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July 14-16, 2008

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Abstract

The 2008 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, pumps and dynamic restraints. The symposium provides an opportunity to discuss improvements in design, operation and testing of valves, pumps and dynamic restraints that help to ensure their reliable performance. The participation of industry representatives, regulatory personnel, and consultants ensures the presentation 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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Statements and opinions advanced in the papers presented at the Tenth 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, Metric units have been used as an expression of current industry practice with English units also indicated where possible.

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A Novel Fouling Mitigation Method for Jet Pump Inlet Mixers

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ABSTRACT

GE-Hitachi Nuclear Energy has developed an advanced technology for preventing electrostatic deposition of charged particles on the metal surfaces of BWR components. The approach is based on modifying the electrochemical property of the metal surface to mitigate the oxide fouling on the surface. Fouling is a thick/dense oxide crud layer, which deposits over time, on components exposed to high flow water, such as the nozzles and throat areas of jet pump assemblies, impellers, condenser tubes, and steam generator parts. This crud buildup substantially reduces water velocities and degrades the performance of the recirculation system. This degradation will affect the efficiency of the plant because the recirculation pumps must be run at a higher speed to maintain core flow. Degraded jet pump performance can also cause extreme jet pump vibration and damage to jet pump components. Eventually, the jet pump inlet mixer (including the nozzles and throat area) must be either mechanically cleaned with a high pressure spray or replaced. This is a costly and time consuming procedure. Recently it has been found that the fouling can be mitigated by applying an antifouling coating such as TiO_2 to the metal surface. Since the fouling oxide particles have a zeta potential of the same sign as the TiO_2 coating, the oxide particles are not attracted to the surface. Thus, applying a few micron layer of TiO_2 coating by chemical vapor deposition (CVD) improves the efficiency of the jet pump by mitigating the fouling mechanism. It is a cost effective method, which reduces the oxide fouling mechanism, increases core flow and potentially eliminates the cleaning process.

I. INTRODUCTION

In a boiling water reactor (BWR), an annular space is defined between the core shroud and the reactor pressure vessel wall. As shown in Figure 1, jet pumps are located in the annular space for recirculating coolant through the reactor. Jet pumps, which contain no moving parts, provide an internal circulation path for the core coolant flow. Typically a substantial number of jet pumps, for example, from sixteen to twenty-four, are installed in the annular space. A jet pump consists of four main sections: the nozzle section, inlet mixer section (throat barrel), flare and diffuser (Reference Figure 2). A high velocity coolant flow moves through the nozzles and discharges into the inlet mixer. Oxide corrosion products are part of the coolant flow, and over time these oxide particles build up on the inside 304 stainless steel surfaces of the nozzles and the inlet mixer forming a layer of "crud" (fouling) [1-3]. The fouling is caused by charged oxide particles suspended in the coolant, which interact with the metallic inner surface of the jet pump. The greatest degree of fouling is observed in areas that experience a high velocity flow rate, i.e., nozzle and throat, shown in Figure 2. In the event that the fouling layer becomes excessive, the performance of the jet pump will be degraded. Degradation of jet pump performance can result in jet pump vibration and a reduction in core flow capability. Eventually the nozzles/inlet mixer must be mechanically cleaned or replaced during regular maintenance and refueling outages. During the cleaning process, the jet pump is moved from the reactor to the fuel pool where it is exposed to a high pressure water spray, which removes the crud layer. Over the life of the plant, the jet pumps may require several cleaning procedures to restore core flow to optimum conditions. It is noted, however, that repeated cleanings will not restore flow to its original licensed condition. As such, it became compelling to find a method that would address the above problem. This paper provides a fundamental description of the oxide fouling mechanism and discusses the novel method of applying an antifouling coating to the jet pump inner surfaces. This antifouling layer repels the charged particles from the reactor coolant and thus mitigates the fouling.

II. THEORY OF OXIDE FOULING

Metal oxide surfaces in the reactor coolant develop surface electrical charges and also ions in solution, some of which are formed from metal oxide corrosion particles. As shown in Figure 3, those ions having a charge opposite to that of the oxide surface form a layer that is strongly attracted and attached by electrostatic interaction. The interaction is strong when the particle is within the radius of action of the electric field generated by the oxide surface. The surface-particle bond will prevent the particle from being dragged away by the flowing water. There are several factors affecting the range of the electric field: (1) the conductivity of the aqueous solution in contact with the oxide surface, (2) the solution temperature, and (3) the solution velocity relative to the oxide surface. The combination of low conductivity, high temperature, and high flow velocity tends to favor the extension of the electric field into the bulk solution, and consequently the higher deposition of metal oxides (fouling) on the surface.

A parameter that relates to this phenomenon is referred to as the zeta potential [4-7]. The zeta potential accounts for the fouling noted on jet pump surfaces. For fouling to occur the zeta potential of the oxide surface must be opposite in sign to the zeta potential of the ions in solution. Thus one method to avoid fouling is to apply a coating to the metal oxide surface that has a zeta potential of the same sign as the ions in solution.

In Figure 4 is presented the isoelectric point of surface (IEPS) curve for several oxides [8]. Since the zeta potential depends on the solution pH, the pH at which the surface/particle charge is zero is called the pH of zero charge (pzc) or the isoelectric point of surface (IEPS). In neutral water, metal oxides having an IEPS > 7 have a positive surface charge, whereas those having an IEPS < 7 have a negative surface charge. From a review of the IEPS curve, it is evident that the Fe_3O_4 particles will be attracted to the Fe_2O_3 metal oxide surface. It is further noted that the oxides including Ta_2O_3, TiO_2, and ZrO_2 have IEPS values close to that of the Fe_3O_4 particles. As such, the zeta potentials of Ta_2O_5, TiO_2, ZrO_2 and Fe_3O_4 will have the same sign in the BWR environment. Thus by coating the metal surface with an antifouling layer such as Ta_2O_5, TiO_2, or ZrO_2, the attraction between the charged oxide particles will be eliminated. Upon evaluating the above antifouling oxides with regard to properties, ease of deposition, and the BWR operating conditions, TiO_2 was chosen as a coating of interest.

III. EXPERIMENTAL

III.A. Preparation of Test Specimen

An experiment was setup to simulate the fouling mechanism in actual operating conditions of the reactor cooling water systems. A high temperature/ high flow loop as shown in Figure 5 was fabricated for that purpose. The high flow dynamic conditions occurring at the jet pump nozzle and inlet mixer were simulated by using a venturi tube. The ID surface of the high flow venturi tube was mechanically cleaned, washed with acetone and distilled water, and preoxidized in 288 ^{o}C water containing 250-300 ppb O_2 for 3 weeks before testing in a given water chemistry condition. Also, a piece of 304 SS tubing (¼"or 1/8" diameter x 5' long) was connected to the outlet of the test venturi tube for examining the oxide surface morphology before and after the iron injection. The venturi tube was used for the iron oxide deposition test by measuring the pressure change and relating this to the degree of fouling. This high flow venturi loop generates a linear flow rate up to 35 ft/sec (10.5 m/sec).

III.B. Water Chemistry Control

A desired concentration of Fe as Fe $(NO_3)_3$ in high purity water was injected directly into the inlet flow of the high flow venturi tube using a PTFE tube contained in a 0.125 cm OD SS tube. The concentration of Fe in the inlet and outlet water was also measured by inductively coupled plasma mass spectroscopy (ICP-MS) during the experiments. Inlet water to the autoclave and the effluent stream from the autoclave were monitored for resistivity and desired O_2 and H_2 concentrations.

III.C. TiO_2 Coating by CVD

Following the above baseline study, a venturi tube and 304 SS tubing were coated with Titania by the chemical vapor deposition (CVD) method. The above experiment was repeated with the coated specimens. The titanium dioxide (TiO_2) coating was evaluated for coating thickness, which was determined to be about 1.5 microns. The presence of Ti on the surface was examined by scanning electron microscopy (SEM) and x-ray photoelectron spectroscopy (XPS).

IV. RESULTS AND DISCUSSION

IV.A. Iron Oxide Deposition on 304 SS Surface

Figure 6 shows the effect of water chemistry condition on the differential pressure change in the venturi tube (no coating on the ID surface) as a function of immersion time in 288^{o}C water at a flow rate of 20 ft/sec (6 m/sec) before and during 250-300 ppb Fe injection. Initially, the preoxidized venturi tube was immersed for 3 days in 288^oC water containing 250 ppb O_2 without Fe injection. The purpose of this approach was to retain the oxide structure formed during preoxidation that might have changed during the shutdown of the system used to preoxidize the venturi tube.

The increase in pressure change with Fe injection indicates the possible build-up of Fe oxides on the ID surface of the venturi tube, and a further injection shows the steady state conditions on pressure change. Another increase in pressure change was observed when the water chemistry changed from 250 ppb O_2 to 150 ppb H_2. This pressure response in the NWC/HWC environment indicates that the use of the venturi tube provides a useful tool to perform the oxide deposition behavior in high temperature and high flow water. Figure 7 shows the oxide surface morphology of the 304 SS tubes before and after the Fe injection test. It is clearly evident that Fe oxide particles were deposited on the ID surface of the tube specimen after Fe injection.

IV.B. Iron Oxide Deposition on TiO_2 Coating Layer

Figure 8 shows the XPS results for the TiO_2 coating prepared by chemical vapor deposition CVD. The coating is a uniform TiO_2 layer with about a 1.5 μm thickness. The pressure change in a TiO_2 coated venturi tube was measured as a function of immersion time in 288^oC water at a flow rate of 20 ft/sec (6 m/sec) with and without 250-300 ppb Fe injection, as shown in Figure 9.

During the iron injection in 300 ppb O_2, the unstable pressure change was observed, suggesting that the possible build-up and spalling-off of Fe oxides on the ID surface of the coated venturi surface was occurring. When the water chemistry changed from 300 ppb O_2 to 150 ppb H_2, an increase in pressure was measured that resulted from the iron oxide deposition. However, the degree of pressure change for the TiO_2 coated venturi surface is much smaller than the one observed for the non-coated venturi surface. Figure 10 shows the surface morphology of the TiO_2 coating before and after the Fe injection test. It is clearly evident that a much smaller amount of Fe oxide particles were imbedded on the TiO_2 coated surface after Fe injection compared to the uncoated SS surfaces.

As shown in Figure 11 (comparison of Figures 6 and 9), it is evident that the TiO_2 coating layer decreases the iron oxide deposition. Jayaweera et al. measured the zeta potential of various oxides in high temperature water and, based on the same sign of zeta potential and similar IEPS between TiO_2 and magnetite, suggested a TiO_2 coating as a possible candidate oxide coating to prevent corrosion product deposit [6]. Also, Ta_2O_5, Cr_2O_3, WO_3, and ZrO_2 are suggested as potential anti-fouling coatings.

V. SUMMARY

Anti fouling coatings, such as TiO2, were examined in high temperature and high flow water under various water chemistry conditions. A TiO2 coating prepared by CVD showed the beneficial layer for decreasing the iron oxide deposition under NWC and HWC conditions.

ACKLOWLEDGMENT

The CVD Titania coating was produced by Dr. Raj Israel and his group at GE Consumer and Industrial Products. Lauraine Denault and John Chera at GE Global Research Center performed the surface analyses.

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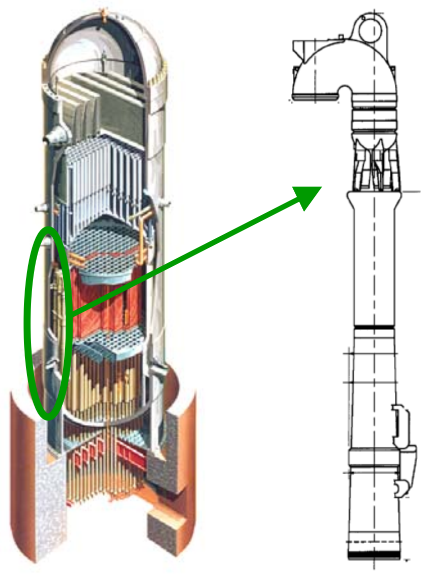

Figure 1. Jet pumps are located in the annular space for recirculating coolant through the reactor.

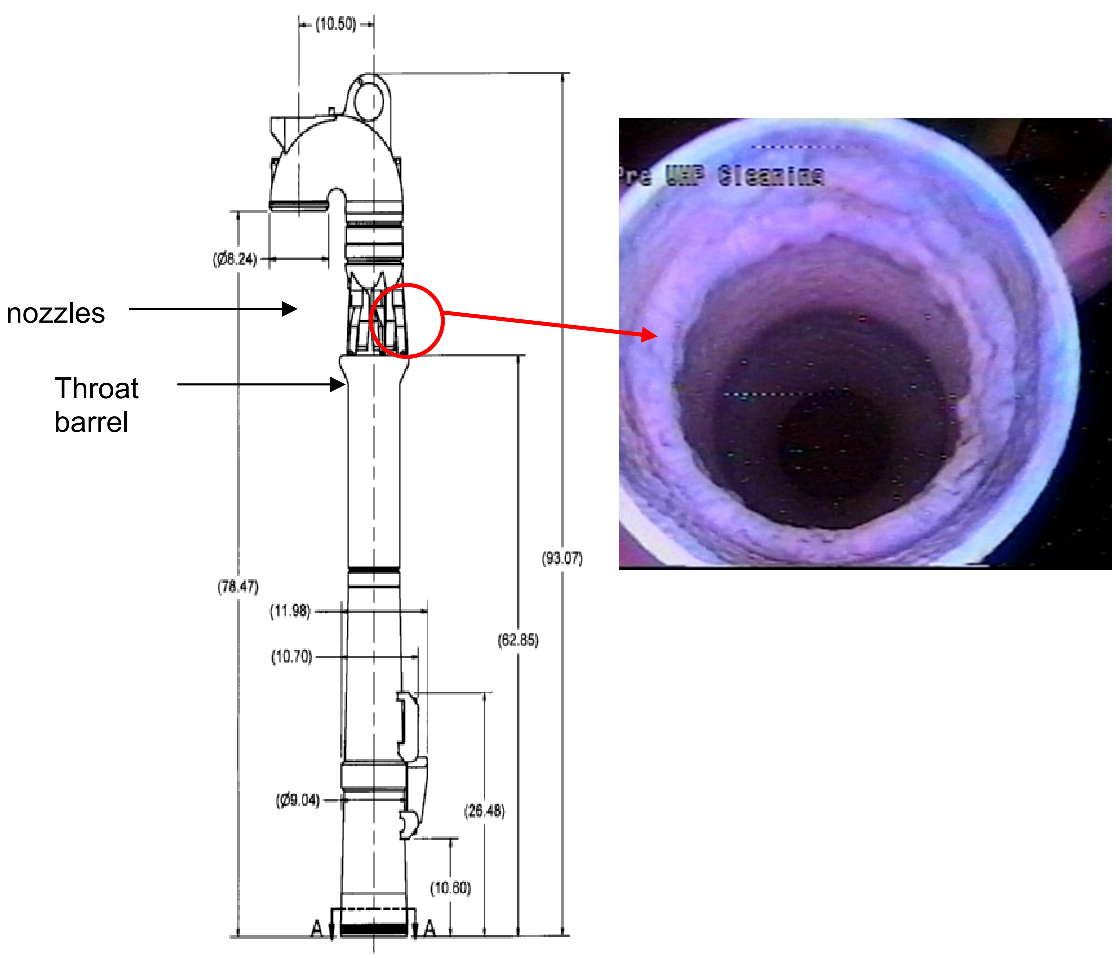

Figure 2: Schematic of jet pump and crud deposition on jet pump nozzles

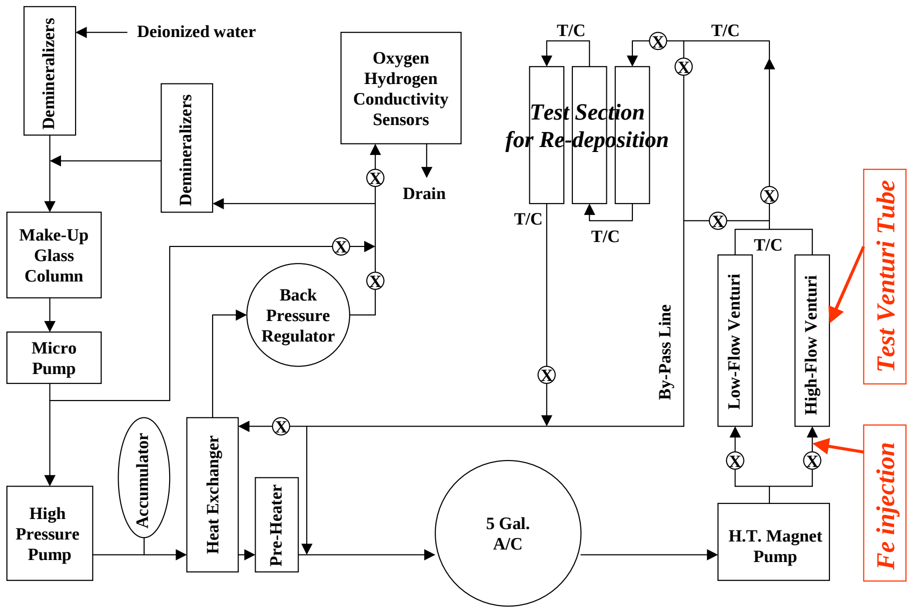

Figure 3. Experimental arrangement of the high temperature and high flow loop for oxide deposition.

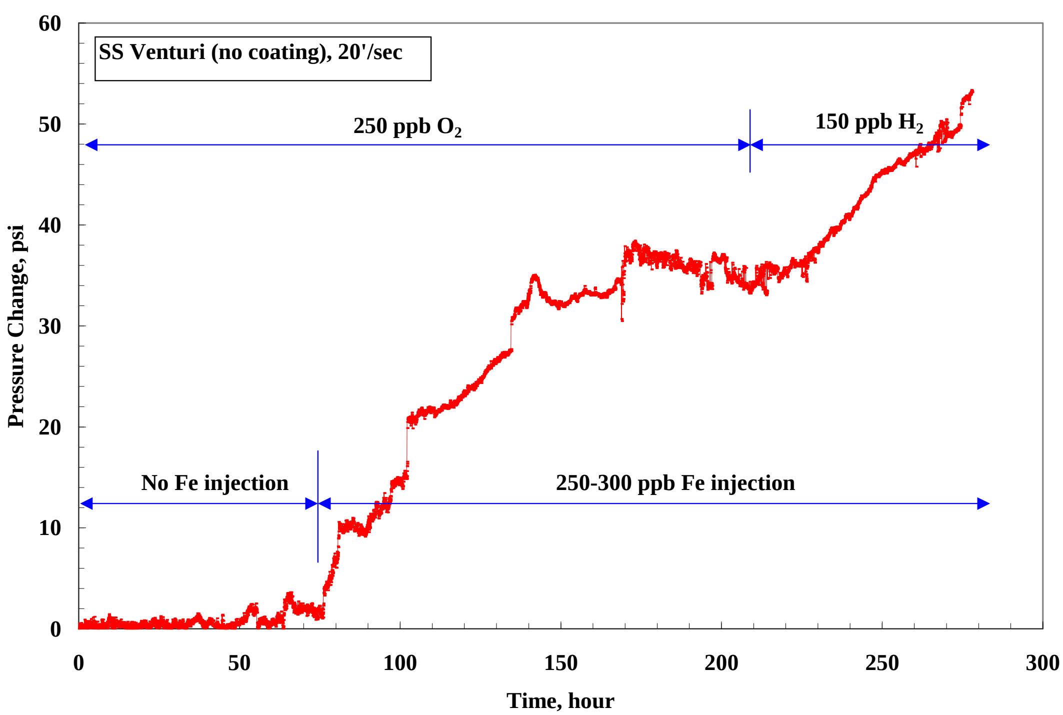

Figure 4: Effect of water chemistry condition on differential pressure change on 304 SS venturi tube as a function of immersion in 288^{o}C water with and without injection of 250-300 ppb Fe at 20 ft/sec (6 m/sec).

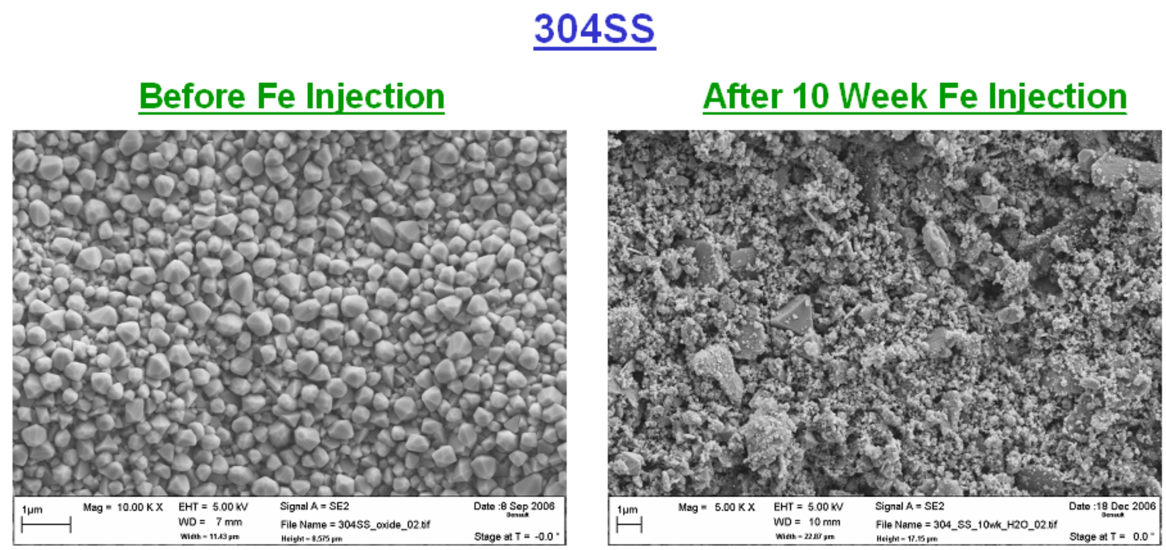

Figure 5. Oxide surface morphology of 304 SS before and after the Fe injection test in 288°C water under various water chemistry conditions.

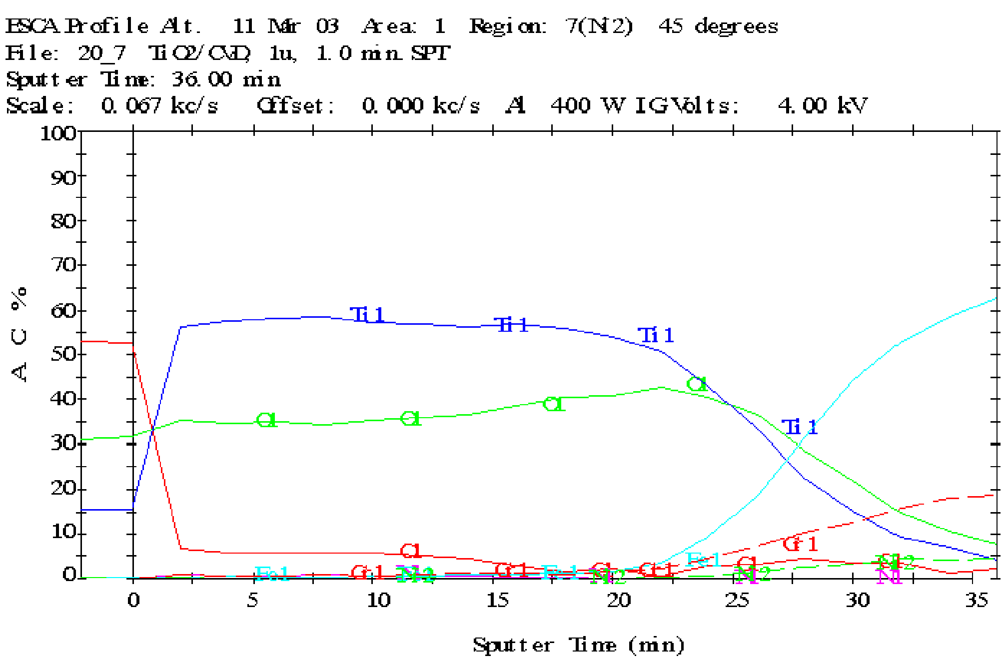

Figure 6: XPS confirmation of Stoichiometry of TiO_2 coating by CVD on 304 SS.

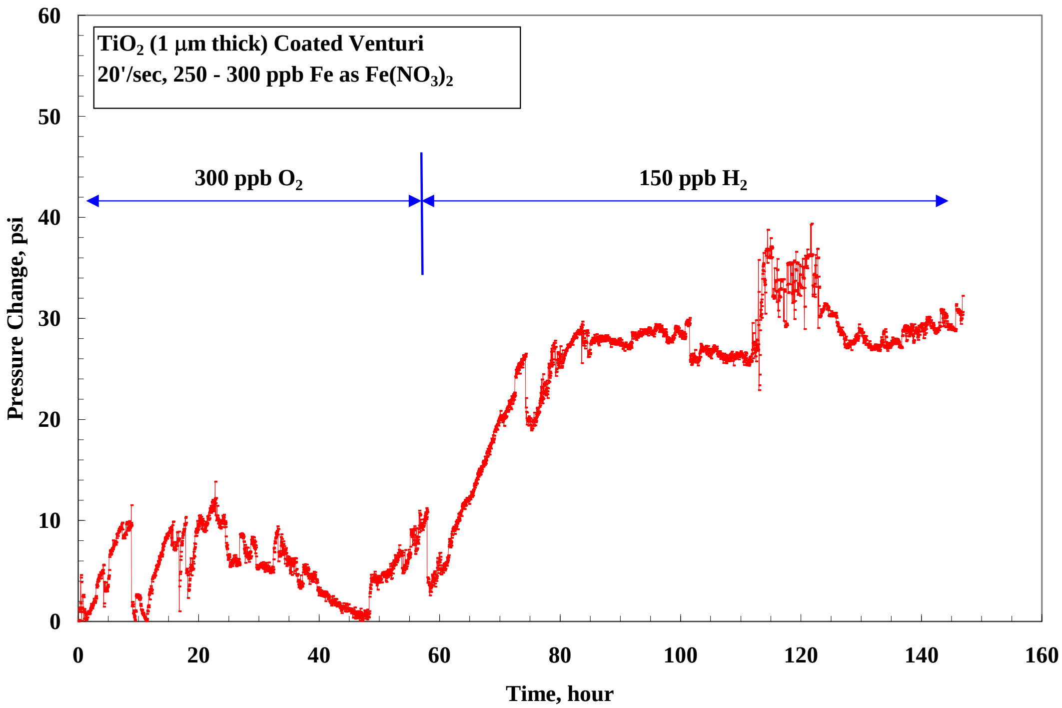

Figure 7. Effect of the water chemistry condition on the differential pressure change for the TiO_2 coated venturi surface as a function of immersion in 288^oC water with injection of 250-300 ppb Fe at 20 ft/sec (6 m/sec).

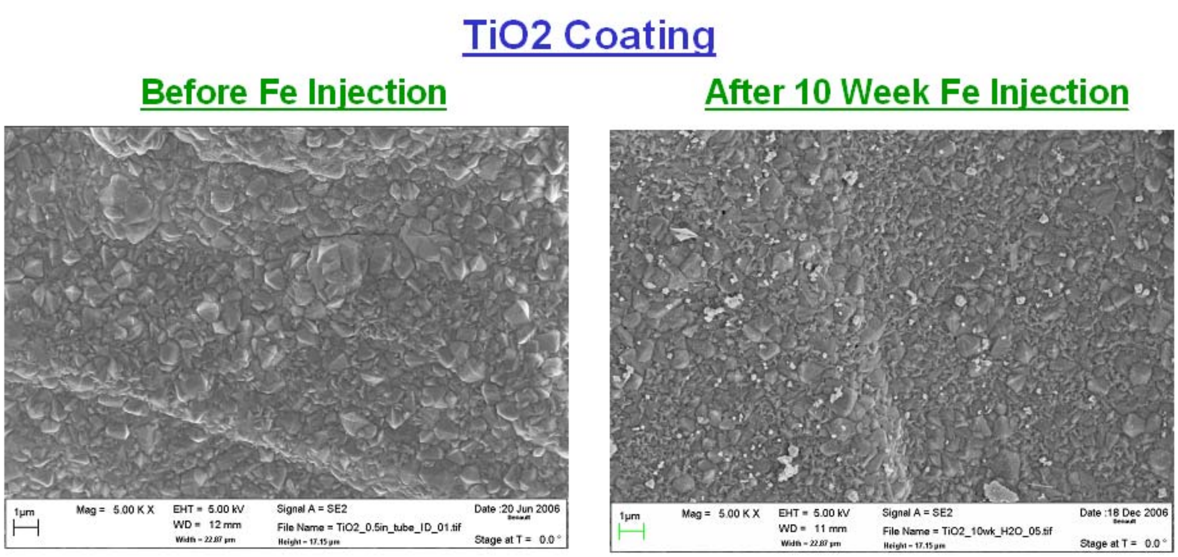

Figure 8. Surface morphology of TiO_2 coated 304 SS before and after Fe injection test in 288^oC water under various water chemistry conditions.

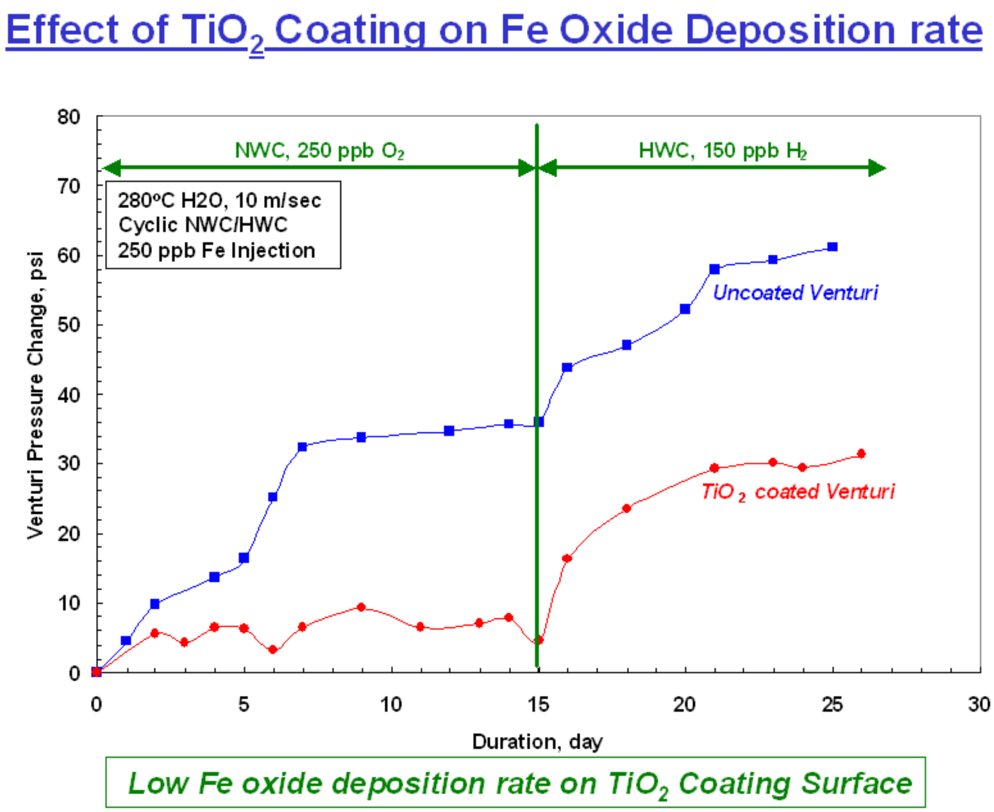

Figure 9. Comparison of 304SS and TiO2 coated 304SS venturi surfaces under the 250 ppb Fe injection test in 288^{o}C water under normal water chemistry and hydrogen water chemistry.

THE ROLE OF ACOUSTIC ANALYSIS IN UNDERSTANDING PUMP-INDUCED VIBRATIONS IN REACTOR RECIRCULATION PIPING

Brian J. Voll
Sargent & Lundy[LLC]

Abstract

Reactor Recirculation (RR) system components in Boiling Water Reactor (BWR) plants have experienced damage and failures attributable to flow-induced vibration. Susceptible components include small bore branch piping, valve operator components and other piping appurtenances, as well as valve internal components. In some instances, significant noise and vibration has been transmitted to locations outside of the primary containment structure where the RR piping is located. The most significant vibrations are typically associated with RR pump impeller vane passing frequencies (VPF). Furthermore, in plants with variable speed RR pumps, the most significant noise and vibration occur in specific RR pump speed ranges, indicating an interaction of the VPF with RR system acoustic modes.

The RR piping configuration is relatively complex from an acoustic standpoint, typically consisting of two RR loops connected by the reactor pressure vessel (RPV), with residual heat removal (RHR) and reactor water cleanup (RWCU) large bore branch piping. Therefore, RR system acoustic modes cannot be determined by simple calculations. Rather, acoustic modeling of the RR system is necessary to provide an understanding of the RR system acoustic characteristics. This paper describes how RR system acoustic analyses were used for a BWR plant to investigate system responses as a function of RR pump speed, including the prediction of responses at elevated pump speeds, as an aid for evaluating vibration severity, and as a tool for investigating potential system modifications.

1.0 INTRODUCTION

A typical reactor recirculation (RR) system in a boiling water reactor (BWR) plant, including residual heat removal (RHR) shutdown cooling and reactor water cleanup (RWCU) branch piping, is shown schematically in Figure 1. The system shown consists of two separate loops, each with its own recirculation pump. The flow in each loop is controlled by RR pump speed (alternatively, a flow control valve is used in some plants). During normal plant operation there is no flow in the RHR shutdown cooling lines. Therefore, the RHR piping between the RR loops and the RHR isolation valves are dead legs. The physical configuration of a typical RR loop, including the RHR shutdown cooling lines, is shown in Figure 2.

Industry experience has shown that increased containment noise and vibration of components associated with the RR system have occurred during certain operating conditions. These incidences of increased noise and vibration have been attributed to RR system operation in particular RR pump speed ranges. Examples of such incidents are provided in U.S. Nuclear Regulatory Commission (NRC) Information Notice 95-16 [1] and General Electric (GE) Services Information Letter (SIL) No. 600 [2]. Many of the incidents occurred at increased RR pump speeds, which may be pertinent for plants considering operating scenarios where increased core flows are required or where increased RR pump speeds are required to maintain flow due to increases in pressure drop across the core.

As a result of the increased vibrations, damage to and failure of piping in-line components and piping attachments have occurred. Much of the damage has occurred in the vicinity of the isolation valves on the RHR shutdown cooling lines attached to the RR loops. In one case, a through-wall crack developed in a cantilevered branch line near the RR pump suction inlet. The predominant vibration frequencies have been shown to correspond to the RR pump impeller vane passing frequency. Typically, the RR pumps have five vane impellers and the increased vibrations have corresponded to RR pump speeds anywhere between 1200 rpm and 1600 rpm (for variable speed pumps), depending on the plant. Therefore, typical vane passing frequencies (VPF) corresponding to the increased noise and vibration range from approximately 100 to 133 Hz.

Since the increased noise and vibrations occur in certain RR pump speed ranges, the evidence suggests that in those speed ranges the VPF coincides with RR piping system acoustic modes. Furthermore, components that have structural natural frequencies near RR pump VPFs that coincide with RR system acoustic natural frequencies are most susceptible to damage and failure. Systems with variable speed RR pumps, as opposed to systems with fixed speed RR pumps and flow control valves, are more susceptible since the excitation frequency varies and is more likely to coincide with system acoustic modes at one or more operating conditions.

A common type of acoustic resonance in piping systems corresponds to organ pipe, or quarter-wave, frequencies of dead-ended piping legs. In this case, pressure pulsations generated at the branch opening at a certain frequency are reflected from the closed end of the piping leg and are reinforced, resulting in a standing pressure wave with significant amplification. The quarter-wave frequency is dependent on the speed of sound in the fluid and the length of the dead-ended pipe leg. Harmonics of the quarter-wave frequency occur at odd multiples of the fundamental frequency. A typical source of quarter-wave acoustic resonance is vortex shedding at branch openings. For example, vortex shedding at safety relief valve branch openings can excite quarter-wave frequencies in the branch and cause valve chatter and vibrations that lead to wear and damage of the valve components.

Piping quarter-wave frequencies can be easily calculated if the pressure pulsation source is at the branch opening of a dead-ended leg with no branches, the piping has a uniform cross-section and the speed of sound in the fluid is constant. Examples of closed ends in the RR loops and RHR shutdown cooling lines are the capped ends on the RR pump discharge ring headers and the RHR shutdown cooling return and supply line isolation valves. However, the sources of the VPF pressure pulsations are the RR pumps, which are not at a branch opening of a dead leg. In addition, the RR piping configuration contains many branches and area changes. Finally, the RHR shutdown cooling lines are typically long enough for significant temperature decays to occur, which results in a varying speed of sound along the length of the RHR shutdown cooling lines.

RR system acoustic natural frequencies are thus dependent on the overall system configuration and thermodynamic properties, and cannot be calculated using simple means. Rather, an acoustic analysis computer program is required to determine the RR system acoustic natural frequencies. This paper describes how acoustic analyses were used to investigate RR system responses as a function of RR pump speed, including the prediction of responses at elevated pump speeds, as an aid for evaluating vibration severity, and as a tool for investigating potential system modifications.

2.0 ANALYSIS OF SIMPLE ACOUSTIC MODELS

Acoustic analysis results for two simple models are presented to demonstrate the limitations of simplified methods for determining acoustic frequencies and the need for acoustic analysis programs to calculate acoustic responses for all but the simplest configurations.

The first model, shown in Figure 3, is a simple open-closed cylinder with a constant area cross-section, similar to a safety relief valve branch. The acoustic frequencies for this model, determined using an acoustic analysis program, are shown in Figure 4. The fundamental frequency is 10 Hz, with harmonics at 30 Hz, 50 Hz, 70 Hz, etc. For this model, the acoustic frequencies can also be calculated using the following formula [3]:

$$f_i = \frac{ic}{4L}$$

where,

f_i = ith mode acoustic frequency
i = acoustic mode (1, 3, 5, etc.)
c = acoustic velocity
L = cylinder length

For the model shown in Figure 3, c = 1200 ft/sec and L = 30 ft. Therefore, the calculated fundamental acoustic frequency is 10 Hz. The second mode is 30 Hz, the third mode is 50 Hz, the fourth mode is 70 Hz, etc. Thus, the results from the formula and the acoustic analysis program are in agreement.

The second model, shown in Figure 5, is the same as the first model except that the cylinder area is reduced by a factor of four for the second half of the cylinder length. The acoustic frequencies for this model, determined using the acoustic analysis program, are shown in Figure 6. The fundamental frequency has shifted from 10 to 14 Hz, while the second mode has shifted from 30 to 26 Hz. The higher modes have also shifted. Using the formula for an open-closed cylinder, the frequency shift is not predicted since c and L did not change. Even substituting L/2 for L, where the area change occurs, does not produce the correct results. For even more complicated configurations, such as the RR piping configuration, it is then understandable that the acoustic frequencies can not be predicted and that the acoustic characteristics can only be determined using an acoustic analysis program.

3.0 RR SYSTEM ACOUSTIC MODEL EXAMPLE

The following example illustrates how acoustic analysis results can be used to help understand RR system acoustic characteristics, including how acoustic responses change as a function of RR pump speed, and how the acoustic model can be used as a predictive tool and as an aid for investigating potential system modifications.

An acoustic model of an RR system was created which included both RR loops, the RHR shutdown cooling lines up to the closed isolation valves and the RWCU branch piping. In addition, a representation of the RPV was included in the model to allow for cross-talk between the loops. One-dimensional modeling was used for the RR system, meaning that acoustic responses could vary along the length of the piping, but were constant throughout each piping cross-section. This type of modeling was sufficient for the range of VPFs under consideration and greatly simplified the acoustic model, resulting in faster analyses and more readily providing forcing functions that could be applied to piping structural models.

Even with one-dimensional modeling, the acoustic model contained several hundred analytical node points for which acoustic responses were calculated. The challenge with a model of this complexity is determining how best to represent the results so that they provide meaningful and useful information. This is similar to the challenge in understanding structural response characteristics of a complex piping system. Several methods, as discussed below, were employed to help understand the RR system acoustic characteristics.

One method used was to apply a white noise function at the RR pump locations to excite the system acoustic modes in the frequency range of interest. Then, responses were plotted at various locations to identify frequencies at which the largest acoustic responses would be expected. The acoustic frequency response at one of the RHR shutdown cooling isolation valves is shown in Figure 7. The RR pump speeds corresponding to the maximum response frequencies are also shown.

Another analysis method used to help determine the RR system acoustic responses involved the application of harmonic functions at the RR pump locations that covered the range of VPFs corresponding to the RR pump speeds of interest. Acoustic responses were then plotted as a function of RR pump speed. The acoustic responses at the RHR shutdown cooling isolation valve locations versus RR pump speed are shown in Figure 8. This type of representation, showing acoustic response peaks and valleys, provides an indication of which RR pump speed ranges may result in the most significant acoustically-induced noise and vibration.

Another method used for visualizing the harmonic analysis results was to create pressure distribution animations, or movies, for each RR loop. These movies showed the pressure distribution throughout each loop as a function of RR pump speed. With this type of visualization, the locations of the pressure nodes (minima) and antinodes (maxima) for each pump speed and the change in pressure distribution and amplitudes as a function of pump speed could be quickly identified.

A snapshot of the pressure distribution in one of the RR loops at a pump speed where the pressure amplitudes were relatively low is shown in Figure 9. The darker colors in the plot represent pressure nodes and the lighter colors represent antinodes. A snapshot of the pressure distribution in the same RR loop at a pump speed where the pressure amplitudes were near their maximum values is shown in Figure 10. The locations of the pressure maxima, represented by the brighter colors, are consistent with areas where component damage due to excessive vibration was found.

In order to validate the RR acoustic model and enable it to be used as a predictive tool, benchmarking of the model against vibration and pressure measurements was performed. Since the output of the acoustic analyses is dynamic pressure, measurements of dynamic pressure at key locations in the system are preferred for benchmarking. The installation of dynamic pressure transducers in the piping was not feasible for this particular project; therefore, strain gages were installed to obtain measurements proportional to dynamic pressure. The strain gage measurements were compared with the acoustic analysis results, and adjustments to the model were made to improve correlation between the analysis and measured values. An example of a comparison between the acoustic model dynamic pressure output and the strain gage measurements near one of the RHR shutdown cooling isolation valves is shown in Figure 11.

Once the acoustic model was benchmarked, it was used to predict system acoustic responses at RR pump speeds that were higher than those at which the RR pumps had previously been operated. These analyses indicated that maximum acoustic responses already observed would not be exceeded at higher pump speeds. The acoustic analysis results were also used as input for structural analyses used to evaluate predicted vibration severity.

The benchmarked acoustic model was also used to investigate the effects of potential modifications intended to reduce system acoustic responses. The largest analytical and measured responses, and the most significant component damage, occurred in the vicinity of the RHR shutdown cooling return isolation valves. Therefore, the investigative acoustic analyses focused on modifications in those areas.

Results for one example, where the effects of changing the location of an RHR isolation valve were investigated, are shown in Figure 12. These analysis results indicate that the RR pump speeds where the acoustic peaks and valleys occur did not change significantly as a result of moving the valve. Therefore, the overall system characteristics did not change significantly. Locally to the valve, however, the predicted dynamic pressure amplitudes did change, although they tended to decrease at some pump speeds and increase at other pump speeds.

Making major modifications, such as changing the location of an isolation valve in a congested drywell, was determined in the end not to be feasible. However, since the acoustic model, supported by test data, identified the RR pump speeds at which the highest acoustic responses would occur, it was determined that qualification of susceptible components for the maximum system responses, and modification of susceptible components to avoid structural resonance where necessary, would be sufficient. Therefore, the acoustic model did prove to be a useful tool for better understanding the system acoustic characteristics and determining the most effective modifications.

4.0 CONCLUSIONS

Complex piping systems have acoustic characteristics that cannot be determined using simple methods. In most cases, this is not a concern because the excitation mechanisms, such as pressure pulsations at pump impeller VPFs, occur at fixed frequencies. If unacceptable system responses were to occur, they would be identified and corrected during initial system operation. In some cases, however, the excitation frequency can vary. For example, in RR systems employing variable speed pumps, the VPFs can vary widely. This almost ensures that system acoustic modes will be excited under certain operating conditions. Unless the amplitude of the excitation is very low, the potential for unacceptable system responses is real, especially when component structural resonant frequencies coincide with system acoustic resonant frequencies.

In cases where unacceptable system responses are occurring due to acoustic excitation, and where the excitation frequencies are variable, understanding the system acoustic characteristics is critical for determining effective corrective actions. Even when making modifications in systems where unacceptable acoustic responses have not been occurring, understanding the effects of the modifications on system acoustic characteristics is important in systems where variable frequency acoustic excitation mechanisms exist. This is also true when considering changes in system operation that affect excitation frequencies, for example, increasing RR pump speeds to increase core flow or maintain flow due to increases in pressure drop across the core.

System acoustic characteristics in complex systems can be better understood through the use of acoustic analyses. This has been demonstrated for RR systems with variable speed pumps. Various methods can be used to better visualize and interpret the results of acoustic analyses. These methods include generation of plots showing peak acoustic response frequencies, plots of acoustic response amplitude trends as a function of excitation frequency and animations showing system-wide acoustic responses as a function of excitation frequency. Benchmarking of acoustic analysis results with dynamic pressure measurements is important if acoustic analyses are to be used as predictive tools, including investigation of the effects of potential modifications intended to reduce acoustic responses.

5.0 REFERENCES

1. NRC Information Notice 95-16, Vibration Caused by Increased Recirculation Flow in a Boiling Water Reactor, March 9, 1995.

2. GE Nuclear Energy, Increased containment noise and vibration at increased recirculation pump speed, Services Information Letter (SIL) No. 600, May 15, 1996.

3. Blevins, Robert D., Formulas for Natural Frequency and Mode Shape, Krieger Publishing Company, 1995.

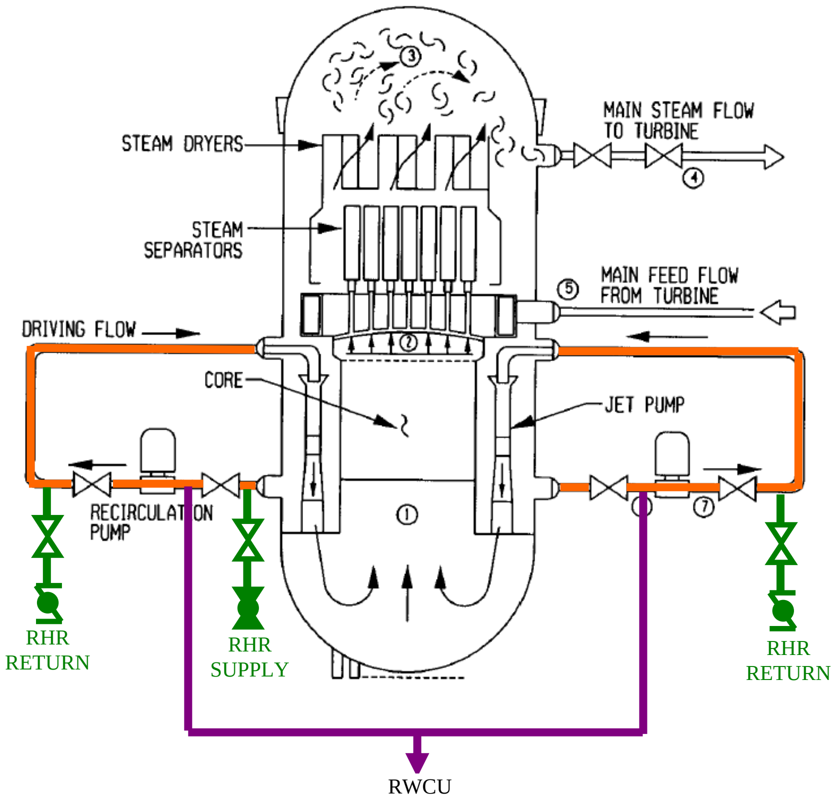

Figure 1. Schematic of a BWR RR System

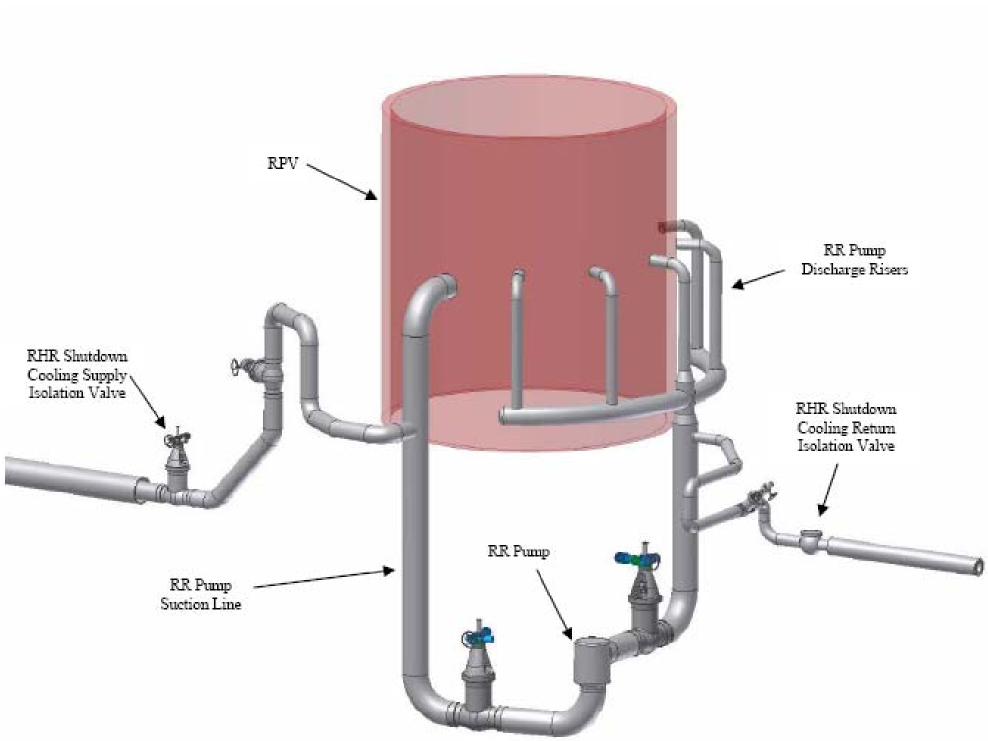

Figure 2. RR Loop Physical Configuration

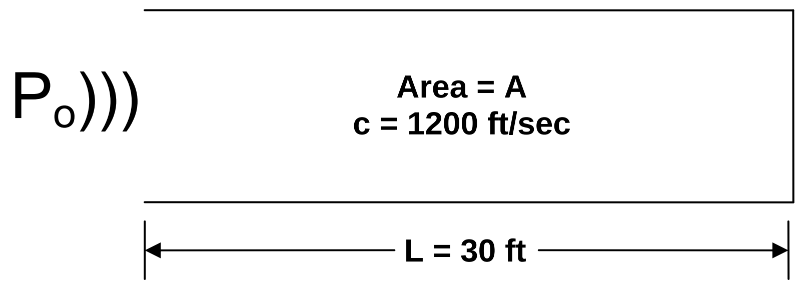

Figure 3. Open-Closed Cylinder of Constant Area

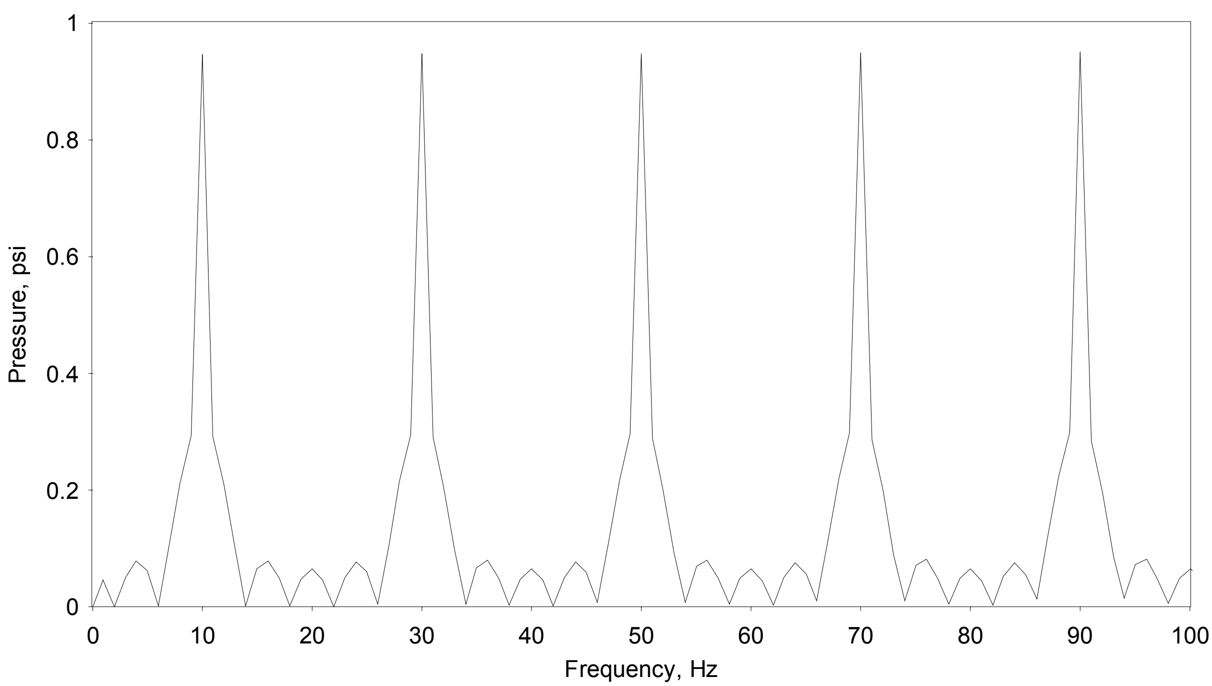

Figure 4. Acoustic Frequencies for Constant Area Open-Closed Cylinder

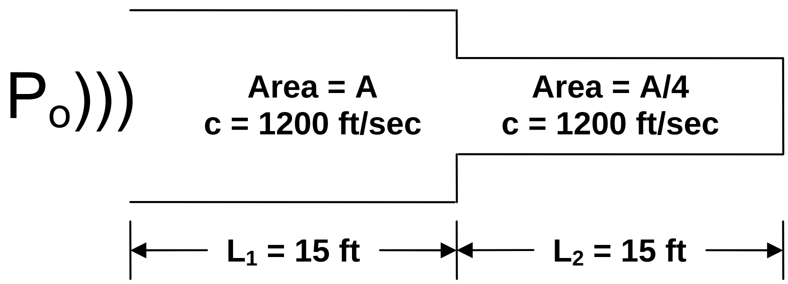

Figure 5. Open-Closed Cylinder with Area Change

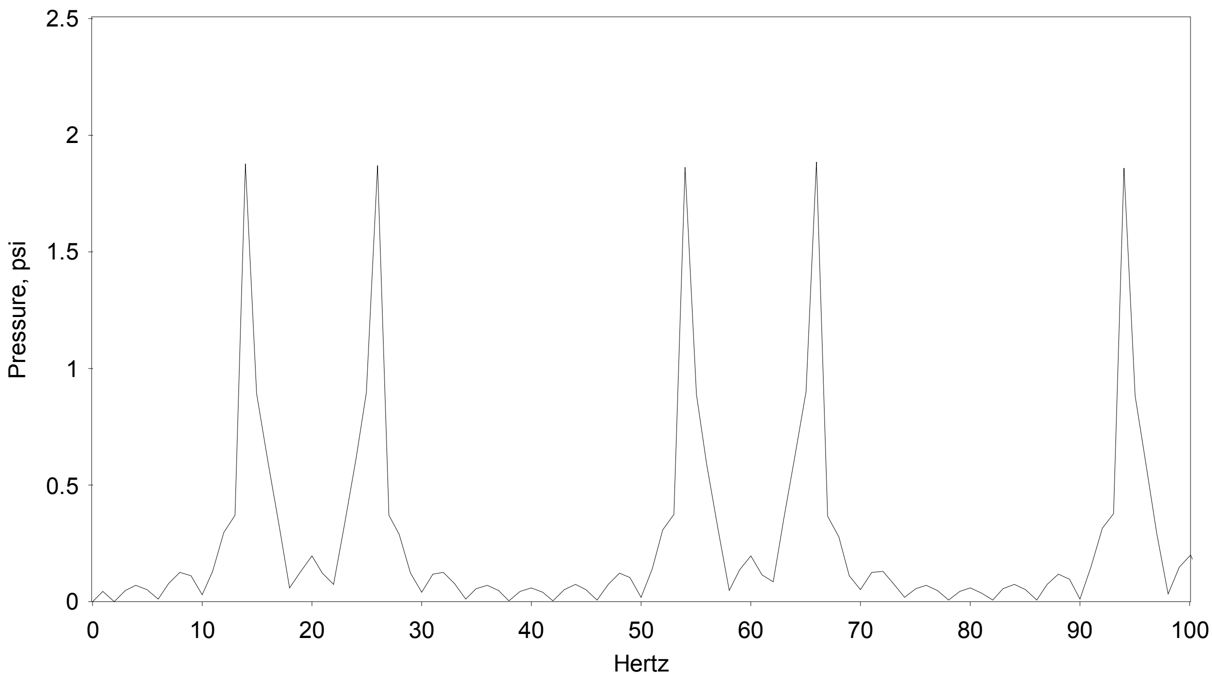

Figure 6. Acoustic Frequencies for Open-Closed Cylinder with Area Change

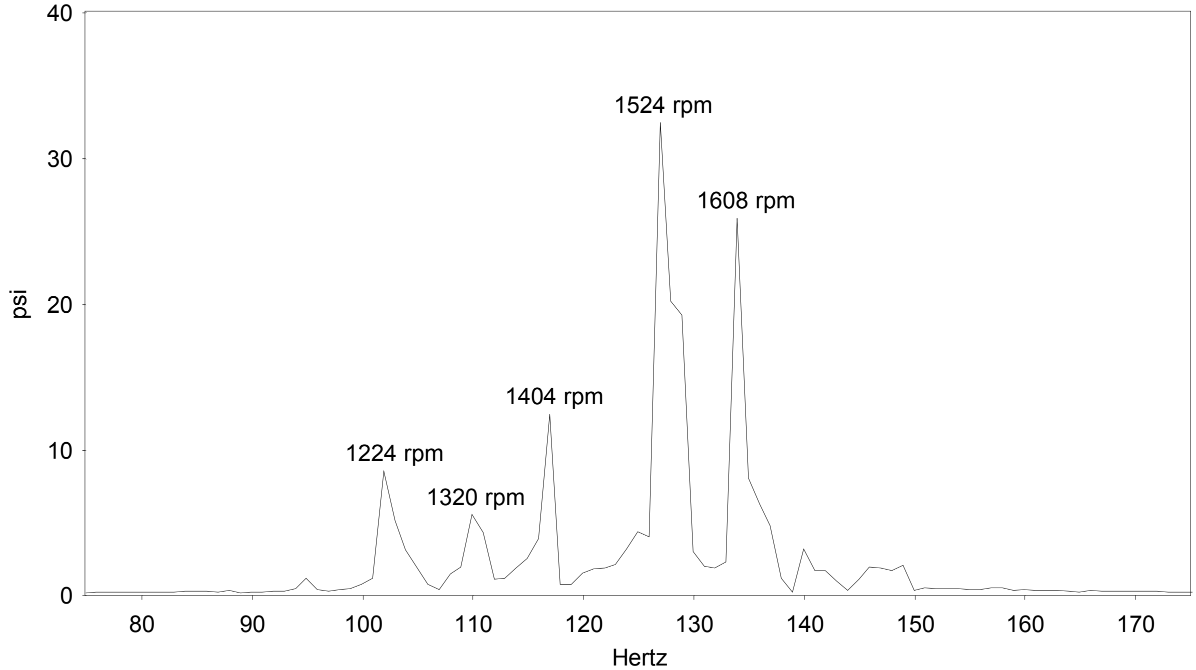

Figure 7. Acoustic Frequencies at RHR Isolation Valve Location

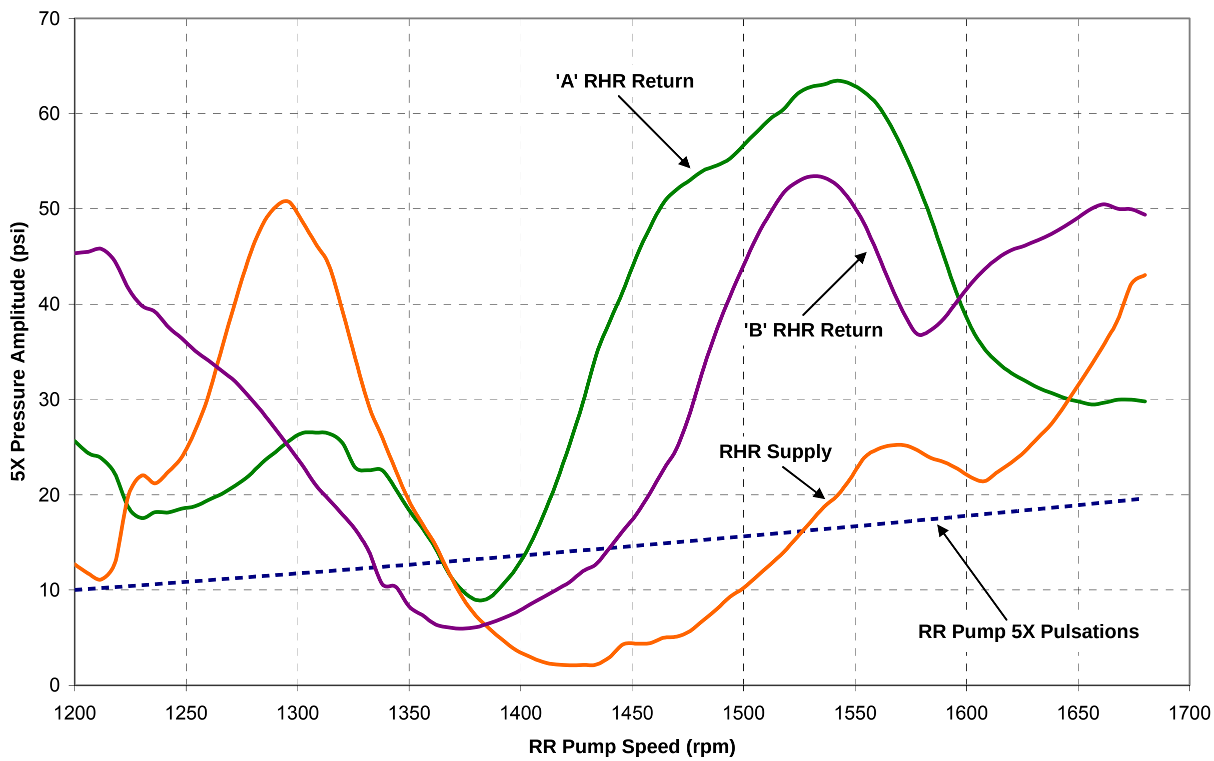

Figure 8. Comparison of Acoustic Responses at RHR Isolation Valve Locations

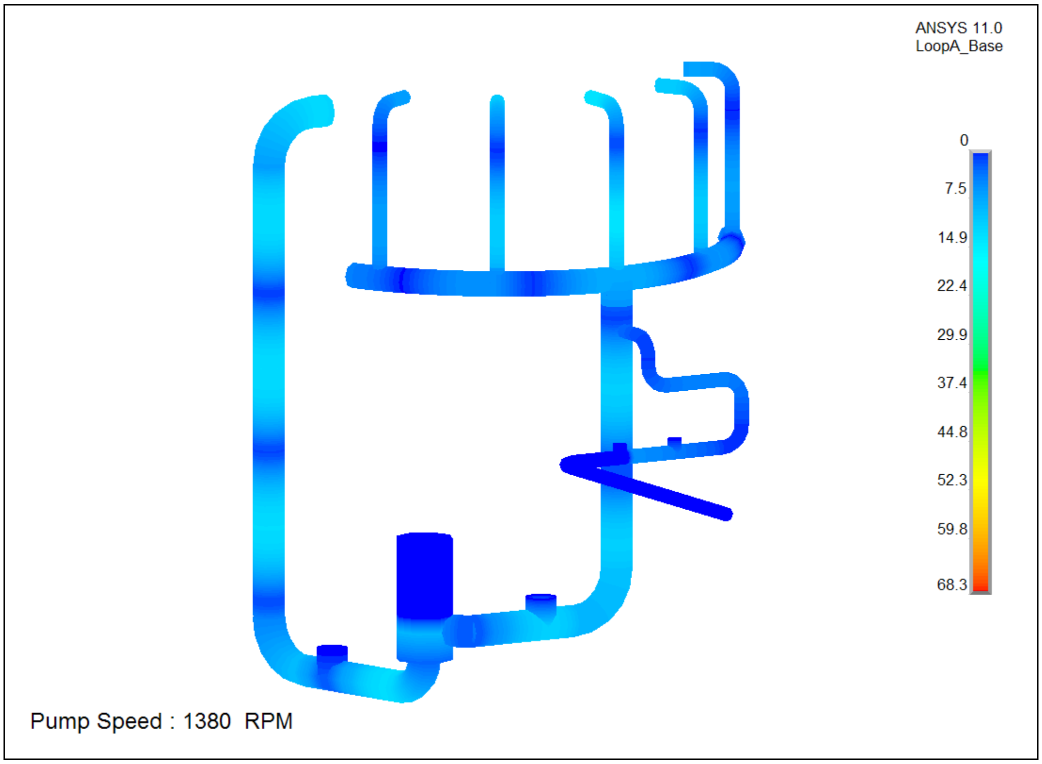

Figure 9. RR Loop Acoustic Responses at “Low Response” Pump Speed

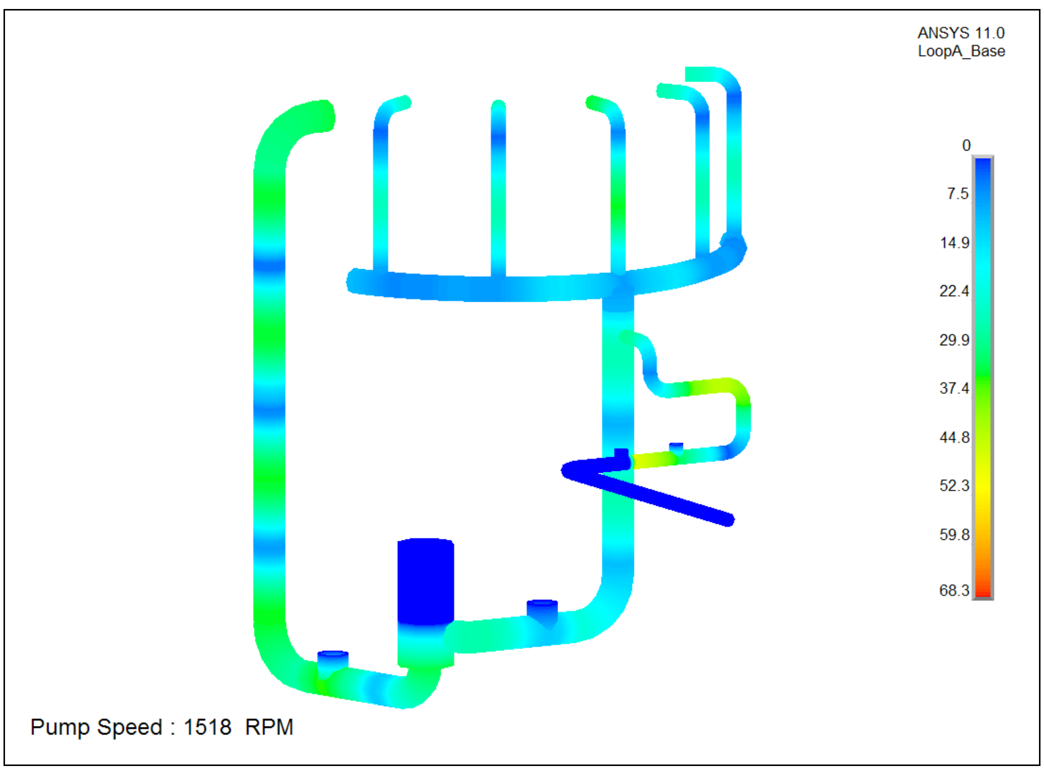

Figure 10. RR Loop Acoustic Responses at “High Response” Pump Speed

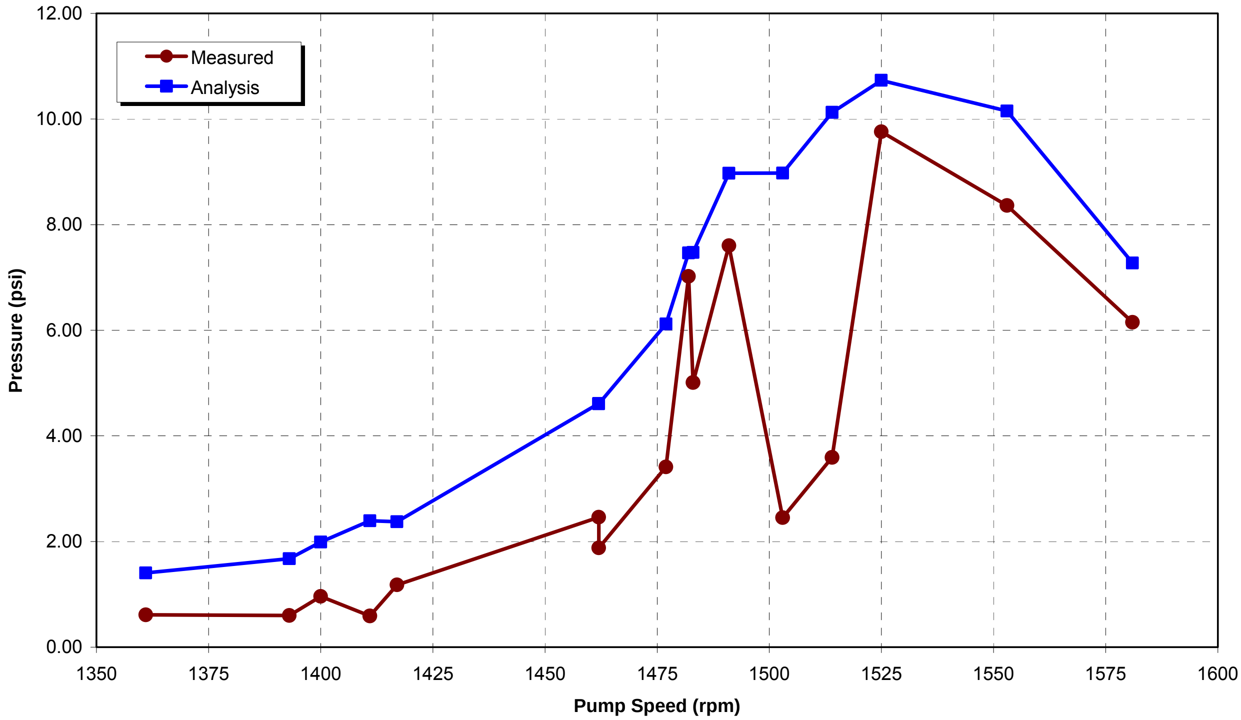

Figure 11. RR Acoustic Model Benchmark Results

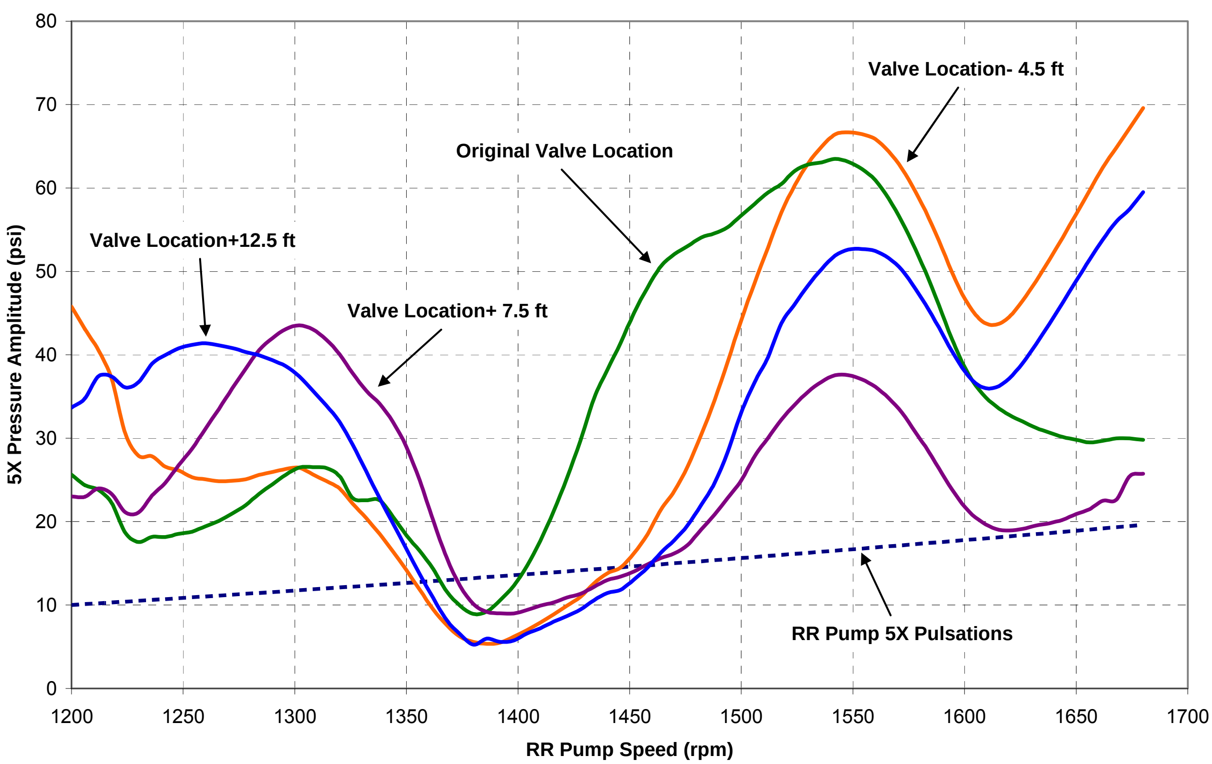

Figure 12. RR Acoustic Model Sensitivity Analysis Results

Pump Inservice Testing Limitations

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Abstract

This paper discusses the limitations of pump inservice testing (IST) that have been identified at U.S. nuclear power plants. These limitations show that relying only on the pump IST program could result in pump damage and/or plant shutdown. This discussion includes information that could have generic applicability for proper pump performance at U.S. nuclear power plants.

Pump Inservice Testing Limitations

INTRODUCTION

The NRC staff has encountered a number of pump inservice testing (IST) limitation issues. This paper discusses events that have occurred at several U.S. nuclear power plants that demonstrate that IST alone may not guarantee the operational readiness of pumps. This discussion includes information that could have generic applicability for proper pump performance at U.S. nuclear power plants.

POINT BEACH UNIT 1 TURBINE DRIVEN AUXILIARY FEEDWATER (TDAFW) PUMP

On May 1, 2007, during post-maintenance testing (PMT) of the 1P-29 TDAFW pump following a 10-year overhaul of the turbine, oil was observed to be leaking from the outboard end of the turbine at the turbine trip disc housing. The test was aborted when the reason for the oil leakage could not be readily determined. Troubleshooting was initiated and it was determined that the leakage was from inadequate journal bearing to bearing housing clearance, caused by an improperly manufactured bearing. The journal bearing had been replaced as part of the overhaul. The old bearing was reinstalled and the oil leakage ceased.

On June 9, 2007, the TDAFW pump was run in accordance with the quarterly IST procedure. During this test, the TDAFW pump turbine outboard bearing reached a temperature of 233 degrees Fahrenheit (°F), which was 8°F over the bearing high temperature alarm setpoint of 225°F (temperatures over 250°F required the pump to be shut down). A condition report was written indicating the temperature was still increasing when the pump was secured from the IST; however, no immediate actions were taken to address the high bearing temperature, which had not stabilized at the time the pump was secured. On June 11, 2007, licensee personnel reviewed the data from the June 9, 2007 IST and raised the concern that the bearing did not appear to have reached an equilibrium temperature, as evidenced by the continuing rate of increase. Based on these issues, the licensee re-performed the IST on June 12, 2007. During the test, the turbine outboard bearing temperature reached 249.5°F and the operators aborted the test, shut down the TDAFW pump, and declared the pump inoperable. As a result of an initial investigation, the licensee determined that the turbine outboard bearing was contacting the turbine shaft and oil analyses indicated the oil was subjected to thermal stress and contained moderate wear debris.

Over the next 72 hours, the licensee attempted to identify the cause and repair the TDAFW pump. The licensee identified turbine bearing upper housing alignment issues, a potential radiative heat issue from the turbine casing insulation towards the outboard bearing, and potential turbine bearing clearance issues. However, the licensee was not able to identify the cause and repair the TDAFW pump within the allowed Technical Specification outage time of 72 hours and began a Unit 1 shutdown June 14, 2007 due to the inoperable TDAFW pump.

Over the next 48 hours, the licensee again attempted to troubleshoot and repair the cause of the high outboard bearing temperature. The licensee identified an improper thrust bearing axial clearance, the coupling shim pack stretch was identified as incorrect, and the licensee's procedure for overhaul had an incorrect setting for the turbine wheel lap. The licensee also replaced the outboard bearing cooler. On June 17, 2007, the licensee commenced what was essentially a turbine overhaul, which was completed on June 19, 2007.

On June 20, 2007, two runs were performed with the TDAFW pump. On the first run, the maximum stabilized outboard bearing temperature while on recirculation was 230.8°F. The turbine insulation was adjusted, and a second recirculation run produced a maximum stabilized outboard bearing temperature on recirculation of 228°F.

On June 21, 2007, additional insulation configuration changes were made, and recirculation runs were performed on June 21 and June 22, 2007. Both runs produced a maximum stabilized outboard bearing temperature of 223°F.

On June 23, 2007, bearing stabilization at full flow (400 gpm) yielded a maximum outboard bearing temperature of 226°F. Based on the NRC inspectors' questions concerning the effects of increased service water temperatures, the licensee performed an operability determination which concluded the maximum service water temperature would result in a 12.5°F rise, for a temperature of 238.5°F. The TDAFW pump was returned to an operable status.

The licensee performed a root cause evaluation for this event. The licensee stated that the root cause for the oil leaking out of the TDAFW pump outboard bearing was that the licensee received a bearing from the vendor with an incorrect chamfer. The material receipt process did not verify critical bearing dimensions because there were no drawings or dimensions available.

The licensee stated that the root cause for having the elevated operating temperatures with the TDAFW pump outboard bearing was a failure to correctly revise the turbine overhaul procedure. In addition, the licensee determined that the root cause for not identifying the condition immediately following the maintenance was the failure to perform an adequate post-maintenance test. Specifically, the post-maintenance test should have included running the TDAFW pump until bearing temperature stabilization was achieved for both the pump and turbine, following the maintenance overhaul.

The licensee stated that other issues identified during the investigations were:

Tolerance of recurring equipment problems, which desensitized the organization to the condition

Inadequate equipment performance monitoring and trending

Numerous organizational failures to consider risk significance

Poor project planning and execution

Limited Terry Turbine expertise

Shortcomings in the skill of the craft

Procedure inaccuracies

Poor procedure use and adherence practices throughout the organization

The American Society of Mechanical Engineers (ASME) Code for Operation and Maintenance of Nuclear Power Plants (OM Code) requires that these pumps have a quarterly IST consisting of measuring flow rate and/or differential pressure, and vibration. Bearing temperature measurements are not required for IST. Measuring bearing temperature until temperature stabilization would have identified this bearing problem earlier.

COLUMBIA GENERATING STATION SERVICE WATER (SW) PUMP

On June 14, 2005, following a start of the SW pump SW-P-1A, control room operators noted that SW flow to the Division 1 residual heat removal heat exchanger was out of specification. Operators also noted the pump discharge head and pump motor current were lower than normal and declared the pump inoperable. A subsequent surveillance test on the pump determined that its performance had degraded and was operating at the intersection of the alert and action ranges of its performance curve. The licensee proceeded to replace SW-P-1A with an available spare pump.

During disassembly of the pump, the licensee determined that intergranular stress corrosion cracking (IGSCC) failed the pump shaft end flanges on two of the shaft sections allowing the shaft sections to drop. This condition caused the pump impeller to contact the pump suction casing, which resulted in substantial wear of the pump impellers and degraded pump performance. The shaft drive keys remained captured between the shaft keyways and coupling sleeves such that the shaft segments and impeller continued to rotate. The licensee conducted a metallurgical examination of the damaged pump shaft and also identified axial cracking on the impeller pump shaft segment and two diagonal cracks on the top column shaft.

The metallurgical examination determined that the shaft material, TP410 martensitic stainless steel, was susceptible to tempering embrittlement (shaft material was tempered at 970°F, which was conducive to tempering embrittlement). Tempering embrittlement reduced the corrosion resistance of the shaft material, thereby, increasing the material's susceptibility to IGSCC. The licensee determined that SW pump SW-P-1B was also susceptible to the same failure mechanism as identified in SW pump SW-P-1A. However, it had not exhibited performance degradation based on past surveillance test results.

Interim corrective actions included additional monitoring of SW pump SW-P-1B to verify pump performance until a replacement pump could be procured and installed. A subsequent inspection of the as-found condition of SW pump SW-P-1B determined that the pump shaft had degraded in a manner similar to SW pump SW-P-1A, due to IGSCC, and that the pump impeller had degraded due to wearing on the suction casing.

A detailed evaluation of SW pump SW-P-1A historical computer data determined that although pump performance had met IST acceptance criteria, pump performance had slowly degraded from as early as August 2000 and as late as December 2001. Similarly, a detailed evaluation of

SW pump SW-P-1B historical computer data revealed that the pump had slowly degraded since August 2003.

The ASME OM Code requires that these pumps have a quarterly IST consisting of measuring flow rate and/or differential pressure, and vibration. Vibration measurements are required to be taken on the upper motor-bearing housing. Vibration analysis using more sophisticated tools (i.e., transducer on pump bowl, phase angle analysis, obtaining vibration data directly from the pump shaft) might have been capable of identifying this degraded condition prior to the marked decline in pump hydraulic performance.

POINT BEACH SW PUMP

On February 2, 2008, an auxiliary operator on rounds noticed, by visual and audio perception, that the vibration of SW pump P-032E did not seem normal. The operator contacted engineering, and vibration measurements were taken on the pump. When the vibration levels were discovered to be in the required action range, the pump was immediately secured and declared inoperable.

The pump shaft was removed and inspected. Significant wear was discovered on the pump shaft (40 foot length) at the Cutless bearing locations. The wear was only on one side of the shaft. Approximately ¼-inch of metal was worn away at the highest wear area in the lower end of the third intermediate shaft.

SW pump P-032E was last rebuilt on June 7, 2006. Motor frame vibration data obtained on December 5, 2007 suggested that some very minor changes in stiffness were starting to occur.

The apparent cause was determined to be most likely the licensee’s installation and rigging practices. Since there is no opening in the roof of the screen house, the pump has to be taken apart and put back together in the small SW pump room. The manner in which the maintenance personnel were rigging the 40-foot shaft to install it in the pump was bending the bottom of the shaft, causing the third intermediate shaft to have a run out of approximately 19 thousandths of an inch. Rigging practices were changed so that the shaft is no longer bent in order to install or remove it from the pump.

The ASME OM Code requires that these pumps have a quarterly IST consisting of measuring flow rate and/or differential pressure, and vibration. Vibration measurements are required to be taken on the upper motor-bearing housing. Vibration analysis using more sophisticated tools (i.e., transducer on pump bowl, phase angle analysis, obtaining vibration data directly from the pump shaft) might have been capable of identifying this degraded condition earlier.

CONCLUSION

The purpose of this paper is to make licensees aware that IST alone is not a guarantee of the operational readiness of pumps. Proper maintenance activities, inspections and possibly additional testing should be performed in addition to IST. Licensees may want to review their maintenance and inspection programs for the pumps in their IST program to insure that they are adequate in order to alleviate problems such as those described in this paper.

REFERENCES:

ASME/ANSI, Code for Operation and Maintenance of Nuclear Power Plants, 2001 Edition and 2003 Addenda:

Subsection ISTB, “Inservice Testing of Pumps in Light-Water Reactor Power Plants.”

NUREG-1482, Revision 1 “Guidelines for Inservice Testing at Nuclear Power Plants,”June, 2004

Section 55a of Part 50 to Title 10 of the Code of Federal Regulations, (10 CFR 50.55a) “Codes and standards,” January, 2007

Development of a Generic Systems Testing Standard

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Luminant Power

Abstract

ASME (American Society of Mechanical Engineers) OM (Operations and Maintenance) Part 28 was developed as a generic standard for performance testing of systems in Light Water Reactor power plants. It represents the culmination of a long-term effort by the ASME OM Subgroup on Functional Systems to produce a series of system testing standards, including OM Parts 15, 20, 25, and a major revision to OM Part 2. This paper presents the background leading to the development of this document, and the governing philosophy that provided the basis for the overall approach. The structure of the document is discussed, and the Subgroup's vision for implementation is summarized. Future standards actions following publication of Part 28 are also presented. Finally, potential enhancements to Part 28 to address current trends in the nuclear industry are proposed.

Background

The genesis of ASME OM Part 28 (Performance Testing of Systems in Light Water Reactor Power Plants) began in 1991, when a working group was reconstituted to complete OM Part 15 (Performance Testing of Emergency Core Cooling Systems in Pressurized Water Reactors). As we progressed through development of a philosophical basis for the standard, completion of drafting, a very challenging balloting process, and resulting enhancements to the standard, it became clear that the fundamental approach contained in Part 15 could be applied to other process systems. As a result, when the Part 20 (Performance Testing of Emergency Core Cooling Systems in Boiling Water Reactors) working group lost membership and momentum, the Part 15 working group proposed to the OM Main Committee that we add Boiling Water Reactor expertise and assume responsibility for completing Part 20. This was an offer the Main Committee could not refuse.

Following publication of Part 15, as Part 20 was nearing completion, the working group evaluated strategic goals. Because the documents were philosophically and structurally consistent, it had become clear that Parts 15 and 20 could be combined into a single Emergency Core Cooling System (ECCS) testing standard. Similarly, we believed that a generic system testing standard could be developed since the basic methodology contained in Parts 15 and 20 could be deployed for any process system. Both concepts were discussed with the OM Special Committee on Standards Planning in March, 1998. The result was deferral of a generic standard but endorsement of the concept of a combined Pressurized Water Reactor/Boiling Water Reactor (PWR/BWR) ECCS testing standard. Subsequently, a scope statement for the combined ECCS testing standard was successfully balloted, authorizing development of OM Part 25 (Performance Testing of Emergency Core Cooling Systems in Light Water Reactors). The resulting document was produced, balloted, and published after completion of the Part 20 project. During work on Part 25, the working group was also requested to complete an in-process revision to OM Part 2 (Performance Testing of Closed Cooling Water Systems), which had fallen victim to the rigors of the development and balloting process.

Successful progress on Part 25 convinced the renamed Subgroup on Functional Systems that a single, generic systems testing standard was not only feasible, but desirable. Such a document would have broad applicability across a wide spectrum of plant systems, and would allow existing OM system testing standards and guides (Parts 2, 17, and 25) to be subsequently deleted, thus consolidating published system testing guidance. It would incorporate direction on service water and auxiliary feedwater system testing, avoiding development of two proposed additional OM standards. In addition, any future identified need for additional systems testing guidance could be addressed through a focused revision to a single document rather than development of a new, somewhat redundant product. This would all serve to increase the efficiency of the OM system standards and guides development, approval, implementation, and maintenance process. Following endorsement by the Special Committee on Standards Planning in March, 1999, scope, need, and value impact statements were prepared. After a typically arduous, but ultimately successful, balloting process, production of Part 28 was approved in March, 2000 and commenced in 2003 following completion of other open Subgroup work. The balloting cycle was initiated with Review and Comment transmittal in May, 2007, and completed with Board on Nuclear Codes and Standards approval in March, 2008. The new standard will be published in 2008.

Part 28 Philosophy

The scope statement for Part 28 is applicable to safety-related systems and systems important to safety. However, use of Part 28 for a given system or systems within the scope does not mandate application to all systems within the scope. In addition, this standard could be deployed for any other system requiring a performance testing program based on Owner evaluation of regulatory issues, Maintenance Rule insights, Probabilistic Risk Analysis insights, or other considerations. Part 28 does not mandate specific tests for all plants. While the process to be used to develop a testing program is prescriptive, the specifics are left to the owner to define, justify, and document. Additional flexibility in implementing the requirements is provided by allowing engineering evaluations, taking credit for applicable existing testing programs, or modeling when testing is not practical.

The fundamental elements described in Part 28 for developing and implementing a testing program for a selected process system are as follows:

- Establish the boundaries for the selected system or subsystem
- Identify performance requirements from licensing and design basis documentation
- Identify testable characteristics that represent performance requirements
- Establish test acceptance criteria for each characteristic
- Develop test procedures that include test acceptance criteria and test frequencies, and perform required testing, inspections and engineering analysis
- Evaluate test data and implement corrective action as appropriate
- Document and retain a Test Plan and test results

Testing of support systems, such as those providing power, heat rejection, or chemical addition, is not within the scope of Part 28. However, any potential interaction that may affect operation of the system being tested must be addressed by the test program.

A key element is the definition of system performance requirements consistent with the plant licensing and design bases. The purpose of a Part 28 testing program is to verify that the installed system is capable of meeting these requirements. Therefore, a thorough search of applicable design and licensing basis source information is essential.

Since it is often impractical to directly test design basis capability, it becomes necessary to identify testable characteristics that can be used to confirm that system performance requirements are met. An example is verification of the ECCS performance requirement to deliver a minimum safeguards flow to the reactor core versus reactor coolant system backpressure. It is neither practical nor desirable to directly confirm this capability. However, related testable parameters can be extracted from the supporting analysis for pump total dynamic head versus flow, system discharge resistance, branch line discharge resistance imbalance, and parallel flow path discharge resistance.

Part 28 is focused on verification of system level rather than component level performance requirements. It is assumed that the latter are addressed by other codes, standards, or owner programs. The standard also documents that implementing a Part 28 test does not relieve the owner from meeting more limiting criteria associated with component testing programs.

Maintaining nuclear safety is the paramount operating consideration at a nuclear generating plant. Verification of the capability to meet design basis performance requirements for a safeguards system, via deployment of a Part 28 testing program to assess integrated inservice system performance, can clearly enhance nuclear safety. However, creating an integrated test program that does not place the plant at risk is particularly challenging. Accordingly, pointed direction is provided in the Owner's Responsibilities section of Part 28:

- Develop the test program within the bounds of the plant's design basis
- Consider required test conditions and test consequences
- Minimize the impact of testing on plant risk
- Define appropriate contingency actions to manage plant risk during testing
- Use engineering evaluation or analysis in lieu of testing that would be impractical, cause detrimental interactions, or conflict with the design basis.

Additional reinforcing guidance on risk management is provided in the section on development and implementation of test procedures.

Part 28 requires definition of acceptance criteria for each testable system characteristic. The criteria are derived from the limits assumed in the analyses supporting the plant design. Correct performance of this step is critical in assuring design basis capability through testing. The owner is required to account for differences between analytical and test conditions considering the system configuration and boundary or process fluid conditions. In addition, test instrument loop accuracy must be addressed. However, explicit accuracy requirements are purposely not provided. This allows the owner flexibility on the choice of test instrumentation. Either the measured data or the test acceptance criteria must be adjusted conservatively to account for instrument accuracy prior to comparison. As a result, the lower the accuracy, the tighter the allowable performance band. Detailed guidance is provided in Appendices B and C for dealing appropriately with instrument accuracy.

A properly designed and implemented Part 28 test program will allow assessment of integrated system performance. However, simultaneous testing of all system components and subsystems may be impractical and is not required. A combination of separate tests whose results are evaluated in the aggregate is an acceptable approach. In addition, testing is to be performed at conditions as close as possible to expected least margin operating conditions. Where testing at these conditions is not feasible, or could risk equipment damage, analysis is performed to account for the differences.

Part 28 specifies a 5 year +/- 25% initial test interval for inservice testing. After each test, the subsequent test interval is established based on remaining margin, system performance trending, modification and maintenance history, service conditions, operating experience, safety significance, and risk assessment. The interval may be adjusted up to a maximum of 10 years. Retesting is required following replacement, repair, maintenance, or modification activities that could impact the system's ability to meet its performance requirements. This performance-based approach to the test interval allows the owner flexibility to adjust the interval in accordance with need.

Documentation of the bases for the test program is required for any system tested in accordance with Part 28. This documentation constitutes the system Test Plan. In addition, documentation and retention of procedures, results, deficiencies, data evaluations, and corrective actions is required.

Industry need for system testing standards has been a topic of continuing interest ever since the initial balloting of Part 15. The OM system testing standards and guides published to date were intended to address a need for guidance to establish integrated system testing requirements which would, if satisfactorily performed, demonstrate a system's readiness to perform its safety functions. This is the essence of Operability. There is copious operating experience from regulatory inspections and plant incidents describing cases where the installed system configuration failed to meet the design basis because system level performance criteria were not being tested. Nuclear Regulatory Commission Information Notice 97-90 describes a number of such incidents, and cites OM Parts 15 and 20 as resources for testing program guidance. Part 28 is intended to provide one-stop shopping for such information in a clear, consistent, consolidated, and accessible format.

Structure and Implementation

The format of Part 28 is as follows:

- Paragraphs 1 through 7: Generic testing program development guidance
- Appendix I: Testing guidance for ECCS in BWRs (adapted from Part 25)
- Appendix II: Testing guidance for ECCS in PWRs (adapted from Part 25)
- Appendix III: Testing guidance for Auxiliary Feedwater Systems
- Appendix IV: Testing guidance for Closed Cooling Water Systems (adapted from Part 2)
- Appendix V: Testing guidance for Emergency Service Water Systems (Open Cooling Water Systems)
- Appendix VI: Testing guidance for Instrument Air systems (adapted from Part 17)
- Appendix A: Operating Experience Lessons Learned on testing
- Appendix B: Guidance for Testing Certain System Characteristics
- Appendix C: Measurement Accuracy of System Characteristics

This structure provides flexibility for the owner to use the document for any fluid system in the plant. Specific testing requirements are provided for Emergency Core Cooling, Auxiliary Feedwater, Closed Cooling Water, Emergency Service Water, and Instrument Air Systems in Appendices I through VI. The generic process for developing an integrated system testing program is captured in the main body, consisting of paragraphs 1 through 7. As documented in the Owner's Responsibility paragraph, Part 28 may be applied to any system. If the system is one of those with its own mandatory appendix, the Specific Testing Requirements section directs the user to the appropriate mandatory appendix containing additional system-specific guidance to augment paragraphs 1 through 4. The user is then directed back to paragraphs 6 and 7. If the system does not have a mandatory appendix, the main body of Part 28, in conjunction with Appendices A, B, and C, provides a sufficient framework to develop a system testing program.

The Future

Consistent with the overall strategy, OM Parts 2, 17, and 25 will be deleted from the OM Standards and Guides document following publication of Part 28 since their content has been incorporated into the document.

What might the future hold for Part 28? If the promise of new nuclear construction comes to fruition, several opportunities exist. As Technical Specifications are developed for new plant designs, prescriptiveness could be reduced and value increased by adopting Part 28 testing regimens in lieu of some current surveillance tests. This approach could give the owner the flexibility to construct a performance based testing program focused on performing the appropriate testing to ensure that the physical plant configuration supports the underlying design basis. In addition, since testability issues exist with some current plant designs, there is an opportunity to utilize existing test program operating experience to optimize new plant designs for testing of active systems, thus allowing more comprehensive inservice testing to be conducted without additional burden.

The Subgroup on Functional Systems has already conducted conceptual discussions on the potential benefits of a risk-informed approach to integrated system testing. Utilizing the guidance already developed by the OM Committee for risk informed inservice testing, Part 28 could be revised to allow a risk-informed approach to be used when developing a test program for a chosen system. This would involve safety significance categorization on both a component and a subsystem level, using appropriate risk-importance measures. The results would be applied to the development of testing program scope and methods, as well as definition of required testing intervals, evaluating the impact on defense in depth, safety margins, and risk impact. Such an enhancement would serve to build on industry progress in risk informing existing processes while optimizing a Part 28 testing program.

Conclusion

OM Part 28, which will be published later this year, provides single point access to guidance for developing an integrated testing program for any process system. The document is the culmination of a long-term strategic effort to develop, ballot, and publish ASME OM consensus standards for integrated system testing. Potential future opportunities for Part 28 include integration with Technical Specification surveillance requirements for new generation nuclear units, and incorporation of guidance to risk-inform system testing program.

Session 1(b): Valves I

Session Chair: Kevin G. DeWall, Idaho National Laboratory

Evaluating MOV Margin and Performance Using MCC-Based Motor Torque Measurements

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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 such a method.

In 2006, EPRI issued Report 1011919, *Use of MCC-Based Motor Torque Measurements for Periodic Verification of Motor-Operated Valves*. This report developed, justified and validated, with limited data, the MCC-based Motor Torque Periodic Verification (MTPV) method. The methodology uses MCC-based measurements (most importantly, motor torque) in PV of MOVs and is applicable to torque-switch controlled closing strokes of rising stem MOVs with alternating current (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 are compared to these motor torque limits to verify adequate setup and to determine margin.

At the time EPRI Report 1011919 was published, limited test data were available for validation of the methodology. However, the available test data (data from 4 similar gate valves) showed that the predicted thrust based on measured motor torque matched the measured stem thrust favorably. After publication of EPRI Report 1011919, EPRI identified additional nuclear plant MOV test data (data from 11 other MOVs) which could be used for a more extensive validation of the MTPV method. The results of this expanded validation verified those from the earlier assessment based on limited data. EPRI Report 1015069, documenting the final MTPV Methodology and the expanded method validation, will be issued in 2008.

In addition to the validation analyses included in the final report, the data from the 15 in-plant MOVs were evaluated to determine the potential effect of the MTPV methodology on MOV setup limits. Based on this evaluation, the MTPV method is useful for MOVs that have an operational margin (margin to lower limit) and a margin against structural damage (margin to upper mechanical limit) of at least 25%. There is no constraint with regard to margin against motor torque capability and in fact use of MTPV is likely to improve margin in this category. Accordingly, 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.

BACKGROUND

In 1989, the US Nuclear Regulatory Commission (NRC) issued Generic Letter 89-10, *Safety-Related Motor-Operated Valve Testing and Surveillance*, which required nuclear power plants to review and validate design basis requirements for safety-related 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 required facilities to develop a PV 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 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 EPRI to develop, justify and validate an MCC-based 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 paper.[1]

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 are 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 differential pressure (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.

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[1] Note that the MTPV method is independent of the specific methods used for determination of motor torque, so long as measured motor torque is expressed with a justified uncertainty. For example, plants currently using diagnostic systems to measure motor power could apply the MTPV method if the motor power data are converted to motor torque and measurement uncertainty is addressed.

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. The parallel test data are 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 Control Switch Tip (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 CST. 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[2] (the terms in the equation are detailed under "Nomenclature" at the end of this paper),

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[2] 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.

$$F_{MOV} = \frac{\left(MT_{MEAN,CST} - MT_{HOTEL}\right)}{TH_{CST}} = \frac{(FS)}{(OAR)(EFF)} \qquad (1)$$

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 typically based on 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. Specific methods for determining the reduced voltage motor torque capability are defined within a plant's MOV program and these methods may vary between plants. For analyses included in this paper, the reduced voltage motor capability is determined using the following equation.[3] This value of motor torque at reduced voltage is adjusted to account for uncertainties, as shown in Figure 1.

$$MT_{VRED} = \left(MT_{NOM}\right)\left(\frac{V_{RED}}{V_{NOM}}\right)^2 \qquad (2)$$

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)[4]:

- torque switch repeatability
- thrust measurement uncertainty
- motor torque measurement uncertainty
- stem factor uncertainty
- actuator efficiency uncertainty
- inertial thrust uncertainty
- rate of loading (ROL)

[3] 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.
[4] Note that not all of these uncertainties are applicable to each limit.

Most of these uncertainties are identified within existing plant MOV programs. A value for the actuator efficiency variation, an uncertainty not typically quantified by plants, has been justified by EPRI based on analyses of in-plant MOV data. Plants may use the EPRI-justified value for this uncertainty or may elect to justify their own value for this term.

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; specifically, motor torque measurement uncertainty and actuator efficiency 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 CST is compared to the upper and lower motor torque limits (see **Figure 6**).

Acceptable Changes in Motor Torque Between Baseline and Subsequent Tests

In the MTPV method, the setup and general conditions of an MOV need to remain consistent between the baseline and subsequent tests to ensure a consistent relationship between motor output torque and actuator thrust. Maintenance activities that could potentially affect the relationship between actuator thrust and motor torque are not permitted between the baseline test and subsequent tests. Examples of specific maintenance activities which are not permitted are detailed later in this paper (see *Conditions Requiring a New Baseline Parallel Test*). However, in the absence of these activities, the measured motor torque at CST would be expected to remain relatively constant between tests, and any variation in motor torque is attributable to either an unexplained degradation effect or the following uncertainties.

- Motor torque measurement uncertainty
- Actuator efficiency uncertainty
- Torque switch repeatability

Accordingly, a change in measured motor torque at CST greater than the effect of the uncertainties listed above may indicate a problem with the MOV (e.g., degradation). Based on analyses of the 15 validation MOVs, an allowable variation in measured motor torque at CST is ~20%. That is, an MCC-only test performed subsequent to the baseline test may only be used to satisfy an MOV's periodic verification requirements if the measured motor torque at CST is within 20% of the baseline motor torque at CST. Variations greater than 20% may indicate a significant change in the MOV performance and further engineering evaluation of the valve or an at-the-valve test may be required to determine valve operability and margin.

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).

$$\mathrm{MARGIN} = \frac{(\mathrm{MT}_{\mathrm{MEAN,\,CST}} - \mathrm{MT}_{\mathrm{LL,\,CST}})}{(\mathrm{MT}_{\mathrm{LL,\,CST}} - \mathrm{MT}_{\mathrm{HOTEL,\,2nd\,TEST}})} \qquad (3)$$

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, an alternative test (e.g., at-the-valve) is needed to assure positive margin.

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. The reduced voltage motor torque capability represents the *maximum* motor output torque that the MOV is allowed to demand under design basis conditions. Since the upper limit could be defined by either the MOV mechanical limit or the reduced voltage motor torque capability, the MTPV method requires comparison of the *Maximum Motor Torque at CST* to both limits. If the *Maximum Motor Torque at CST* is less than the controlling (lowest) 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)

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, 15 MOVs met the MTPV applicability requirements and had test data with stem thrust and motor torque measurements from two separate tests. These 15 MOVs included flexible wedge gate valves, double disk gate valves, and globe valves ranging in size from 1 to 14 inches. The actuators for these MOVs included SMB-0, SMB-00, and SMB-000 and the motors ranged in size from 5 to 25 ft-lbs (460V and 575V motors).

Validation of the method is based on a comparison of "Measured Thrust at CST" to "Predicted Thrust at CST based on measured Motor Torque". This comparison is made for results obtained in the second test. This validation approach does not determine or confirm an *absolute* accuracy for the MTPV method because there is no standard reference measurement (measured stem thrust has an associated measurement uncertainty). However, a correlation between thrust determined with the MTPV method and stem thrust measured directly validates use of the MTPV method in lieu of direct stem thrust measurement.

For 14 of the 15 MOVs, the difference between the predicted mean thrust at CST ($TH_{MEAN, CST}$) and the measured stem thrust at CST (TH_{CST}) was within the uncertainty associated with measurement of motor torque and stem thrust. **Figure 7** shows sample data comparing measured thrust to predicted thrust with consideration of measurement uncertainty for 5 of the validation MOVs. As summarized in Table 1, the predicted mean thrust at CST ($TH_{MEAN, CST}$) was within 24% of the measured stem thrust at CST (TH_{CST}), for these 14 MOVs.

For one of the 15 MOVs (MOV 3), the deviation between predicted and measured thrust was 63.4%. For this outlier, the deviation can not be explained by the combination of uncertainties for stem thrust measurement and the MTPV method. It is important to note that the change in measured motor torque at CST between the baseline and second test for this MOV was 41%. This variation in measured motor torque exceeds the MTPV acceptance criteria for a valid MCC-based periodic verification test (> 20% change in motor torque) and indicates that either the MOV setup has changed between tests or that there may be a problem with the MOV. In practice, this test result would indicate to the user that further engineering evaluation or an at-the-valve test is required.

Of the 14 MOVs that had a predicted thrust consistent with expected uncertainty, one of these (MOV 2) also had a change in motor torque from baseline to second test that exceeded the criterion of 20%. Specifically, the observed change in motor torque was 23%. Although this MOV would have been "screened out" from use of MCC-based results in periodic verification, further evaluation showed that the results fell within the expected and acceptable uncertainty range for this MOV. This result shows that the 20% screening criterion is suitable, and on the conservative side.

Figure 8 shows the correlation between thrust determined with the MTPV method and stem thrust measured directly at the valve. The perfect agreement line (45^o line) is included on the figure. As shown in the figure, the data for 14 of the 15 MOVs fall near this line. The data validate that the MTPV methodology determines thrust consistent with direct stem thrust measurement, with some additional associated uncertainty. This additional uncertainty is included in the MTPV method during determination of upper and lower MOV setup limits.

EFFECT OF MTPV METHOD ON TYPICAL MOV SETUP

The general procedure for determining MOV setup limits based on motor torque measurements in the MTPV method is analogous to that used for determining setup limits based on stem thrust measurements in current PV methods. However, the application of data uncertainties to these limits is dependent on the parameter measured during MOV testing (i.e., stem thrust or motor torque). As such, the motor torque setup limits for an MOV are different than just a simple factor times the MOV's analogous thrust limits.

The 15 MOVs used in the MTPV method validation were evaluated to determine the potential effect of the MTPV method on an MOV's upper and lower setup limits. Upper and lower limits to be compared with measured thrust were determined with consideration of uncertainties typically used in industry (measurement uncertainty, stem factor uncertainty, rate of loading, etc.). Upper and lower limits to be compared with measured motor torque were determined in accordance with the MTPV method, with consideration of similar uncertainties. The motor torque limits were then converted to thrust limits using the MOV factor (determined from the baseline test data).[5]

Upper Limit

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. This reduction is due to the higher measurement uncertainty for motor torque compared to thrust. The observed reduction in mechanical upper limit ranged from 9 to 19% for the 15 MOVs.

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. 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. The observed improvement ranged from 7 to 121% for the 15 MOVs.

Lower Limit

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. This increase in the lower limit is due to the higher measurement uncertainty for motor torque compared to stem thrust. The observed increase in the lower limit ranged from 5 to 24% for the 15 MOVs.

Setup Window

The upper and lower limits discussed above define an MOV's setup window.

As shown in the figures, the setup window between the mechanical upper limit (left upper bar) and the required thrust lower limit (lower bar) is smaller in the MTPV method than the setup window determined using direct stem thrust measurements. However, the setup window between the reduced voltage upper limit (right upper bar) and the lower limit is larger in the MTPV method. This improvement is due to a reduction in the uncertainty of the reduced voltage limit as discussed above. .

CONCLUSIONS

Based on the validation analyses and the evaluations related to the effect of the MTPV method on typical MOV setup, the MTPV method satisfactorily determines Operational Margin and MOV 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. Specifically, the MTPV method would

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[5] In plants, data uncertainties will be accounted for and applied in accordance with individual plant procedures, which may differ slightly from the procedures used in this analysis. Nonetheless, the general trends discussed here will be consistent regardless of the specific procedures followed.

be most useful for MOVs that have an operational margin (margin to lower limit) and a margin against structural damage (margin to upper mechanical limit) of at least 25%. There is no constraint with regard to margin against motor torque capability, and in fact, use of MTPV is likely to improve margin in this category. Accordingly, 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.

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1. Limitorque SEL-3.
2. Limitorque Technical Update 05-01, "Actuator Output Torque Calculation SMB/SB/SBD Actuators/3 Phase Motors," January 2005.
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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

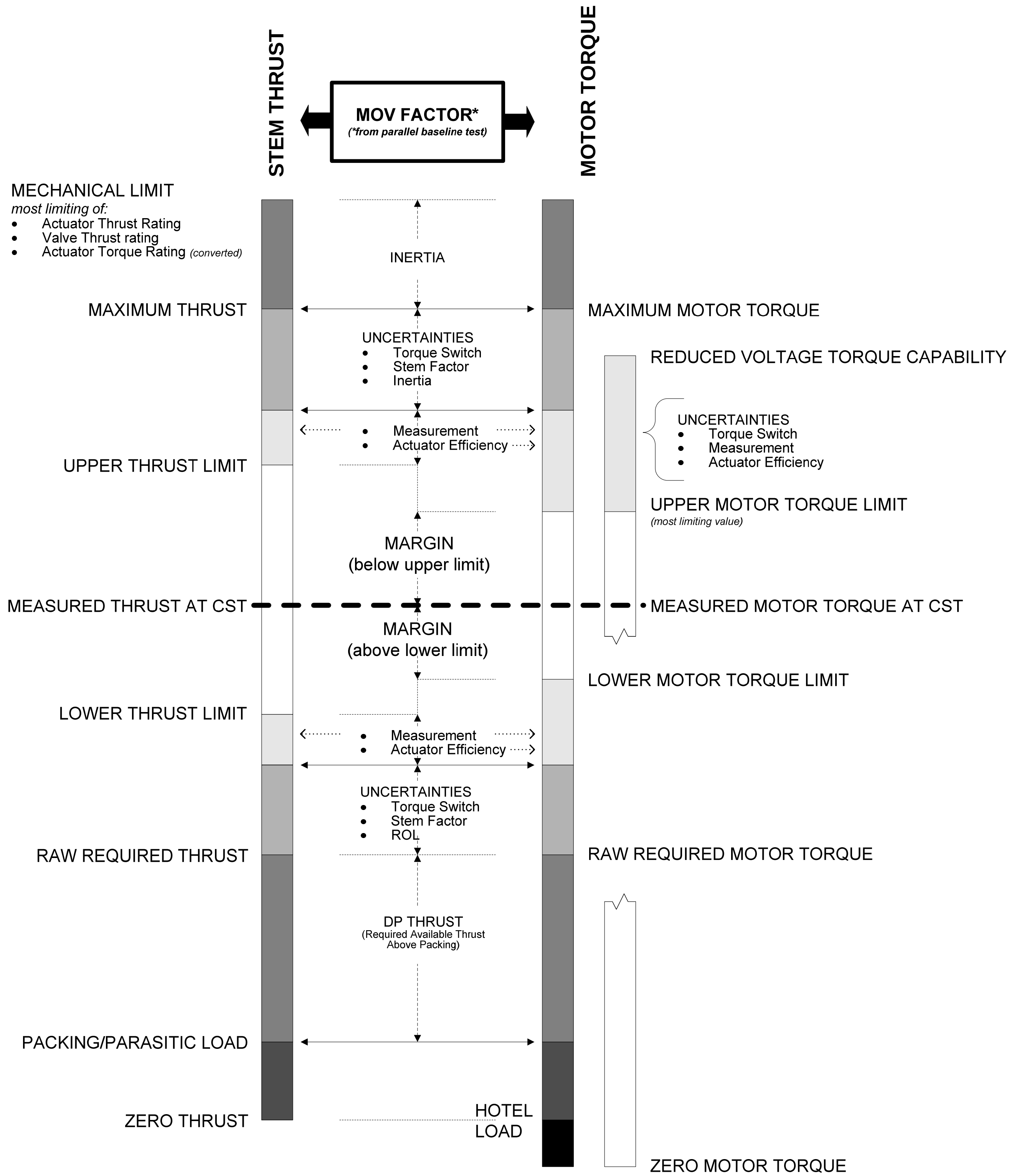


Figure 1
Motor Torque Periodic Verification Method Limits and Margin

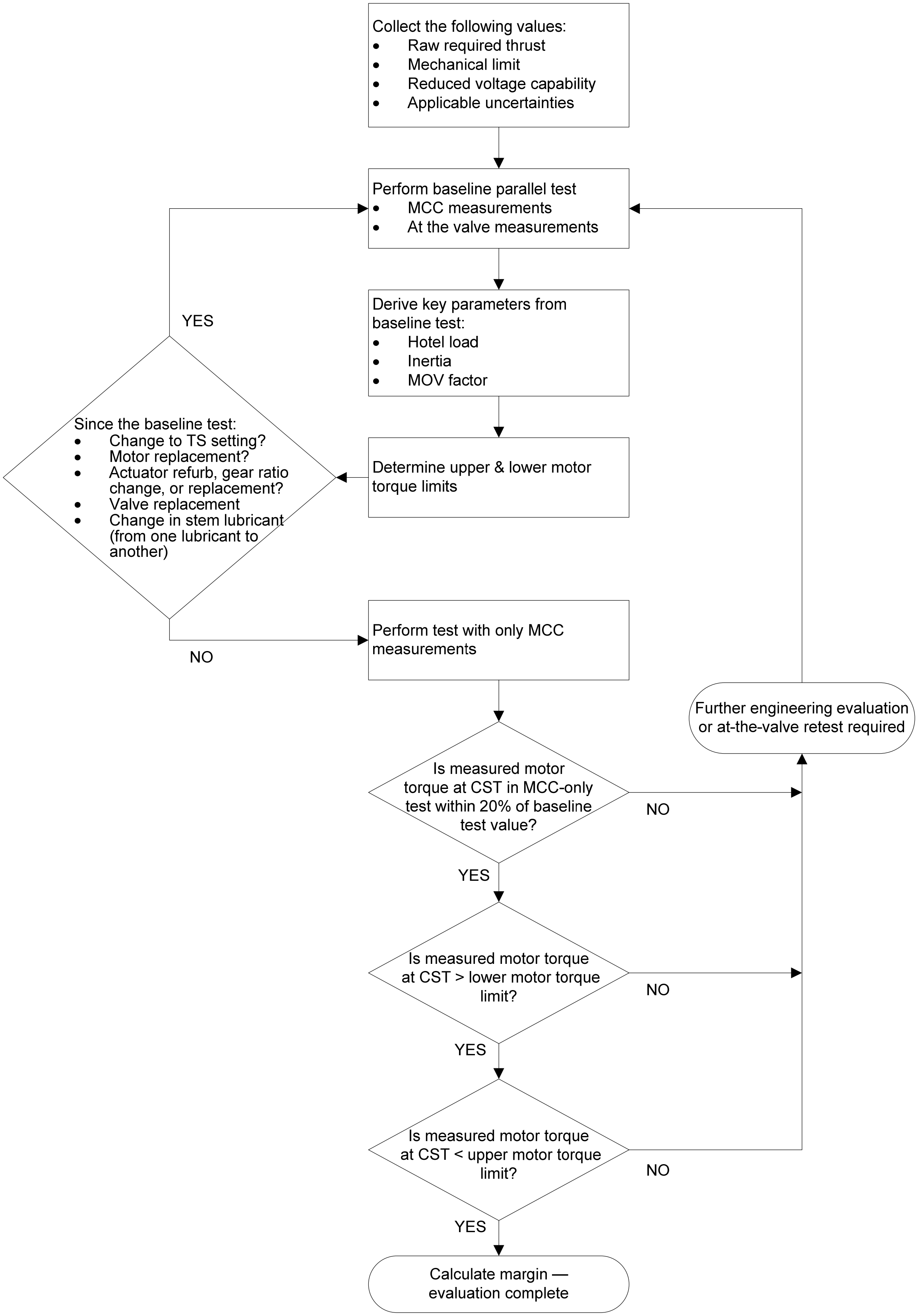


Figure 2
Motor Torque Periodic Verification Method Flowchart

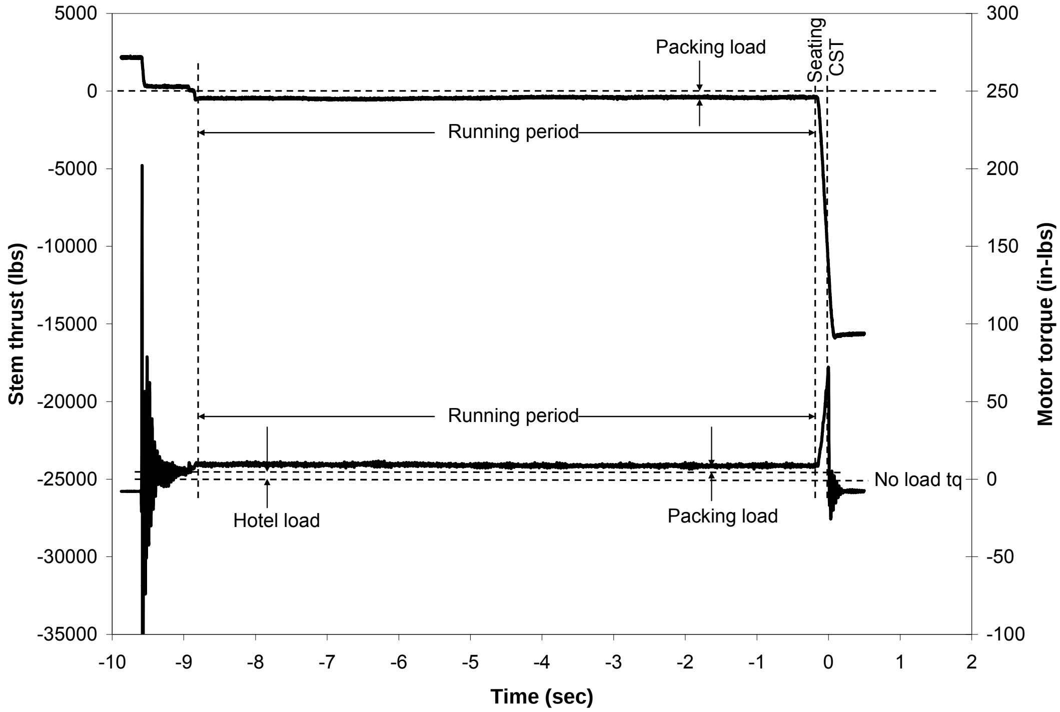


Figure 3
Overlay of Measured Motor Torque (bottom trace) and Stem Thrust (top trace)

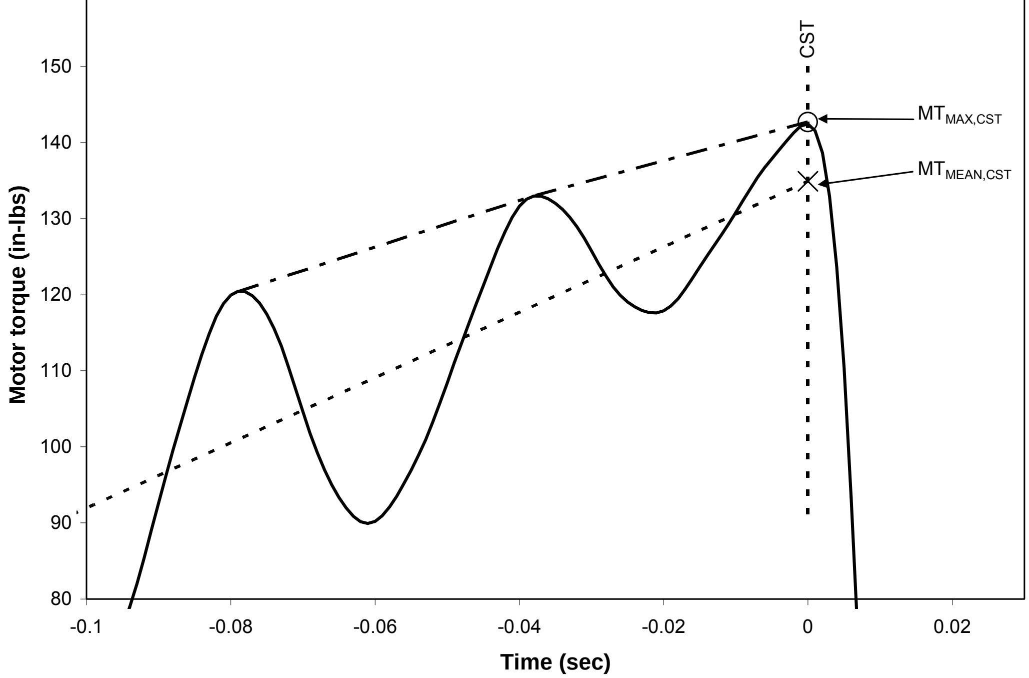


Figure 4
Measured Motor Torque Near CST – Example with Significant Oscillations

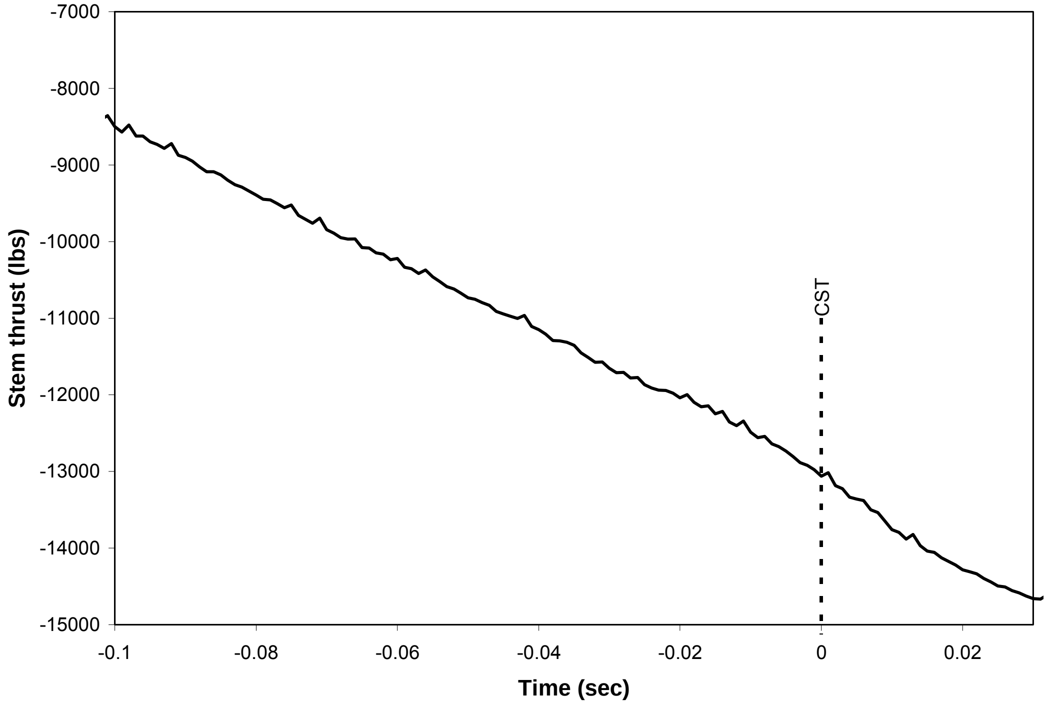


Figure 5
Measured Stem Thrust Near CST for Example Corresponding to Figure 4

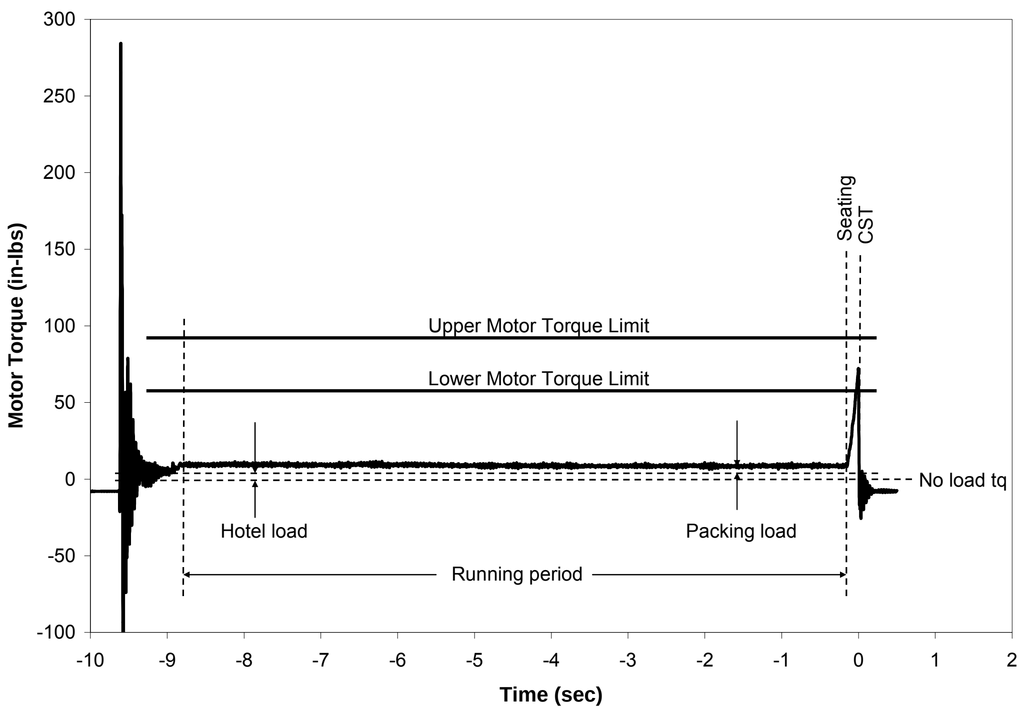


Figure 6
Measured Motor Torque from MCC-Only Test

Table 1
Percent Difference Between Predicted Mean Thrust at CST based on measured motor torque ($TH_{MEAN, CST}$) and Measured Stem Thrust at CST (TH_{CST})

Valve	$(TH_{MEAN, CST} - TH_{CST}) / TH_{CST}$
MOV 1	-3.1%
MOV 2	23.2%
MOV 3	63.4%
MOV 4	24.0%
MOV 5	-0.1%
MOV 6	-13.3%
MOV 7	18.9%
MOV 8	-0.7%

Valve	$(TH_{MEAN, CST} - TH_{CST}) / TH_{CST}$
MOV 9	11.5%
MOV 10	8.1%
MOV 11	-13.4%
MOV 12	-2.2%
MOV 13	4.3%
MOV 14	6.3%
MOV 15	0.5%

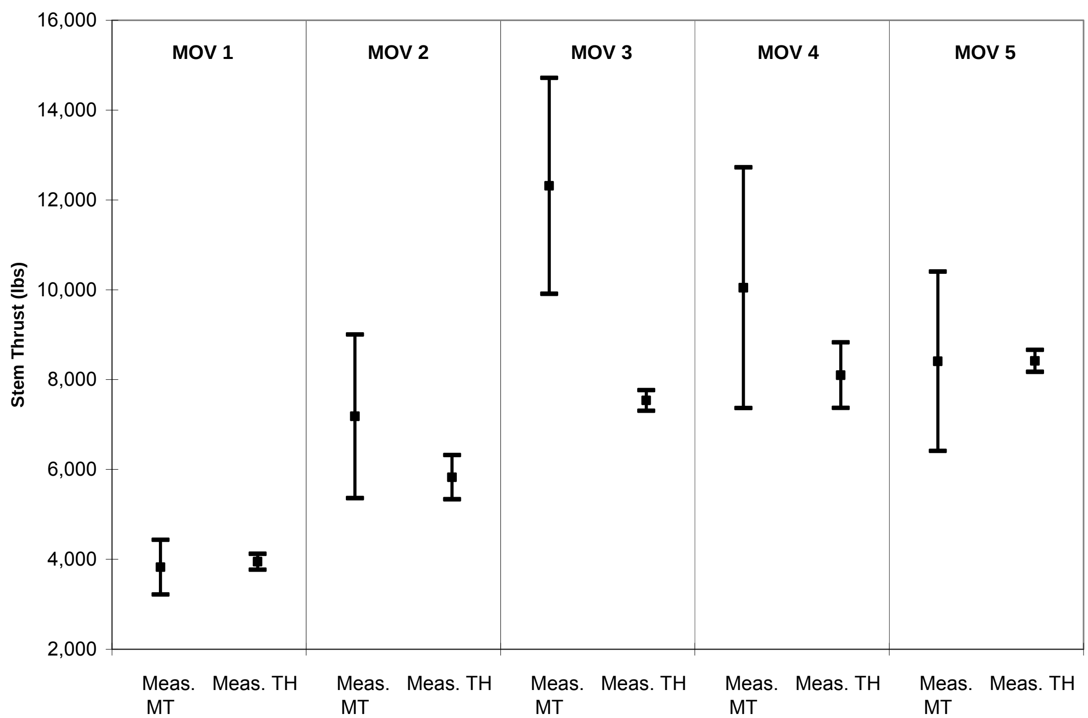


Figure 7
Comparison of Predicted Mean Thrust at CST (based on measured motor torque) to Measured Stem Thrust at CST, Including Measurement Uncertainty (MOVs 1 – 5)

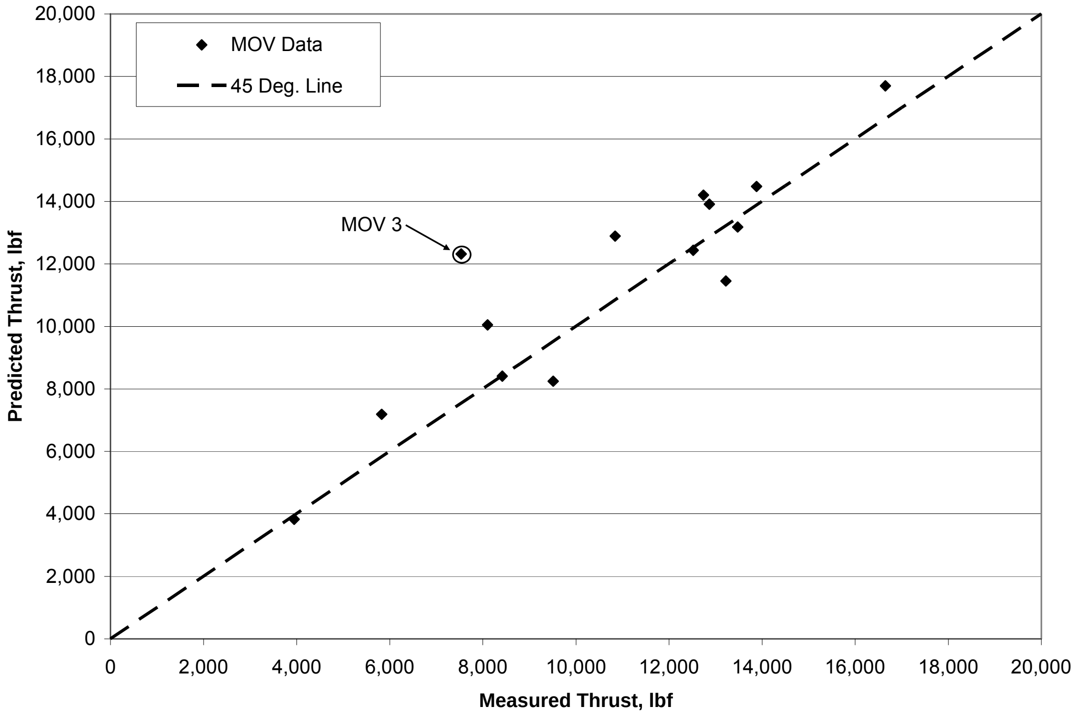


Figure 8
Graphical Comparison of Predicted Mean Thrust at CST (based on measured motor torque) to Measured Stem Thrust at CST

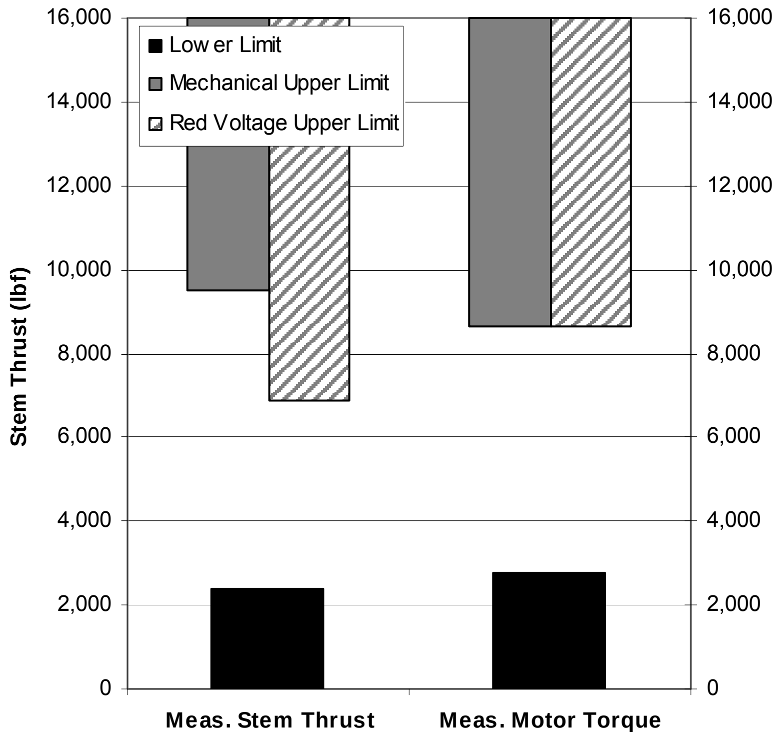


Figure 9
Comparison of Setup Windows for MOV 1
(Stem Thrust vs. Motor Torque Measurements)

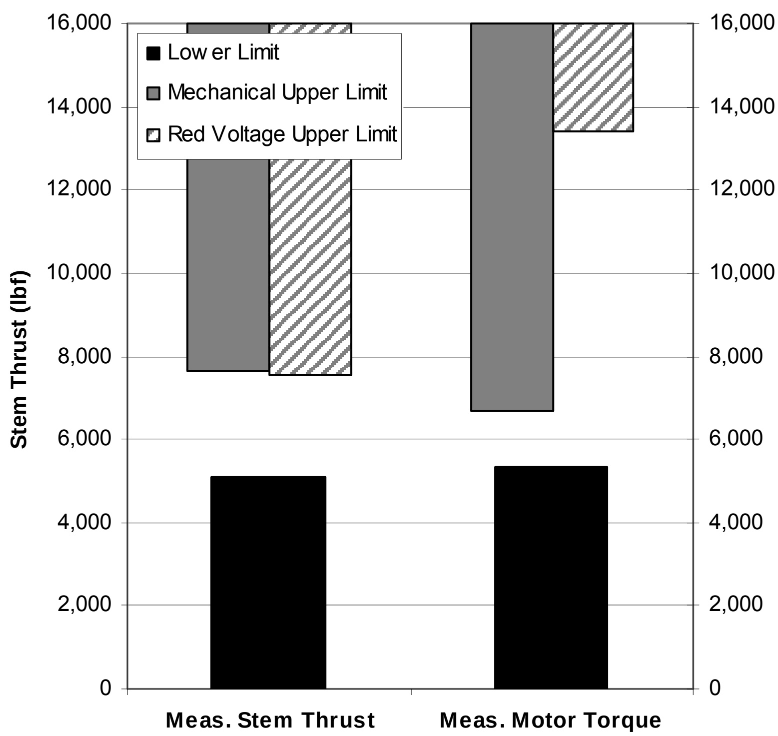


Figure 10
Comparison of Setup Windows for MOV 2
(Stem Thrust vs. Motor Torque Measurements)

Motor Operated Valve Testing for Dummies

Shawn Comstock, Palo Verde

Abstract

Motor Operated Valve (MOV) degradation is a recognized and expected phenomenon. Mechanical components with rubbing surfaces simply wear out over time with use, especially if they are in a corrosive environment. Modern tools utilize data acquisition systems to more accurately determine what's going on inside of these components. The modern tools provide graphs of performance from sensors installed at various locations. Analysis of these traces can determine when it's time to work on an MOV before it breaks.

Diagnostic test equipment can tell you a lot about wear in a valve by simply measuring the time between events shown in a trace. These performance times are compared to each other in successive tests to determine wear extent. The method works the best when a new valve of the same model is available for baseline testing. This condition monitoring approach can reveal wear from stem nuts, stem to disc interfaces, and other general wear between mating gear parts.

When an alternating current electrical motor powers the MOV, the time between events will remain constant if the test conditions are duplicated. In essence, the MOV is as reliable in this respect as an analog clock. Small changes in the event times reveal degradation. This paper will discuss a specific application of this wear measurement technique that was successfully used to avoid costly and unplanned inspections to address a common cause failure mechanism.

Introduction

There have been numerous papers and presentations about motor operated valves in the nuclear industry for the last 20 years. During this time, the test equipment and methods for assessing design basis capabilities has evolved significantly. The application of this technology can be difficult in the absence of adequate training or fundamental knowledge about how a motor operated valve works. However, a simple comparison of successive measurements and comparison of times between events can reveal degradation to anyone with a fundamental grasp of trend analysis.

This paper will describe a specific failure at Wolf Creek Nuclear Operating Corporation to show the basic technique of analysis used with modern test methods. The application of a stroke time method to estimate and predict wear between mechanical parts will also be described to show the reader one method that can be used to address the problem of determining when a valve requires attention.

Case History

There were two identified issues on valves used in a service water screen wash application and relating to unanticipated loss of material from the valve disk at the connection point to the valve stem. The valve tagged as EFHV91 actually failed. A weekly task at Wolf Creek is to verify flow through the service water screen wash valves to keep debris from accumulating on the traveling screens. When the expected screen wash spray did not occur, troubleshooting began to determine the cause. Work was planned to evaluate the condition of valve EFHV91. The work evolution began with a diagnostic MOV test using the Crane Nuclear VIPER system with a stem mounted strain gage. The EFHV91 as-found test revealed a trace that had never been seen before on any valve at Wolf Creek. There was an "unexplained" slippage (or spike) in a region of the valve stroke. The As-Found test was determined to be unsatisfactory based on the presence of the anomaly.

After it was recognized that flow was not passing through the valve, disassembly revealed that the disk had separated from the valve stem. A temporary modification allowed the removal of the valve disk from the system for a limited period of time. Our main concern after identifying the problem with the EFHV91 valve was to verify we did not have a simultaneous problem with the EFHV92 valve in the other safety system train. Once it was verified that the valve EFHV92 opened, the valve was tagged in its open position while repairs commenced on EFHV91. The EFHV91 valve was replaced with a totally new valve (including body and all trim).

The following discussion will describe the way that the valve should work (using the new valve installed in EFHV91) and will compare the new valve's performance to the condition of the VIPER trace obtained from the sister valve that is tagged as EFHV92.

Figure 1 illustrates what the internals of these valves look like:

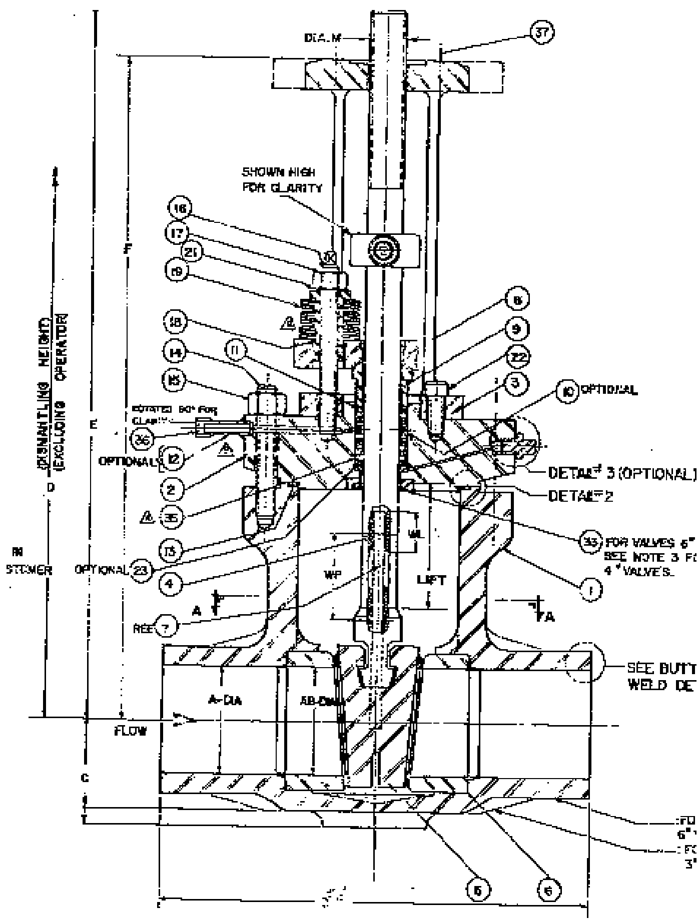


Figure 1 – MOV Detail

The point of interest is the area where the stem connects to the disk. The stem is machined to fit within a slot in the disk. For this particular gate valve, the points of contact between the stem and the disk in the opening direction are referred to as the "ears." The stem is machined to fit within the slot in the disk to open and close the valve. There are clearances associated with this connection that are not defined by available documentation, but the following analysis of the MOV diagnostic traces explains the situation found on EFHV92 and projected the condition of similar valves in the plant. For comparison purposes, the test of the new valve will be the foundation of the analysis. The section of the VIPER trace in Figure 2 shows the opening signature on the new valve installed at location EFHV91.

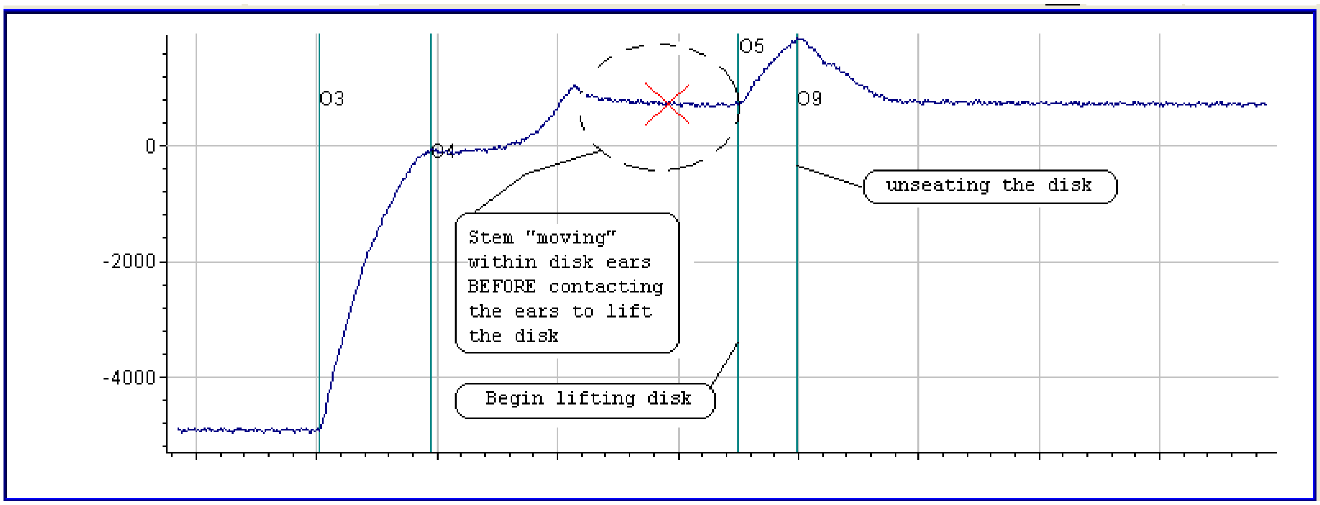


Figure 2 – Crane Nuclear Viper Trace of New Valve

This is a "textbook" trace of how a gate valve should OPEN. The MOV analyst assigns standard markers to MOV traces for evaluation. The points of interest on Figure 2 are marked with the standard markers and a custom mark, which mean the following in terms of MOV operation:

- O3 - This is called the "start of stem decompression," it basically means that the closing force has been removed, and the stem is beginning to relax. The Limitorque is "turning," but there is nothing yet moving in the valve.
- O4 - This is called the "end of stem decompression" (or opening "zero"). Now the stem is "relaxed," but it has not yet started to "lift."
- The "circled" area on the trace is the point where the STEM begins to "lift." The small peak on the left depicts the amount of force necessary to overcome the static friction between the stem and the valve packing. After the peak, the stem is on its way to the disk ears to lift the valve. The stem is now "moving."
- O5 - This is the point where the stem contacts the disk ears and begins pulling on the disk. The valve is still CLOSED.
- O9 - This is the point where the disk "unseats" (or cracks). Once the disk breaks free the force required to move the valve to the OPEN position drops off substantially and the disk is lifted to the full OPEN position.

Everything that happens within the MOV is a function of time, gear ratio and assembly tolerances. Even though we do not know the exact dimensions of the internal gaps, we do know the output RPM of the Limitorque, the valve stem thread pitch, and the valve stem thread lead.

Limitorque Technology Basics

Stem pitch (P) describes the ratio of threads per inch. These valves have 5 threads over a 1 inch measurement, which yields a pitch of 1/5 or 0.2 inch. The stem lead (L) is the amount of travel the stem will have with one full revolution of the Limitorque. A valve's pitch is multiplied by the number of thread starts to determine the value of L. This valve stem has 2 thread starts; therefore the value of L is 0.4 inch. One full revolution of the MOV stem results in 0.4 inch of travel.

The other two parameters to know when determining a motor operated valve's performance characteristics are the motor revolutions per minute (RPM) and the actuator's overall ratio (OAR). This enables the determination of the output rpm from the Limitorque operator. In this case, the motor is 1,700 RPM and the actuator has a 100 OAR, which means that the valve stem itself sees 17 RPM.

Correlation of Basics to Analysis

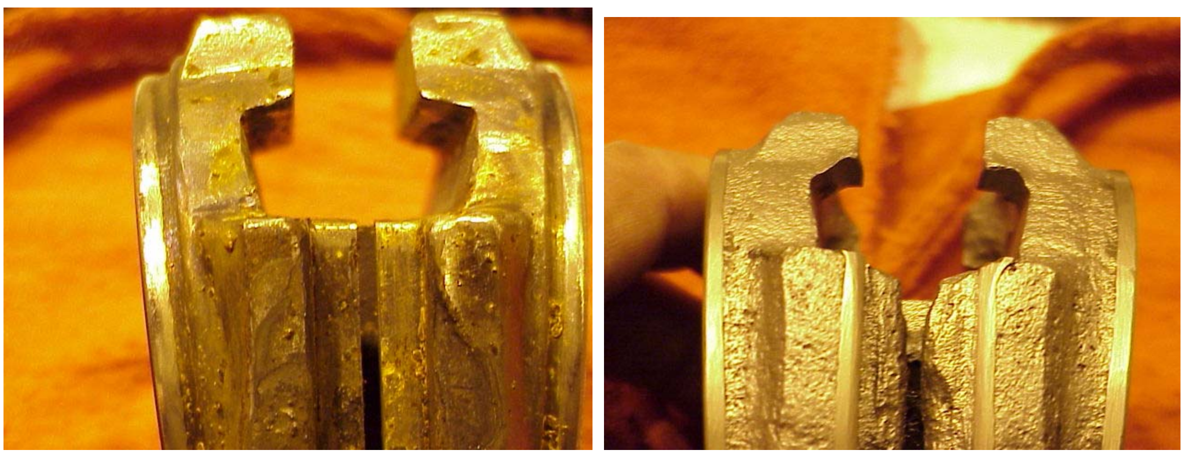
Understanding the relationship between motor and drive sleeve RPM as well as stem lead allows us to convert MOV diagnostic test event time intervals to measurement of internal clearances. The Limitorque is nothing more than an automatic micrometer and we can convert event times to distance using the following formula:

$$\frac{M_{rpm}}{OAR} \bullet \frac{1 \bullet min}{60 \bullet sec} \bullet (event_{time}) \bullet L$$

This formula allows us to convert the time between selected events of the MOV diagnostic traces to determine internal clearances. For the new valve on EFHV91, we know that the TIME between the beginning of stem MOVEMENT (the "peak" to the right of O4 and to the left of O5) and the O5 mark (where the stem actually begins pulling on the disk) is 0.171 seconds. This means that the DISTANCE that the STEM MOVES in the disk slot is equal to 0.0194 inch. This is a sensible answer for the clearance between the stem and disk ears (slightly more than 1/64 of an inch).

MOV Disk Comparison

The preceding discussion is meant to establish how to measure the clearance between the valve pieces that are moving at a particular time. The fact that there is a new valve installed on EFHV91 establishes the "new" condition of this particular model of MOV. This performance can be extrapolated to the other 5 valves of the same model in the same service conditions. The next few pictures compare the new disk to the disk removed from EFHV91 (the old valve that failed):



NEW Disk OLD Disk

Figure 3 – Disk Slot Comparison

The pictures above illustrate the differences between the new disk and the one that had been in service for over 20 years. The disk in the new valve was inspected before installation and it was determined that the machined area of the disk slot is flat. Notice that the area of the slot in the old disk is rounded. The areas at the top of the disk are the "ears" that the T-handle on the valve stem contacts. The disk slot ears on the old disk had worn to the point that the valve stem T-handle could be pulled through this area.

What Caused This Problem

This wear pattern was introduced by a combination of valve design, service conditions, and frequent operation. The disk itself is made of carbon steel. The stem that fits into the slot shown above is made of stainless steel. In combination with Wolf Creek's lake water, this establishes a classic condition of galvanic corrosion when the valve is open. The disk slot ear is the piece of material that serves as the sacrificial anode. This, in combination with operation of the valve on a weekly basis, significantly accelerated the normal wear rate of the valve disk's slot ears.

Trace Shape on Sister Valve

After EFHV91 was repaired, testing and inspection of the EFHV92 valve commenced. Figure 4 shows what the as-found test from EFHV92 looked like:

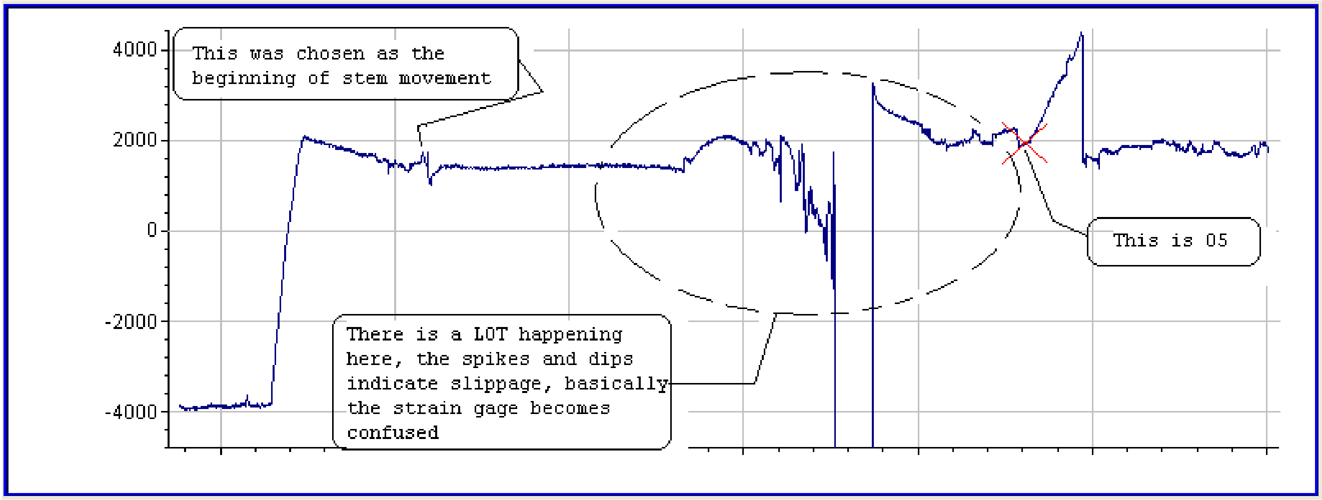


Figure 4 – EFHV92 As-Found Trace

This trace made no sense at first. It was initially believed that the strain gage was malfunctioning. Electricians reported a loud noise during opening that they initially believed was coming from the Limitorque. As Engineers studied the trace, it was determined that the strain gage was functioning properly and the trace actually was following the slippage of the stem within a damaged disk slot.

The TIME between the point chosen for the beginning of stem MOVEMENT on the EFHV92 valve and the O5 point was 3.450 seconds. This is 20 TIMES the time that the new valve took to travel this distance.

Applying the equations given above to determine the measurement of the distance reveals that the stem traveled 0.391 inch before it began to pull on the disk ear to unseat the valve. Examination of the disk removed from EFHV92 reveals that this gap is observable in the removed piece. There was enough material remaining to allow the stem to grab the disk and unseat the valve.

This trace was VERY UNIQUE. The MOV Engineer at our site has reviewed diagnostic traces for over twenty years and had never seen one look this strange. At first we were confused, but the lessons learned on EFHV91 led to the determination that the diagnostic test was a failure.

What About the Other Four Valves?

Given the results on the first 2 valves, the challenge remained on how to evaluate the condition of the other four valves that are exactly the same design and in the same system. Fortunately for Wolf Creek, the other 4 valves had been tested within the last year. Table 1 illustrates the results of those tests using the stroke time method for the wear evaluation of the disk slot ear gap:

Table 1 – Disk Slot Gap Evaluation

Valve	Test date	O5 time	Stem starts to move	Clearance time	Computed clearance
EFHV97	12/6/06	2.548	1.554	0.994	0.113
EFHV98	2/14/07	2.079	1.477	0.602	0.068
EFPDV19	6/6/07	2.427	1.532	0.895	0.101
EFPDV20	8/14/07	2.522	1.633	0.889	0.101

These four valves have been installed for several years. EFHV97 and EFHV98 have been in-service since the plant came on line (over 20 years). EFPDV19 and EFPDV20 were both replaced approximately 10 years before the evaluation. The four other valves are not operated at the same frequency as the first two. Typically, the other 4 valves are operated quarterly rather than weekly (about 12 times less). Valves EFHV97 and EFHV98 are vent valves that are only exposed to water during pump starts, so there are very few opportunities for galvanic corrosion to occur. Valves EFPDV19 and EFPDV20 are normally closed, so the disk slot ears are not in contact with the valve stem t-handle. All valves will exhibit some wear in service, so it is not inconsistent that the measured gaps of these valves would be greater than the gap measured on a brand new valve assembly.

The MOV Weak Link Analysis for these six valves computes the strength of the disk slot ears. Analysis of the calculation for the strength of the ears reveals that even with a 50 % reduction of the section thickness of the upper ear thickness that the valve disk is capable of withstanding over 140% of the design basis loading. The disks removed from EFHV91 and EFHV92 reveal no sign of load deformation from the stem. The disk on EFHV91 disconnected from the valve stem because there was no material at the connection point for the valve stem to pull. The EFHV92 valve exhibits significant material loss (much more than 50% of the thickness) yet the valve disk ears were still in contact with the stem and the valve opened.

Conclusion

Using the “computed clearance” from the new valve installed at location EFHV91, the following criteria can be applied to these six valves for diagnostic acceptance:

The maximum time for stem movement initiation to beginning to pull on the disk can be established at

0.171 (sec) the time that the new valve took to take up the clearance
+ 2.6 (sec) the time it would take to move the stem 0.3” - 50% of thickness remaining
2.771 seconds would mean that 50% of the design dimension of the disk ear remained

The analysis of the MOV diagnostic traces, computation of internal clearances, and review of the structural weak link calculation reveals that the acceptance criteria (for determining 50% of remaining disk ear thickness) is not critical. All MOV tests are reviewed and compared with historical data. The situation that occurred on EFHV91 would have been identified if the valve was tested on a more frequent basis and if the MOV Engineer recognizes the significance of the change in the signature. Review of the historical traces for the six valves is tabulated below:

EFHV91:

Year	Clearance Time (sec)
1994	1.229
1997	1.590
2000	2.242

EFHV92:

Year	Clearance Time (sec)
1994	1.048
1997	1.456
2000	1.947
2007	3.450

EFHV97:

Year	Clearance Time (sec)
1994	0.815
1997	0.912
2000	0.923
2005	0.946
2007	0.994

EFHV98:

Year	Clearance Time (sec)
1994	0.397
1997	0.443
2000	0.502
2007	0.602

EFPDV19:

Year	Clearance Time (sec)
1994	0.637
1997	0.250
2000	0.421
2005	0.707
2007	0.895

EFPDV20:

Year	Clearance Time (sec)
1994	0.840
3/97	1.066
10/97	1.083
10/97	0.280
2000	0.433
2007	0.889

All of the valves reveal changes in the “clearance time,” but EFHV91 and EFHV92 stand out because these times basically DOUBLED between 1994 and 2000. It appears that the trend was set in the first test sequences but it was not identified by MOV review of the data.

The other four valves do exhibit lengthening of the “clearance time,” but the increases are much more gradual. There is no indication on the EFHV97, EFHV98, EFPDV19, or EFPDV20 traces taken in the past year that any “slippage” or excessive wear exists. None of these valves approach the derived maximum time of 2.8 seconds (signifying 50% wear of the disk ear).

Solenoid Valve Testing Using Commercial Valve Diagnostic Platforms

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ABSTRACT

Proper maintenance and testing of components critical to plant operation and safety is essential to maintain the capacity factors expected of Nuclear Power Plants (NPPs) in today's competitive energy markets. Diagnostic testing of Motor Operated Valves (MOVs) and Air Operated Valves (AOVs) have been performed for over 25 years. This paper identifies diagnostic test methodologies that have been developed using an existing diagnostic platform to perform accurate stoke time test measurements for Solenoid Operated Valves (SOVs).

These tests may be used to perform accurate stroke time testing. Additionally the capability exists to support troubleshooting and preventative maintenance activities by identifying malfunctions of valve internals and degradation of electrical components once baseline performance is established.

INTRODUCTION

Process SOVs are installed in systems that are relied upon to perform critical plant functions to support safe plant start-up, operations and shut down. They are also relied upon to mitigate design basis accidents and to maintain the plant in a safe shutdown condition.

The historical technique for measuring stroke time for Power Operated Valves (POVs) has been performed by monitoring position indication switches using a stopwatch. This technique is adequate for relatively slow acting MOVs and AOVs but is inadequate for SOVs due to their inherent nature. Most process SOVs complete their full stroke in less than a half of a second and in many cases in less than 100 milliseconds. This extremely fast stroke time makes stop watch measurements meaningless since the actual stroke time and health of the SOV cannot be ascertained.

BACKGROUND

Operating events involving observed or potential common mode failures of AOVs, SOVs, and MOVs were first documented in NUREG-1275, Operating Experience Feedback Report. Events that specifically involved SOVs were identified in Volumes 2 and 6 of NUREG-1275. The findings of NUREG 1275 lead to the establishment of programs to verify the capability of MOVs and AOVs to stroke under design basis conditions. These programs were not established for SOVs.
Safety related POVs are required to be tested in accordance with the following codes/regulations:

- 10CFR 50.55a(f) – In-service Testing requirements
- ASME OM Code Subsection ISTC 51.50 SOVs
- ASME Section XI IMV 3400

These tests normally include the following:

Pre-service Test

Post Maintenance Test
Position Indication Verification
Stroke time Test

NUREG 1482, Rev 1 was issued in January 2005 to incorporate changes in the 2003 edition of Title 10, Part 50 of the Code of Federal Regulations. In NUREG 1482, Rev 1 Guidelines for In-service Testing at NPPs - Section 4.2.3 Stroke Time for SOVs the NRC stated: "*The NRC staff recommends that licensees should use advanced diagnostic techniques to obtain stroke-time measurements in accordance with the frequency provisions of the Code, and should also use those advanced techniques or maintenance programs to monitor the degradation of SOV performance. In addition, the staff recommends that the technique should evaluate actual disk movement and not only movement of the pilot valve or valve stem.*"

DIAGNOSTIC SYTEM REQUIREMENTS

To date, few NPPs have adopted the recommendations in NUREG 1482. This is due in part to the unavailability of diagnostic equipment and software specifically designed for SOV use.

To address the NRC staff recommendations in NUREG 1482 Rev. 1, the development or modification of existing valve diagnostic instrumentation and software is necessary. Some commercially available valve diagnostic systems have the capability to perform SOV testing with the addition of sensors and software modification.

The following capabilities for a SOV diagnostic system were identified as the minimum requirements to ensure accurate stroke time measurements and over all health of the SOV:

- Operating Coil Current/voltage
- Coil Pick Up/Drop Out Voltage
- Position Switch Settings indication
- Acoustic monitoring for disc/poppet position

An additional requirement included the ability of the diagnostic system to be used insitu with minimal impact on the SOV and plant operation.

Sensors capable of performing the above should be mounted to cause minimal intrusion on the SOV being tested or its associated wiring/conduit and be readily accessible for connection to the Signal Analysis Module/Data Acquisition Module (SAM/DAM). Sensor mounts may be either permanently or temporarily installed on the SOV. Fastening methodology of permanently installed mounts should be capable of handling the ambient and process fluid temperatures usually associated with process SOVs.

DIAGNOSTIC EQUIPMENT/SENSORS SELECTION/SETUP

Diagnostic testing was performed using a CRANE diagnostic system (Figure 7) on process and air pilot SOVs. The SAM/ DAM consisted of a Universal, Contact and Acoustic module and a laptop computer that contains the diagnostic software (SSW 5.3). Sensors included a hall effect probe for line current measurement, an Eddy Current sensor to measure coil magnetic field strength, and an accelerometer to verify valve stroke completion. These sensors are not considered esoteric and can be used on most valve diagnostic systems.

The SOV(s) were fitted with permanent sensor mounts. The Accelerometer (Acoustic Sensor) was mounted on the valve body and the Eddy Current (Puck) sensor mounted on the solenoid coil slightly above center. The sensor mounts were attached using high temperature/strength epoxy. A clamp on current probe (hall effect probe) was connected to the power leads of the coil. Leads were also connected from the four position indication switches to the Contacts Module of the DAM.

Sensor placement was not specifically optimized to receive the strongest signal. However, signal strength from both the accelerometer and Eddy current sensor were adequate. Figures 2, 4 and 6 show sensor placement on the SOVs tested. Insitu sensor placement is likely to be influenced by accessibility to the SOV and insulating protective material surrounding it. The SAM/DAM was configured to display four trace fields. One field is assigned for each sensor and the SOV position indication switches were attached to the contacts module.

TRACE ANALYSIS

Interpretation of diagnostic traces depends on SOV type (Direct, Pilot or Balanced), seat type (hard/soft), electrical source (AC, AC rectified or DC Voltage), process medium fluid pressure, and fail-safe position - normally closed (NC) or normally open (NO). Three types of SOVs were bench tested, a NC direct globe, NC direct gate and a NC piloted globe. Their internal configurations are identified in Figures 1, 3 and 5 respectively.

The open and close diagnostic trace for a NC 2-Way Direct Lift globe valve is identified in figures 8 and 9 respectively and a NC 2-Way Direct Lift Gate in figures 10 and 11 respectively. These valves are both direct acting and therefore their diagnostic trace characteristics are similar. The four trace fields are Eddy Current, Current, accelerometer and switch contact from top to bottom.

Powering the coil for either a Normally Open (NO) or NC coil can be characterized in four distinct steps, Initial current draw, plunger movement, increase to steady state and steady state (holding). Each phase is clearly identifiable on the current and eddy current trace.

NC DIRECT GLOBE/GATE VALVE

The open trace(s) figures 8 and 10 clearly identify initial excitation of the coil by the current draw and development of magnetic field flux in the coil. The beginning of this event is identified on the Eddy current and current trace fields by marker O0. Initially, current and magnetic field strength increases until the plunger begins to travel through its stroke, at this point there is an identifiable decrease in current draw (Back EMF produced as a result of the plunger traveling through the magnetic field) until the valve hits its stop. Once opened, the current increases until the solenoid coil becomes fully saturated. The slope and time characteristics of the trace can be used to establish baseline performance characteristics for the SOV.

Upon completion of its stroke the plunger hits its stop and an acoustic signal is generated. This signal verifies completion of the valve stroke as indicated by marker O5. This is also verified by the open position switch signal changing state on the OSW trace field. The open stroke time is determined by measuring the elapsed time between Marker O0 and O5.

Signal conditioning (enveloping and filtering) of the eddy current trace was not performed but is necessary to fully utilize this non invasive monitoring method of measurement. Although not as apparent as the current trace, the change in magnitude and slope is readily apparent.

The close trace clearly identifies initial cessation of current to the coil and collapse of the magnetic field in the coil. The beginning of this event is identified on the Eddy current and current trace fields by marker C0. The solenoid plunger does not immediately stroke upon removal of power to the SOV. The magnetic force holding the plunger in place is maintained until the return spring force overcomes the collapsing magnetic field force. When the solenoid plunger travels through its stroke and impacts its seat an acoustic signal is generated which signifies completion of the valve stroke as indicated by marker C6. The strength of the signal varies based on seat type (hard or soft). The completion of stroke is also verified by the close position switch signal changing state on the CSW trace field. The close stroke time is determined by measuring the elapsed time between Marker C0 and C6.

PILOT GLOBE VALVE

The open and close diagnostic trace for a Normally Closed (NC) 2-Way Direct Lift Pilot Assist globe valve is identified in figure 13 and 14 respectively. The diagnostic technique for identifying stroke time is the same as the direct gate/globe valves discussed earlier with minor differences in the acoustic signature since movement of both the pilot and main disc may be identified.

PERFORMANCE MONITORING

SOV performance characteristics can be used to establish base line performance, performance monitoring and trending, and eventually as a diagnostic tool to predict SOV failure. Performance monitoring and trending could be used to predict the following failure modes:

- Coil Degradation/Failure
- Rectifier Failure/ "Transorb" Surge Suppressant failure
- Return/Plunger Spring Failure
- Gummed/sticking Components
- Jamming/friction due to debris (e.g. magnetite, erosion/corrosion by-products)
- Seat Leakage
- Faulty Position Indication Switch and or Setting

When comparing traces it is imperative that process conditions are considered to account for differences in process medium characteristics e.g., pressure, viscosity, specific gravity.

Significant differences can exist between a static and dynamic trace. This is especially apparent on piloted SOVs since pressure equalization must occur before the main poppet strokes. This is a result of the time delay between the pilot and main disc opening since upstream and downstream pressure is being equalized. This must be considered if traces are to be used for trending or for diagnostic evaluation. Additionally, the same process medium that is present during insitu testing should be used in performing initial baseline test to ensure proper comparison of traces.

CONCLUSION

Testing of the SOVs using commercially available diagnostic equipment will allow the user to obtain accurate stroke time measurements. Testing performed to date is a first step in developing diagnostic tools for trouble shooting and preventive maintenance. Further testing and development of the software is required to fully utilize the diagnostic capability and to enhance the interpretation of the diagnostic trace.

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Operating Experience Feedback Report – Performance of Safety related Power Operated Valves Solenoid Operated Valve Problems, NUREG-1275

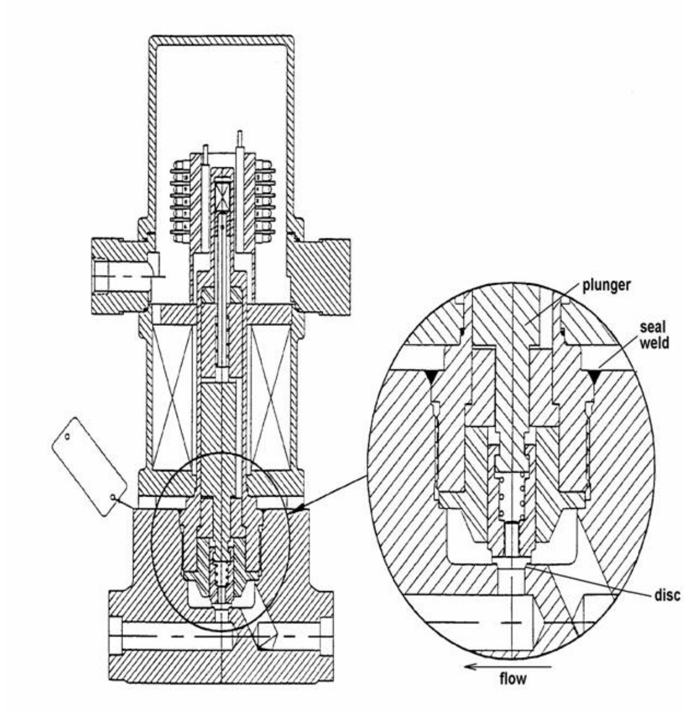
NRC, Code of Federal Regulations, Title 10, Part 50

ASME OM Code Section ISTC 51.50

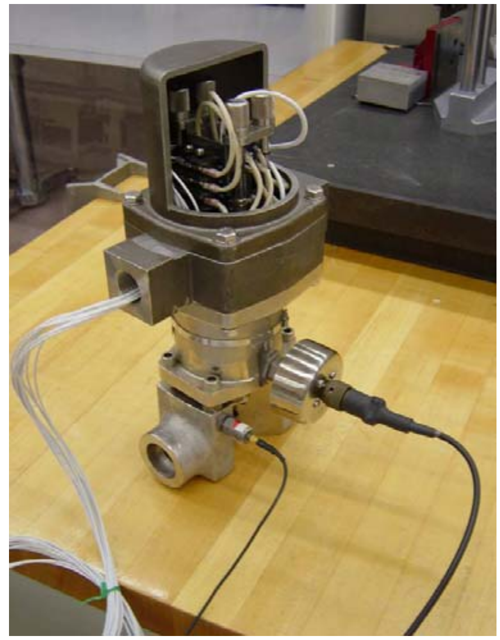
ASME B&PV Code, Section XI, IMV3400

Carow, James, Non Intrusive Stroke Timing and Diagnostic Testing of Solenoid Operated Valves, NUREG/CP-0137

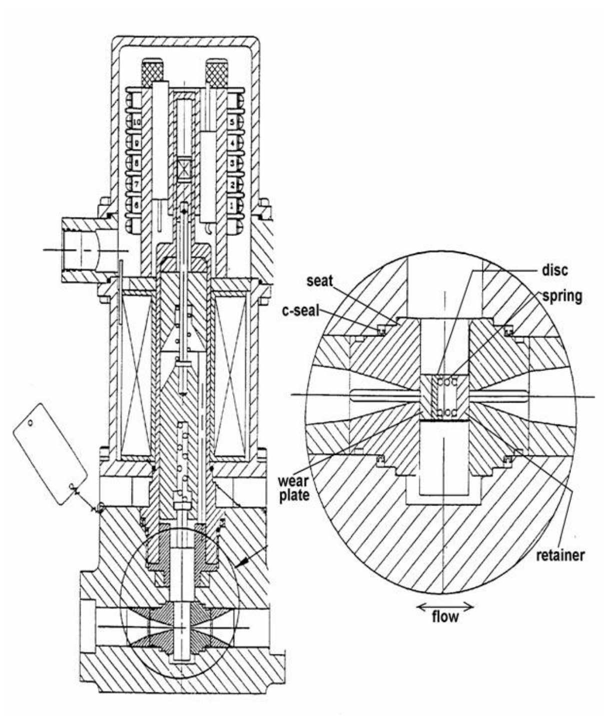
Direct Lift Globe (Figure 1)



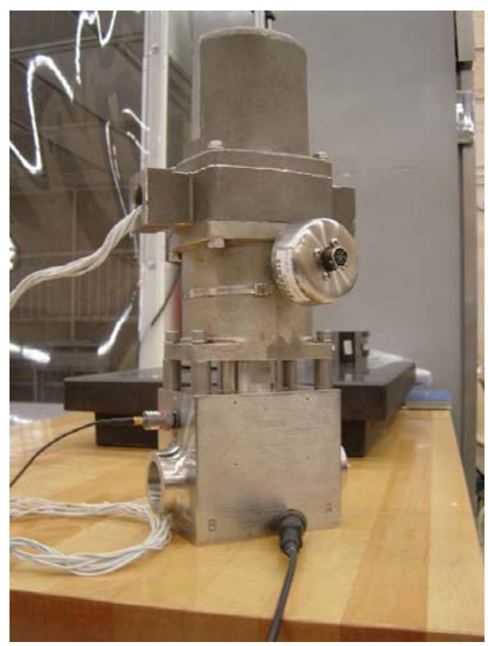
Globe Sensor Placement (Figure 2)



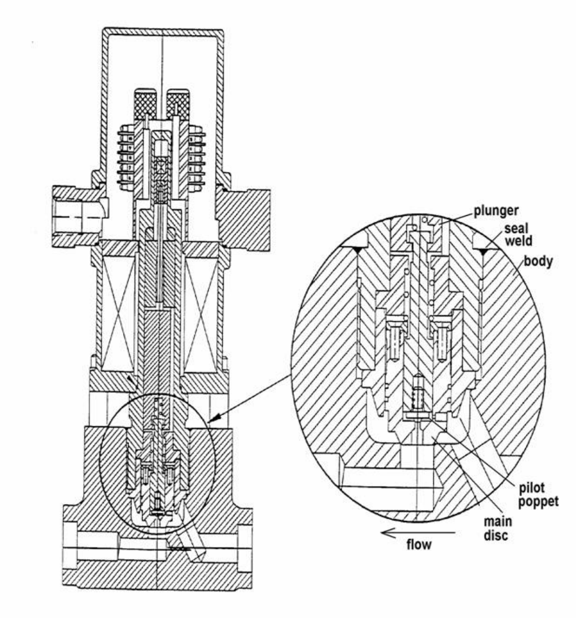
Direct Lift Gate (Figure 3)



Gate Sensor Placement (Figure 4)

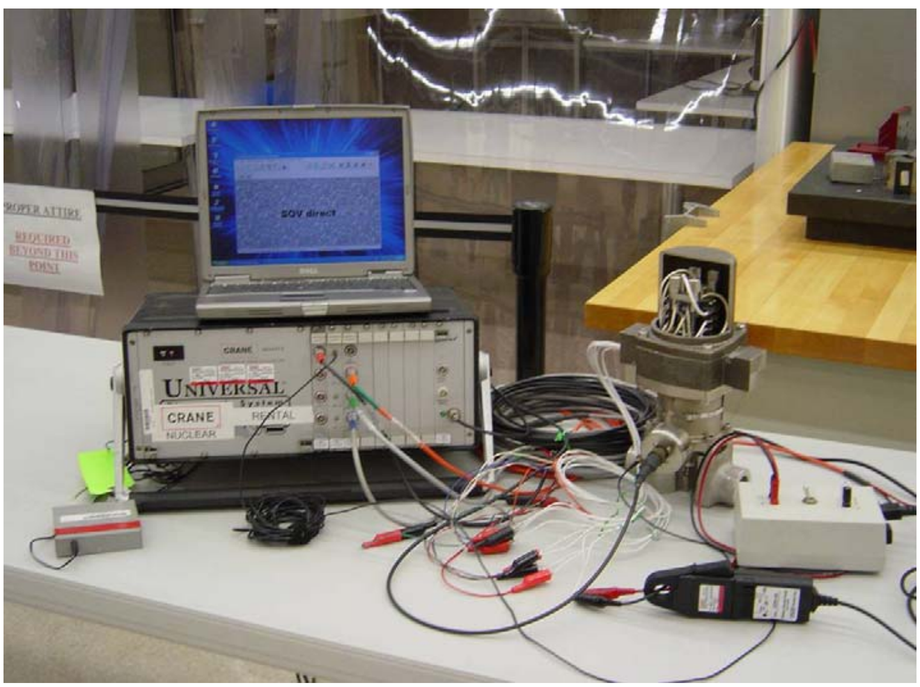


Direct Pilot Assist (Figure 5)

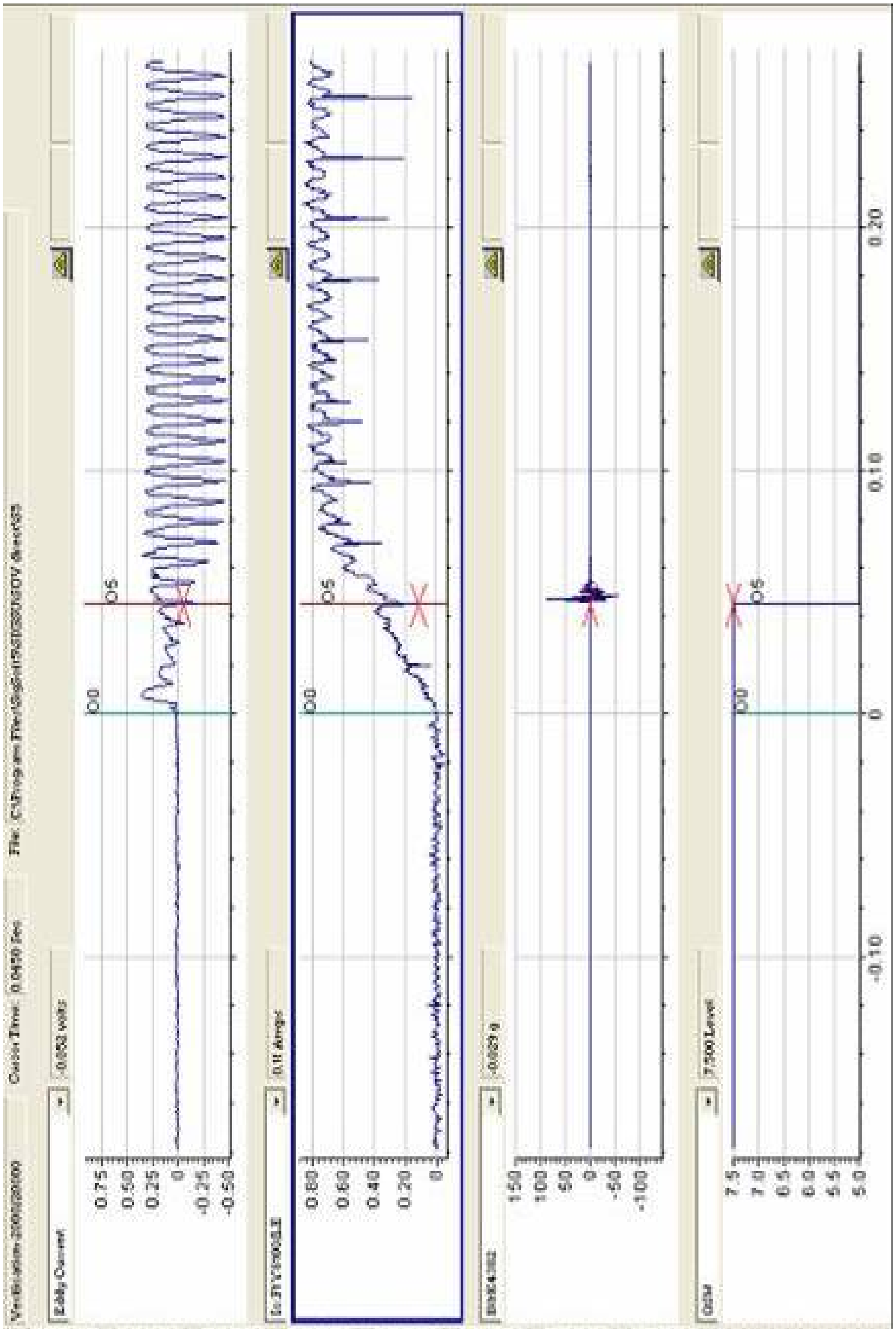


Sensor Placement (Figure 6)

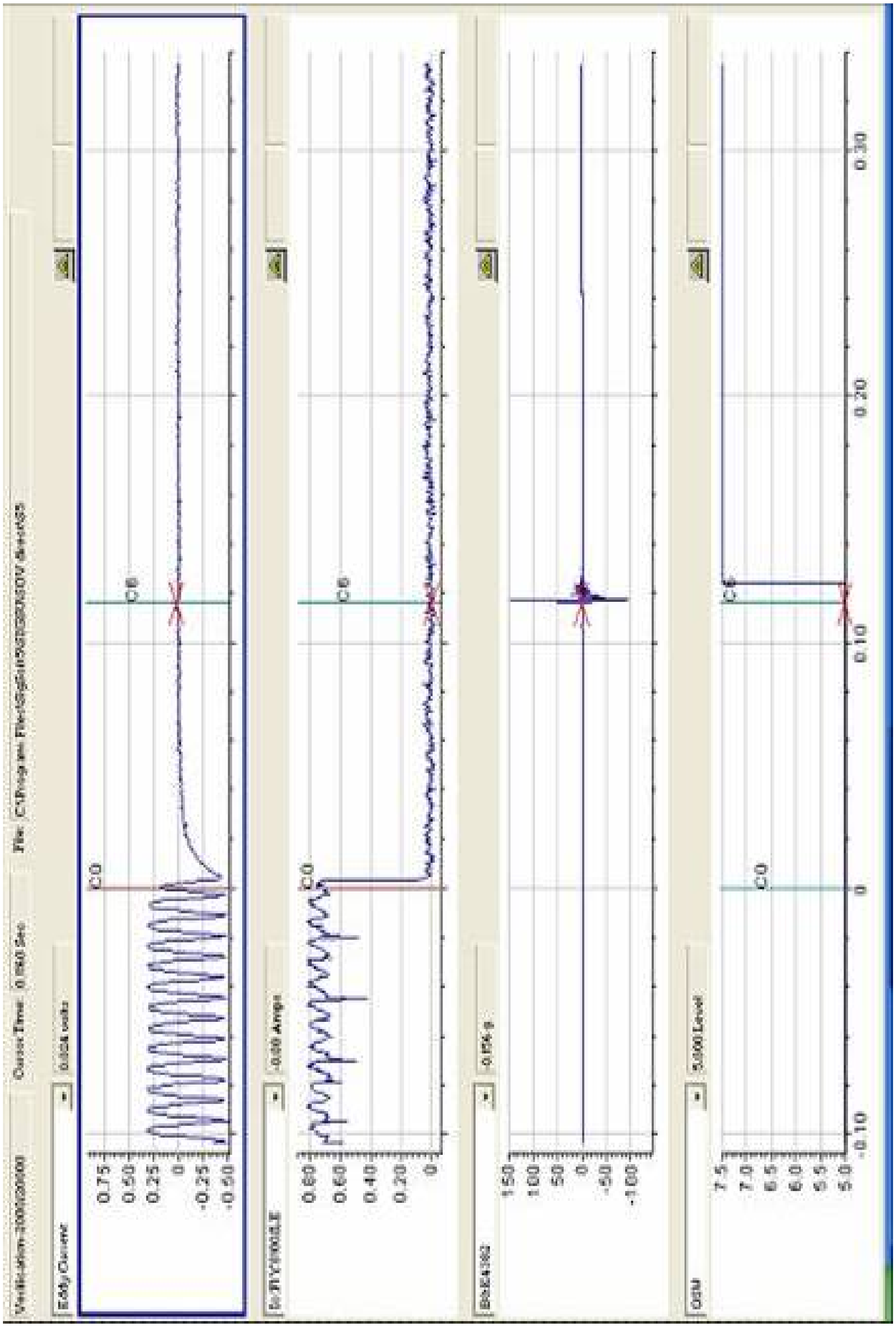
SAM/ DAM (Figure 7)



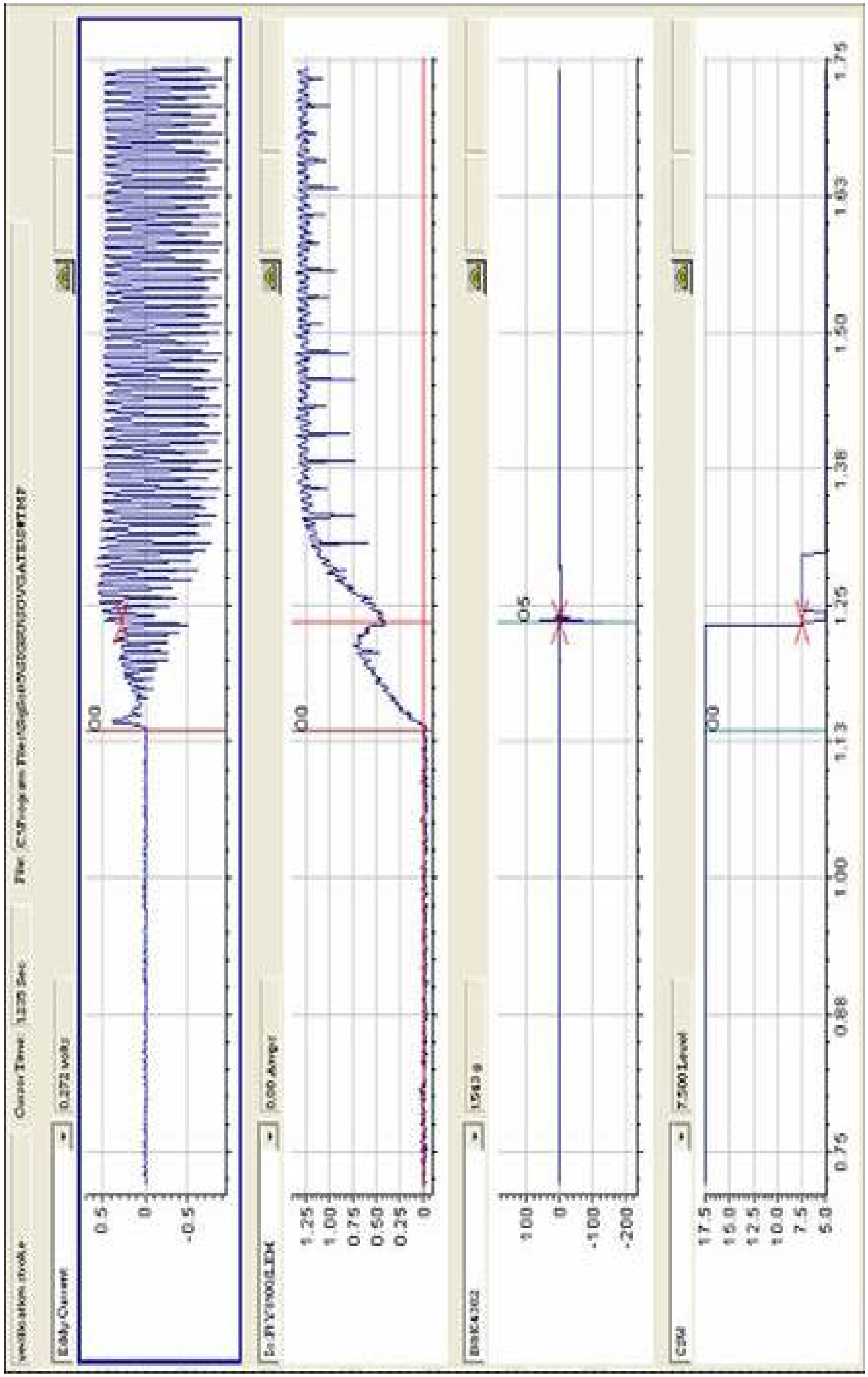
Direct Lift Globe – Open trace (Figure 8)



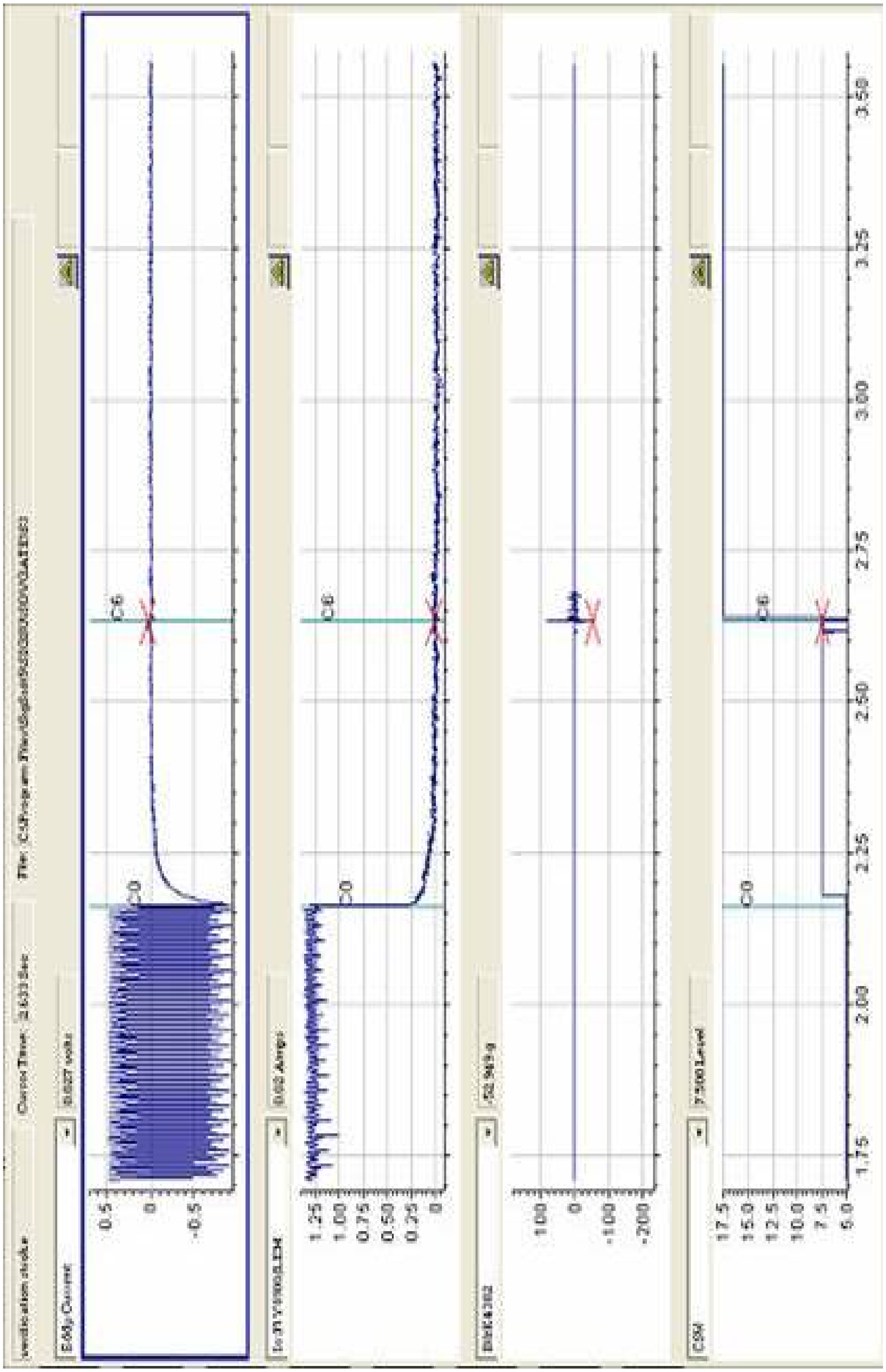
Direct Lift Globe – Close trace (Figure 9)



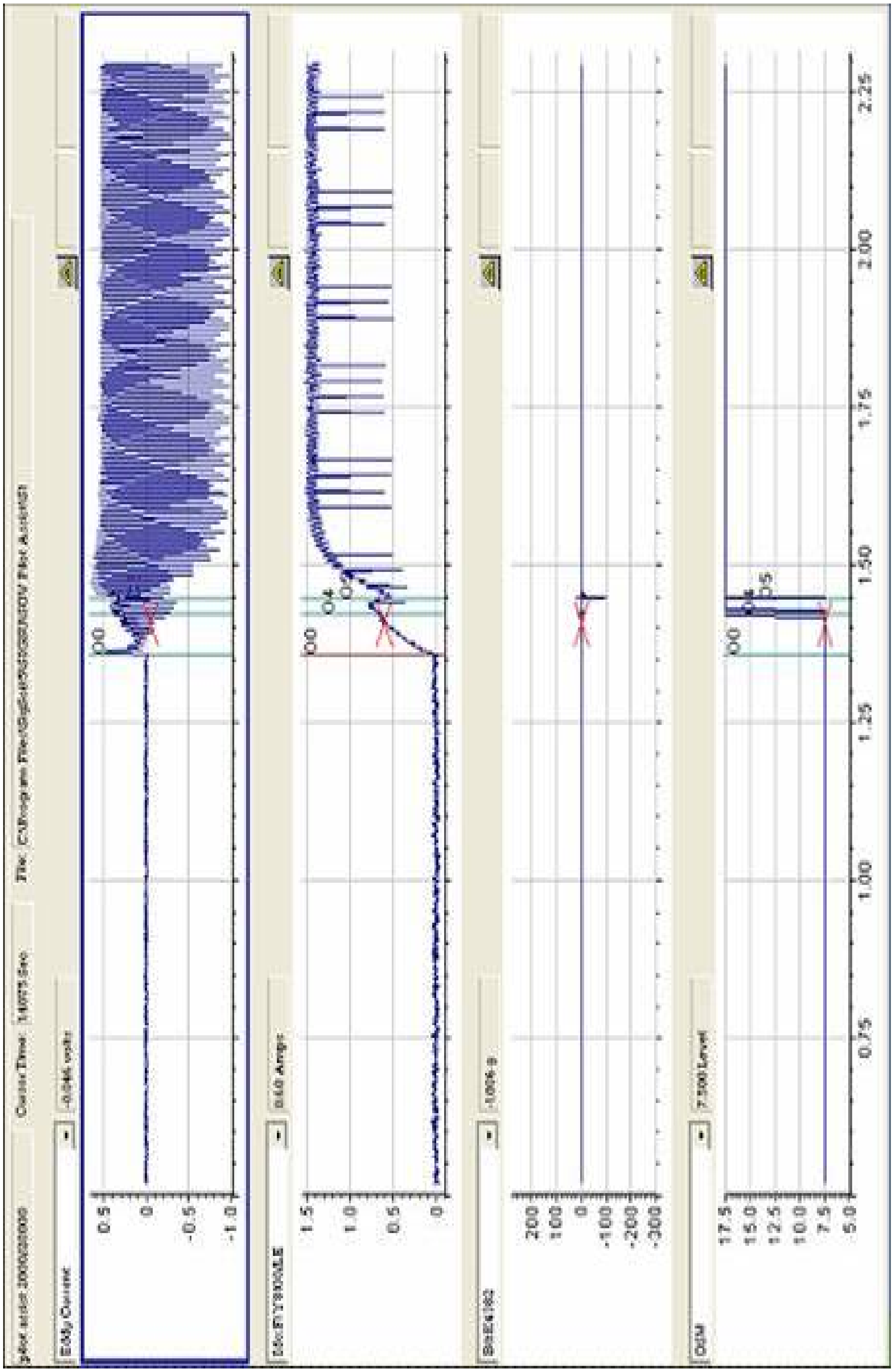
Direct Lift Gate – Open trace (Figure 10)



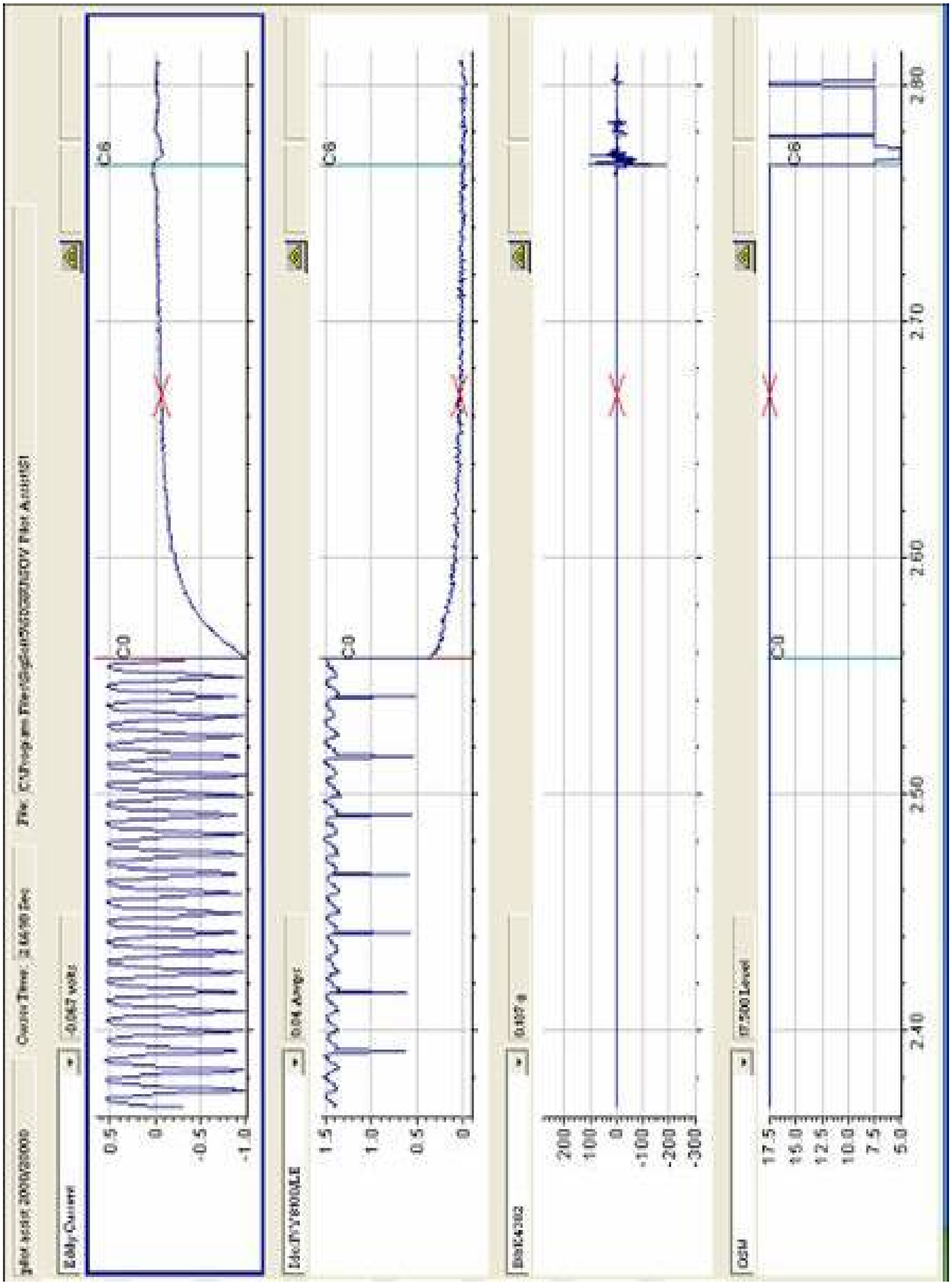
Direct Lift Gate – Close trace (Figure 11)



Direct Lift Pilot Globe – Open trace (Figure 12)



Direct Lift Pilot Globe – Close trace (Figure 13)



Fatigue Analysis of Electric Actuator Torque Train Components -
Avoid Modifications and Maintain Margin

By
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Abstract

Actuator thrust rating extension was emphasized and successfully implemented during execution of NRC Generic Letter 89-10 motor-operated valve (MOV) programs, primarily to address inertial overshoot after control switch trip. During testing to justify the higher thrust ratings, a variety of torque train components failed. Torque train fatigue predictions are much more involved than thrust fatigue predictions. This is because during a single valve stroke (defined as open to close to open, or close to open to close) the actuator experiences just one thrust cycle, but the torque train components are subject to a fatigue cycle during each worm revolution that occurs at stress levels above the material endurance limit.

While thrust fatigue is proportional to the number of valve strokes and the seating and unseating thrust magnitudes, torque fatigue life is proportional to the actuator torque load time history. To successfully model the torque fatigue life, the actuator load history is expressed as a series of linearly increasing ramps from zero torque to the maximum torque magnitude over a certain time period and linear constant maximum torque magnitude dwells, again over a certain time period. Analytical formulas and stress concentration factors are used to calculate stresses at critical locations on the worm and the worm shaft. These stresses are combined to form a stress tensor, which is used with the von-Mises-Henkley failure theory to enter a modified stress-cycle curve (extended to include both high and low cycle fatigue). Modified Goodman Criteria are used to model mean stress effects, and Miner's Rule is used to compute cumulative damage. Since the expected life is based on cycles to failure, a safety factor is applied for design purposes. The safety factor generally used in these analyses is 5.24, which is based on ASME Section III.

MOV calculation revisions to support the industry Joint Owner's Group (JOG) MOV periodic verification (PV) program may require either extended actuator torque ratings or actuator replacement to achieve margin goals. Evaluation of extended actuator torque requirements beyond the current ratings can provide a cost-effective alternate to actuator replacement in many situations. Plant examples are provided to show the range of results that can be expected from torque train evaluations and the load history features that lead to more favorable results.

Introduction

During implementation of the U.S. Nuclear Regulatory Commission (NRC) IE Bulletin 85-03 [1][6] recommendations, utilities performed diagnostic testing of safety related motor-operated valves (MOVs) to ensure that the torque switches and limit switches in the actuators are properly set to allow these MOVs to perform their safety-related opening and closing functions. It was discovered that during plant operation, a significant population of MOVs was being subjected to cyclic thrust overloads that exceeded manufacturer's ratings. To address utility-specific overload conditions, Kalsi Engineering performed evaluations for a number of plants using fracture mechanics analysis, fatigue analysis, and cyclic overload testing of actuator components to justify continued operation.

Under the sponsorship of several nuclear power plant utilities, a *generic* test program was conducted to determine the capability of Limitorque actuators to withstand inertia related overloads above rated

[6] Numbers in brackets [ ] are reference numbers

capacity, and to qualify them for higher thrust ratings. Qualification testing was performed by Kalsi Engineering, Inc. (KEI) on four Limitorque SMB actuators (SMB-000, SMB-00, SMB-0, and SMB-1) during Phase I of the program. All of the thrust related components of the actuators successfully completed the overload test goals. However, certain torque related components (i.e. worm, worm gear, worm shaft, and worm shaft bushing) in some of the actuators required interim replacement due to fatigue damage or excessive wear. The results of the testing were reported in a proprietary KEI document [2]. The report included a recommendation for further testing of torque related components and for a detailed fatigue analysis so that a generic approach is developed to extend test results to actual Motor Operated Valves (MOV's) including gear train combinations used in actuators that are not tested. A follow-up program, designated Phase II, was initiated to retest the SMB-000 actuator and to extend the testing to an SMB-2 actuator. During the Phase II program the development of analytical capability to utilize SMB test results for the quantification of torque limitations for the SMB-000 through SMB-2 actuators, as well as for the SB/SBD-000 through SB/SBD-2 actuators, was undertaken [3-9].

The Phase II program resulted in the development of a software package [10-12] to determine the torque and fatigue life of the actuators within the test scope. This software package was qualified for use on nuclear safety related projects by following the procedures of KEI's quality assurance program.

Test Details

Test Fixtures

Five sizes of Limitorque actuators (SMB-000, -00, -0, -1, -2) were tested using several different fixtures designed to perform cyclic testing at 200% of the thrust rating, stall testing, and seismic testing [2].

Figures 1 and 2 show the details of the test fixtures used for testing the SMB-000 through SMB-2 actuators. The fixtures allowed the MOV stiffness to be simulated by using appropriate Belleville spring stacks in both the opening and closing stroke directions. The stiffness for each actuator size was based upon a survey of representative MOV stiffness levels obtained from in-situ static thrust traces provided by program participants for gate and globe valves.

The simulation of the MOV stiffness in the fixture is not important for the life of thrust components but it has a significant effect on the life of torsional components. This is because the thrust train components are subjected to only one peak stress cycle during each MOV cycle regardless of the stiffness. However, as shown in Figure 3, the torque train components (e.g. worm and worm shaft) are subjected to multiple stress cycles, and the MOV/actuator stiffness does affect the number of alternating load cycles (and cumulative fatigue damage) for the actuator *torque train components* during the load ramp.

Test Plan

Briefly, the test plan consisted of the following major steps:

1. 2,000 Cycles of Overload Testing at 200% of Rated Thrust
   - Disassemble and perform initial inspection of all actuator components
   - Complete 500 load cycles, disassemble and inspect
   - Complete 2,000 load cycles, disassemble and inspect
2. Stall Tests
   - Perform 5 load cycles at full motor stall in both open and close directions
   - Disassemble and inspect
3. Seismic Qualification Tests
   - Perform a matrix of seismic qualification tests
   - Disassembly and inspect
4. Extended Overload Testing to 4,000 Cycles
   - Complete 2,500 load cycles, disassemble and inspect
   - Complete 4,000 load cycles, disassemble and inspect

During testing, the actuators were completely disassembled for visual examination, dye penetrant examination and dimensional inspection (if necessary) of critical components at several intermediate steps identified above. The objective of these intermediate inspections was to detect evidence of any crack initiation, monitor crack propagation and to look for any wear-related degradation of internal components, which could provide the basis for developing maintenance/replacement interval recommendations.

Test Results

Thrust Train Components

The allowable number of design cycles, and the corresponding allowable design loads for actual plant applications was based upon applying appropriate margins or factors to the 200% of the thrust loads and 4,000 cycles used in the Phase I qualification testing. Even though there are no requirements to apply *any* margin according to the applicable IEEE-382/344 qualification standards available at the time of the testing [13, 14 and 15], it was considered prudent to voluntarily apply a recognized approach for margins used for other equipment e.g. mechanical/structural components, pressure vessels, hydraulic machinery etc. After reviewing various approaches used in different industries, it was decided to use the ASME Section III, Division 1, Appendix II approach which provides factors for allowable number of design cycles and design loads based upon qualification test cycles and loads using one or more test specimens [16]. Based on this approach, the Limitorque SMB-000, -00, -0 and –1 actuators were qualified for extended thrust beyond their current ratings for 2,000 cycles.

Torque Train Components

During thrust overload qualification testing of 4,000 cycles, torsional failures (e.g. as shown in Figure 4), as well as excessive wear of certain components in some actuator sizes, were encountered. A summary of the torque train component failure encountered during Phase I and II is given in Table 1. It should be noted that the loading on the torque train components was significantly above the torque ratings on two of the actuators. The actual levels of loading on the torque related components was different for each actuator, and it ranged from 96% to as high as 141% of the rated torque based on actual gear ratios and motor sizes that were used in the test actuators (for the same Limitorque actuator size, there are different torque ratings for different gear ratios). Based upon technical recommendations by Kalsi Engineering and consensus of the sponsoring utilities, it was decided to continue testing to achieve the thrust qualification goal by replacing the prematurely failed non-thrust train components, and to address the torque train components fatigue life issue based on a rigorous first principles approach and develop appropriate recommendations under Phase II of the program.

Wear of Torque Train Components - Bushing Wear and Impact on Fatigue

Replacement of components due to wear was necessary to achieve the thrust test goal. Other than moderate wear of the SMB-000 worm, only excessive wear of the worm shaft bushing was observed. Worm shaft bushings are used in SMB-0, -1, and –2 size actuators. The bushing is located in the motor end flange and supports the end of the worm shaft to support the lateral forces generated between the motor pinion and the worm shaft gear. Excessive wear of the bushing allowed the worm shaft to slap inside the bushing and created high fluctuating stresses in the area of the retaining ring groove. After repeated failures of the worm shaft, the bushing was replaced at periodic intervals. The periodic replacement eliminated failure of the worm shaft.

Model Development

The loading of the actuator torsional components in MOV applications is a very complex phenomenon. Load magnitudes depend on the type of valve used in the application, packing friction, fluid dynamic forces, the closing and opening stem forces, stem screw lead dimension, worm gear ratio, worm/worm gear friction coefficient, and the limit switch helix drive gear forces. The number of cycles that the torsional components experience depends on the number of valve operations, stem loads, valve stiffness, stem screw lead, and worm gear ratio. Indirectly, it may even be a function of the spring constant of the torque switch spring stack in that the contact point on the worm is displaced by a small amount as worm axial force is increased and the spring stack compressed. This displacement has the tendency to distribute the load over the worm tooth and slightly change the cycle characteristics of any loaded tooth location.

Axial load and torque applied to the stem was measured simultaneously by a strain gage load/torque cell during Phase 1 and Phase II testing. Typical measurements of stem thrust and torque profiles imposed on the actuators are shown in Figure 5. The profiles have well-defined segments with the following characteristics. As the actuator moves the valve stem from some intermediate position A to the valve closed position C, there is a time interval A-B of free travel (zero stem thrust and torque) before the stem thrust and torque start to increase linearly to their maximum magnitude at C. After a short pause in the fully closed position with constant non-zero thrust and torque, time interval C-D, the actuator torque is reversed to start unloading the stem and stem nut thread to initiate valve opening. Since the torque required unloading the power screw/worm gear/worm system is less than the torque that is required to load it, the reversed torque magnitude is less than the previous maximum required for closing. This torque reversal is very abrupt and is followed by a linear decrease in the stem force and torque (and displacement) from D to F, again passing through a small time interval E-F of free travel. As the stem reaches the valve full open position, the stem force and torque again begin to linearly increase to their maximum values which are equal in magnitude but opposite in direction to their values between C and D. After a short pause at the fully open position, the torque again reverses to the smaller breakaway torque, and stem and stem nut threads begin to unload and the stem begins its travel to the fully closed position. This sequence of events forms the basic actuator load cycle that was repeated continuously to a total of 4,000 cycles during Phase 1 testing. Since the pause times do not affect fatigue life, the load cycle can best be represented by neglecting the periods of zero or constant loads as shown in Figure 6. The torque loading can be seen to consist of four approximately linear ramp increases to local peaks.

Since the loading cycle definition is totally dependent on the specific application, the fatigue analysis program will require a detailed input of valve stem loading, as well as of actuator details including stem nut, worm gear, worm, worm shaft, and limit switch helical drive gear dimensions.

A detailed description of the model was provided previously and will not be repeated here [17].

The computation of fatigue life of the torsional components of the Limitorque actuators requires determination of the applied loads and moments, analysis of the generated stresses, and application of a failure theory. The applied loads and moments are computed from a mechanical model of the affected components and a predetermined set of valve stem thrust and/or torque levels. The stress components generated by the applied loads can then be evaluated by the use of standard formulas for beam stresses, Hertzian contact stresses, and specialized formulas for gear tooth stresses. The failure theory employed, the distortion energy theory, will utilize the computed stresses in conjunction with material S-N fatigue curves and Miner's rule to calculate cumulative damage and fatigue life.

The critical torsional components of Limitorque actuators have been identified during the Phase I and Phase II test programs to be the worm, and the worm shaft at two locations - at the worm-shaft interior contact point, and at the limit switch helical drive gear. Thus, the mechanical model must describe the worm/shaft assembly in sufficient detail to permit the computation of the external as well as internal loads with reasonable accuracy. Two different mechanical models are required since the actuators of interest here utilize two distinct worm shaft configurations. The SMB-000/00 actuators have a cantilevered shaft design at the motor pinion/drive gear, while the shafts of the SMB-0/1/2 actuators are supported by an oilite bushing mounted in the housing. These differences in the configuration negate the use of geometrical similarity throughout the whole range of actuators. These models are shown in Figures 7 and 8 for the SMB-000/00 and SMB/0/1/2 actuators, respectively. The models show the significant forces and length dimensions for the two worm/shaft assemblies consisting of the worm of length l_W and shaft of length l_{sh}. The forces acting on this assembly are represented by the solid arrows. The externally applied forces act at the contact points of the worm/worm gear, designated F_W, and motor pinion/drive gear, designated F_d. The coordinate system is oriented as shown with the worm contact point on the y-axis, and the pinion contact point offset by the angle θ_d. The bearing reaction forces are designated B_S and B_W for the worm and shaft, respectively. In the case of the model in Figure 8, the additional reaction force at the oilite bushing is designated F_b.

In this mechanical model the forces designated by F_W and F_d are external forces considered to be known variables. Their magnitudes are derivable directly from the given valve stem loading criteria and the worm gear and drive gear parameters. The reaction forces F_d, B_S, and B_W, as well as F_1 and F_2, which represent the worm/shaft contact forces, are internal forces. They are solved for by using static force and moment balances, and elastic deflections where necessary. In the solution for the bearing reactions, the assembly force diagrams of Figures 7 and 8 are used where the contact forces cancel each other. The contact forces are solved by using separate force diagrams for the worm bodies, Figures 7 and 8, where the contact forces become the unknowns.

Although the worm/worm gear contact force is represented in the force diagrams by a single arrow, actually it is distributed over a number of meshing teeth. In the present model, it is assumed to be distributed over three teeth with the central tooth taking some fraction of the total load and the other two sharing the remaining load equally. This partition of load is not shown in the above diagrams because it does not affect the solution for the reactive forces. However, it will have to be taken into account in the computation of bending moments at the critically stressed locations.

Model Validation

The torsional fatigue model was validated against Phase I and II test data as shown in Table 1 and graphically in Figure 9.

Application

The computer computational algorithm performs the following calculations:
Static stresses for worm and worm shaft.
Mean dynamic stresses for worm and worm shaft.
Alternating dynamic stresses for worm and worm shaft.
Fatigue life for worm and worm shaft for each load condition.
Total cumulative fatigue damage due to multiple strokes and multiple load cases using Miner’s Rule.
Remaining life based on torque.

As mentioned previously, the computer algorithm uses analytical formulas and stress concentration factors to calculate stresses at critical locations on the worm and the worm shaft. These stresses are combined to form a stress tensor, which is used with the von-Mises-Henkley failure theory to enter a modified stress-cycle curve (extended to include both high and low cycle fatigue). Modified Goodman Criteria are used to model mean stress effects, and Miner's Rule is used to compute cumulative damage.

Load Input for Torque Model

The fatigue life software is capable of modeling the load history illustrated in Figure 10. The load history is represented by four linearly increasing ramps and four constant load dwells. The load history is valid for left-handed stem threads. The fatigue calculations for the torque carrying components is based on torque applied to the worm and worm shaft, thus it is extremely important that the stem nut rotation be correctly selected.

1. **Ramp 1** represents an increase in compressive (- thrust) closing load and a clockwise (+ torque) rotation of the stem nut. Ramp 1 ends at the beginning of the constant load. For a gate or globe valve this would represent disc wedging or the stem load increase while closing under differential pressure (DP) and flow as the flow is being reduced and the DP is increasing.

2. **Dwell 1** represents the travel under essentially constant closing load and extends until the travel ceases. Dwell 1 is a clockwise rotation of the stem nut. For a gate valve this would represent the relatively constant stem load that occurs between flow isolation and wedging.

3. **Dwell 2** represents the dwell that occurs under constant load when the stem nut rotation is reversed and is turning counterclockwise. In most cases Dwell 2 is very small. The thrust load in Dwell 2 is compressive in nature since the elastic strain has not yet been released.

4. **Ramp 2** represents the part of the stroke that relieves the elastic compressive (-thrust) strain that was generated during the closing stroke. During this part of the stroke the stem nut counterclockwise (- torque) rotation is opposite the closing stroke and the conversion of thrust to torque is based on a stem factor calculated for thread lowering (unloading). This ramp time for this segment is typically very small unless the MOV is very elastic, as might be the case when an SB or SBD type actuator is used.
5. **Ramp 3** represents the buildup of tensile (+ thrust) load during the opening stroke. Ramp 3 ends at the unwedging point of a gate valve. Ramp 3 is highly dependent on the stiffness of the MOV. During Ramp 3 the stem nut is rotating counterclockwise (- torque).
6. **Dwell 3** represents the travel under constant load prior to start of pressure decay. Stem nut rotation is counterclockwise.
7. **Dwell 4** represents the dwell that occurs under constant load when the stem nut rotation is reversed from opening direction to closing direction. In most cases Dwell 4 is very small. The stem nut rotation is clockwise.
8. **Ramp 4** represents the part of the stroke that relieves the elastic strain that was generated during the opening stroke. During this part of the stroke the actuator rotation is opposite the opening stroke. This ramp time for this segment is typically very small unless the MOV is very elastic, as might be the case when an SBD type actuator is used. Stem nut rotation is clockwise (+ torque)

Options for defining the load history

1. One set of stem thrust, ramp time, and dwell time
2. Four sets of stem thrust, ramp time, and dwell time
3. One set of stem torque, ramp time, and dwell time
4. Four sets of stem torque, ramp time, and dwell time

The most direct input of data is to use (option 4) four distinct torque, ramp time, and dwell time values. When the other options are used then the computer program uses internal algorithms for determining the other three sets of values and performs the fatigue evaluation based on Option 4.

Stem Factor

Stem factor is used to convert thrust to torque. Stem factor value depends on whether the actuator is providing a driving force (raising the load) or a restraining force (lowering the load). The equation for lowering the load may become negative when the threads are "non–locking". Whether the threads are "locking" or "non-locking is dependent upon the stem thread geometry. The stem factor for lowering the load should use the same friction coefficient for raising the load.

The relationship between torque and thrust for raising the load is:

$$\text{Torque}_R = \frac{F\, d_m}{24} \frac{(\text{lead} + \mu\, \pi\, d_m \sec(\alpha))}{(\pi d_m - \mu\, \text{lead} \sec(\alpha))} \qquad (1)$$

The relationship between torque and thrust for lowering the load is:

$$\text{Torque}_L = \frac{F\, d_m}{24} \frac{(\mu\, \pi\, d_m \sec(\alpha) - \text{lead})}{(\pi d_m + \mu\, \text{lead} \sec(\alpha))} \qquad (2)$$

In either case the stem factor (SF) is the relationship between thrust and torque

SF=Torque / Thrust (3)

Load History Definition

A typical static load of a gate valve is shown Figure 11. The static trace can be modeled using 4 ramp profiles as shown in Figure 12, but the dynamic trace requires the use multiple ramps and dwell load profiles as shown in Figures 13 and 14.

Cumulative Fatigue Damage

Cumulative fatigue damage due to multiple strokes and multiple load cases is determined using Miner's Rule. The cumulative integral is calculated using the following equation where the cumulative fatigue damage integral is limited to 1.0. The damage due to each applied load is determined ***after*** the design margin has been applied. That is Nn represents the design life and not the failure life and nn represents the actual applied strokes. .

$$\frac{n_1}{N_1} + \frac{n_2}{N_2} + ... + \frac{n_n}{N_n} = \text{Cumulative fatigue damage} \qquad (4)$$

Where

n_n = applied cycles under load case #n
N_n = allowable design cycles under load case #n

Remaining Design Life

The remaining design life due to multiple strokes and multiple load cases is determined using the results of the cumulative fatigue damage (sum of individual usage factors). The remaining design life is calculated in terms of load case 1 only as given below.

$$n_1 = N_1 \text{ (1- cumulative fatigue damage)} \qquad (5)$$

Equivalent Cycles

The equivalent cycles, x, are only calculated when multiple load cases are used. This calculation equates the damage due to loads other than load case 1 to the number of allowable cycles under load case 1

$$\frac{x}{N_1} = \frac{n_n}{N_n}$$

Solving for “x” results in the following equation:

$$x = \frac{N_1\ n_n}{N_n} \qquad (6)$$

Methodology

It is important to determine a conservative estimate of the actuator load history. This includes past and expected future static and DP stroke profiles. It is often sufficient to assume that the most recent static test data would bound any past static test load profiles, and that past DP test data can be used with adjusted seat and guide friction factors to determine load profiles for future DP tests. Figures 15 and 16 show one possible conservative extrapolation technique

There will be four analyses performed for each actuator:

- Analysis 1 – Past Static Strokes
- Analysis 2 – Future Static Stokes (at the new thrust and torque goals)
- Anslysis 3 – Past Dynamic Strokes
- Analysis 4 – Future Dynamic Strokes (at the new thrust and torque goals)

The calculated design life (N) will be determined for Analyses 1, 3, and 4 along with the corresponding Usage Factors based on the estimated number of strokes (n). Miner’s rule will then be used to determine the remaining allowable life for Analysis 2.

Example 1
General Input Data

Worm Mtl	Shaft Mtl	Valve Type	Valve Size mm [in]	Class	Actuator	OAR	Moto r RPM	Thrust Rating KN [lbf]	Torque Rating N-m [ft-lb]
4320	4140	Gate	150 [3]	1500	SB-00-15	59.4	1800	62.28 [14000]	339 [250]

Analysis Goals

Future Torque Goal N-m [ft-lbs]	Future Thrust Goal KN [lbs]	Past Static Strokes	Future Static Strokes	Past Dynamic Strokes	Future Dynamic Strokes
373 [275]	87.2 [19600]	123	298	100	100

Past static strokes were all below the current torque rating, therefore 2,000 valve stroke cycles are permitted by the manufacturer:

Past Max Torque N-m [ft-lbs]	Past Static Strokes (n)	Design Life Strokes (N)	Fatigue Usage Factor (n/N)
295.6 [218]	123	2000	0.0615

Also, the past DP strokes were also below the current torque rating, therefore, 2,000 valve stroke cycles are permitted by the manufacturer:

Past Max Torque N-m [ft-lbs]	Past Dynamic Strokes (n)	Design Life Strokes (N)	Usage Factor (n/N)
288.8 [213]	100	2000	0.0500

Future static strokes are to occur above the 100% torque rating. Based on the new requirements, the following calculation input profiles were used:

Torque1 N-m [ft-lbs]	Ramp1 (sec)	Torque3 N-m [ft-lbs]	Ramp3 (sec)
373 [275]	1.702	-176 [-129.9]	0.191

Likewise, future dynamic strokes are projected to occur above the 100% torque rating.

Based on the new requirements, the following calculation input profiles were used:

Close Direction

Torque 1a N-m [ft-lbs]	Torque 1b N-m [ft-lbs]	Ramp1a (sec)	Dwell1a (sec)	Ramp1b (sec)
288.3 [212.7]	373 [275]	1.369	0.583	1.702

Open Direction

Torque 3a N-m [ft-lbs]	Torque 3b N-m [ft-lbs]	Ramp3a (sec)	Dwell3b (sec)	Ramp3b (sec)
-225 [-166]	-225 [-166]	0.090	1.223	0.495

The usage factor for future dynamic strokes is shown below:

Design Valve Strokes Based on Worm Life (N)	Design Valve Strokes Based on Shaft Life	Estimated Future Required Valve Strokes (n)	Usage Factor (n/N)
104	2000	100	0.96

The remaining allowed future static strokes can no be calculated using Miner's Rule per Equation 5:

Future static strokes = N_{future_static} x (1 – Σ(UsageFactors))

However, for this case the past static and dynamic, and future dynamic strokes would consume all of the fatigue life. One option at this point involve reexamination of input requirements for the future dynamic valve stroke conditions to see if the number of required future valve strokes, the torque magnitudes and/or durations can be reduced. A second option is to replace torque train components after a certain number of dynamic valve strokes.

Example 2

General Input Data

Worm Mtl	Shaft Mtl	Valve Type	Valve Size mm [in]	Class	Actuator	OAR	Motor RPM	Thrust Rating KN [lbf]	Torque Rating N-m [ft-lbs]
4620	4140	Gate	254 [10]	1500	SB-1-25	171.6	1800	200 [45,000]	1152 [850]

Analysis Goals

Future Torque Goal N-m [ft-lbs]	Future Thrust Goal KN [lbf]	Past Static Strokes	Future Static Strokes	Past DP Strokes	Future DP Strokes
1220 [900]	280 [63,000]	110	220	100	100

The torque and thrust increase goals in this example are very modest. As a result, a simplified approach is taken in which the more severe future static and dynamic fatigue are examined to see if they result in a restriction in the original 2,000 valve stroke cycle rating. If unrestricted cycles are determined, then the less severe past static and dynamic strokes would also be addressed. The load profile inputs for static valve strokes are as follows:

Torque 1 = 1220 N-m [900 ft-lb]
Torque 3 = -601 N-m [-450 ft-lb]
Ramp 1 = 4.15 sec
Ramp 3 = 2.05 sec

Since no dynamic data were available, a dynamic load profile was simulated using the design basis system data and conservative seat and guide friction factors in conjunction with the static test data and thrust and torque goals. This resulted in the following dynamic load profile inputs:

Torque 1a = 1136.6 N-m [838.3 ft-lb]
Torque 1b = 1220 N-m [900 ft-lb]
Ramp 1a = 11.7 sec
Ramp 1b = 3.76 sec
Dwell 1 = 7.26 sec
Torque 2 = -305 N-m [-225 ft-lb]
Ramp 2 = 1.04 sec
Torque 3a = -960 N-m [-708.3 ft-lb]
Torque 3b = -960 N-m [-708.3 ft-lb]
Ramp 3a = 2.68 sec
Ramp 3b = 9.9 sec
Dwell 3 = 7.26 sec

The static and dynamic load profile analyses showed dynamic alternating and mean stress levels below the endurance limit of the worm and worm shaft materials. These low stress levels resulted in fatigue life predictions in excess of the 2,000 valve stroke cycle actuator rating. As a result, the sum of the total past and future static and dynamic strokes can be up to 2,000. It was recommended, however, that the worm shaft bushing be inspected for possible excessive wear after 500 valve stroke cycles since wear was observed during the industry sponsored SMB-1 actuator cycle test program.

Conclusion

Actuator torque fatigue analysis is very sensitivity to the valve load profile history. As a result, the analysis is much more involved than that for thrust fatigue which only requires knowledge of the maximum thrust achieved in the open and close direction and is independent of the load profile. A conservative estimate of the past and future static and dynamic valve stroke load profiles can be used to develop input data for the fatigue life model. Depending on the resulting stress levels and number of fatigue cycles, restricted or unrestricted actuator life is possible.

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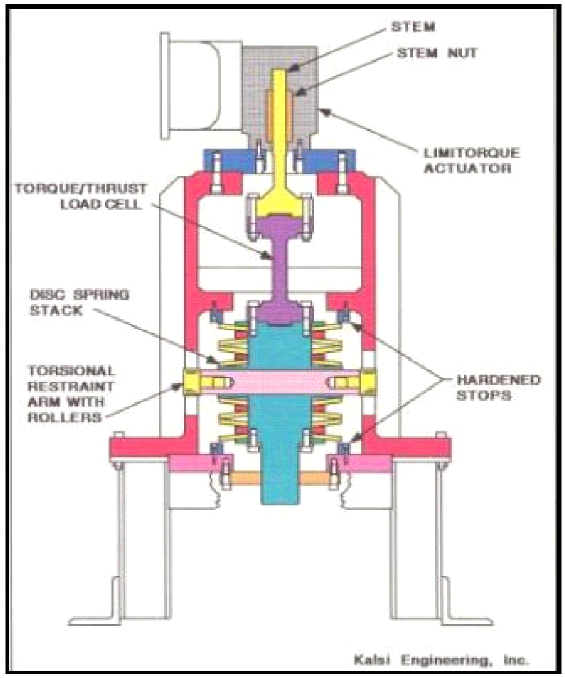

Figure 1 - Test Fixture Cross Section

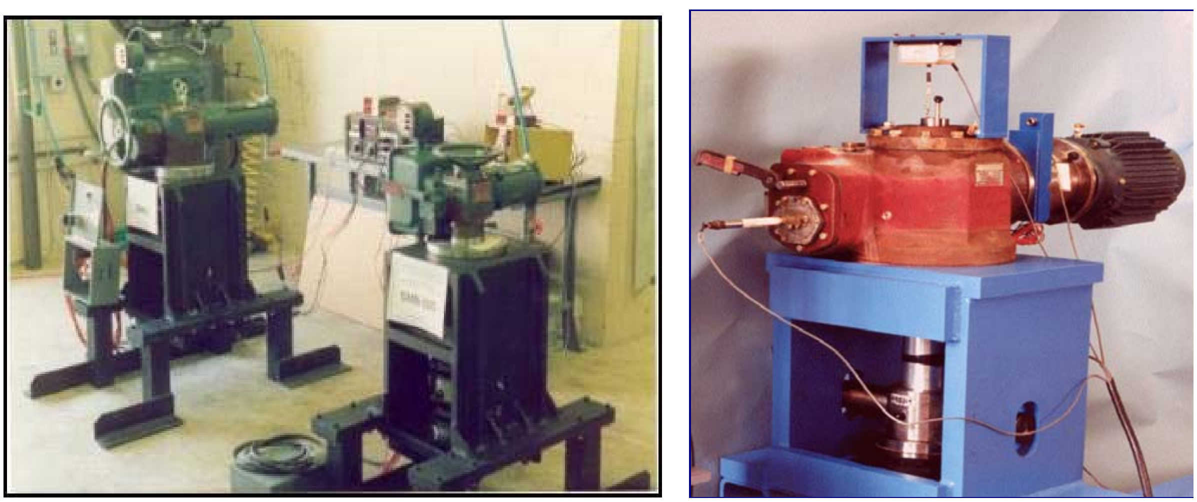

Figure 2 - Test Fixtures Used for Cyclic Testing of Limitorque SMB-000 through SMB-1 Actuators (shown on the left). A higher load capacity test fixture (shown on the right) was used for SMB-2 Testing

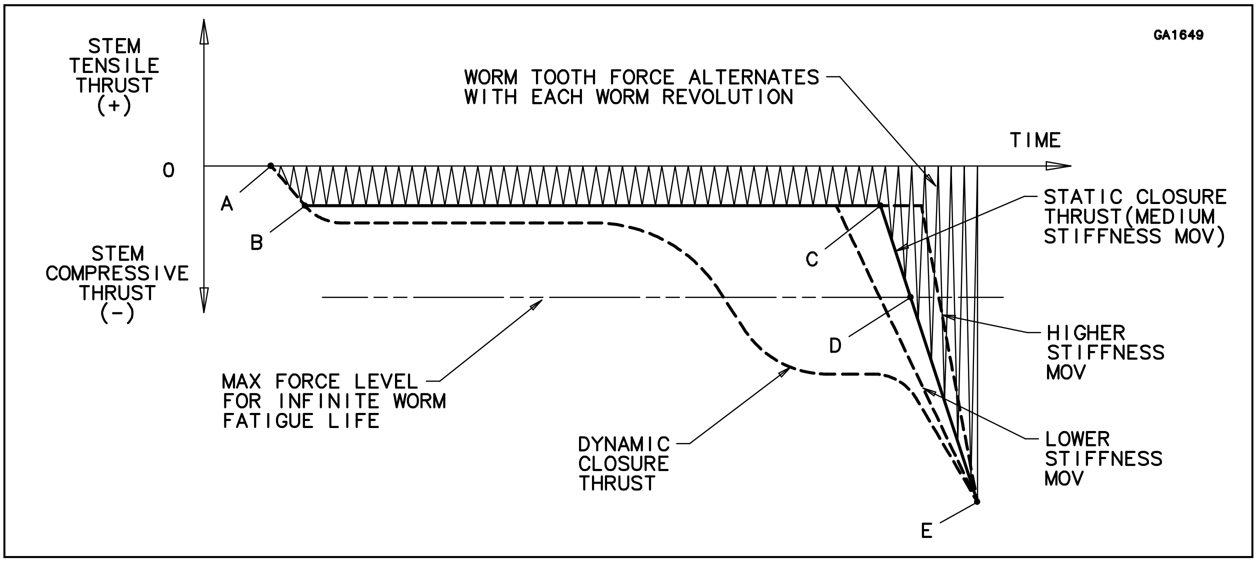

Figure 3 - Thrust Components are Subject to Only One Force Cycle for Each MOV Stroke; In Contrast, Torque Train Components are Subject to Multiple Cycles of Variable Force Magnitude for Each MOV Stroke

Figure 4 - Typical Examples of Limitorque Actuator Torsional Component Failures under Over-torque Conditions during Phase I Testing

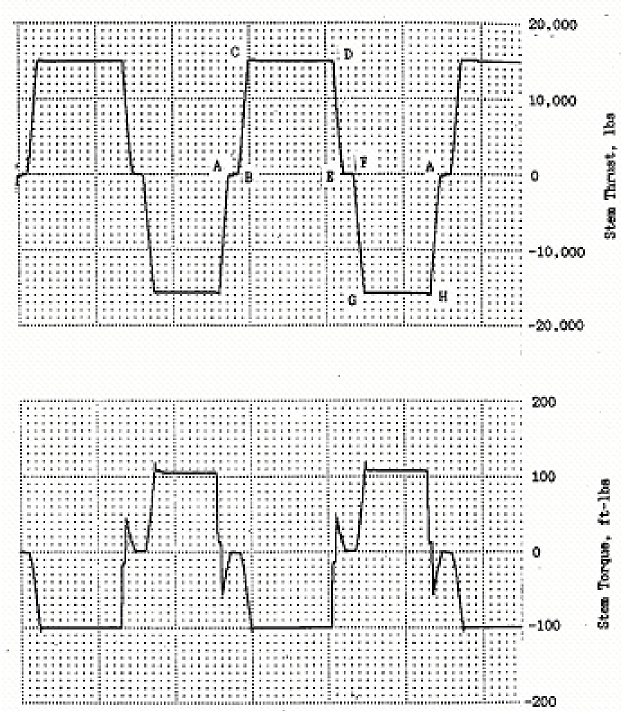

Figure 5 - Typical Stem Thrust and Torque Profiles Measured in Phase I Test Program

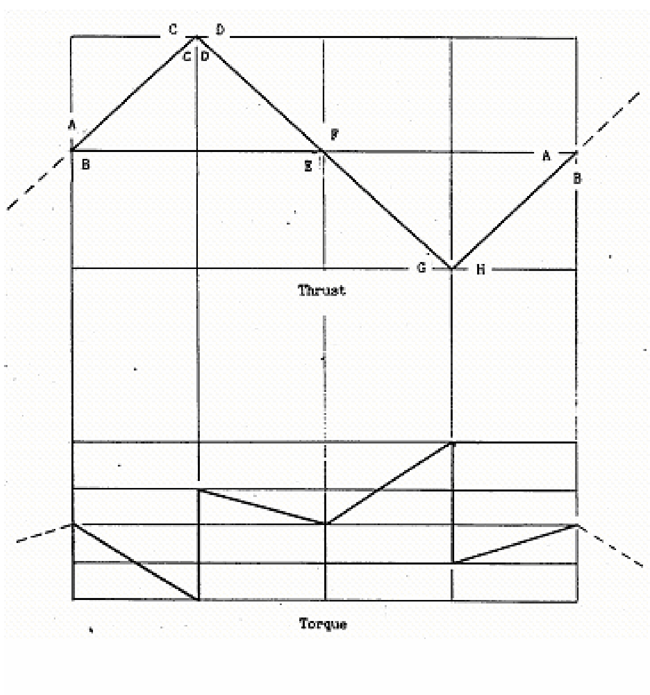

Figure 6 - Effective Load Cycle on Limitorque Phase I Test Actuator Stem

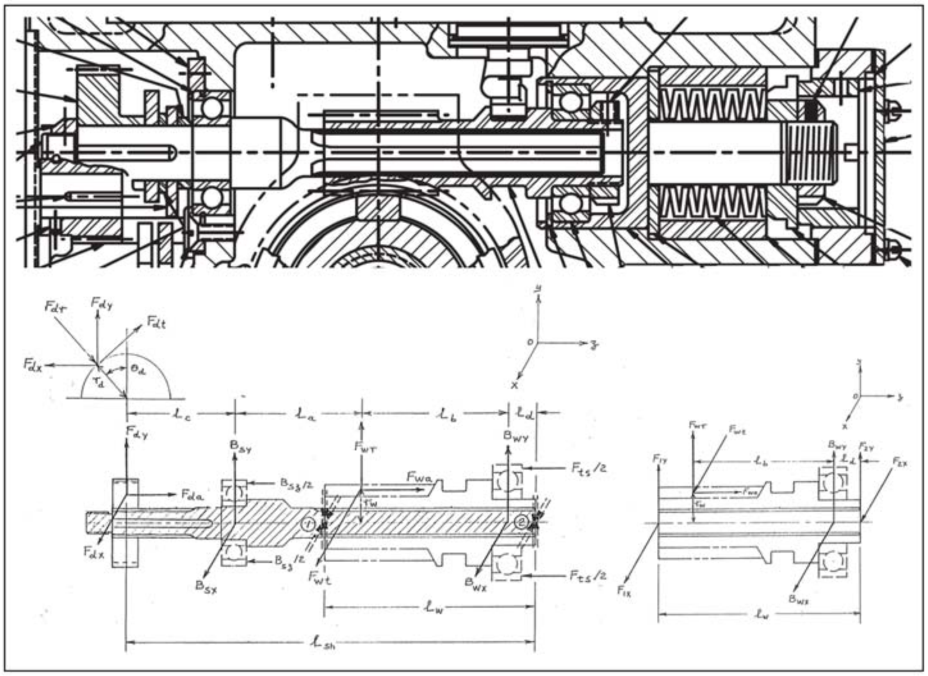

Figure 7 - Mechanical Model and Load Diagram for the Worm/ Worm Shaft Assembly of SMB-000/00 Actuators

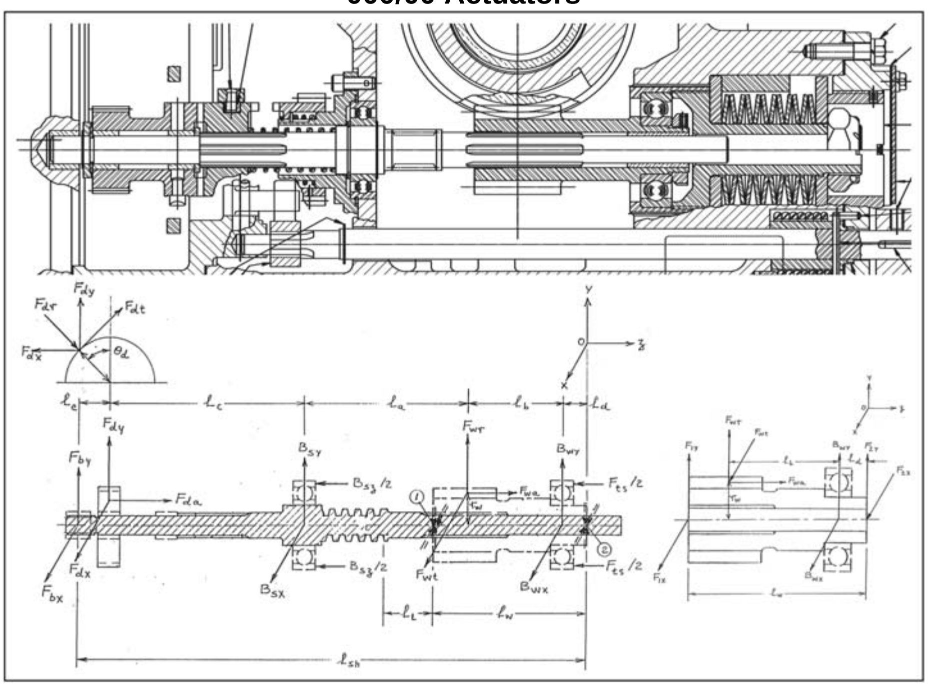

Figure 8 - Mechanical Model and Load Diagram for the Worm/ Worm Shaft Assembly of SMB-0/1/2 Actuators

Table 1
Summary of Torsional Components Predicted and Experimental Fatigue Life Data Used to Validate the Fatigue Life Software

Torsional Component	Average Test Torque (% of Rating)	No. of Cycles to Failure In Test	Predicted No. of Cycles to failure	Ratio Test/Prediction
SMB-000				
8620 worm	117	755	610	1.24
4320 worm	117	2,458	2,039	1.21
4320 worm	117	1,648	2,039	0.81
Worm shaft	106	4,870	4,818	1.01
SMB-00				
Worm	96	3,774	3,767	1.01
Worm Shaft	96	none (>4000)	8,967	*
SMB-0				
Worm	104	none (>4000)	none	*
Worm shaft	104	none (>4000)	3,760	*
SMB-1				
Worm	141	none (>4000)	none	*
Worm shaft	141	none (>1974)	1,489	>1.32
Worm shaft	141	1167	1,489	0.78
Worm shaft	141	714	1,489	0.48
SMB-2				
Worm	113	none (>4000)	none	*
Worm Shaft	113	none (>4000)	none	*
Average, for all tests	NA	2,198	2,321	0.95
Range, for all tests	NA	755-4,870	610-4,818	0.48-1.32

*No failure encountered during 4,000 cycle testing, therefore, ratio can not be calculated.

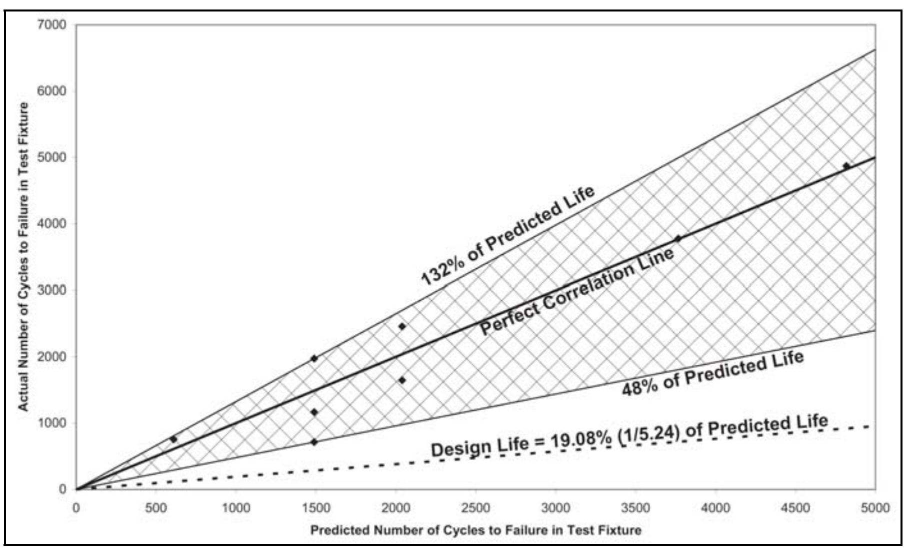

Figure 9 - Software Predictions were Validated against Test Data for Fatigue Life (both failure and no-failure data), and Allowable Design Life of Actuator Torsional Components

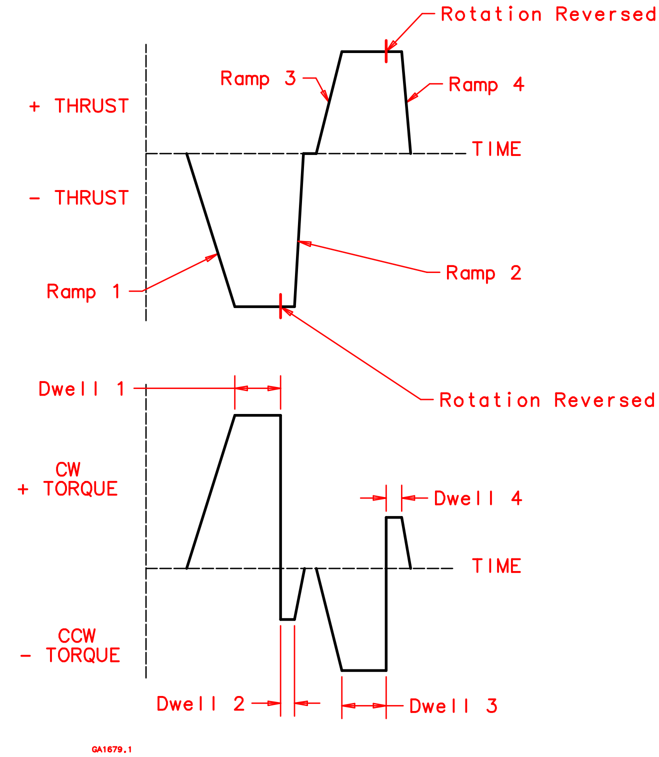

Figure 10 – Definition of Load Profiles used in Torque Fatigue Model

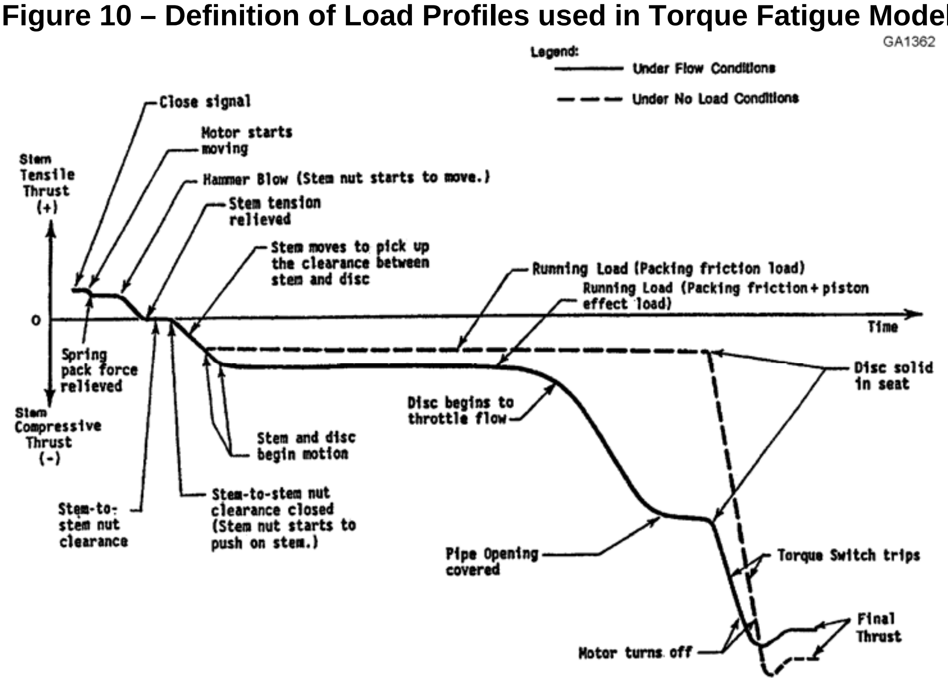

Figure 11 – Close Stroke Direction Gate Valve Signature Showing Differences between Static and DP Load Conditions

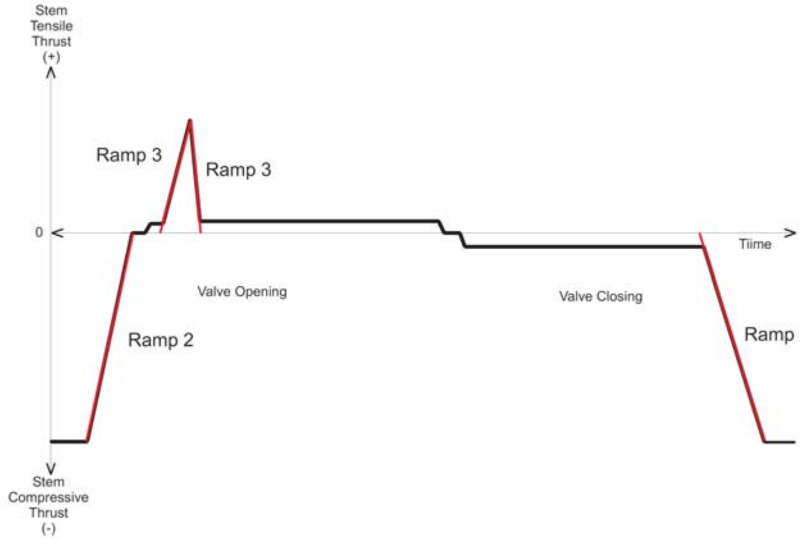

Figure 12 – Ramp Profiles used to Model Torque Fatigue for a Static Valve Stroke

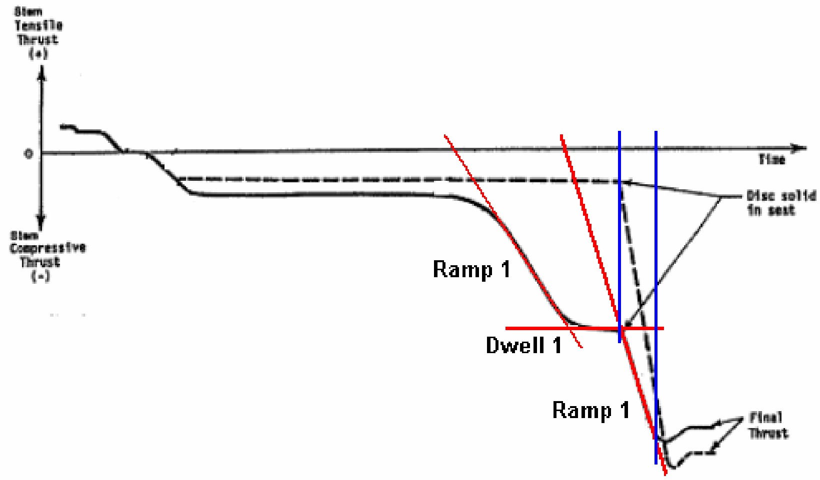

Figure 13: Gate Valve Closing Dynamic Stroke Ramp and Dwell Definitions

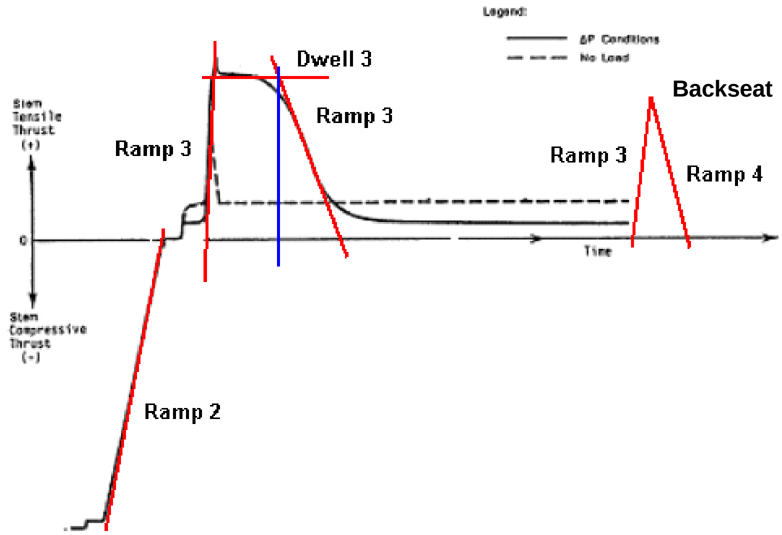

Figure 14: Gate Valve Opening Dynamic Stroke Ramp and Dwell Definitions

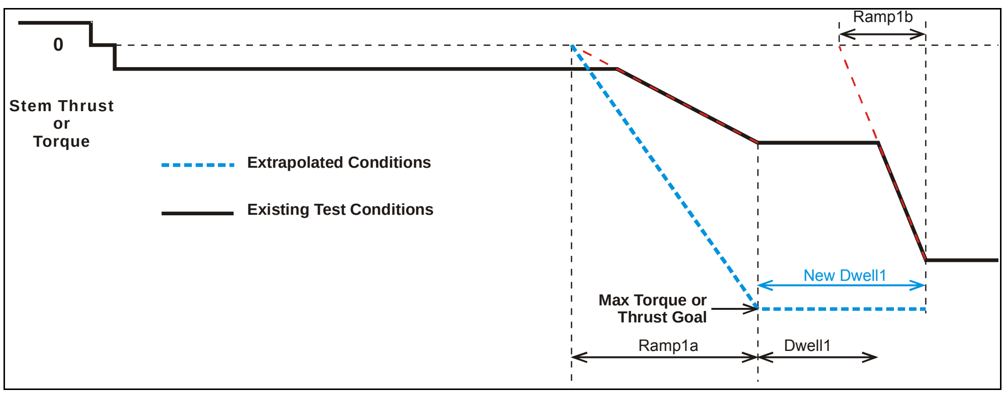

Figure 15 – Approach for Modeling Future Dynamic Close Valve Strokes

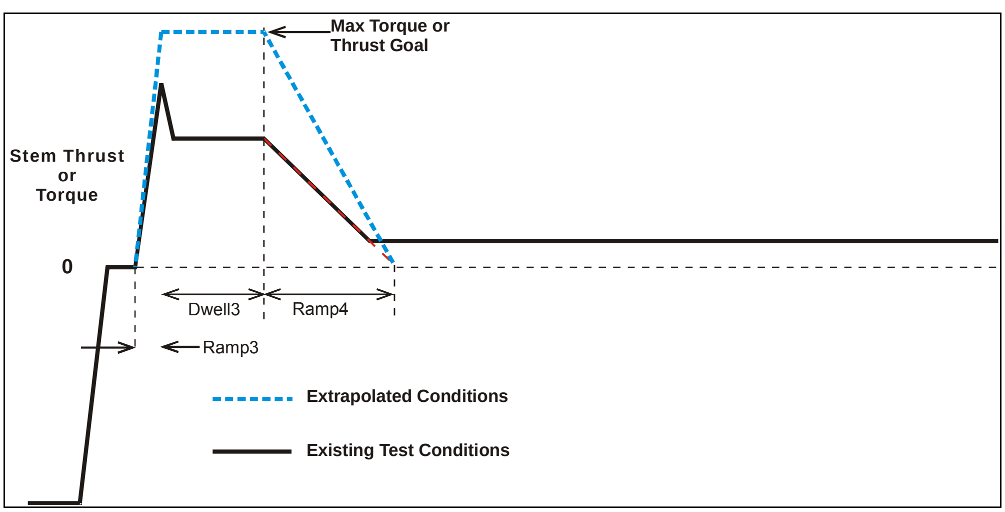

Figure 16 – Approach for Modeling Future Dynamic Open Valve Strokes

Non-JOG Nuclear Station Conformance with the JOG PV Program

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Abstract:

The purpose of the paper is to describe the background and general details of a program that, when completed, would establish that the Motor Operated Valve (MOV) Program in place at Southern California Edison (SCE) San Onofre Nuclear Generating Station (SONGS), Units 2 and 3, is in conformance with the Joint Owners Group (JOG) MOV Program.

On September 18, 1996, the United States Nuclear Regulatory Commission (NRC) issued Generic Letter (GL) 96-05, “Periodic Verification of Design-Basis Capability of Safety-Related Motor-Operated Valves,” requesting each nuclear power plant licensee to establish a program to verify on a periodic basis that safety-related MOVs continue to be capable of performing their safety function within the current licensing basis of the facility. In order to establish an industry wide position on GL 96-05, or long-term MOV performance, the industry conducted the JOG MOV Periodic Verification (PV) Program which provides a justified approach for periodically testing MOVs. The JOG program, had participation by 98 of the 103 operating U.S. reactors, was completed in 2003. A program document was prepared and submitted to the NRC and, subsequently, a Safety Evaluation Report (SER) was issued endorsing the JOG document.

SONGS, Units 2 and 3, elected not to participate in the industry-wide JOG program on MOVs PV because of the large number of valves (WKM design) at SONGS that are unique and dissimilar to valves installed at other nuclear plants. However, to strengthen the provisions on its MOV PV Program SONGS decided to approach Westinghouse for the development of a program that would meet two objectives; demonstrate conformance with the JOG MOV PV Program, and to meet SONGS SER commitment. The planned program is a collaborative effort whereby Westinghouse and SONGS will jointly develop the elements of the final program document. SONGS brings the knowledge of SONGS MOV PV Program while Westinghouse brings JOG MOV PV knowledge.

Session 2 (a): IST I

Session Chair: Craig D. Sellers, Alion Science and Technology

ASME OM Code Subsection ISTE –

A Discussion of the Published Subsection

Craig D. Sellers
Alion Science and Technology
Member ASME OM Committee

Abstract

After many years of work and technical iterations, Subsection ISTE has finally been approved and published. This paper discusses the final requirements for Risk-Informed Inservice Testing of pumps and valves.

Subsection ISTE provides mandatory requirements for owners' who voluntarily elect to implement a risk-informed inservice testing (IST) Program. The Subsection was prepared by combining the component categorization requirements and methodology from Code Case OMN-3 with component specific testing requirements developed, or under development, by the component-specific subgroups. Many of these requirements were based on the existing risk-informed Code Cases.

1.0 INTRODUCTION

Subsection ISTE provides mandatory requirements for owners' who voluntarily elect to implement a risk-informed inservice testing (IST) Program for pumps and valves. The Subsection was prepared by combining the component categorization requirements and methodology from Code Case OMN-3 with the test requirements of the Risk-Informed Component Code Cases, recently approved but unpublished Appendix III for electric motor-operated valves (MOVs), and Appendix IV for pneumatic and hydraulic-operated valves (AOVs/HOVs) which is currently in the course of preparation.

Subsection ISTE does not address dynamic restraints (snubbers). Code Case OMN-10, risk-informed inservice testing of snubbers, provides different requirements for the safety significance categorization of snubbers than Code Case OMN-3. This may be resolved by the incorporation of alternate risk ranking provisions in future revisions.

2.0 TECHNICAL REQUIREMENTS

The requirements of Subsection ISTE are a combination of:

1. The component categorization requirements and methodology from Code Case OMN-3, *Requirements for Safety Significance Categorization of Components Using Risk Insights for Inservice Testing of LWR Power Plants*,(1)
2. The risk-informed IST requirements for check valves from Code Case OMN-4, *Requirements for Risk Insights for Inservice Testing of Check Valves at LWR Power Plants,* (2)
3. The risk-informed IST requirements for pumps from Code Case OMN-7, *Alternative Requirements for Pump Testing,* (3)
4. The risk-informed IST requirements for motor-operated valves from Code Case OMN-11, *Risk-Informed Testing for Motor-Operated Valves,* (4)

5. The new IST requirements for motor-operated valves in Appendix III (5) which was recently approved but is yet to be published, and
6. The new IST requirements for pneumatically and hydraulically-operated valves in Appendix IV (6) which is currently under development but stalled.

2.1 Component Classification Requirements

Subsection ISTE requires components included in the risk-informed IST Program to be classified as either High Safety Significant Components (HSSCs) or Low Safety Significant Components (LSSCs). The component classification requirements of Code Case OMN-3 were incorporated directly into ISTE.

2.2 Pump Test Requirements

The test requirements for pumps are a direct incorporation of Code Case OMN-7.

Pumps classified as HSSCs are required to be tested in accordance with subsections ISTA and ISTB. These are the current test requirements for pumps.

Pumps classified as LSSCs are required to be tested in accordance with ISTA and ISTB except that longer test intervals are allowed and the requirement for the comprehensive pump test is eliminated. In place of the comprehensive pump test, each LSSC pump is required to be Group A tested within ± 20% of pump design flow rate at least every 5 years or 3 refueling outages, whichever is longer.

2.3 Check Valve Test Requirements

Risk-informed test requirements are provided for only the exercise testing of Category C check valves. The seat leak testing requirements of subsections ISTC remain a requirement. The test requirements for check valves are a direct incorporation of Code Case OMN-4.

Check valves classified as HSSCs must be placed in a Condition Monitoring Program in accordance with OM Appendix II. (7)

Check valves classified as LSSCs may either be placed in a Condition Monitoring Program in accordance with OM Appendix II or tested in accordance with the existing requirements of ISTC.

2.4 Motor-Operated Valve Test Requirements

Risk-informed test requirements for electric motor-operated valves refer to OM Code Appendix III which was recently approved by ASME but is yet to be published. Appendix III is the incorporation of Code Case OMN-1 into the OM Code as an Appendix. The seat leak testing requirements of ISTC remain a requirement.

MOVs classified as HSSCs must be tested in accordance with Appendix III (OMN-1) using a mix of static and dynamic testing.

MOVs classified as LSSCs can be associated with other MOV groups. Associated is similar to grouping using significantly relaxed grouping criteria. The test results of the MOV group are then applied to the associated LSSC MOVs. If a LSSC MOV cannot be associated with another MOV Group being tested, the MOV shall be tested in accordance with Appendix III. The test requirements for LSSC MOVs are identical to those of Code Case OMN-11 modified to refer to Appendix III rather than Code Case OMN-1.

2.5 Pneumatically and Hydraulically Operated Valve Test Requirements

Subsection ISTE refers to Appendix IV for pneumatically and hydraulically operated valve (AOV & HOV) test requirements. Appendix IV has been in development for a number of years but progress has stalled. ASME has approached industry groups for support in development of this appendix. At this time, risk-informed IST requirements for AOVs and HOVs do not exist in the OM Code.

2.6 Other Requirements

Subsection ISTE specifies requirements for; Monitoring, Analysis and Evaluation, and Records and Reports. These requirements incorporated directly from Code Case OMN-3.

3.0 REFERENCES

1. OM Code Case OMN-3, Requirements for Safety Significance Categorization of Components Using Risk Insights for Inservice Testing of LWR Power Plants.
2. OM Code Case OMN-4, Requirements for Risk Insights for Inservice Testing of Check Valves at LWR Power Plants.
3. OM Code Case OMN-7, Alternative Requirements for Pump Testing.
4. OM Code Case OMN-11, Risk-Informed Testing for Motor-Operated Valves.
5. OM Code Appendix III, Preservice and Inservice Testing of Active Electric Motor Operated Valve Assemblies in Light-Water Reactor Power Plants. (Recently approved but is yet to be published) and
6. OM Code Appendix IV which is currently under development but stalled
7. OM Code Mandatory Appendix II, Check Valve Condition Monitoring Program.

WHITE PAPER

APPLYING THE OM CODE

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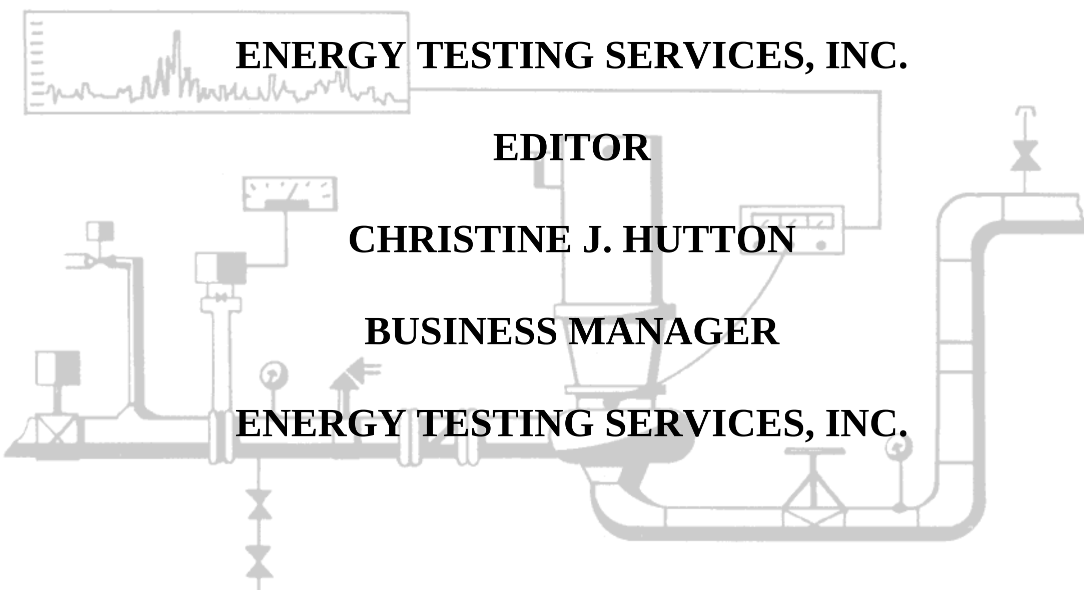

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APPLYING THE OM CODE

ABSTRACT

A major shift in the philosophy of Code implementation is currently taking place in the nuclear industry. These practices have generated uncertainties in the proper usage of the OM Code. This shift can be traced back to the Code's organizational control. The implementation controls of the Inservice Testing of Pumps and Valves were released from the Boiler and Pressure Vessel Committees to the Operation and Maintenance Committees. It should be emphasized that the ASME OM Code has become the Code of Record by its endorsement within the Code of Federal Regulations {10CFR50.55(a)(b)}. Regulatory documents must reference the ASME Operation and Maintenance (OM) Code, thus superseding the ASME Boiler and Pressure Vessel (B&PV) Code, Section XI. The B&PV Code membership has been in existence for a substantial period of time and maintained control of the inservice testing requirements until it turned them over to the OM organization. The B&PV ownership period allowed their members to experience the trials and tribulations of changing or adding to Code with the resulting checks and balances protecting the Code's integrity.

This White Paper will provide numerous examples to support a shift in the application of the OM Code. Additionally, a discussion will be provided detailing the user's responsibilities when applying, interpreting, and implementing the Code's requirements. Using the structure of both the ASME B&PV and OM Codes, this paper will cover and illustrate the missing key ingredient – guidance - for satisfactorily applying each OM Code item. Specific examples will be presented to identify the current and past structure of each Code. After identifying the structure, the paper will provide the necessary information to clarify the proper method for implementing a Code. If the structural guidance within the paper is followed, the information provided ensures the OM Code's integrity for the future. The last topic to be highlighted, a shift in Code usage, examines the current practices when issuing an OM Code Interpretation. The current practices appear to enforce OM Code changes or additions by interpretations which are not within the scope of an issued interpretation. The specific examples to be emphasized within the discussion relate to OM Code Mandatory Appendix I. In conclusion, the author will provide recommendations for changing the current practices, as they apply to the OM Code implementation, if a need is determined to exist.

DISCLAIMER

The following written narrative, with exception of items quoted from specified documents, is the opinion of the author and should only be used in the context to which it applies. This paper is written with the intent to stimulate conversation on the topics being presented. As Webster states, a <u>conversation</u> is an oral exchange of sentiments, observations, opinions, or ideas.

A MAJOR SHIFT

It has always been understood that the Code will not be enforced for many years following its publication. A major change in the Code implementation philosophy is currently presenting itself to the industry. This becomes evident when users who routinely implement the Code appear to be lost and confused about how to satisfy Code requirements. An even bigger indicator is that when a utility is fined for its current practices and interpretations, opinion or other documents are used to support the finding, not the actual Code. If only one user is applying the Code in this manner, it could be considered a one-time event. When numerous users agree with the fined utility's position, the Code Committee should perform an introspective review of its processes and practices to see if there is a developing flaw in the manner in which Code revisions, additions, interpretations and inquiries are being handled. When a problem is detected, probable causes could include extensive familiarity with Code, complexities in changing the Code, inexperience of Code members at the Working Group level, improper representation of implementers at meetings, and other plausible reasons.

If a problem can be detected, the flaw should be traced back to the Code's organizational control. The implementation controls of the Inservice Testing of Pumps and Valves were released to the Operation and Maintenance Organization from the Boiler and Pressure Vessel Organization back in the late 1990s. It should be emphasized that the ASME OM Code has become the Code of Record by its endorsement within the Code of Federal Regulations {10 CFR 50.55(a)(b)}. This change of the Code of Record became evident to all Code users and implementers, because each individual licensee had to change its reference for inservice testing of pumps and valves to the ASME OM Code within Technical Specifications, Technical Specification Bases, and Updated (or Final) Safety Analysis Reports. These major regulatory documents must reference the ASME Operation and Maintenance (OM) Code, superseding the ASME Boiler and Pressure Vessel (B&PV) Code, Section XI.

The B&PV Code membership has been in existence forever (or at least for a very long time) and maintained control of the inservice testing requirements until the turnover to the OM organization. The extensive B&PV ownership period allowed their members to experience the trials and tribulations of changing or adding to Code with the result being the creation of checks and balances to protect the Code's integrity. The B&PV members provided Code users with a well defined Code structure allowing the Code as the primary source, if not the single document to ensure compliance with all the regulated requirements.

With the current ownership being transferred, comments about the effectiveness of the new ownership have been circulating around the industry including:

- "If you need clarification read NUREG-1482 which provides guidance the NRC position for that portion of the Code."
- "That subparagraph is clarified in Code Interpretation No. [etc.]"
- "This is the way the Code is moving, therefore this how you should implement the Code."
- "Everybody is developing their testing schedule in this manner since that Interpretation came out."
- "We are not an inservice testing Code, but an operations Code."
- "We can change a Code faster under the OM organization than the old B&PV committee."
- "The Code was changed to make it easier on the plant."
- "They put that note in so people would not try to use the Appendix improperly."
- Finally, "I do not care what the Code states in Subsection ISTA, we believe the Code meant it this way."

EXAMPLE

> "Subject: OE25262 – ASME Class 1 Pressure Relief Valve Test Frequency (Dresden)
>
> Abstract: Discussions with the NRC staff have identified that the ASME OM Code test interval requirement for ASME Class 1 pressure relief valves is 5 years "test-to-test" frequency. The NRC staff considers Unit 2 to be in non-compliance with the ASME OM Code requirements and Technical Specification requirements for Class 1 Pressure Relief Valves that have exceeded the 5-year "test-to-test" frequency even though the installation time is less than 5 years. To resolve this issue, Unit 2 declared a missed surveillance for two ASME Class 1 pressure relief valves."

The most dramatic shifts occur when the OM Code membership try to make changes within the Code to incorporate specific requirements for different types of similar valves, different testing by type of plant design, complying with ever changing regulatory issues, current approved relief requests, testing predicated on satisfying specific safety system design or accident parameters, etc.

EXAMPLE

The example that shows how far down this road we have traveled is the testing of check valves.

Then - Check valves were to be exercised in the open **or** closed direction to satisfy the Code. The user of the ASME B&PV Code, Section XI, had a very simple mandate. Get the valve open **or** closed, periodically. Of course the first issue was to ensure the check was in its normal operating position of open **or** closed prior to exercising.

Now - Simple check valve exercising requirements require movement verification that must consider safety functions, series pairs, condition monitoring, disassembly, capacity certification, non-intrusiveness, and the mechanical exerciser. Then to top it off there are generic valve testing requirements in which check valves could follow to verify valve obturator movement.

> "ISTC-3530 Valve Obturator Movement. The necessary valve obturator movement shall be determined by exercising the valve while observing an appropriate indicator, such as indicating lights that signal the required changes of obturator position, or by observing other evidence, such as changes in system pressure, flow rate, level, or temperature, that reflects change of obturator position."

The Code for Inservice Testing of Pumps and Valves was originally developed to test two types of pumps (positive displacement and centrifugal) and various categories of valves (A, B, C, D, or multiple categories). The tests were to exercise the pump or valve, checking appropriate parameters to ensure it is operating acceptably. The concept was very simple and many effective inservice testing programs were developed under these rules.

APPLYING A CODE

When a professional opens a Code book he brings a certain working knowledge of Codes with him. This professional maybe a member of the Code Committee, a Regulator, an

Authorized Inspector, an Engineer, a Quality Inspector, Certified Vendor, Licensee Representative, or other professional. This working knowledge can aid or hinder in the implementation of the Code requirements. The only certainty is that the individual is to utilize the Code as written and approved. The most common verb for a user of the Code is that he is applying the Code. Members of the Code Committee have an additional verb at their disposal. They may be interpreting the Code. The inquiry or interpretation of the Code is usually performed at working group level, which is acceptable, because they are not allowed to change or add to the intent of the Code, but to provide clarification. Finally, Licensee Representatives are unique in that they must be the implementing force for Code requirements.

CODE OF RECORD

As discussed in the previous section each user of the Code must know the Code of Record for implementing the testing requirements. When determining the Code of Record, the leading source is the Code of Federal Regulation (CFR) within 10 CFR 50.55a, Codes and Standards, 10 CFR 50.55a(f) "Inservice testing requirements".

"10 CFR 50.55(a)(f)(4)(ii) Inservice tests to verify operational readiness of pumps and valves, whose function is required for safety, conducted during successive 120-month intervals must comply with the requirements in the latest edition and addenda of the Code incorporated by reference in paragraph (b) of this section 12 months before the start of the 120-month (or the optional ASME Code cases listed in NRC Regulatory Guide 1.147, through Revision 14, or 1.192 that is incorporated by reference in paragraph (b) of this section), subject to the limitations and modifications listed in paragraph (b) of this section."

The CFR further distinguishes the Code of Record in paragraph (b), identifying the applicable ASME OM Code Edition and Addenda:

"10 CFR 50.55(a)(b)(3) As used in this section, references to the OM Code refer to the ASME Code for Operation and Maintenance of Nuclear Power Plants, and include the 1995 Edition through the 2003 Addenda subject to the following limitations and modifications."

EXAMPLE

"10 CFR 50.55(a)(b)(3)(vi), Exercise interval for manual valves. Manual valves must be exercised on a 2-year interval rather that the 5-year interval specified in paragraph ISTC-3540 of the 1999 Addenda through the latest edition and addenda incorporated by reference in paragraph (b)(3) of this section, provided that adverse conditions do not require more frequent testing."

ORGANIZATION OF THE ASME OM CODE

The "FOREWORD" provides a discussion of the formation of the ASME OM CODE.

The "PREPARATION OF TECHNICAL INQUIRIES TO THE COMMITTEE ON OPERATION AND MAINTENANCE OF NUCLEAR POWER PLANT"S provides a discussion on developmental business. This includes consideration of written requests for interpretations, Code Cases, and revisions to Operation and Maintenance Code and development of new requirements.

The "COMMITTEE ON OPERATION AND MAINTENANCE OF NUCLEAR POWER PLANTS" identifies the organizational structure and members currently within this organization.

The "PREFACE" provides an "ORGANIZATION OF THIS CODE"

"This Code is published with one Section, entitled Section IST, Rules for Inservice Testing of Light-Water Reactor Nuclear Power Plants. Section IST is divided into Subsections, mandatory appendices, and nonmandatory appendices. Provisions for adding future Sections exist.

Parts 1, 4, 6, 10 from ASME/ANSI OM-1987, Operation and Maintenance of Nuclear Power Plants, are incorporated into this Code as shown below:

OM Code Designation	Previous OM-1987 Designation	
Appendix I	Part 1	Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices
Subsection ISTD	Part 4	Examination and Performance Testing of Nuclear Power Plant Dynamic Restraints (Snubbers)
Subsection ISTB	Part 6	Inservice Testing of Pumps in Light-Water Reactor Power Plants
Subsection ISTC	Part 10	Inservice Testing of Valves in Light-Water Reactor Power Plants"

DESCRIPTION OF SUBSECTIONS AND MANDATORY APPENDICES

"The following description is intended to provide the reader with general information on the scope of coverage and rationale for development of each item described."

A missing key ingredient to the OM Code Designation for Subsection ISTA was not addressed. The following should have been identified:

OM Code Designation	Previous B&PV Code Designation	
Subsection ISTA	Subsection IWA	General Requirements

The Subsection IWA provides guidance. This includes its use for development Subsection ISTA and within the subsection Article IWA-9000, Glossary (i.e., inservice life, pump, valve, etc.).

The "SECTION IST RULES FOR INSERVICE TESTING OF LIGHT-WATER REACTOR POWER PLANTS CONTENTS" provides the overall Table of Contents of the OM Code with a detailed content preceding each OM Subsection.

The "SUMMARY OF CHANGES ASME OM CODE – 2001" is self-explanatory.

This is followed by the actual SUBSECTIONS, MANDATORY APPENDICES, and NONMANDATORY APPENDICES.

ORGANIZATION OF A CODE, SECTION XI

As noted previously, the Inservice Testing of Pumps and Valves has long been governed by the Boiler and Pressure Vessel Code, Section XI (i.e., Edition 1970 through Edition 2001). The Boiler and Pressure Vessel Code established a strict set of rules for the construction and use of the Code. The following is the minimum structure required to successfully apply a Code.

"American Society of Mechanical Engineers, Boiler and Pressure Vessel (B&PV) Code, Section XI.

"SECTIONS"

Section ***I*** – Power Boilers
Section ***II*** – Materials – Specifications
Section ***III*** – Rules for Construction of Nuclear Power Plant Components
Section ***IV*** – Heating Boilers
Section ***V*** – Nondestructive Examination
Section ***VIII*** – Pressure Vessels
Section ***IX*** – Welding and Brazing Qualifications
Section ***X*** – Fiber Reinforced Plastic Pressure Vessels
Section ***XI*** – Rules for In-service Inspection of Nuclear Power Plant Components
Code Cases – Boilers and Pressure Vessels, Code Cases – Nuclear Components, Referenced ASME Standards, Interpretations

DIVISIONS – Section XI consists of three Divisions, as follows:

Division **1** – Rules for Inspection and Testing of Components of Light-Water Cooled Plants
Division **2** – Rules for Inspection and Testing of Components of Gas Cooled Plants
Division ***3*** – Rules for Inspection and Testing of Components of Liquid-Metal Cooled Plants

SUBSECTIONS – The Divisions are broken down into Subsections which are designated by capital letters, preceded by the letters ***IW*** in Division 1, etc:

Subsection	Title
IW***A***	General Requirements
IW***B***	Class 1 Components
IW***C***	Class 2 Components
IW***D***	Class 3 Components
IW***E***	Class MC and CC Components
IW***F***	Class 1, 2, 3, and MC Component Supports
IW***L***	Requirements for Class CC Concrete Components
IW***P***	Pumps
IW***V***	Valves

Subsections are divided into Articles, Subarticles, paragraphs, and, where necessary, into subparagraphs.

ARTICLES – Articles are designated by the applicable letters indicated above for the Subsection, followed by Arabic numbers, such as IWA-1000 or IWB-2000. Where possible, Articles dealing with the same general topics are given the same number in each Subsection, in accordance with the following scheme:

Article Number	Title
1000	Scope and Responsibility
2000	Examination and Inspection
3000	Acceptance Standards
4000	Repair Procedures
5000	System Pressure Tests
6000	Records and Reports
7000	Replacements

"The numbering of Articles and material contained in the Articles may not, however, be consecutive. Due to the fact that the complete outline may cover phases not applicable to a particular Subsection or Article, the rules have been prepared with some gaps in the numbering.

SUBARTICLES – Subarticles are numbered in units of 100, such as IWA-1***100*** or IWA-1***200***.

SUBSUBARTICLES – Subsubarticles are numbered in units of 10, such as IWA-21***30***, and may have no text. When a number such as IWA-11***10*** is followed by text, it is considered a paragraph.

PARAGRAPHS – Paragraphs are numbered in units of ***1***, such as IWA-213***1*** or IWA-213***2***.

SUBPARAGRAPHS – Subparagraphs, when they are *major* subdivisions of a paragraph, are designated by adding a decimal followed by one or more digits to the paragraph number, such as IWA-1111***.1*** or IWA-1111***.2***. When they are *minor* subdivisions of a paragraph, subparagraphs may be designated by lowercase letters in parentheses, such as IWA-1111***(a)*** or IWA-1111***(b)***.

REFERENCES – References used within this Section generally fall into one of six categories, as explained below.

- *References to Other Portions of This Section*. When a reference is made to another Article, Subarticle, or paragraph number, all numbers subsidiary to that reference shall be included. For example, reference to IWA-2000 includes all materials in Article IWA-2000; reference to IWA-2200 includes all material in Subarticle IWA-2200 includes all material in Subarticle IWA-2200; reference to IWA-2220 includes all paragraphs in IWA-2220 through IWA-2222.
- *References to Other Sections*. Other Sections referred to in Section XI are as follows:
  (a) Section II, Material Specifications
  (b) Section III, Nuclear Power Plant Components
  (c) Section V, Nondestructive Examination
  (d) Section IX, Welding and Brazing Qualifications.
- *References to Specifications and Standards Other Than Published in Code Sections*
- *References to Government Regulations.* U.S. Federal regulations issued by executive departments and agencies, as published in the Federal Register, are codified in the Code of Federal Regulations. The Code of Federal Regulations is published by the Office of the Federal Register, National Archives and Records Service, General Service Administration,

and may be purchased from the Superintendent of Documents, U.S. Government Printing Office, Washington, D.C. 20402. Title 10 of the Code of Federal Regulations contains the regulations for atomic energy. The abbreviated reference "10CFR50" is used to mean "Title 10, Code of Federal Regulations, Part 50."

- *References to Appendices*. Two types of Appendices are used in Section XI and are designated Mandatory and Non-mandatory.
  1. Mandatory Appendices contain requirements which must be followed in construction; such references are designated by a Roman numeral followed by Arabic numerals. A reference to III-1100, for example, refers to a Mandatory Appendix.
  2. Nonmandatory Appendices provide information or guidance for the use of Section XI; such references are designated by a capital letter followed by Arabic numerals. A reference to A-3300, for example, refers to a Nonmandatory Appendix.
- *References to Technical Reports.* The following reports prepared at the request of the American Society of Mechanical Engineers and Published by Electric Power Research Institute are relevant to Code-related articles of Section XI. Requests for copies should be directed to EPRI Research Reports Center, Box 50490, Palo Alto, CA 94303."

EXAMPLE

IWA-5250(a)(1)

STRUCTURE	ITEM	TITLE
Section	XI	Rules for Inservice Inspection of Nuclear Power Plant Components
Division	1	Rules for Inspection and Testing of Components of Light-Water Cooled Plants
Subsection	IWA	General Requirement
Article	5000	System Pressure Tests
Subarticle	5200	System Test Requirement
Subsubarticle	5250	Corrective Measures
Paragraph	None	None
Subparagraph	5250(a)	None
Subsubparagraph	5250(a)(1)	None

EXAMPLE

ISTC-5221(a)

STRUCTURE	ITEM	TITLE
Section	IST	Rules For Inservice Testing of Light-Water Reactor Power Plants
Division	None	None
Subsection	ISTC	Inservice Testing of Valves in Light-Water Reactor Power Plants
Article	5000	Specific Testing Requirements
Subarticle	5200	Other Valves
Subsubarticle	5220	Check Valves
Paragraph	5221	Valve Obturator Movement
Subparagraph	5221(a)	None
Subsubparagraph	5221(a)(1)	None

EXAMPLE

Mandatory Appendix I -1320(b)(1)

STRUCTURE	ITEM	TITLE
Section	None	None
Division	None	None
Subsection	None	None
Mandatory Appendix	I[1]	Inservice Testing of Pressure Relief Devices in Light-Water Reactor Nuclear Power Plants
N/A Foot Note	[1]	This Appendix contains requirements to augment the rules of Subsection ISTC, Inservice Testing of Valves in Light-water Reactor Nuclear Power Plants
Article	1000	General Requirement
Subarticles	1300	Guiding Principles
Subsubarticle	1320	Test Frequencies, Class 1 Pressure Relief Valves
Paragraph	None	None
Subparagraph	1320(b)	Replacement With Pretested Valves
Subsubparagraph	1320(b)(1)	None

MISSING TOOL

The key ingredient for properly applying the OM Code is missing. The Organization Of Section IST should provide the Code's structure from Section down to Subsubparagraph and include the use of References. This would include the Articles, Subarticles, Subsubarticles, Paragraphs, Subparagraphs, Subsubparagraphs and References similar to the ASME Boiler and Pressure Vessel Code, Section XI.

Providing this structure maintains the integrity of the OM Code. The structure ensures proper implementation as stated within the reference, *References to Other Portions of This Section:* when a reference is made to another Article, Subarticle, or paragraph number, all numbers subsidiary to that reference shall be included. This statement also implies that if another portion of this section is not referenced within the Article, Subarticle, etc., then the non-referenced portions and their subsidiary portions are not to be included in the Article, Subarticle, etc. All Codes follow a similar structural format (i.e., Code of Federal Regulations), and when implementing the Code requirements, compliance to structure guidance by the user is crucial in satisfying regulations. Therefore, to effectively apply the OM Code to allow compliance with the requirements of 10 CFR 50.55a, Code and Standards, the OM Code Committee should develop a corresponding **ORGANIZATION OF A CODE** similar in nature to ASME B&PV Code, Section XI.

ASME OM CODE COMMITTEE AND ITS MEMBERS

It is the responsibility of the *ASME OM Code Committee and its members* to provide a Code that is clear and concise in detailing the mandated requirements. They shall ensure that changes to the Code are effective throughout the entire document (Subsection through Appendices).

EXAMPLE

"DESCRIPTION OF SUBSECTIONS AND MANDATORY APPENDICES"

"The following description is intended to provide the reader with general information on the scope of coverage and rationale for development of each item described.

"Subsection ISTA: General requirements

"This Subsection is derived from the ASME Boiler and Pressure Vessel Code, Section XI, Division 1, Subsection IWA, and contains requirements directly applicable to inservice testing. Included are Owner's responsibility, duties of Inspector, and records."

This reference to the Subsection ISTA is good with the exception that this current Edition through Appropriate Addenda no longer includes ***duties of Inspector***. The OM Code has eliminated the Inspector oversight of the testing requirements for pumps and valves. This is an easily missed change that should have been identified and removed from the description. Additionally, you would expect that further guidance could have been provided to state that these general requirements are applicable to each of the other subsections and appendices. It is implied indirectly by using the wording, "applicable to inservice testing". Note – This allows minimum repetition of requirements and definitions.

CODE USERS

It is the responsibility of the *user* to apply the Code in the implementation of mandated requirements, as written. This would imply no interpretation or changing of the Code's intent. It should be emphasized that a user may be a member of the Code Committee, a Regulator, an Authorized Inspector, an Engineer, a Quality Inspector, Certified Vendor Supervisor, Licensee Representative, or other professional. Therefore, when a Code member is not performing his responsibilities in conducting standards development business he becomes simply a user (no better or worse than others) and must apply the Code as any other user would be required to.

PREPARATION OF TECHNICAL INQUIRIES

The Preparation of Technical Inquiries discusses the following items: Introduction, Inquiry Format, Code Revisions and Additions, Code Cases, Code Interpretations, and Submittals. The following is from the ASME OM Code:

"INTRODUCTION"

The ASME Committee on Operation and Maintenance of Nuclear Power Plants meets regularly to conduct standards development business. This includes consideration of written requests for interpretations, Code Cases, and revisions to Operation and Maintenance Code and development of new requirements as dictated by technological development. This supplement provides guidance to Code users for submitting technical inquiries to the Committee. Technical inquiries include requests for revisions or additions to the Code requirements, requests for Code Cases, and requests for Code interpretations.

"[…]

"Moreover, ASME does not act as a consultant on specific engineering problems or on the general application or understanding of the Code requirements. If, based on the inquiry information submitted, it is the opinion of the Committee that the inquirer should seek assistance, the inquiry will be returned with the recommendation that such assistance be obtained."

"As an alternate to the requirements of this Supplement, members of the Committee and its subcommittees, subgroups, and working groups may introduce requests for Code revisions or additions, Code Cases, and Code interpretations at their respective Committee meetings or may submit such requests to the secretary of a subcommittee, subgroup, or working group."

"All inquiries that do not provide the information needed for the Committee's full understanding will be returned."

"INQUIRY FORMAT"

Submittals to the Committee shall include:

(a) *Purpose*. Specify one of the following:

(1) revision of present Code requirement(s);
(2) new or additional Code requirement(s);
(3) Code Case;
(4) Code interpretation.

(b) *Background*. Provide the information needed for the Committee's understanding of the inquiry, being sure to include reference to the applicable Code subsection, appendix, edition, addenda, paragraphs, figures, and tables. Preferably, provide a copy of the specific referenced portions of the Code.

(c) *Presentations*. The inquirer may desire or be asked to attend a meeting of the Committee to make a formal presentation or to answer questions from the Committee members with regard to the inquiry. Attendance at a committee meeting shall be at the expense of the inquirer. The inquirer's attendance or lack of attendance at a meeting shall not be a basis for acceptance or rejection of the inquiry by the Committee."

"CODE REVISIONS AND ADDITIONS"

Requests for Code revisions or additions shall provide the following:

- Proposed Revision(s) or Addition(s). For revisions, identify the requirements of the Code that require revision and submit a copy of the appropriate requirements as they appear in the Code marked up with the proposed revision. For additions, provide the recommended wording referenced to the existing Code requirements.
- Statement of Need. Provide a brief explanation of the need for the revision(s) or addition(s).
- Background Information. Provide background information to support the revision(s) or addition(s) including any data or changes in technology that form the basis for the request that will allow the Committee to adequately evaluate the proposed revision(s) or addition(s). Sketches, tables, figures, and graphs should be submitted as appropriate. When applicable, identify any pertinent paragraph in the Code that would be affected by the revision(s) or addition(s) and paragraphs in the Code that would be affected by the revision(s) or addition (s) and paragraphs in the Code that reference the paragraphs that are to be revised or added."

"SUBMITTALS"

Submittals to and responses from the Committee shall meet the following:

Submittal. Inquiries from Code users shall preferably be submitted in typewritten form; however, legible handwritten inquiries will also be considered. They shall include the name, address, telephone number, and fax number, if available, of the inquirer and be mailed to the following address:

Secretary
Committee on Operation and Maintenance of Nuclear Power Plants
The American Society of Mechanical Engineers
Three Park Avenue
New York, NY 10016-5990

Response. The Secretary of the Operation and Maintenance Committee shall acknowledge receipt of each properly prepared inquiry and shall provide a written response to the inquirer upon completion of the requested action by the Committee."

"CODE INTERPRETATIONS"

Requests for Code interpretations shall provide the following:

(a) *Inquiry*. Provide a condensed and precise question, omitting superfluous background information, and, when possible, composed in such a way that a "yes" or a "no" *Reply*, possibly with brief provisos, is acceptable. The question should be technically and editorially correct.

(b) *Reply*. Provide a proposed *Reply* that will clearly and concisely answer the *Inquiry* question. Preferably, the Reply should be "yes" or "no" possibly with brief provisos.

(c) *Background Information*. Provide any background information that will assist the Committee in understanding the proposed Inquiry and Reply."

In recent history the OM Code interpretations have become exclusive to the ASME OM Code review and acceptance, and not the OM Standards being enforced by the ASME B&PV Code. If an interpretation becomes a vehicle to Change or Add to the Code or the brief provisos identifies or specifies items that can not be found in the Code, then the intent of the Code is threatened. Some recent OM Code Interpretations appear to have changed the intent of the Code. The stated question of some inquiries seems to be servicing a specific desire and not simply to clarify the existing Code requirement. It is known that only inquiries that are felt important to Code users are addressed or published, which in itself may also be self serving.

EXAMPLE NO. 1

Interpretation: 04-08
Subject: Appendix I, I-1350(a) (ASME OM Code-2001 through OMb-2003 Addenda)
Date Issued: November 22, 2005
File: OMI 05-01

Question: In accordance with the OM Code 2001 edition with 2003 Addenda, is it a requirement of Appendix I, I-1350(a), that the maximum test interval for a class 2 or 3 valve group that contains only one valve be 10 years.

Reply: No, in accordance with Appendix I, I-1350(a), a minimum of 20% of the valves from a group shall be tested within any 48-month interval. In the case of a valve group of one, this one shall be tested within any 48-month interval."

The ***Question*** is worded to allow the Reply to provide a brief proviso to state that a valve group of one exists. It would seem to be contrary to the Code as the definition of a valve group implies more than one. If the intent is being changed, the CODE REVISIONS AND ADDITIONS should be followed. The following would appear to support that the interpretation is changing the Code intent.

I-1200 Definitions

The following definitions are provided to ensure a uniform understanding of select terms used in this Appendix. Definitions for other related pressure relief devices terms can be found in Appendix I of ANSI/ASME PTC 25, Pressure Relief Devices.

Valve group: valves of the same manufacturer, type, system application, and service media."

Code definition uses "valves", not "***valve or valve(s)***" which infers that a single valve is not a group. When wording becomes an issue then Webster's Ninth New Collegiate Dictionary can be referenced as a standard; it defines "*group* - two or more figures forming a complete unit in a compositions" and "*grouping* - the act or process of combining in groups." Additionally, I-1320(a) and I-1350(a) refer to a minimum of 20% of the valves from each valve group, not to a valve.

<u>EXAMPLE NO. 2</u>

Interpretation: 01-18
Subject: ASME OM Code-1995 with ASME OMa Code-1996 Addenda, Appendix I
Date Issued: June 26, 2003
File: OMI 03-01

Question (1): Are replacement valves for an operating nuclear power plant required by para. I6 to be tested in accordance with para. I7.2, required to be tested 6 months prior to fuel cycle initial criticality?

Reply: No. Paragraph I7.2 does not apply after initial electric power generation. However, the valve must have been tested within 5 years or 10 years as required by I1.3."

The Question is worded correctly for a response (i.e., Reply) of No, but the brief proviso has no bearing on the stated question and as worded the proviso is misleading, not following Code etiquette. The reference to I 1.3 Guiding Principles does have both the 5-year, 10-year and replacement requirements. The problem with this proviso is that the 5 years or 10 years and the replacement are covered by completely different subparagraphs, with neither referencing the other.

Subparagraph I 1.3.3(a) 5-year Test Interval
Subparagraph I 1.3.3(b) Replacement With Pretested Valves
Or
Subparagraph I 1.3.5(a) 10-Year Test Interval
Subparagraph I 1.3.5(b) Replacement With Pretested Valves

Remember that:

"References to Other Portions of This Section. When a reference is made to another Article, Subarticle, or paragraph number, all numbers subsidiary to that reference shall be included. For example, reference to IWA-2000 includes all materials in Article IWA-2000; reference to IWA-2200 includes all material in Subarticle IWA-2200 includes all material in Subarticle IWA-2200; reference to IWA-2220 includes all paragraphs in IWA-2220 through IWA-2222."

Since these are both subparagraphs and no cross-reference to the other is made, they remain separate (i.e., one does not impact the other). The proviso is incorrect as stated or is misleading at best. Additionally, the proviso contradicts I 6, which the Question references.

"I 6 PRESSURIZED WATER REACTOR (PWR) - INTRODUCTION

Replacement valves of the same valve group shall be tested to the requirement of paras. I 7.1 and I 7.4. Replacement valves not of the same valve group previously used shall be tested to the requirements of paras. I 7.1 and I 7.2.

I 7.1 Testing Before Initial Installation
I 7.2 Testing Before Initial Electric Power Generation
I.7.4 Disposition After Testing or Maintenance"

These Subarticles at no point reference Subparagraph I 1.3.3(a) or Subparagraph I 1.3.5(a), again making the proviso incorrect as stated or misleading at best. Subarticle I 7.4 does reference Subparagraph I 1.3.1(e), which goes to show how the Code structure is being followed, as written.

EXAMPLE NO. 3

Interpretation: 01-18
Subject: ASME OM Code-1995 With ASME OMa Code-1996 Addenda, Appendix I
Date Issued: June 26, 2003
File: OMI 03-01

Question (2): In ASME OM Code-1995 with ASME OMa Code-1996 Addenda, para. I 1.3.3(a), the statement is made that, the test interval for any individual valve shall not exceed 5 years. Is this Code referring to installed life rather than test interval?

Reply (2): No."

Review of the Question and Reply seems harmless with a correct answer because it is very obvious that para I 1.3.3(a) discusses the test interval. Now comes the misleading part. Since the proviso in Question (1) misleads you to believe that replacement valves had to follow I 1.3.3(a), the terms "install life" and "test interval" are intermixed within the Question. It appears that someone is trying to place Code requirements on users that do not exist.

It is very clear that the concern was placing Code test interval requirements on replacement valves. The replacement valves have always had separate testing requirements as identified in BWR and PWR Pressure Relief Device Testing, but - as in the case of current Code - the manufacturer's production tests may be accepted.

It should be made clear at this time that replacement valve testing requirements are specifically omitted (i.e., not referenced) from Subarticle I 7.3 Periodic Testing by Article I 6.

I 7.3 Periodic Testing

Periodic Testing of all pressure relief devices is required. No maintenance, adjustment, disassembly, or other activity that could affect "as-found" set-pressure or seat tightness data is permitted before testing. Control ring adjustments are permitted in paras. I 8.1.1(g) and
I 8.1.2(g). Test frequencies are specified in paras. I 1.3.3, I 1.3.4, I 1.3.5, I 1.3.6, and I 1.3.7. When on-line testing is performed to satisfy periodic testing requirements, visual examination may be performed out of sequence."

Therefore the test frequencies are for other than replacement valves (for example newly installed or inservice valves) when referring to Periodic Testing. This shows once more how the Code Interpretation was not correct as stated or misleading at best.

EXAMPLE NO. 4

Interpretation: 01-18
Subject: ASME OM Code-1995 With ASME OMa Code-1996 Addenda, Appendix I
Date Issued: November 15, 1996
File: OMI-95-08

"Question (2): When ending the first 10 year OM-1 test interval and starting the new 10 year OM-1 test interval, does the 48 month requirement of ANSI/ASME OM-1 1981, Part 1, para. 1.3.4.1.2, ASME/ANSI OM-1987 (Part 1), paras. 1.3.4.1(b); ASME OM Code-1990, Appendix I, para. I1.3.5(b); ASME OMc Code-1994, Appendix I, para. I1.3.5(a); and ASME OM Code-1995, para. I1.3.5(a) start over at the beginning of the new test interval for Class 2 and 3 relief valves, excluding PWR Main Steam Safety Valves?

Reply (2): NO, the test interval applies to any 48 month period."

Again this interpretation is contrary to the way the Code is written. It separates the requirements into other 5-year or 10-year test intervals. The subparagraph is unique to the 5-year test interval as stated in header. To then assume it applies to subsequent 5-year or 10-year test intervals would be putting an addition to the Code. The Code and its stated requirements is only applicable as long as the Code remains the Code of Record within the Code of Federal Regulations {10 CFR 50.55(a)(b)}. The test interval requirement as stated may no longer exist in a newer Code.

CONCLUSION

With the above positions being known, the ASME OM Committee should investigate if certain retractions of the interpretations and enhancements would be warranted. The user who has to implement the Code must follow the Code as written. If other sources deviate from the Code in trying to encompass perceived concerns of the industry, concerns should be raised, since the implementer still must follow the Code. Finally, the implementer may go beyond the Code requirements, but in no case should he or she interpret the Code or change or cause additions to the Code's intent.

Nuclear Power Plant Pump, Valve and Snubber Inservice Testing Issues

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Tenth NRC/ASME Symposium on Valves, Pumps and Inservice Testing
July 2008

Abstract

This paper discusses recent issues related to inservice testing (IST) of pumps, valves and dynamic restraints (snubbers) at U.S. nuclear power plants. These issues were identified during the U. S. Nuclear Regulatory Commission (NRC) staff review 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.

Nuclear Power Plant Pump and Valve, and Snubber Inservice Testing Issues

INTRODUCTION

The NRC staff has encountered a number of pump and valve inservice testing (IST), and snubber inservice inspection (ISI) and testing issues since the Ninth NRC/ASME Symposium on Pumps, Valves and Inservice Testing in 2006. This paper discusses (1) smooth-running pump vibration issues; (2) safety relief valve testing frequency issues; (3) check valve sample disassembly and inspection online; and (4) use of Title 10 of the *Code of Federal Regulations* (10 CFR) 50.59 process to change an alternative to the Code requirements authorized by the NRC. The paper discusses the relief requests received related to these issues and the associated NRC safety evaluations of the requests. Some current staff positions and actions in these areas are also discussed. This discussion includes information that could have generic applicability in the implementation of effective IST programs at U.S. nuclear power plants.

SMOOTH-RUNNING PUMP ISSUES

The American Society of Mechanical Engineers (ASME) *Code for Operation and Maintenance of Nuclear Power Plants (OM Code)* - 2001 Edition with 2003 Addenda paragraphs ISTB-5121(e) and ISTB-5123(e) require that the measured vibration values shall be compared with the Code vibration acceptance criteria as specified in Table ISTB-5100-1, Table ISTB-5200-1, or Table ISTB-5300-1, as applicable, to determine if the measured values are acceptable during the performance of Code required pump testing.

Table ISTB-5100-1, Table ISTB-5200-1, and Table ISTB-5300-1 require that, if during an inservice test, a vibration measurement exceeds 2.5 times the previously established reference value (V_r), the pump be considered in the alert range. The frequency of testing is then doubled in accordance with paragraph ISTB-6200(a), “Alert Range Criteria,” until the cause of the deviation is determined and the condition is corrected and the vibration level returns below the alert range. Pumps, whose vibration is recorded to be greater than 6 times $V_{r,}$ are considered in the required action range and must be declared inoperable until the cause of the deviation has been determined and the condition is corrected.

For pumps whose absolute magnitude of vibration is an order of magnitude below the absolute vibration limits in Table ISTB-5100-1, Table ISTB-5200-1, or Table ISTB-5300-1, a relatively small increase in vibration magnitude may cause the pump to enter the alert or required action range. These instances may be attributed to variation in flow, instrument accuracy, or other noise sources that would not be associated with degradation of the pump. Pumps which have vibration reference values (V_r) below 0.05 inch/second are referred to informally as “smooth-running” pumps. Based on a small acceptable range, a smooth-running pump could be subjected to unnecessary corrective action.

The NRC staff has authorized various relief requests related to smooth-running pumps on a case by case basis. At one plant, a safety-related pump with an approved smooth-running pump alternative experienced a bearing failure that was not detected by the inservice testing program, but was detected through enhanced vibration monitoring as part of the plant predictive maintenance (PdM) program. The periodic monitoring had noted an increasing upward trend in vibration that was below the alert range allowed by the approved alternative. If the Code required vibration limits had been utilized, corrective action would have been required. After this event, it was clear to the NRC staff that a simple minimum reference value method alone would not be sufficient to identify degradation of a smooth-running pump. All relief requests related to smooth-running pumps must include a minimum reference (0.05 in/sec) value and a commitment to monitor all pumps in the IST programs with a PdM Program. The PdM program may include bearing temperature trending, oil sampling and analysis, and thermographic analysis, but must include enhanced vibration monitoring. The objective of the licensee's PdM program should be to detect problems involving the mechanical condition of the pump in advance of when the pump reaches its overall vibration alert limit.

One licensee's relief request proposal for smooth-running pumps combined the minimum reference value (≤ 0.05 in/sec) method with a commitment to monitor the IST pump with a PdM program. The licensee proposal to use a PdM program for smooth-running pumps is considered important to safe and reliable plant operation in addition to IST requirements. The program includes periodic vibration monitoring, bearing temperature trending, oil sampling and analysis, and thermographic analysis. The licensee also committed that, if the measured parameters are outside the normal operating range or are determined by analysis to be trending towards an unacceptable degraded state, appropriate actions would be taken. These actions include increased monitoring to establish the rate of degradation, review of component-specific information to identify cause, and removal of the pump from service to perform maintenance. The proposed alternative is consistent with the objective of IST, which is to monitor degradation in safety-related components and take appropriate corrective actions to ensure operational readiness. The NRC staff found that the alert and required action limits specified in the relief request sufficiently addressed the previously undetected acute pump problems. The objective of the licensee's PdM program is to detect problems involving the mechanical condition, even well in advance of when the pump reaches its overall vibration alert limit. Therefore, the licensee's proposed alternative will provide an acceptable level of quality and safety.

Recently, the NRC staff has learned that some licensees are planning to request generic relief from Code vibration limit requirements for all pumps in the IST program should they later be determined to be smooth-running. That is, to request pre-approval to relax vibration limits for certain pumps in the event they are found to be smooth-running prior to such determination actually being made. The NRC staff is not in the position to authorize generic relief for pumps that may or may not meet the definition of smooth-running pumps. Licensees are requested to submit component specific relief requests for each smooth-running pump separately, and provide detailed justification for the proposed alternative to the Code required testing, including enhanced vibration monitoring. The NRC staff does not consider generic relief to implement smooth-running criteria appropriate, and will only consider relief requests on a case by case basis.

FREQUENCY OF SAFETY RELIEF VALVE (SRV) INSERVICE TESTING

The ASME OM Code - 2001 Edition with 2003 addenda, Appendix I, Test Frequencies, Class 1 Pressure Relief Valves, paragraph I-1320, requires that Class 1 pressure relief valves be tested at least once every 5 years.

ASME/ANSI OM Part 1 - 1989 Edition, Section 1.3.3, Test Frequencies, Class 1 Pressure Relief Devices, requires that Class 1 pressure relief valves be tested at least once every 5 years. No maximum limit is specified for the number of valves to be tested within each interval; however, a minimum of 20 percent of the valves from each valve group are required to be tested within any 24-month interval. The test interval for any individual valve must not exceed 5 years.

ASME Code Interpretation 01-18, "ASME OM Code-1995 with OMa Code-1996 Addenda, Appendix I," dated June 26, 2003, clarifies the start of the 5-year test interval. The ASME OM Code Committee position is that the 5-year test interval starts when the Class 1 valve is tested, not when the valve is installed.

Recently NRC has learned that some licensees have used the valve installation date rather than the last test date to start their 5-year test interval. Various licensees have submitted relief requests to extend the SRV 5 year test interval after discovering that test schedules based on the date the valve was installed in the plant, rather than the date of the test, would result in exceeding the 5 year test interval specified by the Code. The NRC staff has reviewed justifications for relief and authorized relief on a case by case basis. Licensees need to demonstrate that the extended period of testing beyond 5 years will not impair the valves' operational readiness. The licensee basis for extension shall include as a minimum that (1) SRVs are routinely refurbished, which provides reasonable assurance that setpoint drift is minimum; (2) SRVs are stored in a controlled environment, which has minimal effect on the set point; and (3) past performance of SRVs for meeting the set point acceptance criteria demonstrates reliable performance.

Licensees are requested to carefully examine the interval or frequency of the inservice testing of SRVs, keeping in mind the ASME Code requirement that Class 1 valve be tested at least once every 5 years. This means that the SRV test-to-test interval shall not exceed 5 years. Licensees must also ensure that plant Technical Specifications related to SRVs testing are met, even if relief from Code requirements is requested.

CHECK VALVE SAMPLE DISASSEMBLY AND INSPECTION ONLINE

Subsection ISTC of ASME/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 applications. 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 Programs."

Paragraph ISTC-3510 states that check valves shall be exercised nominally every 3 months and 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. The 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 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.

USE OF 10 CFR 50.59 PROCESSES TO CHANGE THE NRC AUTHORIZED RELIEF REQUEST RELATED TO INSERVICE INSPECTION & TESTING

The 10 CFR 50.55a, "Code and Standards," defines the requirements for applying industry codes and standards to boiling- or pressurized-water-cooled nuclear power facilities. Each of these facilities is subject to the conditions in paragraph (a), (f), and (g) of 10 CFR 50.55a, as they relate to inservice inspection (ISI) and inservice testing (IST).

10 CFR 50.55a, requires that inservice testing (IST) of ASME Code Class 1, 2, and 3 pumps and valves be performed in accordance with the ASME OM Code and applicable addenda, and inservice inspection (ISI) of ASME Code Class 1, 2, and 3 components (including snubbers) shall be performed in accordance with Section XI, "Rules for Inservice Inspection of Nuclear Power Plant Components," of the ASME Code and applicable addenda, except where alternatives have been authorized or relief has been requested by the licensee and granted by the Commission pursuant to Sections (a)(3)(i), (a)(3)(ii), (f)(6)(i) or (g)(6)(i) of 10 CFR 50.55a.

10 CFR 50.59 requires that licensees (1) evaluate proposed changes to their facilities for their effects on the licensing basis of the plant, as described in the Final Safety Analysis Report (as updated), and (2) obtain prior NRC approval for changes that meet specified criteria as having a potential impact upon the basis for issuance of the operating license.

In accordance with 10 CFR 50.55a, NRC has approved alternatives and granted numerous reliefs from the ASME Code requirements. Once relief is granted the relief request's alternative becomes a part of the licensee's IST or ISI programs and regulatory requirements. Therefore, changing from one alternative to another alternative would require NRC approval. In no case, licensees should use 10 CFR 50.59 process to supersede or overwrite a previously authorized relief request, since 10 CFR 50.55a requires these alternatives to ASME Code requirements be authorized by the NRC.

Following the issuance of 10 CFR 50.36, many licensees created a Technical Requirement Manual (TRM) (other names are used by some licensees) to control certain provisions relocated from the Technical Specifications (TS). As a result, most licensees relocated snubber examination and testing requirements from the TS to the TRM. The TRM requirements are controlled using the criteria in 10 CFR 50.59. In the case of snubber inservice examination and testing, NRC has authorized the use of the TRM snubber examination and testing requirements in lieu of the ASME Code requirements at numerous operating plants.

Recently NRC has learned that at least in one case, the licensee used the 10 CFR 50.59 process to revise the snubber inservice examination and testing requirements of the TRM. The requirements contained in this TRM were approved by NRC to be used as an alternative to the ASME Code requirements. The use of an alternative as authorized by the NRC becomes a regulatory requirement; thus changes to these requirements must be reviewed and approved by the NRC staff pursuant to 10 CFR 50.55a(a)(3) or as an exemption pursuant to 10 CFR 50.12.

Nuclear Energy Institute Procedure, NEI-96-07, Revision 1, "Guidelines for 10 CFR 50.59 Implementation," states that the licensees' activities which are controlled by regulation 10 CFR 50.55a, take precedence over 10 CFR 50.59. NEI-96-07, Revision 1 is endorsed by Regulatory Guide (RG)1.187. Similarly, RG 1.187, Section D, Implementation, states that 10 CFR 50.59 can not be used in those cases in which a licensee proposes an acceptable alternative method for complying with the specified portion of the NRC's regulation. The licensees are encouraged to use caution when revising or changing program or procedures referenced in an approved relief request. Any change or update that supersedes or overwrites an alternative authorized in a relief request must be approved by the NRC unless the requirements of the ASME Code can be met. Utilization of the 50.59 process to change the requirements of an approved relief request is not appropriate.

CONCLUSION

The purpose of this paper is to make licensees aware of a number of pump, valve, and snubber issues that the staff has encountered since the Ninth NRC/ASME Symposium on Pump, Valve and Inservice Testing in 2006. 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:

NUREG/CP-0152, Vol. 5, “Proceedings of the Ninth NRC/ASME Symposium on Valve and Pump Testing,” July 2006.

NUREG/CR-6396, “Examples, Clarifications, and Guidance on Preparing Requests for Relief from Pump and Valve Inservice Testing Requirements.”

NUREG-1482, “Guidelines for Inservice Testing at Nuclear Power Plants.”

GL 89-04, “Guidance on Developing Acceptable Inservice Testing Programs.”

10 CFR 50.55a, “Codes and standards.”

Regulatory Guide 1.192, “Operation and Maintenance Code Case Acceptability, ASME OM Code.”

Nuclear Energy Institute Procedure, NEI 96-07, Revision 1, “Guidelines for 10 CFR 50.59 Implementation.”

Regulatory Guide 1.187, “Guidance for Implementation of 10 CFR 50.59, Changes, Tests, and Experiments.”

ASME/ANSI, *Code for Operation and Maintenance of Nuclear Power Plants*, 2001 Edition and 2003 Addenda:

Subsection ISTB, “Inservice Testing of Pumps in Light-Water Reactor Power Plants.”

Subsection ISTC, “Inservice Testing of Valves in Light-Water Reactor Power Plants.”

Subsection ISTD, “Inservice Examination & Testing of Snubbers in Light-Water Reactor Power Plants.”

Session 2(b): IST II

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Status of Regulatory Activities for Early Site Permits, Design Certifications, and Combined Licenses for New Nuclear Power Plants in the United States

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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.

Abstract

The nuclear industry is preparing for the licensing and construction of new nuclear power plants in the United States. Nuclear reactor vendors have requested and received certification of specific nuclear power plant designs by the U.S. Nuclear Regulatory Commission (NRC). The NRC has updated its regulations and guidance documents to evaluate additional applications for the certification of reactor designs and for combined licenses (COLs) to construct and operate new nuclear power plants. This paper summarizes actions taken by the NRC staff in its review of early site permits, design certifications, and COL applications, and provides a status of the NRC's review of these applications.

I. Introduction

In the past few years, the nuclear industry has initiated many activities in preparation for the future licensing and construction of new nuclear power plants in the United States. Nuclear reactor vendors have requested and received certification of several standard nuclear power plant designs by the U.S. Nuclear Regulatory Commission (NRC) through the rulemaking process of Part 52, "Licenses, Certifications, and Approvals for Nuclear Power Plants," in Title 10 of the *Code of Federal Regulations* (10 CFR Part 52). The NRC is currently reviewing further requests to certify additional standard nuclear power plant designs including two standard plant designs submitted by foreign vendors. At the same time, the NRC has also received an initial wave of applications for combined licenses (COLs) to construct and operate new nuclear power plants for the first time under NRC's newly amended regulations in 10 CFR Part 52.

II. NRC Regulations for New Reactors

The NRC regulations in 10 CFR Part 52, “Licenses, Certifications, and Approvals for Nuclear Power Plants,” provide a process for an efficient and effective licensing review of new nuclear power plants in the United States. The NRC issued 10 CFR Part 52 on April 18, 1989 (54 FR 15372), to reform the NRC’s licensing process for future nuclear power plants. The rule added alternative licensing processes in 10 CFR Part 52 for early site permits, standard design certifications, and combined licenses. The intent of the alternative processes in 10 CFR Part 52 was to allow for early resolution of safety and environmental issues in the licensing process while providing enhanced safety and reliability of nuclear power plants through standardization and, thus, be consistent with the intent of Congress in enacting Section 185b of the Atomic Energy Act, as amended, and the Energy Policy Act of 1992. Using the process of Subpart B of 10 CFR Part 52, the NRC certified four standard nuclear power plant designs during the period from 1997 to 2006.

After issuing the four design certifications, the NRC planned to update 10 CFR Part 52 using insights gained from the implementation of the design certification process (Subpart B). In the meantime, the NRC issued three early site permits using the process of Subpart A of 10 CFR Part 52 and gained additional insights into the implementation of the Part 52 licensing process for early site permits. Utilizing insights gained from the early site permit experience and recognizing the need to address the relationship between the Part 50 and Part 52 licensing processes, the NRC decided that a substantial rewrite of 10 CFR Part 52 and Part 50 as well as other affected regulations in Title 10 was necessary. On August 28, 2007, the NRC issued its final rule (72 FR 49352) amending its regulations applicable to the licensing and approval processes for nuclear power plants. These amendments clarified the applicability of various requirements to each of the licensing processes by making necessary conforming amendments throughout the NRC’s regulations in Title 10. The amended rule became effective on September 27, 2007.

Applicants for constructing and operating a nuclear power plant may also continue to follow the licensing process in 10 CFR Part 50. However, it is anticipated that most nuclear power plant applicants will follow the process established in 10 CFR Part 52.

Part 52 consists of several subparts that specify rules for (1) Early Site Permits (ESPs), (2) Standard Design Certifications, (3) Combined Licenses, (4) Standard Design Approvals, and (5) Manufacturing Licenses. These are summarized below.

An ESP is defined in Part 52 as a Commission approval for a site or sites for one or more nuclear power facilities. An ESP is a partial construction permit. An ESP allows a nuclear power plant applicant to initiate limited work at a site or sites for one or more nuclear power facilities. Although an ESP applicant does not need to specify a particular nuclear plant design, as in construction permit applications, it does need to provide sufficient surrogate design information (developed to bound the nuclear plant design or designs that are being considered by the applicant) so that the NRC can make a determination on the acceptability of the site and the environmental impacts, and determine whether designs bounded by the surrogate design information provided by the applicant can be qualified for the proposed site. An ESP application is subject to the procedural requirements in 10 CFR Part 2, including requirements for a mandatory hearing.

A standard design certification is issued under the rulemaking process described in Part 52 for Commission approval of a final standard design for a nuclear power facility. The design certification application is required by 10 CFR 52.47(b)(1) to contain proposed inspections, tests, analyses, and acceptance criteria (ITAAC) that are necessary and sufficient to provide reasonable assurance that the facility has been constructed and will be operated in conformance with the design certification and the Commission’s rules and regulations. Because design certifications are issued using the rulemaking process, public participation is achieved through public comments on the proposed rule. There is no mandatory hearing for design certifications, although the Commission may determine that a legislative hearing on a design certification application be held at its discretion.

A combined license (COL) is a "one-step" combined construction permit and operating license with conditions issued under Part 52 for a nuclear power facility. A COL applicant may, but need not, reference an ESP, a standard design certification, standard design approval, or a manufacturing license issued under Part 52. The COL application is required by 10 CFR 52.80(a) to contain proposed ITAAC that are necessary and sufficient to provide reasonable assurance that the facility has been constructed and will be operated in conformance with the Commission's rules and regulations, including those related to a referenced ESP or design certification. A COL application is subject to the procedural requirements in 10 CFR Part 2, including requirements for a mandatory hearing to allow public participation in the licensing process.

A Standard Design Approval is an NRC staff approval issued under Part 52 of a final standard design for a nuclear power reactor. The approval may be for either the final design for an entire reactor facility or the final design of its major portions. As stated in 10 CFR 52.143, the NRC staff will publish a determination in the *Federal Register* as to whether or not the design is acceptable, subject to appropriate terms and conditions, and will make a report on its analysis of the design available on the NRC website.

A Manufacturing License is a license issued under Part 52 authorizing the manufacture of nuclear power reactors but not their construction, installation, or operation at the sites on which the reactors are to be operated. A Manufacturing License application is subject to the procedural requirements in 10 CFR Part 2, including requirements for issuance of a notice of hearing.

The NRC intends that the reactor licensing process will maintain the safety of licensed plants, provide a predictable licensing process, ensure meaningful public participation, enhance the safety of future plants, and establish an independent and credible regulatory process.

III. Acceptance Reviews of New Reactor Applications

The NRC staff performs an acceptance review of a design certification or COL application prior to docketing the application and initiating a detailed technical review. The staff has prepared NRO Office Instruction NRO-REG-100, "Acceptance Review Process for Design Certification and Combined License Applications," to provide guidance to the NRC staff for conducting the acceptance review. As part of the acceptance review, the staff evaluates both the completeness and technical sufficiency of the application to determine whether the planned schedule for performing a detailed technical review can be achieved.

The NRC staff typically interacts with the design certification or COL applicant during the acceptance review process to ensure a clear understanding of the application. The staff determines whether the review depends on the completion of a parallel review (such as completion of the review of a related topical report). Following the acceptance review, the staff prepares a letter to the applicant indicating the results of the review and specific areas that might need additional attention if the application is accepted for docketing.

IV. Regulatory Guidance for Preparing COL Applications

The NRC staff prepared Regulatory Guide (RG) 1.206, "Combined License Applications for Nuclear Power Plants (LWR Edition)," dated June 2007 to provide guidance to potential applicants to use in preparing COL applications for the construction and operation of nuclear power plants. As stated in the Supplementary Information provided in the *Federal Register* notice describing the revision to 10 CFR Part 52 (72 FR 49352, 49387), the NRC does not require applicants to evaluate their facility against RG 1.206. However, the NRC believes that RG 1.206 can provide useful guidance to COL applicants in preparing their applications and that use of this guidance will facilitate the NRC's review.

The regulatory positions in Section C of RG 1.206 are divided into four parts. Part I of Section C in RG 1.206 addresses information requirements specified in 10 CFR 52.79 with guidance for applicants not referencing a certified design. Part II of Section C in RG 1.206 addresses information requirements specified in 10 CFR 52.80, including ITAAC and the environmental report. Part III of Section C in RG 1.206 provides guidance for COL applicants who reference either a certified design or both a certified design and an ESP. Part IV of Section C in RG 1.206 addresses miscellaneous topics for COL applicants including, for example, an acceptance review checklist, COL application format, change processes, operational programs, and regulatory treatment of nonsafety systems.

The guidance in RG 1.206 is intended to minimize requests for additional information, streamline the review process, and reduce inspection activity. With respect to operational programs (such as inservice testing, motor-operated valve testing, and environmental qualification), RG 1.206 provides one method for applicants to fully describe their operational programs to satisfy Commission policy discussed in Commission paper SECY-05-0197, "Review of Operational Programs in a Combined License Application and General Emergency Planning Inspections, Tests, Analyses, and Acceptance Criteria."

V. Guidance for NRC Staff's Technical Review of Design Certification and COL Applications

The NRC issued Standard Review Plan (SRP) NUREG-0800, "Review of Safety Analysis Reports for Nuclear Power Plants," to establish areas of review, acceptance criteria, and review procedures that the NRC staff uses in its review of light-water, nuclear power plant designs to ensure that NRC regulations are met. The SRP is not a substitute for the NRC regulations and compliance with it is not required. However, an applicant needs to identify differences between the design features, analytical techniques, and procedural measures proposed for its facility and the SRP acceptance criteria. The applicant also needs to evaluate how the proposed alternatives to the SRP acceptance criteria provide an acceptable method of complying with the NRC regulations. The regulations in 10 CFR 52.47(a)(9) and 52.79(a)(41) specifically require that an applicant for a design certification or COL, respectively, evaluate the facility against the SRP revision in effect 6 months before the docket date of the application.

The NRC issued Revision 3 to the SRP in March 2007 to provide updated guidance for NRC staff reviewing applications for design certifications and COLs. The staff included lessons learned from licensing reviews and operating experience related to current nuclear power plants. For example, the NRC staff revised SRP Section 3.9.6, "Functional Design, Qualification, and Inservice Testing Programs for Pumps, Valves, and Dynamic Restraints," based on lessons learned from valve operating experience and testing programs from operating nuclear power plants in the U.S.

VI. Licensing Status of Early Site Permits

Under Subpart A of 10 CFR Part 52, the NRC has issued ESPs for three sites—the Clinton site in Illinois, the Grand Gulf site in Mississippi, and the North Anna site in Virginia. The NRC staff is currently reviewing an ESP application for the Vogtle site in Georgia.

It should be noted that the four ESPs submitted to the NRC to date have been for sites on which operating nuclear power plants currently exist. The NRC has not yet received an ESP application for a "green-field" site (i.e., a new site with no nuclear plant built on it).

VII. Licensing Status of Design Certifications

Under Subpart B of 10 CFR Part 52, the NRC has certified standard nuclear power plant designs for General Electric's (GE's) evolutionary Advanced Boiling Water Reactor (ABWR), ABB Combustion Engineering's evolutionary System 80+ pressurized water reactor (PWR), Westinghouse's passive AP600 PWR, and Westinghouse's passive AP1000 PWR. These four standard plant designs are codified in Appendices A, B, C, and D of 10 CFR Part 52 for the ABWR, System 80+, AP600, and AP1000, respectively.

The NRC is currently reviewing, for the first time, a proposed amended design certification application for the AP1000 certified design. The NRC is also reviewing a design certification application for the General Electric-Hitachi (GEH) Economic Simplified Boiling Water Reactor (ESBWR). This is the first passive BWR standard plant design submitted to the NRC for design certification. Finally, the NRC staff has initiated its technical review of two foreign design certification applications. The first is AREVA's Evolutionary Power Reactor (EPR)—an evolutionary PWR design from France. The second is Mitsubishi Heavy Industries' U.S. Advanced Pressurized Water Reactor (US-APWR)—an evolutionary PWR design from Japan.

VIII. Licensing Status of COL Applications

The COL application represents a combined construction permit and operating license for a nuclear power plant. The objective of this "one-step" COL application process is to resolve all safety and environmental issues before major construction of the nuclear power plant begins. Prior to fuel load, the licensee (also referred to as the "COL holder" before the plant begins commercial operation) must verify that the facility was constructed in accordance with the license and NRC regulations.

The initial COL application that references a certified design is referred to as the "reference-COL" (R-COL) application. Subsequent-COL applications are referred to as "S-COLs" and will essentially mimic the R-COL applications, thus, requiring less NRC staff review time and effort where issues have been resolved during the R-COL application review. The NRC staff has incorporated this reduced resource obligation in the review schedule of all S-COLs.

To date, the NRC staff has received several COL applications under the Part 52 process. As of April 2008, the submitted COL applications for new nuclear power plants include:

Calvert Cliffs Unit 3 EPR (R-COL)
South Texas Project Units 3 and 4 ABWR (R-COL)
Bellefonte Units 3 and 4 AP1000 (R-COL)
North Anna Unit 3 ESBWR (R-COL)
William S. Lee Units 1 and 2 AP1000 (S-COL)
Shearon Harris Units 2 and 3 AP1000 (S-COL)
Grand Gulf Unit 3 ESBWR (S-COL).

IX. NRC Inspection Program

A new construction inspection program (CIP) is being developed for plants licensed in accordance with the requirements of 10 CFR Part 52. The introduction of ITAAC into the Part 52 licensing process creates a design-specific, pre-approved set of performance standards that the licensee must meet and that the Commission must find have been met, before the licensee can load fuel and operate the plant. Therefore, the major focus of the CIP is on the licensee work being performed to support the completion of the ITAAC. However, additional inspections of quality assurance verification activities and operational programs will be needed to provide assurance that these activities and programs are in compliance with NRC requirements. The CIP is being developed to address the specific needs associated with verifying the successful completion of the ITAAC as well as to incorporate lessons learned from previous NRC construction inspections.

The NRC staff has described the new CIP in several Commission policy papers. For example, an overview of the CIP is provided in Commission paper SECY-06-0014 (May 13, 2006), "Description of the Construction Inspection Program for Plants Licensed under 10 CFR Part 52." Commission paper SECY-07-0047 (March 8, 2007), "Staff Approach to Verifying the Closure of Inspections, Tests, Analyses, and Acceptance Criteria through a Sample-Based Inspection Program," indicates the staff's approach to verifying the closure of ITAAC through a sample-based inspection program. Commission paper SECY-07-0049 (March 8, 2007), “Construction Inspection Roles and Responsibilities,” describes the roles and responsibilities of NRC’s Headquarters and Region II construction inspection staff.

X. Conclusion

The nuclear industry (both domestic and foreign) and NRC staff are currently fully engaged in design and licensing activities for new nuclear power plants in the United States and are preparing for plant construction in the next few years. The NRC is currently reviewing many early site permit, design certification, and COL applications for new standardized nuclear power plants under 10 CFR Part 52. The NRC staff has been updating its regulations, regulatory guidance, and inspection procedures in preparation for this initial wave of new nuclear power plants. The NRC staff intends to perform its review of early site permit, design certification, and COL applications in an efficient and effective manner in order to make timely safety determinations for the certification and licensing of new reactors in the United States.

Regulatory Issues Related to Inservice Testing Programs Under 10 CFR Part 52

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Abstract

The U.S. Nuclear Regulatory Commission (NRC) reviews information related to inservice testing (IST) and motor-operated valve (MOV) testing programs provided by applicants for Design Certifications and Combined Licenses (COLs) under Part 52, "Licenses, Certifications, and Approvals for Nuclear Power Plants," in Title 10 of the *Code of Federal Regulations* (10 CFR Part 52). This paper discusses the regulatory requirements in 10 CFR Part 52, and guidance in Commission papers, the NRC Standard Review Plan, and Regulatory Guide 1.206, "Combined License Applications for Nuclear Power Plants (LWR Edition)," regarding IST and MOV testing operational programs for new nuclear power plants.

I. Introduction

The nuclear industry is preparing for the licensing and construction of new nuclear power plants in the United States. The U.S. Nuclear Regulatory Commission (NRC) reviews information provided by applicants related to inservice testing (IST) and motor-operated valve (MOV) testing programs for Design Certifications and Combined Licenses (COLs) under Part 52, "Licenses, Certifications, and Approvals for Nuclear Power Plants," in Title 10 of the *Code of Federal Regulations* (10 CFR Part 52). The NRC is currently reviewing requests for Design Certification of new nuclear reactors and COL applications under 10 CFR Part 52. As part of this review, the NRC must reach a safety conclusion regarding IST and MOV testing programs for Design Certification and COL applications.

II. NRC Regulations

The NRC regulations in 10 CFR Part 52 provide a process for licensing of new nuclear power plants in the United States. Part 52 specifies rules for (1) Early Site Permits (ESPs), (2) Standard Design Certifications, (3) COLs, (4) Standard Design Approvals, and (5) Manufacturing Licenses. Recently, the NRC revised Part 52 to improve the process for Design Certification and COL reviews. It is anticipated that most applicants for the construction and licensing of new nuclear power plants will follow the process established in 10 CFR Part 52.

ENCLOSURE 2

The NRC regulations in 10 CFR 52.47(a)(9) require that Design Certification applications include an evaluation of the standard plant design against the Standard Review Plan (SRP) revision in effect 6 months before the docket date of the application. Regarding operating experience more recent than the SRP revision to be applied, the NRC regulations in 10 CFR 52.47(a)(22) require that Design Certification applications include information necessary to demonstrate how operating experience insights have been incorporated into the plant design.

The NRC regulations in 10 CFR 52.79(a)(11) require a COL applicant to provide in its safety analysis report, at a level sufficient to enable the NRC to reach a final conclusion on all safety matters that must be resolved before COL issuance, a description of the programs and their implementation necessary to ensure that the systems and components meet the requirements of the American Society of Mechanical Engineers (ASME) *Boiler & Pressure Vessel Code* (BPV Code) and the ASME *Code for Operation and Maintenance of Nuclear Power Plants* (OM Code) in accordance with 10 CFR 50.55a. The NRC regulations in 10 CFR 52.79(a)(41) require that COL applications include an evaluation of the standard plant design against the SRP revision in effect 6 months before the docket date of the application. In addition, the NRC regulations in 10 CFR 52.79(a)(37) require that COL applications include information necessary to demonstrate how operating experience insights have been incorporated into the plant design. Regulatory Guide (RG) 1.206, "Combined License Applications for Nuclear Power Plants (LWR Edition)," discusses the consideration of operating experience for COL applications referencing a certified design in Paragraph C.III.1.9.4, "Operational Experience (Generic Communications)," on pages C.III.1-13 to 14.

The NRC regulations in 10 CFR 50.55a(f)(4)(i) state that inservice tests to verify operational readiness of pumps and valves, whose function is required for safety, conducted during the initial 120-month interval must comply with the requirements in the latest edition and addenda of the ASME Code incorporated by reference in 10 CFR 50.55a(b) on the date 12 months before the date scheduled for initial fuel loading under a COL issued per 10 CFR Part 52 (or the optional ASME Code cases listed in NRC Regulatory Guide 1.192), subject to the limitations and modifications listed in Section 50.55a.

III. Operational Programs

The focus of the NRC review of operational program information is different for Design Certifications and COLs. A Design Control Document (DCD) supporting a Design Certification provides general information on operational programs with allowance for flexibility by the COL applicant when developing plant-specific operational programs. The COL application needs to support an NRC decision that the operational programs at the proposed nuclear power plant will provide reasonable assurance of the safe operation of the plant.

With respect to IST and MOV testing operational programs, the NRC staff review of the DCD for a Design Certification focuses on accessibility for the performance of IST activities. The staff also reviews the general description of the operational programs in the DCD to ensure an adequate foundation for the plant-specific operational programs to be developed by COL applicants. The DCD for a Design Certification may indicate that the COL applicant will develop the IST and MOV testing program.

The Supplementary Information provided with the revision to 10 CFR Part 52 in the *Federal Register* (72 FR 49352, 49385-49387) discusses the requirement in 10 CFR 52.79(a) for descriptions of operational programs that need to be included in the Final Safety Analysis Report (FSAR) to allow a reasonable assurance finding of acceptability. The FR notice discusses the information to be provided in COL applications for operational programs where ITAAC are not necessary. This information is summarized below.

COL applicants should fully describe IST, MOV testing, and other operational programs as defined in Commission Paper SECY-05-197, "Review of Operational Programs in a Combined License Application and General Emergency Planning Inspections, Tests, Analyses, and Acceptance Criteria

[ITAAC]," to avoid the need for ITAAC for the implementation of those programs. Commission paper SECY-05-0197 defines operational programs for new nuclear power plants as programs that are required by regulation, are reviewed by NRC staff for acceptability with the results documented in the safety evaluation report, and will be verified for implementation by NRC inspectors. Examples of operational programs related to pumps, valves, and dynamic restraints include preservice testing, IST, MOV testing, and environmental qualification programs.

In a Staff Requirements Memorandum (SRM) dated September 11, 2002, to SECY-02-0067, "Inspections, Tests, Analyses, and Acceptance Criteria for Operational Programs (Programmatic ITAAC)," the Commission provided direction to the NRC staff that a COL applicant is not required to have ITAAC for an operational program with the exception of emergency planning. The Commission stated that 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 staff at the COL stage. The Commission noted that the burden is on the applicant to provide the necessary and sufficient programmatic information for approval of the COL without ITAAC.

In an SRM dated May 14, 2004, to SECY-04-0032, "Programmatic Information Needed for Approval of a Combined License Without Inspections, Tests, Analyses, and Acceptance Criteria," the Commission stated that "fully described" for an operational program should be understood to mean that the program is clearly and sufficiently described in terms for scope and level of detail to allow a reasonable assurance finding of acceptability. The Commission noted that 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.

In SECY-05-0197, the NRC staff concluded that operational programs could be fully described in a COL application and recognized that operational programs, such as the IST program, would not be available at the time of COL issuance. The description of the program would contain the information necessary for the staff to make a reasonable assurance finding on the acceptability of the operational program in the review of a COL application. The staff also proposed license conditions to provide certainty for the NRC as to when the operational programs will be implemented.

With the requirement in 10 CFR 50.55a(f)(4)(i) that the IST program comply with the Code edition and addenda incorporated by reference in the NRC regulations 12 months before fuel load, a COL applicant may describe the IST and MOV testing operational programs based on the ASME Code edition and addenda applicable at the time of submittal of the COL application. The COL applicant could then provide milestones for developing and implementing the final operational programs. The milestones should allow sufficient time for NRC inspections to review the operational programs prior to relying on equipment to perform their safety functions. The NRC staff is developing inspection procedures for IST and MOV programs to provide guidance for NRC inspectors to review those programs at nuclear power plants licensed under 10 CFR Part 52.

IV. Regulatory Guide 1.206

In early 2007, the NRC staff completed Regulatory Guide 1.206, "Combined License Applications for Nuclear Power Plants (LWR Edition)," to provide guidance for COL applicants to use in preparing their applications. As noted in the Supplementary Information provided in the *Federal Register* notice for the revision to 10 CFR Part 52 (72 FR 49352, 49387), the NRC does not require applicants to evaluate their facility against RG 1.206. However, the NRC believes that RG 1.206 can provide useful guidance to COL applicants in preparing their applications and that use of this guidance will facilitate the NRC's review.

In Paragraph C.I.3.9.6 of RG 1.206, the NRC staff provides guidance for the submittal of COL application information on functional design, qualification, and IST programs for pumps, valves, and dynamic restraints in plants not referencing a certified design. In Paragraph C.III.3.9.6 of RG 1.206, the staff provides such guidance for a COL applicant referencing a certified design. The guidance is intended to minimize requests for additional information, streamline the review process, and reduce inspection activity. Several highlights of RG 1.206 are discussed below.

The success of an IST program depends on the adequate design and qualification of pumps, valves, and dynamic restraints. In the 1980s, operating experience at current nuclear power plants raised concerns regarding the capability of MOVs to perform their safety functions under design-basis conditions. In response to those concerns, the NRC issued Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance," to request that nuclear power plant licensees develop programs to verify the design-basis capability of their safety-related MOVs through dynamic testing where practicable. Also in response to valve operating experience, the NRC issued GL 95-07, "Pressure Locking and Thermal Binding of Safety-Related Power-Operated Gate Valves," to request that nuclear power plant licensees verify that the design and application of their safety-related power-operated gate valves were adequate to avoid pressure locking and thermal binding that might prevent those valves from performing their safety functions. Testing and research in response to those generic letters by the nuclear industry and the NRC identified significant inadequacies in valve design, qualification, and testing. The NRC discussed the applicability of lessons learned from resolution of MOV concerns to other power-operated valves (POVs) in Regulatory Issue Summary (RIS) 2000-03, "Resolution of Generic Safety Issue 158, Performance of Safety-Related Power-Operated Valves Under Design-Basis Conditions," and Information Notice 96-48, "Motor-Operated Valve Performance Issues." In response to the valve qualification weaknesses, the American Society of Mechanical Engineers (ASME) revised ASME Standard QME-1, "Qualification of Active Mechanical Equipment used in Nuclear Power Plants," in 2007 to incorporate those lessons learned into the provisions for POV functional qualification. The updated ASME standard also provides guidance for the functional qualification of check and relief valves, pumps, and dynamic restraints. The COL applicant should provide information with respect to its proposed nuclear power plant to enable the NRC staff to evaluate the functional design and qualification of safety-related valves and dynamic restraints to determine whether the proposed nuclear power plant will meet the regulatory requirements to provide reasonable assurance that those components will be capable of performing their safety functions. RG 1.206 provides guidance in Paragraph C.III.3.9.6.1, "Functional Design and Qualification of Pumps, Valves, and Dynamic Restraints," for applicable information that could help during the NRC staff review of the COL application based on the acceptance criteria in SRP Section 3.9.6.

In light of the weaknesses in the IST provisions in the ASME BPV Code and OM Code for quarterly MOV stroke-time testing, the NRC issued GL 96-05, "Periodic Verification of Design-Basis Capability of Safety-Related Motor-Operated Valves," to request that nuclear power plant licensees establish programs to assure the capability of safety-related MOVs to perform their design-basis functions over the long term. In addition, the NRC revised 10 CFR 50.55a to require that nuclear power plant licensees supplement the MOV stroke-time testing specified in the ASME OM Code with a program to periodically verify the design-basis capability of safety-related MOVs. In light of past weaknesses in the IST provisions for MOVs at nuclear power plants, the COL applicant should provide information to enable the NRC staff to determine that a program will be established that assures the capability of safety-related MOVs to perform their design-basis functions over the long term. Paragraph C.III.3.9.6.3.1, "Inservice Testing Program for Motor-Operated Valves," of RG 1.206 provides guidance for information regarding the IST Program that periodically verifies the design-basis capability of safety-related MOVs that could help during the NRC staff review of the COL application based on the acceptance criteria in SRP Section 3.9.6.

Lessons learned from the resolution of weaknesses in the design, qualification, and testing of MOVs are applicable to other POVs used at nuclear power plants. In discussing the MOV lessons learned applicable to other POVs in RIS 2000-03, the NRC staff states that the regulations provide adequate requirements to ensure verification of the design-basis capability of safety-related POVs. Paragraph C.III.3.9.6.3.2, "Inservice Testing Program for Power-Operated Valves Other Than Motor-Operated Valves," in RG 1.206 provides guidance for information regarding the application of lessons learned to the IST program for safety-related POVs other than MOVs that could help during the NRC staff review of the COL application based on the acceptance criteria in SRP Section 3.9.6.

Paragraph C.III.3.9.6.5, "Relief Requests and Alternative Authorizations to ASME OM Code," in RG 1.206 provides guidance for information regarding those components for which the COL applicant is requesting relief from or an alternative to the ASME OM Code. The staff considers that the design of new plants should minimize the need for relief from the ASME OM Code with respect to such items as materials and geometry because efforts can be made at this early stage of the design to address most limitations encountered with the current operating fleet. However, the staff recognizes that the ASME OM Code might change and that a licensee might need to submit a request for relief in the future.

V. Standard Review Plan

The NRC regulations in 10 CFR 52.47(a)(9) and 52.79(a)(41) require that an applicant for a Design Certification or COL, respectively, evaluate the facility against the revision to Standard Review Plan (SRP) NUREG-0800, "Review of Safety Analysis Reports for Nuclear Power Plants," in effect 6 months before the docket date of the application. The NRC revised the SRP in March 2007 to provide updated guidance for NRC staff reviewing Design Certification and COL applications. The SRP revision incorporates experience from licensing reviews and operating experience related to current nuclear power plants.

Each SRP section establishes criteria that the NRC staff responsible for the review of applications to construct and operate nuclear power plants uses in evaluating whether an applicant meets the NRC regulations. The SRP is not a substitute for the NRC regulations and compliance with it is not required. However, an applicant needs to identify differences between the design features, analytical techniques, and procedural measures proposed for its facility and the SRP acceptance criteria. The applicant also needs to evaluate how the proposed alternatives to the SRP acceptance criteria provide an acceptable method of complying with the NRC regulations.

The revised SRP Section 3.9.6, "Functional Design, Qualification, and Inservice Testing Programs for Pumps, Valves, and Dynamic Restraints," incorporates lessons learned from operating experience at nuclear power plants for those components. This operating experience includes the results of MOV testing performed in response to GL 89-10 and GL 96-05, and research programs developed to resolve MOV performance issues. The guidance and acceptance criteria in SRP Section 3.9.6 for the staff review of COL applications are consistent with the guidance in RG 1.206 for the description of IST and MOV programs in COL applications.

VI. Conclusion

The nuclear industry is preparing for the design and construction of new nuclear power plants in the United States. The NRC staff has developed guidance for describing operational programs as part of COL applications in RG 1.206, and the review of the program descriptions in the updated SRP. Operational programs need to be clearly and sufficiently described in terms for scope and level of detail to allow a reasonable assurance finding of acceptability. The NRC staff will use the information provided by the applicants based on the acceptance criteria in the SRP to reach a safety conclusion on operational programs for Design Certification and COL applications.

ABSTRACT

FOURTH TEN-YEAR INTERVAL IST PROGRAM UPDATE

EDWIN I. HATCH NUCLEAR PLANT – UNITS 1 & 2

Dennis M. Swan, Southern Nuclear

Edwin I. Hatch Nuclear Plant went into commercial operation on January 1, 1976 followed by Unit 2 on September 5, 1979. Georgia Power Company received NRC approval to align the ISI/IST intervals for both units in 1986 along with the first program updates. Therefore, both units were updated for the fourth ten-year IST interval in 2005 utilizing the ASME OM Code, 2001 Edition through 2003 Addenda. This paper provides a summary of the major obstacles encountered during the update process.

ISSUES DISCUSSED

1. Early submittal of alternative for pump IST and NRC response.
2. Use of contractor to develop program update and reasons for doing such.
3. Software utilized for program update.
4. Relief Requests needed for update.
5. Initial submittal including request for Phased in Implementation.
    a. Use of NUREG-1482 for formal and content
    b. Use of NUREG-1482 to justify phased in implementation.
        i. 1-year to be in 100% compliance
        ii. Schedule for procedure updates and effective date
        iii. Consideration for outages
6. NRC Safety Evaluation Report (SER).
7. Procedure Updates and Implementation
    a. New Pump IST requirements.
        i. Grouping of pumps.
        ii. Group A IST.
        iii. Group B IST.
        iv. Comprehensive Pump Test.
        v. Preservice Pump Test.
        vi. Instrumentation issues.
    b. New Valve IST requirements.
        i. Additional scope for relief valves.
        ii. Bi-directional testing of check valves.
8. Lessons Learned
    a. Plan and schedule.
    b. Early communications with NRC.
    c. Dedication of personnel for procedure updates.
    d. Procedure templates.
    e. Unexpected issues.
    f. Partners in Crime.
9. Current Status and Open Issues

Title: Barriers to Implementing Snubber Examination and Testing Programs

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Mr. Shutt is a licensed professional engineer of over 27 years experience with Duke Energy Corporation. He is the snubber program manager for the Catawba Nuclear Station in York, SC and serves as the Duke Energy fleet subject matter expert with regard to snubbers. A past president of the Snubber Users Group (SNUG), he currently serves on the SNUG board of directors and is responsible for the planning of the users group conference agendas. Credits include primary authorship of a number of SNUG white papers and guidance documents, preparation and presentation of training materials, and interfacing with regulators on behalf of the users group on technical specification issues. Additional snubber related responsibilities include membership on the ASME Operating and Maintenance Code Subgroup ISTD (In-service Testing of Dynamic Restraints) and ASME Qualification of Mechanical Equipment Subcommittee on Qualification of Active Dynamic Restraints (QDR).

Abstract

As is the case with many programs in the nuclear power industry of the present day, there are significant barriers to the implementation of high quality snubber examination and testing programs. This is true on both a single plant level as well as from an industry-wide perspective. In an effort to reduce the burdens associated with these barriers, the Snubber Users Group (SNUG) has undertaken the task of identifying and addressing barriers facing the industry in this arena. This paper will outline the more significant of these barriers identified to date and provide information regarding actions being taken or recommended to address the concerns caused by them.

First of all, from an industry-wide perspective, one of the most obvious obstacles is the inconsistency in program requirements and the implementation of these requirements. For the most part, each plant has its own set of requirements and their own interpretation of how best to satisfy them. Snubber examination and test requirements originated as commitments in Technical Specifications and have evolved into a multitude of plant-specific Technical Specifications, owner-controlled documents, Operating and Maintenance Code requirements, and ASME Section XI relief requests. Whereas there are some general commonalities in the various program commitments, there are many more differences. Even among programs with similar commitments there are significant differences in how the specific requirements are interpreted and satisfied, largely due to plant-specific design basis assumptions. This makes it difficult to make meaningful comparisons between programs.

On the plant level there are also a number of barriers that render it difficult to plan, implement, and maintain a successful long term snubber program. Significant areas of concern include the loss of experienced personnel due to an aging workforce, high personnel turnover rates, limited training opportunities specific to snubber examination and testing programs, and a lack of resources – both financial and personnel related. Coupled with these obstacles is a general tendency on the part of plant management to relegate snubber program issues to a “second tier” status in terms of priority. The snubber issues of the early 1980’s that led to the current requirements were successfully addressed and there have been few new issues requiring significant resources to be focused on snubber programs. As a result, many snubber programs have become minimal in nature, thereby causing such programs to be incapable of detecting long term degradation in an aging component population.

This paper will identify significant issues and detail actions that the Snubber Users Group (SNUG) is taking to address some of these concerns. This includes plans for future activities and recommendations for removing or reducing obstacles on both an industry and plant level. Information and references provided will prove useful in increasing the effectiveness of snubber examination and testing programs.

Introduction

In order to effectively identify and address barriers affecting the implementation of high quality snubber programs, it is important to understand how and why snubber programs were established in the first place. Once this starting point is established it is equally important to understand how program requirements have changed over time. From an industry-wide perspective, one of the most cited concerns is the inconsistency in both program requirements and implementation practices from one plant to another. Many plants have requirements and practices unique to their program, or their own interpretation of how to best satisfy industry requirements. In large part, this is due to the evolution of the use of snubbers and how associated issues developed along with the increased usage. For this reason it is important to get a historical perspective on snubber usage and associated program requirements. With such a perspective established it is then possible to more clearly identify and understand those global issues affecting industry consistency with regard to implementation of snubber examination and testing programs.

After establishing that context, the more localized plant level barriers can then be defined. These barriers are more plant specific in detail, but often have a common cause that can be addressed more globally. The Snubber Users Group has sought to identify both industry-wide and plant level barriers and has both current and planned actions to address these barriers.

A Historical Perspective

In the early days of the nuclear power industry there were no clearly identified program requirements for snubbers. Early plant designs and older vintage plants had a very limited number of snubbers installed. Snubbers are generally utilized to provide seismic restraint of pipe and equipment while allowing for unrestricted movement due to thermal expansion. Older plants were initially designed using rather simplified seismic criteria that did not require extensive use of seismic pipe and equipment restraints. Over time the situation changed and the use of snubbers increased as plant design criteria itself evolved to be more seismic intensive. As more plants came into actual operation in the late 1960's and early 1970's, more and more snubbers were used in their design. There was a rush to get plants into commercial operation, and incorporating snubbers into the design allowed for quicker and less expensive analysis methods to be utilized. The use of snubbers made piping systems more flexible during normal operating conditions while still maintaining the rigidity needed for seismic design criteria. Large numbers of snubbers were used in many plants, numbering several thousand in some cases. By including large numbers of snubbers in the initial design, stress analysts were capable of executing what was at the time very expensive mainframe computer analysis with a high probability of satisfying the design criteria on the first iteration.

In the early stages of the industry, the few snubbers that were utilized were almost exclusively hydraulic devices which were simple in design and function. There was no significant performance history of snubbers in this application and no perceived concerns with their function. Examination and test requirements were not part of original plant design criteria as there were no identified issues related to these components in the early days of the nuclear power industry.

As more plants came on line and actual service time began to accrue, some snubber reliability issues began to emerge. Since most of the snubbers installed at this point were hydraulic, the initial concerns centered on snubber fluids and leakage. As a result of these concerns, regulators began to implement requirements for examination and testing of hydraulic snubbers in order to provide some degree of assurance with regard to reliability and operability. These initial requirements were fairly simple in nature and applied only to hydraulic snubbers.

When examination and test requirements were imposed on hydraulic snubbers, designers began to utilize mechanical snubbers in greater numbers in order to avoid both the requirements for testing and examination and the issues related to hydraulic snubber designs. This became a common practice as mechanical snubbers were effectively marketed as life-of-plant components with no maintenance, examination, or test requirements. This perception, along with the rush to complete the design and construction of licensed plants, resulted in great numbers of the mechanical components being installed in the 1980's.

In turn, concerns with mechanical snubbers themselves began to emerge after a cycle or two of actual plant operation. Smaller sizes proved to be fragile and susceptible to damage during plant construction. The larger sizes also soon begin to exhibit limitations under certain conditions as well. As experience accumulated, test and examination requirements were imposed on mechanical snubbers as well as hydraulic snubbers. It also became obvious that maintenance and service life monitoring programs were needed for both snubber types if they were to be utilized in long term service conditions.

From this one can see that examination and test requirements for snubbers did not start out as part of a clear comprehensive program, but rather evolved in response to emerging issues. Basic requirements were first identified in generic bulletins and notices issued by regulators; were later incorporated into technical specifications; and then later were codified into various ASME code formats. Table 1 of this document provides some comparisons of the various requirements as they evolved over time and in the various formats. The result of this evolutionary process was that individual plant snubber programs were generally based on the requirements that were in place at the time of licensing, or were adapted in response to the most recent regulatory notices. This is why there are so many different programs in existence today. Licensees have largely continued to grow their programs from their initial practices that were based upon original commitments. What originated as responses to industry operating experience and regulatory bulletins became commitments in generic Technical Specifications that have since evolved into a multitude of plant-specific Technical Specifications, owner-controlled documents, Operating and Maintenance Code requirements, and ASME Section XI relief requests. Whereas there are some general commonalities in the various program commitments, there are many significant differences. Even among programs with similar commitments, there are great differences in how the specific requirements are interpreted and satisfied, dependent in large part to plant-specific design basis assumptions. Maintenance and service life programs are even more divergent due to differences in factors such as availability and experience level of personnel resources, plant configurations, and managerial philosophies.

This does not mean that snubber programs in general are not effective, only that there are some unique and significant barriers that make it more difficult to implement and maintain quality snubber examination and testing programs than is the case for many other components. There are barriers associated with these programs that can be addressed and minimized to increase the consistency and quality of the programs.

Industry Wide Inconsistencies

Programmatic differences create a number of difficulties for the industry. Inconsistencies between programs across the industry have led to challenges for regulators and ASME code authorities as well as for utility personnel.

In general, regulatory guidance and consistency is complicated by the many varying and inconsistent snubber programs in place across the industry. Any task associated with an individual plant request or concern requires the respondent to invest time in learning the unique aspects of that particular program. A similar request from a different plant might require the same effort to be reinvested all over again. Large numbers of plants with differing licensing basis backgrounds can result in numerous requests for relief that can overburden regulators. This can lead to delays and frustrations on the part of both parties as responses may require more time and questions since each request has its own unique background which a reviewer must become familiar with.

Consistent enforcement can also be difficult when each plant has somewhat different programs and base commitments. These differences become even more exaggerated when factoring in the diversity of the program owners themselves in relation to education and experience. A program owner who has been at the helm of their program for a long period of time and has many years of testing and examination experience tends to have a more defined and direct approach to managing their program, whereas a relatively new and inexperienced program owner may be somewhat vague in their program management. The multitude of programmatic approaches can result in misunderstanding of concerns due to differing interpretations or terminology; and can serve to mistakenly escalate minor concerns to major issues that unnecessarily strain resources of both regulator and licensee. Issues are not only difficult to address in this environment, but they can even be difficult to identify. With so many programmatic differences, it can become problematic to clearly define a genuine generic concern versus a program or plant specific issue. As a result, the corrective actions that are required may be unfairly cumbersome on some plants simply due to the fact that their program is different.

Similarly, inconsistent plant programs impact the role of ASME code personnel as well. Consistent code implementation is difficult since each plant interprets code requirements in light of their specific programs and historical basis information. Developing or revising code guidance is greatly complicated by having so many various inputs and impacts to consider. Utility representatives on code committees naturally tend to model their efforts on their own programs, and reconciling differences can be a long drawn out and frustrating process. In addition, code inquiries and requests for interpretations are becoming more commonplace as plants update to requirements based on later code editions than that of their programs previous requirements. Addressing such requests is resource intensive and competes for the time code personnel have available to complete other needed activities.

On the more technical level, inconsistencies among plants contribute to a number of concerns. Benchmarking, industry operating experience, accurate dissemination of information, and resource sharing can all be problematic areas if differences in programs are not carefully considered.

Benchmarking a particular program can be a difficult task with regard to identifying a comparable program to benchmark against. Significant differences in design and licensing basis information can make comparisons between implementation practices meaningless. Benchmarking against the best of programs is of limited value if the programs differ too greatly. For instance, consider the benchmarking efforts of a plant with mostly mechanical snubbers and whose snubber program is based on an older Technical Specification and an associated ASME Section XI Relief Request (from IWF-5000 requirements). Such a plant would likely find few areas of common ground on which to draw meaningful comparisons when compared to a plant whose population consists of primarily hydraulic snubbers and whose program is based on Section ISTD of the OM Code. This is not to say that such efforts are wasted, but the greater benefit is to benchmark a similar program. The inconsistencies across the industry make this a more difficult task for many.

In a similar fashion, it is sometimes difficult to share or interpret industry operating experience accurately. Generally, an operating experience item is distributed through the industry via a “clearing house” style organization, with little background or detail provided. While this is generally a good practice, the effectiveness of the method is often lost in the area of snubbers due to that very lack of specifics. In many, if not most, instances, a significant problem in one program is not an issue in another simply due to the requirements of the particular program. In that case, the posting of the item is meaningful in that it results in a review for relevance. However, if the situation occurred at a plant where the event was not meaningful with respect to their program requirements, it would likely not be posted at all. In that situation, the industry is deprived of meaningful data. While it is recognized that this limitation extends to items and programs other than snubbers, the fact that snubber programs are so very specific and so different makes this a larger stumbling block for snubber program effectiveness. It is very difficult to glean meaningful operating experience specific to snubbers from the normal industry operating experience programs.

This leads to the associated concern of accurate dissemination of information. It is not only operating experience items that are often inaccurately or incompletely shared across the industry. General information, trending data, and technical information is also often difficult to convey in a meaningful manner. Many plants have accumulated an abundance of information relevant to their programs and to the snubber models installed in their plant. However, due to the snubber program evolving independently, much of this data has not been shared because of a perceived lack of common ground and the very real issue of multiple and varying criteria.

Related to this concern is the issue of resource sharing. While many other types of programs in the industry have developed methods to share resources such as personnel and equipment, the snubber industry has encountered problems in these areas. It is difficult for personnel to go from one plant to another and contribute meaningfully due to the fact that the processes, procedures, equipment, and even snubber types are generally different. And again, due to the low profile visibility of most snubber programs, the caliber of personnel typically assigned to assist the program owner in implementation of their program is typically the personnel with the least experience, since more experienced personnel are usually assigned to the higher profile tasks.

Plant Level Concerns

In addition to the challenges presented by program inconsistencies, there are also a number of obstacles at the individual utility and plant level that render it difficult to plan, implement, and maintain a successful long term snubber program. Significant areas of concern include a lack of experienced personnel due to an aging workforce, high personnel turnover rates, limited training opportunities specific to snubber examination and testing programs, and a lack of resources – both financial and personnel related. As previously mentioned, due to many of the early issues related to snubbers being resolved over the years, snubber programs have operated smoothly for the most part in recent years and have therefore become somewhat invisible. As a result, there has been a general tendency to relegate snubber program issues to the status of “lower tier” items. These factors have led some programs to become minimal in nature, and many programs may be limited in the ability to manage long term degradation of an aging component population.

As is the case in many areas of the nuclear power industry, personnel with significant snubber related experience and “tribal knowledge” are aging out of the work force. Due to the previously discussed evolutionary nature of snubber programs, many of them are extremely dependent upon experienced personnel who know the history and background of their programs particular requirements. In addition to the loss of historical information due to retirement and other attrition factors, technical knowledge and skills accumulated over the years are also being lost. This is true not only in the area of program management, but in the realm of “hands-on” craft labor as well. Many programs depend heavily on the ability of field personnel to detect degradation by a sense of “feel” that can only come from experience.

These losses of experienced field personnel are often made even worse by their combination with a high turnover rate of personnel assigned to manage snubber programs. For a number of reasons, it is not uncommon for the responsibility of managing a snubber program to be rotated through different individuals on an almost annual basis. This may be the result of retirements, transfers, planned rotations, or simply the result of a shrinking work force. Whatever the reason, the effect on a program for a component so heavily dependent upon experience related knowledge is significant.

Another significant challenge at the plant, or working level, is a limited number of training opportunities for those who are engaged in snubber related tasks. This applies to both those who are newly assigned to snubber activities and the more experienced personnel. There are few formal training programs available to the industry to provide or expand on snubber program related knowledge. As is the case with any plant component, snubbers have evolved over the years and replacements have become more technologically advanced. Even many of the more experienced snubber program personnel must keep abreast of emerging data and changing technology to be able to understand what is happening to their snubber population and manage it effectively as their population ages and the newer components change with technology.

Of course, much of the above is closely related to a very prevalent and important factor affecting the entire industry. A general lack of resources affects every aspect of a component program. Aging workforce and high turnover issues are made worse by a lack of resources. In an ever more cost conscious environment utilities are looking to reduce the expense associated with man power as well as other areas. Filling vacant positions often becomes a series of shuffles in an attempt to preserve budget allotments. In addition, budget issues impact the ability to replace aging equipment and components on an effective schedule.

In today's environment, the reality is that "the squeaky wheel gets the grease". Valuable resources are largely limited to those items or issues that are perceived as being of the highest importance at the moment. The fact that snubber related issues have not been seen as significant for a number of years has led to snubber programs being perceived as lower tier programs that can be continued on a minimal budget. The snubber issues of the early 1980's that led to the establishment of snubber requirements were successfully addressed and there have been few new issues requiring resources to be focused on snubber programs. It is ironic that the in-depth efforts to address earlier snubber issues were so effective that it is now difficult to focus attention on issues regarding the continued functioning of those programs. It is a legitimate fear that this focus may not be regained until a program (or programs) finds itself in a position of addressing issues neglected for some of the reasons stated above.

Snubber Users Group Actions

The SNUG is a self supported industry users group in which the members represent working level program owners and implementation personnel at operating nuclear power plants. As an independent entity with no organizational ties to other industry groups, SNUG has the freedom to address those issues that are forefront on the agenda of those doing the actual program work, and has a great depth of industry wide experience and knowledge inherent in the membership.

The SNUG mission statement reads: "The Snubber Users Group (SNUG) will strive to improve the reliability and cost effectiveness of snubber programs in the nuclear electric utility industry. We will accomplish this by providing a forum for the open exchange of technical information and ideas between members." SNUG feels that the open exchange of information and the utilization of subject matter experts represent the best avenues for addressing many of the issues described previously.

It is true that a number of the barrier issues are historical in nature and thus difficult to resolve. However, there are some steps that can and are being taken to alleviate some of the concerns and mitigate others. Generally, those issues that revolve around industry inconsistencies are the most difficult to address in any short term viewpoint, simply due to the nature of the issues that have evolved. However, SNUG has already taken some actions towards the goal of industry consistency and quality, and others are in various stages of implementation.

SNUG Conferences

SNUG holds conferences on a semi-annual basis to provide multiple opportunities for members to attend and discuss issues and concerns. The twice yearly format is utilized in order to provide as many opportunities as is possible for members to attend, given that both scheduled and unscheduled outages make it difficult for many to attend on any given annual date. The conference agendas are planned to include a variety of opportunities for discussion as well as the sharing of information and experience. Each agenda includes time for members to share their latest outage information, discuss operating experience, special topic panel discussions, question and answer sessions, and both utility and vendor presentations on topics of interest.

The conferences generally are scheduled for three days of general conference, and often a fourth day is allocated for training sessions or vendor seminars. Although these conferences cannot address all of the issues inherent in the historical differences in snubber programs, the information sharing is very valuable as plants slowly evolve to more common ground. In addition to the information shared during the formal conference sessions, the informal sharing of information between snubber program personnel, their industry peers, snubber product and service vendors and even regulators in break, lunch and other non-session related time periods has proven to be invaluable to all involved.

Working Groups

In order to better address specific needs and concerns the users group has recently created several standing working groups to focus on particular areas of interest. These include working groups on knowledge retention and industry training, code and regulatory issues, and industry good practices. In addition to these standing groups, other short-term working groups are created as needed to address emerging concerns. One example of a current short-term group is the working group on SF-1154 hydraulic snubber fluid Part 21 concerns. All of these groups serve to focus expert attention on the given areas in order to provide the industry with consistent information and recommendations. In most cases where a specific issue is being addressed, the end result will be a position paper (White Paper) providing recommendations that have been agreed upon by the majority of snubber program personnel and that can be implemented across the industry.

Training

The SNUG Board of Directors has concluded that the most pressing issues in the current snubber arena center around the loss of experienced personnel, high turnover rates, and the retention of knowledge that is unique to snubbers. To address these issues, SNUG has tasked the Training and Knowledge Retention Working Group with developing an ongoing series of training sessions for snubber personnel. The goal is to utilize the core of experienced subject matter experts within the group to pass on their knowledge and experience to others. To this end, SNUG has previously developed and provided training sessions on a number of topics that are offered on a recurring basis.

These include a seminar for snubber program owners, geared primarily for the inexperienced but informative for others as well. That seminar provides background information on the history of snubber programs, how the requirements originated and developed, the general basis of snubber programs, as well as more detailed information on implementing test plans and examination programs. This seminar is a half day session that is generally offered during a conference every one or two years.

SNUG has also provided training on the use of a program assessment guide that the group has developed and published. The guide is intended as a very snubber specific assessment template that can be utilized in conjunction with other more generic industry assessment tools to provide a clear picture of program strengths and weaknesses. Training on the guide is provided to promote consistency among programs and to provide program owners a clear picture of what constitutes a complete program.

Training has also been offered on understanding and implementing the requirements of ASME Operating and Maintenance Code Section ISTD. It is anticipated that ISTD will eventually be the predominant governing document for snubber programs, as more and more plants convert to ISTD programs over the course of their ten year Inservice Inspection (ISI) code updates. For this reason SNUG has developed a short training course as an overview of ISTD, and is developing plans for a more extended and detailed session. A large percentage of personnel serving on the ISTD Subcommittee are either SNUG members or routinely attend conferences. This availability of knowledgeable sources makes this a valuable and informative tool for attendees. It is hoped that such training will make the eventual transition to ISTD easier and more consistent for the industry.

Other training opportunities have been provided on a number of specific subjects. A recent example is a “hands-on” session on the topic of visual examinations and preventive maintenance hand stroking of snubbers. This session was led by actual utility craft personnel who demonstrated procedures and techniques as well as having class participation using a mockup assembly.

In addition to training offered directly through SNUG, the organization has also partnered with vendors to provide seminars pertinent to their products. These seminars are offered in conjunction with SNUG conferences and are offered free to conference attendees.

Future plans with regard to training include the aforementioned expanded ISTD seminar, failure analysis training, a service life monitoring and preventive maintenance seminar, and continued vendor seminars. These multiple offerings by SNUG will continue to be made available at future conferences based upon requests and perceived need.

The SNUG leadership is convinced that the utilization of the group's resources for training members of the industry is a key to developing consistent programs of high quality. The users group is currently in discussions with the Electric Power Research Institute (EPRI) to determine if there are ways to partner in providing more formal training opportunities outside of the conference setting. Requests from utilities to provide subject matter experts for on-site training are also being considered.

Library

To complement the training efforts, SNUG also has an ongoing endeavor to develop and expand its document library. Located on the group's web site (www.snuginc.org), the current SNUG library already contains a sizable amount of information helpful to program owners. Examples of items available to members are two very extensive reference manuals, one for hydraulic snubbers and one for mechanical snubbers. These manuals were compiled over years of effort and each is thousands of pages long. Also available in the library are white papers produced by previous working groups to address specific issues. Among these are:

- Vibration and Fretting Corrosion in Mechanical Snubbers (White Paper). This is an extensive study of issues associated with vibration and in particular how to identify vibration degradation and distinguish it from other commonly misapplied failure causes. This paper resulted from an industry-wide trend of wrongly diagnosing vibration issues as lubrication degradation.
- Lubrication Working Group Report. This voluminous report also resulted from an increasing trend of assigning blame for mechanical snubber degradation to lubrication issues, especially in high temperature environments.
- End Attachment Gap White Paper. This paper addressed issues raised by a regulatory inspector regarding spacer washers and end gaps on snubbers and struts. The paper compiled data from manufacturers to verify that the use of spacer washers was not a critical requirement.
- Drag Testing of Mechanical Snubbers (White Paper). This paper addresses recommended practices regarding drag testing methods and criteria for mechanical snubbers.
- Program Assessment Guide. This guide (discussed previously under training) can be used to perform assessments or to simply identify key elements of a snubber program.
- Program Template. This document was developed to provide a template for a comprehensive program document to describe a plant's snubber program in detail. The purpose of the end document is to serve as a guide to the program and its bases information.

In addition to the above listed documents, the official SNUG web site (www.snuginc.org) also contains copies of presentations given at past conferences, plant program overviews provided by member plants, various vendor documents, SNUG responses to NRC information notices, and other miscellaneous guidance documents. Appendix 1 provides a sample listing of documents available to members on the SNUG web site.

Planned additions to the library include documentation of the response to the current SF-1154 hydraulic fluid issue, a compilation of various industry good practices, and possibly a commentary on ASME OM Subsection ISTD.

Good Practices
As mentioned previously, one of the current SNUG activities is to compile a document (or set of documents) outlining good practices in a number of snubber program areas. The intent of this endeavor is to promote consistency across the industry in as many areas as possible. Currently no such document exists, although there was an INPO effort to create such a document many years ago. That effort was abandoned at the draft stage after significant industry feedback regarding numerous inaccuracies and inconsistencies.

Operating Experience
SNUG is actively working to address industry inconsistency and a lack of effective use of operating experience. In many cases, snubber-related operating experience items transmitted through the normal industry processes are inaccurately communicated or do not provide enough detail to be meaningful. SNUG is working to make active sharing of operating experience a deliberate part of each conference. As part of each conference agenda, a significant block of time is allocated for attendees to informally share any experiences that may seem helpful to other programs. As part of this agenda item, the normal industry OE postings are reviewed as well, and the involved parties are asked to provide detailed information and lessons learned.

An additional tool for the sharing of operating experience is the use of the users group web site forum. Members (and associated vendors) can post questions and information on the electronic forum. Other members are notified of the posting via email and can then respond for others to see. This interactive tool is effective in both seeking and offering helpful information and data.

Benchmarking
SNUG provides cost effective means for member plants to benchmark against other programs. At every conference a number of member plants are asked to provide an overview of their program by completing a preformatted template for presentation. By sharing this information other attendees can spot similarities to their own programs, pick up ideas, and identify areas where meaningful comparisons can be made. In addition, copies of the accumulated presentations are available on the SNUG web site for overall comparison.

In addition, each conference represents a simple form of benchmarking simply by the presence of numerous program owners and the agenda driven discussions and reports. Each conference agenda includes reports from every plant that has undergone an outage testing period since the previous conference. These reports offer excellent opportunities to compare and discuss different programs and methods of implementation. SNUG conferences also include panel discussions that permit attendees to ask questions about specific programs and specific practices. These interactions, although limited in context, offer broad overviews of other programs for attendees as a beginning for effective in-depth benchmarking efforts.

Regulatory Issues
SNUG has worked in cooperation with regulatory agencies on a number of issues over the years, and continues to do so in an effort to further a consistent approach to establishing and enforcing program requirements. One example of this was the extensive collection of data for the basis of Generic Letter 90-09, which permits a significant extension of the interval between visual snubber examinations. Another more recent example is the cooperation with the NRC in the development of a new snubber Technical Specification LCO that provides more opportunities to test snubbers without impacting the availability of supported systems. In most cases, a representative of the NRC is in attendance at SNUG conferences at least once each year. This presence allows an opportunity for interaction and cooperation for the benefit of all involved.

ASME Code
SNUG leadership works closely with ASME personnel to provide information and feedback on code issues and concerns. The conferences also provide an opportunity for attending members to speak

freely about any concerns or issues that they may have with regard to ASME code usage. Since many of the members of the ASME ISTD sub-committee also attend SNUG conferences on a regular basis, this is an excellent chance to get real-time feedback on the code and address questions outside of the formal ASME process.

Resources (Subject Matter Experts)
As the industry faces a loss of experienced personnel in almost every area, the value of such experience is becoming greater. SNUG has realized that within its membership resides a great deal of the available experience in the snubber field. The group is currently developing concepts for a realistic way to utilize this experience for the overall benefit of the industry by assembling and making available subject matter experts on an as-needed basis. Options being evaluated include providing peer teams for assessments, on-site “consulting”, or document reviews. One vision is for SNUG to be a single point of contact for such expert assistance when requested. Although a formal program is still in the conceptual stage, SNUG already provides assistance of this type on a limited and informal basis.

Another possible future development is the use of SNUG as a resource pool for not only knowledge but as a clearinghouse for equipment, parts, and tools that might be shared among members. Although this concept may prove cumbersome due to legal issues, it is worthy of being evaluated as a long term possibility.

Conclusion

Obviously there are a number of issues that adversely affect the ability of the nuclear power industry as a whole to implement consistent and high quality snubber examination and testing programs. Existing programs for the most part adequately address the basic program requirements, but are hindered in an effort to be consistent in both practice and quality across the industry. Varying program bases, historical differences, and a continuing loss of experienced and knowledgeable personnel are all barriers to achieving the goals of industry consistency and quality.

The Snubber Users Group (SNUG) has increased the organization’s focus on these issues and is intent on contributing to the goal of consistent, high quality snubber programs. They intend to do this by continuing the proven practices that promote industry cooperation, as well as developing new initiatives to meet the emerging needs of the industry. In addition to the continued use of conferences to promote sharing of technical information and experience, SNUG is expanding its effort to provide training to the industry and to better utilize the experience base within the SNUG membership. The leadership of SNUG realizes that this is a vital role within the industry and is committed to increasing the focus and resources of the organization to that end. Through the refining of existing activities as well as innovative new programs, the barriers that currently inhibit consistent quality can and will be removed or significantly reduced.

Table 1
Evolution of Snubber Program Requirements

	Early TS	Later TS (TRMs)	Section XI (pre-1989)	Section XI (1989+)	ASME OM-4	ASME ISTD
Scope	All Safety Related Hydraulic Snubbers	All Safety Related Snubbers + Snubbers Affecting SR Systems	Class 1, 2, 3, & MC Snubbers with Load ratings < 50 kips	Class 1, 2, 3, & MC Snubbers	All Safety Related Snubbers	All Safety Related Snubbers
Boundary	Pipe to Building Structure	Pipe to Building Structure	Pipe to Building Structure	Refers to OM-4	Pipe to Building Structure	Pin to Pin
Certifications	None	None	VT required	VT Method required	Per Owner's Requirements	Per Owner's Requirements
Test Plans (Failure Mode Grouping Requirements)	10% or 10 No FMG	10%, 37 (others, 88,55,etc) No FMG	10% No FMG	Refers to OM-4	10%, 37 Mandatory FMG	10%, 37 Optional FMG
Drag Testing	Hydraulic Snubbers Only Breakaway & Running	Mechanical Snubbers Only Breakaway & Running	Hydraulic.& Mechanical Snubbers Breakaway & Running	Refers to OM-4	Mechanical Snubbers Only Breakaway & Running	Mechanical Snubbers Only Running
Service Life Monitoring	None	Yes	None	Refers to OM-4	None	Yes

Appendix 1
Sample Document Listing from SNUG Web Site (www.snuginc.org)

1. Alternative Acceptance Criteria
2. Benefits of Specialized Snubber Training
3. Evolution of Snubber Testing
4. Lubrication Working Group Report
5. Member Plant Program Overview Reports
6. NRC Generic Letter 90-09
7. NRC Operability Letter 02/28/86
8. NRC Operability Letter 05/27/86
9. NRC Paper on 72 hours
10. Review of GI-113 with ACRS
11. Snubber Program Document
12. SNUG By-Laws
13. SNUG Policy Statements
14. SNUG Program Assessment Guide
15. Understanding Breakaway Drag
16. Wald's Sequential Probability Ration Plan
17. White paper on Alternate Visual Examinations
18. White Paper on Drag Testing
19. White Paper on Snubber-Strut End Attachment Gaps
20. White Paper on Vibration and Fretting Corrosion in Mechanical Snubbers

Appendix 2
Acronyms and Definitions of Terms

10% Plan	A test sample plan where the initial sample makes up 10% of the subject population.
37 Plan	A statistical sample plan where the initial sample consists of 37 snubbers chosen at random from the subject population.
Breakaway Drag	Generally used to describe the force required to initiate movement of a snubber, or the test used to determine said force.
FMG	Failure Mode Group – a group of all snubbers determined to be subject to a particular failure mechanism or cause. The use of such grouping requires that appropriate corrective actions and/or further testing be applied to the entire group separate from the original subject population.
ISTD	ASME OM Code Section ISTD – this section governs the In Service Examination and Testing of Dynamic Supports (Snubbers)
OM-4	Common term used to refer to ASME OM Code Part 4, which preceded ISTD as the OM section governing snubber examination and testing.
Running Drag	Generally used to describe the force required to maintain movement of a snubber at a prescribed velocity, or the test used to determine said force.
Section XI	Section XI of the ASME Boiler and Pressure Vessel Code
TRM	Technical Requirement Manual – this term generically refers to Owner-controlled documents that typically replaced Technical Specifications for snubbers. Other terms for these documents include Operating Requirements Manual, Selected Licensee Requirements Manual, Program Requirements Manual, etc.
TS	Technical Specification
VT	Generally used to describe ASME Section XI visual examination requirements or methods. The term derives from the examination levels described in Section XI (VT-2, VT-3, etc.)

Session 3(a): Pumps II

Session Chair: Robert G. Kershaw, Arizona Public Service Company (APS)

Alternate Method to Measure Cooling System Flow in Nuclear Power Plants

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Introduction

Nuclear Power Plant safety system monitoring requires the ability to determine flow in conduits where hydraulic conditions are sometimes not suitable for use of typical time of flight ultrasonic flow meters. These hydraulic conditions have resulted in applications where the 2% accuracy required is called into question due to instability or measurements which do not correspond to other data. This paper discusses an alternate method to determine flow using an ultrasonic Cross-Correlation technology. The methods detailed in this paper have been applied to similar flow measurements in the CANDU nuclear power plants. The following areas will be addressed:

- Underlying physics of Cross-Correlation flow meter
- Description of hardware and software including installation process
- Application issues (sensitivity to various parameters)
- Demonstration of meter accuracy
- Description of meter uncertainty
- Application example

Underlying Physics of Cross-Correlation Flow Meters

1.1.Turbulent flow structure

Single phase turbulent flow in a pipe can be considered as random assembly of changing structures, formed by the turbulent velocity field, and called Turbulent Eddies. Motion of such eddies along the pipe constitutes the motion of the liquid as a whole and defines volumetric or mass flow rate.

Typical structure of turbulent flow with clearly identified eddies is shown in Figure 1. In many engineering applications, real structure of the turbulence is approximated by a time-average model, as is illustrated in Figure 1.

1.2.Principle of Operation

The simplest design of a Cross-Correlation Ultrasonic Flow Meter (UFM) consist of two ultrasonic beams transmitted through a pipe at a certain distance apart along the pipe length, as is shown in Figure 2. Each beam is affected (modulated) by the moving eddies, and after de-modulation generates a time signal.

For any specified time interval **T**, the time signal is completely defined by the set of eddies, which were affecting the beam during this time interval. The set of eddies is unique and not repeatable because it is a subset of a random assembly. A second beam, which is positioned on the pipe a distance **L** downstream of the first one, is affected by the same assembly of eddies with a certain time delay τ^*. Eddies in the selected set do not form exactly the same pattern when they move along the pipe; therefore, the signal generated by the second beam is not exactly identical to the first one.

However, there is a measurable correlation between the two. Delay time between signals, **τ*** can be determined by a mathematical process which compares of two signals.

Assuming that **X(t)** and **Y(t)** are two signals generated by the interaction of the eddies with upstream and downstream ultrasonic beams respectively, and introducing signal **Y(t +τ)** which is equal to signal **Y(t),** but is shifted in time by time-shift τ**.** To obtain quantitative estimation of the similarity between **X(t)** and **Y(t +τ)** within time interval **T**, an average of the difference between two signals, **Δ(τ)** can be calculated as follows:

Equation 1:

$$\Delta(\tau) = \frac{1}{T}\int_0^T [X(t) - Y(t+\tau)]^2 \, dt$$

The maximum similarity between signals is achieved at point τ **=** τ***** where **Δ(τ)** is minimum. From Equation 1 it follows that the minimum of **Δ(τ)** corresponds to maximum of function **R(τ),** which is defined by the following equation:

Equation 2:

$$R(\tau) = \int_0^T X(t) \cdot Y(t+\tau) dt$$

Function **R(τ)** is called the Cross-Correlation Function. Time delay τ***** between two time signals is calculated as a position of the maximum of a Cross-Correlation Function **R(τ).** Transport velocity of the set of eddies (measured velocity $\mathbf{V_m}$) is calculated as follows:

Equation 3:

$$V_m = \frac{L}{\tau}$$

L - Distance between two ultrasonic beams.

In general case, measured velocity $\mathbf{V_m}$ is not equal to the cross-section average flow velocity $\mathbf{V_a}$. The relation between $\mathbf{V_m}$ and $\mathbf{V_a}$ depends on liquid properties (Reynolds Number) on hydraulic characteristics of the flow (piping geometry and meter location) on meter design (number of ultrasonic beams and their orientation) and on signal processing algorithm (signal frequency band, filters characteristics). The ratio of the bulk flow velocity $\mathbf{V_a}$ to the measured flow velocity $\mathbf{V_m}$ is defined as Hydraulic Factor **C**. Similarly as with other flow measurement instruments, the value of **C** has to be obtained based on laboratory calibration or other analysis. If **C** is known the flow rate in a pipe **F** is calculated by the following Equation:

Equation 4:

$$F = \rho \cdot A \cdot C \cdot \frac{L}{\tau^*}$$

ρ - Fluid density

A - Pipe cross-section area

Theory of flow measurements, using cross-correlation technique, was developed in the 1960’s and is presented with greater detail in Reference 2.

2.3 Comparison with Conventional Transit Time Meters

The basic difference between the cross-correlation and transit time technologies is the way that each of these meters measures the velocity of the fluid within the pipe. The transit time technology injects an ultrasonic signal diagonally through the fluid and then measures the difference in the time that it takes for the ultrasonic pulse to travel upstream versus downstream. It can then be shown that the difference in these times is approximately proportional to the velocity of the fluid in the pipe. In this approach the ultrasonic path changes direction on each interface between transducers, pipe wall and liquid inside the pipe. Therefore, it is very sensitive to temperature change and to transducer installation discrepancies due to alignment and pipe conditions. Measured time difference, which is on the order of a few microseconds, has to be measured with accuracy of few nanoseconds.

The cross-correlation meter measures velocity of the fluid by determining the time that it takes for the fluid to pass distance between two ultrasonic beams. An ultrasonic beam is injected perpendicular to the axis of the pipe, rather than diagonally as is required for the transit time meter, providing very robust and stable ultrasonic path, which is not effected by the temperature change and is not sensitive to transducer alignment. For typical industrial applications the magnitude of the delay time is in order of 30 – 100 milliseconds, which can be measured very accurately.

A significant difference between the time of flight meter and the cross correlation meter is the precision required for measurement of the time difference (see Figure 3).

Another significant feature of the cross-correlation technology is that the bulk flow velocity is not derived from measurements of local characteristics of the velocity profile. The cross-correlation meter measures directly the volumetric flow rate by measuring the time that is required for a known volume to travel through the pipe cross-section. The movement of that volume is determined by tracking the movement of turbulent eddies contained in the volume. Therefore, the meter is not directly affected by radial or angular velocity components in the flow. Effective sampling area of the flow is defined by the size of eddies, but not by the diameter of the ultrasonic beam, as it is in transit time meters.

Hardware-Software Components and Installation Process

The cross-correlation UFM consist of the following components: (a) a transducer, which is mounted on the pipe and supports two transmitter- receiver pairs of ultrasonic probes; (b) a Signal Conditioning Unit (SCU), which generates transmitter signals to drive piezoelectric crystals and provides demodulation of the carrier ultrasonic signal; (c) a multiplexer which allows monitoring of up to eight transducers; and (d) a computer, which provides interface between analog and digital components of the signal processing, analog signal processing, and user interface. Hardware components of the meter are shown in Figure 4.

Software package of the meter provides user interface and supports the following functions:

- Hardware configuration, such as selection of electronic filters frequency band and ultrasonic frequency band
- Optimization of signal processing parameters, such as sampling rate, data acquisition time, signal rejection/acceptance criteria
- Input of pipe and flow parameters, which are required for flow rate calculation, such as pipe internal diameter, distance between ultrasonic beams, pipe and transducer thermal expansion coefficients, etc.
- Presentation of historical flow trend
- Diagnostics of the system performance and other trouble-shooting tools
- Data storage

Installation of the system does not require flow interruption and can be performed on a hot or cold pipe. Time of installation significantly depends on required accuracy, and may vary from 15 minutes to a few hours. Typical installation time for one transducer is 2 – 4 hours. Normal installation process consists of the following steps:

- Assessment of the piping configuration for selection of the location for transducer installation. The location should be accessible, and upstream piping geometry should not include features that may produce unpredictable effect on flow. For example, effect of a partially closed valve on downstream meter might be difficult to predict if valve position is not known or is not stable.
- Arrangement of the work-space, such as construction of platform, radiation protection etc.
- Preparation of pipe surface and pipe measurement. Accurate measurement of the pipe diameter and pipe wall thickness is required to provide accurate input for calculation of the pipe cross-section area; depending on condition of installation, pipe measurement may take few hours. Pipe surface has to be cleaned using sandpaper
- Installation of the transducers
- System tuning and commissioning. This stage includes optimization of the ultrasonic transmission, testing of the electronics components, preliminary data collection and optimization of signal processing parameters.

4. System Validation and Verification in Laboratory and Field Environment

Cross-correlation UFM had been used for various applications in nuclear industry for more than 30 years in Canada, USA, and other countries. Today more than 100 installations are operating world wide.

The first experimental study of the Velocity Profile Correction Factor (VPCF) using the cross-correlation clamp-on ultrasonic flow meter for single-phase flows was conducted in Ontario Hydro between 1985 and 1990. This high temperature test was designed specifically applications in CANDU reactors. Pipe dimension (14 inch and 16 inch diameters), flow pressure, flow temperature and flow velocities were similar to the typical feedwater system in CANDU reactors.

In 1994 - 1996, a study of a cross-correlation ultrasonic flow measurement technology for single-phase flow was conducted by AMAG. The underling physical phenomena were analyzed, and an equation was obtained which described the dependence of the VPCF on Reynolds Number (Reference 3 & 4).

Since 1994 the meter was evaluated in a number of laboratory tests in the following laboratories: Alden Flow Laboratory; National Institute of Standards and Technology in USA; Hydraulic Center of National Research Council (Canada); Chatou Flow Laboratory in Electricity de France, in MHI Takasago Flow Laboratory and others. Results of some of these tests are shown in the figures and tables section and discussed below.

The system is currently being used to measure various safety system flows in the CANDU Power Plants.

Table 1 presents summary of the tests conducted in Chatou Flow laboratory in France on 14” carbon steel pipe at four different Reynolds Numbers. The UFM measurements were obtained in a blind test, where UFM flow readings were submitted to the Laboratory personnel before laboratory reference flow readings were available to UFM operator. In this test, the reference instrumentation was orifice plate with 95% confidence interval uncertainty 0.3%.

A similar blind test was conducted at NIST, where different clamp-on UFMs from five vendors were compared with the NIST weigh tank reference reading. The test was conducted at three different Reynolds Numbers: 400,000, 1,600,000, and 2,900,000. Summary of the results is shown in Figure 5, which was taken from the NIST Report, Reference 5. In this figure the Cross-Correlation Meter data is displayed over “E”.

5. Meter Sensitivity

Detailed analysis of different factors, which may affect meter performance and its reading, was conducted by the cross correlation meter vendor. This vendor study, consisting of a complete system, which combines the meter, the pipe and the flow, was considered based on non-dimensional system analysis and on physical phenomena associated with acoustics, hydraulics and signal processing. Based on this analysis and on operation experience, the following critical parameters were identified:

5.1. Reynolds Number

$$\mathrm{Re} = \frac{\rho \cdot D \cdot Va}{\mu}$$

Reynolds number represents normalized viscosity of the fluid. It affects hydraulic parameters of the flow and, as a consequence of that, affects Hydraulic Factor **C**.

Most significantly the Hydraulic Factor **C** is affected by Reynolds Number in a long straight pipe. Dependence of **C** on Re for such case is described by theoretical equation, which was validated in numerous laboratory tests in broad range of **Re**. (See previous section). This equation shows weak dependence of **C** on **Re**. For example, when Re changes from 1 million to 10 million the change of **C** is about 1%. In the system software the **Re** is calculated by iterations based on measured velocity $\mathbf{V_m}$ and on equation for **C**. In all practical situations, the error in **Re** calculations results in a negligible error in **C**. In total uncertainty balance the uncertainty on **C** as a function of **Re** is estimated conservatively as 0.25%.

5.2. Equivalent Hydraulic Roughness

Hydraulic roughness, similarly to Reynolds Number, affects turbulent and averaged flow characteristics. The Moody Diagram defines dependence of pipe friction factor on Equivalent Hydraulic Roughness. The Theoretical equation for **C** and Moody Diagram can be used to calculate **C** if Equivalent Hydraulic Roughness is known. If hydraulic roughness is not known and can be estimated approximately within a certain range, the corresponding uncertainty of **C** can be calculated, and should be included in total uncertainty. An order of magnitude of the possible roughness effect on hydraulic factor can be seen from the following example: For 12” pipe, made of a typical construction steel, with flow rate of approximately 300 l/s (approximately 5000 GPM) at room temperature, **C** = 0.925, which is 0.4% different from its value for smooth pipe, **C**=0.929.

5.3. Position of the transducer on the pipe

The Position of the transducer on the pipe with respect to the upstream piping geometry, normalized to pipe diameter, together with Reynolds Number and Hydraulic Roughness, defines velocity distribution in the pipe at the location of the transducer.

If the pipe run upstream of the location of the transducer is not long enough, the standard equation for the Hydraulic Factor **C** is not applicable. In this case, the value for **C** should be determined in calibration test. Most typical piping geometry is a combination of elbows. Therefore a single 90-degree elbow was selected as benchmark geometry (**BM**), which is used to compare dependence of **C** on the distance from flow disturbance for other piping configurations. The following example illustrates this approach:

Figure 7 shows real piping where the meter was installed downstream of a combination of elbows. To determine Hydraulic Factor **C** for such piping geometry, two laboratory models were manufactured, a Test Model (shown in Figure 8), and a Benchmark Model. A set of UFM transducers was installed on a straight run test section downstream of the pipe elbows. Transformation of the test loop from Test Model to the benchmark geometry was conducted such that the test section with all transducers remained unaffected, and only the upstream section of the loop was replaced. This approach provides accurate comparison of the test section and benchmark geometry by eliminating uncertainty associated with test section pipe area and with transducer installation. Factor **C** was determined by comparing UFM readings with weigh-tank data. Results of the test are shown in Figure 9. It can be seen that the difference between the Test Model and the benchmark is within 0.5% for the locations between 8 to18 pipe diameters from the nearest upstream elbow.

In general, numerous laboratory tests with different piping geometries have shown that on a distance of 8 pipe diameters and longer, deviation of Hydraulic Factor **C** from its Benchmark distribution is +/-1% for a broad range of flow disturbances.

5.4.Flow Frequency

Flow Frequency is defined as a ratio of flow velocity to pipe diameter, F= Va/D - represents scaling factor for turbulence spectrum in the pipe, and is associated with the size of turbulent eddies.

Demodulated signals, generated by interaction of moving eddies with the ultrasonic beam, are processed using filters with a certain set of frequency bands. Selection of the frequency band of the filters defines a window of eddies' sizes, as they are detected by the meter, and may result in different measured velocities. Appropriate adjustment of the filters eliminates this effect. This is discussed in more detail in Reference 3.

5.5.Pipe Temperature

Pipe temperature may affect flow density, flow viscosity, and acoustical characteristics of the system. Pipe temperature is used as an input for flow rate calculation to account for flow density, for pipe thermal expansion, and for flow viscosity. Possible error associated with last two factors is negligible for typical uncertainties in temperature input. The most significant affect of temperature is associated with flow density; 1^0 C uncertainty in flow temperature results in approximately 0.1% uncertainty in flow density.

Change in speed of sound due to temperature change is another factor, which may potentially affect ultrasonic transmission through the pipe wall and the fluid. In cross-correlation meters this effect can be eliminated by optimization of the range of ultrasonic frequencies depending on specific application. For reactor coolant flow measurements, this allows continuous flow monitoring in temperature range from 20^0C to 320^0C.

5.6.Acoustical noise

Acoustical noise occurs in pipe flow as a result of interaction of acoustical waves. Presence of such noise may displace the position of the maximum of the cross-correlation curve, resulting in flow reading bias. In the last three years, various methods of on-line and off-line signal analysis have been developed, which allow identification of the presence of the acoustical noise and determination of its effect on time delay calculation.

6. Uncertainty Analysis

6.1. Overall Uncertainty

The flow as measured by the cross correlation UFM is calculated based on the following equation:

Equation 5:
$$W = C_f \frac{\rho A L}{\tau}$$

Where:

C_f: Flow Bulk Correction Factor
ρ: Flow Density
A: Pipe cross-sectional area
L: Transducer Spacing
τ: Time Delay

The uncertainty associated with this equation can be expressed as:

Equation 6:
$$\varepsilon_w = \left[\varepsilon_{c_f}{}^2 + \varepsilon_\rho{}^2 + \varepsilon_A{}^2 + \varepsilon_L{}^2 + \varepsilon_\tau{}^2\right]^{0.5}$$

All of the above parameters are loop dependant and are based on measurements taken in the plant during the test. The time delay is comprised of the instrument accuracy of 0.035*ms* along with flow measurement and statistical scatter.

6.2. Flow Bulk Correction Factor (Cf)

Equation 7:
$$C_f = C_p * C_0 * [1 + \Delta C]$$

Where:

C_f = Flow Bulk Correction factor for flow at a point L/D downstream of a flow disturbance
C_p= Piping configuration factor for upstream disturbances (excludes elbow bends)
C_o= Flow hydraulic correction factor at a specific Reynolds number
ΔC= Change in hydraulic correction factor due to upstream elbow bend

The uncertainty associated with the C_o is dependant on the actual pipe configuration upstream of the meter location. Based on the lab analysis compared with in plant testing the prediction for this correction factor is within 0.25% however, for the typical installation discussed in section 7, 0.5% is assumed.

In addition, uncertainty relative to piping configuration and Reynolds number are taken into account. With regard to the Reynolds number the random uncertainty of the inputs to the calculation are also considered. The confidence interval for Reynolds number is determined by taking the derivative of the Reynolds number with respect to velocity, density and dynamic viscosity.

6.3. Flow Density (ρ)

The uncertainty of density is determined by the uncertainty of the pressure and temperature measurement of the fluid. The confidence interval for the density is determined by taking the derivative of the density with respect to temperature and pressure.

6.4. Pipe Cross-sectional area (A)

The uncertainty of the pipe cross-sectional area is determined by evaluating data taken during installation such as pipe wall thickness, pipe temperature and pipe outside diameter measurements. Uncertainty for thermal expansion effects is also considered. The elements of the internal diameter measurements are as follows:

Equation 8:
$$d_{i@y[x°,x°+180]} = d_{o@y[x°,x°+180]} - t_{y[x°]adj} - t_{y[x°+180]adj}$$

Where:

$d_{i@y[x°,\ x°+180]}$ = Inner diameter at location y and position x +180

$d_{o@y[x°,\ x°+180]}$ = Outer diameter at location y and position x +180

$t_{y[x°]adj}$ = $(SV_{sample}/SV_{calref})^* \ t_{y[x°]} + X_{calref}$

$t_{y[x°+180°]adj}$ = $(SV_{sample}/SV_{calref})^* \ t_{y[x°+180°]} + X_{calref}$

$t_{y[x°]}$ = Measured wall thickness at location y and position x

$t_{y[x°+180°]}$ = Measured wall thickness at location y and position x+180°

SV_{sample} = Sound velocity of pipe material sample

SV_{calref} = Sound velocity of calibration block

X_{calref} = Bias correction in wall thickness measurements due to calibration block material property differences

y = location along the pipe

x = radial location around the pipe— 0°, 45°, 90°, or 135°

6.5. Transducer Spacing

The uncertainty of the transducer spacing is determined by evaluating measurement data taken during installation. Uncertainty for thermal expansion effects is also considered.

To determine transducer spacing, measurements are taken on the CROSSFLOW bracket prior to its installation on a pipe or at the time of installation The bracket consists of a male half and a female half that are fastened together when the unit is mounted to a pipe.
Measurements are taken to find the outer diameter separation of the two mounting holes on both male and female sides. Then each mounting hole inner diameter on both sides is measured. There are 6 measurements and the process is completed 3 times. The average of each measurement is found and then an average center-to-center spacing can be found as in equation 9.

Equation 9:
$$L_{spacingmeas} = \frac{L_{ms} - L_{mh} + L_{fs} - L_{fh}}{2}$$

Where:

L_{ms}= average OD separation on male side

L_{mh}= average hole ID on male side

L_{fs} = average OD separation on female side

L_{fh} = average hole ID on female side

6.6. Time Delay

The time delay data is collected by the CROSSFLOW meter and is used in Equation 5 to calculate the flow inside the pipe. The time delay average (t_{delay}), standard deviation (σ_{tdelay}), and statistical uncertainty (ε_{tdelay}) are determined for the collected time delay. In addition to the statistical uncertainty, the total 95% confidence interval for the time delay must also include instrumentation uncertainty and a noise correction uncertainty. The total time delay uncertainty (ε_{txflow}) is found by taking the square root sum of the squares of these uncertainties.

Equation 10: $$\varepsilon_{tflowx} = \left[\varepsilon_{tdelay}^2 + \varepsilon_{CN}^2 + \varepsilon_{tinstr}^2\right]^{0.5}$$

Where:

ε_{tdelay}	=	Statistical uncertainty of time delay data
ε_{tinstr}	=	Uncertainty of CROSSLFOW instrumentation
ε_{CN}	=	Uncertainty noise correction factor

Acquired time delay data and the instrumentation uncertainty depend on certain software and hardware parameters. The optimum configuration of these parameters is determined at the time of installation and tuning.

The Cross Correlation flow meter utilizes multiple samples of the time delay measurement to achieve a relatively low uncertainty due to random error. This is based on the central limit theorem, which basically states that as the sample size (N) becomes large, the following occur:

1. The sampling distribution of the mean becomes approximately normal regardless of the distribution of the original variable.
2. The sampling distribution of the mean is centered at the population mean of the original variable. In addition, the standard deviation of the sampling distribution of the mean approaches the standard deviation divided by the square root of the sample size.

The variability in a measurement system is driven by the standard deviation of the measurements taken. By taking multiple samples of the same measurement, the standard deviation of the sample means is reduced by a factor of one divided by the square root of the sample size.

Example

Flow through a pipe needs to be held to +/- 2 ft/sec (total tolerance = 4 ft/sec)
Allowable measurement system variation = 1%= .04 ft per sec

30 measurements are taken.

Standard Deviation (s) of these samples =0.1 ft/sec
Expected Measurement System Error = 4 * s = 0.4 ft per sec

Take 30 measurements, but each measurement is an average of 100 samples.

Standard deviation (s) of these samples = 0.1/$\sqrt{100}$) = 0.01 ft/sec
Expected measurement system variation = 4 * 0.01 = .04

Table 2 provides typical uncertainty values for the various inputs to the flow equation.

Application Example

Essential Cooling water flow measurement tests were performed at a nuclear power generation facility on October 12-18 2007. The objective of the flow measurement test was to provide a demonstration of the capability of the Cross Correlation method of measuring spray pond flow.
The facility currently uses Annubars for their flow measurement along with ultrasonic time of flight flow meters. Historically, at this utility, flow measurement with a time of flight strap on meter, at locations where there are significant flow disturbances, has been difficult and unreliable. The flow measurements are used to insure design basis flow is supplied to various safety related heat exchangers. The Cross Correlation flow meter uses a different principal of operation which has the ability to measure flow at locations where time of flight meters will have difficulty. As described above,

the cross correlation flow meter is less susceptible to such disturbances and methods are available to correct for their influence.

For the Spray Pond supply line flow measurements the following flow meters were installed as indicated on Figure 10.

- PVSP-1: 10” supply line to the Diesel Cooling Water
- PVSP-2: 20” line downstream of the branch line for the Diesel Cooling Water
- PVSP-3: 24” line upstream of the branch line for the Diesel Cooling Water

Table 3 displays the physical relationship between the meter location and hydraulic disturbances.

It is desirable to measure flow on the 24” supply line before the 10 inch branch line to the diesel cooling water. This location however, is not a favorable location with respect to a predictable flow velocity profile and thus will produce errors in an overall flow measurement. This location will also make it difficult to measure the flow with a time of flight meter since the upstream piping conditions include an out of plane bends which could produce swirl. In order to correct for errors resulting from the piping geometry a simultaneous flow measurement was taken on the 20” line down stream of the branch for the diesel cooling water flow when the flow was isolated. This results in equal flow at the 24” line and the 20” line. The 20” line is a very favorable location due to the significant length of pipe downstream of the upstream out of plane bends. Since the flow measured at the 20” location is the same as the flow measured at the 24” location the difference between the measured flows can be used to quantify the effect of the unfavorable geometry at the 24” location.

The crossflow meter has been tested for stability under non standard conditions such as the 24” location. Once the calibration is performed using the 20” location, the flow profile correction factor will remain stable close to the flow regime where the calibration occurred. The testing performed included hydraulic conditions which produced significant changes to the flow profile. These tests were performed multiple times to determine repeatability of the instrument. Typically the repeatability of the Crossflow meter is on the order of 0.2%.

The flow field (velocity profile) at the 24” location is still developing as the water progresses down the pipe, which causes the requirement for the calibration. However, it is developing in a stable manner such that the mean velocity and radial turbulence profiles are relatively uniform in circumferential orientation. If it is desired to measure the flow at significantly different conditions such as those required to develop a pump curve, further calibrations would be required to bound the Reynolds number effects on the flow velocity profile. The calculated calibration factor for the 24” meter location is shown in
Table 3.

A meter was also installed on the 10” supply line to the Diesel Generator Cooling Water. Measurements were taken with normal flow to the Diesel Cooler. This location is also an unfavorable geometry due to the short distance between the elbow and measurement location. A calibration factor for the 10” supply line can be developed based on the difference between the 20” and the 24” location.

As these flow measurements were only intended as a proof of capability, a formal uncertainty analysis was not performed; however, an abbreviated uncertainty calculation was performed to provide an estimated uncertainty of the flow measurement.

Table 4 displays the results of the flow measurements along with an estimated uncertainty.

The flow measurements are calculated based on actual pipe wall thickness measurements as well as accounting for a reduction of the area by the full 12 mil Plasite® liner. If the liner were a full 12 mils it would account for 0.25% flow reduction on the 20” pipe and 0.2% flow reduction on the 24“ pipe. The 10” pipe liner effect is approximately 0.5%.

The Flow Displayed for the 24" pipe includes flow profile corrections obtained from test with diesel cooling water isolated as described above.

Uncertainty has been estimated for the flow measurements based on all elements of uncertainty with the exception of the 12 mil liner and secondly, accounting for the uncertainty of the liner as a Bias and therefore outside of the radical. The actual thickness of the liner is not known at this time; therefore, the design liner thickness is used for the uncertainty.

8. Conclusion

This paper demonstrates that the use of the cross-correlation flow measurement method is an acceptable alternative for measurement of low Reynolds number turbulent flow conditions with the ability to achieve acceptable uncertainty. The principals of the flow meter are based on established turbulent theory, the installation practices are thorough and well defined, the accuracy has been well documented and the uncertainty determination is rigorous. The application shown is an example of a flow measurement technique that can solve a problematic flow measurement. The results of this flow application were within the expected flow and consistent with the pump performance. The expectation is to use the cross-correlation method to measure flow on other trains of the spray pond flow as well as other flows.

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1. P.A. Durbin, B.A. Petterson, Statistical theory and modeling of turbulent flows, John Wiley & Sons, Inc, 2001.

2. R. S. Beck and A. Plaskovski, "Cross-Correlation Flow Meters", Adam Hinger, Bristol (1987)

3. Y. Gurevich, A. Lopez, R. Flemons, D. Zobin, Theory and Application Of Non Invasive Ultrasonic Cross-Correlation Flow Meter In Harsh Environment, The 9th International Conference On Flow Measurement, FLOMEKO'98

4. Y. Gurevich, A. Lopez, D. Zobin, Performance Evaluation and Application Of Clamp-On Ultrasonic Cross-Correlation Flow Meter CROSSFLOW™ , The 5th International Symposium Fluid Flow Measurement, Washington DC, 2001.

5. To Be Provided

6. Principles and Procedures of Statistics, Steel & Torrie, McGraw-Hill 1960

Figures and Tables

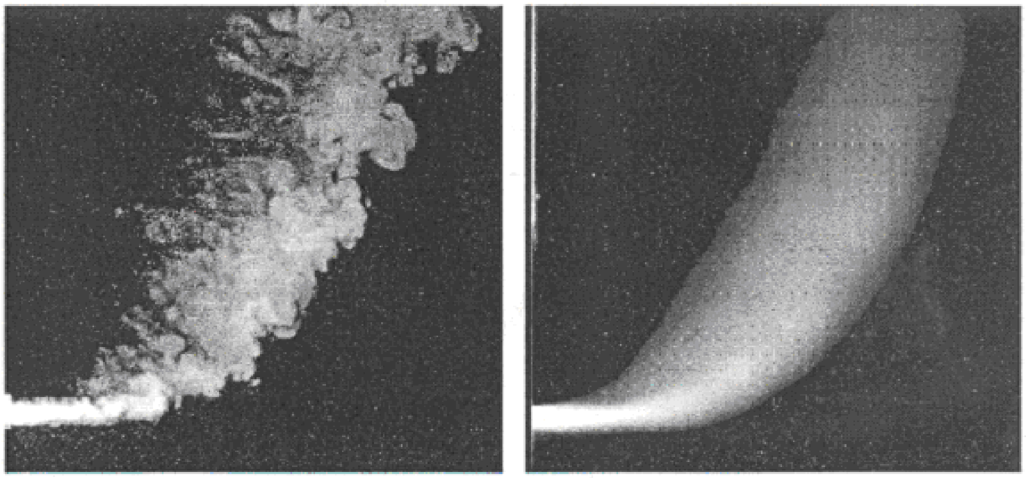

Figure 1. Typical Structure of a turbulent flow and its time-average approximation. (Reference 1)

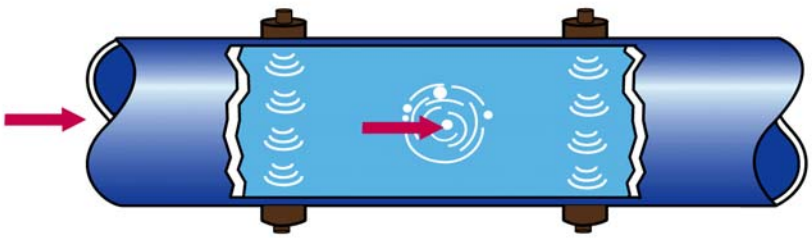

Figure 2. Cross-Correlation Flow Meter Principle of Operation

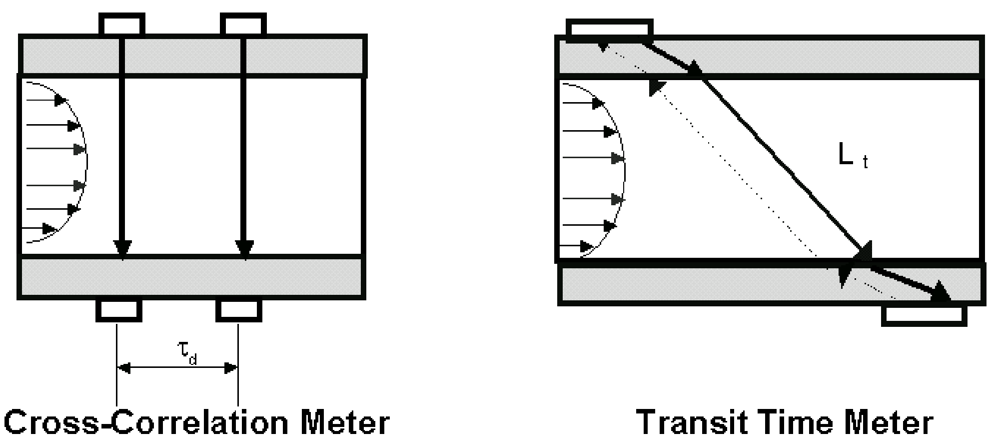

Figure 3: Comparison of Cross-Correlation vs. Transit Time technologies

Figure 4: Typical UFM Hardware

Test #	Date	Reynolds Number	CROSSFLOW	CHATOU	DIFF(%)
1	9-Sep	$1.3*10^6$	1082.73	1084.54	0.17%
2	9-Sep	$1.1*10^6$	901.19	902.7	0.17%
3	9-Sep	$0.86*10^6$	701.21	699.73	-0.21%
4	10-Sep	$0.73*10^6$	589.43	590.53	0.19%

Table 1: Chatou Test Data

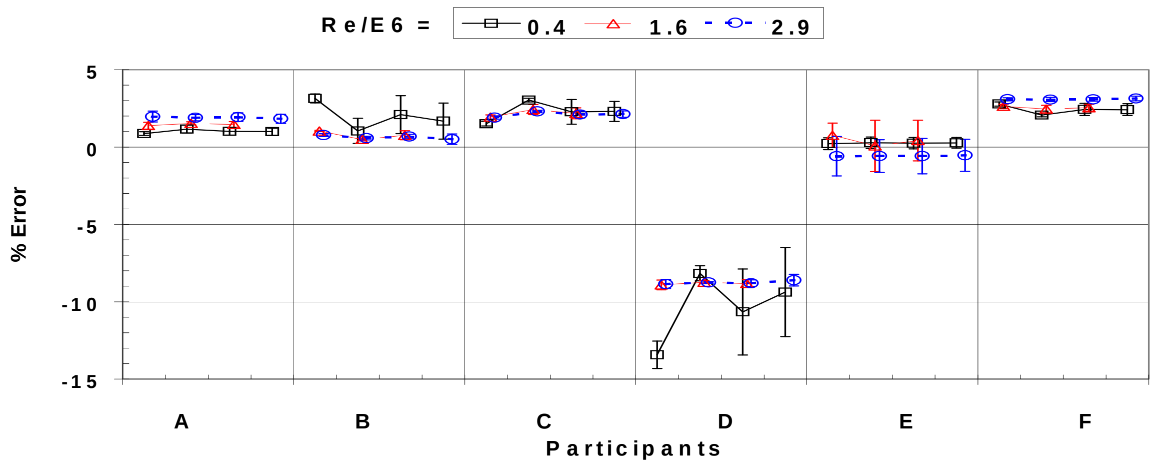

Figure 5: NIST Blind Test

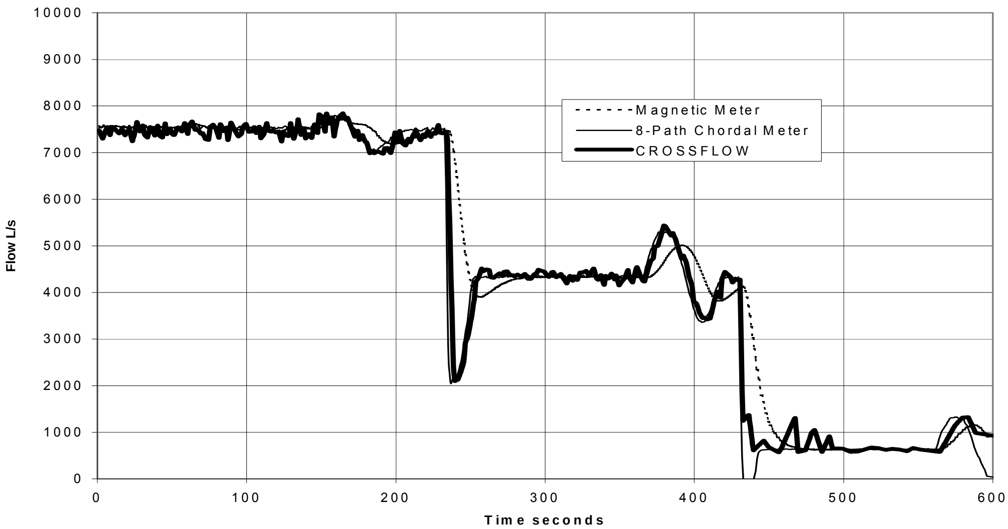

Figure 6: Flow Change Response Testing

Figure 7. Piping Geometry with Combination of Elbows Upstream of the Location of the UFM Transducer

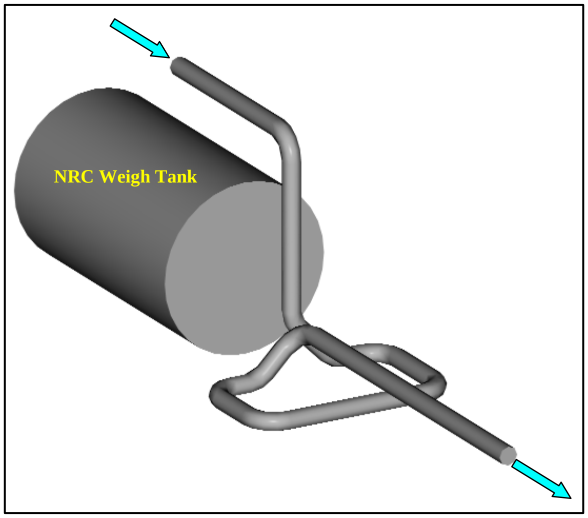

Figure 8. Test Model of the Piping Configuration, shown in Figure 7

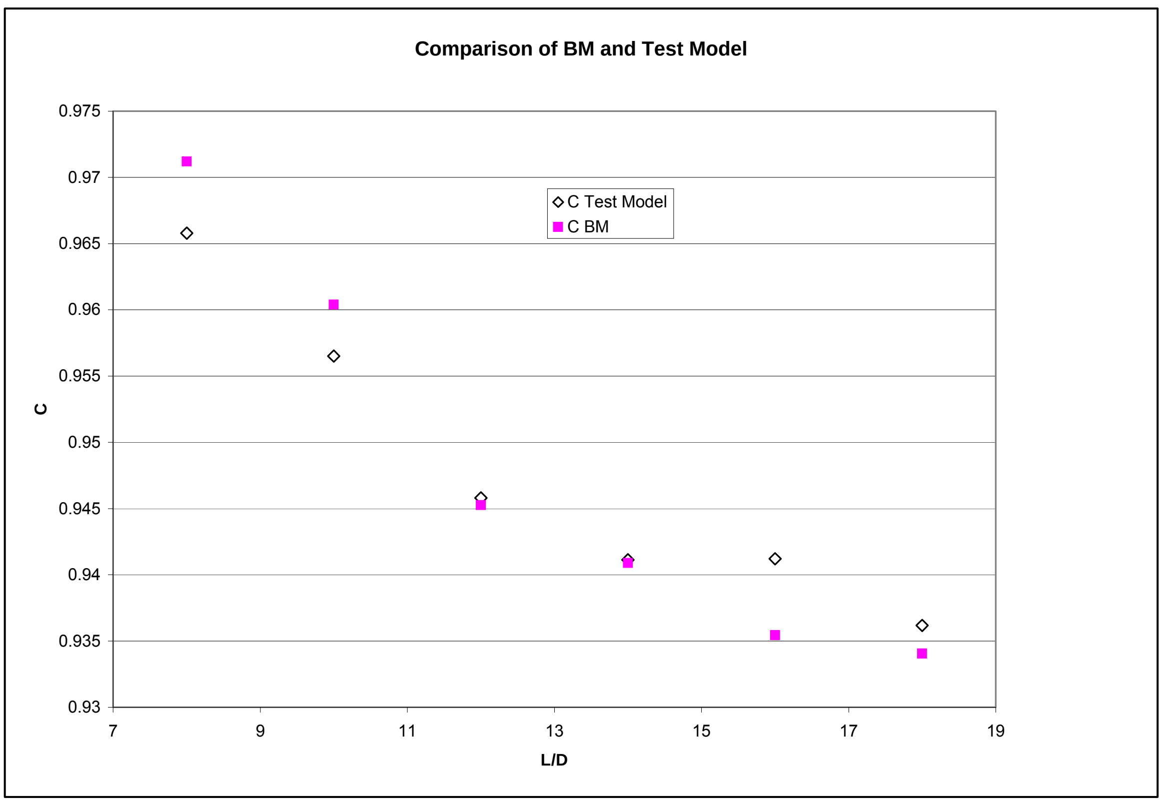

Figure 9. Comparison of Test Model and BM Piping Configurations

Parameter	Uncertainty (%)
C_f	0.63
ρ	0.10
A	0.242
L	0.100
τ	0.128
Flow Freq	0.100
Noise	0.456
Total	0.842

Table 2: Typical Uncertainty Values

Meter Location (see figure 2)	Pipe Length from Disturbance (inches)	Pipe Internal Diameter (inches)	Length Divided by Diameter From Disturbance
PVSP-1	86	10	8.6
PVSP-2	1165	19	61.4
PVSP-3	340	23	14.8

Table 3: Pipe Lengths and Length/Diameter (L/D)

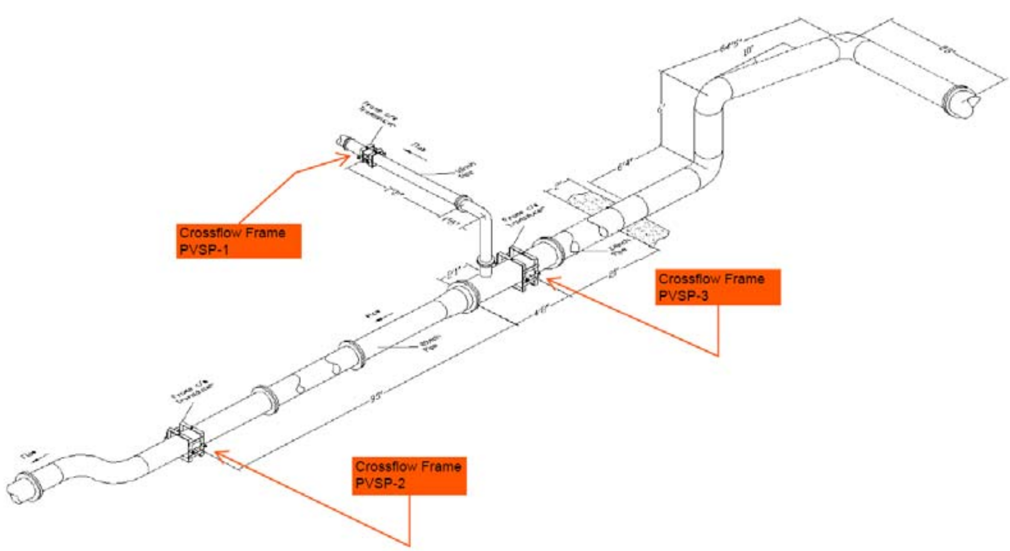

Figure 10: Spray Pond Supply line sketch

True North Consulting - AMAG Spray Pond Supply Flow Measurement Demonstration Demonstration Test Results		
Parameter	**Test 1 Corrected Flow (no liner impact)**	**Test 1 Corrected Flow for 12 mil liner**
20 INCH Line Flow PVSP-2 Flow (gpm)	14780	14743
24 INCH Line Flow PVSP-3 Flow (gpm)	16375	16341
10 INCH Line Flow PVSP-1 Flow (gpm)	1553	1549
Estimated Uncertainty	**Without Liner %**	**With Liner %**
10 INCH	1.59%	2.06%
20 Inch	0.81%	1.06%
24 Inch	1.16%	1.36%

Table 4: Test Results

Test 1 Full Flow DC In service (gpm)	**Test 2 Calibration with DC Isolated (gpm)**
14780	16082
15973	15678
1553	NA
Correction Factor For 24 INCH Meter	2.515%

Table 3: Calibration Factor

APPLICATIONS AND INTERPRETATIONS OF VIBRATION ANALYSIS TO SOLVE PUMP PROBLEMS

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Abstract

The paper presents a field study of two similar pumps, horizontally split casing type, having reliability problems. Vibration analysis was performed on both of them, indicating vibration values exceeding the allowable. Other parameters, such as temperature, were measured. The initial conclusion pointed poor support of the entire structure, which was initially believed to be the root cause of failures. However, upon further investigation, a much simpler explanation to the problem was discovered. The paper presents an unusual twist to problem root cause identification and a solution. Practicing plant engineers, vibration specialists, and maintenance personnel will benefit from this paper, presenting an interesting perspective into balance sophisticated troubleshooting methods against a simpler, and often more pragmatic, approach to machinery problem troubleshooting.

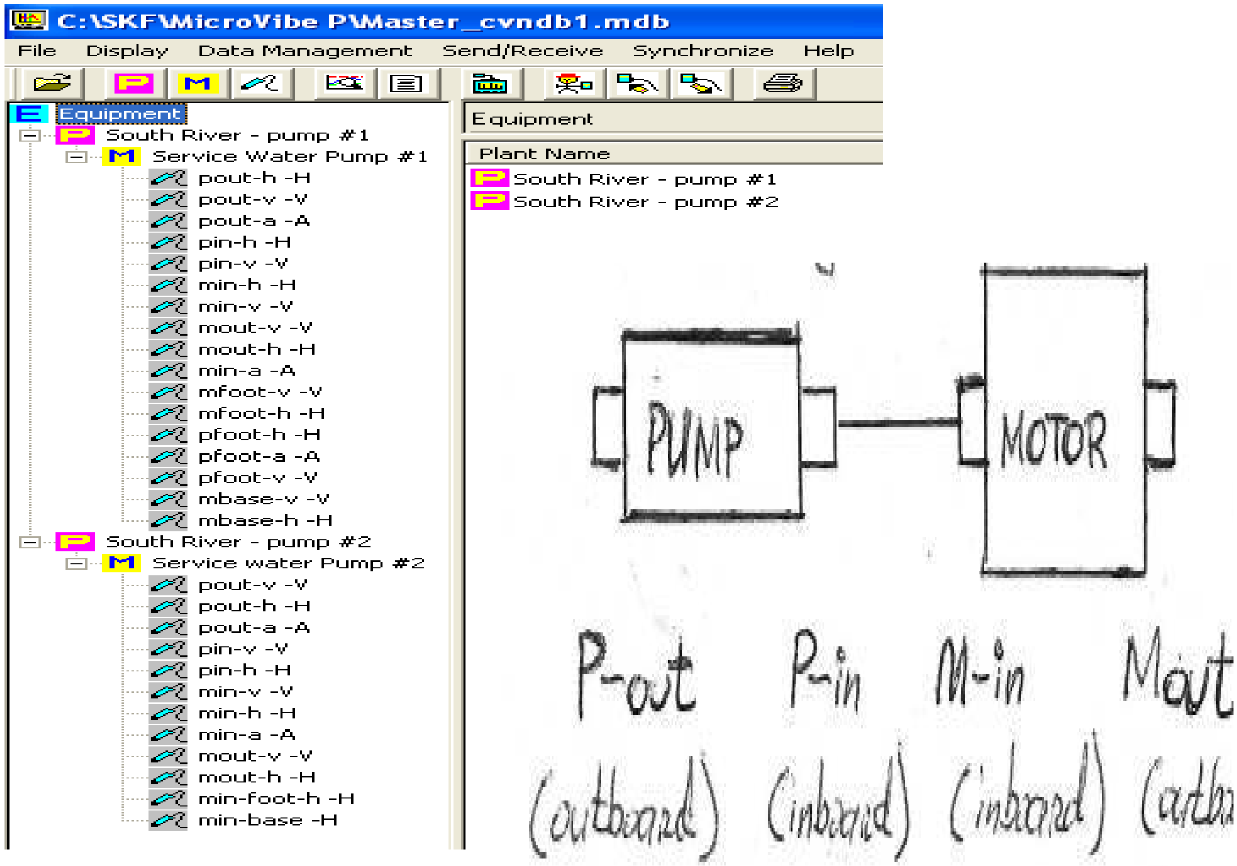
Introduction

A processing plant issued a distress call:

> ***"We got a pump problem. Motor keeps tripping. We had a pump rebuilt recently. Help!"***
>
> *Maintenance department requested troubleshooting assistance with a problematic double suction horizontally split case pump. The problem was accompanied by high vibrations, and a full fast fourier transform (FFT) analysis was being prepared. There was little time to communicate the exact details of the problem, and troubleshooting started before all facts were fully presented. As follows form the paper, a surprise discovery of a root cause followed, a valuable lessen in reviewing the facts first, before sophisticate instrumentation is called for a full investigation.*

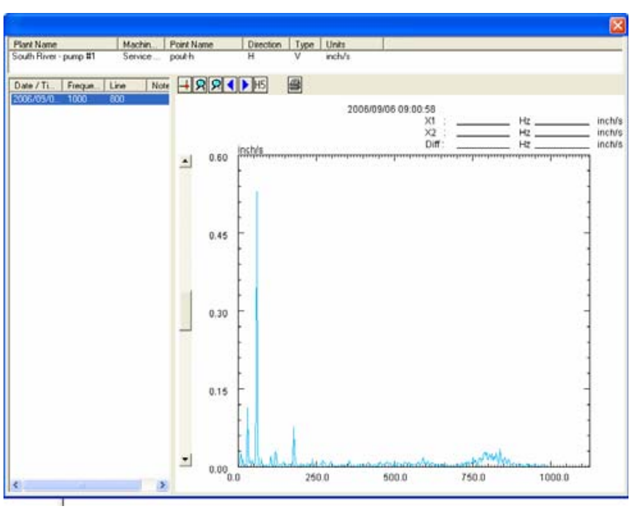
Figure 1: Pump 1 – front   Pump 2 - background

Figure 2: Vibration analysis usually begins with a simple lay-of-the-land review

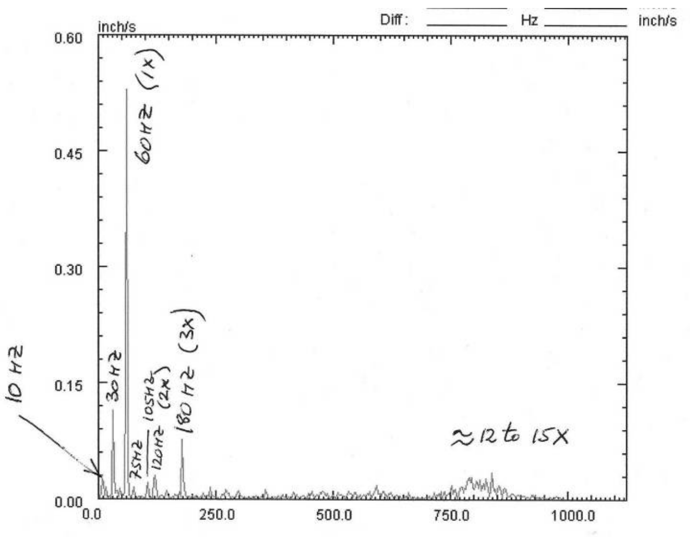


FFT analysis is performed, taking readings at (4) points, as illustrated above: inboard and outboard locations of the pump and motor, in three planes: horizontal, vertical and axial. As the following slides, these locations are identified at each slide, top-left corner, per nomenclature explained on this slide, above.

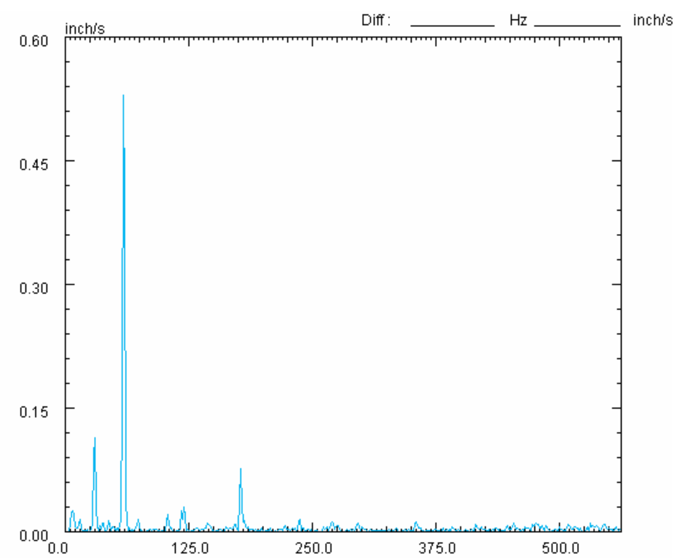
PUMP 1



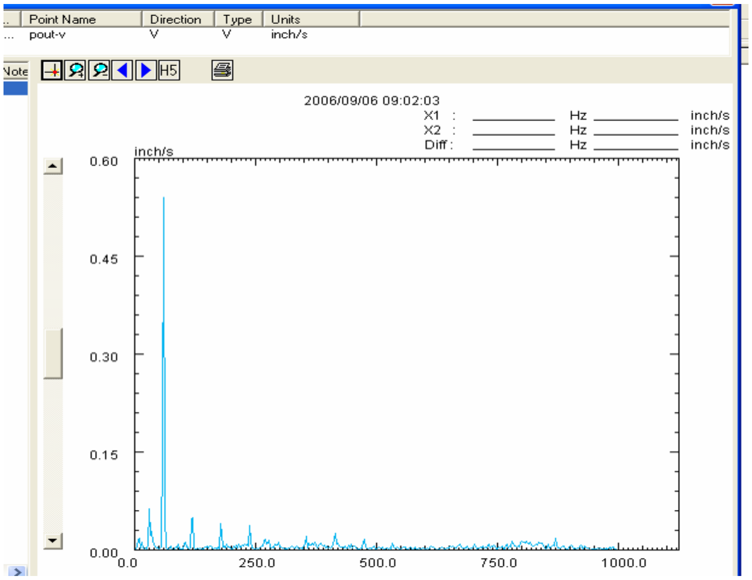
This is how a typical spectral analysis signature looks like for a pump #1.



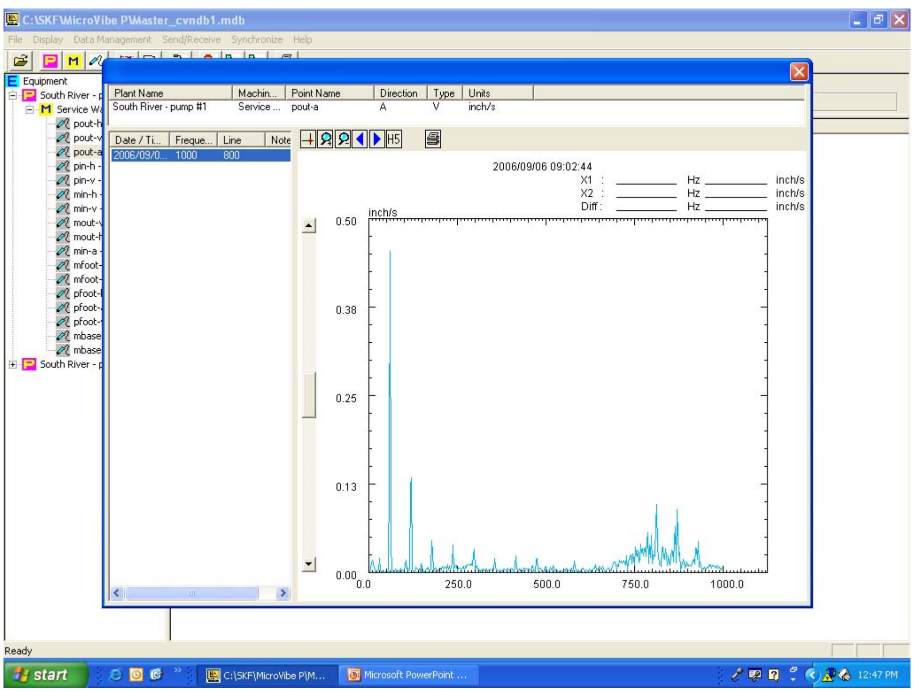
Multiples of the operating speed frequency are noted in their values as well as 1X, 2X, and so on.



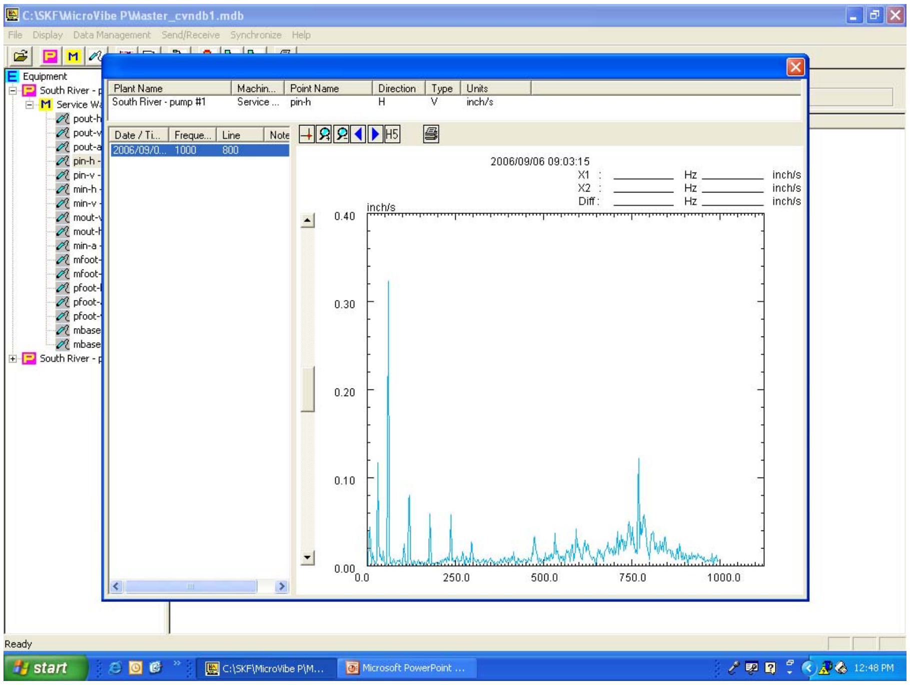
For closer resolution and detail, a frequency axis can be expanded as shown on the figure above.



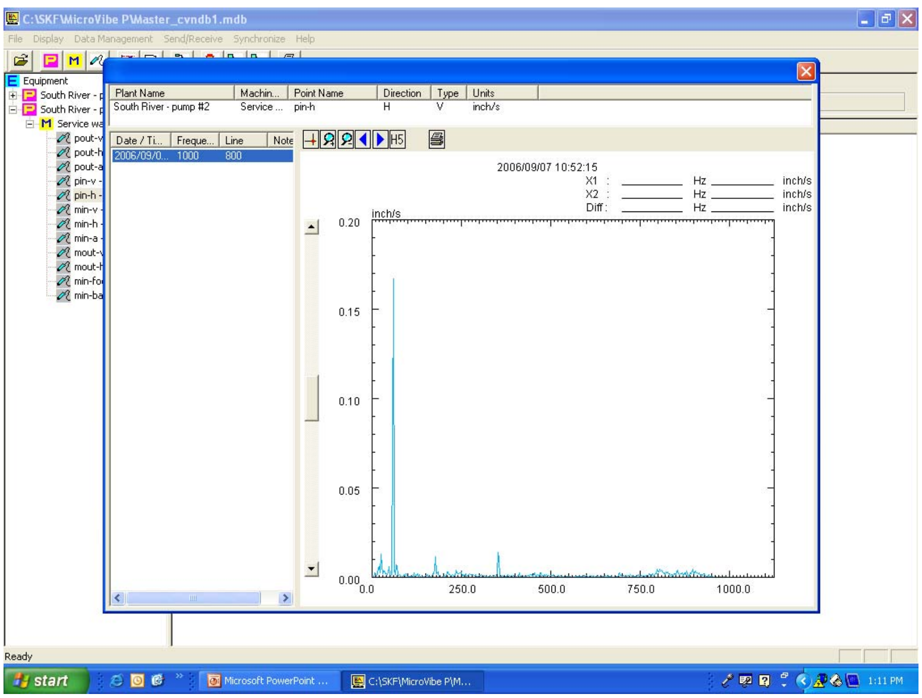
Walking thru the slides, we can see that pump outboard bearing shows high vibration in vertical direction, and 1X (running speed).



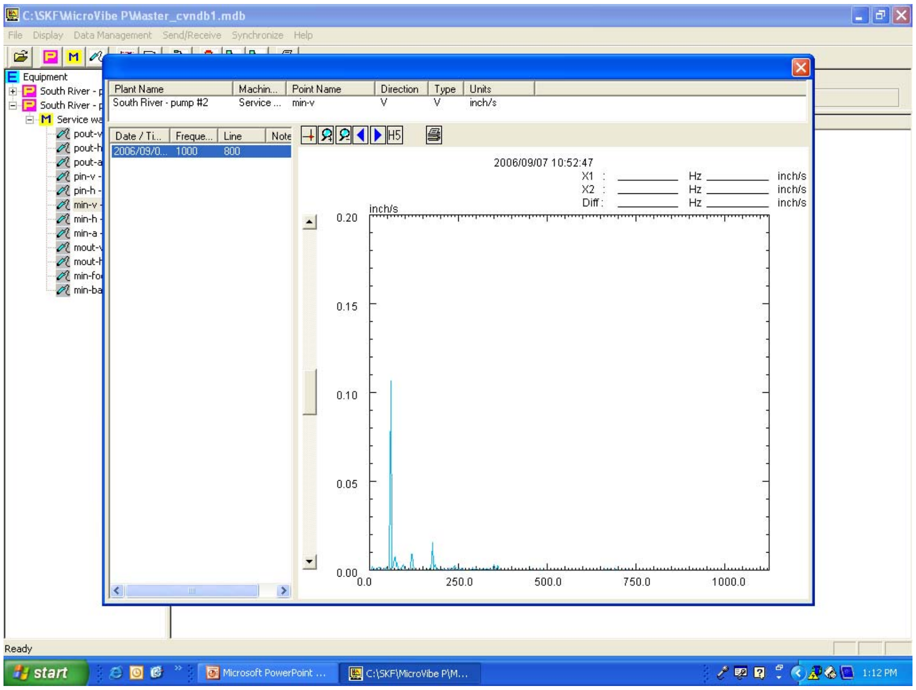
Axial component of vibration is also high…



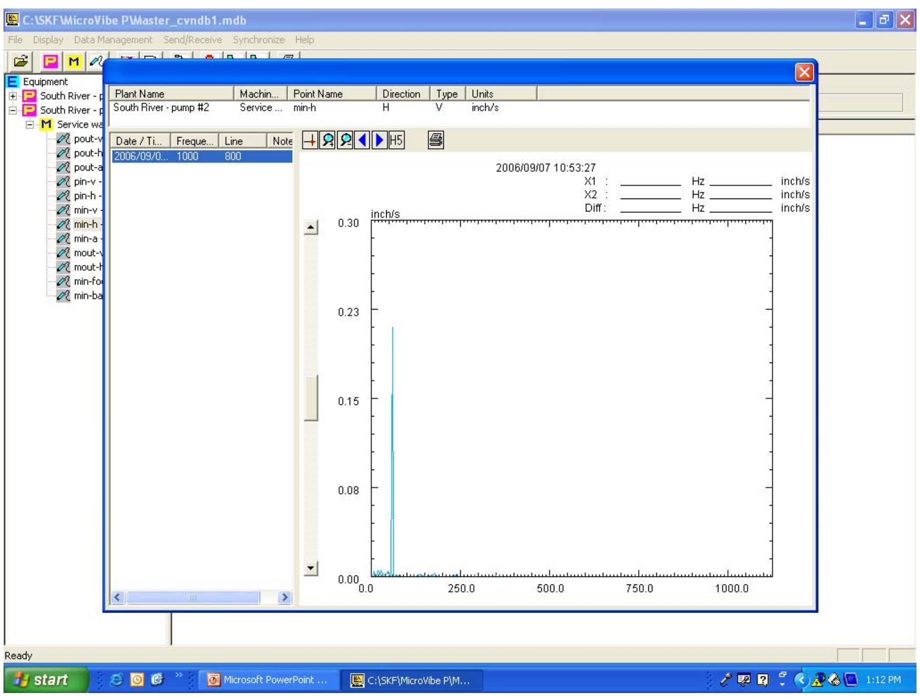
Inboard pump bearing is vibrating somewhat less, but still high.



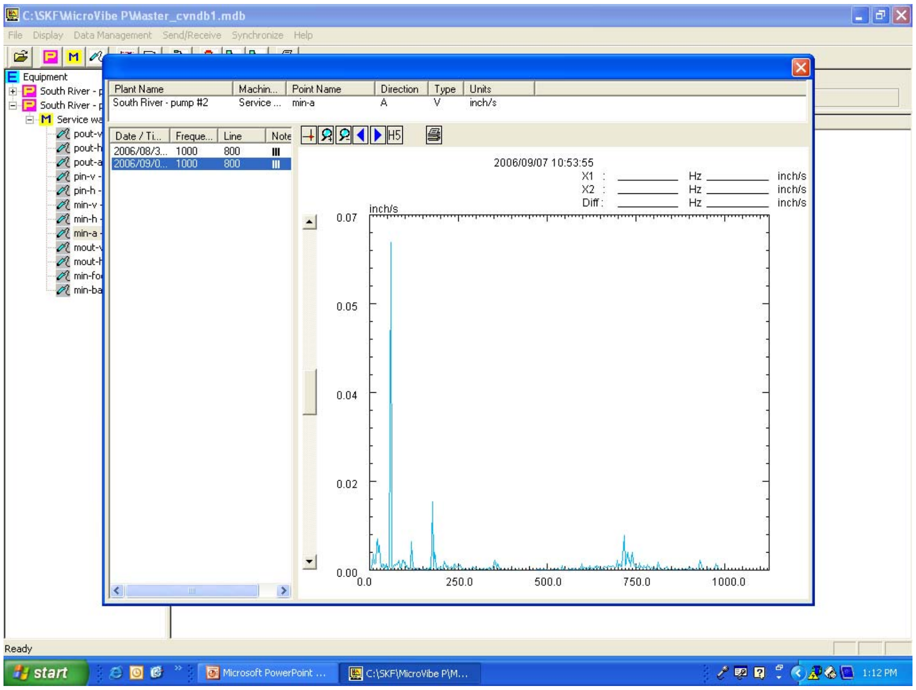
In horizontal direction, vibrations are lower.



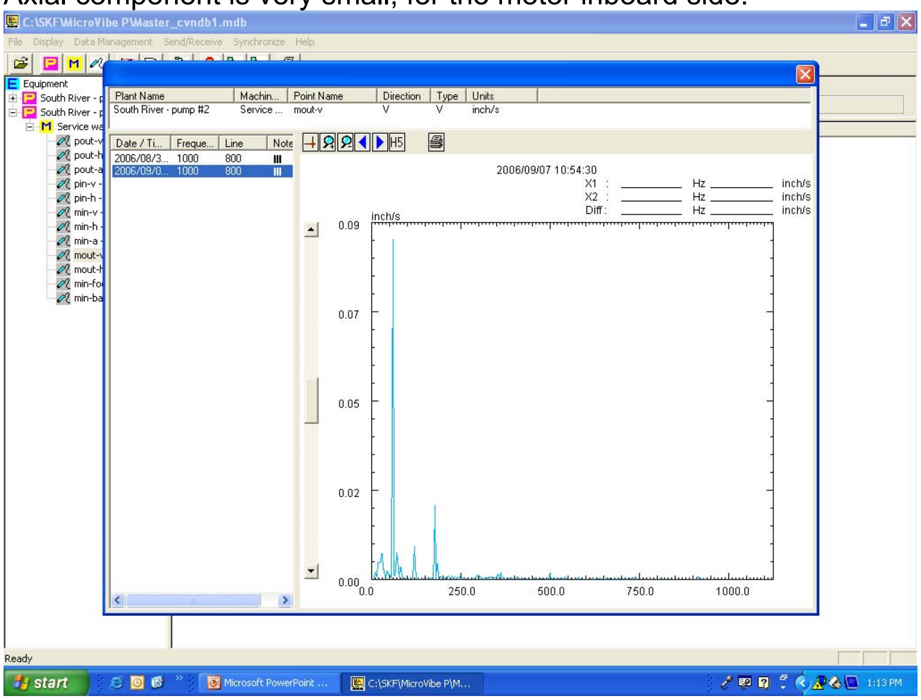
Motor inboard bearing show relatively low vibration in vertical direction.



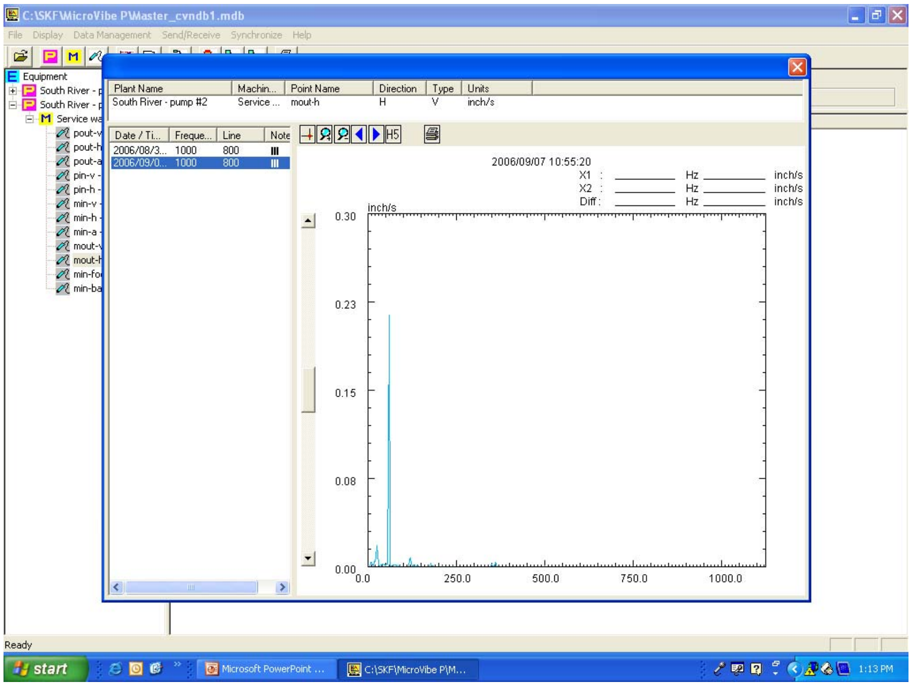
Horizontal distraction is also not excessive.



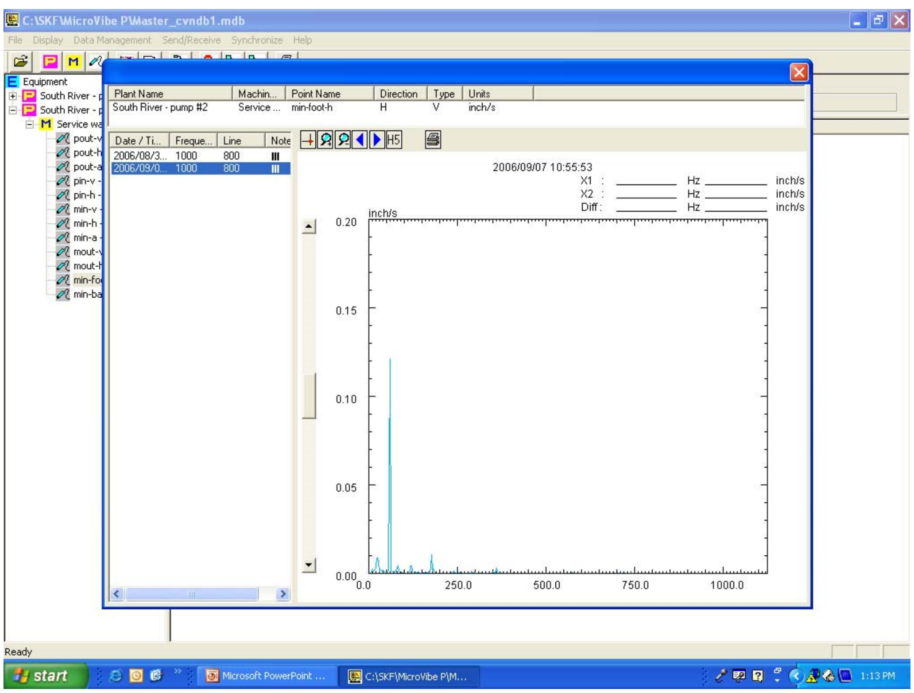
Axial component is very small, for the motor inboard side.



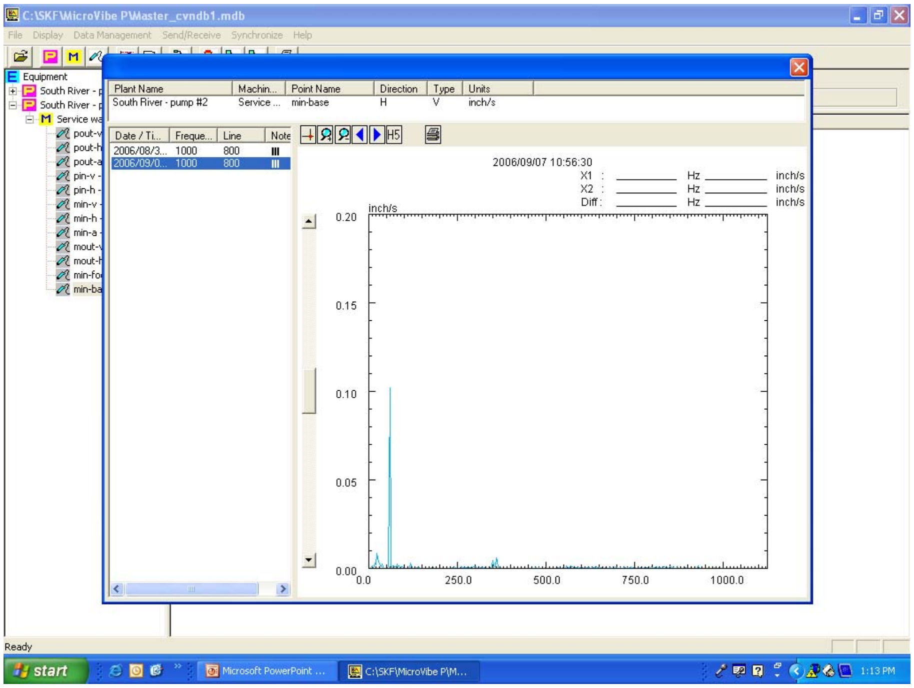
Outboard motor vibration is likewise small.



Horizontal vibration at the motor bearing is moderate.



Vibration at the motor feet are high for what they normally would be expected.



Even the entire baseplate is moving.

Next, let's put this all together at a table and review it:

Pout-h	~ 0.23 in/sec, pk
Pout-v	~ 0.14
Pout-a	~ negligible
Pin-h	~ 0.17
Pin-v	~ negligible
Min-h	~ 0.22
Min-v	~ 0.11
Min-a	~ 0.06
Mout-h	~ 0.22
Mout-v	~ 0.08
Pin-foot-h	~ not taken
Pin-foot-v	~ not taken
Pin-foot-a	~ not taken
Min-foot-h	~ 0.12
Min-foot-v	~ not taken
Min-base-h	~ 0.10
Min-base-v	~ not taken

Observations:

An entire train (motor, pump and supporting structure) exhibits low vibrations.
No value exceeds 0.3 in/sec, which could be considered as an approximate threshold, or an indication, of a beginning of trouble.

What's wrong?

What about temperature?

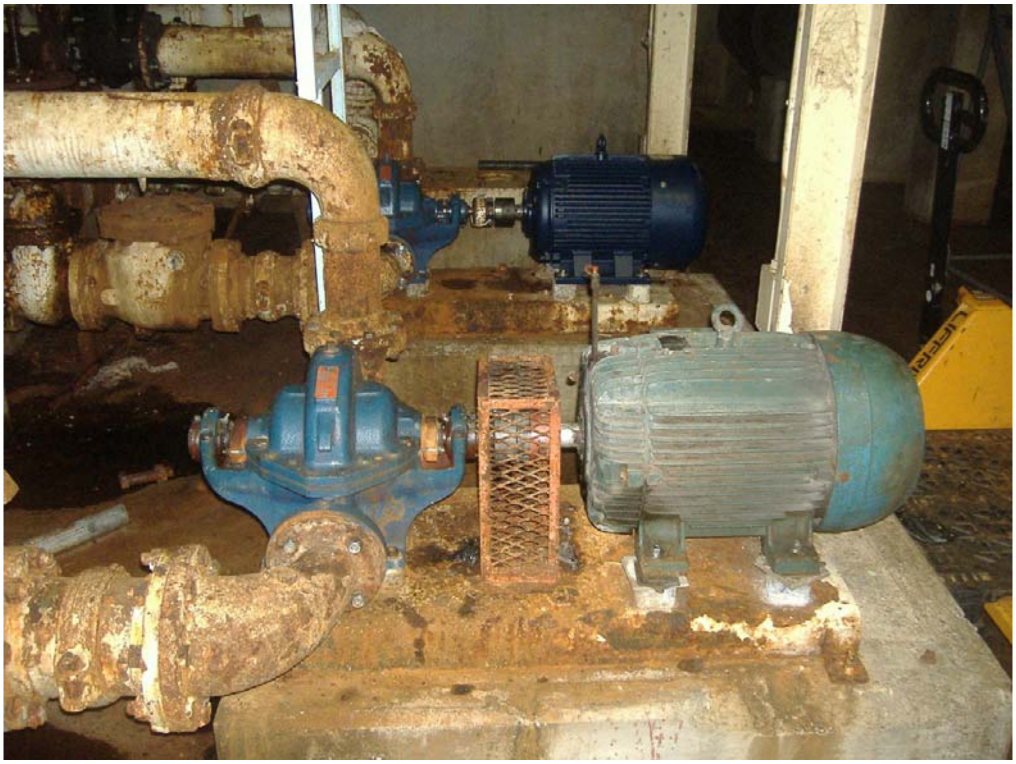
Pump 1
140 deg. F

Pump 2
Pump outboard temperature is too hot, 175 deg. F, and motor inboard bearing is hot also, 165 deg. F. This is somewhat puzzling: and implies that there must be reasons other then vibrations that contribute to a problem.

It is possible that if bearings were not lubricated, or lubricated improperly, then their high temperature might be an indication of lubrication issue, but since a rebuild was done relatively recently, there has not been enough time for hot running bearings to accumulate sufficient damage and begin to exhibit vibrations, which normally follow somewhat later. If so, we should begin to see increased vibrations relatively soon, unless the problem is corrected.

Moisture in windings could be another reason for trips, although normally would be considered an unusual event.

What's wrong with this picture below?



It is misleading: the front unit looks worse, and thus we assumed it is the one with a problem that a maintenance manager reported. Although this unit may indeed have high vibrations which is what we found – it is not the one he complained about!

As he later told us - the new motor for unit #2 was wrong - low horsepower. They replaced a 75 hp motor with a 40 hp wrong motor. No wonder amps cause the motor to trip!

Conclusions

The reason for the motor trip was simple: during the pump repair of unit #2, the motor was replaced with the incorrect horsepower rating. This resulted in amps to be excessively high for the smaller motor, causing trips on amps overload.

Unfortunately, a too-quick-to-act application of vibration analysis missed this simple, yet critical fact, resulting in wasted time in unnecessary analysis of a problem that did not exist. A simpler, more thoughtful review of the facts, before troubleshooting process started, would have uncovered these detailed, and a quick solution would follow.

Work Management & Preconditioning, Compatibility Achieved

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ABSTRACT

The paper will be on the subject of Preconditioning of Technical Specification and ASME Code {IST} & Appendix J required component and system surveillances. The subject of preconditioning has been a recurring issue across the nuclear industry since NRC Information Notice 97-16 was issued on April 4, 1997. Some 10 years later, nuclear plants are still receiving NRC Notice of Violation (NOV's) and Institute of Nuclear Power Operation (INPO) Area for Improvement (AFIs) for occurrences of unacceptable component and system test preconditioning. The paper will illustrate through the use of actions, practices, policies and examples which are complimentary to existing work package planning and reviews, that unacceptable preconditioning can be pre-identified and prevented.

Several other documents have been written attempting to address the issue of preconditioning, when it is "acceptable" and when "unacceptable". Some of these documents include NUREG 1482 (rev 0 and 1), The 1997 NRC IST Workshop Summary, a paper presented during the 7th NRC Pump and Valve Symposium in 2004, and more recently an In-Service Test Program Owners Group (ISTOG) paper on preconditioning.

The primary purpose of this presentation will be to address issues previously identified and addressed either by the NRC or the references discussed above, and to provide discussion of more recent issues associated with preconditioning. Although the aforementioned documents have provided some degree of resolution regarding the determination of the acceptability of preconditioning, areas requiring clarification still exist.

This guidance has been developed with an overall approach for addressing the preconditioning issue. This presentation will provide a summary of the guidance co-developed by utility and consultant personnel in the form of a standardized "general methodology" which can be readily adapted for use at other nuclear plant sites. Utilities that desire to develop and establish site or fleet guidance as to what constitutes acceptable and unacceptable preconditioning may find this paper useful in that endeavor.

Additionally, the paper will illustrate, through the use of examples, the various types of typical maintenance tasks that fall under the NRC umbrella {NRC Inspection Manual, Part 9900} of acceptable preconditioning. Identifying such activities and incorporating them into a plant's work practices as allowable will prevent “reinvention of the wheel”, each time one of these activities appears on the schedule. Ensuring that as-found component and system conditions are preserved, when required, in advance of testing to ensure regulatory compliance, along with optimizing work prioritization and performance, are major goals for presenting this paper.

Introduction

The USNRC in 1997 issued Information Notice (IN) 97-16, "Preconditioning of Plant Structures, Systems, and Components Before ASME Code Inservice Testing or Technical Specification

Surveillance Testing." This notice detailed the NRC's longstanding concern with unacceptable preconditioning of components before the IST or Technical Specification (TS) surveillance testing, and the adverse effect the preconditioning might have on the validity of the tests. Since then, the NRC has included preconditioning in their Inspection Guidelines on Surveillance Testing, 71111.22, which requires an inspection of six surveillance activities (including at least one IST) each quarter and specifically identifies preconditioning as a significant surveillance test attribute to be reviewed. Additionally, the NRC devotes an entire section of the NRC Inspection Manual to "Maintenance Preconditioning Of Structures, Systems, And Components Before Determining Operability," Part 9900.

NUREG-1482, Revision 1, "Guidelines for Inservice Testing at Nuclear Power Plants," Section 3.5 addresses further the importance of testing in the as-found condition. This revision of the NUREG, attempted to provide additional guidance regarding the "the acceptability" and "unacceptability" of preconditioning, and attempted to provide further "guidance" on what facilities needed to do in order to determine if preconditioning was affecting the ability of the owner to detect and monitor for degradation. Specific examples were provided as to what, from a regulatory perspective, constituted concerns associated with preconditioning, in particular those activities associated with inservice testing. This guidance recognizes that the Code does not specifically require all inservice testing to be performed in the as-found condition however, that not testing in the as found condition may result in degradation mechanisms that may not be identified.

Additional papers and presentations were also presented at previous meetings and symposiums attempting to provide the industry with further guidance regarding preconditioning of components and the concern associated with this practice.

Regulatory Basis

Although the Technical Specifications and the ASME OM Code or Section XI do not address preconditioning or as-found testing for all components, the regulatory basis for requiring that components be tested in an "as-found" condition is identified in the Code of Federal Regulations (CFR) in various places. 10 CFR 50, Appendix B, Criterion XI, "Test Control," requires that all testing be performed to demonstrate that structures, systems and components will perform satisfactorily in service. 10 CFR 50, Appendix J, "Primary Reactor containment Leakage Testing for Water-Cooled Power Reactors," requires that prior to containment tests, no repairs or adjustments be made so that the containment can be tested in as close to the "as-is" condition as practical. Appendix J additionally requires that valves be closed by normal operation without any preliminary exercising or adjustments.

Technical Specifications generally do not provide specific guidance regarding the preconditioning of equipment prior to performance of the surveillance testing. 10 CFR 50.36, "Technical Specifications," specifies that surveillance requirements are identified as requirements relating to test, calibration, or inspection to ensure that the necessary quality of systems and components is maintained, that facility operation will be within safety limits, and that the limiting conditions of operation will be met. Although "as-found" testing of components is not specifically identified in the TS, it is required as a result of the NRC regulations described above.

Discussion

Any maintenance (preventive or corrective), performed on components could be considered preconditioning, since it's function is to improve the performance of the equipment. Preventative maintenance programs are normally repetitive tasks that are set up on a regular schedule, which may be more frequent than the functional test. As such, at times, the PM's are performed prior to a component being tested. While this is not expected to be a common occurrence, there is sufficient evidence that it does occur. Preconditioning is often referred to as activities that occur "just prior to" testing; however, using that terminology creates confusion regarding what "just prior to" means (e.g., during the same shift, within 7 days, etc.). For this reason, it is prudent to evaluate any activities performed between successive testing for unacceptable preconditioning.

Several examples of preconditioning have recently been identified by both regulators and individual stations that, in the opinion of many, are NOT truly affecting the ability of the facility to detect and/or monitor degradation of a safety-related component. One such example is a facility was questioned as to whether or not a "tagout boundary" valve when returned to service and exercised prior to performance of an IST required stroke and exercise test constituted "unacceptable preconditioning". The fact that the valve was preconditioned was not in question, the more significant question and one that was disputed for several days was, does this activity constitute "unacceptable preconditioning?" Upon further review, the action was deemed to indeed be "preconditioning" but, "acceptable preconditioning", which is in many cases the outcome of evaluations that have previously been performed as a result of "external" concern.

As a result of this and other identified instances of occurrence, it has now become difficult, at least in the minds of many, what the "actual" concern is regarding preconditioning. To many, it appears to be a "personal preference" or individual determination, to others it is just confusing and creates numerous "issues" which in many cases, detract from the true concern associated with preconditioning. That is whether or not the preconditioning affects the ability to detect or monitor for degradation.

The question becomes, which activities are acceptable and which are not, relative to ensuring the test results are valid. NRC IN 97-16 provided a number of examples of unacceptable preconditioning.

NRC IN 97-16 stressed the importance of obtaining meaningful results during IST in order to determine the degree to which a component has degraded, if at all, and to determine the component's ability to perform its intended function when required. This is especially important with regard to the Code's current initiative of performance based testing. The Code is moving towards requiring condition monitoring rather than providing prescriptive requirements independent of the component's performance or safety function. The Code has already added a non-mandatory appendix for check valves that is performance based. Relief valve testing frequency requirements are also performance based, although the test method requirements are still prescriptive. Other sections of the Code are being similarly being evaluated for the incorporation of performance based testing. Performance based testing is reliant on evaluations of data trends. Test requirements (including test intervals, test scope, and test methods) are determined based on the results of previous test results. If preconditioning has masked the test results, the whole basis for the testing scheme is unsound, and the component may be in a degraded condition longer than a component that is tested on a regular prescriptive schedule.

To obtain meaningful results, it is important to test the components/systems in the as-found condition and to avoid preconditioning to the extent possible. The Code currently addresses preconditioning and as-found testing, but only for snubbers and relief valves, which have some degree of performance based testing. Subsection ISTD-1510 of the Code states that snubbers shall not be adjusted, maintained, or repaired before an examination or test specifically to meet the examination or test requirements. ISTD-5221 states that snubbers shall be tested in their as-found condition regarding the parameters to be tested to the fullest extent possible. Appendix I of the Code discusses as-found testing for relief valves. No maintenance, adjustment, disassembly, or other activity that could affect as-found set pressure or seat tightness is permitted by Appendix I prior to relief valve testing.

ASME OM Code Activities

In 2004, the ASME OM Code Committee attempted to resolve or at least address this concern by providing some sort of definitive position regarding preconditioning. However, as a result of the insistence of "intent" and the difficulty in establishing a "viable" position regarding enforcement of "intent", this action was cancelled with no further activity within the ASME community.

In July 2006 the Inservice Testing Owner's Group developed and published a paper on Preconditioning which, for the most part, reemphasized the concern of the industry and regulators regarding preconditioning and the importance of identifying preconditioning during various aspects of Maintenance (corrective and preventative) and testing associated with SSCs. As a result of recent meeting with the ISTOG (January 2008), the Preconditioning paper is also being revised to provide for a more definitive determination of "as-found". However, the ISTOG is a relatively new group and the membership of the ISTOG is relatively small at present, therefore the paper is ONLY available to ISTOG members of which there are less than half of the utilities in the US. As a result this does not reach a significant population in the nuclear industry at present, nor, does it fully encompass numerous "foreign" owners of nuclear stations. Perhaps in a few years the ISTOG membership may grow sufficiently to provide for the total distribution of the position paper to the whole of the industry but, again there may be some "disconnect" between the regulatory position and the industry position regarding Preconditioning. Therefore, in the long run, this may ONLY provide for a platform for further discussion regarding this topic. The ideal solution would be to have the ASME provide for a more "accepted" position on preconditioning. As stated above this was attempted a few years ago with no resulting Code change. However, once again the banner has been taken up to try and develop and, have accepted definitions and guidance on preconditioning.

The current proposal on the table is to add the following definitions of "preconditioning", "unacceptable preconditioning", "acceptable preconditioning" and "as-found condition", as well as to provide a general requirement in ISTA, as follows:

Add to ISTA-2000, Definitions:

Preconditioning: The modification, maintenance, manipulation, or adjustment of a component before inservice testing. Activities such as cycling, cleaning, lubricating, agitating, or other specific maintenance or operational activities that may be performed prior to or during surveillance or inservice tests which may potentially bias the ability to adequately assess the operational readiness of the component. Preventive maintenance activities performed directly prior to a Surveillance or Inservice Test should be minimized to the extent practicable.

It should be noted that some Preconditioning is unavoidable and the ability to assess the operational readiness of a component should be the primary determining factor in evaluating whether Preconditioning is acceptable or not.

Acceptable Preconditioning: Preconditioning as defined above, which does NOT affect the ability or prejudice the outcome of tests and/or examinations which are performed to detect and monitor for degradation of a component.

Un-Acceptable Preconditioning: Preconditioning as defined above, which may affect the ability and may prejudice the outcome of tests and/or examinations which are performed to detect and monitor for degradation of a component.

As-found condition: The condition of a component between inservice tests without activities that could affect the ability to determine component degradation or bias the results of an inservice or Surveillance test.

Add ISTA-3200, As-Found Testing and Preconditioning

Where practical, components shall be tested in the as-found condition (i.e., the condition representative of its normal standby condition when called upon to actuate during an accident). Measures shall be taken to prevent routine and/or inadvertent actions that may be considered as preconditioning. Un-acceptable Preconditioning shall not be performed on components unless an "as-found" test is performed prior to the activity which results in unacceptable preconditioning.

The major differences in the Code changes from previously proposed Code changes would be to remove the required determination of "intent" from the Code wording, as this is difficult if not impossible to be ascertained.

(It should be noted that these proposed words are still under consideration and have not been approved for publication.)

NRC Guidance

NRC Inspection Manual part 9900, Technical Guidance, Maintenance – Preconditioning of Structures, Systems, and Components Before Determining Operability, provides definitions of preconditioning and questions which may be used and are used by NRC inspectors to determine if preconditioning of a component may be acceptable or would generally be considered unacceptable. The NRC Inspection Manual, Section C.1. provides definitions related to preconditioning and a discussion of acceptable and unacceptable preconditioning and states:

Preconditioning

> "The alteration, variation, manipulation, or adjustment of the physical condition of an SSC before Technical Specification surveillance or ASME code testing. Preconditioning may be acceptable or unacceptable."

Acceptable Preconditioning

> *"The alteration, variation, manipulation, or adjustment of the physical condition of an SSC before Technical Specification surveillance or ASME Code testing for the purpose of protecting personnel or equipment or to meet the manufacturer's recommendations. Preconditioning for purposes of personnel protection or equipment preservation should outweigh the benefits gained by testing only in the as-found condition. This preconditioning may be based on the equipment manufacturer's recommendations or on industry-wide operating experience to enhance equipment and personnel safety."*

Unacceptable Preconditioning

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

The NRC Inspection Manual on Preconditioning, Section D.2, provides a list of questions to be considered when evaluating whether an activity could be considered unacceptable preconditioning:

- -Does the activity performed ensure that the component will meet testing acceptance criteria?
- -Would the components have failed the test without the activity?
- -Does the activity bypass or mask the as-found condition? (Including where trend data could be compromised.)
- -Is the activity routinely performed just before the testing?
- -Is the activity performed only for scheduling convenience?

Some surveillance testing cannot be performed without disturbing or altering the equipment (e.g., attachment of test leads, pneumatic or hydraulic supply lines, disconnection, realignment, and installation of jumpers). Any such disturbance or alteration is expected to be limited to the minimum necessary to perform the test and prevent damage to the equipment. Additionally, some equipment is cycled as a result of other components' test procedures. For example in the case where multiple valves are controlled by one switch, the test procedure must address rotating which valve is as-found tested, otherwise only one of the multiple components is as-found tested. Alternatively, the test may be performed by not using the single switch, but by jumpering each valve separately. Inadvertent preconditioning can also be caused by scheduling testing of systems that are related back to back, for example the HPCI/RCIC and RHR, as well as those that are apparently unrelated. The NRC Inspection Manual provides an example where the diesel generator air start system was scheduled after the EDG test, and the air-start system was operated during EDG pre-test preparation.

For valves, NUREG-1482, Section 3.5, notes that the as-found condition is generally considered to be a valve without pre-stroking or maintenance. It is considered preconditioning when air-operated valve stems are lubricated prior to performing stroke time testing. The practice of operating a pump for warm-up and the practice of venting immediately before performing the surveillance tests are examples of unacceptable pump preconditioning provided by the NRC in the Inspection Manual.

Industry Concerns

This broad definition of acceptable preconditioning could be interpreted to include valve cycling, stem lubrication, and other activities recommended by most manufacturers. However, the NRC specifically identifies these activities as potential unacceptable preconditioning practices, unless performed for the purposes of personnel protection or to prevent equipment damage that may occur if the practice was not performed prior to the surveillance test. The NRC notes that any preconditioning should be evaluated and documented in advance of the surveillance. The evaluation should demonstrate that the ability to assess operational readiness and trend degradation is not adversely affected, thus ensuring design and licensing bases remain satisfied.

Examples of acceptable preconditioning provided in the Inspection Manual include the practice of “barring over” diesel generators to prevent damage due to hydro-locking, and periodic pump venting if technically evaluated, and adequately controlled to ensure the amount of gas vented would not have adversely affected pump operation. Another example of this type of acceptable preconditioning would include vendor recommended periodic pre-lubrication of pump and/or driver bearings.

The effects of scheduling on preconditioning are also discussed in the Inspection Manual. One important consideration is the effect of scheduling seemingly unrelated activities. An example of potentially unacceptable preconditioning caused by test scheduling provided in the Inspection Manual describes a practice whereby a diesel generator was rolled over for pre-lubrication and hydro-locking prevention using the air start system immediately prior to performing the air start valve inservice test (IST) surveillance. Other examples include the performance of a refueling outage frequency simultaneous start of all Auxiliary Feedwater Pumps surveillance prior to the quarterly IST pump and valve test, the practice of “swapping” breakers to facilitate the 4160V breaker maintenance program, and the sequencing of the quarterly Internal Containment Spray Pump and Residual Heat Removal Pump tests as they utilize the same recirculation flow piping.

The NRC Inspection Manual could be interpreted as requiring an evaluation of all maintenance and scheduling activities for unacceptable preconditioning. Performing these preconditioning evaluations would be a major undertaking. As an alternative utilities should have a process for ensuring that scheduled activities do not cause unacceptable preconditioning, as well as documentation of the reviews of the specific maintenance or scheduled activities that may be questioned. This process should be documented in appropriate procedures to identify responsibilities and documentation requirements, and to ensure there is a common understanding of preconditioning throughout the station. Preconditioning is not only an IST program issue as it applies to all surveillance testing and must be understood by work planning and scheduling personnel, the Control Room staff, maintenance personnel and management.

Many utilities are still struggling with establishing a policy and implementing procedure(s) for preconditioning. Some have included a simple precautionary statement in their IST program procedure or have work planning procedure caution statements that focus on the preconditioning aspects presented in IN 97-16; i.e. intent and successful test performance and focus on the component that is having the maintenance performed on. These statements or cautions do not provide the work planner or task implementers the required understanding of preconditioning. Kewaunee Power Station received a non-cited green violation for inadequate procedural controls for preconditioning. The issue began with an inspector's question about the acceptability of using in scope IST valves for an isolation or tagout boundary for corrective maintenance on a non-IST pump minimum flow recirculation line check valve. Following the corrective maintenance the quarterly IST test would be used as a retest to pressurize the recirculation piping. The quarterly test would also perform the stroke time tests for the valves used as isolation boundaries. During subsequent discussions with the regulators the issue soon became how the station considered and evaluated preconditioning, not specifics of this event. The wording of the violation is presented here to provide awareness of the issue and specific concerns of the regulators:

"The inspectors determined, following review of plant procedures, interviews with plant management, and a review of NRC guidance on preconditioning, that plant procedures were not adequate to support proper consideration of preconditioning. These procedures did not discuss or provide a definition for acceptable versus unacceptable preconditioning, did not provide guidance on the scope of components which should be included in such consideration, and did not identify which personnel were responsible for the consideration of potential preconditioning. Therefore, the inspectors determined that, due to inadequate procedures, adequate consideration of preconditioning of ICS-202 and ICS-2B during maintenance activities was not given before the surveillance procedure that tested these valves was conducted"

In response to this violation Kewaunee benchmarked Seabrook Station's preconditioning work control procedures as Seabrook was identified as having a good preconditioning program in the NUREG for the Seventh NRC/ASME Symposium on Pump and Valve Testing. This benchmarking provided valuable information to improve the Kewaunee Power Station work management procedures. A technical position on preconditioning was also placed in the IST program document. This position was based upon information received from Seabrook, informal benchmarking with other sites and the ISTOG position paper on preconditioning. The technical position contains the same basic elements as the work control procedure to ensure a consistent approach to define the aspects of preconditioning is used in both references. A copy of the technical position, for information, follows this paper.

Kewaunee is also evaluating a change in work scheduling. Typically maintenance items are performed early in work week and surveillances performed late in work week. This schedule results in many preconditioning concerns/questions and reversing this schedule by performing the IST surveillances first should minimize preconditioning concerns. However, there are maintenance rule concerns associated with unavailability time that need to be properly considered.

In addition to scheduled maintenance and testing, there also exists the potential for preconditioning as a result of human error or equipment failure and the resulting corrective maintenance and corresponding post-maintenance testing. Inadvertent activities such as these may result in the surveillance test not reflecting the as-found condition. When these unscheduled activities occur, they should be so noted on the surveillance test record/corrective action documentation, and properly evaluated when assessing the condition of the component being tested. The five questions from the NRC Inspection Manual that were presented earlier should be answered in the evaluation.

Potential Resolutions

Programmatic measures which could be taken to avoid/prevent preconditioning of Components/Systems before performance of ASME Code Inservice Testing should be established. Technical Positions that provide the bases and conclusions of why Preconditioning is acceptable when allowing specific activities to occur should also be considered. In addition, activities which could be construed as "unacceptable preconditioning" should be identified and appropriate actions, such as "as found" testing or changes to PM or operational activities which are deemed to "bias" the results of surveillance tests should be performed, so as to minimize the affect of the "unacceptable" preconditioning on the SSCs.

There are several examples of methods being used to address the concern with Preconditioning, that have been incorporated in the industry, some of these are discussed below:

1. A Constellation Energy site has established a technical position that addresses the practice of performing periodic valve lubrication in advance of and/or during IST program AC powered motor-operated valve (MOV) IST stroke time testing/exercising. The technical position is described in the site's IST Program Pump & Valve administrative procedure and also is resident in the repetitive maintenance tasks for the affected IST MOV's. The technical position provides the basis as to why the lubrication maintenance activity has no impact on MOV as-found stroke times and why all concerns of test preconditioning are rendered moot.

2. A Constellation Energy site has IST administrative procedure guidance that prohibits the performance of a) valve stem packing adjustments that will result in exceeding the valve's current "signature torque" value, and b) preventative maintenance which performs valve stem cleaning, valve exercising, or repositioning (i.e. repositioning the valve for a reason other than conducting a surveillance test) prior to conducting the valve test.

3. A Constellation Energy site has IST administrative procedure guidance that prohibits the performance of pump warm-ups, preventative maintenance which performs motor "bump" tests, casing venting or pump barring prior to conducting the pump test.

4. A Constellation Energy site has IST administrative procedure guidance that prohibits performing multiple strokes of power operated valves to obtain valve stroke times when the subject valves are operated from a common "Master" switch.

5. A Constellation Energy site has IST administrative procedure guidance that requires maintenance work package heightened scrutiny when performing electrical maintenance that affects relays, timers, breakers, etc. that may result in unacceptable preconditioning of IST power-operated valves.

6. A Constellation Energy site has administrative procedure guidance that addresses the scheduling and performance of an IST or Technical Specification surveillance in advance of another IST or Technical Specification surveillance where the initial test will result in unacceptable preconditioning of the subsequent test. This practice places special emphasis on train-related preconditioning for components having the same base test interval.

7. A Constellation Energy site has an administrative policy (driven by procedure) that mandates surveillance tests that are performed on an "infrequent" basis (Cold Shutdown or Refueling interval) shall have the associated component "as-found" tested before any maintenance activity is conducted which might affect the component's performance parameters. Exceptions to this policy must be evaluated and documented.

In addition, to the administrative controls that have been put in place as discussed above several facilities have established Cross-Discipline Guidance Documents as well as other “tools” that also provide a site-wide barrier to “unacceptable” Preconditioning. Some of these “tools” are discussed below:

1. Ensure procedure writing guidelines include specific guidance on how to ensure test preconditioning issues are factored in when creating new or revising existing procedures. The result will be a “boilerplate” approach to ensuring sensitivity to unacceptable test preconditioning and its prevention.

2. Consideration should be given to the amount of time it takes for an unacceptable preconditioning activity to have its as-found condition altering impact eliminated. Creation of a matrix which addresses the elapsed time aspect is useful in screening and assessing activities which can be authorized without fear of preconditioning the performance of subsequent tests.

3. Ensure the topic of test preconditioning is an integral part of all site training programs, especially initial training so it is ingrained from the start of employment.

4. Enlist the site’s Work Management System to flag specific work activities that have been identified as contributors to unacceptable test preconditioning so heightened scrutiny of their scheduling and conduct will automatically be initiated.

Other actions or evaluations which could be taken to further reduce the concerns associated with Preconditioning, especially “unacceptable” Preconditioning are listed below:

Special consideration should be given to performing as-found testing of components that can only be tested while the plant is offline or in a refueling outage. Due to their infrequent tests and limited performance data, essentially all preventive maintenance will constitute unacceptable preconditioning by masking the as-found condition. An exception would be corrective maintenance that is intended to restore the component to optimum performance based on previously identified degradation. In such instances, the benefit of an as-found test is of little or no value.

Utilize contemporary preconditioning Operating Experience (OE’s) to reinforce existing policies or define new ways to enhance the site’s ability to prevent unacceptable preconditioning.

Site participation in the various industry Owners/Users groups is vital to staying informed about preconditioning events that may not have risen to the level of formal OE’s. Such groups utilize, periodic meetings, teleconference calls, website bulletin boards and query systems to share, discuss and enlist advice about dealing with test preconditioning, both acceptable and unacceptable.

CONCLUSIONS

In conclusion, for practical purposes, “as-found” testing will yield the most information concerning the ability of the component to perform as required. However, maintenance must be performed to ensure the continued operability of the component. Proper review and documentation of the maintenance is required. Without documentation, the intent of the maintenance can be misconstrued and inadvertent preconditioning can occur. PM activities performed directly prior to an Inservice Test should be minimized.

In most cases, the most effective means of eliminating preconditioning concerns is to perform testing in the as-found condition. Since that approach often is not practical, identifying in advance acceptable acts of preconditioning is encouraged and seen as the most efficient and regulatory compliant manner to address the issue. The acceptability or unacceptability of preconditioning must be evaluated on a case-by-case basis due to the extensive variability in component design, operation, and performance requirements.

Factors to be considered in the evaluation of preconditioning acceptability include component size and type, actuator or driver type, design requirements, required safety functions, safety significance, the nature, benefit, and consequences of the preconditioning activity, the frequencies of the test and preconditioning activities, applicable service and environmental conditions, previous performance data and trends, etc.

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Proposed Changes to the ASME Code Subsection ISTB

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Abstract

This paper describes proposed changes to the ASME OM Code Subsection ITSB, 2004 Edition through the 2006 Addenda, and provides the basis for why the changes are being made. These changes include (1) clarification of comprehensive pump test (CPT) flow rate and addition of periodic verification test; (2) an allowance to substitute the Group A test for the CPT if the CPT requirements for instrument accuracies are met; and (3) a relaxation of the high required action range for CPT hydraulic parameters (1.03 to 1.06).

Session 3(b): Valves II

Session Chair: Dr. Claude L. Thibault, Consultant

Trending Using Nonintrusive Check Valve Technologies

Ernie Noviello-Crane Nuclear

Abstract

Nonintrusive testing (NIT) on service sensitive and safety related check valves has been practically used for about the past fifteen years. NIT technologies are employed to provide check valve heath assessment for various styles, sizes, material composition and service applications during planned refueling outages or on line. NIT technologies provide a highly cost effective and reliable method to meet the requirements of the American Society of Mechanical Engineers (ASME), Code for Operation and Maintenance of Nuclear Power Plants (OM Code) for operability as well as balance of plant check valves for condition assessment. NIT technologies are instrumental in providing an effective means to schedule predictive maintenance based on known check valve problems and provide minimum or no impact on normal plant schedule and operation. Usually, the technologies can be applied during a scheduled system surveillance requiring no additional plant burden.

The NIT technologies for check valves have been validated through the Nuclear Industry Check Valve Group (NIC) Phase 1 through 3 (reliability of degradation detection) and Phase 4 (trending capabilities). An effective NIT program will utilize trending for the early determination of wear to components as a result of service-induced operation. This trending program will optimize valve and plant performance by scheduling necessary maintenance based on scientific evidence of a degraded or degrading condition. Routine disassembly of check valves for the sole purpose of inspection can result in equipment damage, personnel injury, lost time, and incorrect reassembly of which can constitute inefficient plant performance indicators.

Specific attributes exist for the different NIT technologies. Each attribute has value in the overall assessment of valve condition. Knowing what the attributes mean and how to trend NIT data takes understanding of the technologies and attributes. Trending NIT data can clearly assist in proper determination and scheduling of valve maintenance prior to valve failure.

Numerous operating plants have elected to invoke Appendix II, “Check Valve Condition Monitoring Program,” of the ASME OM Code for the check valves in the inservice test program. ASME OM Code Appendix II, Section II-4000(5)(b)(2), requires identification of parameters to be trended for the purpose of condition monitoring activities that would allow adjustment to the established test periodicity. Trending is a requirement and any adjustments to the test periodicity must be justified.

This paper discusses the NIT technologies most successfully used in the nuclear industry as well as their operation and specific parameters that can be trended. Trending will be discussed and operating experience will be shared in regards to degraded parts detection from numerous operating domestic nuclear plants. Trending software developed for NIT technologies will also be presented.

Introduction

Check valve NIT is one of the primary means used to test check valves in the nuclear power industry. In terms of testing to meet the intent of ASME Code Appendix II, it is required that trending be used to optimize activities. ASME OM Code Appendix II, Section II-4000, “Condition Monitoring Activities,” Paragraph 2, states, “Identify attributes that will be trended. Trending and evaluation of existing data must be used as the basis to reduce or extend the time interval between tests or examinations.” In order to take advantage of the optimization of check valve tests in accordance with the interval extension requirements, trends must be used to support the adjusting of test periodicities.

Failure of critical service check valves can have very serious implications including equipment damage requiring shut down of the plant. Trending for accurate valve condition assessment to accurately predict valve condition and the implementation of predictive maintenance prior to valve failure can be accomplished successfully if a clear understanding of the required elements of data analysis and comparisons are made. Significant aptitude, training and experience of test personnel are essential to a successful trending program as data acquisition is the primary key to successful trending. Corrupt or skewed raw data files will inadvertently change the results of the test data resulting in incorrect data trends that can be misleading and or inconclusive.

This paper will discuss ultrasonic, acoustic emissions, and pulsed electromagnetic (eddy current) NIT technologies. Each technology provides its own unique set of attributes in terms of valve condition assessment as well as the ability to trend changes in condition and or valve functionality (refer to Table 1 for NIT attributes that are trended). It is important to state that when trending valve condition, a presupposed trend should not be sought. In other words, trends will not take specific directional changes as related to specific valve degradation. Trends are likely to be as random in direction for one valve as they may be exponential for another valve. It is vital to understand that any change in data points that is clearly not a result of flow variations or other system initiated events should be considered as a true degradation. This of course depends on the valve service and condition of service or demand on the particular valve. It is important to consider that trending is most effective when utilizing at least two different NIT technologies as the information from one technology can be used to support the second technology.

NIT can be performed during three service conditions (steady state, stroke testing or depending on the technology being used, no flow conditions). Steady state is the condition when the valve is in service subjected to its normal operating behavior. This is the best opportunity to observe valve behavior as it relates to the potential for accelerate wear to occur.

Ultrasonic Technology

Ultrasonic testing (UT) of check valves has be occurring for almost twenty years in the nuclear power industry. There are several facets of UT used to support the nuclear power industry from flow measurement devices to critical weld, nozzle, and safe end inspections. Check valve UT is based on a pulse echo instrument. UT relies on the principles of pulse echo sound transmission. It generates high voltage pulses and sends them to a transducer displaying the energy in an A-scan display. The A-scan display indicates the amplitude and the depth of the sound reflections. The amplitude is a relative measure of the amount of reflected energy. The depth is the distance from the transducer to the target point (a point of sound reflection on the check valve disc assembly).

The transducer converts the pulses into sound waves. This is a result of piezoelectric properties of the transducer. If the conditions applied are correct, a large percentage of the sound wave is reflected from the disc assembly back to the transducer. The sound reflected back to the transducer is converted back to electrical pulses which are amplified and displayed on the UT reflectoscope screen as waveforms. In general terms, a sound beam is sent into the valve through the valve body and reflected off the obturator or disc internals. As the disc flutters, any change in movement of the disc is captured and stored as a raw-time trace representing movement in inches. It is this reflected energy that is used in conjunction with the software to determine key parameters such as disc-open angle in degrees off the seat and disc flutter or angular velocity. These parameters provide important information when compared or trended from test-to-test. The output information for the disc-open angle is based on a validated algorithm. This is the most quantifiable method available to predict the affect of disc flutter on valve condition and predict wear primarily to the hinge pin as well as other critical internal components depending on valve design. UT is limited to water-filled systems and is difficult to use on valves that contain larger grain structure such as cast stainless steels.

The initial stable flow test establishes baseline values for angular velocity, total disc travel, and average disc position (disc-open angle). Angular velocity is the total disc flutter measured in terms of degrees/second disc oscillations. The angular velocity obtained during the test can be compared to the stability number classifications in Table 2. Valves with a stability number between zero and four degrees per second are considered stable, stability numbers from five to eleven degrees per second exhibit flutter, and stability numbers twelve degrees per second and higher are considered unstable and may be unsuitable for use in their current application. The total disc travel is directly proportional to the stability number. When the stability number increases, the total disc travel increases. This will increase the wear rate. An engineering evaluation should be performed when the stability number increases or decreases by three degrees per second or if the average disc-open angle changes by five degrees or more in either direction. These changes could be due to various degradation combinations.

Two distinct changes will become obvious when trending UT data as wear to the hinge pin or other internal parts occur. First, the angular velocity will change by at least three degrees/second or may even transition into another distinct stability number classification. Second, the disc-open angle will change by several degrees generally moving closer to the seat. Both of these changes occur as the hinge pin mass becomes less and the friction that exists between the hinge arm and hinge pin is reduced. In other words, the element of friction that maintains a specific disc-open angle and angular velocity during a specific test is likely to change as wear to the hinge pin occurs. This increases the clearances between the hinge pin and disc arm cavity causing the disc position as well as degree of disc oscillations to change. Experience indicates that wear occurs as a result of disc flutter. These changes can be seen when trending UT data (refer to Figure 1 for UT trends determined to be the result of check valve wear).

UT can be performed during normal valve operation as this condition will place the valve in its normal state of operation. It is during this time, that the disc characteristics for a particular flow rate can be determined. It is critical to verify that the same flow rate is used when comparing data. For example, a reduced flow rate may cause increased fluttering and produce a higher stability number and total disc travel. This will give the appearance of a degraded valve. UT can also be used determine disc position by monitoring a fixed point on the disc assembly. This method is normally used to verify disc closed position to meet the requirements of the ASME OM Code. UT is one of the most accurate, effective, and practical ways to meet the ASME OM Code bidirectional test position requirement. Stroke times can also be trended for any potential for hanging up or binding of the disc assembly during valve cycling activities. This is performed using UT to monitor a fixed point on the disc while changing position of the disc from
closed-to-open or open-to-closed positions.

Although hinge pin wear is the primary end result of angular velocity, there are several other conditions related to wear that are obvious when considering UT. Hinge pin wear would be likely the result with swing, duo, and tilting disc check valve designs. The duo disc check valve design is also subject the other variations of degradations due to additional internal parts that do not exist in a swing check valve. Piston and nozzle design check valves do not contain hinge pins; therefore, the wear to these valve designs would be subject to other internal parts such as springs, guides, valve body, and the piston itself (refer to Figure 1 for examples of UT data trends of actual degraded conditions).

Signs of degradation when trending UT data for stable flow operation are as follows:

Worn Hinge Pin

high angular velocity - Excessive angular velocity is the major cause of hinge pin wear. The amount of angular velocity is determined from the test signature and transducer location. In general, a higher the angular velocity results in a higher wear rate. Changes in angular velocity from test-to-test are an indication of wear.

disc position - A degrading hinge pin can create a different value than recorded during baseline testing. Changes from test-to-test are an indication of wear.

Detached Disc

unable to locate the disc at predetermined position - If the valve is tested at the same flow rate as during a baseline test, the inability to locate the disc at the same position will indicate a change in valve operation.

extreme hinge arm flutter - The hinge arm will sometimes swing erratically when the disc is missing. Determining the angular velocity will help identify this phenomenon.

change in full open angle - The hinge arm can move to an abnormal position when the disc is missing. Quantifying the disc position will determine this phenomenon.

Broken Spring (Duo Disc)

disc remains at full open position at reduced flow rates - Tests have shown that a broken or malfunctioning spring will permit the check valve discs to remain open at lower than normal flow rates because there is no spring force to assist in closing. In some cases where there is no reverse flow or pressure, the discs could remain open at no flow condition. Determining an unusually high open-angle at reduced flow rates can identify this degradation.

Loose Disc Stud

extreme disparity between disc and arm flutter frequency - The hinge arm and disc flutter signatures can be compared for gross frequency differences.

backstop tapping - Severe backstop tapping can lead to stud failure and fatigue. The flutter signatures can sometimes indicate backstop tapping. Extreme changes would indicate the potential of wear.

Stuck Disc

unusual disc position - UT can identify discs at unusual positions for given flow rates. (Example - disc 20% open at 100% flow).

disc remains in same position for different flow rates - By monitoring the disc position for change at reduced flow rates, stuck check valves can be identified.

Acoustic Emission Technology

Acoustic emission (AE) technology is based on sound vibrations received from two piezoelectric accelerometers. One accelerometer monitors hinge pin vibrations while the other monitors backstop or seat vibration. The piezoelectric accelerometers convert sound waves from mechanical motion into an electronic signal.

There are three different approaches for using AE technology. The first approach involves raw data that is not processed, filtered, or used in conjunction with software algorithms. The raw AE data is acquired and displayed in magnitude over time. This mode of AE has no preset filters and will acquire essentially all data and is mostly used in conjunction with ultrasonic or eddy current testing during a stroke test. This AE mode allows the user to set the acquisition parameters according to specific needs. The gain, sample rate, acquisition window, and display window can be altered to accommodate specific requirements. The acquisition window and display window need to represent the maximum time required for acquiring all data. The gain will be a function of the magnitude of the event and the sample rate will determine the maximum frequency detectable. Therefore, data acquired at 10 kilohertz (kHz) will have a maximum resolvable frequency of 5 kHz whereas data acquired at 20 kHz will have a maximum resolvable frequency of 10 kHz. Raw data can be processed using a Fast Fourier Transform (FFT) for converting time domain data into spectral data. Impact attributes for analysis include waveform shape and time duration of the event. The shape of the waveform can be used to determine the type of impact such as open/close as seen during stroke testing, or the rubbing or sliding of surfaces as seen from degraded parts and increased clearances.

Raw data that can be trended include impact magnitude and impact frequency. Raw data is more difficult to trend because it is highly susceptible to changing flow conditions. Even in the best case where the flow conditions are deemed repeatable, changes can occur to the magnitude of the raw data. Also understanding what constitutes a change that would be considered relevant for degradation detection is somewhat subjective and less than scientific. Frequency can likely be trended more readily using raw data, however, the information available is highly subjective and may only indicate a change has occurred after valve failure. Studies have shown that changes in valve condition such as a worn hinge pin cannot readily be detected with discernable changes in frequency from test-to-test. However, a drastic change in valve condition such as a detached disc can be readily detected. Significant shifts in the spectrum occur when the mass of an object changes significantly such as in the case of a disc becoming detached. As mass decreases, frequency generally increases, and this can usually be detected. There are factors that can affect the spectrum and how it would be expected to respond in a case such as this. Since check valves are numerous in design with varying regions of body thickness as well as varying material compositions, the placement of accelerometers in a region of additional valve body transition or thickness may produce unexpected frequency responses.

The most effective way to consider the raw data is analysis of the waveform. Accelerated wear waveforms have distinct characteristics. Abundant characteristics can cause some confusion in regards to making universal assumptions about any single waveform. Thus, there are no waveform characteristics that can be accurately trended from test-to-test. The shape of the raw waveform contains the most information concerning its origin and whether or not it is the result of degradation. Generally, the activity of impacting is also another approach for assessment of condition. As parts degrade clearances increase. As clearances increase, there will usually be a change in the total amount of activity as seen in specific time duration. It would be useful to trend this activity in terms of how many impacts are detectable over a fixed period of time.

The second approach for analyzing and trending AE data provides the highest statistical accuracy for determining wear. This method uses a software algorithm for the purpose of data distribution. AE data is obtained when the check valve is experiencing full flow. AE can determine the related affects of flow conditions, present or future, on valve condition and accelerated wear. Impact attributes monitored include rate, energy levels and frequency content. Software tools that provide report comparisons can identify changes in impact rate, magnitude and frequency content. Changes in the impact rate and impact magnitude, in either direction, may indicate changing valve conditions. Also, shifts in the frequency information may also indicate changing valve condition. Not enough information is presently available about this approach. It is difficult to predict frequency responses because so many variables exist during the performance of check valve testing.

The third approach involves the use of AE raw data combined with either eddy current testing or UT during the performance of a stroke test. Software classifies a main event (seating or backstop impact) and events preceding the main event (precursors). Events following the main event are marked as bounces during the stroke test. The precursors and bounces help determine appropriate valve condition. The frequency and magnitude of the precursors and bounces can be used to trend valve condition. The main event is detected by its magnitude in relation to all other event magnitudes and will have the highest energy level noted. The precursors and bounces are detected as they exceed the threshold energy levels by a certain value.

Most, if not all check valves have some designed clearances to allow for proper seating to occur. Theses normal designed clearances may produce some precursor and bounce activity during stroke testing when the valve contacts the seat or backstop. As wear occurs between internal assembly parts, clearances will increase. The precursor and bounce activity will change as theses clearances increase. There may be additional precursors and bounces apparent as the valve degrades. Main event, precursor and bounces can be trended and used to detect degradation.

Figure 2 includes trends of AE attributes for both the raw data as well as the software algorithm that works on the principals of the power spectral density (PSD). As with trending any data, emphasis should only be given to changes in any direction from
test-to-test. Figure 2 also contains AE data trends of actual degraded conditions.

Signs of degradation using AE data are as follows:

PSD/Worn Hinge Pin

hinge pin impacts - Increased clearances can create hinge pin impacts. Changes from previous data would indicate the possibility of wear.

impact rate - Tests have shown that a change in the number of internal impacts from the baseline test may indicate degraded conditions.

> impact root mean square (RMS) - Tests have shown that the average impact amplitude RMS level will usually increase when hinge pins become worn.
>
> flow noise RMS - The RMS levels of flow noise can be monitored from test-to-test to ensure similar testing conditions and proper identification between changes in valve condition from changes in flow conditions.

shift in frequency spectrum - The average frequency spectrum of the total number of impacts identifies frequency shifts that may occur when parts degrade. Frequency spectrums can be overlaid for easy comparisons.

main event precursors - In some cases, as the hinge pin wears, a precursor is created by an impact at the hinge arm/hinge pin interface due to excessive clearances as the valve's hinge arm rotates closed. Precursors are sometimes difficult to produce with degradations in a laboratory environment, so they may or may not occur during actual testing (even for severely degraded conditions).

PSD/Missing Disc

> impact RMS - The impact RMS levels can dramatically increase or decrease when the disc is removed from the check valve. Changes from test-to-test can be trended.
>
> flow noise RMS - The RMS levels of flow noise can be monitored from test-to-test to ensure similar testing conditions. A dramatic change in noise levels from previous tests would indicate a detached disc.

shift in frequency spectrum - Frequency shifts may occur when parts degrade. The frequency spectrum of a backstop impact from a properly assembled disc and hinge arm connection should be different from the backstop impact from a hinge arm with no disc attached. Frequency spectrums can be trended for easy comparisons.

no main event - The absence of a main event can clearly indicate a problem with the closure of a check valve.

shift in FFT - The frequency spectrum of a closure with a detached disk would be different from the frequency spectrum of a closure with a hinge arm and disc assembly.

PSD/Broken Spring (Duo Disc)

impact RMS - When an impact occurs, the software will automatically determine RMS level and average with all impacts detected.

flow noise RMS - The RMS levels of flow noise can be monitored from test-to-test to ensure similar testing conditions and proper identification between changes in valve condition from changes in flow conditions.

shift in frequency spectrum - The average frequency spectrum of the total number of impacts can be used to identify frequency shifts that may occur when parts degrade. Frequency spectrums can be overlaid for easy comparisons.

precursors/rattling - The user can identify any abnormal AE events that occur during a full stroke.

PSD/Loose Disc Stud

backstop impacts - Severe backstop tapping can lead to stud failure. The software will automatically identify the location of the impacts as coming either from the backstop or the hinge pin region.

impact rate - Tests have shown that the number of internal impacts can change if degraded conditions exist. The software automatically determines the number of impacts over a 4 minute test period.

impact RMS - Sometimes, the average impact amplitude will increase when internal degradations are present. The software will automatically determine RMS level and average with all impacts detected.

flow noise RMS - The RMS levels of flow noise can be monitored from test-to-test to ensure similar testing conditions.

shift in frequency spectrum - The average frequency spectrum of the total number of impacts can be used to identify frequency shifts that may occur when parts degrade. Frequency spectrums can be overlaid for easy comparisons.

main event precursors - Theoretically, as the disc stud wears, a precursor event preceding the main event can occur. The precursor is caused by an impact at the hinge arm/disc interface due to excessive clearances as the hinge arm rotates closed.

PSD/Stuck Disc

impact rate - A total reduction of impacts can indicate check valve problems.

impact RMS - Changes from previous test may indicate changing conditions.

flow noise RMS - The RMS levels of flow noise can be monitored from test-to-test to ensure similar testing conditions. A wedged open disc can create much different flow noises than a healthy valve at the same flow rate.

shift in frequency spectrum - Frequency spectrums can be overlaid for comparisons.

no main event - The absence of a main event can clearly indicate a problem with the closure of a check valve.

Pulsed Electromagnetic/Eddy Current Technology

The third technology used for NIT check valves is pulsed electromagnetic or better know as eddy current testing. Eddy current testing uses alternating current from a test coil to induce eddy currents in electrically conducting metallic objects. This is accomplished through the use of coils of wire. The strength of the field is determined by the ampere-turns of the coil. As the temporary field is produced and the check valve disc passes through and disturbs the decaying electrical field, the sensing coils receive the current.

Eddy current testing can be conducted in any medium including air and steam systems. Eddy current technology will not work on carbon steel check valves. The signatures generated from the test displays the full stroke of the valve. Eddy current testing works well on high-pressure class stainless steel valves and fast-stroking valves.

This technology is nonlinear and only provides relative disc position in terms of open and closed; however, testing during NIC Phase 4 has revealed some interesting characteristics that were not expected during the original design and testing of the eddy current test system. More discussion on this will follow in this paper. A valve must be characterized during the baseline test in order to determine actual disc position. A valve characterization can be generated by manually stroking the valve or modeling with an identical valve. Simultaneous comparisons between eddy current and AE tests allow personnel to draw definitive conclusions regarding the operating performance and internal condition of the valve.

The eddy current test system consists of an enclosed module and probe that produces a voltage signal. The analog output of this signal is then passed on to a signal conditioner and processed. The processed signal is then displayed as relative check valve disc position and flutter. The signature from an eddy current test represents a voltage factor. As a check valve remains in an undisturbed position, a voltage will be recorded. Changes in the recorded voltage will occur as changes within the check valve take place. These changes will be displayed by the software as a flat line changing position. Therefore, as the check valve continues to change its position, the line continues to incline or decline depending on the position that the check valve is moving in.

A specific characterization or baseline test must first be conducted. Any change in the measured voltage of the stroke signal that exceeds the eddy current equipment accuracy of 10% during subsequent tests will represent a change (delta) in check valve characteristics. Other subtle signature differences can also be compared from test-to-test in order to determine if valve characteristics have changed. Testing during NIC Phase 4 revealed changes in signal shape representative of signal overshoot that seemed to occur after degradations were implanted into the valves. Certain degradations such as hinge pin wear and disc stud wear caused this signal overshoot to occur. This is likely the result of the clearances allowing the disc to shift in a manner sensitive to the electrical field during initiation of flow that did not exist with the valve in a good condition.

Following a stroke characterization or baseline test, changes not only in stroke voltage but also in degree of signal overshoot have a tendency to represent actual degradations. The NIC Phase 4 testing provided conclusive proof that degradations affect the signal shape in terms of overshoot appearance and magnitude. This is likely the result of the sensitivity of the eddy current testing to shifting valve internals due to excessive clearances caused by accelerated wear. In other words, a baseline test of a valve in good condition would produce a minimum to no signal overshoot provided optimum sensor placement and system balance. Testing with degradations that introduce clearances in the disc arm where the disc stud attaches or hinge pin wear that allows the disc to shift upon initiation of flow will produce these overshoot signatures that differ significantly from the baseline test signatures.

Significant testing experience indicates that some specific check valve designs respond in a similar manner in terms of stroke voltage and signal overshoot. The universal characterization method would require that a specific check valve baseline test be performed. This method is currently being considered. Other check valves with the same size, pressure class, and material characteristics can be tested against the signature generated from the universal baseline test. This would allow a group of check valves to be tested against a known standard. Once validated, this technique will allow assessments such as full open position verification on similarly designed check valves. In other words, the baseline characterization may be performed and data applied to other valves in the group.

Eddy current testing is limited to stainless steel valves and some valves that are primarily stainless steel with some carbon content. This technology works exceptionally well on rapid stroking check valves. Since the eddy current test locates the disc as it changes position and no electronic interface is required to capture the check valve signature, this technology presents itself as the premier method for the testing of rapid stroking valves. Figure 3 contains data examples and trends of degraded conditions based on the NIC Phase 4 eddy current testing program.

Conclusions

The UT, AE, and eddy current technologies described in this paper provide parameters that can be trended and used to predict degradation prior to valve failure. All the technologies require in depth training and experience to use them successfully. Each technology has strengths and weaknesses and it is only through the applied combination of multiple technologies that the most detailed information can be extracted.

UT is the most quantifiable of the technologies. It is used to trend angular velocity in degrees/second and associates a “stability classification” to accurately predict the likelihood of the occurrence of accelerated wear. UT also measures disc-open angle in degrees off the seat. It is a good indicator of hinge pin wear in that as hinge pin wear occurs, the disc-open angle is likely to change. UT requires well trained and experiences personnel to understand the data acquisition as well as analysis and trending of the analysis.

AE testing can measure impact attributes (i.e., rate, magnitude, frequency, precursors, and bounces). These attributes are compiled and compared by software tools. The raw data can contain valuable information to the trained analyst. Software tools are also used to compile and process data in an automated manner. AE testing is relatively easy to use with software tools making the analysis more automated that the other technologies.

Eddy current testing can measure stroke time and voltage or the change in voltage produced. Stroke time and voltage provide valuable information in terms of valve binding, stuck, and missing disc. Significant changes in stroke time or voltage may indicate the presence of any of these degradations. NIC Phase 4 testing revealed changes in signal overshoot that were the direct result of accelerated wear to the disc arm and hinge pin. These degradations produced significant changes in amount of overshoot and were clearly detectable on the data traces.

NIT trending provides a sound capability of degradation detection and allows for optimum predictive maintenance scheduling as well as provides confidence in “good actors”. These technologies provide a sound scientific basic for interval extension as allowed by Appendix II of the ASME OM Code and reduce operational and maintenance costs by minimizing the number of valve tests. Appendix II of the ASME Code allows interval extensions that would allow valve test frequencies to be extended to 10 to 14 years. NIT cannot be used to quantify wear rates and should not be used as a measure of dimensional change for valve internal parts.

References

1. ASME OM Code, Appendix II, Section II-4000(5)(b)(2)-2004 Edition of the ASME OM Code
2. NIC Phase 4 Test Report Dated 2005
3. Crane Nuclear Student Training Manual TM-4, Revision 5
4. NUREG 1482, “Guidelines for Inservice Testing at Nuclear Power Plants, Revision 1 (Paragraph 4.1.2)
5. NRC Information Notice 2000-21, “Detached Check Valve Disc Not Detected by Use of Acoustic and Magnetic Nonintrusive Test Techniques” (Page 3 of 4 - second paragraph)
6. NIC letter dated June 7, 2001 in response to NRC Information Notice 2000-21

TABLE 1 NONINTRUSIVE TEST ATTRIBUTES

CRANE NUCLEAR

	ULTRASONICS	***ACOUSTICS***	***EDDY CURRENT***
STABLE FLOW	Open angle (degrees off seat) and disc angular velocity (degrees/second)	Impact rate (impacts/sec) , magnitude (G's), and frequency (KHz) content	N/A
STROKE	Time (T1) from initial disc travel	Time (T2) to point of impact	Time (T1) from initial disc travel
RAW DATA	Sound path to opposite side of valve, to disc assembly, or distance from a reference point to where the opposite side reflector disappears	Impact magnitude (G's), waveform shape, magnitude, frequency	Full stroke delta from closed to open position
LEAK	N/A	RMS energy levels	N/A

Changes in trends may be the result of several factors. Changing sensor location, test equipment, test conditions including flow rates, test lineups, etc as well as actual valve degradation can result in changes to data points that may constitute adverse trends. A careful evaluation should be performed in the case of adverse trends to verify the actual cause of any change to the data prior to initiating corrective action.

TABLE 2 CHECK VALVE DISC STABILITY CLASSIFICATIONS

CRANE NUCLEAR
Notes on Classification:

1) A "Stable" disc can become "Unstable" or "Excessive" at reduced flow rates.

2) "Unstable" or "Excessive" disc flutter can sometimes be corrected and reduced by adjusting the flow rate. In certain situations, the backstop can be modified (lengthened) to reduce or eliminate disc flutter.

3) Check valves that are required to operate to operate at different flow rates can have several distinct Stability Numbers. The Stability Number quantifies the disc flutter only at the tested flow rate

4) Significant changes (increase or decrease) in Stability Number, without a change in flow rate, can identify changing internal conditions.

5) All valves with disc flutter will experience some wear. It is the purpose of the Stability Number Classification to distinguish between conditions that will lead to accelerated wear, and conditions that will lead to normal wear.

6) Normal wear is considered to occur over many years of service with some check valves remaining functional without replacement of parts.

7) Excessive or accelerated wear is considered to be when internal part replacements are required in less than three refueling outages (considering continuous operation).

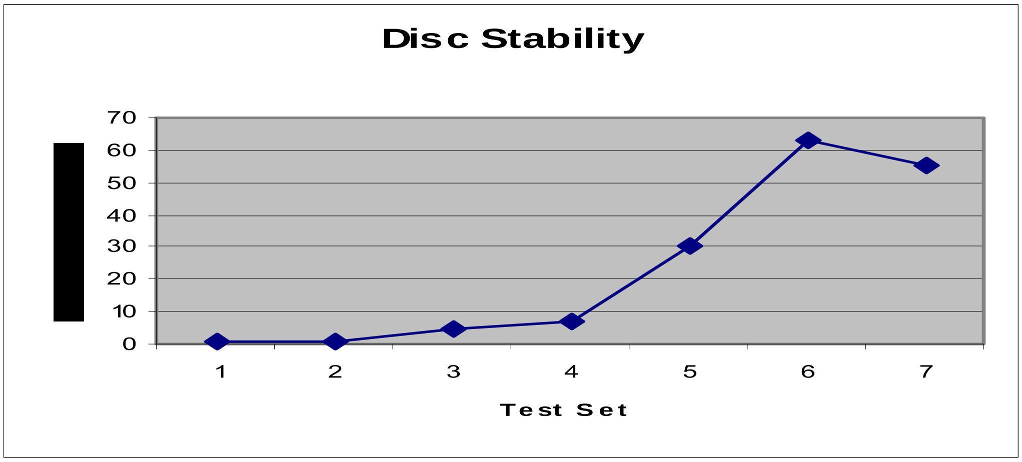
TABLE 2 CHECK VALVE DISC STABILITY CLASSIFICATIONS (CONT)

CRANE NUCLEAR

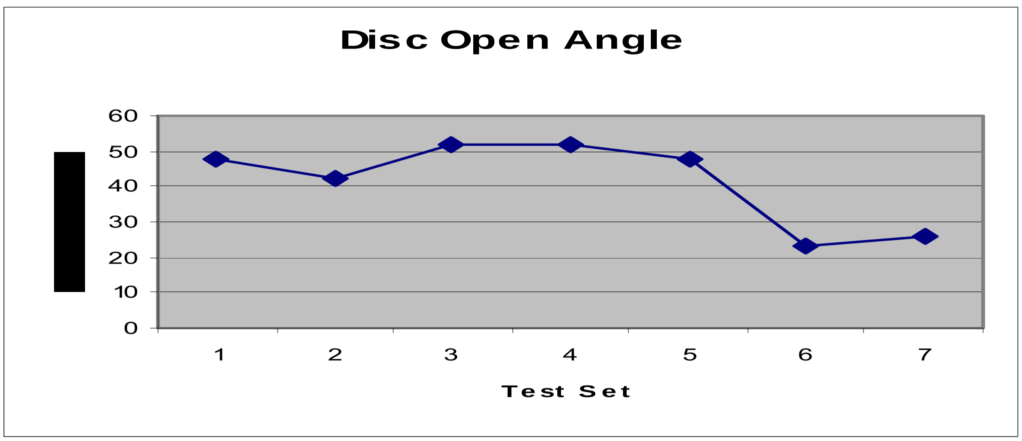
The primary cause of premature degradation of check valves is continuous disc flutter. Accelerated degradation of check valve internals can lead to their failure to perform. The Stability Number represents the amount of disc flutter at a specific flow rate. When a disc signature is recorded, the software automatically calculates the "Total Disc Movement". With inputs from the user, the software then calculates the point of sound beam reflection from the disc or disc assembly and considers this the "Target Point". The "Target Point" and "Total Disc Movement" are used to determine the Stability Number. The Stability Number combines the magnitude and frequency of disc flutter into one term. It is more representative of actual disc flutter then stating average magnitude (degrees) or frequency (Hz) separately since check valves oscillate with random motion.

STABILITY NUMBER (deg/sec)	EVALUATION	EXPLANATION
0-4	STABLE	Represents check valve discs that are either firmly against the backstop (0 deg/sec) or are displaying ordinary flow induced oscillations. Valves in the category will experience only normal (low) internal wear.
>4 - 11	UNSTABLE	Represents check valve discs that are neither clearly Stable nor Excessive. Typically valves in this range are operating under less than ideal flow conditions. Abnormal wear rates are possible depending on the system and operational history.
>11	EXCESSIVE	Represents valves that are experiencing abnormal or destructive flutter. The valves are incorrectly sized, misapplied or are operating at destructive flow rates. Accelerated wear can be expected if the valve continues to operate under these conditions. Valve failure is possible.

FIGURE 1 ULTRASONIC TRENDS



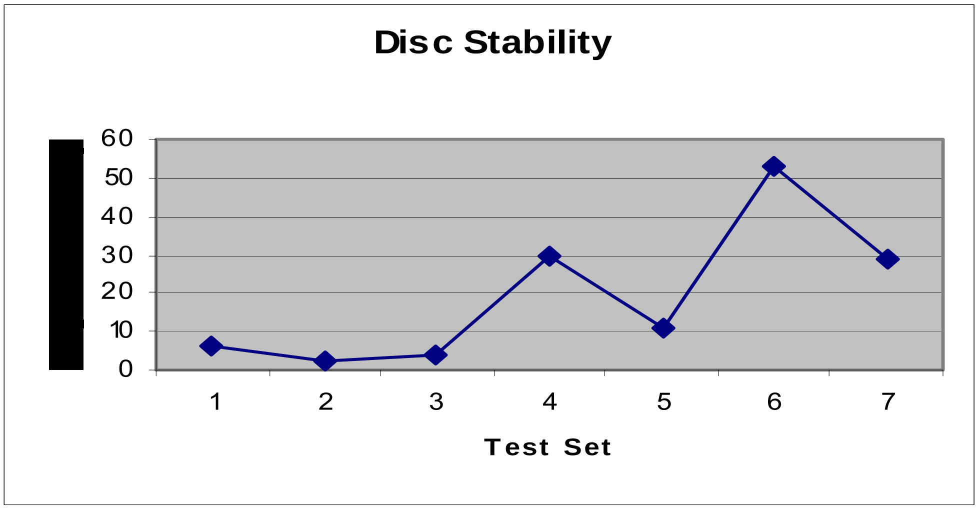
Condensate double disc arm swing check valve at Three Mile Island (TMI) indicating 2 year test intervals or fourteen years of test data. As disc angular velocity increases from test set 3 the wear to the hinge pin support cavity becomes exponential.



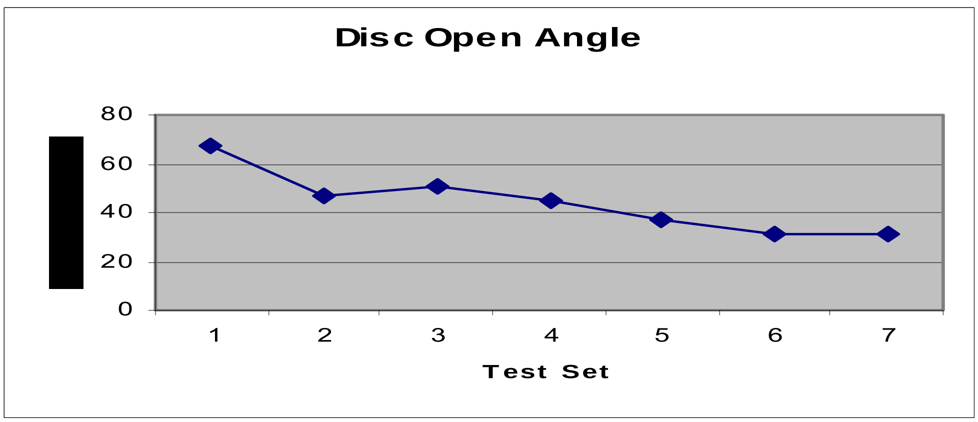
Same valve indicating a decrease in the disc open angle as the wear to the hinge pin support cavity increases.

Severe oblong reshaping of the hinge pin support cavity as a result of the continuous disc angular velocity.

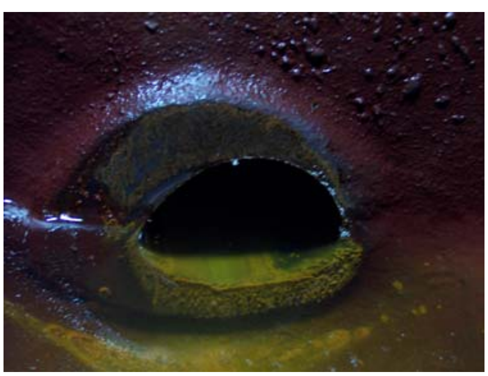
FIGURE 1 ULTRASONIC TRENDS (CONT)



A second condensate check valve at TMI indicating similar trend in that the disc angular velocity began to increase. This trend does however show less exponential change from test set 4 to test set 7.

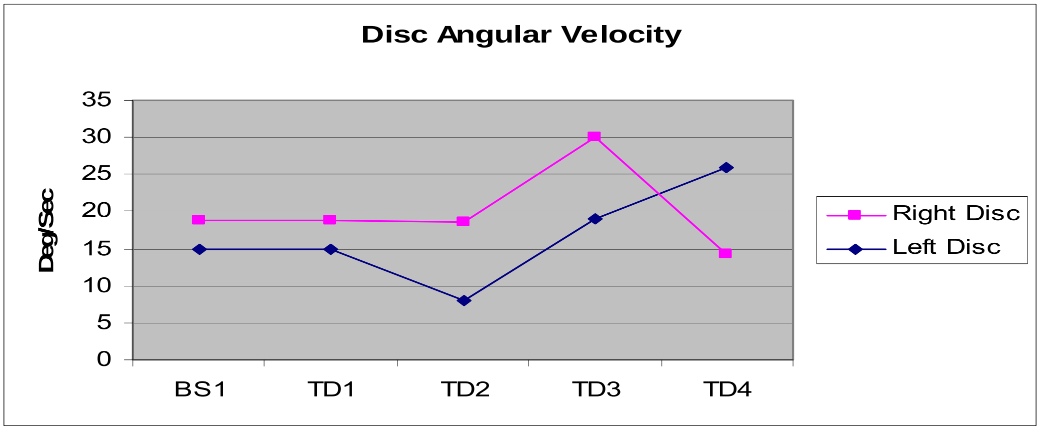


Same valve showing a steady decrease in disc open angle and the angular velocity changes.

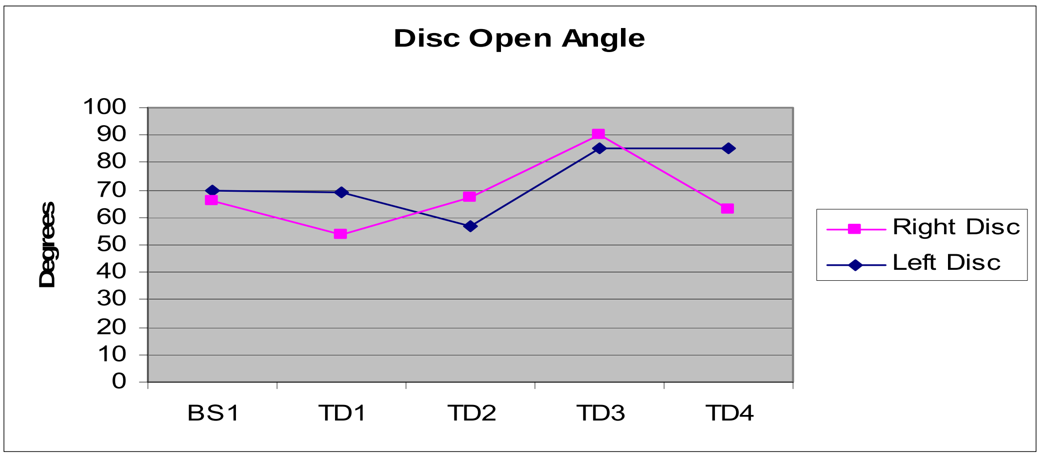


Wear to the hinge pin support cavity.

FIGURE 1 ULTRASONIC TRENDS (CONT)



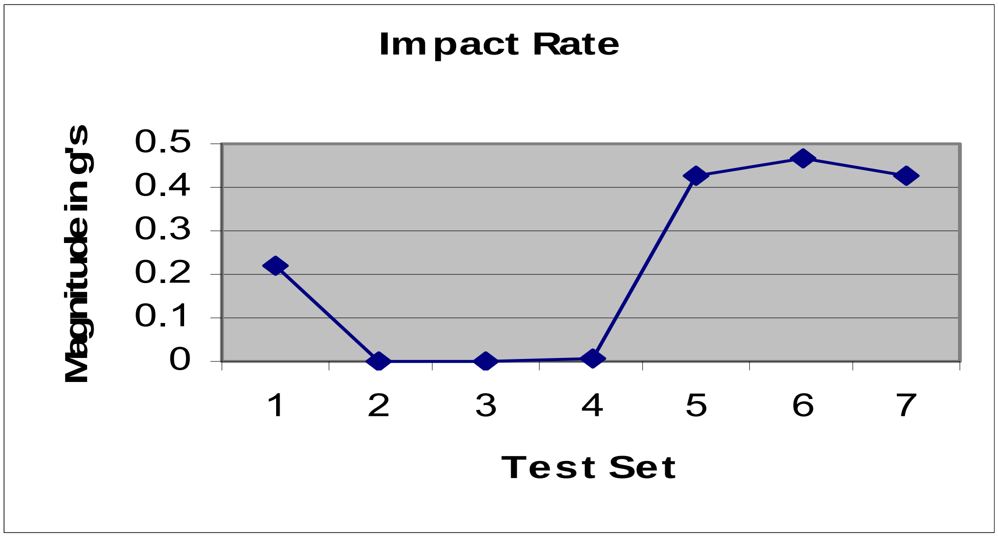
Component cooling water (CCW) duo disc check valve at Diablo Canyon. Valve instability was noted during baseline test concluding the wear was imminent. TD 2 test and beyond indicate changes to valve condition.



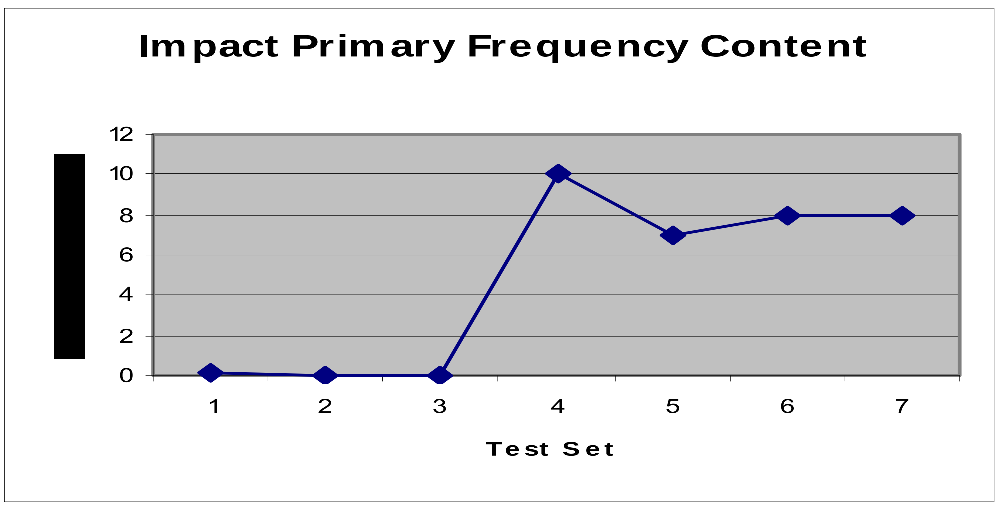
Both discs open angles show significant changes from TD 2 test and beyond.

Disc angular velocity caused the disc support spring to break causing the changes in disc open angle.

FIGURE 2 ACOUSTIC TRENDS



TMI condensate swing check valve PSD impact rate trend over 14 years indicates significant changes after test set 4.



PSD impact frequency indicates shift after test set 3.

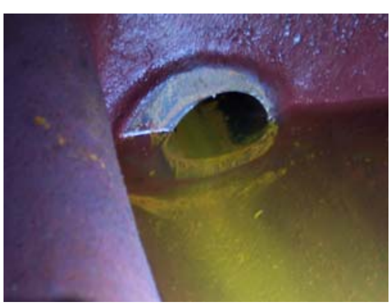
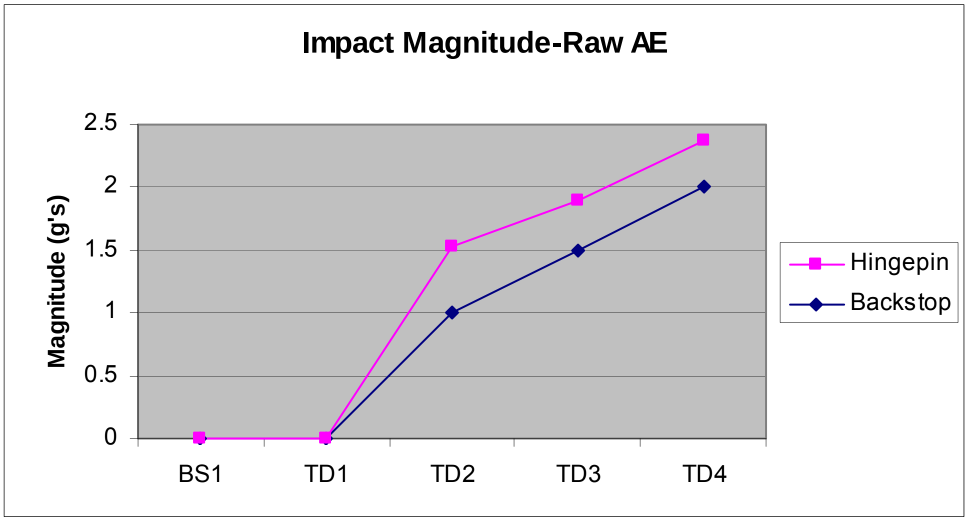
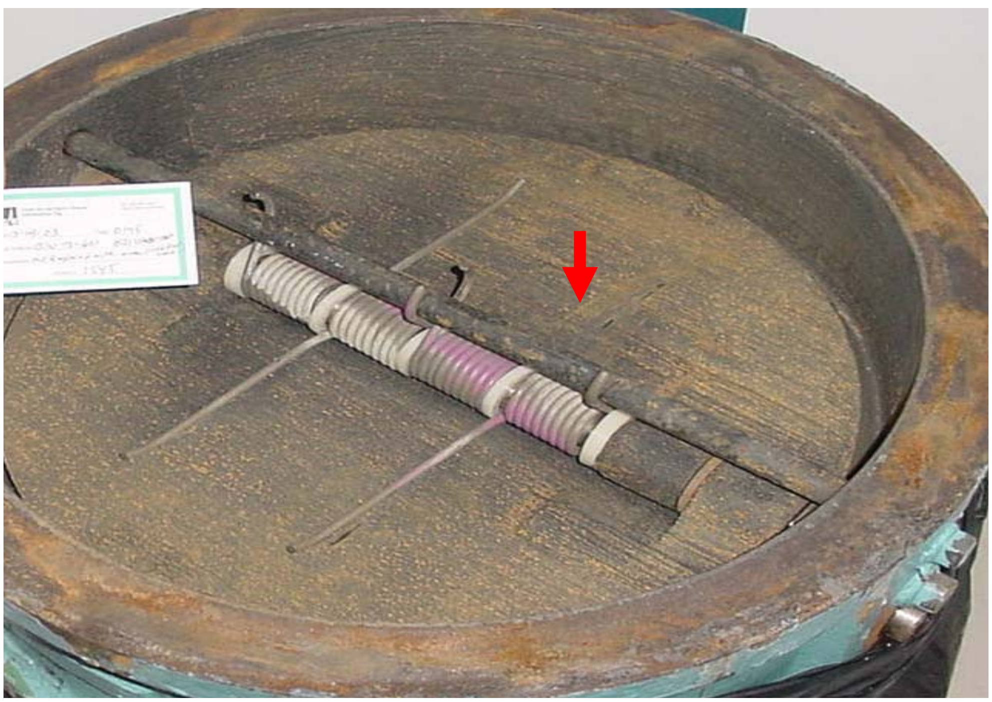

Wear to the hinge pin support cavity.

FIGURE 2 ACOUSTIC TRENDS (CONT)

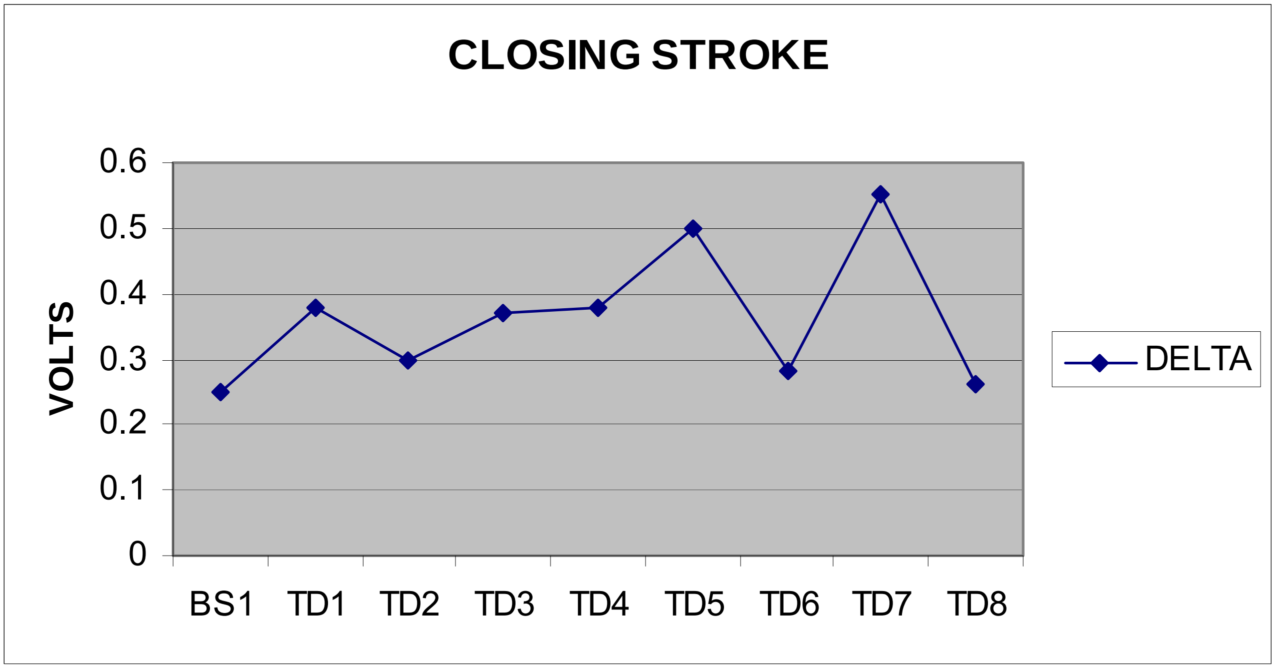


Diablo Canyon CCW duo disc check valve indicates exponential increase in raw acoustic data impact rate after TD1.



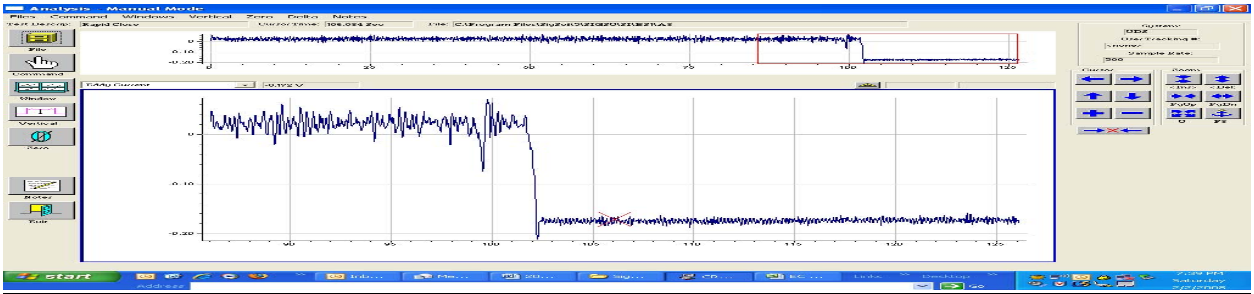
Valve disassembly following trends that indicated wear was present. Broken spring and worn hinge pin was discovered.

FIGURE 3 EDDY CURRENT TRENDS



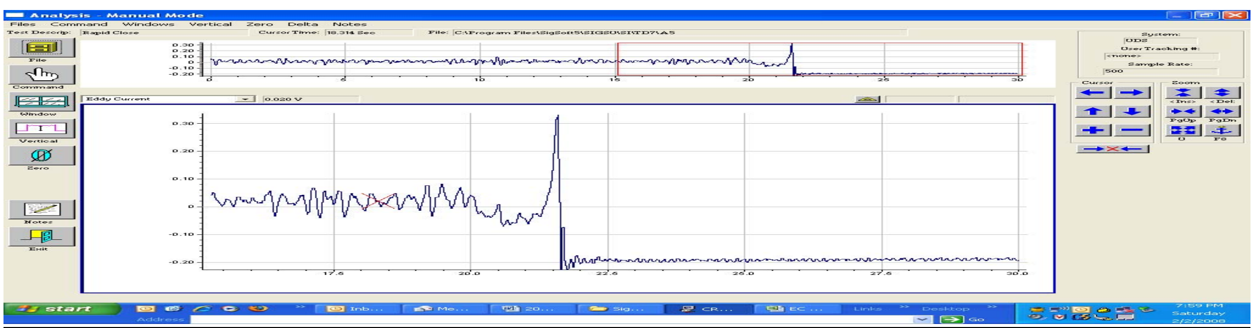
NIC Phase 4 swing check valve eddy current trends of signal overshoot. All degradations except those in TD6 and TD8 are detectable on the trends. Degradations appear in Figure 3 table below.

FIGURE 3 EDDY CURRENT TRENDS (CONT)



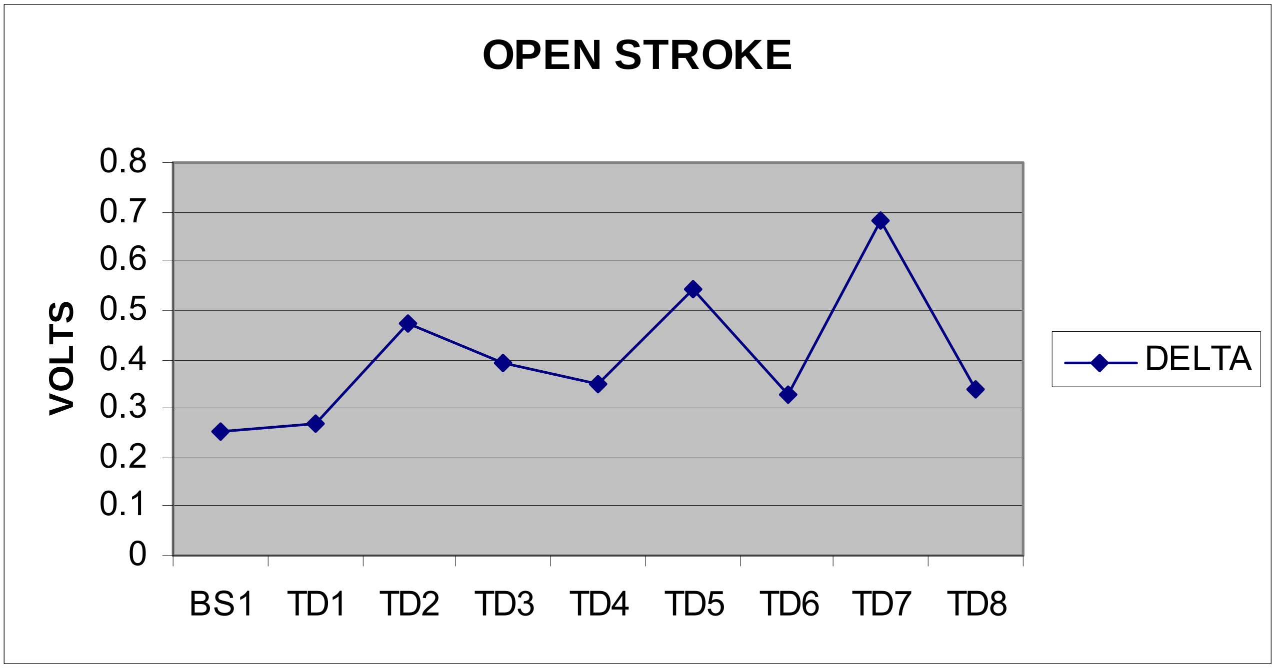
NIC Phase 4 Baseline BS1 eddy current data for swing check valve closing stroke test.

NIC Phase 4 Trending TD3 eddy current data for swing check valve closing stroke test with 50% worn hangar arm degradation. Notice the abrupt change in negative stroke delta as a result of the excessive clearance when the disc seats.



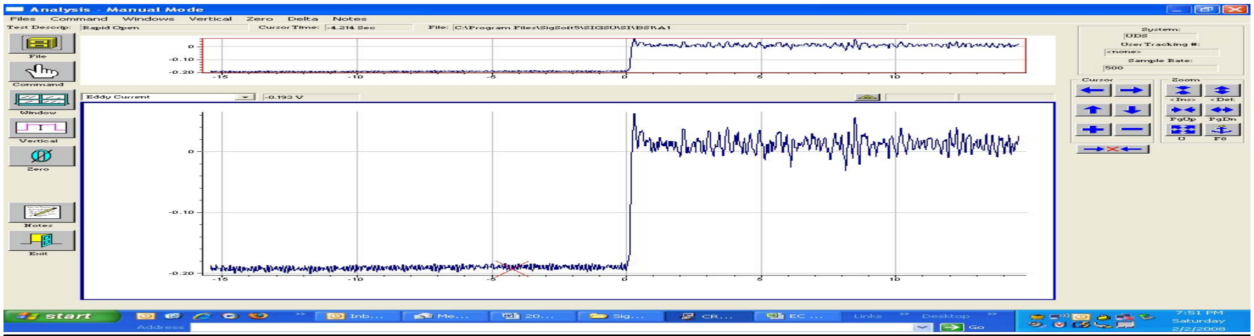
NIC Phase 4 Trending TD7 eddy current data for swing check valve closing stroke test with 15% worn hangar arm degradation. Notice the abrupt change in positive stroke delta as a result of the excessive clearance when the disc opens.

FIGURE 3 EDDY CURRENT TRENDS (CONT)

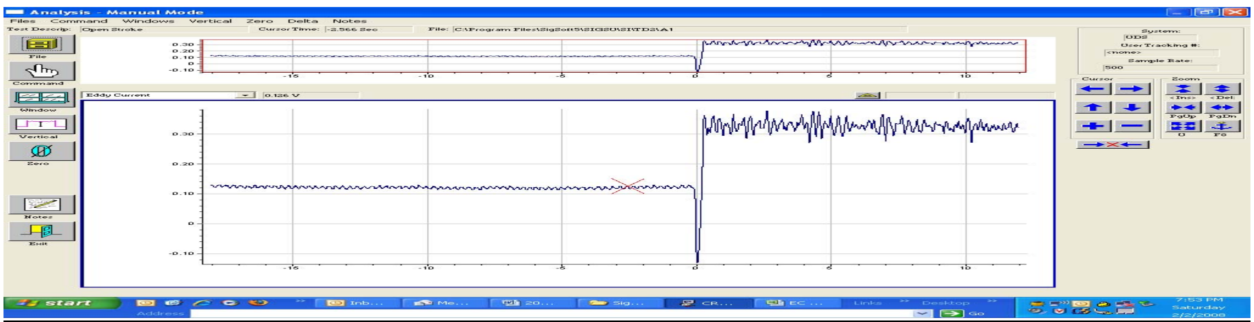


NIC Phase 4 swing check valve eddy current trends of signal overshoot. Test data appears in Figure 3 table below.

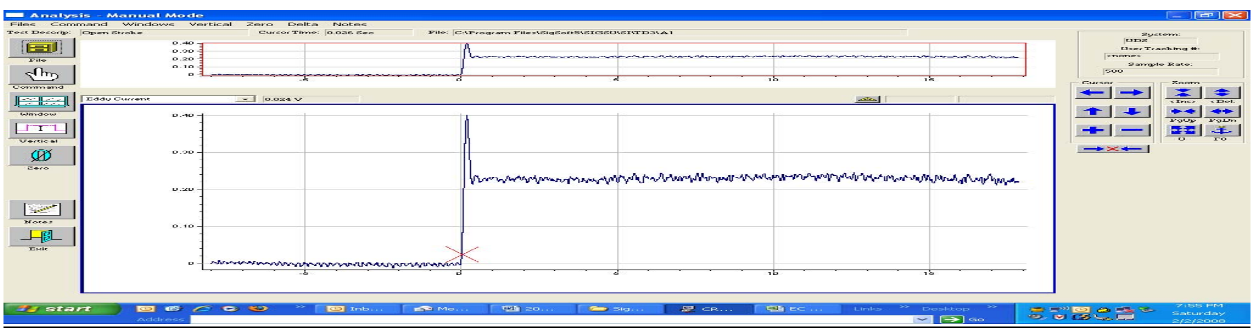
FIGURE 3 EDDY CURRENT TRENDS (CONT)



NIC Phase 4 Baseline BS1 eddy current data for swing check valve open stroke test.

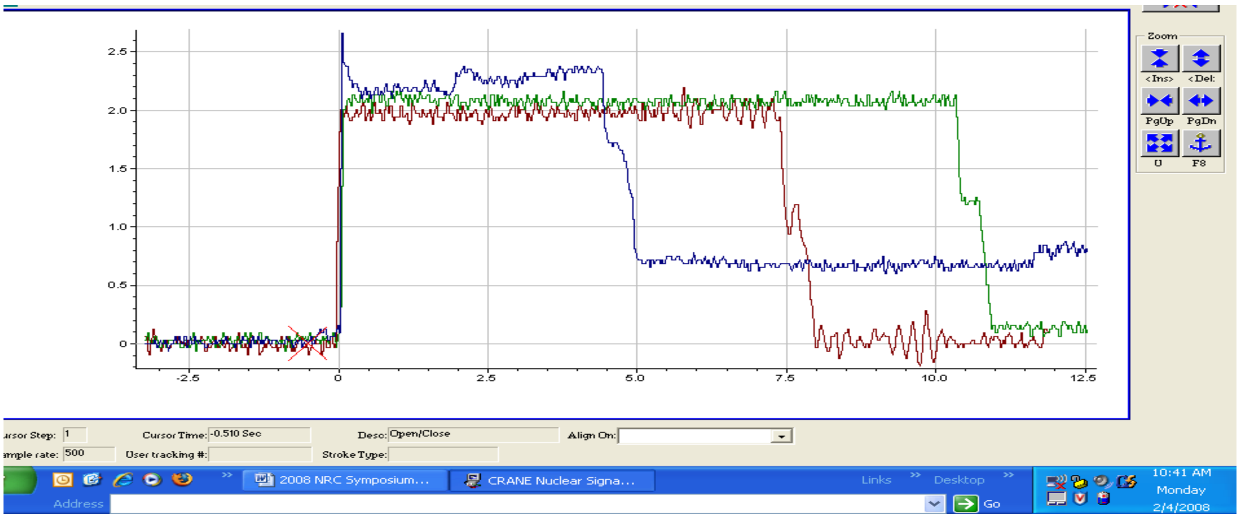


NIC Phase 4 Trending TD2 eddy current data for swing check valve open stroke test with 20% worn hinge pin degradation. Notice the abrupt change in negative stroke delta as a result of the excessive clearance when the disc seats.

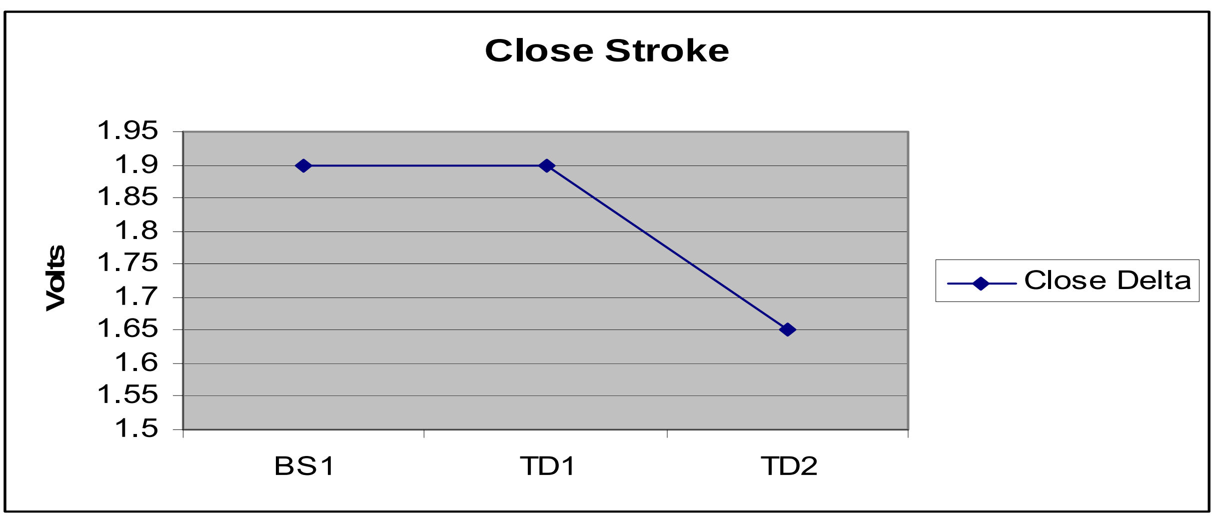


NIC Phase 4 Trending TD3 eddy current data for swing check valve open stroke test with 50% worn hangar arm degradation. Notice the abrupt change in positive stroke delta as a result of the excessive clearance when the disc opens.

FIGURE 3 EDDY CURRENT TRENDS (CONT)



NIC Phase 4 eddy current data overlays from baseline BS1 and trend test TD1, TD2 indicate closing stroke delta change outside 10% as seen in TD2 of the baseline reading indicating some change has occurred in the check valve. This was a piston check valve that had 20% plug guide wear induced. The wear caused the piston to stick partially open. Some overshoot can be seen on the open stroke likely the result of the wear. Notice the symmetry between the other two data sets. BS1 was new valve condition and BS2 was 10% plug guide wear.



Trends of the above data show the TD2 test with 20% plug guide wear outside of the acceptable margin indicating a change in valve condition has occurred. The 10% plug guide wear induced on TD1 was not significant enough to affect valve performance.

Instabilities of non-return valves in low-speed air systems

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Abstract
Practical observations of non-return valve wear in aero-engine cabin bleed systems suggest that such valves are subject to unstable behaviour. A theoretical model for the prediction of non-return valve instabilities in air systems is proposed and a nonlinear state space model of non-return valve and air volume interaction is derived from first principles. Experimental work is used to identify both the dynamic characteristics and the flow properties of the valve which are used to identify the coefficients within the model. Through frequency analysis of unstable valve behaviour the levels of damping within the system are identified. Finally, using a local linearization of the state spaced model an explicit mathematical prediction of valve stability is derived based on system parameters. These predictions are used to generate a map of the transition from stable to unstable system behaviour for low-speed air flow, which is in excellent agreement with experimental data.

1. Introduction

Non-return valves (NRV) are used in a variety of thermo-fluid systems involving gas flow where there is a requirement to prevent reverse flow conditions. Non-return valve wear in the aero-engine cabin-bleed systems has been a recognized problem since the 1970s. Specifically non-return flapper valves, as shown in Figure 1, have consistently failed to meet long-life specifications postulated by manufacturers. The most common failure is wear in the return spring, that can lead to breakage and regular costly inspections. There have also been recorded instances of hinge wear and flap cracking. This combination of failure modes suggest that the valves are experiencing a larger number of opening-closing cycles than expected. Related pressure control poppet valves are also known to experience unstable and irregular behaviour, under certain flow conditions, due to an interaction of the valve with an up or downstream volume of fluid [1]. Results from the present study show that non-return valves experience similar phenomena.

There has been a significant amount of work conducted on the behaviour of non-return valves, where the majority of this work focuses on the behaviour of the valve under a reversing flow condition in hydraulic applications. If the flow velocity through a non-return valve rapidly decelerates and then reverses, it is possible that the valve will close only after a reverse flow condition has been established. If this occurs, fluid momentum can cause a sudden increase in system pressure and valve differential pressure, leading to slamming of the non-return valve [2,3]. The large pressure spikes, and resultant valve slamming, caused by a flow reversal can cause damage to hydraulic systems. When designing such a system, it is therefore important to understand the key parameters affecting non-return valve behaviour. Thorley [4,5] determines relevant parameters using non-dimensional analysis and identifies four key non-dimensional terms which relate flow characteristic to the dynamics of the valve. Valibouse et al [6] approach the problem of check valve slamming by deriving a second-order mathematical model of valve dynamics, which includes the hydrodynamic forces exerted on the flaps. This model is then used to investigate valve behaviour under flow reversal, and a good agreement is found between theoretical and experiment valve behaviour.

These previous investigations show good understanding of valve dynamics under flow reversal, and the second-order mathematical models of the valve are in general agreement with experimental data. However, slamming behaviour does not explain the high levels of wear that are seen on valves in the cabin bleed environment. Rahmeyer [7] suggests that valve flutter and motion under positive flow conditions, where the direction of flow acts to open the flaps, are the leading causes of wear in non-return valves. It is suggested that this flutter is caused by unsteady forces and unsteady separation of the flow within the valve. One proposed solution to this problem is to introduce a minimum flow velocity constrain into the system which ensures that the valve flaps are fully open [8]. By investigating a swing check valve, Rahmeyer[7] models the opening of the valve as a function of the mean steady state flow velocity and uses this model to derive the minimum flow requirement based on valve geometry. Botros et al [9] further refine Rahmeyer's model, for a NPS 4 swing check valve in air, improving the accuracy of the model's coefficients and response through experimental work. This work again focuses on the steady state behaviour of non-return valves and does not consider transients or system stability. In order to understand transient behaviour, as an aid to system design, Pandula et al [10] derive a model of valve behaviour under step changes in flow rate. By modelling the flow characteristics and frictional behaviour in the valve bearing, an accurate model of the system is developed, which is validated using experimental data. This work highlights the need for accuracy when determining the levels of damping in the system, in order to capture the true dynamic behaviour of the system.

Despite generating this understanding of non-return valve dynamics, the literature does not directly address the cause of valve stability and the possibility of fluid-valve interactions leading to flapping behaviour. However, some studies consider such phenomena in poppet valves. Poppet valves are used in a variety of hydraulic system applications as pressure regulating valves. The poppet valve system consists of an upstream supply line, an upstream plenum chamber separated from the supply by an orifice, and valve seat with a spring loaded poppet. An interaction of the upstream volume of fluid with the poppet dynamics is known to cause self-excited oscillations. This can lead to excessive noise, spring breakage or damage to the valve seat [1]. Stone [11] investigated poppet valve behaviour by developing a steady-state model of the system, relating discharge coefficient and reaction forces to poppet and seat geometry. Kasai [12,13] incorporate the dynamics of the up and downstream piping into a system model, which is then used to determine whether the valve is locally stable. Hayashi et al [14-16] extend the idea of local stability numerically, using a Routh-Hurwitz type criterion, to form a global stability map. This map plots the stability of the system as a function of the supply pressure and the lift of the valve head. The geometry of the upstream volume is kept constant during this theoretical analysis.

Although poppet valves perform a very different function to non-return valves, this work suggests that, through an interaction with a volume of fluid, spring loaded valves can exhibit highly unstable behaviour. In this paper, the importance of interactions between a non-return valve and a volume of air are confirmed. A lumped parameter mathematical model of a system containing a non-return valve and a partially restricted volume of air is developed. Experimental apparatus is used to identify the parameters within this model and to examine the nature of valve instabilities. Pandula et al [10] demonstrate that an accurate knowledge of the damping within the valve system is essential in order for any model to accurately capture the behaviour of the system. To address this problem here, we use a frequency analysis method to determine damping coefficients present in the model. By considering the phase lag between the pressure loading and the motion of the valve under experimental transient flow conditions, the level of viscous damping is calculated with a high degree of confidence. While previous system simulations mainly consider numerically solving nonlinear differential equations, this paper uses control engineering techniques to develop a linearized state-space model of the system. With the model in this form, the effects of system geometry and massflow are investigated, and global stability criteria are derived. Based on a Eigen value approach, this method is both robust and highly flexible. These stability criterion predictions are validated, and confirmed by experimental measurements, which show transition from stable to unstable behaviour as massflow and system geometry are altered.

While there are significant results on the stability of poppet valves [11-16] for hydraulic systems, very little information exists on instabilities of non-return valves for air systems. Here we show that non-return valve instabilities are related to valve and downstream volume interactions. This paper thus presents a unique and different approach, compared to all known existing studies, for the understanding of unstable behaviour of non-return valve systems. Through the use of frequency and Eigen value analysis within a state-space system model, the effects of both system massflow and system geometry, specifically the interaction with a downstream volume, on valve behaviour are determined theoretically, and then, validated experimentally.

2. Valve configuration and mathematical models

The non-return valve under consideration is shown in Figure 1. Figure 2 shows a schematic diagram of the valve with different components labelled. The valve is of flapper type design with two spring-loaded independent flaps. The problem of valve interaction with a column of air, as schematically shown in Figure 3, is considered. The variables within the system are P_2, the pressure within the downstream system, the angle of the valve flaps θ (assumed to be symmetric) and the volume V. Within the cabin bleed system the downstream volume is isolated by a pressure regulating valve. In the model this valve is replaced by an outlet restrictor with discharge coefficient and area given by $(C_dA)_{out}$.

2.1 Valve dynamics

For this analysis, it is assumed that the valve sits in a horizontal plane with respect to the gravity vector. The weight of the flaps, along with the spring, closes the valve when no flow is present, while during a forward flow condition a pressure loading acts to open the valve. Valve flap angle is measured from the closed horizontal position, and only one flap is considered due to valve symmetry, by assuming equal pressure loading on each of the flaps. Note that if the valve sits immediately after an offset bend in the pipe, this assumption can not be made, and separate equations of motion would be required for the motion of each flap.

Applying Newton's law of motion to each flap, and one obtains

$$T_F = I\ddot{\theta} \tag{1}$$

where $\ddot{\theta}$ is the acceleration of the valve flap angle. The spring is pre-tensioned so that there is a force component at zero valve angle. This offset is denoted θ_0. Assuming the spring constant, K, is linear, the spring torque applied to each flap is given by

$$T_K = K(\theta + \theta_0) \tag{2}$$

Assuming that each flap is a half disc of uniform thickness hinged along its straight edge, the centre of mass for each flap is then given by

$$R_r = \frac{4r}{3\pi} \tag{3}$$

where r is the flap radius. The gravity induced torque on the valve flaps is therefore of the form

$$T_{mg} = R_r mg\cos(\theta) \tag{4}$$

where m is the mass of each flap. The mechanical damping for the system is assumed to be viscous damping proportional to flap velocity, which leads to the result given by

$$T_c = c\dot{\theta} \tag{5}$$

where c is the damping coefficient. Denoting the torque on the flaps due to pressure loading with T_p and considering a force balance leads to the equation having the form

$$T_p = I\ddot{\theta} + c\dot{\theta} + K(\theta + \theta_0) + R_r mg\cos(\theta) \tag{6}$$

The equation of motion describing the valve flap dynamics is therefore a decoupled second-order system. Note that the cross coupling between the valve spring and hinge is neglected in this derivation.

2.2 Volume dynamics

The non-return valve is modelled as a variable orifice plate in a low velocity flow. The mass flow rate into the volume, through the non-return valve, is therefore defined using,

$$\dot{m}_{in} = C_{dv}A_v\sqrt{\frac{2(P_1 - P_2)P_1}{RT}} \tag{7}$$

where A_v is the area of the valve when fully open. C_{dv} is the discharge coefficient of the valve, defined as the ratio of the actual mass flow rate of fluid $\dot{m}_{in}$ to the theoretical or ideal mass flow rate. As the flaps open, the effective area of the valve increases, and therefore, so to does the discharge coefficient. As a result, C_{dv} is a function of the valve opening angle θ. In order to model the system massflows accurately, this function is determined experimentally, which is discussed later in the paper.

The mass flow rate out of the control volume $\dot{m}_{out}$ is regulated using an outlet restrictor across which the ratio of upstream to downstream pressure is sufficiently high to ensure approximately choked flow. The mass flow rate is therefore given by

$$\dot{m}_{out} = P_2\sqrt{\frac{\gamma}{RT}}\left(\frac{\gamma+1}{2}\right)^{-\frac{\gamma+1}{2(\gamma-1)}}(C_d A)_{out} \tag{8}$$

where $(C_d A)_{out}$ is the product of area and discharge coefficient of the outlet restrictor, defined by the outlet geometry**.** Eq. (8) is used to relate $\dot{m}_{out}$, $(C_d A)_{out}$, and P_2 later in the analysis. The rate of change of the mass of air in the chamber between the non-return valve and the gate valve, which acts as a restrictor, is denoted $\dot{m}_2$, and is given by

$$\dot{m}_2 = \dot{m}_{in} - \dot{m}_{out} \tag{9}$$

Note that $\dot{m}_2 = 0$ with steady flow conditions. From the ideal gas equation with constant temperature, the time-derivative of static pressure is determined using

$$\dot{P} = \frac{\dot{m}RT}{V}$$

and thus, Eq. (9) can be rewritten as

$$\dot{P}_2 = \frac{C_{dv}A}{V}P_1\sqrt{2RT(1-P_2/P_1)} - \frac{P_2}{V}\sqrt{\gamma RT}\left(\frac{\gamma+1}{2}\right)^{-\frac{\gamma+1}{2(\gamma-1)}}(C_d A)_{out} \tag{10}$$

Solving for P_2 in the above equation requires knowledge of C_{dv}, which depends on the valve flap angle θ.

2.3 System dynamics

To combine the Eqs. (1)-(10) requires the knowledge of aerodynamic torques T_P in terms of pressures and angles. It is suggested by Tarnopolsky et al [17], that if a sprung plate is subjected to a steady stream of fluid, that the aerodynamic force acting on that plate is related to the drag coefficient C_θ using

$$T_p = \frac{1}{2}C_\theta \rho D^3 U_\infty^2 \tag{11}$$

Substituting Eq. (11) into (6) then yields

$$I\ddot{\theta} + c\dot{\theta} + K(\theta+\theta_0) + R_r mg\cos(\theta) = \frac{1}{2}C_\theta \rho D^3 u^2 \tag{12}$$

Using $\dot{m} = \rho A u$, the equation for the discharge coefficient can be rearranged into the form

$$u = \frac{C_{dv}\sqrt{2(P_1-P_2)\rho}}{\rho} \tag{13}$$

Then, by combining Eqs. (12) and (13), one obtains

$$I\ddot{\theta} + c\dot{\theta} + K(\theta+\theta_0) + R_r mg\cos(\theta) = C_\theta C_{dv}^2 D^3 (P_1 - P_2) \tag{14}$$

The valve dynamics can now be written in terms of the pressures within the system using

$$I\ddot{\theta} + c\dot{\theta} + K(\theta+\theta_0) + R_r mg\cos(\theta) = f_\theta (P_1 - P_2) \tag{15}$$

where $f_\theta = C_\theta C_{dv}^2 D^3$. Eq. (15) is used to determine valve/flow stability and instability behaviour at particular mass flow rates and air volumes. Note that the valve pressure loading coefficient f_θ is assumed to be dependent only on the valve flap angle θ since it is a product of valve discharge and aerodynamic drag coefficients. Note that steady-state f_θ values are employed for the unsteady stability analysis. In reality, however, f_θ is a time-varying quantity and depends upon past values of θ and P_2. Nevertheless, the unsteady component of f_θ is assumed to play a minor role as individual local stability conditions are considered.

3. Theoretical analysis of valve instabilities

From Eq. (15), $f_\theta(P_1 - P_2)$ acts as implicit non-linear feedback on the system, and as such, can affect both global and local stability. To investigate the effect that $f_\theta(P_1 - P_2)$ has on the stability of the system, consider governing valve Eq. (15). Re-writing Eq. (10) then gives

$$\dot{P}_2 = \dot{P}_{in} - \dot{P}_{out} \tag{16}$$

where

$$\dot{P}_{in} = \frac{C_{dv}A}{V} P_1 \sqrt{2RT}\sqrt{(1 - P_2 / P_1)} \tag{17}$$

and

$$\dot{P}_{out} = \frac{P_2}{V}\sqrt{\gamma RT}\left(\frac{\gamma+1}{2}\right)^{-\frac{\gamma+1}{2(\gamma-1)}}\left(C_d A\right)_{out} \tag{18}$$

To characterize local stability of the valve, consider first, the steady-state condition, where $\ddot{\theta} = \dot{\theta} = 0$ and the valve flaps sit at an angle θ with volume pressure P_2. Then from Eqs. (10) and (15), the following equations are obtained

$$\left(P_1 - P_2\right) = \frac{K\left(\theta + \theta_0\right) + Rmg\cos(\theta)}{f_\theta} \tag{19}$$

$$\left(C_d A\right)_{out} = \frac{C_d A P_1 \sqrt{2(1 - P_2 / P_1)}}{P_2\sqrt{\gamma}\left(\frac{\gamma+1}{2}\right)^{-\frac{\gamma+1}{2(\gamma-1)}}} \tag{20}$$

By defining θ' as the valve angle measured from the steady state angle and P_2', as the gauge pressure measured from steady state condition, Eqs. (16)-(18) are linearized to become

$$\dot{P}_2 = \left(\frac{\partial \dot{P}_{in}}{\partial \theta}\right)\theta' + \left(\frac{\partial \dot{P}_{in}}{\partial P_2}\right)P_2' - \left(\frac{\partial \dot{P}_{out}}{\partial P_2}\right)P_2' \tag{21}$$

$$I\ddot{\theta}' + c\dot{\theta}' + K\theta' - R_r mg\sin(\theta)\theta' = \frac{\partial f_\theta}{\partial \theta}P_1\theta' - \frac{\partial f_\theta}{\partial \theta}P_2\theta' - f_\theta P_2' \tag{22}$$

where

$$\left(\frac{\partial \dot{P}_{in}}{\partial \theta}\right) = \frac{\partial\left(C_d A\right)}{\partial \theta}\frac{P_1}{V}\sqrt{2RT}\sqrt{(1 - P_2 / P_1)} \tag{23}$$

$$\left(\frac{\partial \dot{P}_{in}}{\partial P_2}\right) = \frac{\left(C_d A\right)}{V}P_1\sqrt{2RT}\,\frac{-1}{2\sqrt{(1 - P_2 / P_1)}P_1} \tag{24}$$

$$\left(\frac{\partial \dot{P}_{out}}{\partial P_2}\right) = \frac{1}{V}\sqrt{\gamma RT}\left(\frac{\gamma+1}{2}\right)^{-\frac{\gamma+1}{2(\gamma-1)}}\left(C_d A\right)_{out} \tag{25}$$

Note that Eq. (22) is a second-order ordinary differential equation and can be converted into two first-order equations. This allows Eqs. (21) and (22) to be written as three first order ordinary differential equations in a vector-matrix, which is expressed using

$$\begin{bmatrix} \dot{\theta}' \\ \ddot{\theta}' \\ P_2' \end{bmatrix} = \begin{bmatrix} 0 & 1 & 0 \\ X_1 & X_2 & X_3 \\ X_4 & 0 & X_5 \end{bmatrix} \begin{bmatrix} \theta' \\ \dot{\theta}' \\ P_2' \end{bmatrix} \tag{26}$$

where

$$X_1 = \frac{1}{I}\left[\frac{\partial f_\theta}{\partial \theta} P_1 - \frac{\partial f_\theta}{\partial \theta} P_2 + R_r mg\sin(\theta) - K\right] \tag{27}$$

$$X_2 = \frac{1}{I}\left[-c\right] \tag{28}$$

$$X_3 = \frac{1}{I}\left[f_\theta\right] \tag{29}$$

$$X_4 = \left(\frac{\partial \dot{P}_{in}}{\partial \theta}\right) \tag{30}$$

$$X_5 = \left(\frac{\partial \dot{P}_{in}}{\partial P_2}\right) - \left(\frac{\partial \dot{P}_{out}}{\partial P_2}\right) \tag{31}$$

Eq. (26) is the local linearization of the system dynamics at a steady state condition $\left(\theta, P_2\right)$ for a given massflow. To ensure local stability of the system, the eigenvalues of the matrix λ, given by

$$S = \begin{bmatrix} 0 & 1 & 0 \\ X_1 & X_2 & X_3 \\ X_4 & 0 & X_5 \end{bmatrix} \tag{32}$$

must have all negative real parts. By substituting Eqs. (21) -(31) into the matrix (32), it is possible to determine the Eigen values of the system by solving the equation of the form

$$\det(S - \lambda I) = 0 \tag{33}$$

for a given upstream volume and inlet massflow. By considering the signs of the real parts of the Eigen values, system stability characteristics are determined. To confirm the validity of this analysis, experiments are used to demonstrate a good agreement between theoretical and experimental stability.

4. Experimental apparatus and procedures

4.1 Valve instrumentation

To record the position of the valve flaps under positive flow conditions an optical position sensor is developed especially for this purpose. An optical system is chosen to avoid the need for a physical connection to the flaps, preventing damage to sensitive hardware in case of high flap velocities and flap impacts. It is also employed to minimise disruptions to the air flow through the valve. By shining light across the valve housing into a detector on the other side, a line of sight path is created through which the valve flap passes while opening. As the flap angle increases, more light is obscured from the detector and the output signal reduces. Using one detector in this way greatly reduces the complexity of the supporting electronics and means that the output signal is continuous. The system is calibrated by measuring the output of the photodiodes for a range of known angles, while detached from the rig, and then fitting a polynomial to this data. A bank of rectangular red LEDs provide the light source and a General Semiconductor Industries 10530CAW internally-amplified photodiode is used to detect the flap signal. Internal amplification within this photodiode is essential in producing a signal with acceptably low levels of random noise.

4.2 Experimental facility

To validate the numerical analysis results, a low-pressure experimental facility is used, which is operated by a low pressure vacuum line, and developed and constructed especially for this purpose. Figure 4 shows the rig layout. The flow development lengths on the upstream and downstream sides of the standard orifice plate are 2.0 m and 2.5 m, respectively. This plate is located within a section of piping which is fitted with upstream and downstream static pressure tappings, which are connected to a SensorTechnics BSDX +/-0-100 mbar amplified differential pressure transducer, used to measure pressure drop for determination of the mass flow rate. Following this is the inlet elbow and the non-return valve. Static tappings either side of the valve, connected to a second SensorTechnics differential pressure transducer, are used to determine the static pressure drop across the valve. The mass flow through the orifice plate is determined using the pressure drop across the plate $(P_3 - P_4)$, which is employed within the orifice flow equation. According to the British Standard document BS1042, this equation is given as

$$\dot{m}_{in} = \frac{C}{\sqrt{1-\beta^4}} \frac{\pi}{4} d^2 \sqrt{2(P_3 - P_4)\rho_1} \tag{34}$$

The ratio of orifice diameter to the pipe diameter β is chosen to be equal to 0.5, so that all system mass flow values are measured with maximum accuracy. The flow discharge coefficient *C,* is determined by orifice plate geometry and the Reynolds number of the incoming flow. For the selected β and system mass flows, the British Standard document BS1042 gives C = 0.6.

All pressures are measured using SensorTechnics BSDX +/-0-100 mbar amplified differential pressure transducers. These transducers are zeroed at the beginning of each data acquisition sequence at a zero mass flow rate condition. Each transducer is calibrated using a dead weight pressure testing apparatus. Signals from all instrumentation are sampled and acquired at a frequency of 1kHz using a National Instruments NI6035E card. MATLAB/Simulink software is then employed in conjunction with an OpalRT RT-LAB and QNX operating system to ensure hard real-time operation.

4.3 Experimental uncertainty magnitudes of measured quantities

Experimental uncertainty magnitudes are determined based on 95 percent confidence levels using single-sample uncertainty analysis procedures. The associated uncertainty of differential pressure used to determine mass flow rate is ± 1.2 percent. The experimental uncertainties of differential pressure across the non-return valve, the valve mass flow rate, and the valve flap angle are ± 5.1 percent, ± 0.61 percent, and ± 1.0 percent, respectively.

5. Experimental and analytical results

5.1 Valve parameters

To evaluate the stability matrix, as expressed by Eqs. (32), and to determine the dynamic behaviour of the non-return valve, dynamic parameters of the valve, and the flow characteristics of the system are required. The dynamic parameters are measured directly from the valve, while the flow characteristics are calculated from steady flow experimental data. To determine the spring coefficient *K*, the mass of the flaps *m*, and the spring offset angle θ_0, the valve alone is considered. With the valve flaps stationary, $\ddot{\theta} = \dot{\theta} = 0$, and Eq. (6) reduces to the form

$$T^0 = K(\theta^0 + \theta_0) + R_r mg \cos(\theta^0) \tag{35}$$

Using a spring balance to apply a torque to flap one at various flap angles, and solving this set of simultaneous equations, gives values for the coefficients *K, m* and θ_0. The moment of inertia of each of the flaps is assumed to be that of a half disc of uniform thickness. The resulting valve dynamic parameters, determined through this procedure, are given in Table 1.

5.2 Steady mass flow experiments

To accurately model the mass flow through the non-return valve and determine the effects of pressure loading on the valve flaps, the valve discharge coefficient and valve pressure loading coefficient must be known. To determine these parameters, a range of different system mass flow rates are considered, as the valve flap angle and the differential valve pressure are recorded. When these data are obtained, the valve is initially stabilised by starting the test at the maximum achievable mass flow. The system mass flow is then reduced in small increments to maintain valve stability. Figure 5 shows the variation of flap angle θ *with the system mass flow rate. From this figure, it is evident that, as the mass flow rate through the system increases,* θ *becomes larger as the valve opens in response. It is also evident that the dependence of* θ *on* $\dot{m}$ *is non-linear. Figure 6 shows the variation of dimensional differential pressure across the non-return valve* P_1*-*P_2 *with the mass flow rate. This pressure drop increases as the system mass flow increases in a non-linear fashion.*

The relationship between the steady flow discharge coefficient for the non-return valve C_{dv} and the valve flap angle θ is derived by rearranging Eq. (7) into the form

$$C_{dv} = \frac{\dot{m}_{in}}{A_v \sqrt{2(\Delta P) P_1 / RT}} \quad (36)$$

With this equation, and with knowledge of the relationship between system mass flow rate and flap angle, and the relationship between system mass flow rate and differential valve pressure (given in Figs. 5 and 6), the variation of discharge coefficient C_{dv} with valve flap angle θ can be determined. The resulting non-linear function is shown in Fig. 7. These data show that, as the valve opens, the effective area of the valve increases, giving higher mass flow rate for a given differential pressure.

Tarnopolsky et al [17] suggest that, if a sprung plate is subjected to a steady stream of fluid, then the aerodynamic force acting on that plate is proportional to h_v, the dynamic head of the flow, with the drag coefficient C_θ as the constant of proportionality. This proportionality quantity is shown as it varies with flap angle θ in Figure 8a. To show that C_θ is not affected by flow density, this inlet density is altered by changing the inlet restrictor. Three arrangements are used: no restrictor, restrictor A (where the ratio of restrictor diameter to pipe diameter is 0.50), and restrictor B (where the ratio of restrictor diameter to pipe diameter is 0.25). The resulting ρ_{in} density variations with h_v are shown in Fig. 8b. The application of sequential restrictors thus reduces the inlet density ρ_{in}. With this in mind, Fig. 8a shows that the relationship between dynamic head and angular position is independent of flow density.

Using results in Figs. 5 and 6, and the valve pressure loading coefficient $f_\theta = C_\theta C_{dv}^2 D^3$ can be determined. This is needed in regard to the terms on the right-hand side of Eq. (14). The resulting f_θ function as it varies with valve flap angle θ is shown in Fig. 9. Recall that $f_\theta (P_1 - P_2)$ is the pressure loading on the non-return valve. From these data, it is evident that, as the valve opens, less torque is present on the valve flaps for a given pressure difference.

5.3 Unsteady flow experiments

Pandula et al [10] show that accurate measurement of valve damping is essential to model unsteady system behaviour. In the stability analysis of the present non-return valve, the mechanical damping is assumed to be viscous damping, which is proportional to flap velocity, and characterized by a valve damping coefficient. The damping torque is equal to this valve damping coefficient *c* times the first derivative of valve flap angle, as given by Eq. (5). To determine magnitudes of the valve damping coefficient *c* accurately, unsteady experiments, when the valve is flapping, are employed. When the valve is subject to such unsteadiness, the valve flaps are effectively oscillating sinusoidally, driven by a sinusoidal pressure loading force, given by $F = f_\theta (P_1 - P_2)$. Experimental data indicates the presence of a time lag between the loading force, *F*, and the flap response *θ*. By considering the dynamics of the valve, this time lag can be related directly to the damping coefficient, *c*. By determining experimentally the level of time lag present in the system, it is possible to calculate the damping coefficient, *c*. To establish the relationship between the system time lag and the damping coefficient, the transfer function *G(s)*, between the loading force, *F*, and the flap angle *θ* is required. *G(s)* is determined by taking the Laplace transform of Eq. (22), the linearized valve position equation, which gives

$$G(s) = \frac{\theta'}{F} = \frac{1}{Is^2 + cs + K - R_r mg \sin(\theta)} \quad (37)$$

Any dynamic system will have a time lag, between the input function and the output system response, which is given by

$$t_L = \frac{\arg(G(jw))}{w} \quad (38)$$

where arg(*G(jw)*) is the argument of the transfer function evaluated at the driven frequency and *w* is the driven frequency. For the non-return valve system, the argument of the transfer function is then given by

$$\arg(G(jw)) = \arctan\left(\frac{wc}{K - Rmg\sin(\theta) - w^2 I}\right) \tag{39}$$

Eqs. (38) and (39) can be rewritten in terms of the damping coefficient, *c*, as

$$c = \frac{1}{w}\tan(wt_L)\left(K - R_r mg\sin(\theta) - w^2 I\right) \tag{40}$$

The time lag, t_L, between the pressure force *F* and the flap response *θ* is determined experimentally by considering the phase shift between the pressure variation across the valve and the valve flap position at the resonant frequency of the system and by employing the Blackman-Tukey approach [18]. The Blackman-Tukey approach can be used to compute the phase between the input and output of a system, at a given frequency, by evaluating the argument of the ratio of the output-input cross-spectrum, $\Phi_{F\theta}$, and the input-input cross-spectrum, Φ_{FF}, at the driven frequency. The cross-spectrums are determined by recording the pressure loading force *F* and the valve position *θ* for 25 seconds at a sample rate 1000Hz, during unstable valve behaviour. These data are then windowed using a Hanning window of the same length. The input-input cross-spectrum Φ_{FF} is calculated by correlating the windowed input signal with itself before taking the Fourier transform. The output-input cross-spectrum $\Phi_{F\theta}$ is similarly determined by correlating the windowed input signal with the windowed output signal before taking the Fourier transform. Using this approach, the time lag between the system input and output is given by

$$t_L = \frac{1}{w}\arg\left(\frac{\Phi_{F\theta}}{\Phi_{FF}}\right)_{w=w_n} \tag{41}$$

The damping coefficient is then calculated from t_L using Eq. (40) and is equal to $c = 0.0097$.

5.4 Stability map and comparisons between experimental and analytic results

Variations of valve flap position and system mass flow rate with time, which illustrate unstable valve behaviour, are shown in Fig. 10. From these data, it is evident that, as the system mass flow rate increases slowly, the system becomes unstable. This is apparent as the valve flaps begin to swing rapidly between fully closed and fully open, with violent impacts occurring at both locations. This behaviour is self-exciting and continues indefinitely, unless the system massflow is increased above a threshold level. For these experimental conditions and configuration, this threshold is present at approximately 0.15 kg/s.

For a given downstream volume, this threshold can be predicted using Eigenvalue stability criteria and a linear state-space model of the system. For a given mass flow rate, the steady-state flap position θ and system pressure P_2 are calculated using Eqs. (19) and (20). These values are then used to evaluate the state-space matrix values within Eqs. (32), which are the Eigen values which determine whether the system is locally stable or unstable. By repeating this analysis for different mass flow rates and downstream volumes, the boundary between stable to unstable valve behaviour is determined. This stability boundary is denoted in Fig. 11 by a black line, where mass flow rate is given as it depends upon dimensional volume downstream of the valve *V*. The unstable region lies below this line. For the range of volumes considered, this stability boundary is approximately linear, with the threshold mass flow rate increasing with size of the downstream volume. Note that the influence of the downstream volume on valve behaviour is negligible when stable conditions are present.

Experimental data, obtained using the low pressure test facility shown in Figure 4, are used to validate this theoretical analysis. These data are first obtained for stable conditions using relatively high mass flow rates. For a given downstream volume, the mass flow rate is then reduced to sequentially lower values. This is done slowly to ensure that stable valve behaviour is maintained. For each mass flow rate, a disturbance is introduced by applying a restrictor plate is to the inlet, which is then removed. This introduces a small step disturbance in valve position. If the system is stable at this operating point, the valve flaps return to the initial position which existed prior to the perturbation. If the system is locally unstable, the flaps rapidly and violently oscillate. This procedure is repeated at different mass flow rates to determine the experimental conditions which correspond to the change from locally stable to locally unstable behavior. The use of a fixed restrictor in generating the input disturbance ensures that the test is repeatable. The input flap angle step disturbance varies with the massflow conditions and is approximately 2 degrees for the lowest massflow of 0.1 kg/s and approximately 3.5 degrees for the highest massflow of 0.27 kg/s and is thus considered to be a local perturbation on the system and suitable for validation of our locally linearized model. The size of the downstream volume is also varied by changing the duct length. The resulting measured points of stability and instability are shown in Fig. 11. Crosses indicate experimental conditions corresponding to stable valve behaviour, while circles indicate experimental conditions which correspond to unstable behaviour. It is clear from Fig. 11 that the experimental results and theoretical predictions are in excellent agreement.

The generally accepted solution to the problem of non-return motion is to ensure that the valve is held fully open for all possible flow conditions [8]. This is achieved by placing a restriction on the minimum flow velocity in the system. Although this method is broadly successful, the condition that the valve is fully open is overly restrictive. The present stability map data are thus useful because they allow relaxation of this constraint on minimum flow velocity, which provides a definitive means to maintain stable valve behaviour when the valve is not fully open.

6. Summary and conclusions

The performance of non-return valves in air systems is investigated and it is shown that, when the downstream ducting is confined by a restrictor, sprung non-return valves can exhibit unstable behaviour. This behaviour is a result of an interaction between the dynamics of the downstream volume and the dynamics of the non-return valve. In order to investigate this unstable interaction, a non-linear state-space model of non-return valve and air volume system is derived from first principles. Working with this model, it is shown that the local stability of the system is dependent on the volume of the downstream ducting, and upon the massflow rate through the system. By linearizing the state-space model and using control engineering techniques, the stability of the system is determined analytically, allowing a stability map to be produced.

Experiments on the valve are performed using a highly accurate valve flap position sensor in conjunction with a low-pressure flow facility to determine the flow performance characteristics of the valve. The variation of discharge coefficient and loading coefficient with valve flap angle are determined from the resulting experimental measurements. A novel frequency analysis method is also used to determine the damping coefficient of the valve from experimental data. A step disturbance in valve position is introduced to perturb system behaviour for a range of mass flow rates to determine the stable-to-unstable boundary for different downstream volumes. The resulting experimental stability boundary shows excellent agreement with the theoretical predictions, validating both the model and the stability criteria employed.

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Flap Radius	$72x10^{-3}$ m
Flap Inertia	$6.2x10^{-4}$ kgm^2
Flap Mass	0.47 kg
Spring Constant	$1.178x10^{-3}$ Nmrads^{-1}
θ_0	2.05 rad

Table 1. Non-return valve dynamic parameters.

Figure 1. Spring loaded twin flapper non-return valve.

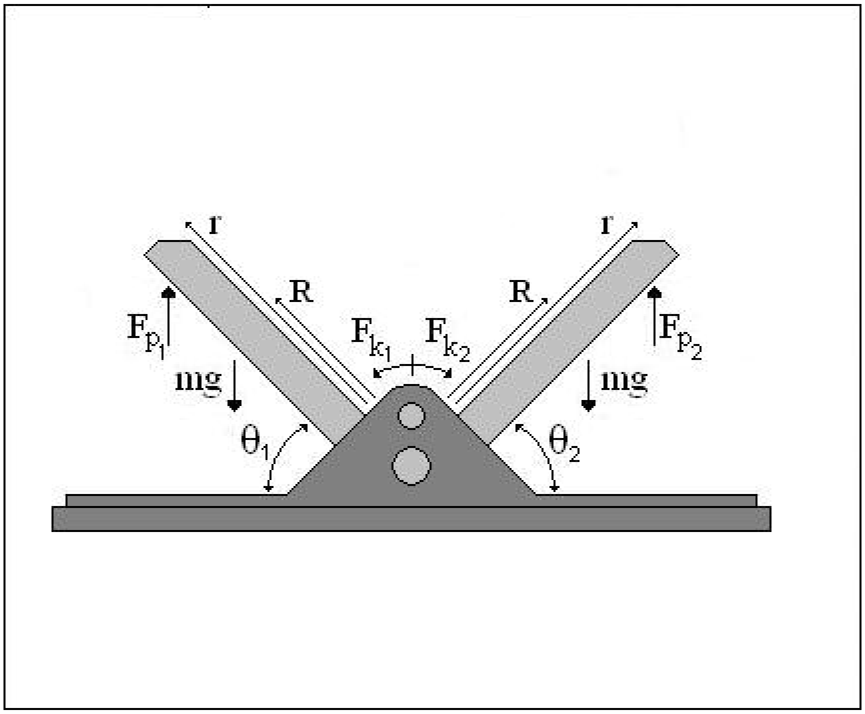

Figure 2. Non-return valve geometry and loading.

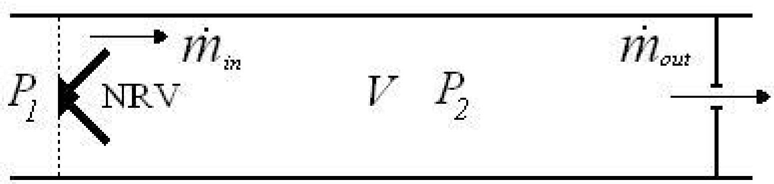

Figure 3. Flow configuration, parameters and arrangement.

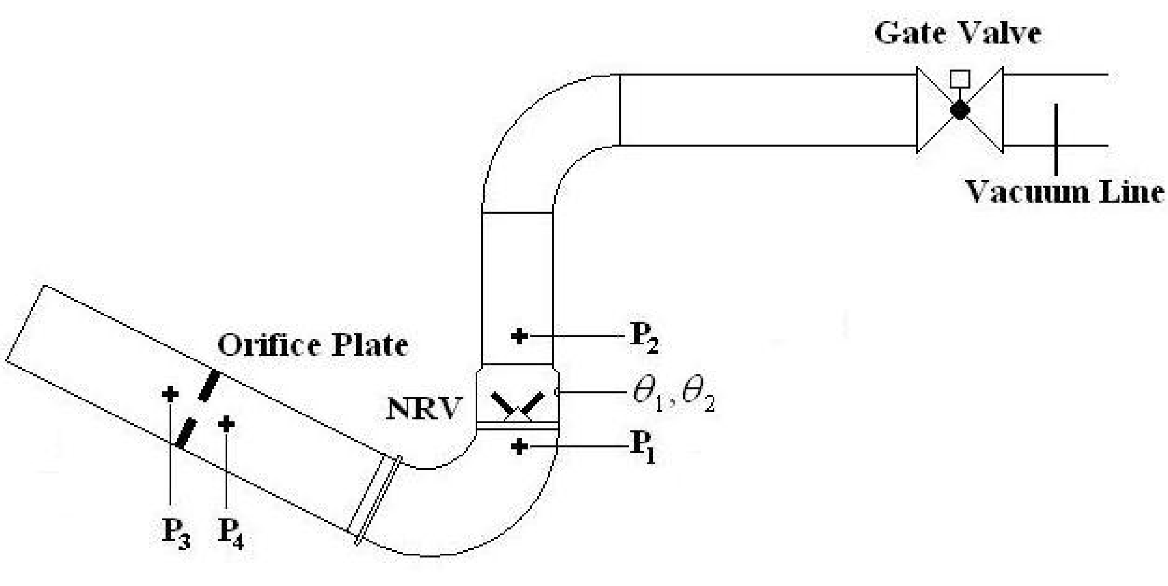

Figure 4. Schematic of experimental setup and transducers.

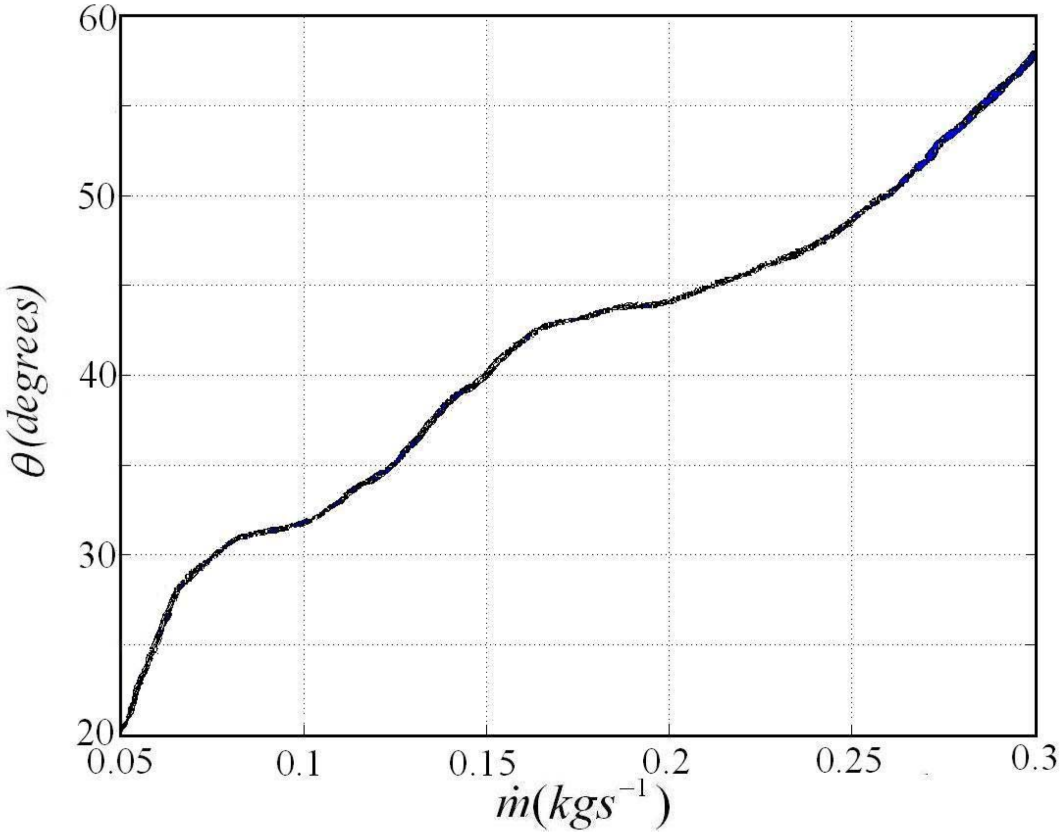

Figure 5. Variation of valve flap angle with system mass flow rate under stable operating conditions for all downstream volumes at different mass flow rates.

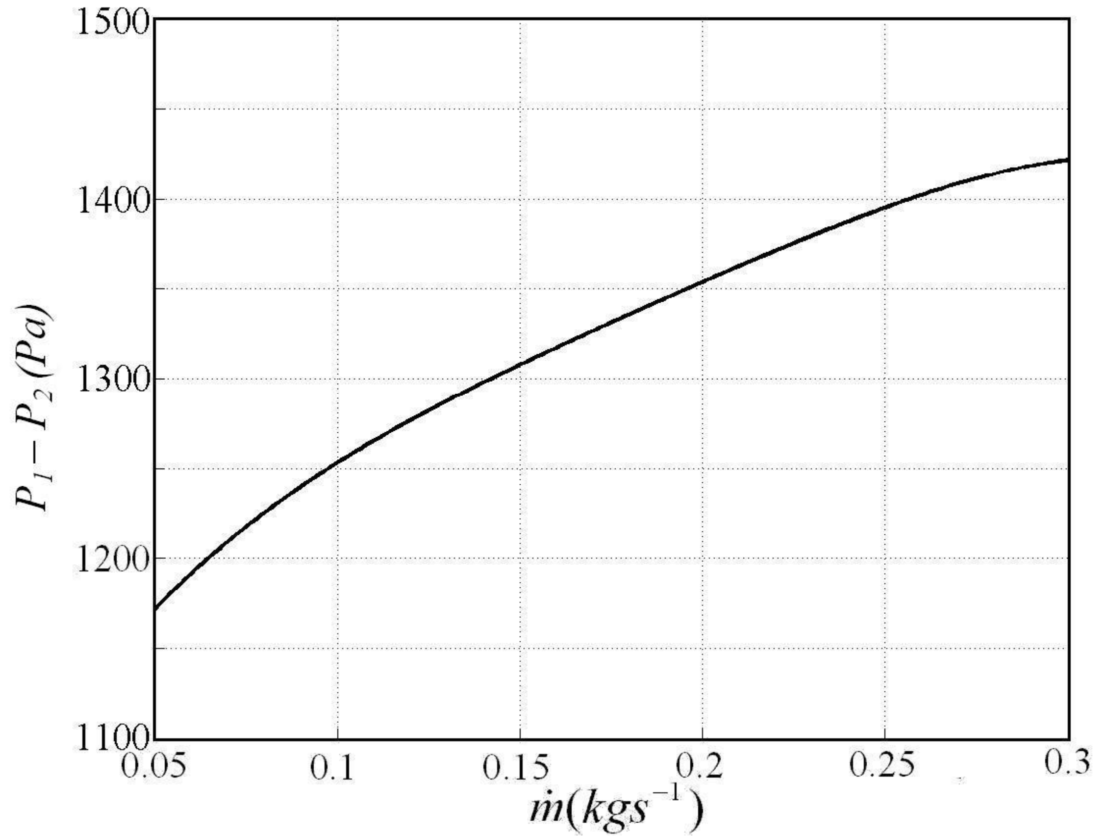

Figure 6. Variation of differential pressure across the valve with system mass flow rate under stable operating conditions for all downstream volumes at different mass flow rates.

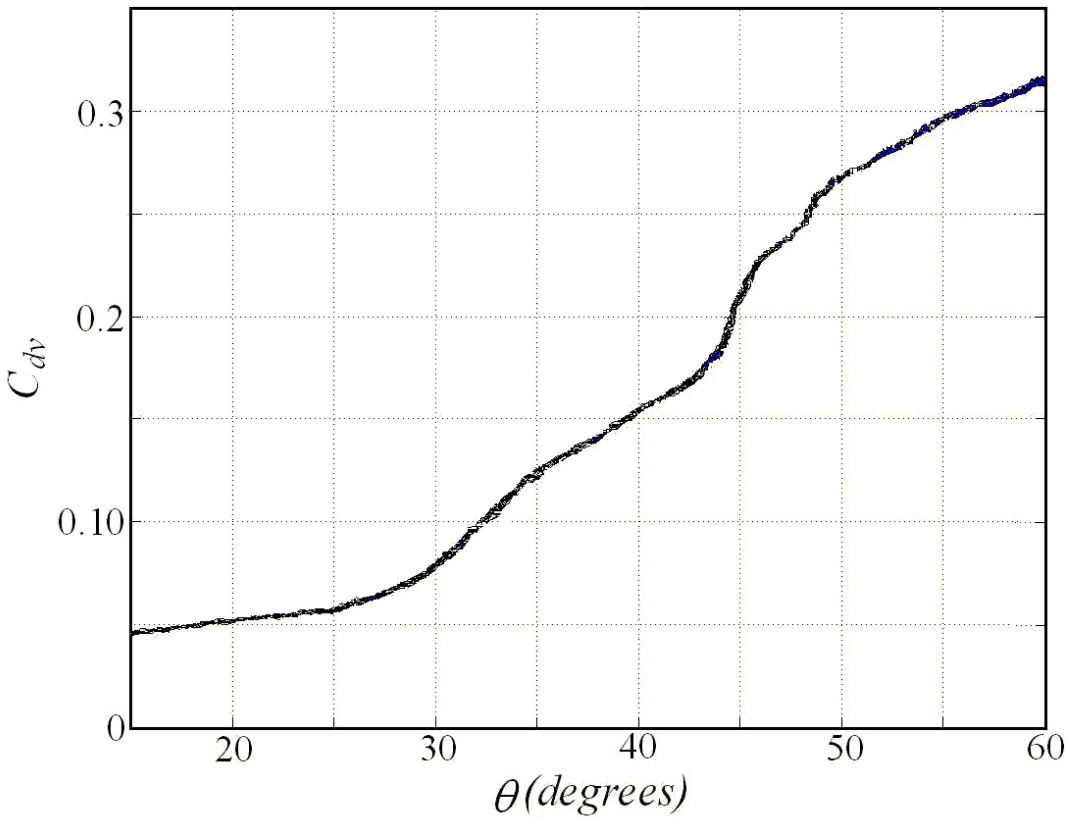

Figure 7. Variation of the static discharge coefficient with valve flap angle under stable operating conditions for all downstream volumes at different mass flow rates.

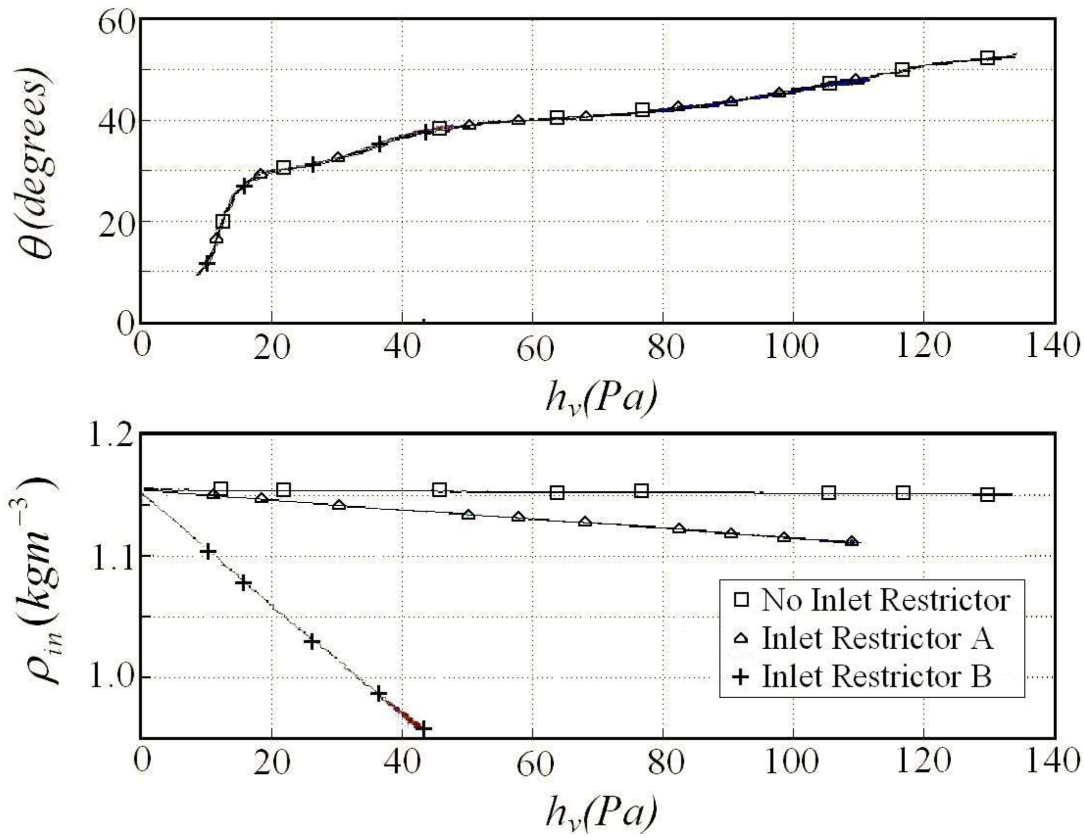

Figure 8. Variation of dynamic head with valve flap angle (a), for a range of inlet densities (b) under stable operating conditions for all downstream volumes at different mass flow rates.

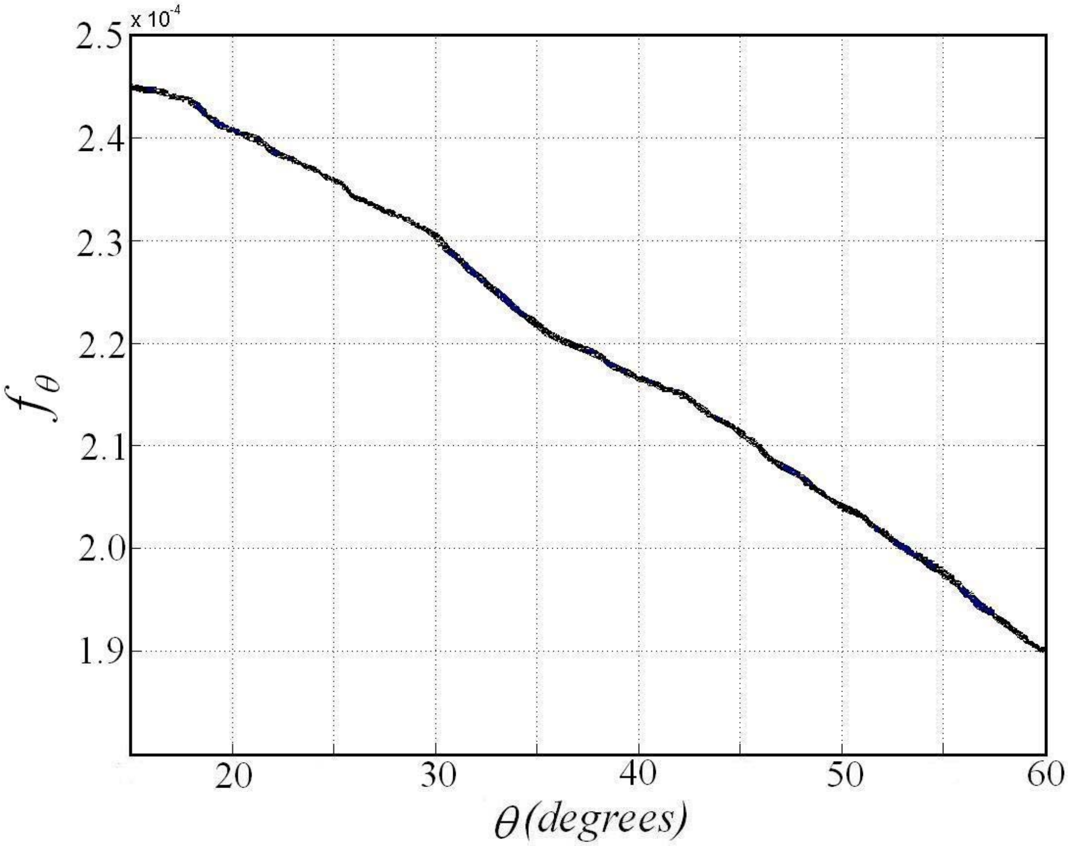

Figure 9. Variation in valve pressure loading coefficient with valve flap angle under stable operating conditions for all downstream volumes at different mass flow rates.

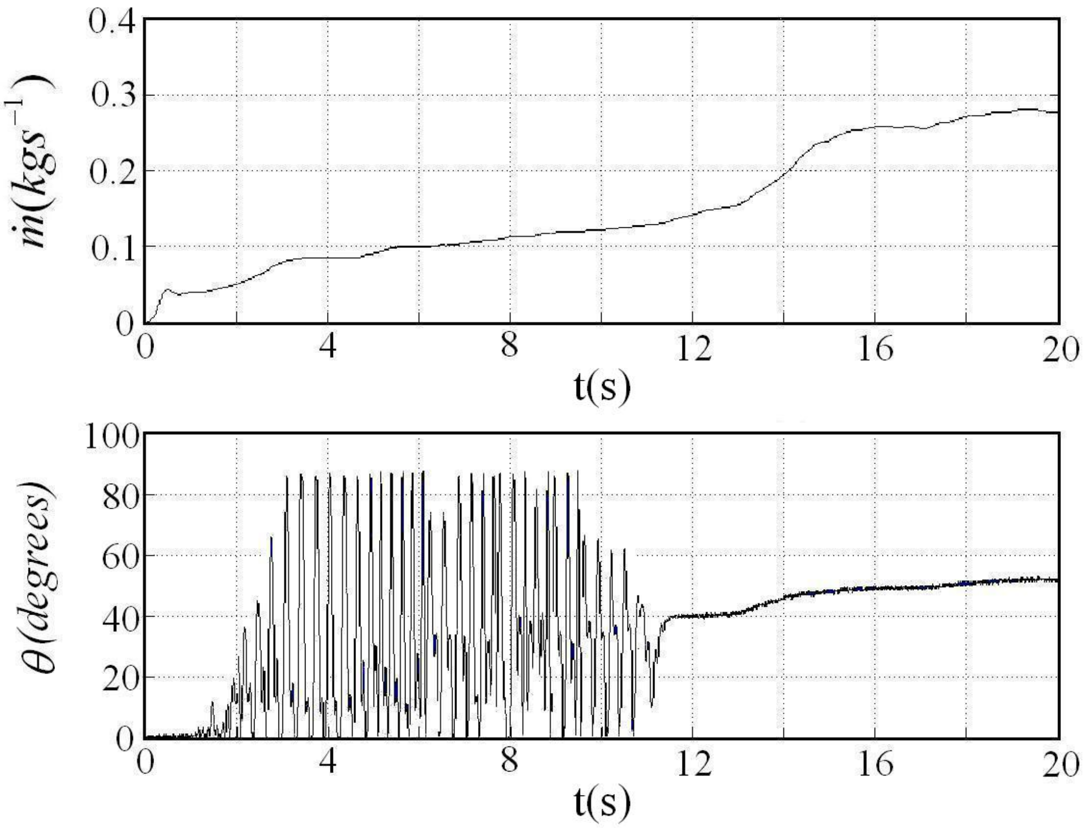

Figure 10. Example of valve instability as mass flow rate varies with time (top) and as valve flap angle varies with time (bottom) for a downstream volume of $0.034m^3$.

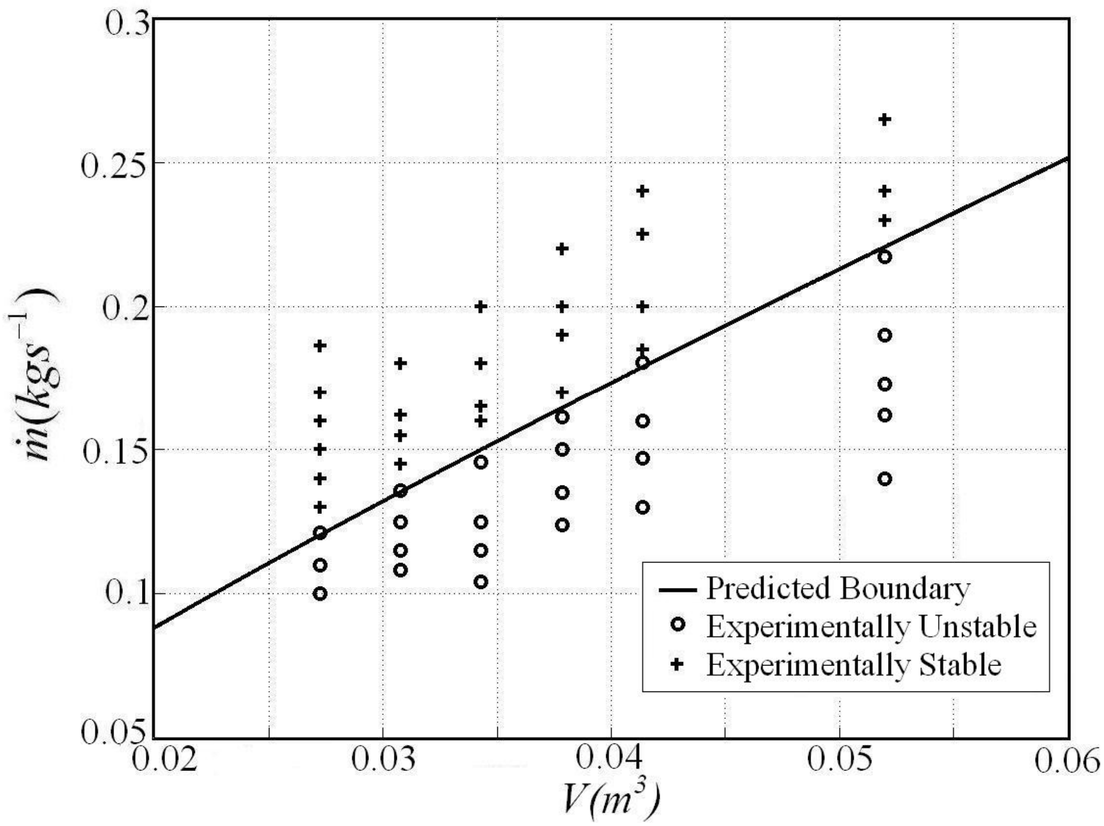

Figure 11. Theoretically predicted transition from stable to unstable behaviour (line) along with experimentally measured stable data points (o) and unstable data points (+).

Refined Globe Valve Thrust Prediction Model To Bound Midstroke Predictions of AOV Margins

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Abstract

Globe valve thrust prediction models are used to calculate margin between the required stem thrust to operate the valve and the available actuator thrust for air-operated valves (AOVs) in nuclear power plants. The "Bounding Thrust Zone" approach to model the required thrust is compared to the previous "Two-Point" model. The "Bounding Thrust Zone" model was introduced in 2000 by Kalsi Engineering, Inc. (KEI) to address the mid-stroke variations in required thrust due to flow-induced forces acting on the disc that can reduce AOV margins. Mid-stroke effects on required stem thrust was identified as one of the industry issues in the Electric Power Research Institute (EPRI) AOV Evaluation Guide issued in 1999.

For convenience, the flow-induced forces acting on a globe valve disc are decomposed into the stem rejection force, *FSR*; the differential pressure force, *FDP*, (the pressure force on the disc in the axial direction); and the transverse force (the pressure force on the disc acting perpendicular to the stem axis). For unbalanced valves, the change in the valve thrust requirement is largely due to the *FDP*, as such, the dependency of the *FDP* with regard to disc position is of primary interest.

Kalsi Engineering, Inc. (KEI) conducted an experimental investigation to better understand the dependency of the "knee" location in the Bounding Thrust Zone model on the flow characteristics, because the "knee" location can dictate the minimum margin in AOVs. The dependency of the *FDP* on the trim type was studied using a four-inch unbalanced Fisher globe valve in the flow-under the plug orientation. During the testing, the resultant stem load was measured for a quick open and equal percent trim set. The static friction forces and stem rejection forces were removed and the *FDP* results are discussed.

Test results showed that the location of the "knee" in the Bounding Thrust Zone can extend up to 55% of the stroke from the closed position for an equal percent cage. The results also showed that the relationship between the valve differential pressure and disc position was a good indicator of the relationship between the *FDP* and the disc position. Therefore, the "knee" location in the Bounding Thrust Zone for unbalanced valves can be estimated using the measured and/or predicted valve differential pressure. Furthermore, the methodology and considerations for determining the appropriate "knee" location methodology are discussed.

Introduction

In March 2000, the U.S. Nuclear Regulatory Commission issued the Regulatory Issue Summary, RIS 2000-03, "Resolution of Generic Safety Issue 158: Performance of Safety-Related Power-Operated Valves Under Design Basis Conditions" [1]. Previous studies by NRC [2, 3] and INPO [4] had identified common cause failures and problems with the operation of Air Operated Valves (AOVs), which can lead to safety concerns, reactor scrams, reduced plant efficiency and increased maintenance costs. The Joint Owners Group for Air Operated Valves (JOG AOV) developed programmatic guidance to the U.S. plants to implement AOV programs to verify the ability of the AOVs to perform their function under design basis conditions [5]. Technical guidance for evaluating AOVs under normal operating conditions and design basis conditions was provided by Electric Power Research Institute (EPRI) in 1999 [6]. The EPRI Guidance for AOVs was based on an earlier comprehensive research implementation program related to MOVs [7], and lessons learned from a pilot program related to AOVs at four U.S. plants as described in Reference 6.

EPRI identified several technical issues that may have direct impact on the evaluation of AOV margin [6]. One of the issues identified was that the mid-stroke load in AOVs may be greater than the fully seated loads, not only for quarter-turn valves, but also for globe and gate valves. For butterfly valves, this issue has been addressed by the development of more accurate, validated models which provide bounding required torque predictions at each disc position throughout the stroke [7, 8]. For globe and gate valves, the AOV Evaluation Guide provides equations to calculate required valve thrust and AOV margin at two disc positions: fully open and fully closed positions. Because the actuator output thrust/torque for typical diaphragm actuators or piston actuators also changes over the stroke, the actuator output for some air actuator designs may be lower in mid-stroke positions, thereby reducing the mid-stroke AOV margin to less than that calculated at the fully open and fully closed disc positions.

To address this potential mid-stroke non-conservatism for linear valves (globe and gate), a “Bounding Thrust Zone” methodology was introduced by Kalsi Engineering, Inc. and incorporated in the AOV evaluation software in 2000 [9]. This paper describes the Bounding Thrust Zone methodology and includes the key results from flow loop tests performed on a commonly used cage-guided globe valve with an unbalanced plug design with two different types of flow characteristics trim. The trim/flow characteristics are shown to have a significant influence on the mid-stroke thrust requirements. The paper also provides guidance for bounding the mid-stroke thrust requirements by determining the location of the “knee” in the Bounding Thrust Zone based on flow characteristics, valve design, actuator design, flow direction, fluid media, and operating conditions.

Comparison of Two-Point and Bounding Thrust Zone Models

Two models are discussed below and are illustrated using the required thrust for a flow-over the plug valve to illustrate the potential weakness of the Two-Point model and the enhancement offered by the Bounding Thrust Zone model.

The AOV Evaluation Guide provides the following equations for required stem thrust to open and close a globe valve [6]:

Opening Stroke

$$FO = FDS + FP + FUS + FSR + FDF + FDP$$

Closing Stroke

$$FC = FDS + FP + FUS + FSR + FDF + FDP + FSL$$

where

FDS	-	Stem thrust due to disc and stem weight
FP	-	Stem thrust due to packing friction
FUS	-	Stem thrust due to friction at upper seal (balanced plugs only)
FSR	-	Stem thrust due to stem rejection load
FDF	-	Stem thrust due to disc-to-body/cage friction (cage guided designs only)

FDP - Stem thrust due to ΔP

FSL - Stem thrust due to sealing load

For unbalanced globe valves, *FDP* is the dominant force component, and the maximum required thrust typically occurs at or near the fully seated position, where the valve differential pressure is greatest. The *FDP* force is calculated using maximum ΔP across the valve multiplied by either the seat area or guide area, whichever is applicable based upon guidance for "seat based" or "guide based" designs provided in Reference [10].

The *Two-Point model* (Figure 1.a) based on the AOV Evaluation Guide provides a simple approach to predicting the required thrust including the effects of the flow-induced forces. This Figure shows the required stem thrust for an unbalanced stem-guided globe valve with flow-over the plug orientation. This valve was used in a steam dump application at a fossil plant. The peak thrust (point A) for this valve occurs at approximately 30% open. In the Two-Point model, the maximum peak thrust is taken into account by transposing the peak force magnitude to the fully closed position, A'. Even though this approach provides a bounding value for the maximum required thrust at point A', the model does not bound required thrust over a portion of the stroke as indicated on Figure 1.a.

The *Bounding Thrust Zone model* (Figure 1.b), allows the user to bound the valve required thrust curve. The Bounding Thrust Zone extends the maximum required thrust (at A') from the closed position to a *user-defined location* referred to as the "knee" and identified by point C on Figure 1.b. The predicted maximum thrust value used over Bounding Thrust Zone is based on the flow-induced forces acting on the disc at the seated position multiplied by a factor to account for mid-stroke effects. The thrust value used over the Bounding Thrust Zone includes the flow-induced forces and frictional forces (like packing), but excludes mechanical loading (i.e., the seat load, *FSL*) between the disc and seat. Seating effects are included for closing stroke analyses at the fully closed position.

The location of the "knee", if incorrectly positioned, can result in an incorrect evaluation of the margin and possibly yield positive margin when negative margin exists, or vice versa. As shown in Figure 2.a, *it is important to note that the location of the "knee" is important only for opening stroke margin evaluations for the flow-over the plug orientation regardless of actuator action* (i.e., direct acting actuator or reverse acting actuator). This is because for the opening stroke, the actuator output decreases as the valve travels from the closed to open position. Whereas for the closing stroke, the actuator output decreases as the valve travels from the open to close position. For the closing stroke, the minimum margin always corresponds with the closed position. Since the actuator thrust curve always decreases during the opening stroke, the actuator action only determines whether the opening stroke is spring powered or air powered.

Methodology for Predicting "Knee" Location

The Two-Point model and Bounding Thrust Zone model are similar in one respect in that each model predicts the maximum required thrust and imposes that maximum required thrust value at the seat position. However, the Bounding Thrust Zone model allows the user to bound the entire thrust curve by defining a "knee" position. The required thrust is maintained constant from the seat position to the "knee", thus providing the means to bound the required thrust curve. This approach requires a priori knowledge of the behavior of the thrust curve.

For most on/off isolation valves, which commonly have trim with quick open flow characteristics, the "knee" location typically does not extend beyond 15%. However, for controlling valves that often use linear or equal percent flow characteristics, the maximum thrust requirement can extend out to and beyond 55% open [11,12].

The methodology provided here illustrates how the location of the "knee" can be estimated using the valve flow coefficient, *Cv*, (or valve flow resistance, *Kv*) and resulting valve pressure drop to provide more accurate valve predictions for unbalanced globe valves.

For unbalanced discs, the *FDP* is typically the dominant thrust component. The "knee" location is strongly dependent on the change in *FDP* with respect to disc position. The *FDP* is influenced by key factors including the clearance between the disc head and cage/body wall, the difference between the effective pressure area based on the maximum disc head diameter and the seat diameter, the length of the "dead zone" between the seat and cage-port or outlet duct opening (if applicable), and the flow characteristics defined by the disc nose contour or cage port geometry. The effect of these variables is discussed later.

Predicting the "knee" location using the valve differential pressure curve is possible because *FDP* is largely dependent on the valve differential pressure. The *FDP* is primarily governed by the pressure acting on the disc. However, the *FDP* can be related to the valve differential pressure by Equation (1) below.

$$C_{DP} = \frac{F_{DP}}{DP_v \, A_{seat}} \quad (1)$$

In Equation (1), *FDP*, is made dimensionless by the valve differential pressure, *ΔPv*, and the seat area, *Aseat*. The seat area is based on seat diameter; therefore, the coefficient at the closed position is equal to unity. For valves with minimal clearance between the disc load and cage/body wall, *CDP* increases from a value of unity at the seat to larger value based on the maximum disc diameter.

From Equation (1), it is clear that a change in *FDP* (i.e. - location of the "knee") occurs only when either *ΔPv* or *DCP* have a sharp change in their respective slopes. Because *CDP* is dependent on disc position, which also affects the *ΔPv* and the corresponding pressure acting on the disc, it is reasonable to assume that both *ΔPv* and *CDP* undergo a noticeable decrease near the same disc position, which is close to the true "knee". Since *CDP* can only be obtained through testing, the *ΔPv* curve (based on experimental measurement or analytical prediction) can be used to estimate the "knee" location. As such, the differential pressure ratio (*ΔPR*) (i.e., the ratio of the *ΔPv* to system *ΔP*) curve for the quick open, linear and equal percent trim are compared in Figure 3 using their respective valve flow coefficients and a reasonable value for the upstream and downstream resistances (i.e., *Kup* + *Kdown* = 10). The *Kv* values for the three trim sets used to calculate the *ΔPR* in Figure 3 are based on the 4-inch Fisher ES valve *Cv* values given in Reference 15.

The valve trim characteristics, and consequently the *ΔPv*, can be established by either the cage contours or disc-head contour (see Figure 4). The flow characteristics of the Fisher ES valve are determined by the cage, which is of direct relevance as a four-inch ES valve was tested by KEI. The Fisher ES quick open, equal-percent, and linear cages are shown in Figure 5. The *ΔPR* curve shown in Figure 3 was calculated using a simplified approach. The *ΔPv* is dependent on many factors including the internal valve body geometry, flow characteristics, operating conditions, and flow state (i.e. incompressible, compressible, cavitating, or flashing). However, a simplistic model relating the valve differential pressure with the disc position for non-cavitating incompressible flow is given by

$$\Delta P_R = \frac{\Delta P_v}{\Delta P_{sys}} = \frac{K_v(x)}{K_{up} + K_v(x) + K_{down}}. \qquad (2)$$

In Equation (2), *Kv* is the valve resistance and is a function of disc position, *x*; *Kup* and *Kdown* are the upstream and downstream resistances respectively; and *ΔPsys* is the total system differential pressure. For convenience, $\Delta Psys$, Kup and $Kdown$ are maintained constant with respect to the disc position.

The valve resistance related to the flow coefficient (see Reference 14) by Equation (3) given as

$$K_v = \frac{891 d^4}{C_v^2}. \quad (3)$$

In Equation (3), d is the reference diameter (typically the pipe internal diameter) in inches and Cv is the valve flow coefficient in GPM/(psi)$^{1/2}$. It should be noted the Kv is dimensionless despite d and Cv both having dimensions.

From Equation (2), it is seen that the ΔPv is proportional to the ratio of the valve resistance to the total system resistance. Therefore, the ΔPv can be much larger for the equal percent and linear trim compared to the quick open trim for a given system differential pressure, as seen from Figure 3. This is especially true between 20% to 80% open where the relationship between the valve resistance and disc position for the equal percent and linear trim deviate most from that of the quick open trim. The difference in the ΔPv over this part of the stroke suggests a potential difference in the behavior of *FDP* throughout the stroke for the three trim types.

From Equation (2) it can be seen that as the total system resistance goes toward the valve resistance (i.e., $Kup+Kdown \rightarrow 0$) the *DPR* value throughout the stroke goes toward unity. In other words, the full pressure drop occurs across the valve at each disc position, but the quick open trim will have a smaller required thrust than the equal percent trim, ignoring any possible effects due to cavitation.

Experimental setup

The test valve was installed in a water flow loop at KEI shown in Figure 6.a. During the testing the stem load; upstream, downstream, diaphragm, and upper body gallery pressures; flow rate; and stem position were measured. The stem load was measured using a Crane ForceLink (shown in Figure 6.b), which is used in place of a standard valve marriage block (the connecting piece between the actuator and valve stem). The flow rate was measured using an orifice plate located upstream of the valve. The upstream pressure was measured two pipe-diameters upstream of the valve, and the downstream pressure was measured six pipe-diameters downstream of the valve. The stem position was measured using a linear voltage differential transducer (LVDT) that was mounted to a bracket on the actuator. The LVDT wire was attached to a rod extending from the marriage block.

The valve was tested under one static condition and three sets of dynamic conditions with both the quick open and equal percent trim installed. The three different dynamic conditions were defined by the maximum valve differential pressure at the closed position. The three valve differential pressures used were 30, 60 and 90 psi.

Experimental results

The flow coefficient, *Cv*, was calculated from the measured flow rate and valve differential pressure. The calculated *Cv* values for the equal percent and quick open trim are compared to values from Reference 15 in Figure 7.

The static stoke test results are summarized in Figure 8. The diaphragm pressure and measured packing load indicate that the valve was in good operating condition.

The *FDP* was extracted from the measured force obtained using the ForceLink. The *FDP* for the Equal Percent trim with the corresponding valve differential pressure is compared in

Figure 9. From Figure 9, it can be seen that the *FDP* begins to decay at about the same location (~55% open) as the valve differential pressure shown in Figure 3.

The *FDP* coefficient for the equal percent and quick open trim are compared in Figure 10. From Figures 9 and 10, it can be seen that the "knee" in the equal percent and quick open disc head force coefficient can be located from their respective valve differential pressure ratio curves. Note that the differential pressure curves provided in Figures 3 and 9 are system dependent and a decrease in system resistance would result in a shift in the differential pressure curves to the right and thus extend the "knee" position towards the open position. Conversely, as the total system resistance increases, the "knee" shifts toward the close position.

It can be seen from Figure 3 that for the quick open trim, the valve differential pressure is nearly constant from 70% to 100% open, as is the disc-head force coefficient, which indicates the disc head force is nearly neutral over this range. This is in contrast to the equal percent trim whose differential pressure continues to decrease along with the *CDP* over this same region.

Model validation and discussion

As previously mentioned, the Bounding Thrust Zone provides enhanced modeling capability for mid-stroke positions for the opening stroke of flow-over the plug valves. However, the discussion provided below is applicable to peak thrust predictions for all valve orientations and stroke directions.

Model Validation

A validation of the Bounding Thrust Zone model was made using the experimental data for the equal percent cage. The model predictions were made using the following key parameters: The Bounding Thrust Zone "knee" location (of 55%), maximum disc head diameter, port diameter, packing friction, upstream and valve differential pressure at the full open and full closed positions. The comparison of the test data with the model predictions are given in Figure 11. The comparison shows that prediction bounds the test data throughout the stroke, yet excessive conservatism is not present.

Discussion

Correct application of the Bounding Thrust Zone requires a bounding prediction of the maximum required thrust and a bounding prediction of the "knee" location. The maximum thrust and "knee" location depend on the following factors:

Whether the disc is balanced or unbalanced,

Flow characteristics (i.e., equal percent, quick open, linear),

The difference in area based on the maximum disc head diameter and seat diameter. (This is especially important when a small clearance is present between the disc and seat.),

The clearance between the disc head and the cage/body wall

The presence of body guide ribs, which directs the flow over the disc instead of around the disc (body ribs are used to guide the disc in some valves in place of a cage),

The extent of the "Dead Zone." (The "Dead Zone" is the stroke length between when the disc lifts from the seat and clears the outlet or cage port),

Body style (i.e., T- or Y-pattern),

Effects on the thrust and ΔPv of the flow state (i.e., presence of cavitation for incompressible flow or sonic or transonic sonic flow for compressible flow), The change in the valve resistance with respect to the total system resistance as a function of disc position.

The effects of some of these factors have been studied; however, a comprehensive study of the entire set has not been performed, thus only general guidance can be given for many of these factors. The important point to identify when evaluating these factors is whether they prolonged or increase differential pressure across the disc or whether they cause an increase in the area acted on by the differential pressure.

For example, the peak thrust, when not based on the *FDP* value at the seat, often occurs due to an increase in the disc area exposed to the high-pressure fluid and/or an increase in the differential pressure across the disc compared to the valve differential pressure.

In the previous example, the increase in area could occur as the disc lifts from the seat. The initial area the pressure acts on is based on the seat diameter; however, if the disc width is greater than the seat diameter (as shown in Figure 12) an increase in the “effective area” can occur. The actual increase in effective area is dependent on the maximum disc diameter and the clearance between the disc head and body. For example, the disc head in Figure 12.a has a relatively large clearance between the disc and body wall whereas the disc in Figures 12.b and 13 has a small clearance. If the clearance between the disc and cage/body wall is small and creates large resistance compared to the restriction between the disc and seat, a high differential pressure can act across the entire disc face, increasing *FDP* compared to the seat position. If the clearance between the disc and cage/body wall is large and creates small resistance compared to the restriction between the disc and seat, a local reduction in the pressure on the disc can occur and decrease the effective area, decreasing *FDP* compared to the seat position.

In some cases, the average differential pressure acting across the disc can be larger than the valve differential pressure. This occurs when a local drop in pressure is present either above the disc for flow-under the plug valves (or below the disc for flow-over the plug valves), but is later recovered as the flow leaves the valve outlet.

Conclusions

For MOV predictions only a single point bounding prediction is required due to the nearly constant actuator output. However, for AOVs, which have a position dependent output, the required thrust prediction model must bound mid-stroke effects. The Bounding Thrust Zone model provides the means using the “knee” position to bound the required thrust throughout the stroke. The “knee” position in the Bounding Thrust Zone model can be predicted using either the measured or analytically predicted valve differential pressure.

The Bounding Thrust Zone model is most applicable to controlling valves not associated with isolation, as most isolation valves are quick open and likely have a relatively small bounding thrust zone (less than 15%). However, the refined model provides higher accuracy for valves that can have a significant pressure drop across the disc at the mid-stroke positions due to trim restrictions and is required for open stroke margin predictions for flow-over the plug valve.

To facilitate model refinement, a four-inch Fisher ES valve was tested with a quick open and equal percent cage. Based on the test results, the *FDP* for the two cages was found to be strongly dependent on the valve pressure drop, which in turn is related to the valve and system resistances.

The Bounding Thrust Zone model predictions were made using the measured thrust from the equal percent cage. The measured valve differential pressure indicated a “knee” position of 55%, which was used in the Bounding Thrust Zone and yielded bounding predictions.

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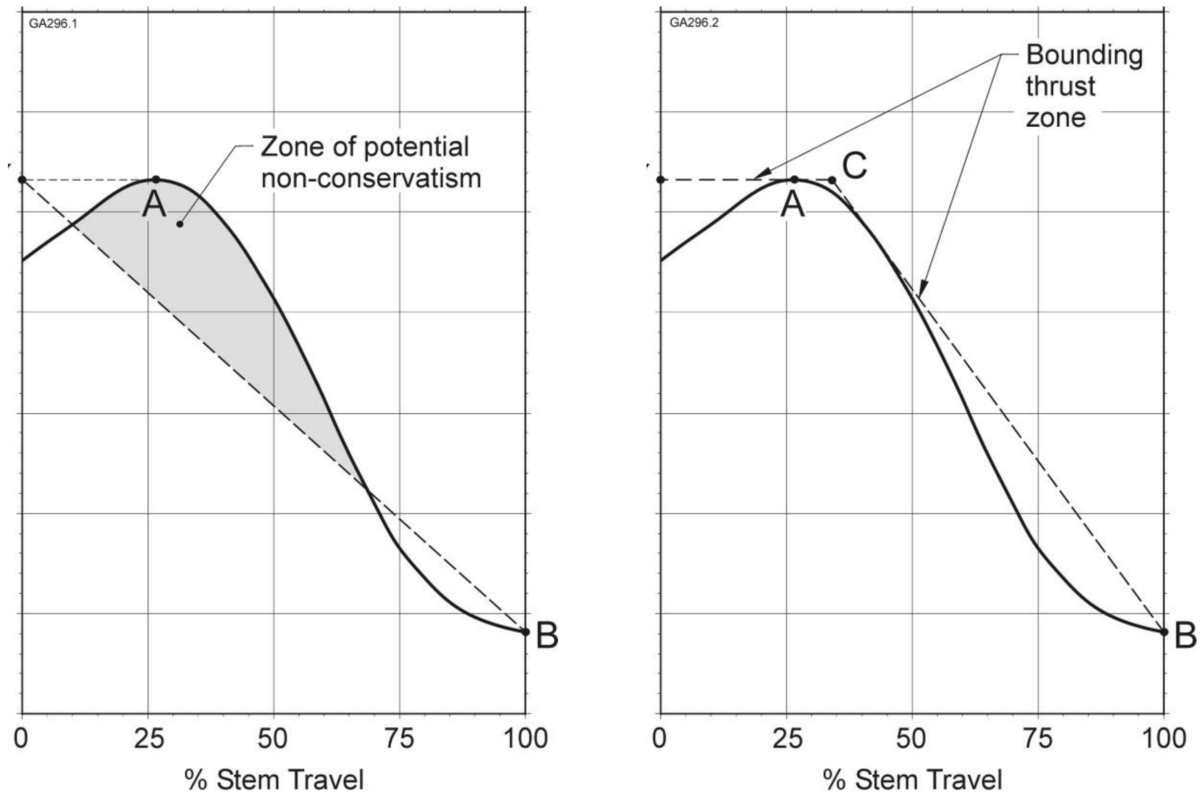


a) Two-Point Model

b) Bounding Thrust Zone Model

Figure 11

Comparison of Two-Point & Bounding Thrust Zone model

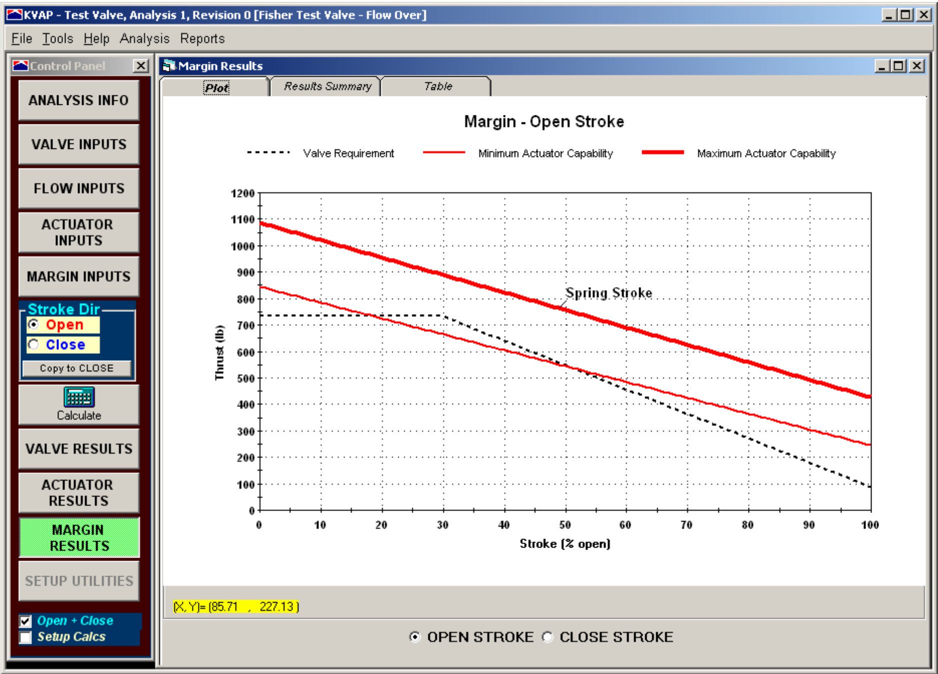


Figure 12.a

The location of the "knee" in the Bounding Thrust Prediction dictates the minimum margin between the required stem thrust & available actuator force for opening stroke direction

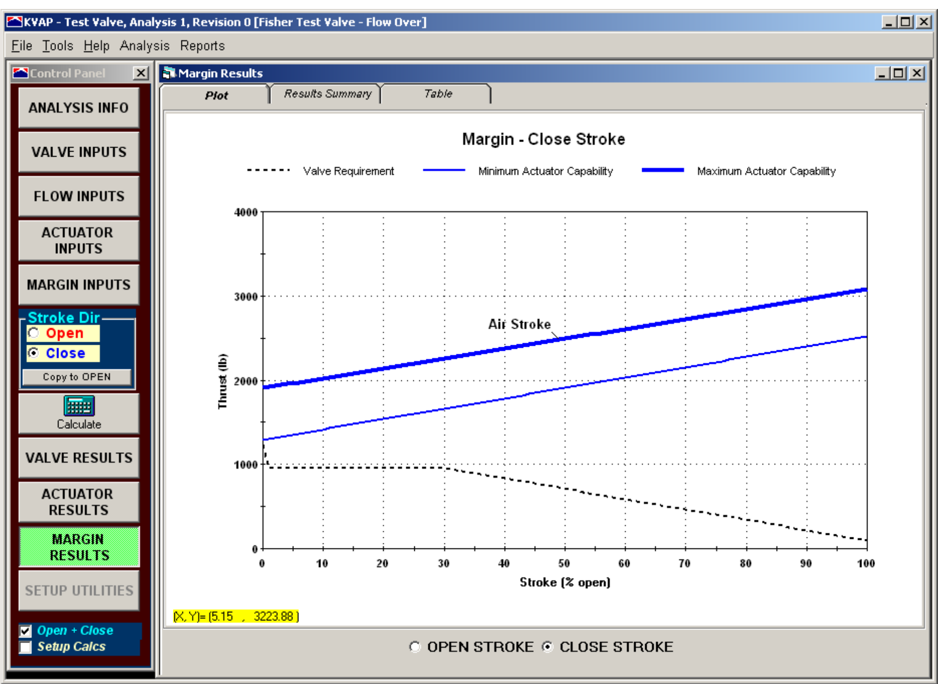


Figure 2.b

The location of the "knee" in the Bounding Thrust Prediction does not dictate the minimum margin between the required stem thrust & available actuator force for closing stroke direction

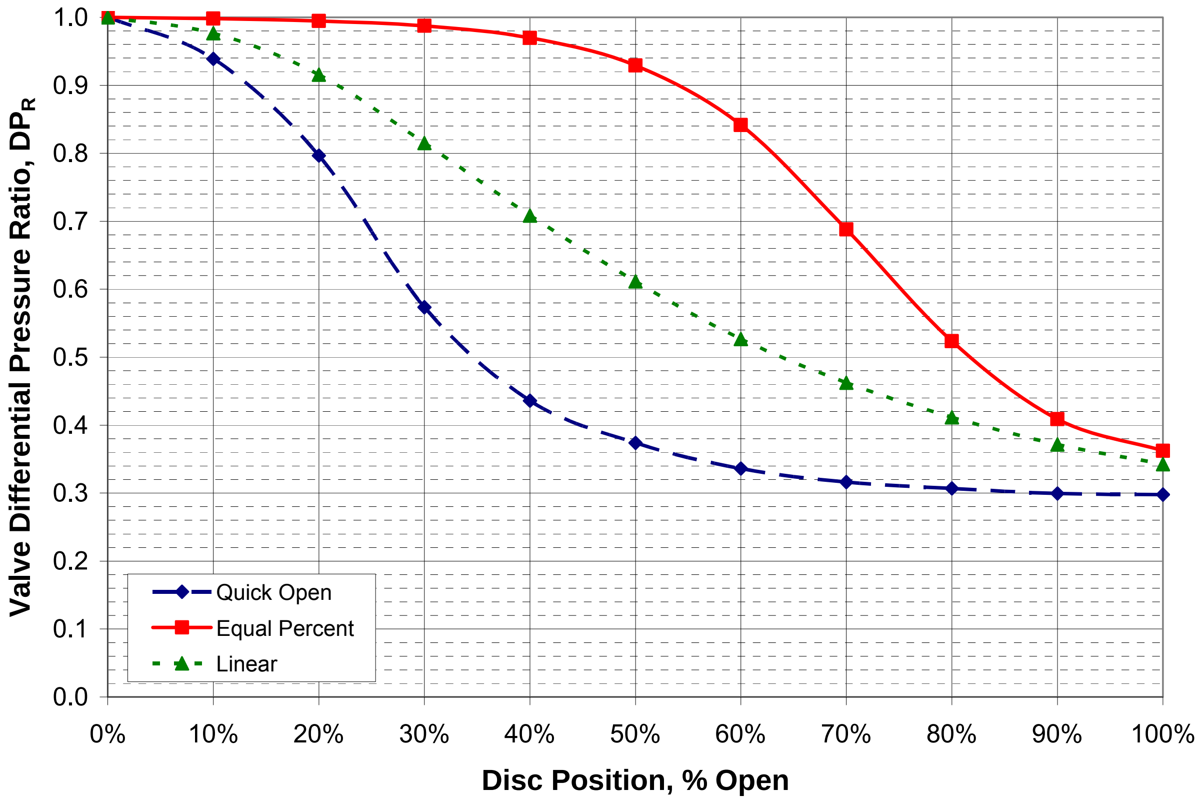


Figure 13

The Valve Differential Pressure Ratio (ΔPR) for the three trim sets is given for a value of Kup + Kdown =10. The equal percent trim results in the largest differential pressure across the valve.

a) Fisher ES trim

B) Fisher EZ trim

Figure 14

Illustration of the a) Fisher ES trim and b) Fisher EZ trim (taken from Reference 13). The flow characteristic for the ES trim are set by the cage, whereas the characteristics of the EZ trim are set by the disc curvature.

a) Quick Open b) Equal Percent c) Linear

Figure 15

The a) Quick Open, b) Equal Percent, and c) Linear trim for a Fisher Design ES valve are shown (taken from Reference 13).

a) The 4-inch Fisher ES valve installed at KEI

b) The Crane ForceLink used to measured thrust and LVDT used to measure stem travel

Figure 16

The test valve installed at the KEI flow loop

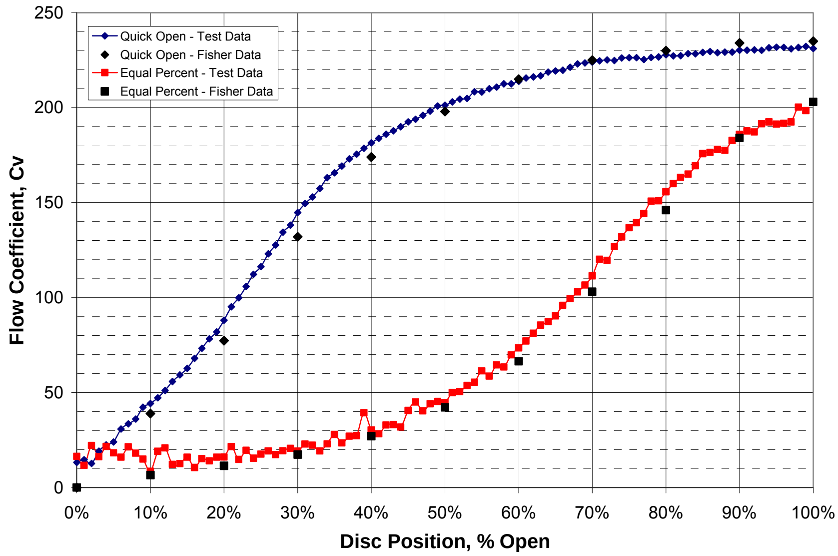


Figure 17

Calculated values of the valve flow coefficient are compared and found to be in good agreement with published data (see Reference 15).

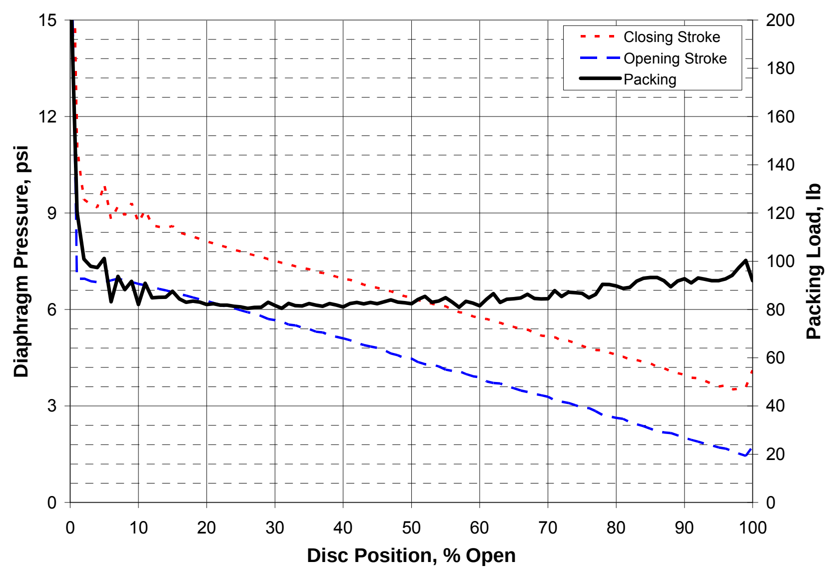


Figure 18

Static stroke results showing the actuator diaphragm pressure and measured packing thrust.

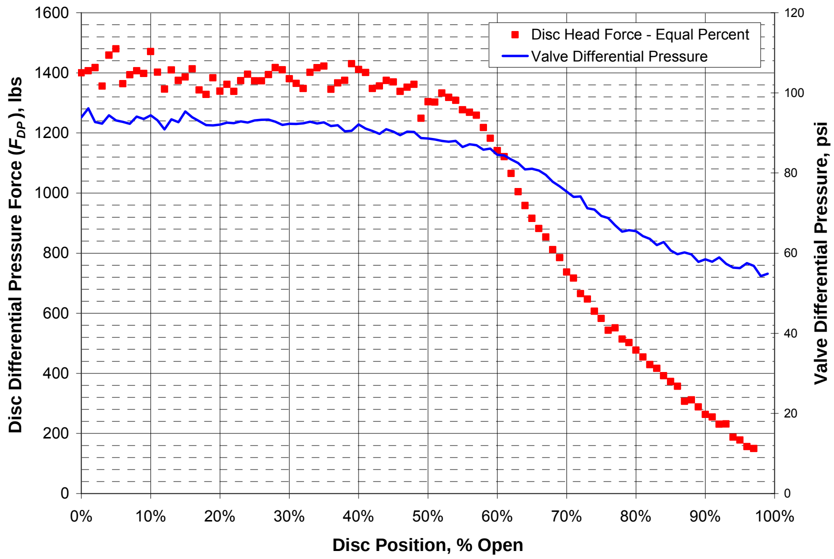


Figure 19

The FDP and valve differential pressure for the equal percent trim are compared. The FDP and valve differential pressure are generally flat from the closed position until about 50% open, at which point both values begin to decrease. The estimated “knee” position is 55% based on a tangent intersection of the differential pressure curve.

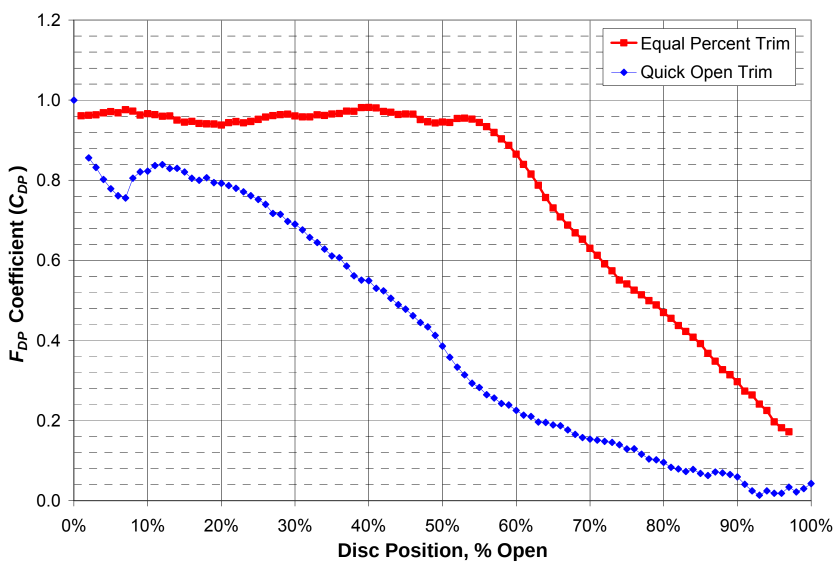


Figure 20

The FDP coefficient for the equal percent and quick open trim are compared. The coefficient curves have the same general behavior as the DPR curves shown in Figure 3.

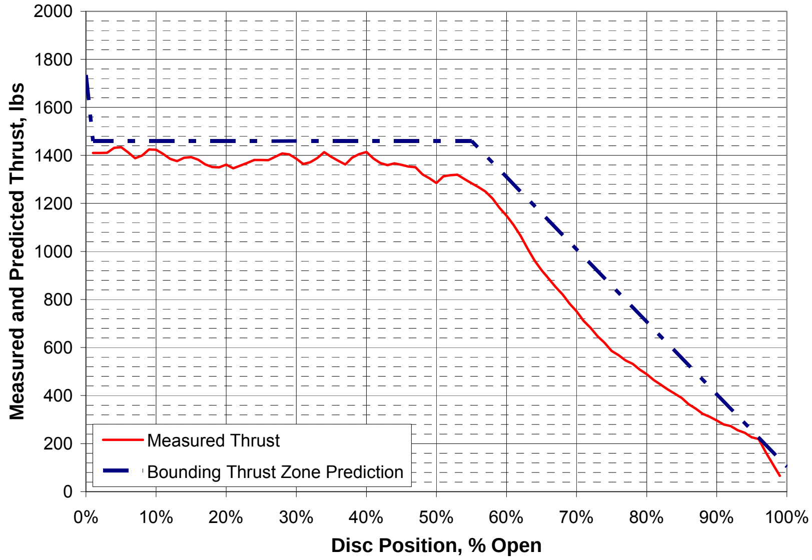
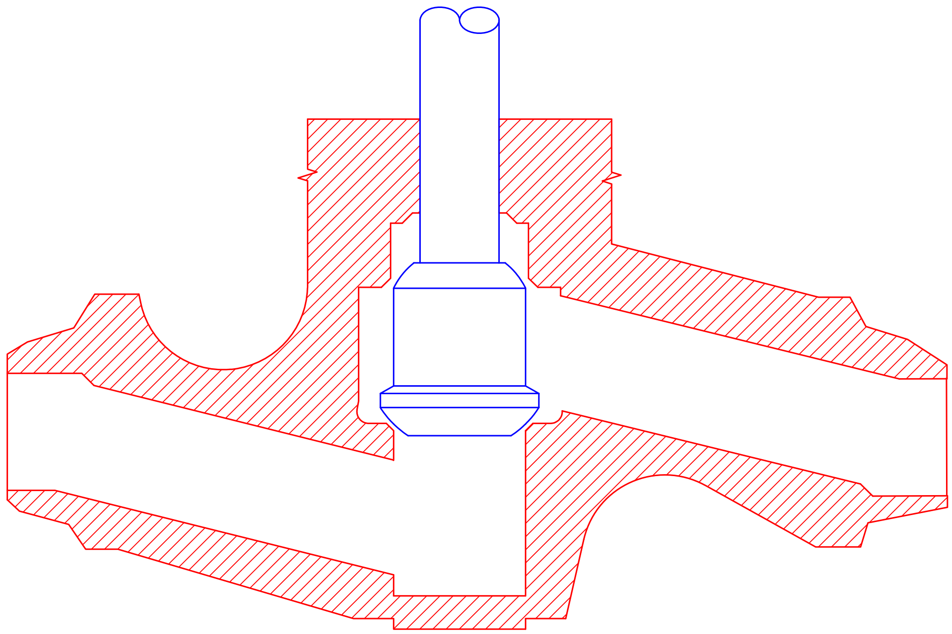
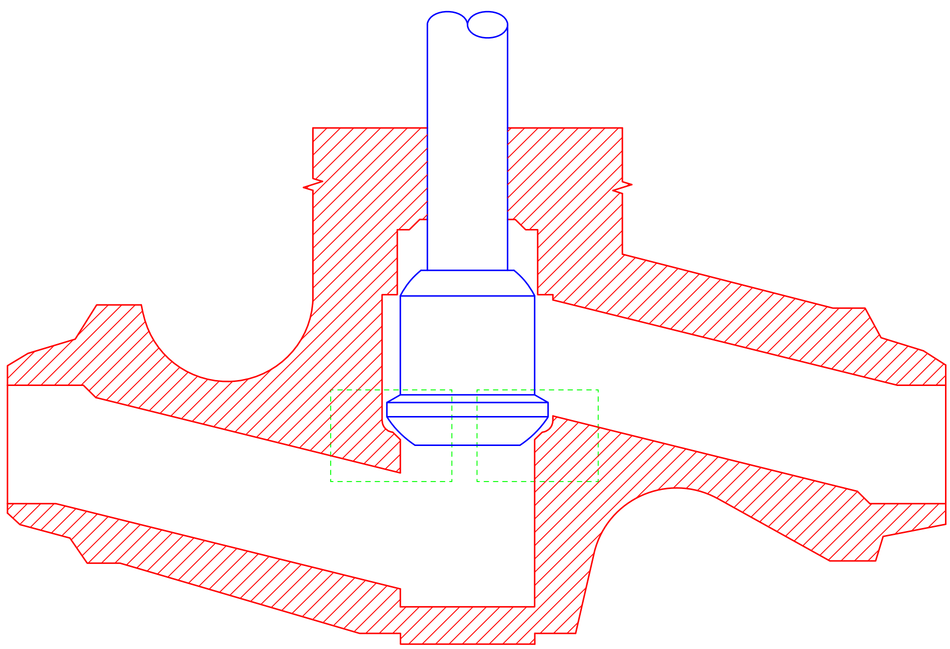


Figure 21

Comparison of the Bounding Thrust Zone model predictions with the measured required thrust. Correct placement of the “knee” provides a bounding prediction at each disc position throughout the stroke.



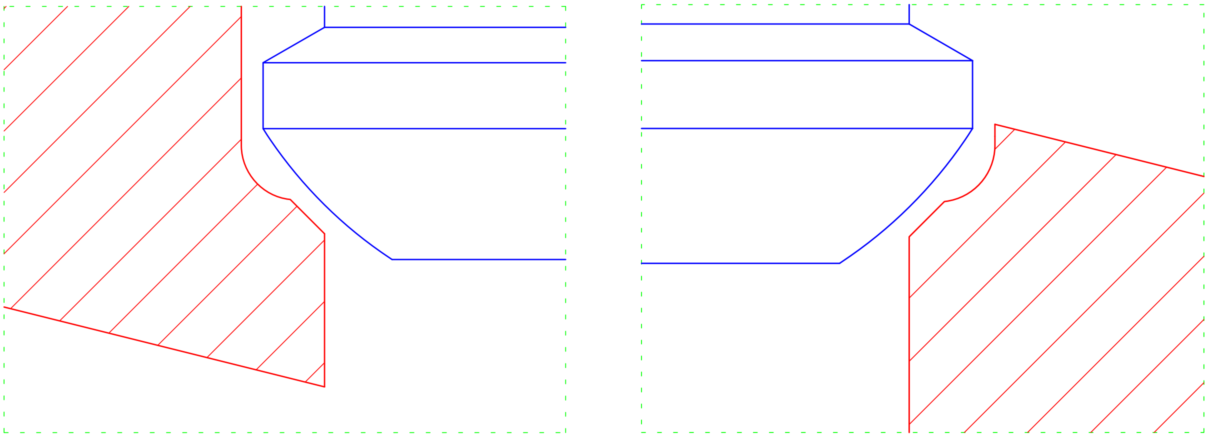
a) Minimal resistance due to the wide disc head to body wall clearance



b) Increased resistance due to the narrow disc head to body wall clearance

Figure 22

The disc head diameter, seat diameter, disc head to cage/body wall clearance and trim characteristics can affect the maximum thrust location and value, thus affecting the “knee” location used for the Bounding Thrust Zone Model. (Detail views shown in Figure 12)



a) Enlarged view of left detail from Figure 12.b

b) Enlarged view of right detail from Figure 12.b

Figure 23

The disc head clearance and “dead zone” (i.e., distance the disc must travel before clearing the outlet port for flow-under the plug valves) can affect the maximum thrust value and location.

Pressure Relief Valve Set Point Accuracy

Chad R.H. Dupill
DeLuca Test Equipment

Abstract

Testing methods and devices vary widely within the industry when testing for the set point of Pressure Relief Valves. Testing devices within Nuclear Plants range from Shop built devices, Hand Pumps, and Manufacturer built test devices among others. The purpose of this paper is to discuss some of the different design characteristics and how those design characteristics can affect your set point when testing for the set point of Pressure Relief Valves in both gas and liquid service. This paper references The American Society of Mechanical Engineers (ASME) Pressure Relief Devices Performance Test Code 25-2001 (PTC 25-2001), The National Board Inspection Code (NBIC), and plant testimonials. A special thanks to J. Alton Cox (Vice Chairman, NBIC Sub-Group Pressure Relief Devices) for his contributions in guidance, and experience.

Introduction

When using a Pressure Relief Valve (PRV) test bench or bench testing as it is referred to in the ASME PTC 25-2001 Section II, there are several design characteristics found on commonly used test benches that can affect the observed set point when testing Pressure Relief Valves. The purpose of this paper is to discuss some of these characteristics and how they can affect the determination of the set point when conducting a bench test.

The first design characteristics addressed is the importance of following the recommendations of ASME PTC 25-2001 on internal contours of fittings, adapter, and reducers between the test vessel and test device (see figure 1) and the importance of having a sufficient ASME code pressure vessel that provides sufficient volume, as well as large Stainless Steel Tubing with minimal and subtle bends (see figure 2). Following these design recommendations are important when discussing the effects of not having sufficient volume as well as the effects of media turbulence to the outcome of a set point test.

Second, this paper will address PRV seat alignment when testing PRV's for both Air and Liquid as well as the importance of having sufficient volume. This will followed by a discussion addressing the trapped air or "Air Bubble" between the test valve disk and water when conducting a liquid service PRV test and the affect it has on the accuracy of the "as found" test and a review of commonly used air evacuation methods.

Definitions

NOTE: For the purpose of this paper we are going to define the following as:

<u>High Volume test bench</u> – PRV Test device with at a minimum capacity volume of 28 Liter [1 cubic foot] test vessel and a minimum piping (between the Test Vessel and the Test Connection for the Valve being tested) diameter of 5 cm [2 inches].

Low Volume test bench – PRV Test device with a test vessel less than 28 Liter [1 cubic feet] and a piping diameter (between the Test Vessel and the Test Connection for the Valve being tested) of less than 5 cm [2 inches].

Set Pressure – The value of increasing inlet static pressure at which a pressure relief device displays one of the operational characteristics as defined under opening pressure, popping pressure, start-to-leak pressure, burst pressure, or breaking pressure. (The applicable operating characteristic for a specific device design is specified by the device manufacturer.) PTC 25-2001

Opening Pressure – The value of increasing inlet static pressure of a PRV at which there is a measurable lift, or at which the discharge becomes continuous as determined by seeing, feeling, or hearing. (For liquid Service) PTC 25-2001

Popping Pressure – The value of increasing inlet static pressure at which the disk moves in the opening direction at a faster rate as compared with corresponding movement at higher or lower pressures. PTC 25-2001

Start-to-Leak Pressure – The value of increasing inlet static pressure at which the first bubble occurs when a PRV is tested by means of air under a specified water seal on the outlet. PTC 25-2001

Chatter – abnormal rapid reciprocating motion of movable parts of a pressure relief valve in which the disk contacts the seat. PTC 25-2001

First Steady Stream – Set pressure for liquid Service Pressure Relief Valves. When gravity overcomes cohesion the water drops straight off the Pressure Relief Valve outlet; an observable, repeatable Phenomenon. NB 18

Cohesion – the attraction of like molecules

Note: For the purposes of this paper Test Bench, Test Device, Test Stand, and/or Test Unit are interchangeable.

Internal Contours of Fittings, Adapter's, and Reducers between Test Vessel and Test Device

When testing and setting liquid and gas service Pressure Relief Valves (PRV's), minimal flow turbulences are necessary in order to get the most accurate reading when determining a PRV set point. ASME PTC 25-2001 states that "The pressure relief device to be tested shall be installed on a test vessel with adapter fittings (flanged, screwed, welded, etc.)." See Fig. 1 for recommended, acceptable adapter fitting contours for minimum inlet pressure drop from PTC 25-2001.

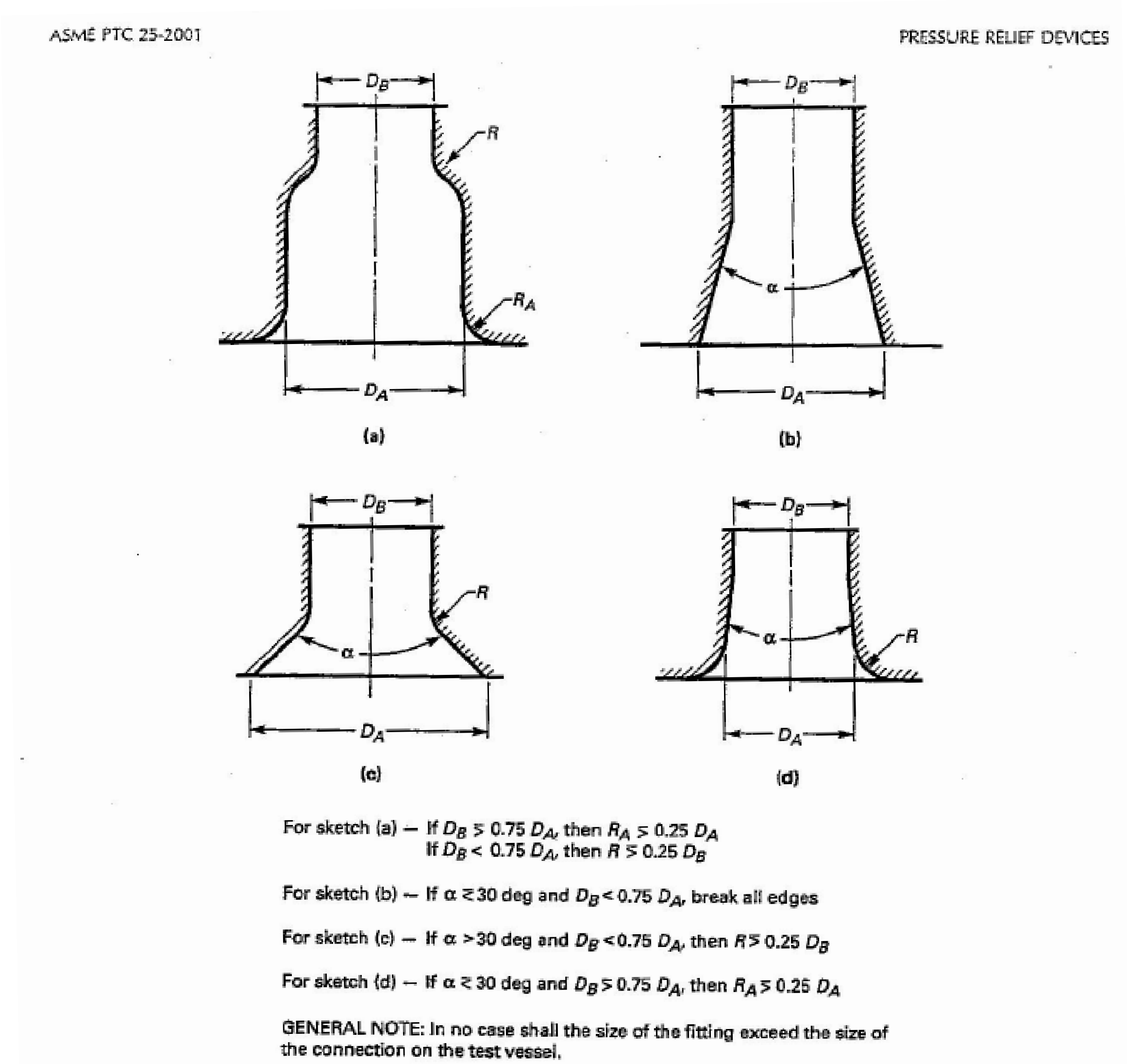


Fig. 1 Recommended Internal Contours of Fittings, Adapters, and Reducers between Test Vessel and Test Device

For example as the medium (either liquid or air) travels from the test vessel, or associated piping, to the test valve it is recommended that there are as little flow interruptions as possible so that the inlet pressure drop is minimized which in turn impacts the accuracy of the test. The NBIC also states that "any intervening piping between the test vessel and the pressure relief valve should be as short and straight as possible and be of adequate size to minimize inlet pressure drop." Refer to Fig. 2 from ASME PTC 25-2001 of the recommended arrangement for testing valves with incompressible fluids. The test unit depicted in Figure 2 gives a very good example of how the route and size of piping as well as the flanged reducing adapter's are implemented in the design of the test unit.

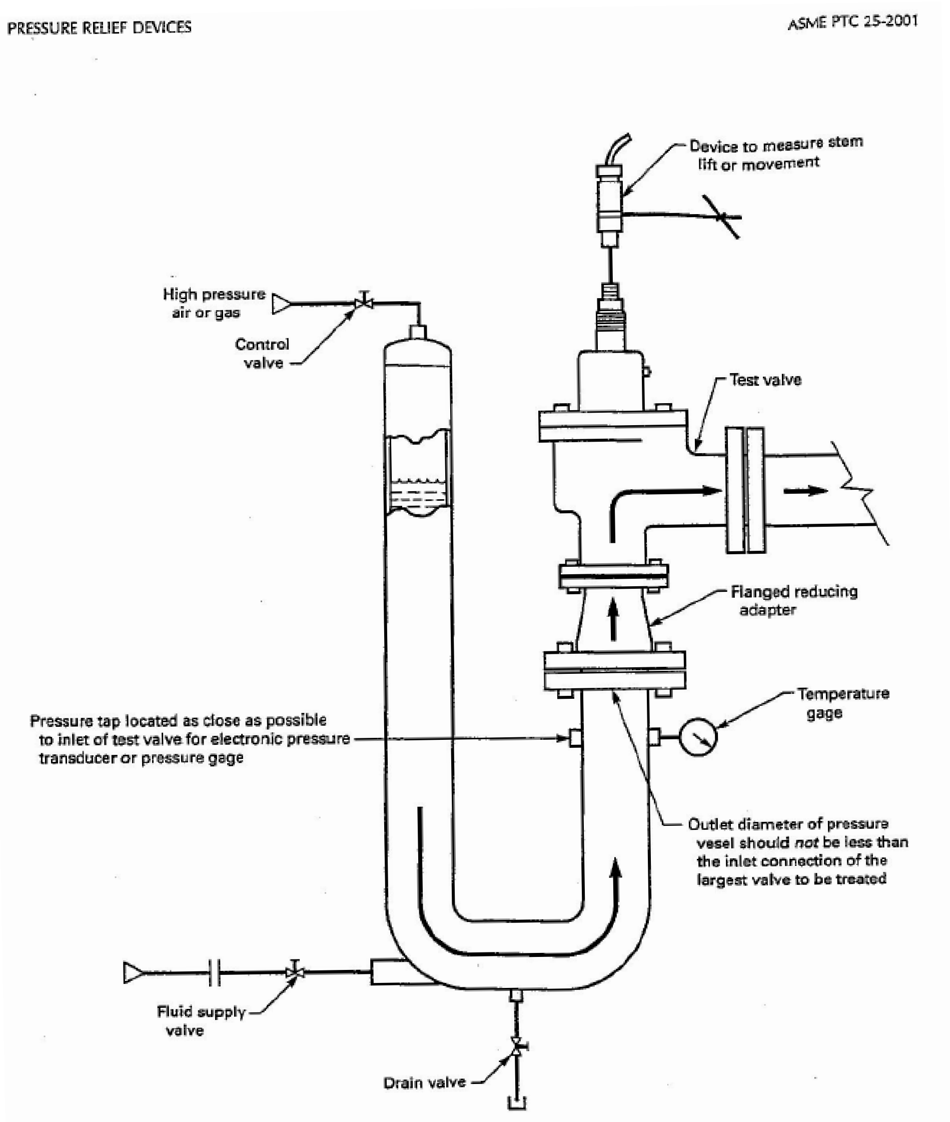


Fig. 2 Recommended Arrangements for Testing Valves with Incompressible Fluids

ASME PTC 25-2001 goes on to state that "Other adapter fittings may be used provided the accuracy of the test is not affected." This statement leaves Test Bench Manufacturers or builders some leeway when designing their units but, is something that users should be aware of when testing because the pressure drop may be significant enough to actually affect their results. Finally, ASME PTC 25-2001 and the NBIC recommend that when testing a PRV on a test bench the piping, adapters, and fittings leading up to the test valve need be taken into account in order to minimize the pressure drop as well as minimize the affect of flow turbulence on the final test results.

Recommended Volume Capacity

Test vessels, piping size, and the size of test port vary to a large degree when comparing one PRV test bench to another. For example DeLuca Test standard model test benches are offered with 56 Liter [2 cubic foot] stainless steel test vessel, 7.6 cm [3 inch] piping and test port. On the other hand another well known Test bench manufacturer's PRV test benches is offered standard with a 4 liter [less than .2 cubic feet], 1.3 cm [½ inch piping] and test port. These differences are significant because the size of the vessel, piping, and test port affects the volume that the test bench has the capability to act on the valve which can affect the results of the test. Successful test's can be conducted with lower volume test units by experienced technicians but may affect the results of the test specifically when conducting liquid service PRV test's and possibly damage the PRV seat by inducing chatter when testing with air.

When conducting a liquid service PRV test it is necessary to determine where the actual set pressure is by visually determining when you have achieved a first steady stream (see Fig. 3). If the test bench being utilized does not have sufficient volume it can be difficult for the operator to determine when they are actually achieving set pressure. For example in Figure 3 below it can be very difficult, on a low volume test bench, to differentiate between a "Pre-leak" where cohesion causes the water to curve back toward the PRV inlet (which is not the PRV opening pressure) and a first steady stream.

Note: It is necessary to have sufficient volume and flow path to accurately perform liquid service PRV Testing

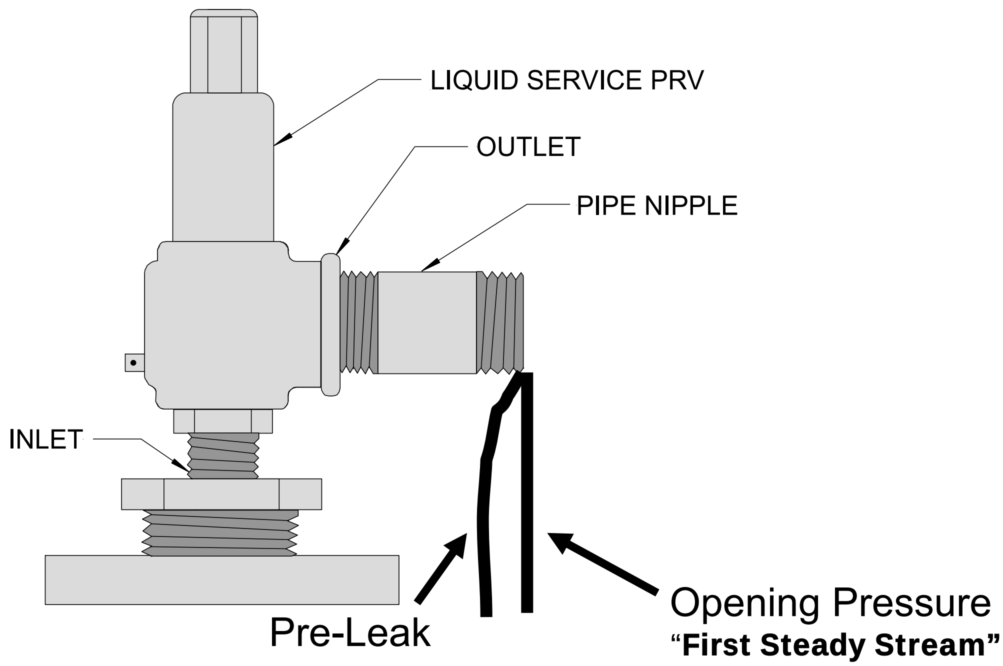


Fig. 3 Liquid Service PRV Testing

Rich Booth, with Vermont Yankee, has experience with both Low and High volume test benches and stated that "Accuracy and repeatability of new relief valve test equipment (High Volume) exceeds the capabilities of the old equipment (low volume test unit). A three (3) inch test bench port and accumulator now provides the volume to control relief valve lift at set point more accurately versus the previous reciprocating pump test bench and 3/8 inch tube. The low volume test bench could in some instances only "burp" the valve before dropping below the lift point. Determination of manufacturer's set points (i.e. first steady stream) was subjective, susceptible to inconsistencies and sometimes not achievable".

Gary Caudill with VC Summer is also a strong proponent of high volume testing and has seen improved results by using his high volume test bench (3 inch diameter piping along with a 4 cubic foot Stainless Steel test vessel) when compared to the previous method of testing with a low volume test bench.

PRV Seat Alignment

One of the primary issues with performance within a PRV design is alignment. Seat and disk alignment is critical for PRV performance. Valve manufacturers routinely employ radial surfaces called “bearing points” to allow for movement between the disc and the disc holder (commonly referred to as “disk rock”). Following a bench test or system excursion, this movement allows the disc to re-align with the nozzle upon reseat, forming a tight seal. Low Volume test benches do not generate the lifting forces required to permit the disc to realign upon closure and consequently results in a leaking PRV.

Alton Cox (NBIC Sub-Group Pressure Relief Devices) recently participated in a liquid test with an ASME Consultant for a “UV” Certificate Holder which highlights the importance of volume in regards to PRV seat alignment. Alton stated that they “had to gush a liquid PRV …. to get it to stop leaking. We had 31 drops per minute (allowable was 2.5 dpm) after three successful lifts. Then we gushed the valve (sent into overpressure) and the PRV had zero drops after that.”

Air Evacuation

This portion of the paper will address how air can be trapped in the nozzle between the disk and the water in the test bench when testing liquid service PRV’s. And then addresses a couple of methods used in the industry to evacuate the trapped air or “air bubble” in the system.

The trapped air beneath the seat of a PRV installed on the test bench skews the first “as found” test. The air caught in the nozzle is compressed during the initial test by the liquid. The compressed air or “air bubble” then creates a “burp” just ahead of the liquid which follows it out of the PRV. This has the effect of yielding something other than a steady stream which in turn affects your initial “as found” test (Pre conditions the valve).

There are a couple methods commonly used to evacuate the compressed air out of the PRV prior to initiating the test. One of these methods is to literally rotate the entire valve and clamping station 180 degree’s in order to allow the air to escape, sometimes referred to as pre-conditioning the valve. This method is effective but can be tedious and time consuming. Another method commonly used is to implement a test plate adapter with a bleed valve that is placed in between the test plate itself and the test valve. One then would pressurize the system with the bleed valve open until all of the air is evacuated. After all air is evacuated one would then close the bleed valve and initiate the test. DeLuca incorporates a standard air evacuation tube design developed by Gary Caudill at VC Summer in cooperation with DeLuca Test Equipment.

There are several methods to evacuate the air bubble when conducting liquid service PRV tests when bench testing. The important point to remember is that some sort of air evacuation method should be used when conducting liquid tests in order to conduct a valid as found test.

Conclusion

In conclusion The National Board Inspection Code along with The American Society of Mechanical Engineer’s Performance Test Code 25-2001 provide input and recommendations in regards to bench testing Pressure Relief Valves and gives a basic guide on test bench layout and characteristics. Although these recommendations provide a base to follow they can still be open to interpretation. These interpretations will be evident in the differences in both the design’s and capabilities of PRV test benches found throughout the industry. These differences in regards to the internal contours, fittings, and adapters along with a PRV test benches Volume Capacity and some sort of air evacuation method should be taken into account when bench testing PRV’s. Finally, PRV test bench users should be at least aware of these differences and the impact they could have on the accuracy of their set point results.

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Curt Dupill, Dupill Group / DeLuca Test Equipment
Dean Mathews, DeLuca Test Equipment

References

1. ASME Performance Test Codes 25-2001, Pressure Relief Devices
2. National Board Inspection Code, Part 3, Sections I and II, 2007

WHITE PAPER

DEVELOPING A PRESSURE RELIEF VALVE TEST SCHEDULE

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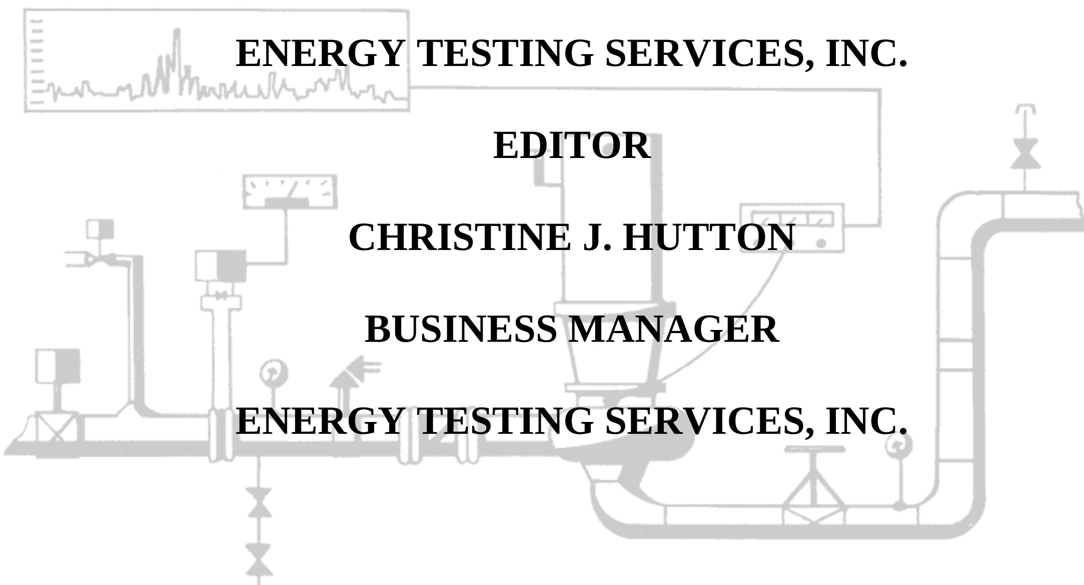

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DEVELOPING A PRESSURE RELIEF VALVE TEST SCHEDULE

DISCLAIMER

The following abstract narrative is the opinion of the author and should only be used in the context to which it applies. The narrative provides an overview of the Operations and Maintenance (OM) Code requirements for scheduling safety and relief valves tests. This paper continues the line of questioning presented in APPLYING THE OM CODE. Therefore, in order to properly explain the processes for scheduling tests, numerous references to the previous narrative may be required.

INTRODUCTION

The paper will provide Code requirements and bases to support recommended practices when developing a safety and relief valve test schedule. A review of the historical perspective within the ASME Code, Section XI (safety and relief valve test scheduling) will provide an understanding of previously accepted practices.

The current OM Code subparagraph to be covered is I-1320(a), *5-Year Test Interval,* which is very similar to I-1350(a), *10-Year Test Interval*. The header I-1320(a), *5-Year Test Interval,* pertains to the Mandatory Appendix I, Inservice Testing of Pressure Relief Devices. Therefore, even though inservice is not specifically stated in the header, the 5-year test interval is an inservice test interval assigned to Class 1, Pressure Relief Valves. The OM Code Subsection ISTA-3000, General Requirements, delineates test and examination program requirements for inservice test intervals. Each sentence of I-1320(a) will be covered with a method of implementation. The paper will provide a step-by-step approach for compliance with test scheduling for safety and relief valves. Step 1 will detail a process for specifying thermal relief valves; these valves are excluded from the I-1320(a) and I-1350(a) scheduling requirements. Step 2 provides a process for grouping safety and relief valves into a specific valve group. Valves not placed in a valve group would be excluded from the I-1320(a) and I-1350(a) scheduling requirements pertaining to groups. Using Steps 1 and 2, a glossary of scheduling terms would be recommended in order to consistently apply scheduling guidance. Then, using the material presented, an example of a I-1320(a) test schedule will be provided.

HISTORY

Historically, pressure relief valve scheduling can be separated into three phases. Phase 1 would involve the time frame between the 1970's through mid 1990's. Phase 2 would include the mid 1990's through the mid 2000's. Phase 3 includes the current time frame from the mid 2000's until the present.

Phase 1: Using the ASME Code, Section XI

The ASME Code, Section XI, Edition 1983 was the last Section XI to provide the testing requirements for the safety and relief valves:

"IWV-3510 SAFETY VALVE AND RELIEF VALVE TESTS

"IWV-3511, Test Frequency
Valves shall be tested at the end of each time period as defined in Table IWV-3510-1.

"IWV-3512, Test Procedure
Safety valve and relief valve set points shall be tested in accordance with ASME PTC 25.3-1976. Bench testing, with suitable hydraulic or pneumatic equipment, or testing in place with hydraulic or pneumatic assist equipment, is an acceptable method under PTC 25.3-1976. Valves so tested are not required to be additionally leak tested in accordance with IWV-3420.

"IWV-3513, Additional Tests
When any valve in a system fails to function properly during a regular test, additional valves in the system shall be tested as determined by an arbitrary assumption that a 12 month operating period has passed to another refueling, and the additional valves shall be tested to make the cumulative total tested at least N/60 X total valves in this category, where N now includes the additional 12 months (see Table IWV-3510-1 for definition of N). If any of these additional valves fails to function properly on test then all valves in the system in this category shall be tested.

"NOTE:
(e) N_1, N_2, N_3, etc., are the numbers of months from startup to first refueling, second refueling, third refueling, etc. When N is a number larger than 60 all valves which have not been tested during the preceding 5 year period shall be tested. The following period shall then be considered to be the same as startup to first refueling for purposes of determining test frequency, with the added requirement that at each refueling all valves which have not been tested during the preceding 5 year period shall be tested. The subsequent period will be considered the same as the first refueling to the second refueling, etc., with N determined by counting months from the new starting point."

The ASME Code, Section XI, established a 5-year test interval for all valves. The additional testing was system related and lacked guidance for additional tests. Also, the process of calculating the number to be tested was based on a refueling outage of 12 months.

Phase 2: Using the ASME/ANSI OM-1987 Part 1 (Standard)

The ASME/ANSI OM-1987 Part 1, initially used and provided the testing requirements for the safety and relief valves:

"1.3 Guiding Principles

"1.3.3 Test Frequency, Class 1 Pressure Relief Devices

"1.3.3.1 Pressure Relief Valves
(a) Initial 5 Year Period.
(b) Subsequent 5 Year Periods
All valves of each type and manufacture shall be tested within each subsequent 5 year period with a minimum of 20% of the valves tested within any 24 months. This 20% shall be previously untested valves, if they exist.
(c) Replacement With Pretested Valves
(d) Acceptance Criteria
(e) Valves Not Meeting Acceptance Criteria

"Table 1, Class 1: Pressure Relief Valve Testing Schedules, First 5 Year Period[1]

Time Period	Minimum Cumulative % of Valves of Each Type and Manufacture to be Tested
Startup – 12 Months	0
13 Months – 24 Months	25
25 Months – 36 Months	50
37 Months – 48 Months	75
49 Months – 60 Months	100
NOTE: (1) No maximum limit is specified for valve tests within any specific time period of the above table; however, a minimum of 20% of the valves of each type and manufacture shall be tested within any 24 months. This 20% shall be previously untested valves, if they exist.	

"1.3.4 Test Frequency, Class 2 and 3 Pressure Relief Devices

"1.3.4.1 Pressure Relief Valves
(a) Initial 10 Year Period.
(b) Subsequent 10 Year Periods
All valves of each type and manufacture shall be tested within each subsequent 10 year period with a minimum of 20% of the valves tested within any 48 months. This 20% of the valves tested within any 48 months. This 20% shall be previously untested valves, if they exist.
(c) Replacement With Pretested Valves
(d) Acceptance Criteria
(e) Valves Not Meeting Acceptance Criteria"

"Table 2, Class 2 and 3: Pressure Relief Valve Testing Schedules, First 10 Year Period[1]

Time Period	Minimum Cumulative % of Valves of Each Type and Manufacture to be Tested
Startup – 24 Months	0
25 Months – 48 Months	25
49 Months – 72 Months	50
73 Months – 96 Months	75
97 Months – 120 Months	100
NOTE: (1) No maximum limit is specified for valve tests within any specific time period of the above table; however, a minimum of 20% of the valves of each type and manufacture shall be tested within any 48 months. This 20% shall be previously untested valves, if they exist."	

The ASME/ANSI OM Standard Part 1 was implemented by ASME Code, Section XI, Subsection IWV. This Standard established the first 5-year and 10-year test intervals by Safety Class. The standard also created the initial and subsequent test interval. The unique position on additional testing was taken because the requirement only pertained to the initial test interval. It is important to note that any test failure required the valve to be repaired or replaced and the cause of the failure to be determined and corrected. The requirements for testing appear to be current with the ASME OM Code, Section IST, except that the initial and subsequent testing were grouped into test intervals. The wording for replacements has been consistent.

Phase 3: Using the ASME OM Code, Section IST, Mandatory Appendix I

This Appendix contains requirements to augment the rules of Subsection ISTC, Inservice Testing of Valves in Light-water Reactor Nuclear Power Plants. The OM Code is covered in detail by providing a narrative of the individual sentences.

ORGANIZATION OF THE ASME OM CODE

The FOREWORD discusses the formation of the ASME OM CODE.

The PREPARATION OF TECHNICAL INQUIRIES TO THE COMMITTEE ON OPERATION AND MAINTENANCE OF NUCLEAR POWER PLANTS discusses developmental business. This includes consideration of written requests for interpretations, Code Cases, revisions to the Operation and Maintenance Code, and development of new requirements.

The PREFACE includes:

"ORGANIZATION OF THIS CODE

"This Code is published with one Section, entitled Section IST, Rules for Inservice Testing of Light-Water Reactor Nuclear Power Plants. Section IST is divided into Subsections, mandatory appendices, and nonmandatory appendices. Provisions for adding future Sections exist.

Parts 1, 4, 6, 10 from ASME/ANSI OM-1987, Operation and Maintenance of Nuclear Power Plants, are incorporated into this Code as shown below:

OM Code Designation	Previous OM-1987 Designation	
Appendix I	Part 1	Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices
Subsection ISTD	Part 4	Examination and Performance Testing of Nuclear Power Plant Dynamic Restraints (Snubbers)
Subsection ISTB	Part 6	Inservice Testing of Pumps in Light-Water Reactor Power Plants
Subsection ISTC	Part 10	Inservice Testing of Valves in Light-Water Reactor Power Plants”

As presented in the previous paper, “Applying the OM Code,” a missing key ingredient of the OM Code was a cross-reference to Subsection IWA. The following should have been identified:

OM Code Designation	Previous B&PV Code Designation	
Subsection ISTA	Subsection IWA	General Requirements

Subsection IWA provided guidance. This included the Section’s use for development Subsection ISTA. Subsection IWA provided necessary terms in Article IWA-9000, Glossary (for example, “inservice life”, “pump”, “valve”, etc.), and necessary clarification in Article IWA-7000 Replacements.

The format of current ASME OM Code, Mandatory Appendix I, is consistent with that of other portions of the Code. An example:

Example: Mandatory Appendix I -1320(b)(1)

STRUCTURE	**ITEM**	**TITLE**
Section	None	None
Division	None	None
Subsection	None	None
Mandatory Appendix	I[1]	Inservice Testing of Pressure Relief Devices in Light-Water Reactor Nuclear Power Plants
N/A Note	[1]	This Appendix contains requirements to augment the rules of Subsection ISTC, Inservice Testing of Valves in Light-water Reactor Nuclear Power Plants
Article	1000	General Requirement
Subarticles	1300	Guiding Principles
Subsubarticle	1320	Test Frequencies, Class 1 Pressure Relief Valves
Paragraph	None	None
Subparagraph	1320(b)	Replacement With Pretested Valves
Subsubparagraph	1320(b)(1)	None

SUBPARAGRAPH

To properly implement the subparagraph of interest, each of its sentences should be evaluated against the Code's intent, as written. The subparagraph to be discussed is I-1320(a), *5-Year Test Interval*, which is very similar to I-1350(a), *10-Year Test Interval*.

"I-1320(a) 5-Year Test Interval. Class 1 pressure relief valves shall be tested at least once every 5 years, starting with initial electric power generation. No maximum limit is specified for the number of valves to be tested within each interval; however, a minimum of 20% of the valves from each valve group shall be tested within any 24-month interval. This 20% shall consist of valves that have not been tested during the current 5-year interval, if they exist. The test interval for any individual valve shall not exceed 5 years."

The header "I-1320(a) 5-Year Test Interval" pertains to Mandatory Appendix I, Inservice Testing of Pressure Relief Devices. Therefore, even though "inservice" is not specifically stated in the header, the 5-year test interval is an inservice test interval assigned to Class 1 Pressure Relief Valves.

It is important to note at this point that Subsection ISTA-3000, General Requirements, delineates test and examination program test intervals.

"ISTA-3120 Inservice Test Interval

- The frequency for inservice testing shall be in accordance with the requirements of Section IST.
- The inservice test interval shall be determined by calendar years following placement of the unit into commercial service.
- The inservice test intervals shall comply with the following, except as modified by
- ISTA-3120(d) and ISTA-3120(e):

(1) Initial Test Interval: 10 years following initial start of unit commercial service; and
(2) Successive Test Intervals: 10 years following the previous test interval.

- Each of the inservice test intervals may be extended or decreased by as much as 1 year. Adjustments shall not cause successive intervals to be altered by more than 1 year from the original pattern of intervals.
- In addition to ISTA-3120(d), for units that are out of service continuously for 6 months or more, the inservice test interval during which the outage occurred may be extended for a period equivalent to the outage and the original pattern of intervals extended accordingly for successive intervals.
- The inservice test intervals for component replacements, additions, and alterations that may be required during the service lifetime of the unit shall coincide with the remaining intervals, as determined by the calendar years of unit service at the time of replacement, addition, or alteration."

First Sentence:

"Class 1 pressure relief valves shall be tested at least once every 5 years, starting with initial electric power generation."

The first sentence fits right into the intent of Subsection ISTA-3120(b):

"The inservice test interval shall be determined by calendar years following placement of the unit into commercial service."

Therefore, this first sentence is referring to an inservice test interval and it provides a start date. Since operation of plants beyond the commercial service date, numerous plants have experienced long duration shutdowns and other extended outages. The inservice test interval therefore should coincide with a known commodity, and the date for performing the Inservice Testing of Pumps and Valves Program upgrade for compliance with {10CFR50.55(a)(b)} is known and fixed by most plants. This date should become the interval start date in lieu of the placement of the unit into commercial service/initial electric power generation. The inservice test interval would be determined by adding a 10-year increment to the Inservice Testing of Pumps and Valves Program upgrade date.

<u>Intent</u>: This sentence establishes a fixed interval for inservice testing. It has a 5-year duration with an unique start date and end date. Therefore, when the inservice test interval (10-year) is established, the 5-year interval can be determined by dividing the inservice test interval by 2, creating two 5-year intervals.

Second Sentence:

"No maximum limit is specified for the number of valves to be tested within each interval; however, a minimum of 20% of the valves from each valve group shall be tested within any 24-month interval."

The second sentence can be separated into two parts:

Part 1 - No maximum limit is specified for the number of valves to be tested within each interval, which is still referring to the 5-year interval.

<u>Intent</u>: A valve may be tested more than once in a 5-year interval.

Part 2 – A minimum of 20% of the valves from each valve group shall be tested within any 24-month interval. Therefore, we know when a 5-year interval starts, because it's the date the Inservice Testing Of Pumps and Valves Program upgrade compliance with {10CFR50.55(a)(b)}, and it should be easily separated from the 5-year interval into 24 month intervals. It is important to note the sentence is referring to a specific
5-year interval.

<u>Intent</u>: At least 20% of the valves in each valve group must be tested in the known fixed 24-month interval. This implies that more than one 24-month interval exists in a 5-year test interval (two 24-month intervals). It also implies multiple valves in a valve group.

Third Sentence:

"This 20% shall consist of valves that have not been tested during the current 5-year interval, if they exist."

This sentence again discusses the current 5-year interval, for which we have a starting and ending point, as discussed above. This sentence refers back to the previous sentence and emphasizes that this discussion pertains to the 5-year interval only.

<u>Intent</u>: A review of tested valves in a group is to be made and if a valve has not been tested within the current 5-year interval, it has priority on testing within the 24-month interval. Additionally, the sentence recognizes that all the valves of a group may have been already tested (expanding the requirement's scope) and will need to be tested again.

Fourth Sentence:

"The test interval for any individual valve shall not exceed 5 years."

This sentence gets specific to a valve and not the 5-year test interval, and specifies, that each valve must have been tested within 5 years. A 5-year test interval may be extended as allowed by Subsection ISTA:

"ISTA-3120 Inservice Test Interval

"Each of the inservice test intervals may be extended or decreased by as much as 1 year. Adjustments shall not cause successive intervals to be altered by more than 1 year from the original pattern of intervals.
In addition to ISTA-3120(d), for units that are out of service continuously for 6 months or more, the inservice test interval during which the outage occurred may be extended for a period equivalent to the outage and the original pattern of intervals extended accordingly for successive intervals."

Intent: The testing of the individual valve must be within 5 years of the previous periodic test. The periodic testing of valves is pertinent to inservice valves. By definition certified spares have been manufactured or sent out for maintenance or repair. The as-found testing of a certified spare valve does not exist.

"I-3300 Periodic Testing

"Periodic testing of all pressure relief devices is required. No maintenance, adjustment, disassembly, or other activity that could affect 'as found' set-pressure or seat tightness data is permitted prior to testing. Control ring adjustment is permitted per I-4110(g) and I-4120(g). Test frequencies are specified in I-1320, I1330, I-1340, I1350, and I1360. When on-line testing is performed to satisfy periodic testing requirements, visual examination may be performed out of sequence."

BUILDING A TEST SCHEDULE

When determining a schedule for inservice testing of pressure relief devices, the use of the following subsubarticles would be mandatory to develop individual test frequencies by Class and Type:

- Subsubarticle ISTA-1320, Test Frequencies, Class 1, Pressure Relief Valves
- [I-1320(a) 5-Year Test Interval];
- Subsubarticle ISTA-1330, Test Frequency, Class 1, Nonreclosing Pressure Relief Devices;
- Subsubarticle ISTA-1340, Test Frequency, Class 1 Pressure Relief Valves That Are Used for Thermal Relief Application;
- Subsubarticle ISTA-1350, Test Frequency, Class 2 and 3 Pressure Relief Valves
- [I-1350(a) 10-Year Test Interval];
- Subsubarticle ISTA-1360, Test Frequency, Class 2 and 3 Nonreclosing Pressure Relief Devices;
- Subsubarticle ISTA-1370, Test Frequency, Class 2 and 3 Primary Containment Vacuum Relief Valves;
- Subsubarticle ISTA-1380, Test Frequency, Class 2 and 3 Vacuum Relief Valves, Except for Primary Containment Vacuum Relief Valves; and
- Subsubarticle ISTA-1390, Test Frequency, Class 2 and 3 Pressure Relief Devices That Are Used for Thermal Relief Application.

Again, this is an inservice testing Code, therefore the device (including the valve) being tested would be specific to which device was installed in a specific plant location, not a serial number or other trait.

Determine the current IST Pumps and Valve Program 10-year interval start and end date. This will identify the Inservice Test Interval [ISTA-3120] start and end date. This start date coincides with the {10CFR50.55(a)(f)(4)(ii)} date plus 12 months.

“10CFR50.55(a)(f)(4)(ii) Inservice tests to verify operational readiness of pumps and valves, whose function is required for safety, conducted during successive 120-month intervals must comply with the requirements in the latest edition and addenda of the Code incorporated by reference in paragraph (b) of this section 12 months before the start of the 120-month (or the optional ASME Code cases listed in NRC Regulatory Guide 1.147, through Revision 14, or 1.192 that is incorporated by reference in paragraph (b) of this section), subject to the limitations and modifications listed in paragraph (b) of this section.”

THERMAL RELIEF VALVE SELECTION PROCESS

PURPOSE

To develop a practice for selecting pressure relief valves to use for thermal relief application only. Pressure relief valves with a thermal relief function only shall be classified as thermal relief and then removed from the valve grouping process. This unique classification of thermal relief will be used within the SRV Program and IST Program documents.

RECOMMENDATION

It is recommended that the following guidance be used to determine which pressure relief valves are to be classified for thermal relief application only, thus satisfying the OM Code Mandatory Appendix I requirements. Using the criteria specified within ASME Code (Addenda 2003) I-1200 Definitions, this valve shall only have one function of overpressure protection that is from thermally induced changes. This function is to protect against overpressurization from fluid expansion caused by changes in fluid temperature. Therefore pressure relief valves used to protect isolated components, systems, or portions of systems from overpressurization sources or in-leakage of high energy fluids would be beyond the scope of thermal pressure relief classification.

DEFINITIONS

“Thermal Relief Application: a relief device whose only overpressure protection function is to protect isolated components, systems, or portions of systems from fluid expansion caused by changes in fluid temperature

“Valve Group: valves of the same manufacturer, type, system application, and service media.”

DETAILED STEPS

Step 1. Locate all pressure relief valves within the scope of the IST plan for the site.

Step 2. Determine what relief protection function(s) are associated with the valve as it pertains to its location within the system or portion of a system.

Step 3. If the pressure relief valve serves only to protect the isolated component, system, or portion of a system from fluid expansion caused by changes in fluid temperature, the valve is to be classified as a thermal pressure relief.

Step 4. Once all the thermal pressure relief valves have been identified, remove them from the valve grouping process. A unique classification, “thermal relief,” can be used, if desired. This classification as a thermal relief device would also be used to perform failure analysis for frequency of replacement.

RELIEF VALVE GROUPING PROCESS

PURPOSE

To develop a practice for grouping pressure relief devices used to satisfy the ASME OM Code Mandatory Appendix I test-frequency requirements.

RECOMMENDATION

It is recommended that the following guidelines be used to develop a standard grouping process for relief valves. The most important criteria when creating relief valve groups is that a group should ideally contain two to three valves. Grouping by definition requires valves of the same manufacturer, type, system application, and service media to be identified as a valve group.

DEFINITIONS

“Valve Group – valves of the same manufacturer, type, system application, and service media {ASME OM Code, Mandatory Appendix I-1200, Definitions}.

Thermal relief application - a relief device whose only overpressure protection function is to protect isolated components, systems, or portions of systems from fluid expansion caused by changes in fluid temperature {ASME OM Code, Mandatory Appendix I-1200, Definitions}.”

Types Of Reclosing Pressure Relief Devices - (a) Pressure Relief Valve, (b) Safety Valve, (c) Relief Valve, (d) Safety Relief Valve, (e) Pilot-Operated Pressure Relief Valve, (f) Power-Actuated Pressure Relief Valve, (g) Temperature-Actuated Pressure Relief Valve, and (h) Vacuum Relief Valve {ASME/ANSI PTC 25 Section 2, Definitions and Description of Terms, Section 2.3, Types of Devices}.

DETAILED STEPS

Step 1. Locate and list all pressure relief devices within the scope of the IST plan for the specific unit.

Step 2. Determine what relief protection function(s) are associated with the device as it pertains to its location within the system or portion of a system.

Step 3. If the pressure relief device serves only to protect the isolated component, system, or portion of a system from fluid expansion caused by changes in fluid temperature, the device is categorized as a thermal pressure relief and eliminated from the valve grouping process (see the THERMAL RELIEF VALVE SELECTION PROCESS section earlier in this paper).

Step 4. If the pressure relief device serves any other functions than just protecting from fluid expansion caused by changes in fluid temperature, the valve shall be placed in a unique valve group.

Step 5. Identify and record grouping information for each valve: manufacturer, type, service medium, and system application.

Step 5a. Determine the manufacturer as follows: ANSI/ASME OM Part 1, Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices, I-5100 BWR Records and Record Keeping Requirements & I-9100 PWR Records and Record Keeping requires that the Owner shall maintain a record that includes the following for each valve covered by this Appendix: (a) the manufacturer and manufacturer's model and serial number, or other identifiers and (b) a copy or summary of the manufacturer's acceptance test report, if available. Therefore the manufacturer is a known term specified in the OM Code and this is a single item not to be separated into anything more than manufacturer.

Step 5b. Determine the type as follows: ANSI/ASME Performance Test Codes (PTC) 25, Pressure Relief Devices -1994, Section 2 – Definitions And Description Of Terms, 2.3 Types Of Devices, 2.3.1 Reclosing Pressure Relief Devices, *(a) Pressure Relief Valve, (b) Safety Valve*, *(c) Relief Valve*, *(d) Safety Relief Valve*, *(e) Pilot-Operated Pressure Relief Valve*, *(f) Power-Actuated Pressure Relief Valve*, *(g) Temperature-Actuated Pressure Relief Valve*, and *(h) Vacuum Relief Valve*. Therefore the type is a known term from the PTC Standard, and this is a single item not to be separated into anything other than one of the above types of valves.

Step 5c. Determine the service medium as follows: ANSI/ASME OM Part 1, Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices, I-4110 Steam Service, I-4120 Compressible Fluid Service Other Than Steam, I-4130 Liquid Service, I-8110 Steam Service, I-8120 Compressible Fluid Service Other Than Steam, and I-8130 Liquid Service requires the test medium be identified as the normal system operating fluid. Examples include: steam, compressible fluid (for example, nitrogen, air, or hydrogen) and liquids (such as water or oil). Therefore the service medium is a known term specified in the OM Code, and this is a single item not to be separated into anything other than the normal system operating fluid.

Step 5d. Determine the system application as follows: This term is the most useful in separating valves into valve groups. System application allows the owner to classify valves as he desires. Examples could include: valve actuation set-pressure (high or low), location (environment), orientation (vertical/horizontal), backpressure compensation, tail piece, service medium impurities (chemicals, raw water, borated water and etc.), and others. Therefore the system application is not a specified term and the owner may use this term to enhance the grouping process as desired.

Step 6. Place each pressure relief valves, except thermal relief valves, in a unique valve group.

Finally, if only one valve remains this valve des not fit the criterion of a valve group and is tested as a stand alone component (that is, once every 5 or 10 years as applicable).

See the next section, REPLACEMENT TEST-TO-TEST, which actually develops a test schedule. It is important to note that replacements have always been addressed by Section XI (Article IWA-7000, Replacement) as a program. Replacement scope provides the rules and requirements for the specification and construction of items to be used for replacement. Replacement includes the addition of components, such as valves, and system changes, such as rerouting of piping, within the scope of the Division. The reasons for replacement may include;

a) "discrepancies detected during inservice inspection
b) regulatory requirements change
c) design changes to improve equipment service
d) changes to improve reliability
e) damage
f) failure during service
g) personnel exposure
h) economics
i) end of service life
j) discrepancies detected during maintenance"

This program (IWA-7130, Replacement Program) would include:

a) "the applicable Edition and Addenda of Section XI;
b) a description of the items being replaced and the Codes and Code Cases to which they were constructed;
c) a description of the work to be performed;
d) the Code Edition, Addenda, and Code Cases applicable to materials, design, manufacture, and installation.
e) any special requirements pertaining to materials, welding, heat treatment, and nondestructive examination requirements;
f) the test and acceptance criteria to be used to verify the acceptability of the replacement;
g) the documentation required by IWA-7500;
h) the application of the ASME Code Symbol Stamp in accordance with IWA-7330."

The definition of inservice life is provided in the Article IWA-9000 Glossary

" *Inservice life* – the period of time from the initial use of a item until its retirement from service."

REPLACEMENT TEST-TO-TEST

The attempt to apply a test interval requirement on a valve using the valve's service life in lieu of the proper inservice life is outside of a realistic application of the Code of Record, as it pertains to testing of pumps and valves. Service life testing is addressed in Subsection ISTD, Preservice and Inservice Examination and Testing of Dynamic Restraints (Snubbers) in Light-Water Reactor Nuclear Power Plants.

- Subarticle ISTD-6500, Testing for Service Life Monitoring Purposes.
- NONMANDATORY APPENDIX F, Dynamic Restraints (Snubbers) Service Life Monitoring Methods

It is important to note that inservice testing of Dynamic Restraints is also addressed in:

- Subarticle ISTD-5200, Inservice Operational Readiness Testing

The use of the term "service life" is specifically addressed in two other portions of the ASME OM Code:

- Subarticle ISTA-1500, Owner's Responsibilities, Subparagraph ISTA-1500(j) (requires the retention of all test and examination records for the service lifetime of the component or system)
- Subsubarticle ISTC-5260, Explosively Actuated Valves

To assess the impact of placing the last test date on replacement valves, not the replacement installation requirements are addressed below. This applies to the inservice periodic testing intervals of either the Subsubarticle ISTA-1320 Test Frequencies, Class 1 Pressure Relief Valves [I-1320(a) 5-Year Test Interval] or Subsubarticle ISTA-1350 Test Frequency, Class 2 and 3 Pressure Relief Valves [I-1350(a) 10-Year Test Interval].

Plausible (Test-to-Test Interval) Scenario

Initial Conditions: A single-unit PWR has three (3) installed Class 1 Pressurizer Safety Relief Valves to be removed and will undergo a full complement replacement with certified spares of the same type and manufacturer. A manufacturer/approved testing facility recertified (set-pressure tested) these replacement valves six (6) months prior to the start (January 2007) of the new 10-year testing interval. The Owner recently obtained approval of their fourth ten year Pump and Valve Inservice Test Program.

Inservice Event Timeline For Scheduling Tests During The First 24 Months Of The First Five Year Test Interval: The Owner determines an acceptable strategy for minimizing the upcoming refueling outage duration would be complying with I-1320 Test Frequency and scheduling a full complement replacement. The Inservice
Class 1 Pressurizer Safety Relief Valves are to be removed from the plant during the refueling outage, scheduled four months (September 2007) into the interval and replaced with recertified spares. The removed (replaced) Class 1 Pressurizer Safety Relief Valves are to be expedited to the manufacturer/approved testing facility and tested within three (3) months (December 2007). These replaced valves then will undergo refurbishment over the next six (6) months receiving recertification by the manufacturer/approved test facility in June 2008. The valves are shipped to the site where the Owner places the certified spare valve in the warehouse following the acceptable receipt visual examination and verification of the manufacturer's/vendor's Certification-of-Compliance test documents. The following unique designators and dates are assigned to the Class 1 Pressurizer Safety Relief Valves as follows: initial installed (inservice) valves designated Group A; replacement (certified spare) valves designated Group B; and the replaced/refurbished (recertified) valves designated Group AA.

Valves Group	Inservice Date	Service Date	Removed From Service Date
A	Before June 2007	Before June 2007	Sept 2007
B	Sept 2007	Jan 2007	N/A
AA	N/A	Jun 2008	N/A

The implementation of the ASME OM Code, Mandatory Appendix I, requires the Owner to satisfy specific valve test dates to address test interval frequencies. These compliance dates will vary when applying the installation date or recertification test date.

Each valve (serial number) would be assigned an unique title during different periods of its service life for compliance with the ASME OM Code requirements, such as:

(1) Inservice Valve: describes a valve that is installed in the plant at a valve location which experiences normal system operating conditions (fluid, pressure, temperature) that may affect one or more of the operating characteristics over time.
(2) Purchased Valve: describes a valve procured, receipt inspected, and accepted under the Owner's Quality Assurance Program. Upon acceptance by the Owners, this valve would be designated as a Certified Spare Valve.
(3) Pretested Valve: describes a valve satisfactorily tested but not installed in the plant at a valve location and considered a certified spare.
(4) Refurbished Valve: describes a valve that has undergone modification, rework, repair, or routine servicing which requires recertification to the manufacturer's specifications.
(5) Certified Spare Valve: describes a valve that has been either purchased or refurbished having manufacturer's/vendor's Certification-of-Compliance test documents and available at the plant for use.
(6) Replaced Valve: describes a valve removed from operation, replaced by a pretested valve, and no longer considered an inservice valve.
(7) Retired Valve: describes a valve removed to be scrapped / trashed.
(8) Untested Valve: describes an inservice valve, in a Valve Group, which has not been tested within the existing 5-year test interval.
(9) Additional Valve: describes an untested inservice valve, if it exists, within the valve group requiring testing because of an as-found testing failure during valve group expansion.

The OM Code, Mandatory Appendix I, establishes the following test-frequency requirements for Code compliance follows:

Five (5) Year Test Interval: The OM Code, Mandatory Appendix I, I-1320 Test Frequencies, Class 1 Pressure Relief Valves, states that valves in each Valve Group shall be tested at least once every 5 years, starting with the assigned date (mo/yr) in which the current IST Program must satisfy 10CRF50.55(a)(f). Additionally, the OM Code states that the PWR Main Steam Safety Valve shall be tested at the same frequency as the Class 1 Pressure Relief Valves (at least once every 5 years).

20% Selected Amount: Established within the OM Code, Mandatory Appendix I, Inservice Testing of Pressure Relief Devices in Light-Water Reactor Nuclear Power Plant it and states that a minimum testing criterion for each interval must be satisfied.

Five (5) Year 20% Selected Amount: Established that a minimum of 20% of the inservice valves from each Valve Group shall be tested within any 24-month interval; where possible, these shall consist of untested valves.

Five (5) Year: The OM Code, Mandatory Appendix I, I-1320 Test Frequencies, Class 1 Pressure Relief Valves, states that the test interval for any individual valve shall not exceed 5 years from the last tested date. Additionally, the PWR Main Steam Safety Valve shall be tested at the same frequency as the Safety Class 1 Pressure Relief Valves (at least once every 5 years).

100% Selected Amount: Established within the OM Code Mandatory Appendix I, Inservice Testing of Pressure Relief Devices in Light-Water Reactor Nuclear Power Plants; it states that no maximum testing criterion exists during plant outages. No maximum limit is specified for the number of valves to be tested within the Five (5) Year Test Interval, with the exception that an additional test failure of a valve among the 20% Selected Amount requires the remaining untested inservice valve(s) of a valve group to be tested.

Using the valve designators (A, B, and AA), both the inservice interval and test-to-test interval Code compliance replacement requirements can be identified in order to show the hardships and financial burdens being placed on the Owner:

Compliance Date For Group B Valves:

Group B Valves - a Manufacturer's Test Date of January 2007 and an Installation Date of September 2007.

Compliance Requirement	Compliance Date
Installation Date	9/2007
Test Interval Start Date	6/2007
Test Interval End Date	6/2017
1st Five (5) Year Test Interval Start Date	6/2007
1st /1st Five (5) Year 20% Selected Amount	6/2009
2nd /1st Five (5) Year 20% Selected Amount	6/2011
1st Five (5) Year Test Interval End Date	6/2012
1st Five (5) Year Test Date	9/2012 or 1/2012
2nd Five (5) Year Test Interval Start Date	6/2012
1st /2nd Five (5) Year 20% Selected Amount	6/2014
2nd /2nd Five (5) Year 20% Selected Amount	6/2016
2nd Five (5) Year Test Interval End Date	6/2017
2nd Five (5) Year Test Date	9/2017 or 1/2017
Manufacturer Tested Date	1/2007

The following would apply if two refueling outages (for example, 9/2007 and 3/2009) are planned within the first 5 Year Test Interval. When scheduling the first refueling outage, testing the following would apply:

Installation Test Date 9/2007 Planned Full Complement Replacement 9/2007

Number of Valves	1st Five (5) Year Test Interval Selected Amount	1st /1st Five (5) Year 20% Selected Amount	2nd /1st Five (5) Year 20% Selected Amount	1st Five (5) Year Test Date Selected Amount
3	3	3	N/A	3

Manufacturer Test Date 1/2007 Planned Full Complement Replacement 9/2007

Number of Valves	1st Five (5) Year Test Interval Selected Amount	1st /1st Five (5) Year 20% Selected Amount	2nd /1st Five (5) Year 20% Selected Amount	1st Five (5) Year Test Date Selected Amount
3	0	0	N/A	0

Results:

1st Five (5) Year Test Interval Selected Amount – is satisfied using the installation test date but not met with the manufacturer test date. Therefore all certified spares would have to be retested prior to installation within the 5-year time frame (the 1st Five (5) Year Test Interval) for establishing a newer manufacturer test date.

1st /1st Five (5) Year 20% Selected Amount – is satisfied using the installation test date by any of the valves but not met any valve using the manufacturer test date. Therefore one of the certified spares would have to be retested prior to installation within the first 24 months when using the manufacturer test date.

2nd /1st Five (5) Year 20% Selected Amount – is not required to be satisfied until the next scheduled refueling outage. Both the installation test date and manufacturer test date would require a valve to be replaced/tested. Replacement with a certified spare would satisfy the requirement using the installation date but would not meet the requirement using the manufacturer test date. Therefore one certified spare would have to be retested prior to installation within the second 24 months when using the manufacturer test date.

1st Five (5) Year Test Date Selected Amount – is satisfied using the installation test date for each valve but not met for any valve using the manufacturer test date, because the manufacturer test date would expire during the interval. Therefore each valve would have to be retested prior to installation or removed from service and tested.

The result of using the installation date is that the minimum number of pressure relief valve tests would be required but would still satisfy test frequency requirements providing an acceptable level of safety and quality. When using the manufacturer/approved vendor test date, an additional 4 up to 6 tests would be required for valves experiencing no normal system operating conditions (fluid, pressure, temperature). These additional tests would be performed on Owner designated certified spares.

CONCLUSION

The ASME OM Committee should look into the current interpretations presented in this paper. The user who has to implement the Code must follow the Code as written. When sources other than the Code must be applied, satisfying newer trends, a Committee evaluation should be performed. This evaluation is to ensure the Code is not being Added to or Revised. The implementer may go beyond the Code requirements, but in no case should he interpret or Change or Cause Additions to the Code's intent. Finally, a user should not have to locate and incorporate external guidance for compliance with 10CFR50.55a(f) requirements (i.e., applied Code as written and accepted in the Code of Federal Regulation).

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The 2008 Symposium on Valves, Pumps, Snubbers, 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 the inservice testing programs at nuclear power plants, including design, operation, and testing of valves, pumps, and snubbers 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, pumps, and snubbers at nuclear power plants.

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