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July 17-19, 2006

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Board on Nuclear Codes and Standards
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Abstract

The 2006 Symposium on Valves, Pumps and Inservice 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, programmatic and regulatory issues associated with inservice testing programs at nuclear power plants, including the design, operation and testing of valves and pumps. The symposium provides an opportunity to discuss improvements in design, operation and testing of valves and pumps that help to ensure their reliable performance. The participation of industry representatives, regulatory personnel, and consultants ensures the presentatation of 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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Disclaimer and Editorial Comment

Statements and opinions advanced in the papers presented at the Ninth NRC/ASME Symposium on Valves, Pumps, and Inservice 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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A Method for Characterization of Resonant Response and The Prevention of Critical Pump Failure

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Abstract

The U.S. industrial sector depends highly on the reliability of critical function machinery. Of significant concern to industry is the dependability and maintainability of its turbo machinery. This is particularly important to the nuclear utilities industry where the unexpected loss of a critical pump can lead to prolonged unscheduled downtime and severe cost implications. Although no clear method exists that can absolutely predict pump failures, vibration monitoring, in conjunction with the use of decomposition results, can be used to predict and potentially prevent complete pump failure. This paper examines a method to characterize system resonant behavior associated with pumps and pump systems by the use of spectral decomposition. In addition, discussions of operational procedures are reviewed that can help reduce the risk of complete pump failure once the resonant regions are identified through the use of the decomposition routine.

History

For more than 50 years, the Electro-Mechanical Corporation (EMD) of Curtiss-Wright, formerly Westinghouse EMD, has built critical function pumps for the U.S. Nuclear Navy and the commercial nuclear industry. As part of its legacy with over 100 years of Westinghouse innovation, EMD designed and manufactured the main coolant pump for the first nuclear-powered submarine, the U.S.S. Nautilus and the first U.S. nuclear power station in Shippingport, Pennsylvania. Over the years, given the stringent requirements of the nuclear sector, EMD has developed state-of-the-art sound and vibration data analysis techniques that aid in the performance evaluation of turbo-machinery that EMD produces.

Today, now a part of the Curtiss-Wright Corporation, EMD continues to develop, design and supply advanced electro-mechanical products for the U.S. Navy and the nuclear utility industry by supplying reactor coolant pumps, seals, motors, and control rod drive mechanisms. Recently, EMD has expanded its product offering to include hazardous nuclear waste mixing and transfer pumping systems. The focus of this paper will highlight the decomposition process as conducted on a Submersible Mixer Pump that was developed by EMD.

Background

For most industrial applications, sound and vibration concerns are typically an afterthought as long as the machine is operating at its intended condition and its performance has not been compromised. Accordingly, the vibration performance by a pump in operation in industry is typically not an issue that generates much concern, unless the vibration levels are such that it is a strong indication of impending failure. The core business of EMD is focused on pumps and other turbo-machinery, typically for non-commercial applications. EMD specializes in canned motor design with a large emphasis placed on sound and vibration performance. Given EMD's product line, it is in the best interest for the company to possess state-of-the-art design and analysis tools.

One such state-of-the-art tool that is utilized by EMD is the spectral decomposition process. The concept of spectral decomposition was originally conceived by Weidemann [1] over 30 years ago and was further explored by Pennsylvania State University / Applied Research Laboratory (PSU/ARL) [2] and [3]. The process was then optimized and further refined by EMD for use as an analytical tool for sound and vibration analysis. The spectral decomposition process is a Matlab based routine used to differentiate between vibration sources (rotating machinery) and structural/system response (resonances).

The original intent of the decomposition was its usefulness in generating source spectra that was free of system response. A by-product of the process generates the system response curve that accurately characterizes a dynamic system response. The results of the structural characteristics obtained by the spectral decomposition method yields a more effective structural response than that obtained by traditional static impact test methods. This is because the decomposition method utilizes data in its dynamic form as generated by a vibration source such as a pump. Using the actual pump excitations in the decomposition form is more accurate than data collected during an impact test using a calibrated hammer. This is simply due to the fact that the hammer impacts might not excite all of the frequency regions of the structure that would be excited during the pump operation.

Introduction

The use of spectral decomposition is not limited to pumps. Rather it can be applied to any type of turbo-machinery where data is collected at multiple speeds. The principles of the method are based on Weidemann's acoustic similarity laws [1] and were later utilized and refined by Mongeau as a method to investigate rotating stall in centrifugal turbomachinery [2]. Jonson and Young also used the method for separating hydrodynamic excitation from structural responses for unsteady thrust measurements on a rotor driven by a dynamometer [3].

The output of the decomposition routine produces two distinct components of the original frequency spectra; vibration source and system response. The vibration source spectra is comprised of Strouhal related components while the system response is related to the Helmholtz components. Strouhal effects are useful for the identification of sound generation mechanisms because they are related to the hydrodynamic portion of the spectra rather than the vibration characteristics of the machine environment.

The aforementioned methods focus on the decomposition method as a process to develop the hydrodynamic component of the measured spectrum in the frequency domain. Mongeau focused only on sources generated by fluidborne sound mechanisms. Other sound mechanisms such as structural vibration, cavitation and vibration generated by mechanical components were excluded from the original scope of the investigation.

Specifically, Mongeau's method for characterizing aerodynamic sound sources was focused on microphone measurements of a centrifugal water pump impeller with various discharge configurations. Upon the completion of the Mongeau work, it was assumed, but not proven, that the method of decomposition could be used on hydro-acoustic sources such as hydrophone measurements and on other types of turbomachinery besides pumps. In addition, it was also unclear if the method could be used for the study of hydrodynamic events as measured by accelerometer instruments. Jonson and Young applied the decomposition technique on measurements of unsteady rotor thrust made in a flow channel using a force crystal.

Mongeau's original work was successful in characterizing the phenomenon of rotating stall associated with a centrifugal pump impeller having no casing or diffuser. The use of spectral decomposition had proved to be a useful tool for investigation of aerodynamic sources and led to a separate paper written by Mongeau that focused strictly on the method of decomposition itself [4].

In 2001, EMD used the principles of the Mongeau investigations to develop a spectral decomposition method utilizing Matlab and data collected on a main coolant pump. As with the other earlier works in decomposition, the EMD spectral decomposition routine was developed primarily to isolate and to aid in the identification of hydro-acoustic sources. The original results of the EMD decomposition revealed hydro-acoustic sources that were not apparent by simple review of the autospectra. The method worked on both hydrophone measurements and accelerometer measurements. The source spectra produced through the use of the decomposition procedure generated spectra that were void of the acoustic frequency response of the system.

Continuing with the approach used by Mongeau with microphone data, EMD calculated the decomposition of other instrumentation including hydrophones, load cells, strain gages, and accelerometers. However, the focus of this paper will be of data collected with accelerometers.

The benefit of conducting decomposition on turbomachinery data is that, as with any vibration measurement being performed, system response is coupled with the forced excitation that is often the true focus of the measurement. In some cases, the system response is often higher in amplitude than the source, which is being measured. This causes the source data to be obscured by the system response, making it very difficult to evaluate the source spectra by itself. Often during the hydrodynamic design process, engineers at EMD are interested only in the hydraulic vibration source spectra without the effects of the system response. Most tests conducted by EMD are conducted in a factory setting that has a fixed mounting method and its own unique system response. When a unit is installed in its field application, its mounting configuration and system response may be different from the factory test. Hence, it is important to be able to evaluate the pure source of the unit devoid of any system response that would not be representative of the actual machine installation. Until the method of spectral decomposition was utilized, separation of the two distinct spectra had not been completed at EMD.

This paper will demonstrate the benefits of using spectral decomposition as a method of not only differentiating source from system response, but will focus on the usefulness in the ability to identify resonant regions of a pumping system without the need of conducting a traditional static impact test.

Resonance Avoidance

The word resonance was derived from the Latin word "resound", meaning to become filled with sound or sound loudly. The acoustic textbook definition of resonance as defined by Kinsler and Frey [5] is as follows: "the resonance frequencies of any mechanical system are defined in general as those frequencies for which the input mechanical reactance goes to zero." One interpretation of this textbook definition of resonance can be simplified as the buildup of large vibration amplitudes that occurs when a structure or an object is excited at its natural frequency. In the engineering field, it is typically desirable to avoid resonant behavior because the resultant response in most cases is an unwanted effect. Resonant response can be destructive and cause subsequent damage if the given amplitude is large or even just slightly elevated but of long duration. The destructive nature of resonant behavior is well known and can be cited in books, journals, and symposiums of engineering related organizations throughout the world.

Often, a machine or structure is monitored throughout the duration of field operation for changes in performance. The structural response is also calculated over time to determine if changes have occurred which could be an indication of potential issues that would lead to critical failure. This constant monitoring is not always possible as the costs of conducting continuing tests to determine changes in the structural characteristics are prohibitive.

Determining Resonant Frequency

Multiple methods exist to determine the resonant frequencies of a structure. The two most common methods are by calculation though the use of computer software and the other being empirical means through testing. The calculation method is typically done with finite element analysis (FEA) conducted during the design phase of an assembly. During the design stage, complete assemblies and subassemblies are analyzed to optimize environment boundary conditions. This type of analysis is quite involved and requires the use of computer-based programs. Even though computers have advanced the predictive capabilities, the process is still lengthy and requires proper assumptions for boundary conditions prior to getting valid results. The result of FEA analysis is often validated through experimental testing in the form of a static impact test with a calibrated hammer or a dynamic test with the use of a shaker. Static impact tests are typically more common and are conducted with a calibrated hammer and accelerometers placed on the machine/structure. The impact of the hammer excites the structure while the accelerometers measure the response of the structure due to the excitation. Impact testing is an adequate method for most engineering applications.

However, one of the weaknesses of impact testing is that it requires the engineer to have access to the equipment being tested, which is not always possible. For example, the area in which a test needs to be conducted may be hazardous and not allow for human entry which may include a pump located in a nuclear waste tank or a coolant pump located in a reactor containment building. In addition, a typical impact test may not identify resonant regions that are not excited by the hammer, yet are excited by the forced excitation of the machine. Even if the conditions were ideal and a static test could be conducted, the optimal resultant response may not be obtained due to differences in boundary conditions between the operating and non-operating conditions.

Development of an Alternate Approach

Although most typical industrial pump or turbo-machinery applications do not have strict criteria for fluidborne or structureborne sound and vibration, it is a major concern with many of the products developed and produced by EMD. Due to these concerns, EMD has developed state-of-the-art techniques that are used to evaluate centrifugal pumps and other turbo-machinery design and manufactured by EMD. Many sound and vibration related issues have hampered the review of autospectra data in one form or another. One particular issue has been sound propagation phenomenon such as system resonant responses contaminating the autospectra data being collected.

In most cases where fluidborne sound and structureborne vibration is of concern, it is in the best interest of the investigator to remove the effects of the vibration propagation from the measured data and reveal the spectrum of the vibration source alone. The method developed by EMD and discussed herein is called spectral decomposition. Unlike typical uses of the spectral decomposition process where the intent is to eliminate the system response from the spectra, this paper discusses the use of spectral decomposition to extract the underlying turbomachinery source in order to evaluate the system response in its absolute form. The extracted system response will then be compared to data acquired during an impact test to validate the decomposition process. The system response is then reviewed in frequency content with sensitive areas identified. Avoiding operational conditions that may excite resonant modes will ultimately increase the longevity and life of a machine. Ignoring such resonant modes can lead to premature malfunctions including complete failure.

Theory

Acoustic similarity laws are used to isolate the vibration propagation characteristics of the system from measured vibration spectra. In order to use the similarity laws on a centrifugal pump or other type of turbo-machinery, the vibration spectra must be measured over a range of rotational speeds but with a constant flow coefficient. The trends in the spectral shape common to all spectra can then be identified because they are stationary with changing speed and thus can be used to generate the structural response of the machine. The isolated structural response spectra forms the function G(He,Ø) and is related to the Helmholtz frequency. The Helmholtz function can then be used to extract the pure source spectra by eliminating the G(He,Ø) from the original spectra. The resulting function formed from this whitening is the function F(St,Ø) and is related to the Strouhal frequency and is varying in frequency with operating speed.

Following Weidemann [1] and Mongeau [2], the acoustic parameter is assumed to be given by the linear product of two non-dimensional functions, as follows:

$$\frac{Sxx(f)}{[\rho_o V^2_{tip}]^2 (D/V_{tip})} = F(f/Bn)G(f/f_o) \quad (1)$$

where, Sxx(f) is the Vibro-acoustic auto-spectrum (x may be pressure, acceleration, force, etc);

$[\rho_o V^2_{tip}]^2 (D/V_{tip})$ normalizes the pressure or acceleration response autospectra by a combination of variables directly proportional to the hydrodynamic force spectrum where ρ_o is the fluid density, Vtip is the tip speed, and D is the pump impeller diameter;

F(f/Bn) is the hydrodynamic component of the vibro-acoustic autospectrum, which varies with the dimensionless Strouhal number f/Bn, where f is frequency and Bn is the number of vanes. The Strouhal number is defined as:

$$\text{Strouhal Number} = \frac{fD}{(Vtip)} x \frac{\pi}{n} = \frac{f}{BPF}, \quad (2)$$

where f = frequency
D = impeller diameter
N = number of vanes

Vtip = tip velocity = π x rpm/60 x D

G(f/fo) is the structural-acoustic component of the vibro-acoustic autospectrum, which varies with dimensionless frequency f/fo, where fo is the reference frequency which may be based on a structural or acoustic phase speed and a characteristic dimension. This function is related to the Helmholtz number, He, and is defined as the ratio of the impeller diameter and the acoustic wavelength. It characterizes acoustic phenomena such as resonance, directivity or sound wave reflection. The Helmholtz number is defined as:

$$He = \frac{fD}{a}, \quad (3)$$

Decomposing the vibro-acoustic system response into hydrodynamic and structural-acoustic components requires several spectra over a broad range of operating speeds at a constant advance ratio so that the relationship between the noise levels and flow parameters can be extracted. An operating speed ratio of 2:1 with at least six speeds is recommended to generate accurate results. In addition, good signal to noise ratios are required over all frequencies and speeds. In the example vibration problem of a Submersible Mixer Pump, the measured vibration spectra, Sxx(f, V_{tip}, Ø, x) depend on the rotational speed of the rotor, the pump operating condition, and the accelerometer location.

A function of the Strouhal number, F should vary with the pump operating conditions and should remain independent of the measurement location, since any sound propagation effect between the source and the receiver is included in function G. Therefore, the dependence on Ø and x/D in Equation (1) must be removed by using the equation in its rewritten form:

$$\frac{Sxx(f, V_{tip}, \phi, x)}{[\rho_o V^2_{tip}]^2 (D / V_{tip})} = G^2\left(He, \phi, \frac{x}{D}\right) F^2(St, \phi), \quad (4)$$

By removing the dependence on Ø and x/D in Equation (4), Equation (1) can again be rewritten as:

$$20 \log G\,(He) =$$

$$Lpp\,(f, V_{tip}) - 30 \log V_{tip} - 10 \log \frac{\tilde{n}_o^2 D \Delta f_{ref}}{p_{ref}^2} - 20 \log F(St) \quad (5)$$

and then simplified to:

$$= Lpp\,(f, V_{tip}) - 30 \log V_{tip} - 20 \log F(St) - 90.7 \quad (6)$$

where Lpp is the amplitde level spectral density and defined as

$$L_{pp} = 10 \log [S_{xx} \Delta f_{ref} / p^2_{ref}], \quad (7)$$

By utilizing equation (6), the levels are made non-dimensional and can be plotted against the Helmholtz number. In simple terms, the output of equation (6) can be used to generate the Helmholtz related terms which are also the structural terms that are used to calculate the resonant regions. For a full interpretation of the development of the method, it is recommended that the Mongeau [2] and [4] work be reviewed. It is not the intent of this paper to derive the equations generated by Mongeau, rather, it is to review the results of the Mongeau work to establish a method to develop autospectra that are accurate representations of the structural system resonance.

Application of the Decomposition Process

Although the spectral decomposition process was originally developed for use by EMD to isolate the vibration source of pumps, the tool can be used on most types of turbo-machinery. The example that follows shows the tools versatility as the process was used on the Submersible Mixer Pump (SMP) to isolate and identify the SMP system resonant response regions. The SMP is designed to prepare the contents of nuclear waste tanks for removal by a separate transfer pump for further processing. The SMP unit is a unique pump and is unlike most pumps designed and manufactured by EMD. The pump is a vertical, single stage, centrifugal, canned motor design comprised of three major sub-assemblies: a motor unit assembly, hydraulics and a column assembly.

The SMP hydraulics consists of a suction inlet screen, impeller, diffuser and a casing. The suction inlet screen prevents large particles from being ingested and is bolted to the bottom of the hydraulic casing. After the process fluid is drawn through the suction screen, it travels upward through the suction and is fed into the eye of the impeller as with any typical centrifugal pump design. The flow exiting the discharge provides mixing of the waste material. A small portion of the process fluid is directed past the motor to serve as a coolant and exits from the column above the motor.

The units were required to be qualified through testing that included multiple days of operation at various pump speeds and various operational conditions. The first portion of the qualification testing involved testing the unit in water, followed by testing in a water, Kaolin and sand mixture. The purpose of adding an earth based clay material such as Kaolin into the test tank was to hold the sand in suspension such that the pump could circulate the sand and not allow the sand to simply fall to the bottom of the tank. The sand was inserted to simulate the mixture in which the pumps would likely operate once installed into a production tank. This Kaolin/sand mixture was used as a quasi simulate for actual radioactive nuclear waste sludge. Testing commenced on the first two units without any type of vibration issues. During testing on the third unit however, high levels of vibration were observed during the test when sand and Kaolin was introduced into the test tank. The excess vibration was eventually proven to be caused by a two-fold effect; the existence of a resonant region and an elevated source level that coincided with the resonant region.

Test Setup

Accelerometers were located at multiple unique positions during the testing of the units as highlighted in a layout schematic in Figure (1). Accelerometers were located at the top of the column and at increments of ¼, ½, ¾ of the column length, and at the casing flange. The naming convention used designated any type of accelerometer to be serialized with an “A”, and the direction of accelerometer if in a radial position would be identified with an “R”. The accelerometers that were parallel with the supporting keyway were designated with “X” and the accelerometers located 90 degrees apart or perpendicular with the keyway were identified with a “Y”. A photo of the pump prior to installation is shown Figure (2) indicating the final accelerometer positions. A Bruel and Kjaer Pulse Data Acquisition System was used for the collection of the accelerometer data. The pump levels were presented in terms of velocity in units of inches per second (in/s). The vibration criteria used was in the form an overall RMS value and calculated as the square root of the sum of amplitudes over a given frequency range. The single overall value provides a quick evaluation of the data, which can potentially indicate poor pump performance quickly. The criteria was derived from an API standard, Table 8 [6] and is given as an overall velocity measurement of 0.20 in/s RMS. Using the overall value as a sole indicator of pump performance can be deceiving as it is one value that is a representation of the entire frequency spectrum. A false value or error can be obtained if the proper cutoff frequency ranges are not incorporated or if data is acquired prior to the initial DC bleed off being fully discharged.

Static Impact and Decomposition Results

Prior to initial startup and operation, a static impact test was conducted on the first SMP unit. The objective of the static impact test was to identify resonant regions of the pump and column. Once identified, the pump was to be operated in a manner that any significant operational forcing function would not correspond with any resonant mode of the structure. Coincidence of a significant forcing function with a significant resonant mode would have adverse effects on the performance of the unit and longevity of the machine. The health and lifespan could be dramatically affected by excess vibration levels due to resonant amplification. A sample of the results of the impact testing conducted on the initial SMP unit are show in Figure (7). A calibrated impact hammer was used to excite the structure in 3 directions, X – in parallel with key way, Y – perpendicular to the keyway, and Z – vertical and parallel with the column assembly. The results as shown in this figure show that several resonant regions exist within the normal range of the fundamental rotational frequency of the pump. The significant frequency regions are identified at 6.4 Hz, 15.8 Hz, 18.1 Hz and 27 Hz. Once the resonant regions were identified, the next step was to operate the unit to determine the pump’s vibration signature and identify the significant forcing functions.

Vibration data can be displayed in many formats. For most typical industrial applications, vibration data is normally collected with either velocity probes or accelerometers. The standard format typically used by the commercial industry is to show the data in terms of velocity with units of in/s. The typical velocity spectrum used to analyze machinery is normally displayed in a linear or non-dB format. To further simplify the vibration monitoring, some industrial applications gauge the performance of the unit to a single overall vibration level (square root of the sum of amplitudes over a given frequency range). Unfortunately, the onset of many vibration issues occurs prior to any increase in levels that can be detected in a linear velocity format. This is one of the prime reasons why EMD prefers to review data in terms of acceleration and in a dB format. Often, specifically in the case of bearings, the signature of the bearing is hidden in the midst of other machinery tonal frequencies and harmonics. EMD reviews the data in terms of acceleration to also avoid the ski-slope effect that can sometimes corrupt velocity data. EMD specializes in unique application pumps and other turbo-machinery which gives rise for the need of tonal identification of all tones, not just tones of high amplitude. The data collected on the SMP is shown in acceleration and in dB format in Figure (8). This is a typical spectra that is expected from a normal operating pump. Cursor points are used to identify tones that are tracked throughout the operation qualification testing. Of great significance are the lower frequency (below 50 Hz) discrete tones. The first tone, as designated with cursor point #1 is the fundamental rotational tone, or 1R (RPM/60). Industry often identifies this fundamental rotational tone as 1X. EMD however does not use this convention because more than one discrete tonal frequency is tracked and the 1X designation is overly generic. Some of the many other tones identified and tracked in Figure (8) are electrical tones (2E, 6E, 12E, etc.), hydraulic vane and blade passing tones (RPM/60 * number of blades), and motor structural tones associated with the stator and rotor slot combinations.

Figure (8) shows a single autospectra at one speed for one location. Figures (9) and (10) compare the vibration levels at one accelerometer location for a range of speeds (397 RPM to 1432 RPM). Figure (9) is for the frequency range of 0-400 Hz while Figure (10) is the same data set; however, the frequency range of interest (0-100 Hz) is highlighted. Review of both figures shows the unique complexities of the autospectra. Without the use of signal processing, the task of determining either the isolated source spectra or structural system response would be possible, however extremely difficult.

The spectral decomposition process makes it possible to complete the challenging task of discriminating the system response and isolated source from the autospectra. An illustration of the process is provided in Figure (11). The plot on the left hand side is the raw autospectra at 100% speed at location A9RX. The upper plot on the right hand side is system response output of the decomposition process with the lower plot being the isolated source of the decomposition process. The process provides a clean representation of both the structural system response and the isolated source. The upper right plot is shown in more detail in Figure (12). The resonant regions are identified by the higher amplitude regions, with the major frequencies identified and marked accordingly at 6.4 Hz, 15.8 Hz, and 27 Hz. Comparing these results to the static impact results shown in Figure (7), a strong trend is revealed. The major resonant regions of 6.4 Hz, 15.8 Hz and 27 Hz are accurately and easily identifiable in the decomposition results. The other regions that the decomposition shows as elevated amplitudes are the 7.8 Hz, 13.4 Hz and 18.1 Hz regions, although these are not as dominant as the three aforementioned regions. Differences between the shapes in the two spectra are a result of the differences in excitation. The static impact test is a result of a hammer exciting the column, while the decomposition is a result of pump operational forces exciting the assembly. In order to accurately compare the results, an upper envelope of the static impact tests should be used to compare on a one-to-one basis with the decomposition results. The amplitudes of the decomposition are in dB however. The absolute magnitudes of the results are not quantifiable and in order to be used in place of the static impact values, the results must be adjusted or “grounded” to a known value. The amplitudes are not in an absolute value form because the results are a transfer function generated from the autospectra. What should be taken from this however is the fact that the decomposition routine accurately depicts the frequency content for the resonant regions of the SMP assembly.

To further validate the decomposition output, the decomposition was conducted using data collected at all accelerometer locations. The results were then grouped accordingly, with like oriented accelerometers compared to one another. An astonishingly tight trend was generated from these comparisons with the results displayed in Figures (13) and (14). The decomposition results of the X-direction accelerometers are compared in Figure (13) with the Y-direction shown in Figure (14). Like oriented accelerometers show a strong correlation or system response. Small differences are observed, with most of these differences associated with accelerometers located at the top of the column. The locations of accelerometers A1RX and A2RY were above the mounting position of the column and hence less constrained compared to the rest of the assembly. The differences in location and restraining resulted in the top of the column being able to move differently than the rest of the column, which is why the results from A1RX and A2RY differ slightly from the other accelerometers.

Operational Performance

The overall values were monitored throughout the duration of the test. The overall values in water are extremely low in amplitude for the given range of speeds.

In addition to on-line monitoring of the overall value, the fundamental rotational frequency, 1R (RPM/60) was monitored for potential increases in amplitude throughout the duration of the test. All the units tested were operated over a range of speeds with the maximum speed of 1432 RPM. At the maximum speed, the 1R frequency is 23.8 Hz, which does not correspond to any of the major resonant regions of the column as determined by both the static impact test and decomposition routine on Units 1 or 2.

The units were first tested in clean water then a mixture of Kaolin and sand was added to the tank to simulate actual design conditions. Narrowband data was monitored throughout the test duration. An increase in vibration levels was observed when the Kaolin and sand mixture was added. Figure (15) is a narrowband comparison of vibration levels as monitored in water compared to the water with the Kaolin and sand added. The increase in discrete tones is due to many factors including increased loading both electrically and hydraulically.

The overall values were monitored throughout the duration of testing of each unit as well. Although the overall levels did increase with the addition of the Kaolin and sand as shown in Figure (16), the values were still well below criteria and the pumps were performing as intended. In addition to tracking the overall values, the fundamental rotational tone was also closely monitored for changes in performance which could be indicative of pump degradation. Figure (17) in particular shows the results of the tracked 1R tone during introduction of the sand. Prior to the addition of the sand, the proper amount of Kaolin was already in the tank. The 0% sand represented the point at which no sand was in the tank while 100% sand represents that a total load of 24 tons had been added to the tank (it does not indicate that the mixture was 100% pure sand).

The first two units tested successfully in both the water tests and the Kaolin and sand mixture. Consequently, the review of the decomposition of both units showed small to almost no change in structural response across the entire frequency range of 0-2500 Hz. It was not until testing of the third unit that vibration issues started to develop.

Importance of Resonance Avoidance

During the testing of the third SMP unit, the 1R value increased in amplitude dramatically at all pump speeds as sand was added to the tank mixture. The increase in 1R amplitude on the third unit was higher than the levels observed on the other units and higher than the expected trend that the 1R amplitude is expected to follow with increasing speed (f^2 relationship). At the maximum pump speed, the 1R frequency was calculated as 23.8 Hz. A graphical representation of the rotational amplitude values plotted as a function of speed is shown in Figure (18). Of significance is the dramatic increase of the 1R amplitude value as the speed of the unit is increased and the 1R frequency approaches the resonant region of 24 Hz. Unit 3 was tested in clean water and the levels were well behaved. The performance of the unit was normal until the sand was entered into the tank, at which point the unit exhibited elevated 1R tones.

A decomposition calculation was completed immediately and showed that, unlike the first two units tested which both had a significant resonance near 16 Hz and 27 Hz, the third unit had a shifted resonance with its significant peaked centered at 24 Hz. The result of the decomposition for one accelerometer location of the third unit is shown in Figure (19). These results were compared to the results of the first unit. This shifted resonance frequency centered at 24 Hz unfortunately also corresponded to the rotational frequency at maximum speed which explained the higher than expected 1R values at the same speed.

The other SMP units did not appear to have as strong a response at this particular frequency nor did the other units exhibit the increase in 1R amplitudes as the third unit did. The overall levels exceeded the criteria when the unit was operated in the viscous mixture, as shown in Figure (20). A detailed review of the autospectra showed that the increase in overall levels was mainly attributed to the increased 1R value with the larger portion of the spectra showing small increases. The 1R values themselves at maximum speed were high enough to control the amplitude of the overall values. When the 1R increased in amplitude during testing of the third unit, the column and structure vibrated excessively when the pump was operated at full speed.

The increase in 1R amplitudes was perplexing because the increase was not strictly due to resonance amplification. The excess vibration of the column when the unit was operated at full speed was due to resonant amplification caused by the shifted resonant region and elevated 1R. However, the root cause of the increased 1R tone observed with sand insertion was still unidentified. Figure (21) indicates that the 1R values are normal when the unit is operated in water and increase dramatically when Kaolin and sand were added to the tank. Increases in vibration performance were noted throughout the test duration on all units when sand was added into the tank but not to the excess that was observed on the third unit.

Investigation of Increased 1R Tone

Although the existence of a shifted resonance was identified as causing resonance amplification, the increase in the 1R levels across all speeds was in question. Several potential theories had been discussed as to the cause of the increased 1R value.

It was not until another unit was tested that the root cause of the increased tones were eventually determined. During the testing of the following unit, the vibration levels were closely monitored during sand insertion. The beginning of the sand insertion started in the morning with over 24 tons of sand added to the tank by the afternoon. The vibration levels increased in amplitude during the insertion of sand, however, not to the same magnitude of increase as experienced with the third unit.

In conjunction with an increase in vibration levels during sand insertion, it was noted that the flow rate generated from the motor cooling water/bearing discharge holes was reduced. In addition, the motor current was steadily dropping throughout the day during the sand insertion period. As the current was dropping, an increase in speed was noted by an increase in the 1R frequency. This observation at first seemed to be counter-intuitive as the loading on the unit was expected to increase as the sand was added. This led to a conclusion that the SMP was being unloaded in some fashion and was doing less work.

Other trends that seemed to be common among all units were being developed. One particular trend was an observed increase of both the overall and 1R amplitudes when the unit was operated at lower speeds for long periods then ramped to full speed. An additional similarity also existed when flush water was introduced into the column, the vibration levels were reduced only to return to an excessive level a short time later. Both trends were repeatable that led to the generation of a list of several potential causes. At the top of the list included blockage of the suction screen which would cause the pump to operate at an off design point.

With potential theories in hand and still perplexed by the increased vibration levels, an experiment was conducted on the unit in the tank with the full compliment of the Kaolin and sand mixture. The experiment that was conducted included using a pressure washer to clear any potential build up of mixture at the suction inlet screen. During the pressure washing of the screen, the flow from the bearing discharge holes was restored, the motor current values returned to normal levels and the vibration levels were reduced. It was then clear that the elevated vibration amplitudes, including the increase in the 1R values, were caused by thick Kaolin settlements that had accumulated at the suction screen choking off fluid to the pump. Figure (22) shows a significant reduction in measured 1R performance when the suction screen was cleaned. Cleaning of the screen with the pressure washer also shed light on the temporary decrease in vibration levels and increased flow from the bearing holes after a flush was conducted. Prior to startup, a flush is conducted on an SMP unit to lubricate the bearing surfaces. During the flush, clean water is pumped down the column into the bearing surfaces and discharged through the bearing fluid discharge holes. The majority of the water exits the pump/column through the pump casing discharge outlet while a much smaller portion is able to penetrate and exit through the suction screen, clearing a small amount of accumulated debris. This also accounts for why the bearing water flow rate returned to a fraction of the expected flow rate every time a flush was completed.

The effects of operating a pump off its design point are widely known. When pumps operate off design, generally to the left of the best efficiency point (BEP) of the pump head-flow curve, many well known negative effects occur. These effects include low efficiency, high radial loads, audible noise and excess vibration. The excess vibration is typically caused by distorted flow through the impeller. At off design conditions, large adverse pressure gradients can be generated by large incidence values at the impeller leading edge causing flow separation. In addition, blade passage vortices and horseshoe vortices at the leading edge of the impeller blades increase in strength and lead to sizable losses and flow blockage. This, in turn, leads to flow recirculation and backflow in the impeller resulting in significant pre-rotation at the impeller inlet. These flow instabilities can be strengthened and the severity of the problem increased by the introduction of non-uniformities in the inlet flow field.

The cause for the increased 1R values was due to a hydraulic imbalance caused by the choked suction screen. This elevated 1R tone coinciding with the main resonant region of the structure led to excessive vibration. When the unit was decreased in speed by even just a small fraction, the 1R value dropped off dramatically, although the levels were still much higher with the sand and Kaolin mixture than the levels recorded in clean water alone.

Conclusions

In many cases, the characteristics of the system resonant response is desired and can be acquired by conducting a static impact test using a calibrated hammer and accelerometers located on the test unit and structure. The static impact test requires the user to have direct access to the unit being tested. During the test, the hammer is used to excite the structure and therefore, contact with the structure is needed. However, not all applications allow the test engineer to have direct access to the unit, as the environment of which the machine is located may not allow access.

The spectral decomposition routine can generate a system response curve that does not require direct access by the test engineer, therefore not subjecting the user to dangerous conditions. Only the instrumentation would be subject to the damaging environmental conditions. In addition, by using spectral decomposition to generate a response curve, a more accurate representation of the system is produced, given the fact that the forcing function used to excite the system is not a hammer, but the operation of the unit itself. The resulting response curve can be considered the dynamic response curve as opposed to the response curve as generated by static means, and is therefore more representative of the system and hence more accurate. The desire to have excitation forcing functions avoid resonant regions has been well founded in the history of machinery and structures. In order to avoid damage that can be caused by resonant response, it is necessary to first identify the potential resonant frequency regions. As shown in this paper, the spectral decomposition tool can accurately identify the resonant regions of turbomachinery structures and systems, even though it was originally developed for the evaluation of the source forcing function without the effect of the system.

This discussion also has shown that operation of a pump off its design point can be of major concern. Due to the changing environment conditions which lead to a choked off suction condition, the pump was forced to operate at an off design condition. Operation at this off design condition led to hydraulic instability, which led to an increase in the hydraulic 1R tone. Although this increase in 1R was observed on multiple units, it only proved to be detrimental when the system resonant frequency coincided with the 1R frequency at high pump speed operation.

The case study reviewed here has shown the existence of an elevated 1R tone corresponding to a significant resonance led to excessive vibration. This increase in vibration caused audible noise with vibration levels high enough to be detrimental to overall pump health. The identification of the resonant region in using the decomposition process was a critical step that facilitated the investigation of determining the cause of the excess vibrations.

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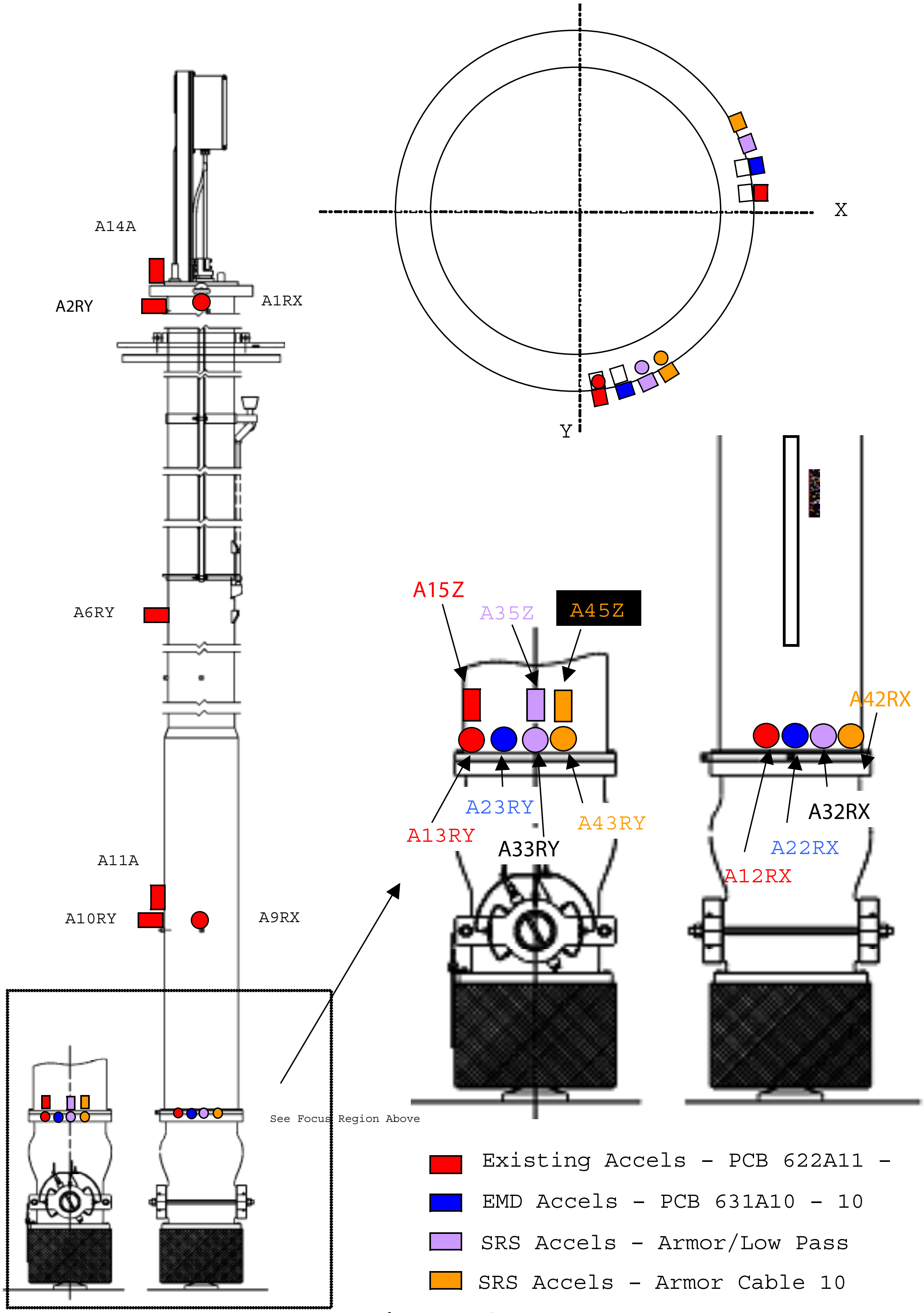
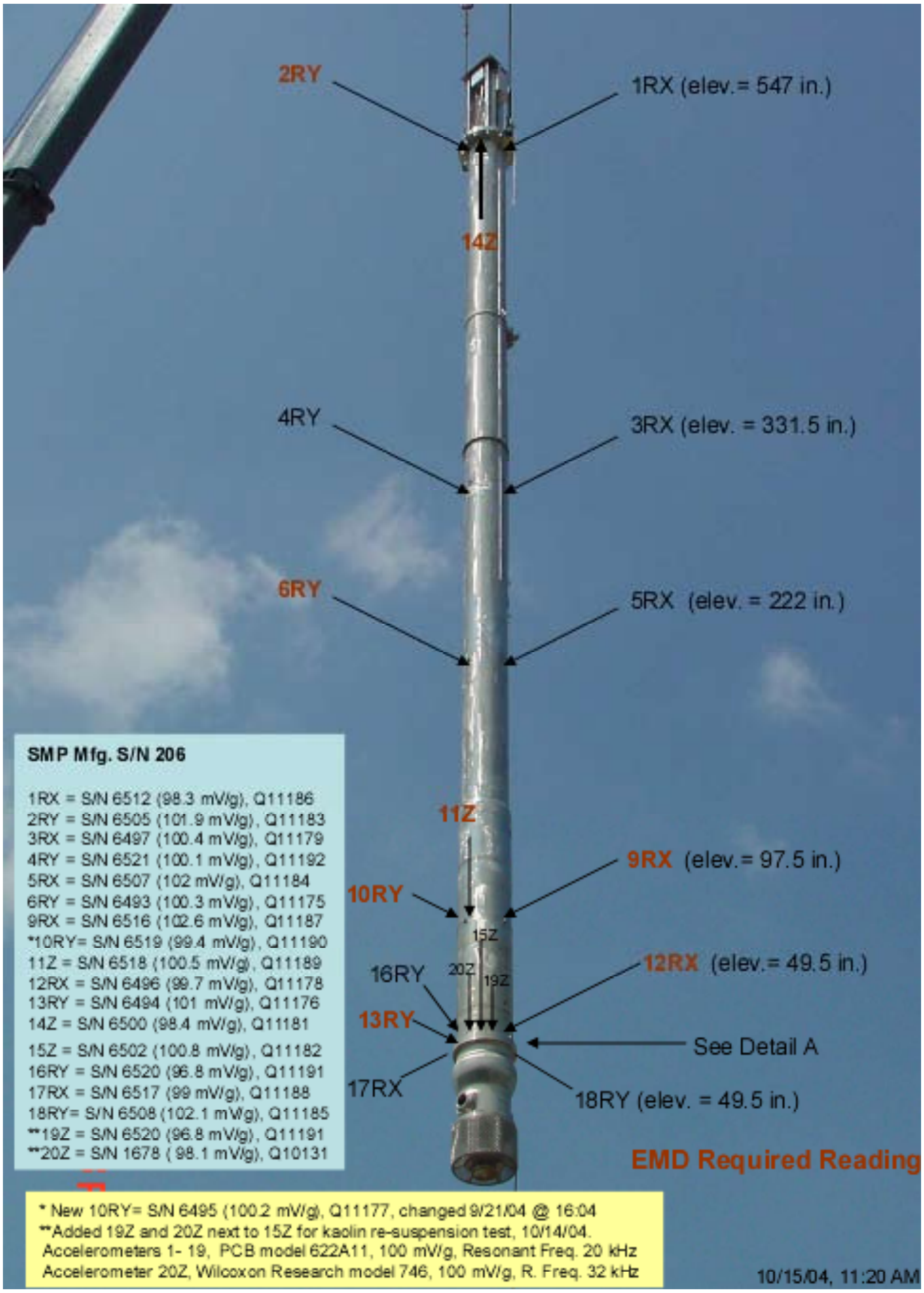


Accelerometer Setup

Figure (1)



Final Accelerometer Locations
Figure (2)

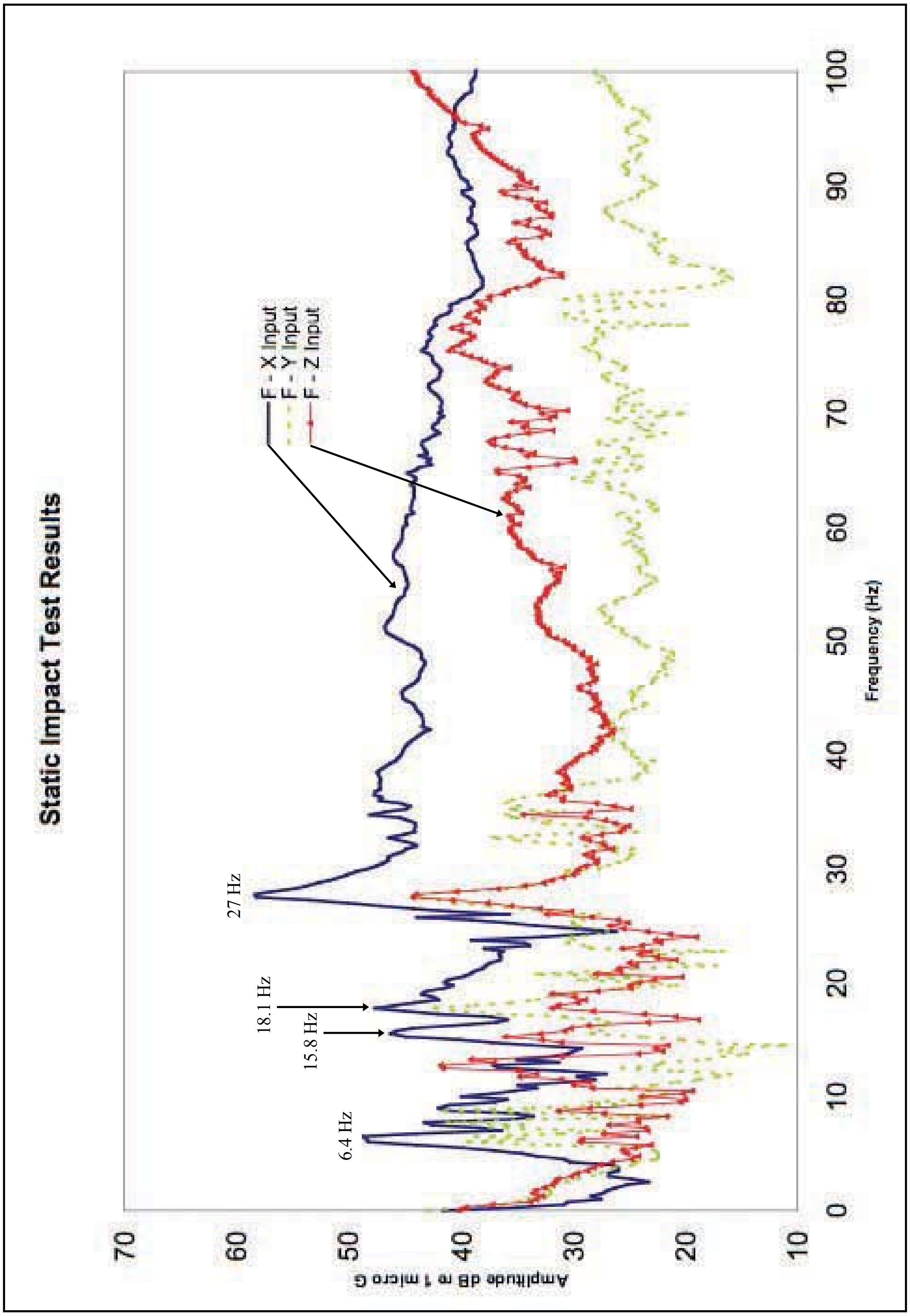
Conducting Initial Static Impact Test
Figure (3)

SMP Qualification Testing

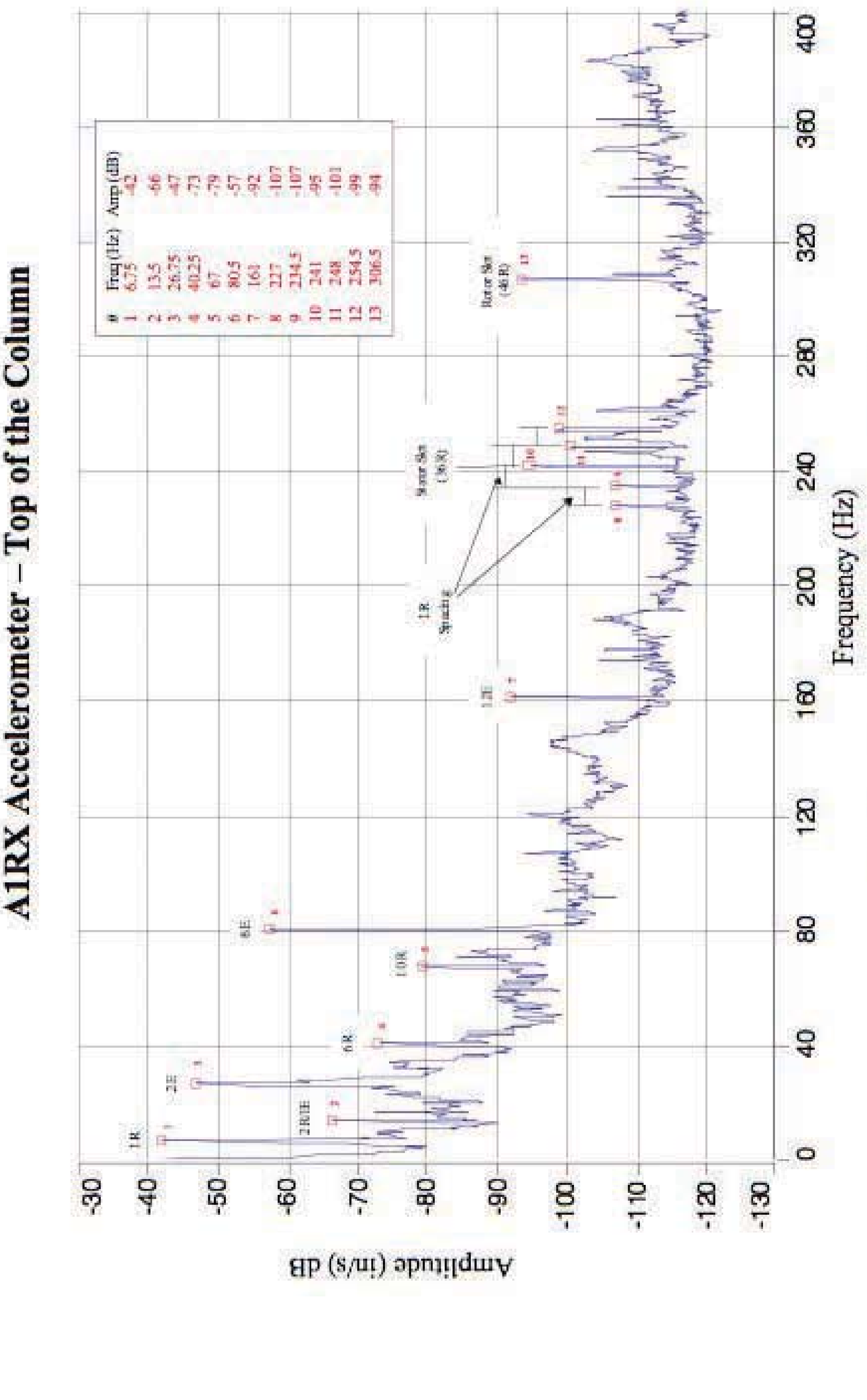
Figure (4)

SMP Operation in Water
Figure (5)

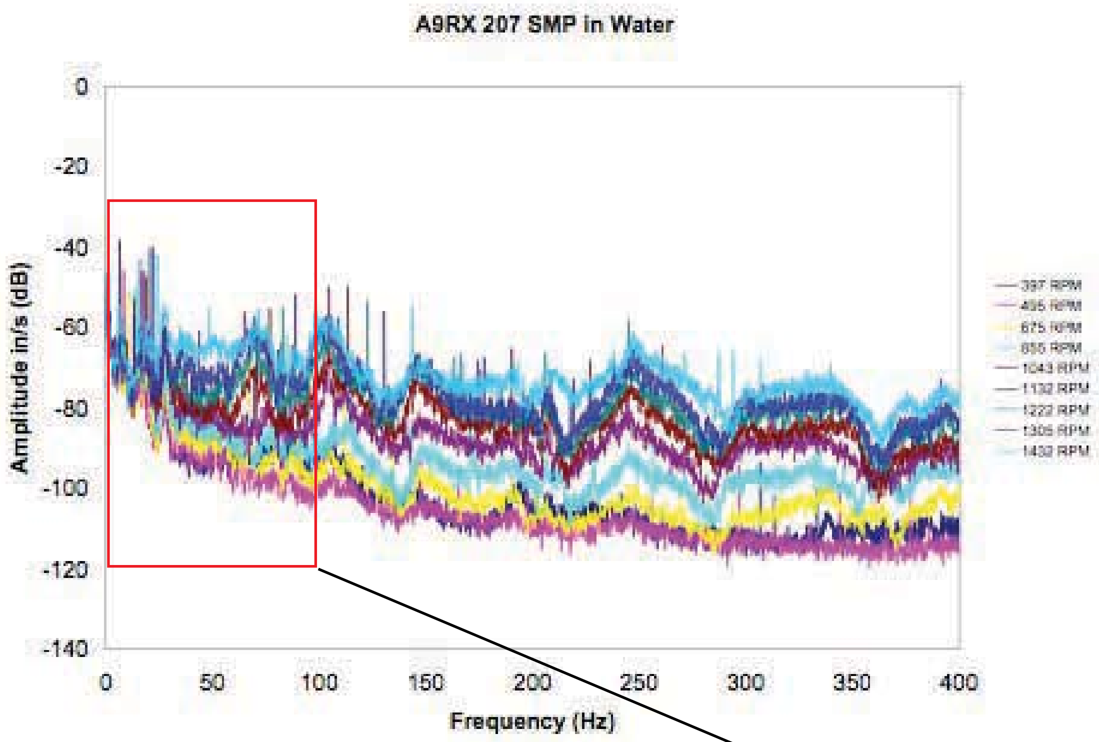
SMP Operation in Kaolin and Sand Mixture
Figure (6)



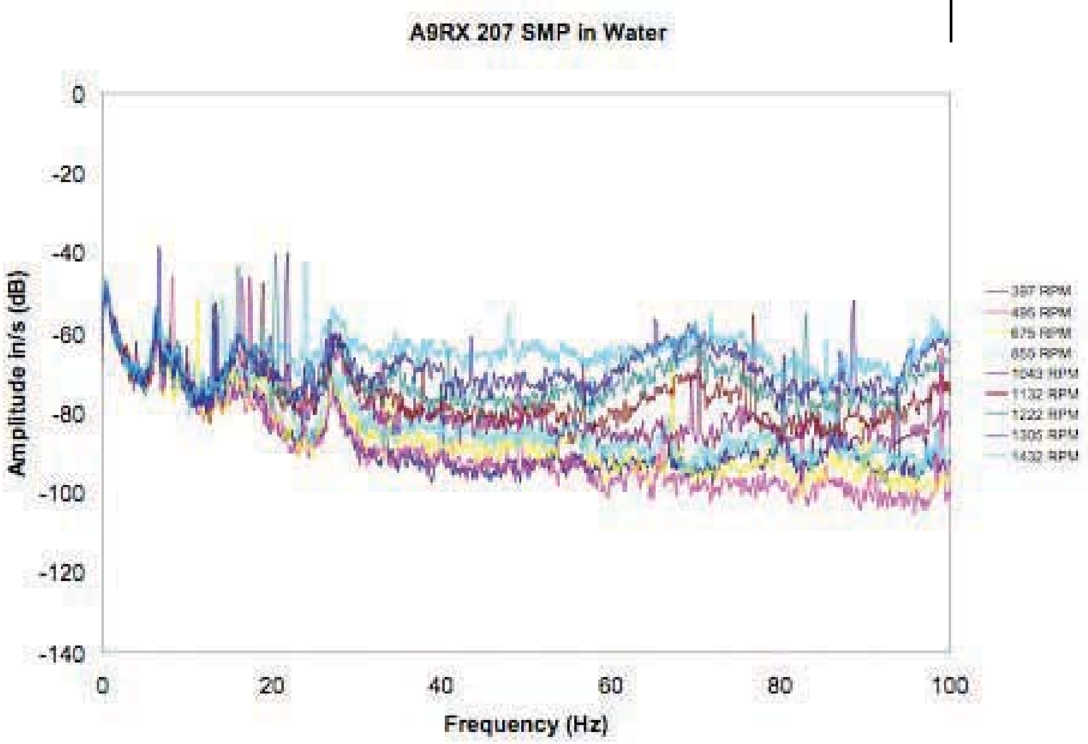
Static Impact Results on an SMP Unit
Figure (7)



Typical Accelerometer Response of an SMP Unit with Tonal Identification



SMP Operational Autospectra at Multiple Speeds
Figure (9)



SMP Operational Autospectra – Zoom of Frequency Range of Interest
Figure (10)

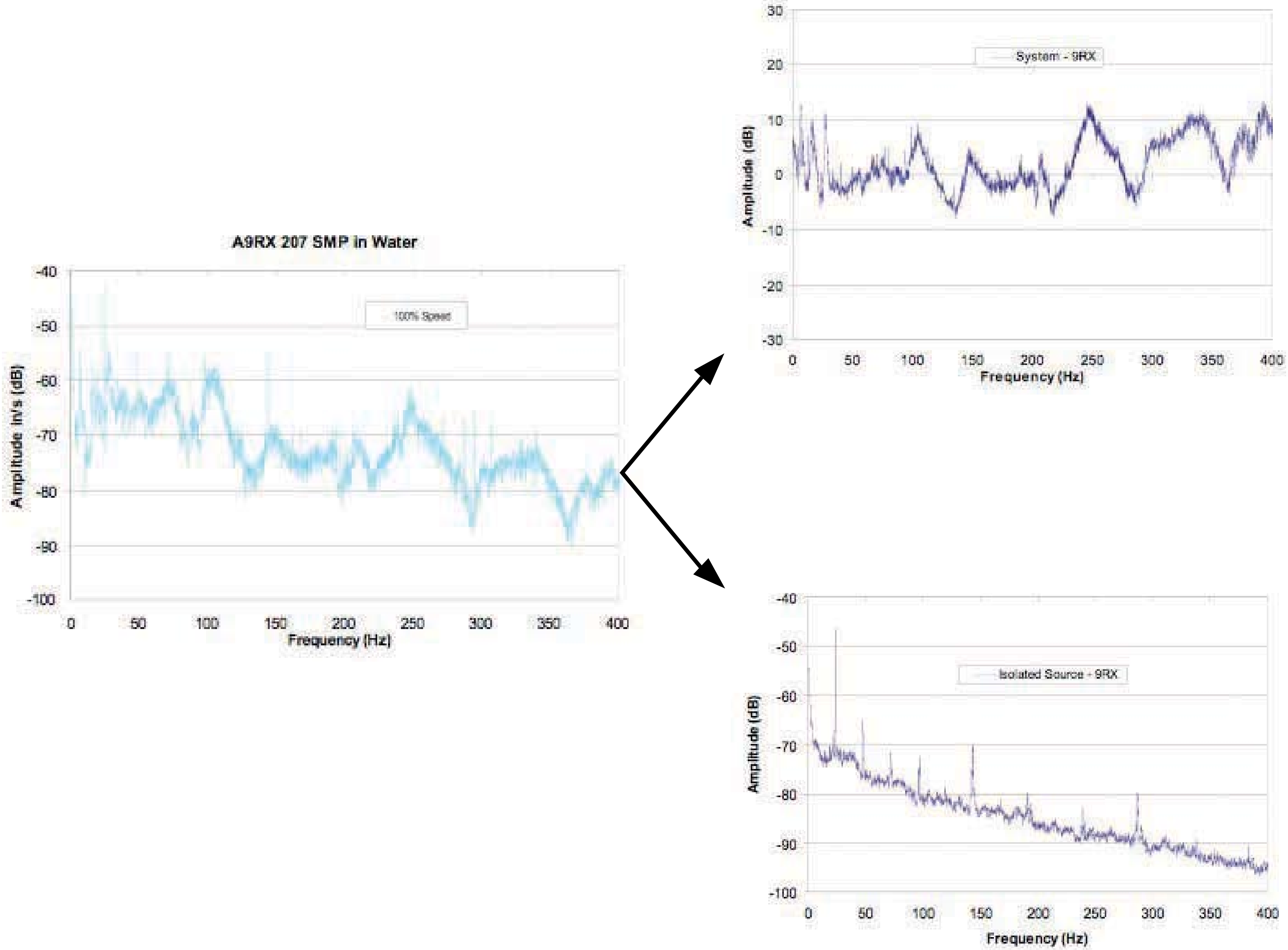
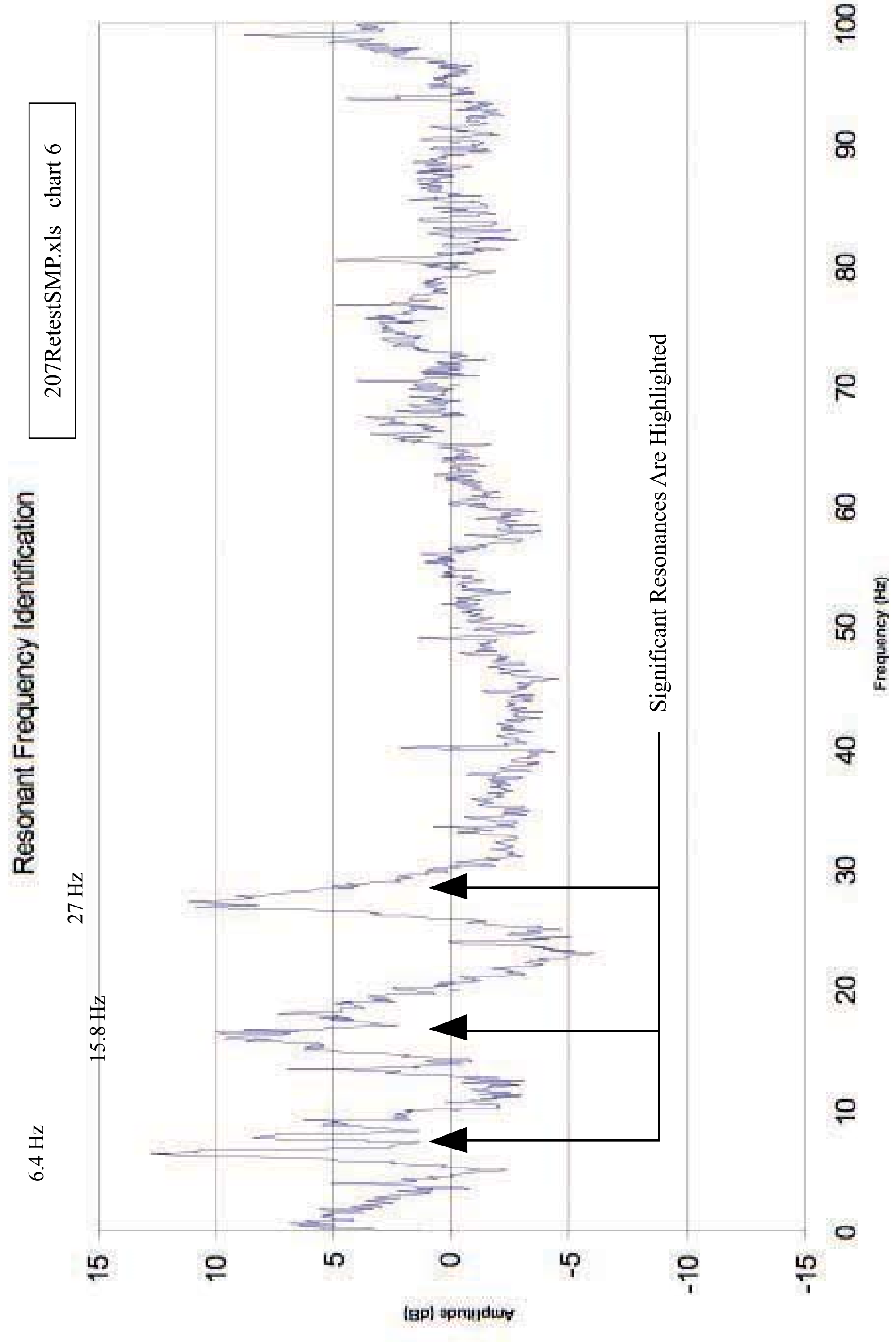
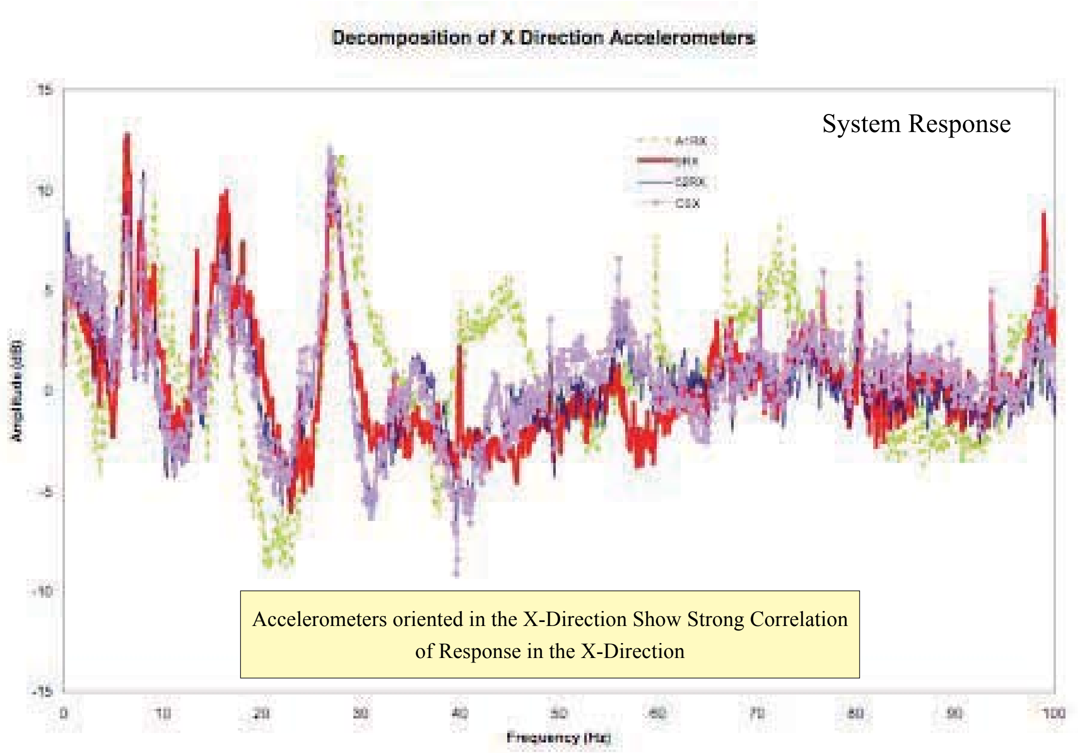


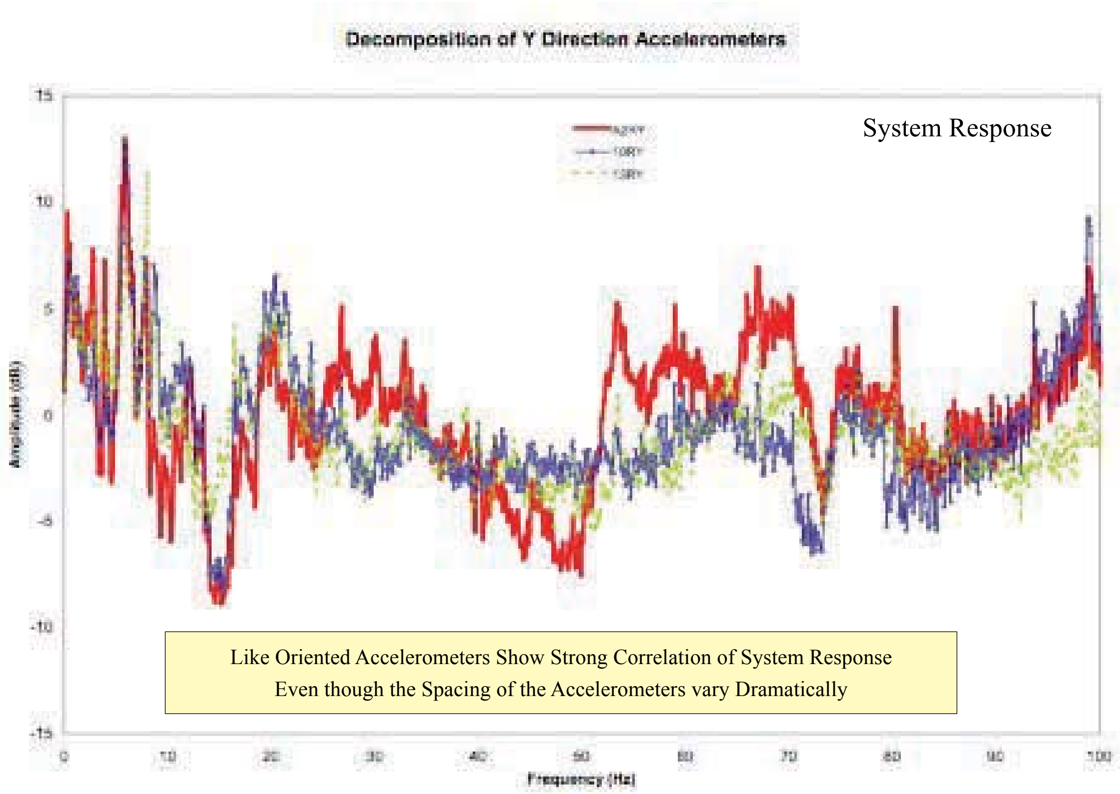
Illustration of the Decomposition Process
Figure (11)



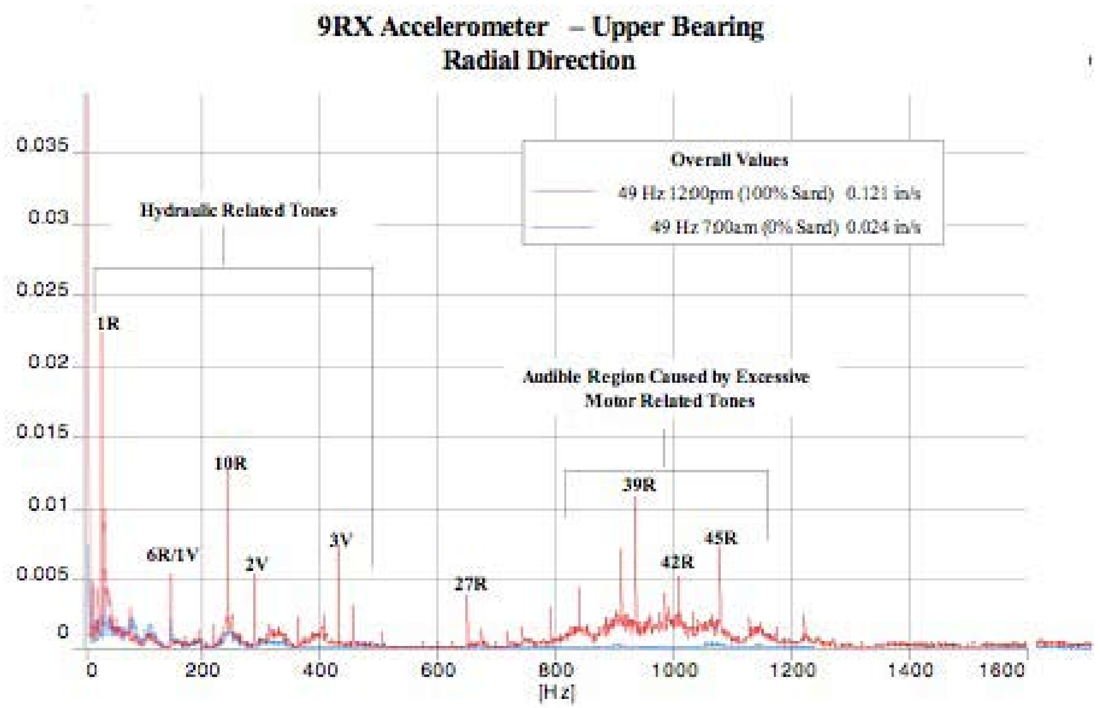
System Response Results of the Decomposition Process on an SMP Unit
Figure (12)



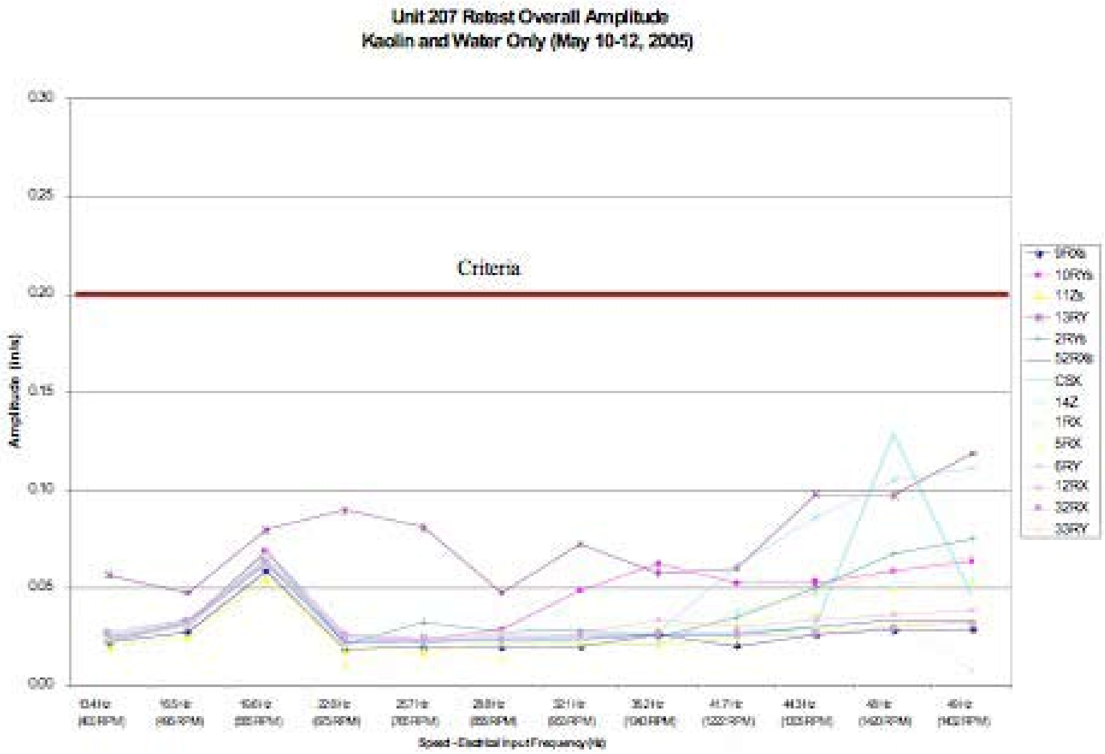
Combined (X –Direction) Oriented Decomposition Results on an SMP Unit
Figure (13)



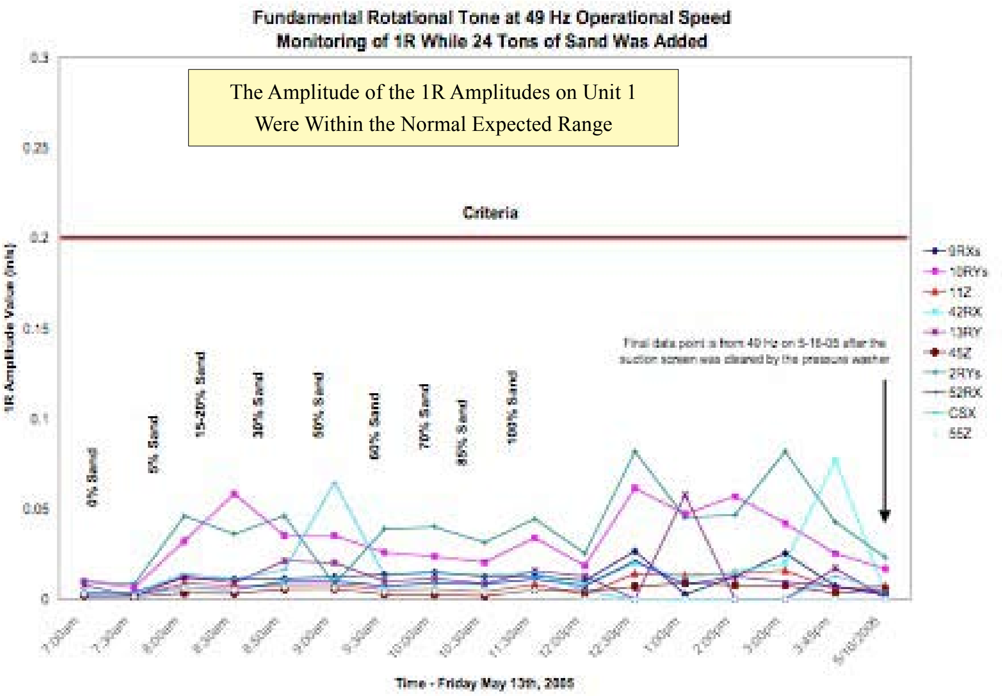
Combined (Y –Direction) Oriented Decomposition Results on an SMP Unit
Figure (14)



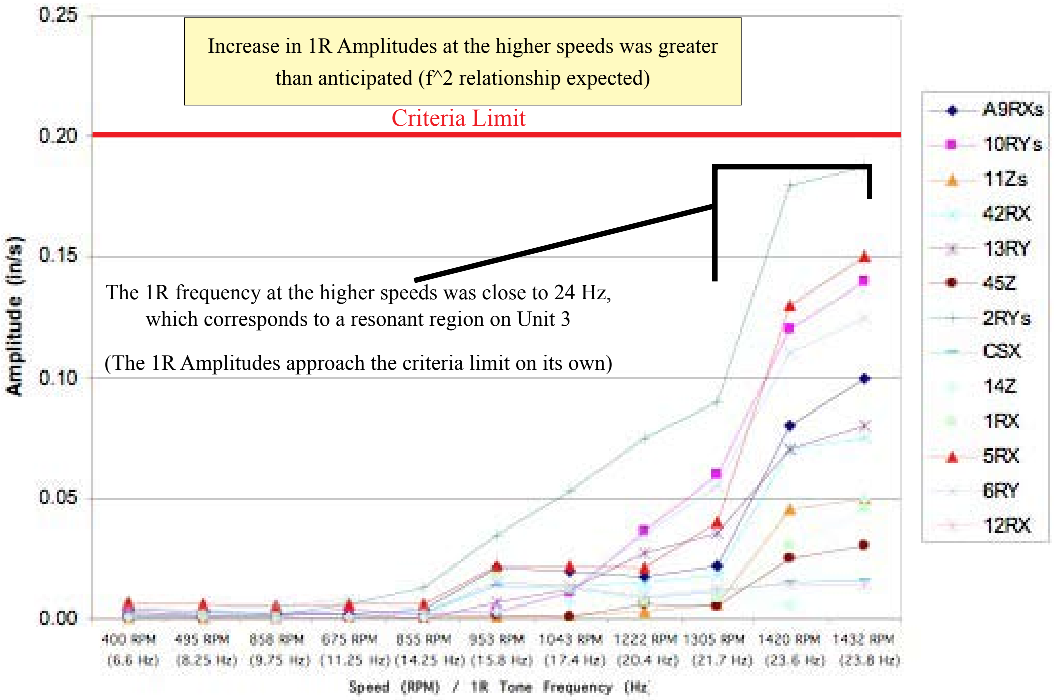
Effect of Kaolin and Sand Mixture on Narrowband Vibration Performance
Figure (15)



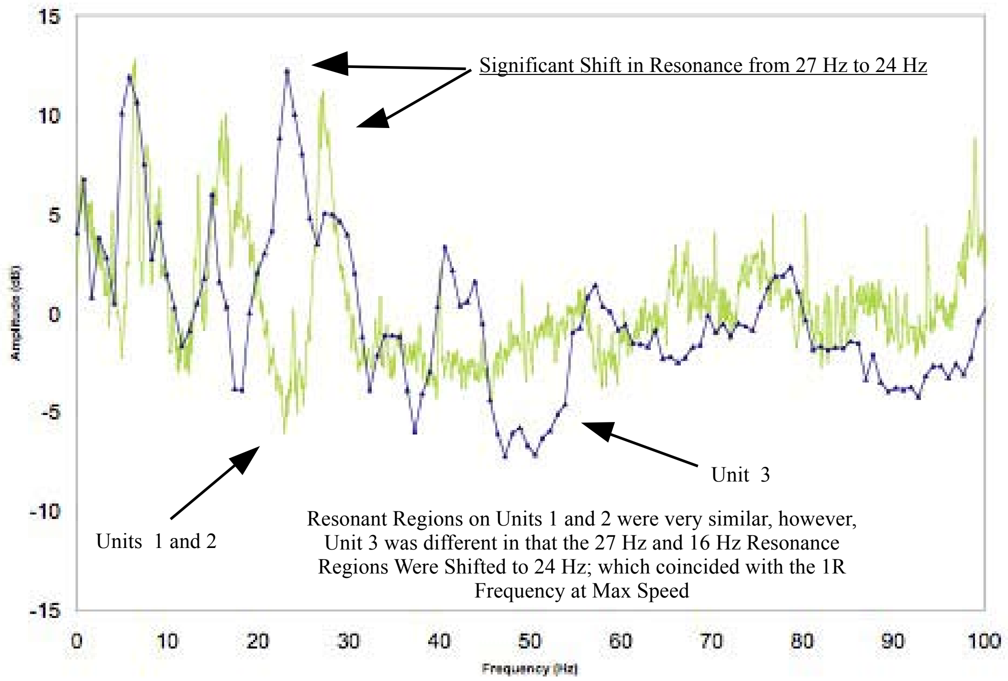
Effect of Kaolin and Sand Mixture on Overall Vibration Performance
Figure (16)



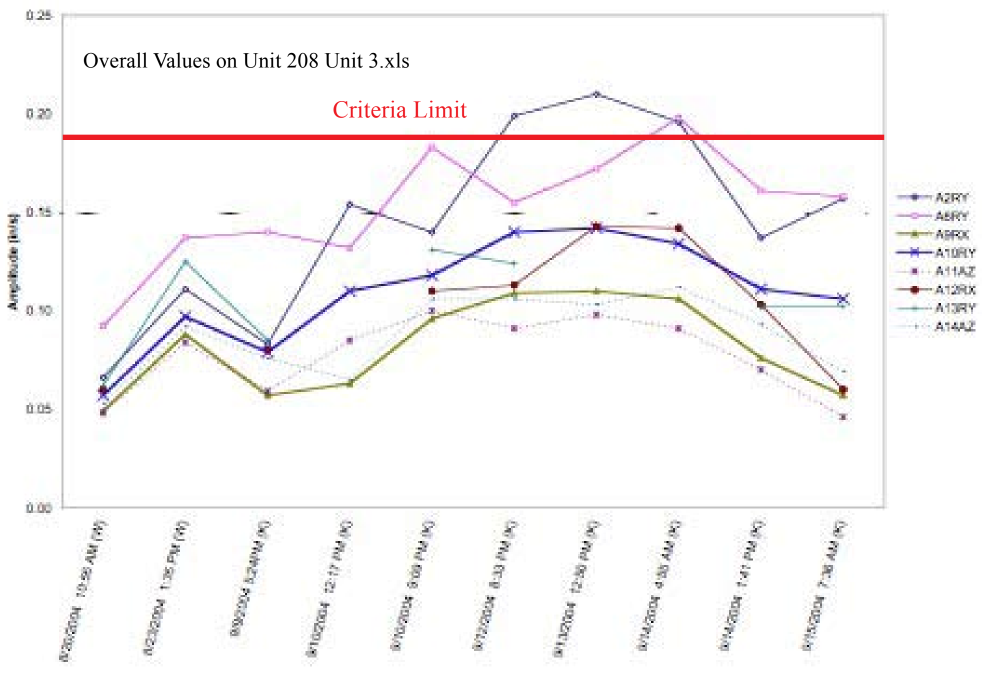
Effect of Kaolin and Sand Mixture on Narrowband Vibration Performance
Figure (17)



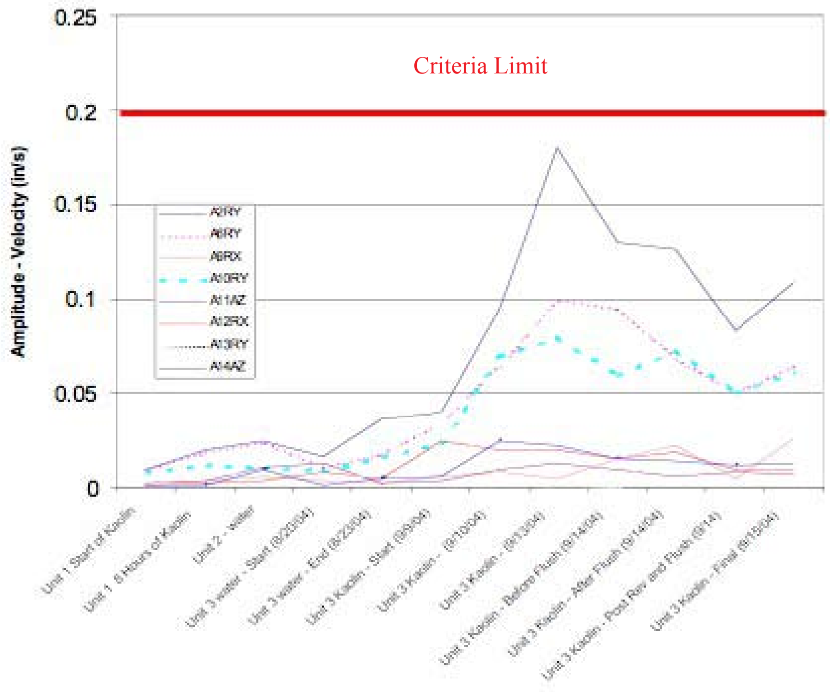
Effect of Kaolin and Sand Mixture on Overall Vibration Performance
Figure (18)



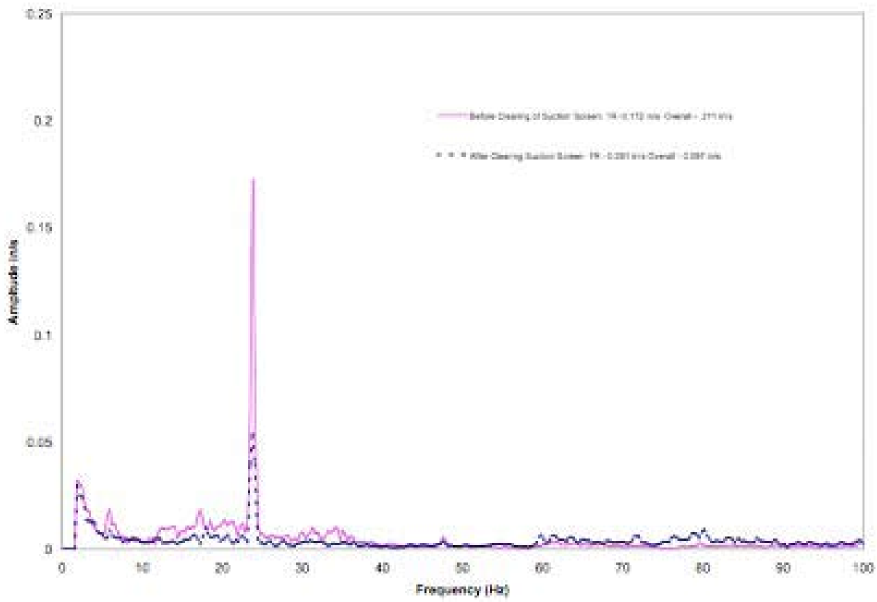
Effect of Kaolin and Sand Mixture on Overall Vibration Performance
Figure (19)



Effect of Kaolin and Sand Mixture on Overall Vibration Performance
Figure (20)



Fundamental Rotational Vibration Performance at Maximum Speed
Figure (21)



Effect of Flushing the Suction Screen on the 1R Amplitude at Max Speed
Figure (22)

AVAILABILITY OF INPUTS REQUIRED FOR PWR ECCS AUXILIARY COMPONENT-SPECIFIC EVALUATIONS IN SUPPORT OF GSI-191

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Abstract

Generic Safety Issue 191 (GSI-191) raised concerns about the potential for debris ingested during post-LOCA recirculation into the Emergency Core Cooling System (ECCS) and Containment Spray System (CSS) to affect the performance of systems, structures and components. The possible impacts of debris ingestion include blockage of close clearance flow paths and wear of component surfaces. This paper describes the challenges associated with obtaining inputs for the downstream effects evaluations of auxiliary components. Data collection for each evaluation starts with the identification of the flow paths of the sump debris-laden fluid, proceeds to the identification of the associated components, and ends with the determination of the as-installed conditions. Industry efforts to address GSI-191 have uncovered a number of issues relative to data availability to support the evaluations. Lessons learned from plant-specific evaluations of downstream effects of components will be discussed.

Introduction

In the event of a Loss of Coolant Accident (LOCA) in a Pressurized Water Reactor (PWR), the Emergency Core Cooling System (ECCS) begins injecting water from the accumulators and refueling water storage tank into the Reactor Coolant System (RCS) in order to cool the core; and into the containment spray system (CSS) in order to lower the pressure in the containment (for a large break LOCA). This is the injection phase. Following a LOCA, water is continuously discharged from the break and from the containment spray nozzles and collects inside the containment. Once the accumulators and refueling water storage tank are depleted, the ECCS is realigned to take suction from the containment sump for the recirculation phase. During this phase, some of the debris generated by the break flows with the reactor coolant through the ECCS and CSS into the reactor vessel and into containment, then back to the sump, where it is recirculated through the ECCS, CSS, and reactor vessel again. Typically, a containment sump contains one or more screens in series that are designed to filter debris in order to minimize the ingestion of particles in the ECCS, CSS, and reactor vessel.

Concerns have been raised about the potential for debris that passes through the containment sump screens and is ingested into the ECCS and CSS as to how it would affect the performance of systems, structures and components. Possible impacts include blockage of small flow paths, pump seizure, and the wear and abrasion of component surfaces. In September 2004, the Nuclear Regulatory Commission (NRC) issued Generic Letter (GL) 2004-02 (Reference 1) to address GSI-191 (Reference 2), Post-Accident Containment Sump Performance. GL 2004-02 requested licensees to perform a “downstream effects” evaluation of their ECC and CS systems.

To perform this evaluation, licensees need both applicable information and an evaluation methodology. Therefore, the Westinghouse Owners Group (WOG) sponsored the development of a methodology and data collection effort to evaluate the effects of debris ingested into the ECCS and CSS during post-accident recirculation phase. Westinghouse was also contracted to perform a number of the downstream effects evaluations. This included evaluations of the auxiliary components, including all valves, pumps, heat exchangers, orifices, spray nozzles, and instrumentation lines that could be subjected to containment sump debris-laden fluid.

Data Collection – Step 1

In order to evaluate ECCS and CSS auxiliary components for plugging and wear, a number of different inputs are required. The first step of the data collection process is to request input data from the plant. A generic list of required inputs for the evaluation is typically provided with the initial discussion. With subsequent discussions, a second list of more specific inputs is created based on the project scope and is sent to the plant. These lists are generally not specific enough and the requested data may be difficult to retrieve, causing delays in beginning the evaluations. As evaluations were completed and lessons learned, these input lists became more refined. However, not providing plants a complete and well-defined list of required inputs has been identified as one of the causes of schedule delays in the evaluation process.

Another challenge in this process is that once the input data is sent by the plant, the engineers have the time-consuming task of reviewing the documents and collecting the specific data needed in order to perform the evaluations. This task is made more difficult by the fact that when the input data is provided, it is often unorganized, and electronic files are named with non-descriptive titles, so that it is impossible to know what is contained in the files without considerable effort.

Data Collection – Step 2

The second step of the data collection process is to identify the flow paths during ECCS cold-leg and hot-leg recirculation. The inputs required for this step are the plant Piping and Instrumentation Diagrams (P&ID's), Emergency Operating Procedures (EOP's), and the applicable System Descriptions, or Design Basis Documents. Information about which valves open and close during the injection, and the cold-leg and hot-leg recirculation phases is required to determine the correct flow paths of the debris-laden fluid.

For some of the plant-specific evaluations, the plants had difficulty providing inputs in a timely manner before the evaluations began, due to the tight schedules associated with the projects. For these cases, historical data was usually used, when available. This was done in order to utilize the time spent waiting for inputs. However, in some cases, there was no time savings, but actually a time penalty due to rework. For example, an engineer working on a valve evaluation used the original P&ID's to initially determine the flow paths. Once the plant input was provided, it was discovered that one valve which was indicated to be closed upon receipt of an "S" signal on the original P&ID's, was shown in the Design Basis Document to be open on an "S" signal, and furthermore was to be de-energized to remain open during post-LOCA injection and recirculation modes. This also opened up another flowpath that included more valves, orifices, and a heat exchanger which needed to be evaluated.

Data Collection – Step 3

The third step in the data collection process is to identify the different auxiliary components located in the flow path, using the P&ID's and System Descriptions. Latest revisions of these plant documents are required in order to identify all existing components and their respective identification numbers.

The ID number is especially important for valves and orifices in order to locate the appropriate drawing of the component. As some engineers found during their data collection, historical component ID's sometimes differ from the plant ID's. Furthermore, in some of these cases, the documents provided by the plant referenced both plant and original ID's. For example, in one case, data collection for an instrumentation line evaluation revealed that plant specific ID's were used on the P&ID's. However, since the original instrumentation drawings were provided by the plant, the NSSS designer's ID's were used on the drawings. This required additional input from the plant in order to match the plant component ID with the designer's historical component ID.

Similarly, when valve drawings were not provided by the plant, historical drawings existing in the NSSS vendor files were used, if available. Like the case described above, additional input had to be requested from the plant to match component ID's.

In some cases, components were mistakenly omitted in evaluations because they were not identified in any of the supplied or historical references. For example, for one 2-unit plant, the number of containment spray nozzles in each unit differed. However, all references provided, including the FSAR, System Descriptions, and P&ID's, indicated that there were similar numbers of spray nozzles in each unit. This caused schedule delays due to rework once the error was discovered during the review process.

Data Collection – Step 4

The fourth step is to collect all necessary information about the components to be evaluated. The required information includes component materials in contact with the debris-laden fluid, internal dimensions of parts and components through which the fluid flows, and, in the case of throttle valves, the position at which each valve is set. This information is obtained from numerous sources including drawings, equipment manuals, vendor catalogs, System Descriptions, and plant Final Safety Analysis Reports (FSAR).

One challenge for this step is obtaining up-to-date information about the components. As stated above, many plants did not provide complete inputs before the evaluations began, and in these cases, the NSSS designer's historical information was used. However, many plants have had to rebuild some of their pumps, replace heat exchangers, add or replace valves and orifices, etc., and in those cases, the historical data may not match the current plant configuration.

Another challenge that applies to the valve evaluations is to identify the position of throttle valves when the plant cannot provide the information. Determining the actual throttle valve position involves extra calculations, for which the methodology is documented in Reference 3. There were also some throttle valve evaluations for which the information required to calculate the throttle valve position was not known. In these cases, the minimum position to avoid plugging was assumed for these valves, and that position was used for the erosion evaluation. The minimum position to avoid plugging is determined based on the sump screen hole size. This also presented some difficulty because although plants knew they would change their sump screen, they did not necessarily know the size of their future sump screen holes.

When plants did not have the complete valve dimensional data required, the engineers performing the evaluations contacted the vendors to obtain the dimensions needed. In some cases, the engineers were able to determine that two different plants had identical valves. If the data required was available for one plant and not the other, it was assumed to be applicable to both plants.

For plants with 2 units, sometimes data was only supplied for one, or incomplete data was supplied for both units. In these cases, the engineers assumed one unit was the same as the other. This assumption had to be verified by the plant, and additional information was supplied, as necessary. The verification process was generally tedious and time-consuming.

Data Collection – Step 5

The fifth step is to perform the evaluation. There are some additional inputs required for the evaluation, including material hardness numbers and debris characterization and concentration.

Material hardness numbers are used in the abrasive and erosive wear calculations. The hardness numbers proved to be difficult to identify accurately, since they can vary for a given material depending on the heat treatment or process involved with manufacturing the part. These details are not given in most available vendor drawings; therefore, information from industry codes and standards, as well as historical data were collected and applied consistently in all component evaluations performed in support of GSI-191.

Debris characterization and concentration are also inputs to the analysis. The size and type of the debris are important, as well as the concentration of debris in the containment sump.

Conclusion

The authors conclude that the problems incurred during the gathering of inputs for the plant specific evaluations would in large part not exist had there not been tremendous time pressure to complete the evaluations. Plants were provided with a large-scale list of inputs required for the evaluations. This list was sometimes general and unclear, and resulted in inputs from plants that were often unorganized and incomplete. Furthermore, due to the number of inputs required, plants often did not have time to collect the references and send them before the evaluations began.

Therefore, historical documents were sometimes used, assumptions had to be made, and open items were left in calculation notes for plant verification.

Once a few evaluations had been completed, subsequent evaluations were easier to complete. Engineers were more familiar with the data required, and were able to refine the input requests. Furthermore, once the input data was collected, the engineers had gained the experience to know what documents may contain the specific information they needed. In the case of the throttle valves, a methodology was developed to calculate the throttle valve position, when the position was not known by the plant (Reference 3). Finally, the engineers had a better understanding of which assumptions were reasonable.

Although many lessons have been learned, and common pitfalls recognized, data collection for evaluating auxiliary components for downstream effects remains a difficult and time-consuming task for both plants and analysts. Many plants do not have all of the data needed for the evaluation, and even when they do, it takes time to find the documents required and to extract the required data from those documents. Westinghouse is currently implementing a program to analyze and improve the process involved with performing downstream effects analyses. Going forward, these improvements will facilitate the data collection process and improve the efficiency of the evaluations.

Acronyms

ECCS – Emergency Core Cooling System

CSS – Containment Spray System

LOCA – Loss of Coolant Accident

RCS – Reactor Coolant System

P&ID’s – Piping and Instrumentation Diagrams

NSSS – Nuclear Steam Supply System

FSAR – Final Safety Analysis Report

WOG – Westinghouse Owner’s Group

EOP’s – Emergency Operating Procedures

Acknowledgements

The authors are grateful for the constructive comments of Messrs Gary Corpora and Rolv Hundal, both of Westinghouse Electric Company, Pittsburgh, PA, USA.

References

1. NRC Generic Letter 2004-02: Potential Impact of Debris Blockage on Emergency Recirculation During Design Basis Accidents at Pressurized-Water Reactors, September 13, 2004.

2. Generic Safety Issue (GSI) 191, Assessment of Debris Accumulation on PWR Sump Performance.

3. L. I. Ezekoye, W. E. Densmore, “An Approach to Estimating PWR ECCS Throttle Valve Positions in Support of GSI-191 Evaluations,” presented at the NRC/ASME Symposium on Valves, Pumps, and Inservice Testing, July 2006.

Emergency Core Cooling Pump Performance with Partially Voided Suction Conditions

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Abstract

This paper summarizes a full-scale test program conducted for the Palo Verde Nuclear Generating Station. The test program investigated the impact of a partially voided pump suction flow path on pump mechanical and hydraulic performance. The test program included a multi-stage horizontal centrifugal high pressure safety injection pump and a single stage vertical centrifugal containment spray pump. Air was injected into the pump inlet flow stream at a rate sufficient to produce a desired inlet air volume fraction at the pump suction. An observation port was provided in the high pressure pump suction piping immediately upstream of the pump to allow observation of the flow regime. Pump hydraulic and mechanical performance was monitored during the process. This testing allowed the impact of the inlet air volume fraction on pump performance to be determined.

Prior scale model testing was used to predict the transport of an air volume initially trapped in a horizontal segment of the containment sump outlet line through a vertical down-comer and subsequently into the Emergency Core Cooling System (ECCS) and Containment Spray (CS) pump suction lines. The range of inlet air volume fractions at the pump suction bounded those predicted during the scale model testing. A range of pump flow rates that encompassed the flow rates expected during postulated loss-of-coolant-accident (LOCA) events were investigated in the test. This paper provides a description of the test facility and test processes, along with an overview of the impact of inlet air volume fraction on pump performance for each of the pumps tested.

Introduction

The Palo Verde Nuclear Generating Station (PVNGS) identified a potential concern where an air volume could be introduced into the suction lines to the Emergency Core Cooling System (ECCS) pump and Containment Spray (CS) pump. A three phase test program was developed to address this concern. This paper discusses the third phase of the test program which was a full-scale test program in which the ECCS and CS pumps were operated under the simulated inlet air volume fraction fluid conditions. The prior phases involved scale model testing of the PVNGS specific suction piping network that established the potential extent of air the pumps would have to experience.

The purpose of the testing was to determine if temporary performance degradation occurs during the ingestion of an inlet air volume fraction, and to identify if any permanent degradation of performance after un-voided inventory returns to the pump. Based on the results of the scale piping network testing phases it was apparent that the pumps would possibly experience significantly more air than had been documented in the available literature.

This paper provides a description of the test facility and test processes, along with an overview of the impact of inlet air volume fraction on pump performance for each of the pumps tested.

Background

PVNGS had identified a scenario in which a pocket of air was trapped between a pair of closed motor operated isolation valves and a check valve. This pocket of air was in the containment building emergency sump recirculation flow path to the ECCS and CS pumps that could be drawn into the operating pump suction during a postulated design basis loss of coolant event.

The first phase of the test program investigated the manner in which the liquid outflow from the sump interacted with the trapped pipe air volume, and the ability of the liquid outflow to transport air through the piping network; specifically, the flow pattern of the two-phase mixture in the piping down-comer.

The second phase of the test program investigated the nature of the two phase flow pattern that is produced in the pump suction piping for the High Pressure Safety Injection (HPSI), Low Pressure Safety Injection (LPSI), and CS systems' pumps after transportation of the initial trapped air volume from the horizontal piping through a vertical down-comer and to the pump suction lines. This work predicted a range of inlet air volume fractions at the various pump suction lines.

Both the first and second phase of testing were performed by scale model testing. That testing program is described in Reference [1].

The third phase of the test program took the findings of the first two phases and performed full scale pump testing where the predicted inlet air volume fractions were introduced to a HPSI and CS pump during a range of operating conditions. This range of operating conditions encompassed the flow rates expected during postulated loss-of-coolant-accident (LOCA) events.

Test Article Description

The equipment to be tested consists of two pump / motor assemblies;

An actual spare PVNGS HPSI pump and motor was used and is an Ingersol-Rand model 4 x 11 CA8 described as follows:

Motor (CA):

Westinghouse Electric Frame 5810H

Class 1E

Rated at 1000 Horsepower (HP), 3-Phase, 60 Hz, 4000 Volts

Speed: 3553 rpm (revolutions per minute)

Weight: 4,800 lbs

Motor Identification Number: 17535LN01

Pump (CA):

4x11CA-8

Nameplate Head = 2850 feet (ft)

Horizontal shaft

Nameplate Rated flow = 900 gpm (gallons per minute)

Weight: 4,400 lbs

Suction diameter: 10" sch 40

Discharge diameter 4" sch 80

Pump Serial Number: 117814

A photograph showing the HPSI (CA) pump and motor installed in the test loop during facility construction is illustrated in Figure 1.

To simulate the CS pump and motor, a salvaged pump (an equivalent PVNGS LPSI pump and motor) was obtained. This pump is an Ingersol-Rand 8x20 WDF described as follows:

Motor (WDF):

Westinghouse Electric Frame 55010-P39

Rated at 500 HP, 3-Phase, 60 Hz, 4000 Volts

Speed: 1776 rpm

Weight: 4,500 lbs

Motor Identification Number: IS-78

Pump (WDF):

Nameplate Head = 335 ft

Vertical shaft

Nameplate Rated flow = 4300 gpm

Weight: 4,400 lbs

Suction diameter: 14” sch 40

Discharge diameter 8” sch 40

Pump Serial Number: 087634

A photograph showing the CS (WDF) pump and motor installed in the test loop is illustrated in Figure 2.

Test Facility Description

The test facility is a two closed loop system consisting of a 30,000 gallon pressure vessel with one loop for each test specimen pump. One loop is the test loop for the HPSI (CA) pump / motor and the piping and control valves are sized based on the supplied pump curve. The second loop is the test loop for the CS (WDF) pump / motor and the piping and control valves are sized based on the supplied pump curve. Each loop is fitted with an air injection device (described later) in the pump’s suction piping. Piping between the location of the injection device and pump inlet simulated actual plant orientation.

The pressure vessel has the ability to be pressurized to a specified pump suction pressure. This pressure vessel pressure can be adjusted and controlled.

The test medium was de-ionized water under ambient conditions.

The overall test loop is illustrated in Figure 3 and a General Arrangement drawing is provided in Figure 4. A 4” 900# globe control valve was installed downstream of the HPSI (CA) pump to provide pump flow adjustment through the test sequence. The inlet piping is 10” schedule 40 and the outlet piping is 4” schedule 80.

An 8” 300# globe control valve was installed downstream of the CS (WDF) pump to provide pump flow adjustment through the test sequence. The inlet piping is 14” schedule 40 and the outlet piping is 8” schedule 40.

The flow control valves for each test loop are illustrated in Figure 5. Flex connectors were installed both upstream and downstream of each of the test specimen pumps in the supply and return pipe lines to allow for minor thermal expansion.

Air injection was provided by introducing compressed air into the water flow using a specifically designed air nozzle to disperse the air that enters the suction piping at the side of a 90 degree elbow to inject the air in the flow direction. The air supply was provided to an air control valve at 100 psig from an air compressor. The actual pipe insert into the suction line and the air injection controls system are illustrated in Figures 6 and 7, respectively.

A section of transparent piping was provided in the 10” suction line for the HPSI (CA) pump as illustrated in Figure 8. Video was recorded of the sight glass results during the air injection testing.

The air volume was determined by measuring the volumetric flow of water in the inlet piping prior to the location where the air was input. The mass flow of the air was also measured prior to the location where the gas enters the inlet piping. The ratio of the volumetric flow of air to the total flow (air and water) in identical units provides the void fraction.

Instrumentation

Following the HPSI (CA) test specimen pump and motor installation and alignment, the instrumentation was installed. A similar instrumentation approach was used for the CS (WDF) test specimen pump and motor, but is not included here.

The following table summarizes the instrumentation used for the test program and the identification numbers (TAG) used by Wyle Laboratories:

Test Program Description

The intent of the testing was to determine if temporary performance degradation occurs during the ingestion of a void fraction, and to identify any permanent degradation of performance after un-voided inventory returns to the pump.

Each pump was tested individually. Initially, each pump was run through a standard multi-point performance curve to baseline the pump performance.

A test matrix of test conditions for the inlet air volume fraction for each pump was developed, based on the Phase 1 and 2 scale model tests. The test matrix used a graded approach for the air volume fraction for the HPSI (CA) pump. The graded approach to the level of air volume fraction, meaning a carefully controlled step wise increase in air volume fraction at successive initial pump flow rate conditions, was necessary since the ultimate final extent of air volume injection being tested was significantly beyond that previously known to be documented and the pump's limit of air fraction tolerance would likely result in mechanical failure.

Each test case run injected the air for a sufficient time to simulate the trapped air volume's transport through the pump.

Test Results

The test matrix is shown in Figure 9. The HPSI pumps were tested over a range of flow rates that were representative of the expected variation in operating point following postulated post-accident conditions. The flow rates chosen for the HPSI pump were the design (best efficiency) point of 900 gpm, the nominal system full flow rate of 1300 gpm, and a reduced flow rate of 600 gpm. The CS pump was only tested at the nominal system full flow rate of 4900 gpm, which corresponds to its expected post-accident operating point.

A multi-point performance run was performed for each pump to base-line its performance prior to testing under voided conditions.

The test runs in the matrix were performed by first establishing the pump liquid flow rate at the desired value. Air was then injected at a gradually increasing rate until reaching the desired air volume fraction, held at that rate for a specified time, and then gradually decreased. The pump was then run for a specified amount of time to detect any change in performance.

Figures 10a-d show the results of the HPSI performance tests at 900 gpm. The pump performance has been normalized to the nominal pump head at 900 gpm. It is noted that there is very little change in pump developed head with 6% air volume fraction at the pump suction conditions. The impact on pump head becomes more pronounced as air volume fraction is increased. It is noted that the pump head oscillates at a fixed frequency for each air injection test. This is due to the fact that air collected at the elbow immediately upstream of the vertical pump suction nozzle and was periodically swept downward into the pump. This phenomenon was observed at the sight glass in the pump suction piping. The air volume fraction was defined as the ratio of air volumetric flow rate to liquid volumetric flow rate at the pump suction. The air injection rate was steady during the test, but the liquid flow rate fluctuated as air was periodically collected and purged from the pump suction pipe. This gives rise to the oscillatory nature of the air volume fraction measurement.

The magnitude of the change in pump head increased as the air volumetric fraction increased. In all cases, the pump performance returned to its base-line value at the conclusion of the tests. The normalized performance of the HPSI pump at 600 gpm and 1300 gpm was very similar to the performance at 900 gpm and is not shown. The HPSI pump was shown to be remarkably tolerant of air ingestion; continuing to produce significant, albeit degraded-from-base-line, discharge head and flow at air volume fractions approaching and exceeding in some cases 30%. In all cases, the pump performance returned to its base-line value at the conclusion of the tests.

The CS pump test results are shown in Figure 11. The pump performance has been normalized to the nominal pump developed head at 4900 gpm. The magnitude of the change in pump head increased as the air volume fraction increased. Since the CS pump suction enters from below the pump, the air did not collect in the pump suction piping. The pump performance during the test does not demonstrate the pronounced oscillatory nature characteristic of the HPSI pump test. On a relative basis, the CS pump performance was more sensitive to air ingestion than the HPSI pumps.

Test Conclusions

As discussed in the test results, both the pumps were subjected to the postulated air inlet volume fraction established from prior scale piping network testing phases that was significantly more air than had been documented in the available literature.

The multi-stage HPSI pump was remarkably tolerant of air ingestion. Limited industry data (i.e., as reported in NUREG/CR 2792, Ref. 2) had suggested that multi-stage pumps would be more tolerant of air ingestion than single-stage pumps. This test program produced substantial evidence that this was in fact the case.

The test results clearly show that pump performance was impacted by the introduction of the air as illustrated in the Test Results section, but that the pumps continued to operate and move the voids through the pump casing and returned to nominal performance once the voids were fully passed. It was evident that the pumps sustained no mechanical damage during the repeated test cycles since, following the air inlet volume fraction testing, the pump performance was compared to test results taken prior to air injection. As shown in Figure 12, the pump performance curve is the same before and after the air fraction testing. A post test disassembly visual inspection of the HPSI pump confirmed that no mechanical damage had occurred.

It is concluded that no pump degradation occurred during the defined total quantity, large air volume fraction testing. No pumps were harmed in the completion of this test.

References

1. *Scale Model Testing of Air Transport through Pump Suction Piping;* Mark Radspinner (Arizona Public Service), and Robert Hammersley and Robert Henry (Fauske and Associates, Inc.); Proceedings of the Ninth NRC/ASME Symposium on Valves, Pumps and Inservice Testing, 2006.
2. NUREG/CR-2792, “An Assessment of Residual Heat Removal and Containment Spray Pump Performance Under Air and Debris Ingesting Conditions,” September 1982.

HPSI (CA) Pump Loop Instrumentation:

TAG	PARAMETERS
INLET_TEMP	Inlet Water/Air Temperature
AIR_TEMP	Inlet Air Temperature
EXHAUST_TEMP	Outlet Water/ Air Temperature
INLET_PRESS	Test Pump Outlet Pressure
EXHAUST_PRESS	Test Pump Outlet Pressure
MIX_PRESS	Test Pump Outlet Pressure
MOTOR_AMPS_A MOTOR_AMPS_B MOTOR_AMPS_C	Motor Current (three phases)
MOTOR_VAC_A MOTOR_VAC_B MOTOR_VAC_C	Motor Voltage (three phases)
RPM	Motor Speed (rpm)
3Z	Motoro Vertical Vibration Bearing 3
3X	Motor Horizontal Vibration Bearing 3
4Z	Motor Vertical Vibration Bearing 4
4X	Motor Horizontal Vibration Bearing 4
2Z*	Pump Vertical Vibration Bearing 1
2X*	Pump Horizontal Vibration Bearing 1
1Z*	Pump Vertical Vibration Bearing 2
1X*	Pump Horizontal Vibration Bearing 2
1Y*, 2Y*, 3Y, 4Y	Axial Velocity
H20_FLOW	Water Flow rate
AIR_FLOW	Air Flow rate
MIX_FLOW	Water/Air Flow Rate on Discharge Pipe

Bearing Name	Bearing Description
Bearing 1	Pump Radical Bearing (inboard)
Bearing 2	Pump Thrust Bearing (outboard)
Bearing 3	Motor Bearing (inboard, coupled)
Bearing 4	Motor Bearing (outboard)

* The labeling of accelerometer 1 and 2 does not correspond to the bearing numbering system. For the purposes of this test program, the following bearing nomenclature was used:

Figure 1– Photograph showing Installation in Test Loop for CA Pump and Motor Test Specimen during facility construction

Figure 2 – Photograph showing Installation in Test Loop for WDF Pump and Motor Test Specimen.

Figure 3– Overview of Test Facility with 30,000 gallon pressure vessel and Enclosure containing two Test Loops and two Test Specimens.

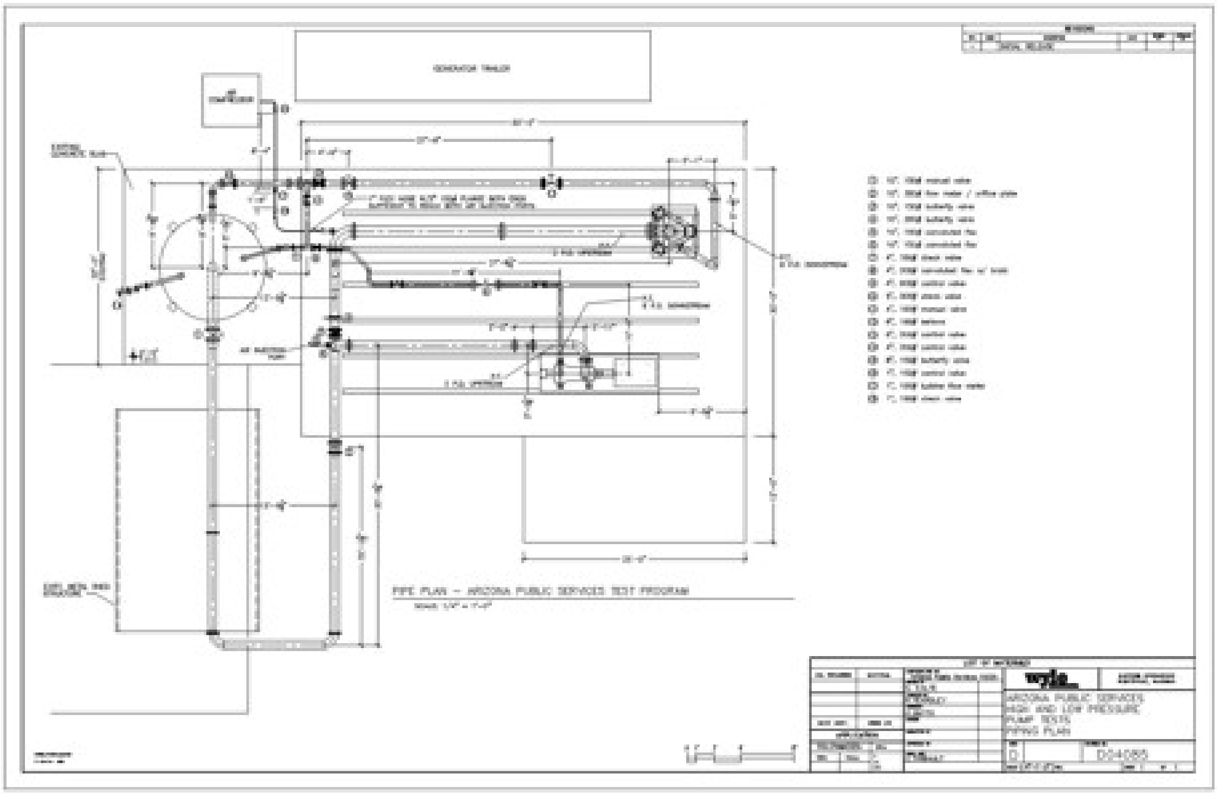


Figure 4 – General Arrangement Drawing for the Flow Facility

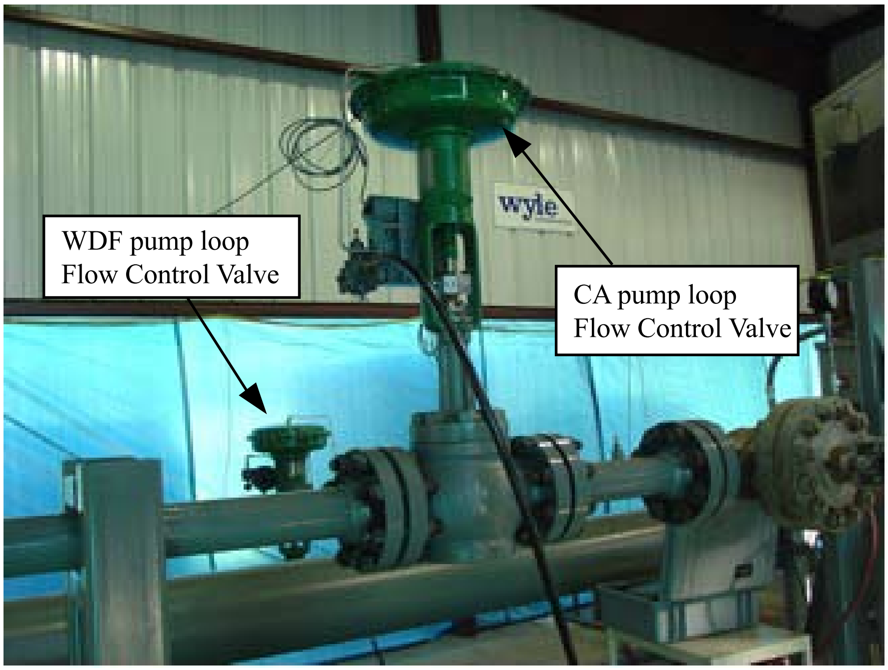


Figure 5 – Photograph showing flow control valves in CA and WDF pump test loops

Figure 6 – Photograph illustrating pipe insert that attaches to suction line for air injection into either CA or WDF pump suction line.

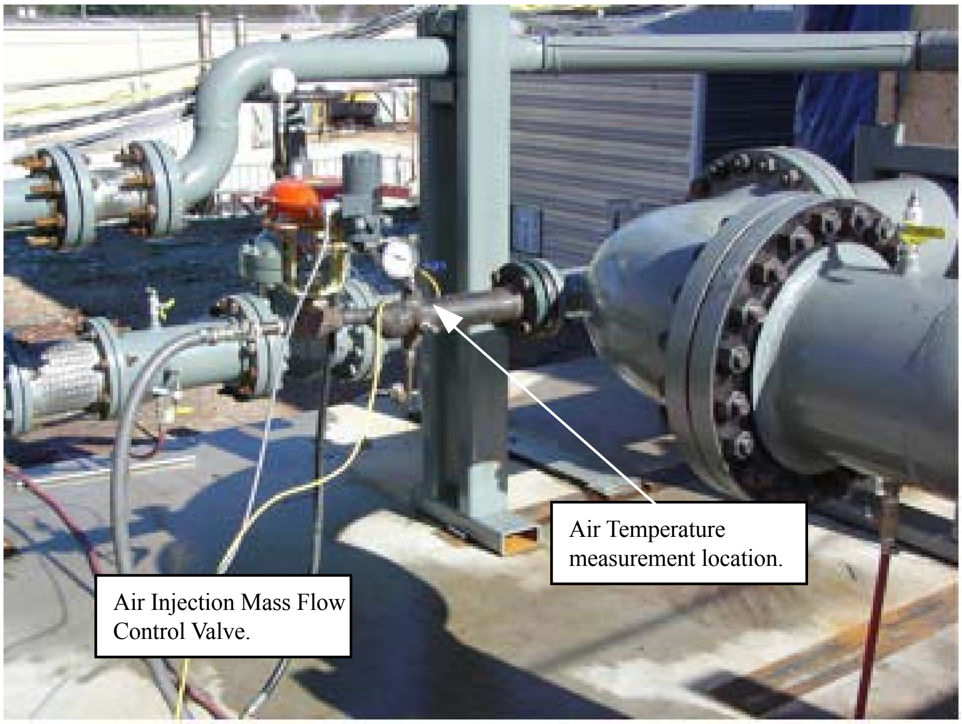


Figure 7 – Photograph illustrating Air Injection System inserted into WDF suction line.

Figure 8 – Photograph illustrating Sight Glass in suction line for CA pump.

Figure 9

CA and WDF Pump Test Matrix

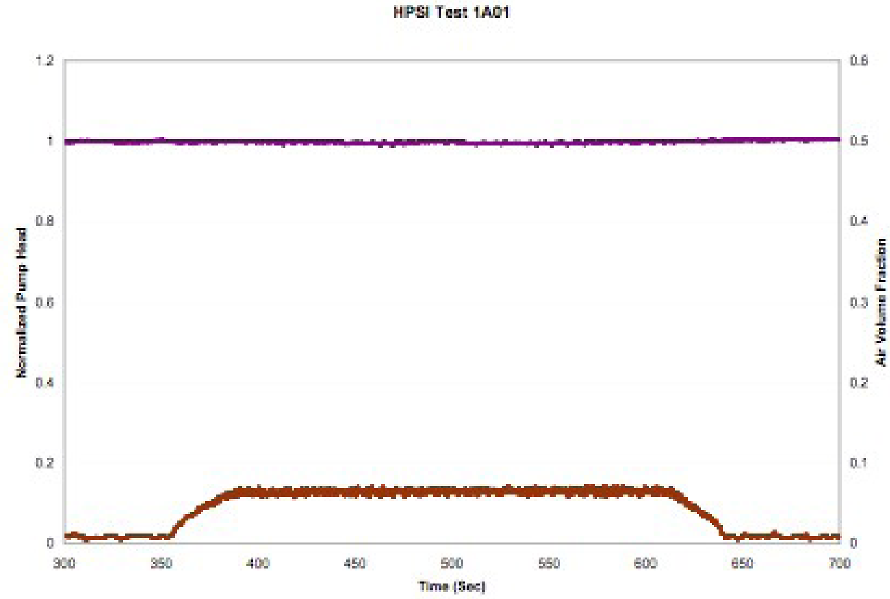
CA Pump Test Matrix

Case Number	Test Date	Data File	Initial Conditions		Void Conditions			
			Pump Flow rate (gpm)	Inlet Pressure (psig)	Starting Air Mass Flow (kg/s)	Target Air Volume (seconds)	Air Injection Duration (seconds)	Maximum Air Injection Ramp Rate (kg/s^2)
1A	12/13	HPSITEST1A01	900	22	0.011	0.05	200	0.083 max
1B	12/13	HPSITEST1B01	900	22	0.022	0.10	100	
1C	12/14	HPSITEST1C01	900	22	0.030	0.15	60	
1D	12/14	HPSITEST1D01	900	22	0.037	0.20	60	
1Drerun	12/14	HPSITEST1D03	900	22	0.037	0.20	60	
2A	12/15	HPSITEST2A01	600	22	0.009	0.05	204	0.083 max
2B	12/15	HPSITEST2B01	600	22	0.018	0.10	102	
2C	12/15	HPSITEST2C01	600	22	0.024	0.15	76	
2D	12/15	HPSITEST2D01	600	22	0.029	0.20	63	
2E	12/15	HPSITEST2E01	600	22	0.037	0.25	49	
3A	12/13	HPSITEST3A01	1300	22	0.018	0.05	164	0.083 max
3B	12/14	HPSITEST3B01	1300	22	0.036	0.10	72	
3C	12/14	HPSITEST3C01	1300	22	0.045	0.15	60	
-	12/15	HPSIPOSTTEST01	all	22	0	0	N/A	N/A

WDF Pump Test Matrix

Case Number	Test Date	Data File	Initial Conditions		Void Conditions			
			Pump Flow rate (gpm)	Inlet Pressure (psig)	Starting Air Mass Flow (kg/s)	Target Air Volume (seconds)	Air Injection Duration (seconds)	Maximum Air Injection Ramp Rate (kg/s^2)
4A	12/16	TEST4A01	4885	22	0.028	0.03	180	
4B	12/16	TEST4B01	4885	22	0.056	0.06	180	

Figure 10 a & b – HPSI (CA) Pump Performance



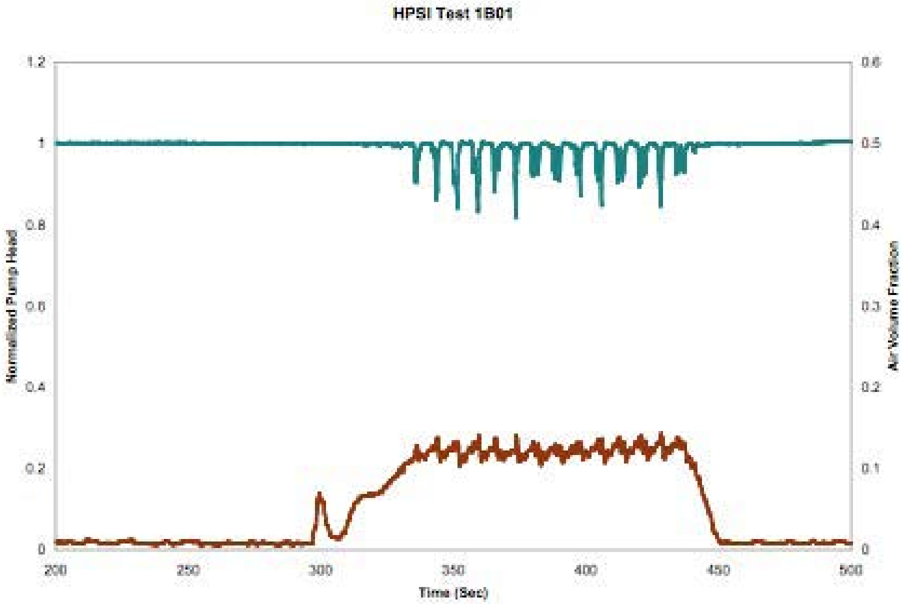
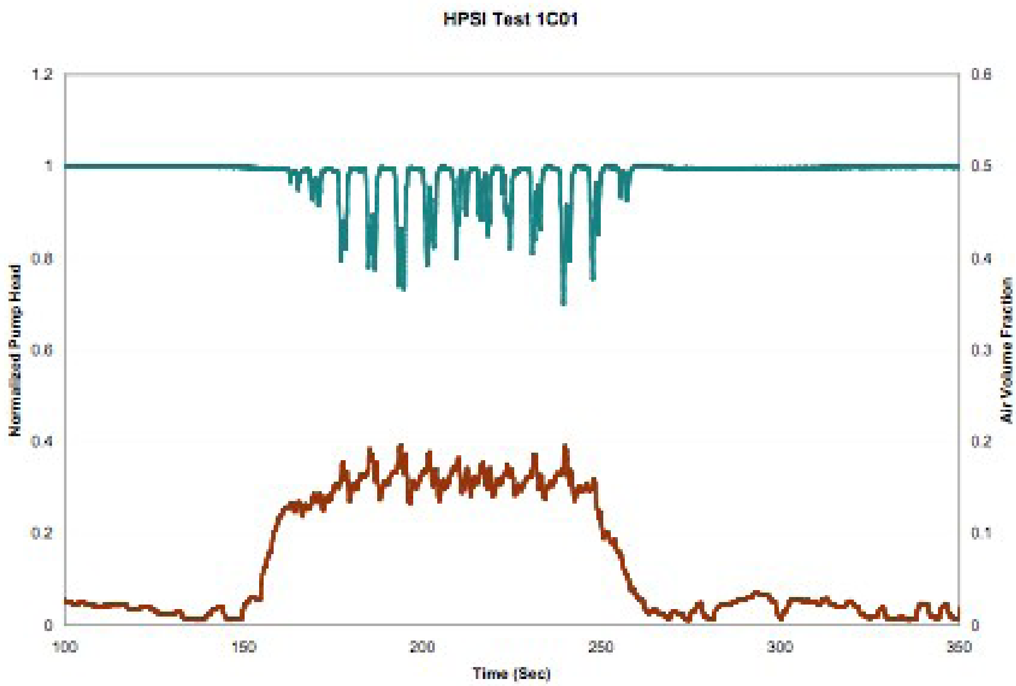


Figure 10 c & d – HPSI (CA) Pump Performance



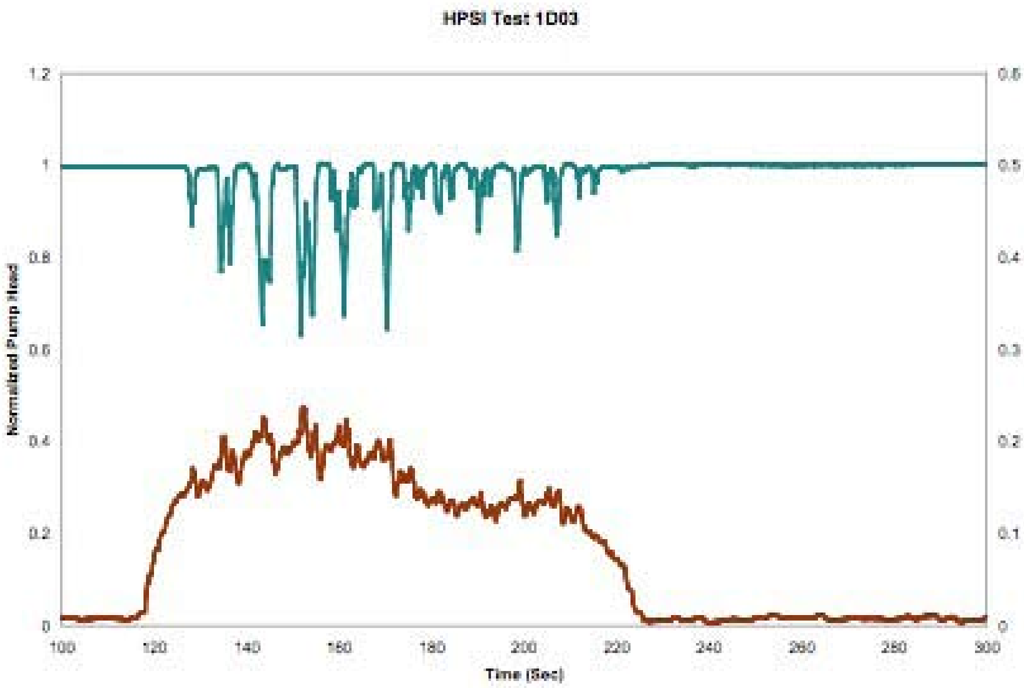
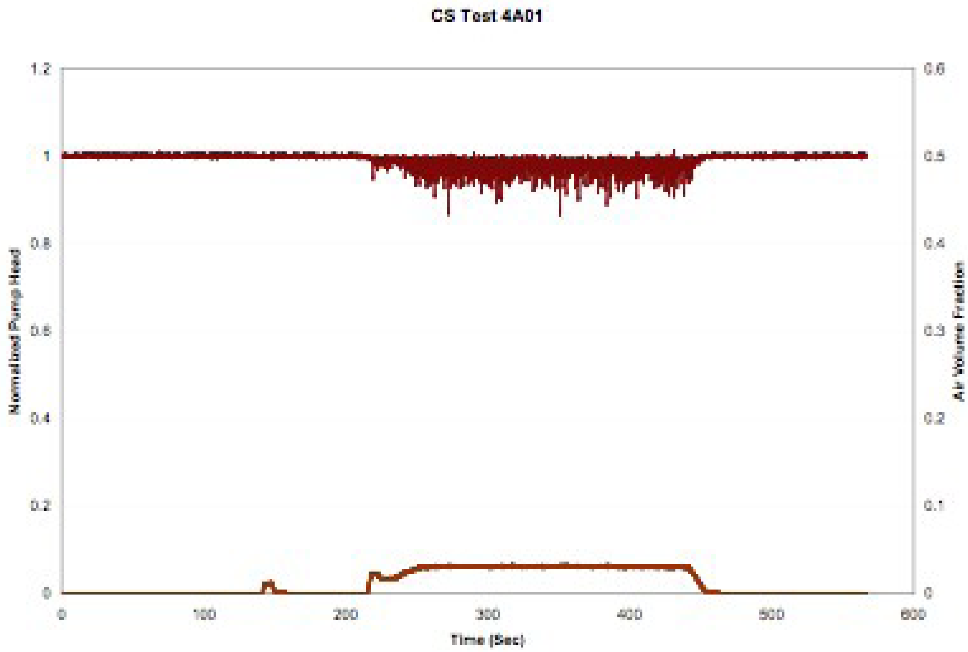


Figure 11 – CS (WDF) Pump Performance



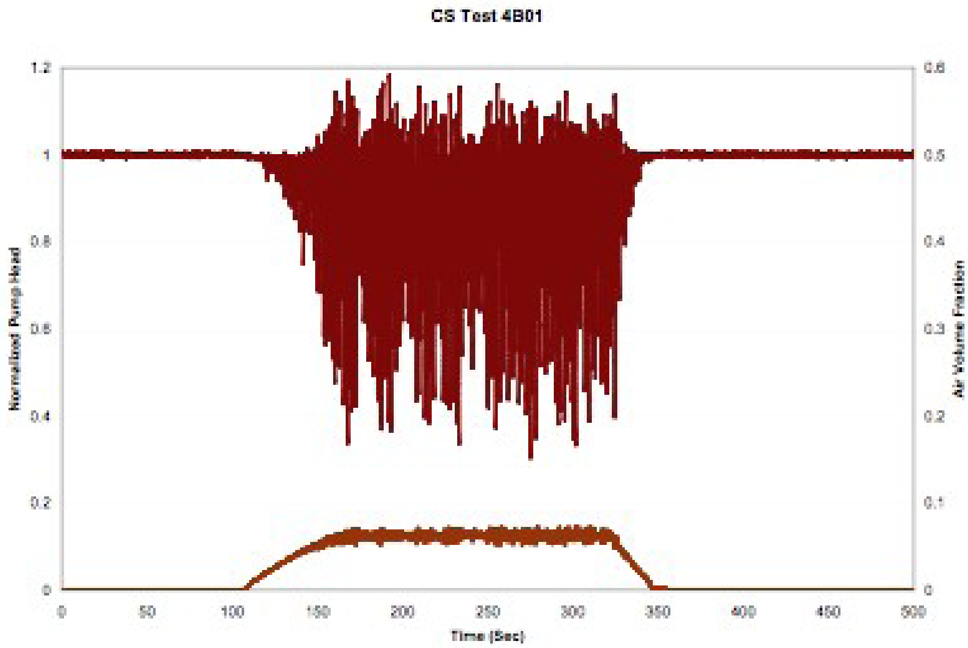
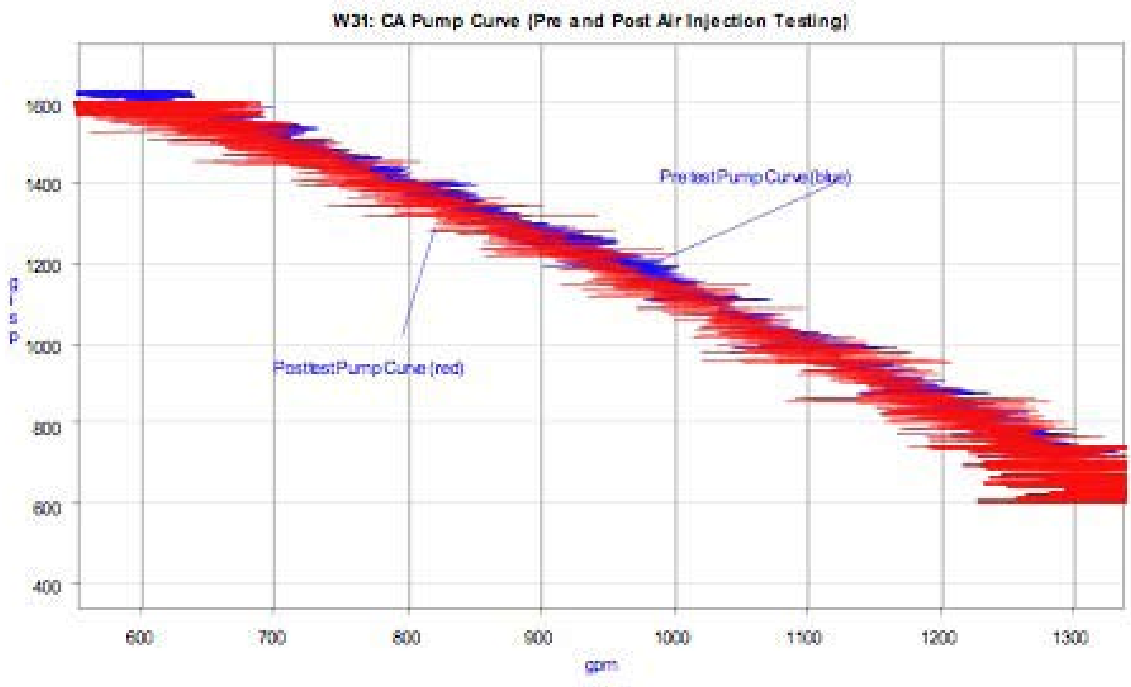


Figure 12 – HPSI (CA) Pump Curve



Proper Pump-to- Piping Procedure – 10 steps

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It should be realized that piping issues directly affect the pump's life and its performance. Bringing the pump to the pipe in one operation and expecting a good pump flange or vessel fit is a very difficult, if not impossible, task. When bringing the pipe to the pump the last spool (suction side and discharge side, each) should always be left until the pump has been leveled in placed and rough aligned. The final alignment will be a "free bolt condition", and, as may sound like a surprise to some, no "come-alongs" would be needed. As an ultimate investment in common sense and proper attention to details, - your pumps will last longer, with fewer failures of seals, shafts, bearings and couplings. More equipment uptime, and less lost production, will result in significant savings in dollars, and fewer headaches.

Step 1

At this point the pipe should be securely anchored just before the last spool, to prevent future growth towards the pump's flanges.

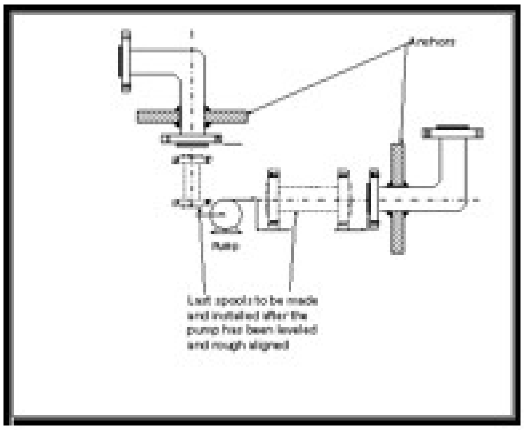

Figure 1 Typical anchors for the pump piping

The final piping lay out should not be finalized until certified elevation drawings are received from the engineering group or from the pump vendor. Once the final certified prints are received the final isometrics can be completed and the piping takeoff can be done.

The delivery of the equipment can either be early or it can be late in arriving at the site. When the equipment is late it is critical to have certified elevation prints of the equipment. The certified prints that the isometrics required for the piping takeoffs can be made without impacting the construction schedule. If the equipment is early, it will arrive at the site prior to the construction team needing it for installation. In such cases, early preparations must be made for long term storage. It is customary to use oil mist lubrication to keep the equipment in as-shipped conditions during the storage. The pressurization of the bearing housing with a small pressure (even 1 pound per square inch [psi] over atmosphere would do) prevents moisture and contaminants from entering the sealed areas and damaging the components. The early delivery of equipment to the site has the advantage of allowing for verification of the actual measurements.

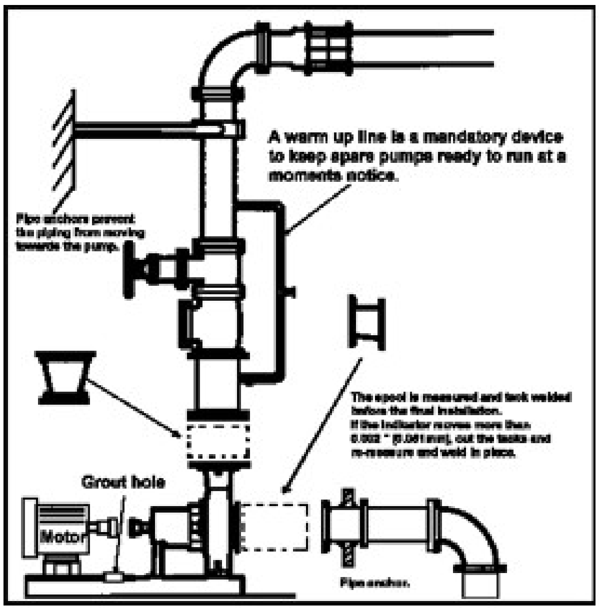

Figure 2 Rough alignment phase (note that the motor and the pump are not coupled yet and the baseplate is still sitting free, not grouted)

Step 2

Once the location of the equipment is set, the baseplate can be put in place, leveled and rough-aligned, with the equipment mounted. Rough alignment of the equipment should be done prior to building the grout forms.

Step 3

Once you are satisfied with the rough alignment, remove all the equipment (pump, motor gearbox, etc.) from the baseplate. Level the baseplate to maximum out of level of 0.025” (0.06 mm) from end to end in two planes. Use machined pads as the base for the leveling instruments. Inspect the foundation for cleanliness and, if not clean, use solvent to remove grease and oil.

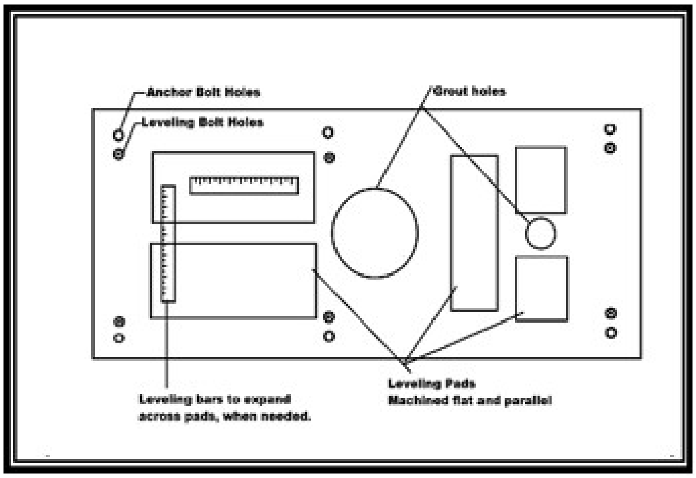

Figure 3 Baseplate leveling pads and grout location

Step 4

Allow time for the cleaning substances to evaporate. Form the base using the appropriate techniques to allow for the weight, temperature rise, and fluidity of the grout material. Grout the base using epoxy grout. Allow the grout to cure, following the grout manufacturer’s recommendations. This normally requires 24 hrs at 80° F (27°C). Remove the forms and clean all sharp residue and edges from the foundation.

Step 5

The rough alignment step, which we mentioned above, is critical to minimize the changes that will be required to appropriately fit the piping to the pump. At the last stage, when the final spools are installed, the final alignment will be achieved with small adjustments. This will minimize the adjustments required on the motor feet/bolts. Unfortunately (motor manufacturer’s take heed!), motor hold-down bolts are often too tight and allow only for small adjustments to the motor before becoming bolt bound. Motor manufacturers could improve this situation significantly if motor feet were slotted, by design, rather than drilled for bolts. Figure 5 shows the tightness of space available to insert the foot hold-down bolt.

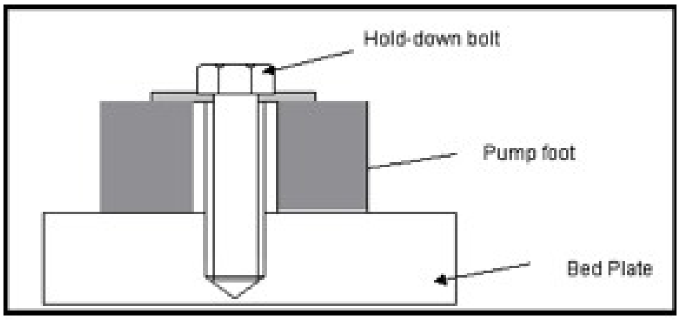

Figure 5 Potential bolt-bound situation due to tight clearances between bolt, feet and base

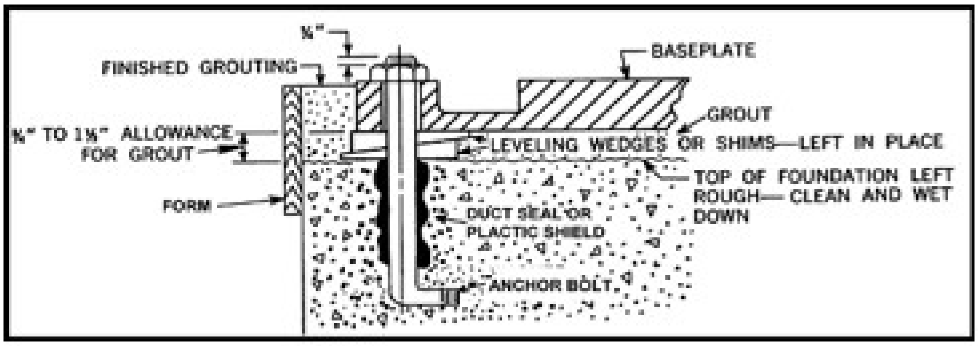

Figure 4 Typical anchor bolt and leveling wedges

This illustrates once again why good alignment at step 3 can save time and the cost of having to alter motor feet (a nightmare) by slotting or reaming.

Step 6

Reinstall the pump and the motor on the baseplate. Rough align the equipment again, using reverse indicator or laser alignment or similar accurate techniques.

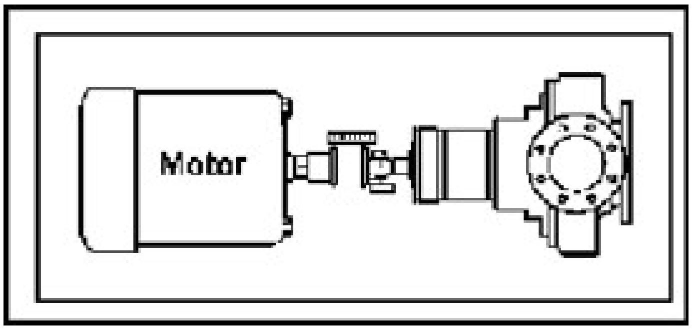

Figure 6 Rough alignment after grouting

It should be now easy to fine-tune the motor movement within the allowable alignment target without becoming bolt bound. This is possible because the rough alignment during the prior step (Step 4) was completed. Note: Never install shims under the pump feet. If the shims are lost or misplaced then alteration to the piping may be required to get the pump within the required alignment specification. The normal procedure is to place 0.125” (3.2 mm) thick shims under the motor feet. This allows for adjustments that will be required during final alignment.

Step 7

Make up the final spool pieces for the suction and discharge spaces. Bring the piping to the pump now.

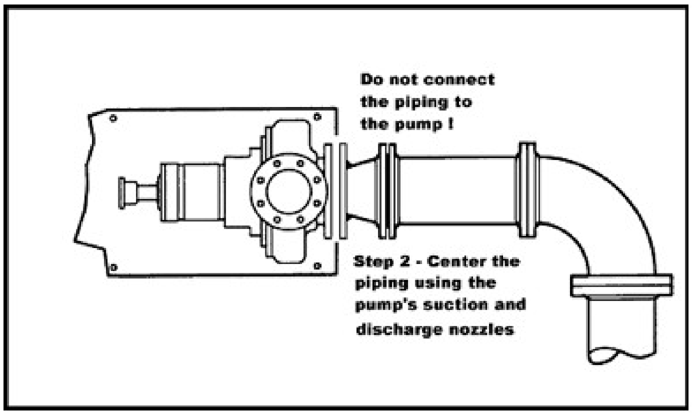

Figure 7 Illustration of the final connection of the suction piping.

Step 8

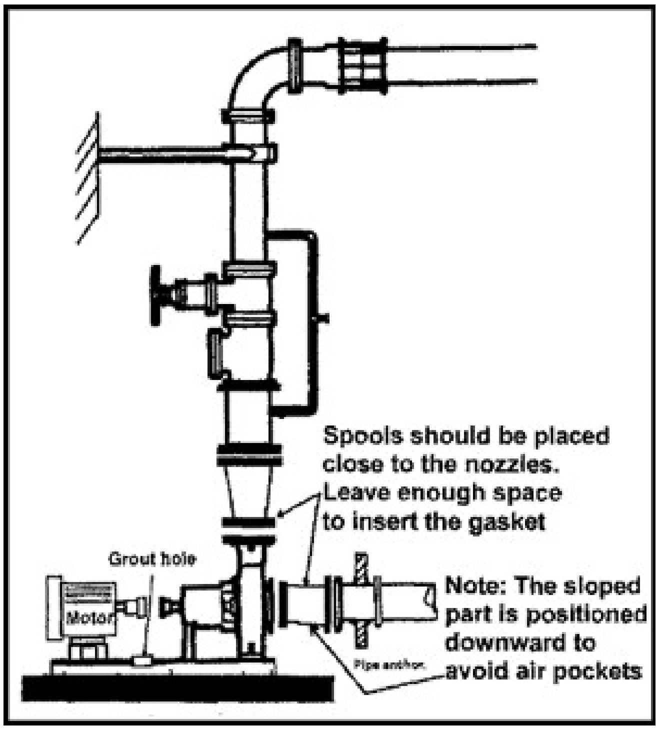

Figure 8 Final piping

As a final alignment step, bring the piping to the equipment; take final measurements, tack weld the spools in place. At this time the spools can be removed and taken back to the hot work permit area to finalize the weld. Leave a square and parallel gap between the flange faces. The gap should be wide enough to accommodate the size of the gasket required, plus 1/16 - 1/8", depending on piping sizing. (This is the only distance over which the piping will be pulled. However, because it is properly anchored before the spool pieces, this length is short, and stresses are minimized). Final align the equipment, taking into account hot and cold operating conditions, using two indicators on the pump shaft coupling area.

Step 9

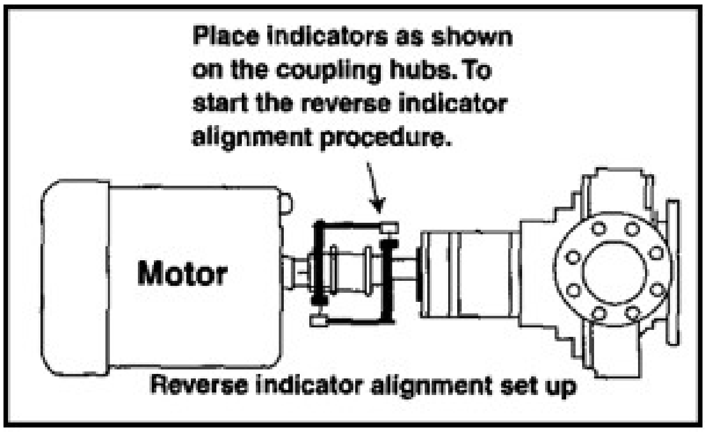

Figure 9 Overhead view of the motor and pump

As the piping is tightened into place, the shaft shall not be moved more than 0.002" (0.005 mm), otherwise modify the spool pieces until the piping misalignment is fixed.

Several clues are common to piping misalignment. These clues come via the way of mechanical seal and/or bearings running hot, and failures. A quick analysis of the failed parts can clearly show the evidence of piping misalignment. To make a final confirmation of the symptoms, unbolt the piping while measuring the movement in the vertical and horizontal plane. Again, the piping that moves more than 0.002" (0.005 mm) must be modified to correct the situation.

Step 10

Place an indicator in horizontal and vertical planes, using the motor and pump coupling.

Uncouple the pump and motor, while watching the indicator movement. Start unbolting the flanges, and continue watching for movement in the indicators. If the needle jumps over 0.002" (0.005 mm) the piping has to be modified to improve the pump's performance.

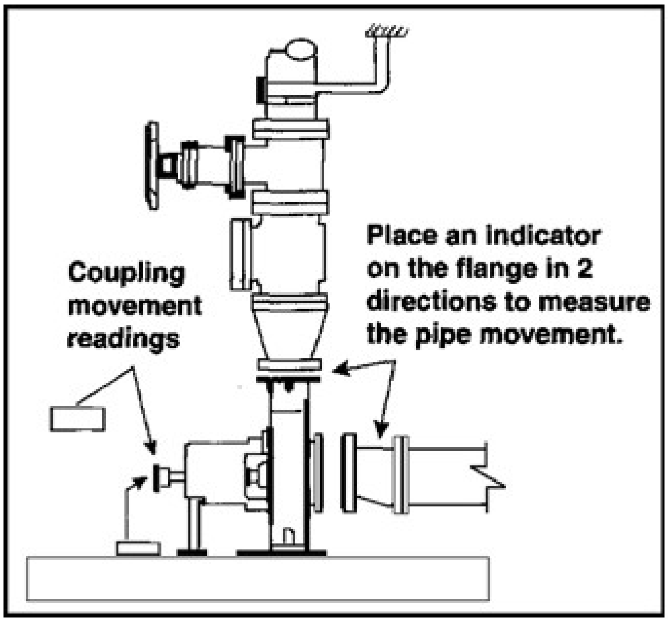

Figure 10 Piping alignment verification

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Dr. Nelik has 25 years experience with pumps and pumping equipment. He is a Registered Professional Engineer, and has published over fifty documents on pumps and related equipment worldwide. He is a President of Pumping Machinery, LLC company, specializing in pump consulting, training, and equipment troubleshooting. His experience in engineering, manufacturing, sales, field and management includes: Ingersoll-Rand, Goulds Pumps, Roper Pump and Liquiflo Equipment. He teaches pump training courses in the US and worldwide, and consults on pumps operations, engineering aspects of centrifugal and positive displacement pumps, maintenance methods to improve reliability, improve energy savings, and optimize pump-to-system operation.

Questions and feedback are appreciated and can be forwarded to him at www.PumpingMachinery.com

Session 1(b): Valves I

Session Chair

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The Case for a Kinetic Energy Criterion In Control Valves - Part 3

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Abstract

This paper discusses 470 installed control valves that failed and were made successful by only changing the energy level in the fluid jets exiting the valve trim. Controlling the fluid jet energy is accomplished by retrofitting the original trim with a trim that controls the jet exit velocity. The retrofit of the valve trim was made because the valves were not performing their control function. The statistics associated with the retrofit of these valves are presented. The statistics include the causes of valve failure, trim jet energy levels, and a regression analysis of the energy data before and after trim replacement. Analytical expressions are provided to estimate the kinetic energy expected from a valve supplier that has not used the guideline kinetic energy criterion. The criterion used in designing the retrofit valve trim was to limit the kinetic energy density of the fluid jet exiting the trim to 480 kilopascals (kPa) [70 pounds per square inch (psi)] or less. Without this criterion being imposed, valves are supplied that have fluid jet kinetic energies up to 50 times this level and on average exceed it by a factor of 20. These high fluid jet energy levels are the root cause of the valve's failure to perform to control expectation. Failures include physical damage to the valves causing excessive maintenance as well as piping system vibration and noise.

Introduction

This paper continues the discussion of the very positive experience gained in retrofitting 470 control valves throughout the world. Table 1 shows a breakdown of the number of valves and designs in the retrofit database.

A "retrofit" is a replacement of a valve's flow control trim. The flow control trim is most often the valve's cage or the plug and seat. Every effort is made to use the original valve body without alteration. The retrofit trim can be installed without removing the valve from the pipe line. Occasionally, a valve body is found to be damaged and repair is necessary prior to the retrofit.

The decision to retrofit an installed control valve is quite significant and a user will need a strong motivation in order to take this step. The problems and causes are presented that have pushed the user to implement a retrofit of a valve. These causes were discussed in more detail in Part 1, Reference 1. To summarize, the most frequent problems included poor control, cavitation, noise, vibration, erosion, excessive maintenance and stem breakage. Reference 2 looks at the retrofit data base by separating the data into the general categories of liquid and gas applications. In general, the gas applications had slightly lower fluid jet kinetic energy levels for the original valve designs; however, still not within the 480 kPa [70 psi] guidelines.

A number of retrofitted valves have been discussed in the literature before (see References 3-5). These referenced cases are those in which definitive measurements of vibration were made on the control valve before and after the retrofit. The magnitude of the improvement could be determined by comparing similar measured results. The “before and after” measurements provide strong support for a kinetic energy criterion for the fluid exiting the valve trim.

The database for this study includes those few cases in which measurements were made as well as more than 450 other cases in which only anecdotal feedback is available. In none of the cases has the feedback been negative regarding the performance of the retrofit. The population of valves covers the entire industrial control valve application base when viewed from the range of valve inlet pressures.

Statistical comparisons are made of the fluid kinetic energy exiting the trim for the valve design prior to retrofitting and after the trim are replaced. The comparison is made for the entire population of designs and then the liquid and gas applications are separated for further review. The valve trims range from top guided, cage guided, all forms of drilled hole configurations as either single cylinder to as many as 7 concentric cylinders, axial staged (multi-stage, single-path) to other forms of multi-stage, multi-path configurations. Valve outlet sizes ranged from one to 36 inches (25 to 900 millimeters [mm]).

What is the Kinetic Energy (Density) Design Criterion?

The kinetic energy density combines the influence of fluid density with velocity of the jet exiting a valve trim. The term “density” is used to qualify the kinetic energy because the energy level is per unit volume. The term “density” is intended throughout this paper whenever kinetic energy is used. Kinetic energy density is defined as follows:

$$KE = \frac{\rho V^2}{2M}$$

Where: M = 1000 for metric or 4636.8 for imperial units

KE is in kPa or psi

V is in meters per second (m/s) or feet per second (ft/s)

p is in kilograms per cubic meter (kg/m^3) or pounds per cubic feet (lb/ft^3)

The kinetic energy density criterion was first introduced in 1997 at the “Summer” meeting of the Fluids Engineering Division of the American Society of Mechanical Engineers (ASME). The criteria are summarized in the *ISA Control Valves –Practical Guides for Measurement and Control*, Reference 6.

Kinetic energy density is expressed using the same units as pressure and is sometimes called the dynamic pressure. The velocity in this expression is the average trim outlet jet fluid velocity and the density of the fluid at the exit of the trim. For most applications, the kinetic energy criterion is 480 kPa [70 psi] or lower.

The application of the kinetic energy criterion is an addition to the traditional control valve design process. That is, all decisions are made regarding materials, capacity sizing, body- trim- and actuator- selection and adjustments made for erosion, cavitation and/or noise. Then a check of the trim design is made to be sure the kinetic energy level meets the design criterion. If the energy level is too high, then additional flow resistance through the trim is used to provide an additional reduction in the fluid jet velocity. The increase in resistance for the retrofitted valves is achieved by adding more stages of pressure drop. For all of the valves in the database, the energy control trim included a tortuous path, multi-stage, multi-path trim. An example of this type of trim is shown in Figure 1. Each right angle turn of the Figure 1 trim causes additional pressure drop and reduced channel fluid velocity. Capacity requirements are met by assuring enough passages are available to meet the flow needs.

Kinetic Energy Level, Before and After Retrofit

Figure 2 presents the fluid exit kinetic energy of the original valve trim and the energy control trim ranked from the highest energy levels to the lowest. This figure shows the dramatic reduction in the kinetic energy from the failed original trim to a level that assures a good control valve application. The average reduction in kinetic energy was from 3.3 MPa [480 psi] to 300 kPa [44 psi]. The maximum kinetic energy was 22.6 MPa [3280 psi], which is almost 50 times the recommended criterion. Overall, the average kinetic energy ratio between the original trim and the energy control trim is 21. With these high jet kinetic energies, it is not surprising that a lot of damage was taking place in the control valve and associated piping prior to retrofitting. Figure 2 includes both liquid and gas applications, and the deviations from 480 kPa [70 psi] are discussed in Reference 1. In general, when the energy for the Energy Control Trim exceeded the criterion, the original valve body did not have enough space to allow packaging more pressure drop stages. A judgment to proceed with the retrofit was then made based on the magnitude of the reduction in energy level and a consideration of the specific valve application. For the cases in which the energy levels for the original trim designs were less than the criterion, there were usually other flow conditions that governed the need to retrofit the trim.

Figures 3 and 4 present the flow cases for the liquid and gas valves, respectively. In these figures, the Energy Control Trim values have not been ranked but have been shown superimposed on top of the energy level for the original trim. This direct comparison shows the magnitude of reduction in trim energy level that has taken place in the retrofitted valves. The kinetic energy deviations for the energy control trim from the criterion are discussed in Reference 2. General statistical measures for Figures 2 through 4 are listed in Table 2.

For the liquid cases, the average kinetic energy ratio of the original trim to the energy control trim was 27. This is a significant difference in fluid energy levels between the original valve trim and the energy control trim. These kinetic energy levels, acting over fairly small areas, can cause significant forces with the dense liquids and one can see why valves and pipelines will vibrate unless these high levels of energy are significantly reduced.

For the gas flow cases the average kinetic energy ratio of the original trim to the energy control trim was 8.7. The energy levels for the original and energy control trim gas cases are much closer together but still significantly different. The lower energy level for the gas valves is attributed to generally imposed requirements for noise control for these valves. In many applications, the noise control requirement will cause the kinetic energy criterion to be met without additional reduction in the fluid jet velocity exiting the valve trim. However, many times the noise levels to an outside observer are met because of the thick pipe walls and insulation encountered in industrial processes. Internal energy levels can therefore be quite high leading to unbalanced forces and vibration of the valve parts and piping systems. These high energy levels then can lead to excessive maintenance and/or erratic control.

As demonstrated by the need to retrofit all of these gas valves, considering only the noise requirements is not enough to assure a good control valve application. The additional design criterion of jet energy must be imposed even when noise control is an installation requirement.

The impact of the high kinetic energy density for the original valve designs is illustrated by the causes and complaints associated with these valves before retrofitting the trim. The reasons for the retrofits are presented in Tables 3 and 4. The number of valves in these tables is higher than the total valves retrofit because the users provided multiple causes regarding the motivation for the retrofit. The impact of high fluid energy levels is apparent in these two tables. The tables also suggest that the valve supply industry and users are doing a much better job in controlling cavitation for liquid valves than is done for noise associated with gas valves. Vibration is fairly dominate in both cases and is the likely cause of the stem breakage for both fluid categories. A significant argument against using a kinetic energy control criterion for gas valves is that the noise requirements imposed for them is sufficient. This cannot be concluded from Table 4 as not only is noise the dominant cause but vibration associated with these lower density applications is a second most frequent complaint. Vibration for the liquid valve applications is also significant because of the apparent catastrophic failure brought on by the more frequent stem failures.

Regression Analysis

A quick method of estimating the kinetic energy density for an application is offered by regression modeling of the large database for the original trim and the energy control trim. The database contains several variables. Statistical methods of variable selection were used to select the most appropriate and statistically significant variables in estimation of the kinetic energy. Some variables were transformed, using the Box-Cox transformation, to achieve a sound linear relationship between the response and the predictor variables in the linear regression model. The following is the linear regression model that was constructed for the Original Trim:

Original Trim

$$KE_O = e^{Fn_O} \quad (1)$$

Where :

$$Fn_O = 1.0817 - 0.0181 Size + 0.000124 P_2 + 0.811(\ln \Delta P)^* A + 0.6985(\ln \Delta P)^* B + 0.5982(\ln \Delta P)^* C + 0.524(\ln \Delta P)^* D$$

where e is the natural logarithm base, ln represents the natural logarithm, KEO is the kinetic energy density in psi, Size is the nominal valve outlet size in inches, ΔP is the valve pressure drop in psi, P_2 is outlet pressure in psi, and the variables

$$\ln(\Delta P)^* A, \ \ln(\Delta P)^* B, \ \ln(\Delta P)^* C, \text{ and } \ln(\Delta P)^* D$$

denote the interactions between $\ln \Delta P$ and each of the trim types A, B, C, and D. These trim types are described in Table 5. For example, if the original trim is a top guided valve, Trim A, only the term $0.811(\ln \Delta P)^* A$ would be considered and the last three terms in the equation would be zeroed (i.e., the contribution of the last four terms to the log of kinetic energy would only be $0.811(\ln \Delta P)$). Thus **A, B, C,** and **D** have a value of zero or one depending upon the trim type being considered. To convert units, use the equivalence of 1 psi = 6.895 kPa.

Equation 2 gives the linear regression model for estimating the kinetic energy for the energy control trim.

Energy Control Trim

$$KE_{ECT} = e^{Fn_{ECT}} \quad (2)$$

Where :

$$Fn_{ECT} = 1.81 + 0.2782(\ln \Delta P) - 0.1559(\ln \rho_2) + 0.0002613(\rho_2)$$

In this model, KE_{ECT} denotes the kinetic energy density for the Energy Control Trim in psi, ρ_2 is the outlet fluid density in lbs/ft³ (1 lb/ft³ = 0.624 gm/l). Note that using the inlet density for liquids and adjusting gas inlet density for pressure difference is sufficient for this calculation.

To gauge the variability of the kinetic energy estimates obtained from Equations 1 and 2, we have calculated confidence bands for the estimates. Because these confidence bands have mathematically complex multidimensional expressions, we do not report them here. However, a reasonable estimate of the 90 percent confidence bands for the original and energy control trims can be obtained by $KE_O \pm 21\%$ and $KE_{ECT} \pm 28\%$, respectively where, as mentioned earlier, KE_O and KE_{ECT} are obtained from Equations (1) and (2). Even though the Energy Control Trim has a higher percentage spread in the confidence band it is applied to the much smaller value of kinetic energy associated with that type of trim.

The above equations allow the user to quickly obtain an estimate of the expected kinetic energy for a valve proposed by a supplier who is not using the kinetic energy criterion in their design process. Figure 5 shows the relative magnitude of the estimates from Equations 1 and 2 for a set of pre-specified values of the variables. Specifically, in Figure 5, the outlet pressure is held at 2760 kPa [400 psi], as the inlet pressure ranges from 3.4 to 13.8 MPa [500 to 2000 psi]. Figure 5 is plotted using the pressure difference. For the original trims, Equation 1, a valve size of 6 inches is selected for the illustration. Smaller valves will have larger estimated kinetic energies and doubling the valve size to 12 inches will decrease the estimated kinetic energy by a bit more than 10 percent. Density is only required for Equation 2, the Energy Control Trim, and for the illustration in Figure 5 a gas density of 0.0624 gm/l [1 lb/ft³] is selected. For heavier densities, Equation 2 will produce lower estimates of kinetic energy. With the estimates from the above equations, a judgment can be made as to whether additional action to reduce the energy level needs to take place to assure a low risk control valve application. When there is a need for a more accurate method of estimating the kinetic energy, the methods outlined in Reference 7 are quite reasonable. Reference 7 calculations require more detailed knowledge about the actual trim geometry being proposed.

The low kinetic energy results of Figure 5 are expanded in Figure 6 to illustrate more clearly the differences between the more tortuous flow path trims. Also a curve has been added for liquids with a density equal to 1 kg/l (62.4 lb/ft³) to show the difference expected between a gas and a liquid fluid. As noted the multipath, multistage trim types of Figure 1 meet the kinetic energy criterion over the entire range of pressure drops. Only the trim types C and D will meet the criterion for a pressure difference exceeding 690 kPa (100 psi); however their ability to meet the criterion diminishes quickly with increasing pressure drop. One is reminded that there is a 90 percent confidence band around the original trims of about 21 percent of the calculated value.

Conclusion

Figures 2 through 4 clearly show the benefits of controlling the fluid jet energy exiting the valve trim. Failing valves that had very high kinetic energy levels were made into designs that met the control needs of the application just by minimizing this energy to 480 kPa [70 psi].

A linear regression analysis of the large data base provided a means to roughly estimate the fluid jet energy exiting from the valve trim. A comparison of this estimate against the kinetic energy criterion of 480 kPa [70 psi] will allow a judgment of the risk associated with the valve application to be made. The estimated kinetic energy can also be calculated for valves in the field that are causing a lot of maintenance or are already considered to be problems to help diagnose the potential root cause.

Database Audit:

The authors welcome an audit of the retrofit database and the supporting information by any user or person responsible for the specification of control valves in process design. All information will be made available at the California company location for review. However in the interest of confidentiality for our customers and due to the proprietary nature of the information, we would specify that there be no copies made of the information.

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Table 1 - Retrofit Database

Total valves retrofitted	**470**	**%**
Number of liquid valves	329	70
Number of gas valves	141	30
Total Designs retrofitted	**140**	
Number of liquid designs	90	64
Number of gas designs	50	36

Table 2 – Trim Exit Kinetic Energy Statistics, kPa [psi]						
	All Flow Cases		Liquid Cases		Gas Cases	
	Original	Energy Control	Original	Energy Control	Original	Energy
Maximum	22600[3280]	2345[340]	22600[3280]	1380[200]	13700[1990]	1280[186]
Mean	3310 [480]	295 [43]	3530 [515]	220 [32]	2780 [403]	430 [63]
*Std. Dev.	3750 [545]	260 [38]	4160 [605]	185 [27]	2590 [375]	290 [42]
Ave. Ratio	21		27		8.7	

*More than 2/3 of the data fall within one standard deviation of each of the means.

Table 3 – Causes/Complaints, Liquid Valves				
	Compliants		Valves	
	#	%	#	%
Controllability	37	23.9	135	24.5
Erosion	30	19.4	107	19.4
Leakage	30	19.4	67	12.2
Vibration	18	11.6	64	11.6
Cavitation	22	14.2	53	9.6
Stem Break	6	3.9	41	7.4
Capacity	4	2.6	21	3.8
Noise	3	1.9	20	3.6
Bonnet leak	1	0.6	15	2.7
Maintenance	5	3.2	12	2.2
Other	5	3.2	11	2.0
Galling/Wear	4	2.6	5	0.9
Total	**155**		**551**	

Table 4 – Causes/Complaints, Liquid Valves				
	Compliants		Valves	
	#	%	#	%
Noise	25	25.0	71	26.0
Vibration	18	18.0	54	19.8
Leakage	10	10.0	44	16.1
Stick/gall/wear	7	7.0	34	12.5
Controllability	9	9.0	17	6.2
Erosion	11	11.0	14	5.1
Other	6	6.0	11	4.0
Stem Break	3	3.0	7	2.6
Hi Maintenance	2	2.0	7	2.6
Vendor support	1	1.0	6	2.2
Capacity	5	5.0	5	1.8
Stroke Speed	3	3.0	3	1.1
Total	**100**		**273**	

Table 5 – Description of Trim Types for Equation 1	
Trim Type	**Description of Trim**
A	Single large orifice such as for a top guided valve or a single cage with 2 or more large holes for flow control.
B	Drilled hole cages. The holes are small, usually much less than 1 inch (25 mm), Single and multiple concentric cages. Up to 7 concentric drilled hole cages are included in the database.
C	Axial multistage trim.
D	Other multistage trims with 4 or more pressure drop stages.

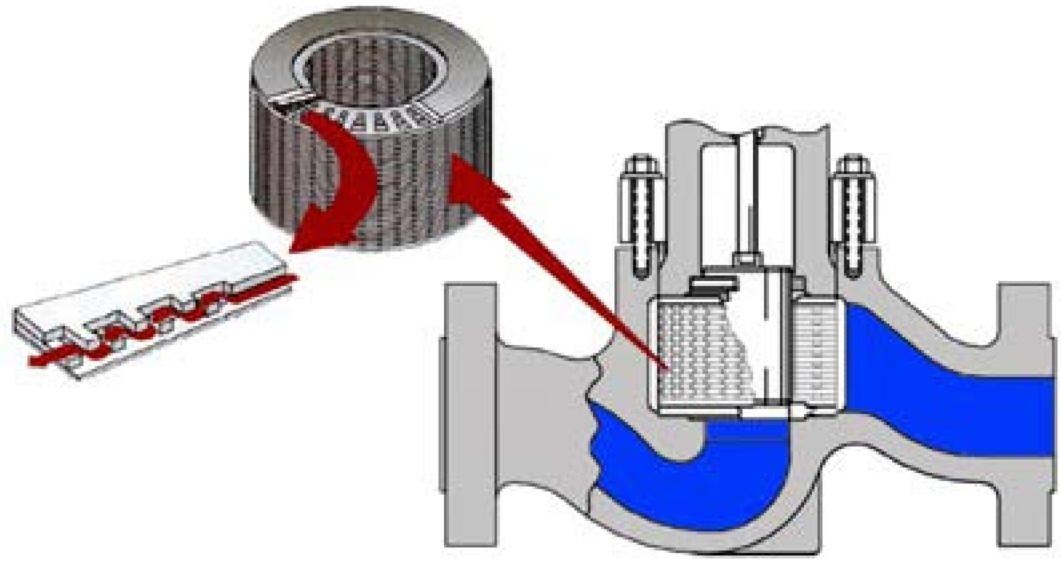

Figure 1 – Energy Control Trim for Retrofit

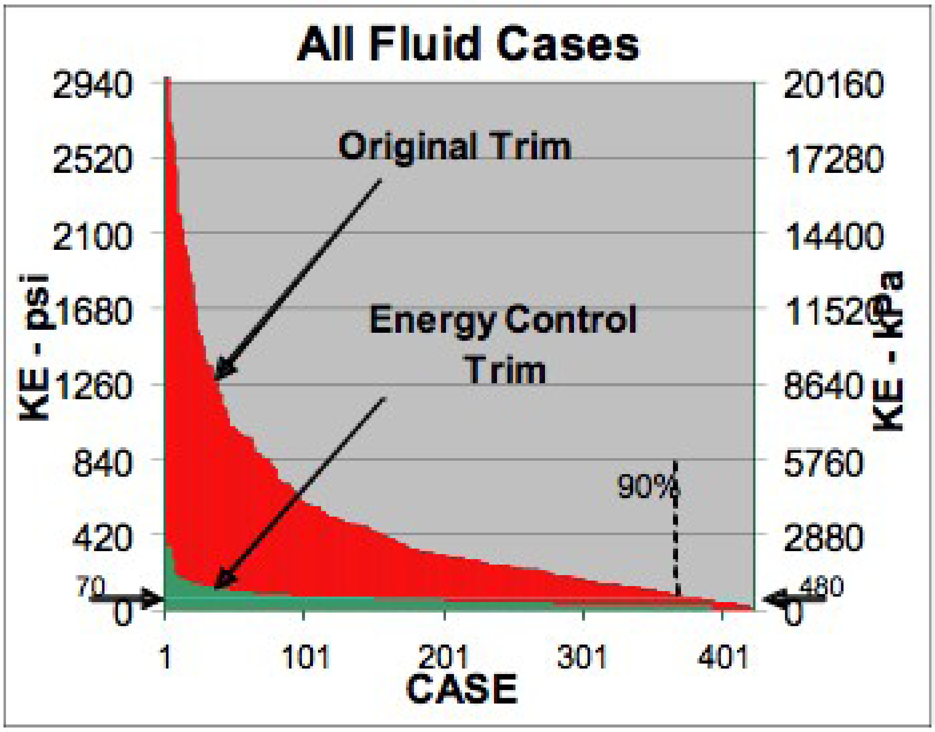

Figure 2 – Kinetic Energy Before and After Retrofit

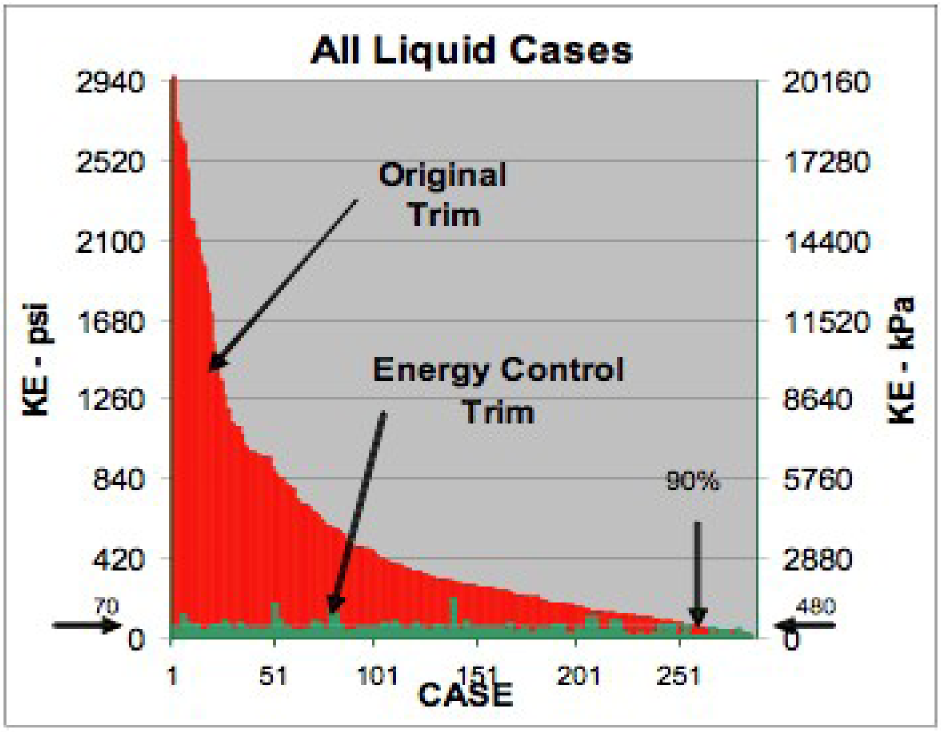

Figure 3 – Kinetic Energy Before and After for the Liquid Cases.

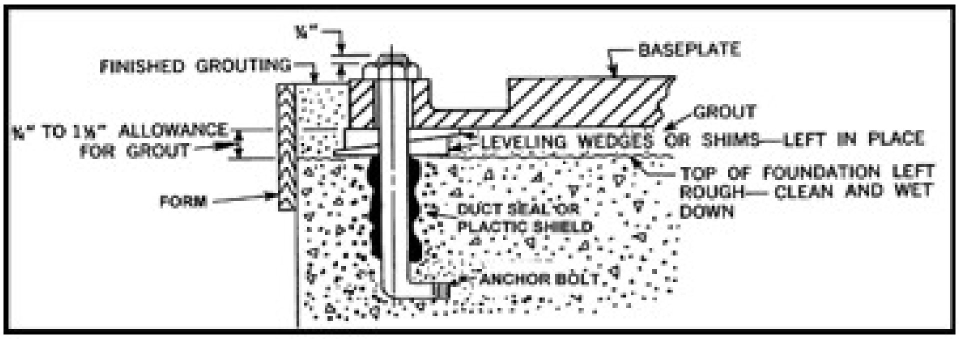

Figure 4 – Kinetic Energy Before and After for the Gas Cases.

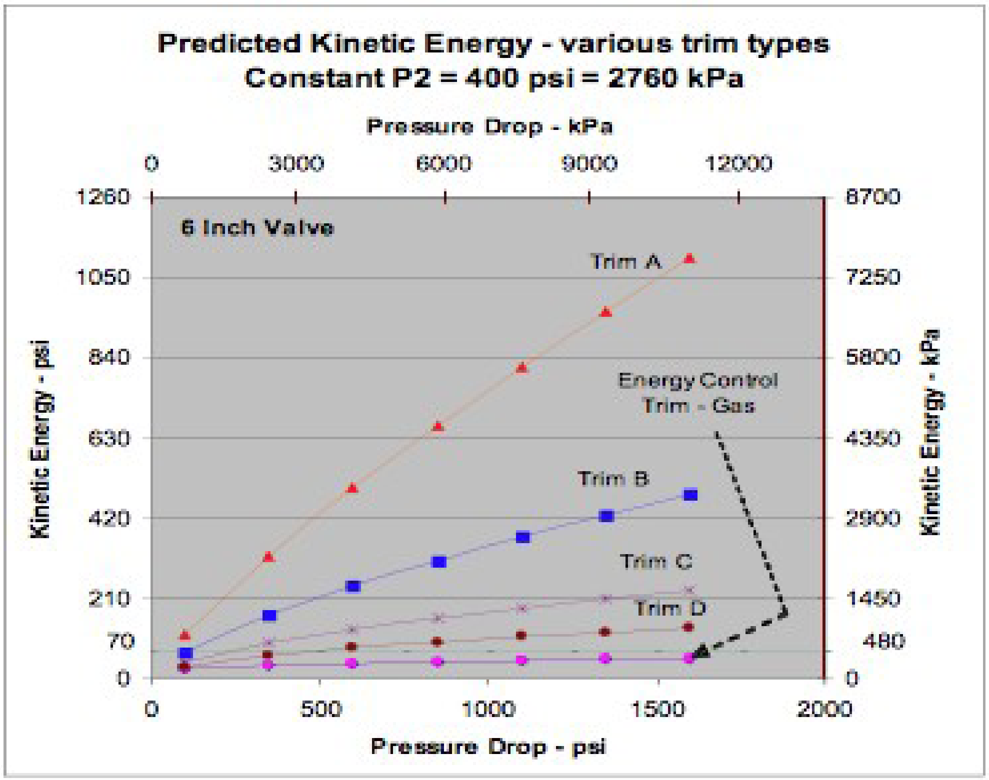

Figure 5 – Estimated Kinetic Energy from Equations 1 through 3.

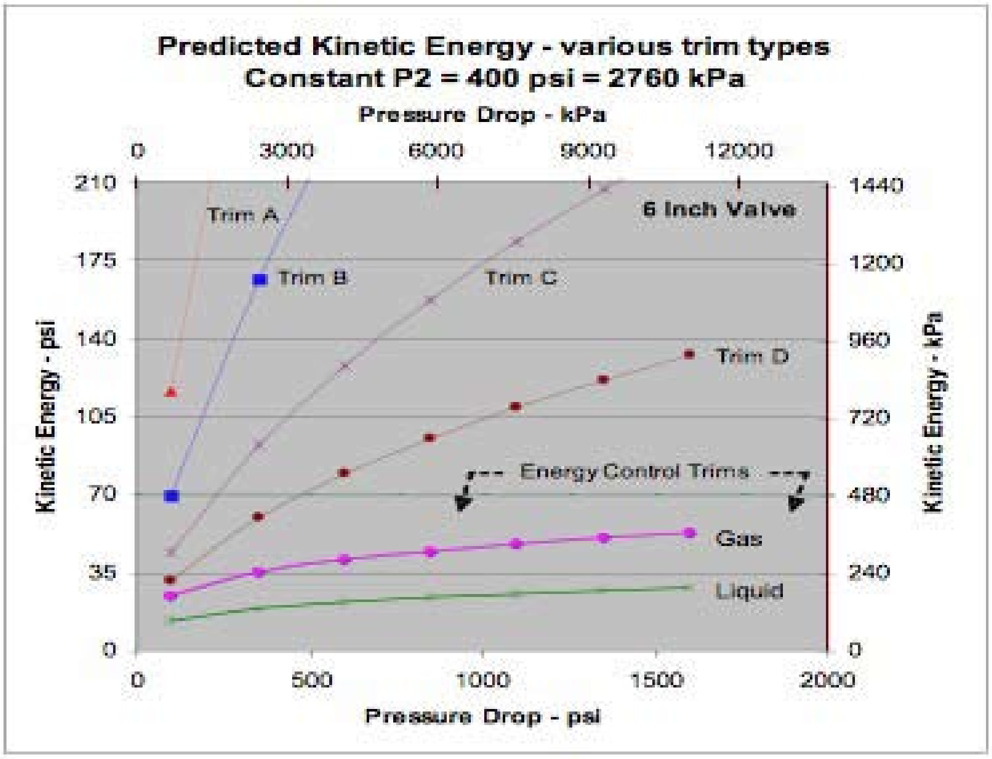

Figure 6 – Lower Portion of Figure 5 Expanded with a Liquid Curve Added.

Valve Leak Reduction Program

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Abstract

Valves are a major source of external leaks in power plants and the consequences of these leaks result in forced shutdowns, equipment degradation and contamination. Recent Operating Experience reports (OEs) and shutdowns from valve packing and gasket failures provide a snapshot of the problems that still exist in the industry. In some cases, these problems are getting worse as equipment ages and skilled mechanics leave the workforce.

This purpose of the paper is to provide the basics for proper selection, design and installation of new-age valve packing, gaskets and pressure seals. The matrix of a successful leak reduction program for valves will be defined as well.

Introduction

Effectively sealing valves is a challenge that every station faces. Valve manufacturers were not held to a stringent specification or standard when designing the stuffing box, gasket seal area, or pressure seal area. Most designs are still based on the use of asbestos packing/gasket materials or metal pressure seals. Some Original Equipment Manufacturers (OEMs) placed a strong emphasis on the design of the valve in relation to packing and gasket performance while most do not. Most stations do not rely on OEM supplied packing systems and more are moving away from OEM supplied gaskets and pressure seals due to performance and high cost.

The challenge is to define a program where the optimum packing system or gasket is used on the myriad of valves that exist in power plant. The answer lies in first making sure the basics are covered then getting into specific valve types and manufacturers.

Design Basics for Packing, Gaskets and Pressure Seals

There are three basic design rules for all sealing products:

1. Use sealing products that do not degrade under operating conditions
2. Adequately load the packing or gasket
3. Contain the packing or gasket system

Nearly every packing and gasket failure can be traced to one or more of these basics designs not being met. The body of the paper will focus on the proper selection and use of packing systems, gaskets and pressure seals. In addition, the basis of a program will be defined and the LeakManager and SmartSeal web based Leak Management and Repair program explained.

Developing a Leak Reduction Program

Valve packing, gaskets and pressure seals represent a large portion of identified leaks in plant and some of the most severe in regards to plant operation. Some plants have an overall leak reduction or monitoring program while others have programs for valve packing, air-operated valves (AOVs), motor-operated valves (MOVs) and check valves, and handle leaks within these groups. Few stations have a single program for all leaks. The overall status of leak programs in the industry can be summed up by:

- No industry standard or regulatory mandate
- No real definition on “what is a leak”?
- Every site has their own version and interpretation

- Responsibility for program falls into different areas of the plant
- Several plants have a program for tracking leaks or fixing leaks, but not both
- Lack of coordination between maintenance, engineering and procurement on program implementation
- Lack of knowledge and training in proper design, selection and installation of new age sealing products
- Reliance on OEM supplied spares that may or may not be correctly designed for the application
- Obsolete or damaged sealing products in the warehouse
- Lack of information on drawings, manuals, procedures and specifications
- Consultants want to tell you where your problems are – not how to fix them

Successful Leak Reduction Programs all have the same elements. These include:

- Support from all levels through management
- A coordinator(s) that has a high level of credibility, passion and competence (internal, external or combination resource)
- A sealing products company as a partner
- Updated procedures and specifications
- Training Program
- Current sealing products materials and technology
- Quality inventory
- Computer program to track, trend and define sealing product future, installed and historical configurations
- Integrated with other programs such as Work Management RCM [reliability center maintenance], PdM [predictive maintenance], Lube Oil walkdowns, etc…

Typically, it takes a significant event(s) or poor audits to get management's attention. However, there have been numerous industry events and issues that are making management more likely to support a program:

- More widespread use of OEs have led to several packing and and gasket OEs over the past four years
- Major issue of spiral wound gasket radial buckling has resulted in catastrophic FME [foreign material exclusion] situations in U.S. Pressurized Water Reactor (PWR) Steam Generators and several pump and valve failures with the Institute of Nuclear Power Operations (INPO) and the Nuclear Regulatory Commission (NRC) now involved
- Stations are receiving INPO "strengths - weakness" and Industry TIP (Top Industry Practices) for Leak Reduction Programs
- Leakage leads to equipment degradation and flies in the face of new equipment health mandates and programs

Database Development

AP Services and Insert Key Solutions (IKS) have developed a robust web based database called LeakManager and SmartSeal to define, track, trend and repair leaks. Leak Manager is integrated with the work management system and picks up any leak by searching a work request field or text in the work description.

LeakManager Database

The intent of LeakManager is to route leaks at the Work Request (WR) level before planning to designated program or equipment owners so the leak can be categorized by:

- Leak Type (Oil, Water, Steam and Air)
- Category (1 to 5 leak severity with 1 = none but evidence while 5 = large)
- Equipment or Joint Type
- Corrective Action (repair / adjust / monitor)

A report queue is generated every night for every leak at both the WR and Work Order (WO) level. The user can click on the status and details for each occurrence to get high level information. Reports can be generated to define the number of leaks by dates, cycle, system, equipment type, etc., to provide a constant status of the program.

The user can update the WR and also create the content of the WO (Figure 2 and 3) from the program with formatted to data so quality reports can be generated.

SmartSeal Database

LeakManager is where leaks are categorized and defined while SmartSeal is where the repair is planned and managed. SmartSeal works with any type of packing or gasket regardless of the manufacturer. SmartSeal is a very powerful web based application that provides the following:

- Auto image for valve packing and gasket systems (Figure 4)
- Calculates packing and gasket stress values and gland stud torque (Figure 5)
- Calculates packing friction
- Materials catalog that can be integrated to station inventory system (Figure 6)
- Provides an “installed” / “history” / “future” status for each component
- User Name and Log in defined security levels

SmartSeal can be run from within the utilities’ intranet or hosted on the web.

Gaskets

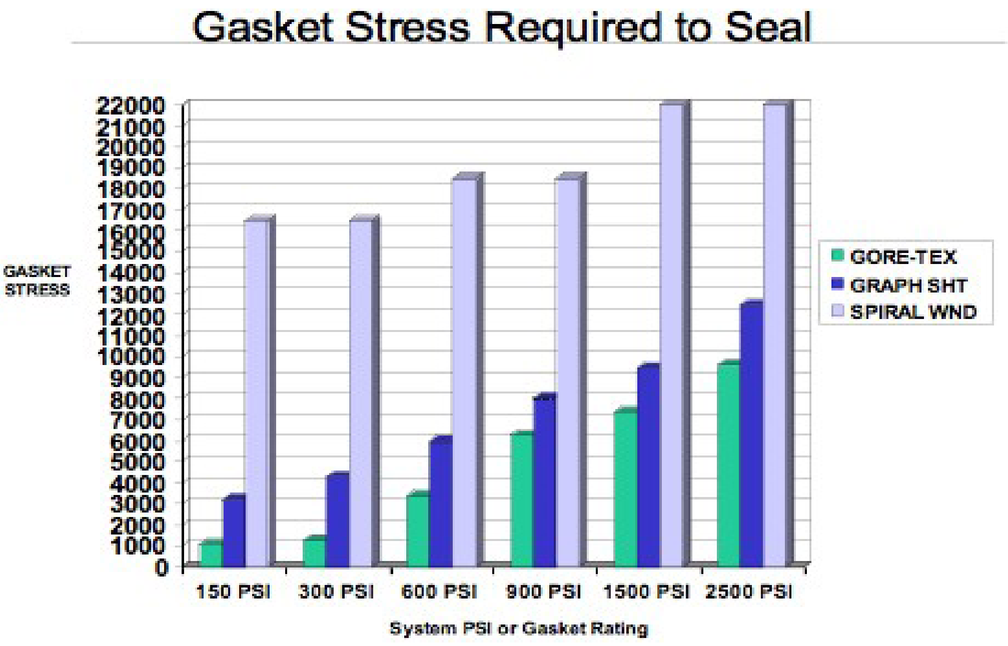
Gaskets or pressure seals are used in valve body-to-bonnet joints and other joints depending on the particular valve type and manufacturer. Valve gaskets come in a variety of types and materials but certainly the most common valve body-to-bonnet gasket is a spiral wound design. Spiral wounds originally used asbestos filler and, like valve packing, have been converted to Flexible Graphite. Flexible graphite has also been the material of choice where asbestos sheet gasket material, metallic gaskets or pressure seals were used.

Flexible graphite is a superior sealing product when compared to asbestos and synthetic non-asbestos products such as fiberglass and Kevlar. However, just as in flexible graphite valve packing systems, the importance of having the correct stress and containing the flexible graphite are critical to success. Plants fail to address this and failures occur that result in leaks and foreign material induced in the piping systems. Flexible graphite spiral wound gaskets, sheet gaskets and pressure seals require a calculated gasket stress and containment.

Gasket Stress

Most stations use a percent of yield on the bolt assuming it generates the required stress to the gasket. This approach is flawed and results in the majority of valve flanges, and even standard flanges, being either over- or under-stressed. The following basics should be used when determining a proper gasket stress:

- Forget the bolt unless yield is approached
- Every gasket material has an ideal stress range that should be used to calculate torque
- Synthetic sheet material’s stress range changes with temperature
- Ideal spiral wound stress is based on seating the gasket – not so with other gasket types
- Published “M” and “Y” values are useless
- Spiral wound gaskets with flexible graphite filler must be contained either by the flange design or inner and/or outer metal retaining rings
- Use flat washers on all bolted joints to reduce friction.



Spiral Wound Radial Buckling

One of the major drawbacks with flexible graphite-filled spiral wounds has been the tendency to buckle inward and induce pieces of graphite and winding material into the piping system. There have been at least eight industry OEs and NRC Information Notice 2004-10 on this subject and several utilities such as Exelon, Duke and Southern Company have addressed, or are presently addressing, the issue. The failures can occur in both American Society of Mechanical Engineers (ASME) Standard B16.5 flange designs (raised face) and other applications where an uncontained spiral wound gasket is used in a gasket joint (like a male/female flange that is open to the bore). The following pictures show some examples of spiral wound failures.

Figure 7 is a failed spiral wound gasket from the suction side of a Boiling Water Reactor (BWR) Boiler Feed Pump (BFP) used on standard B16.5 flange. Figures 8 and 9 are from a failed spiral wound gasket used on a Crane gate valve in a PWR. Pieces of this gasket and others were found lodged in the steam generator.

Inward buckling of spiral wound gaskets is well documented in several cases. Many companies like Exxon/Mobil and DuPont mandate the use of an inner ring to reduce the effect of inward buckling. ASME B16.20 requires the use of inner rings if the user experiences inward buckling. Inward buckling of spiral wound gaskets is mostly due to the incompressibility of the filler materials such as PTFE [Teflon] or graphite which exert excessive radial forces on the inner windings. While inward buckling is difficult to predict, it will be affected by the following factors:

- Gasket geometry (diameter, radial width)
- Type of the filler used
- Gasket density
- Gasket fit in the retaining ring
- Flange surface finish
- Method of loading

Tests performed on 10” Class 300 style CG gaskets [Flexitallic Spiral Wound Gasket with Outer Retaining Ring] showed a much smaller amount of inward buckling on the gaskets made with Thermiculite filler as compared to spirals made with flexible graphite (see Figures 10 and 11).

Gasket Sheet Materials

With the elimination of asbestos-based gasket sheet materials in the 1980s came a flood of replacement gasket materials boosting comparable performance. The products utilize a base product of mineral wool, fiberglass, carbon or Kevlar with a rubber binder such as Nitrile, EPDM or SBR. These products have proved to be a colossal failure in temperatures above 300 °Fahrenheit (°F) continuous service even though many are rated to 800 °F.

As in the case of valve packing and spiral wounds gaskets, flexible graphite has been the accepted substitute for asbestos. The down side of flexible graphite sheet gasket material has been the poor handling characteristic, poor blowout resistance, and corrosion of the flange surfaces. Graphite is one of the most noble materials in the Galvanic Series, and therefore when coupled with any metal may cause corrosion of the metal due to the galvanic corrosion. Graphite is also susceptible to oxidization in temperatures as low as 400 °F.

Thermiculite has shown great promise as replacement for asbestos and graphite. Asbestos was the only material used as the filler for spiral wound and sheet gaskets in nuclear applications prior to being phased out by graphite. Vermiculite and asbestos belong to the same phyllosilicate classification, and therefore it can be expected that Thermiculite will offer a direct replacement without the drawbacks of graphite and heath concerns of asbestos.

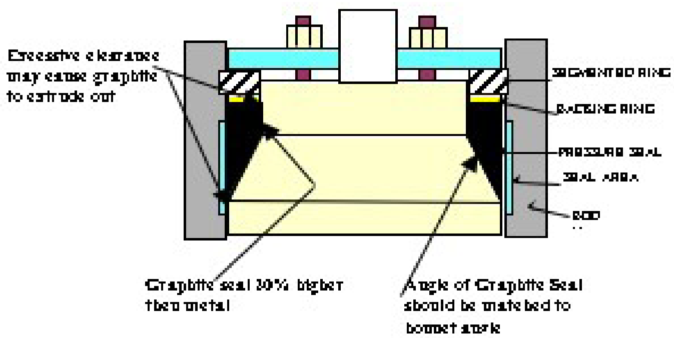
Graphite Pressure Seals

Over the past several years, the superior performance of graphite pressure seals over original metal seals has been consistently demonstrated. Many utilities in this country have instituted wholesale conversion efforts.

Historically, metal seals, if installed properly against clean smooth surfaces, have provided good sealing, but metal seals provide very little margin for typical situations, which are found at an operating power plant. When metal pressure seals are assembled, by the manufacturer under factory conditions, with new parts, highly finished surfaces, and optimum valve position, good performance is obtained. In the aging power plant, where finished surfaces may have minor scratches and washed out areas, where minor valve body distortion has occurred, and valves are found oriented at every angle (particularly with stems horizontal) metal pressure seals provide questionable results.

Though graphite pressure seals will provide superior performance, they must be sized, installed and loaded properly to do so.

CRITICAL POINTS FOR GRAPHITE SEALS



Insure Proper Sizing and Measurements

If pressure seal sizing was based solely on the measurements of the original metal seal, the fit of graphite in the valve may not be acceptable. Particular areas of concern are:

- Difference in angle between the bonnet and the seal
- Excessive clearance at the pressure seal tip
- Inadequate height to accommodate expected take-up (~20%)

Experience from graphite packing should be considered when working on graphite pressure seals:

- Up to 20% consolidation of the graphite should be expected
- Containment of the graphite is critical (In lieu of the braided/composite ring with packing, metal caps and tight clearances are used with pressure seals)
- Take-up is required to handle any future graphite consolidation
- Live-Loading and retorquing can be used to insure long term load is maintain

- Adequate load must be applied to insure the graphite is adequately loaded (4000-8000 psi)
- Torque values should be provided, and with indication if torquing alone will not provide the required load.

Valve Packing Systems

Currently, the most common packing configuration used is combination flexible graphite packing systems with either braided or composite anti-extrusion rings. Teflon packing systems are used in numerous applications in braided or formed Chevron. Die-formed and braided flexible graphite, as well as braided Teflon packing material, will quickly extrude if not contained with anti-extrusion rings. In 1991, Argo developed a braided graphite packing material that uses a combination of flax shaped graphite filament yarn coated with Teflon and graphite. This material, Argo Style 5000, is used as both an end-ring material and “stand alone” bulk packing. Testing at AECL Chalk River has identified the Style 5000 as the best “all around” packing material. Argo Style 5000 is used extensively in both nuclear and fossil with outstanding results.

Every type of packing has attributes and drawbacks. Therefore, it is important to realize what those are so the best packing material can be matched to the application as well as the packing program that exists at a station.

Packing Configurations

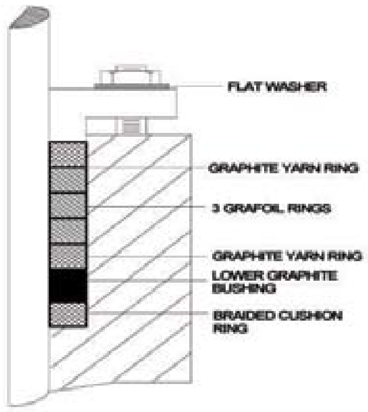
There are hundreds of different packing configurations used in stations but let’s review the most common first. These include:

- Filament Yarn/Flexible Graphite
- Composite/Graphite
- Argo Style 5000/Flexible Graphite

The following pages illustrate the packing configurations, their sealing and friction plots, and advantages/disadvantages of each packing system.

Packing Systems

Standard Five-Ring Packing "EPRI" Set



Performance Statistics

- Friction @ 4,000 psi - 897 lbs
- Friction @ 3,000 psi - 660 lbs
- Consolidation - 33%
- In-Service Consolidation - 2%

Advantages

- Easy to use
- Consistent Performance

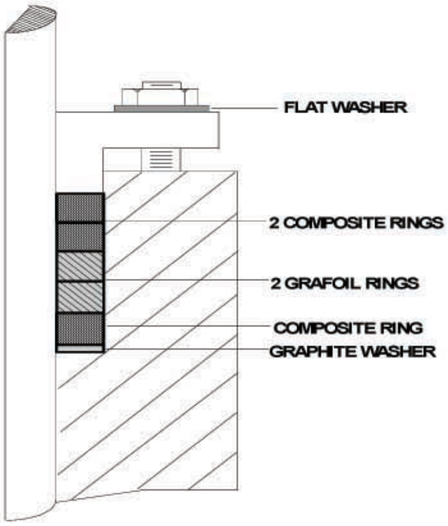
Disadvantages

- High Friction
- Yarn fractures
- High "Break-away" Friction

PPL Susquehanna Valve Packing 1

Packing Systems

Standard Five Ring Set with Composite End Rings



Performance Statistics

- Friction @ 4,000 psi - 425 lbs
- Friction @ 3,000 psi - 360 lbs
- Consolidation - 21%
- In-Service Consolidation - 2%

Advantages

- Low and Repeatable Friction
- Lowest leak rate of any packing system

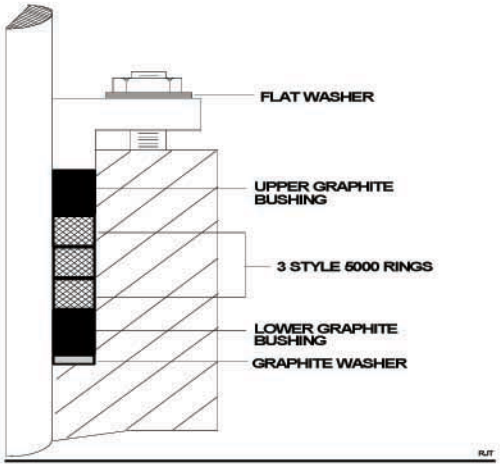
Disadvantages

- Need to apply adequate stress
- Sensitive to high vibration
- "Precision" repack
- Needs to be live-loaded in high travel applications

PPL Susquehanna Valve Packing 10

Packing Systems

Low Friction Style 5000 or PTFE Packing System



Performance Statistics

- Friction @ 4,000 psi - 397 lbs
- Friction @ 3,000 psi - 278 lbs
- Consolidation - 35%
- In-Service Consolidation - 4%

Advantages

- Lowest friction
- Very forgiving and easy to use
- Low break away

Disadvantages

- PTFE issues
- Friction goes up over time as lubricants wear

PPL Susquehanna Valve Packing 11

Graphite Bushings

The most practical material to use to take up space in the stuffing box is high-density Electro-graphite bushing material. It is manufactured from machined halves so it can be installed in-situ and is also drilled/taped for future removal. Graphite bushings are more practical than bushings made from Stainless Steel, Aluminum, or Bronze since graphite will not score the stem and is inexpensive.

Initially, the only function of the graphite bushing was to take up space. However, testing performed by Fisher Controls and Argo in 1992 identified that graphite bushings can improve valve packing and valve performance by performing a bearing function as well. In most valve designs, the packing supports and centers the stem. Any misalignment of the valve stem due to actuator side loading or valve orientation leads to very short packing life as well as potential stem scoring. Placing close clearance bushings both above and below the packing set provides the following:

1) Improved stem alignment

2) Better packing material containment

3) Uniform loading of the packing area (gland followers often have large clearances)

4) Improved valve performance since the stem is centered in the valve. This results in less stem nut wear for MOVs, less gate guide wear for gate valves and overall better seat loads

Packing Gland Stress

One of the most important aspects of a well designed packing system is insuring proper packing gland stress to achieve adequate axial-to-radial transfer of the packing rings that are providing the sealing effect. Packing systems should have the following packing gland stress:

Note: This is for system pressures < 2,000 psi. For higher pressures, use 1.75 times system psi as a minimum, and 2 times system pressure for preferred with combination systems and Style 5000.

These values for packing gland stress have been used by Argo, Ontario Hydro and AECL since the late 1970s. Numerous packing suppliers and some previous test reports recommend much lower gland stresses. In a controlled laboratory environment, it is possible to seal at lower stresses than expressed in this table but in field conditions the failure rate is very high. In 1985, Susquehanna repacked 750 valves during a refuel outage using 2,500 psi gland stress and had a failure rate of 30% (most had to be adjusted after start-up). The following outage 1,115 valves were repacked and the stress value was increased to 4,000 psi preferred and 3,000 psi minimum. The failure rate dropped to 2% with no valves requiring repacking and only valves stressed to the 3,000 psi requiring adjustment.

Conclusion

Elimination of packing, gasket and pressure leaks is possible. A leak reduction program for valves that includes dedicated coordinator(s), training and the LeakManager and SmartSeal Program will provide a basis to create a culture of zero leaks.

Packing Material	**Packing Stress (psi)**	
	Preferred	Minimum
Single Packed Comp/Graphite	4000	3000
Double Packed Valves (all)	5000	4000
Argo Style 5000	4000	2000
Braided Teflon	2200	1700
Chevron Teflon	700	500
Asbestos	8000	5000

Proven approaches in packing, gasket and pressure seal technologies will provide infinite leak free performance.

References

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VanTassell, D.M., “Improved Packing Systems for Critical Valves,” 2001 Electric Power Research Institute (EPRI) Valve Symposium.

Jamalyaria, J., “Use of Thermiculite based materials in nuclear applications,” Flexitallic LLP, March 2006.

Figure 1 – Leak Report Query

Leak related WR or WO is pulled from work management system

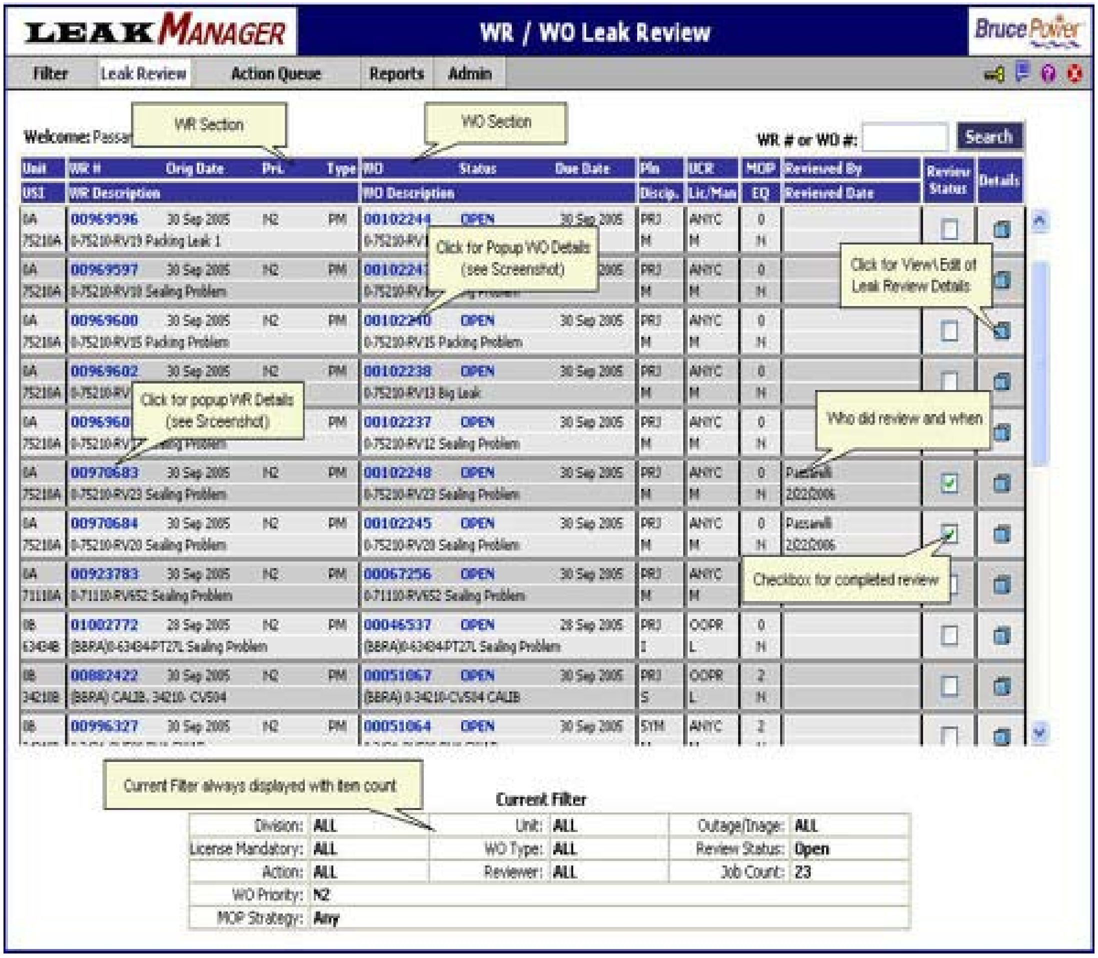


Figure 2 - Leak Review Filter

Filter is set by reviewer so leaks can be categorized by configurable parameters

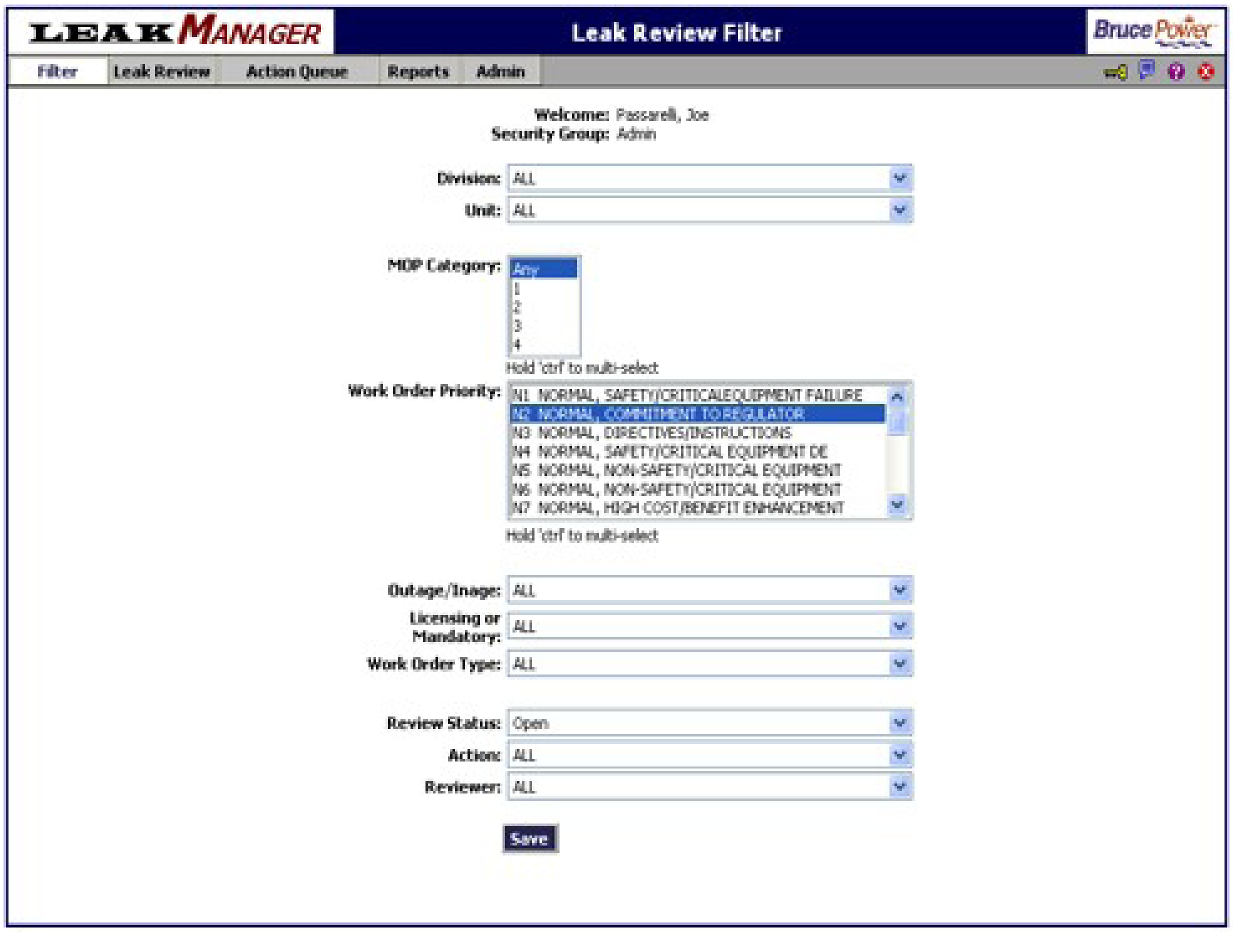
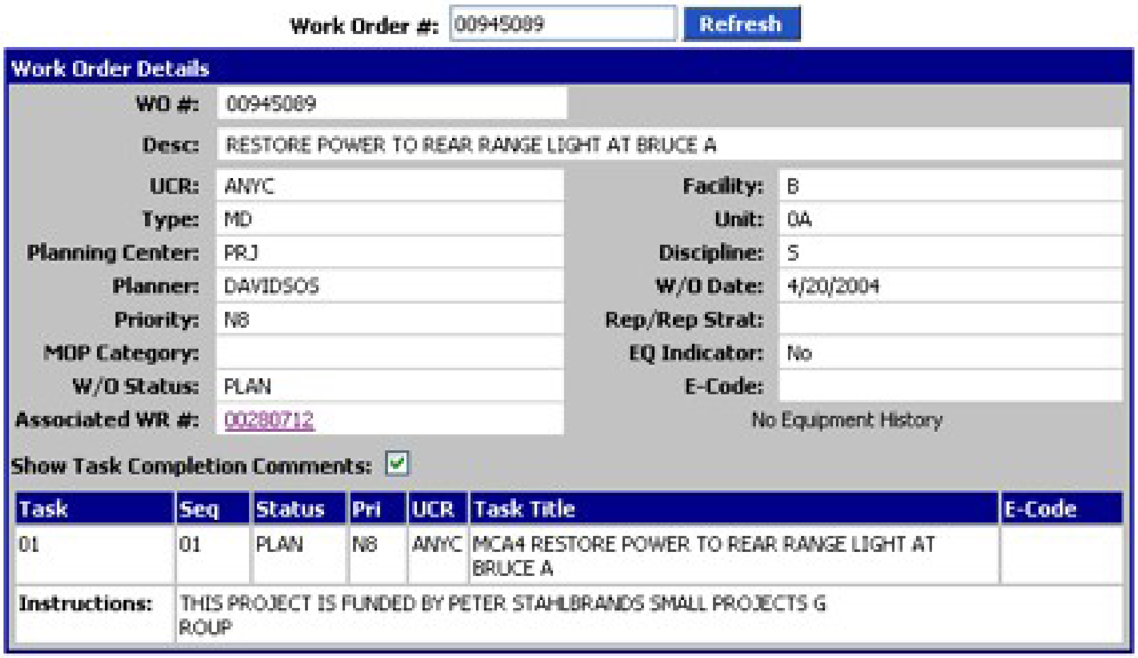


Figure 3 – Example of Work Order Detail


Work Order #: 00945089 Refresh

Work Order Details

WO #:	00945089		
Desc:	RESTORE POWER TO REAR RANGE LIGHT AT BRUCE A		
UCR:	ANYC	Facility:	B
Type:	MD	Unit:	0A
Planning Center:	PRJ	Discipline:	S
Planner:	DAVIDSOS	W/O Date:	4/20/2004
Priority:	N8	Rep/Rep Strat:	
MOP Category:		EQ Indicator:	No
W/O Status:	PLAN	E-Code:	
Associated WR #:	00280712	No Equipment History	

Show Task Completion Comments:

Task	Seq	Status	Pri	UCR	Task Title	E-Code
01	01	PLAN	N8	ANYC	MCA4 RESTORE POWER TO REAR RANGE LIGHT AT BRUCE A	
Instructions:	THIS PROJECT IS FUNDED BY PETER STAHLBRANDS SMALL PROJECTS G ROUP					

Figure 4 – SmartSeal Packing Configuration Page

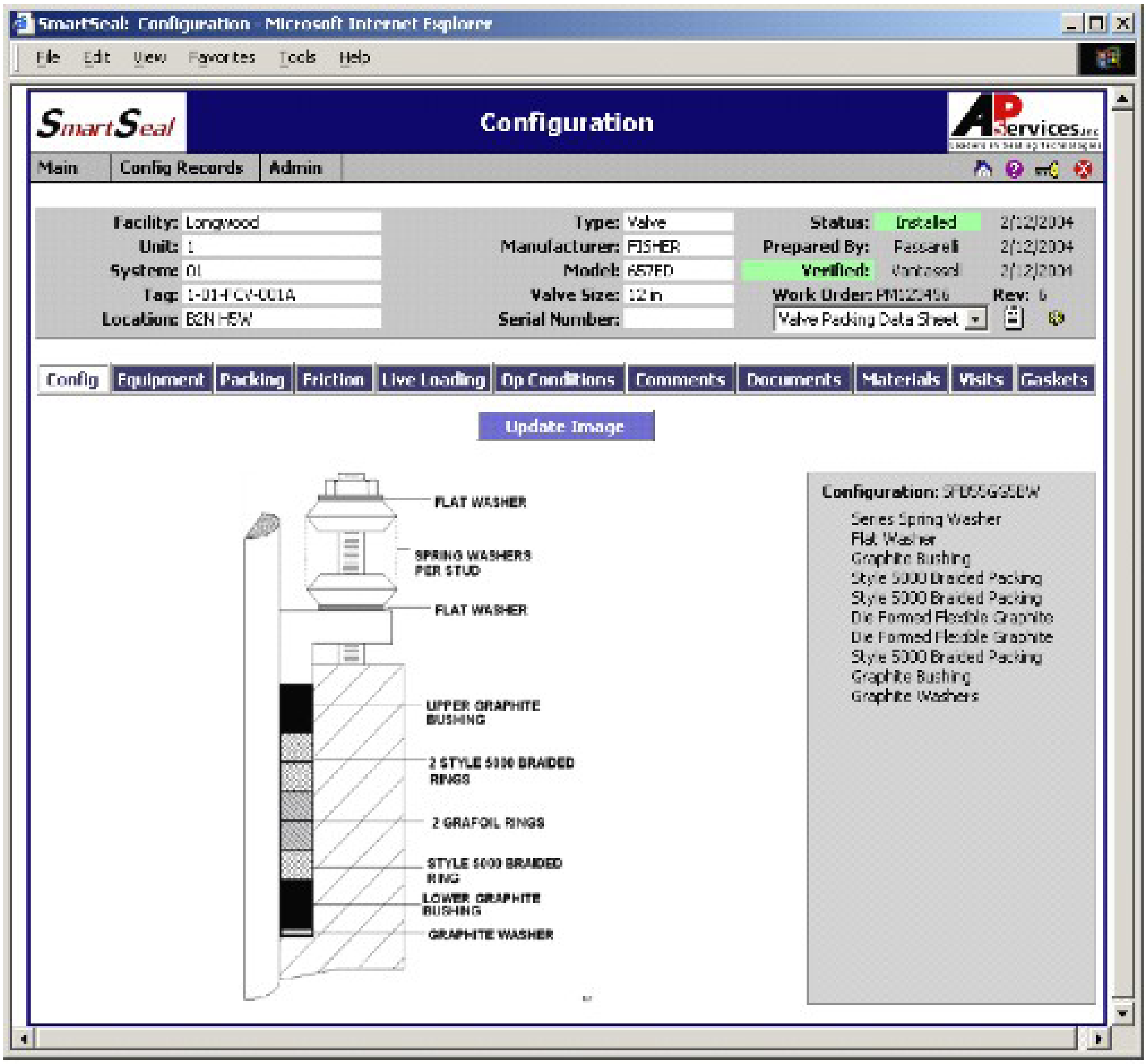


Figure 5 - SmartSeal Packing Torque and Detail Page

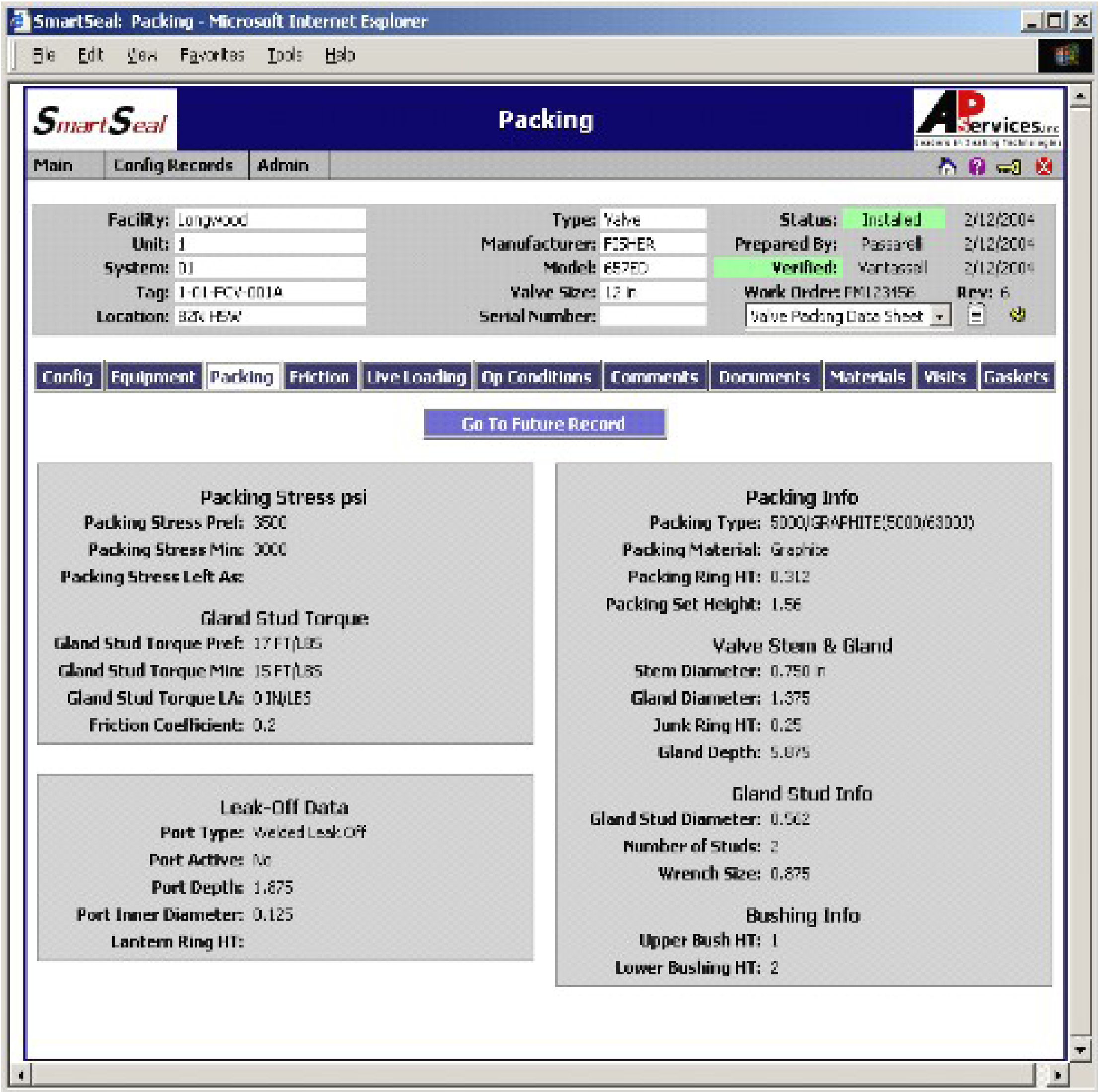


Figure 6 – Screen shot of SmartSeal Catalog

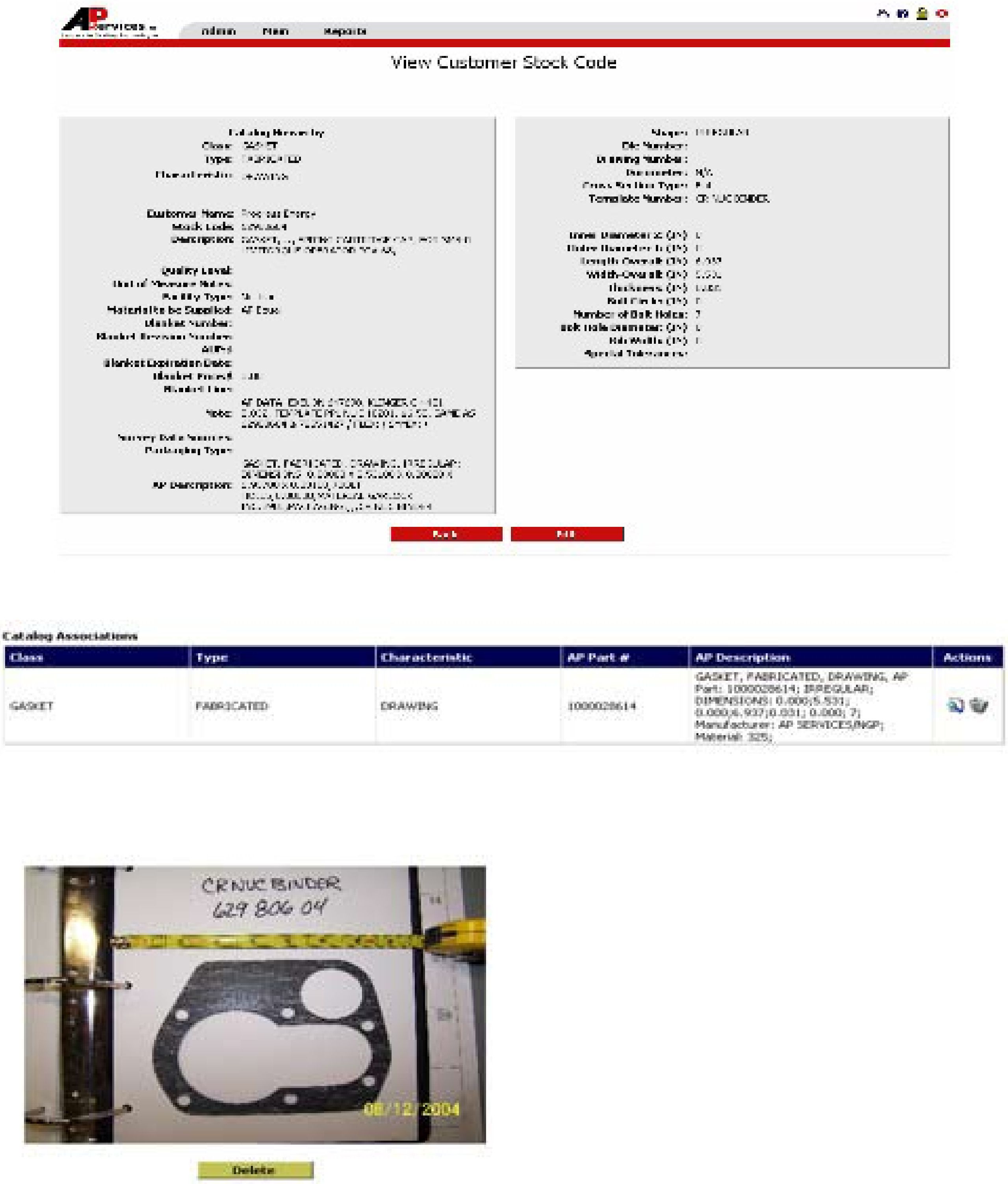


Figure 7 – Failed Spiral Wound Gasket from Suction Side Flange off the BFP in a BWR Station

Figure 8 – Crane Valve looking into bore of valve

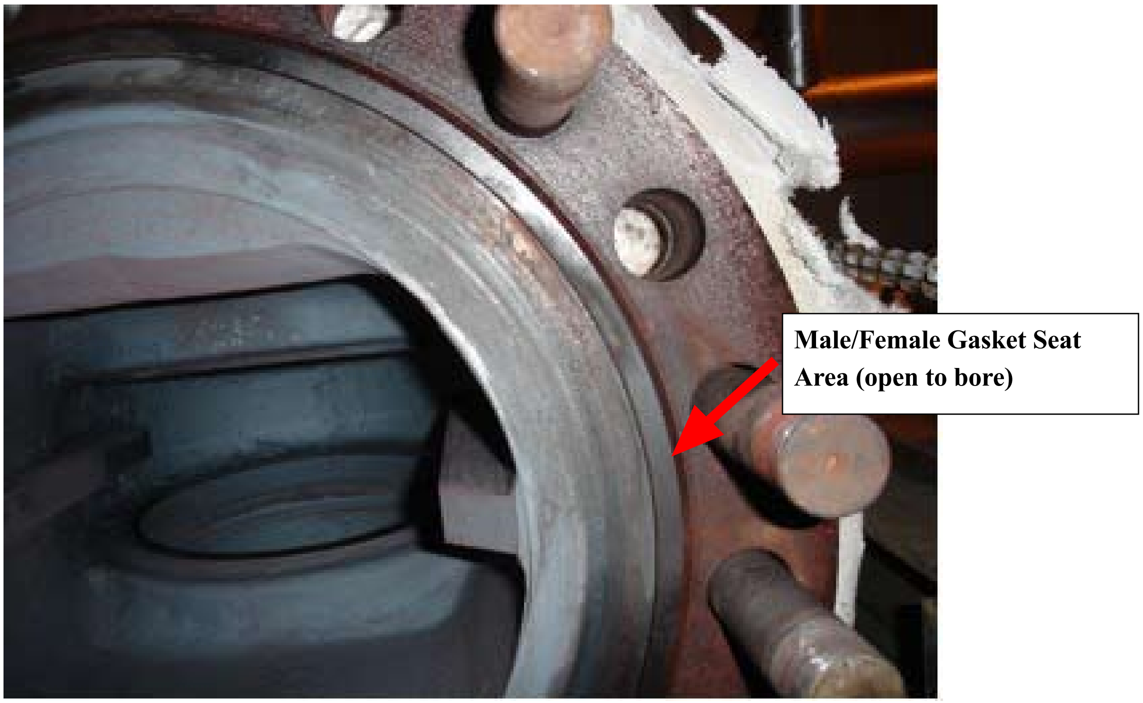


**Figure 9 – Internals of a Gate Valve with extruded gasket.
Gasket pieces also found in Steam Generator**

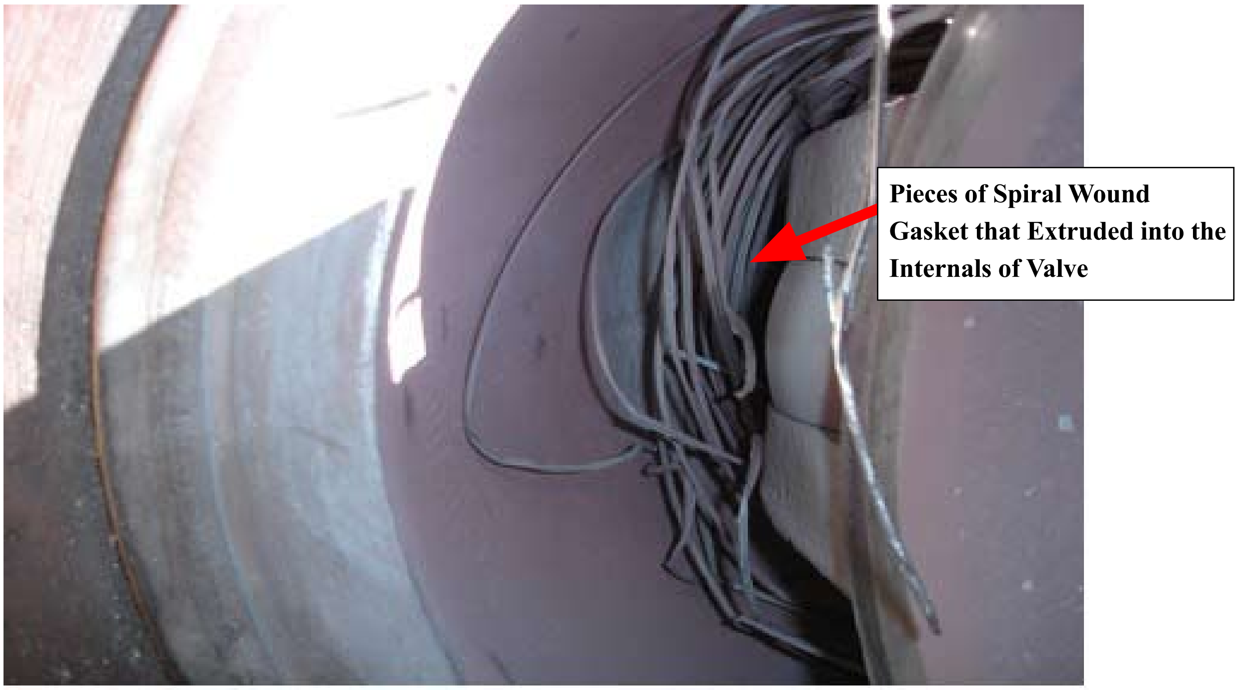


Figure 10 – Testing of spiral wound gaskets with Flexible Graphite filler show considerable radial buckling due to the compressive nature of Graphite

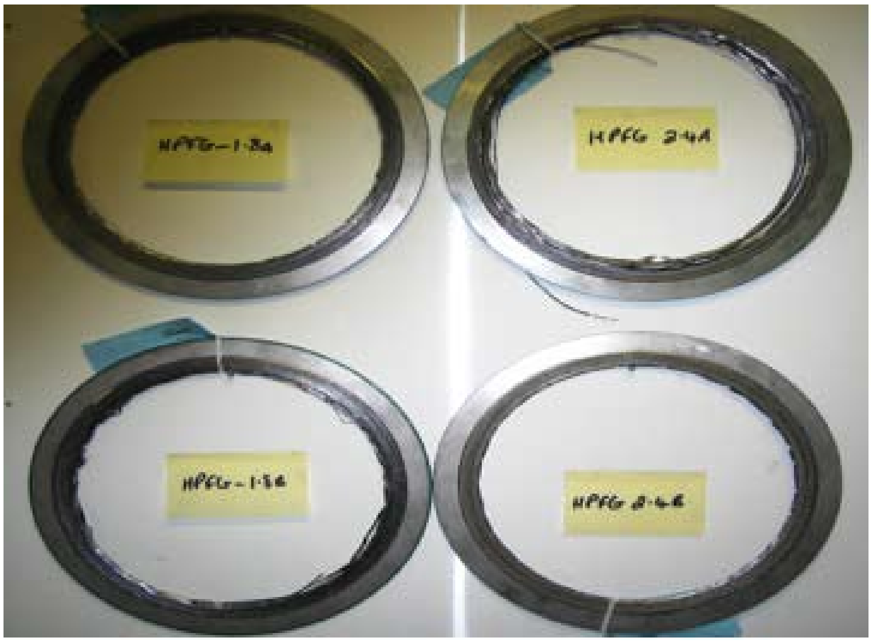
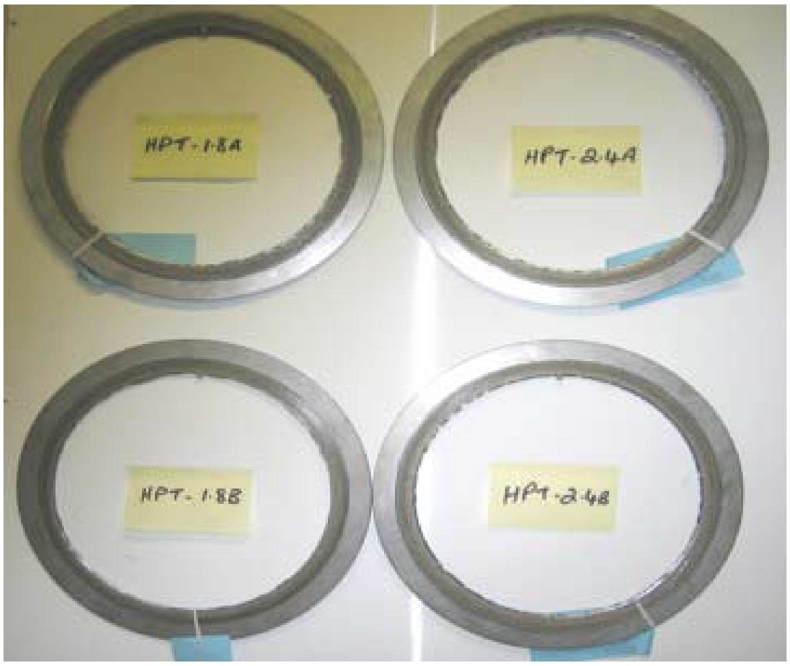


Figure 11 – Compression testing of spiral wound gaskets with Thermiculite filler have considerably less radial buckling



Digital Valve Positioners – The portal to real-time valve diagnostics

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Dresser Flow Solutions – Masoneilan

Peter Koenig
Sr. Product Manager
Enertech – Curtiss Wright Flow Control Corporation

Abstract

Maximizing plant performance, while maintaining stringent nuclear quality and safety standards, demands the utmost level of efficiency from all major assets. This is especially true in today's competitive power generation market where plant management must cope with reduced budgets and the loss of experienced staff. Although some proactive nuclear asset management programs are being developed to address this challenge, the need for technology to improve and facilitate operational decision-making in light of uncertainty and risk is becoming obvious.

Advanced performance digital valve positioners are now available that can be the stepping stone to the "Smart Plant." Critical air-operated control valves, outfitted with the latest digital positioning technology, can significantly increase operational efficiency and establish a solid foundation for enhanced predictive maintenance programs. Process control systems and software packages have also evolved to capture the valuable information produced by these smart valve interfaces, giving operators a complete view of the entire process. The use of standard digital communication protocol eliminates problems associated with field device interoperability with supervisory systems. This minimizes the infrastructure and knowledge required to fully benefit from digital positioners.

This paper will discuss the benefits of microprocessor-based digital control valve technology, and describe the offline and online diagnostic capabilities of advanced positioners. Interfacing with these smart devices will be discussed in order to take full advantage of the available information and create a portal to real-time valve diagnostics.

Introduction

Air operated valve (AOV) reliability improvement programs and valve optimization initiatives have become a critical part of a nuclear plant's overall asset management strategy. Current AOV program objectives include reducing operation and maintenance costs, improving safety, reducing outage scope, and standardizing processes and procedures to improve work efficiency. These extremely challenging goals often exceed the capability of installed conventional technology. To make matters worse, plants must meet these challenges with less available resources. As a result, the nuclear power industry can no longer rely on current preventative maintenance programs to insure trouble-free operations between refueling outages, make additional improvements to equipment reliability, or extend maintenance intervals on safety significant and generation critical components.

The application of field proven digital technology to control valves can have a substantial impact on overall process control while providing the necessary online condition monitoring required to meet these difficult challenges. Digital valve positioners give plant operators the ability to gather critical valve information and provide diagnostic capability for the final control element that is not possible with conventional equipment. The diagnostic data provided with digital valve positioners is a stepping stone to substantial cost saving benefits in control valve maintenance while improving safety, reliability and process control.

Installing a digital positioner on the valve will provide immediate improvements in configuration, calibration, tuning and precise valve positioning control. However, most plants have not been taking advantage of the full potential of these smart devices with respect to continuous monitoring, real-time diagnostics, and multi-variable process information.

The latest generation digital valve positioners, such as the SVI II AP from Masoneilan, can provide valuable online health data and predictive diagnostic information via HART communication protocol. The HART digital signal is communicated over the existing twisted wire pair for the 4-20mA analog channel and uses the same loop power. The digital protocol facilitates the transfer of valuable information in a monitoring system for continuous control valve analysis in order to establish enhanced predictive maintenance programs. In this way, you are able to leverage intelligent device capabilities to improve plant operations, reduce problem-identification to problem-resolution time, and continuously validate loop integrity and control information.

Mechanical (Analog) Vs. Digital

There are significant performance and maintenance benefits of digital positioner compared to their mechanical counterparts. The elimination of many mechanical parts such as bearings, cams, and feedback springs significantly reduces required maintenance and inspection. In addition, the ease and flexibility of mounting allows for standardization of a single positioner for every rotary, reciprocating, and quarter-turn control valve in the plant. The ability to install a remote mount feedback module permits removing the positioner from a valve when location, high temperature, extreme vibration, or radiation levels present dangerous working environments.

Besides having many mechanical parts that require frequent adjustments and maintenance, conventional positioners are only equipped with an analog signal which limits their utility as a data-transfer device. By contrast, microprocessor-based digital positioners have few moving parts and modular construction minimizing maintenance and decreasing drift caused by wear and fatigue. Simple, flexible mounting configurations are adaptable to any manufacturer's control valve using non-contact position sensors. A set-up wizard facilitates easy and repeatable configuration, automatic zero and span, and automatic tuning. This process replaces physical adjustments by technicians with optimized algorithm control for increased precision and consistency.

Finally, digital communication ability provides unmatched improvements in the areas of performance monitoring, alarming, configuration utilities, audit trail documentation, and security and calibration functions. See Figure #1 for a detailed technology comparison.

Like any mechanical system, conventional control valves are subject to failure. Air operated control valves consist of numerous moving parts that are mechanically interconnected and pneumatically actuated. Even though valve manufacturers design and build valves for long, reliable operation, degradation or eventual failure of subcomponents can result from normal wear and tear, improper trim selection, misapplication of the valve, and exposure to harsh environments. See Figure #2 for a depiction of the many areas of potential damage which form the basis of control valve condition assessment. If progressive wear is left untouched or not monitored adequately, it will ultimately degrade process control performance (i.e., reduce thermal efficiency of the plant) and could ultimately lead to catastrophic failure forcing an unplanned outage situation. How much is a precursor warning of such failure worth?

Most nuclear plants have identified those final control elements with single point failure vulnerabilities that pose substantial operational and economic risks. New control valve specifications for these applications call for digital valve positioning systems designed to maximize reliability and fault tolerance while facilitating diagnostic monitoring and maintainability. Based on their criticality and cost saving potential, the following target applications have already been identified for digital positioner upgrades: Feedwater Regulation (Steam Generator or Reactor Level Control), Feedwater Heater Level / Heater Drain, Auxiliary Feedwater, Feedwater Bypass, and Atmospheric and Condenser Steam Dump Valves.

Digital positioning technology allows the gathering and monitoring of critical valve information needed in these applications to diagnose existing or future problems related to the complete valve and actuator assembly. The main types of valve diagnostic information can be divided into the categories of Offline, Continuous Online and Extensive Online.

Offline Diagnostics

Most digital positioners on the market are capable of measuring critical data through diagnostic signatures. The positioner microprocessor samples data from built-in pressure and position sensors at a high frequency to produce Positioner, Multi-Step Test, Standard Actuator, and Extended Actuator Signatures (Figure #3).

The positioner tests produce results similar to the ISA standard for diagnostic testing. The signatures are executed using software or in some designs, like the SVI family, an integral pushbutton interface. Some intelligent valve interfaces can also embed a valve signature in the non-volatile memory of the device. This signature can be used to compare "as-shipped" condition from the valve manufacturer to "as-installed" condition in the field. Along with the actual data points, the device stores the associated friction, spring range and stroking speed which allow baseline comparison with subsequent signatures taken overtime. Comparing overlapping signatures (Figure #4) will help to reveal any possible valve degradation in performance such as poor seating and subsequent valve leakage. Unfortunately, all these tests involve temporarily removing the control valve from its normal operation. This diagnostic mode disconnects the capability of the control system to control the valve, causing operations to intervene and bypass the valve or put the control loop in manual. For this reason, most power plants do not take advantage of the diagnostic capabilities of digital positioners. The majority of valves in a nuclear plant are physically taken out of service, or worked on in the field during plant shutdowns using AOV diagnostic test equipment.

As the final control element, the control valve has the biggest impact on overall loop performance in the plant. Poor control, inadequate responsiveness, or inability to move to a failsafe position can severely hamper operations and ultimately cause plant shutdowns. Conventional analog positioners are simply unable to detect these types of problematic situations. Important valve related information, such as the status of air supply, actuator pressure, valve plug position, and valve "stiction" [sticking friction] could only be obtained by intrusive methods and physical observation. Although there are some loop-tuning packages on the market that can detect poor performing valves, they require sending an artificial signal to move the valve and a travel sensor to precisely quantify the valve performance. More importantly, these tools identify an impaired control valve only after the problem has become serious enough to impact the process.

Fortunately, it is now possible to perform valve condition assessment during normal plant operation without the need to connect external diagnostic test equipment.

Continuous Online Diagnostics

Continuous diagnostic data is saved in the non-volatile memory of Masoneilan's SVI positioner family and it includes valuable data which can be used to identify process control deficiencies, determine premature trim wear and predict eventual valve failure. It can also be used to determine if the valve is not suited for the application. This critical valve information includes: Cycle Count, Accumulated Travel (Strokes), Time Open, Time Closed, and a user defined Time Near Closed. The integration of these values into a historian for trending runtime data (Figure #5) is seamless and easily accomplished because the information is continuously monitored and saved onboard the positioner. Equipped with this type of intelligent valve interface, the valve becomes a virtual "field server" of key control valve information.

Self-initiated device alarms are also part of online diagnostics. Figure #6 shows typical device alerts that are available within an intelligent valve interface. They are grouped into four different categories in order to make it easier to understand what the relation is to the device's health. These categories are: Operation, Communications, Firmware and Circuit. When an alarm is triggered for any of these conditions, online health alarms will notify the supervisory control system (i.e. HART ready host), thereby bridging the information gap for the plant operator.

The valve position error (deviation from setpoint) and position error failsafe alerts are user configurable. The position error alert will be activated if the actual valve position deviates outside the specified range for more than the specified time (e.g., greater than 5% position error for a period of 20 seconds). If the valve offset continues beyond a configured failsafe time, the valve can be made to go into its failsafe position. If the valve is struggling to maintain position, the bias out of range alert is activated. The root cause could be insufficient air supply or an impending failure of the pneumatic amplifier.

Extensive Online Diagnostics

To extensively monitor the health and performance of a control valve during normal plant operation, without disturbing the process by conducting special tests, Masoneilan offers an online diagnostic tool for valves equipped with digital positioners. The online valve diagnostic (OVD) software only requires 0.1% to 2% of normal valve movement to gather the information necessary to conduct diagnostics. The tool reads data from the positioner's sensors periodically on a set schedule. Multiple pressure sensors, built into the advanced positioner, measure atmospheric, supply, I/P output, and actuator input pressures. The non-contact stem travel sensor, temperature sensor and loop current sensor are also utilized for feedback and health status. Information from these sensors is stored automatically in the software and a report is self-generated with an assessment of the valve's performance. Information such as friction, spring range, response time, stiction, lag, RMS error, and oscillation frequency are measured or computed. Using a series of complex algorithms to interpret the data samples over time, the software can determine impending valve failures before they impact plant operations. The fact that the determination is based on actual running conditions is what makes the tool so valuable. It can be utilized to calculate remaining lifespan of packing, o-rings, bellows seals, and actuator diaphragms. See Figure #7 and #8 for examples of some trending data on cycle count and friction tracking. This catapults plant predictive maintenance activities to the next level.

Utilizing online valve diagnostic tools can make the complex task of data and signature interpretation a reality. By providing a synopsis of the effects of in-service conditions on safety significant and economically significant control valves, preventative maintenance programs can be optimized by focusing on specific valves that need attention.

Interfacing With Digital Positioners

Human Machine Interfaces (HMI) with digital positioners in most plants have been limited to handheld communicators and some independent workstations loaded with Original Equipment Manufacturer (OEM) valve software. Until now, the digital positioner's most tangible benefits have been improvement in positioning control, ease of installation, and a reduction in mechanical moving parts. However, continuous device monitoring, real-time device diagnostics, and multi-variable process information represent the true power of digital communication. Integration of this data with plant control, safety and asset management systems has been the challenge. Fortunately, current HART-enabled control system interfaces, remote I/O systems and software solutions can make it easy and cost-effective to unleash the full potential of smart valve interfaces. By doing so, a complete view of the process is possible and the digital positioner can become a portal to real-time valve diagnostics. Figure #9 shows how these systems can integrate into the overall plant network.

Conclusion

Microprocessor-based field equipment has been utilized by the power industry for over twenty years. They have provided greater accuracy and stability, eliminated drift in calibration, and resulted in considerable time savings in the set-up, commissioning and tuning process of many types of instruments. Adding a digital valve positioner to a control valve, such as the SVI family of product from Masoneilan, immediately transforms the valve into a "mini-server" of valuable process information. Using diagnostic information gathered in the non-volatile memory of the device, it becomes possible to monitor cycle count, travel accumulation, and online clocks and alarms. This information can be used to determine if the valve is properly sized and can predict premature trim wear. In addition, the data collected from online valve diagnostic software tools provides a method of calculating the lifespan of the valve, determining the packing maintenance frequency, and predicting future process control issues. Having access to this type of information to perform valve signature analysis and advanced data interpretation can make even the most aggressive nuclear AOV program objectives a reality.

Conventional Mechanical Positioner	Advanced Digital Positioner
Design Optimized for Average Actuator Size	Equal Performance Regardless of Actuator Size
Numerous Moving Mechanical Components	No Moving Parts / Simple, Modular Construction
Calibration is User Dependent	Calibration is User Independent
Requires External Sensors for Valve Information	Valve Diagnostics are Embedded via Digital Communication
Must be Mechanically Connected to the Actuator	Linkage Free Mounting and Remote Mount Capable
Can't Monitor Online Health	Auto-Diagnosis: Continuous Online & Offline

Figure 1 – The many advantages of today's digital valve positioners compared to their mechanical counterparts.

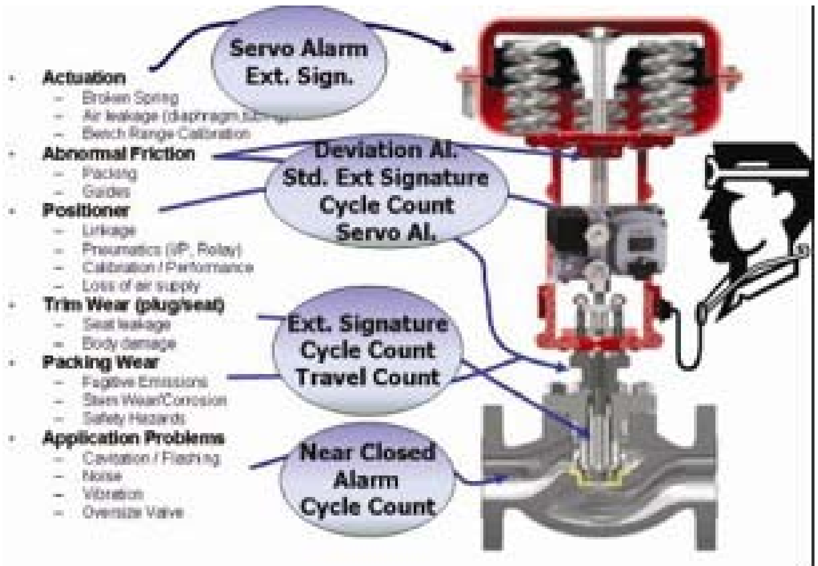
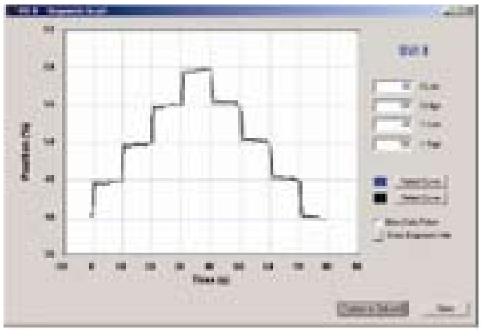
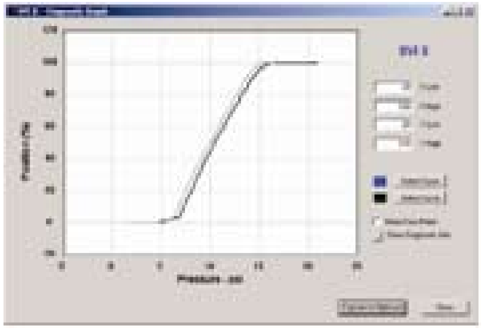


Figure 2 – Complete control valve condition assessment



(A)



(B)

Figure 3 – Examples of Bi-directional Step Test (A) and Extended Valve Signature Test (B)

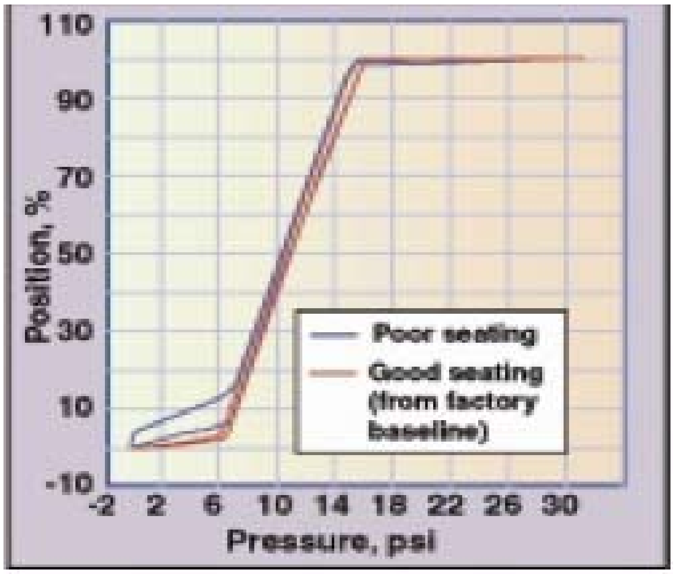


Figure 4 – Current Valve Signature in blue (upper loop) shows degradation of valve seating compared to Baseline Signature in red (lower loop).

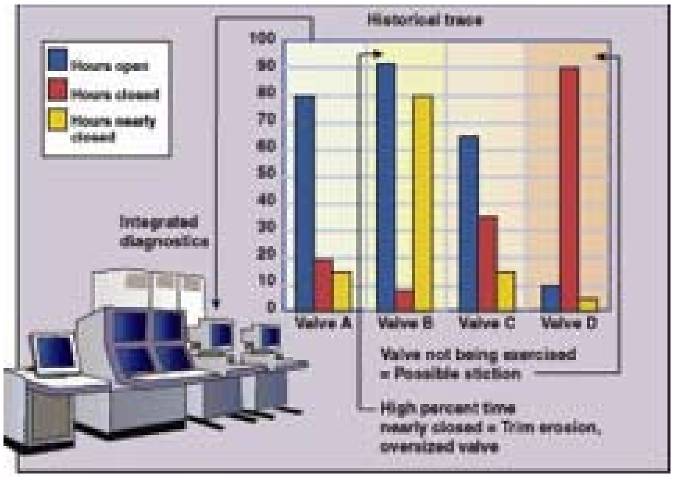


Figure 5 – Tracking valve position over time gives valuable insight into actual operating experience for improved equipment reliability programs.

Figure 6 – Online Diagnostic Alerts – Device's Health

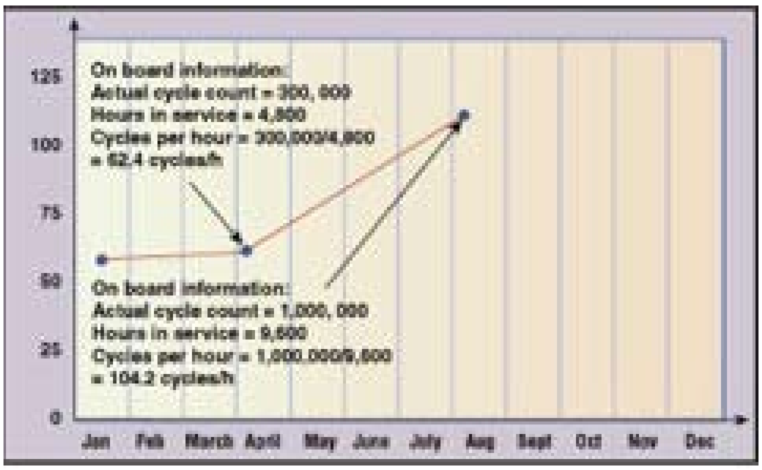


Figure 7 – Data provided by the smart valve interface can be used to set a maintenance schedule based on actual run-time

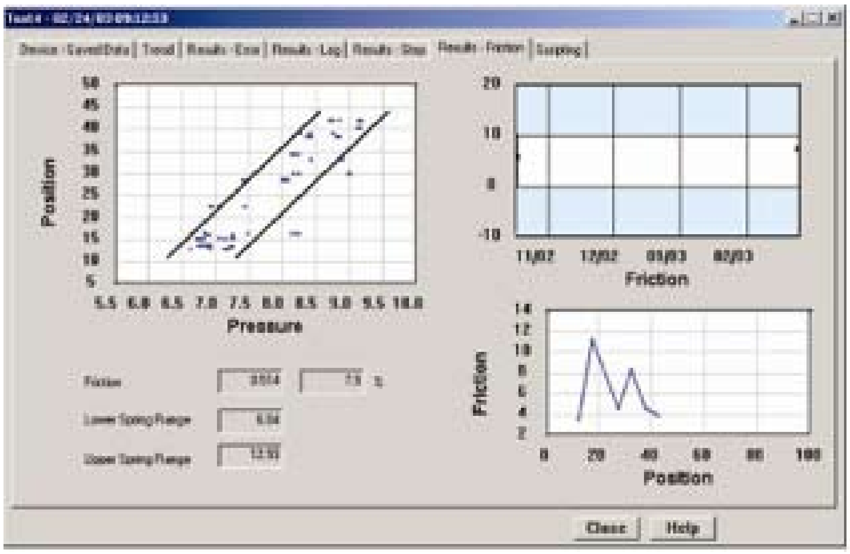


Figure 8 – Online Diagnostics – Friction Monitoring

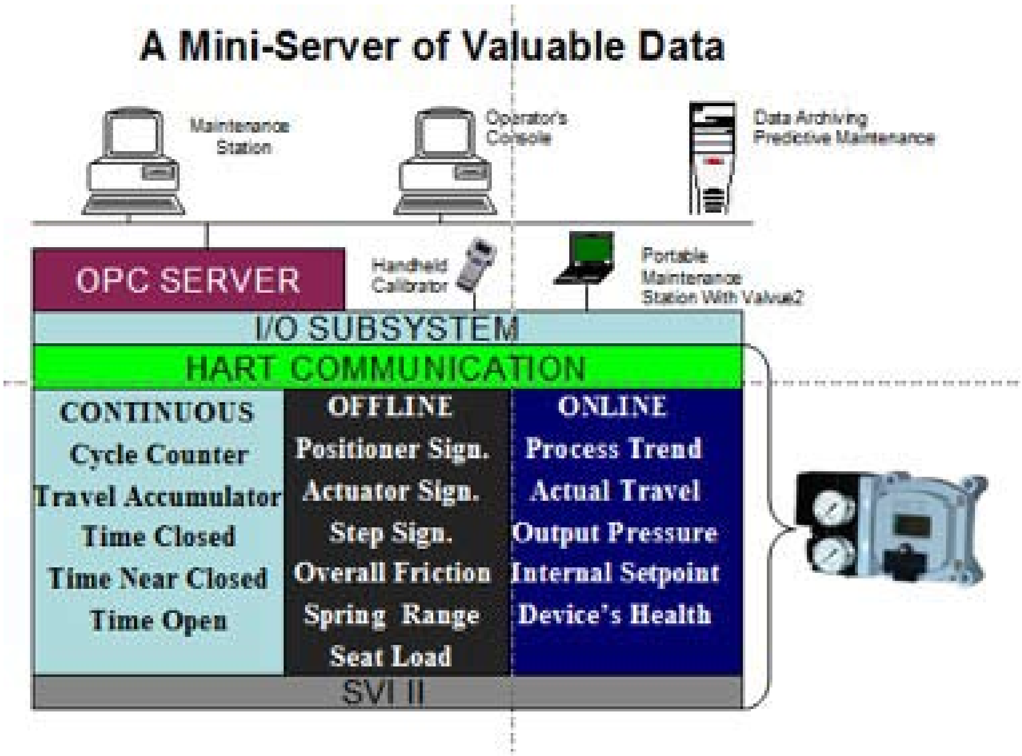


Figure 9 – Example of system Integration into plant network.

Benefits to the MOV Periodic Verification Programs from Other AOV/MOV Industry Initiatives

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Introduction

The Joint Owners Group (JOG) Motor Operated Valve (MOV) Periodic Verification (PV) Program was completed recently [1*] to help nuclear power plants address the U.S. Nuclear Regulatory Commission's (NRC) Generic Letter (GL) 96-05, "Periodic Verification of Design Basis Capability of Safety Related Motor Operated Valves." With 95% participation from the U.S. nuclear power plants that contributed to repeat testing of 176 MOVs under flow and differential pressure (ΔP), the data for this comprehensive program resulted in a detailed technical approach to address the Generic Letter 96-05 recommendations.

Implementation of the JOG MOV PV approach requires a systematic review and classification of each MOV to determine how it is to be tested. For gate valve and butterfly valve applications that are susceptible to increases or variations in required thrust or torque, the users are to add margin allowance or to perform ΔP tests to verify that the performance is stable. Alternatively, threshold values of disc-to-seat friction coefficients for gate valves, or bearing friction coefficients for butterfly valves above which increases or variations will not occur (as specified in the JOG MOV PV approach), can be used to determine the valve requirements. Appropriate allowances for actuator degradation also need to be included in the calculation of MOV margin. The JOG MOV PV approach states the following factors that should be taken into account to determine the "adjusted" actuator output: test equipment inaccuracy, torque switch repeatability, spring pack relaxation, rate-of-loading, and stem lubricant degradation. The JOG MOV PV approach specifies a periodic verification interval for static testing that ranges from 2 years to 10 years based on margin and risk ranking of the MOV. The JOG MOV PV approach requires that MOVs determined to have a negative margin after taking into account the specified allowances for degradation (a) can be dynamically tested every 2 years, or (b) can have their setup modified such that margin is positive.

This paper describes three margin improvement approaches which, based upon application specific MOV evaluation, can provide relief from dynamic testing to static testing, as well as relief by extending the static testing interval in accordance with the JOG MOV PV criteria. These margin improvement approaches are based on (1) validated torque/cycle life prediction models for Limitorque actuators [2, 3], (2) actuator testing on a specially engineered torque test stand [4], and (3) advanced butterfly valve models that have been recently developed to more accurately determine torque requirements [5, 6, 7]. A brief overview of the background of the programs that led to these technical approaches is presented first, followed by application examples/plant experience.

Approaches For Mov Margin Improvement

1. Actuator Torque Capability Increase by Validated Torque/Cycle Prediction Model

Background of Generic Thrust Rating Increase Program

During implementation of NRC IE Bulletin 85-03 and Generic Letter (GL) 89-10, utilities performed tests to verify that the torque switches and limit switches in the safety related MOVs were set properly by using MOV diagnostic test equipment. Many plants discovered that some of the MOVs were being cycled under thrust loads that exceeded manufacturer's ratings. To address this industry issue, a comprehensive, joint utility program was conducted by Kalsi Engineering, Inc. (KEI) from 1990 to 1994 that resulted in generic increases in thrust ratings for the Limitorque SMB-000, -00, -0, -1 and SB-00, -0, and -1 actuators [2]. Figures 1 and 2 show the details of the special qualification test fixtures used in this program. One of the key features of these test fixtures was that the MOV stiffness was simulated by disc spring stacks to ensure that the rotating components within the actuator would be subjected to similar cyclic loads, fatigue and wear that they experienced during actual operation.

The actuators were tested at 200% of the rated thrust, 4,000 opening/closing cycles, and margins based upon ASME Code Section III, Division 1, Appendix II [4] were applied to determine the allowable thrust for 2,000 cycle plant life. This permitted a generic increase in thrust up to 140% of the original Limitorque ratings provided the applicability limits defined in the Limitorque Technical Update 92-01 [10] were met. Thrust levels up to 162% of the rated thrust were permitted to the sponsoring utilities based upon additional application constraints and criteria specified in the proprietary reports [2]. Most U.S. nuclear power plants benefited from these "increased thrust capabilities" by avoiding actuator replacements during NRC GL 89-10 MOV program implementation.

Increased thrust ratings for Limitorque actuators can also be used during the implementation of the JOG MOV PV program if actuator thrust is found to limit the MOV margin.

MOV Application Specific Actuator Torque Increase

During each MOV opening/closing stroke, the thrust carrying components of the actuator (e.g., the actuator housing, housing cover, housing cover bolts) are subjected to one load stress cycle. In contrast, torque train components within the actuator (e.g., worm, worm shaft) are subjected to multiple load/stress cycles of varying magnitude for each MOV stroke as shown in Figure 3. The number of cycles, and the magnitude of the load/stress variations seen by the torsional components for each stroke, depends upon the actuator configuration (e.g., overall gear ratio, worm ratio, stem thread lead and pitch), the MOV stiffness (determined from the static thrust trace obtained during diagnostic testing), and the load profile during the stroke (static or dynamic ΔP). Therefore, unlike the actuator thrust components for which it is possible to provide a generic increase in thrust rating, torque rating increase depends upon the MOV configuration and application, and requires an MOV specific evaluation.

Under the joint utility Limitorque actuator rating increase program, one of the key developments was a validated analytical model, based on first principles, for predicting life of the torsional components. Torsional fatigue life prediction of the actuator components is complex, and is described in more detail in an earlier paper [3]. The model computes all pertinent stress components and their variations as a function of the loading ramp during an MOV stroke. The cumulative damage and fatigue life due to stress cycling under varying alternating stress and mean stress components is computed by using classical fatigue analysis methods. The methodology was implemented in a computer code, LTAFLA (Limitorque Actuator Fatigue Life Analysis Program) and validated against actual test results obtained from five different actuator sizes under a range of torque levels up to 140% of the rated torque. To determine the allowable design life in the actual plant applications, a margin based upon the ASME Code [8, 9] recommended in the methodology [3] is applied.

The software has been recently upgraded and validated to allow a more accurate evaluation of cumulative fatigue damage during a dynamic ΔP MOV closure stroke, for which the load does not increase in a linear ramp, as in the case of a static stroke [12]. The upgrade also includes thrust-rating increase tables and enhanced user friendliness. The new version, called LiFE (Limitorque Fatigue Evaluation), is compatible with Windows 2000/XP [13, 14].

Plant Example

During implementation of the JOG MOV PV program methodology, one of the U.S. nuclear power plants determined that a number of wedge gate valve actuators would have to be replaced with larger size actuators due to the increase in thrust requirements. The application specific details are provided below:

- Charging system isolation valves (10)
- 3" Anchor Darling Double Disc gate valve, Class 1500
- Limitorque SMB-00 Actuator
  - Thrust rating: 14,000 lbs [pounds force]

  - Torque rating: 250 ft-lb [foot-pounds]
  - Worm set Ratio: 45:1
  - Overall Gear Ratio: 49:1
- Stem: 1.25” diameter, 1/3” pitch, 2/3” lead

The diagnostic thrust traces from static and dynamic strokes for this MOV are shown in Figures 4 and 5. Design basis dynamic requirements for these valves prior to JOG implementation and after implementing JOG MOV PV requirements, options considered and results of torque evaluation above manufacturer’s published ratings, are summarized below:

Pre-JOG Requirements:

- Thrust: 15,060 lbs (exceeds 14,000 lbs rating, justified by Limitorque

  Technical Update 92-01)
- Torque: 250 ft-lb

Post-JOG Requirements:

- Thrust: 18,000 lbs (exceeds 14,000 lbs rating, justified by Limitorque

Technical Update 92-01)

- Torque: 300 ft-lb (exceeds 250 ft-lb rating)

Post-JOG Options to Address Torque Requirements:

Option 1: Replace with SMB-0 actuators

Will require seismic re-evaluation and snubber upgrade

Option 2: Evaluate allowable torque for torque train components based upon validated torsional fatigue life methodology, LTAFLA/LiFE

Post-JOG Evaluation Results from Torque Train Evaluation by LiFE:

- Static stroke evaluation results
  - Allowable thrust cycles: 2,000
  - Allowable torque cycles: 2,000
- Design basis dynamic stroke evaluation results
  - Allowable thrust cycles: 2,000
  - Allowable torque cycles: 59 (only one design basis stroke required)

Conclusion

This MOV specific application evaluation showed that under torques exceeding rating, the existing SMB-00 actuators can satisfy the cumulative static stroke cycles and dynamic stroke cycles under design basis requirements for this group of MOVs. The estimated cost savings exceed $400,000 (replacement cost for 10 actuators) plus the additional cost associated with seismic evaluation and piping support/snubber changes, which can be very significant.

In addition to the SMB models of Limitorque actuators, the torsional fatigue life prediction methodology for the worm and worm gear validated under the Limitorque actuator rating increase program is also applicable to HBC actuator models. This can provide a margin improvement in the butterfly valve MOV evaluations while implementing the JOG MOV PV program requirements for which both the bearing friction and seat torque degradation are taken into account.

2. MOV Actuator Test Stand

Background of Test Stand Development

A special MOV actuator test stand was designed by KEI in 1994 to overcome the limitations of the earlier designs identified by the industry users. KEI worked closely with Duke Power Company who had several years of experience in testing MOV actuators on their original torque test stands to verify that the actuator was assembled correctly, detect any problems/anomalies, and determine actuator capability after maintenance. Duke Power Corporation originally started to test each Rotork actuator on a test stand after disassembly/maintenance because it was a requirement by the manufacturer (Rotork Corporation). They found that this procedure was very effective in identifying and eliminating assembly problems which would have affected the MOV performance before installation in the plant. Based on increased productivity and reduced dosage by detecting MOV actuator problems off-line, Duke Power extended this testing approach to include all of their Limitorque actuators.

The following areas of improvements in the torque test stands were identified by Duke Power before the design of the new test stand was started by KEI using a clean slate approach:

(1) Accurate control of a variable torque applied in a specified ramp (time and magnitude) by the brake (dynamometer) from the lowest to the highest torque delivered by the actuator sizes to be tested (e.g., Limitorque SMB-000 through SMB-2).

(2) Applying upward or downward thrust to the stem nut (to simulate stem compression or tension) while verifying actuator performance and its output capability. Even though the actuator manufacturers had considered this effect to be negligible, Duke Power and other industry testing, including Comanche Peak [14] had shown that the actuator output torque is affected by the magnitude and direction of stem thrust.

(3) High accuracy in torque measurements over the entire range from the smallest to the largest actuator to be tested.

(4) Ease of calibrating the torque and load sensors.

(5) Minimizing the set up, testing, and removal time.

A number of conceptual design alternatives were evaluated which led to the final design approach. Duke Power was involved in the design reviews and the evaluation of the first prototype. Design refinements identified from this prototype testing and plant experience were implemented in the final designs, and six units were supplied to Duke Power. Figure 6 shows the cross-sectional details and key features of the Actuator Test Stand.

Actuator Test Stand Capabilities

Load Range

- Torque: 12.5 ft-lb to 3,000 ft-lb, bi-directional
- Thrust: 0 - 75,000 lbs. tension or compression

Torque Resistance

- Industrial pneumatic multiple disc brakes (number of discs engaged dependent on required test range)
- Ramp time - variable (15 seconds minimum)
- Target torque - variable and controllable to larger of ± 1% of target or ± 2 ft-lb

Data Input, Output, Display and Accuracy

- Electronic control panel with touch screen interface
- Operator speed, ± 0.5 rpm [revolutions per minute]
- Thrust load, ± 2% of full scale of the pressure transducer rating
- Operator output torque, ± 0.5% of full scale of the load cell rating
- Dynamometer, ± 0.5% of full scale of the torque cell rating
- Spring pack displacement - sensor dependent
- BNC connector terminals provided for interface with typical data acquisition systems

Plant Experience and Benefits

KEI Test Stands have been used for over 10 years at Duke Power (which has six test stands; two at each of the McGuire, Catawba, and Oconee plants), Bruce Power (which has two stands at Station A and Station B), and Pickering plants to verify performance of each actuator after maintenance. The KEI Actuator Test Stands have provided significant advantage in reducing the maintenance costs and radiation exposure by detecting and fixing actuator problems before installation.

Furthermore, tests performed by the users over the last 10 years have demonstrated 10% - 40% additional capability over the calculated values for the Limitorque actuators (including Technical Update 98-01 [15, 16]) or published values for Rotork actuators [4]. This additional actuator capability can result in a larger MOV margin which can reduce the frequency for periodic verification testing required in accordance with JOG MOV PV methodology.

3. Advanced Butterfly Valve Models

Background

Earlier papers [5, 6, 7] describe limitations of the EPRI MOV Performance Prediction Methodology (PPM) Butterfly Valve Models [17] that were discovered during implementation of the Air Operated Valve (AOV) programs by U.S. nuclear power plants. Specifically, it was found that PPM predictions can have excessive conservatism for certain disc types which can lead to “apparent” negative margin concerns. Tests performed to support the development of EPRI MOV PPM were limited to incompressible flow and generic disc shapes for symmetric and single off-set designs. The generic shapes did not include certain geometric features/variations that are present in different manufacturers' designs (e.g., flat or concave recesses on the flat face of single off-set disc, bosses/projections on the shaft side disc face). For compressible flow, the EPRI PPM model was validated against test data from NRC/INEL testing performed on three valves [18, 19] which basically had similar design features.

A comprehensive program was conducted by KEI to overcome these limitations by performing tests on a much larger matrix of disc shapes which included variations found in the manufacturers' designs and under both incompressible and compressible flow conditions [5, 6, 7]. The KEI advanced models developed under this program provided position dependent accuracy and eliminated excessive conservatisms of the earlier models. Only for compressible flow applications and for certain disc shapes and flow direction, the EPRI MOV PPM model was found to be non-conservative [20].

The advanced butterfly valve models can provide a margin benefit in the MOV applications by reducing the total required torque.

Plant Example

Figure 7 shows the comparison of required torque predictions for an 18-inch double-offset disc containment purge valve (with shaft downstream orientation), to close under design basis LOCA conditions. The AOV actuator was a Scotch-Yoke type with spring return to fail close the valve. The minimum actuator output available from the actuator at various stroke positions had been provided by the manufacturer and verified by the plant engineers. EPRI MOV PPM software indicated a large negative margin throughout the stroke. The advanced butterfly valve models incorporated in the KVAP software, along with an extensive database of torque/flow coefficients, provided a significant reduction (over 40%) in torque requirements and a positive margin throughout the stroke. This eliminated the need for plant modifications that were being planned for 8 valves in this group of Category 1 AOVs.

These margin benefits from the advanced butterfly valve models are also applicable to MOV applications, and can provide a relief in the frequency of PV testing required by JOG MOV PV program.

Conclusion

This paper described three different approaches that offer the potential to address low or negative margin issues encountered when implementing the JOG MOV PV program. Plant examples included in this paper show that application of these advanced, validated models has been successful in eliminating equipment modifications, thus resulting in significant cost savings while ensuring reliable operation under design basis conditions.

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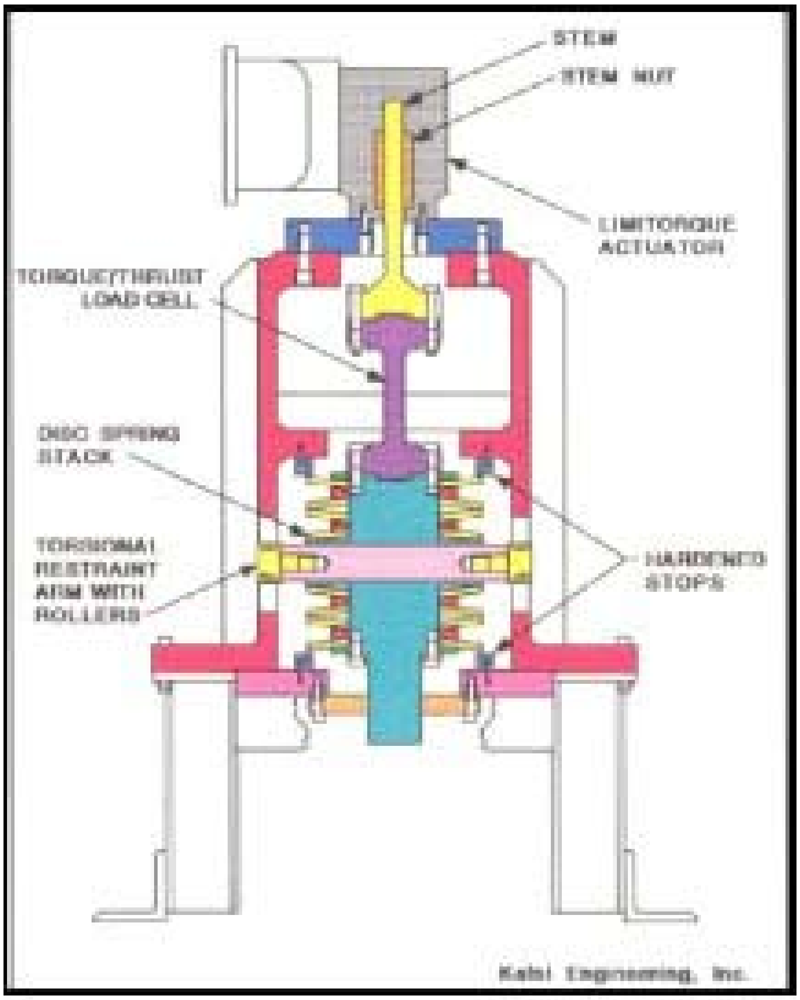

Figure 1 - Details of Fixtures Used for Limitorque Actuator Cycle Testing Under High Loads

Figure 2 - Cyclic Testing of Limitorque Actuator in Progress

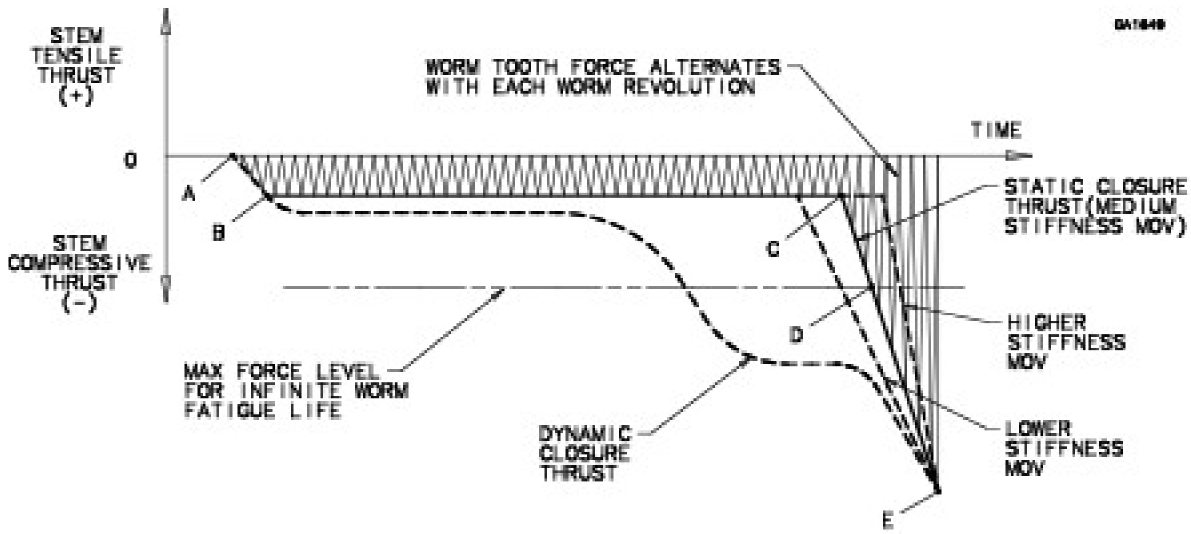

Figure 3 - Thrust Components are Subject to Only One Force Cycle for Each MOV Stroke; In Contrast, Torque Train Components are Subject to Multiple Cycles of Variable Force Magnitude for Each MOV Stroke Which Affects Their Fatigue Life Differently

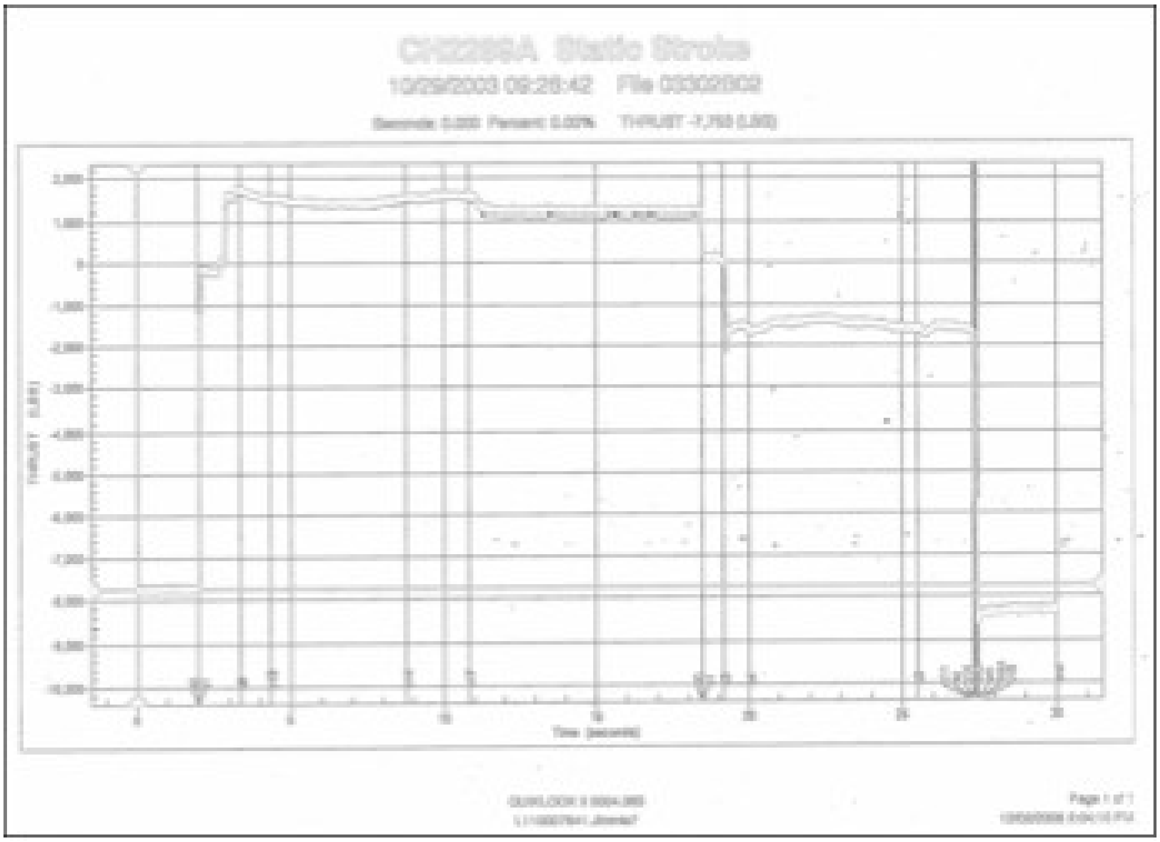

Figure 4 - Static Trace of the 3” MOV Gate Valve

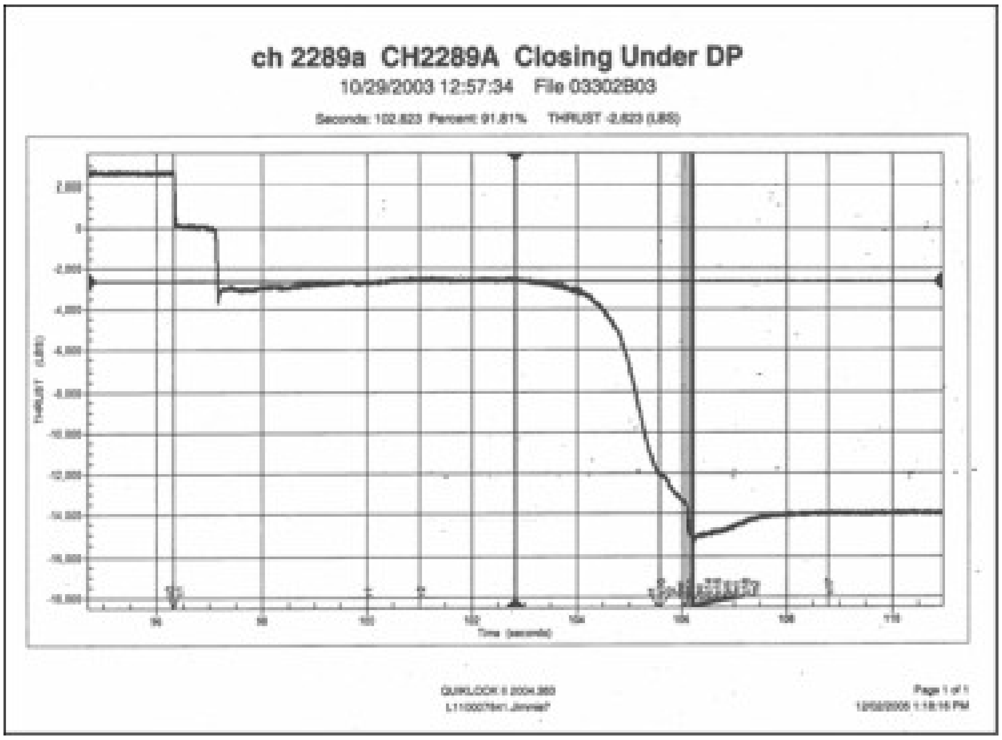

Figure 5 - Dynamic Trace of the 3” MOV Gate Valve

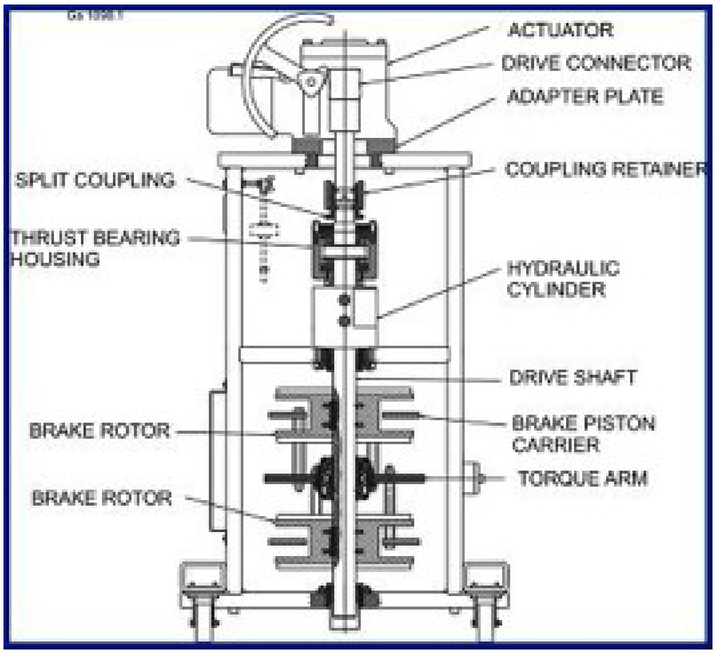

Figure 6 - MOV Actuator Torque Test Stand (with Tension/Compression feature) for Determining Actuator Capability

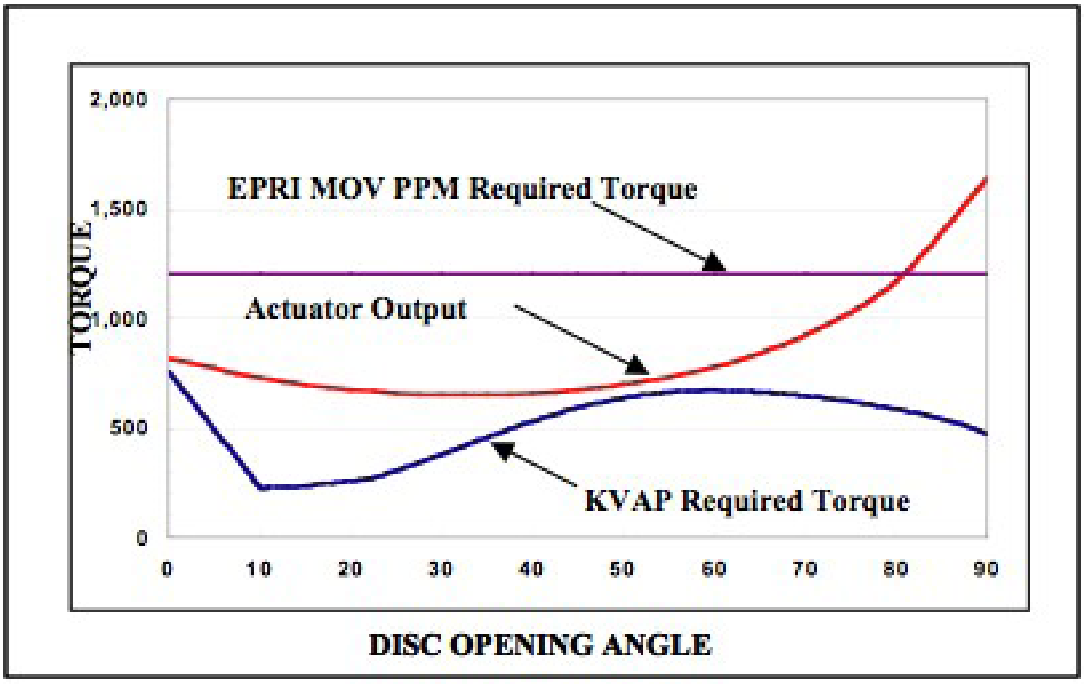

Figure 7 – Margin improvement achieved by use of KVAP models in a compressible flow (containment purge) AOV application at a plant

HIGH PRESSURE VALVES OF THE EPR (EVOLUTIONARY POWER REACTOR)

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Abstract

The high pressure valves of the EPR benefit from the experience acquired on German Konvoi and French N4 plants, and significant improvements were brought to the design of the most sensitive high pressure valves. This paper presents the valves which have the most advanced technologies and especially the pressurizer pilot operated safety relief valves and their new spring loaded pilot SIERION. The valves of the Safety Injection / Heat Removal Systems are also described considering their importance relative to the safety and the maintenance costs of the plants.

1.0 Introduction

The first EPR which is built by Areva NP in Olkiluoto in Finland and which is planned in France in Flamanville is the result of French German co-operation. It incorporates the best technologies of high pressure nuclear valves which can be found at the present time on the American and European markets. Most of them were already used either in most modern plants in France (N4) or in Germany (Konvoi), as no fundamental changes are to be expected in the relatively mature technologies of nuclear valves, taking into account the operating and regulatory constraints. However some, such as the pressurizer pilot valves are quite new. Also some improvements are made to existing designs, to take into account the operating experience feedback and to improve reliability and maintainability.

In this paper, we will present the technologies of the main high pressure valves of the EPR Nuclear Steam Supply System (NSSS) implemented for the Olkiluoto 3 project, that is to say

- The pilot operated pressurizer safety relief valves (PSRVs)
- The severe accident dedicated valves
- The pressurizer spray control valves
- The Safety Injection (SIS) / Heat Removal System (RHR) isolation valves

2.0 Pilot Operated Pressurizer Safety Relief Valves

2.1 Principle of operation of the PSRVs

The PRSVs are pilot operated, operating according to the depressurization principle (see Figure 1). In normal operation, the main body is pressurized with pressure above the seat and on either side of the disc control piston, which is located inside a control cylinder. The tightness between the piston and the control cylinder is insured by piston rings, while borings of small diameter through the control piston allows the draining of condensate water and pressurization of the control volume. As the surface of the control piston is greater than that of the seat, the PSRV opens when the control volume pressure falls to about 60% of the system pressure, following the opening of the pilot, whose discharge area is greater than that of the control piston borings.

After closure of the PSRV' pilots, the pressure in the control volume builds up, via the control piston borings, up to about 90% of the inlet pressure. The disc moves down very quickly owing to the difference between the pressure in the valve body and the pressure below the disc. The closure is helped by a return spring whose main function is to insure tightness when venting the primary system under vacuum.

The main characteristic of this PSRV is its ability to discharge any type of fluid without any instability and minimal changes in performance: Ref [1]. The stroke times which are very short are balanced by larger dead times needed for depressurization or re pressurization of the PSRV control volume. Typical opening values at 15 Megapascal (MPa) are :

- dead time : 300 milliseconds [ms] (saturated steam), 150 ms (25°K sub cooled water)
- stroke time 25 ms (saturated steam), 80 ms (25°K sub cooled water)

This PSRV can be activated by up to 4 pilots mounted in parallel, comprising the spring loaded pilots for self actuation: safety function, and solenoid or motor operated pilots for remote actuation: relief functions.

As the PSRV by itself is very reliable either at opening or closure, the prevention against failure to open is obtained either by other PSRVs or by two pilots in parallel.

The prevention against failure to close is done by installing the electric powered pilots in series. The spring loaded pilots can be isolated by two manual block valves.

2.2 Spring loaded pilot SIERION

The new SIERION pilot was developed and tested by Areva NP in its facilities in Erlangen (Germany). It mainly consists of three sub assemblies (see Figure 2).

The first subassembly is the so-called “converter assembly” in which the system pressure is converted to linear motion of the converter rod. System pressure is applied via a pressure sensing line to the interior of the converter bellows. At the bottom of the converter, a preloaded Belleville disc springs stack exerts a force that counteracts the system pressure acting on the hydraulic cross section of the converter bellows.

The second subassembly is the so-called “pilot assembly” which mainly consists of a hollow pressurizing piston moving inside two chambers delimited by bellows. This pressurizing piston is provided on the inside with a seat for the « release» disc and on the outside with a disc for the «refill» seat. The converter rod is guided inside the pressurizing piston and acts on the release disc. The inner annular space around this rod provides an exhaust path for venting the chambers outside. The lower end of the pressurizing piston has latches that engage matching latches on the converter.

The third subassembly is the so-called “actuator assembly” which, like the main valve, is a pilot valve with control chamber, check disc, seat and backseat; and which opens or shuts the control line connected to the main valve control volume.

Pilot Valve Opening

At system pressures of 15 MPa or less, both the refill/release discs and the check disc are in the positions shown on Figure 2. The spring connected to the bottom of the converter is pulling it downwards so that it is resting on a bottom support.

Both the pressurizing piston and the release disc of the pilot assembly are at their lower limits of travel, with the pressurizing piston resting on a mechanical stop and the release disc on its seat. As a result, the control chamber above the plug of the actuator assembly is exposed via the pressure sensing line to the pressure currently prevailing in the system, this serving to hold the plug in its seat. The pilot valve is thus closed. A key feature of this valve design is that all closure elements (piston, disc and plug) are generally seated by the pressure of the system fluid, thus ensuring high specific seating forces.

If the system pressure increases, the hydraulic force inside the converter bellows also increases, compressing the disc spring and causing the converter to move upwards. As a result, the rod rises until it comes into contact with the underside of the release disc. However, due to the force exerted by the spring of the release disc and by the hydraulic force acting on it, the rod is initially unable to unseat the relieving disc. This causes the pressurizing piston to be pushed upwards into its upper seat, taking the release disc along with it.

When the force acting on the converter becomes sufficiently large, the relieving disc unseats and travels its full stroke in a single movement. Since this depressurizes the control chamber above the plug in the actuator assembly, the plug moves to its upper limit of travel (backseat), thereby opening the safety valve. The actuator assembly in the open position is shown on Figure 3.

Pilot Valve Closure

When the system pressure decreases, the converter rod moves down, thus enabling the release disc to reseat under the sole force of its spring. When the pressure has reached a lower level, its latches cause the pressurizing piston to unseat and, in consequence, allows the re-pressurization of the actuator assembly chamber, and the reseating of its plug. The PSRV returns to its closed position as a result of its control volume becoming pressurized again via the borings in its plug.

The main advantages of this new pilot, compared to the pilots of similar PRSVs already installed in France or in Germany are the following :

- Capability to be used with safety valves that operate according to the depressurization principle as well as safety valves based on the pressurization principle, without any of its moving parts having to be modified.

- Parallel pilot valve configurations possible permitting compliance with existing codes and standards regarding redundancy, and permitting remote operation (relief function) via solenoid operated valve (SOV) or motor operated valve (MOV) pilots.

- Very stable operation and constant performances for all types of discharged fluids: superheated and saturated steam, saturated and sub cooled water, steam/H2 mixtures, contrary to the safety valve type pilots (see definition in Ref [2]).

- High reproducible accuracy of set pressure, and negligible dead time so as to be independent of the pressure gradient in the system.

- Very good seat tightness due in particular to operation in warm condensate water and functional capability in the event of postulated leakages of the pilot and of the PSRV itself.

- No connecting lines between the pilots and the PSRV reducing the risk of spurious PSRV opening and the cost of installation.

This pilot was installed for the first time in mid-2005 on PSRVs working on the loading principle in the Swiss Goesgen plant.

2.3 Electric powered pilots

Two types of electric powered pilots are available.

The double SOVs which will be installed in OL3 consist of two single SOVs in series with a solenoid energizing to open. Its force is opposed by Belleville springs which press on the solenoid spindle and on a pilot disc (see Figure 4). When the solenoid is actuated, the force above the pilot disc is removed and it opens as the pressure acts under the disc. The pressure above the main check disc is relieved and the pressure difference below and above causes it to open, which induces the opening of the main PSRV as its control volume is depressurized via the main check valve.

The advantage of the SOVs is their fast acting time which allows using them also for Cold Overpressure Protection, when the RHR is isolated. Their drawback is the need of electrical supplies of enough capacity for keeping them open for a larger time than the one (2 hours in general) allowed by the normal plant batteries.

It is also possible to install MOVs which will stay in position in case of loss of electrical power. Like the SOVs, the MOVs are installed in series and as their thrust is much larger than that of the SOVs they pull directly on the plug of the pilot valve, without the need of an internal pilot. A compact valve actuator SIEKA, qualified according with the KTA Nuclear Safety Standard 3504 can now be provided by Areva NP for installation on valves DN 15 to DN 100 mm.

2.4 PSRVs installation

Fig. 5 shows the installation of the 3 PSRVs on the top of the pressurizer. The PSRVs are directly welded on nozzles on the pressurizer, with a flange at the exhaust.

To prevent hydrogen leakage a hot loop seal is created in front of the PSRVs, owing to scoops welded to the pressurizer inside cladding in front of the PSRVs inlet nozzles. The complete filling of the inlet PSRV piping is insured by condensation of the steam on the colder walls.

Strong natural circulation maintains a homogeneous temperature of the loop seal fluid. This temperature is high enough (> 300°C) to avoid too large discharge forces on the exhaust pipe and supports at the first opening of the PSRV due to water seal ejection.

3.0 Severe Accident Dedicated Valves

One salient characteristic of the EPR is the presence of a dedicated circuit to be able to depressurize the reactor coolant system (RCS) down to a pressure below 2 MPa, to prevent the risk of loss of containment leak tightness following the failure of the reactor vessel after a core melt.

Although the required flow rate of 900,000 kilogram/hour [kg/hr] (1,984,160 pounds/hour [lb/hr]) at 17.6 MPa was the same as that of the 3 PSRVs, they could not be used for the following reasons.

- First, the Finnish Regulations (YVL rules) required a redundant dedicated circuit for coping with severe accidents.
- But also the requirement was to be able to open for a fluid temperature in the pressurizer below 600°C and then to stay open, even for fluid temperatures up to 1000°C.

Indeed, in one severe accident scenario, called “late re-flooding,” the primary pressure may drop to very low values and the PSRVs normally close at a pressure of about 0.5 MPa owing to their return spring.

If water is then sent in the vessel on the molten core, the pressure builds up above 2 MPa if the PSRVs do not reopen.

It would have been very difficult to guarantee their reopening after being heated up to very high temperature while open, knowing that the PSRVs and their electric pilots are qualified only to fluid temperature up to 360°C.

That is why the severe accident line in Olkiluoto will be fitted with an arrangement of two groups in parallel of two MOVs in series: the MOV closer to the pressurizer dome is a parallel slide valve and the other a globe valve. The basis of this choice is :

- Diversity of technologies, as required by YVL rules
- Proven designs, operating up to 600°C in fossil fired supercritical plants
- Guarantee of staying open, with a non-reversible stem nut, even after failure of the electric actuator.
- Possibility of closure under full pressure differential in case of spurious opening.

As said before, the technology of the valves is not original, with however some specific characteristics :

- Stellite is kept as hard facing on the seats,
- Pressure seal bonnet will be used instead of bolted bonnet. Indeed, due to the high body temperature before and during opening, excessive expansion stresses are difficult to avoid and to master on a bolted connection. In this situation, pressure boundary integrity cannot be guaranteed especially as the thread material resistance decreases a lot with temperature and as leakages could jeopardize the actuator operability by convective and radiant heat transfer.
- A thermal shield (plate) will be fixed above the bonnet to avoid too large heat up of the actuator and of the stem/stem nut connection.

- In normal operation and also before opening in nearly all severe accident scenarios, the valves will be kept cold owing to a loop seal. The loop seal will help to keep the valves tight, even with increased H2 concentration in the pressurizer steam phase. That was already the case on previous layout designs of the pressurizer power operated relief and block valves.

4.0 Spray Control Valves

The spray control valves were air actuated ball valves in the actual Areva NP designed operating plants. Most of the French reactors operate with priority to the grid and turbine demands, and not as "base" plants, always at 100% nominal power (NP). As the pressurizer pressure is not constant, spray is frequently actuated as the load swings can reach +,-10% NP. Those numerous actuations induced ball wear (with release of cobalt, as the ball is coated with stellite) and also packing leakage. Therefore, on the EPR, modulating solenoid globe valves will be installed. They have numerous advantages compared to the old design such as :

- Very fast actuating time
- No packing and no risk of external leakage
- Very small wear and no need of stellite hard facing
- Electric actuation with its advantages compared to air actuators: high reliability, compactness, no need of air lines

Two manufacturers were able to supply those valves, both having nuclear references.

5.0 Valves Of Safety Classified Auxiliary Systems

The major change is the replacement of all air operated isolation and control valves (AOVs) by motor operated valves. In addition to increased reliability, the problem of containment pressurization due to spurious or normal air leakage resulting from the operation of control valves is solved. Furthermore, non-intrusive on-line diagnostic is now possible by monitoring the electric power by dedicated modules ("SIPLUG") installed in the valves' control cabinets – Ref [3].

In the auxiliary systems, priority has been given to globe valves up to 10". Indeed the requirement of absolute external tightness relative to radioactive water imposes the use of stem bellows which are best suited to globe valves with short strokes. For larger sizes or when dictated by customer demands for reasons of diversity, small available pressure drop or layout feasibility, wedge gate valves are installed. They are fitted with graphite live loaded packings and seats with iron based hard facing (Norem) instead of stellite to avoid contamination by cobalt 60.

As required by YVL rules, the valves were designed to be operable even after failure of torque and limit switches, corresponding to the stalled motor torque. Therefore, globe valves are fitted with Belleville springs above the stem nut to reduce the load due to actuator inertia while closing and to compensate for stem expansion due to heat up while closing in a hot fluid (see Figure 6).

The actuators are standard multi-turn actuators, qualified for the use inside containment and corresponding to the 1E qualification according to IEEE 382.

The voltage delivered by the power supply system: 400 VAC is controlled in the +5%, -10% range which allows limiting the size of the motors and the maximum delivered torque in case of limit or torque switch failures. Remote couplings between the actuator and some safety related valves, as on French plants, have been deleted, with hand wheels located directly on the actuators. Experience has shown than those remote couplings participated marginally to reduce the exposure during maintenance while decreasing significantly the reliability of the MOVs.

6.0 Conclusion

In conclusion, the technologies of the high pressure valves of the EPR Nuclear Steam Supply System (NSSS) are expected to increase the safety of the plant and to reduce the maintenance costs in spite of the increased number of safety related valves as a consequence of the installation of 4 safeguard systems trains.

The choice of electric actuators, even for valves of small sizes raises their operability margins while allowing the use of rapid on-line non-intrusive diagnostic techniques. Finally, the pressurizer PSRVs and their pilots incorporate all the knowledge acquired through both plant operation experience and test loop qualification, for more than 25 years since the Three Mile Island (TMI) accident.

References

[1] Investigations into the operability of original geometry pressurizer safety valves arrangement during hot blow down at full mass flow rate.

U. Simon, A. Knapp, W. Von Rhein, F. Agemar, W. Hofbeck, R. Puzalowski

Final Report – Research Project BMFT 1500 636/7 – KWU R917/86/005

[2] ASME NB 7000 : Overpressure protection – NB 7170

[3] NUREG CP 0152 Vol. 5: Proceedings of the eight NRC/ASME meeting on pump and valve testing – 1B141: Engineering Based Valve Testing and Evaluation.

Figure 1

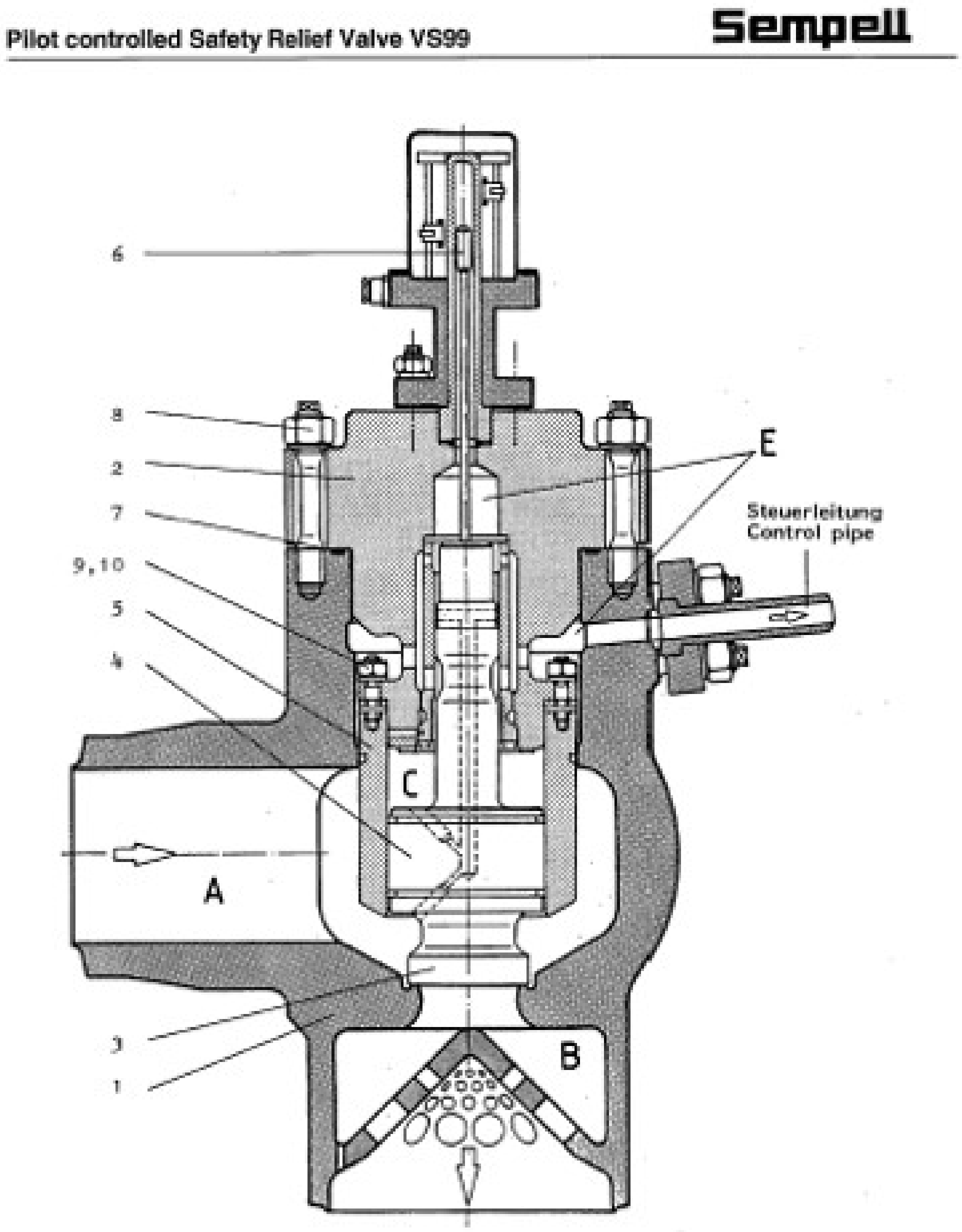

Pilot controlled Safety Relief Valve VS99
Sempell
6
8
2
7
9,10
5
4
3
1
E
Steuerleitung
Control pipe
C
A
B

Figure 2

Sierion Pilot Closed

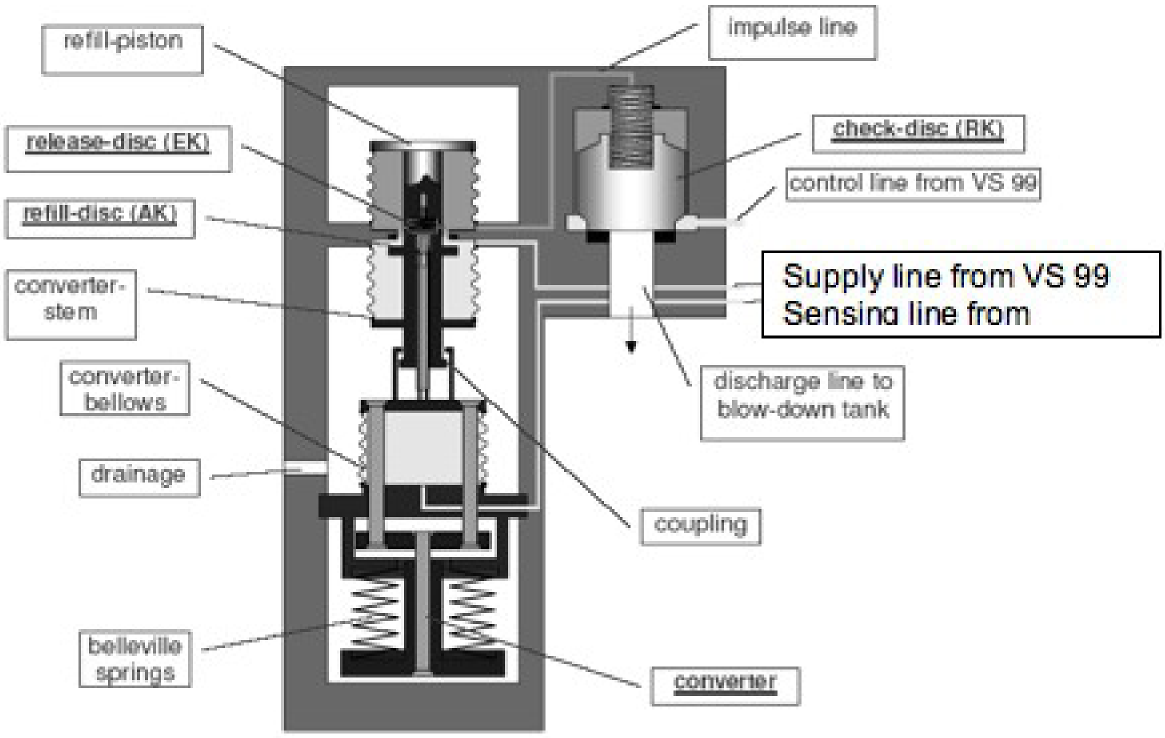


Figure 3

Sierion Pilot Open

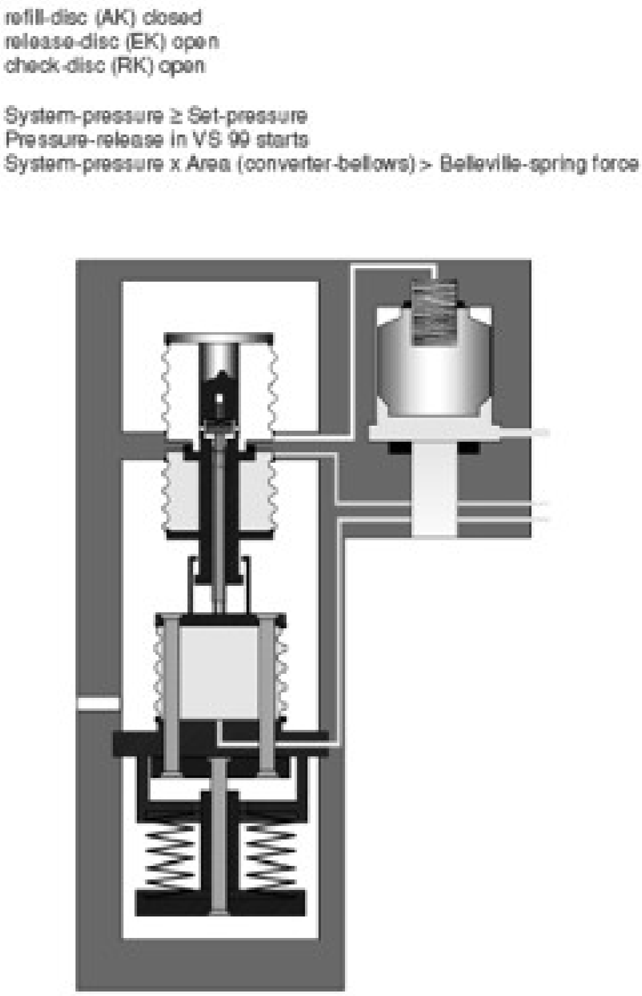


Figure 4

Solenoid Pilot

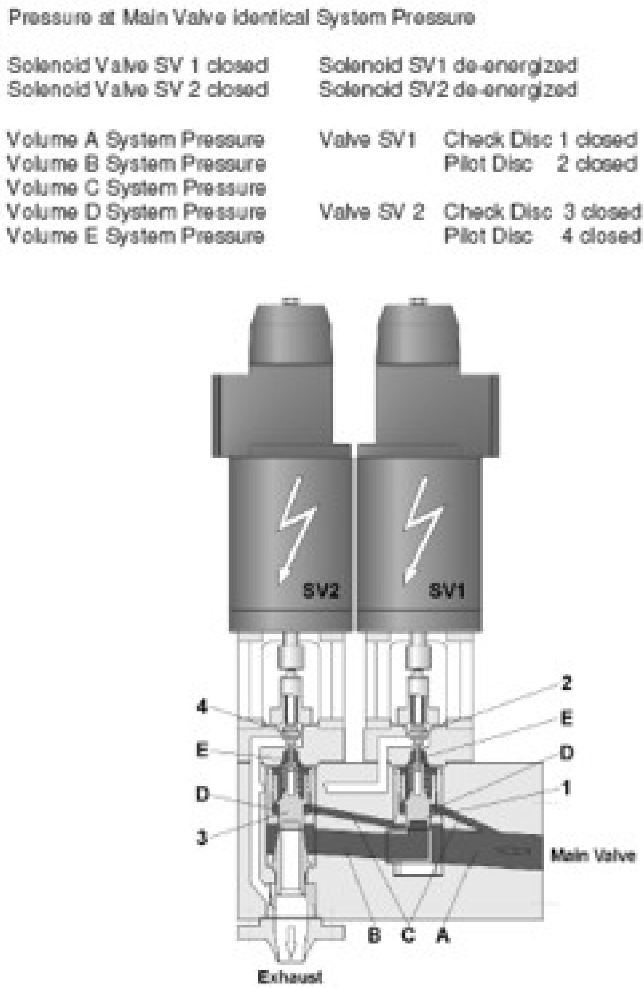


Figure 5

PSRVs Installation

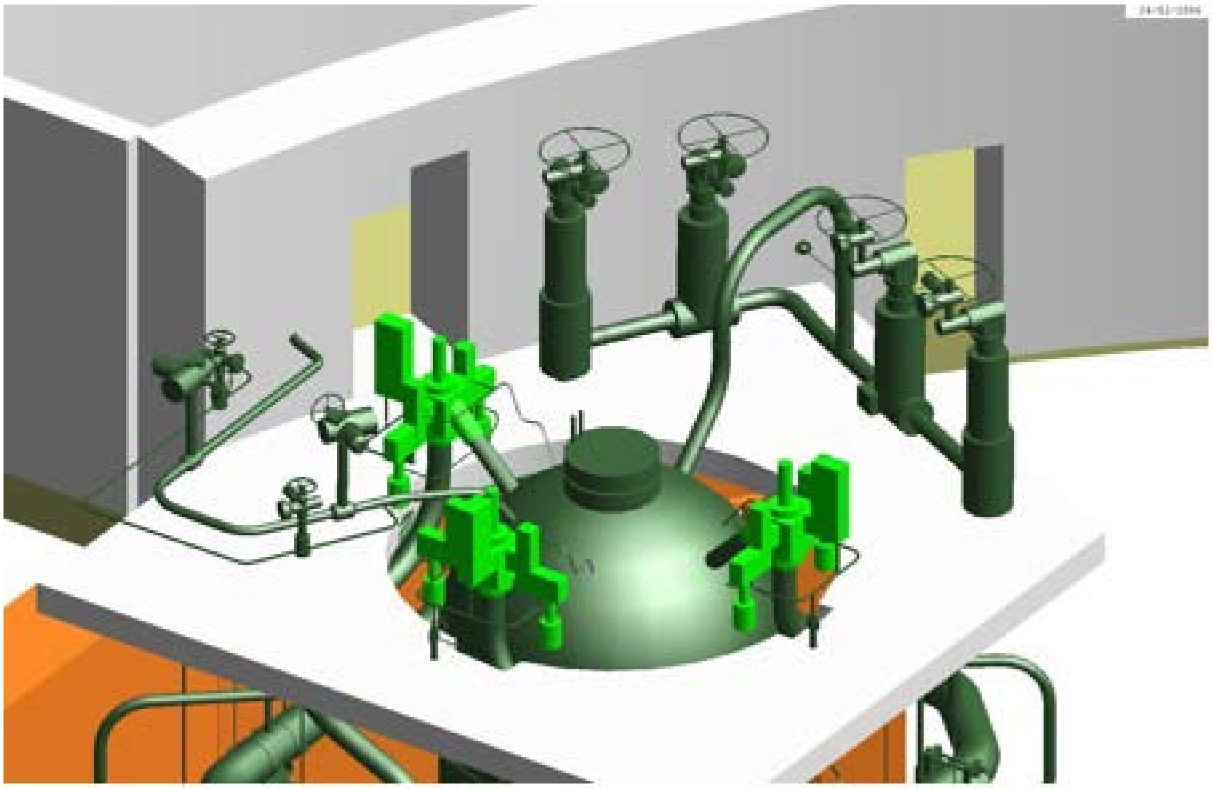
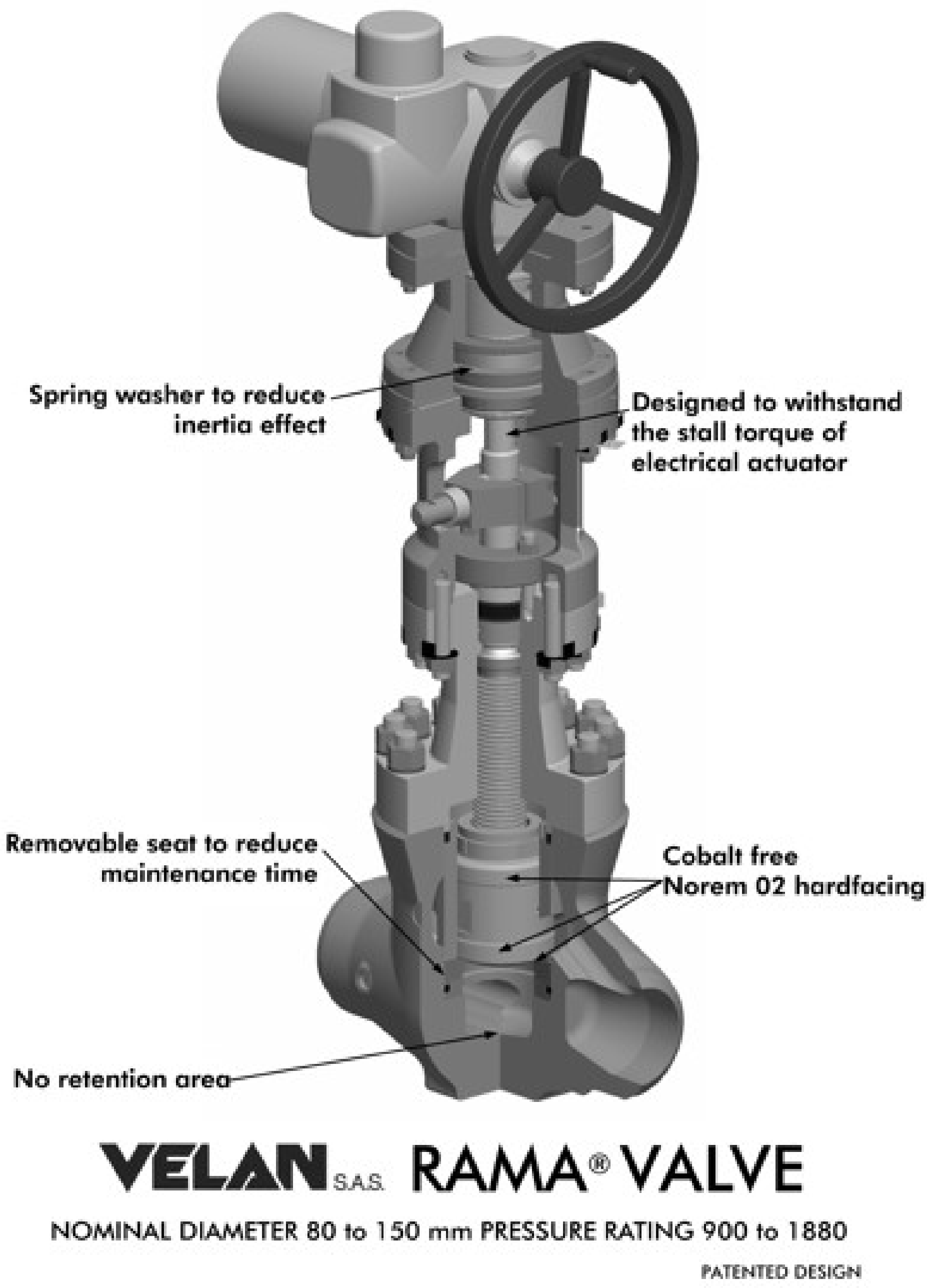


Figure 6


Spring washer to reduce
inertia effect
Designed to withstand
the stall torque of
electrical actuator
Removable seat to reduce
maintenance time
Cobalt free
Norem 02 hardfacing
No retention area
VELAN S.A.S. RAMA® VALVE
NOMINAL DIAMETER 80 to 150 mm PRESSURE RATING 900 to 1880
PATENTED DESIGN

Valve Maintenance & Operation Solutions for NSSS Systems

Part 1: C. Dupill & L. Dupill, Dupill Group, LarsLap USA
L. Larson & B. Carlsson, LarsLap

Part 2: C. Edwards, Vermont Yankee
C. Lampitoc, Triumph Controls, Inc.
L. Dupill, Dupill Group

Introduction:

The demand for safe operation of nuclear facilities is a critical issue. In support of safety issues and As Low As Reasonably Achievable (ALARA) regulations this paper, presented in two (2) parts, offers solutions to two issues of concern. Part 1 addresses the Smooth Edge Criteria on gate valve seat and wedge, particularly Motor Operated Valves (MOVs), utilizing a Radius Grinding Method to resolve this issue. Part 2 covers a specific problem regarding valve operation in an area of high radiation, specifically the Reactor Water Clean Up system at a Boiling Water Reactor (BWR) Plant, and offers a solution to valve operation in contamination thereby addressing ALARA regulations.

This presentation refers to and includes various excerpts from Electric Power Research Institute (EPRI) reports. Copies of these reports are available from EPRI.

ABSTRACT

Problem:

The first part of this paper addresses the need for a smooth meeting area on gate valve seats and wedges. When a gate valve is cycled, the media flow pushes the wedge into the seat causing the sharp inside and outside edges of the wedge to cut the seat as the wedge moves. Use of an actuator (MOV) magnifies this effect. Guide tolerances also contribute to the problem. Too much play causes inward tilting of the wedge, resulting in cutting actions on the valve seat by the wedge edges. These two factors result in major scratching damage to the seat and may cause the wedge to jam on the bottom area of the seat.

The second part addresses valve operation in areas that are difficult to reach, in contaminated and high radiation areas. The focus is on a specific application at Vermont Yankee Nuclear Power Plant. A ¾” isolation valve was being installed on a new pressure equalization line in the Reactor Water Cleanup (RWCU) System located in the holding pump room. This was an ALARA concern as valve operation would expose plant personnel to hot spots (1R/hr range) where the bypass was required. Rerouting the bypass piping to a lower radiation area would be very costly and would require considerable exposure to installation workers. These concerns pointed to the need for a different approach in designing the new bypass line.

Solution Summary

Radius grinding provides the solution addressing the “sharp edge” problem. As stated in EPRI Performance Prediction Methodology (PPM) it was recommended that the “sharp edges” on wedge, seats and guides should be broken during routine valve maintenance and/or inspections. A step by step Preventive Maintenance (PM) procedure will alleviate the cutting actions from the sharp edges. The radius grinding heads are manufactured with two angles as standard (additional angle available upon request) in steps and a ‘rounding’ segment to remove sharp edges on seats and wedges (Diagram 2). In order to reduce the tolerances in the guides, different methods may be used.

Installation of two (2) Remote Mechanical Valve Actuators (RMVA) outside of the RWCU Holding Pump Room provided the solution for Vermont Yankee. An RMVA was installed on the isolation valve for the pressure equalization line around the air-operated inlet isolation valve for each of the two RWCU filter demineralizers.

Conclusion

Utilizing the technique of radius grinding, damage can be eliminated if a smooth meeting occurs between the two seats of the valve. Radius grinding of the wedge and seat is the recommended method for proper entry/exit and seating of the valve. Vermont Yankee demonstrated that the use of the RMVA will reduce radiation exposure of plant personnel when dealing with valves located in a contaminated area.

Part 1 Sub Title:

Addressing the "Smooth Edge Criteria" on Gate Valve Seat & Wedge

HISTORY

In response to NRC Generic Letter (GL) 89-10 (1989), "Safety Related Motor Operated Valve Testing and Surveillance," EPRI initiated an MOV Performance Prediction Program. EPRI submitted Topical Report TR-103237, "EPRI MOV Performance Prediction Program Topical Report" (available through EPRI) for review by the NRC. On March 15, 1996, the NRC issued a Safety Evaluation (SE) documenting its review of the topical report and accepting EPRI's recommendations and methodology.

The EPRI MOV Performance Prediction Program findings for gate valve maintenance are:

1. The edge radii on disk, seats and guide slots are critical to gate valve performance and predictability.
2. Stellite friction coefficients increase with differential pressure valve strokes in cold water to a plateau level, stabilize quickly in hot water and decrease as differential pressure increases.
3. Gate valves with carbon steel guides and disk guide slots with tight clearance might fail to close under blow down conditions.
4. Many existing gate valve manufacturing and design processes and controls, and plant maintenance practices might contribute to poor valve performance.

   (EPRI by permission, MOV PPM, pg. 2, available through EPRI).

EPRI conducted testing on gate valves and found that "In Test No. 33, the sharp edge disk was immediately subjected to a severe load condition and a high valve factor was measured. In Stroke 6, the same load condition was used, but the disk had been previously subjected to lower loading conditions. A lower valve factor was observed due to the edge chamfering that occurred on the lightly loaded strokes." (Information contained in EPRI MOV PPM "Gate Valve Design Effects Testing Results", Test No. 33, available through EPRI).

EPRI determined that there was a significant improvement in performance in the presence of an appropriate chamfer on the gate valve disk and seat edge. "One of the most important results found in the testing was the presence of an appropriate chamfer at the edges of the disk and seat can provide a dramatic difference in performance, especially under severe (1800 pounds per square inch [psi]; 15 feet per second [ft/sec]) disk loading profiles. A direct comparison of results between Chronological Test numbers 54 and 80 show the improvement achieved by providing a 0.060" x 45° chamfer in Geometry 1 design. The valve factor with the sharp edges (Test 54) was found to be 0.70, with very severe damage to the disk and seat faces. With the chamfered edges (Test 80), a valve factor of 0.46 was measured, and only minor damage occurred at the edges of the chamfer near the 4 o'clock and 8 o'clock positions (Information contained in EPRI MOV PPM "Gate Valve Design Effects Testing Results", Test No. 54 and 80, available through EPRI). The actual magnitude of chamfer necessary to provide predictable performance and have low valve factors is dependent upon valve design (guide clearance and guide length), loading profile and valve size." (EPRI by permission).

Since the acceptance of the EPRI's recommendations and methodology in the 1990's, the industry now demands that there be no sharp edges on gate valve seats and disks. Chamfering can result in sharp edges as evidenced in Diagram 1; the stringent requirements of "Smooth Edge Criteria" for gate valves demanded a better solution to this problem.

Causes of Typical Damage to Gate Valves

Cutting action is the most common damage that occurs in the gate valve. When the valve is cycled, the sliding action from the disk across the seat scratches (Refer to Drawing 1 "Cutting Action") the seat at the 4 and 8 o'clock positions on both the disk seat and valve seat. Damage is directly related to the sharpness of the edges of the disk and seat ("sharp edge criteria") - the sharper the edges, the more severe the scratches. Due to the design of gate valves, cutting action scratches will occur. If correct seat lapping/grinding is not performed on a regular basis as part of a preventative maintenance program, these scratches will eventually become deeper and longer, creating a leak path across the entire seat. However, continual seat lapping/grinding will remove material from the seat and wedge and shorten the life of the valve. Slowing the actuator speed on an MOV can reduce scratching thereby extending the valve life.

Cv: Another critical factor affecting seat damage occurs in a pipeline system with a high Cv (600 Class and higher). Gate valves operating with high-pressure media are subjected to a large force that pushes the sharp edge of the disk and seat into each other. This causes severe cutting action damage.

Guide Tolerance is a critical factor when cycling a gate valve. When there is too much play, the disk may tilt towards the seat. Play causes major seat damage and may result in failure of the valve as the disk jams on the bottom of the seat. On manual valves, an experienced operator may be able to back it off and try again; an inexperienced operator or motor actuator will not detect the problem and cannot adapt as necessary. In this case, the valve disk will be pushed downward resulting in a bent or broken stem. Guide tolerances may be lessened by different welding methods. (Refer to Drawing 2)

Solution: Radius Grinding of Gate Valves

As a result of the industry's continued emphasis to address the "Smooth Edge Criteria" through chamfering of sharp edges on gate valves as recommended in the EPRI PPM, the Chamfering Accessory Kit was made available in late 1999, for use with the Model G and FL portable valve grinders, in a valve range of 2 ½" to 43". Due to the new demands of the "Smooth Edge Criteria", a Radius Grinding Accessory Kit has been designed to ensure that there are no sharp edges on gate valve seats and wedges.

Typically, valve seats and wedges were chamfered by cutting a 45° angle, 0.060" from the edge (see Diagram 1). While this lessened the scratching problem, it did not entirely eliminate it.

A more recent solution is the Radius Accessory Kit utilizing mechanically driven technology to grind the inside diameter (ID) and outside diameter (OD) of gate valve seats and wedges. This eases the single 45° chamfer to a 15° and 30° chamfer. It then uses a rounding segment to remove the angle edges providing a smooth curved chamfer (see Diagram 2). All heads are manufactured in permanent diamond. The stationary center of the driving head utilizes a guiding plate that acts as a pilot for the chamfer. This guiding plate will ensure that the angle cut is the same width around the seat.

Procedure for Chamfering and Radius Grinding of Gate Valves

1. Mounting:

- o Mount the bottom plate onto the flange
- o Attach the mounting frame onto the bottom plate
- o Measure the ID of the bore and adjust the guide plate to the correct diameter
- o Choose the correct plates and radius grinding heads to grind either ID or OD of the seats (Refer to Photo 1)
- o Place the drive shaft in the valve and adjust vertical and lateral knobs accordingly

2. Radius Grinding the sharp edge of seat (Refer to Photo 2, Drawing 3)

- o Grind inside of seat 45 degrees (if necessary)
- o Grind inside of seat 15 degrees
- o Grind inside of seat 30 degrees
- o Grind inside of seat with rounded radius segments and 80 grit diamond compound

3. Utilize the above procedure on the outside of the seat (Refer to Photo 3)

4. Radius Grinding radius on OD of wedge/disk (Refer to Photo 4)

- o Grind wedge/disk w. 45 degree wheel if necessary

- o Grind wedge/disk w. 15 degree wheels
- o Grind wedge/disk w. 30 degree wheels
- o Grind with rounded radius segments and 80 grit diamond compound

Alternate method:

- o Chamfer / Grind wedge/disk w. 15 degree wheels
- o Chamfer / Grind wedge/disk w. 30 degree wheels
- o Chamfer / Grind wedge/disk w. 45 degree wheels
- o Grind with rounded radius segments and 80 grit diamond compound

After radius / chamfering grinding the sharp edges on a gate valve, the radius on the ID and OD allow the disk to first slide onto the radius eliminating the cutting action effects. When the gate valve is closing, the OD radii on the wedge and seat will take the role of eliminating the scratching effect on the downward motion. When the gate valve is opening, the ID radii will take the role of eliminating the scratching effect on the upward motion (Refer to Drawing 5).

CONCLUSION: Meeting the "Smooth Edge Criteria"

The required "Smooth Edge Criteria" on the seat and wedge of gate valves has been recognized by both EPRI and the NRC. Chamfering has been the recommended practice on gate valves. Currently, FPL: Seabrook Nuclear Station has implemented the use of the Chamfering Accessory Kit and the specific maintenance procedures program to assure that the edge configuration on the disk seats and guides meet the EPRI recommendations (no "sharp" edges). If the chamfering procedure is correctly performed using the 15°/30°/45° grinding heads on both the seat and wedge, it is not necessary to continue with the radius grinding rounded segments 15°/30°. However, this is the recommended method to ensure a smooth edge on the ID and OD of gate valve seats and wedges.

Part 2 Sub Title: Remote Operation of Valves

HISTORY OF PROBLEM

Many valves in nuclear plants are located in hard to reach, contaminated or high radiation areas. ALARA regulations have pressed the industry to reduce exposure to plant personnel. This portion of the paper will focus on a problem at Vermont Yankee and its resolution.

The RWCU System in BWR plants removes impurities from the reactor coolant using pressure pre-coat type filter/demineralizers (F/Ds). The system for Vermont Yankee, shown schematically in Figure 1, utilizes two half capacity F/Ds in parallel that periodically must be backwashed and recharged with fresh pre-coat material. An F/D is taken off line for this purpose by closing the inlet and outlet isolation valves, air operated gate valves, 80 millimeters [mm] (3") in diameter (refer to Figure 2, V-14A/B & V-16A/B). During this operation, differential pressure (DP) across these valves increases to the reactor operating pressure of 1020 pounds per square inch gage [psig]. The air operators for these valves are not designed to be opened under such high differential pressure, so normally the pressure is first equalized by manually opening small 7 mm (1/4") diameter instrument tubing lines that connect between the two sides of the outlet valves (V-16A/B). This process takes several hours to complete and when the F/Ds are backwashed during normal plant operation, such delays are not a problem.

However, when the F/Ds are brought back on line at the end of a refueling outage, this task is on the startup critical path and this delay becomes very costly. To eliminate this constraint on startup, a decision was made to install a 20 mm (3/4") diameter pressure equalization line directly around each inlet isolation valve (V-14A/B). This pressure equalization line itself is isolated using a 20 mm (3/4") manual gate valve (V-15A/B-refer to Figure 1).

The F/D holding pump cubicle in which the valves are located is a locked high radiation zone since several hot spots in the piping read as high as 1 rem/hour on contact. Because operating the new valves in this environment would be an ALARA concern, two options were investigated to minimize personnel dose.

Option one would involve rerouting the bypass line a longer distance so that the bypass isolation valve could be located near the cubicle entrance where dose rates are much lower (10 to 15 millirem/hr range). However as shown in Photos 1 and 2, the holding pump cubicle is very congested with numerous pipes, valves and instrument line, making installation of the longer runs of piping very difficult and time consuming, and resulting in significant dose to installation personnel. Additionally, auxiliary operating personnel (AO's) would still be required to enter the cubicle to operate the new bypass valves and would therefore still be exposed to some dosage.

Option two was to install extension stems on the new valves enabling remote operation from outside the cubicle. Operations personnel were not receptive to this idea due to past problems when dealing with rigid rod extension stems. Originally the RWCU system had approximately 10 valves in this cubicle operated remotely with rigid reach rods. However, the gear boxes and universal joints employed by this type of reach rod often bound up, requiring that AO's enter the cubicle to free up the linkage or to temporarily remove it to operate the valves locally. The large number of reach rods crisscrossing throughout the room was a nuisance when trying to conduct maintenance on the pumps and instrumentation. As a result of these problems, a decision was made early in the Plant's life, to remove all of these rigid rod assemblies from the RWCU system and to seal up the penetrations provided for them. Photos 7, 8 and 11 show where the grout was used to fill these penetrations. The Operations Department made it quite clear that if the extension stems were used on the new valves, the old style rigid rods with gear boxes and universal joints would NOT be acceptable. (Charles Edwards, Vermont Yankee)

These concerns pointed to the need for a different approach in designing and operation of the new bypass line and valves.

Solution Summary:

After some initial reservations, the Operations Department agreed to try Remote Mechanical Valve Actuators (RMVA) for this project. This decision was based upon several advantages identified for the RMVO's flexible helix cable. The RMVA's continuous loop flexible conduit is completely enclosed requiring no maintenance. No intermediate gear boxes or universal joints are required. As long as the minimum bend radius (~ 300 mm (12") for 10 mm/ ½" diameter cables) is maintained, there is no binding problem. Photo 7 & 11 details how one cable can be bent in several different directions requiring a simple clamp type support every 3 m (10') to 4.5 m (15'). As the cables are flexible, they can be easily configured to avoid interferences with piping, valves, instrumentation and equipment, a particular concern in this holding pump room as evidenced in Photo 5 and 6.

The Remote Mechanical Valve Actuator (RMVA) system was developed in response to a demonstrated need for a highly reliable, cost effective and maintenance free alternative to antiquated reach rod and flexible shaft technologies. The RMVA system meets or exceeds the most demanding Military specifications including high impact shock, vibration, flame resistance and submergibility and is widely used on ships and aircraft.

The RMVA is based upon a simple tension-tension, closed continuous loop, actuating concept. The component common to all RMVA systems are the helix drive cable and drive gear. The highly flexible helix cable is manufactured from high tensile strength steel wires with an outer helical wire wrap. This cable meshes with the drive gear, which is specially machined to match the pitch of the helix cable. It is this precision helix cable/driver gear engagement that enables the RMVA system to efficiently deliver high torque loads over extremely long distances and through multiple planes.

Pre Installation Procedure:

To minimize installation time, cost and dose, Vermont Yankee reused the original core bores already in existence in the cubicle walls that were grout sealed when the old reach rods were removed. Each core bore measured ~ 65 mm (2.5") in diameter as shown in Photo 7, 8 and 11, and for the Alpha train the 10 mm (1/2") diameter cables of the flexible cable fit easily through an opening. It was initially planned to grout the opening in and around the cables and to add a steel plate on the outboard face of the wall if additional shielding was found necessary. However, dose measurements taken outside of the cubicle after the grout was removed (Photo 4) determined that neither grout nor steel plates were necessary for V-15A.

On the Bravo train, the small diameter and flexibility of the RMVA cables permitted the use of an existing piping penetration high in the sidewall of the cubicle. The gap of approximately 25 mm (1”) existing between the pipe sleeve and the 200 mm (8”) diameter pipe running through the penetration provided sufficient clearance to slip the flexible cables through the opening, thus eliminating time, cost and exposure that would have been needed to open one of the original grout sealed bore holes. (Photo 9 and 11).

Installation

The Plant completed installation, as directed by vendor’s Installation Guide and Technical Manual, of both remote extension stem assemblies smoothly and quickly without any problems. Connection of the extension stem to the valve was an easy task. Orientation of the new valve was pre-planned so that several feet of space directly in front of the valve stem would be open for the connection of the necessary hardware. Photos 7, 9 and 10 show the handwheel adapter, flexible coupling and remote operator gear box.

A simple steel angle floor stand was fabricated and installed as a support for the gear box (Figure 3 and structure in Photos 7 and 10). The handwheel adapter was equipped with a built in quick disconnect fitting. This permits ease of accessibility to the valve as the cable and gearbox can be quickly and easily disconnected from the valve and swung out of the way when maintenance is performed on the valve or other components in the area.

Standoff kits were installed at three-to-five foot intervals throughout the length of the system.

The vendor technician crimped the cables, initialized the system and verified that installation was accomplished correctly and that the system was operable.

Conclusion:

Through the use of new technology and creative design options, Vermont Yankee successfully implemented an economic, long term solution reducing radiation exposure to Plant personnel and producing a safer environment at the Plant. The installation in the radioactive areas only required one shift with a total dose of approximately 100 mr received. Had Vermont Yankee chosen the option of running longer bypass lines, it was estimated that the extra installation time would have resulted in a dose five (5) times higher than actually received. Vermont Yankee found that the new Remote Mechanical Valve Operators (RMVA) work well with no operational problems and requiring minimal maintenance as the system is a closed system. It now requires only a few seconds to equalize the pressure across the F/D inlet isolation valves, with no radiation exposure to Plant personnel.

Summary

Both the Radius Grinding technique and the RMVA provide the industry with long terms economical solutions to valve maintenance and operation concerns.

PART 1: "SMOOTH EDGE CRITERIA"

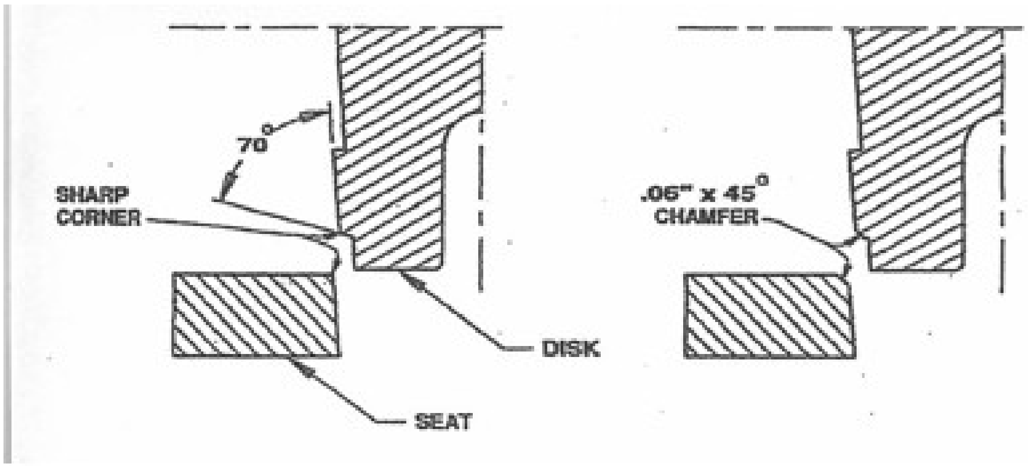


DIAGRAM 1 – EPRI CHAMFERING CRITERIA

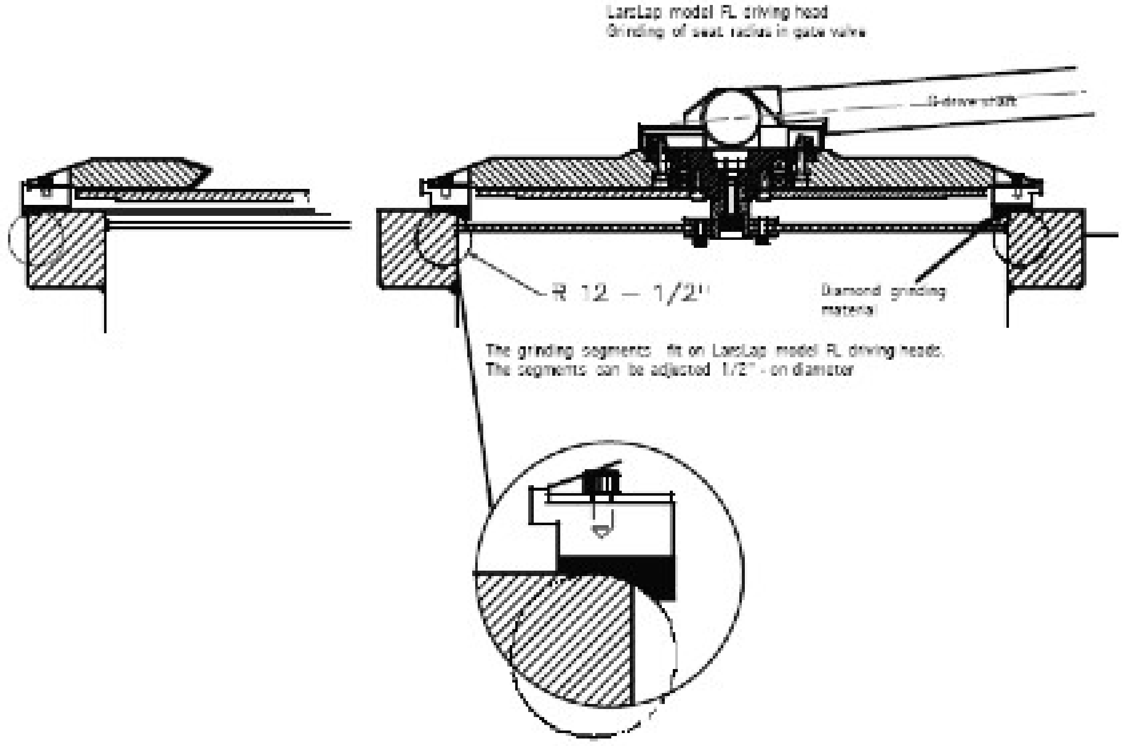
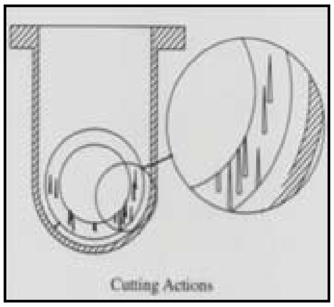
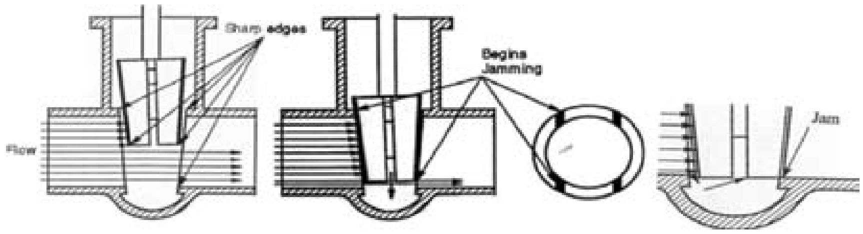


Diagram 2 – Specialty Segment Mounted On FL



Drawing 1 “Cutting Actions”



Drawing 2 “Play in Guide Tolerances”

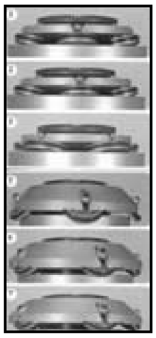
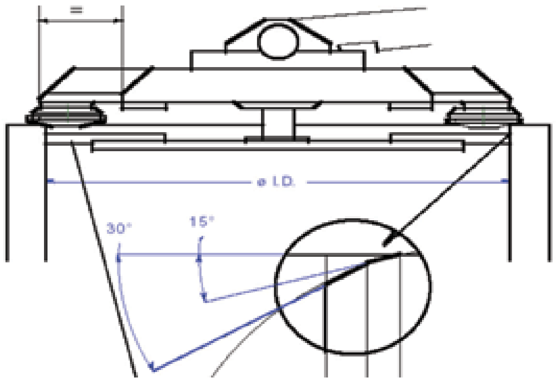


Photo 1 “Radius Grinding Procedure”



Drawing 3 “Radius Grind ID of Gate Valve”

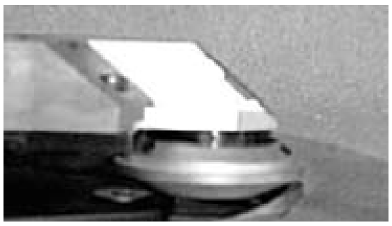


Photo 2 “Radius Grinding ID of Gate Valve”

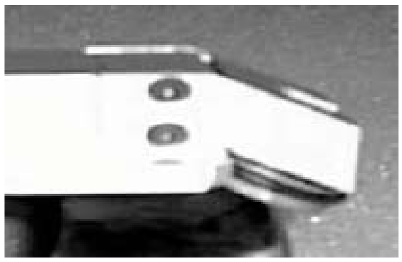
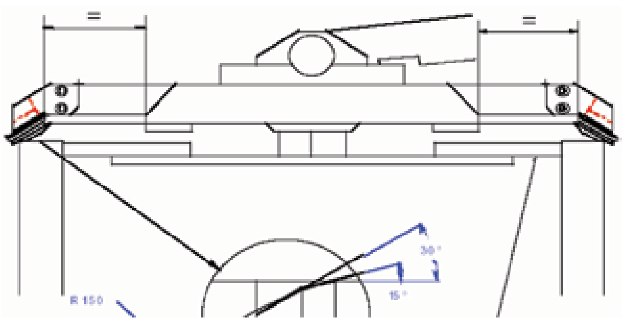
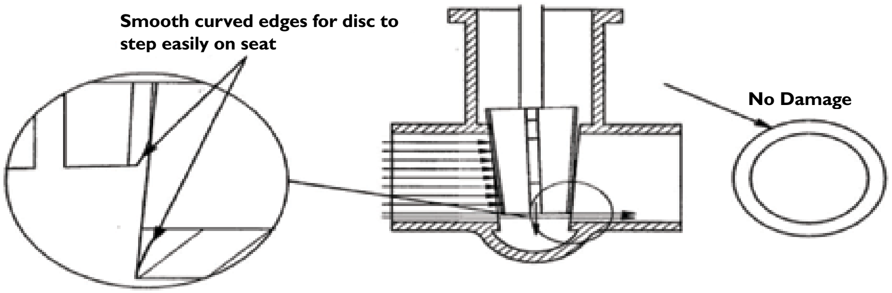


Photo 3 “Radius Grinding OD of Gate Valve Seat”



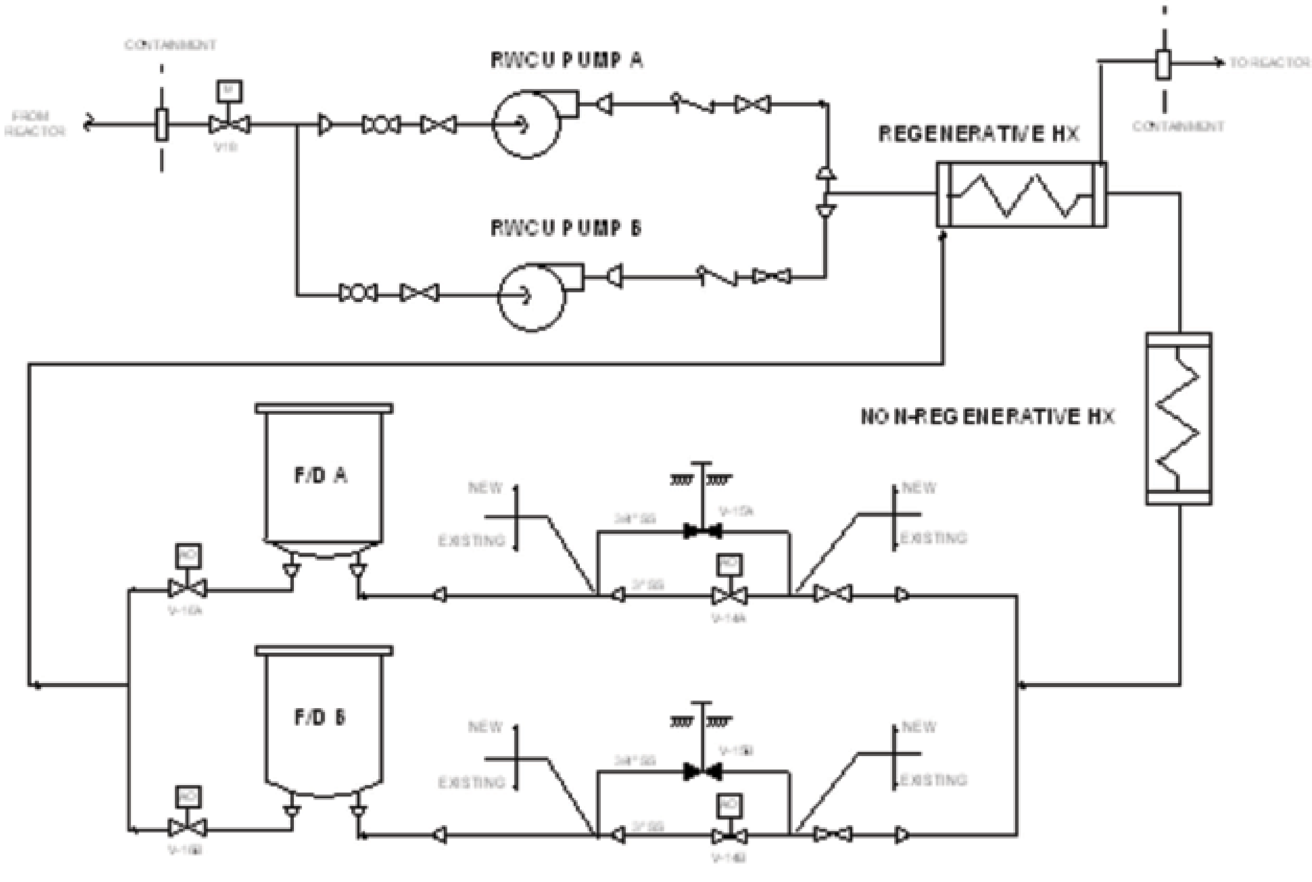
Drawing 4 “Radius Grinding OD of Gate Valve”

Photo 4 “Radius Grinding of Gate Valve Wedge”



Drawing 5 – "Smooth Edge Criteria met by Radius Grinding"

Part 2: Remote Operation Of Valves



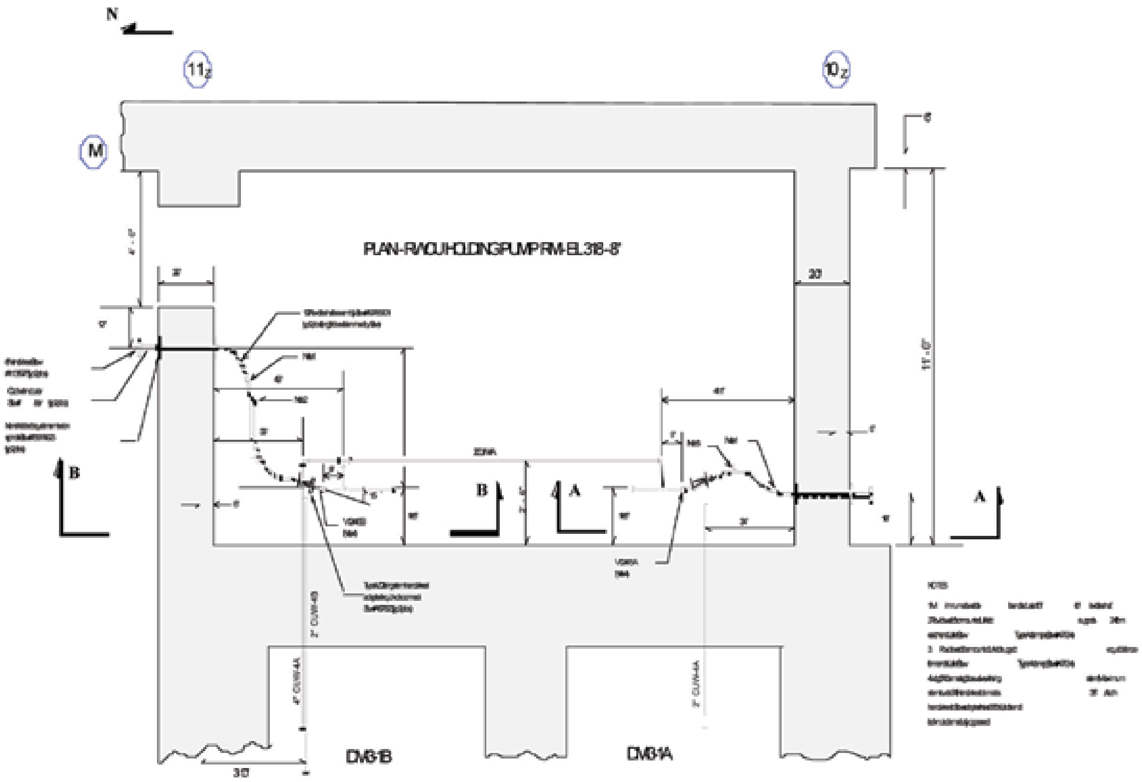


Figure 2 - Plan View Of New Extension Installation

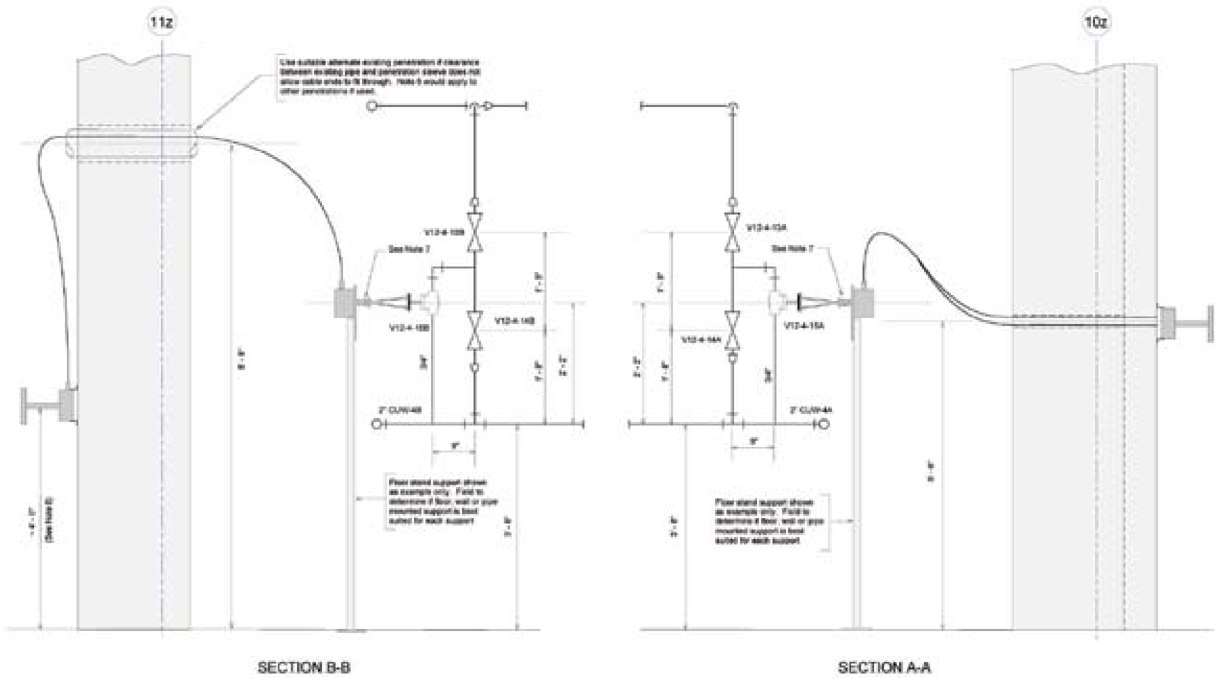


Figure 3 – Cross Sectional View Of New Extension Stem

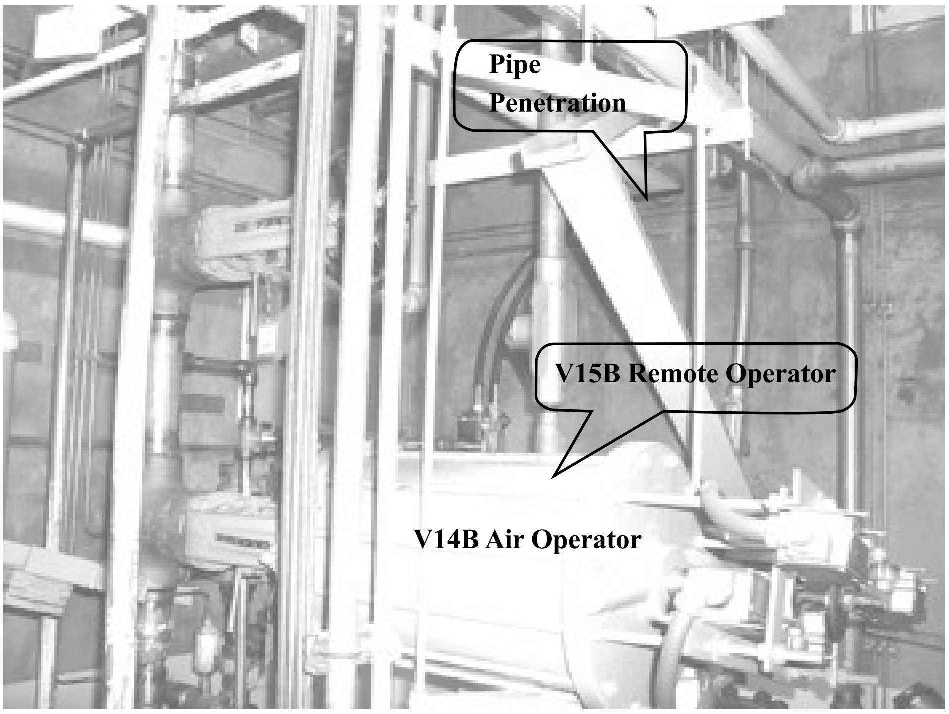


Photo 5

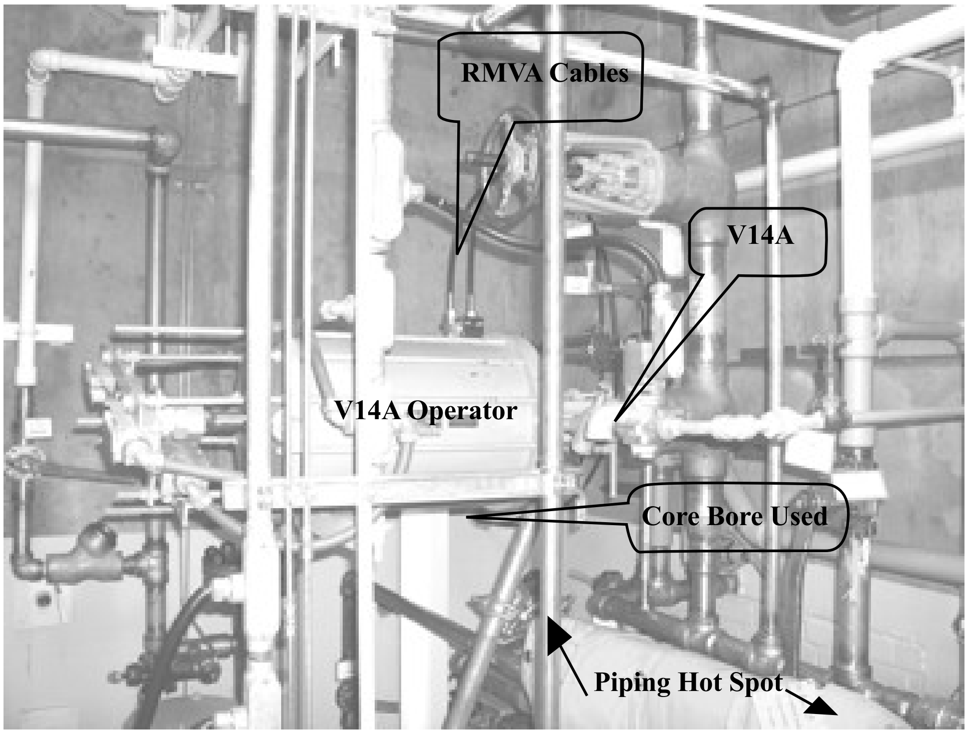


Photo 6

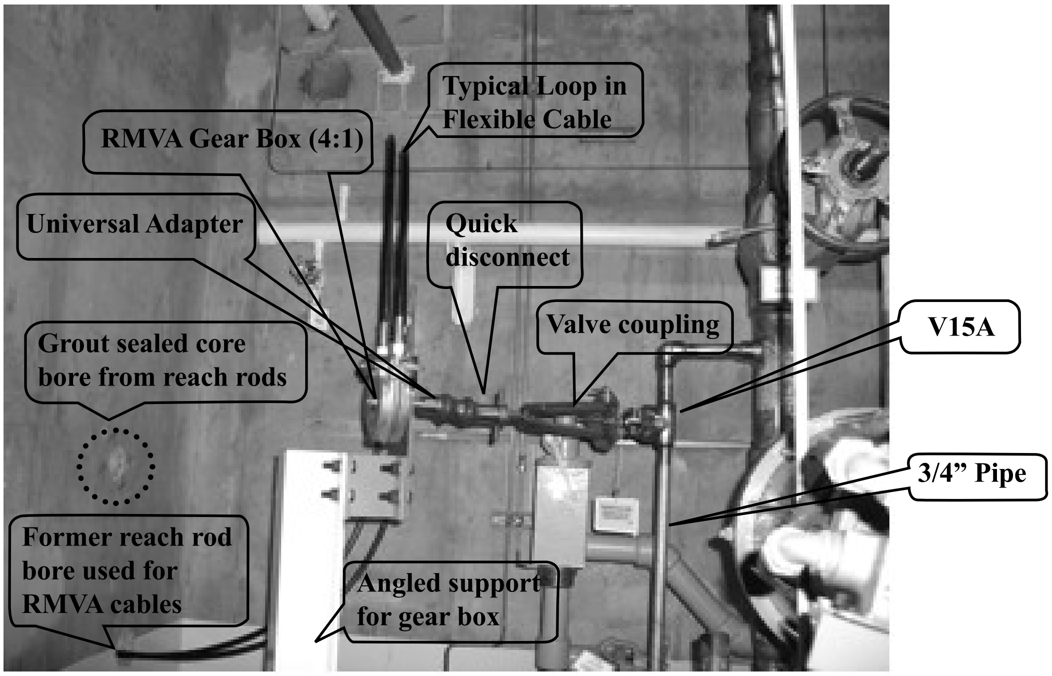


Photo 7

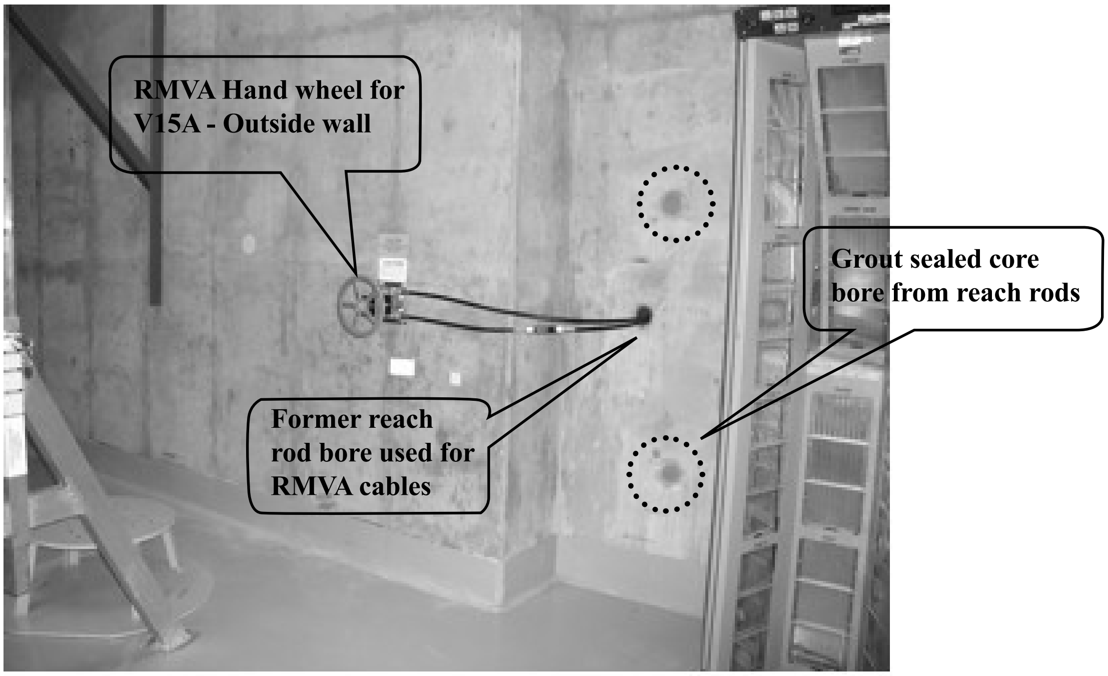


Photo 8

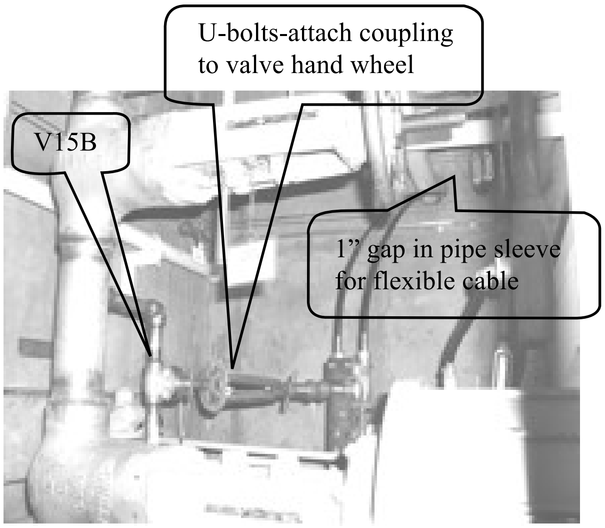


Photo 9

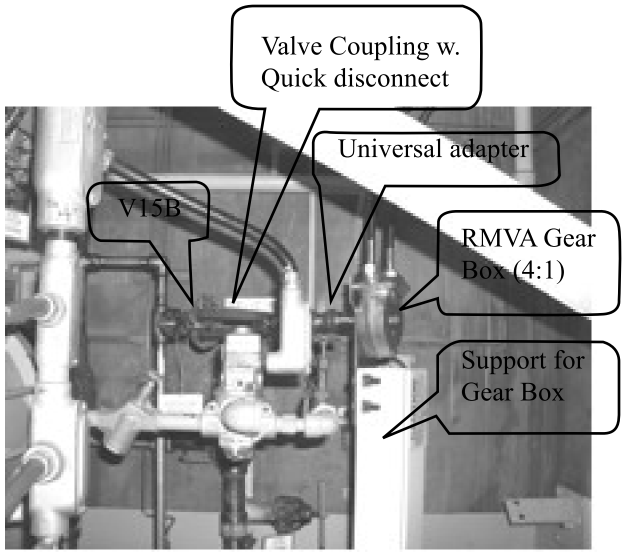


Photo 10

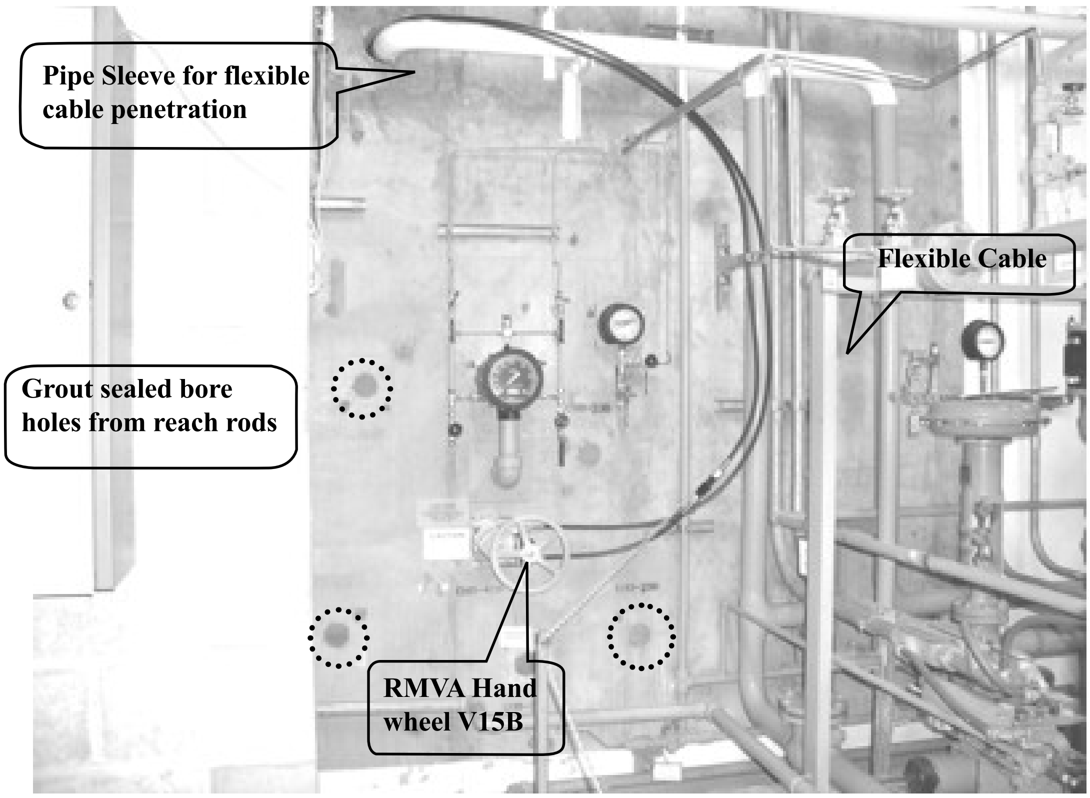


Photo 11

Acknowledgements

William Slover, EPRI

Robert J. Thomas, Seabrook Generating Station

George F. Harding, Mechanical Engineer

References

- NRC GL 89-10, “Safety Related Motor Operated Valve Testing and Surveillance.”
- NRC Information Notice 96-48, “Motor Operated Valve Performance Issues.”
- Seabrook Maintenance Procedures MOV Gate Valve Maintenance (By Permission of Seabrook).

Session 2(a):
Risk-Informed Inservice Testing of Valves & Pumps

Session Chair

Craig D. Sellers

Alion Science and Technology

Lessons Learned during Implementation of Alternative Treatment for In-Service Testing of RISC-3 Pumps and Valves

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Abstract

Nuclear plant owners and operators will be considering whether the risk-informing approach for establishing alternative treatments for safety-related structures, systems, and components (SSCs) can be a benefit to their stations. In part, the decision will be based on the effort required to implement the requirements of Section 50.69 of Title 10 of the Code of Federal Regulations (10 CFR 50.69 rule). The potential for misunderstanding of the allowances and concern for what unanticipated issues may arise are factors that may cause some to hesitate before beginning this new process. This paper will attempt to provide some insights into the issues that were addressed by South Texas Project (STP) during the implementation of the risk-informed exemption for In-Service Testing (IST) of pumps and valves. The 50.69 rule and the STP exemption are generally equivalent, although there are a few significant differences (e.g., the rule requires the categorization process to address common cause failures and known degradation mechanisms, and the STP exemption has more prescriptive treatment requirements for "RISC-3" components defined in 10 CFR 50.69). Although there are some differences between the exemption granted to STP and the requirements of 50.69, most of the issues encountered by STP should be applicable to any site wishing to reduce treatments with the 50.69 rule. This paper will detail five areas of the process with the intention of fostering critical thinking on how these areas can be addressed by each individual site given their own design and operating philosophies.

Reduction of the treatment of some SSCs is contingent upon their minimal contribution to radiological releases or impact on core damage as a result of their failure (i.e., low risk significance) and that there remains sufficient confidence that the SSC will continue to remain functional. This portion of the paper will describe options that were considered for documentation of how reasonable confidence is maintained for the SSCs that were eligible for removal from the regulatory IST program.

The second implementation process addressed by this paper is the maintenance of IST program documents in support of the transition from the full IST program as required by 10 CFR 50.55a to a resource for questions for plant personnel or in support of quality/regulatory audits. Consideration for the duration of the implementation transition was a factor that resulted in a shift in the process.

A major concern for any process change of this magnitude is managing the change so that unintended consequences do not negate the benefits expected by the change. This part of the paper will describe what actions were undertaken to facilitate confidence that removal of pump and valve testing did not result in the removal of testing performed for other commitments.

Valve operability test procedures are used to satisfy "return to service" testing for safety-related valves in the IST program. The "return to service" testing for SSCs that have been removed from the valve testing procedures must be considered. This adjustment to an operational philosophy required active participation in the decision process for the implementation strategy.

The last section of the paper will address the implementation strategy. The method of implementation affects the level of effort required during each step of the process. Two options

for implementation were employed at STP, each with their own benefits and implementation costs. Each process will be described so that others may benefit from the consideration of implementation details.

Introduction

It is not possible to share completely in this one paper the experience gained from the actual implementation of the exemption allowances for in-service testing. Long discussions by site personnel were required to work through issues involving processes that are dovetailed together to form our understanding of the requirements for safety-related components at any nuclear plant. The participants of these discussions would likely have their own ideas of the lessons learned through the implementation of reduced treatments for in-service testing. Anyone interested in implementing a similar process would benefit from consideration of the experiences shared in this paper and then discussing these and other issues with other site personnel to get a broad perspective of the efforts required.

10 CFR 50.69, Risk-Informed Categorization and Treatment of Structures, Systems and Components for Nuclear Power Reactors, was printed in the Federal Register, Volume 69, No. 224, dated November 22, 2004. This new rule allows nuclear plant owners to redefine how special treatment requirements such as In-Service Testing are applied to structures, systems, and components (SSCs). In this paper, dealing with in-service testing of pumps and valves, the term components will be used with the understanding that the rule refers to SSCs more generally. Under paragraph (c) of this rule, components in a complete system are categorized into two groups; components that are high safety-significant and those that are low safety-significant. The details of the categorization process (reference NEI 00-04, 10 CFR 50.69 SSC Categorization Guideline) are not the subject of this paper. However, generally speaking, the categorization process is a blended approach of quantitative analysis using the plant probabilistic risk assessment with a qualitative assessment by experienced station personnel from a broad spectrum of functional responsibilities. The risk categorization process at STP was enhanced to include a review and approval of all components by senior plant management in an expert panel.

The final risk categorization for each component is used to determine the component's eligibility for special treatment reduction in accordance with the rule. Safety-related components that are determined, through the categorization process, to be safety-significant (i.e., Risk-Informed Safety Class 1 (RISC-1)) remain with the full special treatment requirements for safety-related components. Low safety-significant components may be removed from the special regulatory treatment requirements. Safety-related low safety-significant components (i.e. Risk-Informed Safety Class 3 (RISC-3)) are subject to the alternative treatment requirements identified in paragraph (d)(2) of the rule. Paragraph (d)(2) of the rule requires the licensee or

RCE Options Summary

Options	Number of RCEs	Evaluation detail	Plant Review	User interface Comments
RCE by component	250	High	Too many documents, redundant	Direct relation to component, too many documents
RCE by group	125	High	High, with some redundancy	Not a direct relation to component, have to find the right eval.
RCE by component type	6	OK with large eval. for detail	More focused reviewer	OK, with index. Easy to find right eval. if valve type known
RCE for all components	1	Book for required detail	Daunting	Nice to have in one spot, but potentially difficult

applicant to ensure, with reasonable confidence, that RISC-3 components remain capable of performing their safety-related functions. The alternative treatment for RISC-3 components must be consistent with the categorization process and shall include inspection, testing, and corrective action. Plant changes and adverse changes in component performance are reviewed for impact to the categorization of components.

Reasonable Confidence Evaluations

The 50.69 rule requires that licensees develop an alternative treatment approach for RISC-3 SSCs including inspection and testing to ensure, with reasonable confidence, that components remain capable of performing their intended safety functions. The reasonable assurance of component capability that currently exists based on the existing maintenance, testing, inspection and surveillances for these components will provide a beginning point for the development of the alternative treatment to be applied to the RISC-3 components. Reasonable Confidence Evaluations (RCEs) describe the industrial practices currently in use at the plant that provide information for the determination that the components remain capable of performing their intended safety functions as required by the rule. Documentation on testing requirements and maintenance histories were collected to support the basis for maintaining reasonable confidence in the RISC-3 components. Components with the same manufacturer, model, size, and safety functions were combined into component groups similar to the process used for the check valve condition monitoring requirement in Appendix II in the later editions of the American Society of Mechanical Engineers Code for Operation and Maintenance of Nuclear Power Plants (ASME OM Code). The component history from site databases was queried and analyzed for each component of the component group. Maintenance history and failures were identified and reviewed to determine if there are specific concerns for satisfactory performance that currently exist. In all cases, the failure rates were acceptable within normal industry expectations, and the maintenance history indicated that routine maintenance on the components was satisfactory.

The method of documenting the results of the engineering analysis and conclusions about reasonable confidence was considered as the component history was being collected. Originally, it was anticipated that reasonable confidence could be described for each component type (e.g., motor operated valves (MOVs), air operated valves (AOVs), etc.). Tables were developed to identify typical maintenance, inspection, testing, and surveillance activities that were performed for each component type. This resulted in the understanding of how each component type was maintained with typical preventive maintenance activities. It also provided insight into how different components were being tested with existing Technical Specification requirements. It was anticipated that components within a given component type would be maintained and tested in the same general manner so that a model of reasonable confidence for each component type would become evident. Exceptions to the model would be identified and documented. The exceptions to the model would be evaluated to determine whether reasonable confidence for the component group is acceptable or if adjustments to existing activities for these components were required. This evaluation also identified that each component group was utilized during normal operations to varying degrees. In some cases, verification of component performance was proved during normal train rotation activities which occurred more frequently than existing in-service testing program requirements.

The RCEs were documented with engineering evaluations that became a part of the condition reporting process at South Texas. As the RCE for MOVs was being written, it was decided that the discussion of the maintenance history and operational use of each component group would result in an evaluation that would be very large. The discussion of maintenance history and operational use for each component group would typically be a page in length. A discussion on the MOV model for reasonable confidence was also required. How each of 26 MOV groups met the model or the acceptability of any divergence from the model would produce an extensive document. The user of such a document may have difficulty finding the information for the component desired. This system would also require the user to first understand the model for the component-type reasonable confidence and then look to each group to see if there were any exceptions to the model for that component group. These drawbacks in the presentation of the material may be resolved using methods to facilitate the user's search for the required information such as table of contents, summary tables with required information, or references to pages for exceptions where applicable.

RCEs for individual component groups can be written specifically for that group with detailed discussions of the maintenance history, surveillance requirements, and operational use. The basis for reasonable confidence would be clear and evident as the user reviewed the document. Individual group RCEs also supported a concern for change management during the implementation process. More discussion of this concern is provided later in this paper. Documentation of RCEs by individual groups was selected as the process to be used.

However, there were drawbacks to this method that should be noted and considered. The process for collecting reviews and comments from key stakeholders was magnified due to the number of evaluations. Generally, most of the evaluations for a component type, such as MOVs, tended to be very similar. Reviewers might feel that evaluations for each group were the same and their time was not well spent by continuing to review the 26 different evaluations. The following table provides a summary of the considerations for the development of RCEs.

IST Program Documents

IST program test plan and bases documents are maintained to address how code requirements are satisfied at South Texas. Each safety-related component that meets the scoping criteria for inclusion in the IST program can be found in the bases document with its intended safety function(s). The plan identifies the testing requirements and any applicable relief requests for testing interval justifications. These documents are used for resolution of questions concerning the program requirements and are the logical beginning place for understanding the program scope. Most nuclear industry professionals expect to find safety-related pumps and valves that meet the scoping criteria of the OM Code in the IST Plan. Absence of these components in the IST Plan would result in a concern that IST program requirements are not being satisfied. The listing of the RISC-3 components in the IST Plan prevents the erroneous conclusion about the component's status as a component that was scoped into the IST program but is now removed from those requirements. The status, where applicable, of scoped IST components is identified in these program documents to resolve any questions regarding their applicability and status in regards to the IST program requirements. The IST Plan and Bases documents were revised to reference the RCEs approved for individual component groups. The IST Bases document includes the RCE evaluation conclusions for reasonable confidence when the components were removed from the IST program.

Once the components were removed from the IST program, the IST program requirements are no longer applicable. Program documents are not revised to maintain a current basis for reasonable confidence for the removed components. Whenever practical, existing Technical Specification surveillance requirements are selected as part of the basis for reasonable confidence. Technical Specification change control will ensure that the activity remains in place and any changes to the activity will receive a broad review by station personnel. Impacts to the basis for reasonable confidence will be identified and addressed as needed. The Preventive Maintenance (PM) program is an industry practice that includes instructions to station personnel for the development of effective PM tasks with intervals that reflect industry and station experience to maintain component reliability. Controls in the PM program ensure that justifications for task and interval changes are documented so that the basis for reasonable confidence is maintained for the removed components.

One revision of the IST Plan and Bases documents was envisioned at the end of the evaluation process. It became useful during implementation to have an up-to-date list of components that had been removed from IST program requirements based on approval of the RCE for individual component groups. The IST documents were periodically supplemented with a list of removed components to support procedure revisions required to implement the change in testing scope. Up-to-date IST program documents supported the approval of license compliance reviews required for procedure revisions and provided the basis for changes to surveillance testing scope as required.

Change Management Concerns

Implementation of any change, especially one that affects compliance with Technical Specification surveillance requirements, has the potential for unintended and undesirable consequences. The concern was identified early in the process that the valve operability testing procedures are used for requirements other than in-service testing. The removal of stroke time testing of valves was a change that could and, in fact, does affect other testing requirements identified in the Technical Specifications. [Author's note: South Texas is not using the improved technical specifications per NUREG-1431, "Standard Technical Specifications – Westinghouse Plants." The author will point out differences when the author is aware of it; however, there may be other changes of which the author does not have specific knowledge.]

A thorough review of the actual testing requirements for each component was completed and documented in the reasonable confidence evaluations. This process involved a review of the surveillance testing database and procedure scoping statements by knowledgeable plant personnel. Station personnel confidence that all commitments were being identified increased based on the extent of testing being identified on the reasonable confidence evaluations for each component group. Additionally, procedures were reviewed to address any other testing requirements that may be included in the scope of the procedure prior to any revision. This process of procedure and testing review supports the license compliance reviews required for surveillance procedure revisions. Several issues were identified and addressed by an implementation team supported by operations, engineering, and licensing personnel.

STP Technical Specifications require a stroke time test following maintenance on containment isolation valves (this is not a surveillance requirement in NUREG-1431). STP considers that a stroke time test verifies that the component can perform its intended function to close within the design basis stroke time following maintenance. For example, an MOV in a non-safety related system carries waste liquid out of the reactor containment building. The containment penetration and the associated containment isolation valves for the penetration are safety related. The MOV (as one of the containment isolation valves for this penetration) has a safety function to close and be leak-tight. A design basis limit for containment isolation in the safety analysis report is to close within 10 seconds. This valve was categorized as low safety significant and also meets the treatment reduction allowances established for Appendix J, leakage rate testing. Following a typical lubrication and inspection of this valve, STP considers that exercising the MOV and verifying that control room indication is correct for the valve position is sufficient to provide reasonable confidence that the valve is now functional. The surveillance requirement in 4.6.3.1 does not distinguish whether maintenance affects stroke time or not. A literal interpretation of the surveillance requirement requires a stroke time as a surveillance test requirement with surveillance program controls. The capability to perform this test was maintained in the surveillance procedures; however, the code acceptance criteria range for MOVs was removed and the design limit was used as the acceptance criterion.

In a like manner, the verification of the stroke time for valves is required following maintenance that could affect the stroke time, whenever there is a design limit for stroke time included in an overall response time requirement in the safety analysis. These valves that were removed from in-service testing were kept in the surveillance procedures with appropriate stroke time acceptance criteria based on the design limit.

A typical phrase in surveillance requirements is “when tested pursuant to specification 4.0.5.” The implementation team concluded that this phrase is identifying the frequency of testing. “When tested” implies that the test interval is derived from the testing requirements specified by the in-service testing program. Some components are no longer in the IST program as a result of the STP exemption from special treatments. Therefore, a surveillance requirement including the phrase “when tested pursuant to specification 4.0.5” no longer has meaning. NUREG 1431 allows the removal of design information from the technical specifications, so the presence of the functionality criteria in the technical specification does not make it a surveillance requirement. Surveillance requirements that referred back to 4.0.5 are no longer effective when components are removed from the scope of Technical Specification 4.0.5, in-service testing. The technical specification basis for Technical Specification 4.0.5 was revised to include appropriate discussion concerning the status of these types of surveillance requirements when referral is made to 4.0.5, considering the STP exemption allowance.

Operations Return to Service Philosophy

Components are turned over to Operations for return to service operability testing following completion of maintenance and post-maintenance testing by the maintenance craft. Adequate testing of components, which have been out of service for maintenance, is an important aspect of plant operations that is impacted by in-service testing of pumps and valves. Return to service operability testing is generally performed using the valve operability testing procedures which have been written to satisfy IST program stroke time testing. The performance of the surveillance procedure for return to service testing developed into a philosophy that a surveillance test must be performed to prove the component’s operability. For components removed from the scope of IST, operability can be determined by verification with reasonable confidence that the component will continue to perform its intended safety function. At STP, normal industrial practices are used to confirm that the component can perform its function. Use of activities not previously identified as surveillance tests to determine operability created a concern within the operations staff for safety-related components in safety systems that were required to be operable per the Technical Specifications.

The Technical Specification definition for OPERABLE – OPERABILITY states, "A system, subsystem, train, component or device shall be operable or have operability when it is capable of performing its specified function(s), and when all necessary attendant instrumentation, controls, electrical power, cooling or seal water, lubrication or other auxiliary equipment that are required for the system, subsystem, train, component, or device to perform its function(s) are also capable of performing their related support function(s)."

The definition identifies that the capability of the component to perform its function is what determines operability. Verification of the functional performance of the component by means other than surveillance testing meets the requirement for operability for components removed from in-service testing program and technical specification 4.0.5. It was noted that there are safety-related components, other than IST pumps and valves, that are currently returned to service by means other than surveillance testing.

Removal of IST valves from the surveillance procedures affected the Operations philosophy for return to service operability testing. How RISC-3 components would be determined to be capable of performing their function was no longer a simple process of using the surveillance procedure. This is an Operations process that can only be resolved by Operations taking ownership of this process. The return to service process and the basis for operability determination for RISC-3 components was documented in a program procedure to maintain a consistent approach across all Operations crews.

Implementation strategy

The initial implementation strategy adopted after approval of the STP exemption allowance was a partial implementation. This strategy was considered a cautious and deliberate approach in that valve stroke time testing was not removed from the surveillance procedures. However, the exemption from code requirements allowed the relaxation of test intervals without specific code relief. The graduated extension of the testing intervals provided additional confidence that the RISC-3 component failure rate would not increase inordinately when code testing was discontinued.

This partial implementation strategy requires a method to identify the specific partial scope of valves to be tested within the normal valve operability test procedures. This option also maintains operability testing with surveillance procedures in accordance with the current Operations return-to-service philosophy and resulted in fewer procedure changes. Reasonable confidence was maintained since the surveillance procedures were still in use and return to service testing was being performed as usual. Therefore the overall cost to implement this partial implementation strategy is kept to a minimum; however, it also resulted in less benefit from reduced testing than was anticipated with the exemption allowance for in-service testing.

The vision of full implementation of the STP exemption allowance was the complete removal of the RISC-3 components from the In-service Testing program and the extraction of the stroke time testing from the surveillance procedures to the maximum extent possible. Removal of valve stroke time testing resulted in the desire for a more careful review of commitments and testing requirements in the valve procedures as noted in the change management section addressed earlier. The full implementation strategy also resulted in the need to define the return-to-service operability testing since the surveillance procedures no longer contain the valve tests for RISC-3 components. In summary, the full implementation costs were more with greater overall benefit to the plant when the RISC-3 components were removed from the program.

Conclusion

Removal of many of the safety-related RISC-3 pumps and valves from the special treatment requirements results in considerable benefit to the station. The benefit to the station was not described in this paper but deserves to be detailed by specific actions and tasks in a separate paper. However, as a result of the implementation, a significant benefit was derived in that the basis for reasonable confidence for these components identified the overlap of testing and processes that are a part of all nuclear industry stations. The documentation of all methods for verifying component performance provides insights to operations and engineering that can be used to improve reliability of components. During this process, it was discovered that some safety-related components are rich in terms of the demands placed on their performance while other components are demanded sparingly. This information can be used to recognize where alternative treatments are needed to support confidence in the components' performance. The fact that most safety-related components, regardless of their lower safety significance, are proven functional in multiple ways should come as no surprise to nuclear industry professionals. The alternative treatment process identifies the industrial practices at any nuclear station may use to maintain an awareness of the plant and the performance of its many components.

Many of the lessons learned during this project were as a result of nuclear professionals asking critical questions to prevent any slippage in the "nuclear safety first" stance that is prevalent in the industry. Questions about the methods that ensure the component's capabilities result in rigorous processes that bolster overall component performance. The industrial practices at nuclear stations ensure that degraded conditions are identified and corrective action is taken when failures occur.

Application of 10 CFR 50.69 – How a Robust Categorization Process Provides Confidence in Treatment Reduction for Safety-Related, Low Safety Significant Pumps and Valves

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Abstract

Section 69 of Part 50 in Title 10 of the Code of Federal Regulations (10 CFR 50.69), "Risk-Informed Categorization and Treatment of Structures, Systems, and Components for Nuclear Power Reactors," was approved by the Nuclear Regulatory Commission (NRC) on November 22, 2004. This milestone rule provides a structure for categorization of Structures, Systems, and Components (SSCs), and based upon the resultant importance determination, provides guidance for the appropriate treatment of safety significant and low safety significant components.

Much effort is underway within the industry to implement this new rule. This includes a categorization guideline for active components (NEI-00-04) authored by the Nuclear Energy Institute (NEI) as well as an American Society of Mechanical Engineers (ASME) Code Case (N-660) for passive component categorization. In addition, implementation guides (EPRI-1008748, -1009669, -1001234) authored by the Electric Power Research Institute (EPRI) have been developed to ensure consistency in application among diverse users. Even with these available guidelines, there exists hesitation within the industry to transition to 50.69 due to the radical approach to treatment of safety-related, low safety significant components.

This paper provides insight into the soundness of the categorization methodology that serves as a foundation for effective 50.69 implementation. In addition, insight will be provided on the types of treatment reductions that can occur to existing pump and valve testing programs while maintaining an appropriate level of confidence in component performance. Guidance will also be provided in other areas where 50.69 insights apply, and how these insights provide a foundation for effective and defensible decision making. Finally, this paper will provide the current status of both NRC and industry activities, including pilot plant activities and new plant activities, to adopt and implement 10 CFR 50.69.

It is expected that attendees will gain a basic knowledge of the requirements and flexibilities within 10 CFR 50.69, and will gain a greater appreciation of the rule's applications. Attendees will also better understand how to apply these allowances to their current pump and valve testing programs. Insight into costs and benefits of this approach will be communicated.

Introduction

The South Texas Project Nuclear Operating Company (STPNOC) served as the prototype pilot for the development of 10 CFR 50.69, "Risk-Informed Categorization and Treatment of Structures, Systems, and Components for Nuclear Power Reactors." As the prototype pilot, STPNOC submitted a request for Exemption from Certain Special Treatment Requirements to the NRC in July 1999. The intent of this exemption request from the STPNOC license was for full regulatory treatment and controls to continue to be imposed onto SSCs determined to be safety significant, while largely removing regulatory special treatment requirements on those SSCs determined to be low safety significant. In place of these removed controls, low safety significant SSCs would have industrial-type treatments imposed upon them. This request was ultimately approved by the NRC in August 2001.

The critical process that must be completed to effectively implement the allowances granted by the STPNOC Exemption or by the 10 CFR 50.69 rule is the development of a stable and sound categorization process that robustly determines the importance of each SSC within a given system under review. This paper focuses on the categorization process which blends Probabilistic Risk Assessment (PRA) Model insights with deterministic insights resulting in sound, stable importance determinations. These categorization results can be applied to broad-based risk-informed applications, including pump and valve testing. The application to an In-Service Testing (IST) program will be discussed, as well as noting the benefits that can be realized through this process.

The South Texas Project (STP) is a two-unit Westinghouse four-loop Pressurized Water Reactor (PWR) nuclear power plant rated at 1315 MWe output. Unit 1 was placed into commercial operation in 1988, and Unit 2 was placed into commercial operation in 1989. The Station is owned by three separate entities, and managed by the South Texas Project Nuclear Operating Company (STPNOC). The Station is located about 85 miles southwest of Houston, Texas, near the Texas Gulf Coast. Cooling water for the Station is drawn from an above-ground fresh-water reservoir supplied by the nearby Colorado River. The design of the South Texas Project incorporates three safety trains; however, the Station is licensed such that all three safety trains must be available.

NOMENCLATURE

Probabilistic Risk Assessment Model – an engineering tool used for decision-making which models certain components within the plant design that affect the protection of the reactor core and the health and safety of the public.

Reasonable Assurance – a justifiable level of confidence based on objective and measurable facts, actions, or observations, which infer adequacy.

Reasonable Confidence – a level of confidence based on facts, actions, knowledge, experience, and/or observations, which is deemed to be adequate.

Risk-Informed Safety Classifications (RISC) – the segregation of categorized components into specific importance groupings. The four groupings identified in 10 CFR 50.69 include:

- RISC-1 – safety-related, safety significant
- RISC-2 – non-safety related, safety significant
- RISC-3 – safety related, low safety significant
- RISC-4 – non-safety related, low safety significant

Special Treatment Requirements – the additional controls placed on safety-related equipment which exceed the normal controls placed on non-safety related equipment.

CATEGORIZATION BACKGROUND

Historically, nuclear power plants have been licensed with components classified as either safety-related or non-safety related. The safety-related designation is defined in 10 CFR 50.2. This definition focuses on the adequate protection of the reactor core, and on the protection of the health and safety of the public. While these designations have, by virtue of many safe reactor years, served the domestic American licensees and public well, it is recognized that the designation of safety-related or non-safety related are deterministically identified, with no bearing on the extent of the role that a certain component plays in protecting the reactor core or the public. This can result in controls or treatments being imposed on a large number of safety-related components which actually may be contrary to overall safe reactor operations. The additional burden placed on safety-related equipment also unnecessarily imposes costs onto the nuclear licensee which challenges effective, economical production.

It is also recognized that licensees have greatly refined their insights into initiating events and transients that can challenge safe reactor operations. These insights are modeled in detailed engineering tools termed as Probabilistic Risk Assessment (PRA) Models. These models assess the full range of internal scenarios that a nuclear power plant may encounter, and calculates the likelihood that a certain scenario may challenge the reactor core or the safety of the public. By placing appropriate attention to those scenarios that are most significant and/or most likely to occur, the likelihood of such events actually occurring is greatly reduced. This can be accomplished through designing additional engineering controls into the Station, enhancing or developing processes to address the concern, or bolstering the controls placed over activities which challenge the area of concern.

By considering both deterministic insights and probabilistic insights, the resultant categorization properly blends the likelihood of an event occurring and the impact of the event with the knowledge and experience that has been gained through years of plant operations. The resulting importance determination sharpens both the regulator's insight and the licensee's insight into those areas that are truly safety significant.

10 CFR 50.69 codifies this blended categorization approach, and defines the resulting importance determinations as follows:

- Risk-Informed Safety Class (RISC)-1 – safety-related SSCs that perform safety significant functions,
- Risk-Informed Safety Class (RISC)-2 – non-safety related SSCs that perform safety significant functions,
- Risk-Informed Safety Class (RISC)-3 – safety-related SSCs that perform low safety significant functions,
- Risk-Informed Safety Class (RISC)-4 – non-safety related SSCs that perform low safety significant functions

Figure 1 below shows the relationship between these four RISC categories. All safety-related SSCs which are categorized will either be placed into the RISC-1 box (also termed 'Box 1') or the RISC-3 box (also termed 'Box 3'). Safety-related SSCs cannot be placed into either the RISC-2 or RISC-4 boxes unless a design change is performed, and the SSC is redesignated as non-safety related. In addition, a safety-related SSC that is relied upon to satisfy a safety significant function(s) is placed into the RISC-1 box, while those safety-related SSCs that are relied upon to perform only low safety significant functions are placed into the RISC-3 box.

RISC-1 Safety-Related Safety Significant	RISC-2 Non-Safety Related Safety Significant
RISC-3 Safety-Related Low Safety Significant	RISC-4 Non-Safety Significant Low Safety Significant

Figure 1 – The 'Four-Box' Approach to Categorization Outcomes

All non-safety related SSCs which are categorized will either be placed into the RISC-2 box (also termed 'Box 2') or into the RISC-4 box (also termed 'Box 4'). Likewise, a non-safety related SSC cannot be placed into either Box 1 or Box 3 unless a design change is performed to redesignate the SSC as safety-related. Certain non-safety related SSCs may be relied upon to satisfy safety significant functions (e.g., support Station Blackout recovery) and are placed into Box 2. The remainder of non-safety related SSCs that perform only low (or no) safety significant functions are placed into the RISC-4 box.

All safety-related SSCs initially reside in the RISC-1 Box, and may be moved down to the RISC-3 Box through the categorization process. All non-safety related SSCs initially reside in the RISC-4 Box, and may be moved up to the RISC-2 Box through the categorization process.

SOUNDNESS OF THE CATEGORIZATION RESULTS

Licensees are accustomed to the component classifications of 'safety-related' and 'non-safety related' as licensed in their facilities, and largely accept the associated regulatory special treatment requirements imposed upon safety-related SSCs as necessary, though burdensome. For the 10 CFR 50.69 categorization results to have credibility with the regulator and with licensees, a sound process must exist to determine the overall SSC importance. As introduced in the previous section, this blended approach has been approved by the NRC and has been piloted by the South Texas Project (as well as by other industry licensees piloting the 10 CFR 50.69 process). The soundness of this approved categorization process is rooted in the comprehensiveness of the Probabilistic Risk Assessment models, in a consistent categorization methodology, in an effective feedback process, and in the knowledge and experience of licensee personnel. How each of these areas supports the soundness of the categorization results is presented below.

Domestic licensees currently rely on PRA Model insights when addressing a number of regulatory related activities, as well as to communicate the extent of issues with the regulator. To ensure PRA Model consistency among industry users, the American Society of Mechanical Engineers (ASME) and the American Nuclear Society (ANS) have developed, and are continuing to develop, industry PRA standards. In addition, domestic licensees have completed a series of PRA peer assessments to validate that individual plant PRA Models satisfy the requirements specified in the industry standards. During the performance of these peer reviews, any deviations from the industry standards were documented and tracked for resolution. Also, the NRC has issued Regulatory Guide 1.200, “An Approach for Determining the Technical Adequacy of Probabilistic Risk Assessment Results for Risk-informed Activities,” to ensure that industry PRA Models satisfy acceptable quality standards. Domestic licensees are in the process of satisfying the recommendations of Regulatory Guide 1.200. Based on these insights, sufficient industry guidance exists, and adequate NRC oversight is being applied, to ensure that industry PRA Models are properly robust and consistent.

The application of PRA insights into the categorization process is central to the approved approaches. NEI-00-04, “10CFR 50.69 SSC Categorization Guideline,” specifies a categorization methodology that considers PRA insights on Internal Event risks, Fire risks, Seismic risks, External Event risks, as well as Shutdown risks to provide an initial insight into the importance of a specified function or component. In addition, risk sensitivity studies associated with common cause interaction, human errors, increased component failures, etc., are performed in the categorization process to confirm that acceptably small increases in Core Damage Frequency and Large Early Release Frequency are associated with the proposed categorization. NRC Regulatory Guide 1.201, “Guidelines for Categorizing Structures, Systems, and Components in Nuclear Power Plants According to Their Safety Significance,” endorses the NEI categorization process with minor exceptions. Based on the availability of these approved guidelines, appropriate industry guidance exists to ensure that the PRA insights are properly considered in the categorization process.

In addition to the PRA categorization methodology provided in the NEI Categorization Guideline, NEI-00-04 also specifies the deterministic categorization approach. This deterministic methodology is structured to robustly address any potential limitations in the PRA Model, as well as provide a deterministic assessment of all components, including those that are modeled in the PRA. The NEI categorization process includes conservative decision-making into the categorization determination to ensure that minor changes in component performance or any other factors do not result in categorization changes. The NEI process provides a standardized categorization method which results in repeatable, consistent results. This comprehensive approach provides high confidence that each function and component in a subject system is properly assessed and categorized. Once a component is categorized into a specific RISC category using the NEI process, high confidence exists that the component will remain in this RISC category. Of the 94 systems and 78,000 components categorized to date by the South Texas Project, very few components have been noted to change RISC category boxes. Most categorization changes noted to date at STP have occurred following scheduled updates to the PRA Model.

The soundness of the categorization results is also confirmed through both continuous and structured feedback into the process. The continuous feedback process occurs daily as licensees make use of the categorization data. If a plant worker questions the accuracy of the categorization result or its documented bases, this feedback can be provided to the Integrated Decision-making Panel (IDP – the plant group responsible for the categorization process) for reassessment. Feedback can also be provided to the IDP from System Health Reports provided by the system engineer, from proposed design changes prior to implementation, from actual plant performance feedback, etc. However, at least once every two fuel cycles, the licensee must conduct a structured feedback process to confirm the adequacy of the categorization results. This structured feedback considers any changes to the PRA Model, insights from the Corrective Action Program, performance insights, system engineer insights, etc. By incorporating an effective feedback process, the categorization results are both confirmed and assured accurate.

The knowledge and expertise of licensee personnel also ensure the soundness of the categorization process and results. The IDP is composed of station experts knowledgeable in various areas of PRA, operations, maintenance, engineering, etc. These personnel are trained and qualified in the categorization process, and follow a proceduralized process to ensure consistent results. Consensus decision-making is utilized by the IDP, and differing opinions are encouraged to be expressed. The IDP uses conservative decision-making when uncertainty exists about a proposed categorization outcome.

The above overview provides insight into the robustness of the categorization process. This categorization process ensures that repeatable, consistent results are achieved which are soundly based and supported. The categorization outcomes and bases are well documented for future review and assessment. The rigor of the categorization process should instill a high degree of confidence in the adequacy of the categorization results.

Application Of The 10 CFR 50.69 Allowances

Considering the above discussion, the results using an approved categorization process are well based and thorough. The determination that a component is either safety significant or low safety significant should be accepted with confidence based on the robustness of the categorization process and the supporting documentation. Failure to accept the categorization results as well-founded can result in significant uncertainty during implementation of the 10 CFR 50.69 allowances and can significantly impact the benefits of the rule.

For components determined to be RISC-3 (safety-related, low safety significant) through an approved categorization process, 10CFR 50.69 allows reduction of the existing special treatment requirements. For safety-related pumps and valves determined to be RISC-3, the following regulatory requirements no longer apply:

- The In-service Testing (IST) requirements specified by 10 CFR 50.55(a)(f)
- The In-service Inspection (ISI) requirements specified by 10 CFR 50.55(a)(g)
- The ASME Class 2 and Class 3 requirements specified by 10 CFR 50.55(a)(g)
- The Type B and Type C Local Leak-rate Test requirements specified by Appendix J

It should be remembered that 10 CFR 50.69 is a scoping rule – it merely clarifies the scope of components subject to the regulatory requirements. RISC-1 SSCs continue to impose the full requirements of the above regulations, while RISC-3 SSCs are removed from those regulatory requirements. From a treatment perspective, a RISC-1 component's design function must be demonstrated with 'reasonable assurance', while a RISC-3 component's design function must only be assured with 'reasonable confidence'. Reasonable assurance implies some type of demonstration testing to prove with high confidence that RISC-1 pumps and valves will satisfy their design functions when demanded during design basis accidents. Reasonable assurance also implies a rigorous documentation trail to provide objective evidence that the appropriate demonstrations were completed, and that established acceptance criteria were satisfied. It is agreed that this approach to treatment is appropriate for safety-related, safety significant pumps and valves, and RISC-1 SSCs deserve appropriate focus by both the licensee and the regulator.

However, safety-related, low safety significant pumps and valves do not require the same degree of rigor placed on RISC-1 components. A lesser degree of control over RISC-3 pumps and valves is permitted by 10 CFR 50.69, but it is the licensee's responsibility to define when 'reasonable confidence' is achieved. The treatments to be applied to RISC-3 pumps and valves are generally similar to those treatments applied to balance-of-plant SSCs. An effort is currently underway to develop an ASME Standard (proposed OM-29) to offer industry guidance in determining the necessary treatment required to achieve reasonable confidence.

In addition to the special treatment requirements specified above that can be reduced for safety-related pumps and valves determined to be RISC-3, the following additional special treatment requirements can be eliminated for RISC-3 SSCs per 10 CFR 50.69:

- Reporting requirements per 10 CFR Part 21
- Environmental qualification requirements per 10 CFR 50.49
- Maintenance Rule requirements (except for (a)(4)) per 10 CFR 50.65
- Reporting requirements per 10 CFR 50.72 and 50.73
- Quality assurance requirements per Appendix B
- Certain seismic qualification requirements per 10 CFR Part 100, Appendix A

The categorization process insights provide a wealth of information to allow better-informed decisions to occur. The knowledge that a component is low safety significant and is supported by a well-founded basis provides an effective foundation to determine where attention and focus should be placed. In addition, commitments associated with RISC-3 components can be appropriately adjusted to permit increased focus on safety significant commitments and activities.

Status Of Industry And NRC Activities

In addition to the referenced activities completed by the South Texas Project, two other domestic licensees have piloted various aspects of the 10 CFR 50.69 categorization process. Wolf Creek Nuclear Station has completed trial categorization of the Containment Spray system and the Control Room Heating, Ventilation, and Air-Conditioning (HVAC) system. Wolf Creek intends to submit a Topical Report to the NRC on the categorization process that has been completed to date. This Topical Report is targeted for submittal in June 2006. Also, the Surry Nuclear Station has completed trial categorization of the Chemical & Volume Control system and the Main Feedwater system. Surry had intended to submit a License Amendment Request (LAR) to the NRC by the end of 2006 to voluntarily adopt 10 CFR 50.69. However, due to other demands, the ability to submit a short-term LAR is being reconsidered.

ASME continues to work on refining the passive categorization methodology specified in Code Case N-660. Also, the Electric Power Research Institute (EPRI) is continuing to add detail into the broad RISC-3 Implementation Guideline with an expected update to be published late in 2006. EPRI and NEI intend to hold industry workshops on the 10 CFR 50.69 categorization methodology and implementation approaches in the fall of 2006.

The recent approval of NRC Regulatory Guide 1.201, "Guidelines for Categorizing Structures, Systems, and Components in Nuclear Power Plants According to Their Safety Significance," dated May 1, 2006, has essentially completed the NRC short-term activities in support of 50.69. The NRC is prepared to begin the reviews of 50.69 LARs as they are submitted.

Benefits Of A 10 CFR 50.69 Approach

The South Texas Project has completed partial implementation of the approved Exemption from Certain Special Treatment Requirements. The approved Exemption closely mirrors the categorization approach and the treatment allowances specified in 10 CFR 50.69. The STP approach focused on completing baseline categorization while reducing RISC-3 treatment requirements in the areas of IST, Local Leak-Rate Testing (LLRT), Maintenance Rule, Parts procurement, Work Control, Preventive Maintenance tasks, etc. STP has committed to a deliberate implementation approach which assesses feedback to ensure the expected results are achieved. To date, STP has noted no adverse equipment performance trends as a result of reducing RISC-3 treatment requirements. In addition, with the partial implementation of the treatment allowances, STP is realizing annual benefits in excess of $1.2M per year. However, the real benefit noted by STP is the enhanced safety culture that exists at the plant. The readily-available risk information (i.e., component categorization) has fostered heightened understanding of the safety significance of components and activities among a wide range of workers at STP. This heightened 'risk culture' has improved the oversight of safety significant operational evolutions and maintenance work activities, bolstered the focus on planned work details and pre-job briefings when affecting safety significant components, and has heightened the management awareness of risk activities and their effects throughout the weekly scheduled activities.

Conclusion

An approved categorization approach which satisfies the requirements of 10 CFR 50.69 and recommendations of Regulatory Guide 1.201 results in a component importance determination that is robust, defendable, consistent, and repeatable. Licensees who voluntarily adopt 10 CFR 50.69 should have confidence that the rigor of the categorization process and result (i.e., RISC-1, RISC-2, RISC-3, or RISC-4) properly determined the component's overall importance.

Once SSCs are determined to be RISC-3, these components are candidates to be removed from the regulatory treatment programs (e.g., IST Program). The robustness of the categorization process should minimize any concern that components are susceptible to move from a RISC-3 categorization back to a RISC-1 categorization, resulting in the safety significant regulatory special treatment requirements being reimposed.

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References

10 CFR 50.69, "Risk-Informed Categorization and Treatment of Structures, Systems, and Components for Nuclear Power Reactors," dated November 22, 2004.

NRC Regulatory Guide 1.201, "Guidelines for Categorizing Structures, Systems, and Components in Nuclear Power Plants According to Their Safety Significance," dated May 1, 2006.

NRC Regulatory Guide 1.200, "An Approach for Determining the Technical Adequacy of Probabilistic Risk Assessment Results for Risk-informed Activities."

NEI 00-04, "10CFR 50.69 SSC Categorization Guideline."

South Texas Project Units 1 and 2, "Safety Evaluation on Exemption Request From Special Treatment Requirements of 10CFR Parts 21, 50, and 100," dated August 3, 2001 (ADAMS Accession No. ML012040370).

Two Options for a Risk-Informed Inservice Testing Program

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Two methods are available to nuclear power plant licensees to utilize risk-informed insights to focus resources for the inservice testing (IST) program. These methods are defined by Section 69 in Part 50 of Title 10 of the Code of Federal Regulations (10 CFR 50.69), and use of the American Society of Mechanical Engineers (ASME) risk-informed code cases (soon to be incorporated into the ASME Code for Operation and Maintenance of Nuclear Power Plants [OM Code] through subsection ISTE and Appendices). These methods for risk-informing IST programs have different requirements for categorizing the IST components into safety significant components and low safety significant components. This presentation will look at the different categorization methods and provide examples of how one component may have two different risk ranks.

This presentation will discuss a comparison of the risk categorizations for a typical IST program using both methods for all components. The comparison will be presented using existing data where available from risk rankings performed by different licensees using these two methods. It is anticipated that the results will show that 50% of IST components will be low safety significant using the NEI 00-04 guidelines, and 75% of the same components will be identified as low safety significant using the guidance in ASME Code Case OMN-3.

This presentation will describe what treatments would be applied to the components based on the rankings for the typical IST program components as defined above. Specifically, the components that are safety significant using both ranking methods will be maintained in the IST program and the testing requirements identified in the OM Code would be applied. The remaining components which are safety significant per the NEI 00-04 categorization process, are maintained in the IST program. However, based on the low safety significance categorization using OMN-3 guidance, these components may have relaxed testing requirements as identified in the component risk-informed Code Cases. The components that are low safety significant using the NEI 00-04 guideline are eligible for removal from the IST program scope in accordance with the provisions of 10 CFR 50.69.

The presentation is intended to provide insight into both processes to allow better understanding of the differences in levels of testing treatments based on the component's safety significance. It is also a purpose of this presentation to bring to light that several categorization processes have been approved for use by licensees. Given the use of categorization processes in other applications (e.g., motor-operated valves, air-operated valves, etc.), there may be situations where components that have been previously evaluated and categorized by one process to be evaluated in a different process with results that are different, but not unexpected.

Insight into Draft OM-29 – Alternative Treatment Recommendations for Inspection and Testing of Risk-Informed Safety Class 3 (RISC-3) Pumps and Valves

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Abstract

10 CFR 50.69, "Risk-Informed Categorization and Treatment of Structures, Systems, and Components for Nuclear Power Reactors," was approved by the U.S. Nuclear Regulatory Commission (NRC) on November 22, 2004. This new rule, once voluntarily adopted by licensees, effectively removes safety-related, low safety significant components (RISC-3) from the requirements of current American Society of Mechanical Engineers (ASME) Operations and Maintenance (OM) requirements as well as from other regulatory treatment requirements. In place of the current ASME OM requirements, licensees are required to establish and implement an inspection and testing strategy for Risk Informed Safety Class (RISC)-3 Structures, Systems, and Components (SSCs), and to periodically assess the performance results to determine with reasonable confidence that RISC-3 SSCs remain capable of performing their intended functions under design basis conditions. No ASME standards currently exist to assist licensees in defining and meeting these treatment requirements for the RISC-3 active safety functions that were previously treated under a licensee's regulatory In-service Testing (IST) Program in accordance with the ASME OM Code.

To address this need, a new ASME Standard, OM Part 29, is currently being developed. This paper will introduce the new high level recommendations that will be addressed in OM-29 (once approved) for RISC-3 SSCs that were previously detailed in the regulatory IST Program. This paper will identify key terminology (e.g., reasonable confidence) that must be consistently defined and applied within the nuclear industry for successful implementation of 10 CFR 50.69. In addition, this paper will provide insight into the transition from a detailed regulatory IST program to a 50.69 program for RISC-3 SSCs. Finally, this paper will provide the current status of the new OM Standard development, issues that are being addressed by the Team working on OM-29, milestones that are yet to be achieved, and when the new Standard should be ready for publication.

It is expected that attendees will gain valuable insight into the basis for OM-29, and how a 50.69 program for low safety significant, safety-related SSCs can effectively coexist with a regulatory program that will remain intact for safety significant, safety-related SSCs. In addition, attendees will gain insight into the benefits to be gained through implementation of ASME OM-29.

Introduction

The approach discussed in this paper requires that affected SSCs be initially categorized in accordance with an approved 10 CFR 50.69 process. This paper will not discuss the categorization process, but assumes that a robust, sound, stable categorization process has been followed, and that the resulting importance determinations have properly placed components into the appropriate categories. An acceptable categorization process to be followed is presented in 10 CFR 50.69, "Risk-Informed Categorization and Treatment of Structures, Systems, and Components for Nuclear Power Reactors," and is detailed in NRC Regulatory Guide (RG) 1.201, "Guidelines for Categorizing Structures, Systems, and Components in Nuclear Power Plants According to Their Safety Significance." In addition, RG 1.201 references and endorses (with exceptions) a guideline developed by the Nuclear Energy Institute (NEI) to support the categorization process for active functions, NEI-00-04, "10 CFR 50.69 SSC Categorization Guideline."

Once a component is categorized and placed into the proper categorization 'Box' (Risk Informed Safety Class -1, 2, 3, or 4), the scope of components within certain regulatory programs can be adjusted, and component treatments can then be appropriately applied recognizing the safety significance of the component. For SSCs that are categorized as RISC-1, regulatory safety-related controls (including ASME OM requirements) continue to be applied to these components. For SSCs categorized as RISC-2, possible additional treatments are assessed, focusing on the attributes which cause the SSC to be safety significant. For SSCs categorized as RISC-3, the special treatment requirements previously applied (including ASME OM requirements) can be reduced, as allowed by 10 CFR 50.69. For SSCs categorized as RISC-4, industrial controls, as before, continue to be applied.

This paper will focus on components categorized as RISC-3, and will specifically look at safety-related pumps and valves that were previously tested under the ASME OM Codes and Standards. As special treatment requirements are reduced on these SSCs as permitted under 10 CFR 50.69, guidance for consistent industry approaches is necessary to ensure that these components continue to reliably support their designed safety-related functions.

Nomenclature

Reasonable Assurance – a justifiable level of confidence based on objective and measurable facts, actions, or observations, which infer adequacy

Reasonable Confidence – a level of confidence based on facts, actions, knowledge, experience, and/or observations, which is deemed to be adequate.

Risk-Informed Safety Classifications (RISC) – the segregation of categorized components into specific groupings. The four groupings identified in 10 CFR 50.69 include:

- RISC-1 – safety-related, safety significant
- RISC-2 – non-safety related, safety significant
- RISC-3 – safety related, low safety significant
- RISC-4 – non-safety related, low safety significant

Special Treatment Requirements – the additional controls placed on safety-related equipment which exceed the normal controls placed on non-safety related equipment.

The Need For A New ASME Standard

As stated earlier, 10 CFR 50.69 and Regulatory Guide 1.201 were recently issued by the Nuclear Regulatory Commission (NRC). 10 CFR 50.69 is a voluntary rule, and for licensees who choose to adopt the rule through submittal and approval of a License Amendment Request, significant benefits exist for both the regulator and licensee to better focus resources and attention on those components and activities that are truly safety significant. The overall result of implementing a 10 CFR 50.69 approach is enhanced nuclear safety while simultaneously reducing the burden placed on low safety significant, safety-related components.

Prior to the existence of the 10 CFR 50.69 approach, all safety-related pumps and valves with active safety functions were included in a regulatory IST Program per 10 CFR 50.55a(f). The requirements of 10CFR 50.55a(f) impose periodic testing and trending of IST pumps and valves, as well as actions to take when expected test/trend values are not met. The inclusion of components into the IST Program was deterministically driven by the safety-related classification of associated pumps and valves. Applying 10 CFR 50.69 to an existing IST Program, the scope of components subject to regulatory In-service Testing is adjusted to include only safety-related, safety significant (RISC-1) pumps and valves. Existing regulatory controls (including ASME OM requirements) continue to be imposed upon these RISC-1 components. However, for safety-related pumps and valves determined to be RISC-3 (low safety significant), the rigorous controls imposed upon RISC-1 components are no longer necessary – RISC-3 components can be removed from the IST Program scope, and alternate treatment approaches apply.

It is important to note that RISC-3 components remain safety-related following their categorization – they are not reclassified as non-safety related even though they have been determined to be low safety significant. It is also important to note that the design function of the RISC-3 components did not change with categorization – these components are still expected to satisfy, with reasonable confidence, their intended functions under design basis conditions. Therefore, upon implementation of a 10 CFR 50.69 approach onto an existing IST Program, a family of safety-related components with active safety functions will be removed from the scope of the IST Program and will no longer be subjected to the ASME OM requirements. The void that exists for these low safety significant pumps and valves necessitates the creation of some degree of industry guidance to ensure consistency in treatment of these components. The development of OM-29 is focused on proactively addressing this need.

THE SCOPE OF OM-29

10 CFR 50.69, paragraphs (d)(2) and (e)(3), provide some specific guidance for the treatment of RISC-3 SSCs as follows:

> (d)(2) RISC-3 SSCs. The licensee or applicant shall ensure, with reasonable confidence, that RISC-3 SSCs remain capable of performing their safety-related functions under design basis conditions, including seismic conditions and environmental conditions and effects throughout their service life. The treatment of RISC-3 SSCs must be consistent with the categorization process. Inspection and testing, and corrective action shall be provided for RISC-3 SSCs.

> (d)(2)(i) Inspection and testing. Periodic inspection and testing activities must be conducted to determine that RISC-3 SSCs will remain capable of performing their safety-related functions under design basis conditions; and

> (d)(2)(ii) Corrective action. Conditions that would prevent a RISC-3 SSC from performing its safety-related functions under design basis conditions must be corrected in a timely manner. For significant conditions adverse to quality, measures must be taken to provide reasonable confidence that the cause of the condition is determined and corrective action taken to preclude repetition.

> (e)(3) RISC-3 SSCs. The licensee shall consider data collected in 50.69(d)(2)(i) for RISC-3 SSCs to determine if there are any adverse changes in performance such that the SSC unreliability values approach or exceed the values used in the evaluations conducted to satisfy 50.69(c)(1)(iv). The licensee shall make adjustments as necessary to the categorization or treatment processes so that the categorization process and results are maintained valid.

The primary purpose of OM-29 will be to provide the necessary guidance to ensure that the above requirements of 10 CFR 50.69 are consistently understood and satisfied among all industry users that choose to adopt 50.69.

Also, 10 CFR 50.69 introduces the term 'reasonable confidence', yet the NRC chose not to explicitly define this term within the rule language. In addition, an explicit definition of the term 'reasonable assurance' also does not exist within the regulations. The lack of explicit definitions creates a certain degree of regulatory uncertainty when discussing 50.69 and addressing these terms. The intent is that 'reasonable assurance' is necessary for RISC-1 pumps and valves to satisfy their design basis requirements. Reasonable assurance implies some type of demonstration testing to prove with high confidence that RISC-1 pumps and valves will satisfy their design functions when demanded during design basis accidents. Reasonable assurance also implies a rigorous documentation trail to provide objective evidence that the appropriate demonstrations were completed, and that established acceptance criteria were satisfied. It is agreed that this approach to testing is appropriate for safety-related, safety significant pumps and valves, and RISC-1 SSCs deserve appropriate focus by both the licensee and the regulator.

However, safety-related, low safety significant pumps and valves should not require the same degree of rigor placed on RISC-1 components. A lesser degree of control over RISC-3 pumps and valves is permitted by 10 CFR 50.69, but it is the licensee's responsibility to define when 'reasonable confidence' is achieved. By defining these (and other) key terms within OM-29, a common understanding will be established among a wide range of industry users, and the regulator will better recognize how industry consistently applies these key terms.

Even with the term 'reasonable confidence' defined in OM-29, the question that is invariably raised is 'How much treatment is enough to establish reasonable confidence such that a RISC-3 SSC will satisfy its design functions under design basis conditions?'. The NRC was appropriately vague within the 50.69 rule language when discussing 'reasonable confidence', leaving the detail development to experts within the industry. An example of this may be the question of 'What constitutes an appropriate testing activity on a RISC-3 pump?'. Some within the industry may reason that an acceptable bump test of a coupled pump-motor combination is sufficient to verify the operational readiness of a RISC-3 pump, while others may contend that a more detailed test is required. The intent of OM-29 will not be to explicitly define in detail what exactly must be done to achieve reasonable confidence in each and every application; however, sufficient guidance will be offered in establishing a basis of reasonable confidence among industry users. OM-29 will address the types of tests and inspections that can be performed, testing frequency, extent of data to be taken, extent of trends to be maintained, etc., so that a consistent industry position on reasonable confidence is established.

Standard Development

An ASME OM Standards committee has been established to develop the draft OM-29, and to process this proposed standard through the ASME balloting and approval process. This committee includes industry experts in the various fields affecting pump and valve operation and maintenance, and includes expertise in 10 CFR 50.69 development and implementation.

OM-29 is still in the developmental stages, with the committee focused on incorporating the full scope of the proposed standard as detailed in the previous section. No significant technical issues have been identified to date; however, it is recognized that this standard establishes an approach for safety-related component treatment which varies significantly from past historical practices. Based on this fact, it is expected that extensive stakeholder involvement will be required to develop an appropriately worded standard which satisfies the requirements of 10 CFR 50.69.

It is expected that the draft OM-29 will be available for initial balloting by the end of 2006. Based on the extent of comments received through the balloting process, OM-29 should be ready for approval no later than early 2008. This targeted timeframe aligns favorably with industry 50.69 needs. Initial industry applications to adopt 10 CFR 50.69 are expected by the end of 2006, with NRC approval of the initial 50.69 application expected by the end of 2007.

In-Service Testing Program Impacts

As discussed earlier, implementation of a 10 CFR 50.69 approach for safety-related pump and valve testing does not eliminate the need to maintain a regulatory IST Program – the IST Program is still required to programmatically satisfy the testing requirements for the RISC-1 pumps and valves. However, the scope of the IST Program will be reduced by removing the RISC-3 pumps and valves from the regulatory program. The net effect is that increased focus by both the regulator and licensee can be placed on those safety-significant components that remain within the IST Program, while less focus (not to be confused with no focus) can be placed on the RISC-3 components removed from the Program. The overall result is expected to be a net nuclear safety benefit. In addition, the oversight burden on the regulator is reduced, and the testing and administrative burden on the licensee is also reduced, resulting in cost savings.

The treatments applied to RISC-3 pumps and valves removed from the IST Program will be similar to the treatments currently applied to non-safety related pumps and valves. These industrial practices have been demonstrated to be effective by domestic licensees through continued high capacity factors and high reliabilities that are noted in the balance-of-plant. The fact that a component is RISC-3 does not imply that maintenance practices and operational oversight can be eliminated. As stated in 10 CFR 50.69, inspection, testing, and a corrective action program are still required as a minimum. As currently done in commercial applications, it is expected that licensees will apply appropriate treatments to RISC-3 SSCs such that the components will perform as expected when demanded. OM-29 will provide guidance to ensure that licensees are consistent in their practices for RISC-3 pumps and valves.

Conclusion

The Nuclear Regulatory Commission's approval of 10 CFR 50.69 has created an opportunity where licensees can determine the overall importance of SSCs through an approved categorization process. For safety-related pumps and valves treated under a regulatory IST Program, upon implementation of a 10 CFR 50.69 approach, a certain population of these components determined to be low safety significant will be removed from the IST Program scope.

Proposed OM-29 is an effort to proactively develop a consistent, approved methodology to treat these RISC-3 pumps and valves outside of previous special treatment requirements. OM-29, when developed and approved, will provide guidance to ensure that RISC-3 pumps and valves are adequately tested and inspected, and that results are assessed to provide reasonable confidence that these SSCs remain capable of performing their intended functions under design basis conditions.

Unless a standard like OM-29 is developed, licensees who voluntarily adopt 10 CFR 50.69 will be required to determine the needed tests and inspections for RISC-3 pumps and valves without the benefit of an approved industry standard. This situation will invariably lead to certain licensees providing too much treatment to RISC-3 pumps and valves while others may provide too little treatment. OM-29 will ensure consistency in industry application, and will eliminate significant regulatory uncertainty in the implementation phase of 10 CFR 50.69.

Acknowledgements

This paper acknowledges the dedicated people of STP's Risk Management Team, whose vision and tenacity continues to advance the use of nuclear risk insights within the industry, thus promoting the advancement of safe and efficient nuclear power production.

References

10 CFR 50.69, "Risk-Informed Categorization and Treatment of Structures, Systems, and Components for Nuclear Power Reactors," dated November 22, 2004.

Regulatory Guide 1.201, "Guidelines for Categorizing Structures, Systems, and Components in Nuclear Power Plants According to Their Safety Significance," dated May 1, 2006.

10CFR 50.55a(f), "Codes and Standards."

NEI 00-04, "10CFR 50.69 SSC Categorization Guideline."

Regulatory Guidance Supporting 10 CFR 50.69 Categorization Requirements

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Background

The NRC has established a set of regulatory requirements for commercial nuclear reactors to ensure that a reactor facility does not impose undue risk to the health and safety of the public, thereby providing reasonable assurance of adequate protection to public health and safety. The current body of NRC regulations and their implementation are largely based on a "deterministic" approach.

This deterministic approach establishes requirements for engineering margin and quality assurance in design, manufacture, and construction. In addition, it assumes that adverse conditions can exist (e.g., equipment failures and human errors) and establishes a specific set of design basis events (DBEs). The deterministic approach contains implied elements of probability, from the selection of accidents to be analyzed (or not analyzed) to the system-level requirements for emergency core cooling. The deterministic approach then requires that the licensed facility include safety systems capable of preventing and/or mitigating the consequences of those DBEs to protect public health and safety. Those structures, systems, and components (SSCs) at the nuclear power plant necessary to defend against the DBEs are defined as "safety-related," and these SSCs are the subject of many regulatory requirements designed to ensure that they are of high quality and high reliability, and have the capability to perform during postulated design basis conditions.

These prescriptive requirements as to how licensees are to treat SSCs, especially those defined as "safety-related," are referred to as "special treatment requirements." The special treatment requirements were developed to provide greater assurance, beyond that provided by normal industrial practices, that these SSCs would perform their functions under particular conditions, with high quality and reliability, for as long as they are part of the plant. These include particular examination techniques, testing strategies, documentation requirements, personnel qualification requirements, independent oversight, etc. In many instances, these special treatment requirements were developed as a means to gain assurance when more direct measures could not show that SSCs were functionally capable.

Special treatment requirements are imposed on nuclear reactor applicants and licensees through numerous regulations. These requirements specify different scopes of equipment for different special treatment requirements depending on the specific regulatory concern, but are derived from consideration of the deterministic DBEs.

A probabilistic approach to regulation enhances and extends the traditional deterministic approach by allowing consideration of a broader set of potential challenges to safety, providing a logical means for prioritizing these challenges based on safety significance, and allowing consideration of a broader set of resources to defend against these challenges. In contrast to the deterministic approach, probabilistic risk assessments (PRAs) address credible initiating events by assessing the event frequency. Mitigating system reliability is then assessed, including the potential for common cause failures. The probabilistic approach goes beyond the single failure requirements used in the deterministic approach. The probabilistic approach

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This paper was prepared by staff of the U.S. Nuclear Regulatory Commission. It may present information that does not currently represent an agreed-upon NRC staff position. NRC has neither approved nor disapproved the technical content.

to regulation is therefore considered an extension and enhancement of traditional regulation by considering risk in a more coherent and complete manner.

The Commission published a Policy Statement on the "Use of Probabilistic Risk Assessment" on August 16, 1995 (60 FR 42622). In the policy statement, the Commission stated that the use of PRA technology should be increased in all regulatory matters to the extent supported by the state of the art in PRA methods and data, and in a manner that supports the NRC's traditional defense-in-depth philosophy. The policy statement also stated that, in making regulatory judgments, the Commission's safety goals for nuclear power reactors and subsidiary numerical objectives (on core damage frequency and containment performance) should be used with appropriate consideration of uncertainties.

To implement this Commission policy, the NRC staff developed guidance on the use of risk information for reactor license amendments and issued Regulatory Guide (RG) 1.174, "An Approach for Using Probabilistic Risk Assessment in Risk-Informed Decisions on Plant-Specific Changes to the Licensing Basis." This RG provided guidance on an acceptable approach to risk-informed decision-making consistent with the Commission's policy, including a set of key principles. These principles include:

1. Be consistent with the defense-in-depth philosophy;
2. Maintain sufficient safety margins;
3. Any changes allowed must result in only a small increase in core damage frequency or risk, consistent with the Commission's Safety Goal Policy Statement; and,
4. Incorporate monitoring and performance measurement strategies.

In addition to RG 1.174, the NRC also issued other regulatory guides on risk-informed approaches for specific types of applications.

RULEMAKING

On December 23, 1998, the NRC staff recommended in SECY-98-300 that risk-informed approaches to the application of special treatment requirements be developed as one application of risk-informed regulatory changes. This recommendation was Option 2 in SECY-98-300, which the Commission approved in a staff requirements memorandum (SRM) dated June 8, 1999. The stated purpose of the Option 2 rulemaking was to develop an alternative regulatory framework that would enable licensees, using a risk-informed process for categorizing SSCs according to their safety significance (i.e., a decision that considered both traditional deterministic insights and risk insights), to reduce unnecessary regulatory burden for SSCs of low safety significant by removing these SSCs from the scope of special treatment requirements. As part of this process, those SSCs found to be of safety significance would be brought under a greater degree of regulatory control through the requirements being added to the rule, which are designed to maintain consistency between actual performance and their performance credited in the assessment process that determines their significance. As a result, both the NRC and industry should be able to better focus their resources on regulatory issues of greater safety significance.

By an SRM dated January 31, 2000, the Commission approved publication of an Advanced Notice of Proposed Rulemaking (ANPR) and the associated rulemaking plan to evaluate strategies to make the scope of the nuclear power reactor regulations that impose special treatment risk-informed. Following the ANPR stage, in which over 200 comments were received, the Commission approved, in an SRM dated March 28, 2003, issuance of a proposed new rule for public comment. On November 22, 2004, the NRC amended its regulations and adopted a new section, referred to as §50.69, within Title 10, Part 50, of the Code of Federal Regulations (69 FR 68008). This newly promulgated regulation allows power reactor licensees and license applicants to implement an alternative regulatory framework for establishing the requirements for treatment of SSCs using a risk-informed method of categorizing SSCs according to their safety significance. Under this framework, the risk-informed process removes SSCs of low safety significance from the scope of certain identified special treatment requirements, and revises requirements for SSCs of greater safety significance.

RULE OVERVIEW

Section 50.69 represents an alternative set of requirements whereby a licensee or applicant may voluntarily undertake categorization of its SSCs consistent with the requirements in §50.69(c), remove the special treatment requirements listed in §50.69(b) for SSCs that are determined to be of low individual safety significance, and implement alternative treatment requirements provided in §50.69(d). The regulatory commitments not removed by §50.69(b) continue to apply as well as the requirements specified in §50.69. The rule contains requirements by which a licensee categorizes SSCs using a risk-informed process, adjusts treatment requirements consistent with the relative significance of the SSC, and manages the process over the lifetime of the plant. To implement these requirements, a risk-informed categorization process is employed to determine the safety significance of SSCs and to place the SSCs into one of four risk-informed safety class (RISC) categories. The safety functions include both the design basis functions (derived from the "safety-related" definition), as well as functions credited for severe accidents (beyond design basis). The determination of safety significance utilizes an integrated decision-making process, which involves a panel of plant personnel with diverse expertise to consider both risk insights and traditional engineering insights. Treatment for the SSCs is required to be applied as necessary to maintain functionality and reliability, and is a function of the category into which the SSC is categorized. Finally, assessment activities are conducted to make adjustments to the categorization and treatment processes as needed so that SSCs continue to meet the applicable requirements. The rule also contains requirements for obtaining prior NRC review and approval of the categorization process and for maintaining certain plant records and reports.

The overall structure of the rule is as follows:

1. §50.69(a) defines the terms specific to this rule, such as "risk-informed safety class (RISC)" and "safety significant function."

2. §50.69(b) identifies the special treatment requirements that may be removed for SSCs determined to be of low individual safety significance. This paragraph also identifies who may implement §50.69 and provides the submittal requirements for implementation (i.e., via a license amendment).

3. §50.69(c) provides the requirements governing the categorization of SSCs (i.e., the determination of SSC safety significance), which is built around an integrated decision-making process and the use of a plant-specific PRA in providing reasonable confidence that implementation of the rule for various systems will have no more than a small impact on risk throughout the life of its implementation.

4. §50.69(d) applies treatment requirements based on the RISC category assigned to the SSCs. For safety significant SSCs, all requirements are maintained in addition to the §50.69(d)(1) requirements for the beyond design basis functions. For low safety significant SSCs, the special treatment requirements identified in §50.69(b) are removed and replaced with high-level treatment requirements. This paragraph also contains corrective action requirements.

5. §50.69(e) contains monitoring and feedback requirements that are structured to maintain the validity of the categorization and treatment processes over time.

6. §50.69(f) and §50.69(g) contain documentation and reporting requirements.

REGULATORY GUIDANCE

In parallel with the rulemaking activities, the NRC staff interacted with the industry and public stakeholders in the development of regulatory guidance associated with the categorization process required by the rule. In May 2006, the NRC issued for trial use, Revision 1 of RG 1.201, "Guidelines for Categorizing Structures, Systems, and Components in Nuclear Power Plants According to Their Safety Significance," which describes a method that the NRC staff considers acceptable for use in complying with the Commission's requirements in §50.69 with respect to the categorization of SSCs that are considered in risk-informing special treatment requirements. This categorization method endorses, with a number of clarifications, the process that the Nuclear Energy Institute (NEI) describes in Revision 0 of its guidance document NEI 00-04, "10 CFR 50.69 SSC Categorization Guideline," dated July 2005, to determine the safety significance of SSCs and the appropriate RISC category for each SSC.

10 CFR 50.69 does not replace the existing “safety-related” and “nonsafety-related” categorizations. Rather, 10 CFR 50.69 divides these categories into two subcategories based on the SSC’s safety significance. The figure below provides a conceptual understanding of the new risk-informed SSC categorization scheme. The figure depicts the current safety-related versus nonsafety-related SSC categorization scheme with an overlay of the new safety-significance categorization. In the traditional deterministic approach, SSCs were categorized as either “safety-related” or “nonsafety-related.” This division is shown by the vertical line in the figure. Risk insights, including consideration of severe accidents, are used to identify SSCs as being either safety-significant or low-safety-significant (LSS) (as shown by the horizontal line in the figure). This results in SSCs being grouped into one of four RISC categories, as represented by the four boxes in Figure 1 below.

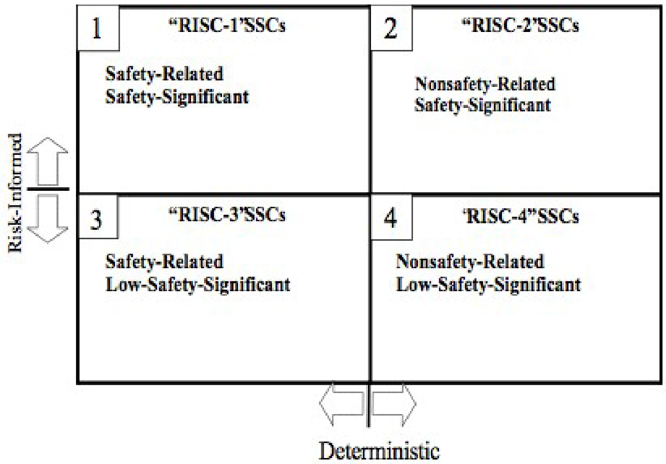

Figure 1. §50.69 RISC Categories

RISC-1 SSCs are safety-related SSCs that the risk-informed categorization process determines to be significant contributors to plant safety. Licensees must continue to ensure that RISC-1 SSCs perform their safety-significant functions consistent with the categorization process, including those safety-significant functions that go beyond the functions defined as safety-related for which credit is taken in the categorization process.

RISC-2 SSCs are those that are defined as nonsafety-related, although the risk-informed categorization process determines that they are significant contributors to plant safety on an individual basis. The NRC staff recognizes that some RISC-2 SSCs may not have existing special treatment requirements. As a result, the focus for RISC-2 SSCs is on the safety-significant functions for which credit is taken in the categorization process.

RISC-3 SSCs are those that are defined as safety-related, although the risk-informed categorization process determines that they are not significant contributors to plant safety. Special treatment requirements are removed for RISC-3 SSCs and replaced with high-level requirements. These high-level requirements are intended to provide sufficient regulatory treatment, such that these SSCs are still expected to perform their safety-related functions under design-basis conditions, albeit at a reduced level of assurance compared to the current special treatment requirements. However, §50.69 does not allow these RISC-3 SSCs to lose their functional capability or be removed from the facility.

Finally, RISC-4 SSCs are those that are defined as nonsafety-related, and that the risk-informed categorization process determines are not significant contributors to plant safety. Section 50.69 does not impose alternative treatment requirements for these RISC-4 SSCs. However, as with the RISC-3 SSCs, changes to the design bases of RISC-4 SSCs must be made in accordance with current applicable design change control requirements (if any), such as those set forth in 10 CFR 50.59.

The safety significance of SSCs is determined using an integrated decision-making process, which blends risk insights, new technical information, and operational experience and feedback through the involvement of a group of experienced licensee-designated professionals. Through the §50.69 categorization process, some safety-related SSCs will be determined to be of low or no safety significance and these SSCs will be categorized as RISC-3 SSCs, while other safety-related SSCs will be identified as safety-significant and will be categorized as RISC-1 SSCs. Likewise, some non-safety-related SSCs will be categorized as safety-significant and be categorized as RISC-2 SSCs and other SSCs will remain of low or no safety significance and be categorized as RISC-4 SSCs. Those SSCs in systems that a licensee chooses not to evaluate using the §50.69 SSC categorization process remain as safety-related and non-safety-related.

RG 1.201 endorses NEI 00-04, which provides detailed guidance on categorizing SSCs for those plants that voluntarily adopt §50.69. Section C of RG 1.201 contains a number of regulatory positions, including:

1. The NRC staff recognizes that the implementation of the entire process described in Revision 0 of NEI 00-04 (i.e., Sections 2 through 12) is integral to providing reasonable confidence in the evaluations required by §50.69(c)(1)(iv). All aspects of the guidance are important and interrelated. Sections 2 through 7 and Section 10 describe the processes used to determine the set of SSCs, for which unreliability is adjusted in the risk sensitivity study described in Section 8, which is used to confirm that the categorization process results in acceptably small increases in core damage frequency (CDF) and large early release frequency (LERF). Section 9 describes the integrated decision-making panel (IDP) function of reviewing and ensuring that the system functions and operating experience have been appropriately considered in the process. Finally, Sections 11 and 12 describe the processes that provide reasonable confidence that the validity of the categorization process is maintained. Thus, all aspects of Revision 0 of NEI 00-04 must be followed to achieve the reasonable confidence in the evaluations required by §50.69.

2. To categorize SSCs under §50.69, licensees must use risk evaluations and insights that cover the full spectrum of potential events (i.e., internal and external initiative events) and the range of plant operating modes (i.e., full power, low power, and shutdown operations). The rule requires at least a peer-reviewed PRA that addresses internal initiating events at full power. Revision 0 of NEI 00-04 allows the use of non-PRA-type evaluations when PRAs have not been performed to address other aspects (e.g., seismic margins analysis). Such non-PRA-type evaluations will result in more conservative categorization in that special treatment requirements will not be allowed to be relaxed for SSCs that are relied upon in these non-PRA-type evaluations. Thus, it should be recognized that the degree of relief provided under §50.69 will be commensurate with the type of evaluation supporting the categorization process.

3. The licensee or applicant is expected to document the technical adequacy of its risk evaluations for this application, in accordance with RG 1.200, "An Approach for Determining the Technical Adequacy of Probabilistic Risk Assessments." Currently the only endorsed standard in RG 1.200 only addresses internal initiating events at full power. Therefore, until standards are endorsed for PRAs covering other scope areas (e.g., external initiating events and shutdown and low power operations), as well as non-PRA-type analyses (e.g., seismic margins analysis), submittals requesting to implement §50.69 will need to document the bases for why the method employed is technically adequate for this application. Once a standard for these other PRA aspects is developed and endorsed by the NRC via revisions to RG 1.200, the NRC expects the licensee or applicant to use that standard as endorsed by the NRC, as part of the plant-specific application, to demonstrate the technical adequacy of the corresponding aspect of the PRA, if it is used in the categorization process.

4. Mechanisms that could lead to large increases in CDF and LERF, which could potentially invalidate the assumptions underlying the categorization process, are the emergence of extensive common cause failures impacting multiple systems and significant unmitigated degradation. However, for these types of impacts to occur, the mechanisms that lead to failure, in the absence or relaxation of treatment, would have to be sufficiently rapidly developing or not self-revealing, such that there would be few opportunities for early detection and corrective action. The NRC staff recognizes that the guidance provided in Section 12.4 of Revision 0 of NEI 00-04 in meeting the §50.69(d) and (e) requirements of inspection and testing, corrective action, and feedback are intended to preclude reaching such unacceptable SSC performance.

5. The NRC staff believes that the guidance in NEI 00-04, as clarified by the regulatory positions in RG 1.201, provides an acceptable approach for categorizing SSCs to support implementation of §50.69.

Through implementation of §50.69, both the NRC and industry should be able to better focus their resources on regulatory issues of greater safety significance, while providing reasonable assurance of adequate protection to public health and safety.

Session 2(b): ASME Code Issues

Session Chair

L.J.Victory, Jr.

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IST BASES DOCUMENTS – THEN AND NOW

-OR-

WHAT HAS YOUR BASES DOCUMENT DONE FOR YOU LATELY?

John J. Dore, Jr.

Dore Technical Resources, Inc.

Owners of nuclear power plants began to develop Inservice Testing (IST) Bases Documents in the 1980s as a means to provide documentation of the reasons for including, or often times more importantly, the reasons for not including various “safety-related” pumps and valves in their Pump and Valve IST Programs. These early attempts were mostly prompted by frequent turnovers in personnel responsible for plant IST Programs combined with notable levels of uncertainty and inconsistency regarding the application of IST requirements. These first IST Bases Documents were based on the requirements of Subsections IWP and IWV of Section XI of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (B&PV Code). Although these requirements were fairly straightforward, the means of applying them to plant-specific designs and licensing applications often were not.

With the introduction of Part 6 (OM-6) and Part 10 (OM-10) of the ASME Operation and Maintenance (OM) Standards in 1988, and their incorporation by reference into the 1989 Edition of Subsections IWP and IWV of Section XI of the ASME B&PV Code, combined with publication of NRC Generic Letter 89-04 (April 1989), “Guidance on Developing Acceptable Inservice Testing Programs,” several options and acceptable alternatives for compliance with IST requirements were identified. This broadening of methods for performing IST using methods acceptable to the NRC made it more prudent than ever to document each component’s safety function or functions, as well as the methods that were used to comply with IST requirements. IST Bases Documents became an increasingly popular means of providing such documentation.

NUREG-1482, “Guidelines for Inservice Testing at Nuclear Power Plants,” published in April 1995, provided much additional guidance and expanded the possibilities for achieving compliance or developing suitable alternatives. In addition, NUREG-1482 (Section 2.4.4) endorsed the IST Bases Document as a means of ensuring continuity of the IST Program when the responsibilities of personnel or groups change. Finally, the incorporation by reference of the ASME Code for Operation and Maintenance of Nuclear Power Plants, 1995 Edition (OM Code-1995) with OMa-1996 Addenda into Section 50.55a(b)(3) of Title 10 of the Code of Federal Regulations [10 CFR 50.55a(b)(3)] in September 1999, followed by the endorsement of later Editions/Addenda, significantly expanded the possibilities for the development and implementation of acceptable IST Programs. Conversely, all of these factors have significantly expanded the possibilities for uncertainty and error, as well.

IST Bases Documents vary widely in quality, content and scope. With so many options and alternatives available, it is more important than ever to develop and maintain a Bases Document that adequately describes the bases and means of conducting IST. This presentation will discuss the various means for compliance with current IST requirements and examine the types of information and level of detail that should be provided in the IST Bases Document.

Appendix J Program Owner's Group (APOG)

John Scranton – *APOG Vice Chairperson*

Entergy Nuclear – FitzPatrick Plant

Abstract:

The Appendix J Program Owners Group (APOG) provides a primary reactor containment leakage testing forum designed to share operating experience and knowledge, identify testing issues pertaining to methodology, practices and component performance as areas targeted for potential improvement, affords a centralized repository for storage of and access to technical information and documents, and assist members with improved implementation of their containment leakage testing program. This paper will discuss how APOG is accomplishing these goals and addressing challenges facing the nuclear industry in Appendix J Program management and implementation.

Background:

The Appendix J Program Owner's Group was formed following a meeting of IST and Appendix J Program Owner's held in June 2003 in Scottsdale, Arizona. Since then, APOG membership has grown to approximately 60% of the nuclear power industry's operating plants. Annual membership meetings have been held at three separate nuclear plant sites; Ginna in 2004, Salem/Hope Creek in 2005 and Brunswick this past June. APOG has elected to hold its annual meetings at member plants to provide a more focused venue to interact and to control logistical costs. These meetings have been a resounding success in opening the channels of communication, facilitating information exchange and providing a forum for identifying germane program management issues and concerns. The face to face interaction allows the Appendix J Program Owners to develop consensus on aspects of program management and implementation. Additionally, vendor, consultant, regulatory, code and other invited guest participation provides an excellent opportunity to be introduced to the latest technologies, standards, regulations and the workings thereof.

A Steering Committee (SC) was organized shortly after the initial 2003 meeting which consisted of volunteer peers and was formalized at the 2004 meeting at Ginna. The SC consists of seven members representing Constellation Energy, Duke, Energy Northwest, Entergy, Exelon, Nuclear Management Corporation and Public Service Electric and Gas.

APOG Charter Summary:

The APOG Charter states in part that the Purpose of the Appendix J Program Owner's Group is to: Review applicable guides, standards, codes, regulations and documents that govern Appendix J programs for uniform application across the industry and provide expert interpretation of these documents; Provide a means to share industry knowledge and resources and exchange technical information relating to the application, testing and maintenance of all components governed by Appendix J; Increase regulatory awareness and provide the Industry with a means to impact legislative issues as well as provide a collaborative effort to promote cost reduction, error reduction, improve performance and maintain safety margins within the nuclear industry.

The APOG Charter also provides the details pertaining to the organization's structure, operation and responsibilities of and to its members.

Technical Papers Developed or Under Development:

Since its initial inception, APOG has been soliciting technical issues from its membership and working to develop Technical Position Papers to address these issues. The APOG Steering Committee meets regularly to discuss the status on these issues and monitor the progress of the task teams working on issues. Some of the issues APOG has taken on are:

- As-Found Local Leak Rate Test (LLRT) requirements
- Use of 25% Grace for Type B and C LLRTs

- Vent and Purge Valves LLRT Frequency
- Main Steam Isolation Valve (MSIV) LLRT issues
- Creation of an Integrated Leakage Rate Testing Planning and Performance Guideline
- Appendix J Program Scope Reduction
- Creation of a Comprehensive Training and Qualification Guideline
- Development of a Consistent Methodology for Assessing Appendix J Components within the Maintenance Rule
- Development of an Operating Experience Database on the APOG web site
- Development of a LLRT Failure Database on the APOG web site

As-Found LLRT Requirements:

It has been noted that many plants perform unnecessary as-found testing. This results in excessive costs and unwarranted radiological exposure. This Technical Position Paper provides guidance on the Regulatory requirements of 10CFR50 Appendix J and associated references related to as-found Type B and C tests on components that are tested at a nominal interval of 30 months. 10CFR50 Appendix J Option B references Regulatory Guide 1.163. This Regulatory Guide in turn references both NEI 94-01, and ANSI/ANS 56.8-1994. 10CFR50 Appendix J Option B is a performance-based rule. The basis for the performance based Appendix J program is NEI 94-01. ANSI/ANS 56.8-1994 specifies testing and program methodology. Under Option B, Type B and C components initially have a base test interval of 30 months. These documents complement each other, differing only in the application of extensions to a base frequency.

Neither Option A or B to 10CFR50 Appendix J, nor ANSI/ANS 56.8-1994 require As-Found Types B and C leakage rate testing for components included in a Type B and C testing program. EXCEPT, ANSI/ANS 56.8-1994, As-Found (Section 3.2.8) and As-Left (Section 3.2.9) leakage rate testing of Types B and C tested components is required to ascertain a pathway's change in Minimum Pathway Leakage Rate (MNPLR) prior to and following adjustments or isolation during the Primary Containment Integrated Leakage Rate Test (PCILRT). Note, that individual component performance is not required, only the changes in the pathway MNPLR.

NEI 94-01 describes an acceptable approach for implementing the optional performance-based requirements of Option B to 10 CFR 50 Appendix J. Therefore, NEI 94-01 sections which address as-found testing do not apply to nominal 30 month interval components.

Use of 25% Grace for Type B and C LLRTs:

The 30, 60 and 120 month intervals associated with Type B and C testing does not necessarily coordinate well with 18 or 24 month fuel cycles. These test intervals may come due at a time when the plant is not in the mode required to support testing (i.e., refuel outage). Rather than shorten these intervals, and lose the benefit of the Performance Based design of 10 CFR 50 Appendix J Option B, the regulation and associated references allow for the application of a 25% grace period. This Technical Position Paper provides guidance on the Regulatory requirements of 10 CFR 50 Appendix J and associated references related to the application of a 25% extension to the Type B and C test intervals. 10 CFR 50 Appendix J Option B references Regulatory Guide 1.163, "Performance-Based Containment Leak-Test Program", which in turn endorses and references NEI 94-01 Rev 0, "Industry Guideline for Implementing Performance Based Option of 10 CFR Part 50, Appendix J".

Section 10.1 of NEI 94-01 states: "Consistent with standard scheduling practices for Technical Specifications Required Surveillance's, intervals for the recommended surveillance frequency for Type B and Type C testing given in this section may be extended by up to 25 percent of the test interval, not to exceed 15 months."

Section 11.3 of NEI 94-01 further clarifies: "An extension of up to 25 percent of the test interval (not to exceed 15 months) may be allowed on a limited basis for scheduling purposes only."

NEI 94-01 states that the basis for utilizing test interval extensions for Appendix J is consistent with standard scheduling practices for Technical Specifications Required Surveillance's. Standard Technical Specifications Bases state the following for application of test interval extension: "these provisions are not intended to be used repeatedly merely as an operational convenience to extend surveillance intervals (other than those consistent with refueling outages)." Therefore, the application of a 25% extension to the intervals allowed in NEI 94-01 is acceptable when tests requiring an outage come due during a non-outage period (i.e., a 60 month interval ends between outages). Additionally, being that RG 1.163 does not endorse section 11.3.2 of NEI 94-01, extending a Type C test beyond three refueling outages would be inconsistent with these guidelines.

Vent and Purge Valves LLRT Frequency:

While some nuclear plants are required by their Technical Specifications to leak rate test their large Vent and Purge valves more frequently than 30 months, this Technical Position Paper is intended to address only the 30 month test interval specified in Section C.2 of Regulatory Guide 1.163. This exception states: "…the interval for Type C tests for main steam and feedwater isolation valves in Boiling Water Reactors (BWRs), and containment purge and vent valves in Pressurized Water Reactors (PWRs) and BWRs, should be limited to 30 months as specified in Section 3.3.4 of ANSI/ANS-56.8-1994, with consideration given to operating experience and safety significance."

The term "containment vent and purge valves" has historically referred to large valves with resilient seats (typically the butterfly style). These valves have a history of exhibiting abnormally high local leak rates due to their large size, aging of the resilient seats, and hardening of the resilient seats for those valves located outside (i.e., exposure to cold weather). These issues were further addressed in a Generic Issue regarding containment leakage due to seal deterioration. The fact that containment purge and vent valves' resilient seats made them more likely to fail leak rate tests, and that most were very large pathways which increased the potential consequences of a failure, resulted in the documenting of this generic issue. This precipitated the requirement to test containment purge and vent valves in excess of Appendix J testing required intervals.

Based on the above information, it is prudent to consider classifying purge and vent valves as "large valves with resilient seats" which are used for venting and purging containment, and consider Section C.2 of Regulatory Guide 1.163 to apply to large bore containment purge and vent valves which are in excess of a diameter specified by the owner. These are the valves that cannot be placed on extended test intervals. Furthermore, it should be considered that Section C.2 of Regulatory Guide 1.163 does not apply to the small bore purge and vent bypass valves under a certain diameter specified by the owner, and that these valves may qualify for extended interval testing provided satisfactory performance has been demonstrated.

Again, this guidance would not apply to components where the above classification conflicts with a plant's Technical Specification, Updated Final Safety Analysis Report (UFSAR), Probabilistic Risk Assessment (PRA) or other Owner requirement or commitment.

MSIV LLRT issue:

This issue was initiated following a Non-Cited Violation (NCV) issued by the NRC in January 2004. This Technical Position Paper was intended to document the justification for performing MSIV LLRT using non-safety instrument air applied to the closing portion (top-side) of the valve actuator, which increases the closing and sealing forces of the valve that can aid in maintaining LLRT results within plant Technical Specification limits. Due to the generic application of this issue to BWR plants, the Boiling Water Reactor Owner's Group (BWROG) was enlisted to support resolution of this issue. A Technical Position Paper was prepared and reviewed by representatives of the BWROG and the paper was provided to the NRC. Due to the proprietary nature of BWROG products, additional information regarding this cannot be published herein. As of this publishing, the paper has not been issued.

Creation of an Integrated Leakage Rate Testing Planning and Performance Guideline:

The performance of Primary Containment Integrated Leakage Rate Tests (PCILRT) is typically considered an infrequently performed test. Therefore, the Appendix J Program Owner's Group has initiated the development of a guideline to enable the plant Program Owner to better manage the preparation and performance of this test. This Guideline will provide a time-line for planning the PCILRT and guidance on the pertinent aspects required to successfully prepare for this test. The guide will also provide a sequentially organized tutorial on the performance of the PCILRT. Additionally, a lessons learned section will provide the plant with necessary Operating Experience (OE) to avoid the pitfalls others have experienced during the planning and execution of a PCILRT.

Appendix J Program Scope Reduction:

In the course of commiserating on the subject of Appendix J Program management with fellow program owners, it has been noted that many plants are vastly different when it comes to the scope of their programs. Some plants test valves that other plants do not and vise-versa. For example, some plants test multiple valves associated with closed loops inside/outside containment (CLICs and CLOCs), while other plants test only one or no valves associated with CLICs and CLOCs. Another scope reduction can be to reduce as-found testing by utilizing the guidance in APOG's TPP on this subject. The Appendix J Program Scope Reduction guideline will document several opportunities for the Program Owner to explore for reducing the number of Local Leak Rate Tests performed. This could result in savings for the utility, which in turn could make additional funds available for Program improvements or other worthwhile endeavors, all without adversely impacting the safety and welfare of the general public.

Creation of a Comprehensive Training and Qualification Guideline:

As with any discipline, the level of training and experience of the responsible individuals is evident in the quality and integrity of their Programs. While the Nuclear Industry workforce continues to age, recruiting younger and thereby less experienced individuals, is becoming a very high priority. In order to ensure consistency across the industry, a new Appendix J Program Owner will need assistance in interpreting the regulations and applying them to his or her plant Program. A Comprehensive Training and Qualification Guide can provide this consistency across the industry, and enable a plant owner to more efficiently and competently fill his or her position.

Development of a Consistent Methodology for Assessing Appendix J Components within the Maintenance Rule:

All U.S. nuclear power plants are required to comply with 10 CFR 50.65; better known as the "Maintenance Rule". Maintenance Rule Programs set performance criteria for Structures, Systems, and Components (SSCs) that are important to safety, and monitor their performance against these criteria. When the performance criteria are not met, a functional failure is documented against the SSC. Recent discussions regarding Maintenance Rule performance criteria associated with Containment Isolation Valves (CIVs) has indicated that there is an inconsistency across the industry as to the application of performance criteria and at what point a CIV would be declared Maintenance Rule (a)(1), which is the category that indicates corrective actions are required to restore the SSC to acceptable performance. The objective of this APOG effort would be to develop a matrix of the various methodologies employed across the industry and establish a standard approach to be used to ensure the requirements of 10 CFR 50.65 are uniformly satisfied. This effort is still a work in progress.

Development of an Operating Experience Database on the APOG web site:

The use of OE is the best way to avoid making mistakes that someone somewhere else has already made. In the past few decades, a vast collection of OE has accumulated. Not all of this OE is as readily accessible as desired, and some of it has not even been documented outside the company that experienced it. The intent of the APOG OE database would be to provide a central location for all OE relevant to Appendix J local leak rate and integrated leak rate testing. APOG members would be able to input and extract OE. While other sources of OE already exist, such as the Institute of Nuclear Power Operations (INPO), this is not intended to replace those sources, but to enhance the accessibility as well as provide a focused subject. Additionally, this database will contain OE that typically "flies under the radar", such as minor lessons learned that have little impact on the industry as a whole, but to the Appendix J Program Owner could mean the difference between a two hour LLRT and a ten minute LLRT of the same type of component. The exact structure of this database is still under development.

Development of a LLRT Failure Database on the APOG web site:

As with the previously mention OE database, it is understood that there are already other sources available for this type of information, however, they are not always readily accessible and may not be structured to meet the need of a specific user. The intent of this database would be to provide specialized sorting and reporting to the APOG member of information relevant to this specialized testing program. The LLRT Failure Database concept was initiated after the 2005 annual APOG meeting and is still in the developmental stage.

Conclusion:

In addition to the above topics, APOG has provided significant input to the efforts to revise NEI 94-01 for institution of a 15 year ILRT interval and the development of the INPO Engineering Program Guide (EPG) for Appendix J Programs.

APOG exists to provide the nuclear industry with a centralized and coordinated organization to support improvements in the area of Primary Containment Leakage Rate Testing. The establishment of APOG has clearly filled a void that existed in the nuclear industry. Evidence of this is the expeditious development of NEI 94-01 Revision 1, the INPO Appendix J Engineering Program Guide, and resolution of some of the technical issues that have historically caused consternation among Appendix J Program Owners. Anyone interested in learning more about APOG and becoming a member site is encouraged to speak with any of the APOG members in attendance, or simply go to www.appendixj.com for more information.

References:

1. 10 CFR 50 Appendix J - Primary Reactor Containment Leakage Testing for Water-Cooled Power Reactors, Option B - Performance-Based Requirements.

2. NRC Regulator Guide 1.163 - Performance-Based Containment Leak-Test Program (September 1995).

3. NEI 94-01, Revision 0 – Industry Guideline for Implementing Performance-Based Option of 10 CFR 50, Appendix J (July 1995).

4. ANSI/ANS-56.8-1994 – Containment System Leakage Testing Requirements.

Code Case OMN-1 Implementation at San Onofre Nuclear Generation Station

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Abstract

This paper will discuss how San Onofre Nuclear Generation Station (SONGS) is implementing the American Society of Mechanical Engineers (ASME) Code Case OMN-1, "Alternatives Rules for Preservice and Inservice Testing of Certain Motor-Operated Valve Assemblies in Light-Water Reactor Power Plants," to determine the operational readiness of motor operated valves (MOVs). The Code Case is an alternative to the ISTC requirements for quarterly stroke-time testing and position verification for certain MOVs. Also, the incorporation of ASME Code Case OMN-1 into the Risk-Informed Inservice Testing (RI-IST) Program implementation methodology used at SONGS and its relationship to Generic Letter (GL) 96-05, "Periodic Verification of Design-Basis Capability of Safety-Related Motor-Operated Valves," will be discussed.

Background

In Bulletin 85-03, dated November 15, 1985, and Supplement 1 of Bulletin 85-03, dated April 27, 1988, the Nuclear Regulatory Commission (NRC) recommended that licensees develop and implement a program to ensure that valve motor-operator switch settings (torque, torque bypass, position limit, overload) for MOVs in several specified systems were selected, set, and maintained so that the MOVs will operate under design-basis conditions for the life of the plant. NRC staff assessments of the reliability of all safety-related MOVs, based on extrapolations of the currently available results of valve surveillances performed in response to Bulletin 85-03, indicated that the program to verify switch settings should be extended in order to ensure operability of all safety-related fluid systems. The NRC staff's evaluation of the data indicated that, unless additional measures were taken, failure of safety-related MOVs and position-changeable MOVs to operate under design-basis conditions would occur much more often than had previously been estimated.

Nuclear power plant operating experience, valve performance problems and MOV research revealed that the focus of the ASME Code on stroke time and leak-rate testing for MOVs was not sufficient in light of the design of the valves and the conditions under which they must function. For these reasons, GL 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance," was issued.

By issuance of GL 89-10, the NRC extended the scope of the program outlined in Bulletin 85-03 and Supplement 1 of Bulletin 85-03 to include all safety-related MOVs as well as all position-changeable MOVs (not blocked from operation). The licensee's program was requested to provide for the testing, inspection, and maintenance of MOVs so as to provide the necessary assurance that they will function when subjected to design-basis conditions during both normal operation and abnormal events within the design basis of the plant. When determining the maximum differential pressure or flow for position-changeable MOVs, the fact that the MOV must be able to recover from mispositioning should be considered. GL 89-10 superseded the recommendations in Bulletin 85-03 and its supplement.

The NRC issued seven supplements to GL 89-10 that provided additional guidance and information on GL 89-10 program scope, design-basis reviews, switch settings, testing, periodic verification, trending, and schedule extensions.

Supplement 6 to GL 89-10 stated that no licensee had adequately justified the use of static test data as the sole basis for periodically ensuring MOV design-basis capability.

On December 1996, the NRC staff determined that SONGS implementation of the MOV program has successfully met the commitments to GL 89-10. The program encompassed 178 MOVs including 82 gate valves, 28 butterfly valves, 20 rotating-rising stem and 48 standard globe valves.

The NRC issued GL 96-05 to discuss the periodic verification of the capability of safety-related MOVs to perform their safety functions consistent with the current licensing bases of nuclear power plants. GL 89-10 and its supplements had provided only limited guidance regarding periodic verification and the measures appropriate to assure preservation of design-basis capability. GL 96-05 provided a more complete guidance regarding periodic verification of safety-related MOVs and supersedes GL 89-10 and its supplements with regard to MOV periodic verification. Although this guidance could have been provided in a supplement to GL 89-10, the NRC prepared this new generic letter to allow closure of the staff review of GL 89-10 programs as promptly as possible.

The NRC believes that various approaches can be taken by licensees to establish a periodic verification program that provides confidence in the long-term capability of MOVs to perform their design-basis safety functions. With each approach, the licensee should address potential degradation that can result in the increase in thrust or torque requirements to operate the valves and the decrease in the output capability of the motor actuator.

In Attachment 1 to GL 96-05, the NRC discusses industry and regulatory activities and programs related to maintaining long-term capability of safety-related MOVs and provide the NRC position regarding American Society of Mechanical Engineers (ASME) Code Case OMN-1.

In January 2000, the NRC staff determined that SONGS have established and implemented a program to provide continued assurance that MOVs within the scope of GL 96-05 were capable of performing their design-basis functions.

Code Case OMN-1

Discussion

NOTES

1. Since the issuance of the RI-IST Safety Evaluation, the NRC regulations in 10 CFR 50.55a(b)(3)(iii) are now Conditions specified in RG 1.192 by NRC approval to use Code Case OMN-1. Also, the SONGS Code of Record was changed to OM Code-1998 with Addenda through 2000 during the 3rd Ten Year Interval update.

2. All procedures referred to in the following discussion as being developed, have been developed and are cross referenced to the Code Case in Attachment 11 of SO23-V-3.50, Administration of the Generic Letter 89-10 Motor Operated Valve Program.

In GL 96-05, the NRC staff stated that, with certain limitations, the method described in ASME Code Case OMN-1 is considered to meet the intent of the generic letter to verify the design-basis capability of safety-related MOVs on a periodic basis. The limitations specified in GL 96-05 were consistent with those specified in 10 CFR 50.55a(b)(3)(iii). Further, NRC conditional approval of Code Case OMN-1 via Regulatory Guide 1.192 specifies these same limitations (conditions), which must be applied.

In the NRC Safety Evaluation issued for the Risk-Informed Inservice Testing (RI-IST) Program at SONGS, the NRC stated that the description of the licensee's program to implement ASME Code Case OMN-1 as an alternative to the quarterly MOV stroke-time testing provisions required by the licensee's Code of Record is approved on the basis that:

a. The NRC regulations in 10 CFR 50.55a(b)(3)(iii) allow licensees to apply ASME Code Case OMN-1 as an alternative to the quarterly MOV stroke-time testing provisions described in the ASME OM Code [Code for Operation and Maintenance of Nuclear Power Plants]. The 1989 Edition of the ASME BPV Code [Boiler & Pressure Vessel Code] applied by Southern California Edison also includes provisions for quarterly MOV stroke-time testing. The NRC staff further finds that the licensee satisfies the two modifications related to OMN-1 specified in 10 CFR 50.55a(b)(3)(iii) as follows:

(1) Section 50.55a(b)(3)(iii)(A) specifies that licensees evaluate the information obtained for each MOV, during the first 5 years or three refueling outages (whichever is longer) of voluntary use of ASME Code Case OMN-1, to validate assumptions made in justifying a longer test interval. In the RI-IST Program Description (IPD) (enclosure to the licensee's submittal dated November 30, 1999), SONGS states that, as a living process, components will be reassessed at a frequency not to exceed every other refueling outage to reflect changes in plant configuration, component performance test results, industry experience, and other inputs to the process. In its submittal dated September 28, 1999, SONGS indicates that the maximum IST interval for MOVs at San Onofre is 6 years or three refueling cycles. The licensee states that testing of groups of MOVs containing more than one valve is based on a stagger test model that evenly distributes component testing over the maximum interval. SONGS reports that its RI-IST program procedures contain guidance to ensure performance and test experience are evaluated to support the periodic verification interval.

(2) Section 50.55a(b)(3)(iii)(B) clarifies the provision in Paragraph 3.6.2 of ASME Code Case OMN-1 for the consideration of risk insights if extending the exercising frequencies for MOVs with high risk significance beyond the quarterly frequency specified in the ASME Code. In particular, licensees will ensure that increases in core damage frequency and/or risk associated with the increased exercise interval for high-risk MOVs are small and consistent with the intent of the Commission's Safety Goal Policy Statement (51 FR 30028; August 21, 1986). The NRC also considers it important for licensees to have sufficient information from the specific MOV, or similar MOVs, to demonstrate that exercising on a refueling outage frequency does not significantly affect component performance. Grouping similar MOVs, and staggering the exercising of MOVs in the group equally over the refueling interval may obtain this information. In its IPD, SONGS stated that high-risk valves at San Onofre initially will continue to be stroke-time tested quarterly, at cold shutdown, or at refueling intervals based on practicability as required by the Code of record. The licensee stated that it might extend these test intervals to refueling cycles when sufficient data are obtained, and analyses through the IPD support such extensions. SONGS further specified that it would develop and proceduralize an approach for determining an MOV test interval that is based on risk ranking, available capability margin, and valve performance history.

In GL 96-05, the NRC staff noted that some licensees are developing risk-informed IST programs as part of a pilot industry effort. The staff stated that licensees need to address the relationship between ASME Code Case OMN-1 and their Risk-Informed IST Programs. In its submittal dated September 28, 1999, SONGS noted that its IPD confirms or adjusts the initial risk ranking developed from the PRA results, and provides a qualitative assessment based on engineering judgment and expert experience to determine the final safety significance categories. This process identifies components whose performance justifies a higher categorization, determines appropriate changes to testing strategies, and identifies compensatory measures for Potentially High Safety Significant (L-H) components, or justifies the final categorization. The process also evaluates the test interval and basis test methodology for Low Safety Significant Components (LSSCs).

In GL 96-05, the NRC staff noted a precaution that the benefits (such as identification of decreased thrust output and increased thrust requirements) and potential adverse effects (such as accelerated aging or valve damage) need to be considered when determining the appropriate testing for each MOV. In the IPD, SONGS states that it will ensure by means of plant procedures that the benefits and potential adverse effects are considered as part of the determination of appropriate MOV testing.

b. ASME Code Case OMN-1 specifies in Paragraph 3.6.1 that all MOVs within the scope of the Code Case need to be exercised on an interval not to exceed one year or one refueling cycle (whichever is longer). This exercising is intended to ensure proper lubrication of each MOV regardless of diagnostic test intervals that might extend beyond this time period. In its submittal dated June 17, 1999, SONGS committed to apply ASME Code Case OMN-1 in its entirety. In its submittal dated September 28, 1999, SONGS acknowledged this provision of OMN-1 and reported that all MOVs in its program will be exercised on at least a refueling outage frequency. SONGS November 30, 1999, IPD includes a provision for exercising MOVs at least once during a refueling cycle.

c. ASME Code Case OMN-1 specifies in Paragraph 3.3.1 that MOV inservice testing be conducted every two refueling cycles or 3 years (whichever is longer) unless sufficient data exist to determine a more appropriate test frequency. The SONGS IPD states that L-H and LSSC

MOVs will be tested in accordance with ASME Code Case OMN-1 and NRC Generic Letters 89-10 and 96-05 commitments at an initial interval not to exceed 6 years until sufficient data exist to determine a more appropriate test frequency. In its submittal dated June 17, 1999, the licensee stated that it was implementing the provisions of Paragraph 6.4.4 of ASME Code Case OMN-1, which provides direction for determining acceptable test intervals. In its submittal dated September 28, 1999, SONGS discussed its use of a stagger test model to obtain data on MOV performance throughout the test interval. SONGS stated that valves identified with reduced margin or degradation rates greater than expected will be subject to more frequent testing. SONGS November 30, 1999, IPD specifies that RI-IST program procedures and MOV trend procedures will contain guidance to ensure performance and test experience from previous tests are evaluated to justify the periodic verification interval.

d. In Paragraph 3.5, ASME Code Case OMN-1 specifies that MOVs with identical or similar motor operators and valves, and with similar plant service conditions, may be grouped together based on the results of design-basis verification and preservice tests. In the IPD, SONGS notes that components will generally be grouped based on system, component type, manufacturer, size, style, and application. In its submittal dated September 28, 1999, SONGS noted that its grouping of MOVs includes such aspects as system conditions and valve internal materials.

e. SONGS IPD states that MOV seat leakage testing for L-H and LSSC MOVs, as applicable, will be performed per the Code of Record, except at a test frequency not to exceed 6 years. In its submittal dated September 28, 1999, SONGS stated that the basis for the extension of MOV seat leakage testing intervals is derived from its IPD. This process includes consideration of performance history, industry history of similar components as available, and risk.

f. ASME Code Case OMN-1 references MOV test procedures and other plant documents containing acceptance criteria for MOV performance. In its submittal dated September 28, 1999, SONGS indicated that development of OMN-1 procedures is in progress. SONGS stated that it intends to incorporate the analysis and evaluation of data sections of OMN-1 as an additional enhancement to the current MOV program independent of the RI-IST program. The NRC staff may review those procedures during an on-site inspection when they are available.

The NRC staff has determined that the SONGS proposed application of ASME Code Case OMN-1, as discussed herein, is consistent with the guidance contained in Regulatory Guide 1.175 and is an acceptable alternative to stroke time testing required by SONGS Code of Record. SONGS has to develop procedures for implementing ASME Code Case OMN-1 at SONGS. The NRC staff may review the procedures during an on-site inspection.

OMN-1 General Requirements – Design Basis Verification Test

A one-time test shall be conducted to verify the capability of each MOV to meet its safety-related design basis requirements. This test shall be conducted at conditions as close to design basis conditions as practicable. Requirements for a design basis verification test are specified in applicable regulatory documents. Testing that meets the requirements of this Code Case but conducted before implementation of this Code Case may be used.

(a) Design basis verification test data shall be used in conjunction with preservice test data as the basis for inservice test criteria.

(b) Design basis verification testing shall be conducted in situ or in a prototype test facility that duplicates applicable design basis conditions. If a test facility is used, an engineering analysis shall be documented that supports applicability to the in situ conditions.

(c) Justification for testing at conditions other than design basis conditions and for grouping like MOVs shall be documented by an engineering evaluation, alternate testing techniques, or both.

(d) The design basis verification test shall be repeated if an MOV application is changed, the MOV physically modified, or the system is modified in a manner that invalidates its current design basis verification test results or data. An engineering evaluation, alternate testing techniques, or both shall justify a determination that a design basis verification test is still valid.

Inservice Test Frequency

(a) The inservice test frequency shall be determined in accordance with para. 6.4.4.

(b) If insufficient data exists to determine the inservice test frequency in accordance with para. 6.4.4, then the MOV inservice testing shall be conducted every two refueling cycles or 3 years (which ever is longer) until sufficient data exists to determine a more appropriate test frequency.

(c) The maximum inservice test frequency shall not exceed 10 years.

Determination of MOV Test Interval per para. 6.4.4

Calculations for determining MOV functional margin shall also be evaluated to account for anticipated time-related changes in performance. Maintenance activities and intervals can affect test intervals and shall be considered.

The interval between tests shall be less than the anticipated time for the functional margin to decrease to the acceptance criteria.

Acceptance Criteria

The Owner shall establish methods to determine acceptance criteria for each MOV within the scope of OMN-1. Acceptance criteria shall be based upon the minimum amount by which available stem torque must exceed the required design basis stem torque. When determining the acceptance criteria, consider the following sources of uncertainty:

(a) test measurement uncertainty and equipment uncertainty (e.g. torque switch repeatability);

(b) analysis, evaluation, and extrapolation method uncertainty; and

(c) grouping method uncertainty.

MOV margins may be expressed in terms of other parameters, such as stem force, if those parameters are consistent with paras. 6.1 through 6.5.

Exercising Requirements

All MOVs within the scope of OMN-1 Code Case shall be exercised on an interval not to exceed one year or one refuel cycle (which ever is longer). Full stroke operation of an MOV, as a result of normal plant operations or Code requirement may be considered an exercise of an MOV, if documented. Alternatively, longer exercise intervals may be used if justified by successful operating experience.

The following reflects Attachment 11, "ASME Code Case OMN-1 and SONGS MOV Procedure/Program Cross Reference" of SO23-V-3.50, "Administration of the Generic Letter 89-10 Motor Operated Valve Program."

OMN-1 Section SONGS Procedure/Program Cross Reference

1.0 Introduction

1.1 Scope SO23-V-5.22.1 MOV Program

Calculation A-94-NM-MOV-POP-VER-001 MOV Population Verification

1.2 Exclusions SO23-V-5.22.1 MOV Program

Calculation A-94-NM-MOV-POP-VER-001 MOV Population Verification

2.0 Definitions SO123-V-3.4, MOV Periodic Verification and Trending Program

3.0 General Requirements

3.1 Design Basis Verification Test Design Program requirement

3.2 Preservice Test Design Program requirement

3.3 Inservice Test M-42652, PM Requirements and Interval for GL 89-10 MOVs

3.3.1 Inservice Test Frequency SO123-V-3.4, MOV Periodic Verification and Trending Program

3.4 Effects of MOV Maintenance Post Maintenance Test Requirements

3.5 Grouping of MOVs for IST SO123-V-3.4, MOV Periodic Verification and Trending Program

Risk Informed IST Program

3.6 MOV Exercising Requirements Risk Informed IST Program

3.7 Risk Based Criteria for MOV TestingRisk Informed IST Program

4.0 Reserved

5.0 Test Methods

5.1 Test Prerequisites SO2323-I-9.30, MOV Analysis and Test System

5.2 Test Conditions MOV Setpoint Calculations

5.3 Limits and Precautions SO2323-I-9.30, MOV Analysis and Test System

Maintenance Orders

5.4 Test Procedures SO123-V-3.4, MOV Periodic Verification and Trending Program

SO23-V-3.50, Administration of the GL 89-10 MOV Program

5.5 Test Parameters MOV Setpoint Calculations

6.0 Analysis and Evaluation of Data

6.1 Acceptance Criteria SO23-V-3.50, Administration of the GL 89-10 MOV Program

6.2 Analysis of Data SO23-V-3.50, Administration of the GL 89-10 MOV Program

SO23-V-3.53, GL 89-10 MOV Motor Torque (UDS MC2) Testing Program

SO123-V-3.4, MOV Periodic Verification and Trending Program

6.3 Evaluation of Data SO123-V-3.4, MOV Periodic Verification and Trending Program

6.4 Determination of MOV

Functional Margin

6.4.1 Determination of

Required Torque SO123-V-3.4, MOV Periodic Verification and Trending Program

6.4.2 Determination of

Available Stem Torque MOV Setpoint Calculations

SO123-V-3.4, MOV Periodic Verification and Trending Program

6.4.3 Calculation of Available Functional Margin SO123-V-3.4, MOV Periodic Verification and Trending Program

6.4.4 Determination of MOV

Test Interval SO123-V-3.4, MOV Periodic Verification and Trending Program

6.5 Corrective Action

6.5.1 Record of Corrective Action MOSAIC

Test Package Reconciliation

Conclusion

SONGS RI-IST Program provides for testing of HSSCs in accordance with the Code test frequencies and method requirements, in accordance with the NRC-approved OMN-1. Similarly, SONGS will test LSSCs in accordance with the Code test method requirements (although on an extended interval) or approved alternative methods. SONGS has not identified any exemptions, technical specifications amendments, or relief from Code requirements, which would require review and approval before implementation of its RI-IST program. Therefore, the NRC staff found this aspect of the SCE's RI-IST program to be acceptable because it is consistent with the acceptance guidelines contained in Section 2.2.2 of Regulatory Guide 1.175.

References

1. Generic Letter 89-10, "Safety-Related Motor Operated Valve Testing and Surveillance," and Supplements 1 thru 7.

2. Generic Letter 96-05, "Periodic Verification of Design Basis Capability of Safety Related Motor-Operated Valves."

3. SO23-V-3.5, "Inservice Testing of Valves Program."

4. ASME Code Case OMN-1, "Alternatives Rules for Preservice and In-service Testing of Certain Motor-Operated Valve Assemblies in Light-Water Reactor Power Plants."

5. "Safety Evaluation by the Office of Nuclear Reactor Regulation Related to the Southern California Edison Request to Implement a Risk-Informed In-service Testing Program at San Onofre Nuclear Generation Station, Unit 2 and 3," dated March 27, 2000.

6. SONGS, “MOV Program Response for Generic Letter 89-10.”

7. SONGS, “Response for Generic Letter 96-05.”

8. NUREG/CP-0152, Vol. 4, “Risk-Informed In-Service Testing at San Onofre Nuclear Generating Station Nuclear Generation Station (SONGS),” by Maureen K. Coveney and William J. Parkison, Data Systems and Solutions and Darryl L. Barney, Southern California Edison.

9. NUREG/CP-0152, Vol. 3, “Why Have Some Plants Embraced RI-IST and Others Not?” by C.W. Rowley, The Wesley Corporation.

10. NUREG/CP-0152, Vol. 3, “Development and Implementation of Code Case OMN-1,” by Robert G. Kershaw, Arizona Public Service Co.

Improving Relief Valve Reliability at Wolf Creek

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Abstract

This paper describes the measures taken at Wolf Creek Nuclear Operating Corporation to improve ASME Class 2 and 3 relief valve reliability. Over a 10-year period between 1995 and 2005, roughly 44% of ASME Class 2 and 3 valves under the Inservice Testing (IST) Program failed to perform within their specified set-pressure tolerances on their initial tests. The causes of these failures are varied, so no single solution has been available to improve overall performance. The unacceptably high failure rate is being addressed by test equipment improvements, improved administrative guidance, changes in maintenance practices, reducing test intervals, and training to improve overall knowledge about these components.

ASME Requirements

The American Society of Mechanical Engineers (ASME) *Code for Operation and Maintenance of Nuclear Power Plants* (OM Code) requires functional testing of ASME *Boiler & Pressure Vessel Code* (BPV Code), Section III, Category 2 and 3, relief valves which protect systems that perform a function to shut down the reactor to the safe shutdown condition, maintain the reactor in the safe shutdown condition, or mitigate the consequences of an accident. Since the 1995 Edition of the ASME OM Code was issued, the requirements for testing relief valves are listed in Appendix I. Appendix I of the ASME OM Code identifies the minimum set of requirements for this testing. The critical testing requirement of ASME OM Code Appendix I is the initial set-pressure test. Failing this portion of the test requires additional tests on valves that are of the same manufacturer, type, system application, and service media.

General Testing Approach

Wolf Creek does not test ASME Class 2 or 3 relief valves while they are installed in the system. Replacement valves have been obtained for each valve group and are stored in our warehouse. Each valve installed in a system is removed and replaced with a valve from the warehouse. The valve is then moved to a temporary storage area and tested later. ASME OM Code, Appendix I, allows a 12-month period between removal and testing if all valves from a group are removed at the same time. If a partial compliment of a valve group is removed for testing, then the time period allowed for testing is limited to a 3-month period. In most cases at Wolf Creek, testing is performed within a 3-month period.

Scope Increase Performance Results

Over Wolf Creek's 2nd 10-year IST Interval, numerous failures occurred on relief valves tested for the IST Program. The failure rate was roughly 44%. The valves with the highest failure rates were addressed first. These valves were not in the IST Program during the 1st 10-year IST Interval and were added as a result of the scope change from ASME BPV Code, Section XI, Article IWV guidance to the ASME OM Code guidance.

Heating, Ventilation and Air Conditioning (HVAC) Relief Valves

These valves tended to skew the overall test results to the unfavorable area due to the high failure rate and increased frequency of testing. Wolf Creek has a Control Room HVAC system with two trains and a Class 1E Electrical HVAC system with two trains that were originally brought into Wolf Creek as a skid. The subject valves were installed on the Freon side of the HVAC system and were determined to meet the scope statement of a system that is used to mitigate the consequences of an accident.

The relief valves were supplied as part of the vendor skid. The vendor reported that the material used for the soft seat of the relief valves had a chemical degradation mechanism and changed to a replacement material for the soft seat in their rebuild kit. This was a little over a year before the valves were added to the IST Program.

The first valve tested failed its initial inservice test. This caused the sample population for testing to be expanded. Two out of four valves failed their as-found tests and a third failed its second as-found test. After this testing round was completed, a rebuild was performed on the failed valves using the new soft seat material supplied by the vendor in the new rebuild kits. The valves were bench tested for set-pressure and seat leakage successfully. A preventive maintenance (PM) task was subsequently implemented to monitor for Freon leakage, which was identified as a potential precursor to a failed valve after the initial performance analysis.

Freon leakage was identified on a valve that had been in service for only 2 years. The valve was removed from service and we couldn't get it to definitively lift on the test bench because of the seat leakage problem. It was at this time that we started to suspect a generic failure mechanism was present that could be related to the new seating material utilized by the vendor. A hardware failure analysis was performed to determine the cause of the reliability problem. These were soft-seated valves that had a very soft rubber material in the disk area. A comparison between a new disk and the old failed disk revealed that the soft material had been flattened, which explained why Freon leakage would occur.

Other problems with one particular system train were identified, which were failures of electronic components in the skid's control panel. This led to suspicion of a control problem. Monitoring of the system with the worst history of overall problems commenced to determine if there was a system control problem that was unexpectedly lifting the valves. A review of system data determined that these valves had never lifted in service, yet the same phenomenon was observed after removing the valve disk from service after about a year.

Vibration monitoring and bump testing of the control panel and relief valve areas determined that these areas were resonant with the compressor. Whenever the compressor operated, the control panel box and the relief valve were put into a situation of high cycle fatigue. Measures were taken to stiffen the control panel and relief valve areas, but they were only marginally effective. Two out of the three remaining vendor skids also had the resonance problem to lesser degrees.

To address this problem, all four skids were removed and replaced with newer systems. This solved the reliability problems of the skid overall, and the dual relief valves installed on the new systems have been performing admirably.

Safety Injection Relief Valves

These valves added to the IST Program during the 1995 update as a result of changes between the ASME BPV Code, Section XI, and the ASME OM Code scope statements. Wolf Creek has two 100% redundant trains of Safety Injection that perform a safety function in an accident to inject water into the Reactor Coolant System (RCS) between the High and Low Pressure safety injection range. There are three valves that protect these systems from the adverse affects of overpressure. The first Inservice Test of a valve from this group in 1996 resulted in an as-found lift above the +/-3% acceptable range. A few days later, the second valve removed for testing from this group lifted below the +/-3% range. Fortunately, the third valve removed for testing on that same day tested within its tolerance.

The valves were inspected to determine the cause of failure, which did not reveal any conclusive results. It did appear that these valves had lifted numerous times while in service. Monitoring was performed during the quarterly pump tests that were subsequently performed. It was found out at this time that the pressure wave from the pump start was bumping open the valves. Although it appeared that there should have been adequate margin between each pump's recirculation pressure and the set-pressure of the downstream relief valves, the phenomena of a pressure wave being generated at pump start had not been recognized until this time.

Consultation with the vendor and a search of industry operating experience revealed that Wolf Creek was not the first plant to identify this type of problem. A safety analysis of system piping was performed to justify an increase in the valve's set-pressure rating. Also, the vendor had a modification that installed a travel stop to ensure the valves would not stick open in a water hammer event, which was the subject of the operating experience that was found. This modification was also implemented on each of the valves as they were removed and replaced.

The test frequency was increased to verify that the modifications were effective. One of the valves failed to lift at the maximum pressure allowed for testing, which resulted in an additional hardware failure analysis. The only unusual result from measurement and inspection was the valve's spindle run-out, which was at .009". The valve was refurbished without replacing the spindle and re-assembled for additional testing. The set-pressure results from these tests revealed that the valve was not performing reliably. The first test was below 3% and the second test was above 3% of the set-pressure. The valve was disassembled, the spindle was replaced with one that had less than .005" of run-out, and the valve was again put on the test stand for adjustment and testing. The valve performed reliably at this point. Based on these results, it appears that spindle run-out can influence the performance of this type of valve, which is a Crosby JRAK style. We continue to have performance issues with this style of valve in this and other applications.

Residual Heat Removal (RHR) Relief Valves

Wolf Creek has two trains of RHR that includes relief valves on the suction side. Callaway is a plant that is of identical design to Wolf Creek and had experienced numerous problems with these valves during their first 10-year IST interval. Wolf Creek did not have the same problems; however, the operating experience from our sister plant was cause for concern. These valves were both selected early in the second 10-year IST interval because of concerns about reliability. Removal, replacement and testing of both valves were performed in 1996. Both valves passed their tests and were subsequently re-targeted for testing in 1999. In 1999, both valves failed their tests. One valve failed slightly below its set-pressure and the other valve failed slightly above its set-pressure. It appeared that the failures were due to simple set-pressure drift. The valves had not been adjusted to the mid-point of the set-pressure range prior to installation and a small amount of drift from where they had been set explained both failures. A removal, replacement and test was performed on one of the valves in 2003 successfully. This valve had been reset to within 1% of its set-pressure prior to installation in the system in 1999. The other of these two valves was removed and tested in 2005. It failed with an as-found set-pressure 2 psig above the 3% tolerance, which prompted yet another hardware failure investigation.

Big Picture Investigation of Failures

The investigation of all relief valve failures from a comprehensive standpoint revealed numerous areas for improvement. We had two test benches that utilized carbon steel accumulator tanks. The test benches were also of different design and had different sized accumulator tanks. They had been in use at Wolf Creek for over 15 years. Use of carbon steel accumulator tanks is not recommended for testing of relief valves. There is a potential to introduce rust products during testing after these tanks have been in service and exposed to liquid. Flushing of the accumulator tanks had been a regular practice that was performed on both test benches, but the age of the tanks and the internal condition created concerns that particulates were being introduced during testing that was reducing the long-term reliability of the valves. Analog instruments with 0.25% accuracy were being utilized with the bench, but the location of the pressure tap relative to the position of the valve on the bench was not at the optimum location on either of the test benches. In summary, our test equipment needed to be updated to modern technology.

It was recognized that our Mechanical Maintenance staff had lost some of their knowledge base about relief valves due to retirement and re-assignments within the company. To further aggravate this problem, relief valve testing is an infrequently performed task and different crews were being used for different tests. It was thought that the exposure of all Mechanical Maintenance staff to relief valve testing would improve the groups overall knowledge that had been lost when the experienced Mechanics left the group. Unfortunately, this created a situation where none of the Mechanics had confidence or proficiency in working on or testing relief valves.

It was also noted by comparison of the serial numbers and the associated test results that some relief valves appeared simply to be more reliable than others. This was attributed to the tolerances of the parts and their interface with each other as originally supplied by the manufacturer. Spindle run-out on certain valves appears at the present time to affect the repeatability of the set-pressure lift for certain types of relief valves. Since this phenomenon is not fully understood, we have reached an agreement with the supplier of this equipment to bring them to the site if this occurs in the future to assist in analyzing the failure mechanism.

In some cases, the relief valves had to be stored for an extended period before being tested. The ASME OM Code allows up to 3 months to complete testing if part of a group is removed and replaced. ASME allows 12 months to complete testing if all of the valves in the group are removed. In at least one instance, a relief valve was stored horizontally rather than vertically in an interim storage location before it was tested nearly 3 months later. This is known to cause problems with subsequent relief valve performance for several types of relief valves.

Pressurizer Code Safety Valves and Main Steam Safety Valves

Problems with ASME Class I Pressurizer Code Safety Valves and ASME Class II Main Steam Safety Valves were virtually non-existent at Wolf Creek over the last 10-year IST interval. The three Pressurizer Code Safety valves are removed and sent to NWS Technologies for testing and general refurbishment every cycle. Main Steam Safety valves are tested in place using the Furmanite Trevitest system and testing crew. Three of these valves are removed and replaced for inspection and refurbishment by NWS Technologies. These two companies have to be given credit for their contribution to the successful performance of these critical valves.

Measures Taken to Improve Performance

To address our relief valve reliability problems, Wolf Creek implemented several improvement initiatives.

1. Two test benches of identical design were purchased and then put into service in 2005. Training was obtained on the use of this equipment from the vendor to the leads in Mechanical Maintenance. Subsequently, the site procedure that describes how to use this equipment properly was revised to add detailed instruction. Mechanics who had not had the training from the vendor worked with Engineering and Maintenance Planning to identify areas of additional guidance above and beyond the vendor's operating procedure.

2. Additional guidance was added to the test procedure to identify as-left criteria. Instruction was added that requires every relief valve with a set-pressure above 50 psig to be set within 1% of its set-pressure tolerance before installation in any system. If this level of performance can't be reliably achieved, corrective action must be initiated to identify the problem for Engineering Evaluation.

3. Training was obtained for the Mechanical Maintenance Department from a member of the National Board of Boiler and Pressure Vessels. This one-day class was provided to all Mechanics over a three-day period to describe valve fundamentals and common problems that have been identified at the safety and relief valve testing facility where he was employed.

4. Pressure instrumentation was obtained for use with the new test benches that are of superior quality. These instruments are digital and have a temperature compensated accuracy to 0.1% of the reading rather than the full-scale.

5. Maintenance instructions for key components in Crosby JRAK valves have been improved with help from the Tyco Valve Corporation. Improved guidance has also been obtained for setting nozzle rings properly on valves that require the rings to first be set to a mid-position. Manufacturing tolerances for spindle run-out have been obtained and applied to the specification for new parts. Maintaining tight control on allowable tolerances will improve reliability of performance for longer periods of time between tests.

6. The performance test results from the last 10-year IST interval have been analyzed for each group of valves. Valve groups that had failures have had their test frequencies shortened for the current 10-year interval based on the failure rate. Wolf Creek will be testing relief valves much more often in several instances during the current 10-year IST interval.

7. Relief valves removed from a system for testing have to be stored in the vertical position. New storage racks were built specifically for interim storage of radioactive relief valves. Non-radioactive relief valves have had this arrangement in place to enable proper storage.

In Summary

Relief valves are tested infrequently and problems with performance can come from many sources, including the equipment used to test the valves. Vibration induced failures in components near relief valves is a warning sign that more frequent replacement and refurbishment may be needed. Performance test results should be used to assess more frequent replacement and testing. Unlike maintenance facilities whose business is to test and refurbish these components on a daily or weekly basis, testing and maintenance of relief valves at a nuclear facility is typically an infrequently performed task. This challenges the plant staff's ability to maintain an adequate proficiency level by comparison. Additional training and administrative guidance sources are necessary to counteract this difference. New pressure measurement gauges and modern relief valve test benches are simply better tools for providing assurance that age and service-related performance drift problems will not impact the reliable performance of relief valves years later. Proper storage of relief valves in the vertical direction is important to ensure that the affect of gravity does not introduce new failure mechanisms of these precision devices. While this is commonly understood by warehouse staff, it is not always understood by those in the field who remove and place these components in an interim storage location. Finally, like any mechanical device, some relief valves will simply perform better than others due to very minor and perhaps unnoticeable differences. Tracking performance by serial number is the best way to ensure that the worst performing valve in the group does not cause successive problems in multiple locations.

Nuclear Valve Packing Performance Testing

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One of the most important challenges in today's nuclear valve arena is the balance between packing performance and its frictional footprint.

We have undertaken a logical testing protocol to gather data on valve packing performance and friction values throughout a full steam thermal cycle. This testing will focus on different packing materials and designs. Also, the research will focus on the relationship between the combined value of the packing coefficient of friction and the ratio of gland load forces. This data could perhaps assist the industry in determining a better model for valve friction calculations. Another facet of this testing will focus on live loading spring heights after a loss of gland load being re-tightened to the original heights for frictional concerns.

Today, it is important to utilize best available packing technology that can enhance air-operated valve and motor-operated valve operability without sacrificing long-term valve sealing. This presentation will discuss nuclear valve packing performance testing.

Session 3(a): Pumps II

Session Chair

Robert G. Kershaw

Arizona Public Service Company (APS)

RCP Vibration Studies: An Examination of Lower Motor Bearing Failures and Their Effects on Shaft Integrity

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AREVA NP

Abstract

Many cases of high vibration and changing vibration in reactor coolant pumps (RCPs) can be traced to one of several types of degradation in the lower motor guide bearing.

This paper presents a comparison of the vibrational characteristics for several failure modes. Included in this list are loss of lubricant, bearing overloading, and time-varying shoe clearance.

Each of the examples studied is from an actual pump which underwent a lower guide bearing failure. In each case the pump was operated for an extended period following the onset of the failure mechanism. This type of operation raises the question of its effect on the fatigue life of the pump shaft and other components. This paper provides a simple assessment of the extent to which fatigue life is impacted.

I. Background

I.1 Design Considerations

Experience has shown that one of the most common failures in nuclear main coolant pumps involves the lower motor bearing. Typically, the lower guide bearing is an oil-immersed, segmented shoe bearing. Figure 1 shows the arrangement for a typical Westinghouse, or similarly designed, reactor coolant pump.

There are several reasons for the frequency of these failures compared to failures in the other bearings. Collectively, the reasons may be summarized in that main coolant pumps are three-bearing machines, wherein the lower guide bearing has the weakest design of the three. Therefore, under conditions such as misalignment where all the bearings have higher-than-normal loads, it is the lower motor bearing that is most likely to fail.

By comparison, the upper motor bearing is typically far-removed from the sources of radial (static) or synchronous (1X) loading, with normal loading applied at the pump impeller.

Further, the upper bearing is typically immersed in the upper oil reservoir. Since the upper oil reservoir provides lubricant for thrust bearings as well as the guide bearing, it is typically an order of magnitude larger than the lower reservoir. Therefore, the time to failure due to oil leakage is an order of magnitude higher for the upper bearing than the lower, and loss-of-lubricant is one of the most common causes of bearing failure in these machines.

The pump bearing is typically a water-lubricated hydrodynamic bearing. It is typically highly overdesigned to accommodate start-up and off-design operation, but spends little time in these operating conditions. With primary coolant as the lubricant, there is minimal potential for a loss-of-lubricant failure in the pump bearing. Both the journal and the bearing are typically coated with a highly wear-resistant surface as well.

I.2 Cases Considered

(1) Normal Operation

This is a reference case against which all the other cases are compared, where alignment and balance and bearing conditions are known to be within the normal range.

(2) Fully-failed Lower guide bearing (loss-of-lubricant)

This case considers the operation of the pump/motor after the lower motor bearing has been completely 'wiped' and provides NO support to the shaft.

(3) Severely Misaligned Lower guide bearing (overloading).

In this case, a loss of coolant accident (LOCA) restraint seized and failed to accommodate the thermal expansion of the primary piping to the pump. As a result, the bottom of the motor became misaligned by at least 0.060 inches relative to the position. The characteristics were sufficiently subtle and unique that it was operated for an extended period.

(4) Unidirectional Lower guide bearing failure (loose guide shoe)

In this case, the lower motor bearing was initially assembled to the proper clearance using jackbolts, but the torque on the locking nuts was inadequate to prevent subsequent loosening of the jackbolts. In this case, the failure was unidirectional in that the jackbolts loosened only in the most-heavily loaded direction. The characteristics were also time-dependent in that the jackscrews continued to loosen with ongoing operation.

II. Vibration and Operating Characteristics

II.1 General Comments

Because of the limitations on available monitoring sites on main coolant pumps and motors and limits on the number of data channels available, the anomalies at the lower motor bearing are most commonly observed in data taken at the pump coupling. Therefore, they are often erroneously attributed to the pump. Correct and timely diagnosis of an impending lower motor bearing failure is facilitated by the use of other data such as bearing temperature, DC gap voltage measurement, seal performance, etc.

II.2 Characteristics

(1) Normal Operation

In a pump operating normally, the vibration signature is dominated by residual unbalance. The shaft orbit is nearly circular, with the two directions typically within two mils of one another. A shaft orbit for an RCP operating normally is shown in Figure 2.

(2) Fully-failed Bearing

In main coolant pumps, full failure of a lower motor bearing is an all-too-common failure mode for lower guide bearings. These are usually the result of ‘wiping’, which is, in turn, usually the result of oil starvation. Oil starvation is typically due to the prevalence of oil leaks in the lower oil reservoir and reliability issues in oil level measuring equipment.

Such a failure is normally easily identified using a combination of dynamic data and bearing temperature data. These characteristics are sufficiently predictable that a well-trained, attentive operator may prevent this from coming to fruition.

A slow temperature rise is usually the first indicator of an oil-starved lower guide bearing. This typically occurs over a period of several hours. There is usually little or no change in vibration level during this event. This is, however, the time in the failure sequence where a well-trained operator may choose to do visual verification of oil levels.

The rate at which the bearing temperature rises will increase within the last few minutes prior to failure. In the same time frame, the vibration levels will start to increase. Soon thereafter, the temperature will spike as will the vibration levels. The spike in vibration levels will often go undetected without continuous monitoring.

In failures caused by oil starvation, it is generally not possible to replenish the oil supply in a manner to mitigate complete failure once wiping has begun. Therefore, the failure will continue until the entire bearing surface is ‘wiped’ sufficiently that it bears no load in any direction and hence a full failure.

Following the spike, the vibration levels and temperatures will stabilize with temperatures returning to normal, and vibration levels settling at a level higher than previously observed with a normal-looking orbit, and minimal harmonic content. In main coolant pumps, the vibration levels may remain acceptable by operating guidelines.

When this occurs, it brings into question the long-term effect of ongoing operation which is discussed in Section III of this paper.

(3) Unidirectional Failure (failure of individual shoes)

When individual shoes fail, the bearing develops stiffness and damping characteristics, normal in one plane of movement, and reduced in the other. In a segmented shoe bearing, there is very little interaction between shoes so the response in the two planes of movement differs strongly.

For the case studied, the failure was first observed by changes in the overall and 1X vibration levels during heatup of the plant. Subsequently, it showed balance sensitivity which varied from one attempt to the next. Once an 'acceptable' balance level was achieved, it then had stable operation until a change in plant conditions precipitated a step change in vibrations.

It again had stable operation until the end of the cycle. At that time, another attempt was made at balancing. By this point, the asymmetry was very prominent, and the historical balance coefficient was successful at reducing vibration in one plane, but did little in the other direction. It also developed a strong sensitivity to component coolant temperature (which cools the bearing lubricant). After approximately two years of operation, it underwent yet another step change in 1X vibrations. At that time, the excessive clearances were identified. The motor was removed from service because of concerns that the same condition could be present in the upper bearing, for which disassembly and inspection require a much greater effort. A series of shaft orbits during the period of operation are shown in Figures 3(a)-3(d).

Because of the progressive nature of the failure and virtually all of the vibration was seen on the pump coupling, the concern of pump shaft cracking was raised and reviewed repeatedly.

There was a point at which the 2X vibration amplitude increased noticeably, giving additional credence to the concern. However, the 2X amplitude stabilized quickly. In retrospect, it is believed that the increase was the result of the bearing behaving non-linearly, stiffening only at high eccentricities.

(4) Severe Misalignment

a. 'Normal' Misalignment

Typically, the lower guide bearings of a main coolant pump are sufficiently rugged to tolerate misalignment due to maintenance errors. Also, they can usually tolerate errors that can, in some pump and/or motor designs, occur in the internal buildup of the motor or pump so that proper motor-to-pump alignment cannot be performed.

In these conditions, one may observe a vibration signature higher in one plane than another. In extreme cases, dry rubs may cause thermal bowing of the shaft, causing vibration amplitudes to increase exponentially; thereby forcing a shutdown.

However, these mechanisms will normally NOT cause immediate damage to the lower motor bearing if it remains well-lubricated, and are unlikely to damage either the pump or motor shaft.

b. Severe Misalignment

The case considered in the current study is of a motor/pump which was, in fact, properly aligned during installation. The misalignment which occurred was the result of binding in a thermal expansion joint at a LOCA restraint. Figure 4 includes a sketch showing the misalignment mechanism. The resulting level of misalignment is higher than would normally result from routine maintenance activities and was at least 0.060 inches. Further, the condition developed when the pump was already running, so an oil film was present. Hence, the evidence collected suggested that hard contact between metal surfaces never occurred. In this case, the motor bearing simply operated in a highly eccentric position.

The vibration characteristics were, in this case, somewhat deceptive. The distortion of the motor stand and other hardware caused the probes to move out of range and saturate. A waveform from the shaft probe in the direction of misalignment is given in Figure 5.

At the time, only the AC component of the vibration signature was being acquired. Therefore, the very large shift in the DC gap voltage went undetected. Those monitoring the vibration signatures believed the apparent probe saturation to be an electrical failure. The first indication that there was a serious problem was when the RCP seal failed from O-ring damage.

Subsequently, the bearing was found to have much of its babbitt surface worn off, and the bearing journal surface was worn heavily, though neither appeared to have made hard contact.

III. Fatigue Considerations

III.1 General Comments

(1) The bending stresses in RCPs and RCP motor shafts are, under normal conditions, quite low, and are only a small fraction of the endurance limit for the materials from which they are fabricated. Nonetheless, shaft fractures and failures do occur and are usually shown to be the result of high-cycle fatigue. Historically, they have occurred in areas where there has been a high stress concentration factor, either as a product of the shaft design or as a result of thermal cycling.

So, with regard to bearing failures and their effects on fatigue life, two questions arise into which this paper attempts to provide insight:

a. Does the failure mechanism cause an increase in the cyclic stress at a location previously identified as a failure site? That is, does the bearing failure mechanism increase the likelihood of a known fatigue failure mode?

b. Does the failure mechanism cause an increase in the cyclic stress at a location NOT previously identified as a failure location to such a level that it may become a failure site? One criterion for ‘too-high’ is whether the cyclic stress exceeds that seen at a known failure site. If a previously unidentified site is shown to have higher stresses than an identified failure site, then it must also be examined to determine if it has the potential for a high stress concentration.

III.2 Modeling Methodology

(1) Assumptions for Individual Cases

a. Normal Operation

For this case, all bearings are considered to have design values for stiffness and damping. There is no misalignment. For this and all the comparison cases, an impeller discharge load of 2000 lbs was considered.

b. Fully-failed Lower guide bearing

This case considers the operation of the pump/motor with NO lower guide bearing stiffness or damping. The pump-motor combination runs with support only at the upper motor bearing and the pump bearing.

c. Severe Misalignment

There is no loss or increase of stiffness considered for this case. The shaft center is considered displaced, for the purposes of this study, 0.060 inches.

d. Unidirectional Lower guide bearing failure

In this case, which represents a bearing with one or two shoes having excessive clearances, the bearing stiffness is considered normal in one plane, and zero in the other.

(2) In each case, it is assumed that rubbing between the rotating and stationary parts at clearance fits does NOT develop. Rubbing sharply changes the dynamic characteristics as well as the shaft stress distribution because the rub location acts as an additional non-linear support.

Rubbing can usually be identified using the vibration spectrum by the presence of a series of harmonics of running speed (1X, 2X, 3X, etc.) or by harmonics of an integer fraction (1/3X 2/3X, 1X, 4/3X, etc.) of running speed.

(3) Normally a shaft is subjected to cyclic stresses by stationary forces such as misalignment or pump discharge loading, and is subjected only to constant stresses by dynamic synchronous forces (e.g., unbalance). The stationary nature of stresses due to synchronous forces assumes, however, that the amplitudes of vibration in the two planes of movement are the same (i.e., circular synchronous whirl).

To illustrate this point, a comparison of the shaft bending stresses for two cycles of shaft revolution is presented in Figure 6. It presents the stress due to (1) a stationary load such as misalignment, (2) synchronous loading (such as unbalance) on a shaft where the response is 'normal', (3) a synchronous load on a shaft where the bearing is fully-failed, and (4) a synchronous load on a shaft where the bearing is failed in one direction only. The figure is provided for comparison and does not represent the actual calculated stresses for the unidirectional failure considered here.

The figure illustrates that the bending stress due to a unidirectional failure oscillates between that for a normal bearing and that of a fully-failed bearing, and thus creates an additional cyclic stress on the shaft. The oscillation occurs at twice the shaft rotational frequency.

Generally, both types of cyclic stress need to be considered to accurately determine the shaft fatigue. However, for cases where the dynamic responses have been shown to remain similar in both planes of movement during 'failed' operation, it has been assumed that the dominant effect on bending stress, and therefore fatigue, is due to the stationary loads. That is, if the measured vibration bears the appearance of circular synchronous whirl, the cyclic loading due to unbalance is ignored.

Of the cases examined in this presentation, the synchronous loading is considered only for the case of the unidirectional bearing failure.

III.3 Results

(1) All Cases

The first question raised with regard to fatigue life is whether the cyclic stresses at known failure sites would be worsened by any of the failure mechanisms considered.

Based on the assumption that rubbing does not develop at points below the lower bearing (such as the labyrinth seals), then NONE of the mechanisms considered will cause an increase in the cyclic stresses at the location for which Westinghouse and pumps of similar design are known to fail.

Because the most common failure site is below the lower-most bearing point, it is affected only by the impeller loading.

Table 1 compares the shaft bending stresses due to static loading in the fully-failed and severely misaligned bearings to those in a normal bearing. The location of the highest bending stress is shown in bold print for each case. The fatigue loading for the unidirectional failure is not easily compared because of the effects of dynamic loading.

Table 2 provides a comparison of the bearing reaction forces for the same cases as in Table 1.

(2) Normal Operation Only

For normal operation of a main coolant pump, the dynamic responses in the two planes of shaft vibration are similar, and are the result of residual unbalance and runout. Under these conditions, the dynamic response contributes very little to the fatigue loading of the rotating assembly. There will be minimal high cycle fatigue loading to the rotating components due to normal radial loading (static) such as pump discharge loading.

(3) Fully-Failed Bearing

In this case, the stiffness of the support assembly is drastically changed. There may be an increase in the unbalance response and/or runout conditions due to the failed support. However, the dynamic response remains approximately axisymmetric. The major change in the cyclic stresses is the result of static loading. While the full failure of the lower guide bearing does not affect the stress below the pump bearing, the cyclic stresses in the shaft seal region increase to a point that they are of approximately the same magnitude as those seen at the maximum stress location in a normal pump shaft. This is not expected to pose a major problem for shaft integrity because there are no sharp thermal transition areas in the seal region. It does, however, suggest a sharp reduction in the life of seal components such as o-rings and rubbing faces, since these do not have the same extensive design margins of safety as the shaft.

(4) Severe Misalignment

Under conditions of severe misalignment, the static response is increased, and depending on the extent of misalignment can cause increased fatigue loading in the rotating components. For the case studied, the shaft fatigue loads are increased most sharply in the upper seal and lower motor shaft areas. This is consistent with findings in the field. During the refurbishment of this motor, the bearing journal was found to be worn to such an extent that the wear was easily visible with the naked eye.

(5) Unidirectional Failure

When individual shoes come loose, the bearing has stiffness and damping characteristics which are quite different in one plane from those in the other. The model assumes that, although the responses in the two planes of movement are different, they are largely independent.

The value of the cyclic stress due to static loading is, therefore, approximated by the average of the cyclic stress with a normal bearing and that seen with a fully-failed bearing. In this case, there is an additional cyclic stress component due to the dynamic loading. The amplitude of this component is equal to the difference between the amplitudes of the fully-failed and the normal bearing cases. Because the two components of cyclic loading occur at different frequencies, the combination of their effects may be combined using methods such as Miner's rule. However, since neither component approaches the endurance limit of any coolant pump shaft material, it is safe to say that the expected life of the shaft will still be infinite following such a bearing failure.

As with the other cases, the increased cyclic loading may be expected to reduce the durability of other components such as the seal. The maximum stress seen in this case is the same as that seen with the fully-failed bearing, so the location of the greatest concern is the same also. Because of the added cyclic stresses attributable to the dynamic loading, the unidirectional bearing failure may be expected to have a more detrimental effect on adjacent components than the fully-failed bearing.

IV. Summary

IV.1 Vibration History

(1) In all cases, vibration data should be augmented with any other data available to support a timely and correct diagnosis and correction.

(2) Vigilance regarding oil level monitoring and bearing temperature are vital tools to assist in preventing the 'wiping' of a lower guide bearing. Vibration changes typically develop a matter of hours before the bearing becomes a 'fully-failed' bearing.

(3) For severe misalignment, caused as in the test case by restraint binding, monitoring of shaft centerline position (DC gap voltage) should be used as corroborating information whenever vibration data appears saturated.

(4) For a unidirectional failure (loosening shoes), the vibration undergoes a series of discrete shifts, increasing 1X running speed component each time. The symptoms are distinct from those due to a shaft crack in that only minimal 2X running speed vibration appears, and that the rate of progression does not increase.

(5) Also for a unidirectional bearing failure, as the failure progresses, the two responses in the two planes of movement become increasingly different from one another.

IV.2 Fatigue

(1) Because the most common failure site is below the lower-most bearing point, it is affected only by the impeller loading. **Based on the assumption that no rubbing occurs, lower guide bearing failures do NOT increase the likelihood of the typical shaft failure mode in Westinghouse pumps where failure occurs at the thermal sleeve anti-rotation pin.**

(2) A fully-failed bearing and a severely misaligned bearing have similar effects with regard to the stresses in the shaft, and cause an increase in fatigue stresses near the upper end of the shaft seal.

(3) With a fully-failed bearing, the cyclic stresses in the seal region may be greater than those seen at the thermal sleeve anti-rotation pin location.

(4) With a severely misaligned bearing, the cyclic stresses in the seal region are likely to be considerably higher than those seen at the anti-rotation pin location.

(5) The stresses in the seal region are still very low, and would only be expected to pose a problem in the presence of a high stress concentration.

(6) In the case of severe misalignment, the stresses at the lower end of the motor are also increased substantially. For the fully-failed bearing, the stresses in the lower motor shaft increase only by approximately 30% for the case studied herein.

(7) Other components such as rubbing face seals or shaft sleeve o-rings, which may not have as large design margins as the shaft itself, may be damaged either by a lower guide bearing failure or by misalignment.

(8) The unidirectional failure, while causing additional cyclic shaft stress compared to a fully-failed bearing, still does not challenge the integrity of the shaft.

Table 1

Comparison Max Shaft Stress for Static Conditions

	Normal	Failed	Misaligned (0.060 inches)
Below Pump Brg.	**1.0 KSI**	1.0 KSI	1.0 KSI
Upper Seal	0.1 KSI	**1.06 KSI**	**2.2 KSI**
Mtr. Shaft Extension	0.2 KSI	0.4 KSI	1.9 KSI

[KSI = kip per square inch, where kip = 1000 pounds force]

Table 2

Comparison of Reaction Forces for Bearing Conditions

(as a fraction of impeller discharge load)

Bearing	**Normal**	**Failed**	**Misaligned (0.060 inches)**
Pump	- 1.4	- 1.13	- 2.1
Lower Motor	+0.5	0.0	+1.85
Upper Motor	- 0.1	+ 0.13	- 0.75

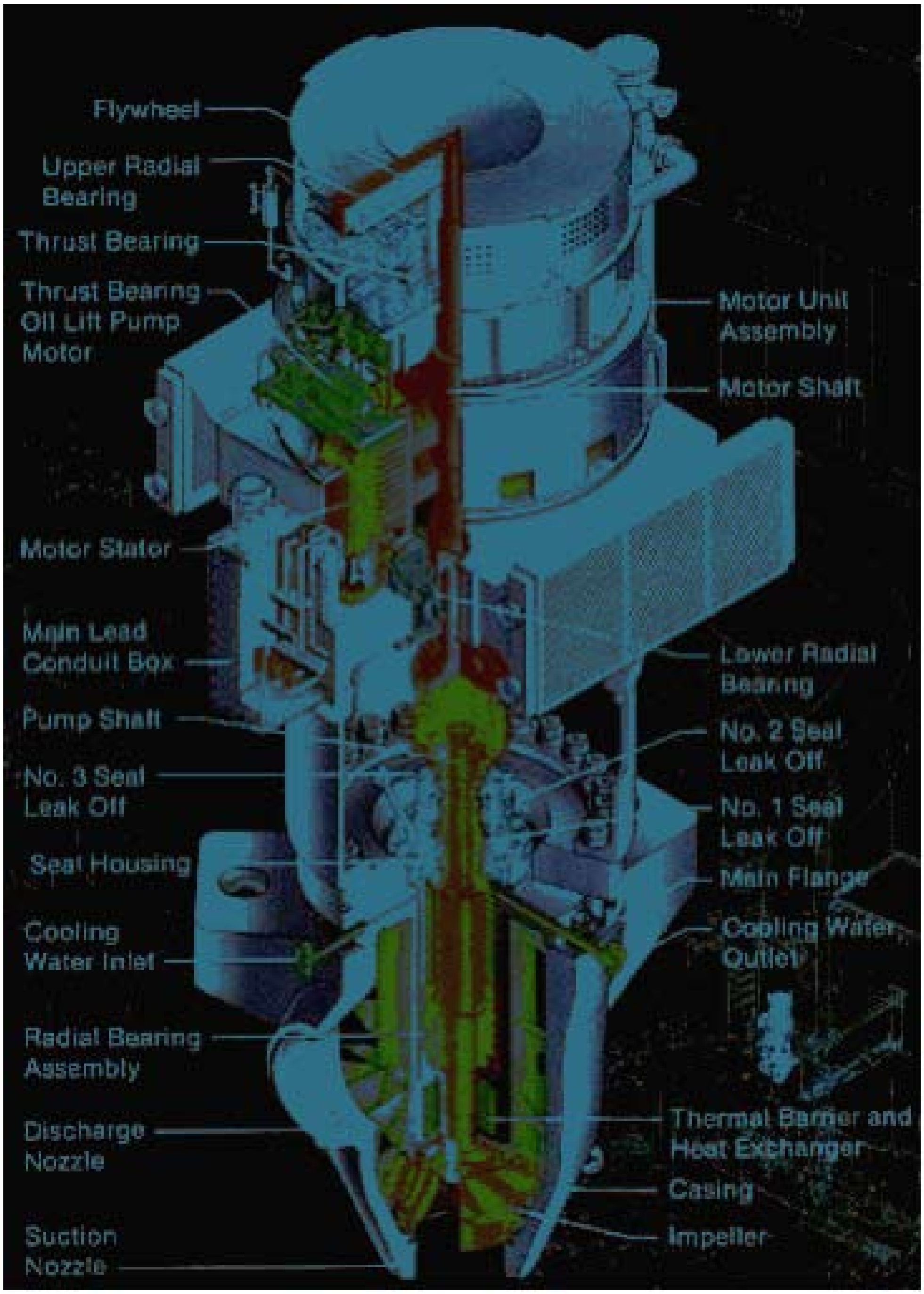

Figure 1 – Typical Westinghouse-Style Reactor Coolant Pump and Motor

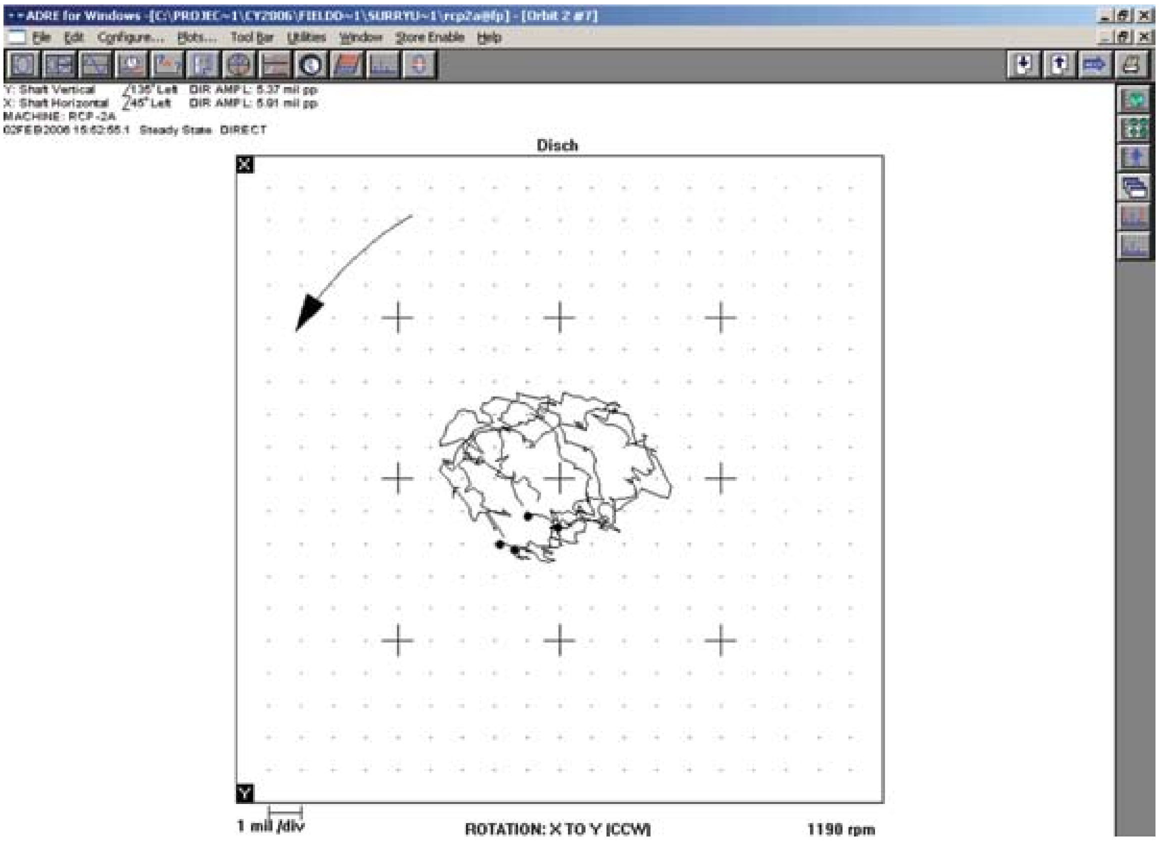

Figure 2 – Shaft Orbit for an RCP during Normal Operation

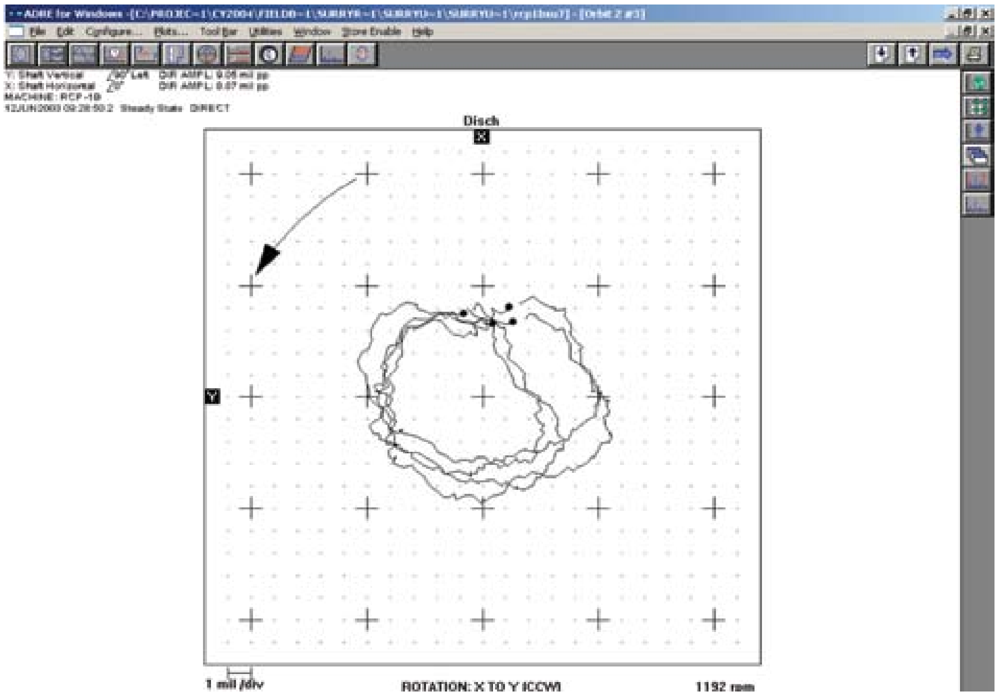

Figure 3(a) – Shaft Orbit for a pump with Unidirectional Bearing Failure (initial operation)

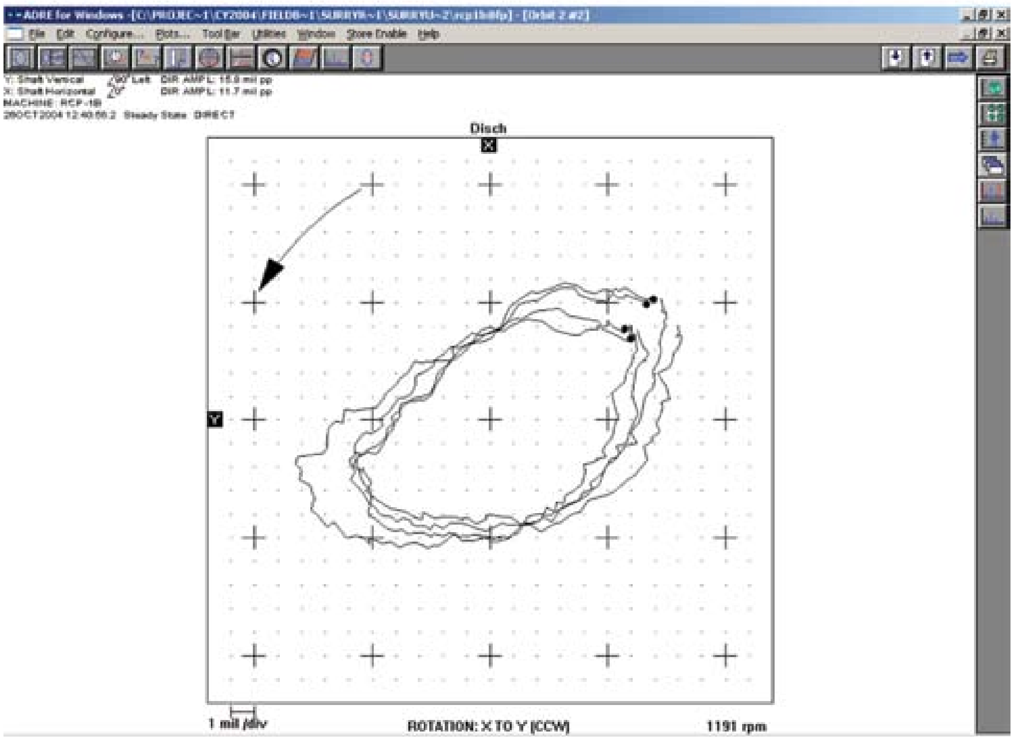

Figure 3(b) – Shaft Orbit for a pump with Unidirectional Bearing Failure (after 14 months of operation)

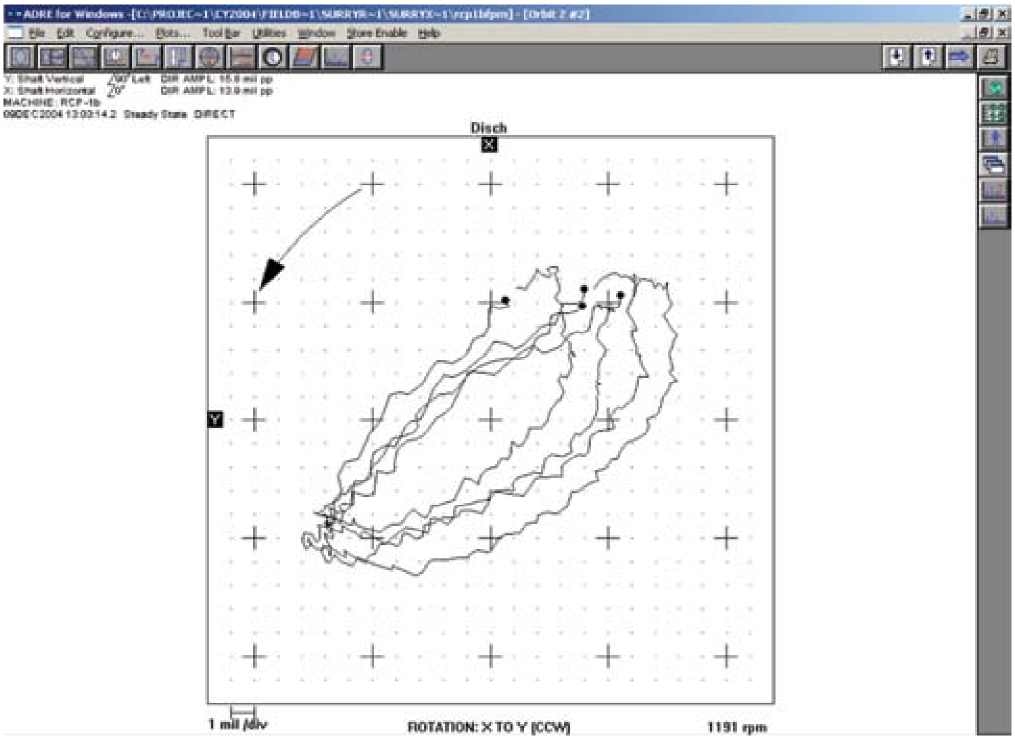

Figure 3(c) – Shaft Orbit for a pump with Unidirectional Bearing Failure (following 15 months of operation and balancing)

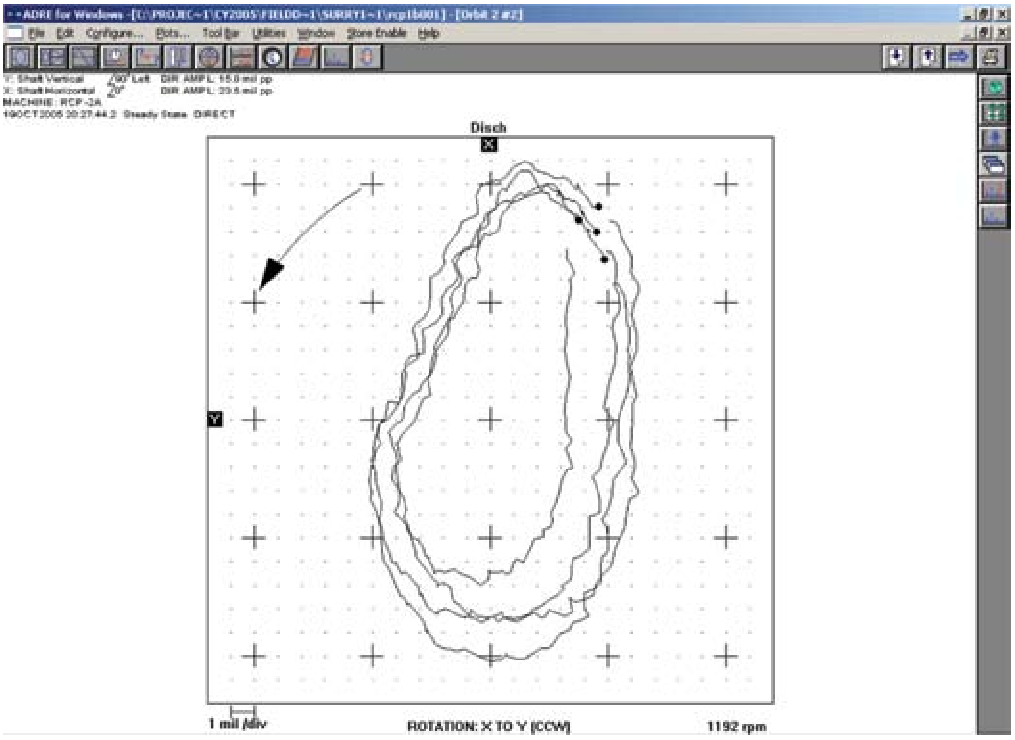

Figure 3(d) – Shaft Orbit for a pump with Unidirectional Bearing Failure (just prior to forced repair)

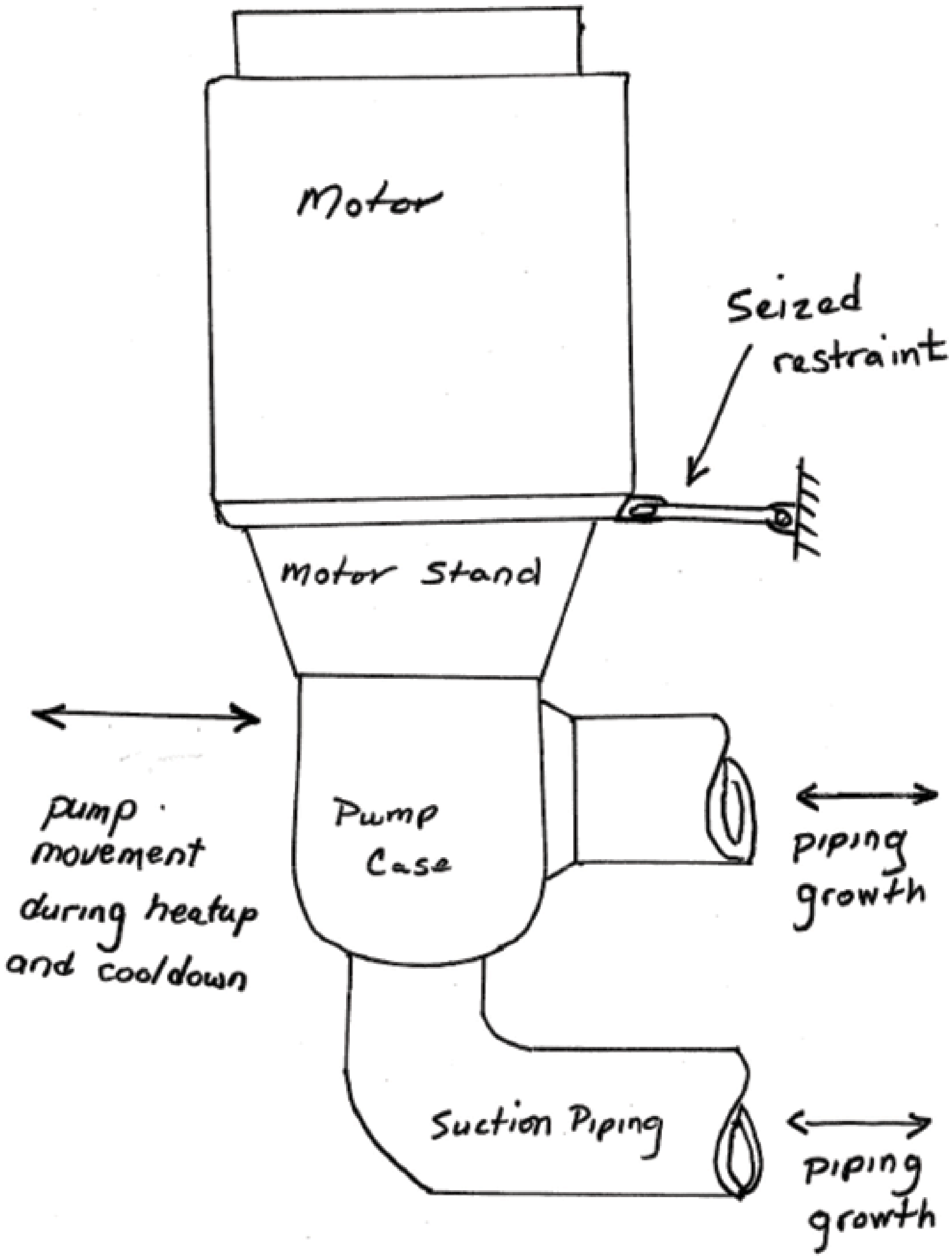

Figure 4 – Simplified Sketch showing Misalignment Mechanism

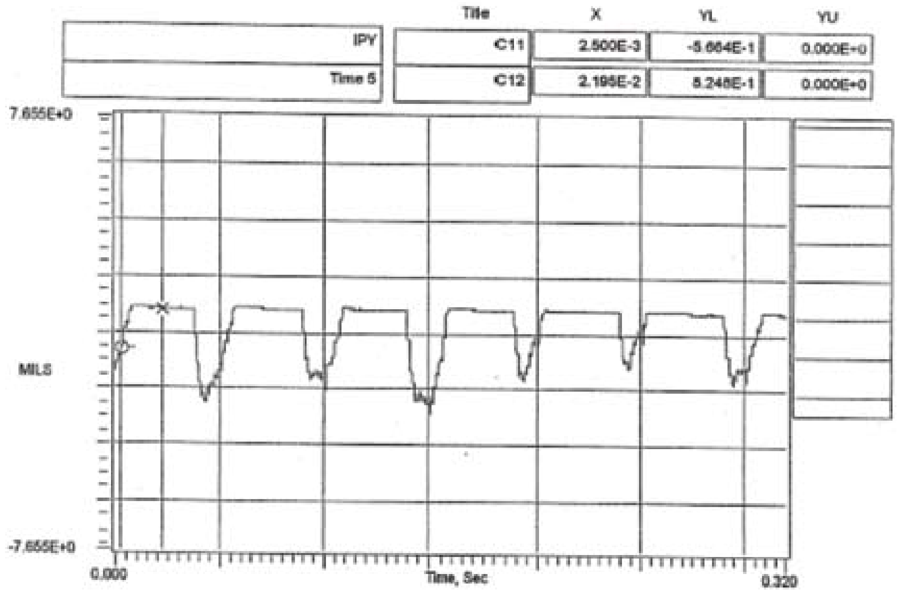

Figure 5 – RCP Shaft Vibration Time History for Severe Misalignment (clipping due to proximity probe out of range)

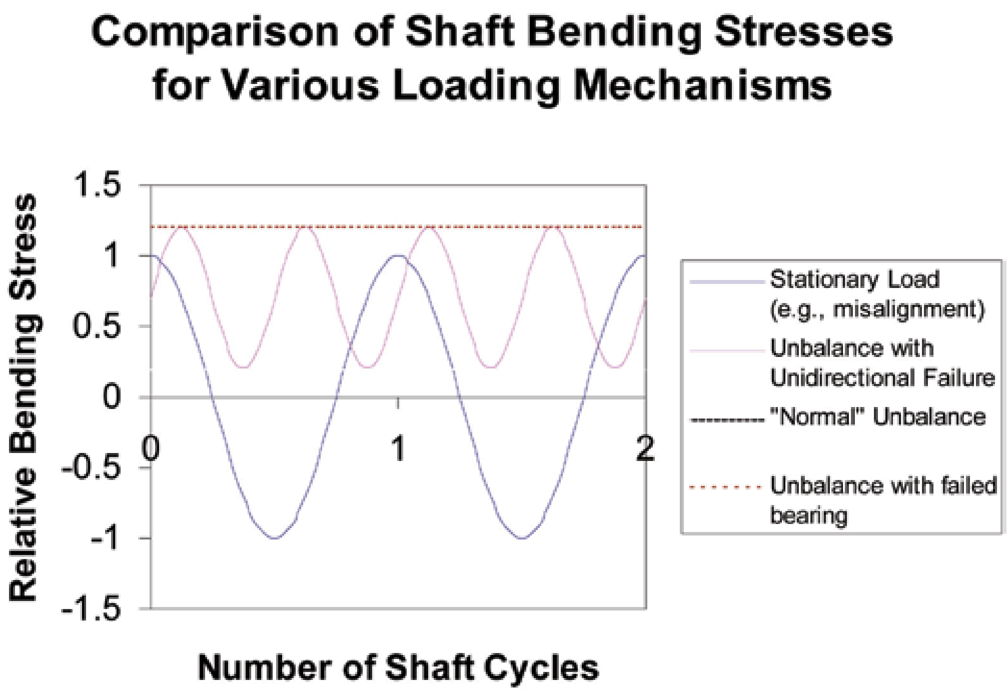

Figure 6 – Comparison of Shaft Bending Stresses for Various Loading Mechanisms

Scale Model Testing of Air Transport through Pump Suction Piping

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Abstract

This paper summarizes a scale model test program conducted for the Palo Verde Nuclear Generating Stations. The test program investigated the potential to transport an air volume initially trapped in a horizontal segment of the containment sump outlet line through a vertical downcomer and subsequently into the Emergency Core Cooling System (ECCS) and Containment Spray (CS) pump suction lines. The testing was conducted in three phases. The first two phases modeled the pump suction transfer from the Refueling Water Tank (RWT) to the containment sump. The first phase investigated the manner in which the liquid outflow from the sump interacted with the air volume, the ability of the liquid outflow to transport air through the vertical downcomer, and the flow pattern of the two-phase mixture in the downcomer. The second phase investigated the nature of the two-phase flow pattern produced in the pump suction piping for the High Pressure Safety Injection (HPSI), Low Pressure Safety Injection (LPSI), and CS systems. The third phase investigated the sensitivity of model scaling factors on the transport process.

A range of containment overpressure and system flow rates were investigated in the tests. The set of conditions that would be expected for a large beak loss-of-coolant accident (LOCA) event were found to result in the air being transported from the horizontal segment into the vertical segment and subsequently to the pump suctions. The two-phase flow pattern in the vertical segment was observed to be liquid continuous with dispersed air bubbles. The two-phase flow pattern in the pump suction lines was observed to approach a stratified state in the lower pump suction header. The ultimate dispersion of the stratified air was found to be specific to the orientation of the HPSI pump and CS pump suction connections off the lower header. The majority of the initial air mass displaced from the pump suction line accumulated in the pump suction header was subsequently discharged through the HPSI pump. Limited air was observed to be discharged through the CS pump for those cases where the HPSI pump was not running with or without the LPSI pump running. Very little air was only intermittently discharged to the LPSI pump.

This paper provides a description of the test facility, test processes, along with an overview of the sensitivity of boundary conditions, system operating parameters, and model scale on the observed transport process and associated flow regimes.

1.0 Introduction

These Phase 2 integral system tests were preceded by a Phase 1 test program (4 inch transparent piping with a single pump) and phenomenological tests (transient tests in transparent 8 inch piping). Both of these showed that (a) air would be transported into and downward through the downcomer and (b) Froude number scaling was not appropriate for the downcomer.

In this Phase 2 one-sixth scaled integral system study, a range of containment overpressures and system flow rates were studied for the containment sump recirculation phase of ECCS operation. A set of conditions that would be expected for a range of LOCA break sizes were investigated to assess the potential for air, initially trapped between the containment sump suction valves, to be pulled into the suction piping for the HPSI, the CS and LPSI pumps. These were examined through three similar, but different, experimental configurations that included operating pumps as follows:

Configuration 2A - HPSI and CS,

Configuration 2B - LPSI and CS,

Configuration 2C - HPSI, LPSI and CS.

The experimental results from the scaled Configuration 2A were used to formulate the test conditions for full scale tests with HPSI and CS pumps performed at a different laboratory. All three of the scaled configurations were used to characterize the response of the plant systems.

2.0 Experimental Apparatus

Figure 1 is a schematic illustration of Configuration 2C which encompasses the other two configurations. This apparatus was constructed with transparent plastic pipe and is a one-sixth scale model of a single train of HPSI, LPSI and CS for the plant system and includes the two isolation valves with one atmosphere of air initially trapped between the valves. The only deviation from the one-sixth scale is the downcomer pipe length. In two-phase vertical downflow, the water velocity determines the potential for downward air transport. Hence, to represent the plant conditions the scaling deviates from the Froude number as given by

$$N_{Fr} = \frac{U}{\sqrt{gD}} \tag{1}$$

to one in which the downward water velocity is the same as in the plant. To accomplish this, the downcomer pipe is 3 inches in diameter instead of the 4 inches dictated by Froude number scaling. (In the above equation, U is the water velocity, D is the pipe internal diameter, and g is the gravitational acceleration.)

Scaling of the plant geometry was followed in terms of the location and geometry of the suction locations for the three pumps. Of particular note is that the suction piping for the HPSI pump is a horizontal pipe at the equator of the pump suction header whereas the CS and LPSI suctions are at a 45° downward angle. As is discussed later, this HPSI suction location is influential in determining the extent of air transported into the HPSI suction piping.

Before initiating these tests with multiple pumps, which permitted flow control to the individual suction connections, other tests were performed with a single pump at one-sixth scale and transient tests at one-third scale with the HPSI and CS pumps simulated. It was these tests that clearly illustrated the importance of the downward water velocity in the downcomer. Moreover, the one-third scale tests revealed a vortex formation at the HPSI suction location with a hydraulic jump immediately downstream of this suction. Whether this occurred in the scale model of the plant was one of the principal objectives of the integral system tests.

To measure the air transported to the HPSI pump, an air separator was installed on the HPSI suction line as illustrated in Figure 1. This separator captures the air, which is measured by a differential pressure sensor (see Figure 2). In addition, this enables the HPSI pump to continue at full flow which is conservative with respect to maximizing the air transport to the HPSI pump. The air accumulation rate in this separator was measured for different HPSI flow rates and used to determine the spectrum of air intrusion rates to be used in the full scale tests with a horizontal shaft, multi-stage HPSI pump.

Due to the location of the CS suction port (downstream of the HPSI takeoff) and the 45° downward orientation, very little air was pulled into this pump suction flow. A small separator at the top of the pipe was used to measure the rate of air ingestion from the pipe suction header. This was used to formulate the test conditions to be examined for the full scale tests with a vertical shaft, single stage pump like that used for the CS.

In addition to the rate of air transport to the pumps, the two-phase flow pattern was also an important parameter for the full scale tests. Consequently, digital video cameras were positioned to observe the transient flow structure in the following locations:

- between the two butterfly valves used for sump isolation,
- at the top of the downcomer,
- at the HPSI suction takeoff from the lower pump suction header,
- along the HPSI suction piping just upstream of the air separator,
- at the CS suction takeoff from the lower pump suction header, and
- at the LPSI suction takeoff at the end of the lower pump suction header.

These direct observations proved to be invaluable in assessing the transient air-water flow patterns as well as in demonstrating the appropriateness of the scaling analyses (Froude number for the horizontal lines and water velocity for the downcomer).

3.0 Test Performance

All of the tests were conducted by initiating flow from the simulated Refueling Water Tank (RWT) to the pump suction header and then to the operating pumps. Since this only establishes the initial condition in the pump suction header, the discharge flows of the operating pumps are returned to the RWT.

The test is initiated when the recirculation actuation signal (RAS) begins to simultaneously open the two butterfly isolation valves of the simulated containment sump. Depending on the pressure in the containment sump at this time, the air is somewhat compressed and transported to the downcomer pipe.

As the isolation valves open, the pipe suction header is exposed to the containment pressure plus the static head of the water in the containment sump and the downcomer piping. This pressure exceeded the pressure in the RWT and caused the check valve on the RWT suction to close. With this action, the pump suction header water supply is transferred from the RWT to the containment sump. Once this occurs, the transport of air through the suction piping is determined by the Froude number in the horizontal piping and the water velocity in the downcomer.

Since the experiment did not include a representation of the Reactor Coolant System (RCS), the pump discharge flow rates were switched from a return to the simulated RWT to the containment sump. This manual (by a test engineer) switchover occurred as the water flow from the sump was observed to fill the region between the butterfly valves and was nearly simultaneous with the audible closing of the check valve on the RWT suction line. After this switchover, each test could be run until the air transport to the pumps was completed. This included those conditions in which the water flow rate from the sump was insufficient to sweep the air from between the butterfly valves and the air rose against the flow backward into the containment gas space.

4.0 Important Test Results

In both the plant system and the experimental facility, the operation of the CS pump provides sufficient flow to sweep the trapped air from between the isolation valves into, and down through, the downcomer piping and then into the lower pump suction header. Therefore, the tests of greatest interest are those with the CS pump operating during the switchover to containment sump recirculation, which is the expected behavior for the plant. Furthermore at the time of RAS, the LPSI pump is automatically shut down while the HPSI pump continues to run with a discharge flow rate determined by the RCS pressure, which would be determined by the LOCA size causing the accident state. Therefore, the major focus for the tests was the air ingestion for those conditions with the CS running at full flow and HPSI flow rates consistent with small, medium and large break conditions within the RCS.

As discussed above, with the RAS signal, the inboard and outboard butterfly valves open simultaneously over a 20 second interval. These valves are oriented stem vertically such that the openings begin at the equator of a horizontal pipe connecting them. As a result of the accident condition, the pressure in the containment plus static sump water level head is greater than the 1 atmosphere (atm) air volume between the valves, hence, the inrush of water from the sump compresses the air. Water can be seen to enter around the sides of the inboard valve and preferentially accumulate in the bottom of the horizontal pipe.

Before the air-water mixture can be transported into the downcomer pipe, the containment pressure needs to exceed the back pressure on the check valve downstream of the outboard valve. This back pressure is caused by the water head in the RWT and, for those accident conditions where this does not occur within about 10 seconds, the air will flow backwards into the containment. Therefore, the accident sequence conditions of interest in these tests are those with a sufficient containment pressure to open the downstream check valve within a few seconds. It is further noted that a higher containment pressure causes more compression of the air volume. Since the primary quantity of interest is the void fraction transported to the pumps, the lowest pressure sufficient to open the downstream check valve as the butterfly valve is opened, would give the maximum potential for the largest void fraction entering the suction location. In the spectrum of LOCA sizes, the small break LOCAs would give the limiting condition. However, the smallest break LOCAs would not be sufficient to quickly open the downstream check valve and the compressed air would be forced back into the containment by buoyancy.

Figure 3 shows the developing two-phase mixture 5 seconds after the motor-operated valves (MOVs) begin to open for a limiting sequence which opens the check valve. A frothy mixture is observed to be generated with an air bubble formed between frothy regions near the outboard valve. Two seconds later the air bubble has been reduced to a small region near the top of the pipe, i.e. most of the air has already been transported into the downcomer. Therefore, these scaled tests with the same Froude number as would occur in the plant system illustrate that the pump suction flow rate, which is principally due to the CS pump, would transport the trapped air into the downcomer pipe.

Digital video observations near the top of the downcomer show that a kinematic shock can be formed with some initial holdup of air. However, as the transient progresses, the air is eventually pulled into the downcomer flow. Similar observations at the bottom of the downcomer reveal a bubbly mixture as the flow exits this pipe and is transported into the horizontal pump suction header. The same flow patterns were observed in the transient one-third scaled tests that were conducted in preparation for these integral system tests. Maintaining the same velocity in the downcomer as the plant would experience caused this flow pattern. If Froude number scaling had been used, the water velocity in the downcomer for the one-sixth scaled test would have been comparable to the bubble rise velocity. Under these conditions, the air would tend to form large bubbles and rise against the flow (Wallis, 1969). In the plant system, the downward water velocity is approximately twice the bubble rise velocity (when the CS is operating) and the air would be swept along with the flow.

Observations from both the one-sixth and one-third scaled tests show that the flow pattern quickly transitions from bubbly to stratified flow as the mixture enters the horizontal pump suction header. This was seen within one to two pipe diameters. This further emphasizes the need for Froude number scaling in the horizontal parts of the system model. Consequently, the flow pattern at the HPSI suction port is stratified as the air begins to collect along the top of the header. As the void fraction in the header increases and the air-water surface approaches the top of the HPSI suction port, a vortex is formed that pulls air into the HPSI pump suction piping. Figures 4 and 5 show that this vortex as observed in the one-third and one-sixth scaled tests respectively. Note the similarity in the conditions at the entrance to the HPSI takeoff and the annular flow pattern developed as the air and water enter the pipe. Similar behavior at these different scales further supports Froude number scaling. The curvature of the opposite wall of the port is normal, and not reversed, and is indicative of a high void fraction, annular flow pattern.

As the air-water mixture enters the HPSI piping, a stratified flow pattern re-develops. This was an important observation for designing the full scale HPSI test facility; particularly for the small break conditions with a reduced HPSI flow due to the elevated RCS pressure. With the 90° downturn at the pump entrance, a reduced flow resulted in very little air entering the HPSI pump when the air separator was replaced with a straight pipe.

Figure 6 illustrates the rate at which the air mass was captured in the separator for different transients. Note that the air accumulates very quickly at the beginning of the transient and tapers off to a relatively slow accumulation rate (some air was observed to exit from solution). To aid in designing the full scale HPSI test, this air accumulation information was interpreted in terms of the rate of accumulation and these are illustrated in Figure 7 which shows the maximum rate develops in the one-sixth scale model within a few seconds of air arriving at the separator. This information was then translated into the most limiting case and interpreted in terms of the full scale test for the design and performance of the full scale experiments. Figure 8 illustrates this limiting air mass flow rate that was used for the full scale test. Recall that this information represents a conservative transfer of air to the pump suction since there was no degradation in the HPSI pump flow rate for the scaled test in which the separator was installed.

5.0 <u>Conclusions</u>

The following conclusions were derived from the three Phase 2 configurations for the one-sixth scaled integral system experiments representing the Palo Verde sump suction line behavior.

Configuration 2A

1. All of the important physical phenomena observed in the one-third scale tests were also observed in the one-sixth scale tests. This demonstrates that the scaling evaluations appropriately considered the governing physical processes.

2. The air-water mixing which occurred between the two butterfly valves in general created a well mixed two-phase bubbly flow pattern which transported the majority of the air out of the horizontal section and into the vertical downcomer.

3. An important aspect of a scaled experiment is to have a vertical downcomer designed such that, like the plant, the downward water velocity is considerably greater than the bubble rise velocity. For these integral tests, this was accomplished by reducing the downcomer diameter from 4 inches to 3 inches. As a result, there was no significant air holdup in the vertical downcomer and the air is transported to the lower horizontal header at the appropriate rate. Furthermore, those tests with HPSI and CS operating showed no bubble coalescence in the reduced diameter downcomer.

4. As the air is delivered to the lower horizontal header, a stratified flow pattern is developed. This flow regime is sustained by continuing downward flow and the experiments demonstrated that the CS pump flow alone will keep the air in the header. With the substantial water head provided by the vertical downcomer in the plant, this adds essentially 1 atm additional overpressure to the static pressure and adds to the compression of the air thereby reducing the air volume.
5. For those conditions with relatively low or no HPSI flow rate, the air occupies the upper regions of the horizontal pump suction header with an essentially uniform void profile along the length of the horizontal header (except at the entrance to the HPSI).

6. For the higher HPSI flow rates, a vortex is developed at the HPSI suction port that, in essence, limits the stratified layer in the suction header to that region from the beginning of the horizontal length to the tee for the HPSI branch. With a vortex at the HPSI take-off, there is a hydraulic jump formed which has a height that approaches the pipe radius. As a result, the hydraulic jump nearly closes off the entire cross section of the suction header downstream of the take-off. Under these conditions, virtually all of the air that is transported from the horizontal header is drawn through the HPSI suction line and this is approximately 60% to 80% of the gas initially resident between the upstream butterfly valve and the check valve in the sump suction line.

7. For all conditions there is little (< 5% void fraction) or no gas transported down the containment spray suction line. Therefore, there is no significant challenge to operation of the containment spray pump as a result of this set of conditions with 1 atm of air initially in the sump suction line.

8. For the flow through the HPSI suction line, the dominant flow pattern is one of stratified flow. This was observed in both the one-third scale tests and in the one-sixth scale integral system tests.

9. Using the numerous experimental tests performed in the integral system, the greatest delivery air mass and mass flow rate to the HPSI pump was developed for each nominal HPSI flow rate. Using the information from these scaled experiments, the effective air delivery rate histories to the HPSI pump were translated to be tested at full scale. Because these data were developed from measurements where there was no feedback on the pump a conservative interpretation is developed. Therefore, this data was applied to the full scale pump in a piecemeal approach that began with the appropriate air delivery rate early in the two-phase transient and then uses the feedback from the measured pump behavior to deduce the longer term air transport conditions. In this manner, the integral behavior for the pump was tested along with the approximate feedback as a result of the pump performance while undergoing air ingestion.

Configuration 2B

While not a design basis configuration, the opportunity to restart a LPSI pump after RAS is permitted within the emergency and abnormal operating procedures (EOP and AOP) for the plant.

1. These scoping experiments which related to the LPSI pump start assuming no HPSI flow demonstrated that the manner in which the LPSI pump was re-started was important.

2. If the LPSI pump were to re-start near runout conditions, considerable air could be drawn into the LPSI suction line.

3. If the LPSI pump re-started near the shutoff head conditions, essentially no air was pulled into the LPSI suction line.

4. It was concluded from these scoping tests that the evaluation for LPSI pump re-start should include a simulation with all three pumps, i.e. HPSI, CS and LPSI. This led to the tests with Configuration 2C.

Configuration 2C

1. These tests were performed with consistent HPSI and LPSI flow rates as if they were pumping to the same RCS pressure. For most of the tests, the HPSI pump was operated until the flow degraded to 50% of the preset initial flow rate, at which time the HPSI pump was isolated. This showed that there was considerable air pulled through the HPSI suction line prior to this isolation which substantially decreased the air in the pump suction header.

2. Experiments were performed to examine the integral response for conditions in which, following loss of HPSI, the control room operators would maintain one train of CS and shut down the other train to start the LPSI pump. These tests demonstrated that a complete shutdown of flow in a single train for a few minutes would enable the air to escape backward up the downcomer, leak through the check valve and flow into the containment sump and hence, to the containment atmosphere. Consequently, there was no air in the horizontal header when the LPSI pump was started.

3. For those experiments with a consistent HPSI and LPSI flow, the LPSI flow was activated at a pressure near that of its shutoff head and the flow rate was incrementally increased at a rate consistent with the RCS depressurization. With the long interval required for the LPSI flow to increase, the air void fractions pulled into the CS and LPSI line during this time were in the range of 2 to 5%. Hence, the air intrusion rates are well within those that have been demonstrated in the open literature (NRC, 1982) to be consistent with successful pump operation.

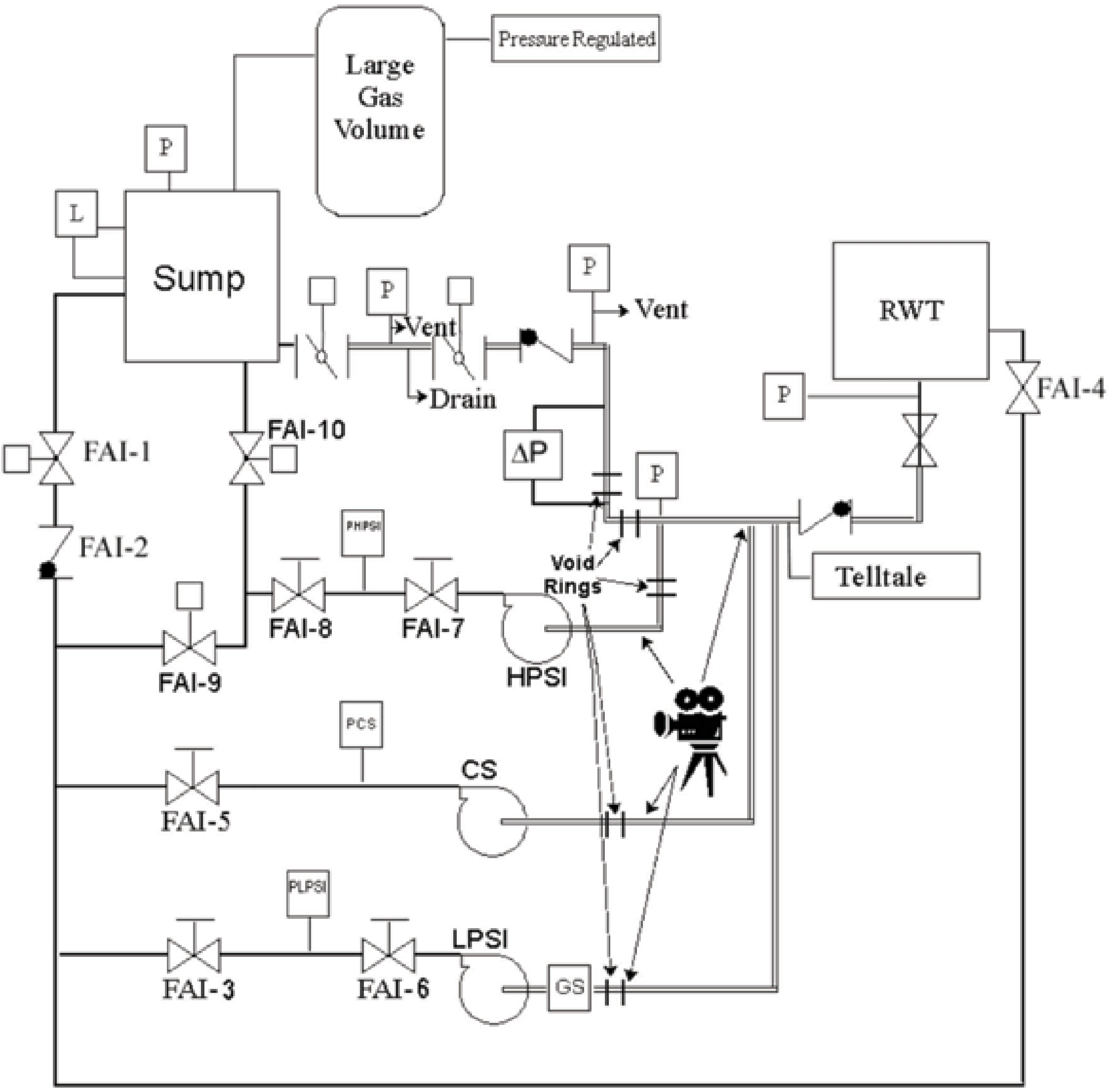
4. In one test configuration the HPSI continued operation during the entire test. This showed a degraded HPSI flow due to air intrusion; however, air and water flow continued. During this time the flow through the HPSI suction line remained in a stratified flow pattern and continued to pull air into the HPSI suction flow. Furthermore, the flow through the HPSI pump continued in a quasi-steady manner without any significant flow rate or pressure oscillations. A key to developing this operating state is that the air intrusion rate is directly related to the degraded pumping rate.

6.0 References

NRC, 1982, "An Assessment of Residual Heat Removal and Containment Spray Pump Performance Under Air and Debris Ingesting Conditions," NUREG/CR-2792, September.

Wallis, G. B., 1969, One-Dimensional Two-Phase Flow, McGraw-Hill, New York.

Figure 1 – Phase 2 test configuration 2C for post-RAS air intrusion.



PCS / PLPSI / PHPSI	=	Turbine Flow Meter
P	=	Pressure Measurement
ΔP	=	Differential Pressure Measurement
GS	=	Gas Separator
L	=	Water Level

Notes

- Double line pipe to be transparent.
- Digital movie cameras to record flow patterns at key locations, i.e., vertical downcomer, horizontal header for the three pumps, and branch lines.
- Telltale to confirm check valve position.

Figure 2 – Cutaway view of air-water separator.

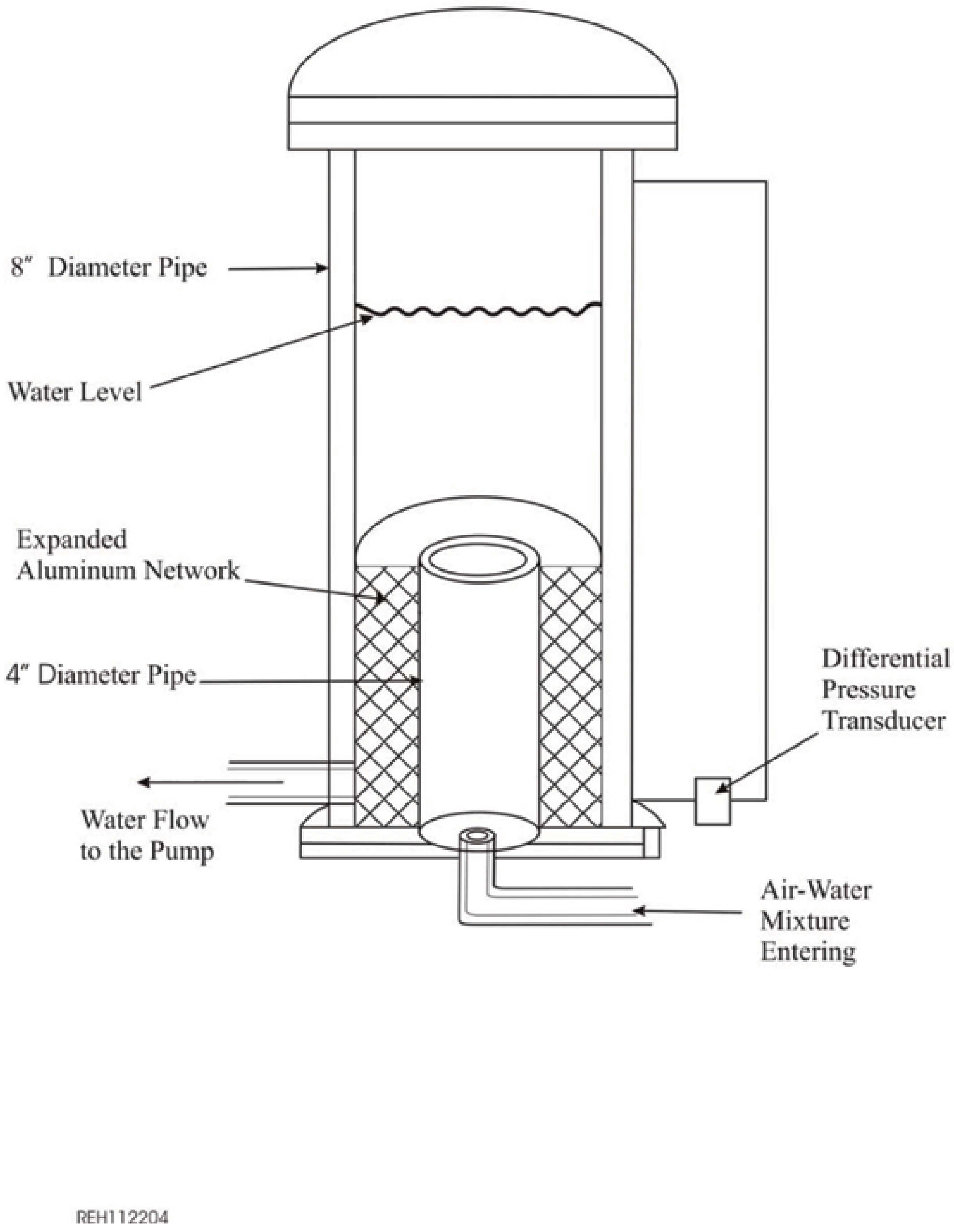


Figure 3 – Test PVA21 5 seconds after MOVs began to open.

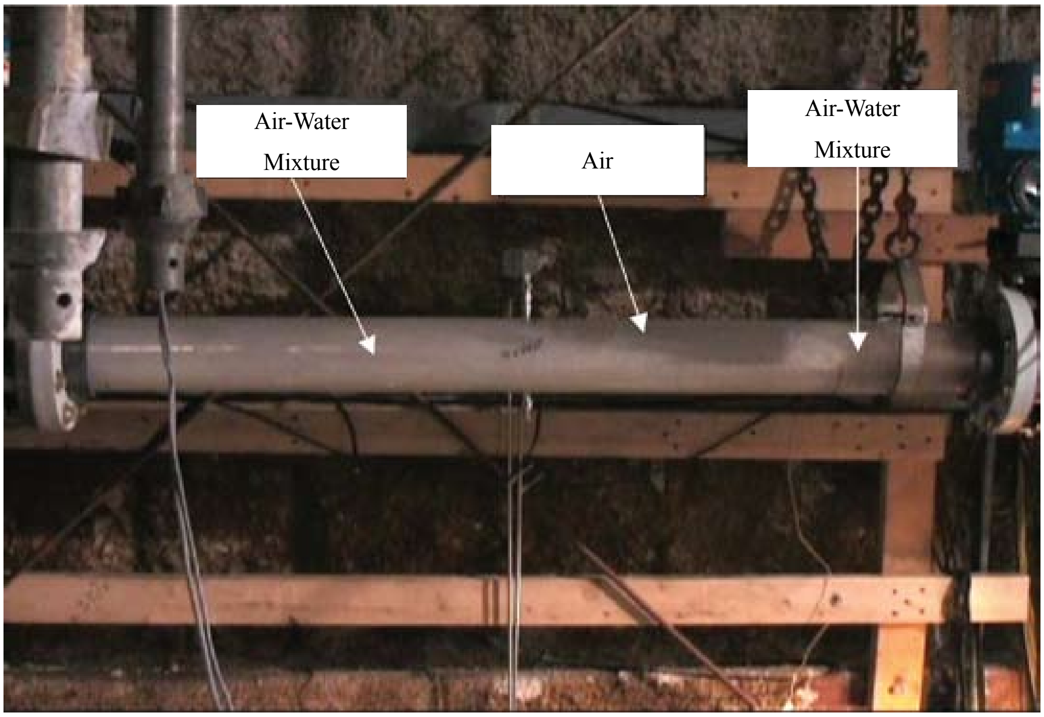


Figure 4 – Air vortex penetrating into sweepolet tee.
Note (a) angle of water swirl in HPSI suction line, (b) water level upstream of HPSI takeoff, and (c) water level downstream of this takeoff.

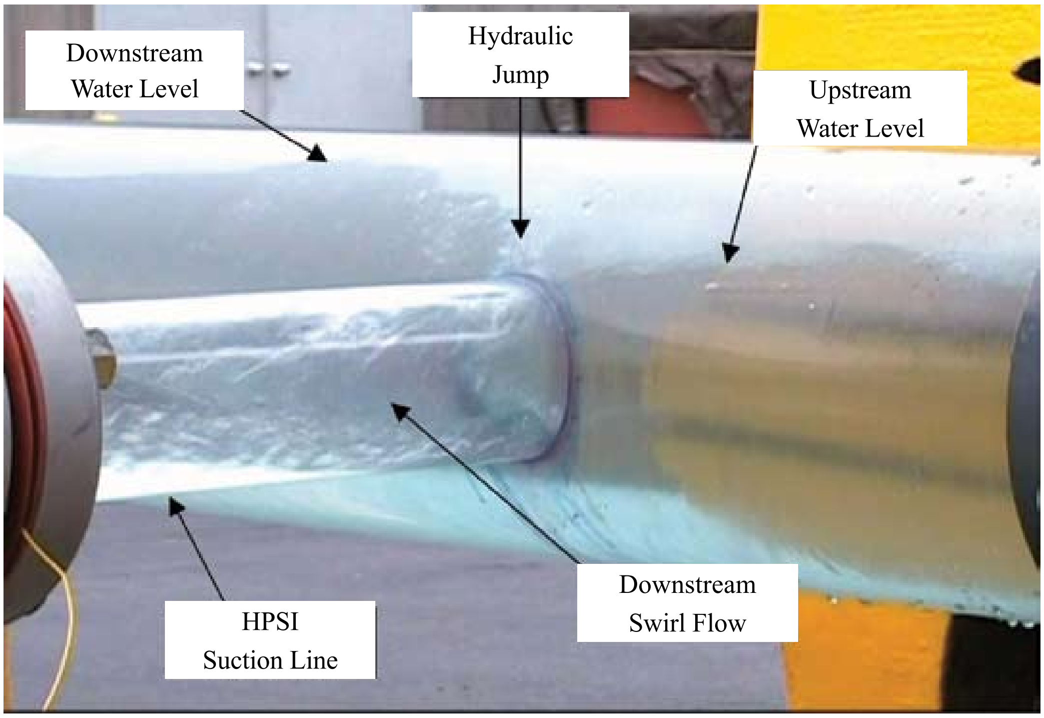


Figure 5 – Vortex in HPSI sweepolet tee for Test PVA21. Note stratified flow pattern in horizontal header and also curvature of opposite pipe wall.

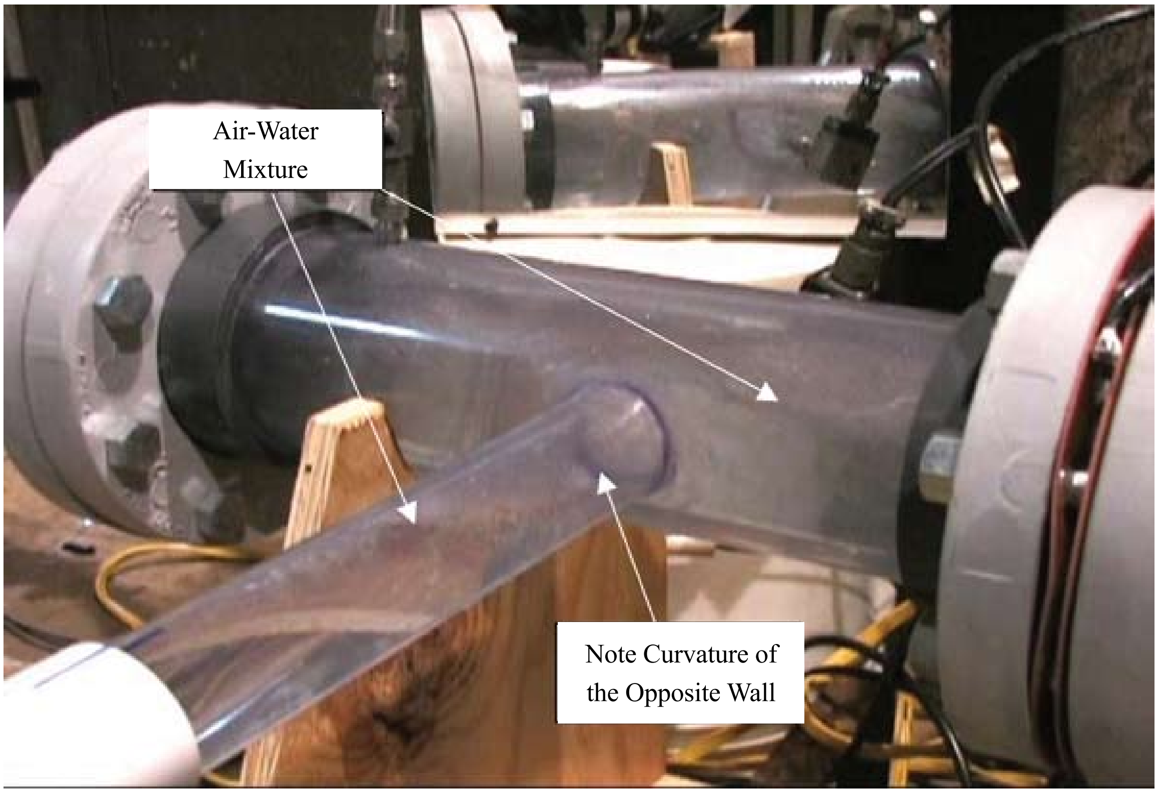


Figure 6 – Mass accumulation histories in air separator for scaled HPSI flow of 1310 gpm. (Nearly system runout conditions)

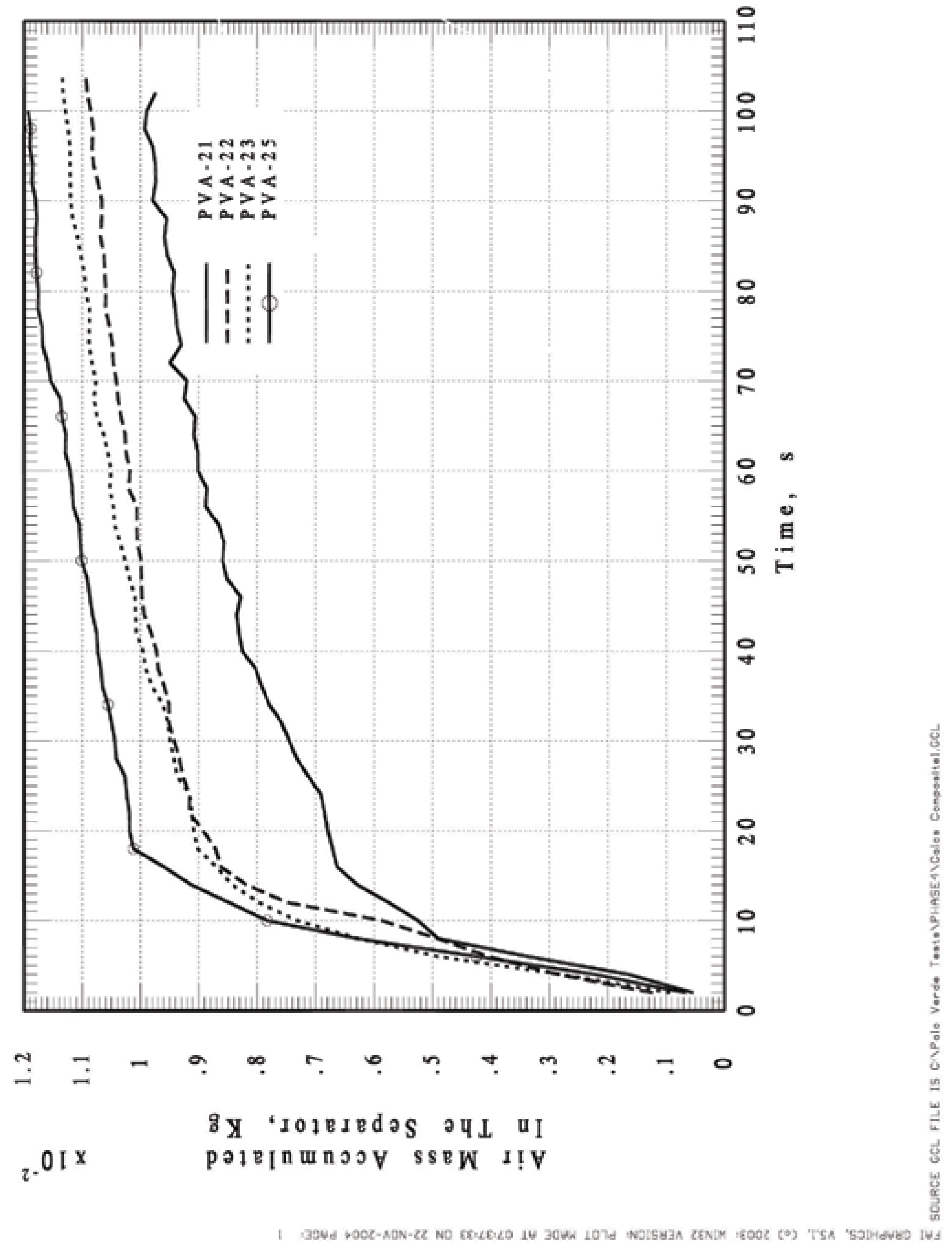


Figure 7 – Air mass flow rates to HPSI pump for scaled flow rate of 1310 gpm.

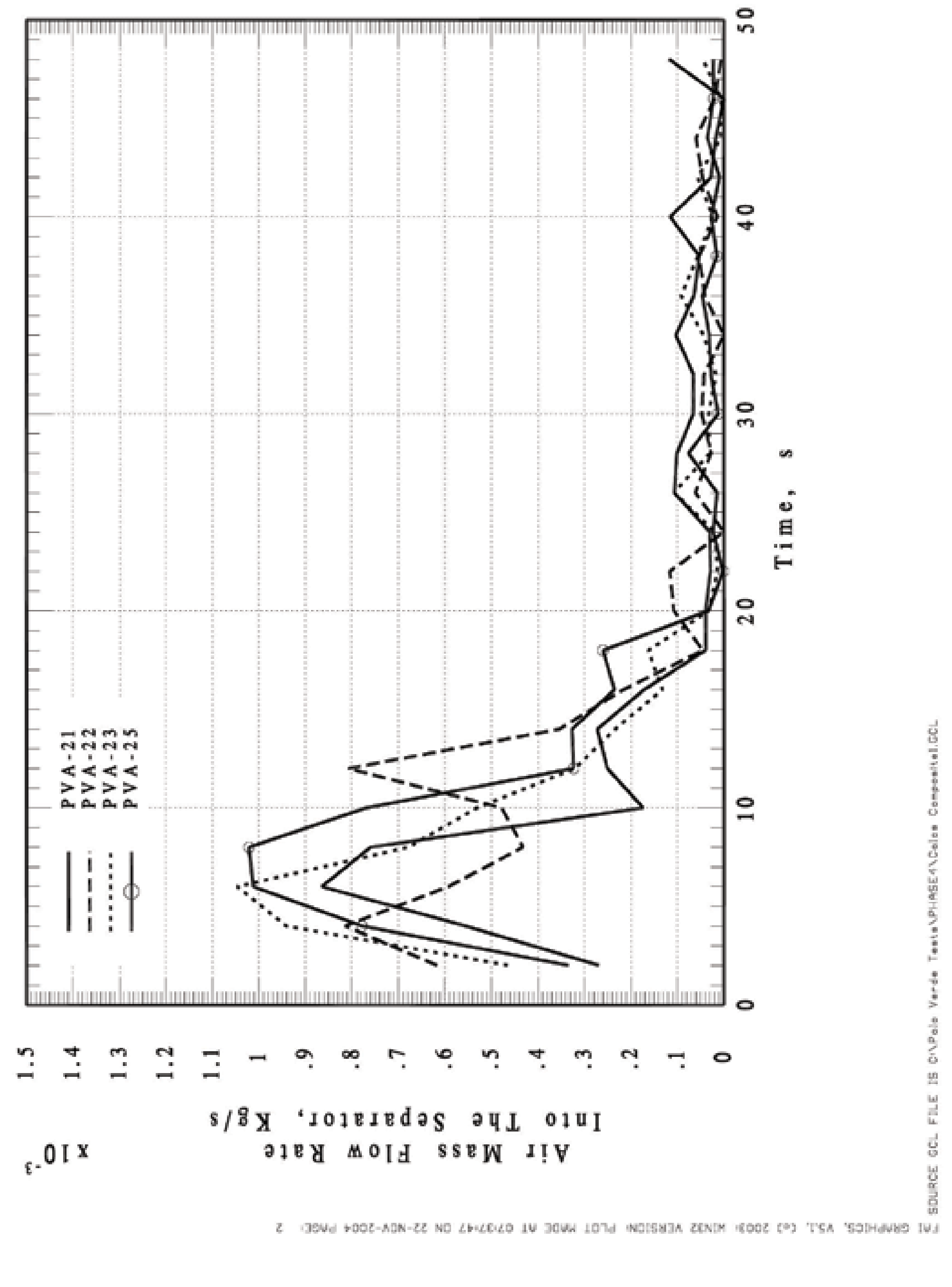
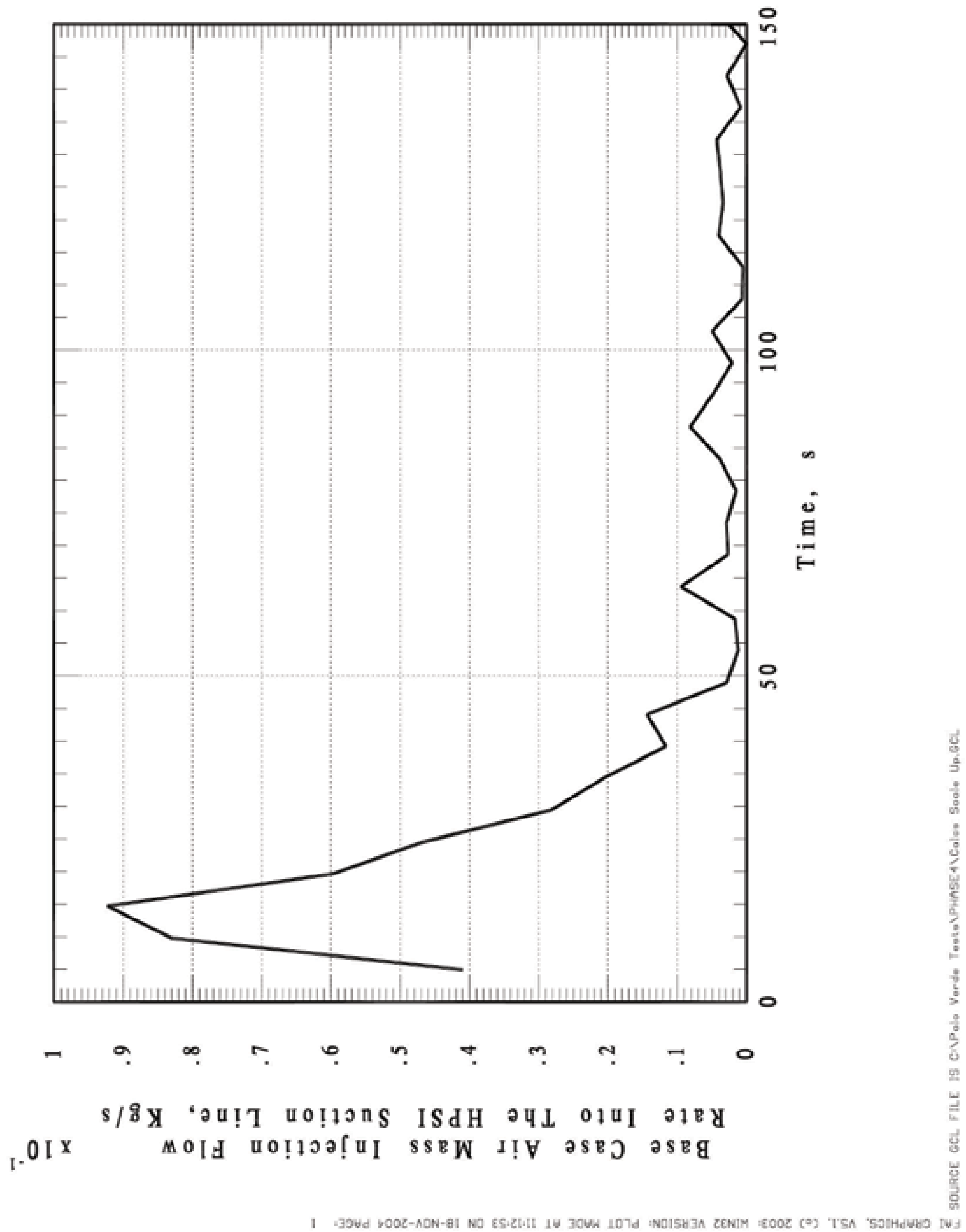


Figure 8 – Scaleup of 1/6th scaled tests to plant condition for full HPSI (1310 gpm) and CS pump flow rates.



AN APPROACH TO ESTIMATING PWR ECCS THROTTLE VALVE POSITIONS IN SUPPORT OF GSI-191 EVALUATIONS

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Abstract

In September 2004, the Nuclear Regulatory Commission (NRC) issued Generic Letter (GL) 2004-02, "Potential Impact of Debris Blockage on Emergency Recirculation During Design Basis Accidents at Pressurized-Water Reactors," to address Generic Safety Issue 191 (GSI-191) "Assessment of debris accumulation on PWR sump performance." GL 2004-02 requested pressurized water reactor (PWR) licensees to perform a "downstream effects" evaluation of their emergency core cooling system (ECCS) and containment spray system (CSS). GL 2004-02 also gave guidance on what analysis had to be completed in order to resolve GSI-191. These evaluations included a wear and plugging assessment of all ECCS and CSS components, including valves. One of the challenges in performing these evaluations is obtaining the positions of throttle valves in the ECCS. Without knowing the position of the valves, it would be impossible to assess the functionality of the ECCS during the postulated event.

The purpose of this paper is to present an approach which can be used to determine the valve position, given certain flow conditions. Working examples covering globe and butterfly valves are provided.

Introduction

In response to the NRC Generic Letter 2004-02, several nuclear power plants requested Westinghouse to complete a downstream effects evaluation of their ECCS and CSS. The ECCS and CSS in a typical Westinghouse PWR provide the ability to cool the reactor core and containment, respectively, by injecting water first from the Refueling Water Storage Tank (RWST) and then later from the containment sump. Figure 1 shows a pictorial representation of a typical Westinghouse PWR ECCS. In the case of a Loss Of Coolant Accident (LOCA), water in the containment sump may become laden with debris generated by the LOCA. This debris laden water creates the potential for blockage of valves, as well as could wear away valve internals at a more rapid rate than with debris free water.

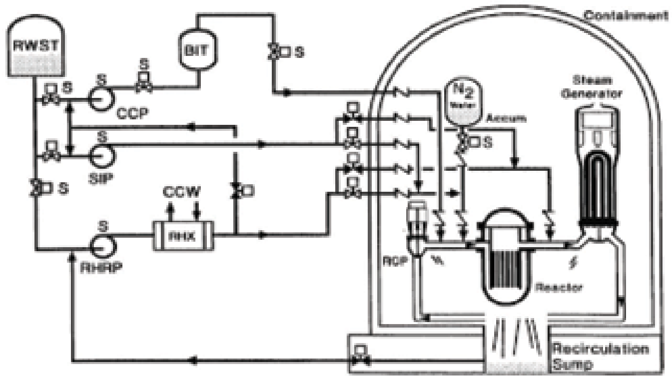

FIGURE 1 – TYPICAL WESTINGHOUSE PWR ECCS

In the process of performing the downstream effects evaluations, it was discovered that in some cases the positions of several throttled valves in High Head Safety Injection (HHSI) systems were unknown. The valves employed in these systems are varied both in size and design depending on the Architect Engineer's preference. Further, the valves can be set at any position for flow balancing. Thus, assessing the effects on the valves relative to the plugging and wear is impossible without either a) perturbing the existing flow balance by manipulating the valve to determine the position relative to full open, or b) performing calculations that should provide a reasonable guidance of valve position provided there is sufficient equipment and system data to support the analysis.

This paper describes an approach to determine the valve flow coefficient, C_V, of two valves and consequentially determine the position, using the C_V. With the valve positions defined, a downstream effects evaluation can proceed. For example, it is possible to assess if plugging can occur if the plug is close to the seat or if the valve is going to wear.

Method Discussion

The approach used to establish the position of the two throttled valves is as follows. First, determine the C_V of the throttled valve by accounting for all pressure drops in the flow path except the throttled valve in question. Second, use the resultant C_V to estimate valve position for both choked and non-choked flow. These steps are as described in further detail below.

Determination of C_V

To determine the C_V of each valve, first determine the characteristics of the flow path in which the particular valve is installed. Flow paths should begin upstream of the valve, at a point of known gage pressure relative to a point of known pressure downstream of the valve. The flow path should be constructed such that all pipe lengths, fittings, meters, valves (other than the throttled valve), and any other structure which could cause a resistance in flow is represented. See Figure 2 for a visual depiction of what a flow path should resemble. The flow path in Figure 2 consists of an orifice (upstream gage pressure point), two valves (including the throttled valve), seven sections of 1" pipe of various lengths, four pipe bends, and one sudden expansion from 1" to 2".

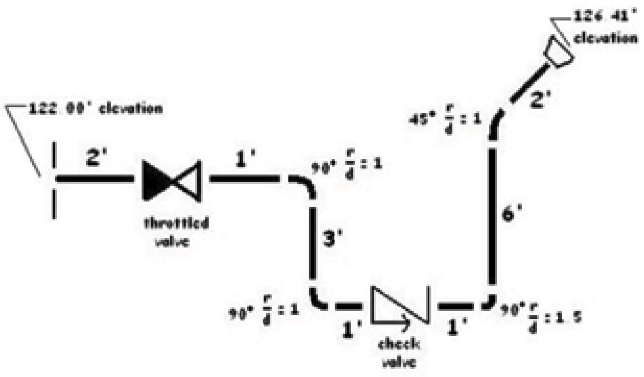

FIGURE 2: EXAMPLE OF FLOW PATH

Once a flow path is constructed, resistance coefficients should be estimated for every component of the flow path, with exception of the throttled valve. Crane Technical paper 410 (Reference 1) provides equations for calculating resistance coefficients for most components.

An overall headloss term, h_L, is then calculated for the entire flow path, using the classical Bernoulli's equation.

$$h_L = (Z_{start} - Z_{end}) + \frac{144(P_{start} - P_{end})}{\rho(g)} \qquad (1)$$

where,

h_L = overall headloss in ft

Z_{start} = elevation at start of flow path in ft

Z_{end} = elevation at end of flow path in ft

$P_{start} - P_{end}$ = gage pressure relative to end of flow path in psi

p= density of water in lb/ft3

g = gravitational constant in ft/s2

Using the overall headloss term, an overall resistance coefficient, $K_{start\text{->}end}$, is then defined using equation 2.

$$K_{start->end} = \frac{2g}{v^2} h_L \qquad (2)$$

where,

$K_{start\text{->}end}$ = overall resistance coefficient

v = velocity of working fluid in ft/s

h_L = overall headloss in feet

The resistance coefficients from each component, with exception of the throttled valve, are calculated and summed as:

$$K_\alpha = \sum_{i=1}^{n} K_i \qquad (3)$$

where,

K_a = total resistance of other system components

K_i = individual component resistance

The individual component resistances for pipes, elbows, fittings, orifices, valves (full open), and other similar components can be estimated from engineering references (1). The resultant difference between $K_{start\text{->}end}$ and K_a represents the valve resistance, i.e. $K_{result} = K_{start\text{->}end} - K_a$.

The resultant, K_{result}, is a conservative estimation of the resistance coefficient corresponding to the throttled valve. Equation 4 is then used to convert the resultant resistance coefficient into a C_V.

$$C_v = 29.9 \frac{d_1^2}{\sqrt{K_{result}}} \qquad (4)$$

where,

d_1 = diameter of valve in inches

K_{result} = conservative resistance coefficient of throttled valve.

<u>Determination of Position</u>

Most throttle valves have well defined flow characteristic curves. Hence, given a C_V an estimated position can then be calculated by using the C_V. A C_V vs Turns Open curve or a C_V vs Degrees Open curve is utilized for the particular valve type in question. Both curves are easily supplied by valve manufacturers. A hypothetical globe valve C_V vs Turns Open curve is shown in Figure 3. A hypothetical butterfly valve C_V vs Degrees Open curve is shown in Figure 4.

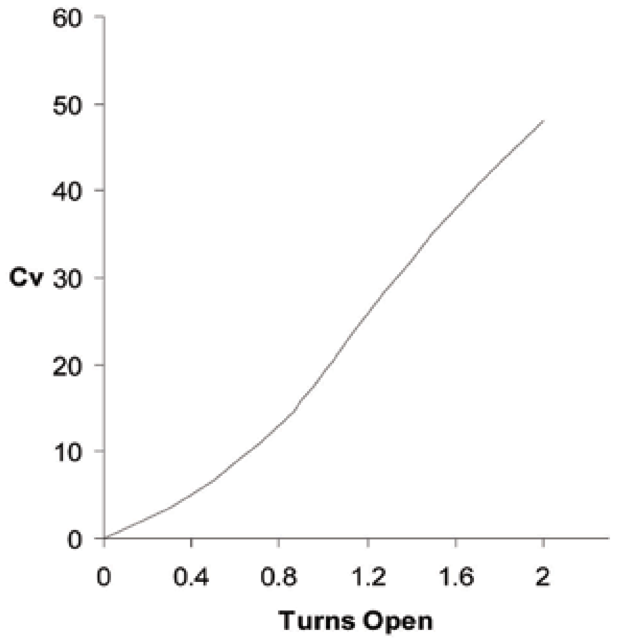

FIGURE 3 – HYPOTHETICAL GLOBE VALVE CV VERSUS TURNS OPEN CURVE

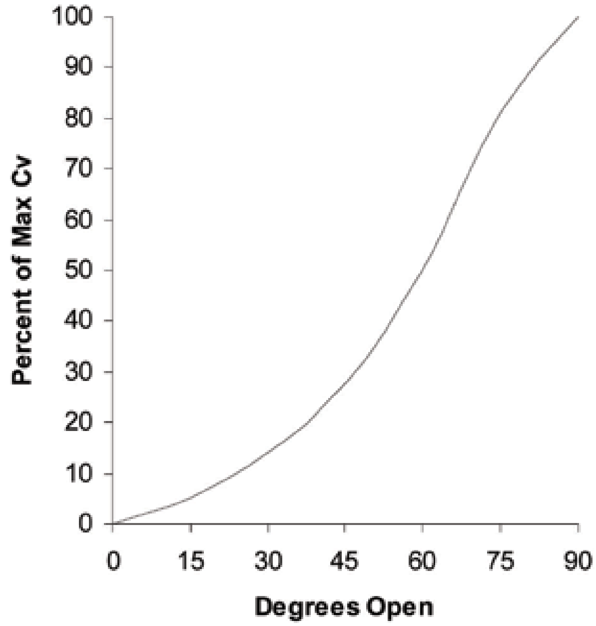

FIGURE 4 – HYPOTHETICAL BUTTERFLY VALVE CV VERSUS DEGREES OPEN CURVE

<u>Incipient Cavitation Index</u>

Given a C_V, a determination should be made as to whether the valve is operating under choked conditions or not. The incipient cavitation index is an appropriate gage as to the existence of choked flow. Manufacturers will routinely publish the incipient cavitation index at which their valves will cavitate. Equation 5 provides a means of calculating the incipient cavitation index of a throttled valve.

$$K_m = \frac{\Delta P}{(P_i - P_v)} \qquad (5)$$

where,

K_m = incipient cavitation index
ΔP = pressure differential across valve
P_i = inlet pressure of the valve
P_v = vapor pressure at fluid temperature

If the calculated incipient cavitation index is greater than the index the manufacturer furnishes for the valve position, the valve must be evaluated for choked conditions. If not, the current lift estimation is valid.

Adjusting for Choke Flow

In choked flow conditions, the choked flow area will be less than the frictional area of the valve for a given inlet condition. Valve manufacturers will typically have information regarding choked flow area and frictional area for their valves. And usually, manufacturers will list the choked flow area and the frictional area as functions of nozzle area.

A simple correlation can be made between the choked flow area, the frictional area and their corresponding lift positions. Equation 6 shows this simple correlation.

$$\frac{A_n}{A_f} = \frac{Lift_{choked}}{Lift_{Cv}} \qquad (6)$$

where,

A_n = choked flow area
A_f = frictional area
$Lift_{choked}$ = Lift position under choked flow
$Lift_{Cv}$ = position calculated without choked flow

Working Examples

Four (4) worked examples are provided to illustrate the methodology.

Problem Statement

For a plant at full power after many years of operation and repeated flow balancing of the ECCS throttle valves, it is not unusual that the record on where the throttle valves are set may not be readily available. However, the flow data exists. Figure 1 is assumed to represent the layout of the ECCS. The following four (4) worked examples are included to illustrate the application of the methodology developed in this paper, using information provided by readily available inputs. In example 1, the valve C_V is calculated based on the system conditions. In example 2, the C_V estimated in example 1 is used to determine the lift for a hypothetical globe valve. In example 3 the valve position in example 2 is evaluated for choked flow. Finally, in example 4 a butterfly valve is evaluated using the same methodology as was applied for the globe valve.

Worked Example #1

This worked example calculates the C_V of a valve using system data and flow balance data. The following inputs are used in conjunction with the flow path shown in Figure 1:

Flow velocity = 5 ft/s

Pipe friction factor = 0.023

P_{start} = 500 psi

P_{end} = 0 psi

Z_{start} = 122.00 ft

Z_{end} = 126.41 ft

p = 62.4 lb/ft^3

g = 32.2 ft/s^2

Utilizing Reference 1, the following resistance coefficients are calculated for the components of the flow path.

$K_{\text{length of line}} = 4.42$

$K_{\text{bend 1}} = 0.46$

$K_{\text{bend 2}} = 0.46$

$K_{\text{bend 3}} = 0.32$

$K_{\text{bend 4}} = 0.34$

K check valve = 1.15

K sudden expansion = 0.56

$K\,\alpha = 7.71$

Equations 1, 2, and 3 are then run to produce the following results.

$$h_L = (122.00 - 126.41) + \frac{144(500-0)}{62.4(32.2)} = 31.42 \text{ ft}$$

$$K_{start->end} = \frac{2(32.2)}{5^2} 31.42 = 80.94$$

$$K_{start->end} - K_{\alpha} = 80.94 - 7.71 = 73.23$$

$$C_v = 29.9\frac{1^2}{\sqrt{73.23}} = 3.49$$

Therefore, a value of 3.49 is determined to be the C_V of the throttled valve.

Worked Example #2

This worked example translates the C_V calculated in worked example # 1 in a lift position for a hypothetical globe valve. The following inputs are used in this working example:

$C_V = 3.49$

Valve Type = Typical Globe Valve

Full Open Turns Position = 2.0 turns

Full Open Lift Position = 0.23 inches

Using the previously determined C_V of 3.49 and Figure 2, the current turns open position of the valve is 0.301 turns open. Then, by use of equation 4, the turns open position is correlated to the lift position.

$$(\frac{0.23in}{2.0turns})(0.301turns) = 0.03in$$

Worked Example #3

This worked example adjusts the lift position of the globe valve calculated in worked example #2 for choked conditions. The inputs for this working example are as follows. These values are hypothetical values for hermetically sealed 2 inch valves.

$A_n = 0.68\ A_m$

$A_f = 0.84\ A_m$

A_m = nozzle area

$Lift_{Cv} = 0.03$ inches

Equation 6 is then carried out to determine the choked lift position.

$$\frac{A_n}{A_f} = \frac{0.68\,A_m}{0.84\,A_m} = \frac{Lift_{choked}}{0.03in}$$

$$(\frac{0.68}{0.84})(0.03) = Lift_{choked} = 0.024inches$$

After adjusting for the choked conditions, the actual lift position of the valve is determined to be 0.024 inches.

Worked Example #4

This worked example translates the C_V calculated in worked example # 1 in a lift position for a hypothetical butterfly valve. The following inputs are used in this working example:

$C_V = 3.49$

Maximum $C_V = 70$

Valve Type = Typical Butterfly Valve

Using the previously determined C_V of 3.49 and Figure 3, the current degrees open of the valve is approximately 15° Open.

ACKNOWLEDGMENTS

The authors acknowledge with thanks the constructive insights provided by Mr. Stephen Swantner and Mr. Joseph Adams, both of Westinghouse, during the development of this paper.

REFERENCES

1. Crane Technical Paper No. 410, “Flow of Fluids through Valves, Fittings, and Pipe,” 1988.

Design and construction of Two-phase coil pump

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Abstract

In this study, the design method and construction of a two – phase coil pump have been investigated. The main characteristics, advantages, various applications of such a pump and parameters influencing its performance have been determined. Experimental results for a small and a large coil pump have been obtained. A theoretical relation has been proposed which can accurately estimate the flow rate for such pumps.

Regarding to the ability of pumping gas-liquid two phase slug flow with void fraction between 0 and 1 ($0< \alpha <1$) the domain of its application is very large comparing to the centrifugal pumps.

Keywords: Helical coil Pumps, two phase flow coil pumps

NOMENCLATURE

d = diameter of the tube

D = base diameter of the drum, 2R

L = length of the drum

n = number of turns

P = pitch of the coil (L/n)

ω = angular speed

θ = inclination of the shaft

ϕ_a = air intake angle

ϕ_w = water intake angle

$\beta = \phi_w / \phi_a$ water to air angle ratio

Q= water flow rate

H= pump head

N= rotational speed

N_R = Critical speed

Introduction:

Rotating Helical coil pumps are very attractive machines for pumping gas – liquid two-phase slug flow. This is not in the case of centrifugal pumps which are limited for passing slug flow. The coil pump has very simple construction and easy to use for operation.

The concept of helical as coil type devices for generating mechanical effort dates back to the days of Archimedes (287-212 B.C.) and Leonardo da Vinci (1452-1519 A.D.). The design of a rotating coil pump was reported in the encyclopedia of arts and science in 1745 and was credited to a Swiss scientist in Zurich named Andrew Wintz.

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The idea seems to have been neglected, until 1975 when the Civil Engineering Department of the Loughborough University investigated development of a coil pump that was reported later in the Journal of the Chartered Mechanical Engineer of the U.K., ref [2].

The design and performance evaluation of a helical coil pump for operating with wind turbines for irrigation purposes and use in rural areas has been studied for more than a decade [≈1980≈1990] at the University of British Columbia and was reported in ref[1].

The same study has been investigated in Hydraulic Research Machinery Institute of University of Tehran. In this study, the focus was made to use a coil pump as a two- phase flow machine. Several prototypes have been designed and tested systematically by changing different geometrical parameters to arrive at an optimum design. The concept of a helical pump appeared to be quite attractive and promising for pumping two-phase slug flow. The main problem of this machine is low efficiency which is the subject of the next step study.

Coil pump construction & test arrangement

The main parts of a coil pump and test rig arrangement are shown schematically in

Figure(1):

1- Drum which has normally a cylindrical shape.

2- Suction and discharge reservoir.

3- Coil which is a flexible tube of desired size and turns several times around the drum. The coil inlet is in the suction reservoir and the coil outlet is connected to discharge tank by using a rotary joint.

4- Shaft and accessories.

5- Variable speed motor which provides the possibility of rotating the pump by a pulley.

Testing arrangement provides a possibility of changing systemically the parameters governing the performance of the pump including inclination of the drum Ø, tube coil diameter d, pitch of the coil P, rotational speed *w* and the ratio of water-air inlet φ_w/φ_a, Figure (2).

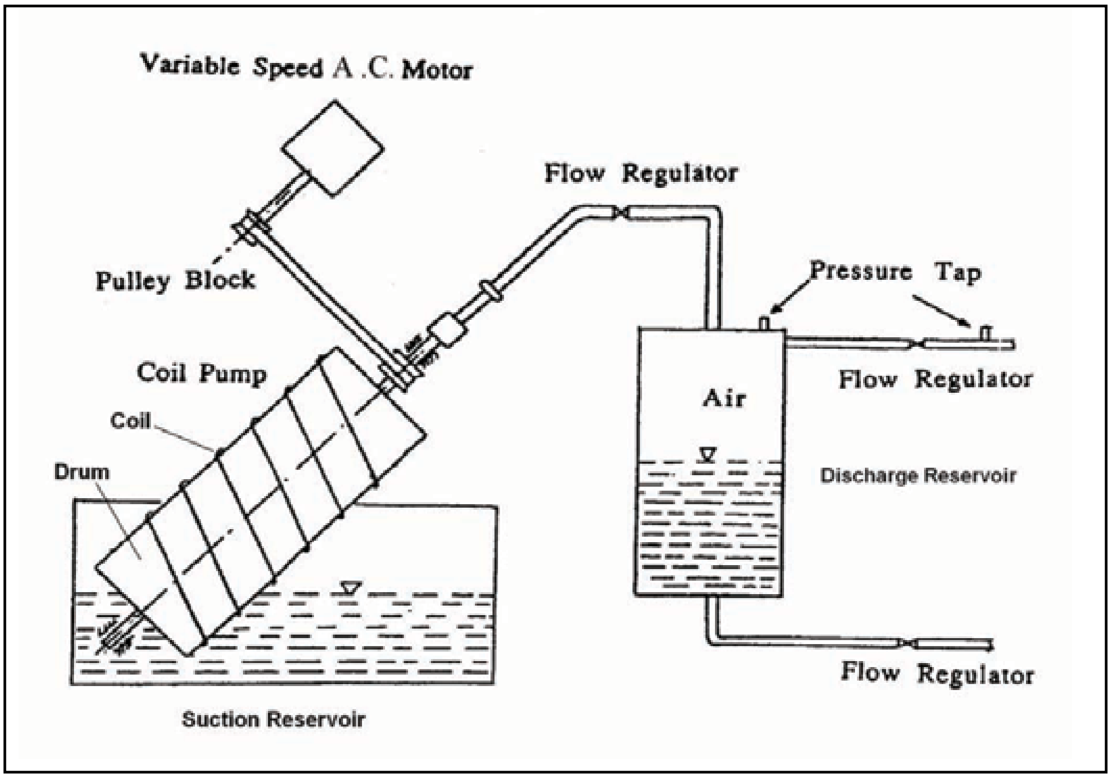


Figure 1 – coil pump and test rig arrangement

The density and temperature of liquid (water) and gas (air) are constant in these experiment tests performed by placing the pump in suction tank reservoir. The water and air intake angle φ_w and φ_a can be varied by changing the water level in the reservoir. By each turn of the drum the alternating slugs of water and air are entering in the coil inlet and transmitted though the discharge pipe. A special tank could be placed at the discharge of the pump for separating water and air.

Experimental results

The influences of many geometrical parameters on pump performance have been studied:

1- Water to air angle ratio, $ß = \varphi_w/\varphi_a$

The water level in the reservoir was observed to have a significant influence on the head and flow rate delivered by the coil pump. For low-water level, the pump delivers more air and its performance is governed by the compressibility effects of the air. But for high-water level, more water passes and the pump has relatively more stable characteristics. The head and flow rate developed by pump versus $ß = \varphi_w/\varphi_a$ are illustrated in figure (3). The water flow rate (Q) increases with a larger value of the ß (the water to air angle ratio (ß) represents the volume fraction of water to air transmitted by the pump). But the head (H) has a maximum around ß=1 to ß=1.2 depending on the different models.

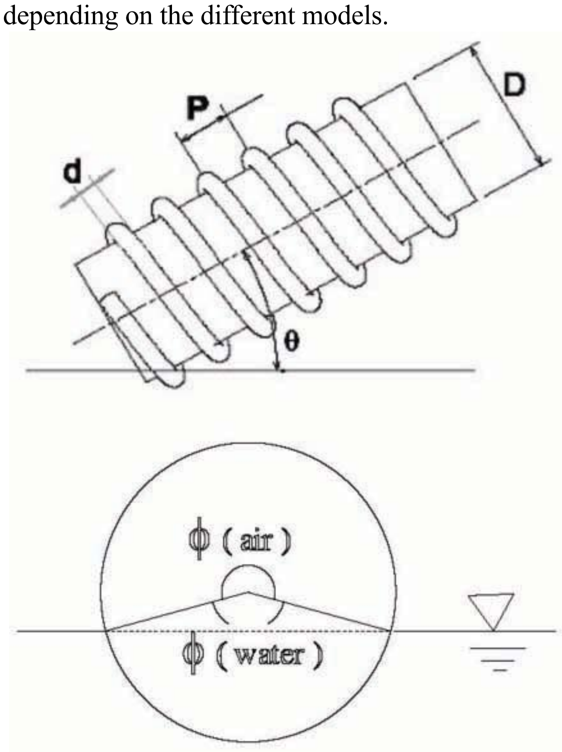


Figure 2 – Geometrical parameters of a coil pump.

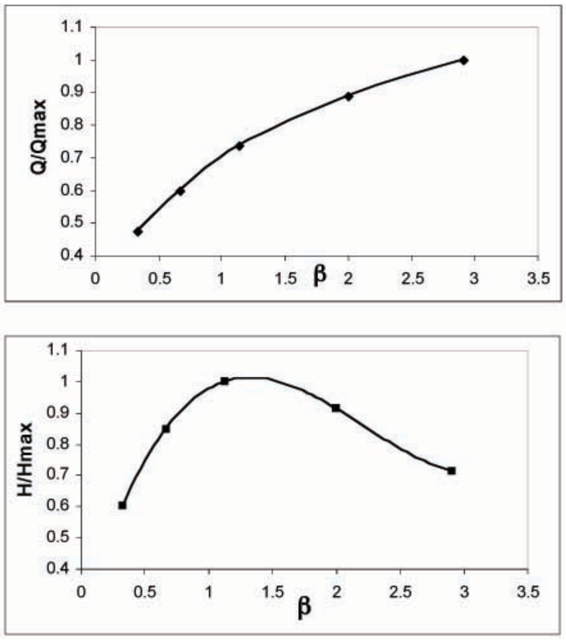


Figure 3 – Flow rate and Head versus β =ϕ /ϕ

2. Rotational speed (N)

The effect of rotational speed (N) on the water flow rate has been shown in Figure (4).

Each coil pump has a critical speed (N_R). By increasing rotational speed, the Head (H) and water flow rate (Q) increased until N_R when the function of pump came to be instable and both Q and H decreased rapidly.

Therefore, it is very important to determine the critical speed of a coil pump for different head and to operate the pump under this speed, Figure (5).

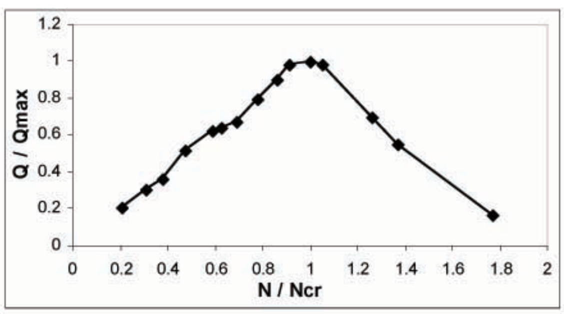


Figure 4 – variation of flow rate with rotational speed

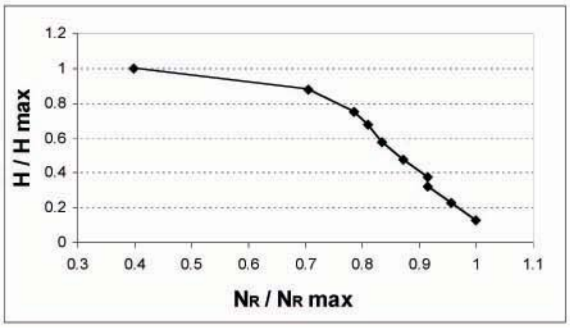


Figure 5 – limit of head delivery with critical speed

3. Coil diameter d

The flow rate increases linearly with square of coil diameter (d^2), but the head is independent of this parameter.

4. Number of turn (n)

The head increases almost linearly by increasing the number of turns of the coil where the pump flow rate (Q) is independent of this parameter, Figure (6), Figure (7).

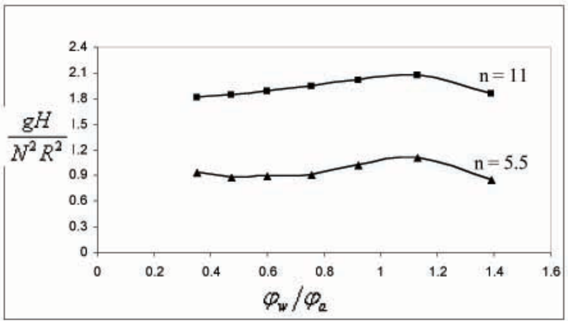


Figure 6 – variation of head with coil turn number

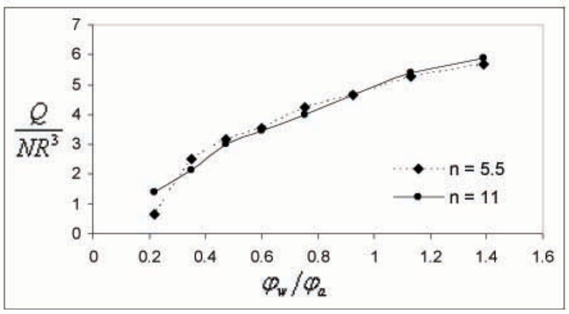


Figure 7 – variation of flow rate with coil turn number

5. Pump performance curve H= f (Q)

The performance curve of coil pump H= f (Q) is almost a vertical line parallel to H axes. See Figure (8)

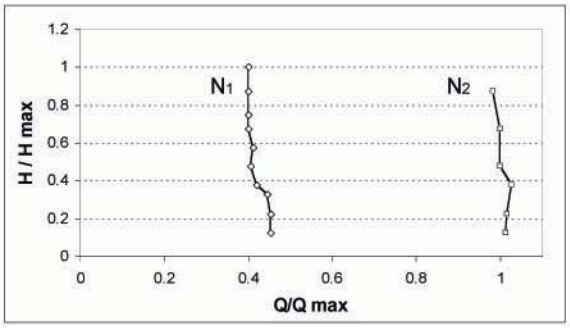


Figure 8 – pump performance curve

Theoretical investigation

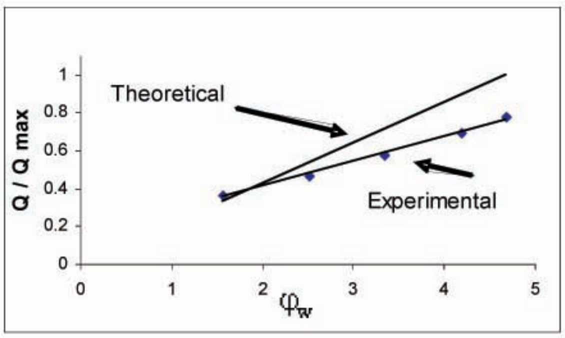
Flow through a rotating helical pipe or simply helical pipe flow have been already studied [3, 4 and 5]. An analytical approach of a two-phase flow rotating coil pump is under investigation. However, the water flow rate of a coil pump could be calculated by simple equations:

$$Q_{Th} = 2\pi \frac{\beta}{\beta+1} \cdot \frac{D}{2} \cdot \frac{\pi d^2}{4} N \qquad (1)$$

$$Q_{Th} = \varnothing_w \frac{D}{2} \cdot \frac{\pi d^2}{n} \cdot N \qquad (2)$$

The values obtained from experimental results and calculated by the equation (1) are compared in Figure (9).

Figure 9 – comparison of flow rate obtained by



theoretical method and experimental results.

The value of flow rate calculated by the theoretical method (Q_{Th}) is always larger than obtained by experiment

(Q_{EX}). However, the ratio of $K = \frac{Q_{EX}}{Q_{Th}}$ is almost constant for different values of $\frac{\varphi_w}{\varphi_a}$ for a coil pump. But this value could be changed for different coil pump depending on the size and other geometrical parameters. For the pump tested, K stays between 0.5 and 0.8.

Therefore the liquid flow rate:

$$Q = 2\pi K \frac{\beta}{\beta+1} . \frac{D}{2} . \frac{\pi d^2}{4} N \qquad (3)$$

$$Q_{Th} = \varnothing_w .K \frac{D}{2} . \frac{\pi d^2}{n} .N \qquad (4)$$

Conclusion

From the above experimental study, one can conclude that head and water flow rate developed by a coil pump are a function of the following parameters:

H = f (ß, N, n)

Q = f (ß, N, d, D)

It is very important to notice that each pump has a critical speed N_R whose function became instable. The performance curve H = f (Q) is almost a vertical line parallel to H axes.

The flow rate of a coil pump can be calculated from equations (3) and (4).

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Accumulator Flow Rate Using Accumulator Gas Pressure for Non-Intrusive Discharge Check Valve Full Open Exercise Stroke Testing

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Abstract

A new method to obtain a lower bound estimate of flow rate from the South Texas Project Electric Generating Station Safety Injection Accumulators (tanks partially filled with subcooled water and pressurized with nitrogen gas) is proposed as an alternative way to test the Accumulators' discharge check valves' full open stroke. With the new method, Containment building entry can be avoided by use of permanently installed plant process level and pressure instrumentation (recorded remotely on the plant computer, the Integrated Computer System). Avoiding Containment activities reduces total dose for the tests by about 670 millirem (mrem) and several person-hours of labor compared to other methods that require special test instrumentation installation and work in radiation areas. The effect of instrumentation uncertainty is included in the flow estimate. The lower bound flow estimate is generally useful where a minimum flow rate must be demonstrated and a sufficient margin to actual flow rate is available.

Introduction

We derived a simple method for measuring outflow from the Safety Injection (SI) Accumulators using the nitrogen gas pressure recorded on the plant computer for use in full open exercise testing of accumulator discharge check valves. Meeting the design flow rate is an acceptable non-intrusive test method for verifying that the check valve disk opens to the position necessary to perform its safety function. The outflow calculated by the simple method is based on isentropic expansion of the nitrogen in the accumulator so we can be sure the outflow we calculate will be lower than actual. Because the method produces the lowest possible flow measurement, it is appropriate to use it to compare against the minimum acceptance criteria (that is, the measured flow must exceed the minimum required value for an acceptable test).

The method is most useful when the installed plant level measurement isn't available over a sufficient range to measure the outflow rate. However, the method can also be used in conjunction with other methods as a confirmation of check valve test results from them.

Background

The South Texas Project (STP) is a two unit, pressurized water electric generating station. Each of the units has three individual SI accumulators installed to inject water into the reactor following a postulated pipe failure large enough to temporarily exhaust the normal water inventory in the reactor. The water is chemically treated with neutron-absorbing boric acid to help prevent return to criticality following a postulated accident. The SI accumulators are filled to a specified level with water such that there is enough air space above the water to allow them to be pressurized with nitrogen. The pressure in the accumulator is kept lower than the pressure in the reactor under normal operating conditions.

In the unlikely event of a postulated large pipe failure, the pressurized nitrogen in the accumulators would force the chemically treated water into the reactor as pressure drops below the SI Accumulators' pressure. By using trapped pressurized nitrogen, the SI accumulators do not require any external motive force to restore the reactor water inventory. During normal operation, higher pressure reactor water is kept out of the accumulators by check valves. If the reactor pressure drops below the accumulator pressure, water gets forced past the check valves due to the pressure difference. A motor operated valve is installed in the accumulator outlet to keep the water in the accumulators against nitrogen gas pressure during refueling or other plant operations that cause the outlet pressure (reactor pressure) to be lower than the accumulator nitrogen pressure. The check valves require testing under the American Society of Mechanical Engineers (ASME) Code for Operation and Maintenance of Nuclear Power Plants (OM Code) during refueling outages.

STP has been testing the SI Accumulator check valves using locally installed, temporary acoustic monitoring instrumentation. To install the equipment, scaffolding is required and several personnel must be in radiation areas for significant periods of time during the testing. We have found that, as the plants age, the dose rates in the areas the test instruments are installed result in significant worker exposures (a little under 700 mrem to test the three accumulators). Significant labor and test equipment expense is also a consideration. The testing method we describe in this article effectively eliminates worker radiation exposure for SI Accumulator discharge check valve stroke testing and significantly reduces time spent in testing activities by taking advantage of existing process instrumentation measuring SI Accumulator pressure and water volume.

Approach

Since the SI Accumulators contain a trapped volume of nitrogen, any change in the contained water volume will result in a pressure change which can be read on the gas pressure measurement instruments. We are interested in cases where the water is exhausting from the tank. As long as sufficient water remains in the tank, the nitrogen will expand in reaction to a piston-like action.

Small amounts of heat addition will occur during an expansion process due to gas cooling below the wall and liquid surface temperature during expansion. The different thermodynamic processes governing gas expansion in normal pressure and temperature ranges (such as in the accumulator) are well known and have been established for many years. The extreme thermodynamic processes for this piston-like action are isothermal (where the heat addition brings the temperature back to the pre-expansion value) and isentropic (reversible and adiabatic). Clearly, as long as highly subcooled liquid remains in the tank, any increase in gas volume corresponds to an equal reduction in liquid volume. If the time duration of the expansion is known, the volumetric outflow rate is the same as the volumetric gas expansion rate. Finally if the expansion is observed over small time increments, the volumetric flow can be calculated. Note that since the check valve test starts with no flow, builds to a maximum, and decays off, averaging the flow over discrete time intervals produces lower than actual flow measurement.

Referring to Figure 1, if the pressure in the tank is measured at regular intervals, Ti, and the gas volume is V, then when the outlet valve is opened, the gas will expand and water will exhaust (assuming the nitrogen pressure in the tank is above the pressure on the liquid free surface at the outlet) as illustrated in Figure 2. The pressure history from a typical test will look similar to Figure 3 while the discharge valve is opened and closed.

Although in fact the nitrogen expansion process is polytropic, we assume isentropic for the purpose of flow measurement. By making this assumption, when we use the pressure history to find the rate of gas expansion we will underestimate the gas expansion (since the process actually will be closer to an irreversible, isothermal process). Clearly, heat transfer from the walls and the water surface will contribute to isothermal expansion. Since no external work is present and no other heat sinks are present, the isentropic process is limiting. The general form of the isentropic expansion process during a time increment from T_0 to T_1 (taking the nitrogen volume to be V_1 at T_1) is defined by the equation:

$$\frac{p_0}{p_1} = \left(\frac{V_1}{V_0}\right)^{\gamma} \qquad \text{(Eq. 1)}$$

The value of g for nitrogen can be taken to be 1.4. Also note that the gas pressure must be in absolute pressure (not gauge pressure). Typically, the plant process computer will record the pressure as gauge pressure. We use measurement of level, pressure, and time difference for flow rate. Both the pressure measurement and level measurement have a degree of uncertainty. The level measurement is used only for the initial level in order to obtain the initial nitrogen volume, V0, and so we don't need to worry about additional uncertainty in the level measurement beyond what exists at the start of the flow measurement. At STP, the level in the accumulators is measured as the total contained water volume, but we are interested in the air volume. Generally, the total contained volume of the accumulator is known from the design documents, so it is easy to find the air volume by simply subtracting the water level (volume) from the total. If there are multiple measurements of level, they can be combined to obtain a more accurate estimate of initial level. However, (in general) one must be careful to properly characterize and combine the uncertainties in each of the measurements to obtain the best estimate of the combination.

For the purposes of the flow measurement, the most conservative direction of the level error is in the direction of lower initial nitrogen volume (inspect Equation 1 with the thought in mind that smaller V_0 calculated from the pressure measurements will produce smaller volumetric increase in the nitrogen space).

Similar to the level measurement, the pressure measurements are subject to uncertainty. In the case of pressure measurement, the uncertainty during the test is important. That is, random fluctuations in pressure could cause the flow rate measurement (increase in air volume) to be larger than without the uncertainty. This could lead to falsely concluding the flow rate was sufficiently high to pass the test (simply due to random noise in the measurement).

As in the level measurement, redundant pressure measurements, where available and properly characterized and combined, can produce much more accurate estimates than a single measurement. When relatively large, random fluctuations are in the pressure measurement, it is possible that using the fractional change in pressure (the pressure ratios at adjacent measurement times) will produce more stable flow rates.

Referring to Equation 1, enumerating successive plant computer measurements of pressure with the subscript i and enumerating successive measurement intervals (i to i+1) with the subscript j then the change in volume for measurement interval j = 0 would be:

$$\left(\frac{p_0}{p_1}\right)^{1/\gamma} = \frac{V_1}{\hat{V}_0} \qquad \text{(Eq. 2a)}$$

taking the error in initial level measurement as ε and $\hat{V}_0 = V_0 - \varepsilon$. Or, keeping in mind we want to use fractional values at each measurement interval, add and subtract 1.0 on the RHS [right hand side] of Equation 2a:

$$\left(\frac{p_0}{p_1}\right)^{1/\gamma} = \frac{V_1 - \hat{V}_0 + \hat{V}_0}{\hat{V}_0} \qquad \text{(Eq. 2b)}$$

let $\Delta V_j = V_{i+1} - V_i$ and rewrite Equation 2b:

$$\Delta V_0 = \hat{V}_0\left[\left(\frac{p_0}{p_1}\right)^{1/\gamma} - 1\right] \qquad \text{(Eq. 2c)}$$

The flow at this interval is then simply the volume change divided by the measurement time interval, Δt :

$$Q_0 = \frac{\Delta V_0}{\Delta t_0} \qquad \text{(Eq. 2d)}$$

After incrementing *i* and *j*, the volume at the beginning of the next time step (V_i with $\hat{V}_o$ as the first) is found by deducting the change in air volume during the previous time increment from the air volume at the start of the previous tme increment:

$$V_i = V_{i-1} - \Delta V_{j-1}$$

Flow rates at successive time intervals are then found from deducting the last change in air volume from the starting air volume and solving for the next change in volume using the pressure ratio for the interval:

$$Q_j = V_i\left[\left(\frac{p_i}{p_{i+1}}\right)^{1/\gamma} - 1\right] \qquad \text{(Eq. 2e)}$$

The error due to random noise in the pressure measurement can be quantified using the pressure instrument outputs at steady state conditions using Equations 2 to obtain, for example, the variance in the output and then applying a confidence interval to obtain a good bound. This error can be simply added to the minimum flow requirement (that is, raise the minimum acceptance criteria by the amount of the random error).

Equations 2 were applied to actual test measurements in STP Units 1 and 2 for the last set of tests performed (three accumulators in each unit having 2 check valves each). In each of the tests, the results using the current method (acoustic measurement) the check valves stroked open satisfactorily. Figure 4 compares the solutions to Equations 2 against the flow limit for the STP SI Accumulator check valves (about 200 Liters per second [L/s]). As can be seen, the limiting flow was met and exceeded for each of the SI Accumulators' tests.

Cost savings

Currently, the approximate total baseline cost for the complete SI Accumulator check valve testing is approximately $38,100 each performance (once every nine months based on an 18 month outage cycle on two units) with radiation exposure included as a dollar cost. The cost breakdown is shown in Table 1. The costs listed in the table are not exact in general, but instead are based on actual costs for a recent check valve test performance in which all three SI Accumulators' check valves were tested in one unit.

The major costs associated with implementing the simplified method come down to development costs associated with creating the plant process computer application and engineering time to develop and verify the method. Table 2 gives a rough estimate of the development costs, inflation, and the rate of return on capital employed as inputs to the analysis. While the inputs to this type of analysis are subject to relatively large uncertainties, in the present case, the extremely short payback (shown below) and low cost of development compared to the ongoing costs don't justify a detailed sensitivity study on the inputs.

We assume ongoing costs associated with the pressure measurement method are negligible compared to the current method, based on automation of the process allowing the Operator to set up and perform the test using the plant process computer Control Room display of installed instrumentation measurements and then simply print out a test report upon completion. Also, plant process computer processing burden (CPU, data storage, and memory requirements) costs are negligible due to the small computational load and the addition of a small number of data points.

The cost of the current method using discounted cash flow is evaluated for a 5 year and 10 year project assuming 3% inflation, 8% return on capital employed, and 18 month cycle duration. The results of this analysis are shown in Table 3 showing that the project pays back in the first year (first outage) and is worth slightly less than $240,000 in five years and about $400,00 in ten years of useful life.

Conclusions

Development of a simple pressure based accumulator exit flow estimation method has been presented. The theoretical basis of the method has been described. Using the isentropic flow assumption and time averaging produces a theoretical minimum flow rate estimate for the conditions of the test.

The cost savings (including radiation dose as a dollar cost) for using the simplified method over the current method at STP has been shown using a discounted cash flow calculation for up to 10 years of useful life. The simplified method is estimated to pay back in the first year of use and is worth slightly less than $240,000 over 5 years of useful life.

Table 1. Approximate costs incurred using the current method per performance each outage (three SI Accumulators), including radiation.		
Item	Effort	Cost
Measurement equipment setup and testing		
Instrument	4 shifts X 2 techs/shift X 12 hours @ $40/hour	$3,840
Engineering coordinator	4 shifts X 1 Section XI coordinator/shift X 12 hours @ $40.00/hour	$1,920
Operations coordinator	4 shifts X 1 Ops coordinator/shift X 12 hours @ $40.00/hour	$1,920
Test measurements	10 participants X 4 hours @ $40.00/hour	$1,600
Other equipment setup		
Scaffolding (primarily labor)	32 hours per 4'X8' footprint @ $35/hour; 2 footprints X 3 trains (SI-38/ 46A,B,C)	$6,720
Test equipment maintenance	Miscellaneous installation costs due to scheduling and parts (mounts, studs, etc.)	$2.000
Radiation exposure	30,000 $/rem X .670 rem	$20,100
	Total	***$38,100***
Cost per year, with 18 month schedule in each unit (multiply Total by 1.33)		***$50,800***

Table 2. Approximate development costs for the simplified testing method (incurred in the first year of use)		
Procedure revisions	1 person-week	$4,000
Method development and qualification	1 person-week	$4,000
Process computer application development	2 person-weeks	$8,000
Software Quality Assurance	2 person-weeks	$8.000
Training (development and training time)	2 person-weeks	$8,000
	Total	***$32,000***

Table 3. Discounted cash flow calculations of the net present value using the current method of testing.			
Year after implementation	Cash flow escalation due to inflation (at 3%)	Discounted cash flow (8% discount rate)	Net present value of the discounted cash flow for the corresponding year(s) after implementation
0	$18,800 ($50,800-$32,000)	$18,800	$18,800
1	$52,324	$48,448	$67,248
2	$53,894	$46,205	$113,453
3	$55,511	$44,066	$157,519
4	$57,176	$42,026	$199,545
5	$58,891	$40,080	$239,626
6	$60,658	$38,225	$277,850
7	$62,478	$36,455	$314,305
8	$64,352	$34,767	$349,073
9	$66,282	$33,158	$382,231
10	$68,271	$31,623	$413,853

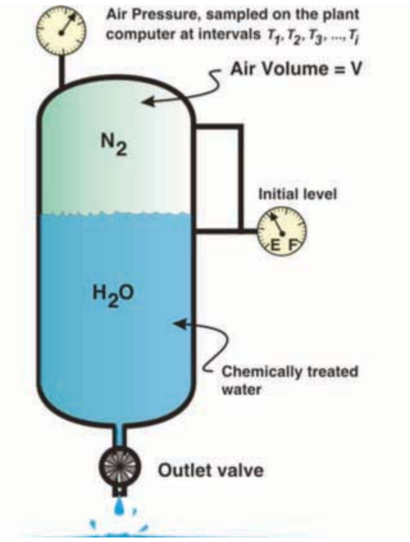

Figure 1 – Accumulator prior to opening the outlet valve.

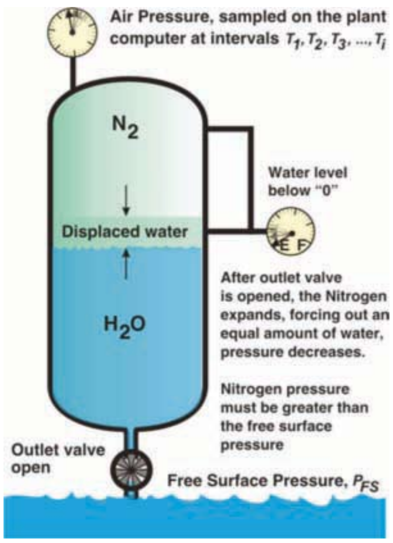

Figure 2 – Accumulator schematic showing displacement of the water during a test.

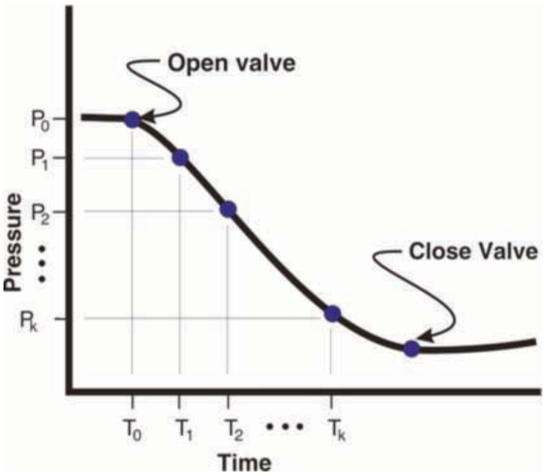

Figure 3 – Typical pressure history of a check valve test showing sample intervals, T1, T2, T3, ..., Tk.

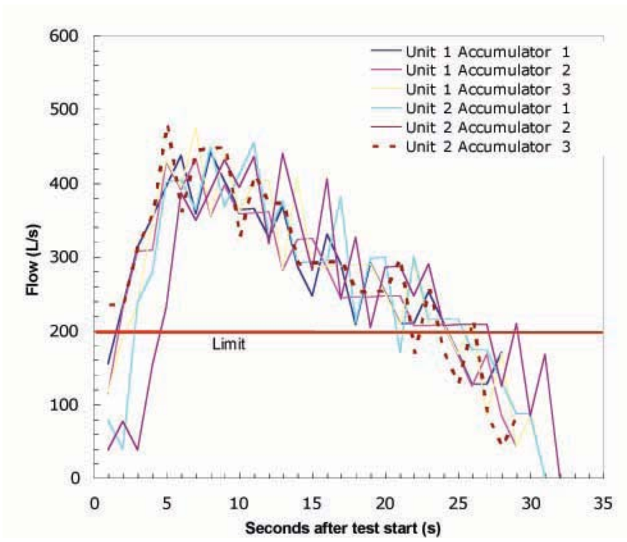

Figure 4 – Flow rates measured in last set of tests in STP Units 1 and 2 in all three SI Accumulatorscompared with the required limit. [3200 gallons per minute (gpm) plus 200 gpm uncertainty = 214.5 L/s]

Session 3(b): Valves II

Session Chair

Dr. Claude L. Thibault

Consultant

Characteristics of New Valve Seats, Including Surface Roughness After Long Time Exposure to Reactor Water Condition

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Abstract

Valve seat aging after long time exposure to reactor water condition has been discussed recently [1]. As the countermeasure for the valve seat aging problem, we have developed the valve "HHV*" with a new valve seat. The evaluation result report of HHV shows that the new valve seat is superior to the conventional valve seat, in terms of corrosion-resistance, coefficient of friction, mechanical-sturdiness, low residual stress, erosion-resistance, low cobalt (Co) release and nondestructive inspection possibility. [2]

To confirm the characteristics of the new valve, an appraisal test at operating plant condition was carried out. We reported that the new valve seat was superior to the conventional valve seat.[2]

Now, we are reporting the new valve seat's characteristics, including the surface roughness and the coefficient of friction due to aging, under the circumstance of the BWR (boiling water reactor) reactor water condition. For the evaluation of the seat surface roughness and the coefficient of friction due to aging, the valve seat specimens (Stellite 6 and new valve seat) were aged in a corrosion autoclave. The result shows that the surface roughness change of the new valve seat due to aging is smaller and the coefficient of friction of the new valve seat due to aging is lower than that of the conventional valve seat. The evaluation result report of the new valve seat shows that the new valve seat is superior to the conventional valve seat in terms of all characteristics including secular change. As described above, it can be said that the use of the valves incorporating the new valve seat in actual nuclear power plants will not only increase the reliability and maintainability of the valves, but also contribute to the increased reliability and maintainability of the plants, in comparison with the use of the valves incorporating the conventional valve seats hardfaced with a Co-based alloy.

*: HHV is the valve with new cobalt base alloy valve seat, whose metal microstructure is different from that of the conventional cobalt base welded overlays.

Introduction

The valves used in nuclear power plants have the seat portions of their valve bodies and the valve casings (hereinafter referred to as valve seats) hardfaced with a Co-based alloy (mainly RCoCr-A, AWS standard) in order to minimize the degradation in the sealing capability and the operational performance of the valves. However, the hardfaced portions are degraded and/or damaged due to surface roughness, cracking, and/or erosion, requiring the repair and/or replacement of the valve seats. In addition, the set pressure drift of safety relief valves, which is called "corrosion bonding," are caused by the adhesion force generated by the corrosion products. Therefore, in terms of the reliability and maintainability of the valves, it is required to minimize the occurrence of the corrosion, cracking and/or erosion of the valve seats. Furthermore, in nuclear power plants, it has been required for a long time to minimize Co release from the valve seats in order to reduce the radiation exposure of the nuclear power plants workers.

In addition to the problems described above, recently it has been pointed out that the friction force on the valve seat made by a Co-based alloy welded overlay increases under the environment of nuclear power plant coolants [1]. The improvement of the valve seat material made by Co-based alloy has been required also for maintaining the operation performance of the valves.

On the other hand, focusing attention on the degradation mechanism of the Co-based hardfacing material(s) used on the valve seats in nuclear power plants, we have demonstrated that the degradation mechanism can be minimized by changing the metallic structure without altering the chemical composition of the metal, and that the valves can be manufactured having the valve seat material with its various Co-based alloy characteristics improved with this method.[2)]

However, the report described above which evaluated the friction force on the valve seats only made evaluations on the friction characteristics of the valve seats associated with frequent operation. The report did not refer to the evaluation of the friction characteristics of the valve seats associated with aged deterioration under the environment of nuclear power plant coolants. Therefore, we re-evaluated the new valve seat hardfaced with a Co-based alloy containing dispersed eutectic carbide (hereinafter referred to as new valve seat, or developed valve seat) in all aspects. We thought that the new valve seat material developed exhibited less surface roughness under a corrosive environment, compared with conventional valve seat welded overlay with a Co-based alloy containing mesh-like eutectic carbide (hereinafter referred to as conventional valve seat), and added the comparisons of the changes in the surface conditions and the friction force between the conventional valve seat and the new valve seat under the corrosive environment. Shown below are the comparisons of the characteristics including aged deterioration and friction force changes between the conventional valve seat and the new valve seat.

2. Valves having new valve seat

2.1 Basic concept of new valve seat material

For valve seats made by corrosion- and wear-resistant, carbide-dispersed alloy, the metallic structure of the conventional valve seat material is composed of mesh-like eutectic carbide and dendrite. The damaging mechanism is as follows: "First, the mesh-like eutectic carbide is corroded and dropped off due to the dissolved oxygen in fluid. Then, the dendrite is damaged and dropped off due to the mechanical force of the flow. The chemical corrosion and mechanical erosion are repeated to expand the damage to the valve seats, which may produce cracks, depending on the condition of the residual stress." In addition, Co release from the valve seats, which is said to be a cause for the radiation exposure of workers in nuclear power plants, is thought to occur in the process of this degradation and damaging mechanism.

The observation of the corrosion, cracks, erosion, and Co release that occurred on the Co-based alloy welded overlay material of the valve seats of the valves in actual operating plants confirmed that each of these phenomena was produced by the damaging mechanism above described (see Fig. 1). As a countermeasure against this degradation and damaging mechanism, we thought that a new valve seat material in which the eutectic carbide particles are dispersed in the metal matrix can control the repetition of the chemical corrosion and the mechanical erosion; thereby, minimizing the possible damage to the valve seats as well as Co release. Shown in Fig. 2 are the comparisons, as described above, of the degradation and damaging models between the conventional valve seats and the new valve.

Valve seats were manufactured according to the development concept for the new valve seats above described. Shown in Fig. 3 is the comparison of the metallic structure of the conventional valve seats with that of the new valves. Both types of the valve seats have the chemical composition falling within the range of RCoCr-A (AWS standard).

2.2 Characteristic evaluation

2.2.1 Required characteristics

The following seven characteristics are required for seats of valves used in nuclear power plants:

① Corrosion resistance: The surface roughness caused by aged deterioration should be less. Such roughness may cause leakage or poor operation.

② Sliding property and antigalling property:

(a) The surface roughness associated with frequent operation should not cause sliding resistance large enough to lead to galling.

(b) The surface roughness associated with aged deterioration (secular change) should be small and not cause sliding resistance large enough to lead to galling.

③ Mechanical Sturdiness: The mechanical sturdiness is necessary to prevent the occurrence of valve seat cracks and thus should be maximized.

④ Lower residual stress: Higher tensile residual stress is one of the factors causing valve seat cracks. Residual stress should be minimized as much as possible and, where applicable, compressive residual stress is preferable alternative.

⑤ Erosion resistance: The resistance to erosion should be high under the service conditions. Especially in nuclear power plants, higher resistance to erosion is required in the circumstance of higher dissolved oxygen at higher temperatures.

⑥ Low Co release property: In nuclear power plants, it is required to reduce the amount of Co release from the Co-based alloy in order to lower the radiation exposure of nuclear power plant workers.

⑦ Ease of work and inspection: The alloy should facilitate the detection of the surface and internal defects by non-destructive inspection as well as the fitting work.

2.2.2 Performance evaluation

We conducted performance evaluation tests on the valves incorporating the new valve seats that were designed to meet the seven requirements described above. Summarized in Table 1 are the results of the tests. In comparison with the valves incorporating conventional valve seats, the valves incorporating the new valve seats exhibit the following characteristics: ① Superior corrosion resistance; ② lower coefficient of friction and less aged deterioration (secular change); ③ higher impact strength and toughness; ④ having residual compressive stress instead of residual tensile stress; ⑤ Much higher corrosion resistance; ⑥ approximately one tenth or less of Co release under the condition which needs to minimize Co release; and ⑦ detection of minor internal defects enabled by ultrasonic test (UT).

Detailed below are the results of the evaluation of the characteristics outlined above. (see Table 1)

(1) Corrosion resistance

Strauss test(s) (JIS G0575) were conducted in order to compare the conventional valve seats with new valve seats. Shown in Fig. 4 is the comparison of the results of the Strauss tests run on conventional valve seats and new valve seats. Specifically, the figure shows the cross sections of the specimens cut after undergoing the Strauss tests. As is evident from the figure, the conventional valve seats were selectively corroded and damaged down to quite a deep level; on the other hand, the new valve seats were hardly damaged. Hence, the new valve seats provide better corrosion resistance than the conventional valve seats.

(2) Sliding property and antigalling property

i) Changes in coefficient of friction caused by frequent operation

In order to evaluate the sliding property and antigalling property of the new valves, the sliding property and antigalling property of new valve seats were compared with those of conventional valve seats by reciprocally sliding a movable piece simulating a valve body against a fixed piece simulating the valve seat on the valve casing side in ordinary-temperature water.

Shown in Fig. 5 is the result of the sliding test performed in ordinary-temperature water. The new valve seats exhibit a smaller coefficient of friction than the conventional valve seats by approximately 30%. As evident from this result, the valves incorporating the new valve seats require less driving force and provide better operational reliability of valves than the valves incorporating the conventional valve seats hardfaced with a Co-based alloy.between conventional valve seats and new valve seats

ii) Changes in valve seat surface conditions caused by aged deterioration (secular change)

In order to compare aged deterioration (secular change) of the conventional valve seats with that of new valve seats under an actual operating environment, a corrosion resistance test was conducted in high-temperature water containing a high percentage of dissolved oxygen (288°C; DO:8 ppm [parts per million]) to investigate the changes in the valve seat conditions. Before the corrosion resistance test, the entire valve seat faces were buffed, and the valve seat surfaces of both conventional valve seats and new valve seats had an average surface roughness (Ra) of 0.03 μm [micrometer].

Shown in Fig. 6 are the measurements of the surface roughness of the valve seat surfaces after a specified test time elapsed. The conventional valve seats exhibited greater surface roughness as the test time went by: The surface roughness Ra of the valve seat faces was 0.11 μm after 2078 hours elapsed. On the other hand, the surface roughness (Ra) of the valve seat faces of the new valves slightly increased to 0.05 µm until 500 hours elapsed from the start of the test, and then remained constant until 2078 hours elapsed. As described in paragraph (1) on Corrosion resistance, the mesh-like eutectic carbide precipitated on the conventional valve seats is distributed continuously in the dendrite gaps. Therefore, corrosion will advance continuously along the carbide into the inside of the valve seats if the carbide in contact with the surfaces of the valve seat faces is selectively corroded and damaged. On the other hand, on the new valves carbide is dispersed in granular conditions. Therefore, corrosion will not advance continuously even if the carbide in contact with the surfaces of the valve seat faces is selectively corroded and damaged. For the reasons described above, it is thought that the surface roughness of the new valve seats did not increase after a lapse of 500 hours in the corrosion resistance test.

From these results, the new valve seats are determined to exhibit less surface roughness across ages and expected to show smaller sliding resistance associated with aged deterioration.

After evaluation of the surface condition of valve seats, abrasion resistance tests for evaluation of coefficient of friction were conducted under room temperature water condition. Shown in Figure 7(1) is the example of the coefficient of friction plots of the valve seat after 1000 hours elapsed in high-temperature water containing high percentage of dissolved oxygen (288°C; DO: 8ppm). The results show that the coefficient of friction of new valve seat exhibited lower than that of conventional valve seat. The results of the abrasion resistance tests of 250 hours and 500 hours show the tendency which is similar to that of 1000 hours. Shown in Figure 7(2) are plots of CF* vs. Initial coefficient of friction of conventional valve seat and new valve seat, aged after 250 hours, 500 hours, and 1000 hours in high-temperature water containing a high percentage of dissolved oxygen (288°C; DO: 8ppm). The results of the test (the plots of CF* vs. Initial coefficient of friction of conventional valve seats) show more widespread distribution, compared to that of new valve seats.

*: CF means the difference of highest coefficient of friction and the initial coefficient of friction at individual abrasion resistance test.

The coefficient of friction threshold values, which are estimated from the CF vs. Initial coefficient of friction plots, are shown below.

Conventional valve seat: 0.48

New valve seat: 0.42

From these results, the new valve seats are determined to exhibit lower and more stable coefficient of friction associated with aged deterioration, compared to the conventional valve seat.

(1) The coefficient of friction plot of the valve seats,

The result of abrasion resistance test(s)

(2) △CF* vs. Initial coefficient of friction plots

*△CF : The difference of largest coefficient of friction and the initial coefficient of friction at abrasion resistance test.

(3) Mechanical Sturdiness

Charpy impact test(s) were conducted at ordinary temperatures in order to compare the mechanical sturdiness of the conventional valve seats with that of the new valve seats. As the specimens for the impact tests, flat specimens and U-notched specimens were used. The test results are shown in Table 2.

As indicated in Table 2, the new valve seats were found to have higher Charpy impact values by two to five times and are tougher than the conventional valve seats. Detailed observation of the occurrence of cracking on conventional valve seats welded overlay on actual valve revealed that cracking mainly started at mesh-like eutectic carbide. From this phenomenon, cracking is thought to occur on the conventional valve seats under a corrosive environment in the following mechanism: The valve seats are composed of mesh-like eutectic carbide and dendrite. The former has less corrosion resistance than the latter. Therefore, the eutectic carbide is first selectively corroded and damaged. Then, the area corroded and damaged is acted upon by the residual stress of the valve seats, becoming a starting point of cracking.

The new valve seat material is at least twice tougher and more corrosion resistant as shown in (1) than the conventional valve seats. Consequently, it can be said that cracking is less likely to occur on the new valve seats than on the conventional valve seats.

(4) Residual stress

Residual stress on valve seats was evaluated by manufacturing a carbon steel disc to simulate the gate valve having a 200A bore, and then hardfacing the disc with the Co-based alloy welded overlay used as conventional valve seat material, and with carbide-dispersed alloy, which is used as new valve seat material. The result was as follows: The conventional valve seats were subjected to high residual tensile stress in the lap-direction; the new valve material was subjected to residual compression stress in both lap- and diameter- directions.

Combined with the results of Charpy impact test described above, it can be said that the new valve seats have significantly less potential to induce cracking and provide higher reliability as valve seats than the conventional valve seats.

(5) Erosion resistance

Erosion resistance evaluation test(s) were conducted in a high-temperature atmosphere (containing approximately 8 ppm of dissolved oxygen) in order to perform comparative evaluation of erosion resistance between the conventional valve seats and the new valve seats. Shown in Fig. 9 is the comparison between the erosion occurrence conditions on the conventional valve seats and those on the new valve seats after the tests that lasted for 48 hours. In comparison with the conventional valve seats, the erosion-damaged conditions of the new valve seats were extremely minor, and the damaged volume of the new valve seats was less than one tenth of that of the conventional ones. As for the jet impact areas and the erosion occurrence conditions, the conventional valve seats were erosion damaged on the jet impact areas and their vicinities. Especially the vicinities of the jet impact areas were severely erosion damaged. On the other hand, new valve seats were subjected to minor damage only on the jet impact areas. It is thought that the superior corrosion resistance of the new valve seats greatly contributes to their better erosion resistance.

(6) Co release property

Co release tests were conducted under the water quality conditions listed below in order to compare the Co release between the conventional valve seats and the new valve seats.

Test conditions

- Temperature: 220 °C
- Dissolved oxygen (DO): 200 ppb [parts per billion]
- Test time: 2000 hours

The test result is shown in Fig. 9.

The result shows the following: The new valve seats exhibited one tenth of the amount of Co release from the conventional valve seats, and thus can significantly reduce the amount of Co release.

The radiation to which nuclear power plant workers may be exposed during the periodic inspection of nuclear power plants mainly derives from radiation crud. The radiation crud is formed in the following mechanism: Co is eluted from the equipment containing Co in the system facilities where primary coolant circulates, is subjected to neutron irradiation while circulating through the reactor core, and then forms a radioactive element named Co60, which is a long-lived nuclide. This substance is deposited on the internal surfaces of the equipment.

Based on this result, it can be said that the new valve seats have the radiation exposure reduction effect almost equivalent to that obtained by using the valve seats made of Co-free material. As shown in Fig. 9, the new valve seats can significantly reduce the amount of Co release. Therefore, in nuclear power plants, the adoption of the valves incorporating the new valve seats will substantially decrease the amount of Co release from Co-based alloy to achieve a drastic reduction of the radiation exposure of the nuclear power plant workers.

(7) Ease of work and inspection

We have requested multiple valve manufacturers to evaluate the ease of work. The result shows that there is no difference in the ease of work between the conventional valve seats and the new valve seats. Furthermore, the valves incorporating the new valve seat allows the evaluation of internal defects of valve seats by ultrasonic testing (UT), though the evaluation of internal defects of conventional valve seats by UT is practically impossible. This ensures that the valves incorporating the new valve seats can be shipped from the factory without any internal defect.

2.2.3 Applicability evaluation

As described above, the valves incorporating the new valve seats have the following characteristics: The va

lves have lower friction, and superior corrosion resistance, erosion resistance, and mechanical sturdiness, as well as lower residual stress. In addition, the valves allow the assurance of the shipment without any internal defect by using the ultrasonic test (UT). Furthermore, the valve seats incorporated in the valves cause less amount of Co release, one tenth of the amount of Co release from the conventional valve seats. Therefore, in nuclear power plants, the valves incorporating the new valve seats will control the roughness of the valve seats caused by corrosion and/or erosion which may occur on the conventional valve seats. This effect will contribute to the reduction of the work for fitting the valve seats during the overhaul of the valves as well as to the prevention of lowered sealing capability and operating performance of the valves and of the cracking on the valve seats during plant operation. Especially, the use of the new valve seats in safety valves will minimize the occurrence of corrosion bonding since the new valve seats can reduce the occurrence of corrosion products. In addition, it was found that, in nuclear power plants, the use of the valves incorporating the new valve seats would provide the effect of lowering the radiation exposure of nuclear power plant workers.

As described above, the valves incorporating the new valve seats are thought to contribute to higher maintainability and reliability of nuclear power plants.

2.2.4 Verification in actual operating plant

In order to verify that the excellent characteristics of the valves incorporating the new valve seats, as described above, can be attained in actual plants, both the valve incorporating the conventional valve seat and the valve incorporating the new valve seat were installed under the same service environment (refer to Table 3 for the environmental conditions) in an actual operating nuclear power plant for approximately one year. Then, the respective valves were removed to perform the comparative evaluation of the conditions of the respective valve seats. Shown in Fig. 10 are the major inspection conditions of the conventional valve seats and the new valve seats.

The result of the major inspection of the conventional valve seats and the new valve seats is as follows: The visual inspection revealed the corrosion and black discoloration of the conventional valve seat. On the other hand, there was little damage found on the new valve seats.

The cross sections of valve seats were investigated for detailed examination of the damaged conditions of the valve seats. The result shows the following: In the conventional valve seats, corrosion advanced along the eutectic carbide into the inside of the valve seats, the surface layer was dropped off, the fitting faces of the valve seats before the test were lost, and the sealing capability of the valves were lowered. On the other hand, on the new valve seats, the marks made during the fitting of the valve seats before the test were observed, and there was little damage found but the partial fall-off of the granular carbide in the surface layer.

The result of verification in actual operating plant shows that the valve incorporating the new valve seat is superior to the valve incorporating conventional valve seat, in terms of

Corrosion resistance, erosion resistance, low aged deterioration (secular change) property and low Co release property.

3. Conclusion

As a result, it was verified that the valves incorporating the new valve seats can also attain superior characteristics in actual operating nuclear power plants to the valves incorporating the conventional valve seats. Based on the findings described above, the use of the valves incorporating new valve seats in actual plants will resolve the pending problems associated with the valves incorporating the conventional valve seats. In conclusion, the following effects can be attained:

a) The reduction of the sealing capability of valves can be minimized by controlling the roughness of valve seats associated with corrosion and/or erosion.

b) The reduction of the operating performance of valves can be minimized by controlling the roughness of valve seats associated with corrosion and/or erosion.

c) The amount of work for fitting valve seats during the disassembly of valves can be reduced by controlling the roughness of valve seats associated with corrosion and/or erosion.

d) The cracking on valve seats starting on surface layer(s) can be minimized by increasing corrosion resistance, by reducing residual stress, and by increasing tenacity.

e) The cracking on valve seats starting at internal defect(s) can be inhibited by ensuring no internal defect.

f) The drift phenomena of the set pressure of safety valves caused by corrosion bonding can be minimized by controlling the occurrence of corrosion products.

g) The radiation exposure of nuclear workers during periodic inspection of nuclear power plants can be reduced by minimization of the Co release quantity.

As described above, it can be said that the use of the valves incorporating the new valve seat in actual nuclear power plants will not only increase the reliability and maintainability of the valves, but also contribute to the increased reliability and maintainability of the plants, in comparison with the use of the valves incorporating the conventional valve seats hardfaced with a Co-based alloy.

Now, 20 or more valves incorporating the new valve seat as mentioned above were delivered as HHVs (Hitachi Hyper Valves) for pressurized water reactors (PWRs) and BWRs, and have been used in the actual operating plants in Japan since 2004.

4. Acknowledgments

We would like to express deep appreciation to all the persons concerned at the Chugoku Electric Power Co., Inc., who cooperated closely in the performance evaluation of "valves incorporating the valve seats made by corrosion- and wear-resistant, carbide-dispersed alloy" which are the valves incorporating the valve seats based on a new concept.

5. References

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2) Yoshihisa Kiyotoki and Mitsuo Chigasaki, "HHV corrosion resistant, withstand erosion and low Co release," Proceedings of ASME/JSME Pressure vessel and piping conference 2004.

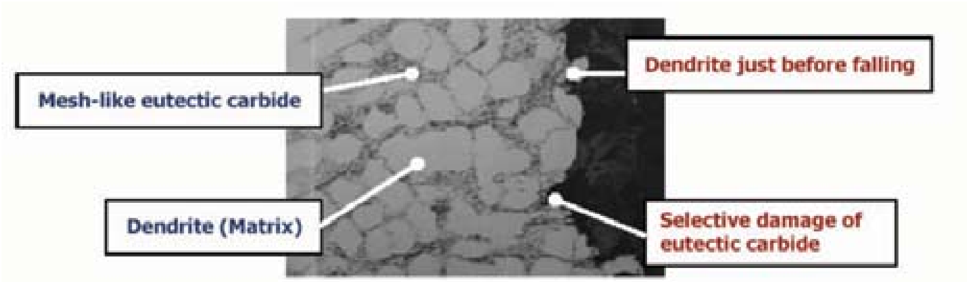
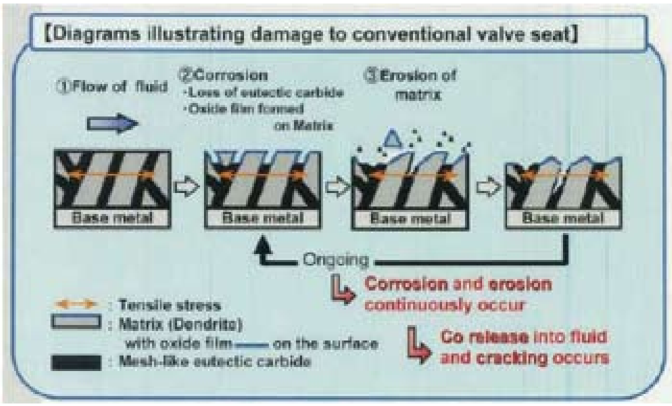
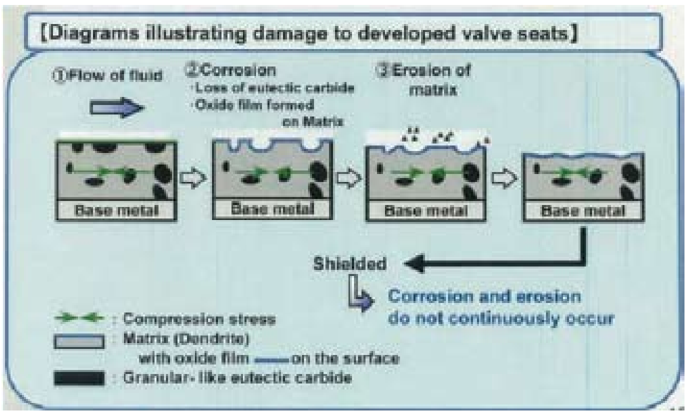


Figure 1 – Case of damage to Co-based alloy overlay welded in an actual operating plant

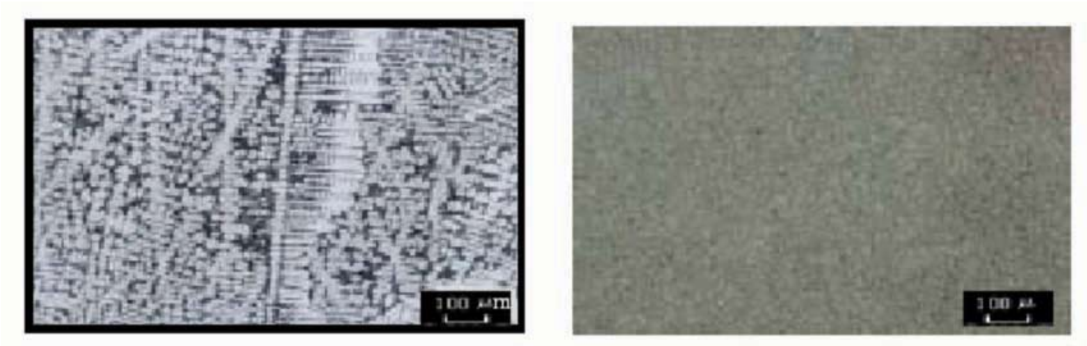


(1) Conventional valve seats



(2) New valve seats

Figure 2 – Comparison of damaging mechanism between conventional valve seat and new valve seat

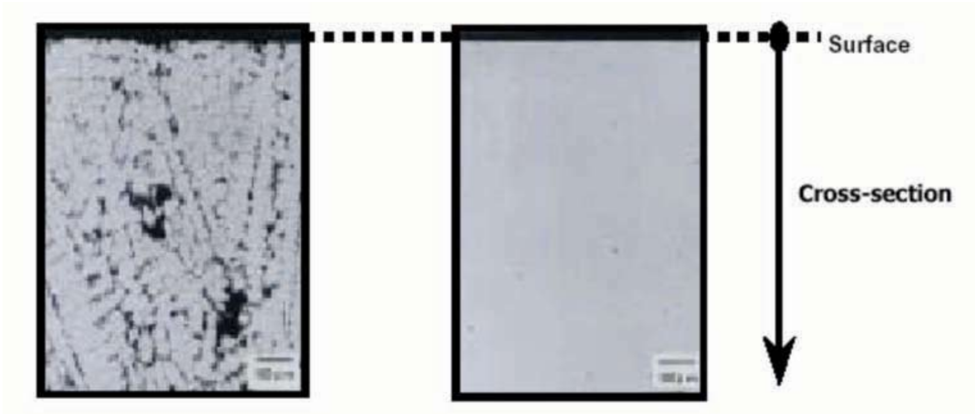


(1) Conventional valve seat **(2) New valve seat**

Figure 3 – Comparison of metallic structure between conventional valve seat and new valve seat

Table 1 Comparison of characteristics between conventional valve seat and new valve seat

Characteristics required for valve seat			Evaluation ○:Requirement satisfied ▲: Requiring improvement	
			Conventional valve seat	New valve seat
①	Corrosion resistance		▲	○[Higher corrosion resistance (Refer to Fig. 4.)]
②	Sliding property, antigalling property	②-1 During frequent operation	○	○[Lower friction (Refer to Fig. 5.)]
		②-2 Secular change of valve seat faces	▲	○[Smaller surface roughness change and lower coefficient of friction (Refer to Fig. 6 and Fig. 7)]
③	Mechanical sturdiness		▲	○[High Charpy Impact values (Refer to Table 2.)]
④	Low residual stress		▲	○[Compressive stress]
⑤	Erosion resistance		▲	○[Refer to Fig. 8.]
⑥	Low Co release		▲	○[Refer to Fig. 9.]
⑦	Ease of work and inspection		▲	○[Ultrasonic test applicable]



(a) Conventional valve seats **(b) New valve seats**

Figure 4 – Comparison of Strauss test results between conventional valve seat and new valve seat

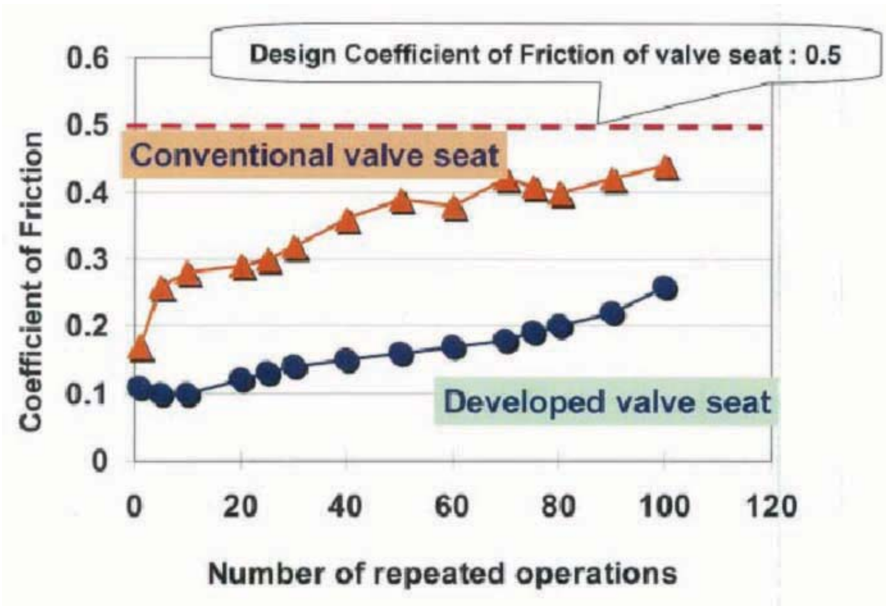


Figure 5 – Comparison of coefficient of friction between conventional valve seats and new valve seats

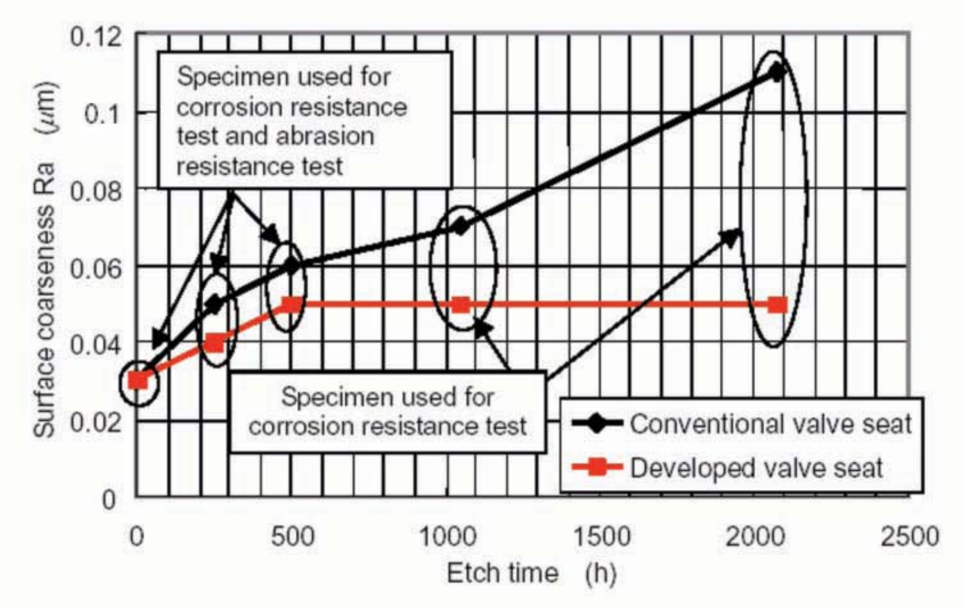
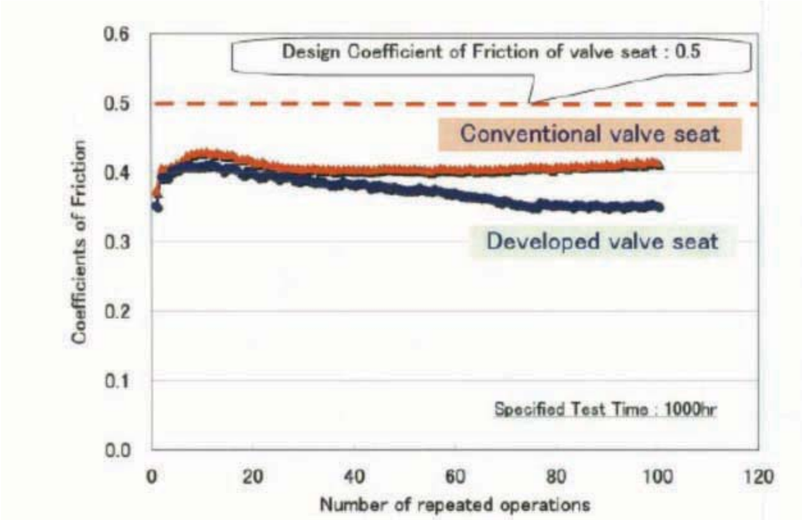
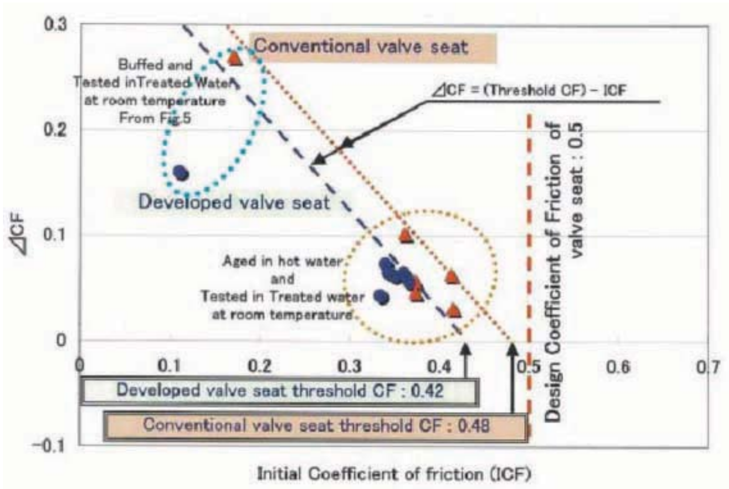


Figure 6 – Surface roughness of valve seat material made by Co-based alloy before and after complete immersion test in high-temperature water



(1) The coefficient of friction plot of the valve seats,

The results of abrasion resistance test(s)



(2) ΔCF* vs. Initial coefficient of friction plots

*ΔCF : The difference of largest coefficient of friction and the initial coefficient of friction at abrasion resistance test.

Figure 7 – Changes of coefficient of friction of valve seat caused by aged deterioration (secular change)

Table 2 Charpy impact test result of conventional valve seat and new valve seat

	Charpy impact test values at ordinary temperatures Joules/centimeter squared (J/cm2)	
	Conventional valve seat	New valve seat
Flat specimen	11.8	59.8
U-notched specimen	3.7	8.2

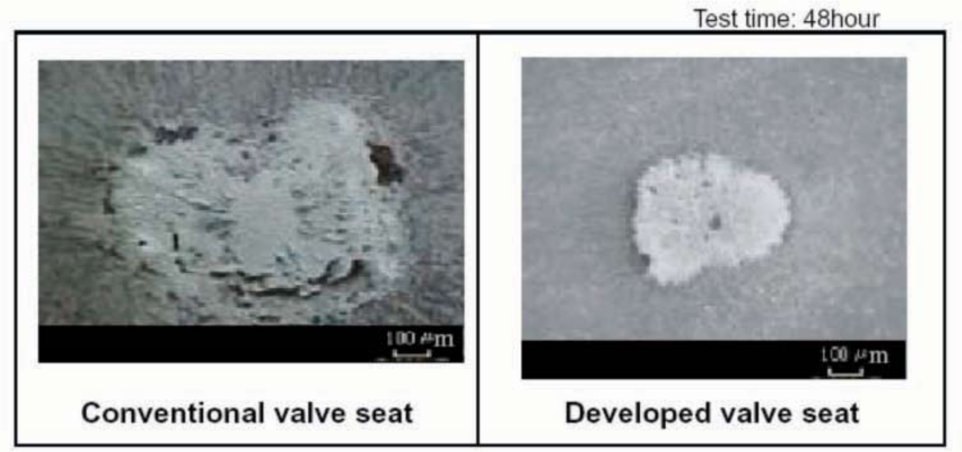


Figure 8 – Comparison of erosion resistance between conventional valve seats and new valve seats

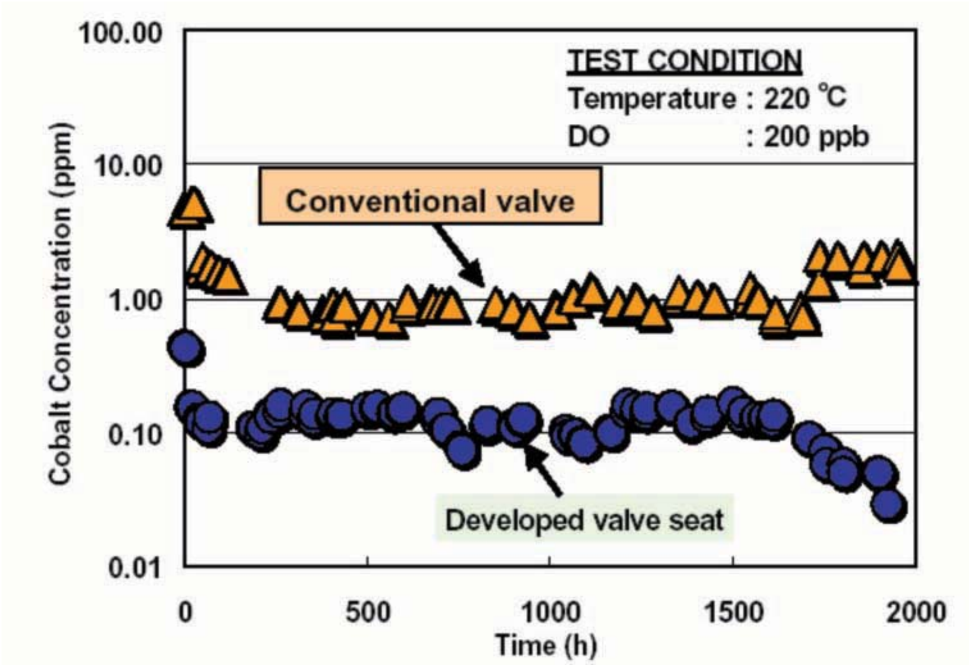
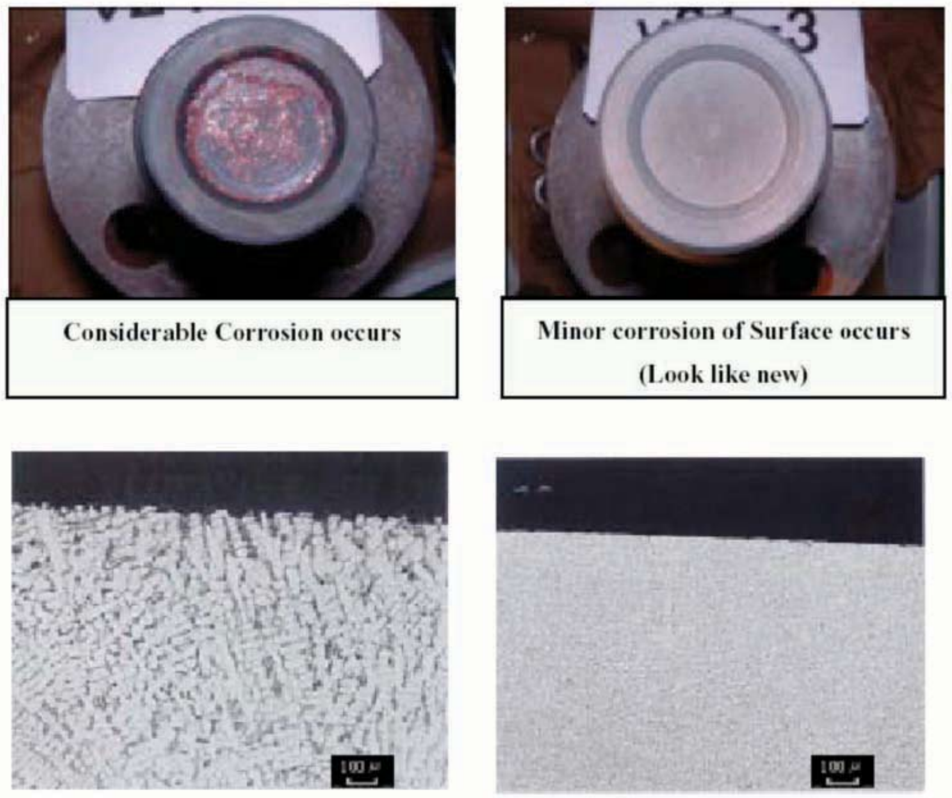


Figure 9 – Comparison of amount of Co release between conventional valve seat and new valve seat

Table 3 Performance verification in actual plant

- Pressure: 7 MPa [Megapascals]
- Temperature: Approx. 285°C
- Fluid: Main steam and its condensed water
- Dissolved oxygen (DO): Approx. 8 ppm



(a) Conventional valve seats (b) New valve seats

Figure 10 – Comparative test in actual plant between conventional valve seat and new valve seat [2)]

(note) Conventional valve seat: hardfaced with RCoCr - A

Friction Factors In Equiwedge Gate Valves Under Flow Interruption Conditions

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Abstract

This paper analyzes flow interruption test data measured on four gate valves ranging from Size 4 to Size 26 regarding friction factors in the body guides, which is a critical input for determining required valve stem thrust for assuring flow isolation. The data was obtained during a QME-1 qualification test program for the Flowserve/Edward Equiwedge Gate Valves with Type A Gas/Hydraulic Actuators. Wyle Laboratories, Huntsville Facility conducted the testing. All the valves are rated as Special Class 900 in accordance with ANSI B16.34 and are Class 2 N-Stamped per the American Society of Mechanical Engineers (ASME) Boiler & Pressure Vessel Code (B&PVC). The actuators are linear piston type and are U-Stamped per Section VIII of the ASME B&PVC.

The conditions for the flow interruption tests and the measured performance data for both the valve and actuator are presented. Comparisons between the required closing force and the available actuator force are made. These comparisons demonstrate that the equipment is capable of reliably performing its intended function of isolating flow. Test results demonstrate a friction factor less than 0.3 during valve closing.

1.0 Introduction

Flowserve and Wyle Laboratories conducted a Qualification in accordance with ASME QME-1-1994 "Qualification of Active Mechanical Equipment used in Nuclear Power Plants" on four Equiwedge Gate Valves each equipped with the Type A Gas/Hydraulic Actuator. The valves were qualified for Main Steam and Main Feedwater Isolation Service in a Pressurized Water Reactor (PWR) Nuclear Power Plant. The test program consisted of the following valve and actuator combinations:

All the valves are rated as Special Class 900 in accordance with ANSI B16.34 1988 Edition, and are Class 2 N-Stamped per the ASME Boiler & Pressure Vessel Code (Division 1, Section III, 1992 Edition including the 1994 Addenda). The actuators were U-Stamped in accordance with Section VIII, Division 1 of the ASME B&PVC (1998 Edition including the 1998 Addenda). Previous to the QME-1 qualification, these valve designs were qualified in accordance with ANSI B16.41-1983, "Functional Qualification Requirements for Power Operated Active Valve Assemblies for Nuclear Power Plants." Numerous qualification tests also were performed during the equipment development and for specific customer applications. In addition, since its introduction in 1978, a large amount of inservice operating history has been obtained.

Valve Size	Actuator Size	Service
26 x 24 x 26	A-100	Main Steam Isolation Valve(MSIV)
4	A-100	Main Steam Isolation Bypass Valve (MSIBV)
20 x 16 x 20	A-260	Economizer Main Feedwater Isolation Valve(EMFIV)
8 x 6 x 8	A-100	Downcomer Main Feedwater Isolation Valve (DMFIV)

The general requirements of the ASME QME-1 Standard for valves and how these requirements were applied to the above listed valves were presented at the NRC/ASME Symposium for Valve and Pump Testing in July 15-18, 2002. The data obtained during the qualification included force measurements from stem mounted strain gages and actuator performance data. Data was obtained throughout the test program including the Flow Interruption and Capability Demonstration. Using the force and performance data, the interaction between the valve and actuator is observed. Since the safety function of this equipment is to isolate the nuclear containment, particularly during a plant accident condition, this paper focuses on the valve closure under flow interruption conditions. The measured data demonstrates the ability of the actuator to isolate flow reliably.

2.0 Service Conditions

These valves are for Main Steam and Main Feedwater Isolation Service in a PWR Nuclear Power Plant. Except for the Main Steam Isolation Bypass Valve, they are maintained in the fully open position during normal plant operation. The Main Steam Isolation Bypass Valve is opened during the startup and shutdown of the plant but is maintained in the fully closed position during plant operation (refer to Figure 1).

The valves were designed for the following conditions:

Their safety related function is to close and provide automatic and positive isolation of the safety related piping and the containment system from the non-safety related piping; therefore, valve opening is not a safety concern. They are required to perform this safety function with sufficient force and within a maximum closure time to achieve isolation before, during and after normal and accident plant conditions.

3.0 Equipment Description

The test valves used in this program were Equiwedge Gate Valves with Type A Actuators (refer to Figures 2 thru 6). These are bi-directional valves that consist of two independent gates separated by a spacer ring. Although a significant differential pressure is sufficient to seal the valve, gate wedging due to the taper in the gates that match the angle of the seat rings aids sealing. The spacer ring maintains flexibility between the gates and prevents binding. The gates are guided throughout the stroke by guides on their sides that fit into grooves in the body (refer to Figure 7). This guiding arrangement prevents contact between the seating surfaces on the gates and seat rings until the valve is approximately 95% from the fully open position such that flow isolation occurs before seating surface contact.

The guide and seating wear surfaces in the valve are hardfaced with a cobalt base alloy (Stellite 21). Flexible graphite is used for the stem packing and the pressure seal bonnet gasket. The valves also have provisions to prevent center cavity over pressurization. This is accomplished by a bypass arrangement on one side of the valve that equalizes

Valve	Normal Operating Pressure (pounds per square inch gage [psig])	Normal Operating Temperature (Fahrenheit [F])	Design Pressure (psig)	Design Temperature (F)
Main Steam Isolation Valve	1055	553	1382	590
Main Steam Isolation Bypass Valve	1155	564	1382	590
Economizer Main Feedwater Isolation Valve	1425	455	2050	500
Downcomer Main Feedwater Isolation Valve	1425	455	2050	500

the center cavity pressure to the high pressure side of the valve. The Type A Valve Actuator is a linear piston actuator composed of hydraulic, pneumatic and electrical systems (refer to Figure 8). Its circuitry is designed to perform either a fast or slow valve closure, an open stroke or exercise cycle. The exercise cycle consists of partially stroking the valve closed (generally 10%) in a slow closure mode and then reopening the valve. The piston rod attaches directly to the valve stem and, by controlling the direction and speed of the piston, the direction and speed of the valve gates are also controlled.

The source of the valve closing force is compressed nitrogen gas contained in a volume on one end of the actuator cylinder. The pressure of the nitrogen is adjusted to suit specific applications. During fast closure, the hydraulic system acts as a classical dashpot so stem closure speed is constant.

The hydraulic system moves the piston in the non-critical direction (i.e., open the valve); this also compresses a fixed mass of nitrogen gas. The hydraulic system also controls the piston speed in the critical direction (i.e., valve closure) while the gas expands to close the valve. The pneumatic system is used to develop the hydraulic force needed for opening the valve and compressing the gas. The electrical system is used to monitor, control and verify the essential parameters and functions of the actuator.

The actuator design was previously qualified to the following Standards:

- IEEE-382, 1985 Edition

Standard for Qualification of Actuators for Power Operated Valve Assemblies with Safety-Related Functions for Nuclear Power Plants

- IEEE-344, 1987 Edition

Recommended Practice for Seismic Qualification of Class 1E Equipment for Nuclear Power Generating Stations

- IEEE-323, 1983 Edition

Qualifying Class 1E Equipment for Nuclear Power Generating Stations

4.0 Flow Interruption and Functional Capability Demonstration

The Flow Interruption and Functional Capability Test was performed to demonstrate the test valve assembly's capability to close against simulated line rupture flow conditions. The testing was performed at the following minimum pressure/temperature conditions:

Valve Size	Pressure (psig)	Temperature (F)
26 x 24 x 26	Saturated	564
4	1390	564
20 x 16 x 20	2100	564
8 x 6 x 8	2100	564

The general test sequence was as follows:

- With minimum motive power to the actuator and the required pressure/temperature test conditions established, the first flow interruption and functional capability test was performed.
  - Immediately after the closure, a seat leakage test was conducted. The test was performed at a differential pressure equal to the test pressure for a minimum duration of 30 minutes.
  - The valve was unseated against differential pressure and opened.
  - The maximum motive power was applied to the actuator and the required pressure/temperature test conditions established.
  - A second flow interruption and functional capability test was then performed followed by a seat leakage test.

All the test valve assemblies successfully closed and seated against the line rupture flow and did not incur any damage.

5.0 Valve/Actuator Performance

Typical performance data for a flow interruption test is shown in Figures 9 and 10. Figure 9 shows the actuator performance data. A review of this figure shows the following:

- The gas force behaves as a non-linear spring. The non-linear behavior is the result of the adiabatic expansion that the gas experiences during a fast valve closure. It is a predictable quantity.
- The hydraulic force varies in a smooth but unpredictable manner. The hydraulic system acts like a classical dashpot and closes the valve at constant stem travel speed while responding to flow resistance. During the closure, there are four distinct transients and, towards the end of the stroke, flow isolation and seating cause two of these transients. Since the hydraulic force is less than the magnitude of the gas force, there is a net downward force acting on the piston.
- The net actuator force is the algebraic sum of the gas force and hydraulic force. As in the case with the hydraulic force, there are 4 distinct transients.

Figure 10 shows the measured valve force during the fast closure. This force represents the algebraic sum of the packing friction force, stem rejection force and the resistive force that results from line rupture flow. During approximately the first 2 seconds of the closure, the valve only experiences the forces due to packing friction and stem rejection; it is fairly constant. However, as the gates progress into the flow stream, the resistive forces due to the flow become significant. Flow resistance reaches a peak during flow isolation and subsequent transition of the gates from the guide rails to the body seats. During and after seat wedging, the stem force results mostly from the net actuator force.

As discussed above, there are 4 distinct transients during the fast closure, and the two towards the end are due to flow isolation and seating. The hydraulic force, the net actuator force, and the valve closing force experience the same transients at the same time. Figure 11 is a comparison between the valve force and actuator force. This comparison demonstrates that, during the fast closure, the net actuator force and the valve force are equal. The hydraulic dashpot causes valve closure at constant stem velocity, and the actuator responds to the force requirements of the valve.

When stem travel stops the hydraulic pressure goes to zero and the final seating force depends totally on the actuator gas pressure. Schematically, the Type A Actuator can be represented as a spring (gas) and dashpot (hydraulic fluid) acting in parallel (refer to Figure 12).

6.0 Friction Factors

The friction factor (μ) is obtained from the following equation, which is consistent with actuator sizing methodology:

$$\mu = \frac{F_S - F_P - P_{UP} A_{STEM}}{\Delta P A_{SEAT}}$$

In the above equation:

F_S - The gross measured stem force

F_P - Measured packing drag force during valve closing from stroke test under no pressure

P_{UP} - Measured upstream pressure

ΔP - Measured differential pressure across the gate

A_{STEM} - Stem cross-section area at packing

A_{SEAT} - Seat area at mean seat diameter

Figures 13 thru 16 show the gross stem forces measured during the flow interruption tests at minimum motive power. Also given are graphs of friction factor, per the above equation, as a function of stroke time at times between flow isolation and hard seating. Although the friction factor should be constant, the figures show a variation. This variation is because the equation is only applicable close to flow isolation conditions. Before isolation, the assumed seat area is not entirely effective, and shortly after isolation, gate wedging occurs for which the equation is not valid. Then, the point of interest is at flow isolation, and results show that the calculated friction factor is generally constant there as presumed.

Table 6-1 summarizes the flow interruption test results.

The information presented for each valve size is:

1. The initial test pressure

2. The final test differential pressure after seating

3. The measured stem force at flow isolation

3. The measured stem force at seating

This force represents the largest force required to achieve valve seating.

4. The measured actuator force after seating

This is the force produced by the actuator on the valve after the valve is seated. The difference between this force and the valve force at flow isolation represents the margin between the available actuator force and the required stem force.

5. Calculated minimum available actuator force

This force is based on the actuator precharge pressure. It is determined during the initial actuator sizing and considered a minimum force because the actuator is driven under minimum motive power conditions.

6. Calculated maximum available actuator force

This force is the same as item 5 except it is considered a maximum force because the actuator is driven under maximum motive power conditions.

Table 6-1 Flow Interruption Test Results

Parameter	Valve Size			
	26 x 24 26	4	20 x 16 x 20	8 x 6 x 8
Initial Pressure prior to Test (psi)	1375	1538	2175	2100
Test Differential Pressure after Seating (psi)	1168	1528	1492	1740
Measured Stem Force at Flow Isolation (lbs)	112000	4280	46900	9300
Measured Stem Force at Seating (lbs)	151263	7277	69675	16816
Measured Actuator Force after Seating (lbs)	160800	8764	109316	18296
Calculated Minimum Available Actuator Force (lbs)	158362	7680	108639	18301
Calculated Maximum Available Actuator Force (lbs)	180703	8766	123990	20887
Calculated Friction Factor at Flow Isolation	.276	.10	.25	.22

7. Calculated friction factor at flow isolation

As noted in Section 3.0 (Equipment Description), the source of the valve seating force is compressed nitrogen gas contained in a volume on one end of the actuator cylinder. The force produced by the actuator at valve seating is dependent on the initial gas pressure when the valve is backseated. When the valve is backseated, the nitrogen gas is contained in a fixed volume so its initial pressure is dependent on its temperature. The minimum motive power condition for the actuator is the gas pressure at the minimum specified ambient temperature which for this program was 50 oF. The maximum motive power condition for the actuator is the gas pressure at the maximum specified ambient temperature which for this program was 122 F.

It should be noted that the final actuator thrust shown in Figure 13 thru 16 are conservatively low compared to what would occur in service. The net actuator force shown is when the equipment is at the minimum ambient temperature. In actual service, a line rupture condition would only occur when the plant is operating. Under operating conditions, the ambient temperature of the equipment and the actuator gas pressure would both be higher; thus producing a higher seating force.

Figures 13 thru 16 and Table 6-1 show the following:

1. The actuator force adjusts to the valve force to maintain equilibrium.

2. During and after gate wedging, the valve stem force is totally driven by the actuator; thus producing a force margin that is dependent on the actuator gas pressure.

3. The stem force at flow isolation and seating is less than the available actuator force.

The calculated minimum required actuator force forms the basis for determining the required nitrogen gas pressure. As discussed above, the initial gas pressure determines the available actuator force for seating the valve. There are several reasons why the calculated actuator force, based on a 0.3 friction factor, results in significant margin. The main reasons are the thermal expansion coefficient of nitrogen, the valve stroke and the packing friction.

When the actuator performs a fast valve closure, the nitrogen gas experiences an isentropic expansion. The gas pressure at the end of the valve stroke is:

$$Pf = Pi \times (Volf / Voli)^{k}$$

where: Pf = Final Gas Pressure

Pi = Initial Gas Pressure

Voli = Initial Gas Volume

Volf = Final Gas Volume = Voli + Piston Area x Valve Stroke

K = Thermal Expansion Coefficient for Nitrogen

The change in gas pressure depends on its volume change and the thermal expansion coefficient of nitrogen. The assumed expansion coefficient is 1.6. However, this is a limiting value for the pressure range used in this equipment. The actual values are lower which result in higher terminal gas pressures.

The valve stroke determines the change in gas volume during a closure. A shorter stroke causes a smaller change in gas volume. This results in a higher terminal pressure. When the actuator is sized, the stroke used in sizing is the actuator stroke. Since the valve stroke is shorter than the actuator stroke, there is a higher terminal pressure.

As noted above, the initial gas pressure determines the available actuator force for closing the valve. If it is desired to provide greater margin over calculated thrust or to size for a greater differential pressure, it is only necessary to increase the initial gas pressure. Comparing the minimum and maximum calculated actuator forces (refer to Table 6-1), there is approximately a 14% increase in the actuator force at valve seating. However, consideration must be given to the resulting stresses in the valve components and to the maximum pressure capacity of the actuator.

7.0 Conclusions

This paper presents the test conditions and performance results for the flow interruption tests, performed during the QME-1 qualification of the Equiwedge Gate Valve. The test data demonstrate that the actuator is capable of reliably performing its intended function of isolating flow. The results indicate a friction factor that is less than 0.3 during closing.

The actuator hydraulic system acts as a classical dashpot such that the valve closes at constant velocity and hydraulic fluid pressure automatically adjusts to compensate for valve resistance forces. Although not presented in this paper, other cycling performed under no pressure or static pressure conditions show similar results.

In all cases, the actuator force exceeded the required force for flow isolation with significant margin. During and after gate wedging, the actuator drives the stem force exerted on the valve. The available actuator force was sufficient to isolate flow and to hard seat the wedge adequately.

The results presented in this paper are based on minimum motive power conditions. In actual service, because of the higher ambient temperature, the motive power to the actuator would be higher; thus increasing the margin over required thrust. The margin can be increased further by increasing the gas precharge pressure in the actuator.

Note that the QME-1 standard and hence test program did not fully address valve preconditioning or for steam aging effects on Stellite surfaces. The observations of valve preconditioning are documented separately in the paper "Observations of Preconditioning during the ASME QME-1 Qualification of the Edward Equiwedge Gate Valve with the Type A Gas/Hydraulic Actuator" presented at the Ninth EPRI Valve Symposium.

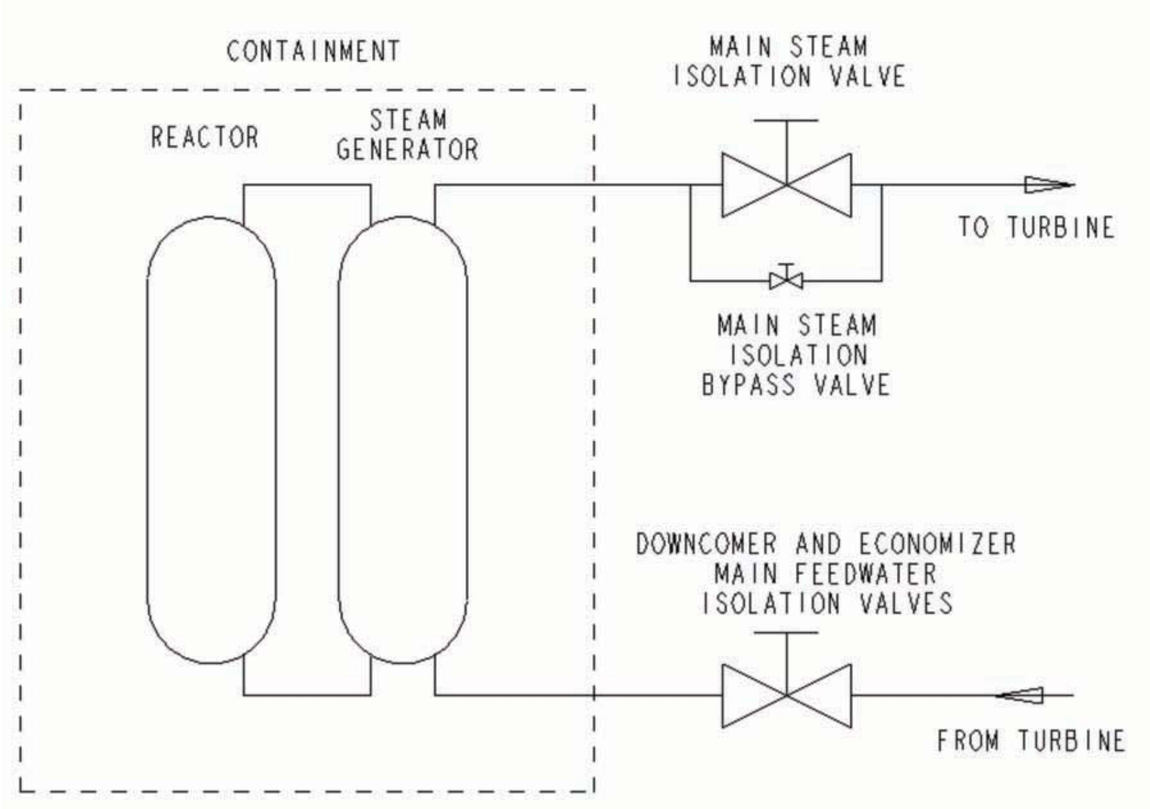


Figure 1 – Schematic for the Main Steam and Main Feedwater Isolation Valves in a PWR Nuclear Power Plant

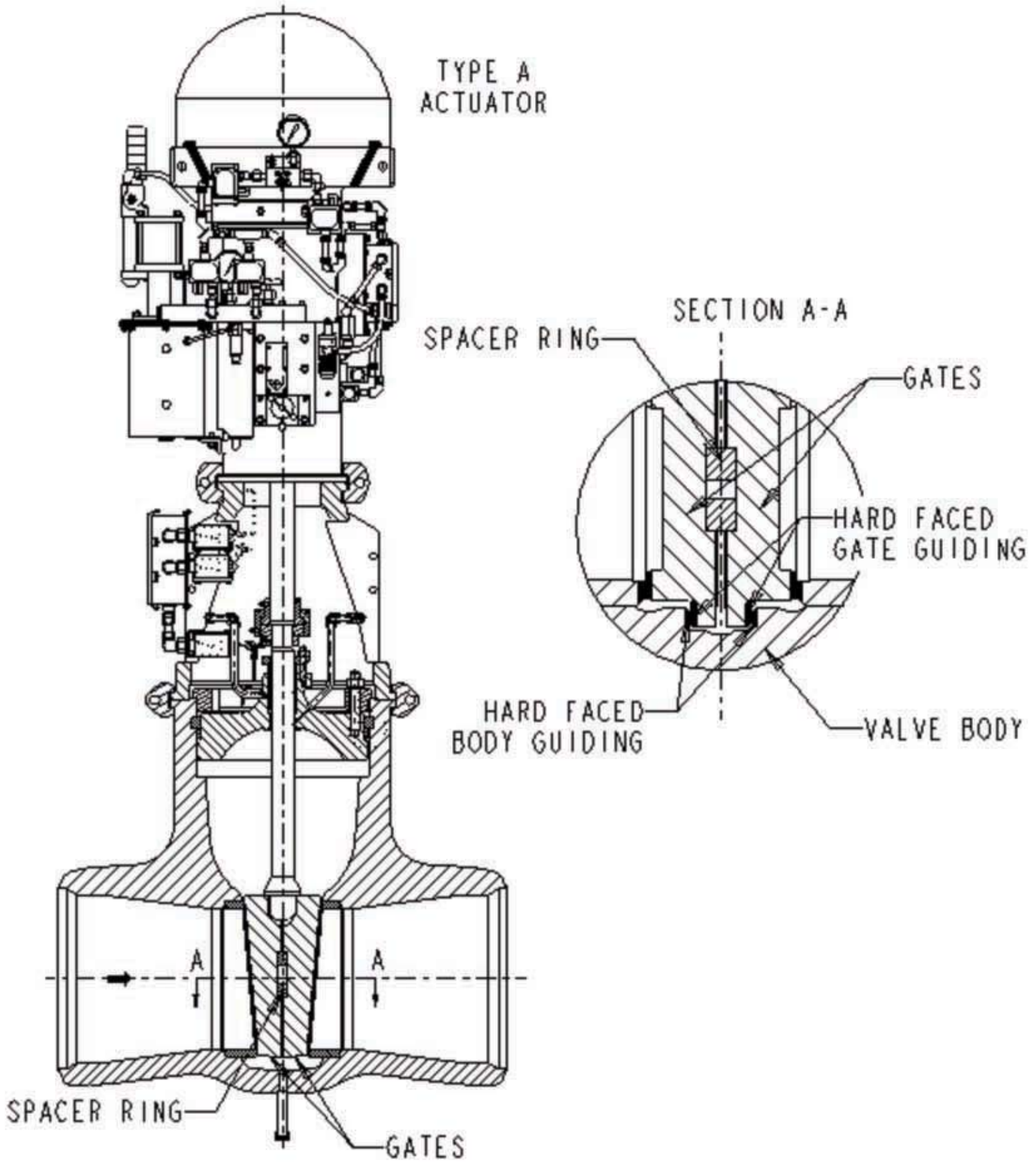


Figure 2 – Cross-section of the Flowserve/Edward Equiwedge Gate Valve with a Type A Actuator

Figure 3

Size 26 x 24 x 26

Main Steam Isolation Valve

Figure 4

Size 4

Main Steam Isolation Bypass Valve

Figure 5

Size 20 x 16 x 20

Economizer Main Feedwater Isolation Valve

Figure 6

Size 8 x 6 x 8

Downcomer Main Feedwater Isolation Valve

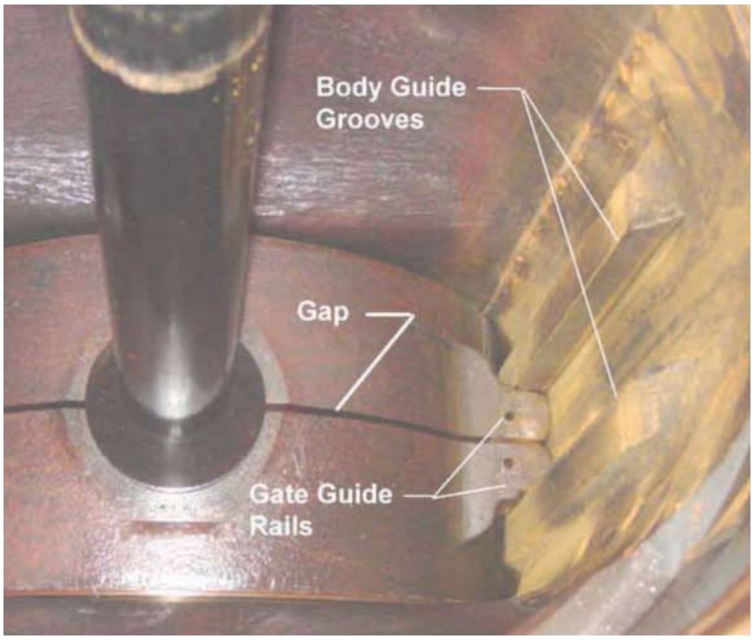


Figure 7

Gate Guiding System

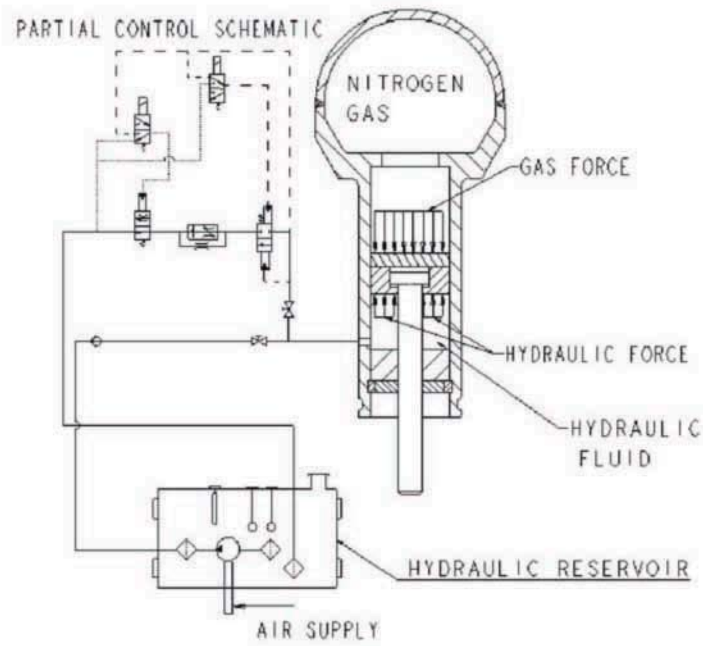


Figure 8

Actuator Cross Section and Partial Control Schematic

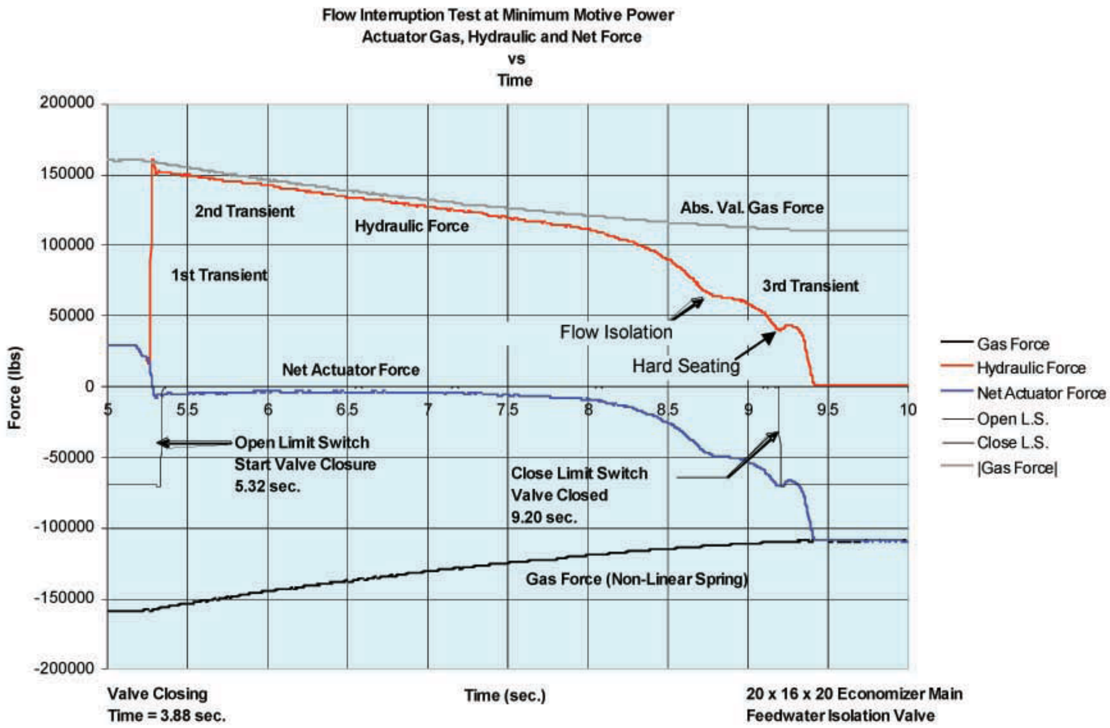


Figure 9

Actuator Performance Data

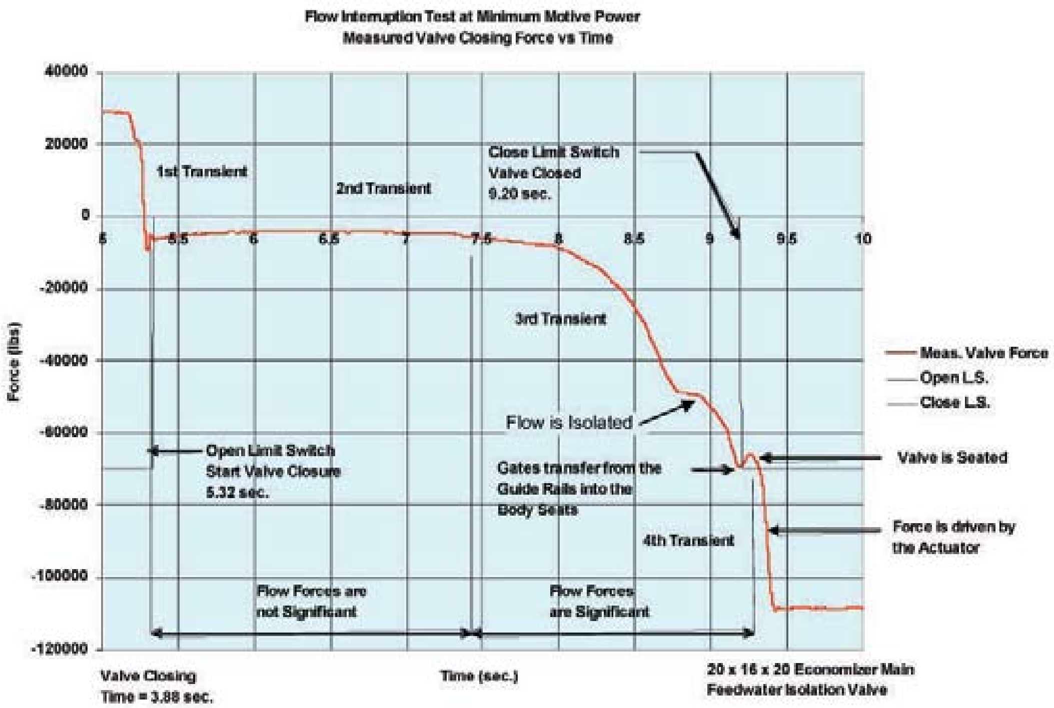


Figure 10

Valve Closing Force

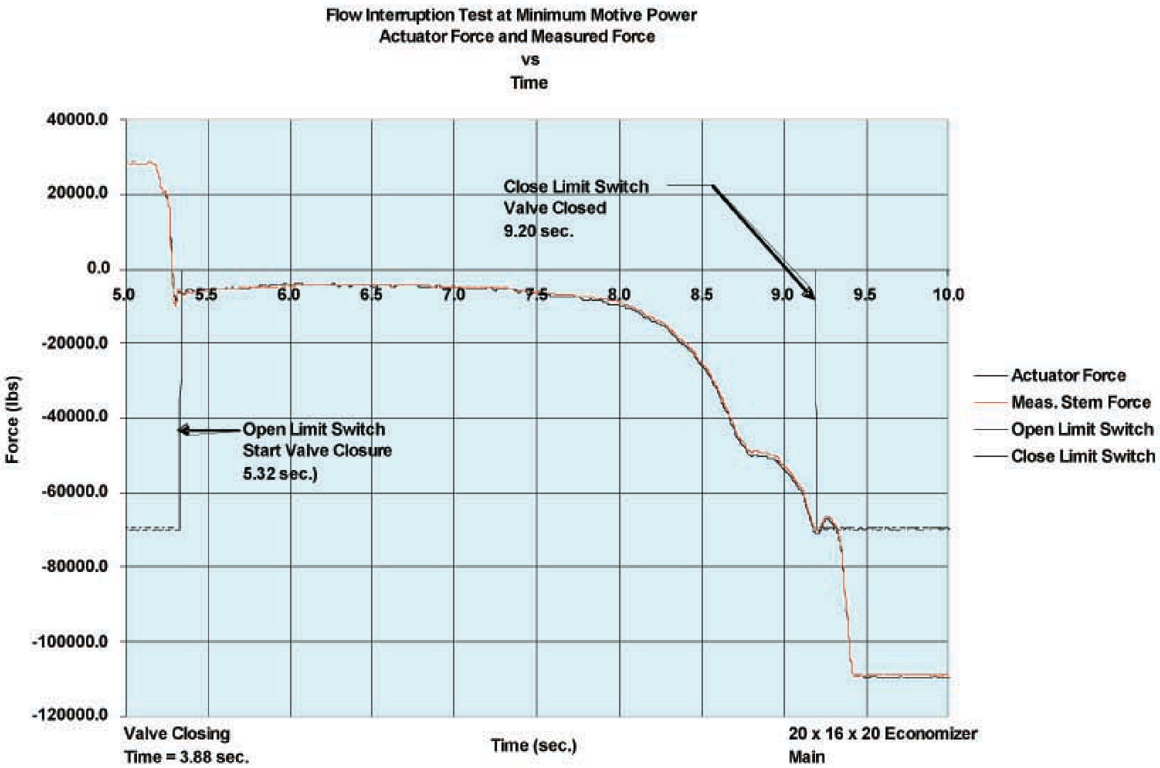


Figure 11

Net Actuator Force and Measured Valve Closing Force

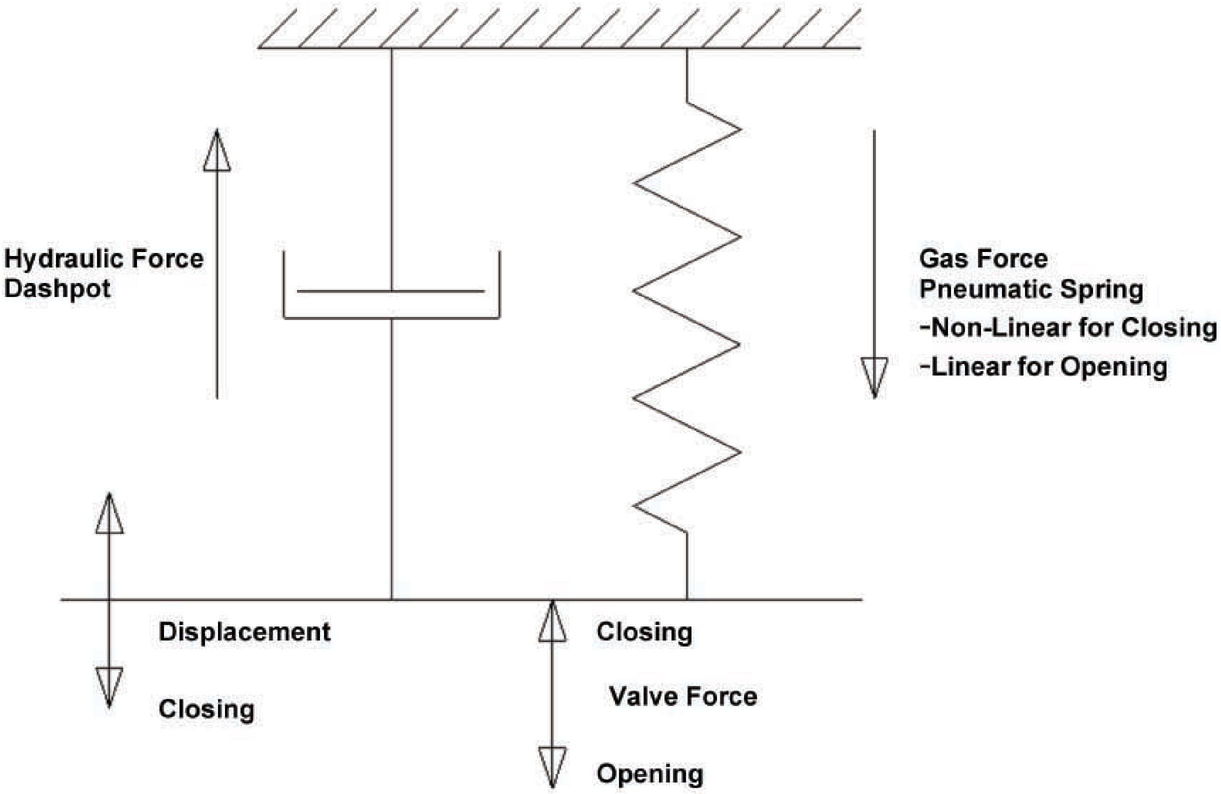


Figure 12

Operating Schematic

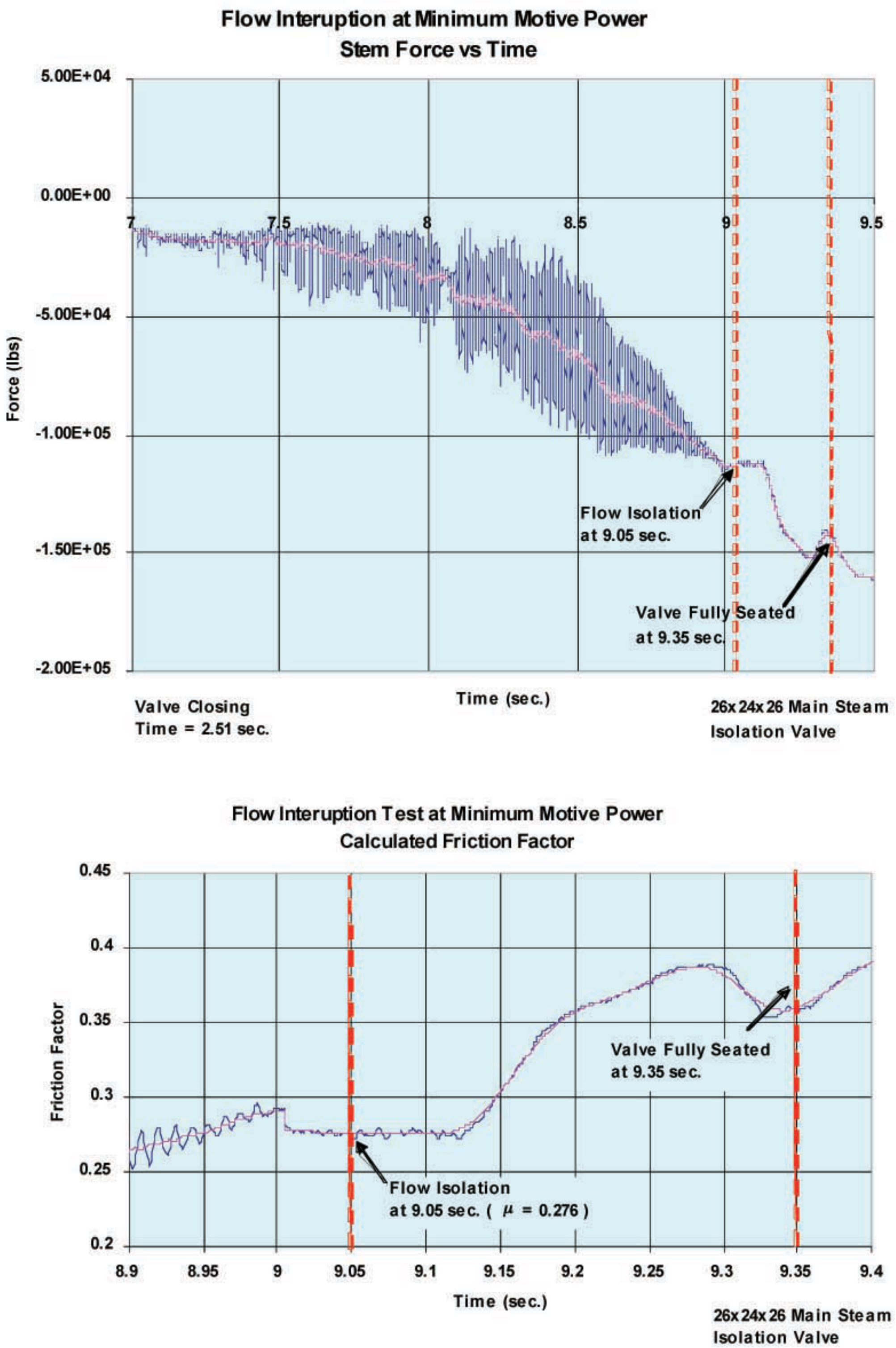
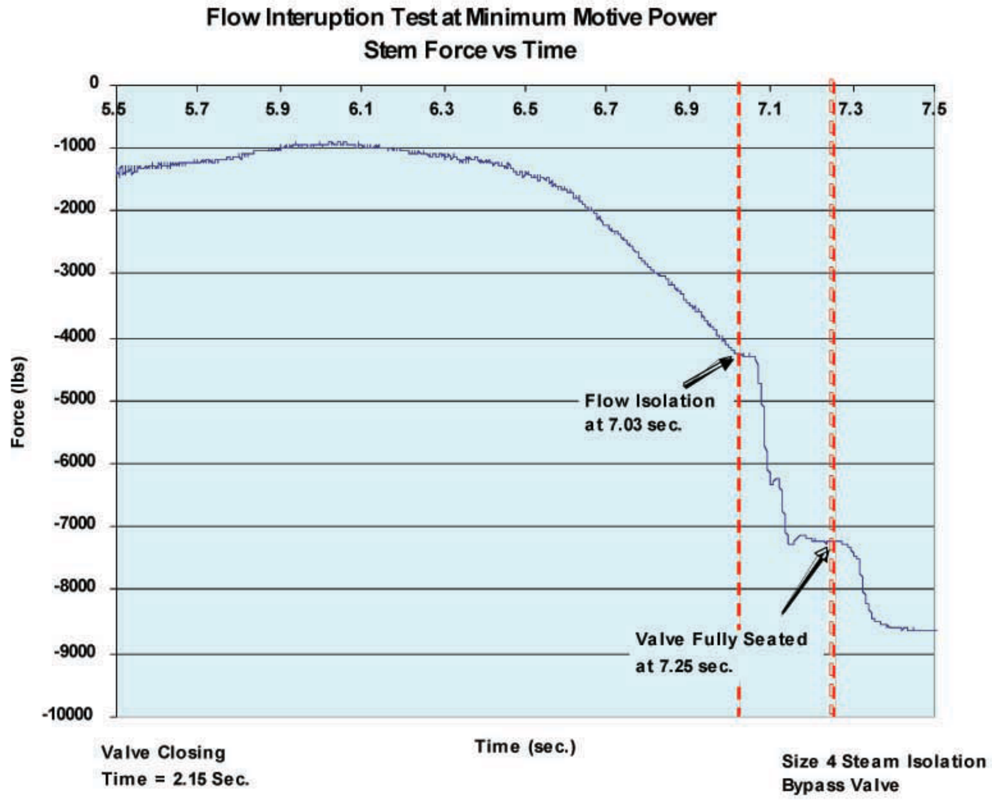


Figure 13

Size 26 x 24 x 26

Valve and Actuator Forces



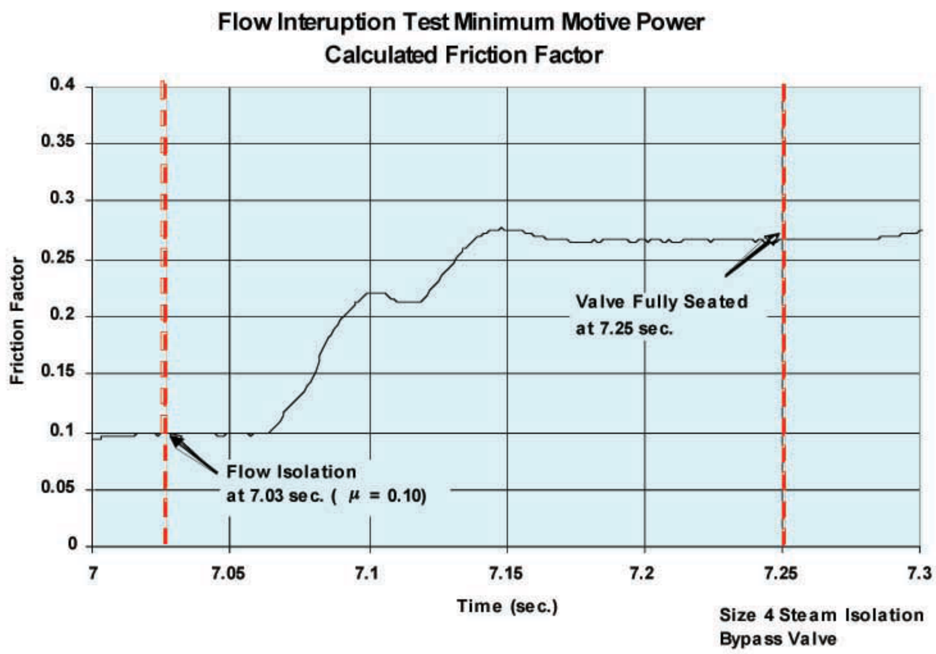
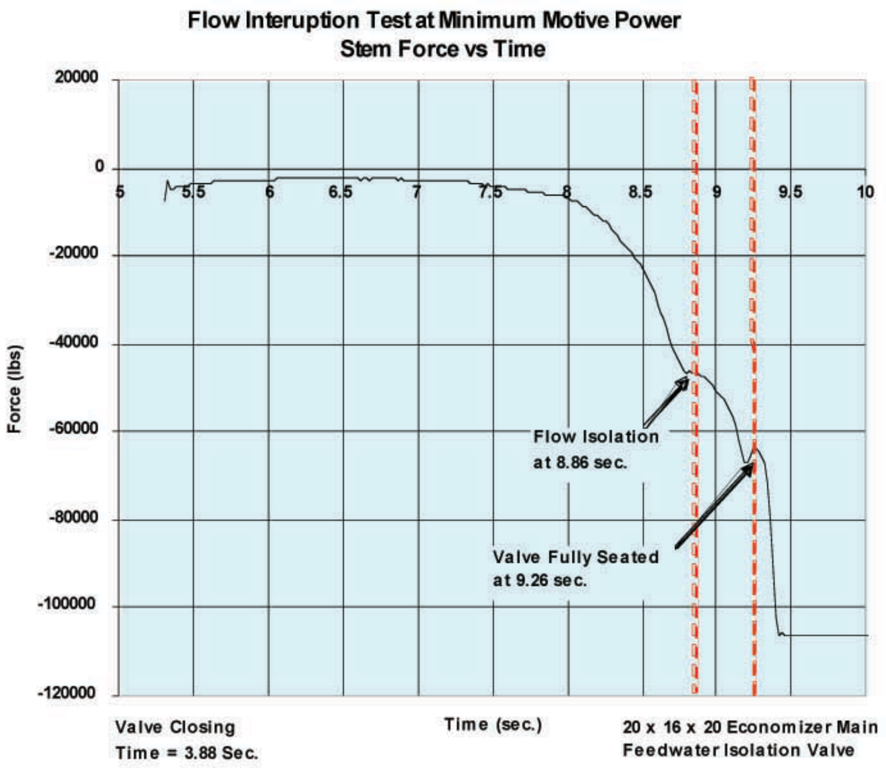


Figure 14

Size 4

Valve and Actuator Forces



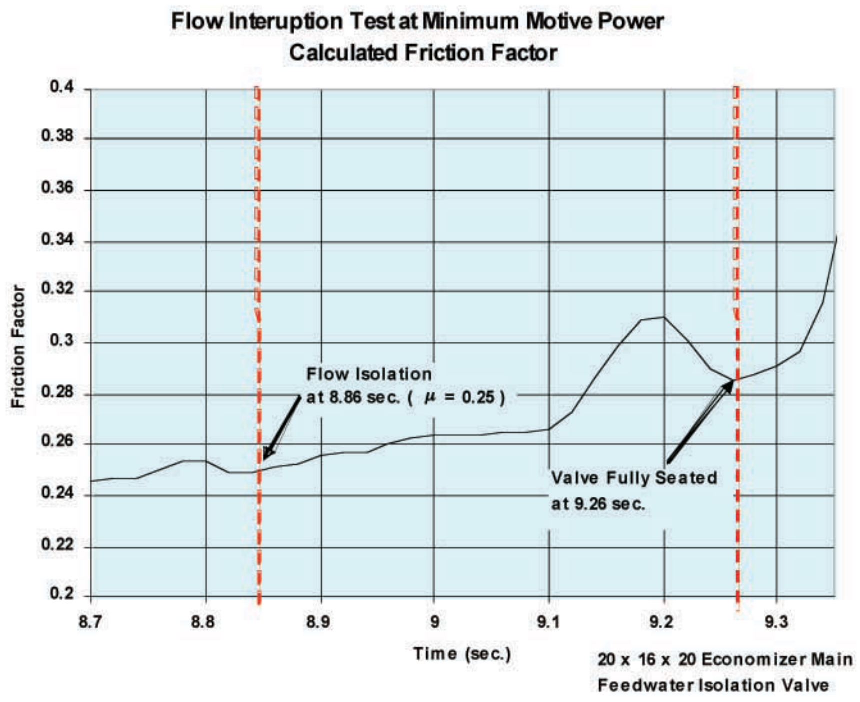
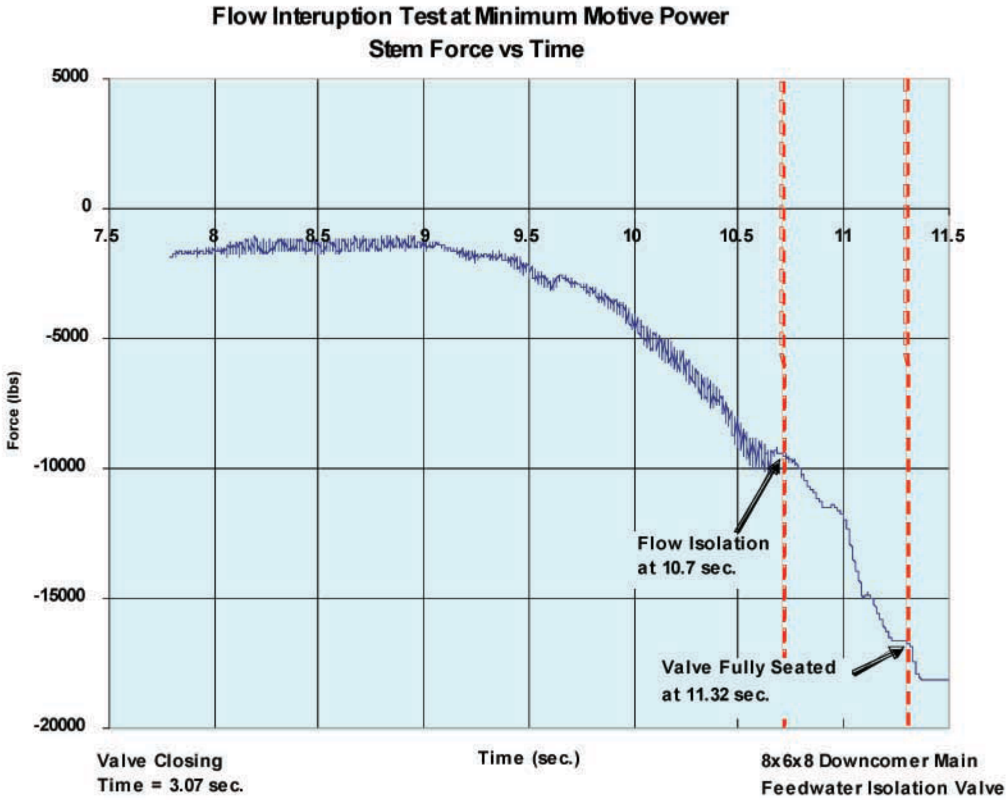


Figure 15

20 x 16 x 20

Valve and Actuator Forces



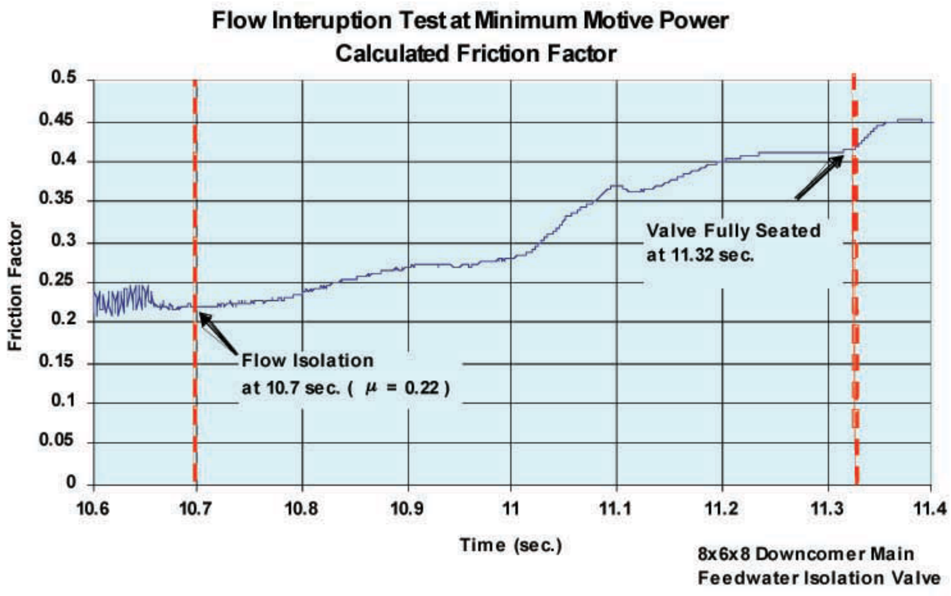


Figure 16

8 x 6 x 8

Valve and Actuator Forces

Avoid letting your Check Valves go to Failure by Trending

Greg Hunter
American Electric Power
D.C. Cook Nuclear Plant

Roger Sagmoe
Nuclear Management Company
Prairie Island Nuclear Plant

Tony Maanavi
Exelon Corporation
Byron Station

Michael Robinson
K&M Consulting, Inc

and

The Nuclear Industry Check Valve Group

Introduction

Trending - what are the attributes that can be trended for determining Check Valve degradation? There are many publications available such as The Maintenance Engineer Fundamentals Handbook, TR-106853 Palo Alto, CA: EPRI, November 1996, Predictive Maintenance Primer Revision Guide, Palo Alto, CA: NMAC, April 1991 1007350, NIC Check Valve Nonintrusive Analysis Guide NIC, Final Report - Revision 0, May 9, 1999, etc. that discuss testing methods and program activities but none provide information activities and attributes that can be trended. The Nuclear Industry Check Valve Group (NIC) has filled that gap. In the summer of 2005, NIC published the "Tracking and Trending Guide for Check Valves." Based on several NIC and Industry test programs along with actual plant experience, NIC along with vendors developed a guide that helps utility personnel apply the proper test for determining the condition of check valves. This paper will discuss performance, condition, and operational/functional readiness activities and attributes. These activities and attributes include, full / partial open, closure, backflow, mechanical exerciser, and seat leakage. Testing techniques/technologies/methods including acoustics, magnetics, eddy current, external inspection, internal inspection, Radiography, etc will be discussed. Yes, trending the proper attribute can keep your check valve from failing.

History

Both the NRC and INPO recommend, as a good practice, trending check valve condition, so incipient failures are identified and maintenance is performed prior to failure. Additionally, based on regulatory activity (NRC issued Information Notice [IN] 2000-21, "Detached Check Valve Disc Not Detected by Use of Acoustic and Magnetic Non-intrusive Test Techniques," and INPO re-emphasized SOER 86-03, "Check Valve Failures or Degradation") there is increased scrutiny and oversight of check valve trending.

In June of 2001 the Nuclear Industry Check Valve Group (NIC) started to determine the trendable attributes of all the testing and test methodologies that are being used to determine check valve conditions. Because of this, several activities were started that included an update to the NIC Check Valve Analysis Guide, the start of a tracking & trending guide for check valves, and the creation of a Technical Advisory group (TAG) for the development and implementation of a program for Check Valve Performance Trending.

In November 2002, the TAG performed its first set of testing and, in 2003, published the results. The results are very encouraging and, as anticipated, the technologies evaluated have trending capabilities which are discussed in detail within the report. This effort was funded by 20 nuclear sites. However, with the incorporation of the initial test results, new milestones were developed and further testing is still being considered as well as the validation of the level of knowledge and training which is required to trend data.

In December 2005 NIC published the "Tracking and Trending Guide for Check Valves." This tracking and trending document builds on the technologies and methods commonly used by utilities for determining check valve conditions including methods described in the NIC Nonintrusive Analysis Guide. A working knowledge of this document will help the reader understand the testing activities and attributes. Training and qualification are also background prerequisites for successfully applying test methods and technologies.

What is the driving force for utilities to seek additional information as to the need for this program? A major reason for this project would be that the NRC has mentioned (in numerous places including the 1999 rulemaking, NUREG-1482, IN 2000-21, and in past meetings) that trending and evaluation of existing data need to be used to reduce or extend testing intervals on check valves. That is what this test program is designed to provide, a solid basis for trending parameters and to provide uniformity throughout the industry. The NRC staff has been very candid about their interest and is receptive to the approach we are taking. If we do not continue with our proposed activities and meet our objectives in a timely fashion, others may set their own criteria which we will then have to follow.

Basics

Before we get started with the discussion about what is currently trendable we need to discuss the basics. For trending to be effective a decision on how the information is going to be gathered and how it is going to be reviewed must be made. The most effective way is in a Check Valve Program which systematically evaluates all available data pertinent to check valve performance for the purpose of maintaining a high level of check valve reliability. It does so by identifying those check valves susceptible to degradation and implementing appropriate inspection, test, or maintenance activities.

A solid Check Valve Program utilizes an approach by which the inspection and testing schedule, developed based upon valve design, application, maintenance history, and industry data, is continually reviewed, in response to the results of these tests and inspections to ensure optimal application of program resources. The objective of this review and optimization is to prevent check valve failures by identifying deficient valve applications, determining the effects of these deficiencies and appropriately testing, inspecting, modifying, or performing periodic maintenance as required, ensuring continued reliability and performance.

Within the program there should be two types of activities and attributes, Performance and Condition. Performance refers to data collected to determine the operational and/or functional readiness of a component and Condition refers to data collected to determine the accumulative effects of aging and degradation. Performance attributes are collected using a variety of activities including but not limited to full or partial open flow with system flow, closure by differential pressure, backflow, or seat leakage.

Condition attributes refer to data collected to determine the accumulative effects of aging and degradation over time.

Technologies & Methods – Performance Vs. Condition

The technologies that we reviewed include but are not limited to acoustics, pulsed EM (EC), DC/AC magnetics (DCM/ACM), ultrasonics (UT), airborne ultrasonic (AU) leak detection, radiography (RT), infrared thermography imaging (IRI), Disassembly and Inspection, and performance based testing; forward flow, differential pressure/ temperature, and quantified leak rate.

The data collected using these various methods and techniques have parameters that are both measurable and trendable. When a measured parameter changes in relation to a known cause at repeated test conditions, it is considered a trendable parameter. The ability of a measured parameter to change in any given direction based on valve degradation is referred to as its trendability.

Check valves can have both performance and condition attributes that are collected at periodic intervals and under differing plant conditions. Whenever possible, data collected should be obtained at similar system conditions (e.g., power level, pressures, temperatures, and flows). Test lineups need to be determined that make the data obtained relevant to the valve and not the system. It is imperative for trending, that test conditions be duplicable to the extent possible, so that any contributions to the measured parameter's change from other possible variables are minimized.

Operational / Functional Activities

These activities are more commonly known as the Surveillance and Monitoring activities. Here are the activities that are used for check valves:

<u>Open Flow (Full or Partial)</u>

Flow is passed through the valve where measurements are directly or indirectly taken. This can be determined by position indicators, pressure drop, or non-intrusive technologies, such as acoustics, magnetics, radiography or electromagnetic technologies.

<u>Close</u>

Upon cessation or reversal of flow the valve obturator moves to the closed position and seats. Closure is confirmed by direct or indirect means using the backflow methods below or by other non intrusive means.

<u>Backflow</u>

Pressure/Flow Profiles taken upstream and downstream of valves can confirm that the valve is closed, leaking by, or flowing in reverse.

Pump Reverse Rotation Check of a stopped parallel pump can be used to confirm that the valve obturator is in contact with the seat by verifying that the pump shaft is not rotating backwards.

Monitoring of System Parameters such as changes in tank levels, system pressure alarms, flow alarms, etc.

Temperature Profiles taken upstream and downstream of a valve using a contact pyrometer or other temperature measurement device can indicate if it is closed. This can be a pass/fail test or criterion applied where temperature is limited to maximum value.

Infrared Thermo Imaging (IRI) analysis is used to detect and analyze temperature differences or gradients. IRI can detect seat leakage when warm fluid is allowed to pass back through a closed disc and produces a temperature gradient. This can be a pass/fail test or criterion applied where temperature is limited to maximum value.

<u>Mechanical Exerciser</u>

An external actuation lever is used to verify the travel of the valve disc, thereby verifying disc travel from valve seat to valve disc backstop.

Seat Leakage

Acoustic (audible range from 0 to 20 kHz) and airborne ultrasonic (non-audible range above 20 kHz to 1 MHz) leak detection methods can provide a means to monitor online seat leakage when a valve is closed and pressure differential is established.

Condition – Based and Predictive Activities

Inspections (Intrusive Techniques)

There are two primary vantage points for collecting inspection data: The first is external and the second internal.

External inspections are more useful for those check valves that have external operating mechanisms and shaft packing than for those with no external moving parts.

Internal inspections provide a direct and proof positive means of collecting condition data.

Disassembly and Inspection

This method is used to verify the ability of a check valve to move through its full stroke via visual observation of the valve internals. It is used when system or plant conditions cannot easily be established to verify operability, if required, at design basis accident conditions. Additionally, the valve internals are visually and mechanically inspected for wear, corrosion, erosion, and other degradation. The information gathered during disassembly and inspection can be used in conjunction with diagnostic testing signature analysis to monitor degradation during subsequent diagnostic testing. This is also the time to collect information and measurements for NIT—verification of seat angle, disc thickness, or collection/confirmation of optimum sensor placements, EC stroke delta by manual disc stroking, etc.

Boroscopic Inspection (Fiber Optics)

Fiber optic or boroscope probes provide a means to perform a visual inspection of valve internals without a complete disassembly.

Seat Leakage

Mass Make-up and Pressure Decay leak rate methods are effective at quantifying and monitoring seating capability and valve condition. Typical valves tested are Appendix J (CIVs - Containment Isolation Valves), Technical Specification High/Low pressure interface valves (PIVs – Pressure Isolation Valves), and OM Code required valves. Leakage rate tests may provide trend data that predict future leakage problems. The leakage rate of a valve may also be a good predictor of future failures associated with hinge pin wear and other valve degradation of the closed disc position changing relative to the valve seat.

Diagnostics (Non-intrusive Techniques / Technologies / Methods)

The various techniques, technologies, and methods used for collecting non-intrusive diagnostic condition data will be examined below. These techniques and methods are highly subjective and based on personnel experience performing the test and analysis of data. For this reason they are open for interpretation and evaluation.

Acoustics

Acoustic monitoring during flow conditions can identify backstop tapping, seat tapping, and relative wear/looseness of internal components. When a valve is seated, acoustic monitoring can detect leakage in some cases, providing a pressure differential across the disc exists. When flow is initiated, acoustic monitoring can confirm the full open position providing the disc or other member impacts the backstop depending on variables that may require evaluation for any specific check valve. When flow is terminated, acoustical monitoring can confirm disc closure providing the disc impacts the seat. This also depends on variables that may require evaluation for any specific check valve. The force imparted during either event has to be sufficient to generate a measurable impact exceeding the background RMS levels. A second technology should always be used to corroborate the open and/or closed positions to ensure the highest level of confidence possible.

Magnetics

Magnetic Flux Analysis (AC or DC) – Magnetic flux analysis can determine disc motion/flutter and assist in confirming open and close positions in conjunction with acoustic monitoring.

Eddy Current (pulsed electromagnetic) analysis can determine disc full open and close under stroke test conditions (after initial baseline test where acoustic impact, characterization from a similar check valve, or disassembly stroke qualified full stroke voltage delta is obtained).

Ultrasonics

Ultrasonic monitoring can determine disc position under flow or detection of disc in the closed position or stuck open under no flow conditions for fluid systems other than steam, gas or air. Ultrasonic monitoring can also detect and quantify disc flutter and indicate conditions that will cause accelerated wear or further degradation of the valve. Manual scanning methods can also be used to confirm that the internals are intact.

Radiography

Standard/Conventional Radiography – Radiographic examination is a diagnostic tool to assess disc position, to confirm that internals are intact, and to some extent to indicate their condition. RT is difficult on cast thick steels where the absorption of the energy is more likely to occur. Precise source placement is required to achieve desired resolution.

Phosphor Plate Radiography – Phosphor plate radiography examination has many advantages over conventional RT. The radiograph is fed into an electronic developer and downloaded to a computer, where, with software the exposure can be manipulated and distances can be determined. Precise source placement is required to achieve the desired resolution.

Leak Detection

Acoustic and Airborne Ultrasonic leak detection methods can provide a means to monitor seat leakage when a valve is closed and pressure differential is established. Note: Ultrasonics for leak detection is not the same as ultrasonics for disc position and disc flutter. They are two different technologies.

Mechanical Exerciser

An external actuation lever is used to verify the travel of the valve disc, thereby verifying disc travel from valve seat to valve disc backstop. Criteria are applied to breakaway and full open torque values to determine if additional frictional load is present or if the disc is attached.

Infrared Thermo Imaging (Thermography)

A technique based on measuring and comparing infrared radiation emitted from various equipment surfaces. Infrared thermography can aid in determining tank levels and internal valve leaks.

Stroke Timing

The valve stroke is measured based on a predetermined test configuration and trended over time to monitor for abnormal changes that may be indicative of valve degradation or a change in test conditions. When using stroke time, test conditions should be similar at each test to minimize timing differences caused by them (for example, measuring the time it takes from a pump trip to the discharge check valve closing, or a motor-operated valve or air-operated valve [MOV/AOV] actuation to the check valve open/close acoustic impact, etc.). Binding may be evident by a longer stroke time, or a loss of disc may reveal itself in a shorter stroke time (e.g., hinge arm in a swing check with no disc attached).

Quantitative Wear Prediction

The need for and benefits of applying quantitative wear and fatigue predictions are significant as they relate to screening and prioritizing safety-related and economically significant check valves. Qualitative data, though generally easier to produce and compile, is varied in consistency and usefulness. Quantitative data requires a greater level of effort to produce, but the end result is generally more tangible and definitive. Wear quantification enhances condition-monitoring activities for safety-related, production-critical and/or economically significant check valve applications by providing an analytical framework for trending valve performance data. For example, it allows for the proper normalization of tracked parameters to account for variations in condition that influence them, thus improving the active feedback process and facilitating problem resolution through planned design changes.

Test Performance and Tracking (Operational Readiness)

One purpose of the Inservice Testing (IST) Program is to assess the operational readiness of pumps and valves in the program. However, caution is advised when using IST results as the sole basis for proving operability. Any other available information that has a bearing on equipment operability should also be considered.

Valve disc movement tests (open and closed) do not generally require the use of reference values. However, it is the owner's responsibility to determine acceptance criteria for the test conditions. Consequently, when using Non-Intrusive Testing (NIT) techniques to fulfill Code check valve testing provisions, a baseline inservice test demonstrating disc movement should be conducted to establish acceptance criteria. The baseline test should indicate valve response when known to be operating acceptably (good condition).

It is the owner's responsibility to verify that a valve is operating acceptably at the time the acceptance criteria is established. A disassembly and inspection, radiograph, back-leakage and flow test, or multiple technologies that provide an effective assessment of condition may be used to establish valve condition. Bi-directional testing should be used whenever possible. Information from inspections and tests from similar valves in the check valve group, industry data, operating experience, and maintenance history should also be considered.

The initial test of a check valve's open or closed safety function using NIT techniques that subsequent tests will be assessed against is considered a baseline test. The baseline test shall only be established when the check valve is known to be operating acceptably. It is also the owner's responsibility to qualify the method/technique(s) used for NIT. NIT baseline data for future comparison is best acquired when a valve is new, or has been rebuilt and restored to a "new" condition.

When applying NIT technologies to prove that a check valve is in an acceptable condition, the use of multiple technologies is recommended, provided no technology limitations exist. In addition, combining an open stroke with a close stroke test increases failure detection and provides the optimum effectiveness for failure monitoring. Test conditions should be used that are easily duplicated and provide repeatable results for trending effectiveness.

The "Generic Implications" section of IN 2000-21, states the following:

"If NIT techniques used to verify the opening or closing capability of safety related check valves are not properly qualified and a baseline established for each individual valve when the valve is known to be operating acceptably, potentially inadequate valve performance may be undetectable in the analysis of NIT results."

IN 2000-21 identifies the consequences of not qualifying NIT to safety-related check valve testing.

Diagnose Valve Health/Condition

<u>Performance</u>

The goal of operational/functional testing is to ensure the readiness of a check valve to perform when called upon. The activities associated with operational/functional testing are performed at a given interval to provide an acceptable level of assurance, or confidence level. The combination of testing needs to verify the valve's ability to stroke open and close for maximum failure detection capability.

<u>Condition</u>

The goal of condition-based activities is to ensure that the valve will perform its functions over a predetermined period of time. The activities selected are periodically performed and data collected at specified intervals that will adequately monitor the check valve's condition. The attributes associated with each activity is extracted from the data and analyzed. A monitoring plan must be determined based on many factors; using predictive or inspection condition-based activities, or a combination thereof. The desired outcome is being able to effectively monitor the aging and degradation effects on the check valve that the system imposes on it over time. A thorough diagnosis of valve health is essential for continued reliable operation.

Conclusion

Now that you have a taste for the testing methods, technologies, and methodologies that are available for check valves, you need to know which are trendable and how to apply them. The attachments contain the determination by the Nuclear Industry Check Valve Group as to which technologies or methods are trendable. However, to apply trending to your program we suggest that you read the "Tracking and Trending Guide for Check Valves," published by NIC in 2005, the "Check Valve Analysis Guide" published in May 1999, and the NIC Phase four report "Evaluation of Non-intrusive Diagnostic Examination Technologies for Check Valve Trending" (NIC-04-Trending) published in October 2003.

Also, just because you did not see a specific technology or method that you are using for trending doesn't mean that it is not acceptable or is not trendible. The technologies or methods that were discussed in this paper are those that the Nuclear Industry Check Valve Group deemed to be the most widely used. If you are using other method(s) than those discussed here, we urge you to attend a meeting of the Nuclear Industry Check Valve Group and share your knowledge. Also, further information may be found at the Nuclear Industry Check Valve Group's Website www.checkvalve.org.

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13. INPO Significant Operating Experience Report, 86-03, Check Valve Failures or Degradations, dated October 15, 1986.
14. Appendix II, "Check Valve Condition Monitoring Program," in the ASME OM Code-1995 Edition through 1996 Addenda.
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Trendable Attributes of Operational / Functional Activities

Activity	Attribute Collected	Measured Parameter	Trendable
Full/ Partial Stroke Open	Full Open	Flow rate or change in tank level per unit time	Yes
	Full Open Indication	Flow or P	Pass/ Fail
	Partial Open	Normal Flow or P	Pass/ Fail
	Position Indication	Position (degrees/inches)	Yes
Stroke Timing	Obturator Stroke	Time in seconds	Yes
	Valve Actuation to check valve opening/closing	Time in seconds	Yes
	Pump Start/ Stop to check valve	Time in seconds	Yes
Close/ Backflow	Pressure Profile	System Pressure readings (ΔP)	Yes
	Flow Profile	System Flow readings (flow met in parallel train)	Yes
	Temp Profile	Temperature Gradient (ΔT)	Yes
	Pump Reverse Rotation Check	Shaft rotation	Pass/Fail
	System Parameters	Tank Levels (Δ Level per unit time)	Yes (depends on capability of instruments to detect a problem)
	IR Imaging	Temperature Gradient (ΔT) and viewable image for comparison	Yes (application dependant)
Seat Leak Detection	Seat Leak Detection	RMS Level (with baseline threshold)	Yes
	Airborne Ultrasonics	Decibel	Yes (but not for all applications)

Trendable Attributes of Condition – Based & Predictive Activities

Activity	**Attribute Collected**	**Measured Parameter**	**Trendable**
Internal Inspection	Wear Measurements (valve internals disassembled)	ID or OD of piece/parts	Yes
	Seat / Disc alignment and contact	Blue Check - contact band position and width	Pass/Fail
		Light Check	Pass/Fail
		Feeler Gauge / Waxed Paper	Pass/Fail
	Internal corrosion / erosion / FAC	Wall thickness	Yes
	External corrosion / leakage	Extent / Amount	No
	Manual Stroke Checks	No binding or hanging up	Pass/Fail
	Visual indications and looseness checks (valve not disassembled and alternate measurements used)	Dial indicator used to obtain gap/clearances between moveable parts (e.g., disc post to hinge arm by measuring side to side and up and down movement to determine gap is less than design clearance). There are other alternate means and techniques used to monitor wear that the CVP engineer can use.	Yes
Boroscopic	Visual for wear/physical damage/seat contact for close General condition checks	Visual and recorded description	Pass/Fail
Seat Leakage	Pressure Decay (App J)	Leakage past seat	Note 1
	Mass M/U (App J)	Leakage past seat	Note 1
	PIV	Leakage past seat	Note 2
	Code	Leakage past seat	Note 2

Activity	Attribute Collected	Measured Parameter	Trendable
Acoustics	Time Waveform	Trace Overlay	Yes using comparison analysis
	Frequency (PSD/FFT/ Waterfall)	Shift in frequency content at stable flow	Yes, but valve dependant
	Impacts	Magnitude, Amplitude, and Ringdown Duration	Yes, but valve dependant
	Impact Rate	# per unit time	Yes, but valve dependant
	Audible Noise	Waveform file played back for audible analysis	No
	Event Origin	Time of Arrival	Yes for larger valves.
Magnetics	DC (used with acoustics)	Voltage Delta/Gauss strength	Used for monitoring disc flutter and stroke, but valve dependant
	AC (used with acoustics)	Voltage Delta	Used for monitoring disc flutter and stroke, but valve dependant
Eddy Current	Full stroke voltage delta	Delta Volts	Yes, once verified
	Stroke Time	Seconds	Yes
Ultrasonic (UT)	Disc Angular Velocity	Disc Angular Velocity	Yes
	Disc open angle	Degrees off the seat	Yes
	Disc Flutter	Change in distance per unit time	No, but severity can be monitored
	Confirmation of internals	Manuals scanning A scan presentation	Pass/ Fail

Activity	Attribute Collected	Measured Parameter	Trendable
Radiography (RT)	Radiograph/digital image	Visual record	No
	Dimensional data	Wear in inches or %	Yes
Leak Detection	Acoustics	RMS Level (with baseline threshold)	Yes
	Airborne Ultrasonics	Decibel (dB) level	Yes (not all applications)
Mechanical Exerciser	Breakaway Torque	Force/Friction in ft lbf	Yes
	Full Open Torque	Force/Friction in ft lbf	Yes
	Position	Degrees (°) rotation	Yes
IR Imaging	IR Imaging Temperature Gradient	Temperature (ΔT) and viewable IR image for comparison	Yes
Stroke Timing	Obturator Stroke	Time in seconds	Yes
	Valve Actuation to check valve opening/closing	Time in seconds	Yes
	Pump Start/Stop to check valve opening/closing	Time in seconds	Yes

Notes:

1. Leakage rates under Appendix J are not trendable as a linear progression, but when leakage approaches a valve's alert limit in a step wise fashion, the reason for change should be pursued. Most Appendix J tests have multiple boundary isolations being tested simultaneously, so other actions are taken to determine where the leakage is occurring. Most administrative limits use the Code guidance for determining when to take action. The typical administrative limit uses the Code specified limit of 7.5 times nominal valve size standard ft^3 per day (air).
2. Pressure Isolation Valves (PIVs) are more apt to be individually tested and leakage may be trended. However, with the allowable limit set at 1 gallon per minute (gpm)—trending may not start until the 1 gpm limit is met. A good overall parameter to track is the 24 hour primary leakage rate. Once seat leakage reaches the 0.02 gpm level, the plant Operations Department is reacting to the effects on plant systems. This action level is used to start additional monitoring activities and investigation.

Flow-Induced Vibration Effects on Nuclear Power Plant Components Due to Main Steam Line Valve Singing

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Abstract

Nuclear power plant components can be subjected to strong fluctuating loads and experience unexpected high-cycle fatigue failures while operating at extended power uprate (EPU) conditions. In particular, physical damage has occurred to steam dryers in certain boiling water reactor (BWR) plants during EPU operation, resulting in the generation of loose parts that could interfere with the functionality of safety-related valves and other components. In addition, steam line safety relief valves (SRVs) and a solenoid valve actuator have been damaged by high-cycle vibration during power uprate operation at BWR plants. The objective of this paper is to discuss the source(s) generating these fluctuating high-amplitude loads, present the methods used to estimate these loads, and discuss monitoring of nuclear power plant components to identify potential adverse flow effects. When turbulent flow of a fluid over a cavity formed by one of the SRVs installed on the main steam line (MSL) locks in to acoustic modes within the cavity, high-frequency, high-amplitude pressure fluctuations can be generated. These pressure fluctuations can be estimated by on-line measurement of strains in gages installed on the MSLs. Acoustic models can then be used to estimate the pressure loads on plant components. The potential for adverse flow-induced vibration effects reveals the importance of assessing the impact of EPU conditions on nuclear power plant components and of monitoring the performance of plant components during power ascension to uprate conditions.

Note 1: Staff members of the U.S. Nuclear Regulatory Commission contributed to the preparation of this paper. It may present information that does not currently represent an agreed-upon NRC staff position. NRC has neither approved nor disapproved the technical content.

1 Introduction

Boiling water reactors (BWRs), such as the ones shown in Figure 1 and Figure 2 (from [1]), use reactor cores to boil water. Wet steam emanates from the boiling water and travels vertically through tube banks, called steam separators, to remove moisture. Above the steam separator, a perforated hooded structure, called steam dryer (see images in Figure 3), further removes moisture from the steam. The steam exits the dryer and collects at the top of the reactor pressure vessel (RPV), then flows into one of four main steam lines (MSLs), where it flows to turbines and generates electricity.

Utilities have been using power uprates since the 1970s as a way to increase the power output of their nuclear power plants. As of July 2004, the NRC had completed 101 reviews of power uprate applications, resulting in a gain of approximately 4,183 MWe (megawatts electric) at existing plants, an equivalent of about four additional nuclear power plants. Over the next five years, the utilities plan to ask for additional power uprates, which would add another 947 MWe to the nation's generating capacity.

Power uprates can be classified in three categories:

(1) Measurement–uncertainty recapture-power uprates are power increases of less than 2% and are achieved by the use of enhanced techniques for calculating reactor power.

(2) Stretch power uprates are power increases up to 7% and usually involve changes to instrumentation settings.

(3) Extended power uprates (EPU) are usually greater than stretch power uprates and have been approved for increases as high as 20%. Extended power uprates usually require significant modifications to major pieces of plant equipment, such as the high pressure turbines, condensate pumps and motors, main generators, and/or transformers.

As of June 2004, EPU operation has been approved for 11 BWR plants, with uprates ranging from 6.3% for Monticello in 1998 to 20% for Clinton in 2002. Seven of these plants have not experienced major problems under EPU operating conditions. But the four remaining plants, Quad Cities Units 1 and 2 (QC1 and 2, respectively) and Dresden Units 2 and 3 with uprates in the range of 17 to 18%, have experienced significant increases in flow-induced vibration in the MSLs and within the RPV. The increased vibrations, along with increased fluctuating pressures within the steam, have led to damage of relief valves and steam dryers in the plants.

A summary of the steam dryer failures was issued by the Nuclear Regulatory Commission (NRC) [3]; and GE, the plant designer, has issued a Services Information Letter (SIL) to owners of GE BWR plants [4]. A schematic of the dryer failures, along with accompanying photographs, are reproduced from [4] in Figure 4. In June 2002, a large cover plate on the outside of the original QC2 steam dryer broke off, and pieces of the plate were carried by the steam through the MSLs. Before and after the failure, increases in moisture content in the MSL steam were evident, indicating that large cracks and/or holes in the dryer were allowing wet steam to flow directly into the MSLs. High cycle fatigue was identified as the cause of the dryer failure, and Exelon, the plant owner, installed thicker cover plates and used stronger welds to repair the dryer. However, in May 2003, moisture carryover in the MSL steam increased significantly again, and the plant was shut down in June 2003 so the dryer could be inspected. This time, large cracks had formed through the walls of the dryer outer bank hood. Also, several braces and tie bars on the top of the dryer had cracked.

Cracks had also formed in the steam dryer at the QC1 plant, and in October 2003 the moisture content in the MSL steam increased. In November, QC1 was shut down, and a steam dryer inspection revealed that a portion of the outer bank hood had broken loose (about 16 cm x 23 cm x 1.3 cm).

While the steam dryers in the QC plants were cracking and breaking, valves on the MSLs were also experiencing higher vibration levels at EPU conditions. An electromatic relief valve (ERV) on a QC1 MSL, along with several MSL support clamps and tie-back supports, failed in November 2003. Recently, in January 2006, several ERVs in the QC1 and QC2 plants were found to be degraded (powered relief mode was not available, but spring safety function was available) due to damage induced by strong pressure fluctuations and vibrations [5]. Although steam dryers do not perform safety-related functions, safety relief valves are responsible for relieving reactor overpressure and must remain functional.

Shortly after the first dryer failure in QC2, Exelon began monitoring MSL moisture content more frequently so that any steam dryer damage could be inferred. The GE SIL 644 [4] recommends weekly moisture content monitoring to all BWR owners, along with periodic inspections of the dryers during refueling outages. However, recent efforts by the QC1 and QC2 plant owners and their subcontractors, along with Entergy, who obtained an EPU license amendment for the Vermont Yankee (VY) nuclear power station, have led to more proactive monitoring of the fluctuating pressure levels within MSLs and RPVs, which should identify potential steam dryer fatigue failures (and potential valve failures) before they occur. In this paper, we discuss the new monitoring techniques, along with the mechanisms associated with pressure fluctuations incident on steam dryers and MSL valves in BWR plants.

To summarize the current understanding of the dryer excitation sources, we draw information from documents submitted to the NRC by Entergy and Exelon, which report measurements and simulations of the steam dryer loading in the QC1, QC2, and VY BWRs. All of the information used is from non-proprietary documents in the public domain.

For more information, see the NRC's Agencywide Documents Access and Management System (ADAMS), at www.nrc.gov.

2 Acoustic and Fluid-Dynamic Excitation of Steam Dryers

Several sources cause the pressure fluctuations acting on BWR steam dryers. The steam flowing through and around the dryer is turbulent, and turbulence induces random, oscillating pressures on the dryer surface, with the magnitude of the pressures increasing with flow speed. The turbulent flow also excites large-scale, low-frequency acoustic modes in the RPV steam volume; these modes, in turn, oscillate against the dryer surface. Finally, various acoustic disturbances in the MSLs, some of which are caused by turbulent flow, propagate through the steam in the MSLs and radiate sound into the RPV steam. The radiated sound impinges on the dryer and can be amplified by acoustic modes in the RPV.

To determine the strengths of the various sources causing pressure fluctuations on the steam dryer, Exelon Nuclear instrumented a replacement steam dryer for their QC2 plant with several pressure sensors, as shown in Figure 5 (from [2], [6] and [7]). The pressure sensors were mounted flush with the surface of small metal domes to reduce localized noise induced by small-scale turbulent flow structures passing over the sensors. Power spectral density measurements of pressure at the original licensed thermal power (OLTP) level (790 MWe) and two locations on the dryer (P12 and P24) are shown in Figure 6, reproduced from [7]. Sensor P12 is located on the lower corner of the hood on the 90 degree side of the dryer (see Figure 5), and sensor P24 is mounted about halfway down the skirt. The graphs indicate that a plateau of energy excites the hood and skirt below about 60 Hertz (Hz), which includes periodic peaks due to acoustic modes within the RPV steam and perhaps acoustic modes in the MSL steam columns. Some of the low-frequency peaks are stronger in amplitude in the skirt region.

High-amplitude pressure tones load the dryer near 150 Hz, particularly in the outer hood region. The fluctuating pressure amplitudes are high (about 0.02 psi^2/Hz [pounds per square inch squared per Hertz] at sensor P12) and increase considerably when the QC2 reactor power increases to EPU levels (930 MWe). Figure 7 shows pressure spectra for sensors P12 and P24 at EPU conditions (also reproduced from [7]), and the results indicate that the peak pressure amplitudes on hood sensor P12 increase to 0.3 psi^2/Hz. Table 1 summarizes the peak spectral levels at four hood sensors (two on each of the hoods). The highest peak pressure loads on the dryer at EPU conditions are not at sensor P24, but sensor P21, with levels of about 0.65 psi^2/Hz (~168 dB Re: 20 μPa). Figure 7 indicates that the pressures at frequencies below 60 Hz, however, increase only slightly on the hood, and change little near the skirt.

The peak pressure spectral levels near 151 Hz (on the 90 degree hood) increase in amplitude by a factor of about 15-18 between OLTP and EPU conditions. The peak spectral levels on the 270-degree hood occur at a slightly higher frequency of 157 Hz and increase by a factor of about 7 between OLTP and EPU conditions. Typical broad-band fluctuating pressures in turbulent flow increase proportionally to the square of flow velocity, while pressure spectra (pressure^2) increase with the fourth power of flow velocity. Steam flow velocities increase linearly with plant power, so pressure spectral levels are proportional to the fourth power of plant power. Given a power increase of 18% (930 MWe/790 MWe) and a corresponding flow velocity increase of 18%, the expected increase in pressure spectral level for turbulent flow is about 94%, or a factor of 1.94. The significantly higher increase in the pressure spectral levels near 151 and 157 Hz observed in the measurements (factors of 7 and 15 on the 270 and 90 degree hoods, respectively) indicates that loading mechanisms other than turbulent flow are present in the QC BWRs. We will present evidence later in the paper that attributes the 150 Hz peaks to flow tones induced in MSL valves.

2.1 Low-Frequency Acoustic Resonances of RPV Steam Volume

Reactor pressure vessels are instrumented with water level sensors, which may be used to qualitatively assess the fluctuating pressures in the steam volume. Two sensors are installed in each of the QC plants, offset by 180 degrees, about 45 degrees from the normal directions of the hoods, and located in the skirt regions of the dryers. Figure 8, reproduced from [8], presents the plots of fluctuating pressures within the QC1 RPV steam volume at the two level sensor locations for 790 MWe reactor power level. As with the instrumented QC2 steam dryer pressures, a low-frequency plateau of energy is evident below about 60 Hz, along with a dominant tone slightly above 150 Hz. Additional peaks appear in the low-frequency range of the level sensor data, some of which are likely due to acoustic resonances within the long (about 60 meters) instrument lines between the RPV and the data acquisition system.

Acoustic resonances of the steam volume within the RPV of the VY nuclear power station have been computed by Entergy and its contractors in support of their EPU application to the NRC. Some of the resonances, extracted from a computational fluid dynamics (CFD) compressible flow model and presented in [9], are shown in Figure 9. (Note that the mode shapes only span the steam around the dryer and do not include the hemisphere of steam at the top of the RPV.) The low-order acoustic modes for the RPV steam volume are generally shaped like half- or full-acoustic waves across the RPV diameter and vertically between the water level and top of the RPV. The modes are clearly more active in the annulus between the dryer skirt and the RPV wall, explaining the increased low-frequency acoustic pressures observed in the skirt region of the QC2 dryer.

Note that the frequencies of the VY RPV acoustic resonances do not match those of the QC plants, since the inner diameter of the VY RPV (5.21 m) is smaller than that of the QC RPVs (6.38 m). The QC RPV acoustic resonance frequencies should be about 80% of those of the VY RPV.

2.2 Flow Tones, or "Singing" in MSL Valves

Several valves are connected to the BWR MSLs, including safety relief valves (SRVs) and main steam isolation valves (MSIVs). These valves perform important safety functions. SRVs reduce reactor steam pressure in the event of primary system overpressurization. MSIVs isolate the reactor system in the event of an MSL break outside the containment.

The SRVs are attached to stubbed pipes, or side branches, which extend perpendicular to the MSLs. A short column of steam in the connecting pipe is exposed to the turbulent steam flowing past the valve, and the fundamental acoustic mode of the steam column in the side branch can couple strongly to flow excitation over the stub pipe opening. The MSIVs are a "Y" configuration valve in the main steamline, with the valve disk oriented at about a 45 degree angle off the pipe axis. The flow through the MSIVs can be a strong source of turbulent excitation.

Figure 10 shows a schematic of an SRV excited by a flow tone excitation (courtesy of the Southwest Research Organization [SWRI], at www.swri.edu [10] and also described in a paper by McKee [11]). The steam flow separates at the leading edge of the stub pipe opening and a shear layer forms. At key frequencies, the effective wavelengths of the shear layer vortices match the diameter of the stub pipe opening, leading to strong coherent excitation of the steam cavity within the stub pipe.

A constant dimensionless parameter, called Strouhal number, can be defined for most shear layers as fD/U, where f is the frequency at which the shear layer oscillates, D is the side-branch opening diameter, and U is the steady flow speed of the shear layer. Ziada and Shine [12] measured the characteristic values of the Strouhal number for flow over circular side branches to be about 0.4, but also observed that Strouhal numbers vary with pipe diameter ratio (MSL diameter/branch line diameter), distance from upstream elbows, and acoustic damping. The characteristic value of the Strouhal number can be used to compute frequencies of strong shear layer loading at specific MSL flow speeds and side-branch opening diameters.

The steam columns within all closed side branches (the stubbed pipe in the SRV) have characteristic acoustic resonance frequencies at (2n - 1)c/(4L), where c is the sound speed in the fluid (about 488 m/s for MSL steam), L is the length of the branch, and n is an integer. The mode shapes of the acoustic resonances have a point of maximum pressure at the closed (valve) end of the branch and a point of minimum pressure at the open end of the branch (intersecting with the MSL). The fundamental (n=1) mode shape is a ¼ acoustic wave across the side branch length, with the next (n=2) mode being a ¾ acoustic wave across the side branch length.

[1] To provide context to these dB levels, the threshold of pain in the human ear is at sound pressure levels of about 140 dB, and most eardrums rupture when sound levels reach about 160 dB. Most window glass breaks at pressure levels of about 165 dB, and residential housing begins to fall apart at pressure levels of 170 dB (fluctuating pressures of about 1 psi).

The minimum pressure at the open end corresponds to a maximum in acoustic particle velocity (which is proportional to the pressure gradient) resulting from the velocity fluctuations within the shear layer. Should the frequency of a side-branch acoustic mode coincide (or nearly coincide) with the shear layer frequency, the acoustic oscillations in the side-branch steam can increase the strength of the vortices in the shear layer considerably, which in turn strengthens the acoustic fluctuations. The feedback and subsequent lock-in between the acoustic and shear layer mechanisms, should it occur, leads to extremely high fluctuating pressure amplitudes, commonly referred to as flow tones, or "singing".

Several of the valves in the QC plants operate at locked-in singing conditions, with tones occurring at various frequencies near 150 Hz. Figure 10 (right side) shows the measured vibration response of a valve on a QC2 MSL against increasing plant operating power (from [13]). A tone near 150 Hz first appears at about 740 MWe and increases significantly in amplitude as power increases to 930 MWe (at a far greater rate than the 94% estimated earlier for turbulent flow excitation between OLTP and EPU conditions). Other valves show similar vibration response but at frequencies between 140 and 160 Hz. The frequency differences may be due to slightly different geometries (including upstream geometries, like elbows) and flow speeds. Given a fluid flow speed of about 60 m/s and a stub pipe diameter of about 0.15 m, the characteristic Strouhal number is (157 Hz)(0.15 m)/(60 m/s) ~ 0.39, which is comparable to the values reported by Ziada and Shine [12] .

In the QC plants, the singing within the valves also excites the acoustic modes of the steam columns in the MSLs, which in turn radiate sound directly against the portion of the steam dryer outer hood near the MSL inlets on the RPV. The MSL acoustic pulsations also couple to the volumetric modes of the steam dome in the RPV to drive the steam dryer in regions away from the MSL inlets. A schematic of the loading mechanisms is shown in Figure 11.

The frequencies of the acoustic modes of the steam columns in the MSLs are integer multiples of c/LMSL, where LMSL is the length of the MSL between the RPV and the turbine. Compared to most BWRs, the MSLs in the QC plants are quite short (50-70 m), and the first acoustic modes appear at low frequencies, (488 m/s)/(50-70 m) ~ 7 – 10 Hz . The acoustic wavelength in the MSL steam at the valve singing frequencies near 150 Hz is about (488 m/s)/(150 Hz) ~ 3 m. It is highly likely that the MSL steam column acoustic modes are excited by valve singing at frequencies near 150 Hz. We will next examine this possibility by using measurements of the acoustic pulsations within the MSLs.

2.3 Measurement of Acoustic pressure fluctuations in MSLs

Early during the investigations of QC steam dryer failures, Exelon attempted to use existing plant instrumentation to quantify the fluctuating pressure loads acting on the steam dryers. Venturi line measurements showed the presence of the singing frequencies, but data measured at other frequencies were unreliable due to low signal-to-noise ratio, and corruption of the signal by acoustic modes in the long instrument lines between the venturis and the data acquisition systems.

Later, signals from strain gages mounted to the MSLs were used to infer internal acoustic pressure by relating pressure to the hoop strain on the outer surface of the pipe wall. However, the gages were mounted only at one circumferential location around the pipe and were measuring more than the hoop, or "breathing" motion of the pipe. The cut-on frequency of higher order acoustic modes in the steam is estimated from $1.84c/[\pi D]$, where D is the pipe diameter [14], and are about 620 Hz in the QC MSLs. Below the cut-on frequency, all acoustic motion is due to plane waves, which induce breathing in the pipe wall. However, the pipes are driven dynamically by many other sources, including turbulent flow impinging on elbows, and mechanical sources throughout the reactor.

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* Ziada and Shine report measured Strouhal numbers not only for single side branches, but for pairs of side branches in tandem (on the same side of a pipe) and coaxially aligned (diametrically opposed to each other). Groups of SRVs are often aligned in tandem along the MSLs in BWRs. The load amplification induced by pairs of side branches can exceed significantly that of a single side branch.

* MSL lengths in other BWRs average 140 m, with some MSLs approaching 300 m in length.

At low frequencies, the pipe wall motion is dominated by its bending and ovaling modes, examples of which are shown in Figure 12. Single strain gages mounted to pipe walls cannot distinguish between strains induced by breathing (the signal of interest) and by bending and ovaling. However, uniformly spaced circumferential arrays of strain gages can filter bending and ovaling signals, retaining only the hoop motion. Figure 13 shows strain gage arrays mounted to one of the MSLs in the QC plants (reproduced from [15]). The time signals of four gages, separated by 90 degree circumferential increments, are summed in each array, normalized by the number of gages, and multiplied by a calibration factor converting hoop strain to internal acoustic pressure (hoop strain spectral density [$\mu\xi^2$/Hz] is multiplied by about 1.9 to compute the acoustic pressure spectral density in the QC plants).

An example of the filtered strain spectrum, along with individual strain measurements at the four circumferential locations, is also shown in Figure 13, reproduced from [16]. At most frequencies, the filtered signal is lower than the individual signals, suggesting that most of the pipe vibrations at low frequencies are caused by structural and hydrodynamic forces throughout the plant, rather than acoustic pulsations within the steam. At some frequencies, however, particularly those around the valve singing frequencies near 150 Hz, some individual strain measurements are lower than the filtered signal, indicating that the individual gage was mounted at a location of low local vibration. Therefore, it is not sufficient to measure strain at a single piping location and assume it represents an upper bound on acoustic signals within the MSL.

The strain signals clearly show the acoustic excitation of the MSL steam columns from the singing valves in the QC2 plant at frequencies at and around 150 Hz. The peaks with highest amplitude occur at about 151 and 157 Hz, which is consistent with the peaks in the pressure signals measured on the instrumented steam dryer (Figure 6 and Figure 7). Also, multiplying the peak microstrain spectral measurements (~ 0.03 $\mu\xi^2$/Hz) by the 1.9 conversion factor yields acoustic pressure spectral densities levels of about 0.06 psi^2/Hz, which are similar to the peak levels measured on the dryer (Figure 7).

For the MSL and dryer, the similarity in fluctuating pressure amplitudes near the valve singing frequencies around 150 Hz implies a strong acoustic coupling between the MSL steam and the steam within the RPV volume. Simple acoustic analysis of the sound power radiated by a flanged, or baffled, open-ended pipe [17] may be used to estimate a power transmission coefficient between the acoustic pulsations within the RPV volume and MSL steam columns. The transmission coefficient, τ, may be computed at any frequency f by combining the sound speed c in the steam and MSL pipe radius a with Equation 9.16a from [17]:

$$\tau = \frac{2(ka)^2}{\left[1+\frac{1}{2}(ka)^2\right]^2 + \left(\frac{8}{3\pi}\right)^2 (ka)^2}$$

where k is $2\pi f/c$. The power transmission coefficient is directly proportional to pressure squared and may therefore be used to estimate the ratio of pressure spectra in the RPV and MSL steam. The power transmission coefficient for the QC plants is shown in Figure 14. At low frequencies, the coupling between acoustic pressures in the RPV and MSL steam is weak (at 50 Hz the transmission coefficient is 0.05). However, near 150 Hz, the coupling is much stronger – about 0.34. It is not surprising, therefore, that strong acoustic pressure pulsation at 150 Hz in the MSL steam couples well to the RPV steam volume.

Acoustic modal analysis may be a more useful tool for predicting potential coupling of excitation sources in the MSLs with the acoustic modes in the RPV. Finite element or boundary element models of the entire steam system could be generated and analyzed to determine how specific MSL and RPV modes couple, in some cases, amplifying the source pressures.

Entergy also installed strain gage arrays in the MSLs of the VY plant. A filtered strain spectrum measured in the VY plant at current licensed thermal power (CLTP) conditions (535 MWe) is compared to spectra measured in the QC2

* Higher-order acoustic modes across the MSL cross section occur when the acoustic wavelengths (sound speed/frequency) in the steam become comparable to the pipe diameter. Since wavelengths decrease with increasing frequency, the high-order modes are said to ‘cut-on’ at specific frequencies.

plant at OLTP (790 MWe) and EPU power (930 MWe) in Figure 15 (reproduced from [18]). The VY data show no evidence of singing valves, but peaks similar in nature to those observed in the QC plants are evident at low frequencies of about 24, 35, 47, and 62 Hz (arrows in Figure 15). We will later explain how the peaks are likely associated with the low-order acoustic modes of the RPV volume, which are excited by turbulent flow within the steam dome and near the MSL inlets.

Both Exelon and Entergy have installed two strain gage arrays on each MSL in the Quad Cities and Vermont Yankee plants, so that relative amplitudes and phase delays between the filtered pressure signals in an array pair can be used to determine the strength and direction of acoustic plane wave propagation. However, the spacing between the arrays limits the frequency at which such processing may be used; the array spacing must be larger than half of an acoustic wavelength (sound speed/frequency). In the Quad Cities plants, for example, the spacing between arrays is about 9 m, establishing a lower frequency limit of (488 m/s)/(2x9 m) ~ 27 Hz. Also, at frequencies where integer multiples of half-acoustic wavelengths correspond to the array spacing, the signals may be reduced in amplitude to the point where they are too small to use (this is when node points, or points of zero amplitude in the acoustic waves coincide with the array locations). To resolve this issue, additional arrays with nonuniform spacing between arrays could be considered (logarithmic distributions are popular for measuring acoustic wave propagation in piping systems).

Along with the technical guidance above, we offer the following practical insights regarding vibro-acoustic data acquisition. While installing strain gage arrays (and other instrumentation) in laboratory environments is straightforward, doing so in a commercial nuclear power plant is quite challenging. The harsh plant environment can cause sensor and instrumentation line failures. Also, extraneous noise signals, such as those due to electrical ground loops and auxiliary machinery often appear in the sensor signals, and must be filtered. So that proper phasing between sensors is maintained, all signals must be synchronized and acquired simultaneously. So that sufficient data are acquired to assess acoustic wave amplitudes and phasing, long data records are required, and adequate storage capacity in the data acquisition system must be budgeted for. Computers with memory sufficient to process the large, multiple data records must be used, and software capable of handling large data records must be exercised.

2.4 Turbulent Flow Excitation of Steam Dryers

At low frequencies, turbulent flow emanating from the dryer vanes convects over the top of the dryer and along the hood outer surfaces on its way into the four MSL inlets. The flow speeds within the steam dome are low, on the order of 5-15 m/sec, as shown in the CFD simulation of the VY plant in Figure 16 (the flow speeds in the MSLs are much higher and range from 50 to 70 m/sec for various power plants at EPU conditions). The dominant pressure fluctuations in turbulent flow are concentrated around frequencies associated with the flow speed and characteristic dimension (a constant Strouhal number, fL/U). Slowly moving turbulence induces low-frequency excitation on neighboring structures. As flow speed increases, the frequency of excitation increases (since Strouhal number remains constant), and the amplitudes of the fluctuating pressures increase proportionally to dynamic head (which is proportional to the square of flow velocity). The total fluctuating force (pressure x effective loading area) applied to dryer surfaces by turbulent flow increases with the cube of velocity, as the loading areas over which the pressures are correlated grow proportionally with increasing velocity.

Direct measurements of the fluctuating pressures on the outer surfaces of the QC2 steam dryer (see Figure 6) show that amplitudes at low frequencies are small with respect to those caused by the valve singing near 150 Hz. The low-frequency pressures measured within the MSLs of the QC and VY reactors are also low with respect to those at valve singing frequencies. However, several low-frequency peaks in the VY MSL strain gage data are likely associated with acoustic resonances in the steam dome, where turbulent flow (either over the dryer surface, in the annulus between the dryer and MSL inlets, or at the MSL inlets) excites the steam dome modes. The steam dome modes couple with the acoustic modes in the MSL steam columns, so that they are visible in the MSL strain gage measurements.

The CFD model used to analyze turbulent flow around the VY dryer and within the steam dome [19] included time-accurate modeling of large-scale turbulence, as well as compressibility effects, so that acoustic modes of the volumes were effectively included in the simulation. Also included in the simulation was the coupling between the turbulent flow and the acoustic modes (the quantitative coupling was not modeled accurately, however, due to computational constraints on the time step used in the simulations). In spite of the modeling inaccuracies, the simulations may be used to qualitatively analyze the coupling of flow turbulence and steam dome acoustic modes.

Figure 17 shows the pressures simulated by the compressible, time-accurate CFD analysis at various locations on the VY steam dryer. The three strong peaks at 32, 46, and 62 Hz are caused by turbulent flow excitation of low-frequency acoustic resonances in the steam dome (there are also weaker peaks at 17 and 22 Hz). Mode shapes reproduced from [9] are also shown in the figure and were computed using the CFD model. The peak frequencies agree well with those measured in the MSLs of the VY plant using the strain gage arrays (Figure 15) *. The CFD model was not employed to find the acoustic modes at the lower frequencies (17 and 22 Hz). The excellent qualitative agreement between the VY pressure measurements in the MSLs and simulated pressures on the dryer is encouraging and provides hope that strong acoustic pulsations within the steam dome can be measured in the MSLs.

* There is, however, the possibility that the low-frequency peaks in the measurements shown in Figure 17 are due not to steam dome modes, but to acoustic modes within the MSL steam columns. If this is the case, further study into how well low-frequency steam dome acoustic modes couple to MSL steam columns should be conducted.

3 Conclusions

Motivated by repeated structural fatigue failures of steam dryers in the Quad Cities BWR plants while operating at EPU conditions, Exelon Nuclear and GE designed stronger steam dryers and instrumented one of them (installed in the QC2 plant) with arrays of pressure transducers. Measurements of the pressures on the dryers, combined with measurements of pressures within the RPV and MSLs, revealed that strong acoustic tones emanating from the SRVs are propagating through the steam in the MSLs into the RPV steam, loading the dryer at several frequencies near 150 Hz. The fluctuating pressure amplitudes approach 1 psi, which is extremely high. Also, low-frequency excitation caused by steam flow turbulence around the dryer and MSL inlets, amplified slightly by acoustic modes within the RPV, is evident in the measured pressure spectra.

Based on the Exelon measurements and other studies performed by Entergy in support of its EPU application for the VY BWR, the current understanding of steam dryer fluctuating loads is summarized in Table 2, along with their propagation paths and means of detection. Most of the dominant pressure loads on the dryer can be detected with level sensors currently installed in the RPV and with circumferential strain gage arrays installed on the outer surfaces of the MSLs (at locations close to the RPV).

Low-frequency fluctuating loads induced by turbulence near the dryer surface are not generally detectable by remote sensors, unless the turbulence couples strongly with acoustic modes within the RPV. Usually, these direct loads are low in amplitude and do not induce fatigue cracking. All measurements and simulations to date have focused on frequencies below 200 Hz. Other high-frequency excitation sources may exist, such as singing of other valves within or downstream of the MSLs. Should such excitation occur, it would be visible in the measurements of the MSL strain gage array.

A frequently asked question is: why are MSL valves singing and steam dryers failing only in the QC1 and QC2 plants, and not at other BWRs already operating at EPU conditions? SRV singing is related directly to MSL steam flow speed and stub pipe diameter and length. The MSL diameters in the QC and Dresden reactors are smaller in proportion to their RPV diameters and rated power compared with most other BWR/3 plants, causing higher MSL flow speeds (see Table 3). The Dresden plants have smaller stub pipe diameters than the QC plants, such that valve singing occurs at power levels lower than EPU conditions (at about 78% of OLTP). Other BWRs with larger MSL diameters may encounter flow-tone problems if the flow velocities within the MSLs are increased to the range where the standpipes are excited.

A key conclusion from these studies is that singing assessments of valves in the MSLs are important for BWR plants considering implementation of an EPU to ensure that strong acoustic excitation does not occur. Also, monitoring of MSL acoustic pressures (such as through the use of MSL strain gages)is important for BWR plants during power ascension from OLTP to EPU conditions to detect the onset of any flow tones within the valves. If a tone occurs, its potential impact on valves (and other MSL components) and the steam dryer needs to be assessed.

Since remote monitoring approaches can only infer integrity of MSL components and the steam dryer, periodic inspection is important in identifying degradation and ensuring long-term component integrity from EPU operation. Walkdowns of MSL components and enhanced visual inspections (EVT-1) of steam dryers during refueling outages are highly beneficial in identifying degradation. Should unexpected fatigue-related damage occur, more study into excitation mechanisms would be warranted, along with more frequent and enhanced visual inspections of MSL components and the steam dryer.

4 Acknowledgements

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Table 1. Peak spectral levels and frequencies at selected instrumented QC2 steam dryer locations at OLTP (790 MWe) and EPU (930) plant power levels (from [7]).

	90 degree hood		270 degree hood	
	P3	**P12**	**P20**	**P21**
Peak Frequency (Hz)	157	157	151	151
Spectral level at 790 MWe (psi^2/Hz)	0.012	0.020	0.025	0.090
Spectral level at 930 MWe (psi^2/Hz)	0.220	0.300	0.160	0.650
Ratio of spectral levels (930 MWe/790 MWe)	**18.3**	**15.0**	**6.4**	**7.2**

Table 2. Overview of fluctuating pressure sources acting on BWR steam dryers.

Frequency	**Cause**	**Source Propagation**	**Source Detection**
Very Low (below 80 Hz)	Turbulant flow over dryer	Directly incident on the dryer	Directly on the dryer (not generally available)
Low (below 80 Hz)	Turbulent flow over dryer	Into low-frequency acoustic modes of the RPV steam volume, which pulsate against the dryer and against the entrances to the MSLs	In the RPV level sensors, and in the MSLs
Mid (80 to 200 Hz)	Turbulent flow and flow instabilities (shear layers) in MSLs coupling to acoustic modes in valve standoff pipes	Into low and mid-frequency acoustic modes in the steam columns within the MSLs, which couple to RPV steam volume modes, which pulsate against the dryer	In the MSLs and in the RPV level sensors
High (above 200 Hz)	Unknown	Unknown	Unknown

Table 3. QC and VY BWR dimensions and parameters.

Quantity	Quad Cities and Dresden	VY
Pressure (MPa)	6.9 - 7.3	6.9 - 7.3
Temperature (degrees C)	282	282
Density (kg./m^3)	36	36
Dynamic Viscosity (Pa-s)	1.9E-5	1.9E-5
Sound Speed (m/s)	488	488
MSL Steam Velocity (m/s)	52 at OLTP (790 MWe) 61 at EPU (930 MWe)	42 at CLTP (535 MWe), 51 at EPU (642 MWe)
MSL Pipe Outer Diameter (m/in.)	0.51 m / 20 in.	0.46 m / 18 in.
MSL Pipe Inner Diameter (m/in.)	0.46 m / 17.9 in.	0.41 m / 16.1 in.
SRV Stub Pipe Diameter (m/in.)	0.146 m / 5.76 in. (QC) 0.117 m / 4.63 in. (Dresden)	0.132 m / 5.18 in.
RPV Inner Diameter (m/in.)	6.38 m / 251 in.	5.21 m / 205 in.

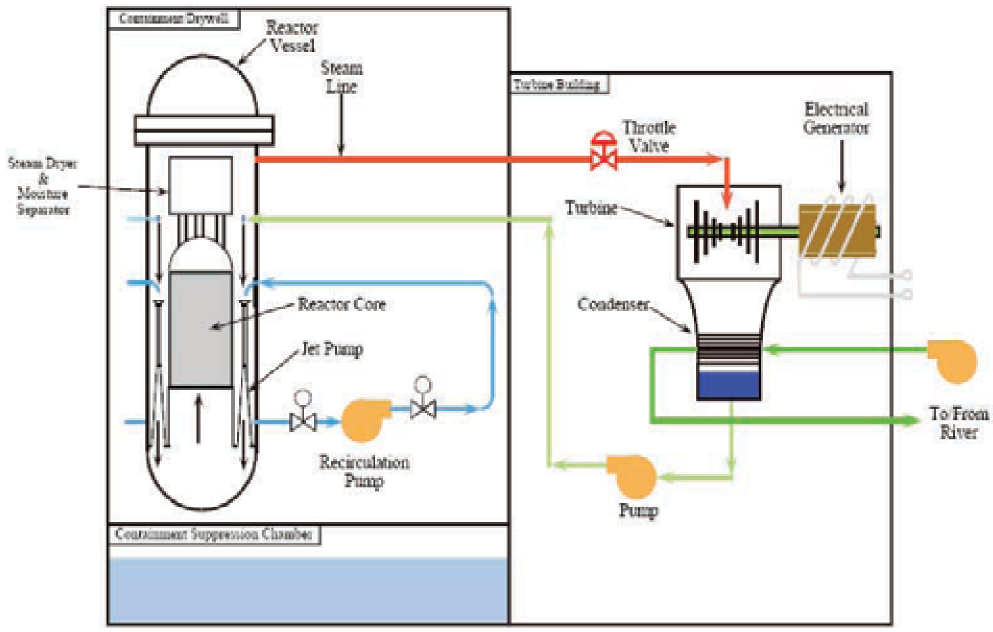

Figure 1 – Schematic of BWR components (from [1]).

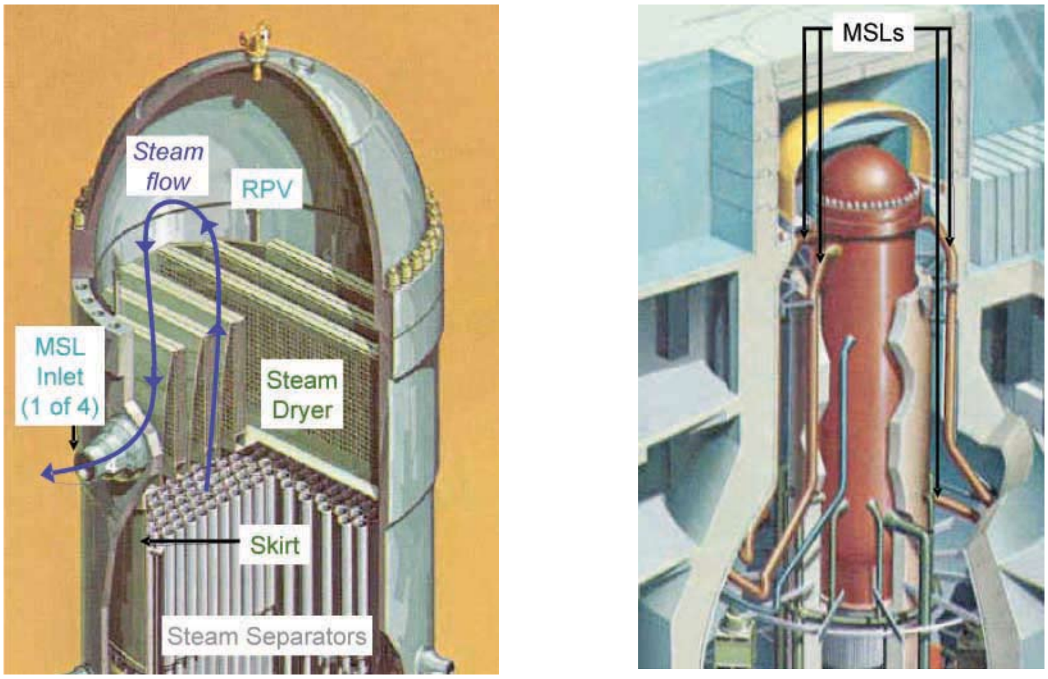

Figure 2 – Artist renderings of BWR with annotations by current authors: left - steam dryer within RPV, right - RPV and MSLs surrounded by containment structures (both images from [1]).

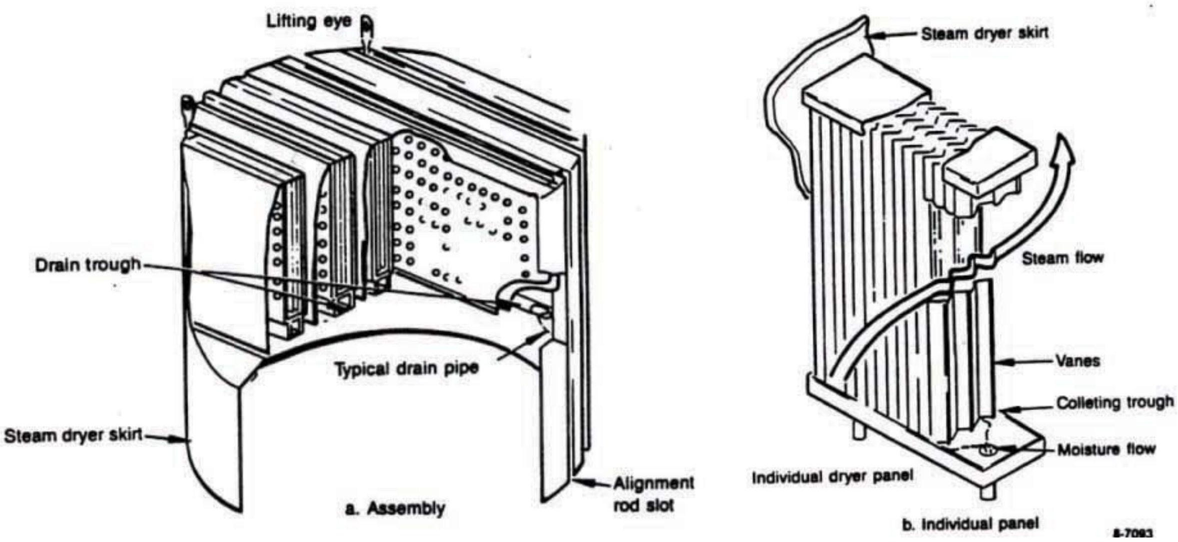

Figure 3 – Schematic of typical original BWR steam dryer: left – assembly, right – single panel.

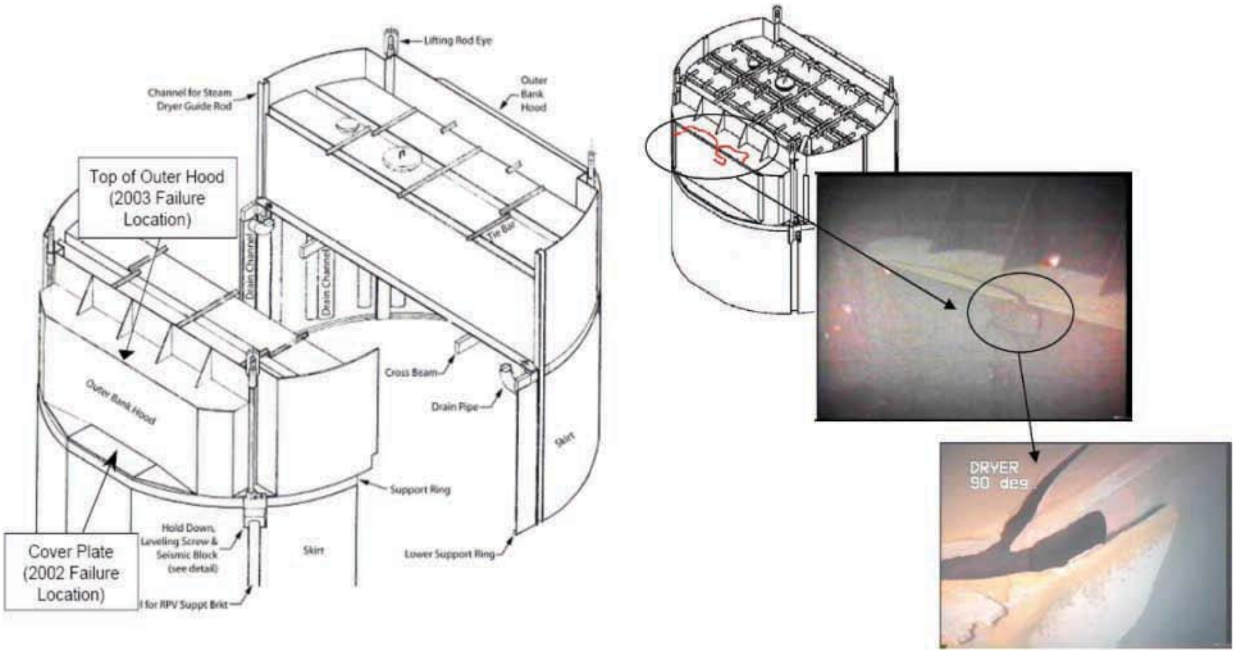

Figure 4 – Structural failures of Quad Cities 2 original steam dryer, from [4].

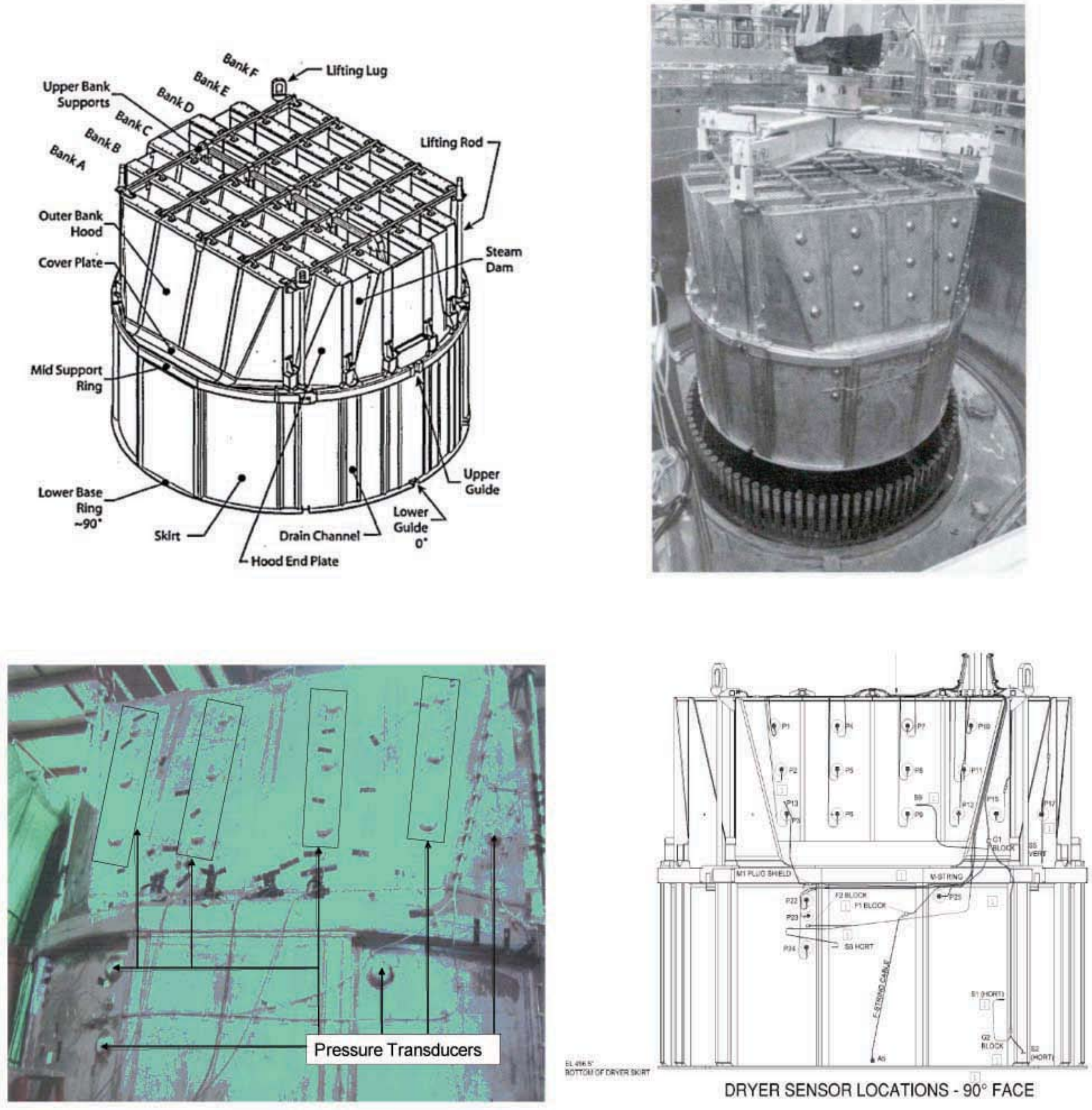

Figure 5 – Replacement steam dryer for QC1 and QC2: top left – schematic from [2]; top right – photograph taken during installation in QC2 (note that the dryer is rotated about 90 degrees between the images); bottom left – photograph of instrumented replacement steam dryer in QC2 plant - from [6]; bottom right – schematic of instrumentation - from [7].

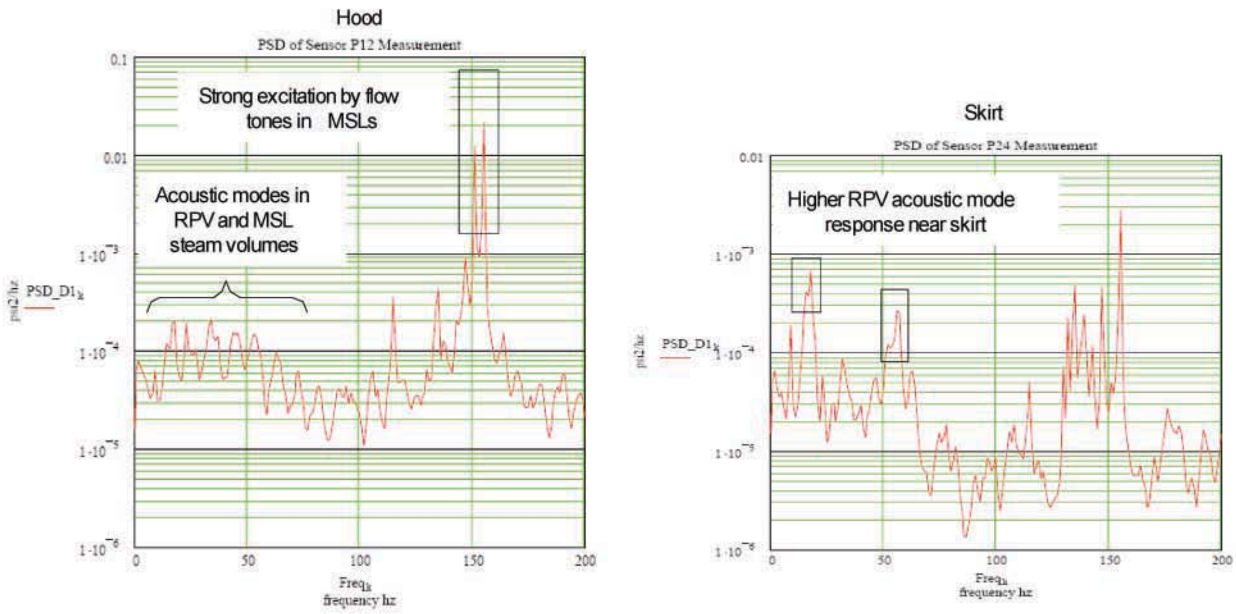

Figure 6 – Pressure spectral densities measured on hood of instrumented steam dryer in QC2 plant (from [7]) at 790 MWe. Sensor P12 is on lower corner of hood, and sensor P24 is on the skirt.

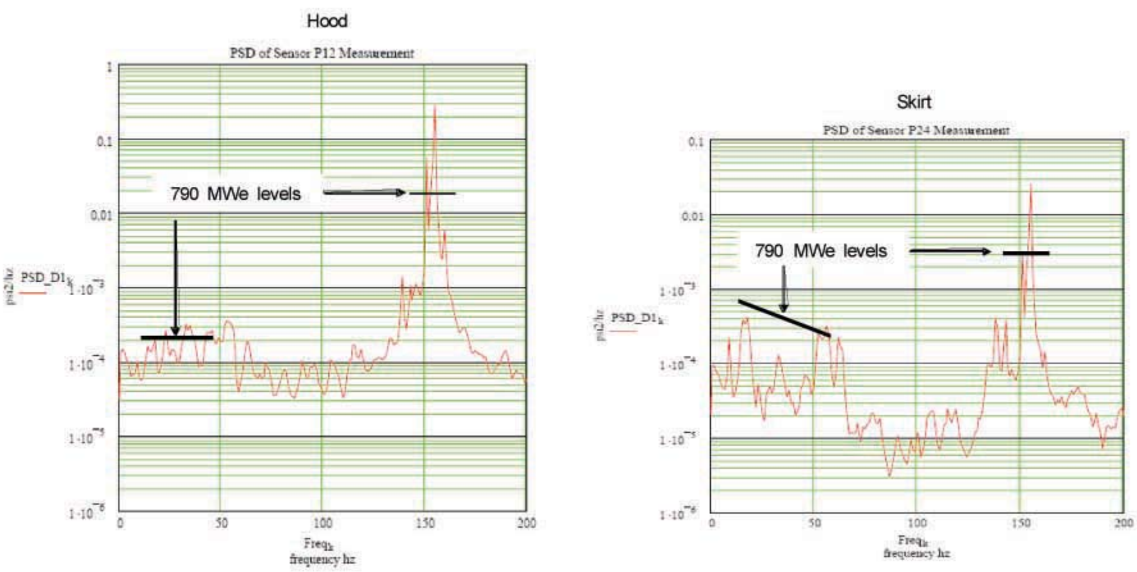

Figure 7 – Pressure spectral densities measured on hood of instrumented steam dryer in QC2 plant (from [7]) at 930 MWe, along with approximate peak levels at low (below 50 Hz) and high (near 150 Hz) frequencies at 790 MWe. Sensor P12 is on lower corner of hood, and sensor P24 is on the skirt.

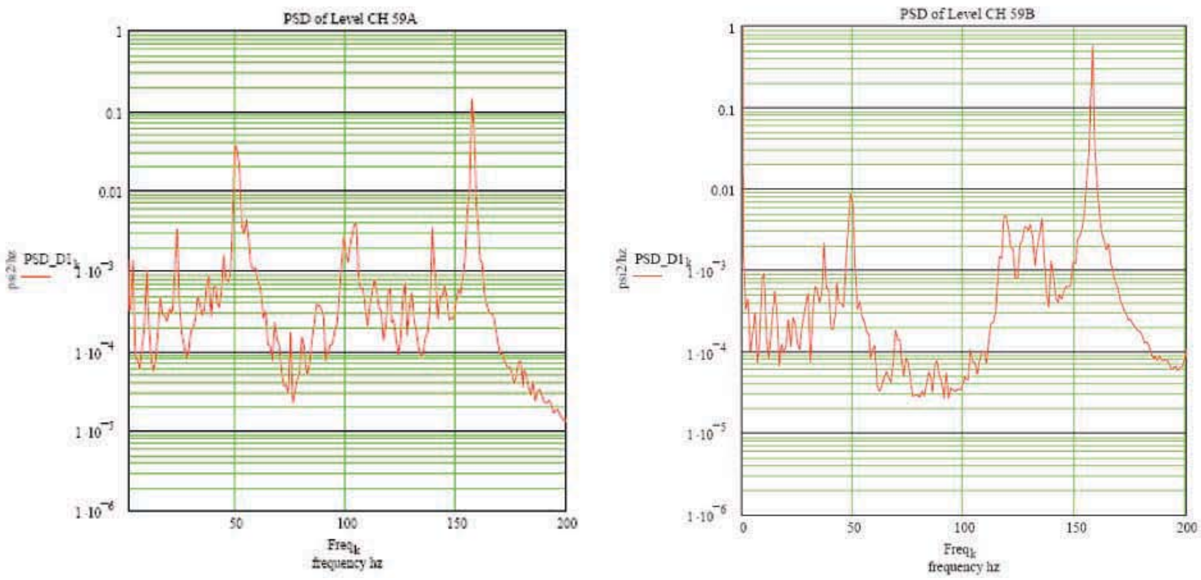

Figure 8 – Pressure spectral densities measured in RPV level instruments near skirts of steam dryers in QC1 plant (from [8]) at OLTP (790 MWe).

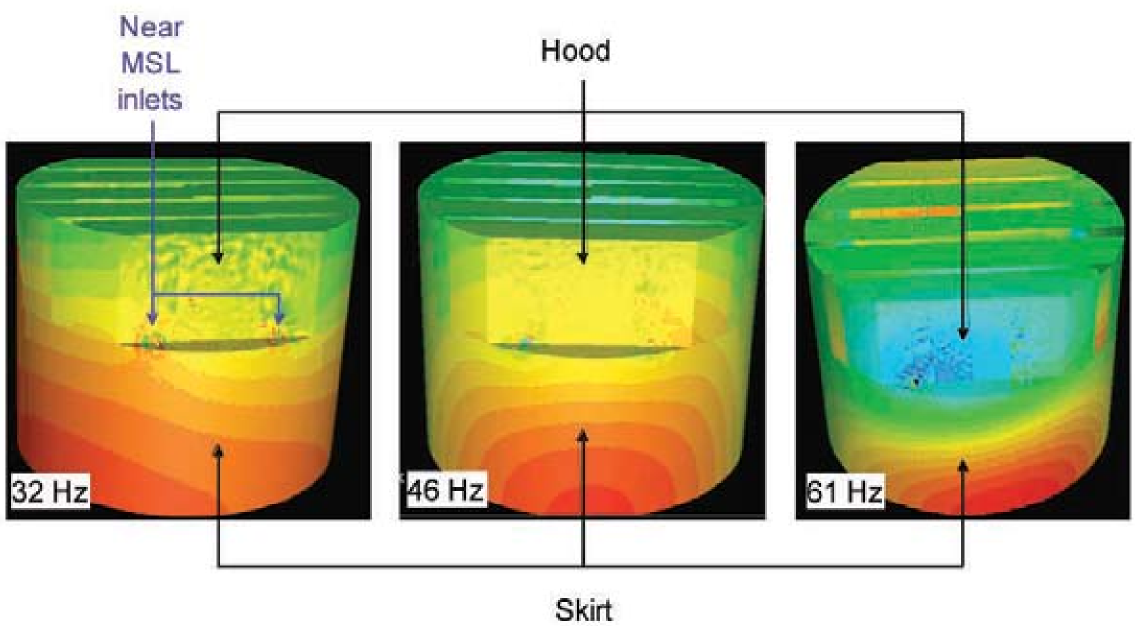

Figure 9 – Simulated RPV steam volume acoustic modes in VY BWR, computed by Entergy [9]. Top half-spherical section of RPV volume has been truncated from the mode shapes.

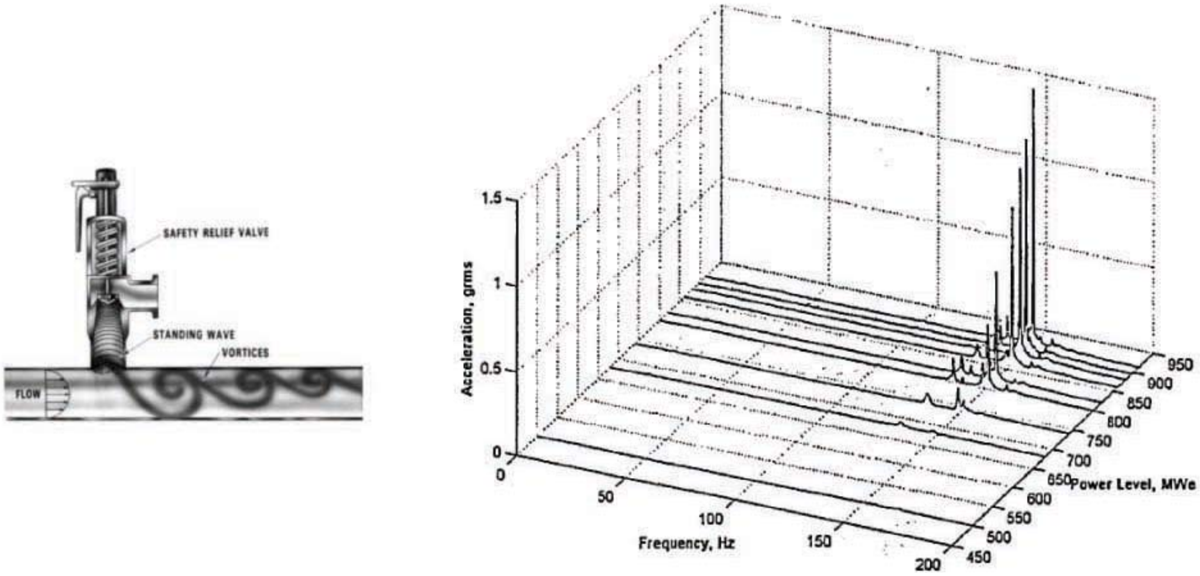

Figure 10 – Left – artist rendering of a "singing" safety relief valve (from [10] and [11]); Right – plot of acceleration measurements (perpendicular to pipe) for Quad Cities 2 electro-matic relief valve (ERV) 3D inlet flange at varying plant power levels (from [13]).

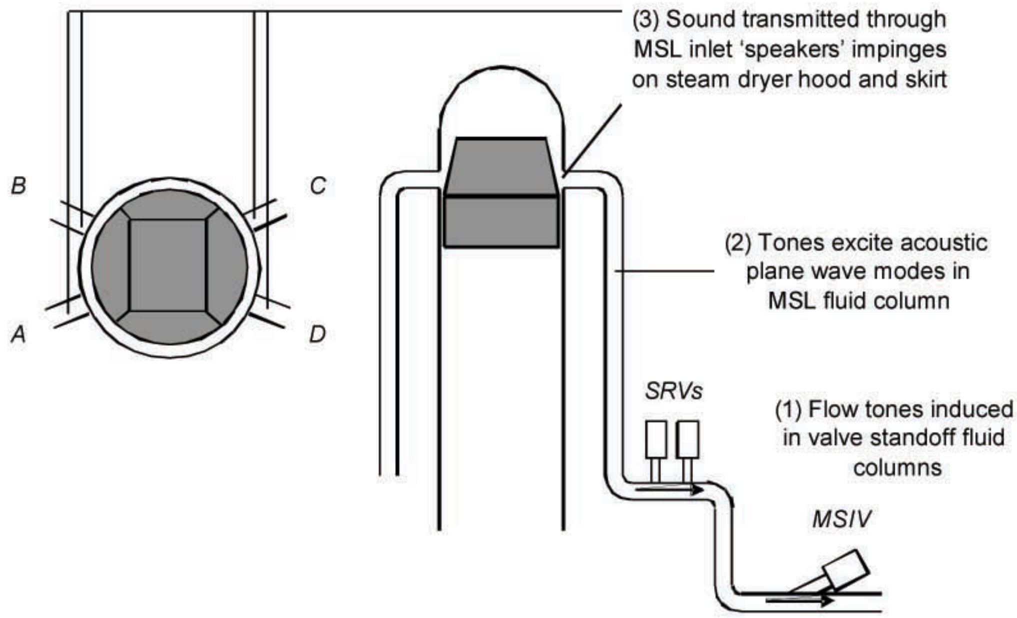

Figure 11 – Valve flow tone excitation of MSL fluid columns and of steam dryer.

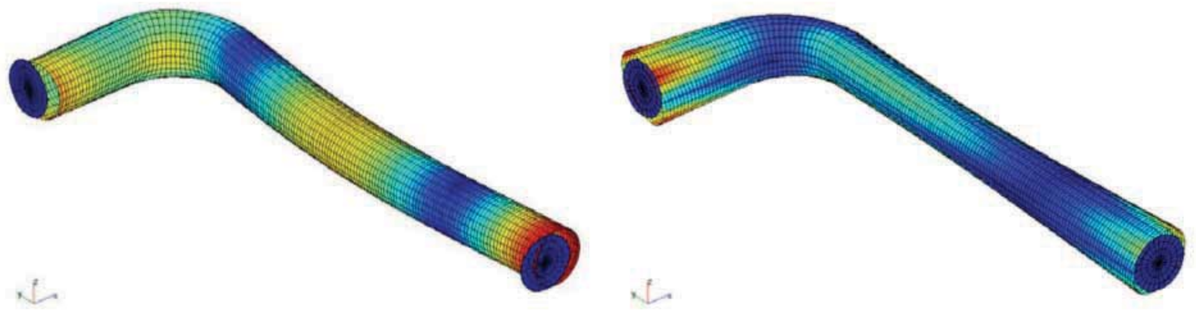

Figure 12 – Typical pipe bending (left) and ovaling (right) modes of vibration. Stationary end "plates" were added to aid in mode shape visualization.

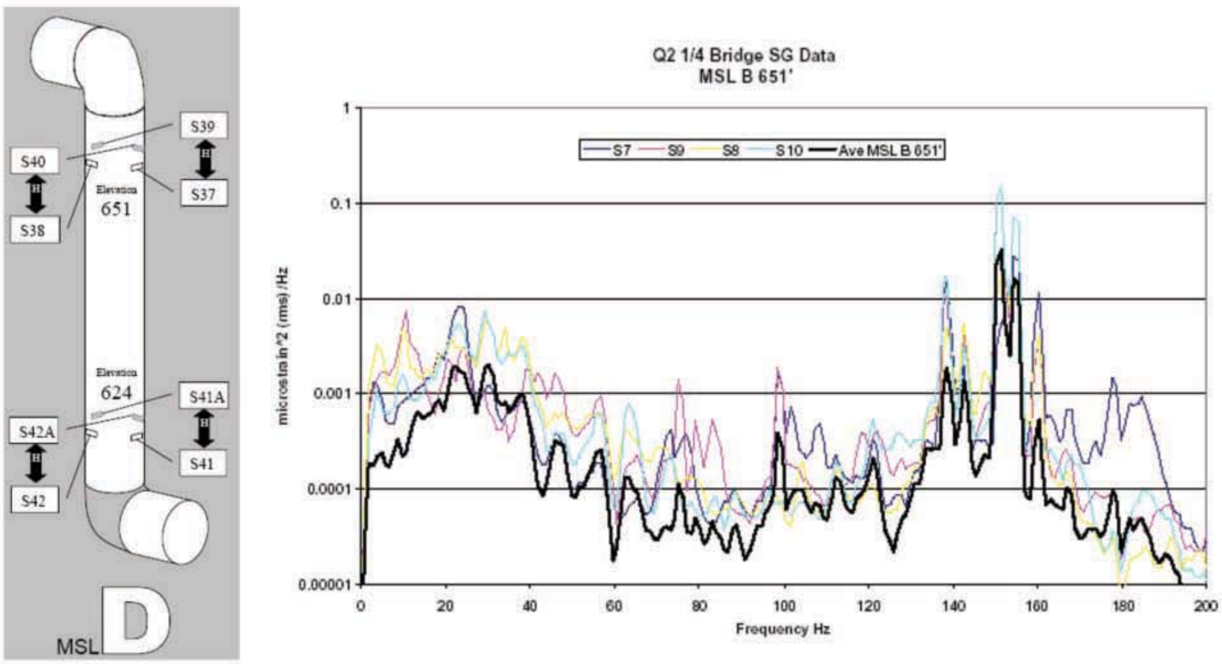

Figure 13 – Left – strain gage arrays installed on MSL in Quad Cities BWR (from [15]); Right - strains measured by strain gage array elements and averaged array on MSL B of the QC2 plant at EPU conditions, from [16]. The average (Ave) spectrum is directly related to the acoustic pressure levels in the steam within the MSL.

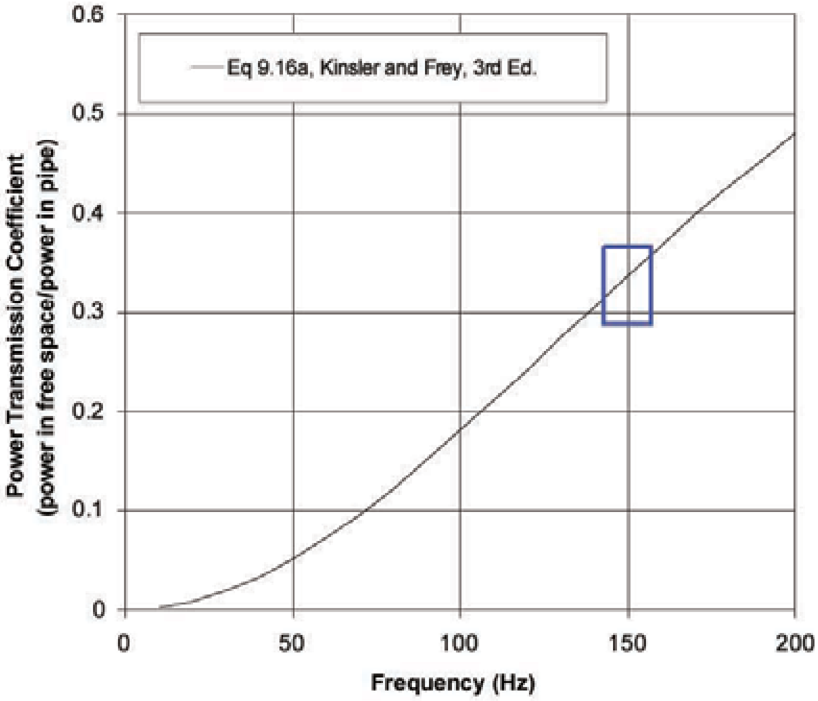

Figure 14 – Power transmission coefficient between RPV and MSL. Steam pipe radius=0.25 m, steam sound speed=488 m/s. Box indicates frequencies of valve singing.

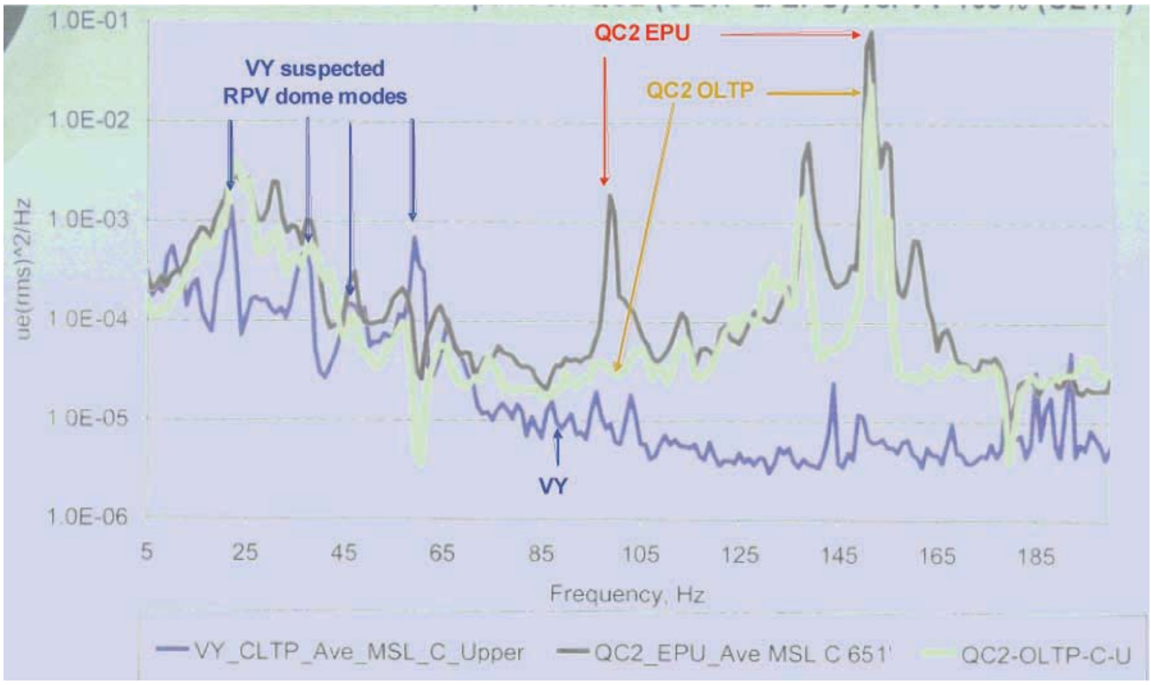

Figure 15 – Averaged strain gage measurements on MSLs at CLTP in the VY plant (lower curve) and EPU (upper curve) and OLTP (middle curve) in the QC2 power plant (from [18]).

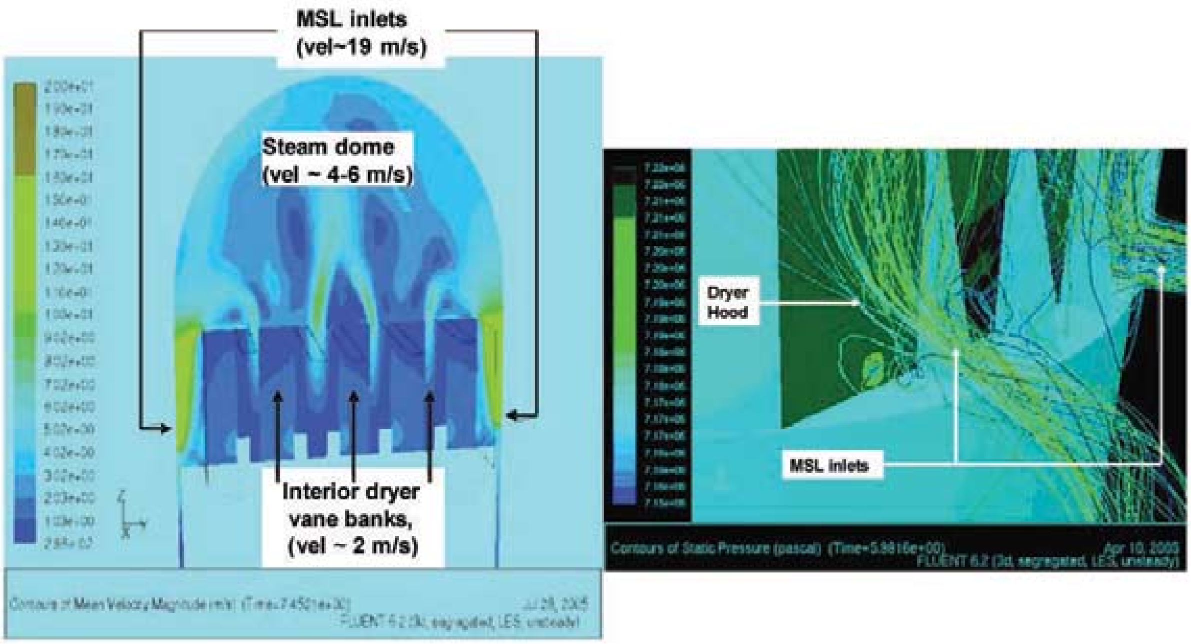

Figure 16 – Left - CFD simulation of flow speeds in VY steam dome (from [9]); Right - flow streamlines into MSL inlets colored with contours of instantaneous pressure (from [19]).

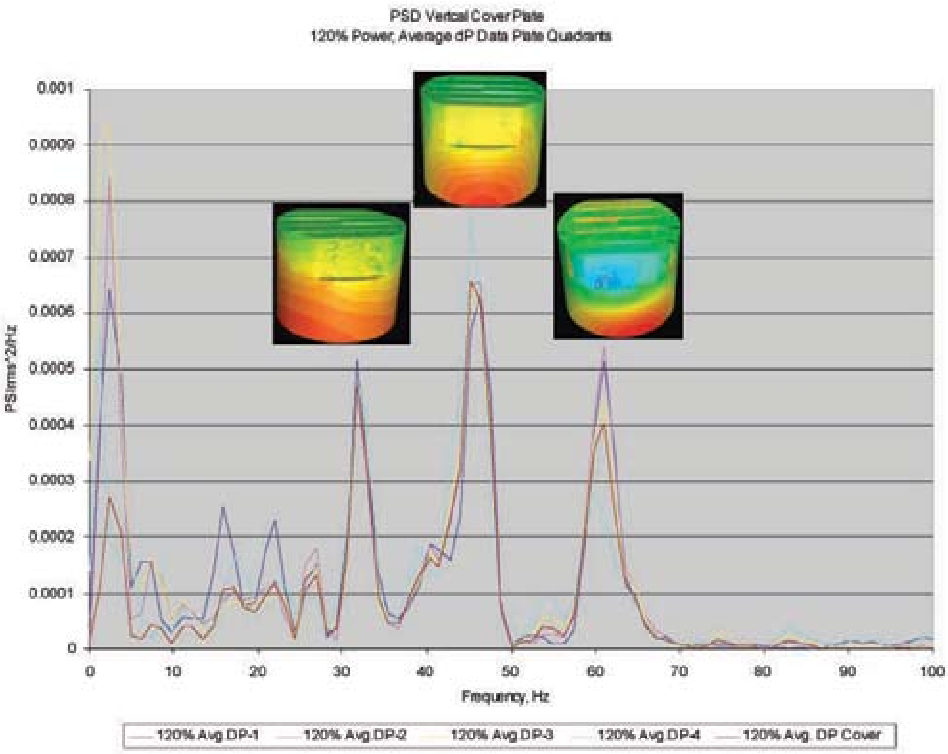

Figure 17 – Pressure loading on VY steam dryer hood estimated using CFD simulations (from [20]), supplemented with acoustic mode shapes of the steam dome volume computed from the CFD model (from [9])

Use of MCC-Based Motor Torque Measurements for Periodic Verification of Motor-Operated Valves

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Abstract

Historically, diagnostic testing of motor-operated valves (MOVs) for periodic verification (PV) has been conducted using at-the-valve tests. Although nuclear power plants have recognized the potential benefits of PV testing conducted at the motor control center (MCC), there is a lack of validated methods for use of MCC-based measurements in PV.

This paper summarizes work funded by Electric Power Research Institute (EPRI) to develop, justify and validate a methodology for use of MCC-based measurements (most importantly, motor torque) in PV of MOVs. The MCC-based Motor Torque Periodic Verification (MTPV) method is applicable to torque-switch controlled closing strokes of rising stem MOVs with AC motors.

The MTPV method uses a baseline "parallel" test with simultaneous motor torque measurements (at the MCC) and stem thrust measurements (at the valve), to determine a relationship between motor torque and stem thrust. Upper and lower thrust limits are converted to motor torque limits using this relationship, with appropriate consideration of uncertainties. Motor torque data from subsequent tests is compared to these motor torque limits to verify adequate setup and to determine margin.

The MTPV method is validated using data from tests of 4 MOVs at a nuclear power plant. For these MOVs, a second "parallel" test provided the necessary data to evaluate how well MCC-based measurements predict stem thrust. For all 4 MOVs, the predicted thrust based on measured motor torque matched the measured stem thrust favorably. Variations were well within measurement uncertainty.

Further, the validation cases showed that the apparent margin is lower with MCC-based measurements compared to at-the-valve measurements, due to greater measurement uncertainty. Accordingly, the MTPV method will be most beneficial for MOVs that have high margin with at-the-valve measurements (> 40%).

Background

In 1989, the US Nuclear Regulatory Commission (NRC) issued Generic Letter 89-10, Safety-Related Motor-Operated Valve Testing and Surveillance, which requested that nuclear power plant licensees review and validate design basis requirements for safety-related motor-operated valves (MOVs) to ensure that these MOVs were capable of performing their required safety-related functions. To ensure continued reliability of safety-related MOVs, the NRC later issued Generic Letter 96-05, Periodic Verification of Design-Basis Capability of Safety-Related Motor-Operated Valves, which requested that facilities develop a periodic verification program to address potential valve and/or actuator degradation.

One component of a successful periodic verification program is regular diagnostic testing of MOVs. Historically, this type of testing has required access to the valve for installation of transducers and other equipment necessary to assess valve performance. This process is time-consuming and limited by accessibility to the plant's MOVs. Although technologies are available that allow diagnostic testing to be performed from a remote location at the motor control center (MCC), this type of diagnostic testing has not been implemented at many sites because widely accepted methods for use of MCC-based testing within a periodic verification program have not been defined.

This paper summarizes work funded by Electric Power Research Institute (EPRI) to develop, justify and validate an MCC-based Motor Torque Periodic Verification (MTPV) method for torque-switch controlled closing strokes of rising stem MOVs, with AC motors. The paper provides a summary of the evaluation of motor torque data obtained from electrical measurements at the MCC and covers use of these (and other) measurements in MOV PV testing.

It is important to note that this work focused on the evaluation of measured motor torque data as it pertains to an MOV's upper and lower setpoint limits. This paper does not address how to measure motor torque from the MCC. Motor torque is assumed to be measured using a vendor-provided diagnostic system with a justified measurement uncertainty. Justification of motor torque measurement is the responsibility of the user (and the diagnostic equipment vendor), and is not included in this report.

Overview

The MTPV method is an approach for comparing measurements of motor torque that are taken at the MCC to pre-determined limits to assess the operational margin of an MOV. The general procedure is analogous to current PV methods based on direct stem thrust measurements:

1. Minimum and maximum "raw" limits are calculated. The minimum limit is based on the required thrust to actuate the valve under its design basis conditions and the maximum limit is based on the load capability of the valve, actuator, and motor.
2. Test equipment accuracy, torque switch repeatability, and other uncertainties are accounted for and used to develop "adjusted" limits.
3. Data is acquired from a test to verify that the measured values fall between the adjusted limits and to quantify the operational margin.

The MTPV method requires a baseline "parallel" test which includes MCC-based motor torque measurements and direct stem thrust measurements from sensors at the valve. Results from this test are used to determine parameters needed to interpret data from subsequent PV tests where measurements are made only at the MCC. All testing (baseline and subsequent tests) is performed with no flow, pressure or DP in the pipe (referred to as "static" testing).

In the MTPV method, motor torque upper and lower limits are determined based on information from the baseline test. These limits are adjusted to account for uncertainties such as test equipment accuracy, torque switch repeatability, etc. In subsequent tests, measured motor torque at control switch trip (CST) is compared to these limits to verify that the setup of the MOV is acceptable, and to quantify the margin for successful operation.

Figure 1 provides a graphical overview of the MTPV method. The left side of the figure shows how limits and margin are evaluated for measurements of stem thrust. The right side of the figure shows how limits are evaluated using measurements of motor torque. Details of this figure are described under "Implementation."

Applicability

The MTPV method is applicable to torque-switch controlled closing strokes of rising stem MOVs with AC motors. Use of the MTPV method beyond these conditions (e.g., limit-switch controlled strokes and opening strokes) has not been validated. Accordingly, users have the responsibility to justify and validate the method for conditions beyond those described in this paper.

Implementation

This section outlines the approach for implementation of the Motor Torque Periodic Verification method. The discussion provides a summary of the methods for (a) analysis of baseline "parallel" test data, (b) development of acceptable upper and lower motor torque limits, and (c) analysis of subsequent MCC-only test data, including determination of margin.

Figure 2 is a flow chart of the process to implement the MTPV method.

Evaluation of Baseline Parallel Test Data

As discussed above, the MTPV method requires an initial valve test (baseline test) which records data simultaneously at (a) the MCC, to determine motor torque and other data (e.g., switch actuation), and (b) the MOV, to determine stem thrust. This parallel test data is used to develop key parameters which relate motor torque to stem thrust for the tested valve. These parameters are needed to establish the minimum and maximum MTPV limits and are discussed further below.

Motor Torque Hotel Load

Motor torque hotel load is the motor torque required to engage the actuator gearing and stem nut, without any load on the stem (i.e., zero stem thrust and stem torque). This load is typically determined from diagnostic testing during the portion of the stroke when the stem nut rotates through its clearance with the stem threads (see Figure 3). As shown in Figure 1, hotel load acts as an "offset" in measurement of motor torque (i.e., hotel load is the small portion of motor torque that is not effective in generating thrust).

Inertial Thrust

Inertial thrust is the additional stem thrust developed after control switch trip due to the inertia of moving parts, primarily the motor. The inertia value from the baseline parallel test of record is used in establishing the upper mechanical limit.

MOV Factor at CST

The MOV Factor is a ratio of measured motor torque (above hotel load) to measured stem thrust, as determined from the baseline parallel test, at control switch trip. This ratio is affected by the stem factor, overall actuator ratio and actuator efficiency. The relationship between MOV factor and these parameters can be expressed as (the terms in the equation are detailed under "Nomenclature" at the end of this paper),

$$F_{MOV} = \frac{(MT_{MEAN,CST} - MT_{HOTEL})}{TH_{CST}} = \frac{(FS)}{(OAR)(EFF)} \quad (1)$$

This value is used as the conversion factor between stem thrust limits and motor torque limits, as shown in Figure 1.

It is important to note that the MOV Factor is based on the Mean Motor Torque at CST for the baseline parallel test. Parallel test data from MOVs often show oscillations in measured motor torque near control switch trip (see Figure 4). However, these data do not exhibit similar oscillations in the measured stem thrust signal indicating that stem thrust is insensitive to these variations (see Figure 5). As such, the MOV Factor, which defines the relationship between measured motor torque and stem thrust, should be determined based on the mean motor torque signal at CST ($MT_{MEAN, CST}$).

Determination of Upper and Lower Motor Torque Limits

For the MTPV method, raw minimum and maximum limits are based on existing stem thrust limits which plants have previously established as part of their MOV programs. These thrust limits are converted to raw upper and lower motor torque limits using the MOV Factor determined from the baseline parallel test, and then adjusted to address sources of uncertainty.

The Upper Motor Torque Limit is the most limiting (i.e., lowest value) of the MOV mechanical limit and the reduced voltage motor torque capability (both adjusted for uncertainties). The MOV mechanical limit is based on of the actuator's thrust and torque ratings and the valve's maximum allowable thrust, whichever is most limiting. This mechanical limit is then adjusted to remove inertia (which is not measured by the MCC-based motor torque signal) and to account for uncertainties, as shown in Figure 1. The reduced voltage motor torque capability is typically calculated using the following equation[2].

$$MT_{VRED} = (MT_{NOM})\left(\frac{V_{RED}}{V_{NOM}}\right)^2 \quad (2)$$

This value of motor torque at reduced voltage is adjusted to account for uncertainties, as shown in Figure 1.

The Lower Motor Torque Limit is based on the tested MOV's required thrust at CST, adjusted for uncertainties.

As discussed above, the raw motor torque limits need to be adjusted appropriately for uncertainties to determine the adjusted upper and lower motor torque limits. These uncertainties may include (but are not limited to)[3] :

- torque switch repeatability
- thrust measurement uncertainty
- motor torque measurement uncertainty
- stem factor uncertainty
- actuator efficiency uncertainty
- inertial thrust uncertainty
- rate of loading (ROL)

[1] Equation (1) is similar to the Limitorque sizing equation (Reference 1), except that Equation 1 accounts explicitly for hotel load and the Limitorque equation uses an Application Factor.

Most of these uncertainties are identified within existing plant MOV programs. In-plant MOV data was used to justify values for those uncertainties which are not typically quantified (i.e., actuator efficiency variation and inertial thrust variation). Plants may use the EPRI-justified values for these uncertainties or may elect to justify their own values for these terms.

As shown in Figure 1, the gap between the upper and lower motor torque limits is likely to be narrower than the gap between stem thrust limits developed for direct thrust measurements. This difference is due to the additional uncertainties associated with the MTPV method, the most significant of which is motor torque measurement uncertainty.

Evaluation of MCC-Only Test Data

Periodic verification tests subsequent to the initial baseline test need only obtain measurements at the MCC. During these tests, the MCC measured motor torque at control switch trip (CST) is compared to the upper and lower motor torque limits (see Figure 6).

Evaluation at Lower Limit and Calculation of Operational Margin

If the measured Mean Motor Torque at CST is greater than the lower limit, the valve is assured to have positive operational margin. The margin can then be quantified using the equation below and the resulting value fed back into the valve's PV program. This determination of margin is consistent with the margin definition within the Joint Owners' Group PV Program (Reference 3).

$$\mathrm{MARGIN} = \frac{(\mathrm{MT}_{\mathrm{MEAN,CST}} - \mathrm{MT}_{\mathrm{LL,CST}})}{(\mathrm{MT}_{\mathrm{LL,CST}} - \mathrm{MT}_{\mathrm{HOTEL,\,2nd\,TEST}})} \tag{3}$$

If the measured Mean Motor Torque at CST is less than the lower limit, then it cannot be assured that the valve has positive margin based solely on MCC testing. Accordingly, a new parallel test is required to satisfy the valve PV requirements and quantify margin, using direct thrust measurements in addition to MCC measurements. If the new parallel test is successful in establishing positive margin, then the parallel test becomes the new baseline MTPV test.

Evaluation at Upper Limit

As discussed above, the Upper Motor Torque Limit is the most limiting (i.e., lowest value) of the MOV mechanical limit and the reduced voltage motor torque capability (both adjusted for uncertainties). The MOV mechanical limit is a thrust limit converted to a motor torque limit using the MOV Factor, which is based on the mean motor torque at CST. However, the reduced voltage motor torque capability represents the maximum motor output torque for the MOV. Since the upper limit could be defined by either the MOV mechanical limit or the reduced voltage motor torque capability, the MTPV method conservatively requires comparison of the Maximum Motor Torque at CST to the Upper Motor Torque Limit. If the Maximum Motor Torque at CST is less than the upper limit, the valve is assured to have margin related to the load capability of the MOV.

Conditions Requiring a New Baseline Parallel Test

Once a baseline test is established for an MOV, this baseline can be used indefinitely going forward, so long as the setup and general conditions of the MOV do not change significantly. The events listed below are judged to significantly alter the setup and conditions of a valve. Accordingly, if any of these events occur after the baseline test of record, the original baseline test is invalidated and a new baseline "parallel" test needs to be performed.

- Change to torque switch setting
- Motor replacement
- Actuator refurbishment, gear ratio change, or replacement
- Valve replacement
- Change in stem lubricant (from one lubricant to another)

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[2] Per Reference 2, for certain motors the exponent in Equation (2) may be 2.5 rather than 2.0. See Reference 2 for additional information.

[3] It is important to note that not all of these uncertainties are applicable to each limit.

Validation

Validation of the MTPV method required measured stem thrust and motor torque data from multiple "parallel" tests of the same MOV. From MOVs with available test data, four similar gate valves (3 inch valves with SMB 00 actuators) met the MTPV applicability requirements and had test data with stem thrust and motor torque measurements from two separate tests. All four of these valves were used in the validation. The validation method included the following comparisons:

- Measured Thrust at CST vs. Predicted Thrust at CST based on measured Motor Torque
- Upper/Lower Limits and Margin calculated based on measured Stem Thrust vs. Upper/Lower Limits and Margin calculated based on measured Motor Torque

Measured Thrust at CST vs. Predicted Thrust at CST

The predicted mean thrust at CST ($TH_{MEAN,CST}$), based on measured motor torque, matched the measured stem thrust at CST (TH_{CST}) relatively well. As shown in Table 1, the maximum deviation from measured stem thrust was 13.3%. This variation is well within the uncertainty associated with determination of stem thrust from measurement of motor torque rather than direct measurement of stem thrust. This is illustrated in Figure 7, which plots Predicted and Measured Stem Thrust, including measurement uncertainties.

Upper Limit Comparison (Limit Based on Stem Thrust Measurement vs. Motor Torque Measurement)

With regard to the MOV Mechanical Upper Limit, the limit calculated in the MTPV method was lower (i.e., more restrictive) than the limit determined using methods which directly measure stem thrust (see Table 2). This reduction in setup "window" is due to the higher measurement uncertainty for motor torque compared to thrust.

However, the Reduced Voltage Upper Limit in the MTPV method was typically higher (i.e., less restrictive) than the limit determined using methods which directly measure stem thrust (see Table 2). This limit is based on motor output torque capability under conditions of reduced voltage. Because the MTPV method directly measures motor output torque, there are fewer parameter uncertainties to apply to the limit than if measured stem thrust is used.

Lower Limit Comparison (Limit Based on Stem Thrust Measurement vs. Motor Torque Measurement)

The Lower Limit calculated in the MTPV method was higher (i.e., more restrictive) than the limit determined using methods which directly measure stem thrust (see Table 3). This reduction in setup "window" is due to the higher measured uncertainty for motor torque compared to that for direct thrust measurement.

Operational Margin Comparison (Margin Based on Stem Thrust Measurement vs. Motor Torque Measurement)

The Operational Margin, or JOG PV Margin, is based on a comparison of the measured thrust or motor torque to the required thrust or motor torque. As expected, this margin is lower for analyses performed with the MTPV method, compared to analyses performed based on measured stem thrust (see Table 4). As discussed above for the Lower Limit, this reduced margin is due to higher uncertainty for measured motor torque than for thrust.

Conclusions

Based on the observations from the validation, the MTPV method satisfactorily determines Operational Margin as well as Motor Torque Upper and Lower Limits. Users should expect a reduction in apparent margin, a reduction in upper mechanical limit, and most likely an improvement (increase) in upper motor capability limit, when using this method in place of direct stem thrust measurement.
Accordingly, the MTPV method would be most beneficial for MOVs that have an operational margin (margin to lower limit) of at least 40% and a margin against structural damage (margin to upper mechanical limit) of at least 20%. There is no constraint with regard to margin against motor torque capability and, in fact, the MTPV method may

be a particularly good PV methodology for evaluation of MOVs whose setup is limited by motor torque capability at degraded voltage.

References

1. Limitorque SEL-3.
2. Limitorque Technical Update 05-01, "Actuator Output Torque Calculation SMB/SB/SBD Actuators/3 Phase Motors," January 2005.
3. "The Joint Owners' Group Program for Motor-Operated Valve Periodic Verification," Proceedings of the Fifth NRC/ASME Symposium on Valve and Pump Testing, July 1998, NUREG/CP-0152, Vol. 2.

Nomenclature

The nomenclature used in this paper is summarized below.

EFF	=	actuator efficiency
F_{MOV}	=	MOV factor
FS	=	stem factor
MARGIN	=	margin above required thrust at CST
MT_{HOTEL}	=	measured motor torque hotel load
$MT_{LL, CST}$	=	lower limit of motor torque at CST
$MT_{MEAN, CST}$	=	measured mean motor torque at CST
MT_{NOM}	=	nominal motor torque capability (motor start torque)
MT_{VRED}	=	motor torque capability at reduced voltage
OAR	=	overall actuator ratio
TH_{CST}	=	measured stem thrust at CST
V_{NOM}	=	nominal voltage
V_{RED}	=	reduced voltage

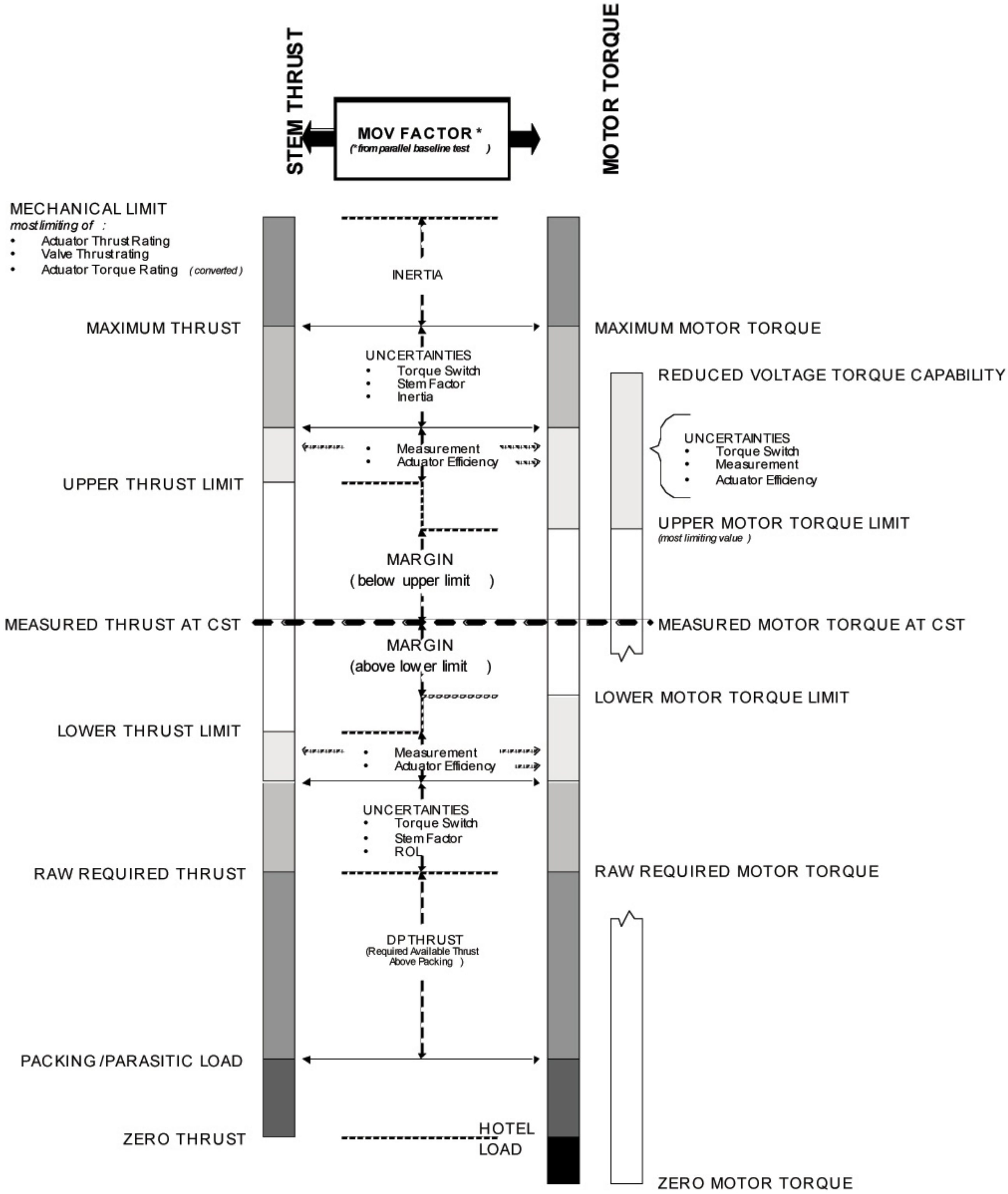

Figure 1 – Motor Torque Periodic Verification Method Limits and Margin

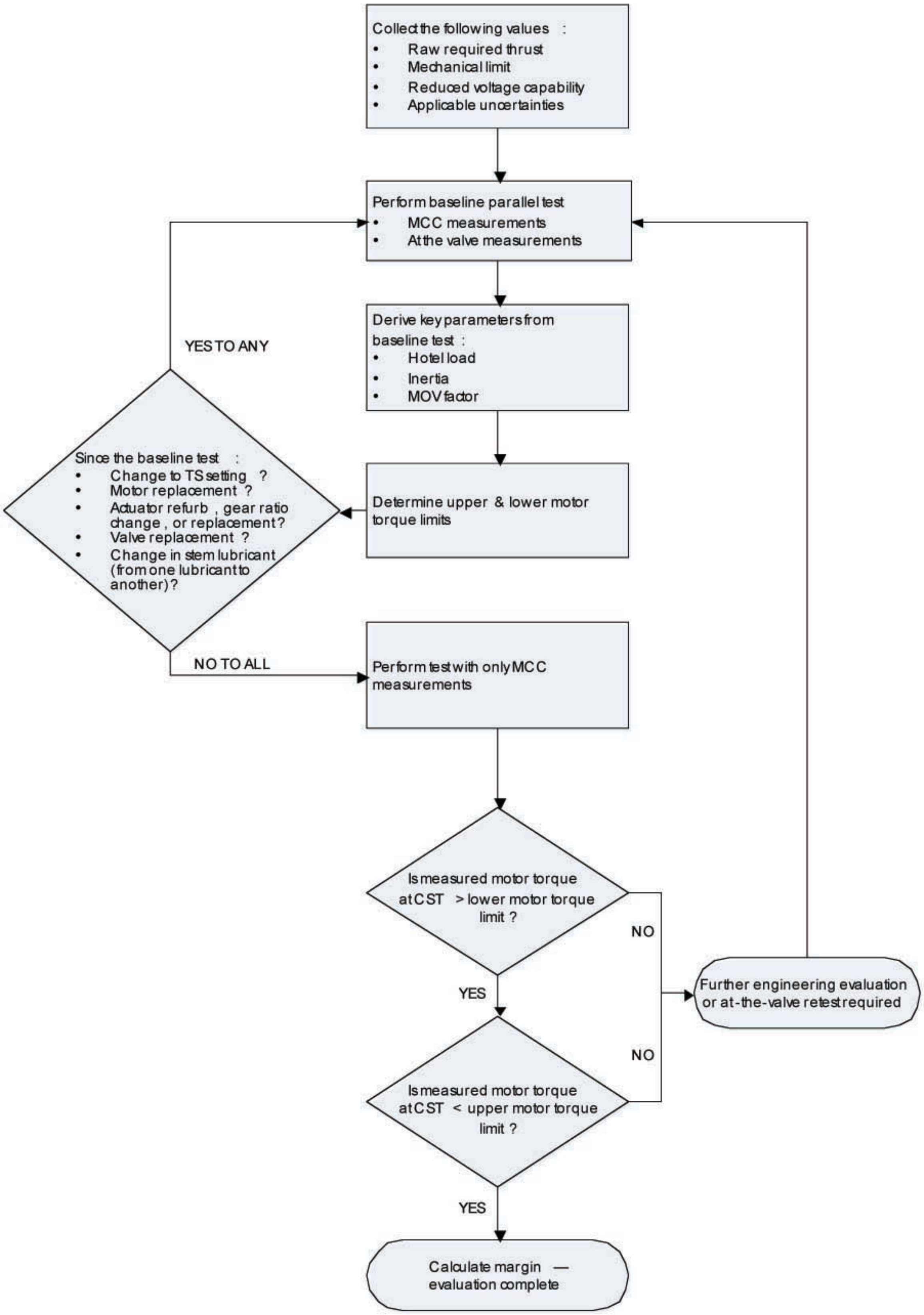

Figure 2 – Motor Torque Periodic Verification Method Flowchart

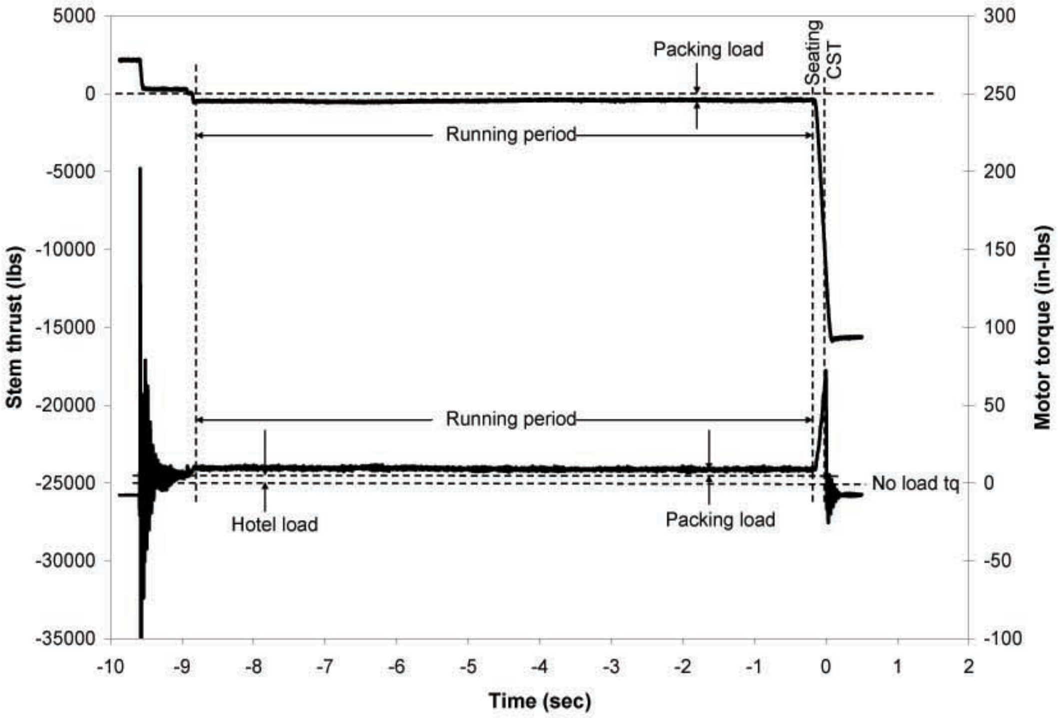

Figure 3 – Overlay of Measured Motor Torque (bottom trace) and Stem Thrust (top trace)

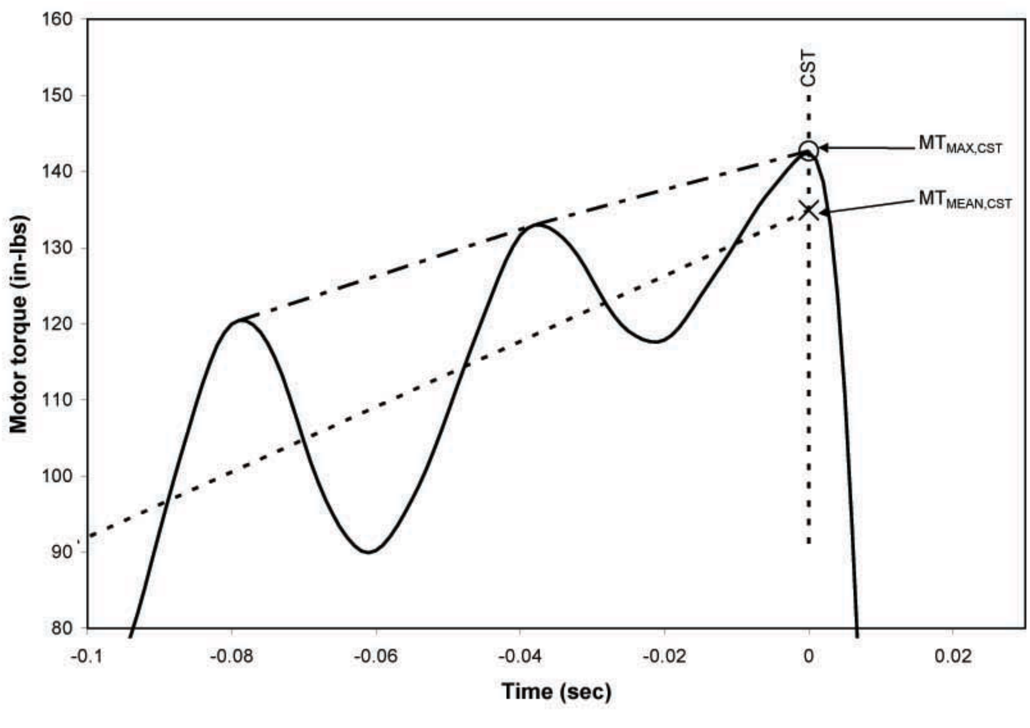

Figure 4 – Measured Motor Torque Near CST – Example with Significant Oscillations

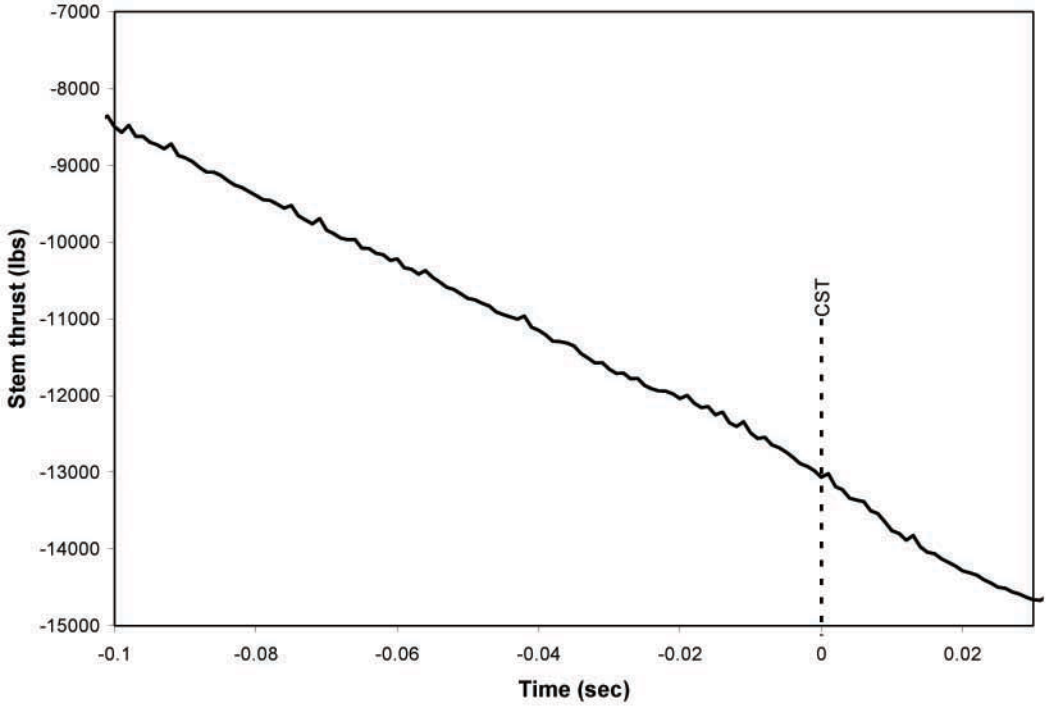

Figure 5 – Measured Stem Thrust Near CST for Example Corresponding to Figure 4

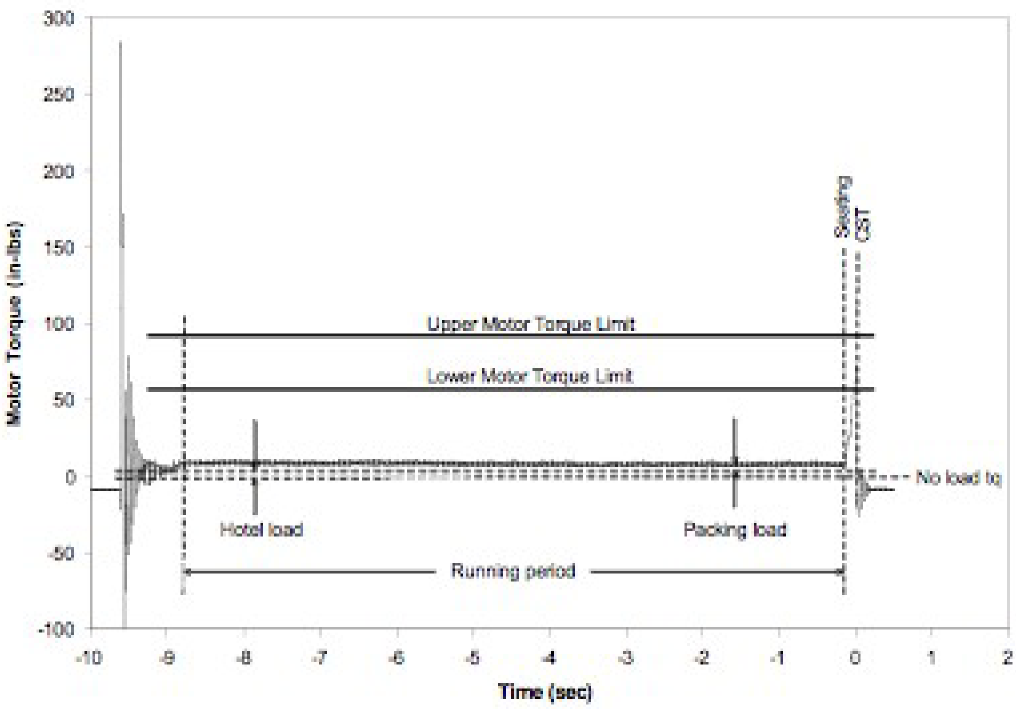

Figure 6 – Measured Motor Torque from MCC-Only Test

Table 1
Measured Thrust vs. Predicted Thrust Based on Measured Motor Torque

Valve	Measured Stem Thrust at CST (TH_{CST}), lbs	Predicted Mean Thrust at CST ($TH_{MEAN, CST}$), lbs	% Difference
MOV 1	12,501	12,202	-2.4%
MOV 2	13,474	13,017	-3.4%
MOV 3	13,224	11,466	-13.3%
MOV 4	12,608	14,174	12.4%

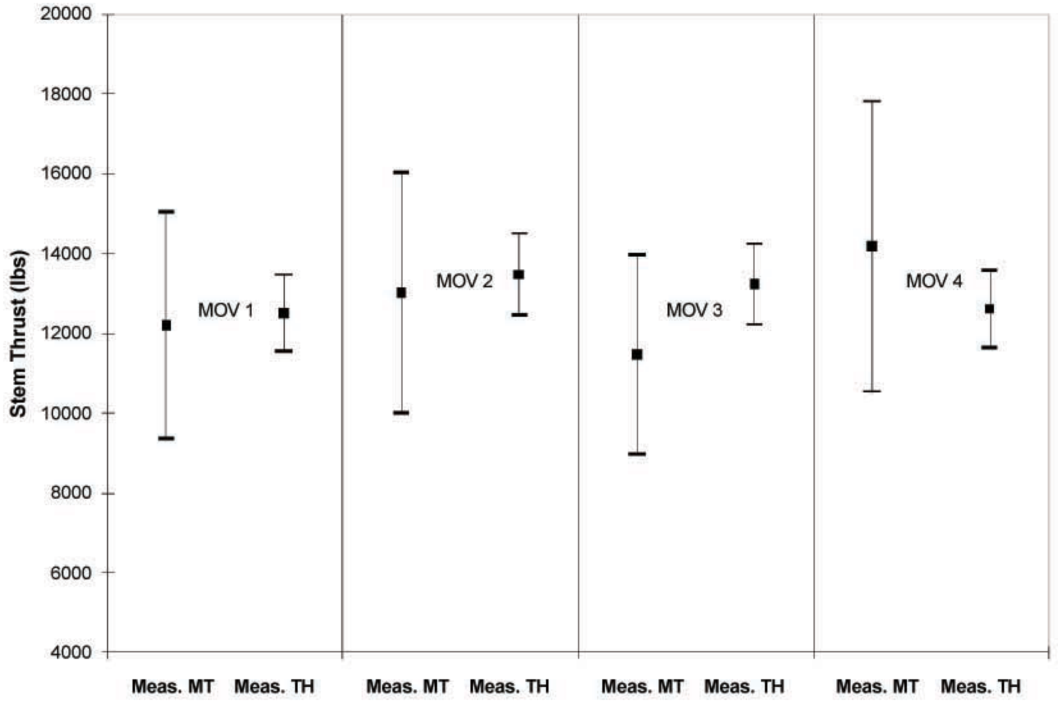

Figure 7 – Comparison of Predicted Mean Thrust at CST (based on measured motor torque) to Measured Stem Thrust at CST, Including Measurement Uncertainty

Table 2
Upper Limit Comparison

Valve	Thrust Upper Limit Based on Measuring Stem Thrust, lbs		Thrust Upper Limit Based on Measuring Motor Torque, lbs	
	Mechanical Limit	Red Voltage Limit	Mechanical Limit	Red Voltage Limit
MOV 1	14,052	19,444	12,040	19,892
MOV 2	14,457	18,241	12,407	20,502
MOV 3	13,151	16,417	11,434	16,028
MOV 4	14,062	23,878	11,477	28,462

Table 3
Lower Limit Comparison

Valve	Thrust Lower Limit Based on Measuring Stem Thrust, lbs	Thrust Lower Limit Based on Measuring Motor Torque, lbs 4
MOV 1	10,527	12,404
MOV 2	7,801	9,573
MOV 3	7,801	9,265
MOV 4	10,527	13,045

Table 4
Operational Margin Comparison

Valve	Operational Margin Based on Measuring Stem Thrust	Operational Margin Based on Measuring Motor Torque
MOV 1	18.8%	-1.6%
MOV 2	72.7%	36.0%
MOV 3	69.5%	23.8%
MOV 4	19.8%	8,7%

[4] Motor Torque Limit converted to Thrust Limit using MOV Factor; TH = $(MT- MT_{HOTEL})/(F_{MOV,CST})$

Elimination of RHR Piping Vibration

Mike Davis and Sekhar Samy

CCI

This paper discusses the vibration and pipe failure problems experienced in the Residual Heat Removal (RHR) system at the Grand Gulf Nuclear Power Plant. The root cause analysis of the problem showed that excessive fluid velocity across the plug and seat ring as it flows through the E21- F003 valve was the prime reason.

Grand Gulf was able to retrofit (replacement of the original valve internals with internals supplied by another company) a new and innovative trim design in this Motor Operated isolation type globe valve with minimal changes to the existing valve. The result is the complete elimination of vibration and control problems.

System:

RHR system in a Boiling Water Reactor (BWR) has several modes including:

- Low pressure coolant injection (LPCI)
- Suppression pool cooling
- Fuel pool cooling assist
- Shutdown (S/D) Cooling

Shutdown cooling RHR throttle valves E12F003A & B are 18", 300# ANSI Powell Globe type with the need to control low flows for extended periods of time to remove decay heat and accommodate in vessel activities. It must have high flow capability at lower pressure drops for the postulated LPCI mode of operation.

At the Grand Gulf NPP, both E12F003A and E12F003B exhibited poor throttle control capability over the years, eventually developing seat and guide damage to both valves from throttle use. E12F003A throttle use in Refueling Outage (RF) 12 resulted in a small bore piping failure and water spill in the RHR room. In addition, there was internal erosion damage found in the valve body and seat. This led to the development of an engineering request to look at various repair options and long term solutions. Solutions considered were:

1. Purchase a new valve body
2. Send the old 3A body to a hot shop and have valve vendor personnel repair it.
3. Repair the valve body at Grand Gulf using their extremely qualified welders.

Option one was outside the time limit. Option two would work but would be extremely expensive and could be time limiting. Option three was the best choice. Grand Gulf has the technicians in house with a hot shop and necessary boring bar for performing the post welding machine work.

The initial choice and determination for RF12 was to repair the body as it was determined to be a repair that could be handled.

Grand Gulf contacted the Original Equipment Manufacturer (OEM) and other valve companies for a long term solution to this problem. Step one was to look at the service conditions and see if this sheds light on the probable causes for valve damage.

In the above service conditions the trim exit velocities for the “Shutdown Cooling” cases are in excess of 200 ft/sec! The Cavitation Index is also around 1.25 for two of the cases and 1.67 for a third. This is an accurate prediction that the process conditions are resulting in cavitation damage. Note: the Cavitation Index is not scaled for pressure or size. So, the conclusion was excessive trim exit velocity is the root cause of vibration and cavitation.

TABLE 1 – SERVICE CONDITIONS

Fluid	Water/Steam					
Critical Pressure	psig	3194				
Critical Temperature	deg F	705.5				
Condition		**S/D Cooling 1**	**S/D Cooling 2**	**S/D Cooling 3**	**LPCI 1**	**LPCI 2**
Fluid State		Water	Water	Water	Water	Water
Liquid Vol. Flow Rate	gpm	2500.0	2500.0	3000.0	7589.0	8635.0
Inlet Pressure	psig	450.0	450.0	425.4	105.5	92.86
Outlet Pressure	psig	173.0	173.0	173.0	101.549	88.0
Pressure Differential	psi	277.0	277.0	252.4	3.951	4.86
Inlet Temperature	deg F	344.0	70.0	344.0	185.0	185.0
Density	lbm/ft^3	55.91	62.39	55.9	60.49	60.49
Vapor Pressure	psig	109.7	-14.33	109.7	-6.303	-6.303
Cavitation Index	σ_1	1.23	1.67	1.25	28.6	20.4
Required Flow Capacity	Cv	142.2	150.2	178.7	3759.1	3856.4
Ported Valve Trim Exit Velocity	ft/sec	214	203	205	24.6	27.3
8 Turn Disk Stack Velocity	ft/sec	44.5	42.1	42.4	No Disk	No Disk

psig = pounds force per square inch gage

gpm = gallons per minute

deg F = degrees Fahrenheit

lbm/ft^3 = pounds mass per cubic feet

ft/sec = feet per second

Solution

Any solution must meet the following criteria:

- No cutting or welding to be performed during implementation
- Provide trim to reduce the S/D cooling velocity to a level below valve industry recommended guidelines
- Do not impact LPCI accident performance & maximum Cv
- Package the trim inside the existing valve body
- Minimal impact on weight and Cg
- No change in stroke length or stroke time
- Solution must be robust & reliable
- Must be capable of implementation with unit on-line

For long term the selected solution was to replace the existing valve stem and plug with a custom throttle plug assembly inserted in the existing valve body. The throttle plug must be designed to reduce pressure in stages and as a result limit the velocity of the fluid in the trim (Figure 1) so that the pressure never falls below the fluid's vapor pressure.

By using multiple right angle tortuous flow paths as shown in Figure 2, it is possible to reduce the trim velocity to acceptable levels. The selected velocity limit to reduce the potential for vibration and cavitation was 40 ft/sec per ISA recommendations in Reference (1).

Each individual flow path has a series of turns that breaks up the pressure drop across the valve into multiple stages, and has expanding passages to reduce fluid exit velocity.

This approach uses a series of flat metal disks to form a trim assembly. Each disk has a flow pattern of successive right angle turns cut into its flat surface. When stacked, these pathways can be matched or mismatched between individual disks to create a labyrinth flow pattern that enables trim to be infinitely tuned to control flow in a manner that maintains positive operating characteristics throughout the valve's operating range (Figure 2). The flow path for each disk is opened as the plug moves within the center opening of the seat ring

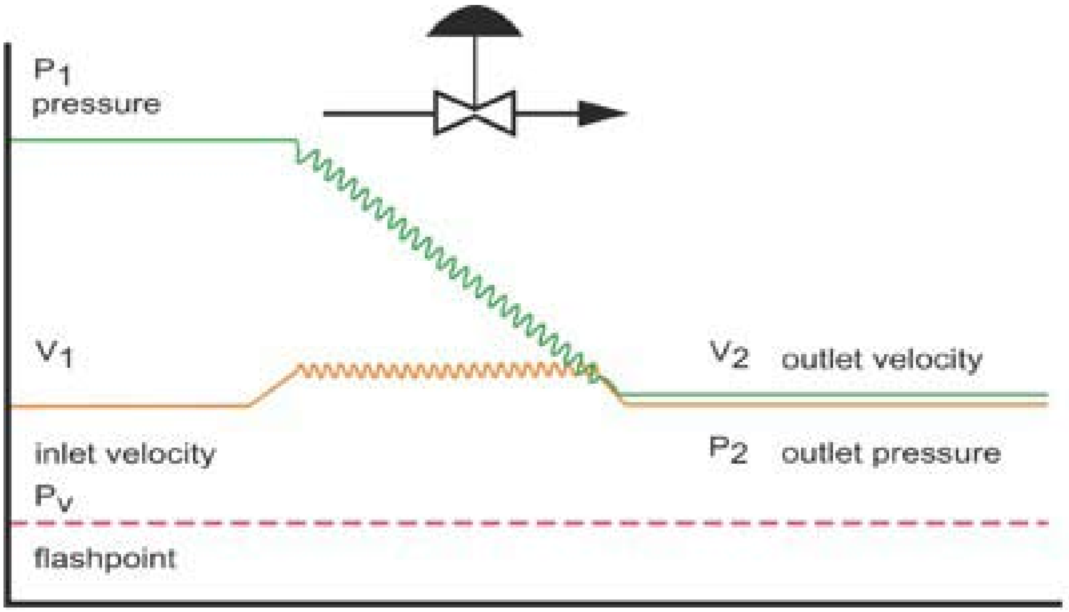

Figure 1 – Flow path in a multi-path multi-stage trim

This flow method controls the damaging effects of velocity in two ways: by dividing the flow into many small streams of low mass flow rate, and by forcing fluid through a series of sharp right angle turns to affect the pressure drop steps.

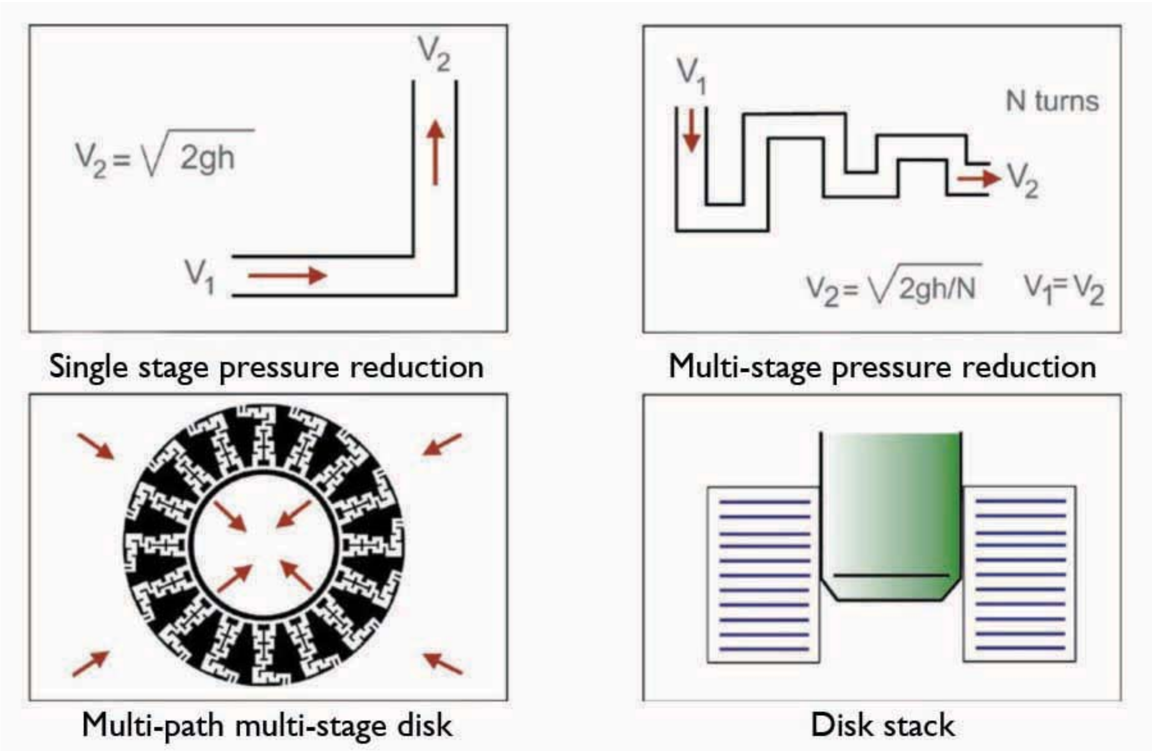

Figure 2 – Multi-Stage Multi-Path Flow Geometry

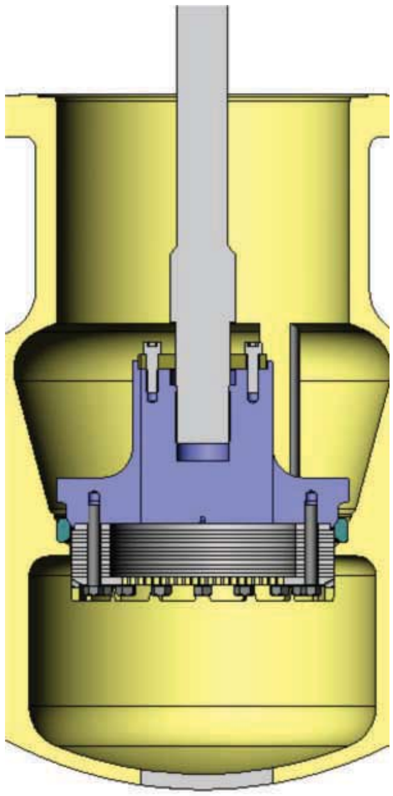

Figure 3 – Plug with Disk Stack

Retrofit

The valve is an isolation type valve with a welded seat ring in the body. Further there is no appreciable place in the valve body to install a disk stack or cage. Therefore the only available solution is to replace the existing William Powell stem and plug with a custom throttle plug inserted in the existing William Powell valve body.

The new solution is to provide a disk stack that is bolted to the plug. For 40% of the stroke the flow passes through the disk stack, this is the range of conditions in the S/D cooling mode. For the remaining 60% of the stroke, the disk stack is retracted from the flow and the flow is through the seat ring, and valve maximum Cv is not affected.

At these stroke positions, the energy in the fluid flow will be sufficiently controlled so that cavitation and vibration are eliminated.

Implementation/Installation

Because 1E12F003A had recently been replaced, thus insuring good internal condition, the decision was made to modify it first while the unit was on-line to provide Operations a Shutdown Cooling loop that could be throttled as needed, prior to and during RF 13. The steps in the retrofit process are as follows:

1. Pre-stage all required tools and test equipment prior to entering the Limiting Condition for Operation (LCO)
2. Prepare work area to accommodate the work scope. De-con-shield, scaffold, rigging.
3. Hang the required tag outs, isolate, and commence LCO and drain down (6 hours)
4. Determine the main and control power from the Limitorque (2 hours)
5. De-tension the bonnet using a multi head hi-torque (2 hours)

6. Unpack the valve once drained (1 hour)
7. Remove the stem nut from the Limitorque and rig for removal from stem (2 hours)
8. Remove mounting bolts and rig the Limitorque from the yoke (2 hours)
9. Remove yoke and bonnet (2 hours)
10. Rig out old valve plug and stem (1 hour)
11. Set up mill and perform valve seat skim cut for preparation to install new trim (4-6 hours)
12. VT visual examination of valve body internal and rail areas (2 hours)
13. Install new plug and stem (3 hours)
14. Blue check seat area (2 hours)
15. Install seal ring and bonnet/yoke assembly (4 hours)
16. Re-pack the valve (2 hours)
17. Fill, vent, restore RHR system to standby line up (4 hours)
18. Install Limitorque, then stem nut, and torque fasteners (3 hours)
19. Re-terminate and rough set the limits on the Limitorque (2 hours)
20. Clear tags and perform proper line up for static and dynamic VOTES diagnostic test (2 hours)
21. Perform Static and In-situ dynamic VOTES test and vibration testing with flow at 0, 20, 40, 60, 80, 100% with E12F048 closed then open (4 hours)
22. Engineering review of test data (4 hours)
23. Return to service (2 hours)

E12-F003A was modified and tested successfully during the second week of December 2003. Testing proved the new plug design was successful in restoring full throttle capability to E12-F003A and that the existing capacity was not affected. Similar retrofit and testing as outlined was successfully completed on the E12-F003B valve in March 2004 during RF 13.

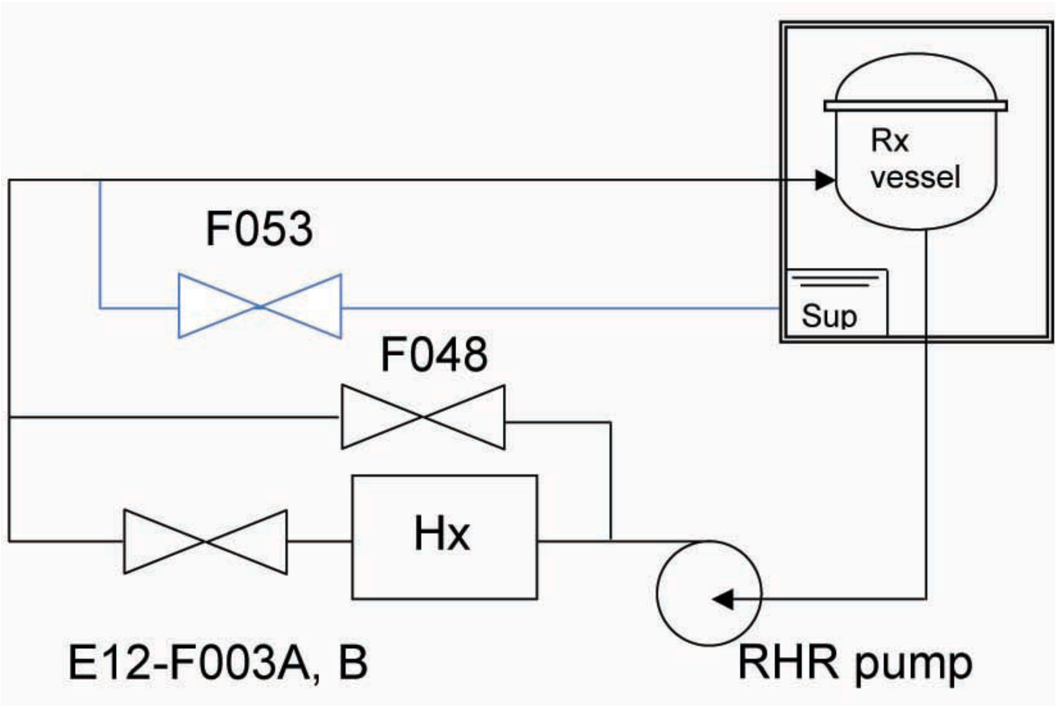

Figure 4 – RHR System Layout

As a post maintenance requirement, the 1E12F003A Limitorque actuator was set up to achieve design requirements for torque and thrust.

Flow

Initial conditions for the flow or dynamic testing had RHR "A" lined up to provide 100% of the pump flow through the 1E12F003A valve. At pump start, the lE12F003A valve was 100% open. Operations closed the valve in a continuous mode with the valve achieving full flow shutoff against RHR A pump shutoff head. Additionally, full flow capability through lE12F003A was satisfied with a recorded value of 7950 gpm (gallons per minute). The next portion of the test performed a step down from full open to significant throttle then back to full open valve position. This established minimum throttle conditions and also established a vibration data baseline. All flow was passed through the lE12F003A valve during this portion of the test. This particular section of the test determined the throttle range to recommend to Operations for long-term use. Beginning at 100% flow, the valve was throttled in steps with the following results

Phase two of the dynamic flow test was performed with 1E12F048A full open as 1E12F003A was throttled closed in -20% increments from 100% open to full close. This

TABLE 2 FLOW THROUGH VALVES

Actual flow	% 1E12F003A OPEN
7950 gpm	100%
7100 gpm	50%
6000 gpm	40%
5000 gpm	32%
4000 gpm	19%
3000 gpm	7%
2500 gpm	2%

TABLE 3 E12F003A VIBRATION DATA

Actual Flow, gpm	Percent 1E12F003A open	Vibration Recorded at Highest Plane, g's
7950	100%	0.041
7100	50%	0.152
6000	40%	0.1
5000	32%	0.1
4000	19%	0.035
3000	7%	0.031
2500	2%	0.1

documented that full flow capability through 1El2F048A of 8200 gpm could be achieved and maintained while 1E12F003A was throttled from 100% to zero.

Additionally, vibration baseline determined no unexpected resonance developed in these various throttle modes.

Vibration recorded at Highest Plane actual flow 7950 gpm was 0.041 g.

Vibration

Pre-retrofit, the vibration of the system was "similar to a train derailing." It was a frightening sound so special instruction were written to only allow low flow throttle for a short duration in order to slow down the harmonic damage. Obviously that did not work in the long term.

As part of the post retrofit test, vibration measurements were taken on the valve body in three planes during the throttling steps with full flow going through lE12F003A, and also with flow shared with lE12F048A full open as lE12F003A was throttled. Preliminary percent lE12F003A OPEN results of the vibration data indicate minimal vibration in all three planes of measurement. Acceleration peaks were less than 0.2 g at all frequencies less than or equal to 100 Hertz. Acceleration peaks remained less than 0.1 g at all frequencies less than or equal 30 Hertz. These conditions satisfied the acceptance criteria of less than 0.3 g at 30 Hertz equivalent.

It should be noted that, during the throttling evolution, the noise level was very acceptable with no impacts noted when greater than or equal to 3000 gpm. At 2500 gpm, there was some low level impact-type sounds which were attributed to the valve being <2% open. Given the total stroke length of the valve (9.1") at 2%, the disc seat and the in-body

TABLE 4 E12F003B VIBRATION DATA

Actual Flow, gpm	Percent 1E12F003B open	Vibration Recorded at Highest Plane, g's
7900	100%	0.081
5000	40%	0.143
4000	29%	0.026
3000	20%	0.036
2500	16%	0.042

valve seat were only approximately 1/16" apart; potentially allowing minor seat contact to create the impact noise. Vibration was still within acceptable limits at this point.

During the last refueling outage in 2005, the valves performed without vibration and fuel pool clarity was maintained.

References

1. "Control Valves – Practical Guides for Measurement and Control," edited by Guy Borden, Jr., and Paul G. Friedman, 1998 edition, published by ISA.

Session 4(a):
Regulatory Issues

Session Chair

Thomas G. Scarbrough

U.S. NRC

POWER-OPERATED VALVE ACTIVITIES AND ISSUES

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Ninth NRC/ASME Symposium on Valve and Pump Testing
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I. Introduction

The safe operation of a nuclear power plant depends on motor-operated valves (MOVs) in fluid systems successfully performing their safety functions. MOVs must be capable of operating under design-basis conditions, which may include high differential pressure and flow, high ambient temperature, and degraded motor voltage. The design of the MOV must apply valid engineering equations and parameters to ensure that the MOV will operate as intended during normal plant operations and design-basis events. Manufacturing, installation, preoperational testing, operation, inservice testing (IST), maintenance, and replacement must be conducted by trained personnel using proper procedures. Surveillance must be performed and testing criteria must be applied on a soundly based frequency in a manner that suitably detects questionable operability or degradation. Moreover, these activities must be monitored by a strong quality assurance program.

The regulations of the U.S. Nuclear Regulatory Commission (NRC) require that components that are important to the safe operation of a U.S. nuclear power plant be treated in a manner that ensures their performance. Appendix A, "General Design Criteria for Nuclear Power Plants," and Appendix B, "Quality Assurance Criteria for Nuclear Power Plants and Fuel Reprocessing Plants," to Part 50 of Title 10 of the Code of Federal Regulations (10 CFR Part 50) contain broadly based requirements in this regard. In 10 CFR 50.55a, the NRC initially required U.S. nuclear power plant licensees to implement provisions of the American Society of Mechanical Engineers (ASME) Boiler & Pressure Vessel Code (B&PV Code) for testing of MOVs as part of their IST programs. In 1999, the NRC revised 10 CFR 50.55a to incorporate by reference the ASME Code for Operation and Maintenance of Nuclear Power Plants (OM Code) for inservice testing of MOVs. The NRC also supplemented the quarterly MOV stroke-time testing specified in the ASME Code by requiring that licensees verify MOV design-basis capability on a periodic basis. In 2004, the NRC issued 10 CFR 50.69 that allows for an alternative approach in establishing requirements for treatment of SSCs at nuclear power plants using a risk-informed method of categorizing SSCs according to their safety significance.

II. MOV Design-Basis Capability

Operating experience at nuclear power plants in the 1980s and 1990s revealed weaknesses in many activities associated with MOV performance. For example, some engineering analyses used in the original sizing and setting of MOVs did not adequately predict the thrust and torque required to open and close valves under design-basis conditions. Both regulatory and industry research programs later confirmed the weaknesses in the initial design and qualification of MOVs. For example, the NRC Office of Nuclear Regulatory Research sponsored an extensive program at the Idaho National Laboratory (INL) to study the performance of MOVs under various flow, temperature, and voltage conditions. In addition, the nuclear industry sponsored a program by the Electric Power Research Institute (EPRI) to develop a computer methodology to predict the performance of MOVs under a wide range of operating conditions. Poor MOV performance also resulted from shortcomings in maintenance programs, such as inadequate procedures and training. Further, testing of MOVs to measure valve stroke times under zero differential-pressure and flow conditions was shown not to detect deficiencies that could prevent MOVs from performing their safety functions under design-basis conditions.

This paper was prepared by staff of the U.S. Nuclear Regulatory Commission. It may present information that does not currently represent an agreed-upon NRC staff position. NRC has neither approved nor disapproved the technical content.

In response to weaknesses in MOV performance, the NRC staff issued Generic Letter (GL) 89-10 (June 28, 1989), "Safety-Related Motor-Operated Valve Testing and Surveillance." In GL 89-10, the NRC staff requested that licensees ensure the capability of MOVs in safety-related systems to perform their intended functions by reviewing MOV design bases, verifying MOV switch settings initially and periodically, testing MOVs under design-basis conditions where practicable, improving evaluations of MOV failures and necessary corrective action, and trending MOV problems. The NRC staff requested that licensees complete their GL 89-10 programs within approximately three refueling outages or 5 years of the issuance of the generic letter.

In support of the regulatory activities to ensure MOV design-basis capability, the NRC Office of Nuclear Regulatory Research identified areas in which research and analysis were needed to assist in evaluating MOV programs at nuclear power plants. For example, the NRC performed research to evaluate (1) performance of MOVs under pump flow and blowdown conditions; (2) output of ac-powered and dc-powered MOV motor actuators; (3) the increase in friction of aged samples of valve materials; (4) methods to determine appropriate values for stem friction coefficient; (5) pressure locking and thermal binding of gate valves; and (6) the effect of ambient temperature on stem lubricant performance. The NRC sponsored flow testing of several MOVs by INL under normal flow and blowdown conditions. The testing revealed that (1) more thrust was required to operate gate valves than predicted by standard industry methods; (2) some valves were internally damaged under blowdown conditions and their operating requirements were unpredictable; (3) static and low flow testing might not predict valve performance under design-basis flow conditions; (4) during valve opening strokes, the highest thrust requirements might occur at unseating or in the flow stream; (5) partial valve stroking did not reveal the total thrust required to operate the valve; (6) torque, thrust, and motor operating parameters were needed to fully characterize MOV performance; and (7) reliable use of MOV diagnostic data requires accurate equipment and trained personnel. The NRC summarizes some of the results of the MOV research program in NRC Information Notice (IN) 90-40 (June 5, 1990), "Results of NRC-Sponsored Testing of Motor-Operated Valves."

To assist nuclear power plant licensees in responding to GL 89-10, EPRI developed the MOV Performance Prediction Methodology (PPM) to determine dynamic thrust and torque requirements for gate, globe, and butterfly valves based on first-principles of MOV design and operation. EPRI described the methodology in Topical Report TR-103237 (Revision 2, April 1997), "EPRI MOV Performance Prediction Program." The EPRI MOV PPM program included the development of improved methods for prediction and evaluation of system flow parameters; gate, globe, and butterfly valve performance; and motor-actuator rate-of-loading effects (load sensitive behavior). EPRI also performed separate effects testing to provide information for refining the gate valve model and rate-of-loading methods; and conducted numerous MOV tests to provide data for development and validation of the models and methods, including flow loop testing, parametric flow loop testing of butterfly valve disk designs, and in-situ MOV testing. EPRI integrated the individual models and methods into an overall methodology including a computer model and implementation guide. On March 15, 1996, the NRC staff issued a safety evaluation (SE) accepting the EPRI MOV PPM with certain conditions and limitations. On February 20, 1997, the staff issued a supplement to the SE on general issues and two unique gate valve designs. On April 20, 2001, the staff issued Supplement 2 to the SE on Addendum 1 to EPRI Topical Report TR-103237 addressing an update of the computer model. The staff alerted licensees to lessons learned from the EPRI MOV program in IN 96-48 (August 21, 1996), "Motor-Operated Valve Performance Issues."

On September 8, 1999, the Nuclear Energy Institute (NEI) submitted Addendum 2 to EPRI Topical Report TR-103237-R2, which described the development of the Thrust Uncertainty Method that takes into account conservatism in the EPRI MOV PPM to provide a more realistic (less bounding) estimate of the thrust required to operate gate valves than predicted by the PPM. In Supplement 3 (dated September 30, 2002) to the SE on the EPRI PPM, the NRC staff concluded that the Thrust Uncertainty Method developed by EPRI is acceptable for the prediction of minimum allowable thrust at control switch trip (or flow isolation) for applicable motor-operated gate valves under cold water applications within the scope of the Thrust Uncertainty Method, based on the NRC staff's review of Addendum 2 to the EPRI Topical Report as supplemented by NEI submittals dated January 5 and December 6, 2001, and June 10, 2002. The NRC staff stated that the Thrust Uncertainty Method may be applied consistent with the criteria specified for the EPRI MOV PPM in EPRI TR-103237-R2 and Addenda 1 and 2 to TR-103237-R2, as supplemented by NEI submittals dated January 5 and December 6, 2001, and June 10, 2002. More recently, NEI has submitted additional addenda to the EPRI MOV PPM that are under review by the NRC staff.

Nuclear power plant licensees implemented the recommendations of GL 89-10 through a combination of design-basis reviews, revision of MOV calculations and procedures, static and dynamic diagnostic testing, industry-sponsored research programs, and trending of test results. The industry expended significant resources to resolve the deficiencies in the design, qualification, and application of safety-related MOVs that led to the issuance of GL 89-10. The results of the GL 89-10 programs and their implementation include (1) MOV sizing calculations and switch settings have been revised to reflect actual valve performance; (2) improved valve performance prediction methods have been developed; (3) valve internal dimensions are being addressed to provide assurance of predictable gate valve performance under blowdown conditions; (4) friction coefficients in new or refurbished gate valves have been found to increase with service until a plateau is reached; (5) MOV output prediction methods have been updated; and (6) personnel training and maintenance practices have been improved. The NRC staff evaluated the MOV programs at nuclear power plants through onsite inspections of the design-basis capability of safety-related MOVs. The NRC staff closed its review of GL 89-10 for each active U.S. nuclear power plant. The NRC staff will be reviewing the GL 89-10 program at Browns Ferry Unit 1 prior to its restart.

On August 17, 1995, the NRC issued GL 95-07, "Pressure Locking and Thermal Binding of Safety-Related Power-Operated Gate Valves," to request that licensees perform, or confirm that they had previously performed, (1) evaluations of the operational configurations of safety-related, power-operated gate valves for susceptibility to pressure locking and thermal binding; and (2) further analyses, and any needed corrective actions, to ensure that safety-related power-operated gate valves that are susceptible to pressure locking or thermal binding are capable of performing their safety functions within the current licensing basis of the facility. The NRC staff completed its review of licensee responses to GL 95-07 through issuance of an SE addressing each active U.S. nuclear power plant.

On September 18, 1996, the NRC staff issued GL 96-05, "Periodic Verification of Design-Basis Capability of Safety-Related Motor-Operated Valves," to provide recommendations for assuring the capability of safety-related MOVs to perform their design-basis functions over the long term. In GL 96-05, the NRC staff requested that licensees establish a program, or ensure the effectiveness of their current program, to verify on a periodic basis that safety-related MOVs continue to be capable of performing their safety functions within the current licensing basis of the facility.

In response to GL 96-05, nuclear power plant owners' groups developed an industry-wide Joint Owners Group (JOG) Program on MOV Periodic Verification to obtain benefits from sharing information between licensees on MOV performance. The JOG described its program in Topical Report MPR-1807 (Revision 2, July 1997), "Joint BWR, Westinghouse and Combustion Engineering Owners' Group Program on Motor-Operated Valve (MOV) Periodic Verification." Elements of the JOG program included (1) an "interim" MOV periodic verification program of static diagnostic testing based on MOV safety significance and capability margin for licensees to use in response to GL 96-05; (2) a 5-year dynamic testing program to identify potential age-related increases in required thrust and torque to operate gate, globe, and butterfly valves under dynamic conditions; and (3) a long-term MOV diagnostic program based on information from the dynamic testing program. On October 30, 1997, the NRC staff issued an SE accepting the JOG Program on MOV Periodic Verification with certain conditions and limitations. Licensees of 98 reactor units have participated in the JOG program. The JOG 5-year dynamic testing program included about 200 valves that received repetitive dynamic tests with at least a 1-year time interval between the tests. On February 27, 2004, the JOG submitted Topical Report MPR-2524 (Revision 0, February 2004), "Joint Owners' Group Motor Operated Valve Periodic Verification Program Summary," providing the long-term recommendations for MOV periodic verification to be implemented by licensees as part of their commitments to GL 96-05. The long-term JOG program includes static diagnostic testing of GL 96-05 MOVs based on their safety significance and capability margin with dynamic testing as determined by the results of the JOG testing program and plant-specific evaluations. The NRC staff is completing an SE on its evaluation of the JOG topical report.

In that the JOG program focused on potential increases in valve operating requirements, licensees address potential degradation in the output of MOV motor actuators by their plant-specific programs. In the late 1990s, the NRC sponsored research at INL to study the performance of ac-powered MOV motor actuators manufactured by Limitorque Corporation, under various temperature and voltage conditions. For the Limitorque ac-powered motor-actuator combinations tested, the research indicated that (1) actuator efficiency might not be maintained at "run" efficiency published by the manufacturer; (2) degraded voltage effects can be more severe than predicted by the square of the ratio of actual to rated motor voltage; (3) some motors produce more torque output than predicted by their nameplate rating; and (4) temperature effects on motor performance appeared consistent with the Limitorque guidance. The NRC study of ac-powered MOV output is described in NUREG/CR-6478 (July 1997), "Motor-Operated Valve (MOV) Actuator Motor and Gearbox Testing." The nuclear industry also evaluated the output capability of ac-powered MOVs at several plants. In response to the new information on ac-powered MOV performance, Limitorque provided updated guidance in its Technical Update 98-01 (May 15, 1998) and Supplement 1 (July 17, 1998) for the prediction of ac-powered MOV motor actuator output. The NRC alerted licensees to the new information on ac-powered MOV output in Supplement 1 (July 24, 1998) to IN 96-48.

Following the NRC review of ac-powered MOV performance, the NRC sponsored research at INL to study the performance of Limitorque dc-powered MOV motor actuators under various temperature and voltage conditions. For the Limitorque dc-powered motor-actuator combinations tested, the research indicated that (1) ambient temperature effects were more significant than predicted; (2) use of a linear voltage factor needs to consider reduced speed, increased motor temperature, and reduced motor output; (3) stroke-time increase is significant for some dc-powered MOVs under loaded conditions; and (4) actuator efficiency may fall below the published "pullout" efficiency at low speed and high load conditions. The research results are provided in NUREG/CR-6620 (May 1999), "Testing of dc-Powered Actuators for Motor-Operated Valves." On June 23, 2000, the Boiling Water Reactor Owners' Group (BWROG) forwarded Topical Report NEDC-32958 (March 2000), "BWR Owners' Group dc Motor Performance Methodology - Predicting Capability and Stroke Time in dc Motor-Operated Valves," to the NRC staff for information. On August 1, 2001, the NRC issued Regulatory Issue Summary (RIS) 2001-15, "Performance of dc-Powered Motor-Operated Valve Actuators," that informs licensees of the availability of improved industry guidance for predicting dc-powered MOV actuator performance. In RIS 2001-15, the NRC staff stated that, based on a sample review, the BWROG methodology represents a reasonable approach to improvement of past industry guidance for predicting dc-powered MOV stroke time and output. The staff considers the BWROG methodology to be applicable to Boiling Water Reactor and Pressurized Water Reactor plants because of the similarity in the design and application of dc-powered MOVs.

Each U.S. nuclear power plant licensee submitted a description of plans for periodic verification of the design-basis capability of safety-related MOVs in response to GL 96-05. The NRC staff reviewed the licensee submittals and conducted sample inspections of GL 96-05 programs. The staff prepared an SE to document its review of the response to GL 96-05 by each licensee. Where a licensee committed to implement the JOG program, the NRC staff relied to a significant extent on that commitment in preparing the SE without the need for plant-specific inspection activity. The NRC staff reviewed GL 96-05 programs of licensees that did not commit to the JOG program by a separate process of submittals and inspections, as appropriate. As licensees implement their long-term MOV programs including incorporation of the JOG program results, the NRC will monitor those programs using Inspection Procedure 62708, "Motor-Operated Valve Capability," as part of the NRC reactor oversight program.

III. Design-Basis Capability For POVs (Other Than MOVs)

The NRC established Generic Safety Issue (GSI) 158, "Performance of Safety-Related Power-Operated Valves Under Design-Basis Conditions," to evaluate whether additional regulatory actions were necessary to address performance issues for POVs (other than MOVs) after MOV operating experience and research results indicated that testing under static conditions was insufficient to demonstrate consistent performance of these valves under design-basis conditions. Operating events involving observed or potential common-cause failures were documented in NUREG-1275, "Operating Experience Feedback Report," Volumes 2 and 6 for air systems and AOVs, respectively. These issues are also discussed in NUREG/CR-6644, "Generic Issue 158: Performance of Safety-Related Power-Operated Valves Under Operating Conditions." Two related documents, NUREG-1275, Volume 13, "Evaluation of Air-Operated Valves at U.S. Light-Water Reactors," and NUREG/CR-6654, "A Study of Air-Operated Valves in U.S. Nuclear Power Plants," are focused specifically on AOVs.

The NRC staff previously requested that the industry verify the capability of AOVs with respect to issues involving the plant instrument air supply system. In GL 88-14, "Instrument Air Supply System Problems Affecting Safety-Related Equipment," addressees were requested to verify by test that air-operated safety-related components will perform as expected in accordance with all design-basis events. All addressees were required to respond to the generic letter with confirmation that this verification had been performed. All responses were received by 1993 and the generic letter was subsequently closed.

In IN 96-48, the NRC staff noted that some of the lessons learned from MOV operating experience and testing are applicable to other POVs. For example, the thrust requirements to operate some gate valves under pump flow and blowdown conditions were higher than predicted by the valve manufacturers. The potential exists for gate valves to be damaged when operating under blowdown conditions such that the thrust requirements can be unpredictable. The effective flow area in some globe valves can be larger than expected and can cause thrust requirements to be higher than predicted. The friction coefficients for sliding surfaces in gate valves can increase with service before reaching a plateau.

In RIS 2000-03, "Resolution of Generic Safety Issue 158, 'Performance of Safety Related Power-Operated Valves Under Design-Basis Conditions," dated March 15, 2000, the NRC closed GSI-158 on the basis that current regulations provide adequate requirements to ensure verification of the design-basis capability of POVs, and no new regulatory requirements are needed. In RIS 2000-03, the staff stated that it would continue to work with industry groups on an industry-wide approach to the POV issue to provide timely, effective, and efficient resolution of the concerns regarding POV performance. If the actions of the industry did not adequately address the functionality of POVs under design basis dynamic conditions, the NRC staff noted that it would take additional regulatory action as appropriate.

The Joint Owners Group on Air Operated Valves (JOG AOV), which is facilitated by NEI, presented a voluntary program to address AOV issues to the NRC staff in a public meeting on June 3, 1999. The JOG AOV program provides guidance to verify valve performance at design conditions and long-term periodic verification of safety-related AOVs categorized as high-risk-significant. For safety-related, low-risk-significant AOVs and AOVs that are not safety-related but are determined to be high-risk-significant, the JOG AOV program also provides guidance for a less-rigorous verification of valve functionality. The methodology to determine valve safety significance, as specified in the industry program, may include such risk insight methods as described in Regulatory Guide 1.174, "An Approach for Using Probabilistic Risk Assessment in Risk-Informed Decisions on Plant-Specific Changes to the Licensing Basis," or programs established to meet the requirements of 10 CFR 50.65, "Requirements for monitoring the effectiveness of maintenance at nuclear power plants," in combination with individual plant examinations and the review performed by a separate expert panel. NRC comments on the JOG AOV program and its implementation were sent to NEI in a letter dated October 8, 1999. Although the program was noted to have limitations, the NRC staff recognized that industry-wide implementation of this program would achieve a uniform level of consistency that would provide increased confidence in the design-basis capabilities of high-risk-significant AOVs in nuclear power plants.

In RIS 2000-03, the NRC staff provided the following list of attributes of a successful power-operated valve design capability and long-term periodic verification program:

1. Include all maintenance rule scope POVs in the program.

2. Verify POVs in their non-safety position are capable of returning to their safety position if the train is assumed operable with the valves in their non-safety position.

3. For air-operated valves, verify guidance in GL 88-14, "Instrument Air Supply System Problems Affecting Safety-Related Equipment," has been successfully implemented, including periodic monitoring of air quality.

4. Evaluate MOV risk-ranking methodologies developed by the Boiling Water Reactor Owners Group and the Westinghouse Owners Group for applicability to risk ranking of POVs at the specific plant, as applicable.

5. Focus initial efforts on safety-related, active, high-risk POVs. Information obtained from these valves and lessons learned may be used to verify and maintain design-basis capability of similar safety-related POVs.

6. Verify methods for predicting POV operating requirements using MOV lessons learned or specific POV dynamic diagnostic testing. Use of the EPRI

MOV PPM must include all guideline aspects of that methodology and not only individual EPRI valve test results.

7. Justify the method for predicting POV actuator output capability by a test-based program established by the vendor, licensee, or industry.

8. Address all applicable weak links, including the actuator, valve, and stem.

9. Ensure quality assurance program coverage.

10. Provide sufficient diagnostics when baseline testing to verify capability. Diagnostics might not be needed if normal plant operation frequently demonstrates design-basis capability.

11. Specify when dynamic or static diagnostic periodic testing is needed.

12. Ensure post-maintenance testing is adequate to verify the capability of all safety-related POVs and risk-significant functions of non-safety-related POVs.

13. Ensure POV maintenance procedures are reviewed to incorporate lessons learned from other valve programs.

14. Upgrade training to incorporate lessons learned from other valve programs.

15. Apply feedback from plant-specific and industry information, including test data, to all applicable safety-related POVs.

16. Establish quantitative (test data) and qualitative (maintenance and condition reports) trending of POV performance with detailed review following each refueling outage.

As noted above, the NRC will continue to work with industry groups to ensure that safety-related POVs are capable of performing their specified functions under design-basis conditions.

IV. ASME Activities On POV Qualification And Inservice Testing Programs

With respect to the qualification of POVs to perform their safety functions, the ASME Committee on Qualification of Mechanical Equipment used in Nuclear Facilities has prepared a proposed revision to Section QV, “Functional Qualification Requirements for Active Valve Assemblies for Nuclear Power Plants,” of the ASME Standard QME-1, “Qualification of Active Mechanical Equipment used in Nuclear Power Plants.” The recent proposed revision to QME-1 reflects valve performance information obtained from nuclear industry programs and NRC-sponsored research since development of the QME-1 standard in the 1980s. The NRC staff is reviewing the latest revision of QME-1 for acceptance and possible endorsement in an NRC regulatory guide.

The ASME BPV Code and the more recent OM Code specifies that stroke-time testing of POVs be conducted as part of the IST programs of nuclear power plants on a quarterly frequency where practical. The NRC and the industry have long recognized the limitations of stroke-time testing as a means of assessing the operational readiness of MOVs to perform their design-basis safety functions. The NRC requires U.S. nuclear power plant licensees implementing the ASME OM Code to supplement the quarterly MOV stroke-time testing specified in the Code with a program to verify MOV design-basis capability on a periodic basis.

In response to concerns regarding the adequacy of MOV stroke-time testing, ASME developed performance-based ASME Code Case OMN-1, “Alternative Rules for Preservice and Inservice Testing of Certain Electric Motor Operated Valve Assemblies in LWR Power Plants,” as an alternative to quarterly stroke-time testing. In Code Case OMN-1, ASME allows periodic exercising of all safety-related MOVs once per refueling cycle and periodic diagnostic testing under static or dynamic conditions, as appropriate, on a frequency determined by MOV performance in terms of margin and degradation rate. In GL 96-05, the NRC staff noted that the method in ASME Code Case OMN-1 could be used as part of a licensee’s response to the generic letter.

ASME subsequently developed Code Case OMN-11, "Risk-Informed Testing for Motor-Operated Valves," to provide guidance for applying risk insights in the implementation of Code Case OMN-1. With respect to AOVs and HOVs, the ASME prepared ASME Code Case OMN-12, "Alternate Requirements for Inservice Testing Using Risk Insights for Pneumatically and Hydraulically Operated Valve Assemblies in Light-Water Reactor Power Plants." Code Case OMN-12 provides guidance for risk-informed inservice testing of AOVs and HOVs as an alternative the ASME Code provisions for these POVs. In Regulatory Guide (RG) 1.192 (June 2003), "Operation and Maintenance Code Case Acceptability, ASME OM Code," the NRC staff accepts the use of ASME Code Cases OMN-1, OMN-11, and OMN-12 with certain exceptions.

Currently, ASME is preparing a revision to Code Case OMN-1 to improve its application by clarifying several aspects of the code case while retaining the safety improvement that is achieved through increased knowledge of the design-basis capability of MOVs obtained from diagnostic testing. In addition, ASME is preparing a revision to the OM Code to revise the IST provisions for AOVs to incorporate lessons learned from industry experience. ASME is also considering revising the IST provisions in the OM Code for MOVs to incorporate the performance-based provisions of Code Case OMN-1.

V. POV Issues

Nuclear power plant licensees need to have effective programs for maintaining the capability of POVs to perform their intended functions. The nuclear industry and NRC staff share POV operating experience at user group meetings, and other public forums. NRC and ASME work to ensure that operating experience is reflected in NRC regulatory communications and Code provisions. Current issues related to proper performance of POVs at nuclear power plants include:

1. Potential preconditioning can mask degradation in POV performance prior to testing.

2. Flow-induced vibration from power uprate operation can cause unexpected and initially undetected degradation of POVs.

3. Maintenance activities can be hazardous to plant personnel because of potential energy stored in mechanical components and fluid systems.

4. Licensees implementing 10 CFR 50.69 will apply less rigorous treatment practices for safety-related POVs with low risk significance that will need to continue to provide confidence in their design-basis capability.

VI. Conclusions

The safe operation of a nuclear power plant depends on POVs in fluid systems successfully performing their safety functions. Based on lessons learned from MOV operational experience and testing programs, the NRC, ASME, and the nuclear industry have taken actions to improve the performance of POVs in nuclear power plants. Performance issues with POVs indicate the need for their continued long-term care and maintenance. The NRC staff will monitor licensee activities related to the performance of safety-related POVs through the reactor oversight program.

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DEVELOPMENT AND IMPLEMENTATION OF OPERATIONAL PROGRAMS IN COMBINED LICENSES

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Abstract

The United States Nuclear Regulatory Commission (NRC) may be receiving several combined license applications in the next few years to license new nuclear power plants. These facilities are expected to be licensed under Title 10 of the Code of Federal Regulations (10 CFR), Part 52, "Early Site Permits, Standard Design Certifications, and Combined Licenses for Nuclear Power Plants." Unlike the current fleet of operating reactors, which was licensed under 10 CFR Part 50, "Domestic Licensing of Production and Utilization Facilities," a combined license would be issued before the plant is built. Verification of the design of the facility would be made by ensuring that the specified inspections, tests, analyses, and acceptance criteria (ITAAC) were completed by the licensee. For operational programs, such as preservice testing, inservice testing and motor-operated valve programs, NRC inspectors would perform a verification of the implementation of each operational program. This issue is discussed in an NRC policy paper, SECY-05-0197, "Review of Operational Programs in a Combined License Application and Generic Emergency Planning Inspections, Tests, Analyses, and Acceptance Criteria," which was issued by the Commission on October 28, 2005. The purpose of this paper is to describe the implications of the Part 52 regulations and commission policy on (1) the development of the preservice testing, inservice testing and motor-operated valve operational programs when the combined license application is submitted, and (2) the implementation of each program after the license is issued.

Introduction

The interest in building new nuclear power plants has grown significantly in the last couple of years. At the time this paper is being published, more than 10 combined license (COL) applications are being planned by utilities in the 2007 through the 2009 time frame, which have currently operating reactors. These applications will be submitted under the new licensing process under the requirements of 10 CFR Part 52. This process allows all the design and siting issues to be addressed before the plant is constructed. Part 52 also allows the COL applicant to reference a certified design incorporated to the appendices of Part 52 and an early site permit (ESP). It should be noted that no COL applicant at this time intends to reference both a certified design and an ESP.

Future construction is being planned at sites with both nuclear plants currently licensed by the NRC and sites where there is no plant currently licensed. Potential COL applications currently indicate that their applications will reference one of three designs:

1) Westinghouse AP1000 (certified by the Commission in January of 2006)

2) General Electric Economic Simplified Boiling Water Reactor [ESBWR] (currently under design certification review by the NRC)

This paper was prepared by staff of the U.S. Nuclear Regulatory Commission. It may present information that does not currently represent an agreed-upon NRC staff position. NRC has neither approved nor disapproved the technical content.

3) Framatome EPR (currently in the early stages of design certification pre-application review, and the applicant plans to submit its design certification application in Fall 2007).

The European version of the EPR is currently being constructed in Finland.

When a COL is issued, the holder will have a license to construct and operate a nuclear plant. This license will include a set of conditions that are referred to as inspections, tests, analyses, and acceptance criteria (ITAAC). ITAAC are a set of inspections, tests, and analyses that, if successfully completed, will verify that the plant has been constructed and will operate in accordance with the Atomic Energy Act, the regulations, and the COL. All ITAAC included in the COL must be successfully completed and verified by the NRC before the licensee can load fuel into the reactor. Although further discussion of ITAAC is beyond the scope of this paper, it is a fundamental part of the Part 52 licensing process and no discussion of Part 52 is complete without the mention of ITAAC.

The NRC staff has proposed in SECY-05-0197 that certain operational programs not have ITAAC. The remainder of this paper will discuss the scope, review, and license conditions associated with operational programs in COL applications.

What is an Operational Program?

The operations of a nuclear power plant contain numerous programs administered by the licensee. A subset of these programs are required by the regulations. SECY-05-0197 focuses on programs that meet three criteria:

1. the program is required by regulation;
2. the program will be reviewed by the NRC staff for its acceptability and the results of this review documented in the staff's final safety evaluation report (FSER); and
3. the program's implementation will be verified by NRC inspectors.

The phrase "operational program" refers to programs that meet these three criteria. Table 1 lists the operational programs that meet these criteria.

Table 1: Operational Programs that Must be Addressed in a COL Application

• Containment Leakage Rate Testing	• Emergency Preparedness
• Fire Protection	• Maintenance Rule
• Operator Training	• Operator Requalification
• Plant Staff Training	• Physical Security
• Access Authorization	• Vehicle Control
• Radiation Protection	• Fitness-for-Duty
• Process and Effluent Monitoring/Sampling	• Reactor Vessel Material Surveillance
• Preservice Inspection	• Quality Assurance - Operations
• Preservice Testing	• Inservice Inspection
• Equipment Qualification	• Inservice Testing
• Motor-Operated Valve Testing	• Safeguards Contingency Plan
• Weapons Training	• Weapons Qualification/Requalification

Fully Describing an Operational Program in a COL Application

In a September 11, 2002, staff requirements memorandum (SRM) for SECY-02-0067, "Inspections, Tests, Analyses, and Acceptance Criteria for Operational Programs (Programmatic ITAAC)," the Commission provided direction to the staff that a COL applicant is not necessarily required to have ITAAC for an operational program with the exception of emergency planning (EP). The SRM stated the following:

[An] ITAAC for a program should not be necessary if the program and its implementation are fully described in a COL application and found to be acceptable by the NRC at the COL stage. The burden is on the applicant to provide the necessary and sufficient programmatic information for approval of the COL without ITAAC.

The Commission defined the phrase "fully described" in a May 14, 2004, SRM for SECY-04-0032, "Programmatic Information Needed for Approval of a Combined License Without Inspections, Tests, Analyses, and Acceptance Criteria," that reads:

In this context, "fully described" should be understood to mean that the program is clearly and sufficiently described in terms of scope and level of detail to allow a reasonable assurance finding of acceptability. Required operational programs should always be described at a functional level and an increasing level of detail where implementation choices could materially and negatively affect the program effectiveness and acceptability.

The staff concluded in SECY-05-0197 that all the programs in Table 1 could be fully described in a COL application. This description would contain the information necessary

Table 2: Operational Programs Related to Inservice Testing

Program	Regulation	Implementation Requirements
Preservice Testing	10 CFR 50.55a (f)	None for commencing program; ASME OM Code. ITSA-2000 defines perservice test period as period of time following completion of construction activities related to the component and before first electrical generation by nuclear heat.
Inservice Testing	10 CFR 50.55a(f)	ASME Operation and Maintenance Code, ISTA-2000: after first electrical generation by nuclear heat.
Motor-Operated Valve Testing	10 CFR 50.55a(b)(3)(ii)	None specified.

for the staff to make a reasonable assurance finding on the acceptability of the operational program in the review of a COL application (i.e., before the plant is built).

Implementation of an Operational Program

SRM-SECY-05-0197 specified that the COL applicant must fully describe the implementation of the operational program in the COL application. The staff must make a reasonable assurance finding on the implementation of the operational program in the review of a COL application.

Most of the operational programs listed above do not have specific implementation requirements listed in the regulations. Therefore, it is essential that the implementation of these programs be reviewed by the staff.

The staff proposed in SECY-05-0197 an implementation condition be included in each COL. It would specify that Section 13.4 of the final safety analysis report (FSAR) contain specific implementation milestones, and the implementation of these operational programs should be fully described in the same section of the FSAR in which the program is fully described. The Commission approved the staff's recommendation.

The staff also proposed a schedule license condition for each operational program. It would require a license holder to submit an implementation schedule for each operational program semiannually starting 1 year after the issuance of a COL. The frequency of submission would increase to monthly when the licensee was within 1 year of scheduled fuel load until the last operational program has been fully implemented or the plant has been placed into commercial service. The Commission also approved the staff's proposal.

Operational Programs Related to Pump and Valve Inservice Testing

Three operational programs are related to inservice testing (IST) of pumps and valves. Table 2 provides the reference to the specific regulation that requires the program and the implementation requirements, if any, specified in the regulations.:

At the time this paper was being drafted, guidance for the information needed for the NRC to review these three operational programs was being developed by the staff in a new regulatory guide for COL applications. The draft of this regulatory guide is scheduled to be issued in summer 2006.

Alternate Treatment for Operational Programs

SECY-05-0197 states that a COL applicant may, at its option, choose to submit a complete program description for any particular program, but omit implementation information and instead include ITAAC. The staff also notes that unique circumstances involving a particular application may raise an implementation issue on an operational program that is best resolved by an ITAAC. The staff expects such circumstances to be rare.

Conclusion

Combined license applications are being prepared for nuclear plants. The first of these applications are scheduled to be submitted in Fall 2007. These applications will be submitted under 10 CFR Part 52 which allows the staff to issue a provisional license to construct and operate a commercial nuclear plant. Operational programs will be reviewed in those COL applications and the staff will make a reasonable assurance finding on the acceptability of the program and its implementation to support the issuance of the COL. The NRC will inspect the implementation of the operational program to ensure that it is being implemented as described in the application. Guidance for including an adequate description of the preservice testing, inservice testing, and motor-operated valve testing programs will be contained in the COL application regulatory guide currently being developed. A draft of the guide will be available this summer.

PUMP OPERATIONAL EXPERIENCE AT U.S. NUCLEAR POWER PLANTS

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Ninth NRC/ASME Symposium on Valves, Pumps, and Inservice Testing
July 2006

This presentation will discuss recent operational experience with the performance of pumps at U.S. nuclear power plants. The presentation will discuss the cause of pump performance issues and the corrective action in response to those issues. The discussion will provide information that could have generic applicability in maintaining the proper performance of pumps at all nuclear power plants.

This presentation will be made by staff of the U.S. Nuclear Regulatory Commission. It may present information that does not currently represent an agreed-upon NRC staff position. NRC has neither approved nor disapproved the technical content.

Pump Air Entrainment
- How Lack of Analysis Can Translate Into
A Potential Safety Issue and Costly Plant Shutdown

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Abstract

Pump air entrainment - a well-recognized pump performance phenomenon that is occasionally not addressed during pump and process system design. The lack of such analysis can result in significant questions regarding the ability of the pump to perform during required system conditions and accident scenarios. This paper discusses the importance of establishing pump design analysis and the potential safety consequences of not having such analysis.

Protection of Safe Shutdown Equipment During Design Bases Events

One of the basic tenets of nuclear power safety is that facilities are designed, constructed, operated, and maintained as described in the facility's Updated Final Safety Analysis Report (UFSAR). The UFSAR describes the design bases events which plants must be designed to withstand and achieve safe shutdown. We all know about the "big ones": large break loss of coolant accident, loss of offsite power, steam generator tube rupture, etc... Occasionally, however, licensees and the NRC find that other events have not been thoroughly evaluated to ensure that plants can safely shut down upon occurrence. Such was the case at the Kewaunee Nuclear Power Plant (KNPP), located near Green Bay, Wisconsin.

The KNPP facility is located within the NRC Region III geographic location and, in 2005, was selected for a pilot engineering inspection. The pilot inspection approach was based on high risk, low margin components to evaluate component acceptability, as opposed to the traditional system-focused engineering inspection. The KNPP plant design is a typical early-vintage, Westinghouse 2-loop Pressurized Water Reactor, with an auxiliary feedwater (AFW) system designed to supply water to the steam generators (SGs) to remove decay heat from the reactor coolant system following postulated design bases events. The AFW system consists of two motor-driven pumps and one steam turbine-driven pump for providing the source of heat removal. The AFW pumps are normally aligned to two non-safety-related 75,000-gallon Condensate Storage Tanks (CSTs). The plant's service water (SW) system provides the Class 1 backup source of water.

In the aftermath of the Three Mile Island accident, the NRC required that licensees evaluate the design of AFW systems to determine if automatic protection of the AFW pumps was necessary following a seismic event or tornado. The primary concern was that an unprotected pump suction source (from the CST, in Kewaunee's case) could result in pump damage prior to the suction supply being shifted to the safety-related water supply.

To address the NRC requirement, Kewaunee installed a low discharge pressure pump trip signal to protect the AFW pumps against loss of suction head. The primary

This paper was prepared by staff of the U.S. Nuclear Regulatory Commission. It may present information that does not currently represent an agreed-upon NRC staff position. NRC has neither approved nor disapproved the technical content.

reason for installing the trip signal on the discharge piping, rather than on the suction piping, was to address concerns regarding the pump operating at sub-atmospheric pressures due to the long suction piping run from the CST to the AFW pumps (approximately 300 feet). Therefore, the low pump discharge pressure trip would be indicative of a loss of suction pressure and hence the pumps would be protected.

Questions Regarding Adequacy of Design and Plant Shutdown

In preparation for the NRC engineering inspection, the licensee documented the lack of a definitive basis for the AFW pump discharge pressure trip setpoints (350 psig for the two motor driven pumps and 100 psig for the turbine driven pump). Notwithstanding the lack of engineering analysis to support the trip setpoints, the licensee initially concluded that the AFW pumps remained operable (primarily based on Net Positive Suction Head [NPSH] considerations) pending further analytical reviews. The NRC questioned the licensee if the potential for air ingestion and pump damage had been evaluated for the pump discharge pressure trip design. This question was crucially important in determining pump and system operability. The primary NRC concern was that the CST supply to the AFW pumps was a common line, and a failure of the non-Class I portion of the line due to a seismic event or tornado could cause a common mode failure affecting all three AFW pumps. This, in turn, would result in air ingestion into the pumps, leading to air binding and failure of the AFW pumps. After extensive re-analysis and evaluations, the licensee ultimately concluded that the low discharge pressure trips would not perform the intended function of protecting the AFW pumps from a loss of suction supply. Accordingly, the plant commenced an extended plant shutdown to address the system operability concerns. Additionally, using hydraulic models developed following the inspection, the licensee also determined that the AFW pumps were not adequately protected from a pump runout condition.

The Solution: Protected Volume and Operator Manual Actions

As the licensee proceeded with developing an AFW hydraulic analysis, conceptual design work began on achieving an acceptable resolution to the air pump entrainment and pump runout issues. The pump air entrainment issue was resolved by establishing a protected volume of water supply to the AFW pumps along with corresponding pump low suction pressure switches. The pump runout problem was resolved by changing the design and licensing basis of the existing pump discharge trip setpoints.

<u>Protection Against Air Entrainment</u>

The protection of the AFW pumps against a loss of normal water supply from the non-safety CSTs required the modification of existing suction piping to add a protected volume. The existing suction piping was re-sized and re-routed to withstand a seismic event and be protected from tornado effects and high-energy line break interactions. In essence, the Class I boundary break was re-established further upstream of the existing Class I break near the pump. Additionally, three suction pressure switches were added on the new protected AFW suction piping to sense a loss of suction pressure and initiate a trip signal to the respective AFW pump. A primary consideration for establishing the low suction pressure pump trip was a postulated catastrophic failure of the non-Class I suction piping. Following such an event, the low suction pressure switches would trip the respective AFW pumps before pump damage occurred. The additional water volume in the suction piping provided a margin for the AFW pump to coast to a stop before the water in the piping was lost.

The pressure switch setpoint development required consideration of the piping pressure drop between the suction pressure switches and the pumps. The setpoint also took into consideration sub-atmospheric conditions that could exist at the AFW pump suction. Pump operation at sub-atmospheric conditions has the potential to damage the AFW pumps due to a loss of seal leak-off which lubricates and cools the pump's packing. Therefore, the licensee's setpoint needed to ensure that the AFW pump suction pressure would be equal to or greater than atmospheric pressure.

<u>Protection Against Pump Runout</u>

The licensee's AFW system hydraulic analysis determined that, with steam generator (SG) pressures above 650 psia, the AFW pumps would not reach run out conditions and actuate the existing low discharge pressure switches. With the exception of a main steam line break (MSLB), all design basis events resulted in SG pressures greater than 750 psia.

The licensee proposed to maintain the existing low discharge pressure switches, but revise the design and licensing basis of the components. The discharge pressure switches would continue to be used for NPSH pump protection. The inadequate available NPSH condition existed due to the AFW pumps being flow limited because of suction line losses. The discharge pressure switch set points were adjusted to trip the pumps before runout condition resulted.

For the MSLB event, the licensee proposed to prescribe local, manual operator actions to isolate the faulted SG and throttle the AFW pump flow. Several considerations were required to ensure that these actions were acceptable. This included operator training, accessability, timeline validations to accomplish the actions, design of discharge valves, and the need for a test demonstrating the capability of the turbine-driven AFW pump at low SG pressures as the plant commenced heat up.

To provide assurance of the acceptability of the AFW system hydraulic analysis and plant modifications, the licensee performed additional actions including a simultaneous start of all three AFW pumps at hot shutdown conditions. Also, each individual AFW pump was subjected to a timed coastdown test at bounding flow rates.

The NRC approved the licensee's resolution of the AFW issues in the form of an Amendment to the facility's Technical Specifications (TS). This regulatory action allowed the plant to recover from the extended shutdown.

The Lessons: Engineering Rigor, Regulatory Impact, Extended Shutdown

Questions by the licensee, regarding the adequacy of the AFW system design, first surfaced in preparation for the NRC's re-vamped pilot engineering inspection. Following extensive discussions and concerns expressed by the NRC, and after several weeks of engineering review of the basis for the discharge pressure trip setpoints, the licensee ultimately concluded that a lack of confidence in the setpoint basis could not support system operability as required by plant TS. Accordingly, the plant commenced a shutdown on February 19, 2005. The engineering challenge of re-design of the AFW system to address the potential loss of suction supply, along with resolving a plant internal flooding deficiency, resulted in an extended shutdown which ended when full power operations resumed on July 4, 2005. Needless to say, while plant and public safety dictated such a plant shutdown, the economic costs of a shutdown in the heart of the Wisconsin winter are both measurable and significant.

The NRC evaluated the risk-significance of the engineering design deficiency and ultimately concluded that the issue was of low to moderate safety, or a White finding, in NRC risk terminology. Accordingly, the NRC will factor the risk significance of the issue, along with any other findings and performance indicators, in determining what column of the agency action matrix the licensee's performance resides. Appropriate additional regulatory inspections will ensue.

As discussed in SECY 04-0071, in 2004, the NRC staff performed an analysis of the previous 3 years of inspection data from the NRC's Reactor Oversight Process (ROP). The analysis was performed to better understand the degree to which NRC inspections and licensee self assessment efforts have been effective in identifying design issues. Of the 17 greater than green design/engineering issues that fell within the scope of the review, 11 were NRC-identified, 2 were licensee-identified, and 4 were self-revealing. Of the 11 NRC-identified issues, 7 involved issues that had been previously recognized by the licensee but had not taken adequate corrective actions.

The staff also performed a review of the results of recent NRC design inspections conducted at Point Beach and Davis Besse; facilities where the licensee had identified significant design issues. The results highlighted the need for aggressive licensee self-assessments in the design area and effective corrective action programs that can evaluate and resolve the identified issues in a timely manner. The results also revealed that in some instances, the NRC had indications of programmatic design/engineering weaknesses, but did not engage further, as the programmatic weaknesses had not yet resulted in issues that could be classified as risk-significant. While this regulatory approach is in accordance with the fundamental element of the NRC's ROP, it re-emphasizes the importance of licensee's corrective action programs and self-assessments efforts.

It is often said that the plant's engineering organization serves as the plant's design and licensing basis conscience. The engineering staff own and maintain the operational margin which is often consumed by poor maintenance practices, operator errors, procedure weaknesses, and degraded components. As stated earlier in this paper, the NRC's pilot engineering inspection focused on such components: high risk, low margin. The erosion of that margin is in many instances a hidden unknown. Some margin is easily discernible; pump flow capacity exceeds requirements by x gallons per minute. Others are not so easily discerned; as is the case with calculational errors or unverified assumptions. There is nothing new in the issues discussed in this paper; but rather it should serve as a reminder that adherence to the well-documented engineering principles of design must be maintained to ensure that design and licensing basis commitments are not victims to other competing priorities.

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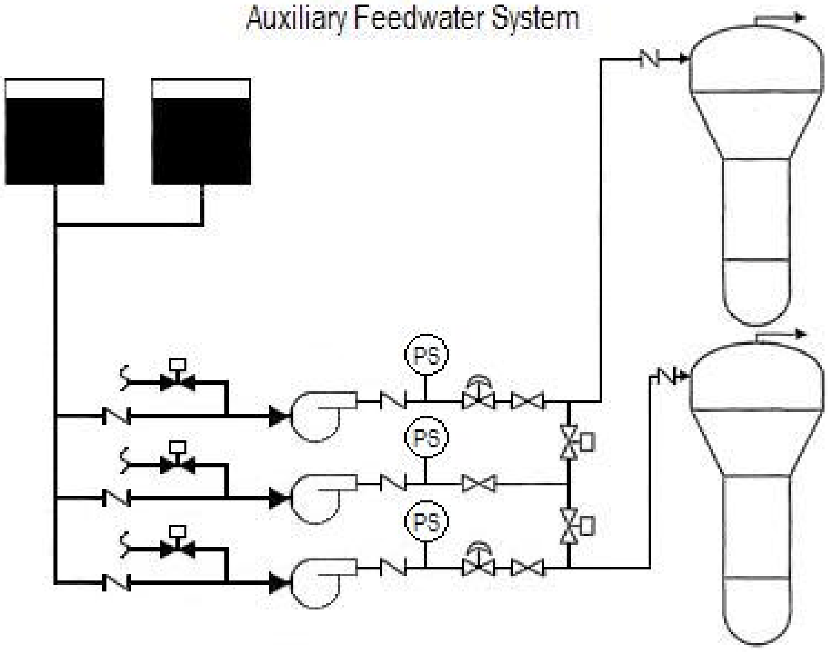
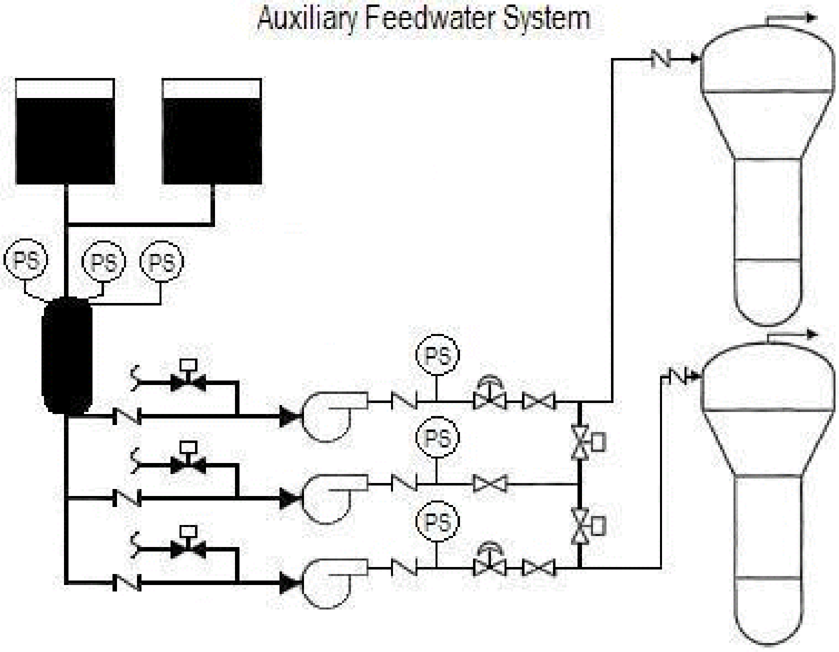
Pre-Modification
Auxiliary Feedwater System



Post-Modification
Auxiliary Feedwater System



Nuclear Power Plant Pump and Valve Inservice Testing Issues

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Abstract

This paper discusses recent issues related to inservice testing (IST) of pumps and valves at U.S. nuclear power plants. These issues were identified during the review by U.S. Nuclear Regulatory Commission (NRC) staff of IST programs and relief requests, and applicable operating experience. This discussion includes information that could have generic applicability in the implementation of effective IST programs at U.S. nuclear power plants.

Introduction

The NRC staff has encountered a number of pump and valve inservice testing (IST) issues since the Eighth NRC/ ASME Symposium on Valve and Pump Testing in 2004. This paper discusses issues involving pump vibration, the frequency range of vibration-measuring transducers, and valve grouping for online testing of check valves. The paper discusses the relief requests received related to these issues and the NRC safety evaluations of the requests. Some current staff positions and actions in these areas are discussed. This discussion includes information that could have generic applicability in the implementation of effective IST programs at U.S. nuclear power plants.

This paper was prepared by staff of the U.S. Nuclear Regulatory Commission. It may present information that does not currently represent an agreed-upon NRC staff position. NRC has neither approved nor disapproved the technical content.

Check Valve Sample Disasembly And Inspection Online

Subsection ISTC of the American Society of Mechanical Engineers (ASME) Code for Operation and Maintenance of Nuclear Power Plants (OM Code) - 2001 with 2003 Addenda, paragraph ISTC-5221(c) allows disassembly of check valves every refueling outage as an alternative means to verify their operability. Instead of disassembly every refueling outage, ISTC-5221(c) provides the option of using a sample disassembly and inspection program for groups of identical valves in similar application. Paragraph ISTC-5221(c)(1) states that grouping of check valves for a sample disassembly examination program shall be technically justified and shall consider, as a minimum, valve manufacturer, design, service, size, materials of construction, and orientation. Further, ISTC-5221(c)(3) states that at least one valve from each group shall be disassembled and examined at each refueling outage, and all valves in each group shall be disassembled and examined at least once every 8 years. The Code requirements are based on Generic Letter (GL) 89-04, "Guidance on Developing Acceptable Inservice Testing Program."

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Paragraph ISTC-3510 states that check valves shall be exercised nominally every 3 months. Paragraph ISTC-3522(c) states that if exercising is not practicable during operation at power and cold shutdown, it shall be performed during refueling outages.

More and more licensees are requesting to disassemble and inspect check valves online to reduce refueling outage time and required manpower during outages. A number of licensees have proposed, as an alternative, to perform the IST disassembly and inspection activities during normal plant operation (online), in conjunction with appropriate system outages, instead of during refueling outages. It is evident that selected refueling outage inservice testing activities could be performed during system outages online without sacrificing quality or safety. In any case, check valves disassembly, inspection, and manual exercising will be performed at least once each operating cycle on a refueling outage frequency. NRC staff has authorized online testing of check valves on a case by case basis.

Recently, the NRC has received relief requests where licensees propose, as an alternative, to perform online sample disassembly and inspection IST activities of check valves in a group.

ISTC-5224 requires that check valves in a sample disassembly program that are not capable of being full-stroke exercised or have failed or have unacceptably degraded valve internals, shall have the cause of failure analyzed and the condition corrected. ISTC-5224 also states that other check valves in the sample group that may also be affected by this failure mechanism need to be examined or tested during the same refueling outage to determine the condition of internal components and their ability to function.

Therefore, when submitting relief requests for check valve group sample disassembly and inspection online, licensees must consider the provisions as specified in paragraph ISTC-5224. Licensees can not defer disassembly and inspection of other check valves in the group. Therefore, online sample disassembly and inspection IST activities for check valves in a group is not recommended unless the allowed outage time (AOT) provides sufficient time to permit the inspection of all valves in the group. The staff has found online disassembly and inspection of valve groups containing one valve acceptable.

Pump Vibration Measuring Instruments (Transucers) Issue

The NRC has received requests from various licensees for relief from the provisions of ISTB-3510(e) of the ASME OM Code for pumps with low pump shaft rotational speeds. Paragraph ISTB-3510(e), "Frequency Response Range," requires that the frequency response range of the vibration-measuring transducers and their readout system shall be from one-third minimum pump shaft rotational speed to at least 1000 Hz.

Most of the licensees stated that procurement and calibration of instruments to cover the lower end of the Code-specified range was impractical due to the limited number of vendors supplying such equipment, the level of equipment sophistication required, and the equipment cost. Therefore, past relief requests were typically authorized pursuant to 10 CFR 50.55a(a)(3)(ii) on the basis that compliance with the specified Code provision would result in hardship without a compensating increase in the level of quality and safety. The NRC staff prepared safety evaluations authorizing these relief requests.

Since then, the NRC has learned that, due to technology advancement and research work performed in the field of instrumentation, vibration-measuring transducers meeting the Code provisions can be easily procured from various suppliers at a reasonably low cost.

Recently, similar relief requests were received from various licensees. After review, requests for additional information, and followup discussion by the NRC, the licensees withdrew the relief request and decided to install a new transducer that met the Code provisions. Therefore, licensees are requested to carefully examine the availability, procurement, and related cost of the Code-required instruments (vibration-measuring transducers) before submitting a relief request in this area to the NRC.

High Pressure Coolant Injection (HPCI) Pump Vibration Issues

The NRC has received a number of relief requests related to HPCI pump vibration measurement criteria shown in the alert and required action ranges of Table ISTB-5100-1. The NRC staff has authorized HPCI pump vibration relief on a case by case basis, after reviewing licensees' additional monitoring and data and other justification. Recently, the NRC has received similar relief requests along with requests for relief from the provision of paragraphs ISTB-5121(d) or ISTB-5123(d), and ISTB-5121(e) or ISTB-5123(e). Paragraph ISTB-5121(d) and ISTB-5123(d) state that "Vibration (displacement or velocity) shall be determined and compared with corresponding reference values. Vibration measurements are to be broad band (unfiltered). If velocity measurements are used, they shall be peak. If displacement amplitudes are used, they shall be peak-to-peak."

Paragraph ISTB-5121(e) and ISTB-5123(e) specify that all deviations from the reference values shall be compared with the range of Table ISTB-5100-1, and corrective action taken as specified in paragraph ISTB-6200. The vibration measurements shall be compared to the relative and absolute criteria shown in the Alert and Required Action Range of Table ISTB-5100-1. For example, if vibration exceeds either 6 V_r or 0.7 inch/second, the pump is in the Required Action Range.

In one relief request, the licensee stated that the peak vibration amplitude was not related to the physical condition or rotating dynamics of the main pump rotor or bearing system. Therefore, the licensee proposed to filter the measured vibration values of the pump, such that filtered vibration values met the Code provisions of Table ISTB-5100-1. As mentioned above, Subsection ISTB of the OM Code specifies that vibration measurements be broad band (unfiltered). A typical spectrum analysis is a means to gather information as to the source of a potential vibration problem. The licensee-proposed alternative to filter the peak vibration values would only hide the vibration peak and would not correct the elevated pump vibration levels. The filtered vibration measurement would only remove the vibration signal from the calculation, not at the pump. The licensee's proposal masked elevated vibration levels by removing them from consideration. Therefore, the staff did not find the licensee's proposed filtering of the peak values acceptable.

The staff found the proposed alternative did not provide an acceptable level of quality or safety because the alternative did not provide reasonable assurance of the long-term operational readiness of the pump. For long-term assessment of the operational readiness of the pump, it is necessary that pump vibration meet the OM Code provisions as specified in Table ISTB 5.2.1-1 without filtration of vibration signal.

In addition, the licensee did not demonstrate that compliance with Code provisions would result in hardship or unusual difficulty without a compensating increase in the level of quality and safety. The NRC staff is aware that the HPCI pump supplier (Byron Jackson) performed inspections and collected vibration data from HPCI pumps at various nuclear power plants and provided various recommendations to reduce vibration levels. The NRC staff has found that some of the licensees who performed the design modification per Byron Jackson recommendations, were able to reduce HPCI pump vibration levels. Although the need to implement the Byron Jackson recommended modifications requires resources, the modification would likely lower the actual vibration levels of the HPCI pump.

Conclusion

The purpose of this paper was to make licensees aware of a number of pump and valve issues that the staff has encountered since the Eighth NRC/ASME Symposium on Valve and Pump Testing in 2004. Licensees who believe that some of the items discussed are applicable to their facilities may wish to review their current IST program and modify their program as appropriate.

References

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ASME/ANSI, Code for Operation and Maintenance of Nuclear Power Plants, 2001 Edition and 2003 Addenda:

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10. SUPPLEMENTARY NOTES

11. ABSTRACT *(200 words or less)*

The 2006 Symposium on Valves, Pumps, and Inservice Testing, jointly sponsored by the Board on 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, programmatic, and regulatory issues associated with inservice testing programs at nuclear power plants, including design, operation, and testing of valve and pumps. The symposium provides an opportunity to discuss improvements in design, operation, and testing of valves and pumps that help ensure their reliable performance. The participation of industry representatives, regulatory personnel, and consultants ensures the presentation of a broad spectrum of ideas and perspectives regarding the improvement of inservice testing programs and methods for valves and pumps at nuclear power plants.

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