnostic rules result in constant monitoring in which the analyst only responds to alerts. Signal correlation provides the ability to screen out the effects of normal operation (e.g. a bearing temperature increasing due to an ambient temperature increase) and still provide early warning of signal problems or degraded conditions. The advantage of this process is that the analyst can assume "health by exclusion" for monitored assets and is directed only to areas where the asset is not performing satisfactorily.

The ability to closely monitor active components such as power plant pumps offers the power plant professional an alternative to routine testing. For high-value assets, or those requiring routine testing, this could translate into savings for the utility. However, proper application of this technology requires analysts who can identify "good" behavior and not train equipment on existing faults. When applied properly, the monitoring process described briefly above can be a powerful tool for power plant owners and operators who are concerned with the "health" of their active components – particularly their pumps. The analyst can create empirical models for each component, test those models, build diagnostic rules for those models then activate those models so that they are monitoring component health in real time.

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DESIGN, TESTING, AND IMPLEMENTATION OF MODIFICATIONS TO THE DAVIS-BESSE HPI PUMPS FOR DEBRIS LADEN WATER OPERATION

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ABSTRACT

Following a loss of coolant accident (LOCA) in a pressurized water reactor (PWR), the emergency core cooling system (ECCS) pumps initially pump cooling water from a storage tank into the containment building. When the storage tank volume is depleted, the ECCS is placed in recirculation mode operation. While in recirculation mode, the ECCS pumps remove heat from containment by drawing suction from the containment emergency sump, directing the flow through a heat exchanger, and then pumping the flow back into containment. Following a LOCA, the containment emergency sump likely will contain debris generated by the blowdown forces of the break on nearby insulation, structures, coatings, and the like. The sump also may contain debris generated by effects of the environmental conditions on materials inside containment. The containment emergency sump design includes a strainer to prevent potentially damaging debris from reaching the ECCS pumps and equipment. However, debris particles smaller than the strainer mesh size may still be transported to the ECCS pumps.

The High Pressure Injection (HPI) pumps at the Davis-Besse Nuclear Power Station were discovered to have flow passages inside the pumps that were smaller than the containment emergency sump strainer mesh. The sump strainer mesh size is 4.8 millimeters (mm) (0.188 inch), while there are flow passages in the pump internals as small as 0.15mm (0.006 inch). Preliminary reviews by FirstEnergy, the plant owner/operator, determined that the pumps may not operate as designed with debris in the pumped water. Of particular concern was the hydrostatic bearing that provides shaft support at the end of the shaft. The bearing design included 2.8mm (0.109 inch) diameter orifices at the inlet to the bearing pockets. Large debris particles could plug these orifices and degrade bearing operation. In addition, the close clearances between the hydrostatic bearing pockets and the bearing shaft sleeve were as small as 0.15mm (0.006 inch) radially. Thus, debris particles may flow to the bearing pockets, but not be able to leave the pockets. Sufficient debris in the pockets could also degrade bearing operation. Additional analysis of the pump determined that the HPI pump has its first critical speed slightly above operating speed with original design close clearances at the stage-to-stage wear rings and bushing support (typically about 0.25mm (0.010 inch) radially). The debris in the pumped water following a LOCA likely will wear these close clearances and increase the clearances. Calculations showed that the first critical speed could decrease to the normal operating speed of 3550 revolutions per minute (rpm) if the clearances increased sufficiently. Pump operation at the critical speed could result in significant vibrations. Finally, pump hydraulic performance would also decrease with increased clearances as stage-to-stage leakage increases. Thus, pumping debris laden water following a LOCA potentially could degrade the ability of the HPI pumps to perform their intended safety function.

This paper describes the results of the comprehensive project to resolve these concerns with HPI pump operation. Overall, the concerns were resolved through a combination of pump modifications, analyses, qualification testing, and in-plant testing. The modifications included changes to the hydrostatic bearing and bearing supply flow path design to ensure proper operation with debris in the pumped water (including adding new strainers in the pump internals), as well as changes to internal pump components to increase their tolerance to debris. Qualification testing was performed to confirm proper function of the design modifications and to determine the wear rates of the pump close clearances. The testing program was performed using full scale pump components along with debris laden water with debris concentrations based on the predicted debris generation in the Davis-Besse containment. Design analyses to support the specific design modifications were performed. In addition, rotordynamics analyses of the pump were performed to demonstrate satisfactory vibration levels with increased close clearances, and hydraulic performance analyses were performed to demonstrate satisfactory hydraulic performance with increased close clearances. In-plant testing of HPI pumps was performed to determine pump vibration levels

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and hydraulic performance with enlarged close clearances to benchmark analysis models and demonstrate acceptable pump operation with the clearances increased.

The results of the analyses and testing demonstrated that the modified pumps would operate as desired and perform their intended safety functions following a LOCA. However, some results developed during the testing indicate that the unmodified pumps may not have been able to perform their intended safety functions for all postulated accidents. In addition, the detrimental effects of debris on pump components were greater than expected. These results are also discussed in this paper.

BACKGROUND

Potential deficiencies were identified regarding the ability of the Davis-Besse High Pressure Injection (HPI) pumps to perform their design functions following a loss of coolant accident (LOCA). The postulated deficiencies concerned the potential for debris in the pumped fluid to affect pump operation. The containment emergency sump screens have a 4.8mm (0.188 inch) mesh. Thus, particles as large as 4.8mm (0.188 inch) could pass through the sump screens and enter the Emergency Core Cooling System (ECCS) piping when the ECCS is in recirculation mode operation taking suction from the containment emergency sump. The HPI pump design includes flow passages smaller than 4.8mm (0.188 inch) diameter that could plug or be impacted by the debris. In addition, the debris could cause wear of the close clearances in the pump, resulting in critical operating speeds near or at the running speed and reduced hydraulic capability. It was postulated that reliable pump operation can not be ensured under those conditions.

Various options to address the identified concerns were considered, including replacement of both HPI pumps and associated motors with new pumps and motors. The selected corrective action was to modify the existing pumps and confirm with test and analysis that the modified pumps would operate successfully with debris laden water.

HPI PUMP CONFIGURATION/DESIGN

The Davis-Besse High Pressure Injection (HPI) pumps were manufactured by B&W Canada and supplied during initial plant construction. The HPI pumps are horizontal, eleven stage centrifugal pumps, powered by 450 kilowatts (kW) (600 horsepower (HP)) electric motors. The design pressure is 13.8 Megapascals (MPa) (2000 pounds per square inch gage (psig)) and the design temperature is 150°C (300°F). The design and manufacturing code was the November 1968 ASME Pump & Valve Code, Class II.

A cross section of the pump configuration is shown in Figure 1. The pump design includes the following key elements:

- The eleven stages are arranged in a “2-9” configuration. Flow enters the pump and immediately enters the first stage. The second stage is adjacent to the first stage and pumps the flow through internal passages most of the length of the pump to the third stage which is located at the end of the pump furthest from the motor. Stages three through eleven are adjacent and pump the flow back to the pump discharge which is located opposite from the inlet.
- The main radial shaft supports are a roller bearing outside the pump, the central volute bushing located between the second stage and the eleventh stage, and a hydrostatic bearing located at the end of the pump adjacent to the third stage.
- There are two wear rings on each impeller, one on the suction side of the impeller and one on the discharge side. In addition to sealing, these rings may also provide radial support.
- The hydrostatic bearing supply flow is from two take-offs on the fourth stage volute. The bearing supply flow is routed through tubing back to the hydrostatic bearing, with the bearing discharge flow entering the third stage suction. The minimum flow cross section between the take-off and the bearing pockets are the orifices in the bearing that are 2.8mm (0.109 inch) diameter.

The HPI pump design includes a number of close clearances that could be impacted by pumping debris laden water. These clearances are as small as about 0.15mm (0.006 inch) normal design. Table 1 provides a summary of the close clearances affected by debris, and identifies the materials of the wearing surfaces.

APPROACH TO RESOLVE PROBLEM

The containment emergency sump screens have a 4.8mm (0.188 inch) mesh. Thus, particles as large as 4.8mm (0.188 inch) could pass through the sump screens and enter the Emergency Core Cooling System (ECCS) piping when the ECCS is in recirculation mode operation taking suction from the containment emergency sump. The evaluation of the HPI pump identified several concerns related to HPI pump operation while in ECCS recirculation mode. These concerns, which involve the impact of debris on flow passages in the pump smaller than 4.8mm (0.188 inch), are summarized in Table 2. The resolution approach for operation with debris laden water must address each of these concerns.

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The overall approach for resolving the concerns with HPI pump operation with debris laden water is based on a combination of design modifications, testing, and analyses. The principal elements of the approach are summarized in Table 3.

MODIFICATIONS

The objectives of the pump modifications were to:

- Prevent debris that could plug the hydrostatic bearing orifices from reaching the hydrostatic bearing.
- Modify the bearing design so that debris that reaches the hydrostatic bearing can exit the bearing pockets so that sufficient flow is maintained through the bearing pockets.
- Minimize the wear of the close clearances while pumping debris laden water.

Several modifications were made to the HPI pumps to improve the pumps' tolerance to debris operation. These modifications are summarized below.

Strainer Installation and Location of Hydrostatic Bearing Supply Take-off

In the original HPI pump design, the take-off for the hydrostatic bearing supply was provided by two holes/ports at the periphery of the 4th stage volute. The pump modifications include adding strainers over the take-off ports to prevent large debris that could plug the bearing orifices from reaching the hydrostatic bearing orifices. In addition, the take-off ports were moved to the impeller discharge hub side of the 5th stage volute under the impeller and just above the wear ring. The radial location of the strainer in the pump and the configuration of the stages is shown in Figure 2. Moving the take-off to this location will reduce the concentration and size of debris available to reach the hydrostatic bearing, as well as increase the flow velocity over the strainer to keep it from plugging, based on the following effects:

- The centrifugal effect from the high circumferential flow velocity produces a radial pressure gradient that tends to "throw" the debris towards the outside of the volute and away from the take-off port which is located close to the shaft.
- The stage-to-stage leakage flow paths in the pump will result in water leaking from the 6th stage back into the 5th stage through the discharge hub wear ring and then past the take-off port. This leakage flow will likely be a major source of water to flow into the take-off port. The clearance between the impeller discharge hub and the wear ring is about 0.25mm (0.010 inch) radially. This close clearance will function as an additional strainer, reducing the amount of debris over 0.25mm (0.010 inch) in size that enters the take-off port.
- At the original location of the take-off port, at the volute periphery, the flow velocity varies depending on the pumped flow rate. At the new location of the take-off port under the impeller the circumferential flow velocity is essentially constant over all pump flow rates. The high velocity keeps the surface of the strainer clear of debris to prevent plugging.

Moving the take-off port to the 5th stage and adding the strainer required modifications to the 4th stage and 5th stage volutes and the hydrostatic bearing supply tubing. The modification activities for the volutes included plugging the flow holes in the 4th stage volute and modifying the 5th stage volute by machining recesses, counterboring, and drilling new flow holes through the solid sections of the volute to fit the strainer. Figure 3 illustrates the modification activities on both volutes.

Two strainers were installed, one over each hydrostatic bearing supply tube inlet. The strainers were welded in machined recesses at locations 180° apart. The strainers are constructed from 3mm (1/8 inch) thick sheet. Each strainer has 434 holes with a minimum diameter on the front face of 1.27mm (0.050 inch). The holes provide ample flow area while minimizing the pressure drop through the strainer. The strainer is tapered as shown on Figure 4. The taper prevents debris that is roughly the same size as the minimum diameter from getting lodged in the hole as it flows through the strainer. One potential concern for a stainless steel strainer is erosion of the surface when hard particles are flowing across it. If there is erosion on the face, the minimum diameter of the tapered hole increases as does the maximum size of potential debris that would enter the hydrostatic bearing. To reduce this effect, the strainers are fabricated from Haynes 25 alloy (AMS 5537) instead of stainless steel. Haynes 25 is a cobalt alloy that has significantly better erosion resistance than stainless steel.

Hydrostatic Bearing

The hydrostatic bearing design was modified to make it more tolerant to operation with debris laden water. The modified bearing design is shown in Figure 5. Features of the modified design include:

- "Escape" grooves were included between the bearing pockets and the bearing outlet. These grooves allow debris larger than the bearing close clearance around the pockets to exit the bearing.

- The bearing pocket configuration was changed from a rectangle to an "8" pattern. The "8" pattern bearing with grooves has comparable load carrying capability as the original rectangle bearing without grooves.
- The "8" pattern is based on an "H" pattern bearing developed by Pump Guinard, the original designer of the pump class. Qualification testing showed that the bearing pockets and orifices in the "8" bearing would not fully plug under debris loading. Furthermore, extrapolation of testing conducted earlier by Pump Guinard from their "H" design showed the "8" bearing would still provide adequate stiffness as a hydrodynamic bearing if flow were lost. Since the bearing is not expected to plug and hydrodynamic capability is not required, this capability was not verified for the HPI pump. The feature was included for defense in depth.

Hardfacing of Wear Components

All critical wear surfaces in the pump were modified to apply hardfacing on the wear surfaces. The following components were replaced with new components with hardfacing:

- Suction wear rings
- Discharge wear rings
- Central volute bushing and central shaft sleeve
- Hydrostatic bearing and outboard bearing sleeve

The impeller hubs at the wear ring locations were already coated with tungsten carbide alloy LW-5 to achieve a wear resistant surface, so replacing that hardfacing was not required.

The replacement stationary wear parts were hardfaced with a 0.75mm (0.030 inch) minimum thickness coating of Stellite 6 on the wear surface. Replacement rotating wear parts were hardfaced with a 0.75mm (0.030 inch) minimum thickness coating of Stellite 12 on the wear surface. The wear combination of Stellite 6 on Stellite 12 was selected based upon (1) experience of reliable performance in safety-related applications, including with Pump Guinard pumps, (2) good corrosion resistance in stagnant PWR reactor coolant environment, and (3) demonstrated successful performance as a wear couple.

The base material of the replacement wear parts was changed to Inconel Alloy 600 to be a suitable substrate for the hardfacing. Inconel Alloy 600 was selected for the base material because its thermal expansion properties are similar to Stellite, its good corrosion resistance in a stagnant PWR reactor coolant environment, and Stellite is easy to apply to Inconel 600.

Design Analyses

Design analyses were performed to demonstrate the acceptability of the modifications. The analyses included:

- Three dimensional finite element stress analyses of the modified pump volute with the new strainer cavity. These analyses demonstrate that the volute stresses are acceptable under worst case loading.
- Three dimensional finite element stress analyses of the strainer. These analyses demonstrate that the strainer will not fail under worst case differential pressure loading.
- Seismic analysis of the volute, strainer, and bearing supply tubing. These analyses demonstrate that seismic loads will not impact pump operation.
- Equivalency evaluations were prepared to demonstrate that the replacement hardfaced parts were equivalent to the original parts in "form, fit, and function". These evaluations were performed considering critical dimensions, materials, and changes from the original design.
- Hydraulic analysis of the supply flow path to and through the hydrostatic bearing. These analyses were performed to demonstrate that the changes have minimal impact on hydrostatic bearing and pump hydraulics and to demonstrate that the hydrostatic bearing would operate similarly following the modifications.
- Failure modes and effects analysis was performed to determine the potential failure modes for the modified pump design and to determine the effects of the failure modes. This analysis showed that no new failure modes are introduced by the modifications or component replacements (with new materials)

QUALIFICATION TESTING

The objectives of the qualification testing were:

- Obtain component-specific wear data for the suction wear ring, discharge wear ring, central volute bushing, and hydrostatic bearing.
- Measure flow rates through the suction wear ring, discharge wear ring, central volute bushing, hydrostatic bearing, and hydrostatic bearing supply strainer.
- Confirm that the hydrostatic bearing orifices and supply pockets do not become plugged with debris to the point that the bearing cannot perform its intended function.
- Demonstrate that the hydrostatic bearing strainers will prevent large debris in the bearing supply flow from getting to the bearing.

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- Demonstrate that the hydrostatic bearing strainers do not become plugged with debris to the extent that they prevent an adequate supply of water from reaching the bearing.

Overview

The qualification testing program was implemented using full scale mock-up fixtures of the critical components of the HPI pump. Pump parts and components that were modified or replaced as part of the modifications were tested using the new design. The qualification test program was implemented as a series of separate effects tests, with each test fixture/test loop representing a separate feature of the HPI pump design. Separate effects mock up testing is a representative approach to place HPI pump parts under the expected detrimental conditions they would face post-LOCA. Based on the comprehensive nature of the separate effects tests and the acceptable results, it was concluded that a test of the actual HPI pump with debris laden water was not required.

The key elements of the mock-up test program include:

- The test fixture designs match the critical characteristics of HPI pump components
- The test fixtures use full scale pump components
- The test program and test fixtures included the capability to pause/re-start tests to determine interim results
- The individual separate effects fixtures provided a flexible platform for evaluating alternate modifications before choosing the preferred design for final testing.

The testing of each component consisted of a series of tests with clean and debris-laden water. Each component was initially tested with clean water to obtain baseline data and assure that the facility and the test fixture were operating correctly. Following the clean water tests, the components were subject to a series of tests with debris-laden water. Following the debris testing, all of the test articles with close running clearances were tested again on clean water to assess the effects of wear on their flow characteristics.

The general arrangement of the test facility is shown in Figure 6. A central tank (Tank 1) capable of holding approximately 34 cubic meters (9,000 gallons) of water was used as the ultimate source of supply. During debris testing, the water in the tank was supplemented by a mixture of debris intended to represent the important characteristics of the debris that might be present in the containment emergency sump at Davis-Besse after a LOCA. The tank was equipped with a total of four agitators – two vertical paddle-type agitators and two submersible pumps – to help keep the debris in suspension.

The test loops for the suction wear ring (Loop 1), the discharge wear ring (Loop 2), the central volute bushing (Loop 4), and the hydrostatic bearing supply strainers (Loop 5) were supplied directly from Tank 1. The hydrostatic bearing tester (Loop 3) was supplied indirectly from Tank 1 via one of the hydrostatic bearing supply strainers in Loop 5. The Loop 5 fixture included two hydrostatic bearing supply strainers. The output of one of the hydrostatic bearing supply strainers was used to continuously supply Tank 2 in Loop 3. The hydrostatic bearing supply pump takes suction on Tank 2 and supplies the hydrostatic bearing. Tank 2 is agitated by the combined action of an external paddle-type agitator and the return of excess flow from the bearing supply pump. A return pump takes suction on Tank 2 and returns water to Tank 1 as necessary to control the water level in Tank 2. Figure 7 shows the configuration of a typical test loop.

Test Fixture Design/Equivalency Evaluation

The test fixtures for the close clearance components in the pump (the wear rings, central volute bushing, and hydrostatic bearing) were constructed similarly. The pump component to be tested was installed in a fixture that recreated the configuration in the HPI pump, and an external pump was used to create a pressure difference across the clearance and a simulated flow through the pump. The flows through the clearance were measured during the testing and the clearances were measured periodically during the testing by disassembling the fixtures.

The test fixture for the hydrostatic bearing supply strainers (Loop 5) was a single stage centrifugal pump with similar configuration and critical dimensions as a stage of the HPI pump. The strainers are installed in the pump volute in the same radial location as in the HPI pump. The design of the Loop 5 pump precluded simulating the discharge side of the volute and the strainer effect from flow passing through the discharge wear ring. As a result, the amount and size of debris present at the strainer surface for the Loop 5 pump is considered to be greater than what would be present in the HPI pump (i.e., a conservative testing approach).

Detailed evaluations were performed to demonstrate that the test fixtures and test loops were sufficiently representative of the HPI pump. The evaluation of the test fixtures and test loops considered the main attributes of design configuration, flow fields, operating conditions, and debris characterization.

Design Configuration

The proper mock-up of the design configuration ensures that the flow areas, wear couples, etc., are suitably representative. The following critical characteristics were evaluated:

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- Key dimensions – the sizes of the fixture components as well as the clearances between components
- Materials – the hardness and strength of the parts used to represent the components

These critical characteristics were confirmed for each fixture by performing detailed receipt inspections of the fabricated fixtures.

Flow Fields

Proper representation of the flow fields is necessary to model the flow of debris through the fixture and the pump, in particular near and through close clearances. Appropriate flow fields were established based on the dimensional and operational characteristics of each test fixture. The operating conditions were controlled via the velocity and direction of inlet flow to each fixture and the motor rotational speed driving the rotating parts. The following critical characteristics were evaluated regarding the flow conditions in the test fixtures:

- Flow velocities (direction and magnitude) into the test fixtures
- Flow directions and profiles near the inlets to the close clearances
- Differential pressures across the close clearances that drive flow through the clearance

The inlet flow velocities were determined by calculation using the flow areas through the HPI pump and test fixture and the simulated pump flow rate. The flow profiles near the close clearances were evaluated using computational fluid dynamics (CFD) modeling leading to qualitative evaluation of the flows through the test fixtures compared to the HPI pump. The differential pressures were determined by calculation using the HPI pump design and the simulated flow conditions.

All flow fields in the test fixtures are representative of the corresponding flow fields in the HPI pumps.

Operating Conditions

The HPI pumps operate at different conditions depending on the size of postulated pipe breaks and the time frame following the pipe break. The selection of the operating conditions for performing mock-up testing is based on an evaluation of these pump operating conditions and the potential for pump degradation.

For the purposes of establishing the qualification test program, the critical characteristics for selecting the pump operating conditions are:

- The simulated pump operating conditions must be comparable to the conditions that would exist following a LOCA when the HPI pumps are performing their required safety functions.
- The simulated pump operating conditions should represent conditions that maximize the potential for pump degradation due to pumping debris laden water.
- The simulated pump operating conditions must match pump operating conditions with the expected debris concentrations that would be present in that operating mode.

The evaluation of the HPI pump operating conditions determined that the limiting conditions for pump degradation are long term boron precipitation control cooling following a large break LOCA. This condition includes the worst case debris in containment following a large break LOCA with pump head/flow conditions of about 57 cubic meters per hour (m3/hr) (250 gallons per minute (gpm)).

Debris Characterization

Selection of the debris for use in qualification testing is a critical aspect of the qualification testing program. Since the testing is performed to investigate the effects of debris on pump internal components, the debris used in the testing must be representative of the debris that could reach the HPI pump following a LOCA.

Determination of the appropriate debris for mock-up testing is a multi-step process involving the following major activities.

1. Debris Generation – Determine the various types and quantities of debris that would exist in a post-LOCA containment environment and have the potential to be transported to the containment sump.
2. Debris Transport – Of the debris that would be in the containment in a post-LOCA environment, determine the type and quantities of debris that could be transported to the containment emergency sump during recirculation operation. For evaluating the HPI pumps, all debris that can be transported to the sump is assumed to pass through the sump strainer and reach the HPI pumps.
3. Debris Critical Characteristics – Establish the characteristics of each debris type that are essential for mock-up testing (e.g., particle sizes, quantities, types of material, material properties, etc.).
4. Representative Debris for Testing – Using the critical characteristics for the debris, select representative debris to be used in the mock-up testing.

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As discussed above, the limiting condition for debris generation is a large break LOCA. The analyses performed for the HPI pump modification testing were based on the analyses performed for the design of the containment sump strainer. The key difference is that the debris generation and transport analyses were modified to ensure that conservative assumptions were used for evaluating conditions downstream of the sump, at the inlet to the HPI pump instead of evaluating conditions at the sump strainer. The result was the determination of the types, sizes, and quantities of the debris expected to flow through the HPI pumps. These results were used to select representative debris materials for the qualification testing. The debris used in the test program is summarized in Table 4.

Test Results

The key results of the qualification testing were:

- As shown in Figure 8, the suction wear rings and impeller suction hubs showed no significant wear or flow increase from operations with debris. This is believed to result from the impeller causing the debris to be "thrown" to the periphery of the volute, away from the suction wear ring.
- The discharge wear ring showed minimal wear, but there was significant wear of the rotating impeller discharge hub. As shown in Figure 9, wear-through of the tungsten carbide coating led to a deep wear groove in the softer hub material. Abrasive wear by accumulation of debris appeared to be the major mechanism for the wear. The discharge wear ring results differ from those of the suction wear rings because there is no impeller at the inlet to the discharge wear ring to force the debris away from the clearance inlet.
- The hydrostatic bearing orifices operated without plugging. The bearing experienced minor and temporary flow-rate reductions during testing, probably as a result of minor debris accumulations in the bearing. Debris, mainly fiber and fine particles captured by the fiber, tended to collect in the bearing, primarily at the pocket islands and in the inter-pocket running clearances. Significant wear of the rotor sleeve occurred as a result of this debris. The sleeve hardfacing was worn to a clearance about two to three times the original clearance, but not through the hardfacing. The bearing showed no tendency to bind during operation and the flow remained sufficient for the test duration. The bearing and shaft sleeve after testing are shown in Figure 10.
- The central volute bushing showed minimal wear, but there was significant wear of the rotating sleeve. The sleeve hardfacing of the outboard sleeve was worn to a clearance about two to three times the original clearance, but not through the hardfacing. The final condition of the sleeve is shown in Figure 11.
- The hydrostatic bearing supply strainers worked well. The flow to the hydrostatic bearing remained constant throughout the testing.

IN-PLANT TESTING

The in-plant testing was performed primarily to support benchmarking of the rotordynamics and hydraulic analysis models. The testing also showed that the pumps operate satisfactorily with the close clearances increased. Two tests were performed. The baseline test was performed using an essentially new pump with design close clearances. The second test was performed using the spare pump, which had the close clearances machined and increased to twice the design clearance. The in-plant tests were expanded pump surveillance tests. Special instrumentation was installed and the pumps were tested over the complete range of operating flows. The pump vibration levels and hydraulic performance were recorded over the flow range.

The baseline testing was performed using the P58-1R HPI pump. This pump had been installed new at Davis-Besse in 2001. Since the pump internal assembly was relatively new, the close clearances in the pump (wear rings, hydrostatic bearing, and central volute bushing) were comparable to the original design clearances (i.e., "1X"). The objectives of the baseline pump test were to:

- Establish a baseline for the nominal close clearance case. This allows subsequent test(s) with increased close clearances to provide direct measure of the effects of increasing the clearances on pump head/flow and vibration.
- Provide detailed measurements for use in validating the rotordynamics model and the pump hydraulic performance model.

The worn pump testing was performed using the spare HPI pump element (P58-1O). This pump element was recently removed from service. The pump element was disassembled and the close clearances in this pump element were increased to twice the normal values. The clearances at the hydrostatic bearing, central volute bushing, and wear rings were all increased. The value of twice normal clearances was selected (1) because typical maintenance approaches with similar rotating equipment is to refurbish the pump when clearances reach twice normal, and (2) it was expected that the actual close clearance wear during recirculation mode operation would be about this amount or less. The objectives of the worn pump test were to:

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- Demonstrate that large amounts of wear of pump wear rings and bearings (wear to twice the normal clearances) does not result in unacceptable pump performance.
- Confirm the rotordynamics model predictions of the effects of increased clearances on pump vibration. This confirmed model could then be used to assure that other potential wear conditions which were not measured in the in-plant test would not result in unacceptable pump behavior.
- Provide additional pump hydraulic performance data for validating the pump hydraulic performance model.

In addition to the baseline and worn pump tests, post-modification testing was performed for both pumps re-installed in the plant. This testing demonstrated that the pumps operate satisfactorily after reassembly with modified and replacement parts.

ANALYSES

Wear, hydraulic, and rotordynamic analyses were performed to demonstrate the modified HPI pumps are acceptable for operation under normal and debris operating conditions.

Wear Analyses

The objective of the wear analyses was to estimate, using the results of the qualification testing, the "worn conditions" of the HPI pump close clearances after pumping debris laden water in the post-LOCA environment. The worn conditions are evaluated using hydraulic and rotordynamic analyses to demonstrate adequate performance of the HPI pumps following a LOCA.

The increases in the HPI pump close clearances following a LOCA were predicted based on analysis and the results of qualification testing using HPI pump components under debris loading conditions. The wear of the close clearances resulted from a combination of erosive and abrasive wear as the debris flowed through the clearance. Abrasive wear is dependent on several factors, including debris type and concentration, surface material and condition, etc. As a result, it is difficult to predict wear rates without testing of the actual conditions. Thus, the predictions for wear of the pump close clearances were based primarily on the results of the qualification testing.

The results of the qualification testing for each close clearance were used to develop predictions for clearance increases during post-LOCA operation. Two adjustments were made to the test results. First, the results were adjusted based on measured flow through the clearance. Since the wear is a function of the volume of debris flowing through the clearance, the wear rates were adjusted based on the ratio of flow through the test fixture clearances compared to the expected flow in the HPI pump. Second, the results were extrapolated to 30 days of operation. (Wear testing durations varied from 21 to 24 days for the various test fixtures). The results of the wear predictions provide a conservative estimate of the clearance increase following a LOCA.

Essentially all wear in all the test fixtures was on the rotating components. The stationary components experienced very little, if any wear. Table 5 provides a summary of the predictions for the worn conditions for the close clearances as a function of time following the LOCA. The results for individual components are discussed below.

- Suction Wear Ring – The suction wear ring and impeller hub surfaces exhibited relatively little wear during the testing. The predicted clearance increase after 30 days is only about 0.10mm (0.004 inch) (diametral), compared to an initial diametral clearance of 0.50mm (0.020 inch).
- Discharge Wear Ring – Measurable wear was seen across the impeller hub during testing. In addition, a deep groove developed near the clearance exit. Based on these results, the length of the wear ring clearance post-LOCA is assumed to be shorter by the length of the groove and the clearance width over the remaining hub surface is predicted to increase by about 0.94mm (0.037 inch) (diametral) after 30 days (compared to an initial diametral clearance of 0.50mm (0.020 inch)).
- Central Volute Bushing – The central volute bushing sleeve exhibited wear during the testing. The predicted clearance increase after 30 days is about 0.64mm (0.025 inch) (diametral), compared to an initial diametral clearance of 0.33mm (0.013 inch).
- Hydrostatic Bearing – The wear of the shaft sleeve during testing was not uniform. Different regions of the sleeve experienced different amounts of wear due to the flow fields in the bearing and the locations were debris became lodged. The first 10mm (0.4 inches) from the axial ends of the bearing wore the least (edge of the bearing). The middle of the bearing under the orifices wore slightly more (center of the bearing). The most wear occurred between the middle of the bearing and the edge of the bearing (central portion of the bearing). The predicted clearance increases after 30 days are between 0.46mm (0.018 inch) and 1.17mm (0.046 inch) (diametral), compared to an initial diametral clearance of 0.36mm (0.014 inch).

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Hydraulic Analyses

The objective of the hydraulic analyses was to demonstrate that the HPI pump would perform satisfactorily and provide the necessary head/flow for all required operating conditions, including following a LOCA.

The approach used to evaluate the hydraulic capability of the HPI pumps was to use a hydraulic model of the pump to predict head/flow capability in the worn condition for comparison to the required capability based on HPI system safety functions.

The HPI pump hydraulic model was constructed from a first principles model of the pump. The model is based on the head/flow characteristics of the HPI pump impellers and includes the leakage flows through the close clearances at the wear rings, central volute bushing, and hydrostatic bearing. The widths of the close clearances are inputs and the model automatically calculates the hydraulic resistance of each leakage flow path based on the flow area and pump conditions. The hydraulic model was benchmarked using measured head/flow performance during in-plant testing.

The required hydraulic capability for the HPI pumps is based on the HPI system safety functions. Immediately following a LOCA, the required head/flow capability is that assumed in the small break LOCA safety analyses (essentially the quarterly surveillance test acceptance criteria). Long term hydraulic capability following the LOCA is based on boron precipitation control requirements. In boron precipitation control mode following a LOCA, the HPI flow through the auxiliary pressurizer spray line is the flow required to remove the decay heat plus an additional flow to prevent stagnation in the reactor vessel and any flow through the pump minimum flow line. The required boron precipitation control flow rate shortly after a LOCA is 57 m3/hr (250 gpm). In the plant safety analyses this minimum flow was assumed to remain constant for the duration of the period after the LOCA. Additional analyses were prepared for this evaluation taking into consideration the decrease in decay heat following the LOCA. This analysis showed that the required flow capability dropped significantly after several days following the LOCA.

The predicted worn conditions as a function of time (Table 5) were used as the input conditions for the hydraulic model to predict pump capability following the LOCA. In addition, the required capability was determined as a function of time following the LOCA. The results of these calculations are shown in Figure 12. This figure shows that the hydraulic capability of the HPI pumps decreases slowly until about 30 days following the LOCA, then remains relatively high after even longer periods. However, the required capability decreases rapidly following the LOCA. Within one day following the LOCA the required capability is less than about 45 m3/hr (200 gpm) at about 365 m (1200 feet) of head. After 30 days the required capability is only about 23 m3/hr (100 gpm) at about 150 m (500 feet) of head. Thus, the modified HPI pumps have considerable margin between the required and available hydraulic capability.

Rotordynamics Analyses

The objective of the rotordynamics analyses was to demonstrate that the HPI pump would operate satisfactorily, without excessive vibration, over the full range of pump flows for the predicted increase in close clearances.

The rotordynamic model of the HPI pump is a finite element model developed with the ANSYS general purpose computer program. The shaft is represented by beam elements, the impellers are represented by lumped masses with rotary inertia, the roller bearing is represented by spring elements, and the hydrostatic bearing and wear rings are represented by stiffness and damping matrices.

The stiffness and damping characteristics of the hydrostatic bearing, wear rings and central volute bushing depend on the pump operating conditions. As pump flow is increased, the differential pressure across each pump stage decreases and the stiffness of the rotor support elements decreases. The stiffness of these elements also decreases as the components wear. The stiffness of these elements was calculated based on the predicted worn conditions. In addition, the discharge wear rings are conservatively not modeled in the rotordynamic analysis because no appreciable flow was measured through the discharge wear rings for over half of the mock-up testing. Without flow, the discharge wear ring does not develop rotordynamic stiffness.

The HPI pump is considered to be acceptable if the vibration amplitudes allow the predicted minimum steady state film thicknesses in close clearances to be maintained. Figure 13 compares the results of rotordynamic analyses for the HPI pump with the new “8” pocket design hydrostatic bearing for nominal design clearances (1X) and the original rectangular pocket hydrostatic bearing for nominal design clearances. The hydrostatic bearing modifications have an insignificant effect on overall rotor-dynamic performance.

The expected rotordynamic performance of the HPI pump with the new hydrostatic bearing was evaluated at Days 10, 20, and 30 following a LOCA event. Support stiffness was based on the increases in clearances shown in Table 5. Figure 14 shows the predicted rotor deflections when the pump is running for the design case and for the three wear conditions. The maximum rotor deflection occurs at the hydrostatic bearing end of the rotor for Day 30 wear.

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The predicted maximum deflection at the suction wear ring closest to the hydrostatic bearing is 0.10mm (0.004 inch). The limit on deflection is set by the suction wear rings. At Day 30, the suction wear ring and hydrostatic bearing radial clearances are predicted to be 0.30mm (0.012 inch) and 0.53mm (0.021 inch) respectively. Metal-to-metal contact would occur first at the suction wear ring closest to the hydrostatic bearing. At Day 30, there is 0.20mm (0.008 inch) of margin (0.30mm – 0.10mm = 0.20mm) (0.012 inch- 0.004 inch = 0.008 inch) in shaft deflection predicted to accommodate rotordynamic vibration.

Figure 15 shows the results of modal analyses. The nominal 1X clearance results from Figure 14 are included for comparison. The Day 10 first mode natural frequency crosses the pump running speed at 90 m3/hr (400 gpm) indicating a critical speed at that point. Day 20 and Day 30 first mode natural frequencies are less than the running speed for all pump flow rates.

Forced response analyses were performed to determine the expected rotor vibration amplitude in the post-LOCA condition. Scoping analyses indicated that the most limiting location in the pump for forced vibration is at the hydrostatic bearing. At this location, the forced vibration displacements in the worn condition were the greatest. Figure 16 shows the forced response results for LOCA conditions. The vibration amplitude at the hydrostatic bearing is plotted as a function of pump flow rate for the three wear conditions. The maximum vibration amplitude is 0.03mm (0.0012 inch). This is well within the 0.20mm (0.008 inch) available based on the rotor deflection analysis.

CONCLUSIONS

The main results of the design, analysis, and testing activities were:

- The modifications to the HPI pumps satisfy all applicable design criteria and assure that the pumps will operate successfully with the defined post-LOCA debris conditions. In addition, the modifications do not negatively affect “form, fit, or function,” so the pump will continue to operate satisfactorily under normal, clean water conditions.
- The limiting condition for HPI pump operation with post-LOCA debris is long term boron precipitation control cooling following a large break LOCA. This condition has maximum debris in the containment emergency sump combined with a relatively low flow, high head pump condition.
- Qualification testing under debris loading of the modified hydrostatic bearing and a mock-up of the hydrostatic bearing supply strainer shows that the bearing and strainer will function adequately.
- Qualification testing of the HPI pump close clearances, coupled with wear analyses, determined that the suction wear rings would experience minimal increase in clearance under debris laden water service. The discharge wear rings showed significant clearance increase along with “grooving” at the exit of the clearance. The central volute bushing and hydrostatic bearing showed measurable wear (approximately three times the original clearance). Essentially all wear was on the rotating component; the stationary components experienced very little wear. These test results were used to determine the worn condition that would exist following a LOCA.
- Rotordynamic analyses for the HPI pump worn condition predicted to exist after 30 days post-LOCA show the pump will function satisfactorily. The maximum predicted pump vibrations are within the acceptance criteria and the bearing support system is adequate to support the weight of the rotating assembly.
- Hydraulic analyses for the HPI pump worn condition predicted to exist after 30 days post-LOCA show that the HPI pump has significant hydraulic margin for the limiting condition of boron precipitation control cooling.
- In-plant testing of an HPI pump modified to increase the close clearances to twice normal values showed satisfactory pump operation with vibration levels no greater than with normal clearances even with the pump operating near critical speed. This testing was used to qualify the rotordynamic and hydraulic analyses.

Based on the results described above, the modified Davis-Besse HPI pumps will operate satisfactorily under normal conditions and while pumping debris laden water following a postulated LOCA.

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Table 1. HPI Pump Close Clearances

Clearance	Design Clearance[1] mm (inch)	Wearing Component/Material[2, 3]	
Central Volute Bushing	0.28 – 0.36 (0.011 – 0.014)	Shaft Sleeve/ Bronze B143, Alloy 903	Bushing/ Hardened 17-4PH SS
Suction Wear Ring	0.48 – 0.53 (0.019 – 0.021)	Impeller Hub/ Tungsten Carbide Coating	Wear Ring/ Hardened 17-4PH SS
Discharge Wear Ring	0.48 – 0.53 (0.019 – 0.021)	Impeller Hub/ Tungsten Carbide Coating	Wear Ring/ Hardened 17-4PH SS
Hydrostatic Bearing	0.30 – 0.38 (0.012 – 0.015)	Shaft Sleeve/ Bronze B103, Grade D	Bearing/ 431 SS

Notes:
1. All clearances are listed as diametral clearances.
2. Smaller diameter, rotating component listed first
3. Materials prior to modifications

Table 2. Concern Summary

Concern	Description
Hydrostatic Bearing Orifice Plugging	The supply flow to the hydrostatic bearing is taken off the fourth stage discharge and directed back to the bearing assembly which is at third stage pressure. There are two supply tubes in parallel; each tube is 9.5mm (0.375 inch) outside diameter, with 1.25mm (0.049 inch) wall. The supply flow enters the bearing assembly where it is distributed circumferentially around the bearing and through orifices into five bearing pockets. There is a single orifice feeding each pocket. The orifice diameter is about 2.8mm (0.109 inch). Since the containment emergency sump mesh is 4.8mm (0.188 inch), debris could pass through the sump mesh, flow to the HPI pump, enter the bearing supply tubing and plug the orifices. If the flow to the bearing pockets is reduced or the resistance of the orifice changes, the bearing may not function properly and reliable pump operation could be impacted.
Hydrostatic Bearing Pocket Plugging	The hydrostatic bearing design includes tight clearances (0.15 to 0.20mm (0.006 to 0.008 inch) radially) between the outlet of the bearing pockets and the shaft sleeve. All flow into the bearing pockets must pass through this clearance to enter the third stage suction flow. Since the containment emergency sump mesh is 4.8mm (0.188 inch), debris could pass through the sump mesh, flow to the HPI pump, enter the bearing supply tubing, flow to the bearing pockets, and then become lodged or stuck in the pockets. If the flow to the bearing pockets is reduced, the bearing may not function properly and reliable pump operation could be impacted.
Close Clearance Wear	The close clearances in the HPI pump are small. The clearances are as small as 0.15mm (0.006 inch) on the hydrostatic bearing, 0.14mm (0.0055 inch) on the central volute bushing, and 0.24mm (0.0095 inch) on the wear rings. A preliminary rotordynamics analysis of the HPI pump showed the pump first critical speed slightly above the normal operating speed for design flow and design clearances. During debris loading conditions, these clearances will wear and increase. When the clearances increase, the critical speed will decrease. Depending on the amount of wear, the critical speed could reduce to the operating speed, resulting in excessive vibration. The increased vibration could impact reliable pump operation. As the clearances increase, the stage to stage leakage through the close clearances will also increase. This increase in leakage will decrease pump hydraulic capability. The resulting pump hydraulic performance in the worn condition must satisfy the applicable design requirements.

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Table 3. Resolution Approach Summary

Concern	Modifications	Testing	Analysis
Hydrostatic Bearing Supply Orifice Plugging and Bearing Pocket Plugging	• A strainer was installed in the pump volute at the take-off for the hydrostatic bearing supply flow to preclude debris larger than the orifice diameter from reaching the orifice. • The location of the take-off for the hydrostatic bearing supply flow was relocated to an inner radius on the pump volute discharge wear ring side to reduce the concentration and size of debris present at the supply line take-off. • "Escape" grooves were added to the hydrostatic bearing pockets to allow debris larger than the bearing clearance to leave the pockets. • The hydrostatic bearing pocket design was modified to increase the bearing stiffness so that the bearing with grooves has a stiffness comparable to the existing bearing.	• Mock-up testing of the strainer in the new supply line take-off location confirmed that the strainer will continue to provide flow (i.e., it will not fully plug with debris). • Mock-up testing of the strainer in the new supply line take-off location was used to obtain the debris loading for testing of the hydrostatic bearing. • Mock-up testing of the hydrostatic bearing assembly was performed with the debris laden water that could reach the bearing to confirm proper operation (maintain adequate supply flow to and through the bearing orifices and bearing pockets).	• The strainer design and volute modification were evaluated to demonstrate structural adequacy. • Rotordynamic bearing analyses were performed to confirm adequate stiffness and load carrying capacity of the new hydrostatic bearing design. • The modified hydrostatic bearing hydraulics (new take-off location and strainer) were included in the evaluation for pump hydraulic performance. • A failure modes and effects analysis was performed to assess the potential for, and implications of, postulated failure modes. • Engineering evaluations were performed to demonstrate that the mock-up testing fixtures, including mockup of the hydrostatic bearing intake flow through the strainer are suitably representative of the HPI pump critical characteristics.

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Table 3. Resolution Approach Summary (continued)

Concern	Modifications	Testing	Analysis
Close Clearance Wear – Operation near Critical Speed Close Clearance Wear – Reduced Hydraulic Performance	• The components with wear surfaces were replaced with new components with hardfaced surfaces to minimize wear during debris operation. The suction wear rings, discharge wear rings, hydrostatic bearing and shaft sleeve, and central volute bushing and shaft sleeve were replaced.	• Mock-up testing was performed of the pump close clearances to determine wear rates while pumping debris laden water. Wear rates were determined for the hydrostatic bearing, suction wear ring, discharge wear ring, and central volute bushing, and associated rotating parts. • In-plant testing was performed for a relatively new HPI pump and a spare HPI pump artificially worn to large clearances (twice the normal design). Flow and hydraulic performance data and detailed vibration data were acquired for both tests to demonstrate that the pumps operated satisfactorily with the increased clearances.	• The results of mock-up wear testing were used to benchmark models to predict the increases in the close clearances during pump operation with debris laden water. • Rotordynamic analyses were performed to demonstrate that pump operation with the close clearances opened to the predicted worn conditions would not detrimentally affect pump operation and vibration levels would be acceptably low. The model was benchmarked using vibration data from the in-plant testing. • Hydraulic analyses were performed to demonstrate that pump performance with the close clearances opened to the predicted worn conditions would satisfy required head/flow requirements. The model was benchmarked using hydraulic performance data from the in-plant testing. • Engineering evaluations were performed to demonstrate that the mock-up testing fixtures are suitably representative of the HPI pump critical characteristics.

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Table 4. Debris Characterization for Testing

Debris Type	Simulated Debris	Material Size Characteristics
Fibers	chopped E-glass fibers	Lengths = 1.6mm, 3mm, and 6mm (0.0625”, 0.125”, and 0.25”) Diameter = 8 μm
Dirt, Dust and Rust	Magnetite (iron oxide)	5.6% = 80 mesh 6.0% = 100 mesh 29.8% = 200 mesh 57.7% < 200 mesh
Concrete	Silica	Type 110, 570 and 2040 silica sand (sieved to obtain proper size distribution)
Coating particles	“Plasti-grit” (urea formaldehyde)	Stream Table Mix (sieved to obtain proper size distribution)
Coating chips	Chopped Plastic Chips (PVC)	Thickness = 0.25mm (0.010”) 0.031” < Diameter < 0.066”
Coating flakes	Plastic Chips (PVC)	Thickness = 0.25mm (0.010”) Diameter = 3mm (0.125”)

Table 5. Predicted Post-LOCA Diametral Clearances

Clearance	Clearance Following LOCA mm (mils)			
	0 Days	10 Days	20 Days	30 Days
Suction Wear Ring	0.51 (20.0)	0.56 (22.2)	0.59 (23.2)	0.61 (24.1)
Discharge Wear Ring	0.51 (20.0)	0.81 (31.9)	1.12 (44.2)	1.44 (56.6)
Central Volute Bushing	0.33 (13.1)	0.65 (25.5)	0.81 (31.8)	0.97 (38.0)
Hydrostatic Bearing Edge	0.36 (14.0)	0.59 (23.3)	0.83 (32.6)	1.06 (41.9)
Hydrostatic Bearing Interior	0.36 (14.0)	0.74 (29.3)	1.13 (44.6)	1.52 (59.9)
Hydrostatic Bearing Center	0.36 (14.0)	0.51 (20.0)	0.66 (26.0)	0.81 (32.0)

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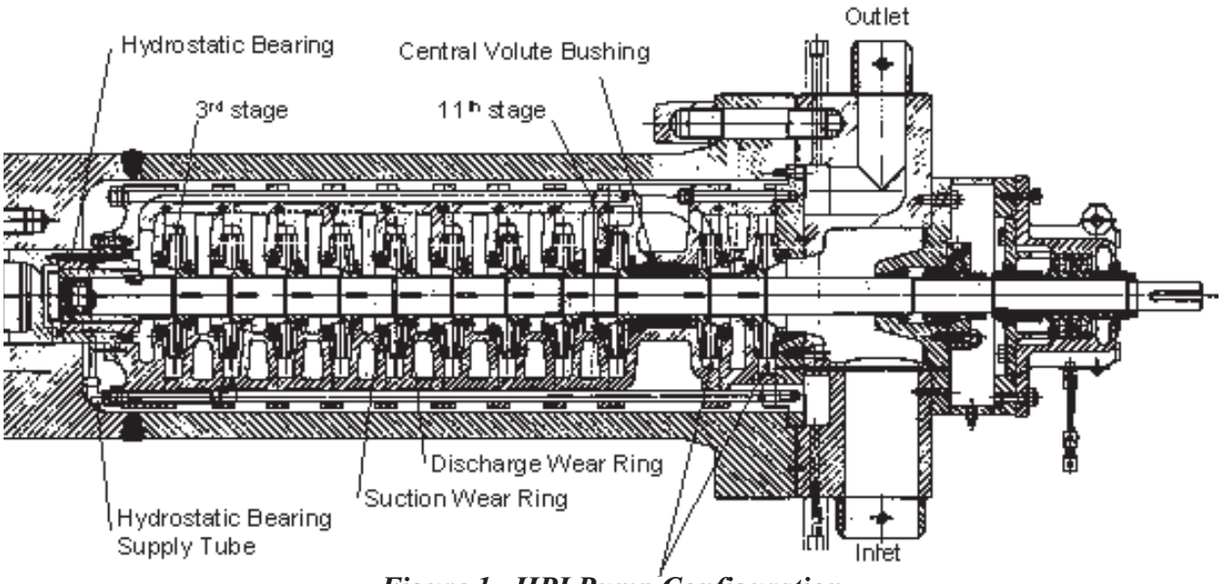

Figure 1. HPI Pump Configuration

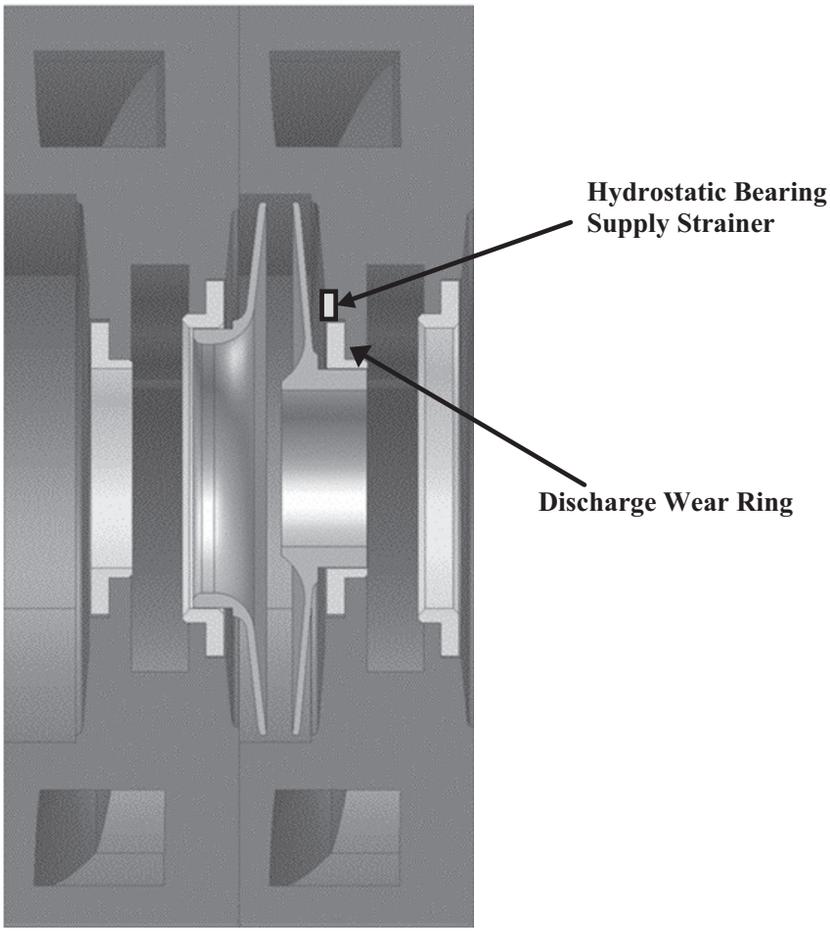

Figure 2. Hydrostatic Bearing Strainer Location

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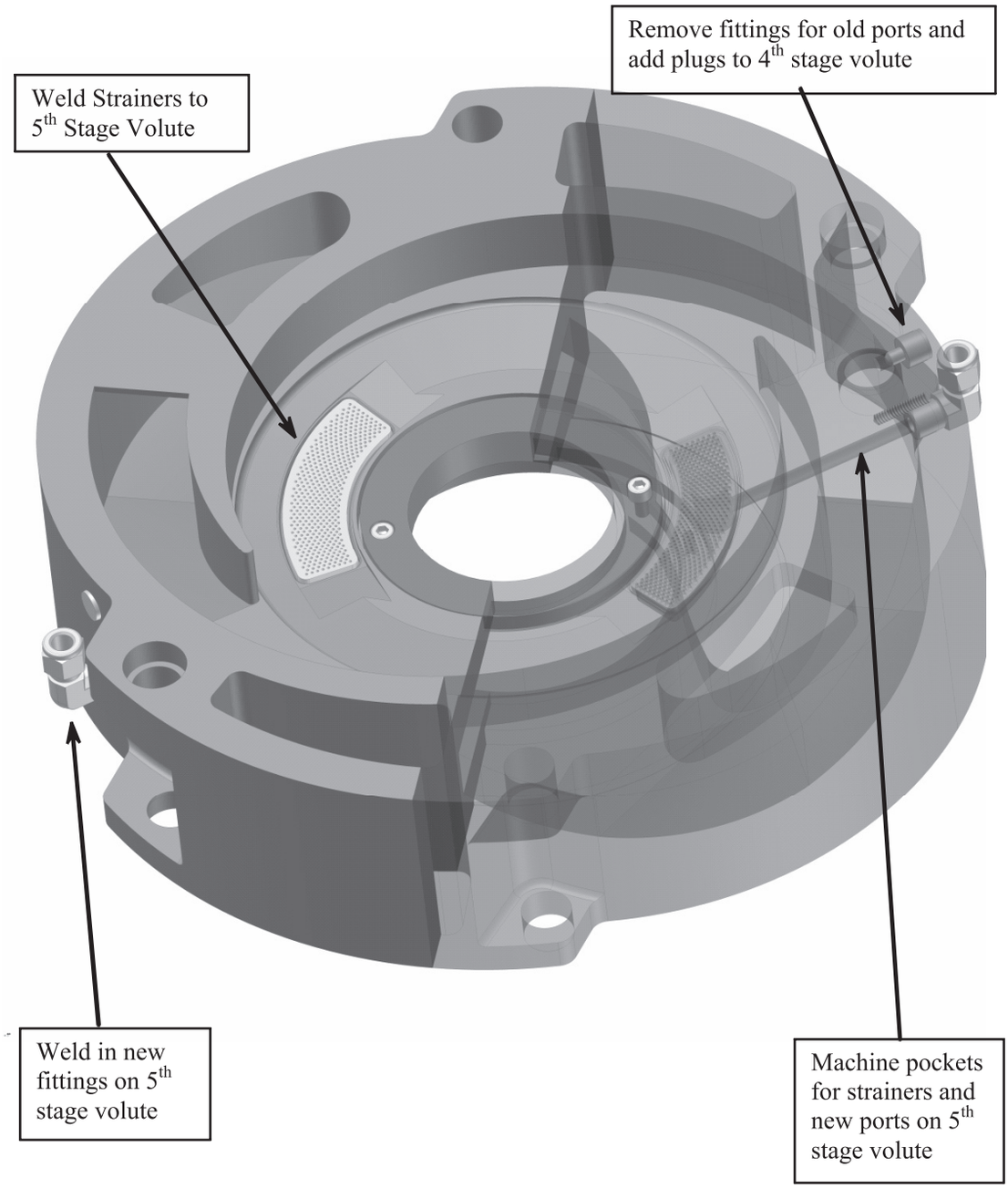

Figure 3. Volute Modifications

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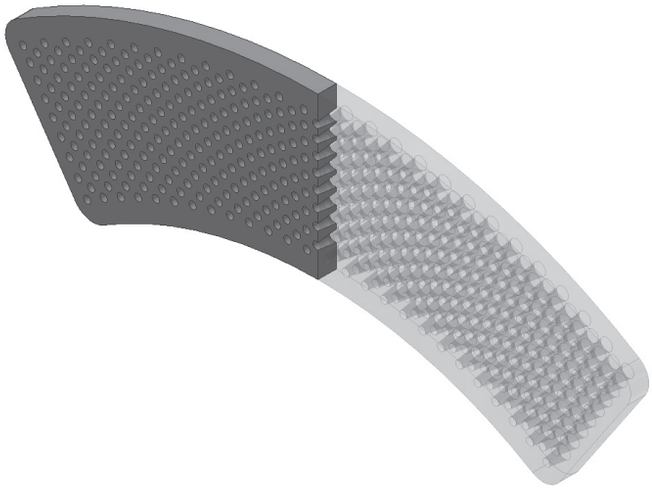

Figure 4. Hydrostatic Bearing Supply Strainer

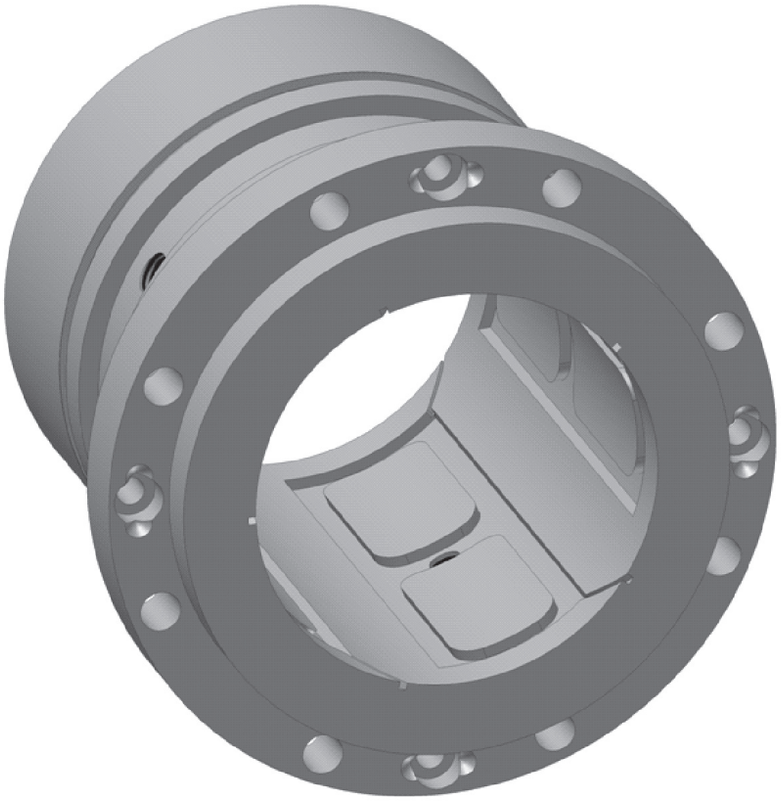

Figure 5. Modified Hydrostatic Bearing Design

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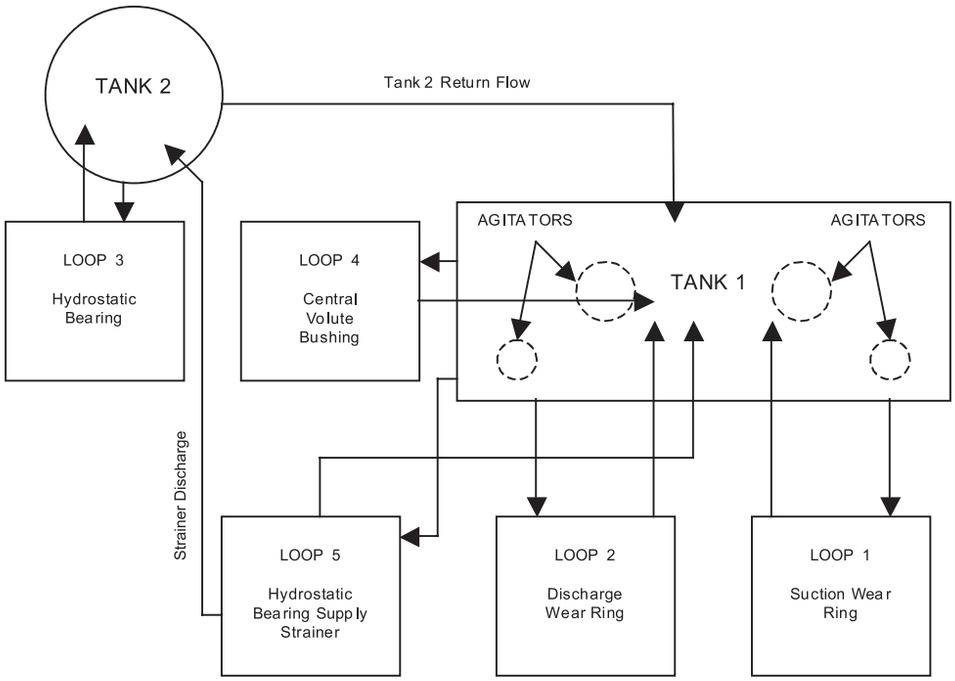

Figure 6. General Layout of Test Facility

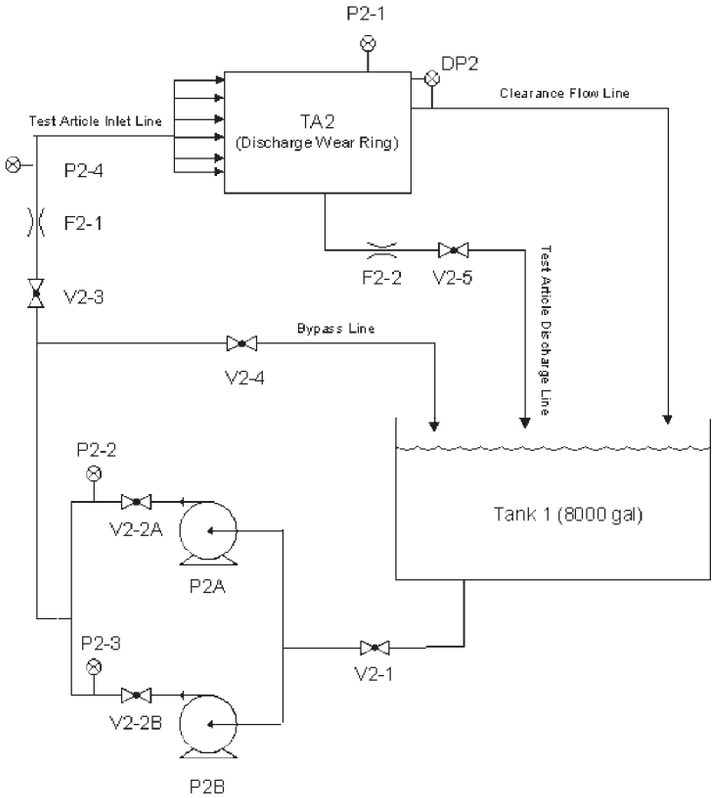

Figure 7. Typical Test Loop

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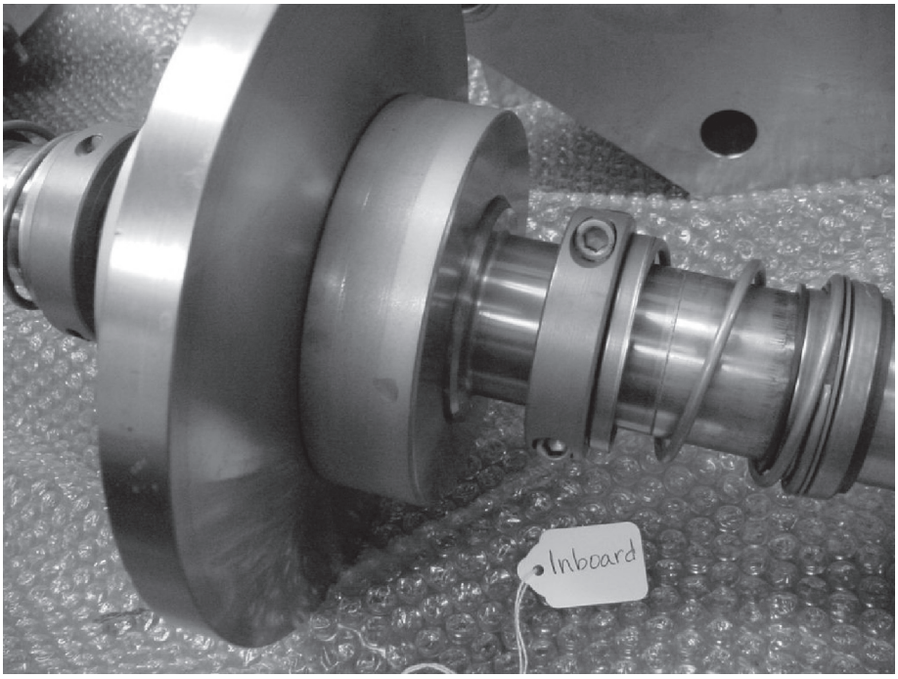

Figure 8. Suction Wear Ring Test Fixture After Testing

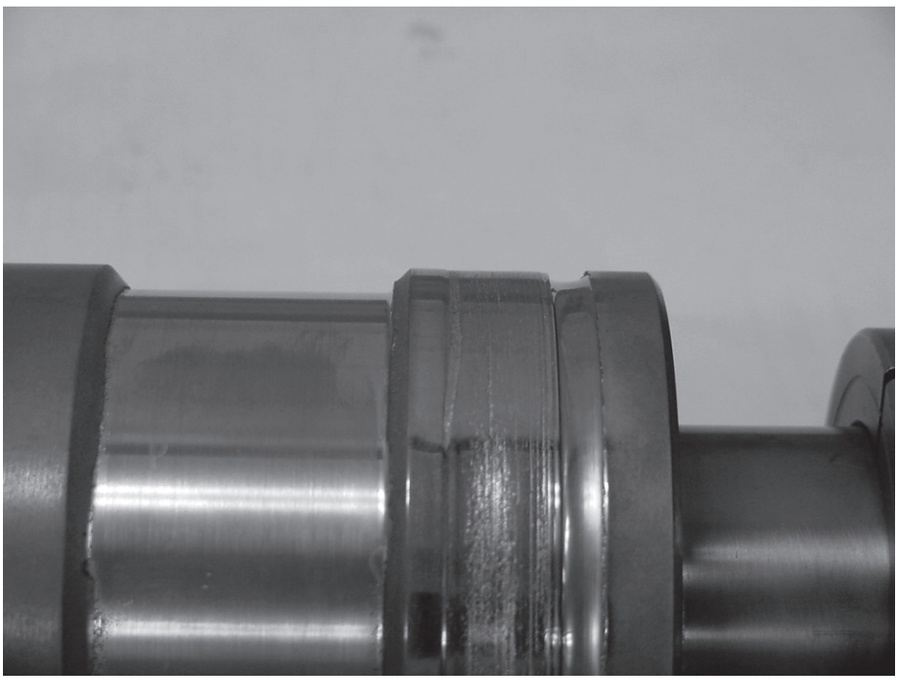

Figure 9. Discharge Wear Ring Test Fixture After Testing

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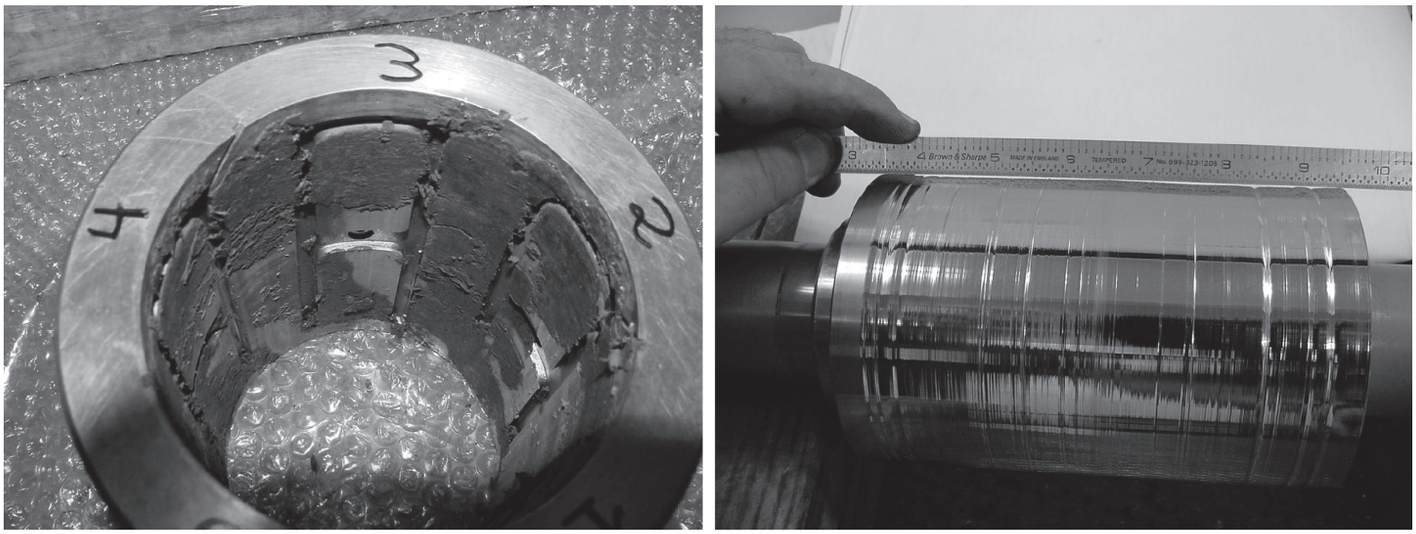

Figure 10. Hydrostatic Bearing and Shaft Sleeve Test Fixture After Testing

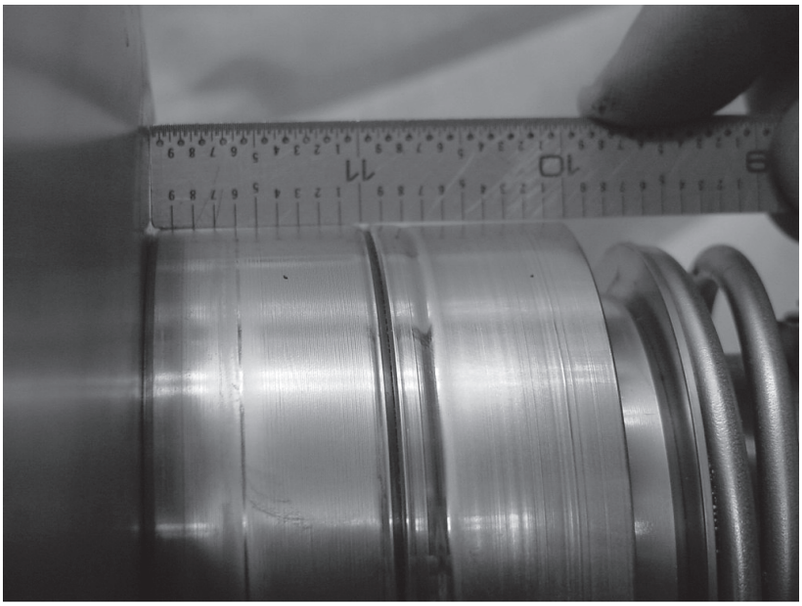

Figure 11. Central Volute Bushing Test Fixture After Testing

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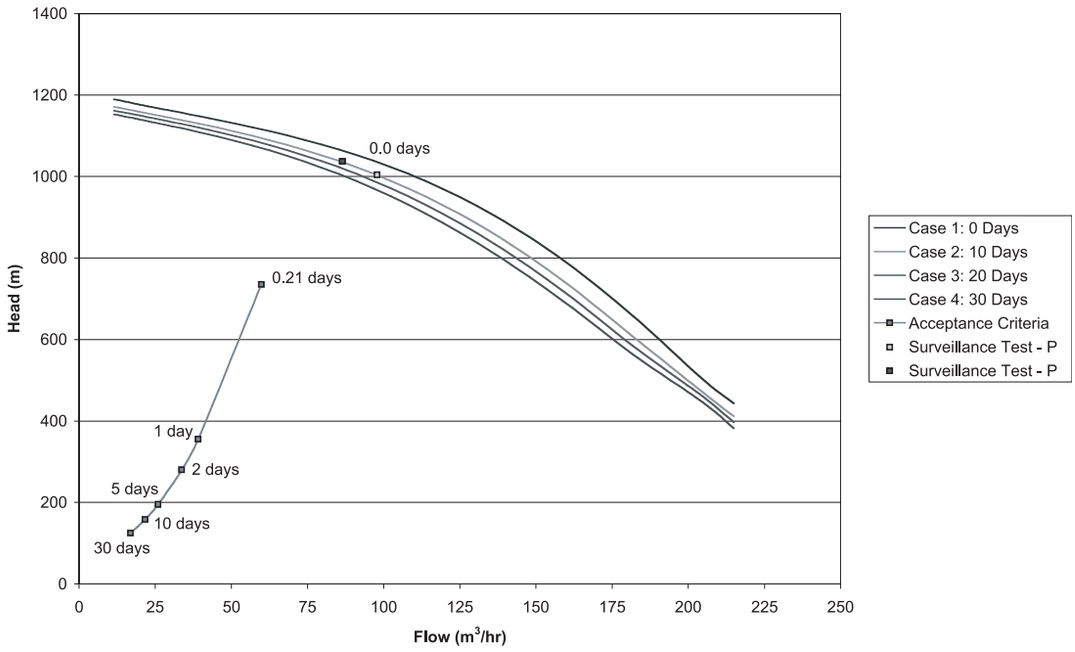

Figure 12. Hydraulic Analysis Results

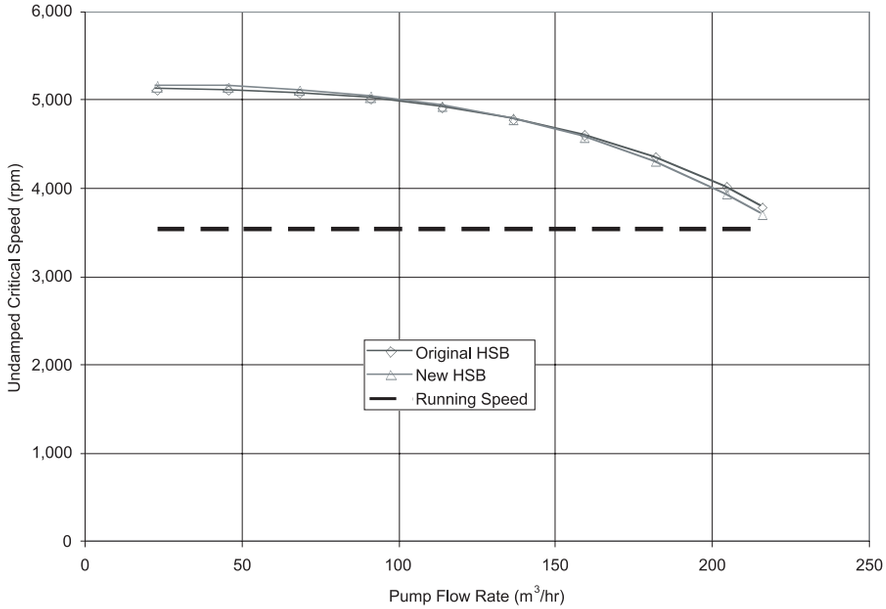

Figure 13. Critical Speed Map Comparing Effect of Original & New HSB's

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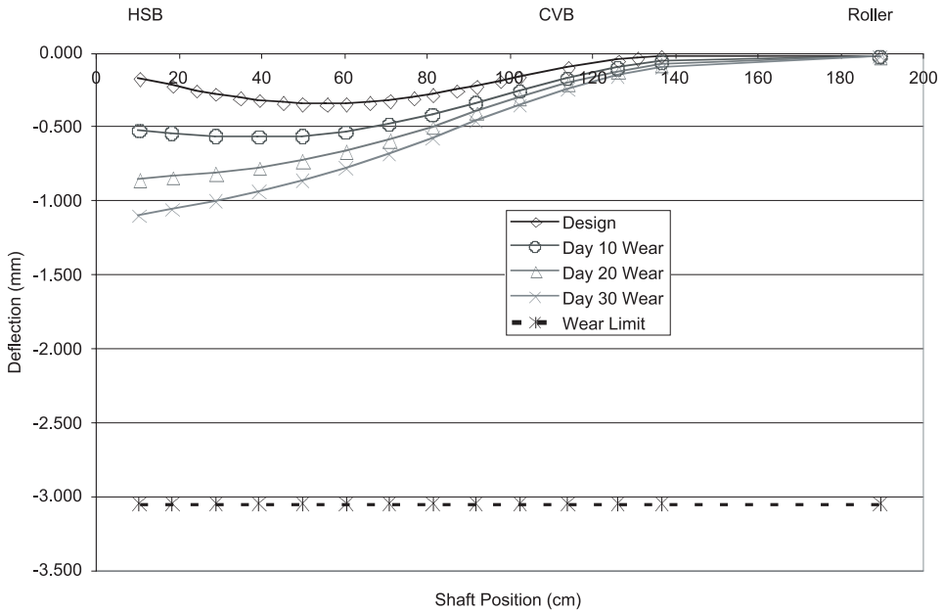

Figure 14. Rotor Deflection for HPI Pump with LOCA Conditions

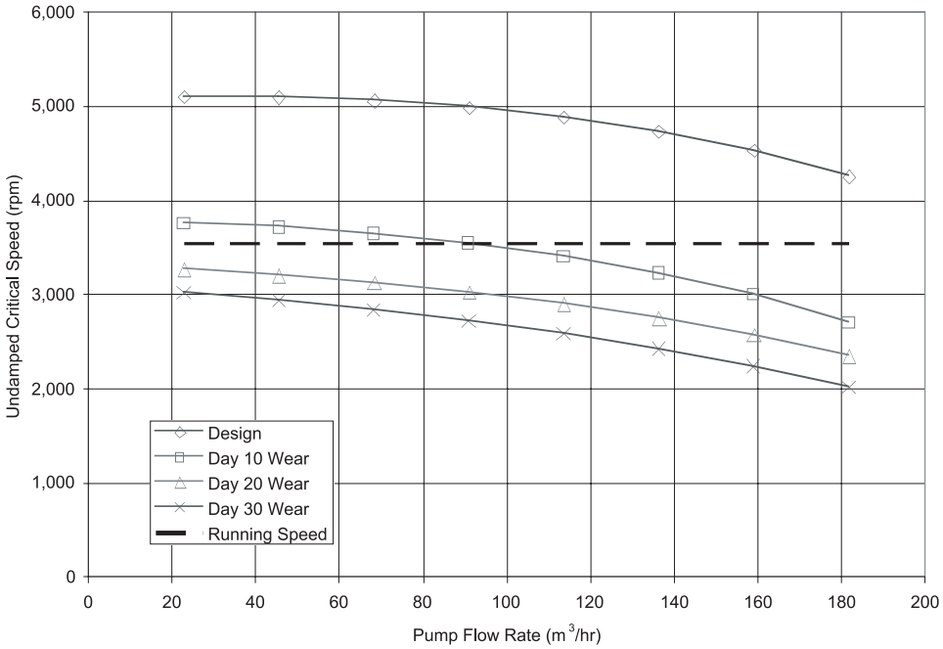

Figure 15. Critical Speed Map for HPI Pump with LOCA Conditions

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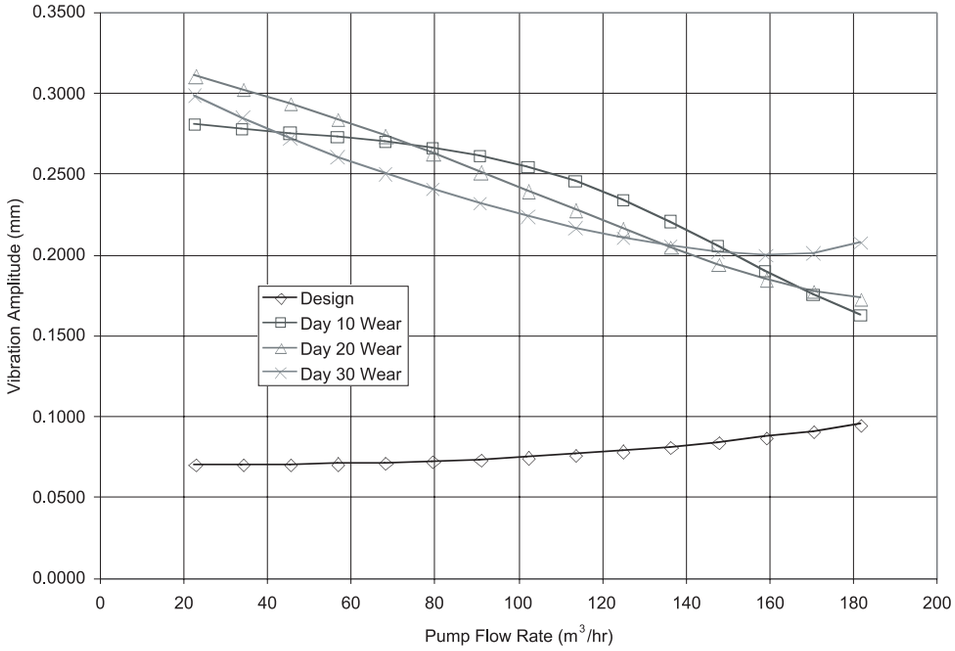

Figure 16. Forced Response Results for HPI Pump LOCA Conditions

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Web-Based Applications for Pump and Valve Condition Monitoring Pump and Valve Seal Designs

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Abstract

In the Utility Industry, companies have had difficulties in applying software technologies to standardize processes, facilitate communication between departments, and automate the generation of Performance Indicator Reports as pertaining to pump and valve condition monitoring and related seal designs and configuration management. This paper describes the development of two Web-Based Applications, PlantIQ™ and SmartSeal™, that have been implemented by large Nuclear producers to address these issues. These applications are an extension of existing infrastructure and are flexible and adaptable to the changing business needs of individual plants and corporate management. They have been designed with a simple, consistent user interface requiring minimal training and the Web-Based architecture keeps implementation costs low and facilitates integration with other systems.

1. Introduction

Pump and valve Condition Monitoring and the Management of their Sealing Designs and Configurations are maintenance processes that can benefit from the application of Web-Based Technologies. Standardization and the sharing of information are key elements to successful programs especially in a distributed environment of multiple sites; however, it has been cost prohibitive to extend the Computerized Maintenance Management System (CMMS) to automate these processes. In recent years, the implementation and deployment of Web-Based Applications as an augmentation to the Information Technology (IT) infrastructure has become cost effective with the advancement in development technologies and methodologies. In addition, it is now possible to provide a full-featured user interface in a browser that was thought only possible in client-server applications.

Condition Monitoring diagnostic technologies, computer programs, analysis techniques, communication flows, and performance metrics can vary significantly between plants, even though they are part of the same company. Analysis results of diagnostic data such as Periodic Vibration, Thermography, Lube Oil Analysis, etc. are often stored in spreadsheet and word processing applications on the hard drives of individual engineers, and are only shared in specific circumstances or discussed briefly in meetings. Data integration tools often lack the database infrastructure to represent the plant Equipment Hierarchy and to store the analysis results for each piece of Equipment and Technology. Plant personnel often spend significant time communicating equipment condition and gathering and organizing information in order to satisfy corporate Performance Indicator (PI) reporting requirements. A single Web-Based Application installed centrally and accessed throughout the Intranet can facilitate these needs.

Pump and valve sealing designs and configurations are detailed and require a significant number of fields and calculations that are specific to the plant application and component types used. The CMMS is effective for creating, scheduling, executing, and closing out Work Orders for repacking and replacing seals in pumps and valves. However, the complicated nature of sealing designs, and the fields and calculations required, have made it impractical and not cost effective to manage this data in the CMMS. Original Equipment Manufacturers (OEMs), sealing vendors, and service providers do provide tools, but they are often limited to specific applications and the materials and pumps and valves they supply and service. Plant engineering and maintenance personnel are therefore left with using tools like Microsoft Excel and Access to create localized databases to manage detailed information regarding pump and valve sealing designs and configurations. These processes and databases, however, can vary significantly between departments and sites, even within the same company.

The application of Web-Based technologies for automating business processes has a number of inherent benefits. With the pervasive use of the Internet for business and personal needs, users have become comfortable and familiar with the controls and the point and click navigational techniques implemented in standard Web-Browsers such Microsoft Internet Explorer. This lowers training costs when business applications are implemented using the same techniques and standards. Information Technology departments benefit from the server-based architecture that has minimal client

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desktop requirements. Change management and software maintenance are streamlined, resulting in lower deployment costs. In addition, integration between Web-Based applications can be extremely simple and low cost, and can result in high value to the end-user.

The remaining Sections of this paper discuss the implementation, results, and conclusions drawn from the collaborative effort of AP Services, Inc. and Insert Key Solutions, Inc. in applying Web-Based Technologies to address the needs described in the preceding paragraphs.

2. Implementation

AP Services, Inc. and Insert Key Solutions, Inc. have developed Web-based applications for automating and standardizing pump and valve condition monitoring and sealing design management. The tools are extremely flexible with configurable fields and forms and can match specific company processes and terminology. They are installed centrally as a single instance that can support an entire enterprise of sites, and can be integrated with existing applications such as diagnostic systems and the CMMS. In addition, user interfaces and the technical architecture have been designed to be simple in order to keep training and implementation costs low. These applications have been commercialized as PlantIQ™ and SmartSeal™, respectively, and are presented in the following Sections.

Condition Monitoring (PlantIQ™)

PlantIQ™ is a Web-Based software application that was developed to specifically address Component Health Reporting in the Nuclear Power Industry. It is focused on satisfying Equipment Reliability guidelines (AP-913) as recommended by the Institute of Nuclear Power Operations (INPO), specifically in the area of Performance Monitoring. The best example of implementation is that it has been installed in a large Nuclear Utility that has 10 distributed sites and 17 total reactors, and it has been implemented for more than 2 years.

Primary users are diagnostic data collectors especially related to PdM technologies, component engineers, and system managers. However, all organizational levels in a company interface with the application and can quickly retrieve information or perform data entry functions. On the Home Page, Figure-1, is a real-time Performance Indicator (PI) that demonstrates the health of the equipment throughout an enterprise of sites. A corporate manager, director, or VP can view this screen to obtain high level status information, and can also drill-down on any of the colored boxes to retrieve and view additional detail. Because of the communication that is facilitated, phone calls to the plant requesting status and related equipment health information are eliminated. Maintenance and Engineering personnel at the plant can therefore be focused on more value-added activities and expanding the Condition Based Maintenance Program to include more components.

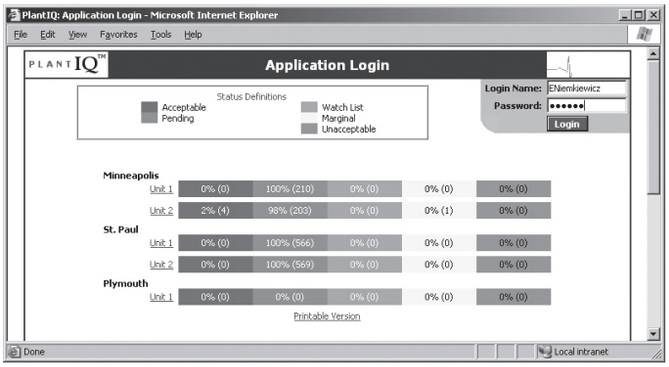
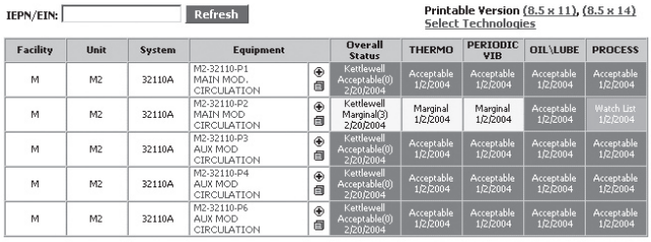


Figure-1 Home Page Performance Indicator

The core feature of the application is the Equipment Matrix, Figure-2, which includes Equipment and Condition Indicators. The equipment is organized in a hierarchical fashion, representing the actual equipment tag hierarchy used in the plant. The rows in the matrix or equipment can be filtered by the user based on complex criteria including System and Equipment Ownership, Component Category and Type, and Status levels. In addition, the Condition Indicators, or columns in the matrix, can also be configured based on the desire or specialty of the user. A person who focuses on rotating machinery, for example, can choose condition indicators such as Periodic Vibration, Thermography, Lube Oil Analysis, Operator Rounds, etc.; and, a switchyard specialist can choose indicators such as Dissolved Gas in Oil (DGA), Acoustics, Thermography, etc. These indicators are persistent upon login and can be easily adjusted by the user.



Facility	Unit	System	Equipment	Overall Status	THERMO	PERIODIC VIB	OIL\LUBE	PROCESS
M	M2	32110A	M2-32110-P1 MAIN MOD. CIRCULATION	Kettlewell Acceptable(0) 2/20/2004	Acceptable 1/2/2004	Acceptable 1/2/2004	Acceptable 1/2/2004	Acceptable 1/2/2004
M	M2	32110A	M2-32110-P2 MAIN MOD CIRCULATION	Kettlewell Marginal(3) 2/20/2004	Marginal 1/2/2004	Marginal 1/2/2004	Acceptable 1/2/2004	Watch List 1/2/2004
M	M2	32110A	M2-32110-P3 AUX MOD CIRCULATION	Kettlewell Acceptable(0) 2/20/2004	Acceptable 1/2/2004	Acceptable 1/2/2004	Acceptable 1/2/2004	Acceptable 1/2/2004
M	M2	32110A	M2-32110-P4 AUX MOD CIRCULATION	Kettlewell Acceptable(0) 2/20/2004	Acceptable 1/2/2004	Acceptable 1/2/2004	Acceptable 1/2/2004	Acceptable 1/2/2004
M	M2	32110A	M2-32110-P6 AUX MOD CIRCULATION	Kettlewell Acceptable(0) 2/20/2004	Acceptable 1/2/2004	Acceptable 1/2/2004	Acceptable 1/2/2004	Acceptable 1/2/2004

Figure 2 – Equipment Matrix

The condition indicator columns in the Equipment Matrix can also be drilled into to reveal more detailed data. Technology Exam reports are behind each colored block and can be manually populated or automatically generated through interfaces to underlying systems. Computerized Maintenance Management Systems (CMMS), Periodic Vibration Systems,

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Lube Oil Databases, and Process Historians are all good candidates for integration. Once the interfaces are built, Technology Exams not only reveal analysis results, but can link directly to detailed diagnostic data.

Another important feature of the Equipment Matrix is a column dedicated to Overall Health. The Component Owner, whose name is displayed in this column, is responsible for reviewing information from all the condition indicators and making an overall judgment. The detailed reports behind the overall status are called Equipment Assessments. Equipment Assessments, as shown in Figure-3, can have associated Technology Exams and File Attachments as supporting evidence, and these are permanently stored in the database. They can be queried as input into future condition assessments based on Component Type, Manufacturer, and from many other attribute fields.

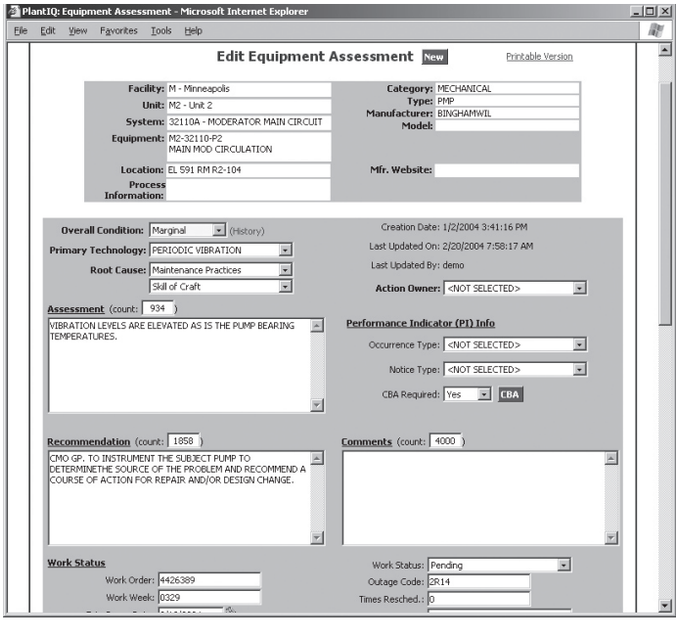

Figure 3 – Equipment Assessment

In addition to acting as 'Case Histories', Equipment Assessments collect the data necessary to generate real-time Performance Indicators (PIs). A sample PI is shown in Figure-4. As Equipment Assessments are created and updated, the summary data for PIs is automatically created. Fields on the Equipment Assessment forms can be customized to the specific process and terminology used by the sites, and custom PI Reports can be created. Reports are created using the Seagate Crystal Report Designer and the users have the ability to set complex criteria when running the reports through the application interface. It is possible for System Administrators to add custom reports to the system with no programming effort.

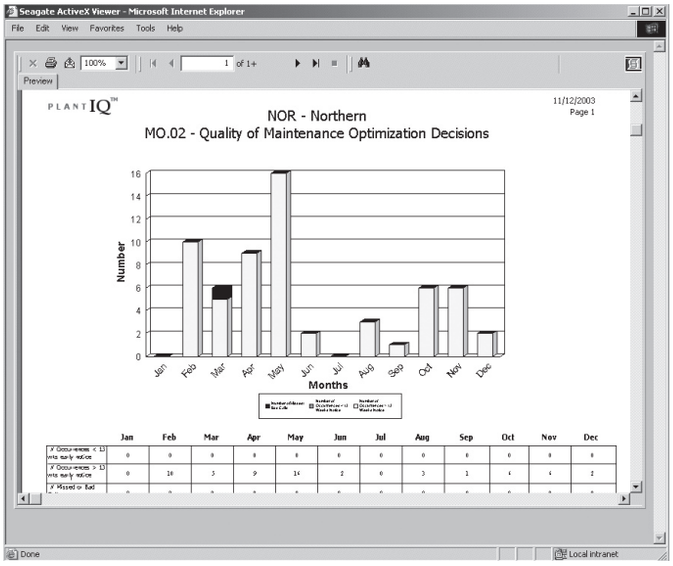

Figure 4 – Example Performance Indicator

Communication is further facilitated through the use of Email Notifications. During the creation or editing of Technology Exams and Equipment Assessments, the user can send an Email Notification with summary information to the Equipment and/or System Owner. If the user requires more information, an embedded hyperlink can be clicked that opens a new browser window and navigates to the detailed report with read-only Security Privileges. In addition, users can request to be notified by emails when specific events on Technology Exams or Equipment Assessments are edited or added.

Seal Designs (SmartSeal™)

SmartSeal™ is a Web-Based application that was developed to manage the documentation, maintenance, and configuration control of pump and valve packing, gaskets, and pressure seals. It provides an extension to the CMMS which is usually generic in its implementation and focused on work planning and scheduling, execution, and closeout. SmartSeal™ integrates equipment, maintenance work control, and procurement data with specific sealing information, to provide a complete view of sealing design and maintenance. Terminology and calculations can be configured with no programming effort in order to comply with changing corporate standards. Since the application is Web-Based, all the information, including equipment and work control data, can be integrated and shared with other applications. SmartSeal™ has been implemented in a large utility with three Nuclear Stations.

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Primary users are engineers in plant maintenance and engineering departments that interface with processes related to pump and valve sealing design and maintenance. Screens have been configured by the utility to meet the needs of component engineers with specific engineering data, maintenance engineers with work planning and history and diagnostic test results, and the maintenance craft with standard packing and sealing datasheets designed for specific groups. The user interface has been designed to provide self-service communication between groups, thereby eliminating unnecessary phone calls and emails requesting status and information.

The application implements a standard process for the verification and revision control of sealing designs and configurations. Each configuration record has a status of Future, Installed, or History and a verification status of Unverified or Verified. A future record contains information about a packing, gasket, or pressure seal configuration that is currently not installed in an operating system. A future record status means that a user can prepare and finalize all packing gasket or pressure seal information before the configuration is installed into an operating system. For a packing configuration record to reach the installed status, it must have originally existed as a future record, has been verified, and then installed. All installed records must be verified. It is not possible for an unverified record to be in the installed status. Once a packing record reaches the status of History, it remains in the system for reference purposes only and cannot be modified.

The software provides a Material Association feature, Figure-5, which is a user-definable catalog that can be searched for specific parts and materials and to associate them to a particular pump or valve. The catalog can be searched by plant stock code, vendor stock code, material types, and specific measurements. Material datasheets and bill of material reports can be automatically generated that include part and material lists, along with the association to specific pieces of equipment.

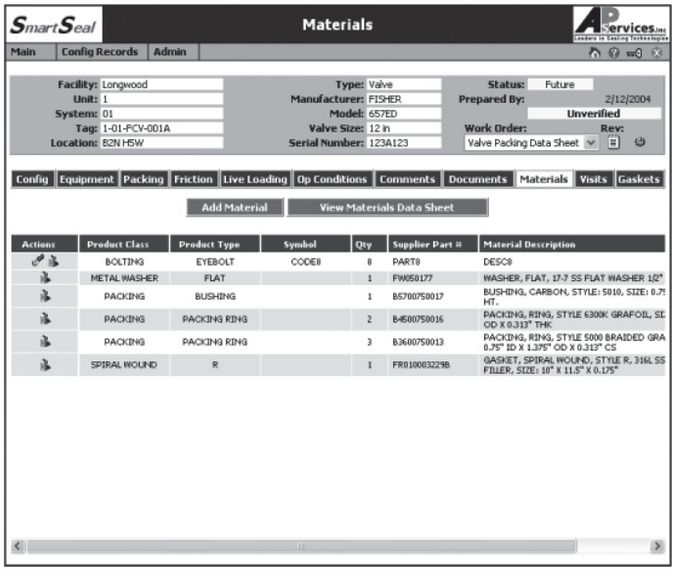

Figure 5 – Example Material Association

One of the key technical challenges that were overcome was to create configuration screens that were completely configurable by the system administrator. A system administrator can add or remove fields, change the sort order and location of fields, change text labels and headings, and modify calculated fields on each of the configuration forms. The format of datasheets used by the craft can be administered with the same degree of control. An example of a configurable form for packing details is included in Figure-6.

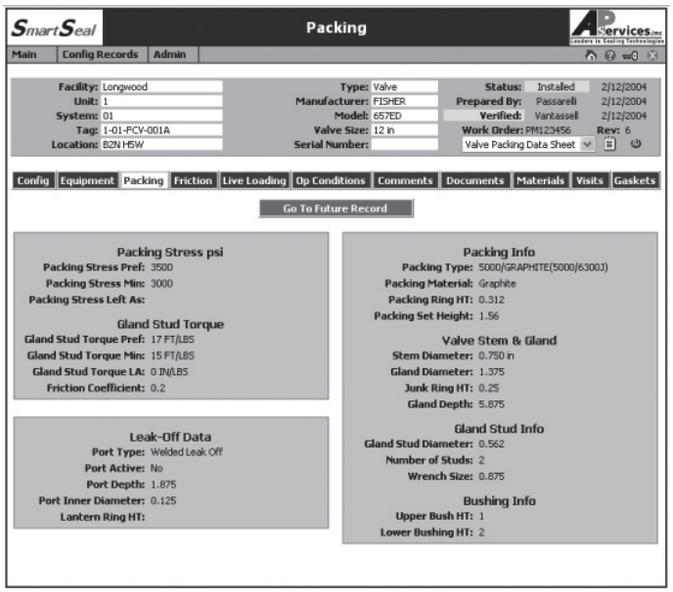

Figure 6 – Example Packing Details

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The application provides a framework for tracking all maintenance activity including diagnostic tests, leak tracking, re-packing, and the results of walk downs and surveillances. These events are stored chronologically and a permanent history is stored in the database for future reference. Figure-7 demonstrates the visit tracking functionality in the application.

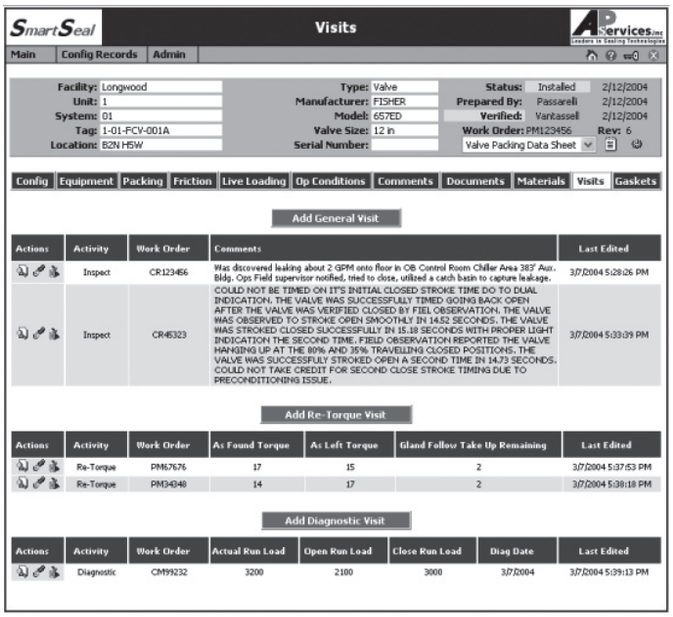

Figure 7 – Visit Tracking

In addition, the software has been implemented with a library of hundreds of configuration images to provide a visual representation of packing and sealing information, and to further aid the communication process. Functionality was created to upload and relate documents such as drawings, manuals, and flow sheets to specific pieces of equipment for easy retrieval. An area for managing gaskets and all the related configuration information also exists, an example of which is shown in Figure-8.

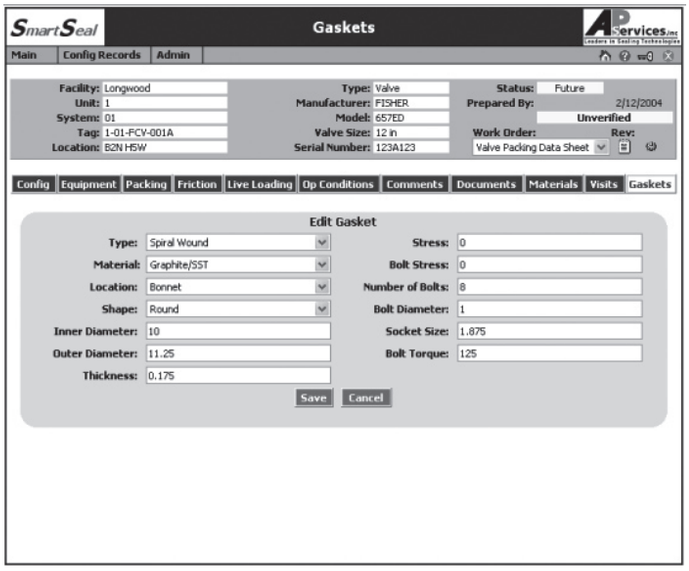

Figure 8 – Example Edit Gasket Detail

Technical Architectures

PlantIQ™ and SmartSeal™ have the same architecture and have been developed using the same technologies and methodologies. They are designed to be installed in a 2-tier environment consisting of a Web Server and Database Server. The desktop interface is displayed with Microsoft Internet Explorer 5.0 with very little reliance on client side controls. PlantIQ™ requires the Seagate Crystal ActiveX Viewer for viewing and exporting reports; and, both applications require MSXML3.0 components which in most cases are already installed on the client PC.

The presentation layers consist of Microsoft Active Server Pages (ASP) that can be rendered by Microsoft Internet Information Services 4.0/5.0. The Web Server operating system can be Microsoft Windows NT or 2000 and should have the latest service packs applied. In addition, PlantIQ™ requires the installation of the Seagate Crystal Reports Web Component Server, and both applications require ASP Upload 3.0 to be installed on the server.

Both applications support and can be installed in the Microsoft SQL Server and Oracle Relational Database Management Systems (RDBMS). The database objects consist of Tables, Stored Procedures, Functions, Indexes and Triggers; and, security is controlled at the application level through tables of user information. Both applications have tools and methodologies for loading and converting data from existing and supporting systems.

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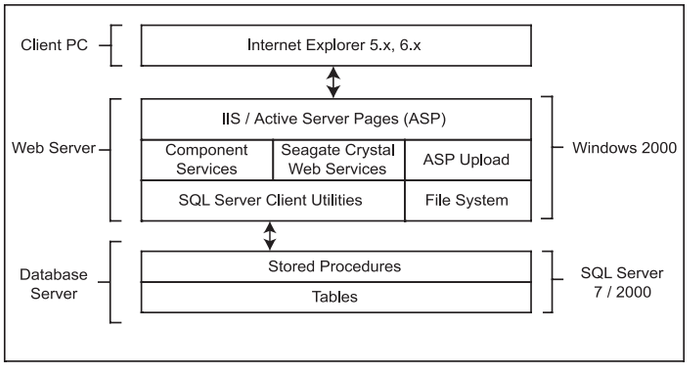

Figure 9 – Technical Architecture

3. Related Work

Condition Monitoring

There are a number of software products that are commercially available that address Condition and Performance Monitoring and have some impressive display capabilities. They are primarily focused on data integration and supporting hard-core analysis though the use of trends, calculations, and the application of statistical models. However, they usually do not promote the integration of all sources of information that are inputs into maintenance decision-making; and, they do not address other key elements that are time consuming to the maintenance engineer and technician.

One common deficiency is the lack of a repository for storing analysis results in a searchable database format. Analysis results are usually textual reports (problem statement, recommendation, Work Order information, action plans, status fields, etc.), and engineers are often left with no choice but to store this information in unsupported spreadsheet files and home-grown Microsoft Access Databases. In these formats, it is difficult to track, trend, and query this data and to share it with individuals at other plants. In the maintenance decision-making process, these reports are as important as the detailed diagnostic data and need to be considered in future analyses. It is very important to be able to search previous occurrences based on component type and other attributes to be able to identify trends and reoccurring problems. PlantIQ™ provides this functionality as a by-product of automating the normal Condition Monitoring process and not as an extra chore bestowed on the engineer.

Because most tools do not store analysis results, they also are not capable of supporting the automated generation of Performance Indicator (PI) Reports. It is at the overall condition assessment level of a piece of equipment where most PI data is generated. Simple elements such as the status, the primary technology indicator responsible for the assessment, cost benefit information, and whether it was detected or missed by the condition-based maintenance program, can be used to generate meaningful results. Here again, PlantIQ™ provides a flexible and powerful reporting infrastructure that supports current and future PI reporting requirements of the company.

Seal Designs

Seal design and configuration management software is readily available in the current marketplace, most of which is provided by individual sealing material manufacturers. These tools tend to focus on vendor specific processes and do not extend to support related procedures and other data consumers. In addition, most have not been developed to address the specific needs of the utility industry and can not be configured to address enterprise needs due to the traditional client-server architecture. Without a Web-Based architecture, data contained in the application is only accessible by those users that have the software installed on their desktop, resulting in a situation similar to storing the data in a Microsoft Excel spreadsheet or Access database. Obscure or non-standard desktop database engines are often used in existing products, making it difficult to integrate data from other sources and equally challenging to make seal design information available to other applications. SmartSeal™ provides utilities with a powerful and flexible software tool that can meet the needs of their current pump and valve sealing program. Because it is web-based and developed on standard enterprise databases, it can be integrated with existing and future applications with minimal effort.

4. Summary, Results, and Conclusions

PlantIQTM and SmartSealTM have been successfully implemented in large nuclear energy producing organizations. The Web-based architecture of the applications helps solve common business process problems such as the sharing of information, standardization of processes, and the integration of data and improve condition monitoring and the management of sealing designs. The following bullets provide additional results and conclusions in support of this summary:

- Software programs PlantIQ™ and SmartSeal™ have been developed and implemented to support corporate and an enterprise of sites with a single, centralized installation. With simple 2-tier architecture, hundreds of users can be supported with acceptable performance. The applications are also capable of monitoring and managing tens of thousands of pieces of equipment with an efficient database design, within standard database platforms such as Microsoft SQL Server and Oracle.

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- Implementing Web-Based applications assists in standardizing processes. Initial presentation of integrated Performance Indicators with data from multiple sites demonstrated inconsistencies in the process. Communication was quickly initiated between the sites and corporate in order to agree on standards, and to close any terminology and expectation gaps. Sites then began to compete in order to improve Performance Indicator results.
- Engineering and Maintenance departments can save significant time and resources by implementing self-service applications that share information and reduce manual communication processes for status and information requests. Incorporating simple functionalities such as email notifications and 'read only' access can provide high value at low implementation costs.
- The generation of Performance Indicator Reports can be automated, saving significant corporate and plant man-hours. These PIs can be real-time and can be created as a by-product of the process, as opposed to an extra task on the engineer.
- With proper planning during development, it is very easy to integrate these Web-Based applications to provide significant value.
- When Web-based applications are implemented, it is easy to integrate condition monitoring data and results with sealing designs and configuration management. The simplest example is creating a reference or hyperlink to sealing design and configuration management data from a component in the condition monitoring tool.

5. Future Benefits

The majority of the integration work between applications with regard to pump and valve condition monitoring and sealing design and configuration management has been internal within a company. Interfaces are created with the CMMS and other diagnostic systems and tools. However, little effort has focused on integration with industry sources such as INPO's EPIX failure database, various Electric Power Research Institute (EPRI) data sources, and OEM databases for product catalog information and purchasing. Technologies such as XML (Extensible Markup Language) which are commonplace in other industries and facilitate business to business integration have not been wide applied in the Utility industry. Exposing information on the Internet with secure sharing and on-demand querying will become more prevalent.

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Reevaluation of Comprehensive Pump Testing and Pump Design Flow Considerations

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ABSTRACT

This paper has been developed as a result of the Task Group of SG-ISTB responsible for reevaluating the Comprehensive Pump Test (CPT) requirements for certain pumps and is considered a work-in-progress. As a result of several inquiries submitted to the committee, the Sub Group has responsibility for evaluating the issues and formulating responses and/or changes to the Code as necessary. The observations and considerations presented in this paper are my own and not to be interpreted as that of ASME, the NRC nor Altran Corporation.

This paper will describe the background of CPT, current industry concerns and issues with the CPT, discussion of alternatives to the CPT, and concluding plans to address future Code changes as necessary.

Coupled with this paper is the evaluation and actions underway to address the term "pump design flow rate". The ASME OM Code 1994 and later editions have incorporated comprehensive pump testing. One of the requirements of the comprehensive pump test is to establish reference values within ± 20% of pump design flow rate. No definition of "pump design flow rate" is provided. This paper will discuss actions underway to address this issue.

Background - Pump Design Flow Rate

The ASME OM Code 1994 and later editions have incorporated comprehensive pump testing. One of the requirements of the comprehensive pump test is to establish reference values within ± 20% of pump design flow rate. No definition of "pump design flow rate" is provided.

The intent of this change to the Code addressed the testing of pumps using minimum flow lines, which have limited ability in detecting pump degradation. Testing of pumps at higher flow rates and on the portion of the pump curve which is well sloped, increases the ability to detect degradation. The Code change to perform comprehensive pump testing within ± 20% of pump design flow rate ensures that the pump is tested at a point at which pump degradation is readily detectable. However, the current Code does not define *Pump Design Flow Rate.*

Typically, the pump designer will select the design of the pump based on the procurement specifications which include required system flow, pressure and temperature. The pump is then designed such that the Best Efficiency Point (BEP) and pump design flow include all system demands and optimizes power consumption, smoothness of operation and component reliability.

In general, the manufacturer will try to design the pump such that the design point is as close to the BEP as possible. This optimizes the performance of the pump. The BEP is typically at a substantial flow rate and on a portion of the curve that is well sloped.

However, for some older plants, cases have been identified where the pumps' have been designed for much more capacity than is required by the system. In theses cases, the BEP flow cannot be achieved by using the as-built system configuration. The pumps can deliver the flow required by the system to perform its safety function, thus the system required flow would be considered the pump design flow.

As an example, a boric acid transfer pump operates during normal power operations at minimum flow conditions to recirculate the boron injection tank contents. Typically, these flow rates may be less 20 gpm (gallons per minute). During accident conditions, the pump must be capable of delivering a higher flow (60 gpm). However, in some cases the pump is designed such that the BEP is more than twice the design point (125 gpm). See Attachment 1, Boric Acid Transfer Pump Characteristic Curve.

In other cases some pumps have been designed such that the rated conditions supplied by the designer are well above the best efficiency of the pump. The attached containment spray pump curve indicates a design point at run out conditions (2000 gpm). This is the manufacturers' rated condition of the subject pump. In this case the BEP is approximately 1300 gpm. The accident analysis flow for the subject pump

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is 1450 gpm (between the rated and BEP flow points). See Attachment 2, Containment Spray Pump Characteristic Curve.

A proposed change to the Code is necessary to alleviate the inconsistencies in what is defined by each plant as "*Design Flow Rate*" for each pump. The following Proposed Code Change would benefit the industry and allow the Owner to determine and document a point on the curve where pump testing may be performed and degradation be detected. This point may be based on system flow requirements, design or rated conditions, Best Efficiency Point, or any other point where testing is effective in detecting mechanical and hydraulic degradation during subsequent testing.

Comprehensive Pump Testing

The ASME OM Code 1994 and later editions have incorporated comprehensive pump testing (CPT) to ensure that pumps are periodically tested at, or near design flow conditions. Typically, the design point is at a substantial flow rate that is on a portion of the pump curve that is well sloped. Degradation at the design flow conditions is more easily detected than at a minimum flow condition where the pump curve is generally flat. Comprehensive Testing was included to consider the requirements of Generic Letter 89-04 that testing at minimum flow was inadequate.

Testing at or near design flow conditions provides reasonable assurance that the pump will perform its intended design function during accident conditions. However, because the comprehensive test interval was extended to two years, the Code requires that more accurate pressure instrumentation be used when performing the comprehensive pump test and tightened the acceptance criteria.

The intent of the present Code addresses the issue of minimum flow testing, which has limited ability in detecting pump degradation. Testing of pumps at higher flow rates and on the portion of the pump curve that is well sloped, increases the ability to detect degradation.

Current Summary of Code Changes for Group A, B and Comprehensive Pump Testing

General

The Owner is required to categorize all pumps as either group A or B. Group A and B tests are required to be performed quarterly, while the CPT is performed biennially for all pumps. Group A and B tests are performed within ± 20% of pump design flow rate if practicable, while the CPT is *required* to be performed within ± 20% of pump design flow rate. For the group A and CPT, a minimum run time of 2 minutes after conditions are stable is required prior to recording the test parameters. No minimum run time exists for the group B test.

Instrumentation

Instrumentation accuracy requirements are the same for all parameters and all test types except that pressure measurement instrumentation for the CPT is required to be ± 0.5 % versus ± 2% for the group A and B tests.

Test Procedure

All tests are performed with the pump operating at a specified reference point. For the group B test, either the differential pressure or flow rate is determined and compared to its reference value. For the group A and CPT, the pump is operated at either the differential pressure of flow rate reference point (set parameter) while the other parameter is determined and compared to the reference value. Vibration measurements are not required for the group B test.

Acceptance Criteria (Centrifugal)

The vibration criteria for the group A and CPT are identical. The hydraulic criteria lower required action ranges are the same (0.9 x Ref) for all tests. The hydraulic criteria upper required action ranges are the same for the group A and B (1.10 x Ref) while the CPT upper required action range is 1.03 x Ref. No alert range exists for the group A and B hydraulic parameters.

Discussion

The CPT was developed and incorporated into the ASME OM Code to ensure that all pumps, required to be in the Inservice Testing Program, are periodically tested within ± 20% of the pump design flow rate. This OM Code requirement institutes the following two fundamental IST requirements.

1. The CPT ensures that each pump is tested periodically at a substantial flow rate point on the curve which is well sloped and where degradation may be easily detected.
2. Also, by performance of this substantial flow test, the pump is verified to be capable of performing its intended design function. It is important to note that the purpose of IST is to assess component operational readiness and not system requirements. In general, the plant Technical Specifications govern the requirements of system operability.

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The intent of the ASME OM Code requirements is to ensure that components (pumps and valves) that are required to perform specific functions in accident mitigation and shutting the reactor are assessed periodically to provide reasonable assurance of operational readiness.

In the case of group A pump testing where the pump is tested periodically at or near design flow (± 20% of pump design flow rate), the pump is tested at a point on the curve which is well sloped and where degradation may be easily detected. Additionally, testing at this point on the curve (substantial flow) provides reasonable assurance of operational readiness. The fundamental requirements of the IST requirements are therefore met by performance of the pump test at or near design flow.

Issues

Several issues exist with the 1994 through 2000 edition of the ASME OM Code regarding the testing of group A pumps.

1. Current quarterly group A pump testing coupled with biennial CPT is far more effective in assessing pump operational readiness than a quarterly group B coupled with a CPT provided the Group A test is performed at substantial flow rate and is at a point on the curve that is well sloped.
2. Group A pump testing at or near design flow (± 20% of pump design flow rate) is far more effective at assessing the pumps' operational readiness than a biennial CPT. Albeit, the pressure instrumentation is more accurate, the frequency of performance will not yield enough data over the life of the plant to equally assess the performance of a routine group A test at or near design flow rate.
3. Several plants have expressed concerns with exceeding CPT hydraulic acceptance criteria while performing a quarterly group A test. Engineering judgment in this case is the only means for a plant to maintain the operability requirements of their respective Technical Specifications. This issue should not reside in the interpretation of the Code.

   (Note: This should not be an issue. The current Code does not address this issue)
4. In general, for all pump types, only the hydraulic acceptance criteria in the latest OM Code differs between group A and a CPT test. The mechanical vibration acceptance criteria and alert ranges are identical for various pump types. While the CPT has an alert range for hydraulic performance, the group A does not (for centrifugal).

   (Note: Although the CPT employs an alert band for hydraulic criteria, the corrective action requirements specify that the test frequency be doubled or the condition corrected. The resultant test frequency would equate to one year.)
5. Pumps that operate routinely, (group A) are in most instances more susceptible to mechanical and hydraulic degradation and failure than pumps that are operated only for testing (group B) and are in standby during normal plant operations. It is recognized that group A pumps, therefore should receive a higher care regime than a group B pump.
6. Group A Quarterly testing at substantial flow rates (at a point on the curve that is well sloped) and where degradation is easily detected, provides reasonable assurance of the pumps' operational readiness. Quarterly pump testing at substantial flow rates represents a better overall test philosophy compared to the method involving a periodic reduced flow test, supplemented by a biennial CPT.

Conclusion

Therefore any pump (Group A or B) that is routinely Group A tested at a flow rate that is equivalent to the comprehensive pump test flow rate need not have an additional comprehensive pump test requirement.

Group B pumps which are not routinely Group A tested, would still require a biennial comprehensive pump test. Note that the Code allows a Group A test to be substituted for a Group B test.

Proposed Code Change – Comprehensive Pump Testing

Remove CPT requirement for any pump that is routinely tested at a flow rate which is equivalent to the CPT flow rate. This can easily be done by adding a Note to the Frequency Table, ISTB-3400-1. The following summarizes the Code change:

TABLE ISTB-3400-1 Add to Comprehensive Test Column "Note 1"

TABLE ISTB-3400-1 Add under GENERAL NOTE "(1) If a Group A test is performed quarterly at a reference flow rate that is equivalent to the comprehensive pump test flow rate, a comprehensive pump test need not be performed."

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Proposed Code Change – Pump Design Flow Rate

The proposed change to the Code is presented below. This change effectively removes all references to the term *Pump Design Flow* and requires the Owner to establish a pump test flow rate that is effective for detecting degradation. The following summarizes the change:

ISTB-1400 Add "(c) establish a pump test flow rate for each pump. In the context of ISTB, the pump test flow rate is determined by considering system flow rate requirements and pump best efficiency point. The specified pump test flow rate shall be effective for detecting mechanical and hydraulic degradation during subsequent testing[1]. The pump test flow rate and its basis shall be recorded in the Pump Records, ISTB-9100."

ISTB-1400 Add "Note 1. Except for positive displacement pumps, this pump test flow rate is at a relatively high flow point on the pump curve where relatively small changes in flow rate results in relatively large changes in differential pressure."

ISTB-3300(e)(1) Revised as "Reference values shall be established at the pump test flow rate for the comprehensive test, if practicable. If not practicable, the reference point flow rate shall be established at a point effective for detecting mechanical and hydraulic degradation."

ISTB-3300(e)(2) Revised as "Reference values shall be established at the pump test flow rate for the Group A and Group B tests, if practicable. If not practicable, the reference point flow rate shall be established at the highest practical flow.

ISTB-5110(a) Revised as "….If practicable, these points shall be from pump minimum flow to at least the pump test flow rate…."

ISTB-5210(a) Revised as "….If practicable, these points shall be from pump minimum flow to at least the pump test flow rate…."

ISTB-9100(d) Add "(d) the pump test flow rate and its basis."

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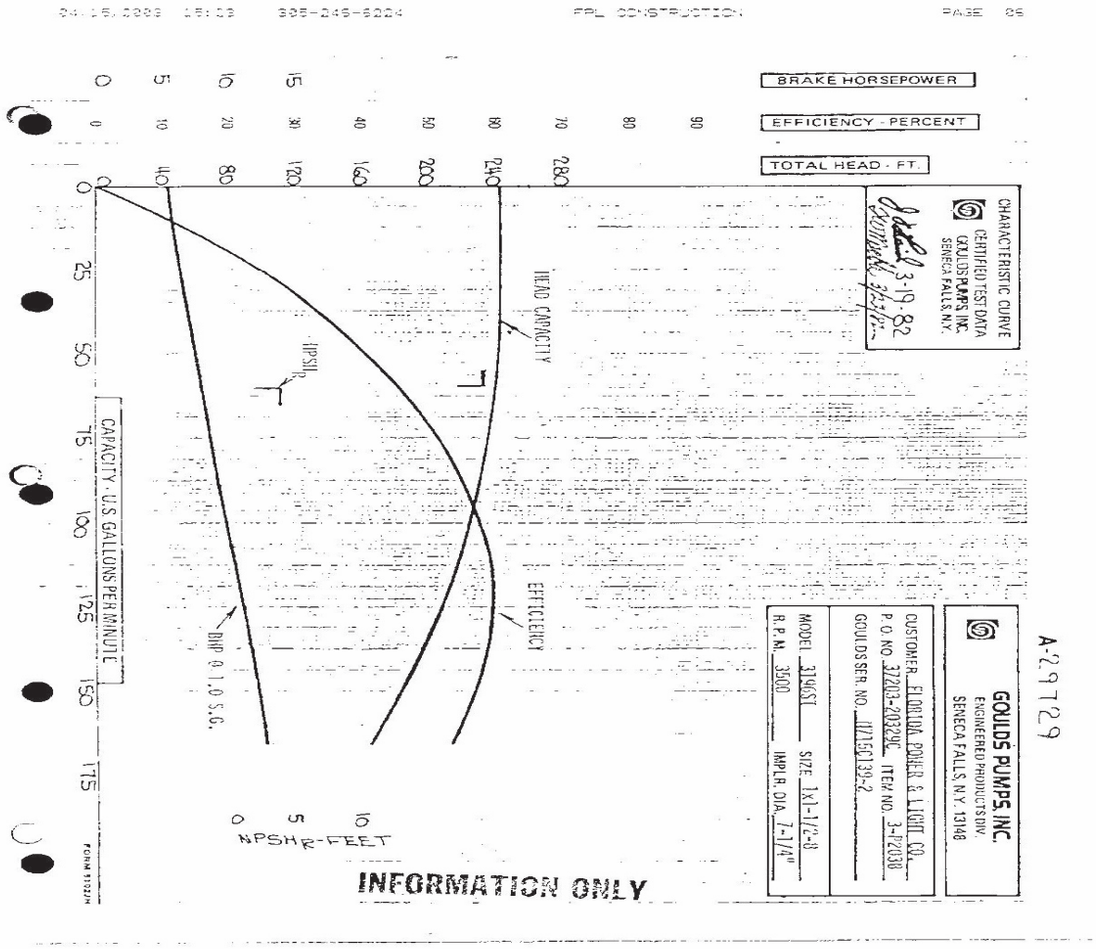
Attachment 1

Boric Acid Transfer Pump Characteristic Curve



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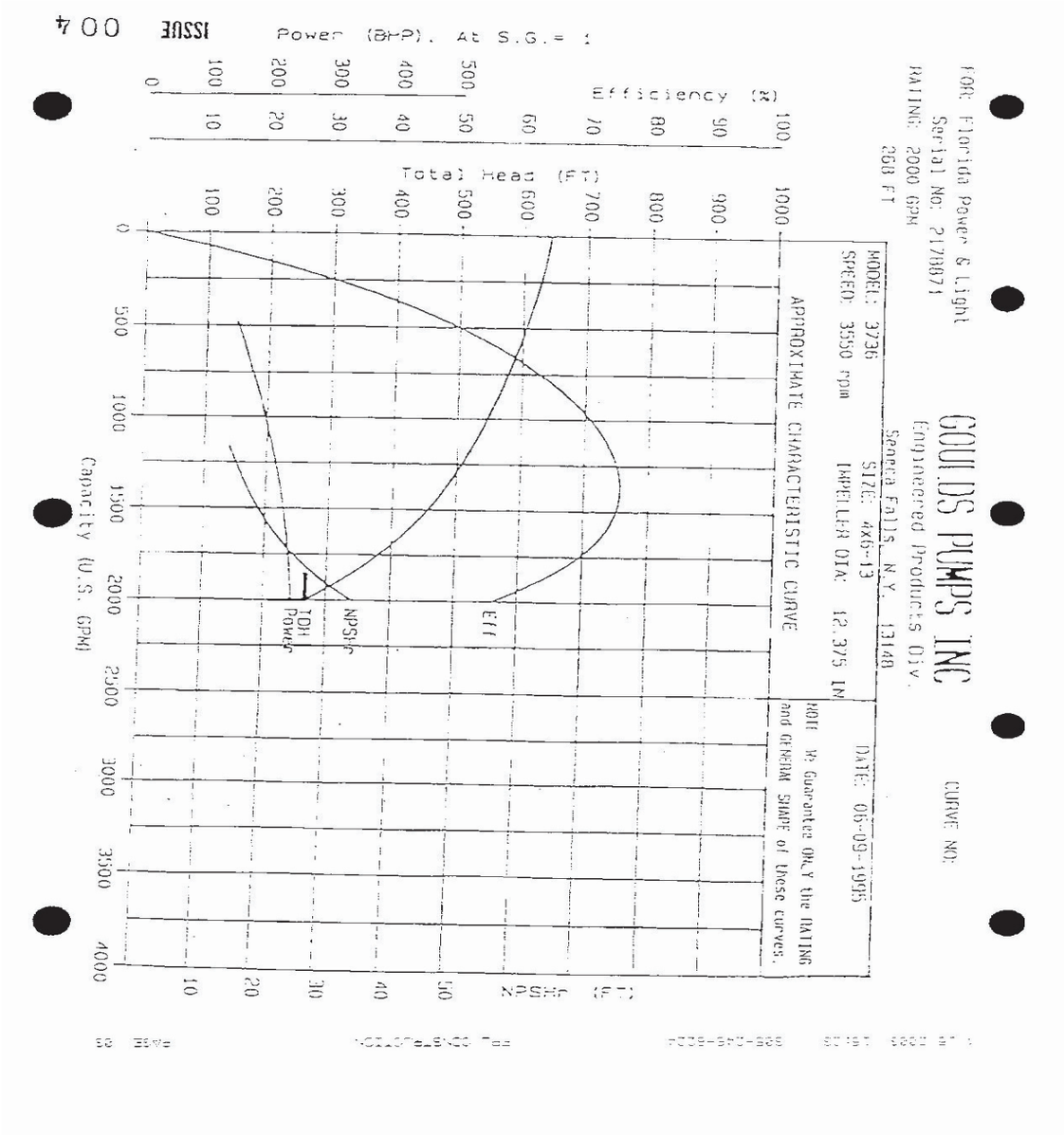
Attachment 2

Containment Spray Pump Characteristic Curve



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Inservice Testing Owner's Group (ISTOG)

Shawn Comstock
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Contributions from
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Abstract

The purpose of the ISTOG is to collect, integrate, and share industry knowledge, resources, and products so that owners will benefit from improved implementation of their inservice testing programs. The benefits of this collaborative effort include cost reduction, error reduction, improved performance, aging workforce knowledge capture for future generations, and increased regulatory influence.

The 8th NRC/ASME Symposium on Valve and Pump Testing will mark the official introduction of the IST Owner's Group (ISTOG). This presentation spot will be an open session of the IST Owner's Group that will cover topics of interest determined by feedback from every IST Owner who chose to participate. At the conclusion of the session, a question and answer period will be held to address specific survey topics more in-depth or to cover areas not addressed by the IST Owner's community that participated in the survey.

Introduction

The IST Owner's group was an idea that initially came to fruition in 2003. Gregg Joss of Ginna organized a joint meeting for Appendix J and IST Engineers in conjunction with the ASME Committee for Operation and Maintenance for Nuclear Power Plants and the Nuclear Industry Check Valve Group at the June meetings in Scottsdale. The meeting was a resounding success and punctuated the need for a group dedicated to the implementation of both Appendix J and Inservice Testing implementation. In the months that followed, Gregg Joss of Ginna and Shawn Comstock of Wolf Creek worked together to organize a Steering Committee. Today, the ISTOG Steering Committee is a 7 member team comprised of Bob Parry, David Chiang, Gregg Joss, Jeff Neyhard, Leonard Firebaugh, Shawn Comstock and Wavel Justice. The IST Owner's Group is open to any interested parties that wish to participate in activities dedicated to the improvement of Inservice Test Program implementation.

True North Consulting's Ron Lippy and Don Horn hosted the first open ISTOG Steering Committee meeting in conjunction with the December Nuclear Industry Check Valve Group meetings held in Orlando, Florida. This meeting included the participants of an IST Training seminar provided by True North Consulting during that week. Significant progress was made in the formalization of the group and those present agreed to formally introduce the IST Owner's Group at the 8th NRC/ASME Symposium on Valve and Pump Testing.

Industry Need

In the nuclear industry, the position of IST Engineer is not a highly sought after responsibility. This is evidenced by the turnover rate, which has averaged about 50% over a three-year period. In addition to the high turnover rate, the more experienced people in this field are getting nearer to retirement every day. Numerous complexities are interwoven into the position of IST Engineer that requires an understanding of multiple ASME Codes and NRC Regulations to be effective in the application of this Program Management responsibility. In addition to this complexity, knowledge about the numerous changes with ASME Codes and NRC Regulations is important to understand how modern IST Program elements have evolved into their present state to avoid the mistakes of the past. The IST Owner's Group seeks to provide an industry support network for the IST Engineer to turn to for answers.

IST Issues of Interest

Different surveys have been conducted to determine what IST Engineers are concerned about. Several different areas of interest emerged. This paper will discuss each one briefly; however, the intent of these discussions is to provide an overview for an interactive discussion at the 8th NRC/ASME Symposium on Valve and Pump Testing rather than an in-depth analysis of each subject. The top two areas of interest

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in 2003 were Preconditioning and Leak Testing vs. Close Testing. The complete list of topics of interest identified to date is as follows (in no particular order):

- Preconditioning
- Risk-Informed IST Implementation/transition guidance
- Flow Loop issues
- Position Papers (Endorsed by ISTOG or Used by utility)
- Condition Monitoring Justifications
- NRC Q&A – guidelines for unwritten processes
- Relief Request templates
- Guidelines for limiting values
- Skid-Mounted How-To (justifications)
- Practical/Practicable differences
- Sample valve passivity (how to justify passive classifications)
- Code Class 1,2,3, Augmented Guide
- Leakage Testing Versus Flow Diversion
- Design Flow Rate guide
- GL 89-04 applicability and NRC new viewpoint
- Code implementation Relief Request Guideline
- Submittal process (program and relief request how-to)
- PMT Guidance
- RCS PIV testing improvement project
- NUREG 1482 development participation
- CV Condition Monitoring How-To
- Terminology Guide
- Condition Monitoring vs Performance Based difference
- Instrumentation accuracies for pump testing
- Compliance with Ambient and Media Temperature Correlation Rules

Preconditioning

by Shawn Comstock

Preconditioning has to be one of the most mobile targets involving the IST Engineer. Preconditioning is described by NRC Information Notice 97-16, "Preconditioning of Plant Structures, Systems, and Components Before ASME Code In-Service Testing or Technical Specification Surveillance Testing"[1]. Some preconditioning is acceptable and some is not.

Acceptable preconditioning is the alteration, variation, manipulation, or adjustment of the physical condition of a plant structure, system, or component (SSC) before Technical Specification surveillance or ASME Code testing for the purpose of protecting personnel or equipment or to meet the manufacturer's recommendations. Preconditioning for purposes of personnel protection or equipment preservation should outweigh the benefits gained by testing only in the as-found condition. This preconditioning may be based on the equipment manufacturer's recommendations or on industry-wide operating experience to enhance equipment and personnel safety. This preconditioning should be evaluated and documented in advance of the surveillance.

Unacceptable preconditioning is the alteration, variation, manipulation, or adjustment of the physical condition of an SSC before or during technical specification surveillance or ASME Code testing that will alter one or more of an SSC's operational parameters which results in acceptable test results. Such changes could mask the actual as-found condition of the SSC and possibly result in an inability to verify the operability of the SSC. In addition, unacceptable preconditioning could make it difficult to determine whether the SSC would perform its intended function during an event in which the SSC might be needed. Influencing test outcome by performing valve stroking, preventive maintenance, pump venting or draining, or manipulating SSCs does not meet the intent of the as-found testing expectations described in NUREG-1482, "Guidelines for In-service Testing at Nuclear Power Plants" (April 1995)[2], and may be unacceptable.

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Leakage Testing Versus Flow Diversion

by Shawn Comstock

Leakage testing versus flow diversion is another topic of interest to IST Engineers. Under the ASME O&M Code, in-scope valves with a specific analyzed leakage limit are classified as Category A valves, which requires a leak test at least every 2 years. Valves in the scope of IST without a specific limit, which often prevent flow diversion, are classified as Category B valves and do not have to be leak tested. IST Engineers are often challenged on the issue of why a valve is not leak tested.

Certain valves have a calculated limit for a specific application. These are ASME Category A valves. Other valves prevent flow diversion and the allowable amount of flow is system dependent, so the amount of leakage allowed at any given time can vary. The IST OM Code is a component-based code and not a system based code (like the Appendix J program); therefore, it is not the purpose of the IST program to perform system leakage tests.

Condition Monitoring Justifications

by Shawn Comstock

Use of an option under the modern IST OM Code for check valve testing involves the research and justification of activities for use in a check valve condition monitoring program. The two purposes of adding check valves to a condition monitoring program are to either optimize test and maintenance activities or to improve reliable performance. Overall industry check valve performance could be enhanced if this documentation is shared between sites on an Internet database. This would also improve the IST Engineer's ability to rapidly implement check valve condition monitoring activities in a comprehensive manner that incorporates the industry's best practices.

RCS PIV Testing Improvement Project

by Shawn Comstock

Every site is required to follow a Technical Specification for quantifying the leakage of pressure isolation valves (PIV) that comprise the reactor coolant system (RCS) boundary. This testing often impacts the refueling outage critical path schedule. Outage Managers working with the Westinghouse Owner's Group have consistently identified this task as a top 10 area for improvement. With the current level of experience and knowledge in our industry about RCS PIV performance, it is believed that the incorporation of a performance based or condition monitoring approach as an alternative in plant Technical Specifications can maintain an acceptable level of safety assurance at a reduced impact to the outage schedule. ISTOG is an organization uniquely positioned to provide the technical expertise for this project.

Practical/Practicable Differences

by Wavel Justice

When ASME Section XI Articles IWP and IWV were replaced by OM Part 6 and Part 10, the term practical was replaced by practicable. The stated reason for the change is that practicable describes that which can be placed into effect and practical describes that which is also sensible and worthwhile. Given enough money and time any test is practicable and you would not need cold shutdown or refuel only testing. Practical is clearly the intent of the Code in many cases.

Submittal Process (Program and Relief Request How-To)

by Wavel Justice

In the past, there have been two major differences in the way Owners have submitted their IST program plans and relief requests. Some have submitted their programs just to be filed (information only) and relief requests to be approved, while others have submitted both their program and relief requests for approval. In the past, specialized contractors were available for detailed program reviews upon request. Today, the NRC typically reviews only the relief requests and performs a spot check of other areas of the program plan. Because the number of relief requests has significantly decreased in the 10-year IST program plans since the adoption of the 1995 OM Code, licensees' IST program plan reviews are reviewed by the NRC staff without the need for specialized contractors.

NRC approval is required to take exception to those Code rules specified in 10CFR50.50a and generally requires specific submittals. However, submittals are not always required. For those plants that are still on OM-6 and OM-10 (Section XI Code, IWP and IWV plants), there are a few NRC approved positions for acceptable Code implementation available as described in Supplement 4 to Generic Letter 89-04 (NUREG-1482). NUREG-1482 contains very specific language that must be followed to document such implementation in IST programs, but acceptability of these methods is specifically addressed. Any plant updating to current 10CFR50.55a rules would be adopting the OM Code, not the Section XI Code (OM-6 and OM-10), and should be aware that some of the guidance in NUREG-1482 for testing may be obsolete or inappropriate for use with the OM Code.

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The NRC staff is currently updating NUREG-1482 to ensure that the inservice testing guidance is consistent with the latest OM Code incorporated by reference in 10CFR50.55a.

For relief request format, it is this writer's opinion that the recommended format in NUREG-1482 described by section 2.5 is acceptable in the absence of new NRC guidance.

Compliance with Ambient and Media Temperature Correlation Rules

by Wavel Justice

The use of documented correlation factors is an alternative to testing valves by simulating ambient temperatures, and using a test medium (fluid and temperature) for which they are designed. Nuclear utility plant owners, through the ASME OM Code and Pressure Relief Device Users Group (PRDUG), are currently addressing compliance with these rules. Several implementation issues are discussed in the "Summary of Public Workshops Held In NRC Regions On Inspection Procedure 73756, Inservice Testing Of Pumps And Valves"[4], Section 2.4. These issues reflect the uncertainty and lack of clarity as to what these OM rules require.

It is believed that failure to have the correlation documentation does not represent any operability or safety concerns. Rules used in previous IST Ten-Year Intervals did not contain the new documentation rules, but yet they were still considered by the NRC, and the nuclear industry, to be adequate for the safe operation of our plants. When the NRC reviewed OM Part 1 for endorsement in 10 CFR 50.55a, they had to consider expediting rule adoption of any new rules that meets certain safety significance criteria. The lack of NRC expedited rulemaking is a fair indicator that the new OM Part 1 documentation rules are not an operability or safety concern. The nuclear industry has discussed the failure to have the correlation documentation required by the new rules for several years without any safety issues being raised.

GL 89-04[6] Applicability and NRC New Viewpoint

by Wavel Justice

The current GL 89-04[6] endorses NUREG-1482[2] which includes 1995 updated responses (Current Considerations) to the original GL 89-04[6] positions, questions, and responses. Another update or overhaul of NUREG-1482[2] is being worked on by the NRC (no specific due date). The NRC/ASME Symposiums and ASME OM Code Committee meetings provide venues for industry folk to get a preview of the future NRC Future Current Considerations. Hopefully, the ISTOG will become a venue for NRC/IST Engineers interfaces that will help form future NRC considerations.

Position Papers (Endorsed by ISTOG & Guidance for Their Use by Utilities)

by Gregg Joss

The ISTOG will develop Technical Positions (TP) on various IST issues deemed important to the ISTOG membership. ISTOG will implement a process for researching and developing TP's with the Steering Committee (SC) having final responsibility for their approval. Once approved, the SC will distribute the TP to all ISTOG members for consideration of adoption at their facility using the 10 CFR 50.59 review process for all associated changes. On an "as deemed appropriate" basis, the SC will create a Topical Report to be sent to the NRC detailing the TP conclusions and associated bases.

NUREG 1482 Revision (Development and Reviewer Role)

by Gregg Joss

ISTOG is very interested in being given the opportunity to become a part of the NUREG 1482[2] document revision and review team. By incorporating an ISTOG review team in the process, valuable program owner and field testing experience will provide a "users" contribution that currently does not exist. In addition, many of the inevitable post-issuance questions and clarification requests could be avoided or resolved while still in the draft revision development or pre-issuance phase of the review process. ISTOG is pursuing NRC permission to provide this type of formal role.

Code Class 1, 2, 3 Versus "AUGMENTED" IST Components

by Gregg Joss

When choosing to include non-Code class components as "augmented" (refer to General Questions 1.1.1, 1.1.2, and 1.1.3 of the "Summary of Public Workshops Held In NRC Regions On Inspection Procedure 73756, Inservice Testing Of Pumps And Valves"[4] and NUREG 1482[2], section 2.2 and Question Group 53 in Appendix A) in the IST program, many different approaches are employed. Approaches range from treating an augmented component identical to a full-fledged Code class 1, 2 or 3 component including all applicable tests and test periodicity, to loosely following the Code requirements with no compensatory requirements

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when Code test provisions cannot be met. ISTOG intends to develop a guide which will assist IST program owners with documenting the inclusion of augmented components and establish a standard approach for establishing the testing requirements of such components utilizing existing regulatory guidance and industry "best practices".

Terminology Guide

by Leonard Firebaugh

Due to turnover of personnel it may be desirable to have a document that compiles in-service testing terminology used in the nuclear industry. This would be a compilation of terms and common acronyms used by various industry groups as well as major documents including the NRC, ASME, ISTOG, Code of Federal Regulations, NUREGs, etc. A brief explanation for each term as well as the source and an example of usage could be given. Industry standardization would not be a goal for this guide.

Relief Request Templates

by Leonard Firebaugh

A recommended format and content for relief requests as well as several examples are contained in NUREG-1482[2]. However it may of benefit to have a set of industry relief request templates written against specific requirements and/or specific equipment types that a utility could pull off the shelf and use with only minor changes. This would especially be useful as an owner is required to implement newer additions of the Code if the templates have been generated by the first wave of owners.

Code Implementation Relief Request Guideline

by Leonard Firebaugh

Industry experience with writing a successful relief request continues to be mixed. Original guidance is contained in NUREG-1482[2] as well as more recent format guidance in an NEI document to which the NRC has agreed. However, recent experience with the NEI format received feedback from the NRC that it did not contain enough information. This ISTOG guideline would set the industry standard for level of detail and format necessary for a relief request that has NRC and industry concurrence.

NRC Q & A – Guidelines for Unwritten Processes

by Jeffrey Neyhard

The ISTOG will provide a guideline that establishes a uniform approach to be used when the IST Program Manager desires to gain insights from the regulator. The ISTOG will work with the NRC to ensure the guidance is consistent with NRC established policies. The intent is to capture acceptable communication processes that are currently undocumented. The scope will be refined as the various undocumented processes are identified.

Guidelines for Limiting Values

by Jeffrey Neyhard

The ISTOG will provide a guideline for the consistent selection of valve stroke time limiting values when no component specific Design Limiting Value is identified. When the Licensing Basis or the Design Basis provide a limiting value, the most conservative of the documented numbers is used as the limiting value for IST. When no documented limiting value is available, engineering judgment is used to obtain a limiting value. Obtaining a limiting value by engineering judgment can be simplified to a formula. In preparing the guideline the ISTOG will compile information from utility sources to ensure the various methodologies are considered. The purpose of the guideline is to ensure uniformity and consistency in the application of engineering judgment to a diverse population of valves.

PMT Guidance

by Jeffrey Neyhard

The ISTOG will provide a guideline for the consistent selection of Pre-Maintenance ("As Found") and Post-Maintenance Tests (PMT) as part of scheduled or corrective maintenance. The guideline is intended to also include Appendix J Owners Group information that provides for consistent decision making when using valve diagnostic data in lieu of performing as-left leak rate tests. The guideline will consist of tables that identify typical IST components, their multiple maintenance activity types and the pre-maintenance and post-maintenance tests to consider ensuring program compliance.

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Sample Valve Passivity (How to Justify Passive Classifications)

by Jeffrey Neyhard

The ISTOG will provide a guideline for justifying passive valve classifications. Industry feedback identified that the approach to passive valve classification is inconsistent between utilities. The questions and answers from both NUREG-1482[2] and the "Summary of Public Workshops Held In NRC Regions On Inspection Procedure 73756, Inservice Testing Of Pumps And Valves"[4] indicate the need for additional clarification and guidance in this area.

Risk Informed IST (RI-IST) Implementation/Transition Guide

by David Chiang

One of the benefits of RI-IST is that the testing frequency of pumps and valves can be extended depending on the component's risk ranking. Typically, High Safety Significant Components (HSSCs) retain their Code specified test frequencies whereas Low Safety Significant Components (LSSCs) benefit from the testing interval extensions as defined in the RI-IST program description. LSSCs are grouped by component attributes and the selected attributes should satisfy NRC criteria provided in NUREG-1482[2]. With the current industry trend of short outages and system train related, it is critical to align the RI-IST components such that they are tested with the system train outage. If the component train and the system train is not aligned, it is then necessary to baseline component train-system train such that they are in synchronization for future outages and testing.

Flow Loop Issues

by David Chiang

In NUREG 1482[2], NRC staff position 9, the NRC has stated its position on using minimum-flow return lines with or without flow measuring flow devices. The NRC has delineated the conditions when flow measuring devices are required. In the 1998 Edition, 1999 and 2000 Addenda of the OM Code[5], ISTB-5121(c), 5221(c) & 5321(c), the Code stipulates that in systems where resistance cannot be varied, flow rate and pressure shall be determined and compared to the reference value. In systems that have non-instrumented minimum flow lines, the licensee will have to seek relief from this Code requirement. It has been found that the NRC is not consistent in granting relief. Some plants have been granted relief and others have been denied. There should be consistency throughout the industry on this issue.

Design Flow Rate Guide

by David Chiang

There has been much discussion within the industry as to the intent of Design Flow Rate. Messrs. Bedi and Colaccino of the NRC in their paper presented at the 7th NRC/ASME Symposium on Valve and Pump Testing (NUREG/CP-0152, Vol. 4)[3] alluded that some licensees have interpreted the Design Flow Rate as the best efficiency point (BEP) of the pump. Some plants, due to their system configuration, cannot test their pumps at the BEP. Other licensees take the position that the Design Flow Rate is the OEM flow rate when the pump was purchased. Furthermore, the Design Flow Rate can be interpreted as the accident flow rate, that is what the pump was designed for. Over the years, the accident flow rate for some systems has changed due to regulatory changes. Therefore it can be seen that there is certainly inconsistency in the interpretation of Design Flow Rate and it needs to be consolidated.

Check Valve (CV) Condition Monitoring How-To

by Bob Parry

Check Valve Condition Monitoring provides the Owner with process flexibility to implement changes in their IST Check Valve Program. This issue will deal with elements necessary to start a Condition Monitoring Program to add value to the station not only in improved performance, but also in optimizing all of the various activities that check valves are subjected to. Some practical applications such as coordinating the Appendix J leak rate Option B Performance testing with the IST closure demonstrations, reducing the number of valves subjected to disassembly, performance monitoring techniques, outage philosophies, etc., will be discussed. These gains offset the costs associated with implementing the program, and offer improved component reliability.

Skid-Mounted How-To (Justifications)

by Bob Parry

Under certain Code Editions/Addenda, integral or skid mounted equipment can be exempted from the Code provided it is tested under another program to verify that the component can perform the intended function. What are the particulars in determining if a component is skid mounted? What does integral mean? Once selected, how are they tested? How are they documented?

This feature is largely associated with the Diesel Generator sub-systems, although other skid systems, techniques and provisions of this program will be discussed and, until the

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ASME OM Sub Group on Diesel Generators completes their effort to establish levels of performance monitoring with trending of specific parameters, this guidance will provide the implementation requirements.

Condition Monitoring vs. Performance Based Difference

by Bob Parry

What are the differences between performance monitoring and condition monitoring? Are some time based? Are some conditional in the selection of activities? Are reviews or analysis of results required for some? Are there requirements for trending with one and not the other? Is there a feedback requirement with one program that suggests alternate activities should be specified at the next available opportunity, or are we just looking for "SATs"? What other programs are called performance based? What programs are called condition monitoring?

Instrumentation Accuracy Considerations for Pump Testing

by Bob Parry

What are the essential elements of an IST Instrumentation program? Is the expectation that the permanent plant devices used for IST remain in calibration for the entire calibration interval? If so, what measures need to be taken? What tracking should be done to ensure requirements are being met? What records need to be maintained? What needs to be done on pump overhauls? What criteria and what bases should be provided to the various support organizations? What documents are essential for such a program?

Conclusion

The IST Owner's Group is an organization dedicated to improving the quality of life for the IST Engineer. The collaboration of industry expertise through ISTOG will improve implementation guidance and industry responsiveness in the field of Inservice Testing.

References

1 - NRC Information Notice 97-16, "Preconditioning of Plant Structures, Systems, and Components Before ASME Code In-Service Testing or Technical Specification Surveillance Testing"

2 - NUREG-1482, "Guidelines for In-service Testing at Nuclear Power Plants" (April 1995)

3 - NUREG/CP-0152, Vol. 4, "Seventh NRC/ASME Symposium on Valve and Pump Testing"

4 – NRC Memorandum - 1997 Summary of Public Work Shops

5 – ASME Operation and Maintenance of Nuclear Power Plants 1998 Edition, 1999 and 2000 Addenda

6 – NRC Generic Letter 89-04, "Guidance On Developing Acceptable Inservice Testing Programs"

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