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

Held at
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Washington, DC
July 17-20, 2000

Jointly Sponsored by:
U.S. Nuclear Regulatory Commission and
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Abstract

The 2000 Symposium on Valve and Pump 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 and regulatory issues associated with the testing of valves and pumps used in nuclear power plants. The symposium provides an opportunity to discuss the need to improve that testing in order to help ensure the reliable performance of valves and pumps. The participation of industry representatives, regulatory, and consultants ensures the discussion of a broad spectrum of ideas and perspectives regarding the improvement of testing programs and methods at nuclear power plants.

Steering Committee

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Acknowledgments

The Steering Committee, the American Society of Mechanical Engineers, and the U.S. Nuclear Regulatory Commission acknowledge the efforts of the Session Chairs, authors, and panel members for their invaluable contribution to the success of the symposium. We also would like to express our gratitude to Commissioner Greta Joy Dicus, Robert E. Nickell, John Ferguson, John W. Craig, and John Groth for their remarks at the opening session. Participation by international presenters and attendees provide a broad perspective to the issues currently under consideration in the United States. We sincerely appreciate the excellent work of Ms. Linda McKenzie and other members of the publications and graphics staff of the U.S. Nuclear Regulatory Commission in preparing the proceedings for the symposium. Finally, gratitude is expressed to all attendees and their sponsoring organizations for a successful and informative symposium.

Disclaimer and Editorial Comment

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

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

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Idaho National Engineering and Environmental Laboratory

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John E. Allen
ASME
Symposium Co-Chair

Eugene Imbro
U.S. Nuclear Regulatory Commission
Symposium Co-Chair

Pumps, Dumps, and Political Humps

Commissioner Greta Joy Dicus
U.S. Nuclear Regulatory Commission

The changing environment at the U.S. Nuclear Regulatory Commission (NRC) continues to influence everything we do. The NRC has become more open and more willing to move away from our more traditional approaches to regulation and instead of asking "Why?", the NRC is asking itself "Why not?" Of course, public health and safety remains our guiding mission but we realize that there may better, more efficient, and more effective ways of helping to ensure safety.

One need only look at some of the recent Commission actions relating to inservice testing of pumps and valves, inservice inspection of piping, and consensus standards to see that the Commission and staff are, at the least, committed to new thinking. And while sometimes our efforts toward regulatory reform are aided by, or the result of, more direct political pressure, political pressure in other areas can slow down some of the regulatory process such as licensing of a high-level waste repository.

The ASME/NRC Working Relationship for the Benefit of the Nuclear Industry

Robert E. Nickell
President, ASME International

ASME International is a standards development organization (SDO) responsible for nuclear codes and standards important to the economic viability of commercial nuclear power plants and ancillary equipment, and for the related health and safety of the general public. The U.S. Nuclear Regulatory Commission (NRC) is the federal regulatory agency responsible to ensure adequate protection of public health and safety from the use of nuclear materials, including commercial power production and the transport, storage, and disposal of nuclear materials and waste. The relationship between the two organizations has been active and beneficial for almost 30 years, since the creation of the NRC in 1974.

More recently, as the result of Public Law 104–113, the National Technology and Transfer Act of 1995, and as the result of the revision of the Office of Management and Budget (OMB) Circular A–119, the working relationship has become somewhat more formalized. PL 104–113 requires federal agencies to participate in the development of appropriate standards, such as the ASME Boiler and Pressure Vessel Code, Sections III and XI, and to use those standards as a part of the regulatory process. The OMB circular provides guidance in both regards.

NRC staff participation in the ASME codes and standards technical and consensus bodies has been, and continues to be, extensive, even in this era of resource constraints. And even though the commercial nuclear power business has matured in the past 30 years, codes and standards activity has diminished only slightly and, in some cases, has actually increased. Two such cases come to mind—first, standards activity related to risk-informed regulation, and second, modifications of existing standards to reflect the extended licensing period of operating reactors from 40 to 60 years. Both of these activities are of extreme economic importance to the continued viability of the commercial nuclear power option to U.S. utilities.

These issues are explored in some detail in the paper, with the conclusion that, while the current working relationship is valuable, an improved format could be even more beneficial to the nuclear power industry. It is proposed that a steering committee, composed of the Vice President, ASME Board on Nuclear Codes and Standards, and the ASME Staff Director, Nuclear Codes and Standards, supplemented by other appropriate ASME nuclear codes and standards volunteers, meet with the NRC Standards Executive, and other appropriate members of NRC staff, to address technical issues relevant to ongoing activity. Representatives of the Nuclear Energy Institute (NEI) and EPRI may wish to have observer status at these technical discussions. These technical discussions would be intended to supplement the current administrative meetings between ASME and NRC. Implementation of this proposal would be anticipated to improve the existing working relationship and would provide even more benefits to the nuclear power industry as a whole.

Future Trends in Nuclear Codes and Standards

John Ferguson
ASME Vice President, Nuclear Codes and Standards

The purpose of this presentation is to provide information on the changing nuclear power industry and on the plans of the American Society of Mechanical Engineers (ASME) for meeting these anticipated changes. The presentation provides a review of the forces impacting the nuclear industry and how these forces impact the use and development of ASME codes and standards for the nuclear industry.

The industry is rapidly changing due to cost pressures, industry reorganization and continuing public pressure from some parts of society to shutdown of the facilities. In this rapidly changing environment, it is critical to anticipate the nuclear power industry needs, provide appropriate codes and standards to meet these needs and provide historical needs for providing public safety.

This presentation identifies the goals of the ASME Board on Nuclear Codes and Standards focused in the three areas of purpose, people and process. These are the critical ingredients needed to deliver relevant codes and standards that incorporate the best operating experience and technological advances.

The Changing Regulatory Environment and Its Impact on Codes and Standards

John W. Craig, Assistant for Operations
Office of the Executive Director for Operations
U.S. Nuclear Regulatory Commission

The U.S. Nuclear Regulatory Commission (NRC) and nuclear regulatory environment have undergone significant changes in the past few years and more changes are expected in the future. In this presentation, some of those changes (e.g., the new reactor oversight process, risk-informed licensing actions, revisions to NRC regulations, and the role of voluntary industry initiatives) are discussed. The presentation also discusses the driving forces for these changes and their implication for codes and standards activities. Of particular interest is the NRC's new performance-based budgeting process that aligns resources with activities that contribute the most to achieving the agency's strategic goals of maintaining safety, reducing unnecessary regulatory burden, increasing efficiency and effectiveness, and improving public confidence. The presentation suggests cooperative actions between the NRC, the nuclear industry, and codes and standards bodies that can help to optimize codes and standards and regulatory activities. Challenges for the NRC and ASME in light of the changing regulatory environment, such as limited resources and the need for timely review of codes and standards, are also discussed. In conclusion, the challenge posed to the consensus standards community in this new regulatory environment is to develop and revise codes, standards, and guides to be endorsed by NRC, and implemented by the industry, that meet the strategic needs of all stakeholders.

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.

Reactor Safety in a Risk-Informed, Competitive Environment—the Human Element

John Groth, Chairman
ASME Committee on Operation and Maintenance of Nuclear Power Plants

Session 1(A)

Safety-Relief Valves/Check Valves

Session Chair
Chris Hansen
Vermont Yankee Nuclear Power Corporation

Success Revisited—Solving Performance Problems Using Nozzlecheck Valves

Gregg Joss—Rochester Gas & Electric, Ginna Station, Inservice Test Coordinator
Jim Zulawski—Rochester Gas & Electric, Ginna Station, Performance Monitoring Supervisor

Abstract

For 24 years, Rochester Gas & Electric's Ginna Nuclear Power Plant had conventional swing check valves installed in its Component Cooling Water (CCW) and Service Water (SW) systems which resulted in persistent system problems when valve disk "slamming" occurred during valve cycling. These check valves are located at the discharge of the CCW and SW pumps and are in parallel configurations. During routine pump swaps, the valves closed violently when flow reversal occurred. This presentation describes the role the nozzlecheck replacement valves played in achieving a permanent and highly successful problem resolution.

Introduction

Swing type check valves were originally installed in Ginna's CCW and SW systems at the discharge of the parallel configured pumps. Signs of excessive force from swing check valve "slamming" and inappropriate valve application were manifested as:

1) CCW System

   - Grout cracking beneath the CCW pump/motor concrete pads and base
   - Damage to motor electrical power cable
   - Abnormally high pump and motor vibrations caused by induced misalignment between the CCW pump and motor
   - Support system pressure gauges and switches frequently found "out-of-tolerance" during periodic calibrations
   - Delays in valve "prompt closure" resulted in inadvertent auto-starts of the standby CCW pump
   - Delays in valve "prompt closure" often resulted in difficulty passing quarterly ASME Section XI check valve exercise/closure tests
   - Cracked valve seat surfaces
   - CCW heat exchanger tube fretting

2) SW System

   - SW heat exchanger tube failures corresponding to SW pump cycling activities (Containment Recirculation Fan and Motor Coolers, Standby Auxiliary Feedwater Pump Room Coolers)
   - Excessive SW header piping displacement (> $^5/_8$ inch)
   - Poor parts availability from the valve manufacturer forced use of customized parts

Investigation

1) To resolve the "slamming" problem, a means of controlling reverse flow, ensuring prompt closure and dissipating the energy emitted from the rapid closure was necessary. The following investigation activities took place in an attempt to identify potential corrective action(s)

   - Explored various replacement type check valve designs (swing, lift, nozzle, piston, tilting disk)

   - Visited Calvert Cliffs which already had nozzlecheck valves installed in the Brine Water (a.k.a. Service Water) and Safety Injection (SI) systems.

     > At that time, Calvert Cliffs was the only plant in the United States to have nozzle checks installed in safety related systems.

     > The SI application solved very severe Hilte bolts/support degradation problems induced by check valve slamming events.

   - Conducted in-depth discussions with various industry valve experts such as:

     > Nuclear Industry Check Valve (NIC) group
     > ASME
     > USNRC
     > Multiple Engineering consultants
     > Numerous valve manufacturers

Technical Approach

1) Ginna Station Design and Technical Engineering personnel prepared for a new style replacement valve by:

   - Testing the existing swing check valve and CCW piping by monitoring test parameters with instrumentation sensitive enough to detect pipe deflection, pressure surge spikes and pipe strain.

2) Comparing the CCW system design and operating performance parameters:

	DESIGN	OPERATING
Flow Rate (gpm)	2980	600 – 2750
Pressure (psig)	150	78 – 90
Temperature (°F)	200	70 – 120

3) Obtaining swing check valve test data during CCW pump activations:

   NOTE 1: The actual CCW test was videotaped to provide audio and visual evidence as well.

   NOTE 2: Testing was repeated three times for statistical validation.

   - Lanyard [pipe displacement (in)] = 0.003–0.005
   - Transducer [pressure surge spikes (psig)] = 1600–1800 (> 10 times expected design magnitude, very brief spikes)
   - Strain Gauge, [pipe strain (in/in)] = not detectable

4) Using a matrix evaluation model to evaluate and assign weighting factors to various objectives and potential solutions.

   - The intent of the matrix is to optimize relationships between cost/ benefit, risk and payback period.

5) Reviewing the "as-found" system isometrics, preparing for differences in replacement valve characteristics.

	Existing Swing Check	Replacement Nozzlecheck
Length, end-to-end (inches)	19½	17½
End configuration	Butt Weld	Flanged
Weight (pounds)	235	176
Flow Coefficient (Cv)	1755	1694
Seismic, Center of Gravity (Cg, inches from pipe centerline)	6	0

6) Reviewing hydraulic flow models, preparing for differences in replacement valve Cv.

7) Reviewing pipe stress models, preparing for differences in replacement valve weight and Cg.

8) Evaluating acceptability of vertical orientation of the CCW swing check valve.

Final Decision

1) A decision was made to install nozzlecheck valves in the CCW and SW systems.

Valve Installation/Testing

1) Installation of flanges and nozzlecheck valves went as planned.

2) Post-installation performance testing of the new valves using the same sensitive instrumentation was not warranted based on their smooth, silent, and virtually "slam-free" operation.

3) CCW and SW pump activations were videotaped to provide before and after comparisons.

Conclusions

1) Installation of the nozzle check valves first in the CCW system and later that same year (1994) in the SW system, has totally solved all reverse flow and "slamming" problems.

2) Other notable post-installation points of interest include:

- In the NRC's 1994 RFO on-site inspection report, Ginna was recognized for "good initiative and long term resolution of a long standing problem."
- All six nozzlecheck valves are fully capable of being tested during plant operation which allowed elimination of a Ginna Station IST Program, NRC Cold Shutdown Relief Request.
- An Operator "work-around" was eliminated since the CCW pump control switch no longer needed to be held to the "off" position for 5 seconds after securing the pump (afforded sufficient time for the pressure spike to dissipate without auto-starting the just secured pump).
- Emergent problems with SW nozzlecheck valves:
  - > Valves close so tightly that a vacuum forms as the valves close and water attempts to flow back down into the screen bay (~ 16 feet) to existing lake level. Resultant pump packing consolidation actually caused

shaft wear (grooves) and required installation of individual vacuum breakers on pump side of the valve.

> There have been no recurrences of the shaft wear since the vacuum breakers were installed.

3) Following **five years of continuous operating service**, one CCW and three out of four SW nozzlecheck valves have been removed and inspected to assess valve wear and overall condition. Inspection results:

- V–723B—CCW Pump "B" Discharge (March 1999)
  - > 100% freedom of movement
  - > No evidence of seat or internal valve wear, valve plug and seat condition described as "like new"
  - > No evidence of degradation internal or external to the valve
  - > Preventative Maintenance (PM) frequency extended from 5 years to 7 years
- V–723A—CCW Pump "A" Discharge (N/A)

  Based on results of V–723B PM inspection, V–723A PM inspection deferred to 2001 and frequency extended from 5 years to 7 years.
- V–4601—SW Pump "A" Discharge (February 1999)
  - > 100% freedom of movement
  - > No evidence of seat or internal valve wear, valve plug and seat condition described as "like new"
  - > No evidence of degradation internal or external to the valve
  - > Cleaned inner body walls
- V–4603—SW Pump "C" Discharge (February 1999)
  - > 100% freedom of movement
  - > No evidence of seat or internal valve wear, valve plug and seat condition described as "like new"
  - > No evidence of degradation internal or external to the valve
  - > Cleaned inner body walls
- V–4604—SW Pump "D" Discharge (January 2000)
  - > 100% freedom of movement
  - > Lapped seat and disc to obtain 360° positive contact [valve is not a Category A IST component]
  - > Replaced all 3 disc springs as a preventative measure
  - > Cleaned inner body walls
- Preventative Maintenance frequency (5 years) **not changed** for V–4601, V–4603 and V–4604 due to evidence of sludge buildup on inner valve body walls caused by "raw water" (Lake Ontario) service environment. This condition did **not** affect valve performance.

V–4602—SW Pump "B" Discharge slated for inspection ≈ September 2000.

Future Nozzlecheck Endeavors at Ginna Station

1) Based on current problems with swing check valves located in the steam supply lines to the Turbine Auxiliary Feedwater Pump (sluggish and often incomplete closure, large backstop slam and excessive corrective maintenance), a modification to replace these valves with normally closed nozzle-check valves will be performed during the September 2000 Refueling Outage.

2) Efforts to improve check valve performance (i.e. back leakage) in the Safety Injection system include consideration of nozzlecheck valves as potential problem solvers.

Investigation of High Lift Phenomenon in Dresser 3700 Series Main Steam Safety Valves

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John Hosler, EPRI
Steven Hart, South Texas Project

Abstract

Several studies have been performed in recent years to investigate a phenomenon of high initial (as-found) lift test pressure in dresser 3700 series Main Steam Safety Valves in nuclear power plants. Building on the result of previous investigations, new investigations have been performed to better understand the cause of sticking and lead into other corrective action options.

The objective of this paper is to present the results of new laboratory investigations and other identified industry trends associated with sticking valves. This information is intended to help identify potential corrective action considerations.

Background

The ASME code requires periodic testing of the main steam safety valves (MSSVs) in pressurized water reactors to confirm valve operability and valve opening within the designated Technical Specification set point tolerances. The set point tolerance is in most cases ±3%, depending on each plants unique specification. Several nuclear power plants with Dresser 3700 series Main Steam Safety Valves (MSSVs) have reported that the in-situ "as found" test pressure exceeded their set pressure tolerance. The initial lift, or first "pop," has been reported to exceed the valves set pressure by up to 7% and occasionally higher. In a few cases, the valve failed to open when the in-situ lifting device was applying maximum load. In these cases, it is unknown to what extent the valve was beyond its set pressure tolerance. In most cases, following the first pop the valve set pressure is reported to return to within the set pressure tolerance.

A sectional view of a Dresser 3700 series main steam safety valve is shown in Figure 1. As seen in this figure, the valve features a disk that is seated against a seat bushing, referred to as a nozzle. A stem holds the disk in place, and a spring that applies a preload to the stem holds the valve closed until the internal system pressure overcomes the spring force at which time the disk lifts and the valves relieves system internal pressure.

The valve disks are manufactured of an A422 martensitic stainless steel and the nozzle is an A316 or A347 austenitic stainless steel. Figure 2 shows details of the disk-nozzle region.

Observations and Conclusions from Previous Investigations

Previous investigations of the performance of these valves were performed for several nuclear utilities including a significant effort sponsored by Pacific Gas and Electric Company [1]. The objective of the work presented in this paper is to build on previous observations and conclusions.

The results of the previous metallographic examinations, valve testing, and finite element analyses provided insight into the phenomenon. The following observations and conclusions were documented:

1. There is clear forensic evidence of a mechanical transfer of material from the nozzle to the disk. Scanning Electron Microscope (SEM) images are shown in Figures 3 and 4.

2. The smeared, elongated configuration of these "scars" reflects a relative disk-nozzle movement in the radial direction.

3. There is significant relative radial displacement between the disk seat and the nozzle seat during heat-up and upon cool down. The primary cause of the relative displacement is the different thermal expansion coefficients of the disk and nozzle materials and changes in the nozzle diameter due to pressurization.

4. The transferred material from the nozzle to the disk is a result of a sticking mechanism between the disk and nozzle. Candidate mechanisms include microbonding and oxide locking.

A recommendation was made to install pre-oxidized Inconel X750 disks as a replacement for the A422 disk material. This material was selected because it has a more favorable coefficient of thermal expansion that would reduce the amount of relative displacement between the nozzle and disk. It is also a highly corrosion resistant material. The disks were pre-oxidized in a steam environment prior to installation.

Scope of New Investigations

This evaluation was performed to obtain information to enable a better understanding of the specific mechanism(s) and/or cause of valve disk-to-seat sticking. A high initial lift is considered in this investigation to be a lift which is at least 2% higher than the subsequent lift. The scope of work presented in this paper includes:

Data Collection—Preparation of a questionnaire and the survey of plants participating in an EPRI collaborative effort as well as selected other plants. The other plants were selected to include those that have not reported high first lifts. The objective was to identify trends that would help in determining root cause.

Analysis of Additional Disks—Additional disks representing a wide range of in-plant initial lift performers were obtained and examined in the laboratory. These examinations were intended to increase the population of examined disks and to be specifically focussed on the oxides at this interface.

Destructive Examination of a Nozzle—A matching disk and nozzle set was destructively examined in the laboratory in an effort to understand how they interface and bond.

Laboratory Examination of an X750 Disk—An Inconel X750 disk removed from service was destructively examined in the laboratory for comparison to the A422 disks.

Survey and Data Collection

Questionnaire

The results from the questionnaire, including telephone interviews, are intended to be integrated with information from the laboratory analysis to lead to the development of a root cause of MSSV sticking. Eleven utilities participated in the survey representing nineteen operating units.

The questionnaire consisted of eight sections and included eighty-seven questions.

- General Plant Characteristics
- MSSV Environment
- Secondary Side Water Chemistry
- Condenser Efficiency/Air In-Leakage
- MSSV Testing Methodology
- MSSV Maintenance
- Post Maintenance Testing
- Valve Off-Site Shipping

Survey Findings Summary

- For the responding plants which observed "sticking," approximately 50% of refurbished valves exhibit high lifts during in-situ testing conducted after 30 to 90 days of service. Fewer valves stick later than this. Exercising valves during operation appear to reduce the potential for sticking. In a limited number of cases, sticking has occurred years after refurbishment—well after the first lift.

- No trends were conclusively identified that related valve sticking to specific installation and operating configuration or parameters, such as set point to operating pressure differential, in-service valve vibration, or feedwater chemistry.

- No trends were identified that related valve sticking to environmental factors such as ambient temperatures, humidity, degree of shelter, or amount/type of insulation.

- Plants that have not experienced valve sticking generally have performed infrequent or no valve refurbishments.

- Some data indicates that a gray, rather than mirror finish, may reduce the potential for sticking. This data, however, is not extensive and further study would be required to be conclusive.

- Pre-oxidized Inconel X750 disks have performed well to date.

- Some respondents do offsite "as found" testing. Although isolated cases may have occurred, the seat sticking phenomenon has not been identified as a significant issue during such testing. This is possibly because the cooling and depressurization process during plant cooldown results in relative movement between the disk seat and the nozzle and/or due to seat flexure which could break the bond.

- Within the limits of delectability, no history of the seat sticking phenomenon has been identified to date during actual plant transients, resulting in MSSV lifts. This could be the result of relative disk-nozzle movement and/or flexure during the increased valve internal pressure which could break any adhesion.

Examination of the Nozzle/Disk Interface

A matching disk and nozzle were examined to better understand the disk-nozzle interface and to identify characteristics associated with disk-nozzle bonding. Figures 5 and 6 exhibit micrographs of regions of the disk and nozzle seating surfaces. The radial scars on the nozzle can be seen to correspond with the transfer of nozzle material onto the disks. A higher magnification optical and SEM examination was performed of a small region of the disk and nozzle surfaces. Figure 7 shows a 4 mil (102 micron) by 30 mil (762 micron) matching section on the disk and nozzle at the ID edge. The nozzle shows a free growth region of oxide of 6 mils (152.4 microns) at the ID edge. It is clear that no disk contact occurs at this location. The next 6 mils (152.4 microns) exhibits a densely packed region of oxide scale. Alternating regions of thick and thin oxide shows a matching pattern between the disc and nozzle clearly indicating adherence between both the disk and nozzle prior to separation. Broken and flattened oxides can be seen further toward the disk/nozzle outside diameter, but they are not as continuous or as tightly packed. Oxide chemistry analyses described below considers the presence of mixed oxides.

Figures 8 shows an image of both the disk and nozzle seats in the area of scars. The nozzle clearly shows evidence of pushed metal deposits with trailing groves. The disk clearly shows corresponding nozzle deposits ("scars") on the seating surface.

The following can be concluded from the examination of this particular disk and nozzle set:

1. The oxide characteristics are greatly influenced by the disk nozzle contact pressures and gaps.
2. There are signs suggestive of oxide bonding between the nozzle and disk (shared oxide scale), particularly at the ID edge of the disk and nozzle.
3. The oxide at the ID edge is quite dense in appearance and may influence the effective pressure area of the disk (mean seat area) should it be tight and continuous enough.

Inconel X750 Disk

Previous Applications

Disks manufactured from Inconel X750 have been successfully utilized at two plants. The performance reported to date is favorable with no high lifts occurring. A preventative program of interim lifts has been employed as a precautionary measure with the longest period of continuous service to date with a successful lift at the end of the period being 330 days.

However, the usage and service time of X750 disks has been somewhat limited. Continued monitoring of performance at plants that have had a successful history with X-750 disks and those currently implementing that option is required.

Comparison of Pre-Service and Post-Service Seat Surface

Prior to initial service, disks manufactured from Inconel X750 were non-destructively replicated in order to baseline the condition of the seating surface for future reference. After being in service for approximately two years, the disks showed evidence of radial scarring. A destructive analysis of a disk was performed to analyze the individual scar material by Energy Dispersive Spectroscopy (EDS) methods.

SEM images of the disk surface are provided in Figures 9 and 10. Figure 10 exhibits an EDS elemental mapping of iron which indicates the scars to be iron based (nozzle) material. The amount of scarring is generally less than typically seen in the original materials.

Oxide Characterization

As part of the investigation to determine the cause of sticking of the safety relief valves, a detailed analysis and characterization of the surface of selected disks and a nozzle was performed.

The oxide scale formed on the surface will be dependent on such factors as oxygen potential of the atmosphere and temperature. These factors not only affect the particular oxide or oxide phases present on the surface but also the thickness of the oxide layers. Research has shown [2] that when austenitic stainless steels are exposed to air at elevated temperatures (above 350°F (177°C)) for two hours, the profile through the surface appears to have an iron oxide outer layer below which there is a chromium rich layer. Figure 11 shows a graph illustrating the change in composition going from the outer surface into the base metal of a 304 austenitic stainless steel (graph is based on data presented in Ref. 2).

Oxide Analyses

The depth profile analysis of contact surfaces from disks and nozzles was expected to provide insight into the role of oxide locking in the sticking of safety relief valves. The depth profiling can be performed by various techniques. Auger Spectroscopy was the method chosen for the depth profiling. Two methods used in this study were line scans and selected point analysis through a prepared cross section and the analysis/etching process described above. In addition, Scanning Electron Microscopy (SEM) with Energy Dispersive Spectroscopy (EDS) was used to examine a cross section through a scar on a disk.

Line Scans

Line scans were performed for selected disks. In this method, cross-sections were cut from the disks in the region of transferred metal (scarring) and the cross section was examined using both SEM/EDS and Auger spectroscopy. Figure 12 shows the SEM micrograph of the region examined and the line scans for the elements iron, chromium, nickel, and oxygen through a scar.

The SEM/EDS line scan through the scar confirmed the presence of nozzle material in the scar. Furthermore, the analysis shows a chromium peak in both the nozzle material in the scar (outer 0.08 mils (2 microns)) and in the disk material. The iron and oxygen rich region between the two chromium peaks appear to have resulted from oxide mixing. The volume analyzed by the EDS technique is of the order of 0.12 to 0.20 mils (3 to 5 microns) in diameter. The method could not provide adequate data on the variation in chemistry in the oxide areas adjacent to the scars.

Depth Profiling

A detailed scanning Auger Spectroscopic analysis of two points on both the nozzle and disk set was performed. The two points correspond to damaged oxide areas near the inside diameter of both nozzle and disk. The two points were carefully selected to correspond to the same contact areas. Thus, Point 1 corresponded to an area with "thick" oxide layer(s) on the nozzle and an area of matching "thin" oxide layer(s) on the disk. In contrast, Point 2 corresponded to an area with "thin" oxide layer(s) on the nozzle and matching "thick" oxide layer(s) on the disk. These points are shown in Figures 13 and 14 for the nozzle and disk respectively. In addition, a non-contact area was analyzed on both the nozzle and disk. This analysis was undertaken in order to establish the characteristics of the non-damaged (free) oxide that was exposed to steam. This region is referred to as a region of free oxide growth.

Figure 15 illustrates the depth profile for Point 2 on the disk. This point corresponded to a location with thick oxide. The profile shows a number of significant features. There were four distinct layers observed. The first outer layer, about 0.5 microns (0.02 mils) deep, contains iron, oxygen, and nickel and appears to be FeO with nickel metal. The second layer contains less iron but some chromium and no nickel. This layer appears to be a mixed iron chromium oxide material and is about 0.02 mils (0.5 microns) thick. The third layer is very similar to the outer layer and is about 0.02 mils (0.5 microns) thick. The fourth layer does not contain nickel. It was not possible to discern the base metal in the sample despite etching for a depth of 0.1 mils (2.5 microns).

The depth profile for the disk Point 2 was combined with the depth profile of the free disk oxide. The result is shown in Figure 16. The reference point used was the initial observation of chromium in the lower layer. The match between the free oxide and the damaged oxide region is excellent below a depth of 0.08 mils (2 microns). In addition, the region between about 0.05 mils (1.3 microns) and 0.08 mils (2 microns) shows an excellent match between the oxygen and iron levels for both the iron and the oxygen. However, in this area there is some nickel. This element was not observed in the free oxide and indicates the transfer of material from the nozzle. Indeed, comparison of the profile of the outer 0.06 mils (1.5 microns) of nozzle free oxide, shows remarkable similarity to the 1-2 micron range identified in the depth profile of Point 2 (damaged oxide) on the disk. Thus, as indicated in Figure 16, there is direct evidence of transfer of nozzle material to the disk in a region of damaged oxide.

The depth profiling of the oxide scale on the surface of the disk and nozzle has illustrated the presence of transferred nozzle material in a region of damage. Furthermore, it is shown that the oxide formed in the non-contact areas of the disk and nozzle (inside diameter surface) are similar to the oxide scales formed on the contact surfaces away from mixed oxide areas. This data is used to illustrate the region of mixed oxides on the disk surfaces.

Results of New Laboratory Investigations

- Observed that seating surface characteristics vary from disk to disk. This is partially due to slight variations in seat contact that can alter crevice electrochemical potential, seating stress profile, and seal tightness.
- Transferred nozzle material onto the disks does not always accompany high initial lifts. No clear relationship between sticking history, degree of sticking, and scar density is observed.
- Evidence suggestive of oxide locking in the form of mixed oxides and fractured

oxides is found on A422 stainless steel disk/nozzle surfaces and is believed to be a primary factor in sticking.

- Oxide accumulation at the ID edge of the seat may result in a mean seat area (MSA) reduction resulting in higher lift pressures for the first lift. The disk/nozzle seating interface characteristics are somewhat variable between disks based on machining tolerances. However, on the paired disc and nozzle set evaluated, a clear area of densely compressed oxide scale was noted at the ID edge of the seat. This tightly packed crevice coupled with the existence of moisture from condensation of steam on the inside surfaces could possibly result in an effective pressure seal, reducing the area exposed to full steam pressure.

Recommendations

Based on the results of this investigation, recommendations for additional investigations and/or mitigating actions are provided. These are as follows:

Use of Pre-oxidized X750

Based on successful performance to date and the laboratory examinations performed of disks removed from service, the use of pre-oxidized Inconel X750 disk material appears to be a viable option to remedy the high lift phenomenon. However, field experience with this material is still somewhat limited and long-term performance has not yet been demonstrated.

Maintenance Schedule

A strong relationship between the frequency of maintenance and the susceptibility to high initial lifts has been identified. Accordingly, performance based MSSV maintenance schedules rather than time based maintenance is recommended. The valves should be refurbished only when required.

Surface Finish

There is some indication that a gray rather than mirror finish reduces the possibility of sticking. Although additional confirmation of this relationship is required, a gray rather than mirror finish should be considered.

References

1. Esselman, Thomas C., Beckham, Peter E., "Safety Valve Operating, Setpoint Pressure, and Testing" ASME Pump and Valve Symposium, April 17, 1998

2. Joshi, A. " Investigation Of Passivity, Corrosion and Stress Corrosion Cracking Phenomena By AES and ESCA," Coatings Corrosion Vol. 3, 1979 pp 55–71, Physical Electronics Industries, Inc.

Acknowledgements

Previous Work

The work presented here builds on the substantial work previously sponsored by Pacific Gas & Electric Company (PG&E) to understand the main steam safety valve seat sticking mechanism and to develop and demonstrate a potential fix. PG&E provided disks for destructive examination as well as test data and technical position papers that were pivotal to the preparation of this paper. Mr. Harry Machado of PG&E is recognized for his support and technical insights that were so valuable to this effort.

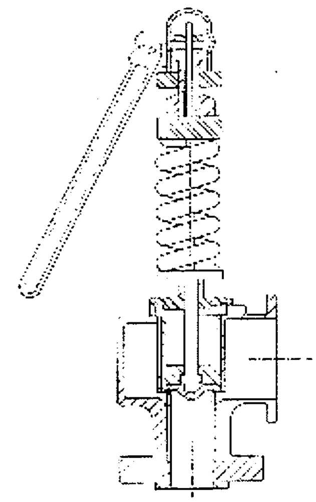
Technical Advisory Group

The utility Technical Advisory Group (TAG) provided technical oversight to the project. The TAG consisted of representatives from each of the six participating utilities. These individuals provided test data, valve disks and a valve nozzle, results from internal root cause evaluations and the benefit of their substantial experience with the MSSV seat sticking issue to support the project. The TAG members are listed below.

Mr. Ronald Cameron
(Baltimore Gas & Electric)
Mr. David Haley
(First Energy Corporation)
Mr. Steven D. Hart
(Duke Energy Corporation)
Mr. Steven K. Hart (South Texas Project)
Mr. James Krueger (UNICOM)
Mr. Steven Quan (Arizona Public Service)

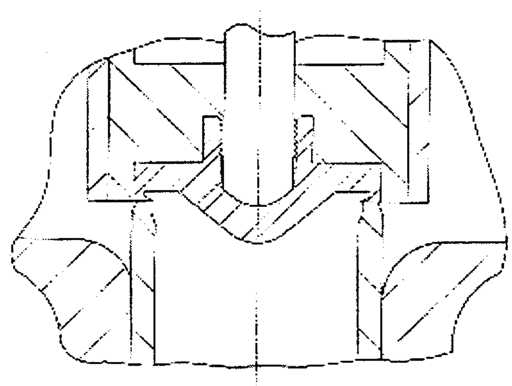
Altran

Recognition is also given to Mr. Peter Beckham, Dr. David Kalmanovitch, Mr. Van Christie and Mr. Gerry Kerzner who contributed significant efforts into these investigations.



STEAM SAFETY RELEF VALVE

Figure 1. Configuration of a Dresser 3700 Series Steam Safety Valve



DISK AND NOZZLE DETAIL

Figure 2. Disk and Nozzle Detail

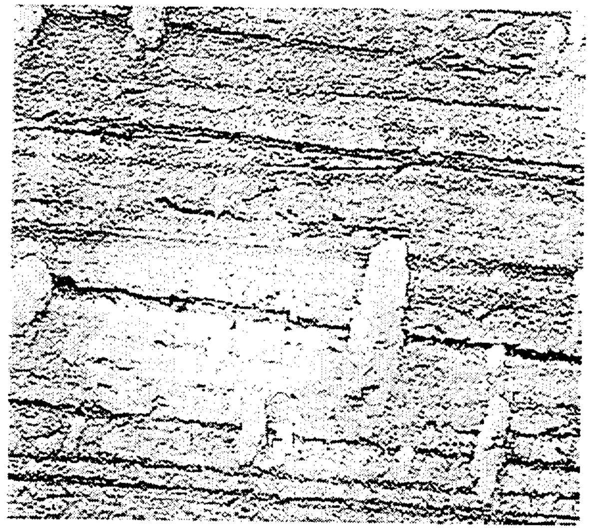


Figure 3. SEM Micrograph of A Disk Surface Showing Radially Oriented Nozzle Deposits (field of view 0.0045 in. horizontal)

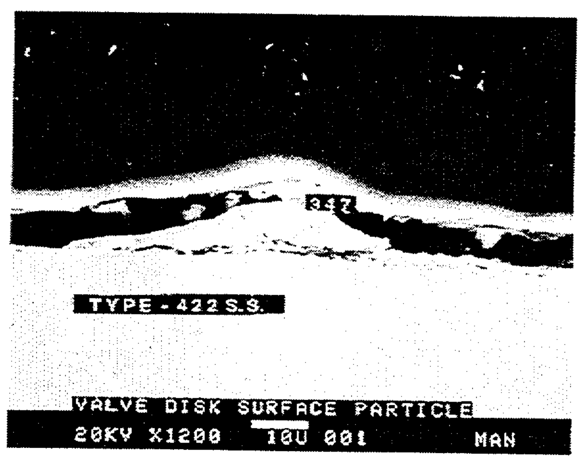


Figure 4. SEM Micrograph Cross Section of a Nozzle Deposit on the Seating Surface of a Disk

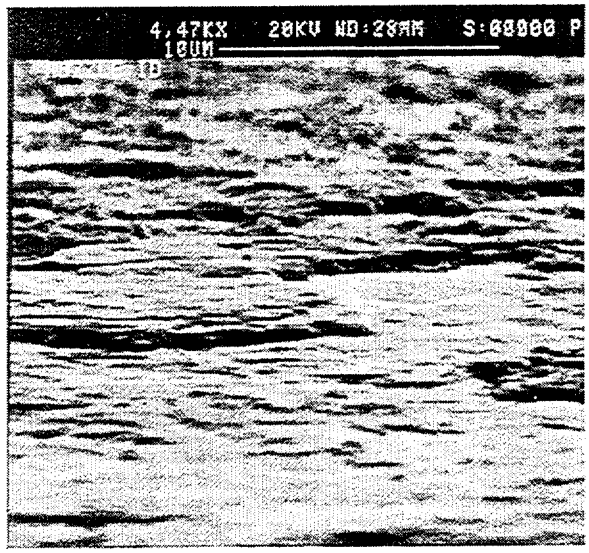


Figure 5. SEM Micrograph of Nozzle Seat Surface Near ID Showing Irregular Oxide Pattern

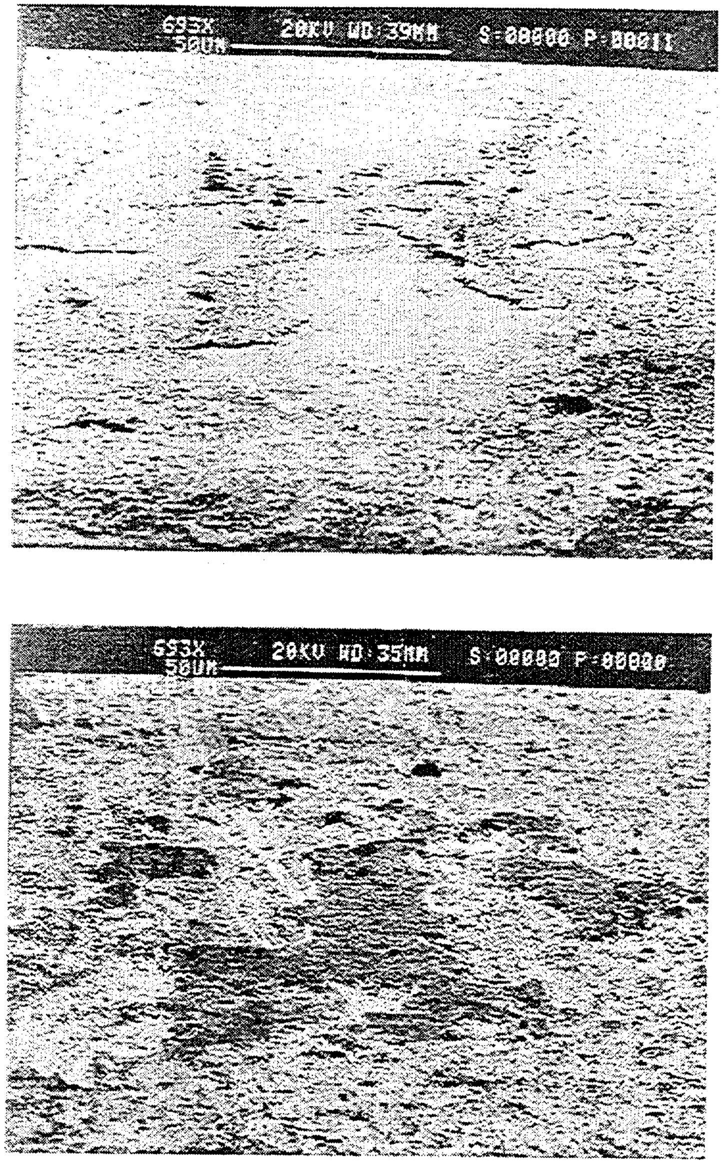


Figure 6. SEM Micrograph of "Pulled" Oxides Between Matching Locations on Disk (top) and Nozzle (bottom)

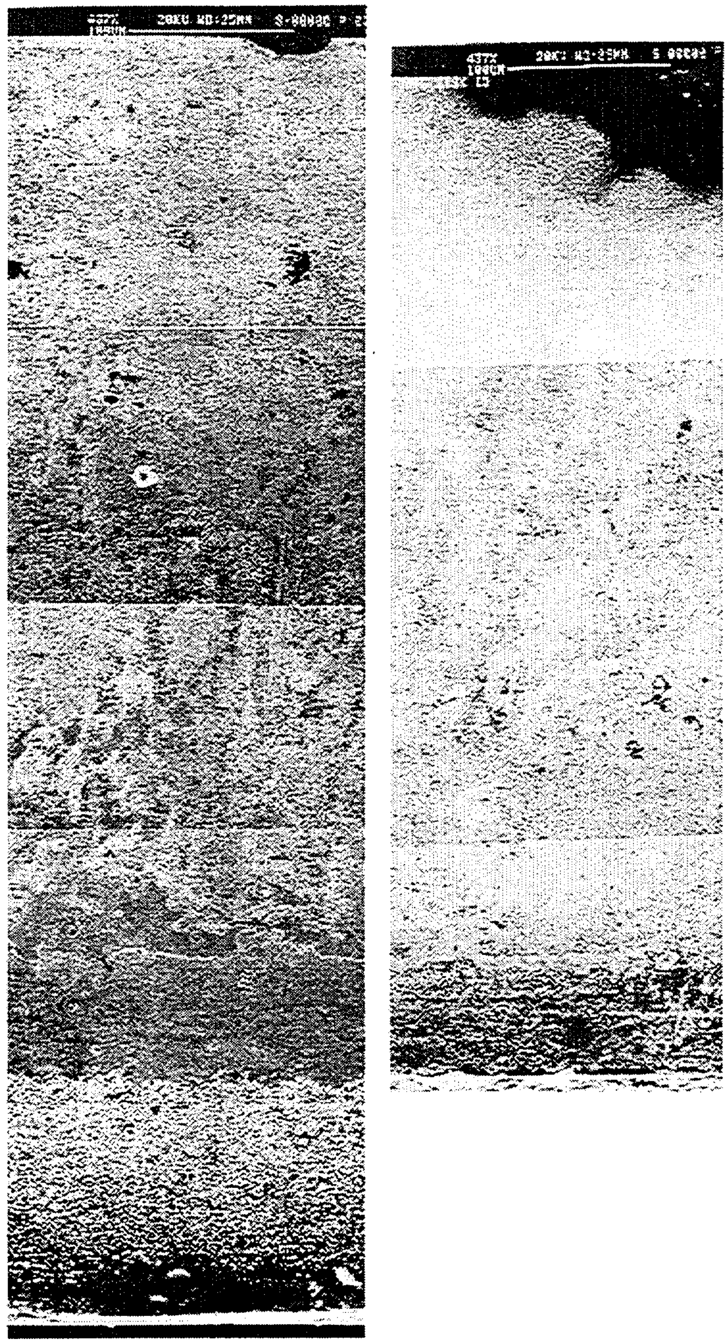


Figure 7. Micrograph of matching nozzle and disk locations at seat OD region (0.004 in wide)

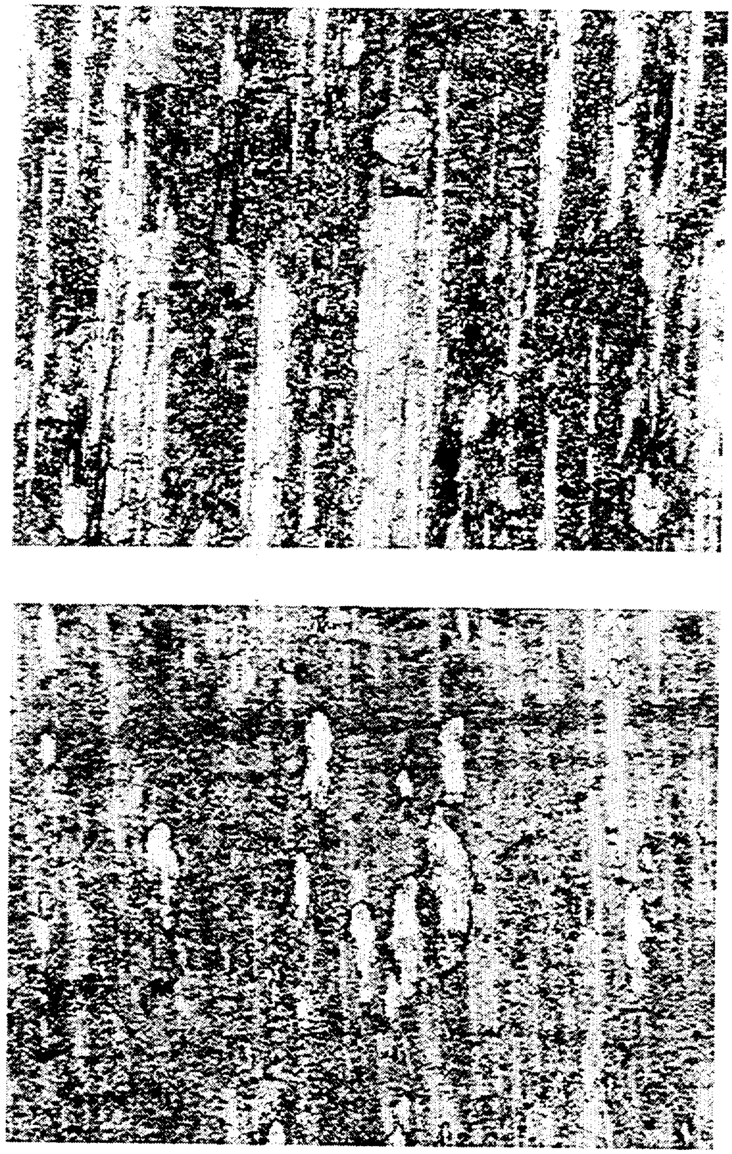


Figure 8. SEM Micrographs Nozzle (top) and Matching Disk (bottom) Showing Spots of Smeared Nozzle Material (OD toward top, field of view 0.009 in. wide)

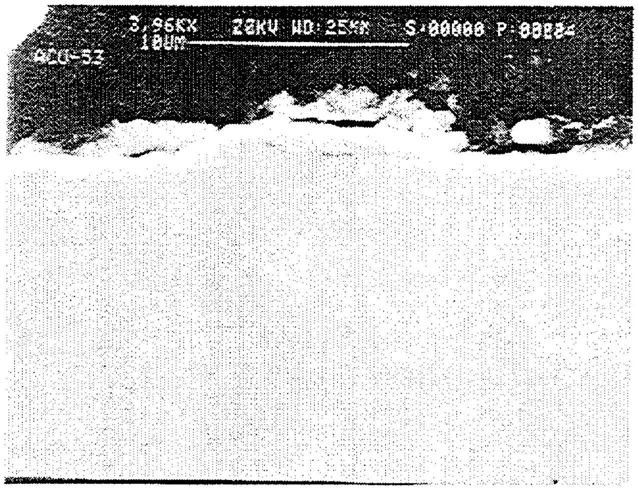


Figure 9. SEM Micrograph of Nozzle Deposit Material Section on X750 Disk

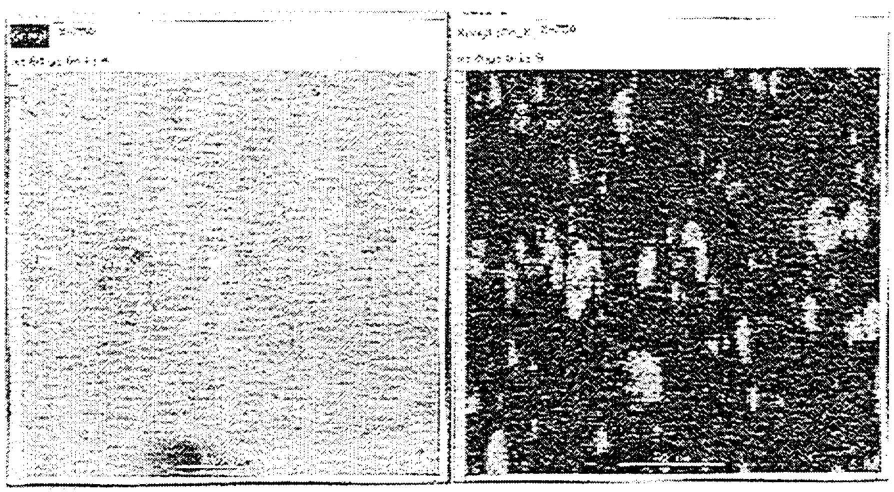


Figure 10. SEM Micrograph of Nozzle Deposit Material Section on X750 Disk With an EDS Iron Map on the Right Showing High Iron in the Radial Scars that Confirms that the Scars are Transferred from the Nozzle

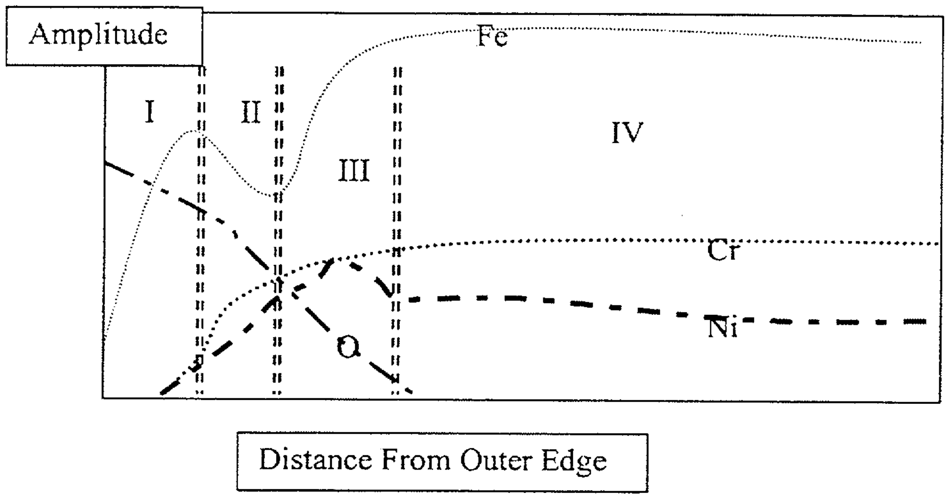


Figure 11 Depth Profile of Oxide Scale Formed on 304 Stainless Steel at 350 °F (177°C) showing Different Zones I through IV (derived from Ref. 2)

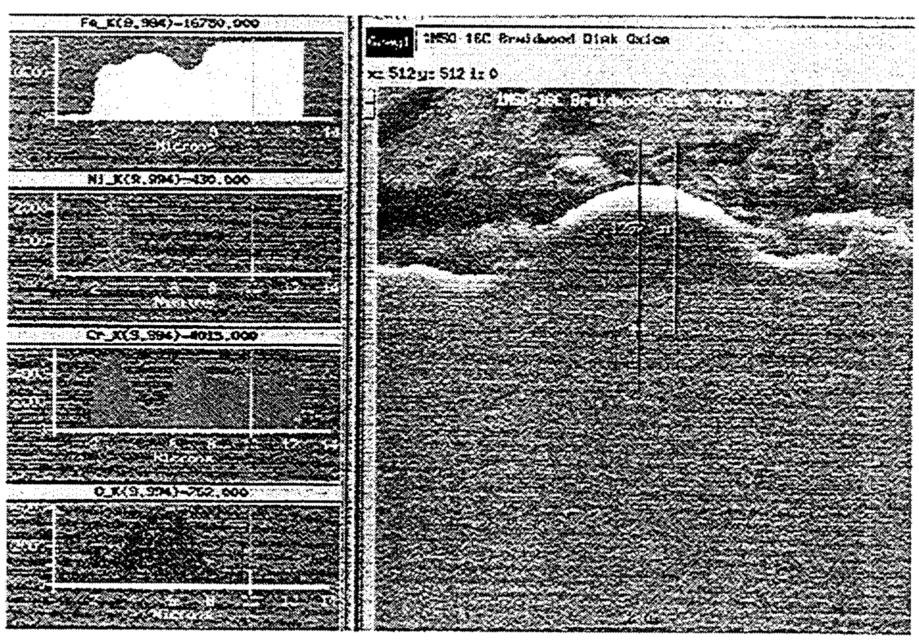


Figure 12. SEM/EDS Line Scan Through Scar

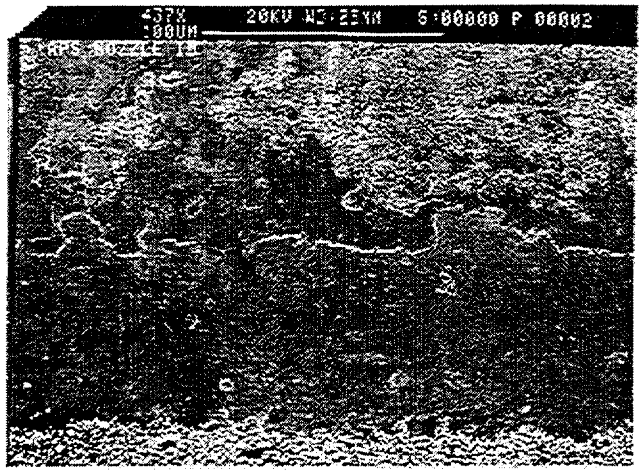


Figure 13. Scanning Electron Micrograph showing Damaged Oxide near Inside Diameter of the Nozzle. Examination Points 1 and 2 Shown. (Scalar Bar = 100 microns)

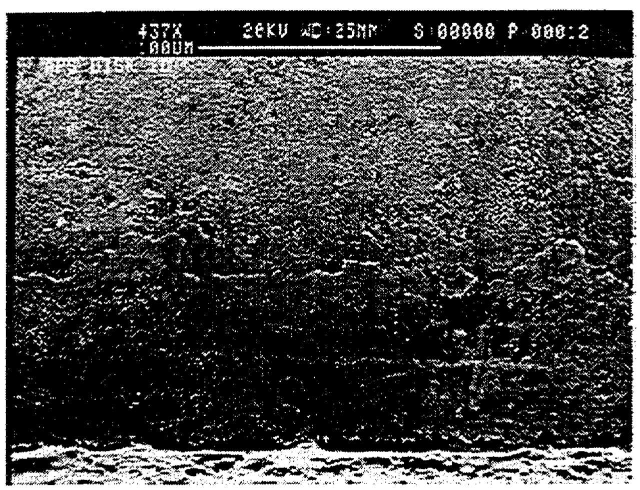


Figure 14. Scanning Electron Micrograph showing Damaged Oxide near Inside Diameter of the Disk. Examination Points 1 and 2 Shown. (Scalar Bar = 100 microns)

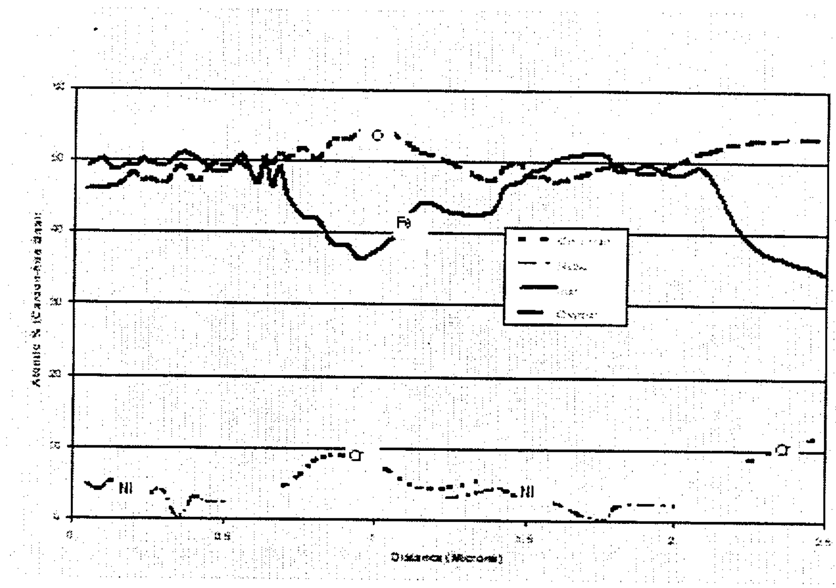


Figure 15. Auger Depth Profile—Point 2 of Damaged Oxide on Disk

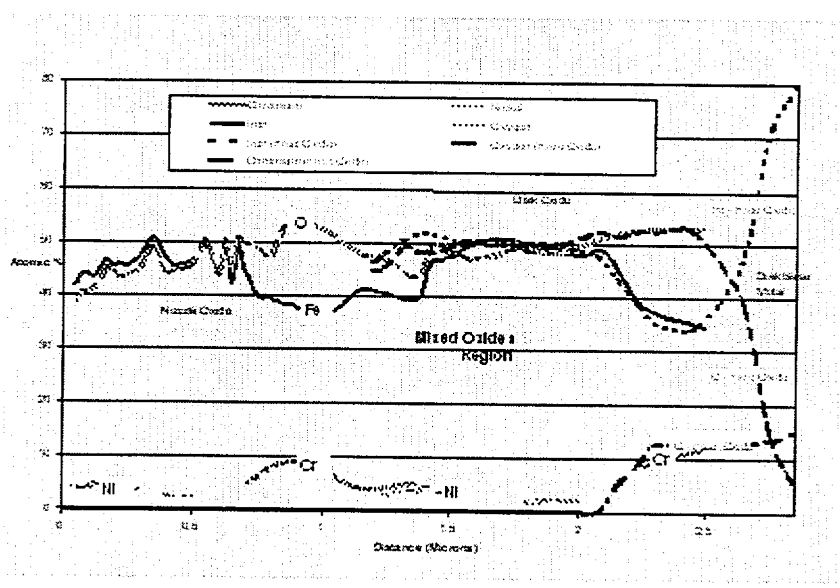


Figure 16. Combined Depth Profiles of Damaged Disk Oxide (Point 2) and Non Contact (Free) Disk Oxide

Check Valve Condition Monitoring at Wolf Creek

Shawn Comstock
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Abstract

Wolf Creek was the first nuclear plant in the U.S. nuclear industry to obtain permission to use ASME OMa Code–1996 APPENDIX II as an alternative to existing inservice testing program Code requirements. The goal of this process is to improve or maintain check valve reliability over the life of the plant through a blend of test, design, and maintenance data analysis. Reliability Centered Maintenance failure mode effects analysis evaluations and performance trending are two critical aspects of a successful program. *The equivalent dollar savings in terms of ALARA and labor costs alone over the current planned life of the plant is estimated to be more than $300,000.* An added bonus is that the up-front costs and uncertainty associated with a risk-based approach are avoided, yet this Code is fully compatible with a risk-based approach.

Implementation of ASME OMa Code–1996 APPENDIX II can reduce the overall cost of testing check valves while at the same time maintaining an equivalent level of safety assurance. Critical path testing and disassembly of refueling outage check valves are two key areas of savings. One such application is the check valves included in the Appendix J Option B (LLRT) program. Although Appendix J Option B allows the extension of test frequency based on performance, the traditional inservice testing program still requires closure verification every refueling outage. Other applications include check valves that are disassembled and inspected every outage. Typically these are high dose jobs which cost outage time and increase labor costs.

Many check valves must be flow tested during outage critical path times because of the nature of their safety function. The valves used in the demonstration process were required to be included in one of these critical path procedures. ASME OMa Code–1996 APPENDIX II can be used to determine an equivalent or more effective means to demonstrate acceptable check valve performance outside critical path, resulting in outage critical path time reduction and ALARA savings.

Introduction

The American Society of Mechanical Engineers (ASME) Operation & Maintenance (O&M) Committee established the Working Group on Check Valves (WGCV) in 1990. Several weaknesses in the Code governing check valve testing were identified by the group. At that point, they set out to create a Code that addressed weaknesses including:

- Code requirement language weakness
- Rigid time-based activity
- Testing not commensurate with safety
- System conditions not considered
- Industry failure data ignored.

While correcting these Code deficiencies, the WGCV also recognized that testing methodologies could be improved by taking advantage of reliability centered maintenance (RCM) philosophies, performance monitoring, preventive maintenance, new technology, and nuclear industry check valve group (NIC) developments.

The result of their hard work is ASME OMa Code–1996 APPENDIX II Check Valve Condition Monitoring Program. This portion of the Code is currently endorsed for use in the latest revision of 10 CFR 50.55a. Figure 1 is a graphic representation of the Code with NRC restrictions. Refer to Figure 1 for an idea of how the Check Valve Condition Monitoring process works.

Implementation at Wolf Creek

Wolf Creek's implementation of ASME OMa Code–1996 APPENDIX II Check Valve Condition Monitoring Program was reviewed initially by a team of industry experts and Nuclear Regulatory Commission (NRC) representatives. The program was found to be acceptable. The program initially included four valves to demonstrate implementation of the process for a self assessment and subsequent NRC review. The four valves selected for the program were the residual heat removal (RHR) cold leg injection check valves. These valves were selected based on data from the Institute of Nuclear Power Operations (INPO) check valve program and the importance to the plant as reactor coolant system (RCS) pressure isolation valves at the low pressure interface. The INPO program grouped these valves in different categories based on initial engineering analysis.

It was originally thought the program would be used to take credit for the INPO check valve reliability program. It was discovered during the research phase that, while using the INPO program is a good starting point, additional analysis and documentation is required. Also, the focus of Wolf Creek's INPO check valve program activities was primarily in acoustic monitoring with disassembly and examination. The ASME check valve program can be used to take credit for a host of other activities such as system pressure or flow trending, inservice inspection (ISI) pressure boundary testing and examination, flow accelerated corrosion trending, preventive maintenance activities, ultrasonic examinations, x-ray examinations, or any activity that provides insights into check valve condition. Most of these activities are already being performed to satisfy program or operational requirements outside of the inservice testing program.

Valve implementation priority

Since the initial demonstration phase, other valves have been added to the program. Certain check valves are tested less often than others. Where practical, check valves are tested quarterly. In many cases, it is not practical to test a check valve during normal operation. These check valves are tested on a refueling or cold shutdown test frequency. This is typically once every 18 months, or one-sixth of quarterly-tested check valves. Valves tested less often in the traditional inservice time-based testing methods provide a lower assurance of operational readiness. Valves tested during cold shutdowns and refueling outages can impose a more significant impact on plant safety and resources than those tested quarterly. Therefore, the current implementation scope is limited to those valves that are tested less often than others, namely those valves that are tested less often than quarterly.

Candidates for the program are chosen based on test frequency length and known or potential impact on plant resources. The amount and type of plant information available is also a consideration since valves with certain types of maintenance and test information are more easily placed in the program. There are 111 Wolf Creek check valves which are tested on a cold shutdown or refueling outage cycle. There are 14 local leak rate tests, 87 exercise close tests, and 74 exercise open tests. Of the exercise close tests, seven take credit for the local leak rate tests. Reducing the close tests by seven, the total number of low-frequency check valve tests is 168.

Purpose statements

The first task in implementing check valve condition monitoring on a group of valves is to identify why valves are being added to the program. The following text provides an example:

"The subject check valves are being added to the check valve condition monitoring program effective immediately. These valves are being added to the program in order to take credit for their historical and continued acceptable level of test performance and maintenance history. The reasons for the acceptable level of behavior, design characteristics, application, and service conditions are documented in the analysis section. The condition monitoring program activities and their intervals to maintain the continued acceptable level of performance are documented at the conclusion of the analysis."

Once the purpose for adding a check valve or group of check valves to the condition monitoring program is defined and stated, an analysis is performed to identify the basis of condition monitoring activities.

Analysis

The maturity of the nuclear industry gives us a big implementation advantage. Virtually all of the information required to perform an analysis is already available in plant records (e.g., test history, maintenance history, design information, industry data). Additional research using the NIC database is performed to identify if problems have occurred in the industry on the same model or similar style check valves or valves in the same application. Once all of the data is gathered, reliability-centered maintenance evaluation is performed to identify failure modes and potential effects. This type of assessment is typically referred to as failure modes and effects analysis (FMEA) in the industry. If not enough information is available to perform an FMEA, the performance improvement section of the Code is used to identify tests, examinations and other activities to gather the needed data. So far, it has not been necessary to use the performance improvement section at Wolf Creek, which began operation in 1985. On valves added to the program, all have had a sufficient amount of data available to perform an FMEA.

Example Analysis

The following is an example of the analysis performed for the check valve condition monitoring program.

The open safety function of valves A, B, C, and D is to allow RHR flow for normal and emergency cool down. The close safety function of these valves is to prevent loss of RCS inventory as an RCS pressure boundary isolation valve and prevent over-pressurization of the RHR header. Check valves in this group are

Westinghouse stainless steel six-inch swing-type check valves. A pipe break upstream of these valves is not a postulated accident as described in the Updated Safety Analysis Report (USAR). RCS pressure is increased gradually, not suddenly. Safety injection (SI) operability is only required in modes 1, 2, and 3 when the RHR system is not normally operating, therefore valves will be seated prior to SI actuation. Based on these observations, these valves are not subjected to reverse flow slamming. A swing check valve is acceptable in this application since 1) fast closure is not required, 2) except for rare operation at RCS mid-loop, this valve operates with little flow variation, and 3) the valve seats tightly which is appropriate since this valve acts as the RCS pressure boundary.

All valves are installed in horizontal runs of pipe, which is appropriate. Installation of check valves A and D does not meet the Electric Power Research Institute (EPRI) guidelines of having at least five diameters of straight piping upstream of a check valve. Valve A has an 18-inch flow orifice three pipe diameters upstream. The close proximity of this orifice and valve can create excessive turbulence that tends to increase valve wear. Calculated corrosion allowance for valve A is .016-inch. This valve is subjected to flows in excess of 661 gallons per minute (GPM). Conservatively, the maximum expected corrosion rate for the body material is two mils per year for this group of valves. The corrosion life before reaching minimum-wall (min-wall) is estimated to be eight years. Valve D has an 18-inch flow orifice 2.9 pipe diameters upstream. The close proximity of this orifice to the valve can create excessive turbulence that tends to increase the wear of the valve. The valve is installed in a horizontal run of pipe. The calculated corrosion allowance for valve D is .039-inch. Corrosion life before reaching min-wall is estimated to be 19 years. These valves are not exposed to harsh service conditions, therefore, actual corrosion rates should be less than postulated maximum. Additional ultrasonic test (UT) examination provides a more realistic estimate of wall thinning.

Installation of valves B and C does meet EPRI guidelines of having at least five diameters of straight piping upstream of a check valve and located more than 10 pipe diameters downstream of control valves, restriction orifices and reducers. These valves have continuously exhibited acceptable seat leakage when tested according to ASME O&M Code Category A leakage requirements. Valve B has a .193-inch valve body corrosion allowance. At maximum expected corrosion rate, valve min-wall thickness allowance should not be exceeded for more than 40 years. Valve C initially had a .023-inch valve body corrosion allowance. At the maximum expected corrosion rate, the valve min-wall thickness allowance from calculation EP–MH–004 should not be exceeded for 11 years. See additional discussion of valve C min-wall under the **Other failure modes** section.

Calculated minimum velocity (Vmin) for full disc lift is five feet per second flow velocity. Vmin at minimum flow was calculated to be 13.2 feet per second flow velocity. This data indicates a stable disc condition should be present. Non-intrusive test results identified tapping, which is contrary to the calculated data indication. However, because of the magnitude of tapping, size of check valve discs, and disassembly examination results, tapping is not a reliability concern to any of the valves.

Improper seating is a credible failure mode for these valves. Technical Specifications limit valve leakage. The check valve is at a high-pressure-to-low-pressure interface. Analyzed turbulence, high flow and small corrosion allowance, and leakage combine to create a credible failure mode of improper seating.

Detached or broken disc is not a credible failure mode. These valves are not subjected to reverse flow slamming transients that could cause the disc to crack or break. The check valve disc is designed to ASME Class 1 requirements. It has a five and one-half-inch diameter and is one and one-quarter-inches thick. Made of high-strength stainless steel, it is unlikely this disc will experience cracking over the life of the plant. Two of these valves may be subjected to turbulence because of location. A worn hinge pin or eroded disc is a possible event; however, it is unlikely that these events would lead to a catastrophic failure. Degradation events would be detected during normal operation and leak testing before catastrophic a failure.

Free or loose parts because of friction or erosion is not a credible failure mode. Two of the valves may be subjected to turbulence because of location. The wire loop-to-bearing block interface could become worn and loose. This degradation would lead to seat leakage, but not catastrophic failure, before being detected. Therefore, this failure mode is bounded by the improper seating failure mode. These valves are not subjected to reverse flow slamming transients that would cause check valve parts to break free.

Restricted motion or reduced flow is not a credible failure mode. These valves are located in a clean system. The simple design of this type of valve does not lend itself to sticking or binding. Conditions in which valves operate combined with design and materials do not lend credence to this type of failure mode.

Stuck closed is not a credible failure mode. These valves are located in a clean system. The simple design of this type of valve does not lend itself to sticking or binding. Based on system design, significant differential pressure may be developed to open these valves.

Stuck open is not a credible failure mode. These valves are located in a clean system. The simple design of this type of valve does not lend itself to sticking or binding. These check valves are not operated in a manner that lends credence to a stuck open failure scenario.

Other failure modes were evaluated. Exceeding min-wall thickness is considered a possible credible failure mode because of the high flows and turbulence to varying degrees for valves A and D. Not enough wall thickness field measurement data has been obtained to confirm or contradict that there is a wall thinning problem. One piece of evidence that tends to refute there may be a wall thinning problem is the absence corrosion or erosion indication when valves were disassembled and examined. For valve B, this is not considered a credible failure mode because of its min-wall thickness, application, and maximum anticipated corrosion rate. For valve C, min-wall thickness is not considered a credible failure mode because of its min-wall thickness, application, and maximum anticipated corrosion rate. This valve was initially provided with a .023-inch corrosion allowance based on the referenced data from calculation EP–MH–004. Subsequent measurements in 1988 identified that a min-wall of 1.25-inches exists. This provides actual corrosion allowance of .06-inch. If maximum expected corrosion rate were to occur each refueling outage, min-wall specification would not be exceeded until 2018. This valve was disassembled and inspected in 1994. No evidence of corrosion or other wear problems were noted. If a wall thinning problem can be confirmed on valves A and D, analysis will be initiated to assess valve C for wall thinning.

EPRI failure probabilities: EPRI uses several references to summarize generic failure probabilities for this style of valve. Wolf Creek uses probabilities provided by IEEE 500:

- External leakage (body-to-bonnet gasket) probability is 5E-8 per hour of service
- Internal leakage (seat and disc leak-through) probability is 9.6E-5 per demand
- Fails to open (when needed for emergency core cooling system flow) probability is 9.6E-5 per demand
- Fails to close (when flow stops or reverses) probability is 5E-7 per hour of service

NIC database failure summary: A database search was performed which identified failures in 32 six-inch stainless steel swing check valves. Of these, three are of the same manufacture and model number. The three data sheets are 95–068, 92–148, and 91–197. Failure for 95–068 was the result of disc and seat area erosion resulting from abnormal wear caused by foreign material. Failures for 92–148 and 91–197 resulted from unknown or normal wear caused by the disc and seat area not being flat. In all cases, the failures were discovered through programmatic leak testing and damage was moderate.

Valve failure importance: Excessive seat leakage results in an intersystem loss of coolant accident (LOCA) through a low pressure line in the RHR system. Failure to isolate is a significant event that could lead to core damage. Exceeding min-wall thickness could result in a diversion of RHR flow or a LOCA. Single failure of excessive seat leakage and exceeding min-wall thickness would not directly result in a LOCA. The upstream series check valve would also have to fail in both instances for this to occur.

Valve A maintenance history

Disassembly and inspection for wear of internal components:

- Refuel I fall, 1986—no damage found.
- Refuel II fall, 1987—disc and bearing blocks badly worn, replaced internals. Work performed because "chattering" sounds heard in line.
- Refuel IV spring, 1990—no damage found. ISI VT–3 exam performed.
- Refuel V fall, 1991—no damage found.
- Refuel VI spring, 1993—no damage found.
- Refuel VII fall, 1994—no damage found.
- Refuel IX fall, 1997—no problems found. ISI VT–3 exam performed with stud PT.

Min-wall corrosion inspection of valve body material:

- Refuel III fall, 1988—min-wall 1.400-inches.
- Refuel VI spring, 1993—min-wall 1.300-inches.

Valve B maintenance history

Disassembly and inspection for wear of internal components:

- Refuel VIII spring, 1996—no damage or abnormal wear found.

Min-wall Inspection for corrosion of valve body material:

- Refuel III fall, 1988—min-wall 1.35-inches.

Valve C maintenance history

Disassembly and inspection for wear of internal components:

- Refuel VII fall, 1994–no operational damage found. ISI VT–3 performed. A gouge on seating surface bonnet gasket area from manufacture or installation identified.

Min-wall Inspection for corrosion of valve body material

- Refuel III fall, 1988—min-wall 1.250-inches.

Valve D maintenance history

Disassembly and inspection for wear of internal components:

- Refuel I fall, 1986—no damage found.
- Refuel II fall, 1987—badly worn, replaced internals. Work performed because "chattering" sounds heard in line.
- Refuel III fall, 1988—no damage found. ISI VT–3 exam performed
- Refuel IV spring, 1990—no damage found. ISI VT–3 exam performed.
- Refuel VI spring, 1993—no damage found. ISI VT–3 exam performed.
- Refuel VIII spring, 1996—no damage found.

Min-wall inspection for valve body material corrosion:

- Refuel III fall, 1988—min-wall 1.42-inches.

Check valve reliability improvement program data

Valves A and D are disassembled and inspected every other refueling outage. Non-intrusive testing has been performed every refueling outage. Additionally, periodic UT inspection is performed to evaluate wear approaching the min-wall thickness.

Test and maintenance analysis

The test and maintenance history since Refuel II indicates a single group of four is appropriate. Knowing the cause of Refuel II degradation has helped confirm valves A and D no longer need to be disassembled and inspected. Refuel II disassembly was performed because an "audible noise" was heard during system operation. The Refuel II disassembly indicated badly worn parts. The Refuel II outage was an unusually long outage. Technical Specifications required maintenance of a higher RHR system flow rate. Technical Specification RHR system flow rate requirement has been lowered. Subsequent disassembly activities during the following several outages has not indicated significant wear. Thus, the requirement for frequent disassembly should be relaxed or eliminated. Audible tapping has also not been heard since this change occurred, indicating an absence of disc tapping. Acoustic monitoring identified no tapping at a magnitude that would cause degradation. These four valves perform the same function. Erosion and bearing block wear rates may vary from loop to loop, but these rates can be trended with different intervals as required. The open flow test data and leak rate test data is acceptable with no trends toward degradation.

Test strategy

Once failure modes and significance are assessed, activities that will potentially mitigate failure mechanisms, assess valve condition, or verify acceptable performance are identified. From all potential activities, a test and maintenance strategy is developed that uses some or all of those postulated. The basis for test strategy is:

- preventive maintenance activities required to maintain continued acceptable performance
- examination activities that periodically assess check valve condition
- test activities that periodically verify acceptable performance.

Example test strategy

From the previous analysis, a test strategy may be developed. The following is an example of an acceptable test strategy that may be applied to those valves included in the previous analysis example.

Technical Specifications leakage test strategy is proven in the nuclear industry to be effective in identifying the credible failure mode of excessive seat leakage. Test data is reviewed by Engineering to ensure there is not a trend toward degradation. An additional UT inspection is performed to evaluate wear rate and determine when the valve body is approaching the min-wall thickness on valves A and D. Test equipment technology and methodology has progressed significantly since the last min-wall measurement. Therefore, this measurement should give a more representative min-wall thickness indication than in the past.

This valve is currently full-flow tested. Based on the evaluation, measuring flow does not provide useful data for failure mitigation. If a check valve disc was stuck closed or restricted because of some unforeseen circumstance, abnormal system operation would be observed because of an unbalanced flow condition. Flow testing will use normal system operation to verify check valves are open. The following details the condition monitoring activities and their implementation frequencies:

- Leak rate testing/leak rate data trend—documented once each cycle.
- Flow testing/normal system operation, in-service operator walk-downs—documented once each cycle.
- Monitoring system pressure—monitoring RHR system pressure and the number of times during a fuel cycle operators vent the RHR system—system health reports evaluate quarterly, documented once each cycle.
- Visual gasket/boron crystallization or leaks; walk-downs performed every outage for RCS Class 1 ISI pressure testing requirements—documented once each cycle.
- Hours of operation/hours and flow can be estimated to assess bearing block or

hinge pin wear—documented once each cycle.

- UT results/min-wall, valve body wear rate—evaluation and actions determined by FAC program.

NRC limitations

The following limitations were taken from the 1999 revision of 10 CFR 50.55a(b)(3)(iv)(A–C):

(A) Valve opening and closing functions must be demonstrated when flow testing or examination methods (nonintrusive, or disassembly and inspection) are used;

(B) The initial interval for tests and associated examinations may not exceed two fuel cycles or 3 years, whichever is longer; any extension of this interval may not exceed one fuel cycle per extension with the maximum interval not to exceed 10 years; trending and evaluation of existing data must be used to reduce or extend the time interval between tests.

(C) If the Appendix II condition monitoring program is discontinued, then the requirements of ISTC 4.5.1 through 4.5.4 must be implemented.

Documentation

The basis of valve groupings, test and maintenance history analysis, failure modes and effects analysis, results of condition monitoring activities, and evaluation of corrective maintenance effects on all phases of analysis are areas that must be thoroughly documented to maintain an effective program.

Check valve condition monitoring effectiveness depends on accurate well-documented records because it is a living program. Following data collection, analysis forming the basis of activities must be re-evaluated. This is necessary to validate assumptions made when determining strategy for assuring continued component reliability.

Looking toward the future

A check valve condition monitoring program allows owners to shift resources from valves performing reliably to valves that are not performing reliably or have the potential to significantly impact plant safety. In the future, Wolf Creek will perform evaluations on all refueling outage and cold shutdown test frequency check valves to determine if they should be put into the condition monitoring program. Potential benefits are improved safety assurance, improved resource allocation, outage scope reduction and ALARA savings.

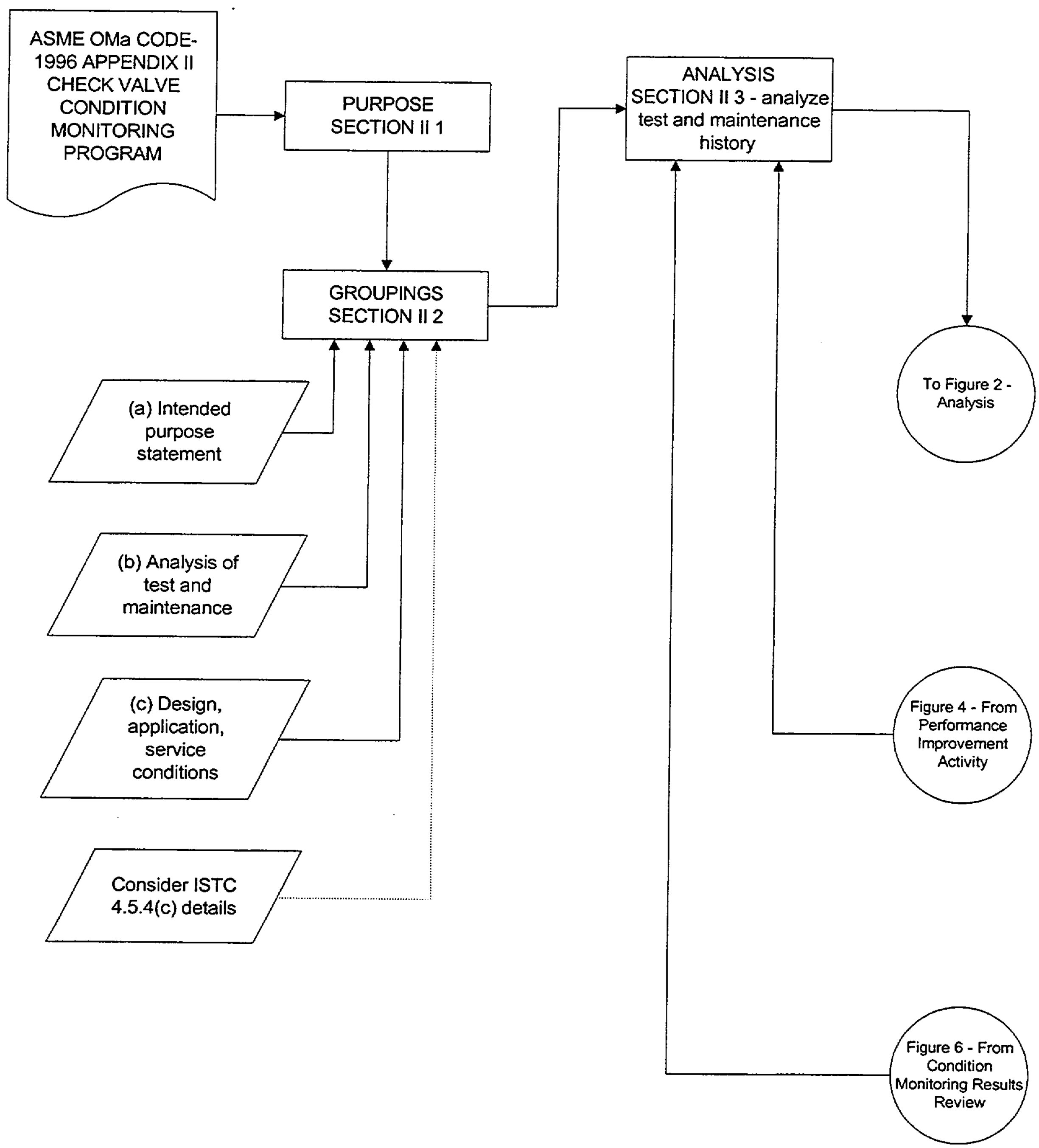

Figure 1 – Begin Condition Monitoring

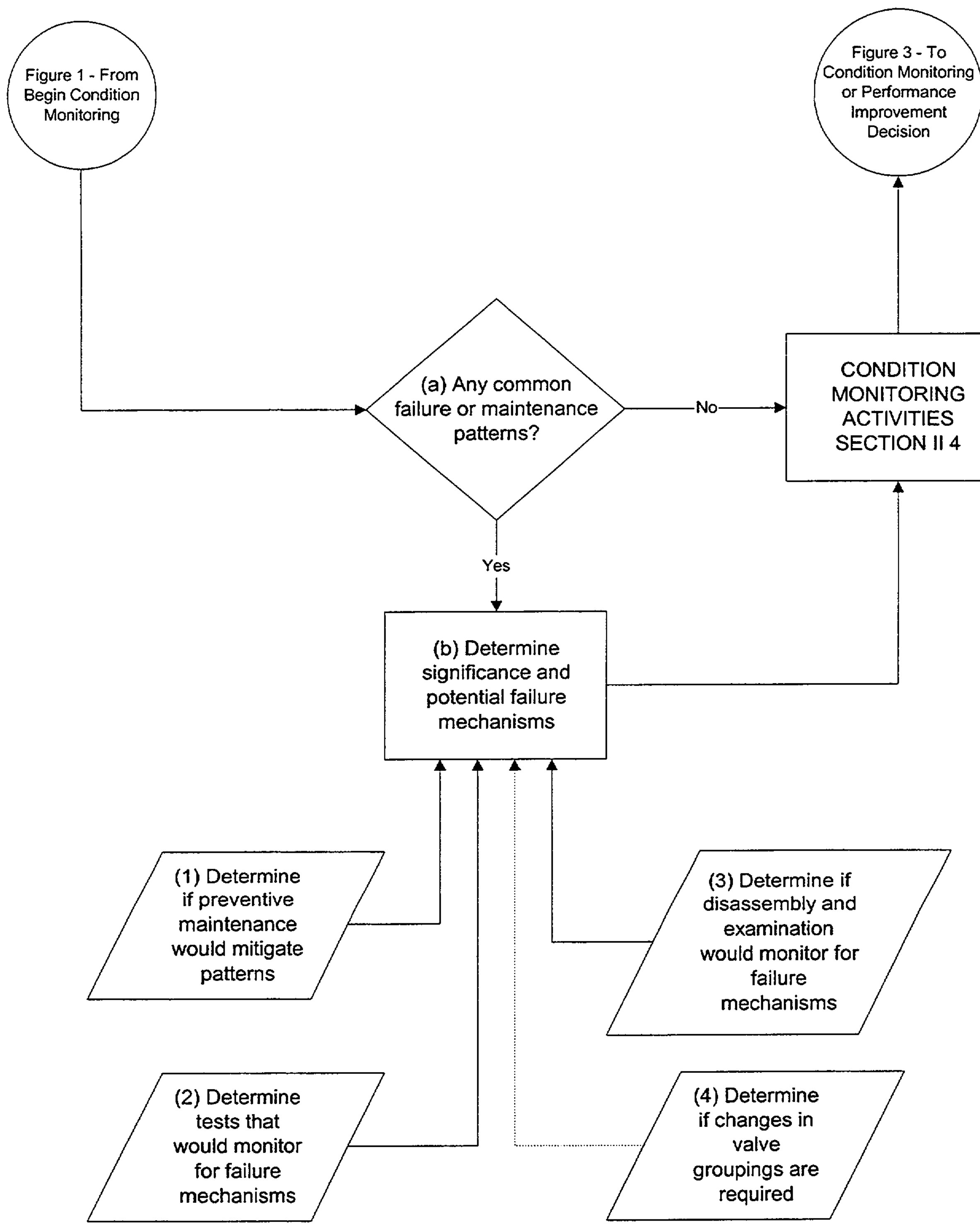

Figure 2 – Analysis

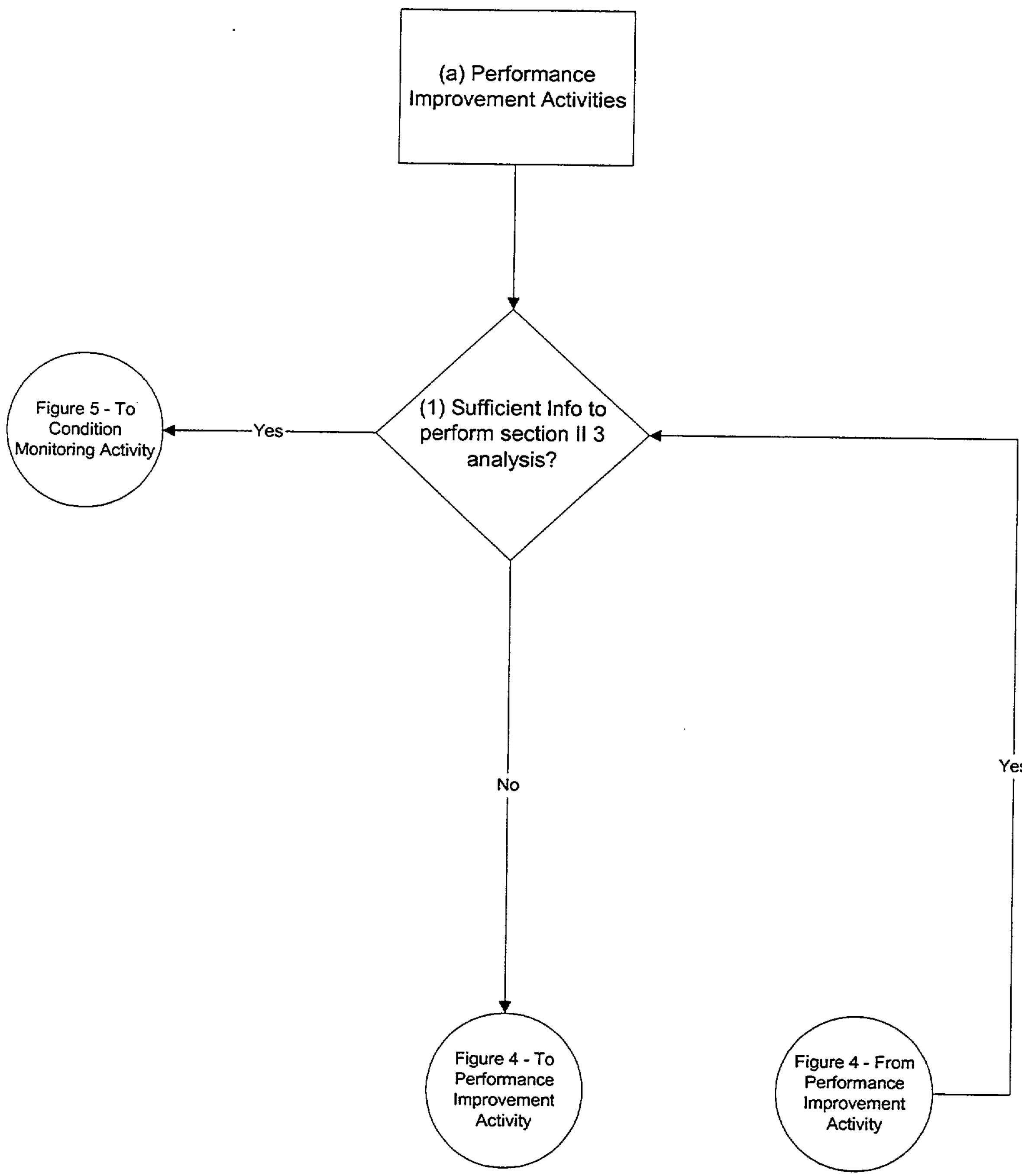

Figure 3 – Condition Monitoring or Performance Improvement Decision

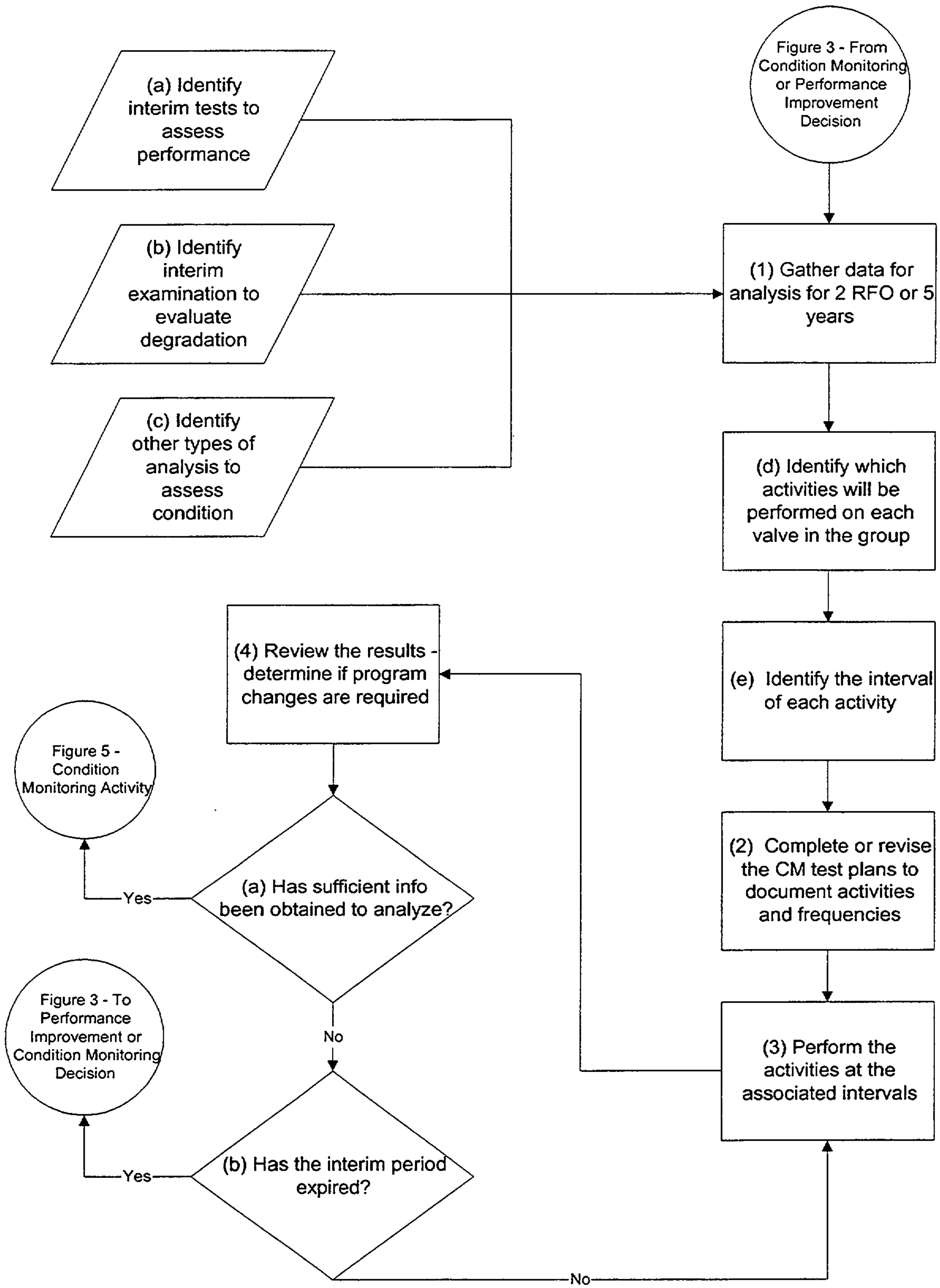

Figure 4 – Performance Improvement Activity Performace

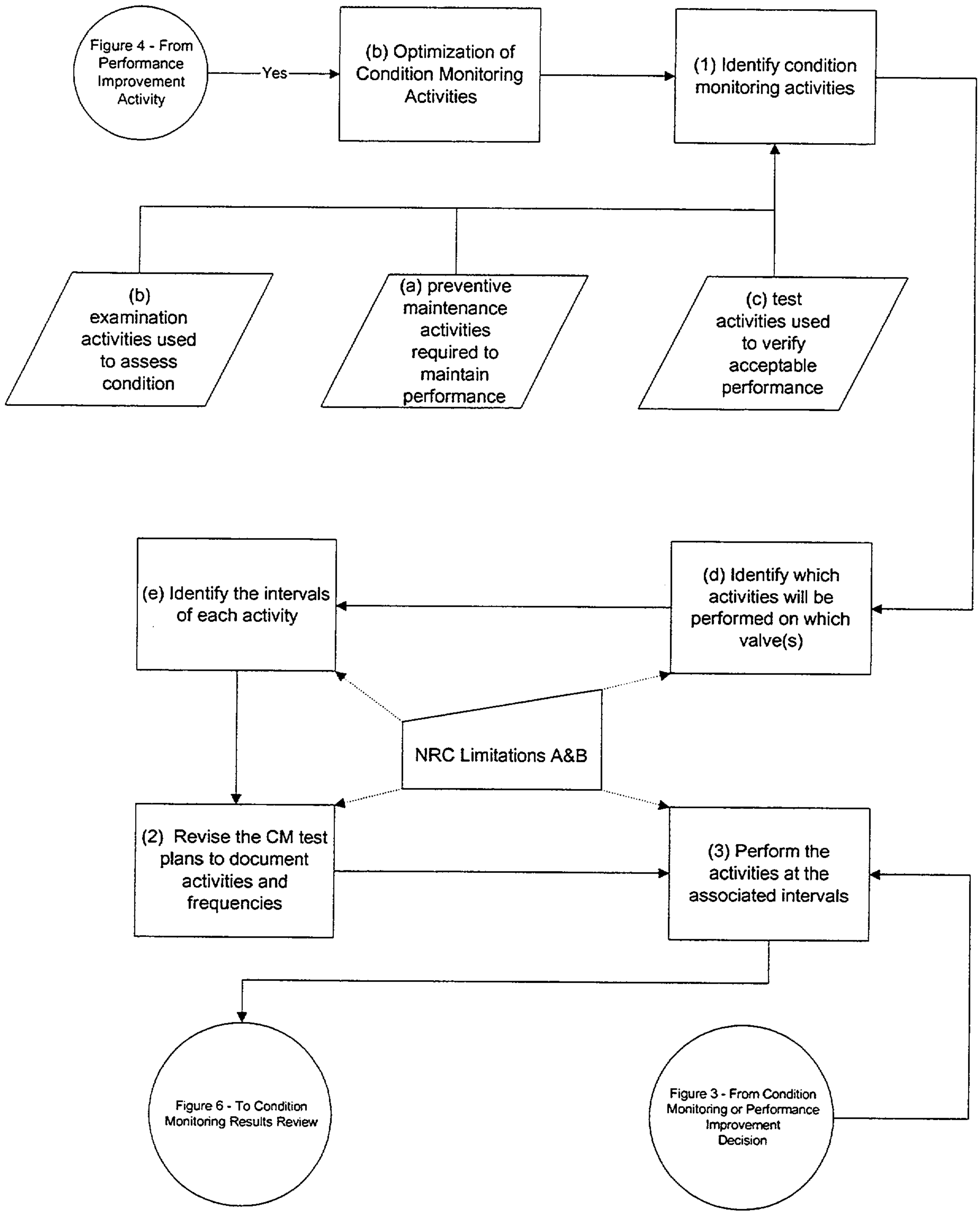

Figure 5 – Perform Condition Monitoring Activity

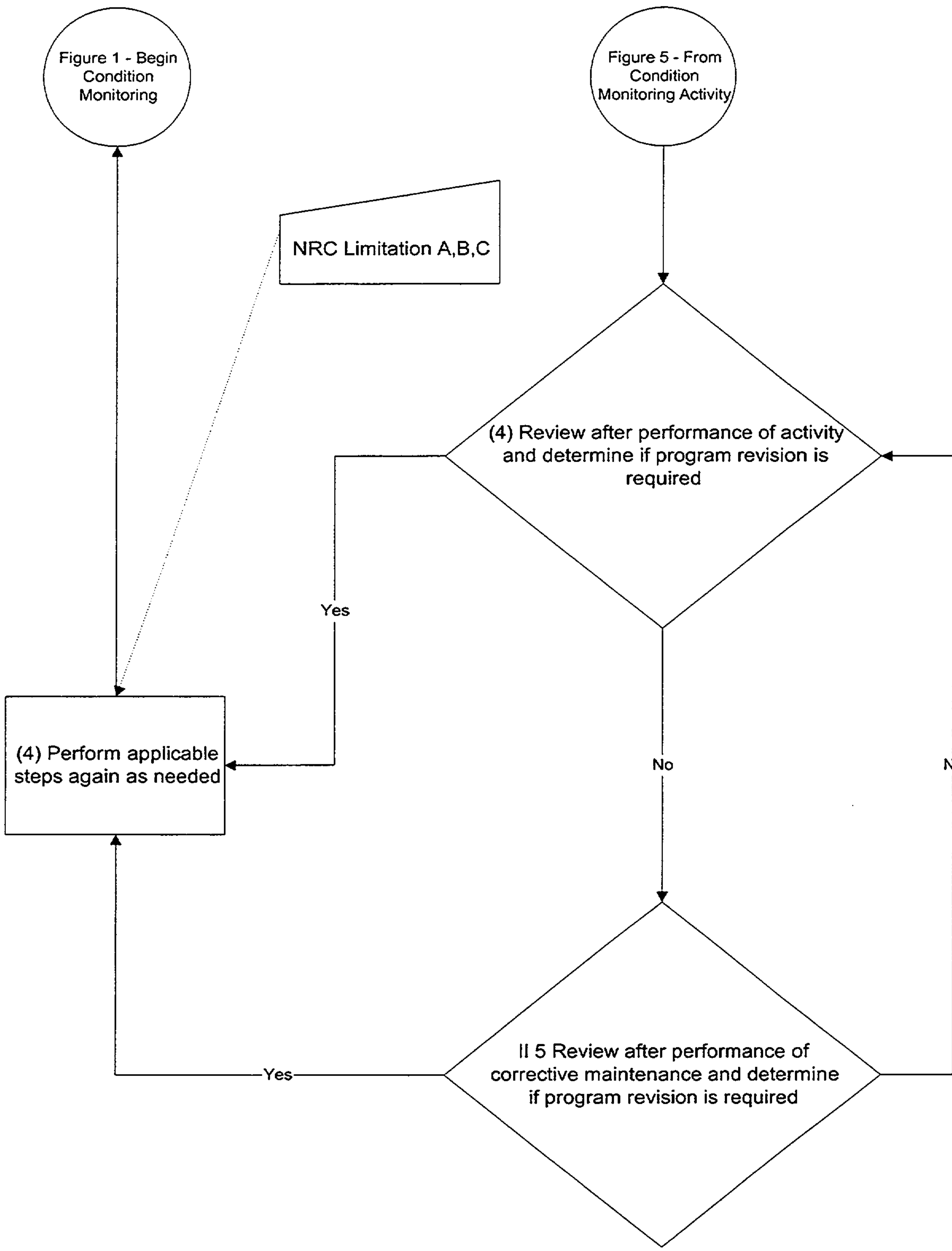

Figure 6 – Condition Monitoring Results Review

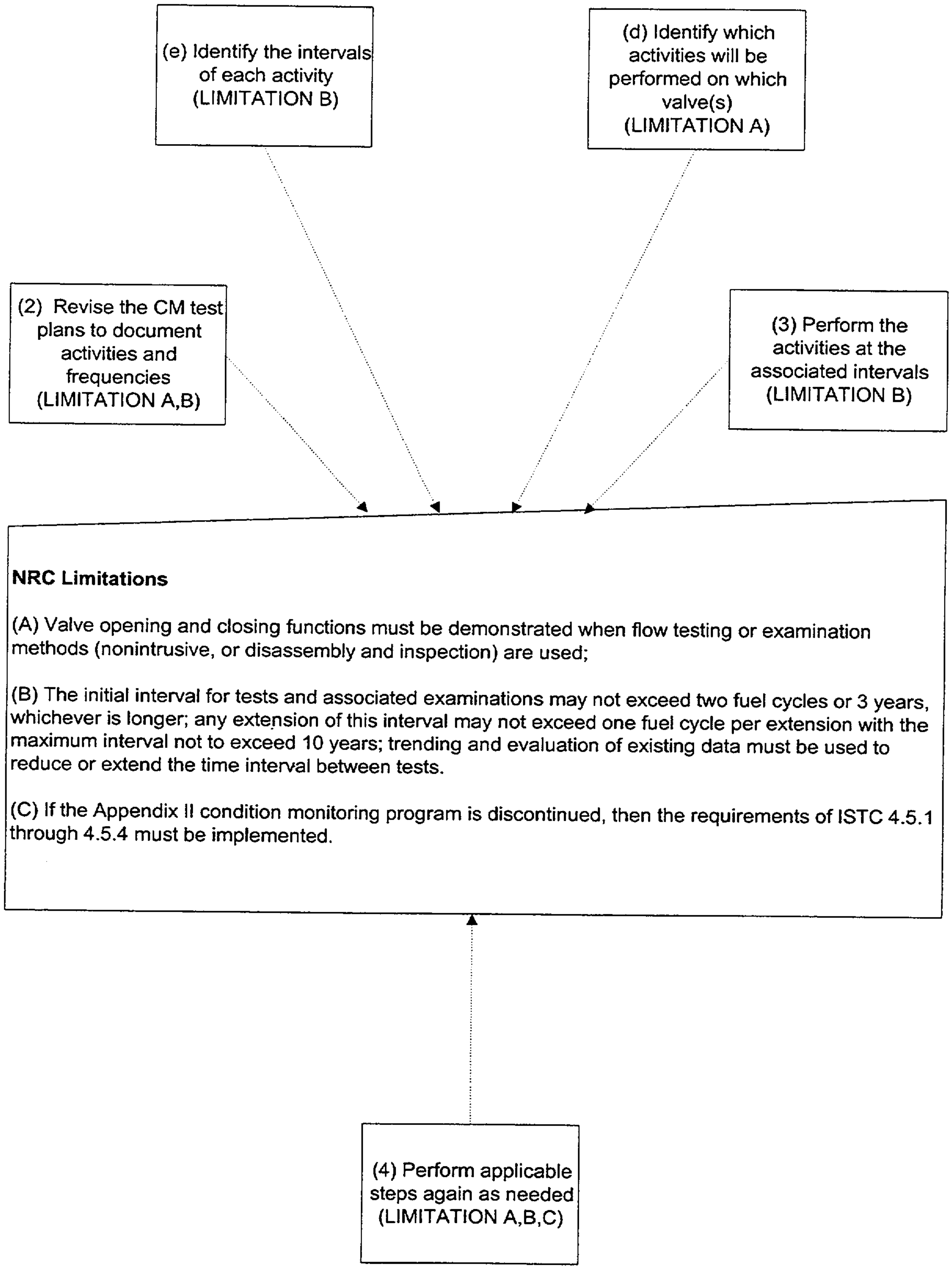

Figure 7 – NRC Limitations Summary

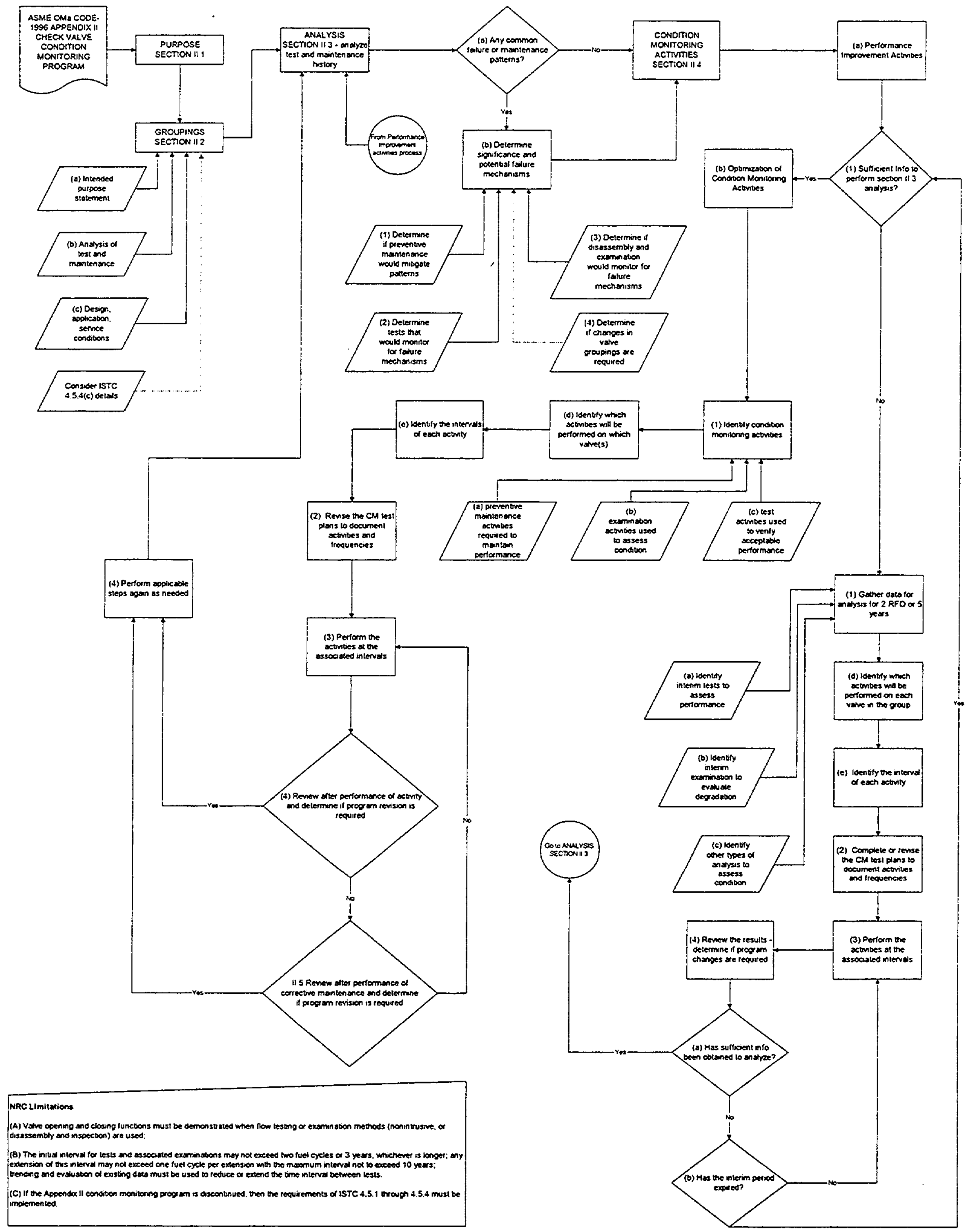

Table 8 – Process Summary

Development of Main Steam Safety Relief Valve for BWR Plants

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Electric Power Development Co., Ltd.

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Okano Valve Mfg. Co.

Abstract

Main Steam Safety and Relief Valves (MSSRVs) are situated inside primary containment vessel to prevent the overpressure in the transient case of BWR. An increase in the electrical output of BWRs results in an increased number of MSSRVs. Typically a 1350 MW-class ABWR power plant requires 18 MSSRVs. Such a large number of valves requires a lot of man-hours for working out the optimum equipment layout design and for maintenance. In addition, the loading of BWR core with mixed oxide fuel could lead to a further increase in the number of these valves.

In order to mitigate such difficulties, the authors have designed, manufactured, tested and confirmed the applicability of a MSSRV with a larger capacity than one available today.

1. Introduction

EPDC (Electric Power Development Co., Ltd.) is advancing the construction of the ABWR whose core will eventually be fully loaded with mixed oxide fuel (Full MOX–ABWR, FM–ABWR) in Aomori Prefecture, Japan. Hitachi, Ltd. is the supplier of the nuclear steam supply system.

The reactor design basically follows the existing ABWR design, but incorporates some modifications to provide for the future loading of high-burn-up MOX fuel bundles.

The modifications were made taking into account the characteristics of the full-MOX core and their resultant effect on the equipment. One of the features of the full-MOX core is an increase in the absolute value of the negative void coefficient. This could lead to an increased magnitude of overpressure in the primary coolant compared to the UO_2-fueled core in the event of overpressure transient events, such as generator load rejection with a failure of all turbine bypass valves to open. The severity of such an overpressure transient would now be mitigated and the number of MSSRVs would be reduced at the same time by the use of a MSSRV which the authors have newly developed. Its capacity is larger than that of valves currently used in the existing BWR plants by 16% (capacity volume 460t/h).

2. Design

MSSRVs are required to possess positive seal capability during normal operation of the plant and high reliability when required to actuate. To meet such requirements, spring-loaded safety valves are used. Spring-loaded safety valves offer rapid and reliable opening characteristics and an optimum configuration that assures a high discharge coefficient. Additionally, the spring-loaded safety valves used in Japan have to have high and stable operability against high back pressures. This is accomplished by the use of a wing disc and by a lip-shaped disc, which provides positive seal tightness.

For the large MSSRV, the authors have reviewed the available information on the above factors and decided to increase the capacity of the valve by enlarging the seat and throat areas, but maintaining the inlet and outlet diameters of the existing valve on which the development is based. Other considerations in the development include the use of the maximum diameter of spring (75mm) obtainable from a spring supplier and the maximum capacity of the valve manufacturer's test facility. Specifications of the large MSSRV are shown in Table 1.

Figure 1 shows the cross section of the large MSSRV. Similar to the existing design, it is composed of a single-coil spring and a wing disc, called a concave disk, with a bendable disk lip for seal tightness. Other materials such as the disc and stem are chosen from readily available Japanese Industrial Standard (JIS) materials rather than specialty materials currently employed.

3. Test Program

The test program consists of a component test whose objective is to derive basic data on the components of the valve and performance tests which confirm valve performance using a prototype.

3.1 Component test

The component test includes the bellows endurance test, spring characteristic test, and coefficient of discharge measurement test.

The bellows is essential in assuring stable operation by providing balanced back pressure characteristics and for achieving seal tightness. The test was carried out by simulating the temperature and valve opening speed of an existing MSSRV.

The spring constant was measured.

The coefficient of the discharge measurement test was required because the ratio of inlet diameter to throat diameter differs from existing valves.

3.2 Performance tests

The following performance tests were planned using a prototype large MSSRV. The scope of the tests was determined in reference to those conducted for the current MSSRV.

- Popping test;
- Performance test of relief valve function;
- Performance test as partof the automatic depressurization system;
- Blowing test;
- Performance test to obtain the relationships of popping pressure to nitrogen gas and saturated steam;
- Effect of hydrostatic test on popping pressure;
- Effect of valve transportation on popping pressure;
- Endurance test under actual thermal function; and
- Natural frequency measurement.

4. Test Results

4.1 Component test

(1) The bellows endurance test was conducted at temperatures expected for the MSSRV discharge line and at a speed of 1.5 m/s which is slightly faster than the actual valve opening. The test was done using three prototype bellows for 1600 cycles. Leak tightness was confirmed visually and by using a helium leak detector.

(2) The measured spring constant was in good agreement with the calculated value. The hardness in the radial direction was 47.8 HRC on the surface and 43.9 HRC in the center, which are within the values specified in the JIS (41.8−48.8 HRC).

(3) The coefficient of discharge (K value) measurement was performed at the National Board Testing Laboratory in the U.S., an ASME license holder, using 9 safety valves of a smaller capacity than the actual size prototype, but with the identical inlet to throat area ratio as the prototype.

From the test results shown in Table 2, the discharge coefficient of 0.97 was derived for the large MSSRV. This test method followed ASME section and JIS B8225. The K values for the flat disc type and wing disc type were measured. The results were as follows:

- K value for the wing disc was larger than K value for the flat disc.
- K value for the flat disc decreased linearly with the increase of the ratio of inlet area to throat area.
- It was supposed that steam flows moved smoothly in the seat area with the use of the wing disc and that the K value depended on the throat diameter minus a displacement thickness of a boundary layer on the inside surface of inlet nozzle. This thickness became smaller when flow acceleration was large (dependent upon the ratio of inlet area to throat area).

The test results are shown in Fig. 4.

4.2 Performance tests

Except for the blowing test, which was conducted at the Wyle laboratory in the U.S., all performance tests were carried out at the factory of the supplier of the prototype valve using the supplier's steam supply facility. Some test findings are described below.

4.2.1 Fundamental tests

(1) Popping test

The popping pressure and leakage through the seat were confirmed to be within the JIS specified values. Three other popping tests were done using nitrogen gas which showed that each popping pressure fell within ±1% of the mean value of the test, and the lift of MSSRV was larger than 29mm. The test results are shown in Table 3.

(2) Relief valve function test

The relief valve function test was carried out simulating conditions which would tend to increase the opening time of the valve. The opening time was found to be less than 0.2 second which was assumed in the safety analysis of the plant. In a performance test of an automatic depressurization system, a 5-cycle continuous opening test was carried out using the actual size accumulator.

(3) Blowing test

Popping characteristic under the full flow condition was checked using the large scale test facility (Wyle Laboratory).The test was done for different popping pressures with each test repeated three times for different back pressures. The test results are given in Figures 2 and 3. The popping pressure remained stable and a clear popping action was confirmed. Blow down during the test was stable. A change in the back pressure between 250–550 psig caused blow down to fluctuate between 10–2%, a value comparable to that which occurs for the existing valves.

4.2.2 Performance test under service conditions

(1) The relationships of popping pressure to nitrogen gas and to saturated steam

Once nuclear plants go into service, the popping pressure of the MSSRV needs to be checked using nitrogen gas rather than steam. To provide for this, the test was performed using both saturated steam and nitrogen gas for three different popping pressures. A correlation factor of 1.033 was obtained, a value very close to that for existing valves. The test results are shown in Table 4.

(2) Effect of hydrostatic test on popping pressure

When a hydrostatic test is required after the plant has entered into service, MSSRVs are gagged during the test. The popping pressure test done before and after the hydrostatic test conditions showed that gagging had no effect on popping pressure.

(3) Endurance test under actual thermal conditions

Endurance test was carried out for 300 opening cycles by using the actuator under actual thermal conditions. The popping pressure, seat leakage and opening time were checked at every 60 cycles of operation. No damage was identified. The test results are shown in Table 5.

(4) Natural frequency measurement

Natural frequency of the large MSSRV was 33–35Hz which was out of the band generated by resonance activity of an earthquake (more than 20Hz). This meant that the large MSSRV was rigid and did not resonate with the vibrations of an earthquake. The both the current MSSRV and the original MSSRV (used by 1980) were confirmed to have operability under high acceleration (9–11G) during the seismic test (see Table 6).

So, the large MSSRV was judged operable under the earthquake.

5. Conclusion

The authors have successfully developed an MSSRV with greater capacity than those currently used, and demonstrated its performance to be very close to current valves. The application of the large MSSRV will allow the decrease in number of MSSRVs required for the FM–ABWR and the man-hours for optimum equipment layout design and for maintenance.

Although the large MSSRV is intended for application to the FM–ABWR, it can be equally used in any BWR.

Table 1 Specifications of MSSRV

Size	Inlet 6B × Outlet 10B
Operating pressure	7.2MPa [gage]
Design pressure	8.6MPa [gage]
Design temperature	302 °C
Set point pressure	7.9–8.2MPa [gage] (safety valve)
	7.5–7.9MPa [gage] (relief valve)
Actuation time (opening)	under 0.2s (relief valve)
Capacity	460t/h

Table 2 Discharge coefficient measurement (wing disc type)

Size of safety valve	Inlet diameter	Throat diameter	Popping pressure	Discharge coefficient
50A	43.1mm	34.0mm	1.27MPa [gage]	0.9631
			1.03MPa [gage]	0.9584
			0.78MPa [gage]	0.9709
40A	38.0mm	30.0mm	1.47MPa [gage]	0.9776
			1.13MPa [gage]	0.9775
			0.78MPa [gage]	0.9836
32A	27.9mm	22.0mm	1.47MPa [gage]	0.9663
			1.13MPa [gage]	0.9639
			0.78MPa [gage]	0.9653
Average				0.970

Table 3 Popping pressure with steam			
Test Number	Popping pressure (MPa)	Lift(mm)	result
1	8.16	30.5	Good
2	8.16	30.5	Good
3	8.16	30.5	Good
Criteria	8.20±1%	≧29	–

Table 4 Correlation between nitrogen gas and saturated steam				
Set pressure (MPa [gage])		7.92	8.06	8.20
Popping pressure for N_2 (MPa [gage])	First	8.18	8.33	8.43
	Second	8.18	8.33	8.43
	Third	8.18	8.33	8.43
	Average	8.18	8.33	8.43
Popping pressure for steam (MPa [gage])	First	7.88	8.05	8.16
	Second	7.94	8.01	8.16
	Third	7.93	8.09	8.16
	Average	7.92	8.05	8.16
Ratio (Ave. N_2/Ave. St)		1.032	1.034	1.034

Table 5. Endurance test results (300cycles)			
Operation cycle	Popping pressure (MPa)	Seat leakage	Opening time
Criteria	8.20±1%	No leakage	≦0.2 s
Before the test	8.15 8.24 8.21	No	0.09 0.09 0.13
After 60 cycles	8.14 8.13 8.14	No	0.09 0.09 0.13
After 120 cycles	8.24 8.27 8.18	No	0.09 0.09 0.13
After 180 cycles	8.19 8.20 8.26	No	0.09 0.09 0.13
After 240 cycles	8.20 8.20 8.14	No	0.09 0.10 0.13
After 300 cycles	8.16 8.15 8.15	No	0.08 0.09 0.13

Table 6 The operability of the large MSSRV under the earthquake					
MSSRV type	Height (mm)	Weight (kg)	Gravity center (mm)	Natural frequency (Hz)	Confirmed operable acceleration(G)
Current MSSRV	1670	1620	740	40	9.6
Original MSSRV	1824	2150	842	31	11
Large MSSRV	1866	1910	779	33–35	(operable under the earthquake)

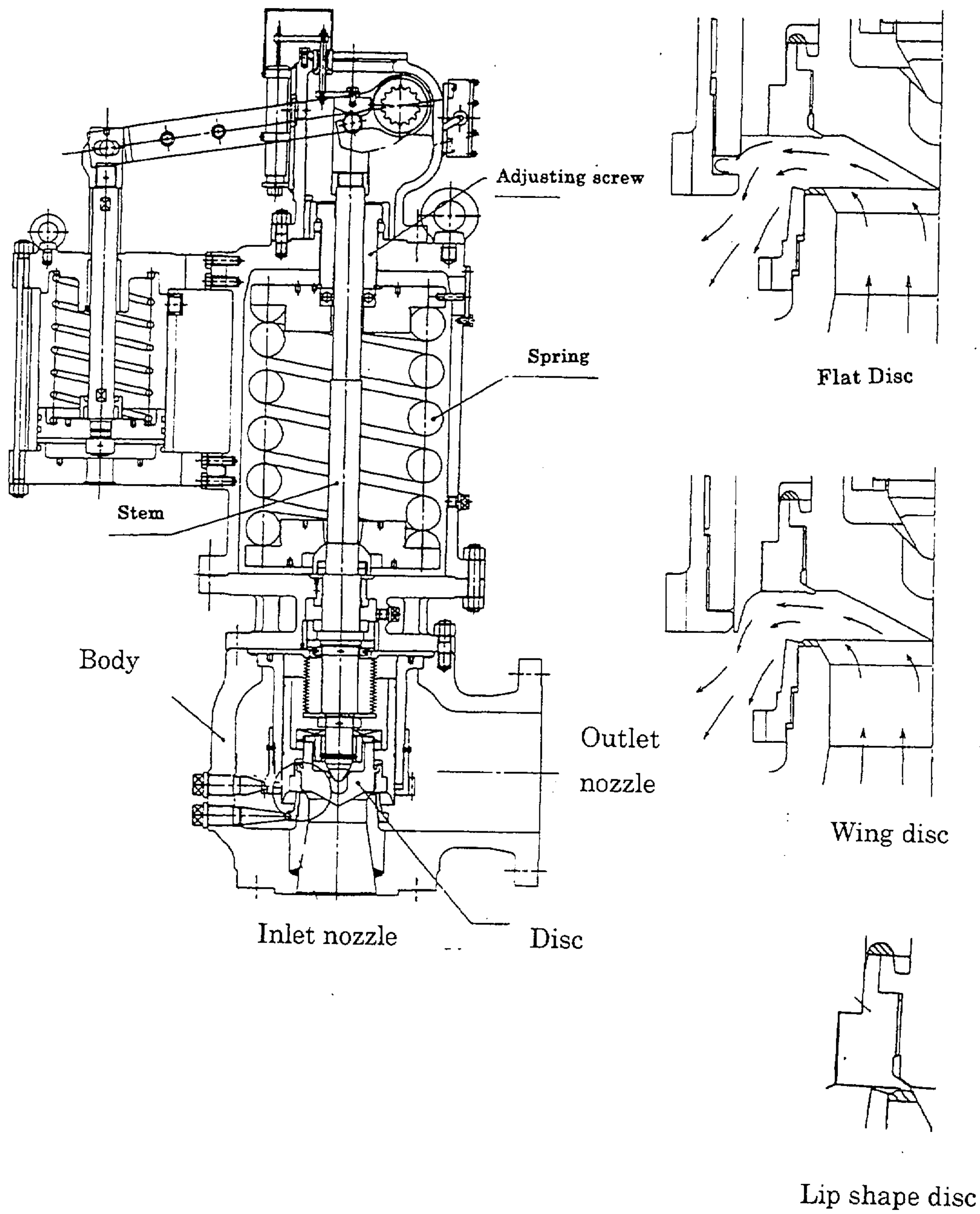

Figure 1. The Cross Section of the Large MSSRV.

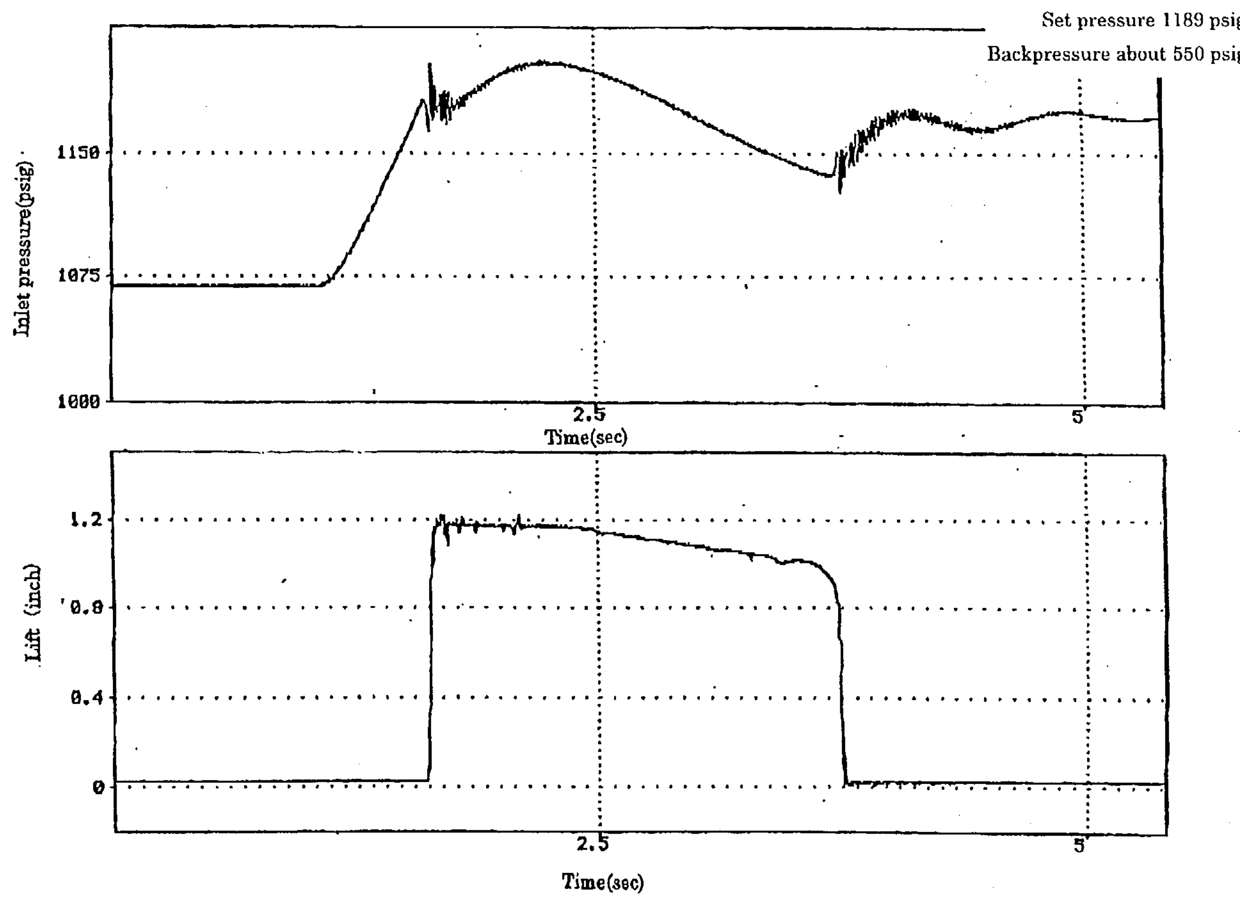

Figure 2. The blowing test result (one example)

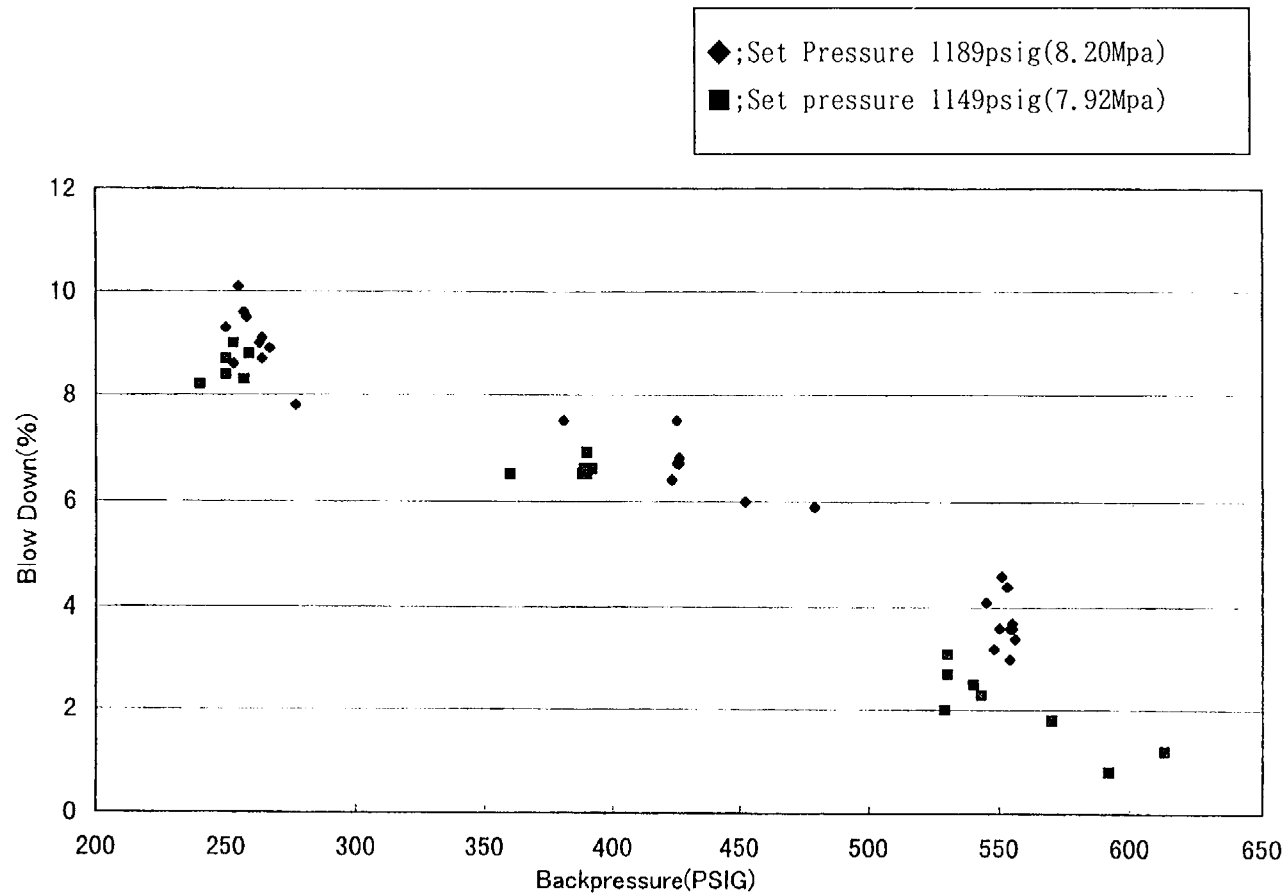

Figure 3. Blow down characteristics

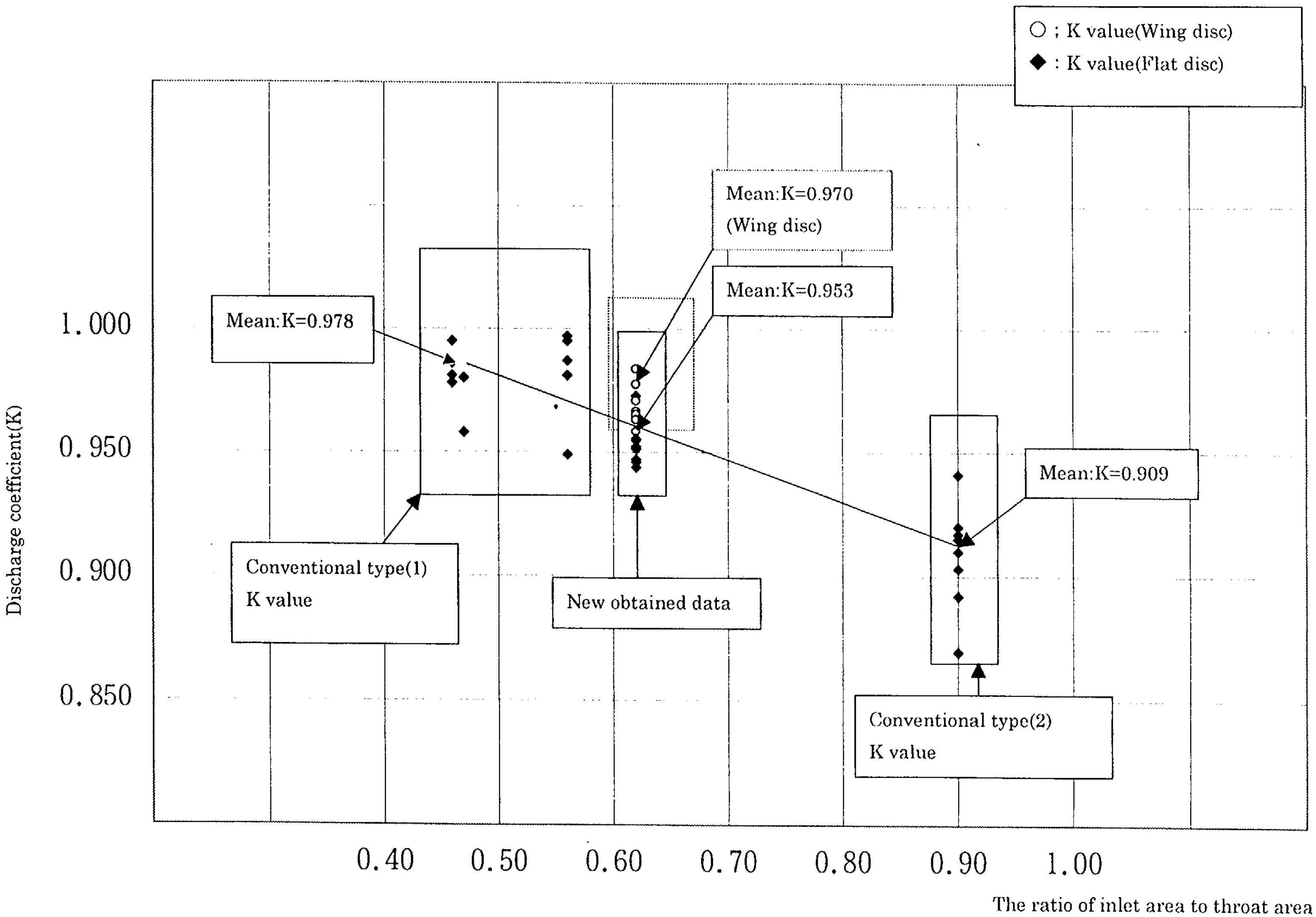

Figure 4. K value for Nozzle area/Inlet area ratio (Flat disc)

Using Non-Intrusive Testing to Eliminate Disassembly and Inspection of Check Valves

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Abstract

Early detection of conditions that accelerate wear in check valves and cause their eventual failure is critical to an effective predictive maintenance program for check valves. This was the initiative for the release of SOER 86–03. The SOER cited a lack of proper methods to determine check valve condition assessment through a structured approach. Conditions that affect proper operation of check valves include excessive disc flutter and undesirable internal impacting. Through non-intrusive check valve diagnostics, these conditions can be identified and trended to ensure proper scheduling of maintenance activities. Non-intrusive check valve diagnostics can also be employed to satisfy the requirements of ASME Section XI for "full stroke exercise" without disassembly and inspection. As a result of the issuance of NUREG 1482 and ASME WG O&M 22, Condition Monitoring Exercise (CME) which provides guidelines for application, these methods provide an approach to a significant cost-saving benefit through reduction in the labor and exposure normally associated with the disassembly and inspection of check valves.

This paper will explain an approach to check valve testing, using various Non-Intrusive Test (NIT) technologies. Acoustic emission (AE), ultrasonic (UT), and eddy current (EC) technologies will be presented in theory. This paper will attempt to provide a better understanding of how these technologies are used to detect check valve degradations through the interpretation of test data and provide comparisons between technologies. An overview of the most common check valve types will be presented. Finally a greater understanding of the principal drivers along with the need for proper training of test personnel for non-intrusive check valve testing in the domestic nuclear power industry will be addressed.

Introduction

Check valves are located in almost every safety and non-safety-related system in a nuclear power plant. These components are susceptible to failure modes generally associated with wear of internal parts. Failure of one of these valves during plant operation or in some cases, under cold shutdown conditions, could significantly affect plant safety. Failure can also result in costly and time consuming maintenance. Operation under conditions that cause the valve disc and hinge arm to oscillate or flutter can lead to degradations which, if uncorrected, may result in failure. A major cause of check valve failure has been excessive wear of the hinge pin, hinge pin bushings, and disc stud. In addition, disc tapping at the backstop or seat may lead to fracture of the disc stud and degradation to the seat.

In the past, two primary methods have been used in an attempt to prevent failure and reduce excessive wear. The first method is proper selection of check valve type. System fluid velocity is determined and a check valve with a calculated minimum flow velocity for stable operation less than the anticipated system flow is selected. However, experience has shown that the analytical or theoretical operating characteristics of check valves frequently do not match their actual operating characteristics. The second method involves a periodic disassembly and inspection program. Once operation has commenced, selected valves are disassembled and their internals inspected for signs of wear. This method is costly, time consuming and may result in unnecessary exposure to radiation. Frequently the disassembled valves show little or no signs of wear, which highlights the inherent inefficiency of an inspection program. In addition, repair and re-assembly of a check valve does not assure proper operation. In some cases it is more detrimental to the check valve. Neither of these methods verifies actual valve performance during normal operation. Check valve diagnostic equipment with powerful and flexible computer software programs, can observe valve performance under all flow velocities as well as measure and quantify valve instability. With proper application of the system, problem valves can be identified based on the valve's actual operating characteristics making it possible to implement design changes and maintenance prior to significant degradation. The system can also be applied to verify the full stroke exercise requirements of ASME Section XI.

Types of Check Valves

The check valve is a valve designed to control the direction of fluid flow in a pipe. Check valves differ considerably in their construction and operation from other types of valves. They are automatic in their operation, and are activated internally by the flow of fluids (either liquid or gases) which they regulate. Check valves permit the flow of fluid in only one direction. If the flow stops or tries to reverse direction, the check valve closes and prevents a back flow. As soon as the flow in the line is re-established, the check valve opens and flow is resumed in the proper direction as before. The five (5) most common check valve types are swing, tilt disc, piston or lift, duo disc, and nozzle.

Swing check valves are the design most commonly used in power plants. They are essentially the best general service check valves available. The flat seat and floating disc features make them the best sealing of all check valves. These same features also make them the easiest to maintain. Their basic straight through design means they impose the least pressure drop on a system. Swing check valves, however, are also the most susceptible to damage from water hammer and low or intermittent flow rates. They do not perform as well as tilt disc check valves (TDC) in high energy, reverse flow conditions. Subjecting a swing check valve to low or pulsating flow may cause severe damage. The most frequent result is damaged seats and disc sealing surfaces. However, it is not unusual for repeated hammering of these valves to result in fatigue failure of the hinge pin and thus the separation of the disc.

Tilting disc (TDC) check valves operate on the following principal: the disc is mounted on two pins just above the centerline of flow. This means that the disc almost pivots about its centerline. The smaller radius of rotation greatly reduces the velocity at the edge of the disc and thus the impact between the disc and seat. In most TDC designs a counterweight is also incorporated. The counterweight adds an additional closing movement and therefore increases the speed of closure. Additional closing speed may be achieved by designing the counterweight in the shape of an airfoil. The flow of the fluid over the airfoil produces a lift that keeps the disc open. When the forward flow ceases, the upward force on the disc disappears and the weight of the disc and counterweight immediately sends the disc to the seat. TDCs are designed and best suited for systems likely to experience water hammer. They are also appropriate when low or pulsating flow is encountered. Although TDCs do not become fully open until higher velocities are reached, the disc tends to remain stable through the entire range. The lift caused by the flow over the airfoil and the aerodynamic counterweight tend to keep the disc steady regardless of the fluid velocity.

Piston or **Lift** check valves are used to prevent flow reversal in piping systems. They are faster closing and handle variable flow conditions better than most types of check valves. Lift check valves are the best choice for high-pressure drop application and are more suitable for operation under less then full open. They mount well horizontally or vertically. Piston type lift check valves should not be used where solid contaminants are present because they could cause sticking. This could prevent proper closing. Frequent maintenance is required if installed near pumps, control valves, in pulsating applications and if left at 50% or less of full open.

Duo disc is a variation of the swing check valve in which two D-shaped pieces pivot about a vertical pin and seat on flat D-shaped seats. A torsion spring acts between the two flaps to help close the valve with low shock. The flaps are directly in the pipe center and are fully exposed to the fluid back flow force to assist closure if the spring action is to slow. Failure of the spring creates potentially serious operational problems. Without the restraining force of the spring, disc flutter and wear will increase significantly.

Nozzle check valves use a spring-loaded circular flat piston that is oriented perpendicular with flow. Since the disc is relatively light, it uses a spring to assist with the closing force. These valves are rapid closing. A low-pressure drop exists around this streamlined design. An annular ring machined into the body of the valves serves as a seating surface. The disc experiences large back seating forces that will allow it to be less sensitive to upstream disturbances.

Non-Intrusive Test Methodologies

Stable flow is a test conducted when the system is at full flow. Observation of a check valve in this state provides useful data about the performance while in service. It may also be performed at less than full flow in the case of minimum flow system operation. Conditions that accelerate wear can be detected in these modes. This allows for corrective action prior to any actual degradation of the internal components occurs. This test also allows for the detection of degraded parts as well as stuck or missing components.

Stroke test is a test conducted by initiating flow through the check valve and then isolating flow. This test allows the detection of free moving internals. It is capable of detecting degraded, missing and stuck components also.

Leak test is a test conducted when the plant is in a desired line-up. Through comparison of acoustic energy levels, leaking check valves can be detected. This approach to leak detection requires close control and monitoring of system and test parameters to ensure that the necessary precision is obtained to determine whether the valve is leaking.

A combination of these test methodologies is recommended to ensure all aspects of degradation detection are applied. Without the combination of both methodology approaches, some degradations may go undetected.

Description of Non-Intrusive Test Technologies

Check valve diagnostic equipment is designed to verify proper operation of check valves and provide a predictive maintenance tool that will identify adverse trends prior to failure. Another application is to determine the ability of the check valve to be exercised through the full stroke. The system relies on a "look" and "listen" approach by viewing the internal moveable components and monitoring the activity generated by these components. To accomplish this task, three different technologies are employed in various combinations: Acoustic Emission (AE), Ultrasonic (UT), and Eddy Current (EC). A combination of these technologies is a sound basis for performance of check valve diagnostic testing. Each technology has strengths and weaknesses, however when combined together they offer high degrees of certainty in proper diagnosis of check valve condition/position. The methodologies advocated in this paper are those with industry proven performance. These methods in a variety of combinations can be used to perform a full gamut of testing on essentially every design of check valve. This range of testing is sufficient to determine the overall condition as well as position of the disc without the need of disassembly. Check valve non-intrusive testing requires no disassembly to correlate the valve condition to the test results to be used a baseline for continued performance of diagnostics, however, this information would be valuable.

Hardware/Software Requirements

A typical check valve diagnostic testing platform consists of a signal conditioner that generates as well as accepts the test signals then processes them before sending out to the device that runs the software program. These devices are typically Pentium computers. The software programs are generally compatible with Windows versions of software allowing more interactions between programs. Data can be stored on permanent drives or a variety of backup methods such as diskette. Generally calibration of these components are required if testing will be performed on safety related plant equipment.

Acoustic Emission (AE)

The acoustic technology is considered a "listen" approach based on sound vibrations received from piezoelectric charge type accelerometers, which convert sound waves from mechanical motion to an electrical signal. Two of the main concerns of acoustic testing are base bending and temperature transients. The Delta Shear accelerometer design employs three (3) piezo elements which provides 18–33 dB better information than compression designs when compensating for base bending. In similar respect, the piezoelectric element has very good temperature transient characteristics.

One design example would be that of High Impedance. Employing a charge amplifier assists in testing check valves at temperatures above 250°F. This type of circuit senses charge and is not affected by cable length. However it is very sensitive to spurious electrical noise in the cable. Therefore the cable needs to be high quality, low noise, high leakage resistant and left undisturbed during testing. If not immobilized, some low frequency noise will be generated. High pass filters could reduce the noise but at the expense of valuable low frequency information. Residual noise increases proportional to cable length but the signal level remains unaffected. Signal to noise decreases as cable length increases.

Significant amounts of research have been invested to determine what reliable parameters can be extracted from the vibrations emitted from a check valve. Acoustics cannot detect a check valve that is fluttering unless impacts are occurring internal to the valve body. Check valve internal components must contact each other to create the vibrations necessary for the accelerometers to sense metal to metal type data. Vibrations with energy levels below those generated from background noise and flow noise would require the use of software filters to differentiate these sources.

This technology is limited when the check valve does not produce events that can be recognized. It is also limited to detection of such failures as a missing disc if a baseline test has not been performed with the disc intact. An acoustics only approach to data collection will greatly reduce the ability to ascertain valve condition/position during the analysis of data. This approach can lead to incorrect conclusions concerning the operability of a check valve. Acoustics can be used with another technology such as ultrasonic (UT) or eddy current (EC) to provide a picture of a check valve in transient.

An important part of acoustic emissions testing is preparation. Once the desired accelerometer mounting location has been determined (a valve drawing is recommended), the area should be cleaned of dirt, loose paint, rust, and any oily film that may interfere with a good mount. The backstop and hinge-pin provide optimum monitoring locations. Once the preparation is complete, the stud can be mounted with epoxy or other approved adhesive and allowed to dry. Some mounting studs are coated to allow proper electrical grounding. Degradation of the coating may produce a signal with undesirable noise levels and interfere with the test results. Accelerometers should be applied only until finger tightened.

Acoustics is very easy to use and usually totally automated with the advanced software programs available today. The data can be automatically compared to previous results for trending valve condition. Automated analysis features of software allow the user to trend check valve performance based on impact RMS levels, long-term energy levels, frequency of impacts and impact rate. Check valve degradation could be the result of changes in these areas from previous data. Examples of this are increased impact magnitudes, increased impact rate, and a shift in the impact frequency domain. The concept is that as parts degrade and clearances increase, the energy will also increase. Software programs capable of performing Fast Fourier Transforms (FFT) determine impact frequency. The FFT is a mathematical manipulation of data acquired in the time domain and converted to the frequency domain by the use of an algorithm. The FFT is very limited in providing useful information concerning changes in check valve condition unless the check valve internal components have been qualified in respect to their nominal frequencies. Another approach is that of a Power Spectral Density (PSD). A PSD is the process of averaging multiple FFT data to provide an average frequency spectrum for a specific impact rate. The PSD provides an overall higher statistical accuracy when comparing data. The concept is that as check valve condition degrades, a shift in the PSD will occur.

Ultrasonic (UT)

Ultrasonic allows the user to determine disc position and disc stability. A modified flaw detection reflectoscope and associated transducers are used to extract necessary information from the check valve disc. The reflectoscope is a pulse/echo type that generates high voltage pulses and sends them to the transducer. The transducer then converts the pulses into sound waves that are applied to the valve being tested. A large percent 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 CRT as waveforms.

The A-scan display indicates the depth and amplitude of the sound reflections. The amplitude is a relative measure of the amount of reflective ultrasonic energy. The depth is the distance from the transducer to the target or disc assembly.

Ultrasonic sound waves can travel through many different mediums. The particle density of a material will determine how well the sound waves travel through it. This is called the velocity of sound for a given material. "Velocity of sound" can be defined as the distance a sound wave will travel through a medium in a certain amount of time.

It is the interface of the metal to air that attenuates the UT signal. Ultrasonic energy is totally lost as a result of this interface. For this reason coupling must be used to eliminate air between transducer and valve body contact surfaces. This relationship prevents the use of ultrasonic on air and steam systems. Additionally the grain structure of some stainless steels attenuate UT signals causing a significant reduction in return signal making it difficult to test. Like other non-destructive testing technologies, ultrasonic requires proper training so the user can be proficient with the technology.

UT can be considered a "look" technology in regard to its ability to monitor disc movement. UT is the most accurate and quantifiable data that can be gathered from check valve testing. This ability allows for determining the valve disc open angle in degrees off the seat and the velocity of movement or "angular velocity." Inputting key data into the software program, which in turn performs the trigonometric functions that result in the disc open angle and angular velocity outputs, performs these calculations. The disc open angle can be calculated to within plus or minus 4 degrees and provide verification of full open without the presence of an acoustic impact. The angular velocity of the disc, which is measured in degrees per second movement, is the best indicator of valve performance. The primary cause of check valve degradation is continuous disc flutter. Accelerated degradation of check valve internals can lead to their failure to perform when called upon. This angular velocity of the moving disc assembly is correlated to a "stability number" classification. This classification determines the operating characteristics of the check valve in terms of "stable," "unstable," or "excessive."

The stability number represents the amount of disc flutter that check valves disc is displaying at a specified flow rate. The stability number combines the magnitude and frequency of disc flutter into one term. It is more representative of actual disc flutter than stating average magnitude (degrees) or frequency (Hz) separately since check valves oscillate with random motion. All valves with disc flutter will experience some wear. It is the purpose of the stability number classification to distinguish between the conditions which lead to accelerated wear and those that result in normal wear. Normal wear is considered to occur over many years of service with some check valves remaining functional without replacement of parts. Excessive or accelerated wear is considered to be when internal part replacements are required in less than three (3) full operational periods between refuel outages.

The stable range represents a check valve disc that is either firmly against the backstop or experiencing ordinary flow induced oscillations. This category of check valves will experience only normal (low) internal wear. The "unstable" range indicates disc flutter that is 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. "Excessive" disc flutter represents valves that are experiencing abnormal disc oscillations. 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 in this category. The higher the angular velocity, the greater the internal wear will be on the rubbing surfaces. This information is extremely useful to select valves for inspection or to predict maintenance activities on a particular check valve. Ultrasonic has the ability to determine a stuck or missing disc, relative stability or angular velocity which is directly associated to wear of the hinge pin, as well as other conditions that cause wear. A wear model is available to predict valve life and wear rates based on the data gathered with ultrasonic testing. Presently this is the most advanced form of predictive maintenance available in the field of non-intrusive testing. UT can ascertain the potential of degradation occurring without the trending or comparison of test data but from one single test data.

UT has been recognized throughout the industry for many years in the determination of critical testing and flaw sizing for service sensitive welds and heat affected zone of piping systems as well as the RPV. It has been and continues to be an integral part of ASME Section V and Section XI. The application of UT for check valve diagnostics redirects the technology for the purpose of check valve condition determination. The same quantifiable qualities are utilized in check valve diagnostic testing. The limitations of UT are that it will not operate in steam or air systems. These limitations usually affect only a limited number of check valves, so that this is not considered significantly limiting.

Eddy Current (EC)

Another "look" technology is eddy current. The eddy current technology generates a temporary electrical field that penetrates non-ferrous materials. Coils of wire are energized with current to create a magnetic field that is pulsed to create transients in the metal. The strength of the field is determined by the ampere—turns of the coil. The movement of the internal components of the check valve disrupts these transients and is sensed with receiving coils to produce a voltage corresponding to internal component movement. The field disturbance can be used to identify a fluttering check valve, however cannot identify the exact mid position of the moveable element. The eddy current test system consists of an enclosed module and probes that produce a voltage-varying signal. The analog output of this signal is processed by the software package. The processed signal is displayed as relative disc position and flutter. When the check valve remains in a stationary position, a voltage is recorded. Changes in the output voltage occur as the check valve disc position is altered. Therefore as the check valve continues to change its position, the voltage continues to vary depending on the direction of disc movement.

Eddy current is non-linear and only provides relative disc position unless a full stroke characterization calibration of a specific check valve has been performed. With the characterization calibration stroke, eddy current can be used to determine if full stroke travel has been achieved by comparison of the full stroke deltas from the calibration as well as the subsequent test data. This characterization stroke can be achieved by manually stroking the valve, or having sufficient flow to identify an acoustic backstop impact. This technology is most commonly used to verify operation of a check valve with regards to full stroke exercise by combining this technology with acoustic emission (AE) by acquiring a time trace while the valve is being cycled.

The advantages of eddy current are the ease of installation and the intuitive traces which are generated. These traces provide a picture of total disc travel. Eddy current can be applied in any medium including air and steam systems. Eddy current works well on high-pressure class stainless steel valves and fast-stroking valves where ultrasonic is limited. Since the eddy current technique itself locates the disc as it changes position and no electronic interface is required to capture the signature, this technology presents itself as the premier method for the testing of these rapid stroking check valves. The eddy current technology is limited to non-ferrous materials (stainless steel) and cannot be used to quantifiably determine disc stability or opening angle.

UT vs D.C. Magnetics for Check Valve Position Determination

Presently the most prominent application of check valve diagnostics is to satisfy ASME Section XI requirements for full stroke exercise. There has been a difference of opinion as to what technology (UT or D.C. magnetics) was best suited to be applied for this determination. A significant amount of research was made in evaluating the use of ultrasonic and magnetic (AC, DC, EC) technologies for determining check valve disc position. UT provides the user with a linear quantifiable means to determine disc position. UT can be applied in a similar means to observe disc travel or in a stand alone approach where test conditions do not produce an acoustic impact. It may also be used to validate the authenticity of the impact event in the case of questions. UT is limited to liquid filled systems. The UT technology is validated to meet 10CFR50 Appendix B criteria. While an additional level of training may be required to understand and implement ultrasonic techniques, the benefits and quantifiable attributes clearly outweigh any additional training requirements. While each technology has certain desirable features and benefits, the ability of ultrasonic to provide quantifiable data is the key attribute.

Conclusion

When applying non-intrusive testing to check valves to avoid disassembly and inspection, several factors are apparent. Disassembly poses many risks in that the re-assembly of the check valve could result in a failure of the valve to properly operate. The disassembly and inspection process has a history of initiating problems to the check valve that would not result from diagnostics. Many plants have experienced broken parts as a result of this activity. Disassembly also produces higher levels of personnel exposure and contamination incidents. Diagnostic sensors can be permanently mounted to further reduce dose accumulation. With disassembly, there is the increased risk of foreign material being introduced into the system. Non-intrusive testing provides a proven alternative to disassembly for the purpose of inspection and testing and eliminates the risks associated with disassembly. The observation of a check valve during normal operating conditions provides valuable information that can be applied for corrective action purposes whereas a visual observation in a static condition cannot always indicate the root cause of wear. Check valve diagnostics utilize proven methods with both lab validation, but more importantly, field verification of techniques that work.

Each technology has strengths and weaknesses. Each can be utilized to provide a complete picture of check valve operability when used in combination with each other to fit each specific situation. The user to avoid misinterpretation of the data must realize the limitations of each technology. Proper training and continued use of the equipment is essential to ensure that users understand all aspects of applying the equipment and interpretation of the results. Misapplication of the technologies as well as improper analysis of the test results will cause confidence in the abilities of check valve diagnostics to waver. When applied and evaluated correctly, check valve diagnostics provide a superior alternative to disassembly and inspection. As non-intrusive test (NIT) data becomes more widely accepted in our industry to satisfy Federal Law, the quality of data gathered is extremely critical. When applied correctly, these technologies are the foundation of a check valve program that exhibits both superior quality and economic value. A consistent result from non-intrusive testing throughout the industry is a significant contribution to the lowering of O&M costs.

The following table summarizes the main advantages and disadvantages for all three technologies discussed.

Technology	Advantages	Disadvantages
Ultrasonic (UT)	Quantifies disc position and disc stability; can be used alone to conclude valve condition	Can only be used in liquid mediums; requires elevated user proficiency
Acoustics (AE)	Results are easily trended to determine degradations; easy to install and acquire data	Cannot be used alone for testing to determine a complete understanding of valve condition
Eddy Current (EC)	Easy to install; can be used in any medium; capable of monitoring entire valve stroke; data repeatable with valve specific stroke characterization; best for rapid stroking valves	Cannot quantify disc flutter or disc open angle; limited to stainless steel

Coupled Fluid-Structure Analysis of a Tilting Disc Check Valve Using Overset Chimera Grids and 6DOF Modeling

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Abstract

An effective CFD methodology is demonstrated to enable simulations such as flow around a tilting disc valve inside a pipe. A chimera overset grid methodology is used for these analyses, in which separate grids are created for the valve and pipe. These grids are then overset on top of each other and flow solution is computed independently in each grid and with the necessary information interpolated between them. This approach is very well suited for this type of analysis, as the grid-remeshing problem, resulting from the valve rotation, is eliminated. The overset grid methodology in conjunction with 6DOF is shown to be very suitable for modeling such internal flow problems with moving bodies.

To demonstrate the efficacy of the suggested approach, a coupled fluid-structure analysis was performed on a tilting disc check valve. The objective of the work was to determine the steady-state valve opening position and pressure drop for various flow rates, and also to estimate the flowrate at which the onset of valve flutter may occur.

The work was performed using MDICE and CDF−ACE+, CFDRC's general-purpose multi-disciplinary software package. This software contains several interconnected modules, which can be implicitly coupled for multidisciplinary simulations. The work described here was performed using the flow and 6DOF rigid body motion modules.

Problem Description

Figure 1 shows a schematic of the problem; it consists of water flowing through a circular pipe past a tilting disc check valve. The valve is closed under the action of gravity, and opened by the pressure difference generated by the flow. The valve also contains a packing material located on the valve disc pin, which provides additional resistance to the valve motion. Unlike the gravity and pressure forces, though, the direction of the packing resistance force changes in time to always resist the valve motion.

Twelve steady-state cases were performed, at flow rates ranging from 100 to 14,000 gallons per minute (gpm). At each flow rate, the valve opening position and pressure drop were determined. In addition, the flow rate for the onset of valve flutter (repeated opening and closing at low flow rates) was estimated by bracketing the flowrate at which the valve first opened.

Geometry Modelling

The pipe geometry is trivial in the regions away from the valve. The diameter of the pipe is 24". The diameter of the valve disc is 28". Hence, in the vicinity of the valve, modifications had to be made to the circular cross section to allow for rotation of the valve. To accommodate this, the pipe cross section was changed to an elliptical shape, in which the valve is allowed to rotate with a very small clearance to the wall. The resulting pipe geometry is shown in Figure 2.

The valve geometry was built in AutoCAD from drawings [1]. The resulting geometry model of the valve is given in Figure 3. This valve model was used for calculation of mass properties (computed in AutoCAD). The pipe and the valve geometries shown in Figure 2 and Figure 3 were used as the starting point for generating the CFD grid.

Grid Generation

The analysis was performed using an overset grid approach (also referred to as Chimera technique), in which separate grids are created for the pipe and the valve disc. The grids are then overlaid on top of each other. Flow solution are computed independently in each grid and then interpolated between the two to enable each grid to perceive the effect of the other. As the valve rotates in response to the applied moments, the valve grid rotates with it as a rigid body. This approach greatly simplifies grid generation issues. It is much simpler to create two independent grids, one each for the valve and the pipe, than to try to create a grid for the single geometry consisting of the valve sitting inside the pipe. In addition, grid-remeshing issues are removed as the valve disc rotates. The combination of the valve geometry, tight clearances with the pipe wall, and large range of motion would render this problem nearly impossible to do with classical grid remeshing techniques.

Figure 4 shows the grid used for the pipe. Half of the geometry was modeled, with a symmetry condition applied along the center plane. A finer grid was used in the vicinity of the valve where a more complicated flow field occurs. The pipe grid contains about 25,000 grid points.

The grid surrounding the valve is given in Figure 5. Also shown in Figure 5 is the grid on the valve disc surface. To allow easy comparison with the valve geometry (Figure 3), the valve surface grid has been mirrored across the center plane to show the entire valve. The valve grid contains approximately 100,000 grid points.

Figure 6 shows the overlay of the valve disc and pipe grids for the closed valve position, and for the valve open position at 14,000 gpm. In the closed position, there is a gap of about 0.15" between the valve and the pipe wall. The valve open position (as shown in Figure 6) is calculated as part of the solution process. This figure demonstrates the rigid body rotation of the valve grid inside pipe.

The Chimera scheme [3] proceeds by cutting "holes" in each grid in the regions where the grid overlaps with wall boundaries, which belong to the other grid. In this case, the cells in the pipe grid that overlap with the valve disc are removed and similarly the cells in the valve grid that overlap with the pipe-wall are removed. This results in the configuration shown in Figure 7.

The cells that are adjacent to these cut boundaries are called Chimera boundary cells. The solution is interpolated between the two grids along these boundary cells in order to make them communicate with each other, allowing an implicit solution of the combined problem.

Numerical Model Setup

For the steady-state calculations, interest in the final valve position and pressure drop for a given flow rate was desired. For these runs, the boundary conditions were fixed mass flow rate at the pipe inlet, and fixed static pressure at the pipe exit. The pipe walls and valve surfaces were modeled with a no-slip boundary condition. Sufficient pipe length was provided beyond the valve to allow for pressure recovery to a uniform level, such that a fixed pressure boundary at the pipe exit is accurate.

Flow rates performed during this analysis ranged from 1,000 to 14,000 gpm. These flow rates corresponded to inlet velocities of 0.216 and 3.024 m/sec, respectively. The fluid (i.e. water) was taken as incompressible with constant properties. The valve disc motion was computed with a 6DOF model, which for this case reduced to one degree of freedom, namely rotation about the valve disc hinge. The valve disc rotates as a consequence of the net torque resulting from the fluid pressure, valve weight, and packing friction resistance. The 6DOF model is inherently time-accurate, so these cases were run as transient solutions until a steady-state solution was reached.

Multi-Disciplinary Analysis With MDICE

CFDRC's MDICE in an advanced computing environment that was developed to perform complex multi-disciplinary engineering analysis. Such analysis typically requires several application modules to be closely coupled. MDICE provides such an environment, which enables direct exchange of relevant data between simultaneously executing engineering analysis modules [4]. Such an architecture enables efficient analysis of multi-disciplinary problems where different analysis modules exist. Each module within the environment, namely, geometry modeling, mesh generation, and flow solver, is responsible for specific tasks within the environment.

In the present work, MDICE is used to couple two CFD−ACE+ modules, which perform the flow computations and a 6DOF analysis module, which performs the analysis of kinematics and dynamics of moving bodies. The two CFD−ACE+ modules work on the pipe grid and the valve grid respectively. MDICE enables the Chimera hole-cutting in the two grids (as shown in Figure 7) as well as the solution interpolation to the Chimera boundary cells in each grid. The 6DOF module helps integrate the forces and moments acting on the valve and predict the motion. All three modules are tightly coupled in MDICE to perform this simulation of flow around the tilting valve inside the pipe.

Results and Discussion

Steady state analyses were performed for flow rates ranging from 1,000 to 14,000 gpm. Results from the numerical analysis are the instantaneous valve position, as well as field values of fluid velocity and pressure, at discrete instances in time.

Results from the steady-state solution for 14,000 gpm are shown in Figure 8. The figure shows the static pressure contours and the flow field velocity contours at the center of the pipe. Similar plots for the 7,000 gpm case are given in Figure 9. As seen, the flow field for the 7,000 gpm case is more complex through the valve region (since the valve is not open as much), which results in a larger pressure drop across the valve. The pressures shown are in N/m^2 relative to a zero pipe exit pressure. A three dimensional view of the solution in the pipe-valve configuration, obtained at a flow rate of 14,000 gpm, is shown in Figure 10, with the valve disc in the fully open position.

A plot of the valve disc-opening angle vs. flow rate is given in Figure 11. The opening angle is measured from the closed position of the valve. After analyzing the results of Figure 11, it was found that the steady state valve disc position was not unique for a given flow rate and was dependent upon the initial flow conditions. For all the cases in the Figures, with the exception of the ones corresponding to flow rates of 5,000, 10,000, 12,000 gpm, the initial flow condition was taken to be that corresponding to the inlet velocity for the given flow rate. In other words, at the start of the simulation, with the valve in the closed position, the flow everywhere in the domain is instantaneously brought up to the desired inlet velocity. The flow impacts directly on the closed valve, resulting in extremely high pressures below the valve disc, which starts the valve disc opening with a large velocity. For the cases of 5,000, 10,000 and 12,000 gpm, the solution was started using an initial solution, which was the converged solution obtained with a slower flow rate. In this case, the valve disc does not see the same large impact that it would have seen had the solution been started with the correct inlet velocity and consequently the valve opening is smaller for these cases and falls outside the general trend.

It is suspected that the packing friction resistance may be the result of this effect. This additional torque is applied such that it is always resisting the direction of motion. If the valve is opened slowly from a low flow rate, the frictional resistance will always be acting to close the valve. However, if the valve is closed from a higher flow rate, the frictional resistance will always be acting to keep the valve open. Since, the resistance will be acting in the opposite direction in each case, the final valve opening will not be the same.

Using this simulation setup, it was found that at flow rates of 1,000 gpm or below, the flow was unable to overcome the torque due to gravity and friction and the valve did not open at all. At 2,000 gpm, the valve had a small opening angle. Hence, it is predicted that flutter or tapping would occur for flow rates less than 2,000 gpm, for the given packing resistance. The actual phenomenon of flutter was not demonstrated in this simulation.

Figure 12 shows the pressure drop across the valve as a function of flow rate. This pressure drop was measured from the beginning to the end of the elliptical pipe cross section, (see Figure 2). This is a nonlinear function of the flow rate and exhibits a local minimum. This local minimum results from the interaction of the flow field and the solid deformation. For a given valve position, a larger flow rate will result in a larger pressure drop. However, a larger flow rate will result in a larger valve opening, which, for a given flow rate, will result in a lower pressure drop. The resulting curve is a function of the interaction of these two phenomena.

In addition, the valve positions for the 5,000, 10,000, and 12,000 cases fall below the general trend, as described above. Therefore, the pressure drops for these cases are larger than they would be if the valve had opened to the larger position.

Figure 13 shows the transient response of the valve position for three flow rates. In each case, the valve quickly opens and reaches a near-steady value. The reason for the fast opening is the initial conditions used in the analysis as discussed above. The velocity throughout the domain is instantaneously raised to the inlet value, resulting in a large pressure difference across the valve at the start of the transient. Once the pseudo-steady position is reached, the valve oscillation about the mean value is about 0.5 degrees.

Additional Analysis Without Friction

After performing the analysis with packing friction, another series of test cases were run to determine if the packing friction was the cause of the non-unique valve opening angles. For these simulations, the friction was set to zero and thus the valve was free to rotate unimpeded about its hinge, damped only by the action of the flow.

The first case run without friction was at a flow rate of 2,500 gpm. The temporal response of the valve-opening angle for this case is given in Figure 14.

The effect of removing the frictional resistance is immediately apparent, as this plot shows that the valve oscillates about an average opening position with a small amount of damping provided by the fluid. The average valve position was calculated to be about 6.5 degrees for this case. The solution that corresponds to the flow rate of 2,500 gpm was used as the starting point to compute the valve-openings at a higher flow rate (4,000 gpm) and one at a lower flow rate (1,500 gpm). In addition, the valve-openings for these flow rates (4,000 and 1,500 gpm) were also computed by starting the simulation with an initial flow field that corresponded to the respective flow rates. The results for these cases are shown in Figure 15.

For each case, the mean value of valve-opening position is the same for both the restarted and non-restarted runs. This demonstrates that, without packing friction, the valve position is a unique function of the flow rate. This leads to the conclusion that the valve packing friction is the cause of the non-unique results seen in the previous runs. The average valve position as a function of flow rate is plotted in Figure 16.

For the flow rates considered, the valve position is a linear function of the flow rate. Also, note that the valve is opened more at 4,000 gpm without friction (12 degrees) as compared to 4,000 gpm with friction (8.5 degrees, see Figure 10). This is to be expected, since the packing friction will act to oppose the opening of the valve. Extraction of precise pressure drop data was difficult because of the transient nature of these solutions. This is due to the fact that the valve exhibited a continually oscillating behavior (about a mean position) due to the lack of packing friction and hence the pressure drop across the valve does not reach a steady state value. Figure 17 shows the transient pressure drop response for the given flow rates of 1,500, 2,000 and 4,000 gpm. The plots shown in the figure do not appear smooth because the data required to post-process the flow field to calculate the valve pressure-drop was not written out every time-step. However, much like the valve opening, it was observed that the pressure-drop across the valve also exhibited an oscillatory behavior about a mean value. For each flow rate an average data was approximated, as shown on the graphs. These average values are plotted in Figure 18. The pressure drop increases monotonically, but not linearly, with increasing flow rate. The exact shape of this curve will depend on the average values extracted from the transient plots shown in Figure 17.

Conclusion

An effective CFD methodology was demonstrated to enable simulations such as flow around a tilting disc valve inside a pipe. A Chimera overset grid methodology was used for these analyses, in which separate grids are created for the valve and pipe. These grids are then overset with the necessary information interpolated between them. This approach was very well suited for this type of analysis, as the grid-remeshing problem, resulting from the valve rotation, is eliminated. The overset grid methodology in conjunction with 6DOF was shown to be very suitable for modeling such internal flow problems with moving bodies.

Several cases of the coupled solid/fluid problem of flow past a tilting disc check valve have been performed. Steady-state results were obtained by running a transient simulation until the solution became steady (i.e. valve stopped opening). For each case, the resulting valve position and pressure drop were calculated. Also, the flow rate at which valve flutter may occur was estimated by bracketing the valve opening flow rate

The solution for the steady-state results indicated that the valve opening position, and hence the pressure drop across the valve, was not unique for a given flow rate. This conclusion was arrived at based on the observation that the valve-opening position was dependent on the initial conditions for the given flow rate. The packing friction may be contributing to this, since the direction of the frictional force can change through-out the run. This was further verified by performing the flow simulations without using the packing friction and allowing the valve to rotate freely. It was observed that the valve-openings in the absence of packing friction were unique for the given flow rate and were not dependent on the initial flow conditions.

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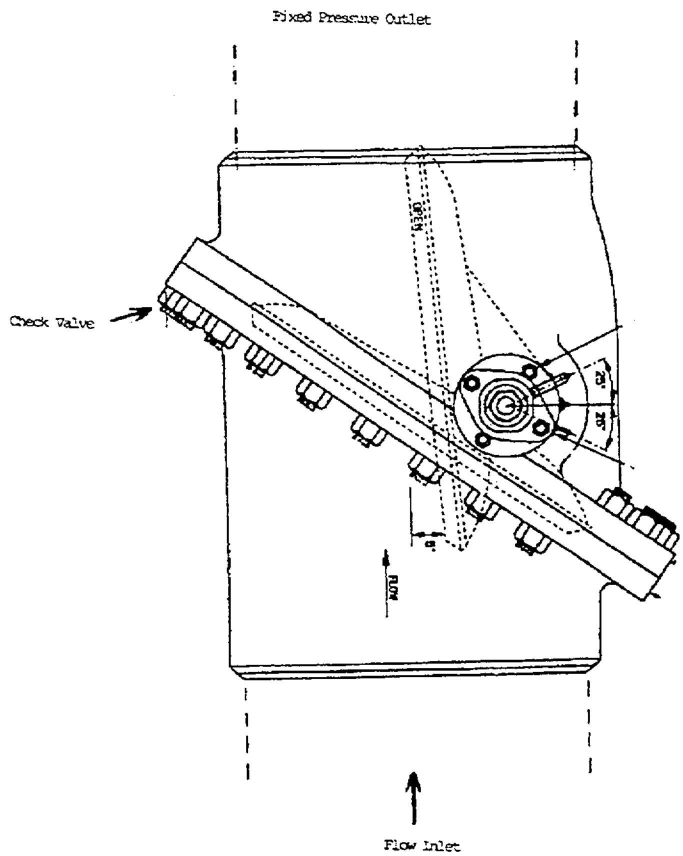

Figure 1. Problem Schematic

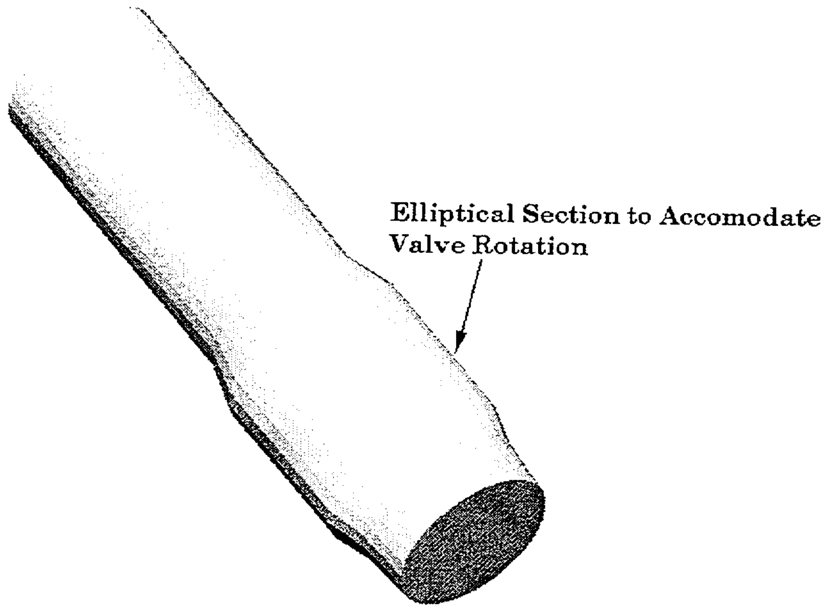

Figure 2. Geometry Modeling of the Pipe

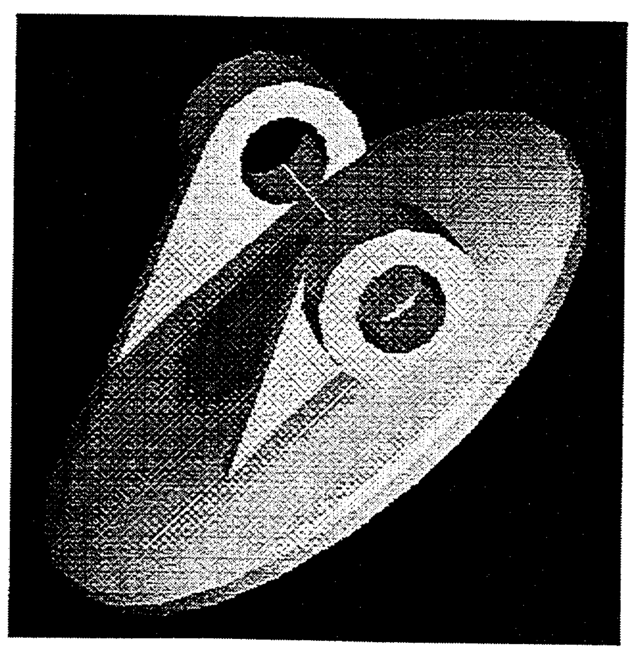

Figure 3. Geometry Model of Disc Valve

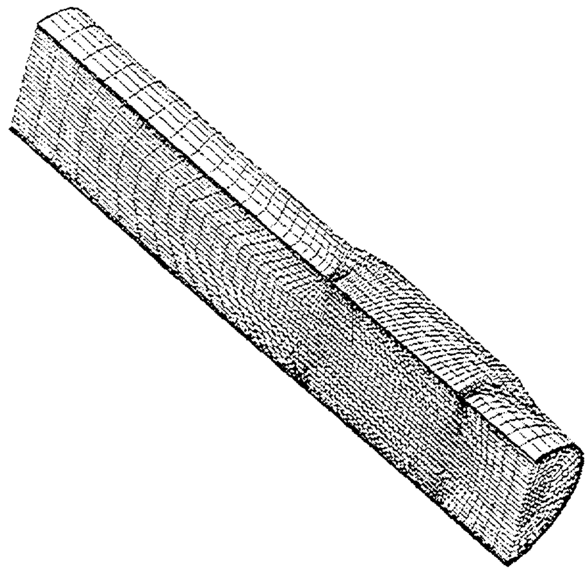

Figure 4. CFD Grid inside the pipe

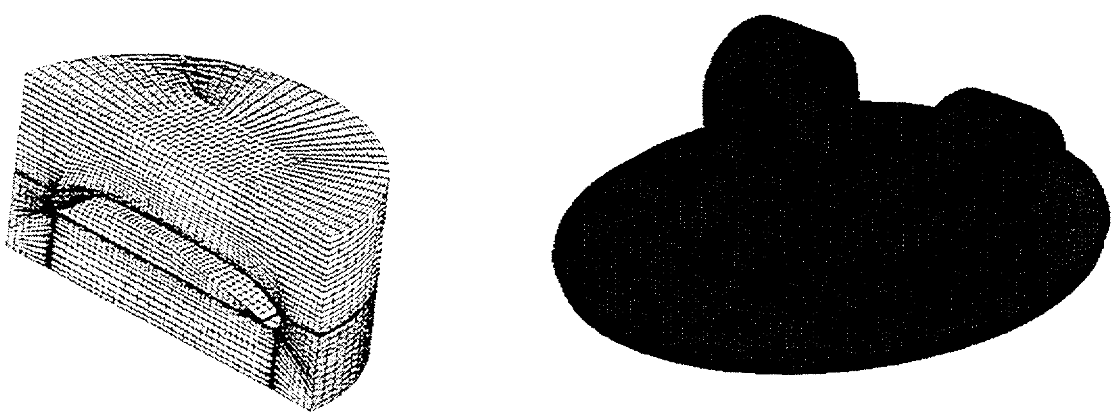

Figure 5. Grid used for modeling valve geometry

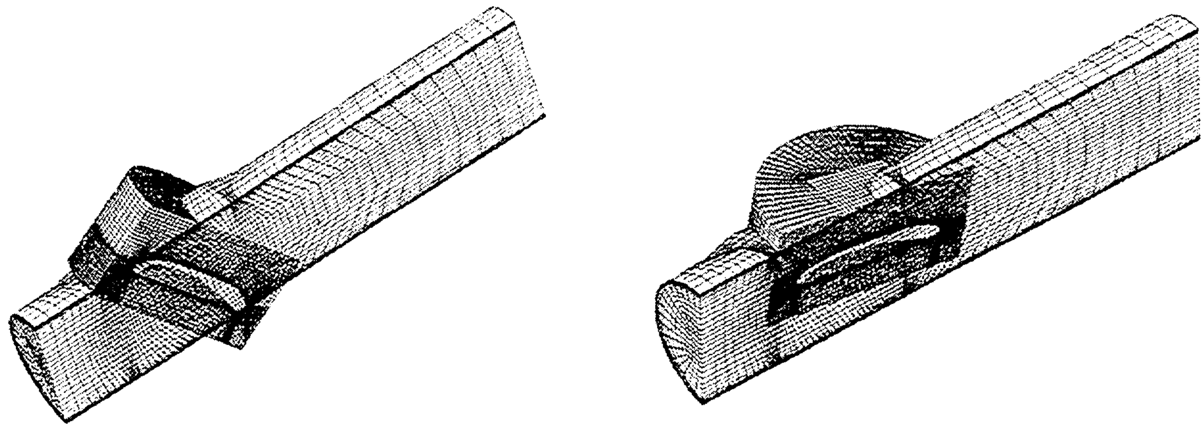

Figure 6. View of Overset Grids used for the Pipe and Valve

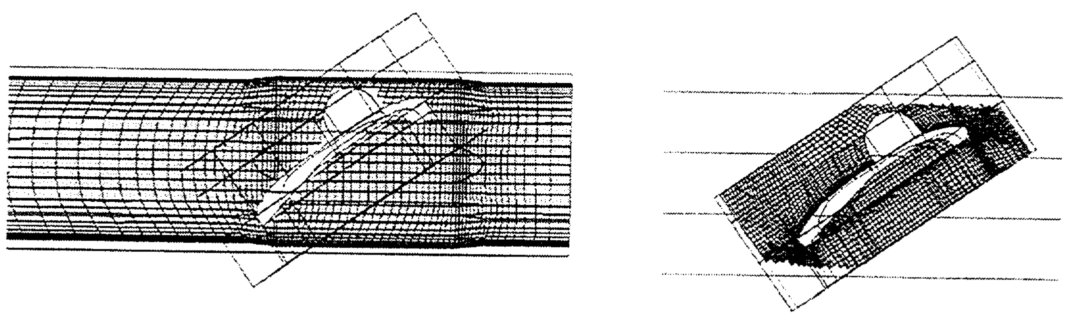

Figure 7. Chimera Hole cutting in the pipe and valve grids

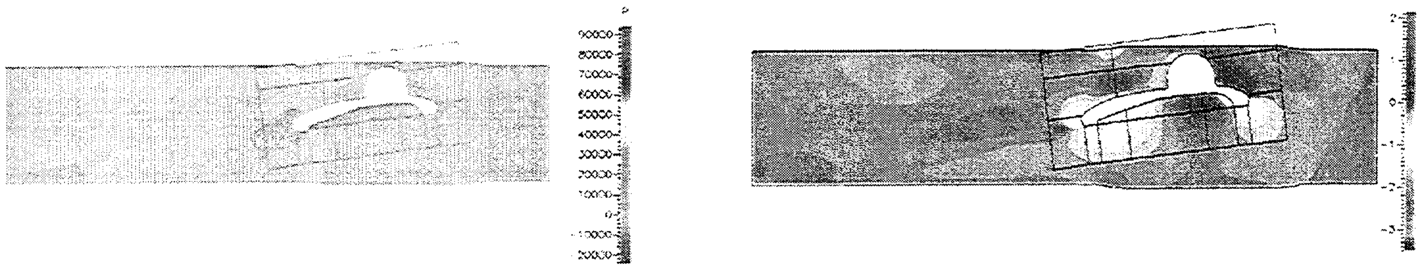

Figure 8. Pressure and Velocity Contours @ 14,000 gpm

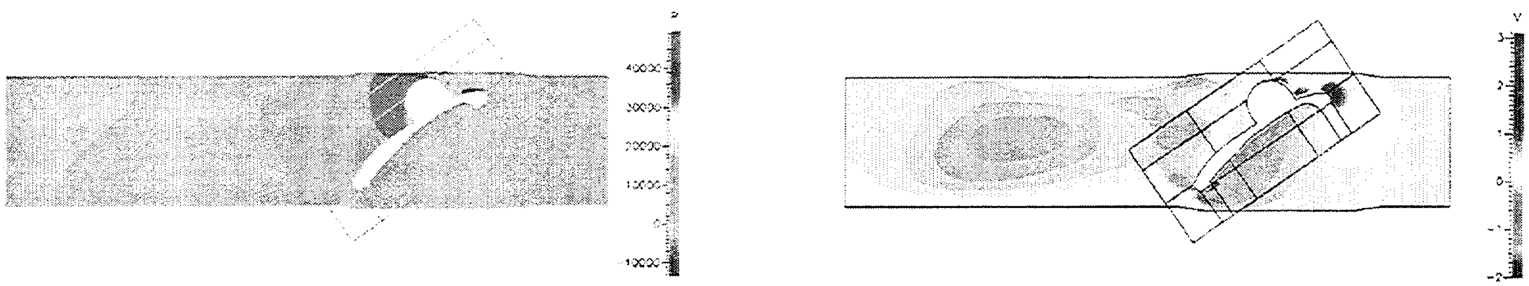

Figure 9. Pressure and Velocity Contours @ 7,000 gpm

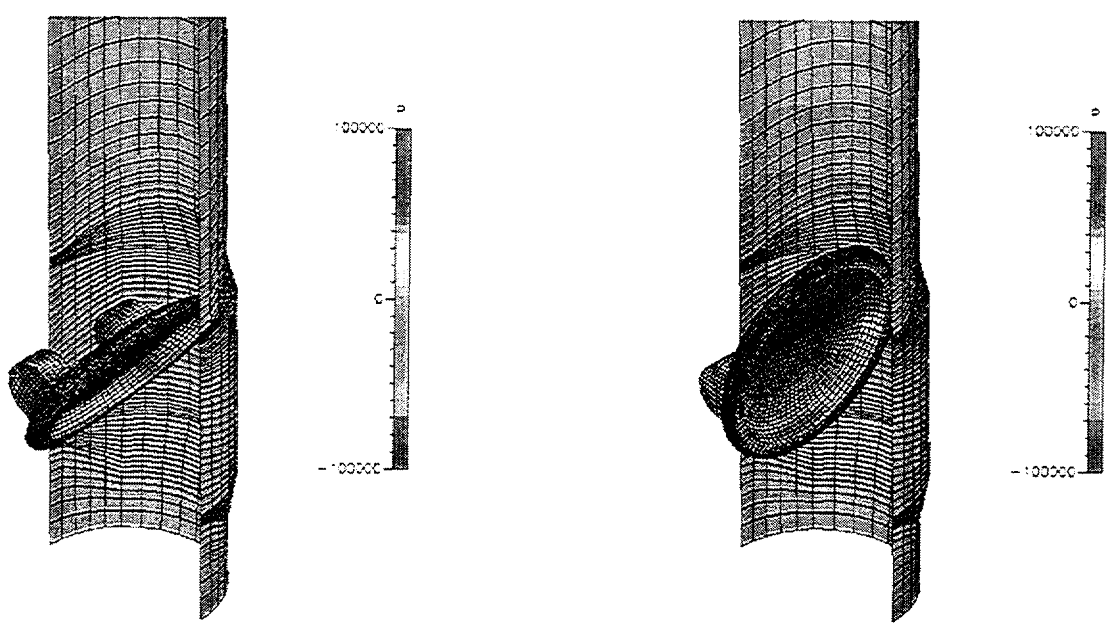

Figure 10. Isometric view depicting the flow simulation at two different positions of the valve. The flow rate in the pipe is 14,000 gpm.

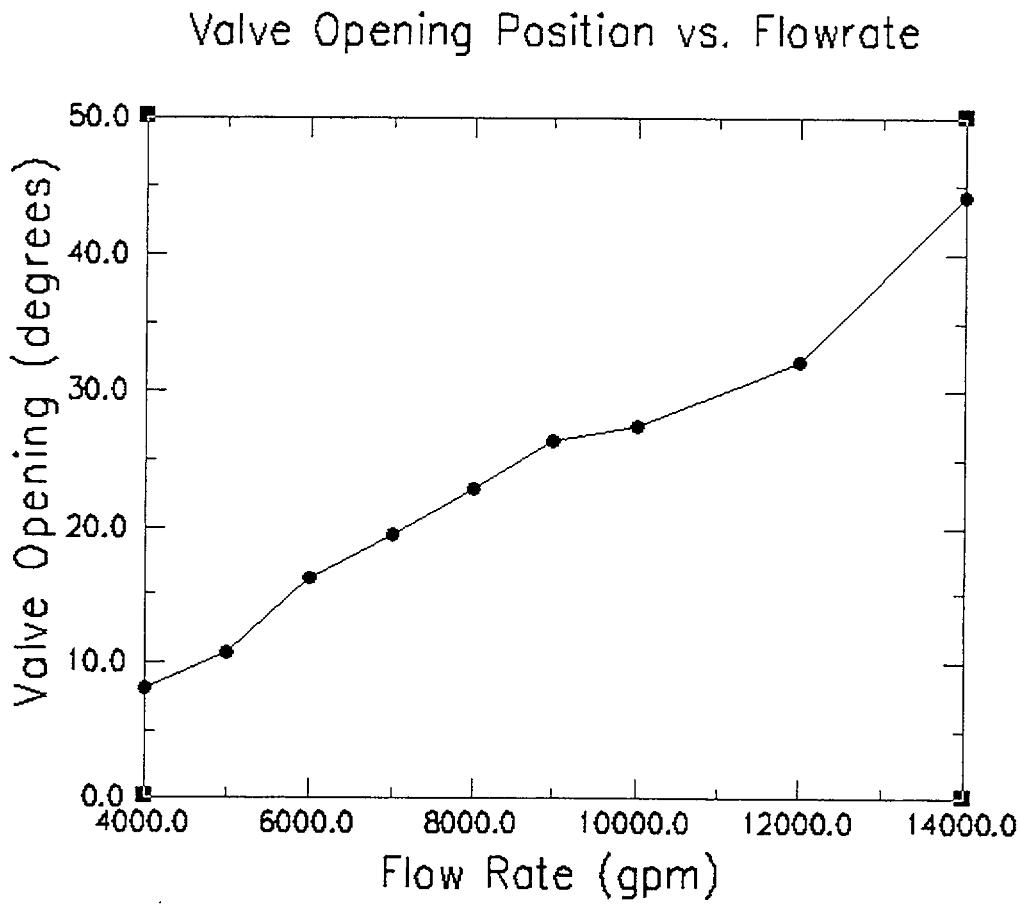

Figure 11. Comparison of Valve Opening Position with Flow Rate

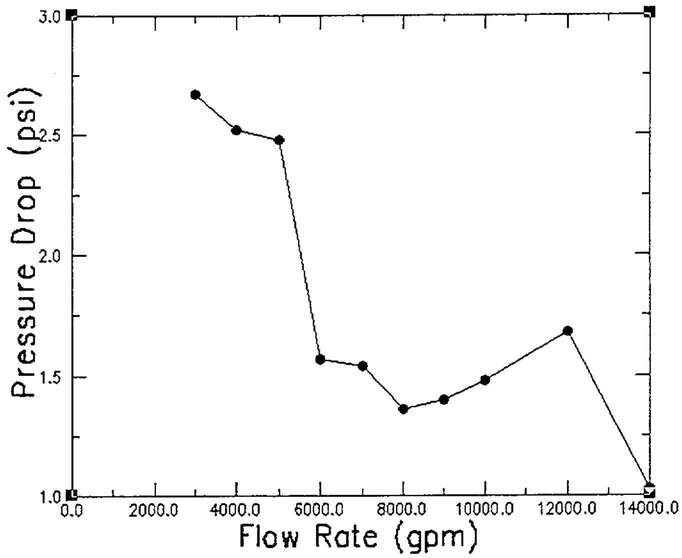

Figure 12. Comparison of Pressure Drop across the Valve for Different Flow Rates

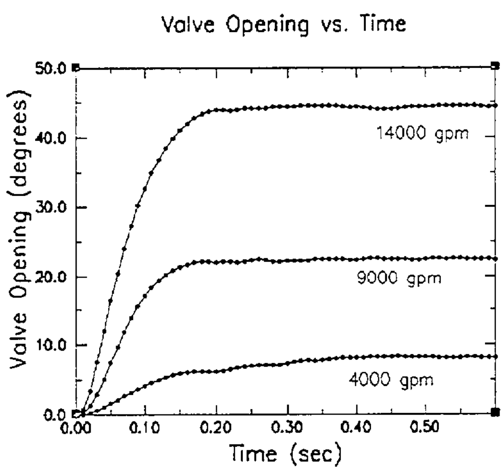

Figure 13. Comparison of Valve Opening at Different Flow Rates

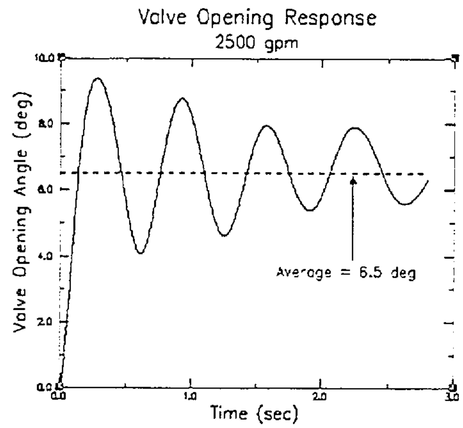

Figure 14. Valve Opening Response for 2,500 gpm.

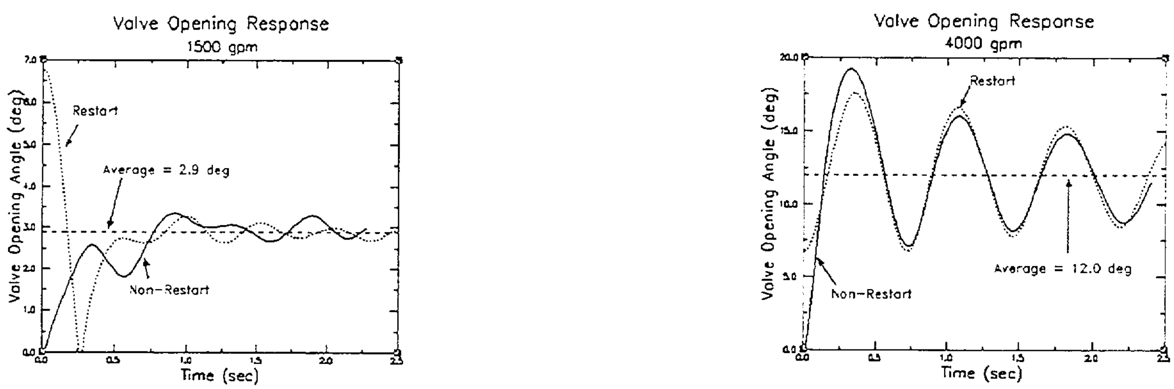

Figure 15. Valve Response for Restarted and Non-Restarted Cases

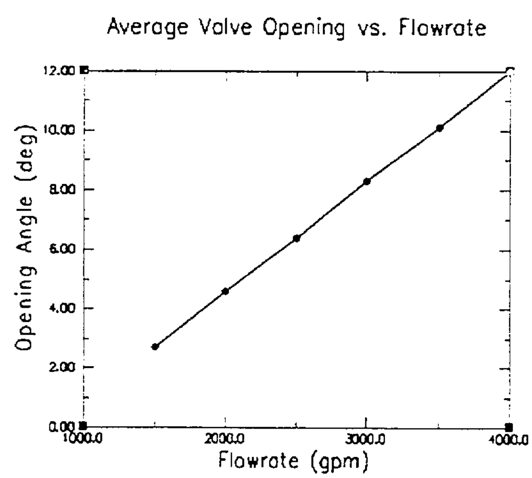

Figure 16. Comparison of Average Valve Opening Position for Different Flow Rates

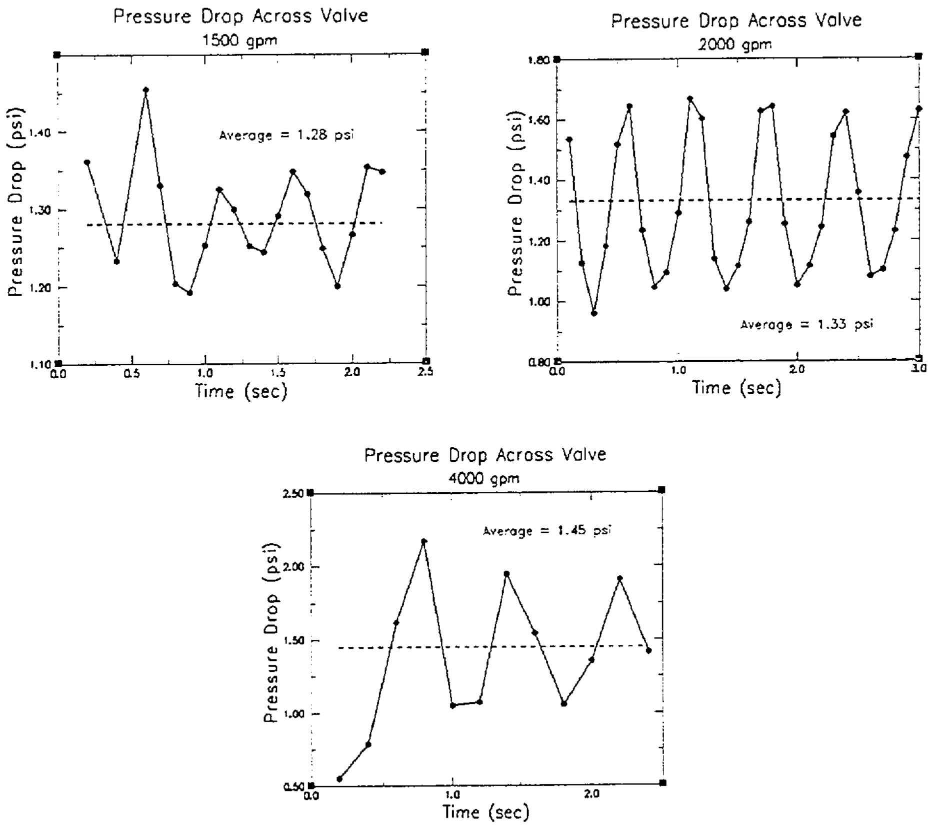

Figure 17. Transient Pressure Drop Across the Valve for Different Flow Rates

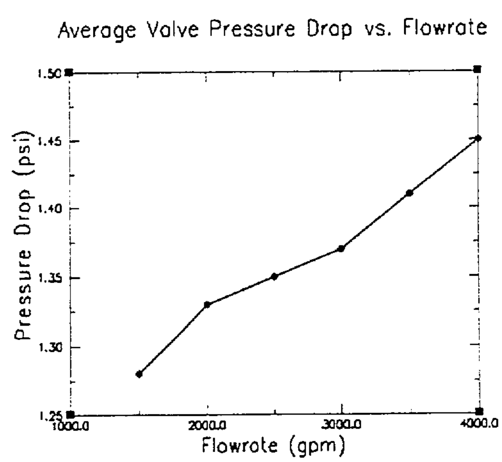

Figure 18. Comparison of Average Valve Pressure-Drop for Different Flow Rates

Session 1(B)

Pump and Valve Performance Issues

Session Chair
R. Scott Hartley
Idaho National Engineering
and Environmental Laboratory

Seat Leakage through Emergency Core Cooling System Check Valves

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Abstract

This paper examines issues regarding the performance of Emergency Core Cooling System (ECCS) check valves. Seat leakage experienced though these check valves has resulted in adverse consequences to plant operations. Actions that can be taken to repair or replace check valves, as well as mitigate the consequences of such leakage, will be addressed.

ECCS check valve seat leakage, well within technical specification limits, has resulted in numerous undesirable consequences. Consequences experienced throughout the nuclear industry, and at Seabrook Station, include the following:

- pressurization of connecting systems, resulting in challenges to safety valves —residual heat removal system (RHR) and/or safety injection system (SI)

- SI accumulator in-leakage and out-leakage, resulting in challenges to technical specification parameters such that:

  - Makeup for out-leakage causes excessive operation of the Safety Injection Pumps
  - Increased sampling frequency for potential dilution from in-leakage is required
  - Boron concentration adjustments from dilution from in-leakage causes increased waste processing
  - gas voiding of ECCS piping (RHR and/or SI) from out-leakage occurs

- Gas collection in ECCS piping caused by check valve leakage from the RCS to RHR and/or SI

Seabrook Station conducted an operating experience review and a comprehensive survey of thirteen (13) Westinghouse 4-loop plant sites on the issue. Although the impacts varied from plant to plant, ECCS check valve seat leakage frequently occurs due to the configuration of the ECCS connections to the RCS and SI accumulators. The manifestation of ECCS check valve leakage impacting operation of these plants is prevalent.

Introduction

Check valves leak to a certain degree; the leakage may be very slight (nuisance) leakage in the gallon per day (gpd) range or less. The technical specification leakage limits for pressure isolation valves (PIVs) is 0.5 gallons per minute (gpm) per inch of nominal valve size, e.g., 5 gpm for 10-inch check valves and 1 gpm for 2-inch check valves. The basis for PIV leakage rate is intersystem LOCA concerns. The specified leakage rates are well within normal makeup and system relief valves capabilities, thereby precluding an intersystem LOCA. To put a typical nuisance leakage rate into perspective, a 4 gpd leak rate equates to about 0.003 gpm. Yet, this low leak rate resulted in significant accumulator level changes and system performance changes over time.

For a system in a standby condition (process flow not in progress), the difference in leakage rates of valves dictates the manifestation of plant operational issues. For example, for a configuration of two check valves in series, if the upstream check valve has a lower leakage rate than the downstream one, the pressure realized in the section of piping between these two check valves rises to the pressure upstream of the first check valve (see Figure 1). In this case, the back leakage rate through the two check valves in series, once an equilibrium is reached, is the rate through the check valve in the upstream location.

The ECCS design at Seabrook Station, as well as at a number of other plants, contains a configuration of check valves such that back-leakage results in one or more of the three operational effects noted above.

Seabrook Station experienced in-leakage to the accumulators commencing in the summer of 1997, shortly after startup from the fifth refueling outage. This in-leakage necessitated actions to preclude exceeding technical specification ranges for accumulator level and boron concentration. In addition, increased frequency of sampling of the accumulators was required.

Seabrook Station has also experienced some of the other issues seen throughout the industry. Relief valves lifted due to pressurization on the SI pump discharge piping resulting from check valve back-leakage during transients, such as plant startups and pump surveillance testing. Difficulties have been encountered during startup operations to seat check valves to minimize accumulator in-leakage and out-leakage. During technical specification venting of the SI pump discharge piping, gas bubbles have been observed. Sampling of the vented stream and further evaluation confirmed that there was some back-leakage to this portion of the SI system from the RCS.

Body

Seabrook Station Operating Experience

Figure 2 shows the portion of the Seabrook Station SI pump discharge piping (cold leg) and SI test header involved in many of the conditions and events experienced at Seabrook Station.

Pressurization of Connecting Systems

Following each refueling outage at Seabrook Station, in-service testing personnel monitor the SI accumulators to determine if the ECCS check valves need to be reseated. Despite these efforts, the SI pump discharge header was at accumulator pressure (approximately 630 psig) for the duration of the first four and the sixth operating cycles. For Cycle 5 (December 1995 to May 1997), efforts to seat the check valves were successful in reducing the SI header pressure to about 50 psig. This represents the static head of the RWST.

The SI pump discharge header relief valves lifted several times after performing ECCS pump and valve surveillances. These activities caused the SI cold leg check valves (SI–V118, SI–V122, SI–V126, and SI–V130) to become unseated. The SI header pressure slowly increased to accumulator pressure of about 630 psig. Occasionally, the RCS first out check valves also became unseated, exposing the SI pump discharge header to reactor coolant system (RCS) pressure. This resulted in lifting the SI pump discharge header relief valves set at 1750 psig.

An investigation of damaged bellows in July of 1996 on SI–V76, train "B" SI pump discharge relief valve, found that the relief valve was challenged by two evolutions. Trend data for SI pump discharge pressure and operator logs revealed that two transients resulted in relief valve lifting and contributed to the damage: (1) during the fourth refueling outage, an accumulator fill operation was conducted, and (2) SI pump to cold leg check valves became unseated during an RCS heat-up evolution. Corrective actions for these events included procedural guidance, using the SI test header to vent the SI pump discharge piping and seat the SI cold leg injection check valves prior to reaching the relief valve setpoints.

In January 1998, another event of relief valve lifting occurred. During a run of SI–P–6–A, its discharge relief valve, SI–V–101, lifted. This allowed a flow path of approx. 40 gpm of RWST water to the primary drain tank. The SI to hot leg gate valve sealing face unseated and allowed the downstream RCS pressure to communicate with the upstream side. The corrective action was to revise the procedure to isolate a pressure locking valve during SI pump surveillance testing.

Following a forced outage in 1998, SI–V–101 developed a seat leak and had to be replaced. The valve condition had been acceptable prior to this forced outage. The seat leakage was caused by challenges to the valve during draining and filling accumulators (conducted to increase boron concentration of the accumulators due to in-leakage) and SI pump surveillances. During the draining and filling of the accumulators, noises were reported, possibly due to pressure waves, during operation of the SI accumulator fill valve (SI–V157).

Similar events of relief valve lifting and damage have occurred elsewhere in the industry. These were further addressed during the industry operating experience review and by the survey of Westinghouse plants.

SI Accumulator In-Leakage and Out-leakage

Significant accumulator in-leakage during steady state operation first occurred at Seabrook Station in the late summer of 1997. In August, it took several hours to stabilize decreasing level in accumulator "D". The pressure between the two 10-inch check valves on each of the "A", "B", and "C" accumulators was at normal RCS pressure (approximately 2235 psig). In contrast to this, the pressure between the "D" accumulator 10-inch check valves was essentially accumulator pressure (approximately 630 psig). Also, the pressure at the discharge of the SI pumps was accumulator pressure. This was indicative that the 2-inch SI to cold leg check valve SI–V130 was leaking more than other valves, including check valves SI–V96 and SI–V71 back at the discharge of the SI pumps (see Figure 2).

As discussed previously, the first four operating cycles at Seabrook Station also experienced accumulator pressure back to the SI pump discharge. However, accumulator level changes, during steady state operation, were not significant. Therefore, the August 1997 occurrence may have been the first indication of a leakage increase through the 10-inch check valves closest to the RCS. Although very small in magnitude (several gpd into the accumulators versus 5 gpm per the technical specification limit), accumulator boron concentration and level parameters would need adjustments to maintain them within the technical specification allowed values. Having SI−V130 more leak tight would reduce leakage into the "D" accumulator, because the accumulator 10-inch check valve (SI−V51) would become the seated valve. Since this valve is more leak tight, as experienced on the other three accumulators, reduction in the in-leakage would be expected.

On August 7, 1997, the SI check valve reseating procedure was stopped, leaving SI−V−131, V−62 and V−70 (refer to Figure 2) open to attempt to allow the system check valves to more fully seat. This vent path through the SI test header was maintained for a couple of days. A review by Engineering of the above lineup revealed that there is a concern in the event of a certain single failure and a postulated pipe break downstream of SI−V131. A single failure of the "B" train solid state protection system would result in a diverted flow through the postulated break. This would add to the operating "A" train SI pump discharge path normal flow, potentially impacting SI pump run-out and its proper cold leg injection into the core. A calculation was performed, which determined that the SI pump run-out values and the flow through the three most restrictive flow paths would remain within technical specification limits.

In September 1997, the "D" safety injection accumulator was experiencing an increasing water level of about 4 to 5 gallons per day. The RCS leakage into the accumulator was diluting the accumulator boron concentration. ECCS check valves had been seated as well as possible. The Safety Injection System operation procedure was revised to provide detailed instructions for seating ECCS check valves. The SI−V131 vent path was successfully used, starting in October of 1997, to divert leakage to the primary drain tank (PDT). The configuration for this diversion was through SI−V131, SI−V62, and SI−V70.

Following a forced outage in December of 1997, the "D" accumulator was experiencing minute out-leakage (an average of less than a quart per day). The 10-inch check valve closest to the RCS (SI−V50) was now seated tightly, such that the leakage through SI−V130 (and/or possibly SI test header AOVs), as limited by the check valves back at the SI pump discharge, was occurring through the now "floated" accumulator 10-inch valve (SI−V51). This provided evidence that there was not significant degradation to the accumulator check valve (SI−V50). A crud problem, being transient in nature (i.e., deposited and removed by flow initiation and isolation), was a potential contributor.

The phenomena of variations on which valves seat, can be partially attributed to the design of the Westinghouse wire-armed check valve (e.g., SI−V50 and SI−V51). They require high differential pressure to keep them tightly seated and are easily "floated." Figure 3 illustrates the nearly vertical disc position for this valve design. Because the disc seats at a nearly 0 degree angle, this valve requires a high differential pressure to seat. Furthermore, the wire arm on the disc is not conducive to tight seating. One of the plants surveyed did bench testing of the valve and found that, at several hundred psi differential, there was 500 ml/minute leakage. This leakage was gradually reduce to zero, once the differential pressure was raised to about 1500 psi.

Shortly after restart from a June/July, 1998 forced outage, accumulators "A", "B", and "D" experienced in-leakage. In September of 1998, nitrogen pressure was adjusted and check valves were seated such that, except for accumulator "D", the levels of the accumulators were stable. Accumulator "D", once again, experienced in-leakage of about 4 gallons per day. Dilution of this accumulator by continued in-leakage would require action.

In October 1998, a task team was formed to: 1) devise an acceptable method to adjust accumulator boron concentration on-line within technical specification allowed outage times, and 2) determine the cause and long term fix to the check valve leakage/accumulator level problems. The task team identified the following issues and actions:

- Develop valve repair recommendations for the March 1999 refueling outage
- Review operating experience
- Conduct survey of Westinghouse 4-loop plants
- Evaluate crud as a contributor to leakage
- Review procedures related to check valve leakage issues
- Develop decision criteria for continued operation in the presence of check valve leakage
- Review operator training
- Develop possible design enhancements
- Define multiple department roles in response to accumulator level changes
- Reassess earlier recommendations on the ECCS check valve leakage issue
- Propose benchmarking trips to plants with best practices

The most valuable results of the above efforts pertinent to generic issues in the industry were provided by the survey of Westinghouse 4-loop plants. Furthermore, many of the other issues listed were resolved, in large part, as a result of this effort. This paper will therefore focus on the survey results.

Gas Intrusion into ECCS Piping

Another manifestation of check valve leakage occurred, during monthly ECCS venting. While performing ECCS venting, the presence of gas was observed in the form of very small bubbles as viewed in the tygon tubing from the SI system. During the static venting of the charging pump, the vented water was observed to be cloudy, as if aerated. The acceptance criterion of the procedure was recently changed to "no gas observed." This condition was attributable to gas coming out of solution via the venting process itself. This degassing of the fluid in the vented stream is caused by depressurization to near atmospheric pressure. As confirmed by ultrasonic testing, it was not indicative of gas pockets within the systems. However, gas was dissolved in the fluid. The source of gas in the charging pump fluid is the hydrogen overpressure imparted on the volume control tank. The SI pump discharge piping to the cold leg was sampled for gaseous content and boron concentration. The results of the presence of hydrogen and diluted boron concentration demonstrated there was back-leakage of RCS fluid through check valves. Studies conducted for an Institute of Nuclear Power Plant Operations (INPO) report addressing potential loss for high pressure injection and charging capability from gas intrusion identified the potential for nitrogen gas pockets in ECCS piping as result of accumulator out-leakage.

Industry Operating Experience

Operating experience documents concerning check valve leakage in the ECCS system were researched. Both INPO reports and NRC reports were reviewed. Due to the proprietary nature of the INPO reports, only the NRC-associated reports are presented in this paper. Table 1 is a summary of the events described in these NRC reports.

Review of these events revealed that documents were generated only in the event of high leakage flow rates, significant relief valve lifting, impacts of safety injection test header valves on the ECCS function, or if leakage resulted in voiding in the upstream portion of an ECCS subsystem. Chronic low level leakage cases, such as what Seabrook Station was experiencing, are not generally reported. The first INPO report of similar "nuisance" leakage was generated in January of 1999.

The only report made by North Atlantic for Seabrook Station was LER 98–002. This LER reported that an error in the use of ultrasonic detectors for flow measurement resulted in the potential for SI pump train A to exceed its maximum flow requirement (pump run-out concern), upon signal failure of the "B" train solid state protection system (SSPS), and the test header valve SI–V131 open. The LER reported that SI–V131 had been open to minimize the effects of check valve back-leakage. This LER was later withdrawn, based on an engineering evaluation that concluded that an assumed postulated break in the NNS test header piping was not a credible event.

Of the eleven (11) reports in Table 1, only one is indicative of accumulator in-leakage. This was a report by Comanche Peak 2 (LER 93–010), where required boron samples were not taken when check valve leakage resulted in accumulator level increase of 15 percent over a 24 day period. This represents close to 40 gallon per day leakage, an order of magnitude higher than what has caused the 1998 dilution of accumulators at Seabrook Station. This report was made only because the technical specification for sampling for accumulator boron concentration was not met. This indicates that there are other examples where accumulator in-leakage is experienced but not reported.

Although not as common as accumulator out-leakage, the survey of Westinghouse plants revealed that accumulator in-leakage has, in fact, occurred at a number of plants. Accumulator out-leakage is more common and has resulted in the condition described in NRC Notice 97–040 of nitrogen gas pockets forming in the ECCS discharge piping. Excessive safety injection pump runs for makeup for out-leakage was another concern identified through the survey. Moreover, the survey of 4-loop Westinghouse plants revealed a number of insights regarding some of the reported, published events, along with evidence of chronic, low-level, nuisance leakage of ECCS check valves and its manifested impact on plant operations. Similarly, a review of several NRC inspection reports further confirmed that chronic low-level RCS and accumulator leakage resulted in undesirable impacts on plant operation.

Survey of Westinghouse 4-Loop Plants

The North Atlantic task team conducted a comprehensive survey of Westinghouse 4-loop plants in order to: 1) gather data on industry operating experiences regarding ECCS check valve seat leakage operational impacts, along with causes and contributors, and 2) ascertain best practices in the areas of check valve seating methodology, and maintenance and design of check valves and systems. It was stipulated that names of plants would not be published.

Of the seventeen (17) surveys sent out, thirteen (13) responses were received. An evaluation of these results follows (each heading is the statement of the question being addressed):

1. Do you have difficulties seating the ECCS check valves on the SI accumulators and/or back to the SI or RHR pump discharge piping?

This question assesses the experience at other plants associated with difficulties in seating check valves to ensure that leakage back to the accumulators and SI or RHR piping do not result in operational impacts. Twelve (12) out of the thirteen (13) respondents indicated that they have or have had difficulties associated with seating ECCS check valves.

The conclusion from the responses to this question is that nuisance ECCS check valve leakage (problems with complete seating) is a very common problem among 4-loop Westinghouse plants. While a number of plants are currently having problems, some plants have corrected, or effectively reduced the problems such that operational impacts are currently not significant. Details of such actions were provided in response to later questions and are discussed below.

2. How long, following a plant startup, does it take to achieve satisfactory ECCS check valve leakage rates, i.e., how long is the ECCS check valve seating activity? What organization (i.e., Operations, IST group, System Engineering) performs this ECCS check valve seating function?

Seabrook Station has experienced difficulties in achieving satisfactory ECCS check valve leakage, such that venting (e.g., via SI–V131) has sometimes been required for a couple of days and/or accumulator in-leakages have been difficult to reduce to a minimal level. A concern existed that only two "experts" from the Component Engineering group are proficient in controlling and resolving ECCS check valve seating issues during plant startups. This question was generated to assess the experience at other plants.

There was somewhat of a correlation of the magnitude of problems experienced during check valve seating and the organization that was in charge of such evolutions. In three cases where there did not appear to be particular seating problems during startup activities, Operations was the responsible organization. In three other cases where problems have been experienced and at a fourth plant site that had two recent problems during startups, System Engineering is indicated as the responsible organization. Three plant sites indicated that Operations was the lead, but depended on System, Support, or IST Engineering to support such evolutions.

There was a wide variety of responses to the question of the time it took to perform the seating. One plant has experience that indicates that they may have check valve seating problems during startup that result in accumulator level changes, but because of accumulator level monitoring limitations, they do not realize it until later. That plant indicated that, about a week following startups, leakage into and/or out of accumulators is often detected. The accumulator level trends are only available via operator logging, recorded once every three hours. In contrast to this, Seabrook Station uses the plant computer to continuously and closely monitor accumulator levels during check valve seating evolutions.

An interesting finding is that one plant site obtained a technical specification change in 1994 that allows them to close an accumulator MOV for 24 hours and be inoperable on boron for 72 hours. Consequently, they perform their PIV testing at normal operating pressure, thus providing better test leakage results. Such a technical specification would also allow for more time to correct an accumulator or check valve leakage problem. The technical specification change was based on a PRA evaluation. The Westinghouse Owner's Group (WOG) is involved in a similar generic technical specification change.

3. Have you employed any check valve seating methods that you consider effective? Are they proceduralized?

Based on leakage concerns for ECCS check valves during startup evolutions and upsets of check valves during surveillances that may cause leakage at other times, Seabrook Station has developed a detailed written guidance for seating check valves. This guidance is included in the safety injection system operation procedure. It involves pressurizing at various points, including the use the SI test header isolation valves (for example, SI–V131) and a safety injection pump. The safety injection pump is also used to "float" check valves in attempts to achieve a better seating condition upon reseating. This procedure is considered effective, but some difficulties are still encountered. The purpose of this question was to determine the experience at other plants.

The responses to this question generally revealed that other plants use the same techniques as used at Seabrook station. Some respondents noted, as Seabrook Station has also experienced, that achievement of a high differential pressure across a check valve is effective in seating check valves.

4. Do you have any ECCS check valves leaking right now? If so how much (is the order of magnitude gpd, gph, or in the gpm range)? What system does it affect—accumulators, RHR, or SI?

The purpose of this question was to determine whether there are current problems and at what type and magnitude at other plants, during steady state operations. Seabrook Station had pressurized SI discharge piping continuously for the duration of each operating cycle except for cycle 5 and during cycle 7 to date.

Nine (9) of the thirteen (13) respondents indicated that there is currently some check valve leakage that manifests itself in plant operation challenges, namely pressurization of upstream systems (both SI and RHR) and/or accumulator level changes.

5. Do you have an administrative limit for acceptable ECCS check valve leakage? Is the value different from the technical specification PIV leakage limits?

The purpose of this question was to determine the industry practices with regard to actions taken in the event of ECCS check valve leakage that is within technical specification limits, but can result in undesirable impact on operation of the plant.

Out of the thirteen (13) respondents, eight (8) indicated that they did not use administrative limits. Technical specification limits were used as limits. One plant site uses an "ALERT" level, based on increasing trends of IST testing results. When such an "ALERT" condition occurs, the valves require more frequent testing. One plant site (plant E) has an informal administrative limit (typically 10 to 25% of technical specification limits) above which the affected valve is scheduled for inspection/repair at the next refueling outage.

Two plant sites indicated that, although they do not have special administrative limits, whenever check valve leakages result in operational problems, action is taken to identify the valves and repair them. It is noted that one of these two plant sites indicated (see response to question 4) that they are currently not experiencing ECCS check valve leakage of any significance. However, they are operating with a pressurized SI test header, attributed to test header check valve and AOV seat leakage.

One plant site respondent indicated, "We are targeting zero for maintenance purposes. If we detect any leakage, we make plans to go in and repair the valve." The only current leakage problem at this plant is accumulator out-leakage to the RHR system at a rate of about 6 to 7 gpd.

The above demonstrates that the industry has not developed strict criteria for nuisance leakage concerns. However, three plants are implementing repairs based on operational impact or a zero leakage policy. Two of these plants appear to be currently experiencing very few operational problems related to ECCS check valve leakage.

6. How does or has check valve leakage impacted your operation (manifested itself)?—Accumulator level changes (up or down)? Boron dilution in the accumulators? Relief valve lifting? Overpressurization of the ECCS system? Do you or have you operated with the RHR or SI pump discharge piping at accumulator or RCS pressure? Do you perform RHR or SI system venting?

The purpose of this question is similar to that of question 4, except that it examines both past and current experience and asks for more specifics on the manifestation of problems. Twelve (12) out of the thirteen (13) respondents indicated that they have or have had one or more of these difficulties associated with leaking ECCS check valves.

One site indicated that they had historically experienced all of the problems cited in the question. Their most visible problem was chronic accumulator out-leakage. It was determined that, in addition to ECCS check valve leakage, there was leakage through AOVs in the SI test header. Based on the consequential accumulator level decreases, the SI pumps were operated frequently in order to refill the accumulators. This resulted in a significant concern associated with potential damages associated with excessive cycling of the SI pumps. The peak number of pump runs occurred in 1994 (about 320). Therefore, this plant conducted a survey of a few other plants (see Attachment 4), which demonstrated that accumulator out-leakage was a common problem. This plant is currently down to about 10 pump runs per year (including quarterly surveillances). It is noteworthy that this same plant is not currently experiencing ECCS check valve leakage of a magnitude that results in a significant operational impact. They modified their 10-inch check valves (different style from Seabrook Station) and conduct check valve seat lapping and careful blue checks.

Of the eleven (11) remaining respondents, the experiences cited in the question are being or have been experienced, as follows:

- Accumulator in-leakage—five (5) respondents
- Accumulator out-leakage—seven (7) respondents
- Boron dilution—three (3) respondents
- Relief valve lifting (challenge to potential overpressure)—seven (7) respondents
- RHR pump discharge pressurization from accumulator and/or RCS source—seven (7) respondents
- SI pump discharge pressurization from accumulator and/or RCS source—ten (10) respondents
- Perform RHR or SI Venting—Two (2) respondents only stated that monthly venting of ECCS discharge piping was conducted, one (1) respondent indicated that venting was conducted to reseat check valves when unseated by system transients, four (4) respondents indicated that venting was conducted to reduce pressure in the RHR pump discharge line, and four (4) respondents indicated that venting was conducted to reduce pressure in the SI pump discharge line.

The conclusion from the responses to this question is that manifestation of ECCS check valve leakage impacting operation of plants is prevalent. Repair efforts, including lapping and careful blue checking, appear to be effective in minimizing problems.

7. What style, vendor, and model do you use for the 10-inch, 6-inch, and 2-inch sizes in ECCS check valves application? Are soft-seated valve designs used—high temperature concerns preclude its use?

It would normally be expected that smaller check valves would result in lower leakage rates than larger check valves. However, based on the experience at Seabrook Station, this is not always the case. The style of the check valve, and perhaps vendor and model, may contribute to the leak tightness of the valves. In addition, the use of a soft-seated design would be expected to be more leak tight.

The responses to this question did not reveal any significant correlation between check valve vendor/styles/models and performance. The six and ten-inch ECCS check valves were all swing type check valves. Some are inclined and others vertical style. The two-inch check valves were all piston type check valves.

None of the respondents indicated that they used soft-seated valves. As determined during Seabrook Station design enhancement efforts, there are limitations associated with the use of soft seats. As one respondent stated, "... they would have to be environmentally qualified for RCS out-leakage (i.e., design temperature $\approx 650°F$) and post-accident conditions with an extended duration of recirculation flow passing through the valve (i.e., integrated radiation dose could be a concern."

The conclusion of these responses is that the vendor, style, or valve type is not that important to better performance. Given that the use of a more leak tight soft seated valve may not be feasible, metal to metal seats are prone to some leakage. This leakage can be minimized, such that operational impact is reduced, through repair of valves.

8. What are the typical leak rates as determined by routine IST leakage testing for ECCS check valves?

In order to result in adverse operational impacts, significant (but still may be very low) leakage would have to be occurring through two or more check valves. For example, the leak rate through one 10-inch valve, by itself may be high, but the leak rate through another one in series may be low enough to preclude significant operational impacts. On the other hand, if one of a series of valves is leaking more than others, it can result in adverse impact because differential pressures are not high enough to provide more leak tightness to a valve. For example, a two-inch valve on the SI cold leg could impact proper seating of the 10-inch vertical check valves. The purpose of this question was to ascertain whether there is a correlation of check valve testing results to operational impacts. This is to determine if lessons can be learned, such that test results can be utilized in this regard.

Unlike at Seabrook Station, most plant sites do not attempt to establish actual leak rates during IST leakage rate testing. In many cases, once the conditions are established such that the testing demonstrates that the valves meet the technical specifications, the "call" for the test leakage is made, and the test is terminated. Furthermore, many plants conduct tests on groups of valves, rather than attempt to establish a flow rate for each valve.

The range of IST flow rates is generally in the gpm range, as opposed to the gpd range, which can result in operational problems. Often, the need for corrective maintenance is determined by operational impact as opposed to test results. Special tests are sometimes conducted to verify that a suspect valve is leaking once its leakage results in adverse operational impacts.

Notwithstanding the above, attempting to establish actual leakage rates as Seabrook Station does, provides more evidentiary data to establish the source of leakage, if it does manifest itself in operational impacts.

The conclusion of the evaluation of the responses to this question is that IST leak test results are not being broadly used to ascertain the condition of check valves. Leak rate test results can provide valuable information. Conversely, plant sites have been successful in the identification of which valves are in need of repair, even with less precise IST testing data.

9. What causes have you determined for ECCS check valve leakage?

Problems cannot be effectively corrected unless causes are determined. The purpose of this question is to determine the industry-learned causes for ECCS check valve leakage. This information may lead to more effective solutions to the problems, including design and/or repair efforts.

The predominant causes cited involved the smoothness of seating surfaces, the need for high differential pressures to achieve and maintain leak tight seating, and distortions of seating surfaces during installations.

10. Has crud ever been determined as a possible cause or contributor to ECCS check valve leakage? If so, how was crud determined to be a contributor? Have special crud cleanup evolutions been implemented and successfully corrected the problem?

The IST data for some of the first out 10-inch check valves at Seabrook Station has demonstrated alternately significant leakage and then zero leakage. This may be indicative of transient crud depositions. If crud gets deposited on the seat, it may result in leakage. The leakage may later be dislodged during a flow test or transient and the leakage rate returned to zero. Other possible causes for changes in leak rate test results are limitations in the ability to apply and maintain the required differential pressure, slight differences in seating and its effect on location of imperfections that may result in leakages, or instrumentation limitations. This question was to determine the experience in the industry as to whether transient crud deposition is a viable cause or contributing mechanism for ECCS check valve leakage.

Eight (8) of the thirteen (13) respondents indicated that crud is not a possible cause or contributor to ECCS check valve leakage. Based on telephone conferences and some written responses, respondents assumed that if the crud was not visible, then it was not a contributor. For example, one respondent indicated that it was not of concern, as crud had not been found upon disassembly. However, it is not necessarily the case that crud would be found, because the crud may be removed by a subsequent flow transient. Of the five (5) remaining responses, evidence of dirt, corrosion layer, or film was noted in four cases and the fifth case indicated suspicion of crud, but it was not a definitive cause.

The conclusion from the above responses is that the contribution of crud to check valve leakage is a viable cause, although it is difficult to definitively establish. Flushing may alleviate the problem; this could not be done on line for first out check valves. It is in effect, however, accomplished via system flow balancing and check valve flow stroking during refueling outages.

11. Have you repaired check valves or implemented new PMs that have been effective in reducing ECCS check valve leakage?

The purpose of this question is to determine best practices in the industry. Note that excessive maintenance can potentially be a problem. Therefore, for practices to be considered among the best, an assessment of correlation of these practices to demonstrated success in minimizing ECCS check valve leakage is necessary.

None of the thirteen (13) respondents indicated that they had an intrusive PM program for ECCS check valves. However, one plant must disassemble and inspect one of the second out 10-inch accumulator check valves in accordance with their IST program (since they cannot full stroke these valves). Most plants have repaired valves, in response to identification of check valve leakage. One plant is implementing a program to reduce the number of disassembly and inspections and doing more non-intrusive PMs. One plant has repaired check valves in other systems, where leakage had been of concern, but there have not been significant check valve leakage concerns in the ECCS system to date.

Four (4) of the plants specifically stated that they conducted repairs based on operational impact during the cycle. Two of these four did testing during the outage to confirm that a particular valve was leaking and was to be repaired during that outage. As evidenced by the response to question 5 and other responses to question 11, other plants also rely on operational impact to make decisions on repairs to valves. One plant's experience illustrates the importance of operational experience versus IST test data. Based on accumulator out-leakage during operation and high SI header pressure (about 600 psig), one of the SI discharge header 2-inch check valves was disassembled and repaired. Its measured IST tech spec leakage before and after this repair was 0 gpm. The repair consisted of a light lapping of the seat. This effort (or possibly flowing through the valve that removed crud) was successful in terminating accumulator out-leakage.

Two plant sites, both with good experience with minimizing ECCS check valve leakage, provided input concerning the quality of repairs and inspections. One plant indicated that they now utilize Prussian blue & try to achieve 360° contact across 1/4 – 3/8" radial surface. They utilize both a Prussian blue and a neo-lube blue check for Kerotest 2-inch piston check valves to provide redundant checking. They now place significant emphasis on ECCS check valve maintenance and on inspections for checking tolerances (hinge-pin, clapper, ...). There is also an increased focus on foreign material exclusion throughout the primary systems. One plant has refined its maintenance procedures since discovering operational problems associated with ECCS check valve leakage. These refinements primarily deal with the valves' seating surface condition, ensuring the best possible disc to seat contact as well as maintaining vendor tolerances.

The conclusion from the above response is that careful inspections, maintenance and checks on the competed work effort are necessary to ensure that the contact between the seat and disc is as complete as possible. Decisions to repair are based on operational impact and validated by IST results.

12. Have you replaced ECCS check valves (like for like) and thereby successfully reduced leakage?

In 1990 and 1991, Seabrook Station changed out parts to SI–V130, such that essentially it became a like for like replaced valve. As discussed previously, manifestation of operational impact of leakage persisted. During the sixth refueling outage, an attempt was made to repair this valve, and it had to be replaced. Due to obsolescence of this valve, the replacement valve was not exactly like-for-like, but is still a piston style, 2-inch check valve. This valve and the other three existing similar valves are, thus far in this cycle 7 (May 1999 to date), exhibiting good performance. The purpose of this question, which was formulated prior to the replacement of SI–V130, was to determine what the industry experience was with check valve replacement.

Out of the thirteen (13) respondents, three (3) have essentially replaced like for like valves, and a fourth respondent plans to replace valves. In three (3) of these four (4) cases, the 2-inch piston check valve (similar to SI–V130 for Seabrook Station) have or are planned to be replaced. The other plant site has replaced a first out, 10-inch check valve. As discussed in response to question 13, some plants have done or are planning modifications to some check valves.

The respondent at one plant site indicated action that is very similar to the action taken at Seabrook Station for SI–V130. They have replaced the 2-inch cold leg injection check valves in both units with a later generation of piston check valves from Edwards. The replacement was with fourth generation check valves that do not have seal welded bonnets, but is considered a like-for-like replacement.

The respondent at the another plant site indicated similar actions for 2-inch piston type check valves. They had four (4) 2-inch check valves scheduled to be replaced for the Unit 2 plant during the refueling outage in the spring of 1999. This included two SI to hot leg Kerotest check valves due to leakage data trending, and two SI to cold leg Kerotest check valves that were to be replaced as determined necessary by special testing once the unit was shutdown. A follow-up discussion with this respondent determined that three of these valves were, in fact, replaced. In addition, two SI to cold leg Kerotest check valves were replaced on Unit 1 in the spring of 2000. These replacements were successful in alleviating high pressure at the discharge of the SI pumps and accumulator level decreases.

The above responses demonstrate that replacement of check valves (like-for-like) has been successful, that damage has been found in the industry beyond the ability to repair valves, and that 2-inch piston check valves have more often had to be replaced than other ECCS check valves.

13. Have you implemented any design modifications to reduce ECCS check valve leakage or to mitigate its effects, such as check valve replacements with different styles, system design pressure changes, SI test header back-pressure control, venting systems, alarms, etc.?

The purpose of this question was to determine industry experience with regard to design modifications to reduce ECCS check valve leakage and to mitigate its effects. This information provides input into possible design enhancements for Seabrook Station. Six (6) out of the thirteen (13) respondents indicated that they have or plan to implement such design modifications.

Two (2) of the respondents made modifications to check valves (one was successful and the other not). One (1) is considering a design change to the 2-inch check valves. These three cases are discussed further, below.

The successful design change to check valves occurred at one of the plant sites. This involved the 10-inch check valves. These were non-Westinghouse bonnet-hung disc valves. The design change is unique to this style of valve. It does not apply to the other plants surveyed, which utilize Westinghouse swing check valves in this application.

An unsuccessful design change for check valves occurred at another site. Upon disassembly of 2" piston check valves, it was found that they had damaged seats. On four (4) of these 2-inch valves, the seat angle was modified. However, the leakage appeared to get worse, so the seat configuration was returned to the original design.

One plant site is considering a design change to the SI to cold leg check valves. They are generally unhappy with the performance of the Kerotest 2-inch piston checks and are considering a replacement style and model. Another plant site in this utility system has the lead on this decision. A possibility is an Anchor Darling 1878 swing check valve. A non-Westinghouse unit has experience with this style of check valve.

Two (2) of the respondents have increased the design pressure of the SI pump discharge piping header. A third respondent indicated that this was pursued by its Unit 2 plant as a contingency to a continued need to vent off the SI discharge piping to reduce pressure (via a plant modification), but was not implemented because of the success in reduction of pressure by the replacement of four 2-inch SI cold leg check valves.

Four (4) respondents have implemented plant modifications to mitigate the consequences of ECCS check valve leakage. In all four (4) of these cases, the design modifications are not currently being utilized because the ECCS check valve leakage has been reduced such that the operational impact is currently alleviated. In these four cases, design changes were made to allow:

- Draining leakage on the back side of check valves via the SI test header to a waste tank
- Continuously venting SI discharge piping to less than 1750 psig
- Venting pressure in both the RHR and SI discharge piping—included a leak-off line with an orifice, routed to the containment sump
- Venting RHR discharge piping via regulating/relief valve to the holdup tank

The operation at Seabrook Station, in the fall of 1997, of continuous venting of the SI discharge piping via SI–V131 (see figure 2) to the PDT is similar to the mitigating design changes described at other plants. Sufficient backpressure must be maintained to avoid draining the RWST.

14. Do you require a containment entry to sample the accumulator boron concentration? Do you have the capability to sample the accumulators from outside the containment?

This question is based on the Seabrook Station experience with in-leakage to accumulators. This resulted in volume changes such that sampling and analyzing for boron was required more frequently. In the case of Seabrook Station, there is no capability to sample the accumulators from outside containment. Therefore, sampling of the accumulators requires a containment entry.

A recent study of the number of containment entries determined that Seabrook Station conducted far more containment entries than the rest of the industry. The frequency of accumulator sampling results in a potential impact on this statistic. If, on the other hand, there is the capability to sample from outside containment and that capability is utilized, an increase in the requirements to sampling because of accumulator volume changes does not impact the number of containment entries.

It was found that all thirteen (13) of thirteen (13) respondents indicated that they had the capability to sample the accumulators from outside containment and that they utilized this capability. Therefore, Seabrook Station may be the only Westinghouse four-loop plant without this capability.

The implementation of a design change at Seabrook Station to provide this capability would be very costly, especially because it is a significant back-fit. The location of the local sample connections is in a relatively low dose rate and low temperature area. The benefits of such a design change, therefore, are not warranted based on the costs.

15. Have you ever raised the boron concentration (e. g., because of dilution from check valve leakage) in the accumulators? If so, what methodology

was used—feed and bleed, drain and fill, etc.?

Seabrook Station needed to raise the boron concentration of accumulator "D", in November of 1998, because of in-leakage from the RCS. The normal design path for this was via drain and fill through the same connection. During shutdowns, this method had been found to be inefficient. Furthermore, there was a concern that the change in accumulator pressure associated with a change in level could result in further upset of check valve seating. Therefore, a procedure change was made that involved a feed and bleed method. The feed path was the normal accumulator fill path and the bleed path was through a sample connection to a hose routed to the containment sump and ultimately to the floor drain tanks. This allowed for a sufficient increase in boron concentration within the eight (8) hour allowed outage time of the accumulator. The purpose of this question was to ascertain what similar experiences may have occurred within the industry, such that additional lessons may be learned.

An additional question was asked, via telephone conferences or follow–up telephone calls, concerning the technical specifications for boron concentration of the RWST and the accumulators. If the lower end of the band of the required accumulator boron concentration is sufficiently below the RWST required concentration (the source of fill and makeup to the accumulators) this would eliminate or minimize the need for raising the boron concentration. The Seabrook Station required range for boron concentration is 2600 to 2900 ppm for the accumulators and 2700 to 2900 ppm for the RWST. Although the lower end of the range for the accumulators is lower (100 ppm) than that for the RWST, it is not significantly low enough to preclude the need for raising the boron concentration with a sustained period of accumulator in-leakage.

Five (5) of the thirteen (13) respondents indicated that they had conducted at least one evolution of raising the boron concentration of an accumulator. Three (3) other respondents indicated not in recent history or not in their recollection. The remaining five (5) indicated that they had not performed such an evolution.

Of the five (5) respondents that had raised the boron concentration of accumulators, three (3) indicated that the evolution was done by feed and bleed and two (2) indicated a drain and fill evolution was performed. Generally, a long time is needed to accomplish these evolutions. There was an event in 1987 where the plant had to shutdown due to accumulator in-leakage at a rate such that this feed and bleed method was not sufficient to increase the boron concentration quick enough to return the accumulator to operable status.

These plants used sample lines (remotely operated unlike at Seabrook Station which provides for local sampling only) for the drain/bleed path. The Seabrook Station method used a hose connection attached to the sample connection for the bleed path. The advantage of this is that a feed and bleed rate of 40 gpm as opposed to 1 to 5 gpm was achieved. This allowed for a shorter time to achieve the desired boron concentration increase.

Of the five (5) respondents who provided boron concentration requirements, only one had a significant difference (more than 200 ppm) between the low end of the accumulator and RWST boron concentration. This plant has a difference of 400 ppm for the lower end of the band.

They stated: "Our Tech Spec Accumulator boron concentration range is 2000 to 2600 ppm, with RWST limited to 2400 to 2600 ppm, so we don't expect to ever have a concentration problem on line unless gross leakage occurs."

Conclusion

The issues at Westinghouse 4-loop plants vary, but ECCS check valve seat leakage is inherent in the design of the ECCS connections to the RCS and SI accumulators. The manifestation of ECCS check valve leakage impacting operation of these plants is prevalent. Vendor, style, or valve type is not that important to better performance. Check valves with metal to metal seats are prone to some leakage.

The most common comment of Westinghouse 4-loop plants in seating check valves is that a significant differential pressure across the valve is required to seat them. Establishment of a large differential pressure is necessary to seat the Westinghouse 10-inch vertical swing check valves. Once seated, these valves have very good leak tightness.

Leakage through the 2-inch piston check valves in the SI system is the most prevalent problem throughout the industry. Considering their smaller size, they are not as leak tight as swing check valves. Proper alignment of parts for proper seating appears to be an issue. In addition, damage of seating surfaces is more frequently encountered. Replacement of these valves has been common, as repairs are often unsuccessful.

Check valve maintenance, including lapping of discs and ensuring vendor-recommended tolerances are maintained, has been effective in reducing check valve leakage. The plants that have had minimal problems pay careful attention to maintenance. Careful inspection techniques (e.g., effective blue checks and tolerance checks) are also important.

The conclusion of this study is that ECCS check valve seat leakage presents significant plant operational challenges. Plant staffs must be diligent in efforts to reduce, monitor, and mitigate the effects of such leakage. Some of the resultant key recommendations to enhance diligence in this regard are:

- ECCS check valves should be disassembled for inspections and repairs at the subsequent refueling outage following determination that trends for IST leak rate tests and operational experience indicate that they are contributing to significant operational impacts.
- The design pressure, and hence setpoints to relief valves, of the SI pump discharge piping should be increased (some plants have already implemented this change).
- Technical specification changes regarding the accumulators (boron concentration and allowed outage time) should be pursued. They provide for easier completion of compensatory actions in the event of boron dilution of accumulators caused by check valve back-leakage. A technical specification change of this nature was developed by the WOG.
- Detailed procedures for seating ECCS check valves, using the SI pumps and the test header, should be developed.
- Plans should be developed to direct undesirable check valve leakage into the SI or RHR to a more desirable

location, such as the containment sump, holdup tank, or refueling water storage tank.

References

1. LER 84–007–00 of Fort Calhoun, dated June 15, 1984
2. LER 86–066–00 of Wolf Creek, dated January 9, 1987
3. LER 90–022–0 of Salem Unit 1, dated August 15, 1990
4. LER 93–010 of Comanche Peak 2, dated October 22, 1993
5. IN 97–40, on Waterford 3 and Sequoyah 1, dated June 26, 1997
6. IN 91–50, S1, on Waterford, Sequoyah, and H.B. Robinson 2, dated July 17, 1997
7. LER 95–01–01 of H. B. Robinson, dated January 8, 1998
8. LER 98–002–00 of Seabrook Station, dated January 22, 1998
9. LER 98–002–01 of Seabrook Station, dated March 20, 1998
10. LER 97–017–01 of Byron 1, dated August 31, 1998
11. LER 1997–017–01 of Fort Calhoun 1, dated March 20, 1998
12. LER 97–005–01 of Braidwood 2, dated March 25, 1998

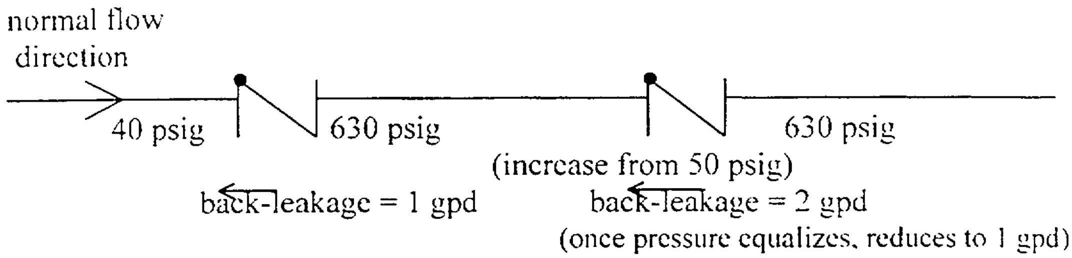

Figure 1. Check Valve Back-Leakage Effect

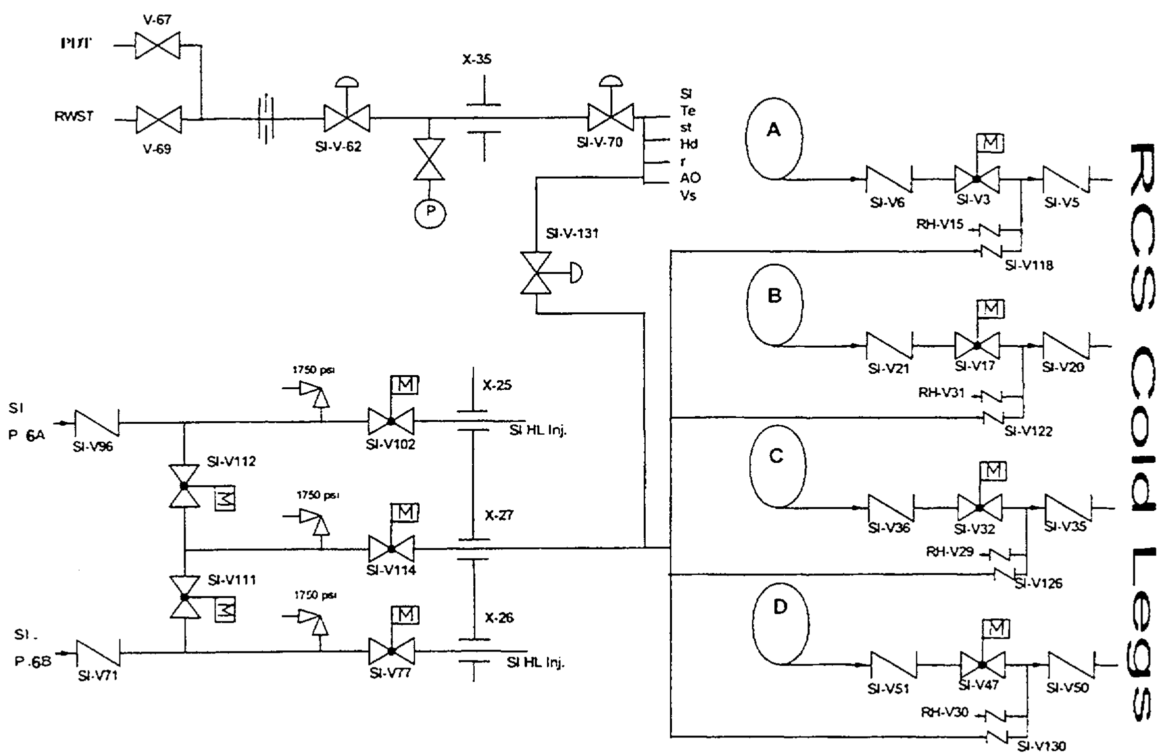

Figure 2. Safety Injection Cold Leg Check Valves and SI Test Header

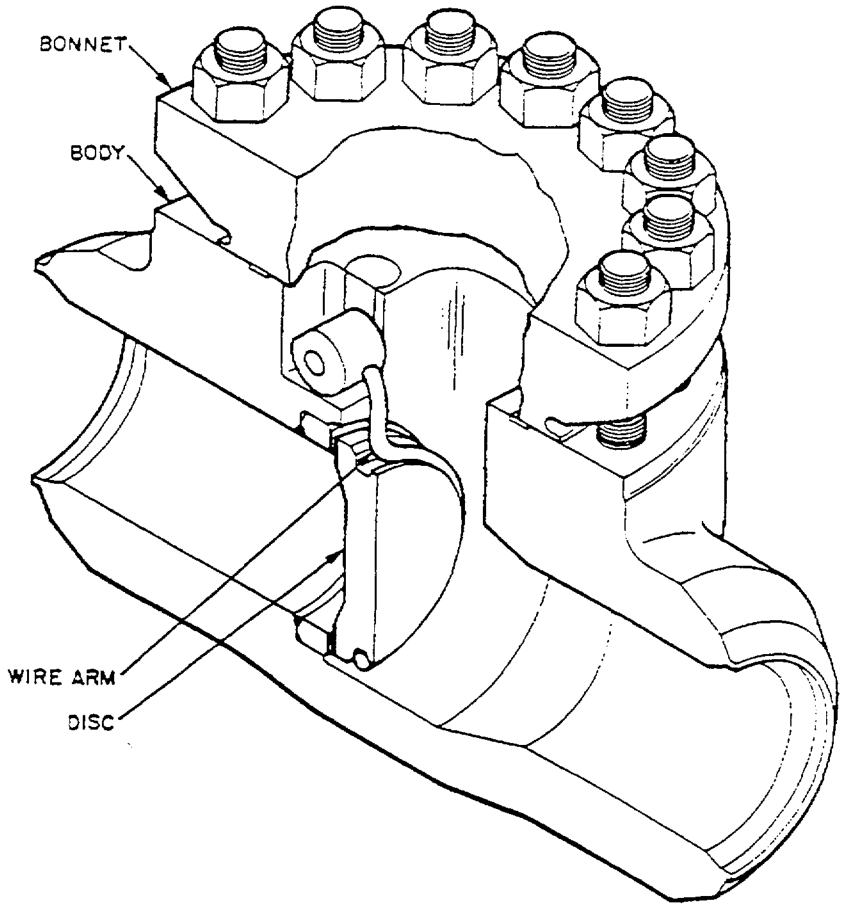

Figure 3. Westinghouse 10 and 6-Inch Vertical Swing Check Valve

Table 1. Industry Operating Experience on ECCS Check Valves			
Report	**Report Date**	**Plant**	**Event**
LER 84–007–00	6/15/84	Fort Calhoun	RCDT relief valve lifts due to check valve leakage, resulting in a radiation-induced ESF actuation.
LER 86–066–00	1/9/87	Wolf Creek	Accumulator Level Drops below Tech. Spec. Values due to check valve leakage because of incomplete seating of the pivotal discs.
LER 90–022–0	8/15/90	Salem Unit 1	Relief Valve Leaking discovered as a result of investigation of accumulator level drop through check valves (two accumulators experienced reducing level at total of less than 1 gpm).
LER 93–010	10/22/93	Comanche Peak 2	Required boron samples not taken when check valve leakage resulted in accumulator level increases of 15 percent over a 24 day period
IN 97–40	6/26/97	Waterford 3 and Sequoyah 1	Nitrogen voiding in LPSI piping due to check valve leakage from SI accumulators
IN 91–50, S1	7/17/97	Waterford, Sequoyah, and H.B. Robinson 2	Same events as described for Waterford and Sequoyah in IN 97–40. Check valve leakage testing at H. B. Robinson results in a partially voided cold leg accumulator injection line.
LER 95–01–01	1/8/98	H. B. Robinson	The use of SI header renders SI inoperable.
LER 98–002–00; 01	1/22/98; 3/20/98	Seabrook Station	An error in the use of ultrasonic detectors for flow measurement resulted in the potential for SI pump train A to exceed its maximum flow requirement (pump run-out concern), upon signal failure of the "B" train solid state protection system (SSTs), and the test header valve SI–V131 open. SI–V131 had been open to minimize the effects of check valve back-leakage. This LER was later withdrawn (Rev. 01), based on an engineering evaluation that concluded that an assumed postulated failure of NNS piping within the test header, was not a credible event.
LER 97–017–01	8/31/98	Byron 1	Check valve leakage from accumulator is 2.8 gpm, exceeding tech. spec. limit of 1 gpm. The leaking valve was 2-inch Kerotest Y-pattern piston check valve.
LER 1997–017–01	3/20/98	Fort Calhoun 1	Nitrogen voiding from check valve leakage resulting in drain of accumulator back to LPSI header.
LER 97–005–01	3/25/98	Braidwood 2	Check valve leakage during surveillance results in SI relief valve lifting and its damage.

High Stress-Strain Hysteresis in 400-Series Stainless Steels

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Abstract

For typical steels, strain is directly proportional to stress throughout the elastic range (Hooke's law), and nonlinearity and hysteresis values in the stress-strain relationship are small as compared to working strain levels. This principle is widely relied on in sensor design and mechanical testing, and is the basis of most strain gage use. However, a recent sensor test program showed more significant hysteresis levels in a group of 400-series martensitic stainless steel valve stem specimens. Although this characteristic doesn't affect the normal functionality of the parts, it can impact strain-based force and torque measurements. The effect appears as a simple mechanical hysteresis, fully reversible in cyclic loading. It shows no time dependency, but is correlated to load event history.

This paper presents the test program data, showing the hysteresis effect in full-cycle material tests and in simulated valve signatures. The impact of the effect on stem force measurements and an analysis technique to minimize its influence are described.

Introduction

Machine forces are often measured by observing strains in their components. This method is common practice in the testing of rising-stem valves, where stem strain is measured, by either strain gages or surface extensometer, to determine force or torque. The certainty of data obtained by this method is determined by three factors:

a) the characteristics of the strain instrument and its circuitry;

b) the correctness of the material constants used in the stress/strain transform, and;

c) the relation of load to strain in the valve stem material.

Instruments are typically calibrated and tested, and material constants for valve stems have been researched. The qualities of stress–strain correlations in valve stem materials, on the other hand, are less well known. The general assumption has been that Hooke's Law is an accurate stress-strain model in the elastic range for the common stem materials, as it is for most steels and other strong metals. For instance, linear stress-strain correlations (expressed as first-order span or sensitivity values) for load cells made from various steels are typically accurate to within one or two tenths of a percent of scale, a small error as compared to the other limits of measurement accuracy. Most steels used in machine parts have stress-strain characteristics similar to this. However, a recent study has shown hysteresis values of two to four percent in some martensitic stem material samples. This characteristic does not impair the functionality of the valve stems, but it can affect some strain-based measurements.

Background and overview

The test program reported here was initially an instrument validation project. The instruments being tested, called Easy Torque/Thrust Sensors or ETTRs, are specialized assemblies of bonded resistance strain gages. The sensors were installed on a group of cylindrical steel tensile specimens (including material from actual valve stems), that were instrumented with conventional strain gages and prepared for use in a biaxial material-testing machine. The strain gage and sensor readings were compared to the testing machine's integral calibration-standard load and torque cells under a variety of loading patterns.

The early tests showed large, consistent errors on 400-series stainless specimens that initially appeared to be an elastic asymmetry or difference between the elastic moduli in tension and compression. The disagreement was between the standard cells and the strain instruments; the ETTR sensors and strain gages all agreed closely, indicating that the cause was either a problem with the data from the machine's integral cells or the strain behavior of the specimen metal.

To check the possibilities that the integral cells were malfunctioning or that the test machine system's internal signal conditioning might be somehow skewing the data, the system was recalibrated by the manufacturer, MTS. This calibration is routine, but was not due for several months. The calibration was done by mounting torque and thrust standards, traceable to the National Institute of Standards and Technology (NIST), in place of the tensile specimens and calibrating the integral cells in situ. The system's signal conditioning circuitry was also checked during this process. The recalibrations showed that, while there are slight asymmetries and hysteresis effects in the integral cells, they are within their specifications, and the values account for less than one tenth of the observed difference in test readings.

When the originally planned instrument tests and the equipment checkout were completed, a second test effort was conducted to characterize the apparent stress-strain asymmetries or hysteresis effects, define their cause and mechanism, and evaluate their influence on the earlier instrument tests as well as on field test data. A standard tension-compression load cell, with its output acquired as a low-level signal, was fixtured in series with the tensile specimen grips to provide a second, independent standard force reading. A series of full-cycle load tests was performed using the thrust-only load patterns from the earlier testing. Lastly, a group of load profiles simulating the events, rates, and magnitudes seen in typical gate and globe valves was run to relate the observed effects to valve testing scenarios.

Materials and Equipment

Materials testing machine

The materials testing machine used in this study is a stationary, computer-controlled hydraulic unit manufactured by MTS, located in the calibration laboratory at the Crane Nuclear, Inc. facility in Kennesaw, Georgia (Figures 1 and 2). The machine is mainly used for calibration of load and torque sensors, and incorporates NIST-traceable calibration standards as integral load and torque cells, which are also used for control feedback.

The machine incorporates one axial and one torsional actuator, both acting on the central axis of the specimen mount. These are capable of 110,000 lb. tension and compression loads and 4100 ft.-lb. torques, independently or in combination; however, the hydraulic specimen fixtures used in this test program are limited to 50,000 lbs. and 1650 ft.-lbs.

Instruments and data acquisition

The primary instruments used in these tests were load and torque calibration standards. In addition to the MTS machine cells, a tension-compression shear web load cell was installed in the load train, providing an independent standard force reading. Data was taken from bonded resistance strain gages on each specimen; these were installed and used in accordance with approved procedures. Data was also collected from the ETT^R sensors (originally the subject of the tests), which were installed and controlled in accordance with a test-specific validation procedure. All data was acquired with a Crane Nuclear Universal^{TM} Diagnostic System (UDS) and its associated cabling and peripherals. All controlled measuring and test equipment used in the test was function-tested and in current calibration.

Test specimens

The ten valve stem material specimens used in the program vary widely in history and control. Table One lists the specimens and their dimensions and materials. The A, B, and C specimens were obtained from the Crane Nuclear facility in Illinois, and were machined from a single nuclear-grade 410 stainless valve stem in their inventory. A certified mill test report was supplied with them, and indicates that the piece was conditioned and treated as typical for valve stem use.

The other seven specimens are all from a collection of test bars maintained for general use with the MTS machine, and were machined and heat-treated during various test programs over a ten-year period. There are no specific, controlled records of purchase, treatment, or testing for these pieces, and those that were heat-treated after purchase were handled as commercial material. The martensitic pieces are known to be commercial 410 or 416, but may or may not be in the condition range typical of valve stems.

Test Design and Methods

After the initial investigation identified a hysteresis effect occurring in the 400-series steels, the test program objectives were expanded to include a second phase. This second phase was planned for examination of these effects and to provide sufficient information to support instrument accuracy statements and evaluation of other potential impact.

The first-phase data was reviewed to determine the need to perform this testing for torque. The hysteresis characteristic was visible in torque in most full-cycle tests, but was generally much smaller as a percent of full scale than the effect in thrust. The general indication is that statements of hysteresis error magnitude for thrust will be bounding for torque. Based on this and the limited time-scope and budget of the project, the hysteresis evaluation was performed with thrust-only tests.

Cyclic testing

A cyclic test set was done on each stem with the same load patterns used in the original instrument tests. These cycle from full-scale tension to full-scale compression, with three continuous cycles per test (see Figure 3). The full scale value for most stems is the thrust load that produces a 20,000 pound per square inch (20 ksi) stress in that diameter. The exception is that the E, K, and D stems were tested at full scale values of +/− 50,000 lbs. because they are large enough in area that their 20 ksi loads would exceed the machine's 50,000 lb. gripper limit.

Cyclic tests of this type deliver the maximum value of hysteresis for a tensile specimen or a tension-compression instrument, since nearly all load-history-dependent effects are proportional to the magnitude of causal load events. An error-versus-load plot of the data generates a hysteresis curve (see Figure 4). The hysteresis values from these full-range-transition tests are shown in Figure 5.

Valve profile tests

The final stage of testing was performed to define the levels and areas of this hysteresis effect in actual valve tests. The full-range, bi-directional cyclic tests provide a good measure of maximum hysteresis values for a specimen, but those extreme load cycles would not typically be seen in a valve test.

Six valve load profiles were programmed for the MTS machine, each designed to represent two consecutive Open-Close-Open (OCO) full-stroke tests. These load profiles are control files for the hydraulic loading system, written as a series of load values, durations, and ramp rates. All were written for the 1.5" diameter stems, with 36,000 lb. total closing force values.

Profiles VV1 through VV5 (Figures 6 through 11) are models of thrust signatures for five generalized valve types, ranging from a shallow-disk globe valve with no unseating force to a flex-wedge gate valve with an unseating force of 30,000 lb. and a 2500 lb. unseating rebound. These five profiles all represent static (no fluid flow) test conditions. The sixth profile, DP1 (Figures 12 and 13), models the thrust signature of the intermediate flex-wedge valve type used in profile VV3 with a 16,000 lb. DP thrust load in both opening and closing. These profiles were each run on the C stem with the additional standard load cell in place in the MTS load train. Figures 6 through 13 illustrate the six load profiles and the resulting error in strain-based thrust as compared to the standard cell.

Analysis

Method

Since this was believed to be a basic mechanical effect, analysis was kept as straightforward as possible. The test data was analyzed by performing a calibration of each strain gage and ETT^R circuit against calibration-standard load cells with data from the cyclic tests, using full-cycle data sets and a linear regression fit. The resulting span numbers were applied to the strain data traces. With the strain and standard traces in like units, the traces were zeroed and each standard trace was subtracted from the accompanying strain instrument trace point by point (Error = Instrument − Standard). This method removes any scaling error, and assuming that any offset is removed, leaves the combination of (non)linearity, hysteresis, and (non)repeatability (LHR error) as a residue. All reported error is calculated by this method, and all loads and error values are reported in percent of full scale, which is set for each stem at the lesser of either the 20 ksi axial stress load or 50,000 lbs.

The residual errors were plotted for the full data set. The resulting maximum hysteresis values (depth of the loop in Figure 4) were calculated for each stem; these are shown versus stem diameter in Figure 5. Analysis of groups of multiple tests of each stem indicated that the effect was highly repeatable and load-history dependent, and is clearly a stress-strain hysteresis when plotted as error-versus-load. Values are approximately an order of magnitude greater than those typically seen in other stainless steels.

Results

This effect is present in all of the 400-series stainless specimens of the study, appearing at two percent in the known stem material (A, B, and C). There is a wide scatter among the hysteresis values in E, F, P, G, and H, but the effect is consistent and repeatable on each specimen. The tests indicate no significant hysteresis levels in the 17–4PH or 316 stainless steels of stems D and K. It is important to note, also, that the data from stems A, B, and C is more significant than that from the other samples because of its known source and history, but that it represents only one piece of material.

This phenomenon is apparently a structural set-type hysteresis, a load-history-dependent shift in the stress-strain curve that does not show time dependency or recovery. Two known hysteresis-like effects, thermoelasticity and primary "fast" creep, were tested for and ruled out; both would show time-dependent behavior at static loads, and none was observed in long-term load tests. The Bauschinger effect, a yield-strength shift phenomenon seen in inelastic strain of some steels when cyclically yielded, was also ruled out since no permanent set occurred. However, the mechanism of this large hysteresis may be due to a similar effect at sub-plastic strain levels, since the observed phenomenon is basically a limited, reversible micro-yielding action.

The action of this effect is fairly complex, since it is a function of load history in both sign and magnitude, but is highly repeatable for a given load profile. The effect on the accuracy of a particular point in the data set is also dependent on where the trace has been zeroed, since the hysteresis effect can be thought of as a motion of the offset or "correct" zero value. In full, symmetrical cycles, the sign and magnitude of the error can be seen to be a direct function of the sign, direction of change, and magnitude of the load, as shown in Figure 4. It follows that any point on that curve could be made "correct" by zeroing at an appropriate level, at the expense of points at the opposite side. For instance, in that full-cycle curve, the error could be minimized at both full scale values by zeroing at a value that splits the force delta between measured full-scale values, leaving the areas around the zero points with errors around half of the depth of the hysteresis curve.

It can also be seen that zeroing at one of the zero-load points will give the two full-scale areas that half-depth error value and put the error at the opposing zero at the maximum value, the full depth of the curve. In the more complex valve load patterns, this is harder to sort out, but two principles hold throughout all of the data:

a) For a valve test with both zero-load points identifiable (a tension-to-compression or "closing" stem nut shift, and a compression-to-tension or "opening" stem nut shift), the maximum value of the structural hysteresis error is the force delta between the two zero-load points, and;

b) For any given small area in the load pattern, there is a "correct" place to zero the trace, at one of the zero plateaus or in the range between their magnitudes, that eliminates the error caused by structural hysteresis for that area.

In the six valve patterns tested (see Figures 6 through 13), hysteresis was generally correlated to the unseating (tension) load history. Hysteresis was minimal in the VV1 "globe valve" tests, to the point where it fell down among the normal LHR/noise error band. Maximum errors occurred in the VV5 "high-unseating flex wedge" tests, and slightly lower errors were seen in the VV2 through VV4 and DP1 flex wedge tests. In all cases where the effect rose above the background LHR/noise error band, there is a clear, repeatable pattern to the error. When the trace is zeroed at the closing, tension-to-compression zero plateau, the above-background error occurs within a zone from the onset of closing force relaxation to the end of the unseating peak. The greatest error occurs at the opening zero plateau, just after unseating. The magnitude of the error is roughly correlated to the magnitude of the tension (unseating) load as a percent of the total seating load.

Conclusions

1) This hysteresis phenomenon appears to be a property of the 400-series materials tested. Any strain-based force or torque measurement on material with this characteristic will be affected, although the accuracy statements of some transducers that were validated on 400-series material specimens will include this effect.

2) For any valve with relatively little load on the tension side of zero load, such as a typical globe valve, this hysteresis effect is likely to be negligible.

3) The maximum material hysteresis error value in a continuous open-closed-open or closed-open-closed valve test is defined by the difference in value between the opening and closing zero plateaus. This is believed to be true for any test where both the opening and closing zero-load values are available.

4) Valve profile testing on known 410 valve stem material indicates that the effect of this hysteresis error can typically be contained to a specific zone by zeroing on the closing stem nut shift of an Open-to-Closed (OC), Closed-Open-Closed (COC), or Open-Closed-Open (OCO) test. This zone, from the onset of closing force relaxation to the end of the unseating peak, can be referred to as the relaxation/pullout zone.

5) The above-normal hysteresis values seen in the relaxation/pullout zones in valve profile tests are bounded by the two-percent maximum hysteresis value obtained from the slower full-cycle tests.

6) When valve profile test traces are zeroed at the closing tension-to-compression zero plateau, normal sensor component accuracies bound the data outside of the relaxation/pullout zone.

Table 1

Stem	Diameter	Material	Max. Full-Cycle Hysteresis, %FS
A	1.5"	Controlled 410 stem material	2.2
B	1.5"	Controlled 410 stem material	2.0
C	1.5"	Controlled 410 stem material	2.0
D	2.5"	Commercial 17–4PH	0.2
E	2.0"	Commercial 400-series SS	2.8
F	1.5"	Commercial 400-series SS	2.7
G	0.875"	Commercial 400-series SS	3.8
H	0.625"	Commercial 400-series SS	3.7
K	2.0"	Commercial 316 SS	0.1
P	1.12"	Commercial 400-series SS	0.5

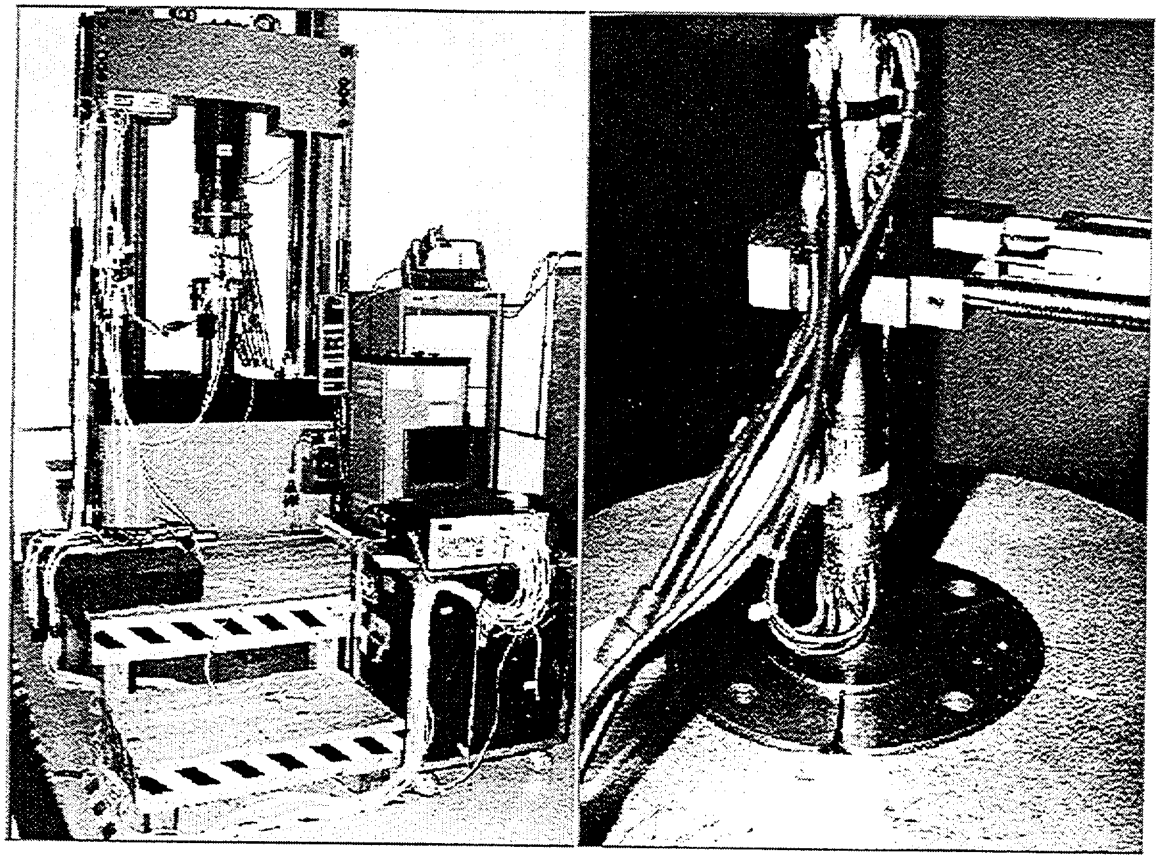

Figure 1: Test area and acquisition system – Stem H

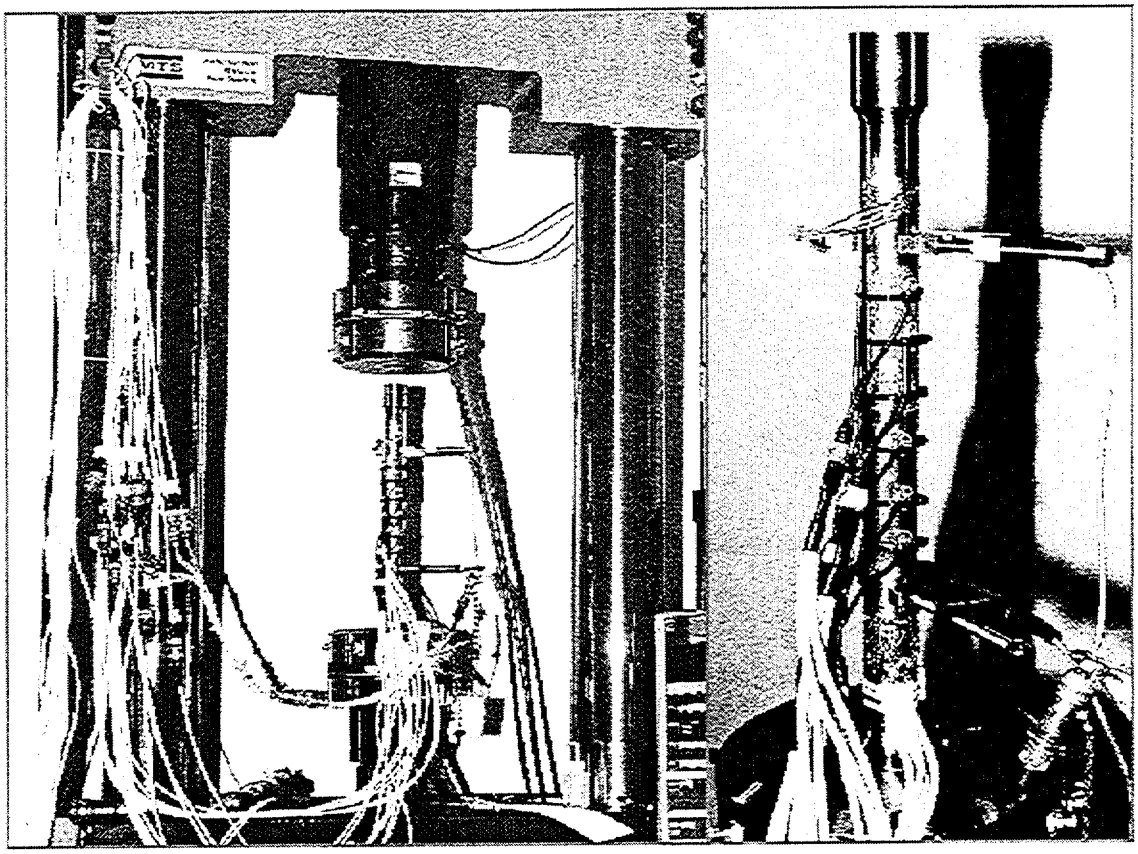

Figure 2: Test area and acquisition system – Stem C

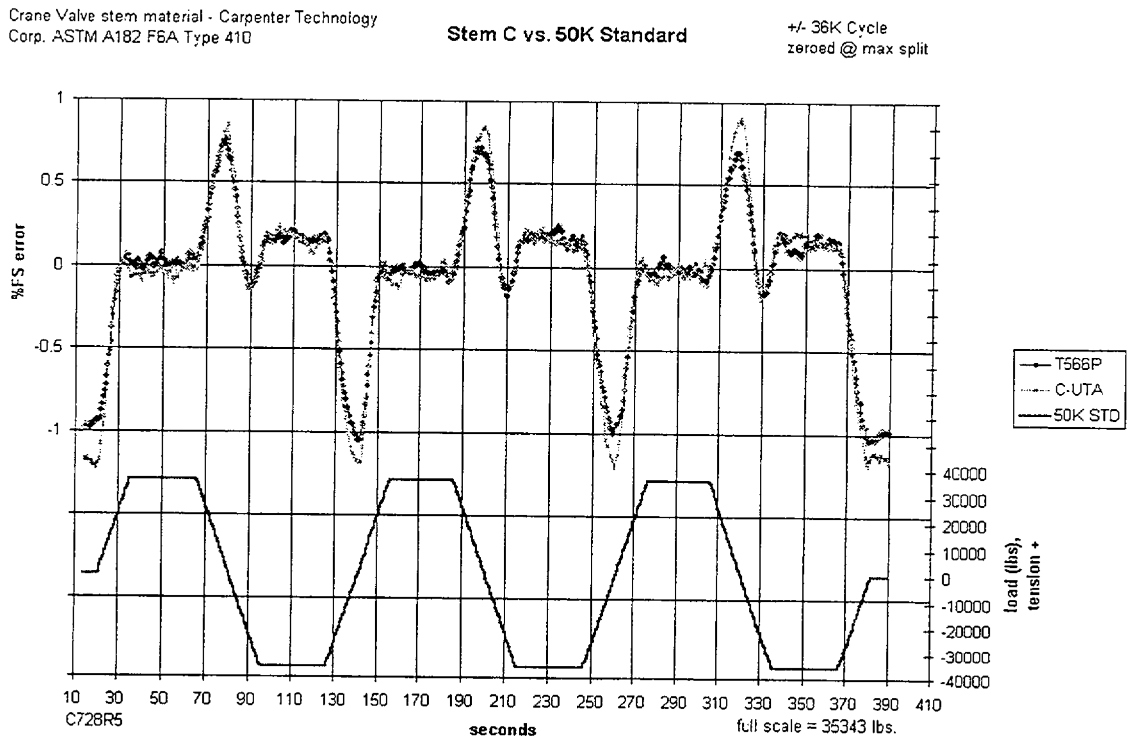

Figure 3: Error and Load vs. Time

Note: the instrument signatures shown in Figures 3, 4, and 6 through 13 are as follows: T566P is the thrust circuit of an ETT instrument, serial number 566. C–UTA is an axial force bridge of bonded resistance constantan strain gages. 50KSTD is an Interface Gold Standard 50,000 lb. calibration load cell, serial number 97167.

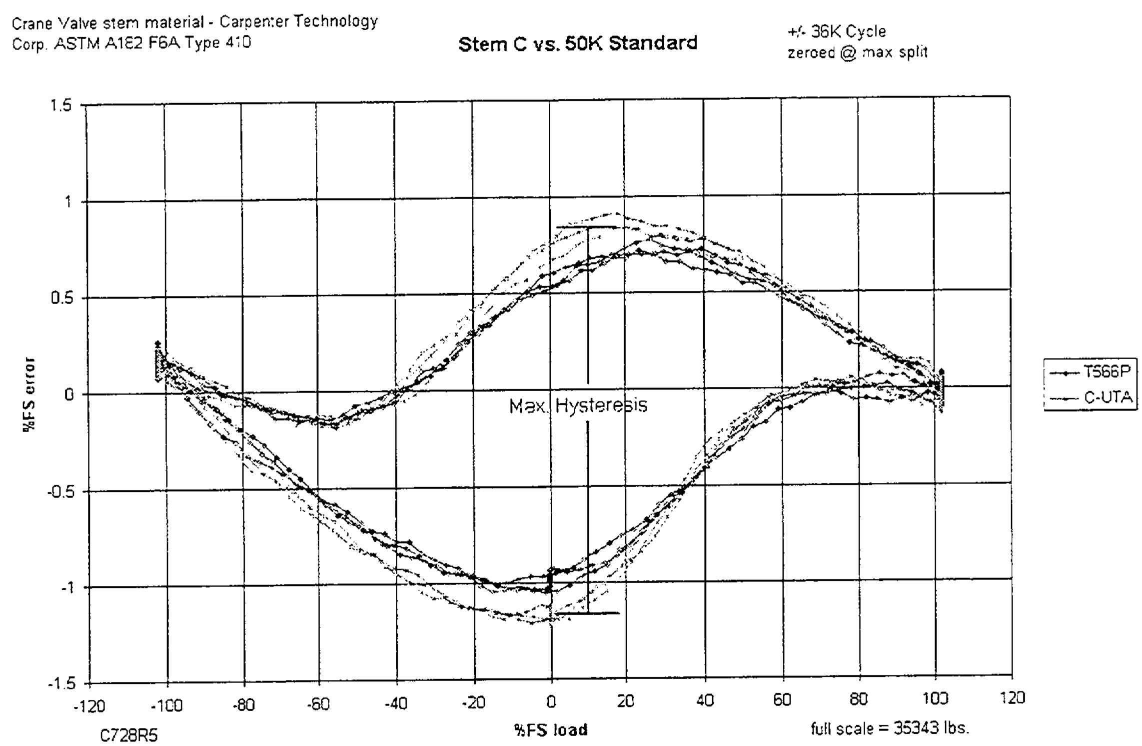

Figure 4: Error vs. Load of Fig. 3 data

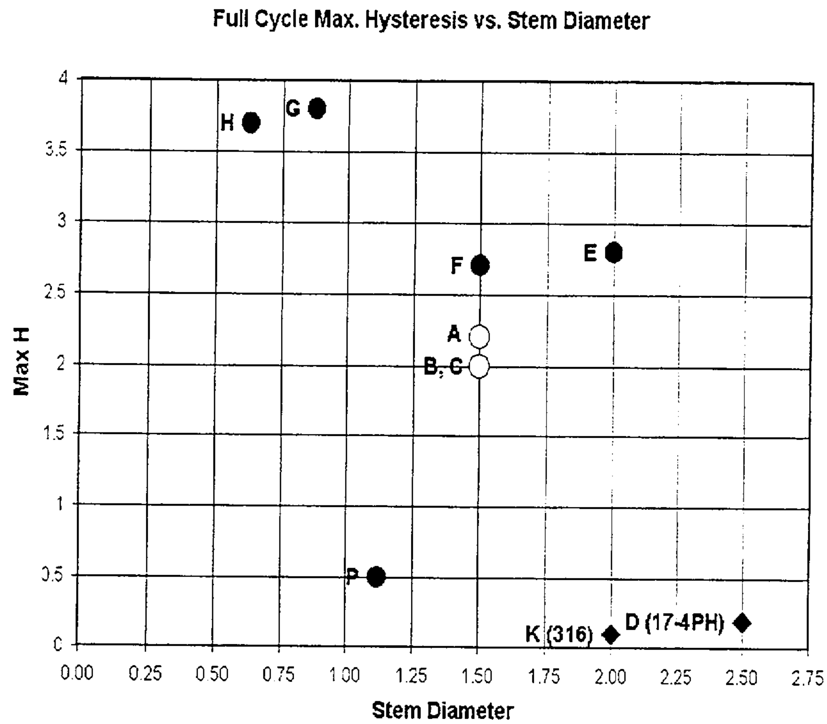

Figure 5: Hysteresis characteristics of test specimens

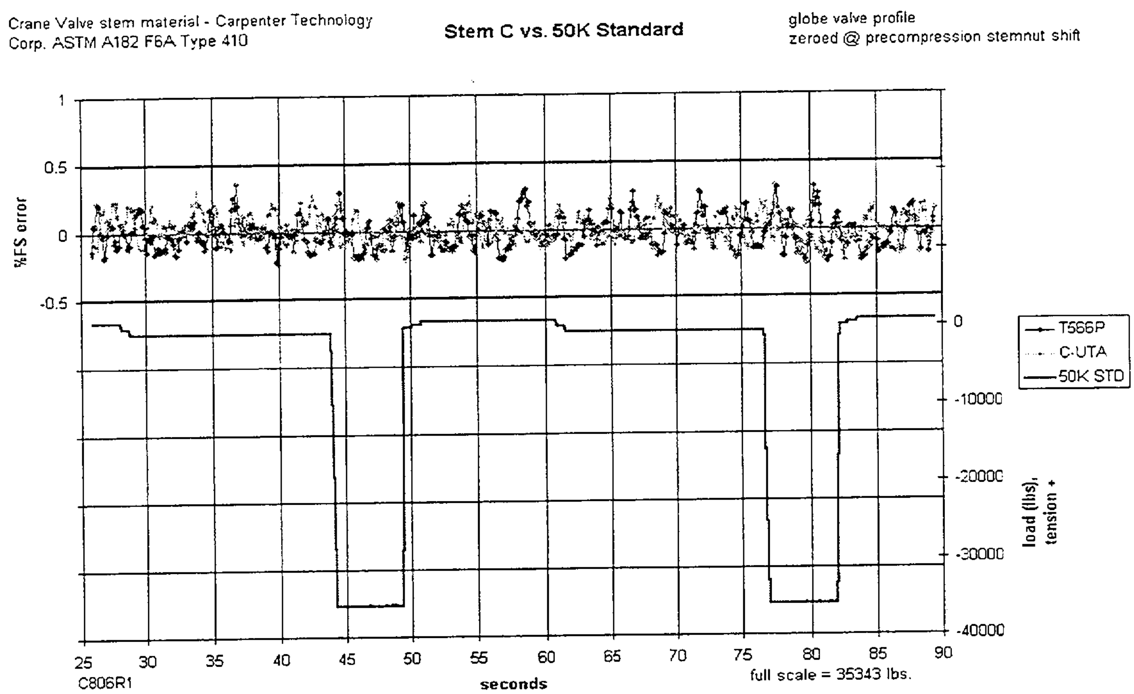

Figure 6: VV1 load profile

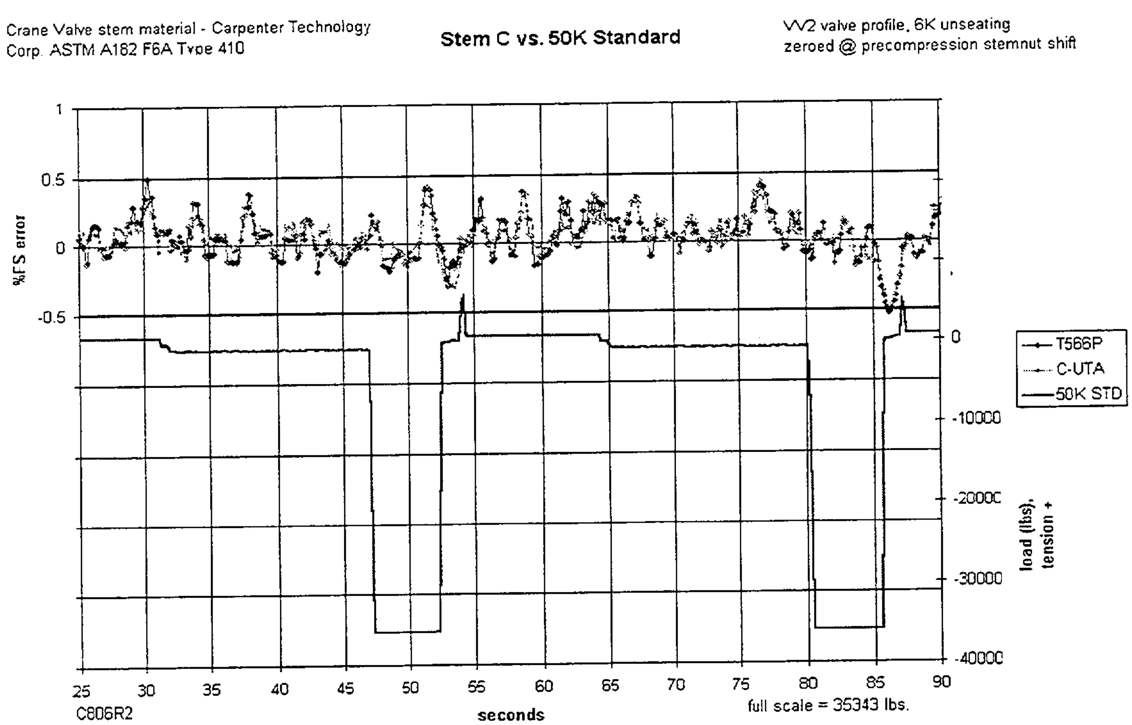

Figure 7: VV2 load profile

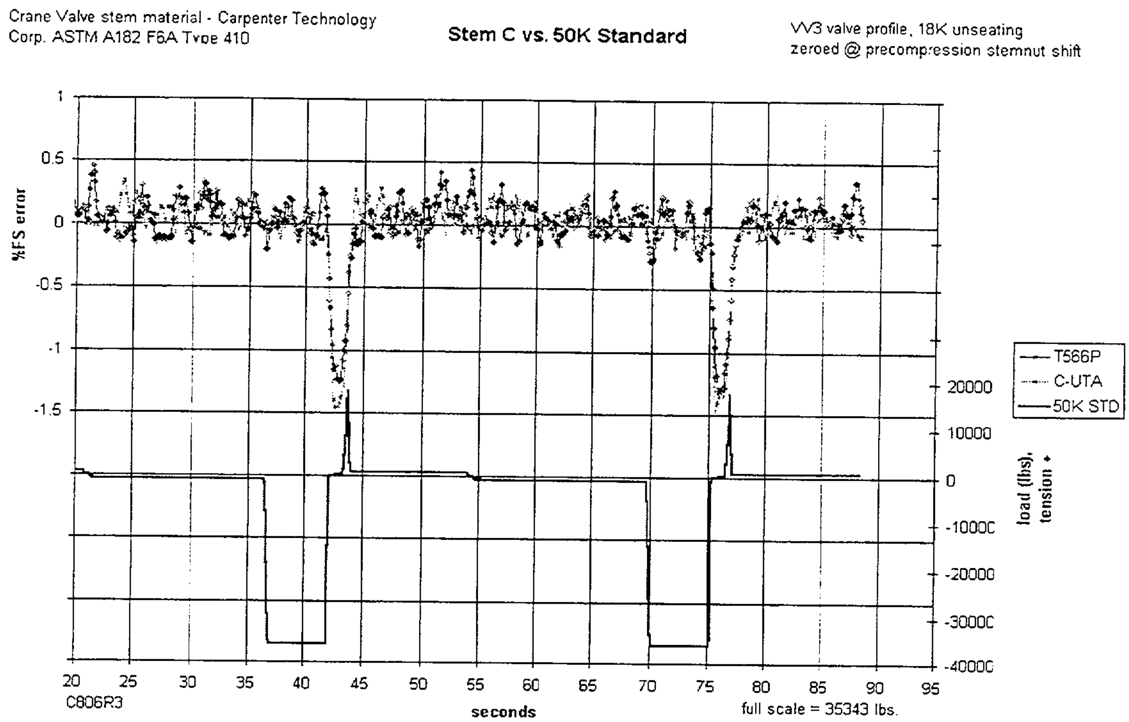

Figure 8: VV3 load profile

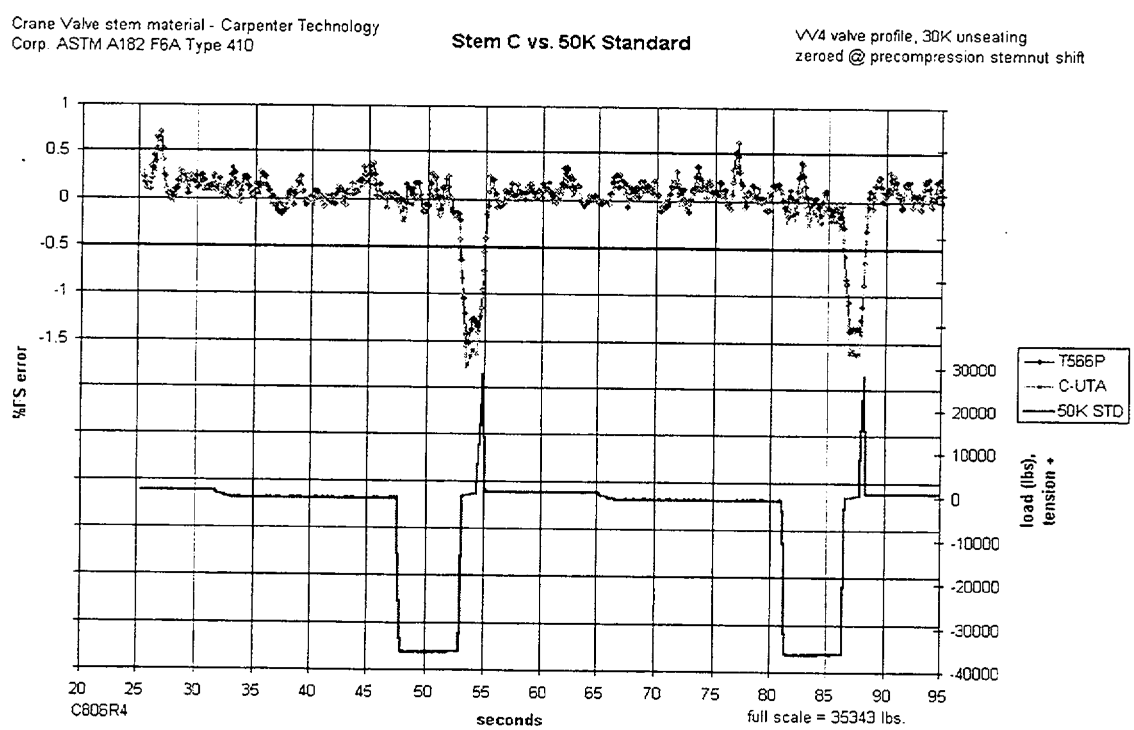

Figure 9: VV4 load profile

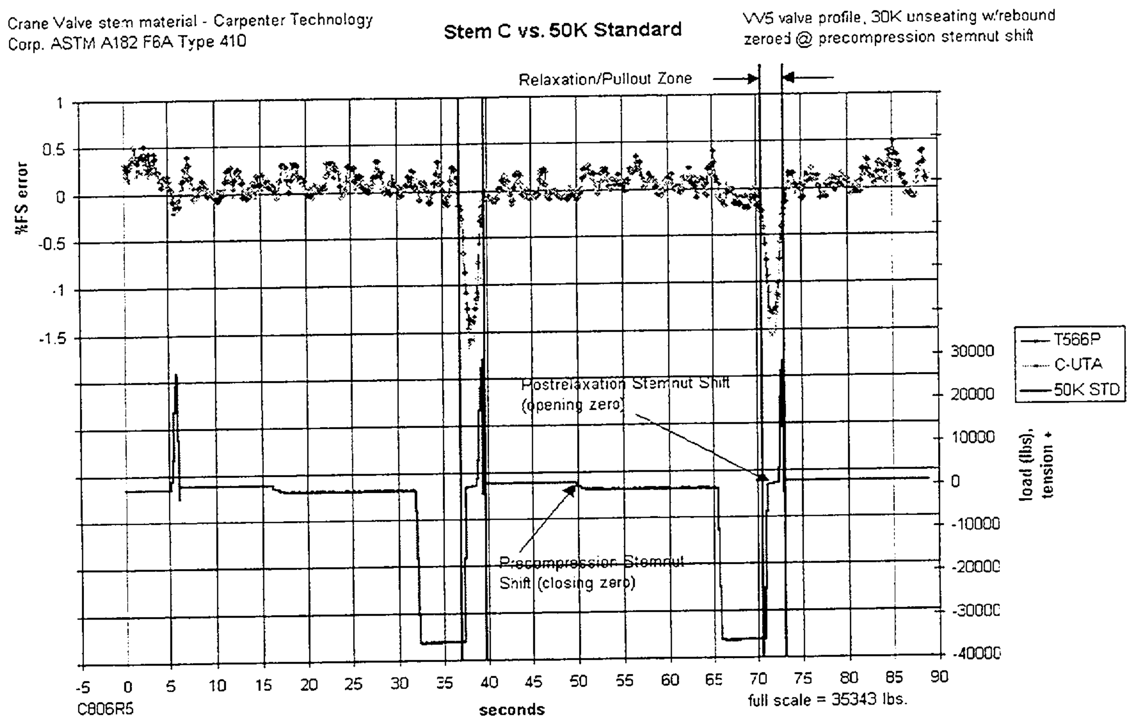

Figure 10: VV5 load profile

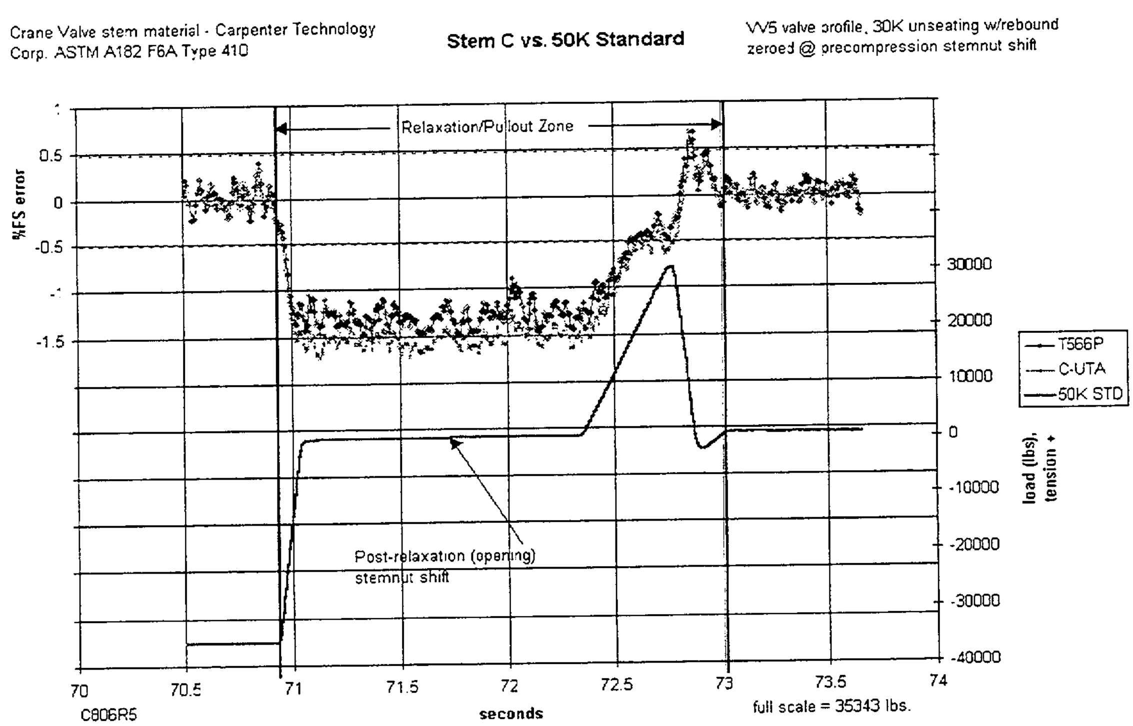

Figure 11: VV5 load profile (detail)

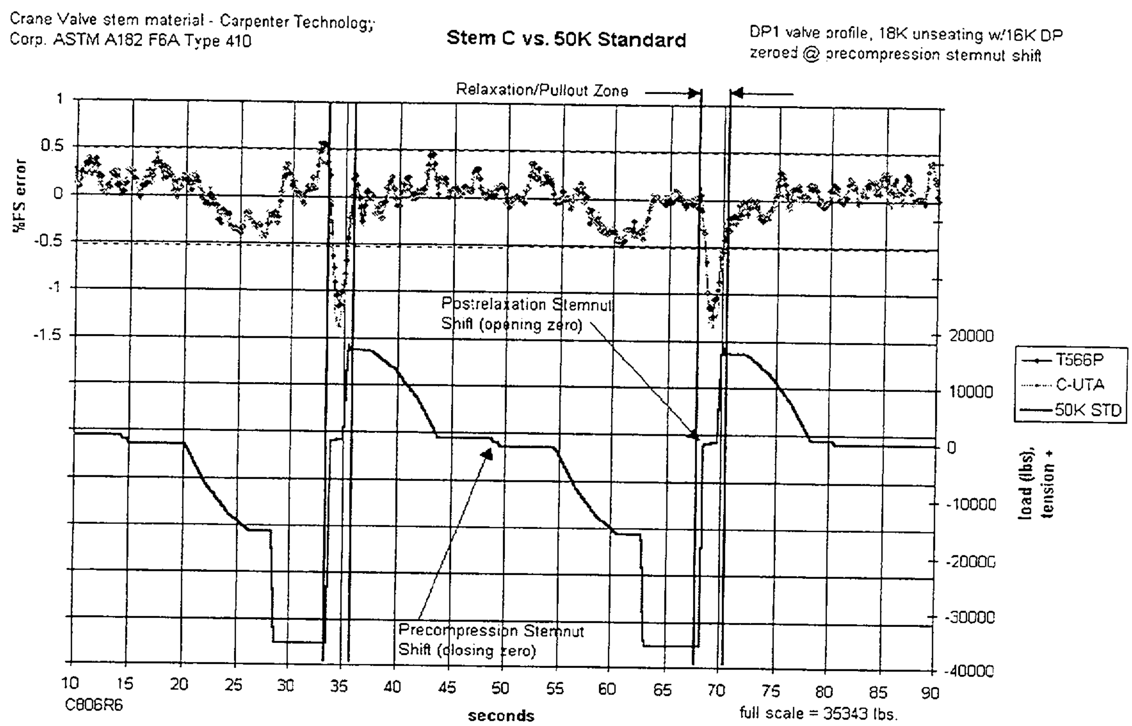

Figure 12: DP1 load profile

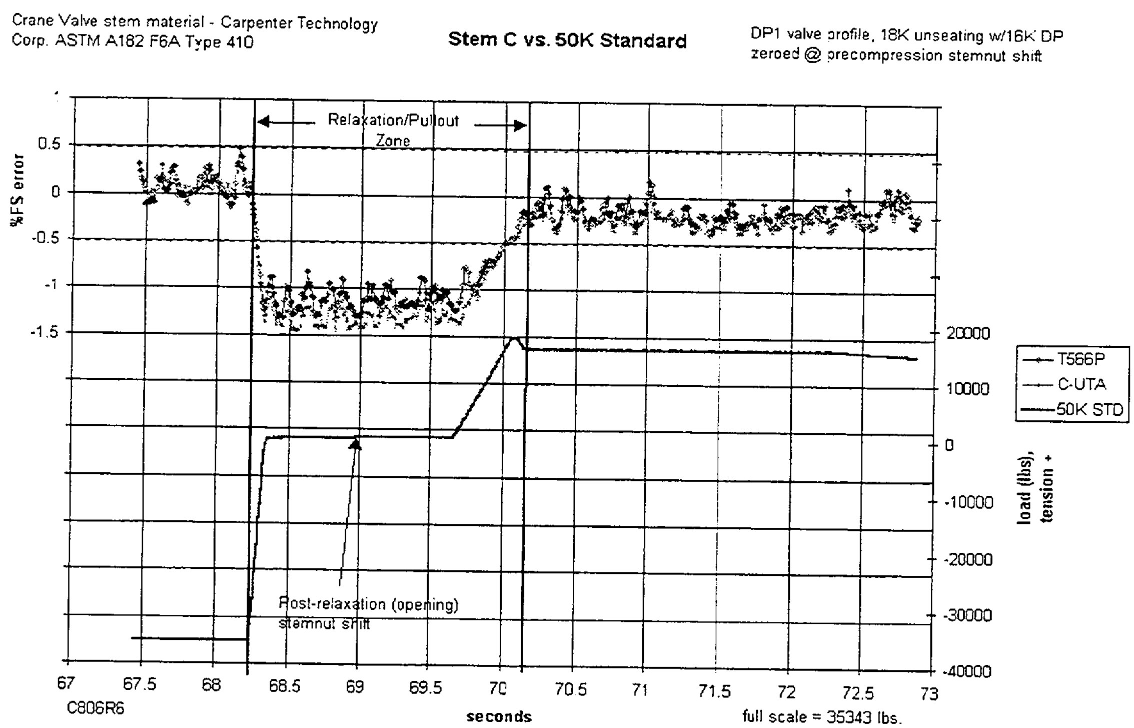

Figure 13: DP1 load profile (detail)

Smooth Pump Vibration Relief—Code OM–6

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Abstract

Palo Verde Nuclear Generating Station performed the 10-year update to its InService Test (IST) program in January 1998. The pump testing requirements were updated from ASME Section XI, Subsection IWP (1980 Edition, Winter 1981 Addenda) to ASME OM–6 (1988). The IWP vibration requirement for pumps with displacement reference values less than 0.5 mil was 1 to 1.5 mils for the Alert Range. However, the OM–6 Code has no fixed minimum Alert since it only allows 2.5 to 6 times the reference value for the Alert range. This created difficulty for pumps that operate smoothly since the normal variance due to system noise and measurement accuracy can send an acceptable pump into Alert condition.

Palo Verde has developed a program that does not penalize pumps for smooth operation. A Relief Request to implement this program was written and approved by the NRC. This paper presents the development and implementation of this program.

I Introduction

The Palo Verde Nuclear Generating Station (PVNGS) updated the IST program on January 15, 1998 by changing from ASME Section XI, Subsection IWP (1980 Edition, Winter 1981 Addenda) to ASME OMa–6 (1988). This is the second 10-year interval for all 3 units.

II Test Requirements

The IST pump test requirements for the first and second 10–year intervals are shown on Table 1 (Section XI, TABLE IWP–3100–2) and Table 2 (OMa–1988, TABLE 3) respectively. Our most common pumps are centrifugal types with speeds greater than 600 rpm. For this type of pump, the prior Section XI requirement for the smoothest pump was an Alert range of 1 to 1.5 mils. This is applicable when the vibration reference value, Vr, is equal or less than 0.5 mil. The newer OM–6 test requirement is for the vibration Alert range to be greater than 2.5 times Vr to 6 times Vr. The vibration collection can be taken as either displacement using mils or velocity in inches per second (ips). Note that there is no fixed lower limit.

III Review

PVNGS reviewed the prior test results to determine if there would be difficulty complying with the lack of a fixed lower limit. A range of the pumps' historical vibration is as follows:

Pump	Typical Vibration Reference Values (ips)
• Auxiliary Feedwater	0.12–0.21
• Condensate Transfer	0.0044–0.0556
• Containment Spray	0.086–0.141
• Essential Chilled Water	0.0075–0.0496
• Essential Cooling Water	0.00295–0.0931

• Essential Spray Pond	0.011–0.213
• High Pressure Safety Injection	0.0667–0.296
• Low Pressure Safety Injection	0.0413–0.319
• Pool Cooling Water	0.0295–0.11

The above shows a large range of vibration reference values. Some of the pumps are very smooth, with reference values significantly less than 0.05 inches per second (ips).

A typical smooth operation pump was reviewed to determine any potential impact of the newer OM–6 Code requirements. Figure 1 graph shows the vibration history for the Condensate Transfer (CT) Pump during the last 12 years of the IST program. The pump is in Unit 3, B train.

Table 3 shows the vibration velocities that were recorded for the outboard axial location, PDA. The axial readings were added in 1997 in anticipation of the newer OM–6 Code requirements. The historical values were plotted from the Master Trend vibration recording system. The range for the last 10 readings was 0.008 ips to 0.0389 ips or about a 1 to 5 range. The OM–6 Code only allows 2.5 times the reference value. Pumps are placed on Alert status if the vibrations are greater than 2.5 times the reference value. Thus, if the reference value had been taken from the 0.008 ips reading of 9/25/97 or the 0.0086 ips reading on 12/16/97, the pump would have been in Alert with the March 12, 1998, reading of 0.0389 ips. This would not have been desirable since the accuracy and repeatability of the low-level vibration measurements are only about 0.050 ips. In addition, flow-induced noise can be a significant portion of a low-level velocity signal of less than 0.050 ips.

The prior Section XI Code requirements were also reviewed. The > 1 mil to 1.5 mil requirement for Alert for all values of displacement vibration of 0.5 mil or less is equivalent to a vibration velocity 0.047 ips if the primary response is at 30 Hz (1800 rpm) or 0.094 ips at 60 Hz (3600 rpm). Thus, PVNGS concluded that the minimum practical level of monitoring the pumps would be a vibration velocity of 0.050 ips. This is roughly equal to the implied Section XI reference values for 1800 rpm pumps and more conservative than the implied reference values for 3600 rpm pumps. A pump relief was written around this value. Thus, pumps with vibration values < 0.050 ips velocity would have an Alert range from 0.125 to 0.300 ips. In addition, PVNGS wanted to take credit for the other monitoring that was normally performed as part of the Preventative Maintenance (PM) program. This program includes the following:

- Spectrum band monitoring
- Bearing acceleration monitoring (on ball and roller bearings only)
- Bearing oil analysis (for oil lubricated bearings)
- Motor current signature analysis (for all but the smallest motors)

If any of these parameters are outside normally expected ranges, an evaluation is performed and appropriate corrective actions taken.

The PVNGS PM Program uses vibration analysis, lubricant analysis, and infrared thermographic analysis, as appropriate, to predict the need for maintenance so that equipment can be reworked prior to failure. The components in this program include those considered important to safe and reliable plant operation, including all the pumps in the IST Program. The intervals for monitoring are based on manufacturers' recommendations, maintenance history, cost effectiveness, and experience. Although the monitoring, analyses, database, and software used in the PM Program do not fall under the PVNGS Quality Program, the PM Program still provides valuable information for assuring the operational readiness of smooth-running pumps.

The pump Relief Request Number 8 (PRR–08) was submitted to the NRC January 13, 1998, for all the IST pumps. The NRC had questions and comments on September 17, 1998, and PRR–08 was revised and resubmitted on December 10, 1998. The NRC requested each applicable pump be listed since a generic relief for all the pumps was not considered desirable. A list of all applicable pumps was added to PRR–08. It contained all IST safety grade pumps except the Charging pump (reciprocating with <600 rpm) and the Diesel Fuel Transfer Pump (submerged and vibration is not monitored). In addition, we clarified how the PM program fits into the PVNGS plan. NRC approved the relief request, PRR–08, in an SER dated July 8, 1999. NRC further clarified that the 2nd paragraph of the "Test Requirement" section is not given any test exception. PRR–08, Attachment A, is enclosed for review. The NRC Evaluation is enclosed as Attachment B. Several of the pumps have utilized the smooth pump relief. The relief request was clarified that it may only be applicable to a single bearing reading location that is low (<0.050 ips), while the remaining readings would not be applicable if they are >0.050 ips. Thus, a pump may take the "smooth pump" relief only for the reading(s) that is (or are) low.

IV Summary and Conclusion

PVNGS noted an oversight in the newer OMa–6 Code and reviewed the historical vibration data to determine if the lack of a minimum value would bring difficulties in future testing. The smoothest pumps would be penalized since Alert values that required increased frequency testing could be smaller than apparent vibration values resulting from system noise and measurement accuracy. A conservative pump relief request was written and submitted to NRC. Several questions were answered and the relief request was revised for clarity. The revised relief request was approved. This has simplified reporting and documentation by eliminating the low-level changes in vibration signals due to system flow noise and measurement accuracy. Therefore, the smoothest pumps are no longer penalized.

V References

1. ASME Boiler and Pressure Vessel Code, Section XI—Division 1, Part IWP–3200, *"Table IWP3100–2, ALLOWABLE RANGES OF TEST QUANTITIES"*

2. ASME Boiler and Pressure Vessel Code, Operations and Maintenance Standards, Part 6, 1988 Addenda, *"Table 3, RANGES FOR TEST PARAMETERS"*

3. NRC Safety Evaluation Report of July 8, 1999, from Mr. Stephen Dembek to Mr. James Levine, APS, with approval of Relief Request No. 8.

Table 1 – Section XI Vibration (Reference 1)					
Test Quantity	**Acceptable Range**	**Alert Range**		**Required Action Range**	
		Low Values	**High Values**	**Low Values**	**High Values**
Vr when $0 \leq$ Vr ≤ 0.5 mil	0 to 1 mil	None	1 to 1.5 mil	None	> 1.5 mil

Table 2 – OMa–1988 Vibration Limits (Reference 2)					
Pump Type	Pump Speed	Test Parameter	Acceptable Range	Alert Range	Required Action Range
Centrifugal and vertical line shaft [Note (2)]	≥ 600 rpm	Vv or Vd	≤ 2.5 Vr	> 2.5 Vr to 6 Vr, or > 0.325 ips	> 6 Vr or > 0.70 in/sec

Table 3. Unit 3 CT Pump Vibration—Axial Direction

List of Trend Points:

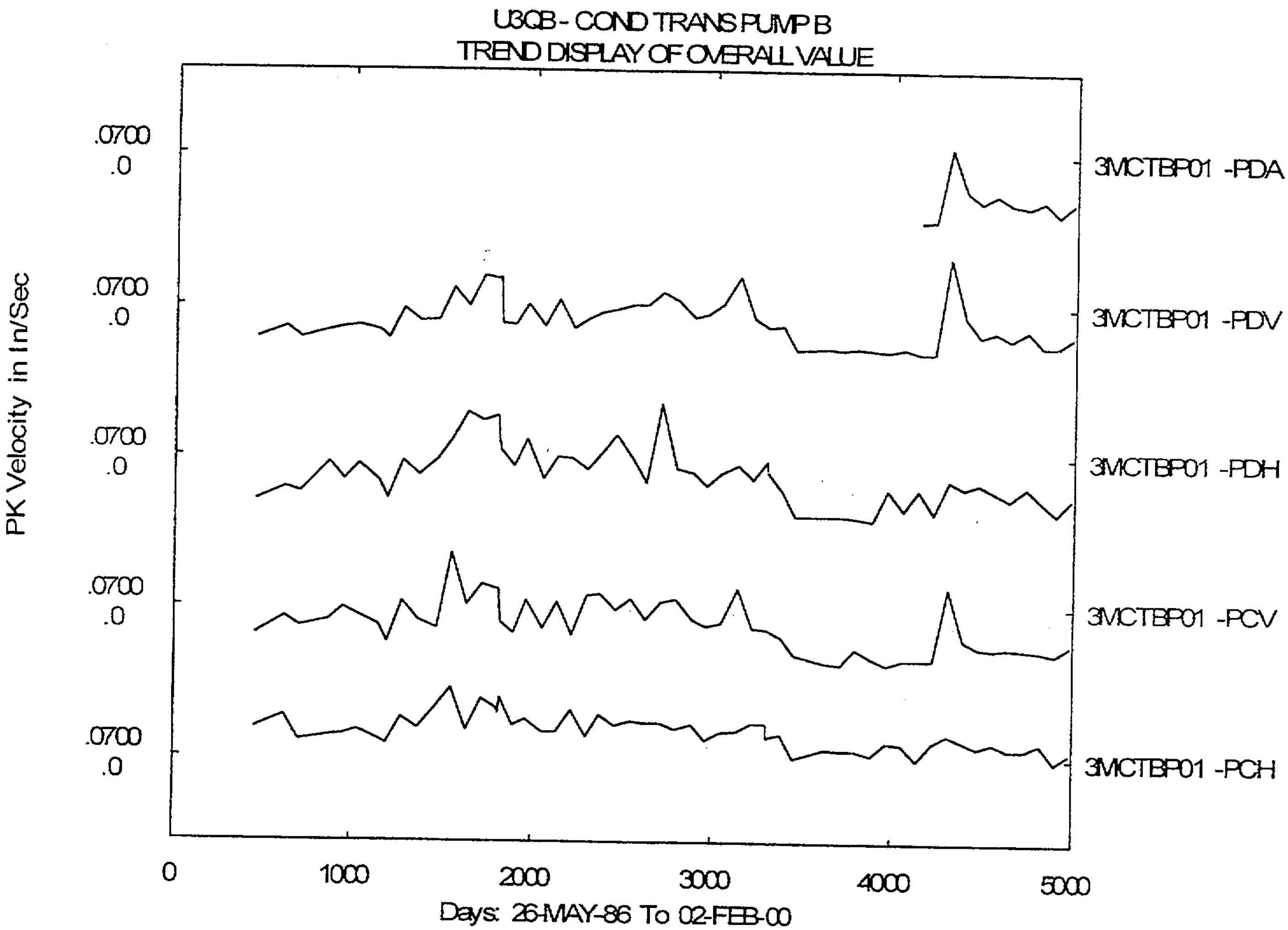
Station: U3QB --> PVNGS UNIT 3 (Q "B")

Machine: 3MCTBP01 --> COND TRANS PUMP B

Meas Point: PDA --> PUMP D AXIAL (ST PT.)

Parameter: OVERALL VALUE (PK In/Sec)

DATE	TIME	VALUE
25-SEP-97	01:27	.0080
16-DEC-97	22:33	.0086
12-MAR-98	03:16	.0389
04-JUN-98	01:48	.0211
28-AUG-98	00:37	.0164
19-NOV-98	02:51	.0200
10-FEB-99	23:17	.0155
06-MAY-99	01:27	.0147
29-JUL-99	03:33	.0170
21-OCT-99	01:17	.0113
13-JAN-00	02:01	.0163



Vibration nomenclature is as follows:

Symbol	Vibration Location	Direction
PDA	Pump Outboard	Axial
PDV	Pump Outboard	Vertical
PDH	Pump Outboard	Horizontal
PCV	Pump Inboard	Vertical
PCH	Pump Inboard	Horizontal

Figure 1. CT Pump Vibration History

Attachment A

I. PUMP RELIEF REQUEST NO. 8 (PRR–08) Smooth-Running Pumps			
Pump ID	**Pump Description**	**Code Class**	**Drawing / Coord.**
AFA–P01	Essential Auxiliary Feedwater Pump (Turbine-Driven)	3	AFP–001 /D06
AFB–P01	Essential Auxiliary Feedwater Pump (Motor-Driven)	3	AFP–001 / B06
AFN–P01	Non–Class Auxiliary Feedwater Pump (Motor-Driven)	N	AFP–001 / H06
CTA–P01	Condensate Transfer Pump	3	CTP–001 / C05
CTB–P01	Condensate Transfer Pump	3	CTP–001 / B05
ECA–P01	Essential Chilled Water Circulation Pump	3	ECP–001 / B08
ECB–P01	Essential Chilled Water Circulation Pump	3	ECP–001 / B04
EWA–P01	Essential Cooling Water Pump	3	EWP–001 / E06
EWB–P01	Essential Cooling Water Pump	3	EWP–001 / E02
PCA–P01	Spent Fuel Pool Cooling Pump	3	PCP–001 / D15
PCB–P01	Spent Fuel Pool Cooling Pump	3	PCP–001 / B15
SIA–P01	Low Pressure Safety Injection (LPSI) Pump	2	SIP–001 / F11
SIB–P01	Low Pressure Safety Injection (LPSI) Pump	2	SIP–001 / B11
SIA–P02	High Pressure Safety Injection (HPSI) Pump	2	SIP–001 / E11
SIB–P02	High Pressure Safety Injection (HPSI) Pump	2	SIP–001 / A11
SIA–P03	Containment Spray Pump	2	SIP–001 / H11
SIB–P03	Containment Spray Pump	2	SIP–001 / C11
SPA–P01	Essential Spray Pond Pump	3	SPP–001 Sh. 1 / C04
SPB–P01	Essential Spray Pond Pump	3	SPP–001 Sh. 1 / C07

Function Various

Test Requirement If deviations fall within the alert range of Table 3, the frequency of testing specified in para. 5.1 shall be doubled until the cause of the deviation is determined and the condition corrected. If deviations fall within the required action range of Table 3, the pump shall be declared inoperable until the cause of the deviation has been determined and the condition corrected. (OM–6 para. 6.1)

Reference values shall only be established when the pump is known to be operating acceptably. If the particular parameter being measured or determined can be significantly influenced by other related conditions, then these conditions shall be analyzed. (OM–6 para. 4.3)

Alternate Testing Vibration parameters that would have reference values ≤0.05 ips may be considered "smooth-running". The Alert and Required Action values for these parameters will be determined as if their reference value is 0.05 ips; that is, the Alert Range will be >0.125 ips to 0.3 ips, and the Required Action Range will be >0.3 ips.

In addition to the Code–mandated monitoring, these pumps are monitored under the PVNGS Predictive Maintenance Program. This program includes the following:

- Spectrum band monitoring
- Bearing acceleration monitoring (on ball and roller bearings only)
- Bearing oil analysis (for oil lubricated bearings)
- Motor Current Signature analysis (for all but the smallest motors)

If any of these parameters are outside normally expected ranges, an evaluation will be performed and appropriate corrective actions will be taken.

Before being treated as "smooth-running" under this relief request, each candidate pump will be evaluated to verify that testing performed under the provisions of this relief request will not prevent the detection of significant pump degradation.

Basis for Relief

The repeatability of pump vibration readings at PVNGS is in the range of 0.05 ips due to hydraulic flow noise in this amplitude range and the repeatability of the vibration instruments. When vibration velocities are less than 0.05 ips, changes have been shown to be non-significant.

At vibration velocities less than 0.05 ips, flow noise and instrument repeatability can significantly affect reference values. Candidates for "smooth-running" status will be analyzed per OM–6 paragraph 4.3 to verify that use of this relief request will not prevent the detection of significant pump degradation.

For displacement reference values less than 0.5 mils, it is noted that the Section XI code in effect for the first interval IST Program (1980 Edition, Winter 1981 Addenda) sets the Alert Range at >1.0 mil and the Required Action Range at >1.5 mil. This implies a minimum reference value of 0.5 mils, which is equivalent to 0.047 ips for 1800 rpm pumps and 0.094 ips for 3600 rpm pumps. The effective reference values proposed for smooth-running pumps are roughly equal to the implied Section XI reference values for 1800 rpm pumps and more conservative than the implied reference values for 3600 rpm pumps. Without this relief request, the Alert Ranges for some smooth running pumps will be reduced by a factor of 10.

The PVNGS Predictive Maintenance (PdM) Program is part of the Preventive Maintenance (PM) Program described in UFSAR section 17.2.3.11.1.6. The PM Program was developed using RCM, NPRDS, EPRI, and INPO guidelines as well as factoring in PVNGS site-specific experience and regulatory requirements. The PM Program and PdM activities are controlled by plant procedures. Each of these pumps has a maintenance plan documented in the PM Program which describes the PM and PdM activities performed on that pump. The performance of the system associated with each of these pumps is monitored and compared to performance criteria under the PVNGS Maintenance Rule Program. This ensures the continued effectiveness of the PM program to minimize component failures and maintain or improve system performance (balance availability and reliability).

The PVNGS Predictive Maintenance Program uses vibration analysis, lubricant analysis, and infrared thermographic analysis as appropriate, to predict the need for maintenance so that equipment can be reworked prior to failure. The components included in this program include those considered important to safe and reliable plant operation, including all the pumps in the IST Program. The intervals for monitoring are based on manufacturer's recommendations, maintenance history, cost effectiveness, and experience. Although the monitoring, analyses, database, and software used in the Predictive Maintenance Program do not fall under the PVNGS Quality Program, the Predictive Maintenance Program still provides valuable information for assuring the operational readiness of smooth-running pumps.

The vibration analysis program monitors the vibration of rotating machinery. In addition to the vibration at pump bearings, the vibration of the driver (turbine or motor) bearings are also collected and trended. Analyzed parameters and methods include vibration velocity, bearing acceleration, bearing high frequency detection, and spectral analysis.

The lubricant analysis program samples lubricants and analyzes them to identify degradation or negative trends. Most testing is performed at the on-site lubrication laboratory, where capabilities include wear debris, chemical composition, and lubrication cleanliness analysis.

In both the vibration monitoring and lubricant analysis programs, recently acquired data is compared with previous data to detect any indicated degradation of equipment condition. If degradation indicates the reliability of operating equipment may be negatively affected, or if acceptance criteria is no longer being met, appropriate corrective action is taken. Corrective action may include: continuing trending of the degraded condition, if the condition is not considered to be immediately threatening to the equipment and can be corrected during a time window convenient to plant operation; additional testing or monitoring to confirm the suspected degraded condition; inspection and repair of the equipment as necessary; changes to preventive maintenance procedures or schedules; or design changes.

PVNGS expends considerable resources on preventive and predictive maintenance. One result of these efforts is pumps that run very smoothly. For example, many pumps in the PVNGS IST Program would currently be candidates for "smooth-running" status under PRR–08, as shown in the table below. To continue to impose Code-mandated Alert and Required Action values on smooth-running pumps unnecessarily penalizes PVNGS for achieving this high level of performance.

PUMP	Typical Vibration Reference Values (ips)
Auxiliary Feedwater	0.12 – 0.21
Condensate Transfer *	0.0044 – 0.0556
Essential Chilled Water *	0.0075 – 0.0496
Essential Cooling Water *	0.00295 – 0.0931
Low Pressure Safety Injection *	0.0343 – 0.319
High Pressure Safety Injection	0.0667 – 0.296
Containment Spray	0.086 – 0.141
Spent Fuel Pool Cooling *	0.0295 – 0.11
Essential Spray Pond *	0.0018 – 0.0316
* Candidates for "smooth-running" status under PRR–08	

Approval Since conformance with the Code requirement cited above has not been determined to be impractical per 10 CFR 50.55a(f)(5) and (f)(6), this relief request is submitted as a proposed alternative per 10 CFR 50.55a(a)(3). It is permissible to implement this relief request immediately on "augmented" pumps, i.e. pumps not within the required scope of IST whose testing is "augmented" by testing them in the IST Program. The provisions of this relief request will not be implemented for the remaining pumps until authorized by the NRC per 10 CFR 50.55a(a)(3).

Attachment B

NRC Evaluation

2.6.3 Evaluation

Overall vibration measurements are required by OM–6 to be taken on all safety-related pumps. Measurements of each pump bearing shall be taken in two orthogonal directions. In addition, vibration in the axial direction shall be taken on each pump thrust bearing. In practice, at least five overall vibration measurement points are performed to comply with the IST on each horizontally mounted safety-related pump. These points are then compared with the Code vibration acceptance criteria to determine if the measured values are acceptable.

Table 3a of OM–6 specifies that if, during an inservice test, the vibration measurement of a particular pump in a particular direction exceeds 2.5 times its reference value established previously, the pump will be considered in the alert range and the frequency of testing will be doubled in accord with paragraph 6.1 until the condition is corrected and the vibration level returns below the alert range. For pumps whose vibration measurement is recorded to be 6 times the reference value, the pump is considered in the required action range and shall be declared inoperable. The vibration reference values are required by the Code to be determined when the pump is in good operating condition.

For pumps where the absolute magnitude of vibration is an order of magnitude below the absolute vibration limits in Table 3a, relatively small increases in vibration magnitude may cause the pump to enter the alert or required action range. These instances may be attributed to variations in flow, instrument accuracy, or other noise sources that would not be associated with degradation of the pump. Pumps that operate in the region are referred to as "smooth-running."

The ASME OM Code Working Group on Pumps has tried numerous times to implement a Code change to establish test requirements for a class of pumps that are defined as smooth-running. These requirements centered on selecting a minimum vibration reference value. All vibration reference values below the minimum vibration specified in the proposed Code change would be assigned the minimum reference value. The Code committees have not reached a consensus on the appropriate minimum reference value and on whether this approach would be sufficient to determine degradation in safety-related pumps. In addition, there has been significant discussion on what other types of pump monitoring activities should be included as compensatory requirements for testing of smooth-running pumps.

Previously, at least one plant has been authorized to use the smooth-running pump methodology similar to the concept described above. The minimum reference value was

0.1 ips. However, this plant experienced significant degradation in a pump bearing that was below the minimum reference value approved in their proposed alternative. Had the current Code requirements been in place, the bearing vibration level would have exceeded the alert range for this pump. The degradation was discovered during vibration monitoring as part of a predictive maintenance program. After this discovery, it was clear to the staff that the simple minimum reference value method alone would not be sufficient to determine pump degradation for smooth-running pumps.

The ASME Code requirements represent the minimum monitoring requirements for safety-related pumps. The staff recognizes that licensees perform a litany of performance monitoring and maintenance activities on rotating machinery at their sites. The staff agrees with the licensee's statement in its basis for requesting relief that plants should not be penalized for maintaining their equipment in excellent mechanical condition. The OM Code committees have attempted to incorporate performance monitoring activities with the minimum reference value concept for smooth-running pumps but could not come to a consensus on the requirements. Some committee members suggested that performance monitoring did not lend itself to be codified.

The licensee's proposal combines both the minimum reference value concept with a commitment to monitor pumps classified as smooth-running in its predictive monitoring program. It states that pumps with vibration reference values below 0.05 ips may be considered smooth-running. The licensee's proposed alert and required action range limits of 0.125 and 0.30 ips, respectively, are consistent with applying the Code vibration multipliers of 2.5 for the alert range and 6.0 for the required action range to the proposed minimum reference value.

The licensee's proposal also describes the predictive monitoring program that is applied to all rotating machinery considered important to safe and reliable plant operation, including all pumps in the IST program. This predictive maintenance program specifies testing activities and test frequencies for each of these pumps. These activities include vibration analysis (i.e., spectral analysis), bearing acceleration monitoring, lubricant analysis, and infrared thermographic analysis. Test activities for many pumps also include the pump driver which, in most circumstances, is not currently required by the Code. Test results are documented and trended in controlled plant procedures implementing the predictive maintenance program. Corrective action will be assessed for each test activity performed and would vary from continued monitoring to repair of the problem depending on the degradation trend and if the parameter being measured exceeds an established acceptance criteria.

The licensee did not directly discuss two significant issues related the classification of smooth-running pumps in its proposed alternative. The first issue concerns the actual pump bearing measured directions and whether all directions measured must be below 0.05 ips for a pump to be considered smooth-running. In a phone conversation with the licensee on February 16, 1999, the licensee clarified that its proposed alternative addresses individual pump vibration parameters that are below 0.05 ips. Parameters below this threshold may be considered smooth-running. The staff questioned the technical basis to allow one or more parameters to be classified smooth-running while other pump vibration parameters may have overall vibration parameters well above 0.05 ips. The licensee stated that the design of the entire structure, including pump, pad and baseplate, as well as the supports and piping, may result in one or more directions being significantly stiffer than the other. In addition, vibration sources, such as misalignment, may be prevalent in only one direction. Therefore, the combination of stiffness and vibration contributors may result in one or more measured vibration directions having a significantly less overall vibration measurement than other measured directions on the pump. The staff has noted in its inspection of IST programs at select plants that overall vibration levels may not necessarily be uniform over all measured vibration parameters. Because the purpose of IST is to determine degradation in safety-related components, the proposed alternative should not prohibit the detection of pump degradation. The licensee stated in its proposed alternative testing that the provisions of this relief request will not prevent the detection of significant pump degradation.

The second issue concerns the overall vibration level at which the pump would not be considered smooth-running. The alert and required action limits specified in the relief request sufficiently address acute problems with the pump that were not previously detected. The staff assumes that the intention of the licensee's predictive maintenance program is that problems involving the mechanical condition of the pump would be detected long before the pump reached its overall vibration alert limit. If the pump overall vibration is allowed to degrade such that the vibration level is maintained above 0.05 ips for a significant period of time, although the pump is clearly still operable, the classification of such pump as smooth-running is clearly called into question. In the February 16, 1999, phone conversation, the licensee stated that each parameter would be assessed on an individual basis. However, for vibration parameters above 0.05 ips, that particular parameter would no longer be considered smooth-running. During the February 9, 1999, phone call with the staff, the licensee stated that it would document its approach to both of the staff's concerns in its IST program. A revision of the proposed alternative to include this information will not require additional review by the staff.

The licensee has established reasonable overall vibration levels to consider individual pump parameters as smooth-running. Each pump that has a parameter that is classified as smooth-running will be required to be included in a predictive maintenance program which is controlled, tracked, and trended by plant procedures. All parameters for a pump that is smooth-running will be evaluated in the licensee's predictive maintenance program, whether they are smooth-running or in excess of the 0.05 ips threshold. Data from the predictive maintenance program will be used to determine degradation and corrective action for smooth-running pumps. The licensee's proposed alternative testing to the Code test frequency requirements of paragraph 5.3 provides a reasonable assurance of operational readiness for the reasons stated above.

The licensee has not provided a basis for relief or proposed alternative testing for establishing reference value requirements of OM−6, paragraph 4.3. Reference values established in accordance with the licensee's proposed alternative would not violate the requirements of paragraph 4.3, as the licensee implies in its basis for relief. When questioned about this aspect of the relief request in the February 16, 1999, phone call, the licensee stated that it had not intended to propose an alternative to the requirements of OM−6, paragraph 4.3. Therefore, the licensee's request for relief from the requirements of paragraph 4.3 is denied.

2.6.4 Conclusion

The proposed alternative from the test frequency and acceptance criteria requirements of OM−6, paragraphs 5.1 and 6.1, respectively, from the pumps listed above is authorized pursuant to 10 CFR 50.55a(a)(3)(i) based on the alternative providing an acceptable level of quality and safety. The licensee's request for relief from the reference value requirements of paragraph 4.3 is denied.

Application of Generalized Pressure Locking Methodology to Improve Thrust Margins in Motor- and Air-Operated Wedge Gate Valves

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Abstract

Two earlier papers by the authors describe the development of a validated first principles-based "generalized" pressure locking methodology that accurately predicts the increase in unwedging thrust due to changes in pressures upstream, downstream, and in the bonnet of wedge gate valves. An increase in unwedging thrust due to pressure changes can be caused by two distinct phenomena:

(1) The "traditional" pressure locking phenomenon in which the bonnet pressure is higher than either the upstream or the downstream pressure, and the disc design is such that both the upstream and the downstream disc faces can move independently in response to the respective DPs across them (e.g., flexible wedge, split wedge, and double disc designs). The traditional pressure locking phenomenon has been well understood by the industry for many years. Solid wedge discs and flexible wedge discs with a pressure equalizing hole are considered to be immune to traditional pressure locking phenomenon.

(2) The "pressure-induced binding" (PIB) phenomenon, in which the elasticity of the structure and the sequence of pressure changes in the system can cause an increase in the seat/face reactions. The magnitude of increase depends upon the magnitude of pressure changes and stiffnesses of the valve body, disc, stem, yoke, and topworks. This phenomenon is applicable to both flexible wedge disc designs (even with a pressure equalizing hole) and solid wedge disc designs. The disc pinching phenomenon was identified relatively recently; the phenomenon itself and its significant contribution to the unwedging thrust are not currently recognized by the industry.

This paper compares the results of a matrix of analyses performed for several different application conditions that were analyzed by (1) the generalized pressure locking methodology that has been validated to accurately predict unwedging thrust increase due to both traditional and PIB phenomenon, and (2) the ComEd methodology that has been widely used by the industry. The matrix of analyses (64 cases) included parametric variations in body-to-disc thickness ratios, three different pressurization/depressurization sequences, and solid and flexible wedge disc designs (both with and without a pressure equalizing hole).

The results show that the ComEd methodology is accurate for cases in which the ratio of the valve body/disc stiffness is high; however, large inaccuracies are encountered for cases in which the ratio is

below a minimum range. To account for this uncertainty in predictions, and to bound the test results, relatively large margins need to be imposed on all predictions based on the ComEd methodology. This paper shows that use of the generalized methodology can reduce the uncertainty in prediction to less than 10 percent, thereby reducing the margin to be imposed between the predicted unwedging thrust and the actuator output. This can be particularly important for AOVs that typically have lower actuator output thrust than the MOVs.

Verification by Testing of Main Feedwater Control Valves for Russian NPPs

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Abstract

A jointly developed NPS 400 mm (16”) feedwater control valve in the scope of modernisation and safety improvements of Russian built nuclear power plants was successfully tested. The first tests were performed on a 1:2.5 model at the test laboratories of Chekhov Power. The test showed that the valve would fulfil the required capacities (flow coefficient CV) as well as the required characteristic. The second tests were performed on a 1:1 valve at the WNJJAES laboratories in Kashira as a requirement of the Russian Authorities. The tests confirmed the model test results and the control valve was certified for application for the VVER 1000 PWRs.

Introduction

Feedwater Control Valves of the Russian built large PWRs known as VVER 1000 are subject to replacement as to improve the plants performance. Fig. 1.1 shows a specification of a TACIS project as an example of modernisation and safety improvements of VVER 1000 NPPs. The valves to be replaced are 16” gate valve type control valves with a down flow sleeve to protect the pipe, see Fig. 1.2. A pressure balanced disk of a cage type control valve was developed as a joint project of Chekhov Power and Siemens/KWU, see Fig. 1.3. In addition to fluid mechanical and stress analysis the design had to be flow tested to assure fulfilment of the requirements. A 1:2,5 model test was performed at the test laboratories of Chekhov Power showed that the valve performed reliable and fulfilled the requirements.

The Russian Authority (GAN) required a 1:1 test at the WNJJAES Kashira test facility as an independent verification. The results showed that the design does not only fulfil the requirements in an excellent way but shows a sufficient margin for possible future system improvements.

Requirements on VVER 1000 Feedwater Control Valve

Table 2.1 is an excerpt of the design and testing requirements on the VVER 1000 feedwater control valves.

The control valve has to fulfil also primary and auxiliary functions. The most important requirements are:

Primary Requirements

- Design of external and internal surfaces of control valves shall allow the total removal of sediments, corrosion products, dust and other impurities, after removal of thermal insulation.

- Design of control valve components, contacting the working medium, shall allow maximum working medium draining.
- Correct design shall avoid vibration of the valves or its components. Clearance between components shall be reduced to a minimum. In order to avoid high fluid velocities in the control valve, the valve body diameter shall under no circumstances be less than half of the inlet pipe diameter.
- Parts exposed to friction shall be made of different steel types or receive a mechanical or thermal surface treatment approved by the Purchaser in order to avoid any jamming. The stem part of the rod shall be perfectly smooth and the permissible diameter tolerances contacting the packing shall be ≤ 0.05 mm. The friction factor to be considered for alloy 6 is 0.4 on liquid and 0.6 in steam.
- For all bolted assemblies it shall be made sure that the bolts break before damaging the thread in the body or the bonnet.
- The valve design shall avoid any cavitation. If service conditions do not exclude the possibility of cavitation, it shall be eliminated by adequate choice of internal parts:
    - an exact sizing
    - hardened plugs, seat, cages
    - special cage or design,
- Valve's piston shall be unstressed in any range or piston motion.
- Design of the valve shall allow turning of electric drive around the vertical piston axis of the valve during assembly. The turning of the drive shall be possible in steps of 45°.
- Connections of the control valve to feedwater pipes shall be welded.
- Branch pipes of valves shall be prepared for welding according to the requirements of paragraph 6.8 of OTT–87.
- Feedwater control valve shall be maintainable without cutting out from feedwater pipe.
- Valve seat shall be removable.

Secondary Requirements

- The unrestricted motion of the control piston;
- ensure a defined time of full stroke;
- ensure a defined relative leak rate through the working unit;
- provide the leak-tightness relative to environment;
- position indication of the valve piston.

Furthermore the Russian Technical Standards for NPP have to be fulfilled, see Fig. 2.1.

Verification Requirements

In addition to the thermohydraulic calculations and stress analysis new designed valves have to be verified by testing. Further requirements are stated in the technical specification and the OTT–87. The following shows an example of the requirement in a technical specification.

1. The control valve head sample of the ordered lot shall undergo the acceptance tests.

   Programme and procedure of acceptance tests shall be developed by the Supplier and approved by the Purchaser. The programme of acceptance tests shall include the tests, confirming the calculated discharge characteristic or discharge coefficient and also the definition of the control valve speed of response.

   The acceptance tests are carried out by the Acceptance Commission on the Supplier's testing stands. By the common decision of the Purchaser and the Supplier the acceptance test (or some part of them) may be performed in a specialised testing organisation (testing centre).

2. Each feedwater control valve from the delivery lot undergoes approval tests according to a programme, which includes the following tests:

   - Strength and tightness tests of the parts and welds operating under fluid pressure;
   - Serviceability and smoothness of the movements;
   - Leak tightness relative to the environment;
   - Speed of response;
   - Other tests, stated in the programme of approval tests of the Supplier.

   The programme of approval shall be consulted with the Purchaser.

   The approval tests shall be carried out in the presence of the Purchaser (res. the End User) and the Russian Authority GAN representatives.

   The test requirements according to OTT−87 chapter 12 can be seen in Fig. 3.1.

Test Facility and Test Results

The official WNJJAES Russian "Experimental Centre for Atomic Equipment" in Kashira with a long history of valve testing was the given test facility for the feedwater control valves, see Fig. 4.1.

At Kashira a water flow loop is available consisting of two parallel pumps with a flow capacity of 10^3 m^3/h (4400 gpm) with a pump head of about 30 m (98 feet) at zero flow. The straight pipe up flow of the test valve is at least 40 x NPS ($\geq$ 16 m). The test loop is designed for cold water $40^\circ C \pm 20$ ($104^\circ F \pm 36$). Available pipe diameter for testing covers the range from NPS 150 mm (6") up to NPS 600 mm (24").

The minimum required valve capacity of $K_{v100} \geq 860$ m^3/h($C_V \geq 994$ gpm) had to be achieved at a design pressure difference of 0.5 MPa (73 PSI).

The design valve of $K_{v100} = 860$ m^3/h (C_V = 994 gpm) was achieved at about 83% of the valve stroke. Therefore a reasonable margin was available to about 950 m^3/h (1098 gpm) in addition to the excellent fulfilment of the flow characteristic.

The superiority of the modern cage design in suppressing cavitation was verified as well as the efficiency of the pressure balanced disk.

Table 2.1: Specific Design Data for VVER 1000 Feedwater Control Valve			
Description		**Dimension**	**Value**
Nominal passage	D_n	mm	400 (16")
Design pressure	P_d	MPa	11.8 (1711 psig)
Design temperature	t	°C	250 (482 °F)
Internal flow characteristic			linear
Maximal flow of medium through the valve at the minimal pressure difference and working parameters	G_{max}	t/h	1760
Flow control range		% of Gmax	3 ÷ 100
Maximal valve pressure difference in closed state (within start up)	dP_{max}	MPa	11.8 (1711 psig)
Maximal valve pressure difference in transients	dP_{tra}	MPa	≤ 4 (580 psid) (100 hours per year)
Minimal valve pressure difference in open state (within start up)	dP_{min}	MPa	0.5 (73 psid)
Nominal valve pressure difference allowed in the control range	dP_{nom}	MPa	0.7 ÷ 1.5 (102 ÷ 218 psid)
Relative leak rate through the piston assembly		% of K_{v100}	0.1
Full stroke time		sec	25 to 30

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Project: TACIS 93/94 On-site Technical Assistance

Replacement of four Steam Generator Control Valves on the Balakovo NPP Unit 1

R 1.02/94 D

TECHNICAL SPECIFICATIONS AND REQUIREMENTS

Approved by:	EUROPEAN COMMISSION	Date	Signature
AGEEV	GOSATOMNADZOR RF	27.03.96	
KUTYURIN	VGNIPKII ATOMENERGOPROEKT	25.03.96	
POGOV	OKB GIDROPRESS	25.03.96	
KRILOV	ROSENERGOATOM	26.03.96	
IGNATOV	BALAKOVSKAYA NPP	22.03.96	

Figure 1.1: Technical Specification for Feedwater Control Valves for VVER 1000

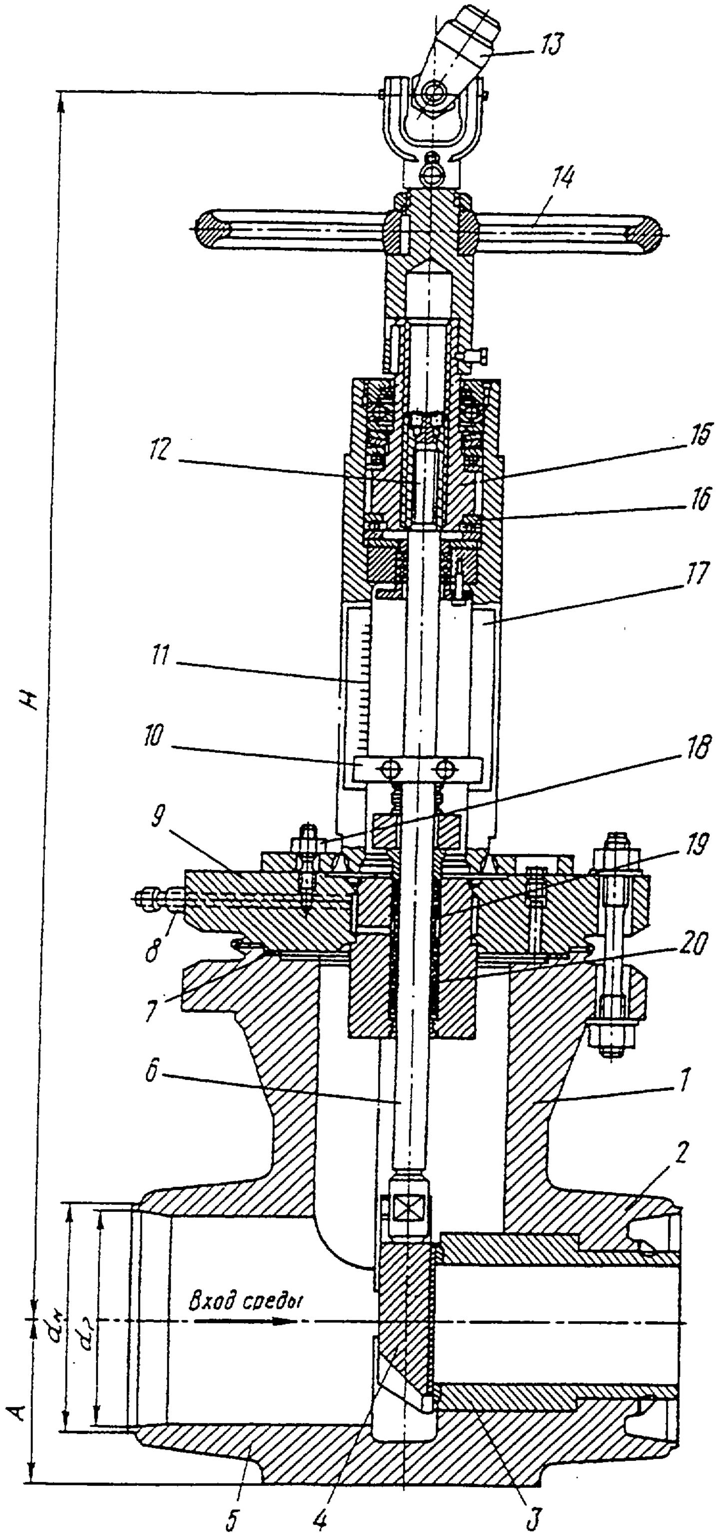

Figure 1.2: Example of an as-built Feedwater Control Valve

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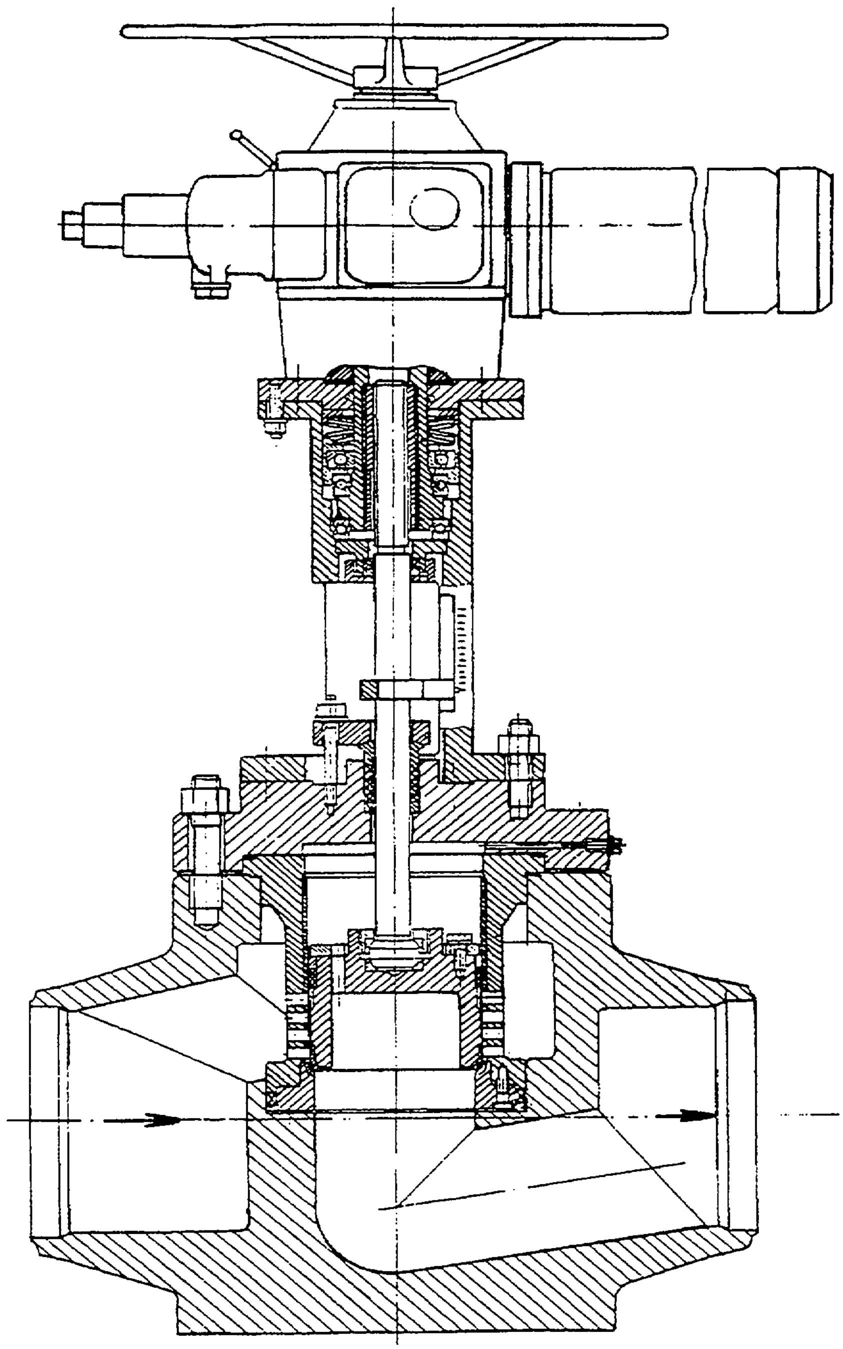

Figure 1.3: Jointly Developed Feedwater Control Valve

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1. Design, manufacture, tests and inspection of the components and the control valve as a whole shall be carried out in accordance with the requirements and criteria stated in the following standards:

(ПН АЭ Г ≡ Regulations and Norms for Atomic Energy of GOSATOMNADZOR of Russian Federation)

ПН АЭ Г-5-006-87. Design regulations for anti-seismic nuclear power plants.

ПН АЭ Г-7-002-86. Regulations and norms applied in nuclear power. Norms for strength analysis of equipment and pipelines of nuclear power units.

ПН АЭ Г-7-008-89. Regulations and norms applied in nuclear power. Regulations for design and safe operation of equipment and pipelines of nuclear power units **(see annex)**.

ПН АЭ Г-7-009-89. Regulations and norms applied in nuclear power. Equipment and pipelines of nuclear power units. Welding and overlaying. Basic propositions.

ПН АЭ Г-7-0 10-89. Regulations and norms applied in nuclear power. Equipment and pipelines of nuclear power units. Weld joints and overlays. Regulations of inspection.

ПН АЭ Г-7-0 11-89. Regulations and norms applied in nuclear power. General propositions for providing the safe operation of nuclear power plants (ОПБ-88).

ПН АЭ Г-7-025-90. Steel castings for nuclear power units. Regulations of inspection.

ОТТ-87. Valves for equipment and pipelines of nuclear power plants. General specifications, dated 9 April 1987, with amendment dated 9 Nov. 1991, asserted by Ministry of Nuclear Power Industry (МАЭП) and State Nuclear Power Supervisor (Г ПАН) of (former) USSR **(see annex)**.

Figure 2.1: Excerpt of Russian NPP Standards

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12.3. Valves testing

12.3.1. I&C used in testings shall be checked for consistency with passports or other technical documentation which list the main parameters of this equipment. The pressure gauges used at testings shall be operable and sealed.
The accuracy instrumentation class shall provide the truth of testing results. Examined values shall lie in limits of the second third of the pressure gauge indication scale. When testing the products, it is prohibited to use instrumentation of the elapsed time of obligatory checks.

12.3.2. Prior to testing the instrumentation rack communications shall be washed to exclude penetration of mechanical impurities into a tested product. Cleaning and washing of the instrumentation racks are performed in compliance with instructions of the manufacturer.

12.3.3. Once manufactured each unit of the valve including the bellows sealed valves shall be subjected to the hydraulic (pneumatic) testing strength and tightness of the component materials and welds pressed by surroundings in compliance with "NPI Rules".
The hydraulic and pneumatic testings shall be performed in compliance with point 5 of the "NPI Rules".
For casted bodies and components the hydraulic (pneumatic) testings are performed at an environment temperature of not less than 5° C. In this case Tbr is not calculated.

12.3.4. It is allowed to manufacture the valves (P<10 Pa) not being in contact with radioactive mediums, without plugs for air removal in case, int the case that when filling them with water of T=20° C, P=0.1 MPa (1 kgs/cm^2), the air volume does not exceed 30% of the valve internal cavity volume.

12.3.5. Testing strength and tightness of materials and welds should be performed prior to painting the valves.

12.3.6. Components and units of the bellow valves should be subjected to test strength and tightness prior to assemble in compliance with drawing indications to avoid damages. The bellows shall be protected against compression and extension.

12.3.7. This point is excluded.

12.3.8. The valves as a unit shall be subjected to the hydraulic testing for leak-proofness of stuffing box glands and packings, head-body contact, upper valve sealing (for valves with outlet of control leakages through the stuffing box) and the valve penstock in compliance with the testing program and method.
The value of the experimental liquid pressure shall be suited to indication of the assemble drawing and TC for a valve, but not less than P.

Figure 3.1: Test Requirements According to OTT−87

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МИНИСТЕРСТВО РОССИЙСКОЙ ФЕДЕРАЦИИ

ПО АТОМНОЙ ЭНЕРГИИ

ВНИИАЭС

Государственный научно-испытательный центр

оборудования атомных станций

STATE EXPERIMENTAL CENTRE FOR ATOMIC EQUIPMENT

КРАТКОЕ ОПИСАНИЕ ПРОИЗВОДСТВЕННОЙ БАЗЫ

г. Кашира
1995 г

Figure 4.1: Official Russian Experimental Centre for Atomic Equipment

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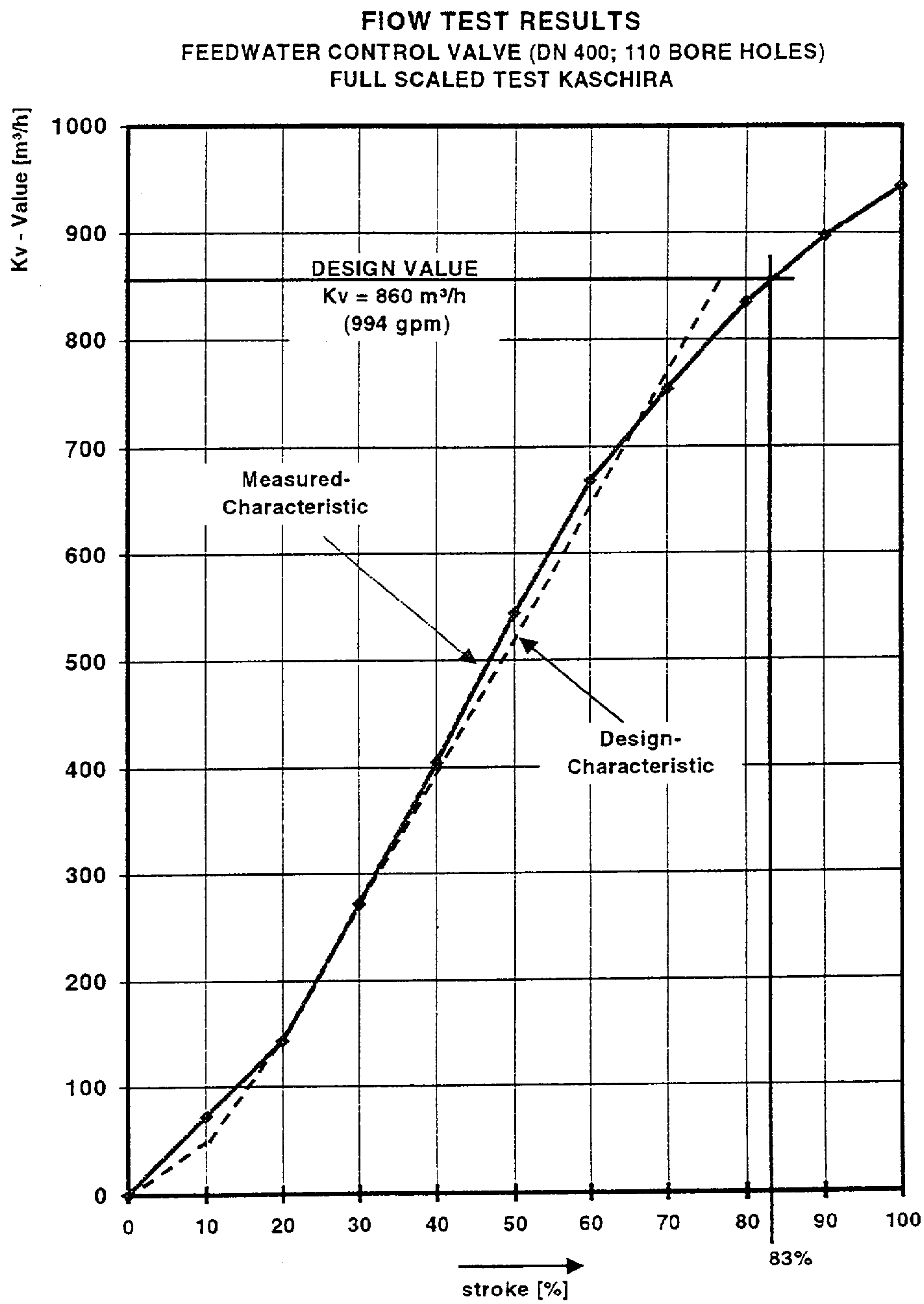

Figure 4.2: Flow Test Results

Session 2(A)

Risk-Informed IST of Valves & Pumps

Session Chair
Gilbert L. Zigler
ITS Corporation

Why Have Some Plants Embraced RI–IST and Others Have Not?

C. W. Rowley
The Wesley Corporation

Abstract

The root issue for building and operating a nuclear power plant has always been the economics of generating electricity. To make an operating nuclear power plant more economical, regulatory mandated "get well" extended outages must never occur, forced outages must be avoided, short refueling outages must be the norm, and plant "programs" must be synergistically integrated. Each operating plant has numerous "programs" to operate and maintain the plant. Thus synergistically coupling pump and valve IST with these additional component programs is clearly desirable. What are the obstacles to the achievement of this synergy? First obstacle is that the requirements for various programs have both safety and economics as the driver. Second obstacle is that frequently different plant organizations are responsible for these various component programs. Third obstacle is that the responsible personnel for the two programs that need to be coordinated quite frequently have substantially different levels of expertise and available time to take on the pro-active effort to capture this synergy.

There has been a somewhat lengthy industry effort to use performance to achieve this desired synergy. More recently there has been a tremendous effort to apply risk to both improve safety and improve operating economics. Ideally plant component (specifically pumps and valves) programs would achieve synergy via both risk and performance. The "proactive" operating nuclear power plant has learned that good safety practices and good economic practices are synergistic with each other. One reason why there are only two plants with NRC approved RI–IST programs is that most plants are still working on avoiding forced outages and reducing refueling outage length. Integrated, risk-informed, and performance-based active component programs is the way to achieve further operating economics while maintaining safety.

1. The Root Issue

The root issue for building and operating a nuclear power plant has always been the economics of generating electricity. That is why there are no nuclear plants in the western states where mine-mouth coal is plentiful. The "expansion" in nuclear power plant operating costs over the years has made some plants uneconomical. Some of that "expansion" was due to layering of added plant "programs" to address regulatory issues. Those plants have a choice: reduce operating costs or shutdown. The recent trend in the U.S. to deregulate electrical generation has forced the economic issue to the forefront of management attention.

What can an operating nuclear power plant do to make itself more economical? The number one initiative is to operate the plant safely so that long term "get well" outages are not necessary. What do you think that the cost of the recent extended outage for D. C. Cook Station has been to date? ....hundreds of millions of dollars and the cost meter is still ticking! The number two initiative is to get rid of forced outages, so that the plant operates continuously from refueling outage to refueling outage. Fortunately there are numerous plants within the nuclear industry that have accomplished this goal. The number three initiative is to shorten those refueling outages from ninety days to twenty or thirty days (or even shorter). South Texas has had several refueling outages in the twenty-day timeframe. There are many plants now getting into the thirty-day timeframe. Most plants have been working on these number two and three initiatives for several years, because the impact on operating economics is huge.

So when the safely operated plant has a rolling availability of 85 to 95%, then it has to look for the number four initiative, which is to make its various "programs" more efficient. When the "programs" become more efficient, the operating staff can be reduced, which is where substantial operating economics can be achieved. However if the operating staff is reduced before the "programs" are made more efficient, then the possibility of operating the plant is an unsafe manner becomes far more likely...a no no (see number one initiative above).

Of course the number five initiative is to find more available megawatts within the plant via design changes, more accurate instrument calibration, different safety analysis, etc. For many plants this has already been done, but for those operating plants with the "alligator fighting" approach, this may be yet another management initiative to achieve better operating economics.

2. Nomenclature

AOV	Air Operated Valve
ASME	American Society for Mechanical Engineers
AUG	AOV Users Group
CIV	Containment Isolation Valve
CV	Check Valve
EPIX	Equipment Performance Information Exchange
EPRI	Electric Power Research Institute
GL	Generic Letter
GQA	Graded Quality Assurance
INPO	Institute for Nuclear Power Operations
ISI	Inservice Inspection
IST	Inservice Testing
MOV	Motor Operated Valves
MUG	MOV Users Group
NIC	Nuclear Industry Checkvalve (Users Group)
NPRDS	Nuclear Plant Reliability Data System
NRC	Nuclear Regulatory Commission
NSSS	Nuclear Steam Systems Supplier
PRA	Probabilistic Risk Assessment
RCM	Reliability Centered Maintenance
RG	Regulatory Guide
RI–IST	Risk-Informed Inservice Testing
SOER	Significant Operating Event Report
SSC	Structures, Systems, and Components
U.S.	United States

3. "Programs" for the Operating Plant

Each operating plant has numerous "programs" to operate and maintain the plant. Some programs are driven by operating economics, while others are driven by nuclear safety. The pump and valve IST program is just one of the nuclear safety programs, but as a key aspect of nuclear plant defense-in-depth and safety margin, it is mandated by the Tech Specs. Over the years other pump and valve component reliability programs have crept into the plant for various reasons. These additional component programs are frequently not coupled to pump and valve IST. Thus synergistically coupling pump and valve IST with these additional component programs is clearly desirable, since this addresses the number four initiative described above in Section One. What are the obstacles to the achievement of this synergy?

First obstacle is that the requirements for various programs have both safety and economics as the driver. Then there are multiple safety driven programs. Some good examples are IST versus NRC GL 89–10 for MOVs and IST versus INPO SOER 86–04 for CVs. Sometimes the economic driven programs are not complementary with the safety driven program. A good example of this is IST versus predictive maintenance for pumps.

Second obstacle is that frequently different plant organizations are responsible for these various component programs. Managerially it is a well-established fact that synergistically coordinating programs across departments is far harder to accomplish than within the same department. This is because departments inherently tend to over-optimize their assigned goals and objectives at the expense of other departments.

Third obstacle is that the responsible personnel for the two programs that need to be coordinated quite frequently have substantially different levels of expertise and available time to take on the proactive effort to capture this synergy. All too often one of the responsible personnel is relatively new to the position and is "fighting alligators", thus having no time to "drain the swamp".

4. Common Overall Themes to Capture Synergy

Over the past ten to fifteen years there has been a somewhat common industry effort to use performance to achieve this desired synergy. Good examples of this are the NPRDS and EPIX equipment data bases. In addition there has been a proliferation of user groups, such as NIC, MUG, and AUG. Finally in the mid-1990s, the NRC issued a new rule (10 CFR 50.65) to increase the incentive for improved maintenance performance. Clearly this effort on performance has created high component reliability, but multiple "program" synergy has been elusive and thus staffing improvements have been minimal.

Over the past five years there has been a tremendous effort to apply risk to both improve safety and improve operating economics. This has been possible because every U.S. operating nuclear power plant has a PRA, due to the response to NRC GL 88–20. EPRI, NSSS owners-group, NRC, and ASME have all worked together to apply risk to various "programs". This is a very promising tool to actually achieve synergy between "programs" because risk can be used for both safety improvement and economic improvement. To date the PRA tool has mostly been used to address safety risk, since assessing plant vulnerability to severe accidents was the goal of GL 88–20. For example the NRC has issued RG 1.174 to address risk-informed regulatory programs in general and a series of four program specific RGs for IST, ISI, GQA, and Tech Specs.

5. Integration of Component Programs

Ideally plant component (specifically pumps and valves) programs would achieve synergy via both risk and performance. Of course the original plant design placed a lot of pumps and valves in various plant locations to achieve certain system functions. Failure of some of these pumps and valves has an immediate impact on safety or on the generation of electricity. Other pumps and valves have a much more delayed effect on the plant. For example a large leaky CIV may only have an impact when the containment needs to be isolated. Since nuclear power plants have lots of standby systems for nuclear safety, hidden failures are of big concern. Thus IST is a major aspect of defense-in-depth to find and fix these hidden failures.

The PRA identifies in an integrated fashion those SSC that are most important to the plant via these functional systems. Please recall that risk is defined as:

Risk = Probability of Failure x Consequences of the Failure

Therefore system risk tends to be driven by the risk of the active components, such as certain pumps and valves, which typically have probability of failure in the order of 1E−03 to 1E−04. Passive components such as tanks and pipe typically have a probability of failure several orders of magnitude lower than the active components. So to improve plant performance ergo system performance, we need to make those active components more reliable...ideally all of those active components.

However, in the real world where resources are not infinite, we can use risk to focus resources on the active components that impact safety (by using the PRA) and generation of electricity (with simplified risk models). A good example is AOVs in most plants. There typically is close to 1000 AOVs in the plant, but perhaps 20 have a high safety risk and 50 have a high economic risk. The ideal goal of a preventive maintenance program might be to make all 1000 AOVs reliable (i.e., replace the elastomers periodically), but practically the goal is to avoid inservice failures for those 20 high safety risk AOVs and those 50 high economic risk AOVs. To avoid that inservice failure, perhaps periodic diagnostics of those 70 AOVs is warranted to monitor degradation in addition to periodically replacing the elastomers.

6. The "Proactive" Plant

The "proactive" operating nuclear power plant has learned that good safety practices and good economic practices are synergistic with each other, especially in the active component arena. Chances are this "proactive" plant has a well-trained operating crew that understands the system impact on plant risk, hence they have train outages and yet allow some on-line maintenance. Chances are this "proactive" plant has some integration of its design basis, licensing basis, and PRA basis (e.g., the risk significant portion). Chances are this "proactive" plant has a well thought out maintenance program, based on risk and performance (RCM is a good start for performance). Note that in reality IST is merely a subset of a good preventive maintenance program, but due to historical reasons it is subsumed into the Tech Specs.

7. Conclusions

One reason why there are only two plants (Comanche Peak 1/2 and San Onofre 2/3) with NRC approved RI–IST programs is that most plants are still working on the number one, two, and three initiatives discussed in Section One above. Clearly a base-loaded plant generating electricity 85% to 95% of the time is the primary goal of plant management. When that goal is achieved, then efficiently integrating plant "programs" is the effective way to achieve staff reduction...the major factor in reducing operating costs.

Integrated, risk-informed, and performance-based active component programs is the way to achieve further operating economics while maintaining safety. Since IST is a Tech Specs mandated program for nuclear safety defense-in-depth and safety margin, converting this program to RI–IST and PB–IST while integrating with the other plant component reliability programs is a great way to achieve further improvement in plant operating costs.

ASCÓ NPP Risk-Informed Inservice Testing Program for Check Valves

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Abstract

Inservice Testing programs for nuclear components at Spanish nuclear power plants are performed in compliance with the requirements of 10CFR 50.55a(f), Section XI of the ASME Code and ASME OM Code.

Current IST programs are essentially based on a set of deterministic considerations (for example, the Code defines the same IST requirements for a Class 1 valve and for a Class 3 valve). Most of the tests have to be performed every three months with no difference among classes.

This presentation describes an alternative approach applying risk insights from PRA to make changes to the ASCO NPP check valves IST program. The basic concept of the Risk-Informed Philosophy is the so-called "integrated approach," which can be defined as combining the insights derived from probabilistic risk assessments (PRA) with deterministic system and engineering analyses (through the Expert Panel) in order to focus attention on issues commensurate with their importance to nuclear safety.

The main result of the ASCO NPP RI–IST program is an optimization of the current IST program by focusing resources on high safety significant valves (new IST methods) and reducing the effort on low safety significant valves (extended test intervals). The RI–IST program for the low-safety significant valves is a performance-based program, which defines the IST strategy based on the performance of the component.

Introduction

In Spain, the Nuclear Regulatory Body (CSN) requires the use of all the rules and regulations endorsed in the country of the reference plant, for the definition of the Inservice Inspection and Testing Programs for the Nuclear Power Plants. For this reason, the ISI/IST programs for the Spanish NPPs are mainly based on U.S. regulations (ASME Code for Operation and Maintenance (ASME–OM), all the NRC provisions such as Generic Letters, Regulatory Guides, etc.).

Operating in the new deregulated market requires that nuclear utilities management pays more and more attention to the cost of operating and maintenance of the plants in order to be able to deliver electricity at competitive prices.

In order to target more efficiently the inservice and testing activities, technologies for risk assessment of systems and components have been developed rapidly over the past two decades.

Check valves are critical components in the operation of nuclear power plants. Recognizing that check valves failures can result in significant operating transients, increased maintenance cost, and/or decreased system reliability, ASCO NPP has defined an inservice testing program for this type of valves taking into account risk information.

The requirements for inservice inspection and testing programs are moving towards risk informed methods, with this new approach the effectiveness of these programs will be improved.

The main objective is to perform better inspections maintaining or increasing safety levels. For doing this, testing methods for high-safety significant components should be improved, for low safety significant components the new programs will be "performance based."

The result of the ASCO NPP RI–IST Program for Check Valves is an optimization of the current IST program, focusing resources on these valves which are high safety significant (with new testing methods) and reducing the effort on those who are low safety significant (mainly with bigger IST intervals). The RI–IST Program for the Low Safety Significant Valves is a performance-based program, IST strategies are defined based on the performance of the valves.

This paper describes the process followed by ASCO NPP for the definition of the Check Valves RI–IST Program.

Risk-Informed IST Program

The philosophy underlying a risk-informed IST policy has been described in detail elsewhere (ASME Documents, ASME Code Cases, US NRC Regulatory guides) and is not described here.

For the definition of the RI–IST program the following rules and regulations were used as references:

ASME Code Case OMN–3: "Requirements for Safety Significance Categorization of Components Using Risk Insights for Inservice Testing of LWR Power Plants" which provides the methodology for ranking components into high and low safety significance categories and evaluating the change in risk from the proposed program.

ASME Code Case OMN–4: "Requirements for Risk Insights for Inservice Testing of Check Valves at LWR Power Plants" which provides testing requirements for the high safety significance and low safety significance check valves. The HSSC valves require implementation of a condition monitoring program while the LSSC valves require a bi-directional exercise test.

US NRC Regulatory Guide 1.174: "An Approach for Using Probabilistic Risk Assessment in Risk-Informed Decisions on Plant Specific Changes to Current Licensing Basis" and Regulatory Guide 1.175: "An Approach for Plant-Specific, Risk-Informed Decisionmaking: Inservice Testing" that define the four basic steps and the five fundamental safety principles for any risk-informed application and specifically for the risk-informed inservice testing programs.

The ASCO risk-informed IST process, consists of the following elements:

- Scope definition.

- Risk ranking, utilization of PSA Techniques to categorize check valves on the basis of importance measures.
- Integrated decisionmaking (Expert Panel categorization)—blending deterministic and probabilistic data to obtain a final categorization of valves as either a high safety significance valve or a low safety significance valve.
- Definition of testing strategies—development/determination of test frequencies and test methods for each valve.
- Evaluation of change in risk—evaluation of the cumulative impact of test strategies on total plant risk.
- Implementation and corrective action program.

Scope Definition

The purpose of inservice testing (IST) at a NPP is to assess the operational readiness of pumps and valves that perform a specific function in shutting down the reactor to a safe shutdown condition, in maintaining the safe shutdown condition, or in mitigating the consequences of an accident.

The Scope of the traditional IST program for ASCO NPP includes all the valves that perform the functions described above.

For the RI–IST program, the scope of check valves to be analyzed consists of all the valves included in the traditional IST program and all the valves included in the PSA. With the junction of these two groups of valves the Scope of the IST program is defined.

This new scope showed in Fig. 1 includes valves not considered in the traditional IST program and that may be important to safety, particularly those modelled in the ASCO PSA but which are not currently within the actual test program.

Check Valves Risk Ranking

This step of the process consists of a preliminary classification of the valves taking into account risk information. For doing this, the Level 1 PSA was used (at power and other modes PSA). The impact of the valves on the Level 2 PSA was estimated qualitatively. All the sensitivity analysis required by the CC OMN–3 were performed and the results were presented to the Expert Panel.

The results of this risk classification are the following:

- 21 check valves were classified as HSSC (RRW > 0.005 and/or RAW > 2). Of this initial HSSC group, 5 check valves are not included in the traditional IST program, the others (16 valves) are included and testing according the ASME Code.
- 24 check valves were classified in an intermediate group and the PSA personnel required the expert panel to define the final classification for these valves. This intermediate group is form by all the check valves with RRW and/or RAW greater than the threshold values for any of the sensitivity analysis. The check valves that are included in the Level 2 PSA are also included in this intermediate group.
- 151 check valves were classified as LSSC.

Integrated Decisionmaking: Expert Panel

For the establishment of the Expert Panel the following activities were performed:

Development of the expert panel administrative procedure in order to define the minimal requirements for the Expert Panel including, quorum requirements, documentation requirements, approval process, and documentation of any disagreement with the final check valves classification.

Expert Panel worksheets development. These worksheets have integrated PSA related information with detailed check valve descriptions, failure modes/cause analysis data and other deterministic information.

Training course for all the members and alternates. This training course has the following main task: PSA Training, decisionmaking process training and risk-informed technology training.

Once the Expert Panel was made up, the final classification of the check valves was established as showed in Figure 2.

Test Strategies

The objectives of the test strategies differ depending on the safety classification of the valve.

For the HSSC group of valves the objective is to identify and to trend the degradation that could lead to the failure mode that resulted in the HSSC.

For the LSSC group of valves the objective is to verify the operational readiness of the valve. For doing this it is necessary to define a "performance based" program.

The process for the definition of the testing strategies, taking into account the above objectives, is the following:

- Failure modes and causes analysis in order to identify the critical failure mode of each valve and the potential causes for this failure.
- Test effectiveness. It is important to analyze the effectiveness of each test testing method in order to determine their capability to detect the postulated degradation for each valve before it could lead to a failure.
- Definition of group of valves, using detailed check valves data sheets, maintenance and test history, and documentation from other plants.
- Definition of the adequate test strategy for each valve or group of valves. This includes the definitions of the test method, test interval, sample requirements, etc.
- Evaluate the risk impact of the strategies defined, in order to check that the new program does not have a negative impact on safety.

RI–IST Program for HSSC Check Valves

Each valve will be full stroke tested with the frequency defined in the current program (3 months, cold shutdown or refuelling outage).

For the check valves that are CIV or PIV, the results of the seat leak test will be used as an indicator of possible valve degradation.

All the valves will be included in a disassembly and visual inspection program. This program will be performed on a sample basis in each group. All valves in a group should be tested in a 6-year or 4-refuelling outage period.

In areas of high radiation and big valves, the use of non-intrusive diagnosis is an appropriate substitute for identifying degradation.

RI–IST Program for LSSC Check Valves

Each valve will be full stroke tested with a 10-year frequency.

For the check valves that are CIV or PIV, the results of the seat leak test will be used to analyze if the full stroke frequency should be modified.

If there is any valve which the safety function is opening and it is not possible to perform a full stroke test, then a partial stroke will be performed and the valve will be included in a disassembly and visual inspection program. A summary of the strategies are showed in Figure 3.

Implementation and Monitoring Program

A sampling strategy is applied to determine if the scope of testing needs to be expanded to other similar check vales when unacceptable degradation is discovered.

When a functional failure or unacceptable degradation is found, a failure mode effects analysis is used to determine if other similar check valves, not included in the original sample, need to also be included in the test sample.

For LSSC check valves, particularly those with extended test intervals, it is possible to take credit for normal plant operation and maintenance tests in the verification of the operability of the valve.

There is a transition plan before extending test interval to maximum defined values in order to allow for experience to be gained before a component test interval may be extended too far.

It is essential in the RI–IST process to use a feedback process to validate the effectiveness of test methods and appropriately modify the test strategy.

For LSSC check valves where a functional failure has been found after the test interval has been extended, it is necessary to return the component to a shorter interval, such as each refuelling outage, and only extending the interval after acceptable performance has been achieved for 2 consecutive tests.

Results and Conclusions

The new inservice testing program for the ASCO NPP check valves requires the performance of more effective testing for the HSSC valves. For the LSSC check valves, there is a reduction in the number of required tests based on the good performance of the valves.

The new program has some valves that are not tested by the current code

The objectives of the new program go beyond the traditional program ones.

Figure 4 shows the results of the groups of valves defined.

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3. ASME Code Case OMN–3, "Requirements For Safety Significance

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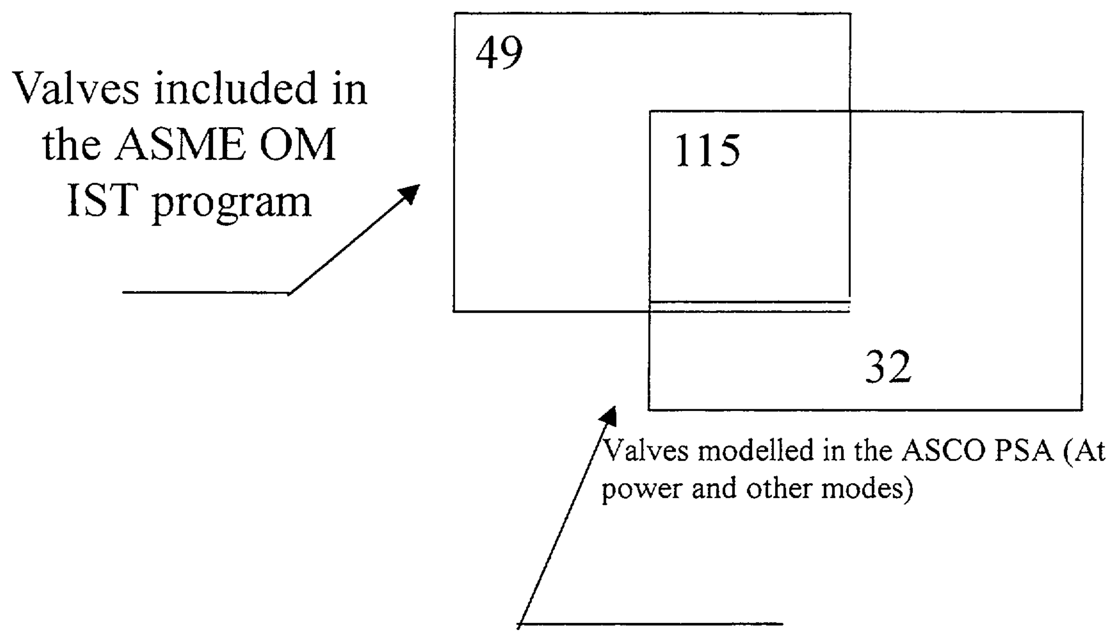

Figure 1 Scope of the New IST Program

HSSC CHECK VALVES	LSSC CHECK VALVES
39 There are 2 valves not included in the ASME OM IST program, the other 37 are included	157 There are 30 valves not included in the ASME OM IST program

Figure 2 Final check valve classification

HSSC PROGRAM	LSSC PROGRAM
Full stroke to each valve with current frequency	Full stroke* to each valve with a 10-year frequency
AND	* If the full stroke test is not possible, a partial stroke will be performed and the valve will be included in a disassembly and visual inspection program(1) on a sampling basis. 100% of the valves in each group will be tested with a 10-year frequency.
Disassembly and visual inspection(1) on a sampling basis 100% of the valves in each group will be tested with a 6 year (or 4 refuelling outage) frequency.	
Additionally	
Results for maintenance activities, seat leak test and other program in place will be analyzed to determine the degradation (HSSC) or the functional readiness (LSSC) of the valve	

Figure 3 Summary table

HSSC	LSSC
1 Group of 1 Valve 3 Groups of 2 Valves 3 Groups of 4 Valves	4 Groups of 1 Valve 5 Groups of 2 Valves 7 Groups of 3 Valves
1 Group of 8 Valves	12 Groups of 4 Valves
1 Group of 12 Valves	2 Groups of 5 Valves
	2 Groups of 6 Valves
	1 Group of 7 Valves
	1 Group of 8 Valves
	1 Group of 9 Valves 2 Groups of 12 Valves

Figure 4 Groups of Valves

Risk-Informed Part 50

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The U.S. Nuclear Regulatory Commission (NRC) staff is conducting an effort to risk-inform the "special treatment" requirements that reside principally in Part 50 of Title 10 of the *Code of Federal Regulations* (10 CFR Part 50). This presentation will provide the status of the NRC staff's efforts as of July 2000. It is anticipated that the discussion will include a description of the background for the effort; the risk-informed categorization of structures, systems, and components (SSCs); the regulatory treatment of SSCs given their risk-informed categorization; schedules for completion; the pilot program; and key issues to be addressed as part of the rulemaking. The NRC staff provided recommendations and plans to the Commission in SECY–99–256 (dated October 29, 1999), "Rulemaking Plan for Risk-Informing Special Treatment Requirements." The NRC issued an advance notice of proposed rulemaking (ANPR) for public comment on March 3, 2000. The comment period for the ANPR ended on May 17, 2000.

Development and Implementation of Code Case OMN–1

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Introduction

Risk informed initiatives first appeared in the ASME Code as part of the OMN–1 Code Case, which was approved for use in 1995. This Code Case is entitled "Alternative Rules for Preservice and Inservice Testing of Certain Electric Motor-Operated Valve Assemblies in Light-Water Reactor Power Plants OM Code–1995, Subsection ISTC." In the OMN–1 Code Case, risk based criteria for motor operated-valve (MOV) testing is addressed in Section 3.7. This Code Case allows risk-informed initiatives to be used in determining the type and frequency of performance testing, acceptance criteria and exercising frequency for applicable MOVs.

As part of the ASME Operations and Maintenance Committee efforts to develop a comprehensive approach to using risk informed initiatives, the OM–8 Working Group (the O&M Working Group on MOVs) submitted another Code Case that has been recently approved and issued as OMN–11. This Code Case addresses different approaches for High Safety Significant Component (HSSC) and Low Safety Significant Component (LSSC) MOVs. The OMN–11 Code Case can only be used by owners that have chosen to implement the OMN–1 Code Case in lieu of the testing requirements of ASME OM Code Subsection ISTC.

This paper traces the development of these code cases and looks possible reasons behind why they have not been aggressively implemented.

Background

There are many approaches to component operation, testing and maintenance. The following are several that are used today:

- Operate to failure and then replace or repair
- Deterministic based testing and maintenance
- Performance based testing and maintenance
- Risk based testing and maintenance
- Risk informed testing and maintenance

The operation, maintenance and testing of MOVs used in nuclear power plants are covered under the ASME OM Code. MOVs specifically covered include those MOVs required to perform a specific function in:

- Shutting down a reactor to the safe shutdown condition
- Maintaining the safe shutdown condition
- Mitigating the consequences of an accident

For safety-related equipment, at nuclear power plants, the option of operating to failure and then taking corrective action is considered inappropriate and is not to be an acceptable option. The OM Code for MOVs was put in place to require owners to regularly assess the operational readiness of certain MOVs in their facilities.

Development of ISTC for MOVs

The ASME Code that is applicable to MOVs was developed in the 1970's and early 1980's. At that time, deterministic based testing and maintenance was considered to be the best available approach and was, therefore, implemented in the ASME OM Code. A deterministic approach is one in which components or systems are classified or coded according to pre-established criteria. Then all the components or systems in a classification or category are required to meet a predetermined program of testing and maintenance. In the case of MOVs, the classifying criteria for ASME Code was whether that MOV had a specific function in shutting down a reactor, maintaining safe shutdown conditions or in mitigating the consequences of an accident. Once an MOV was classified, the predetermined corresponding testing and maintenance requirements were applied to that MOV. This is a deterministic approach because the testing and maintenance requirements for each MOV were specified purely on the basis of the category chosen for the MOV. The testing is considered deterministic because the testing requirements for all the components in a classification are the same without regard or consideration for those components on a case-by-case basis.

The MOV testing requirements of the ASME OM Code Subsection ISTC were chosen based on testing strategies adopted before the development of MOV diagnostic testing equipment. ISTC requires the following for applicable MOVs:

- Position verification
- Quarterly exercising
- Stroke-time acceptance criteria for the quarterly exercise
- Leak rate testing (if required for a particular MOV)

Development of NRC mandated MOV Programs

During the early 1980's, the nuclear industry began to develop an awareness of problems associated with the operation of MOVs. The Nuclear Regulatory Commission (NRC) issued numerous concerns and cautions. The NRC issued a series of industry directives that resulted in utilities developing MOV Programs. Those directives included:

- IEB 85–03 (Use of MOV diagnostic testing)
- GL 89–10 (MOV Program development)
- GL 89–10 Supplements
- GL 96–05 (Periodic verification)

The NRC mandated MOV Program requirements included:

- Design basis analysis
- Use of diagnostic testing technologies
- Design basis testing
- Determination and control of MOV setpoints
- Preventive maintenance program changes

- Post maintenance and post modification testing
- Retest requirements
- Trending analysis
- Periodic retest program

Development of the OMN−1 Code Case

During the early 1990's, the ASME OM−8 Working Group, developed Code Case OMN−1 to bring the OM Code up to date with what was occurring in the industry in the area of MOV testing and the MOV diagnostic technology that had been developed. This was an effort driven by the development of technologies and testing strategies that needed to be addressed in the ASME Code. The OMN−1 Code Case is performance based. Testing requirements and frequencies are determined using MOV classification along with input that considered MOV design and capabilities, operational use and environment, the maintenance program, and the MOV testing program. Use of MOV grouping was provided for and encouraged in the OMN−1 Code Case to take advantage of the similar nature inherent in the MOV population at nuclear power plants. The OMN−1 Code Case also encourages the use of engineering evaluations when determining the testing strategy and frequency for each MOV or for groupings of MOVs. The testing frequency is based on MOV design, MOV capability margin and what the owner knew about the degradation rate for a particular MOV or group of MOVs. The OMN−1 Code Case replaces the ISTC requirements for quarterly stroke-time testing and position verification. OMN−1 exercising requirements are performed in lieu of ISTC exercising requirements. The ISTC leak rate testing requirements are unaffected by Code Case OMN−1.

Other changes to the ASME Code to change from the traditional deterministic testing to more performance based testing have occurred or are in progress.

NRC endorsement of the OMN−1 Code Case

Prior to the approval of the OMN−1 Code Case, utilities had to maintain dual testing programs on their safety-related MOVs. One testing program to meet the requirements of ASME Code and a separate set of testing to meet the requirements of the NRC mandated MOV Program. The OMN−1 Code Case allows the use of the NRC mandated program with some minor additions, such as exercising each MOV once per refueling cycle to meet both the NRC and the ASME requirements for safety-related MOVs.

Although stroke-time testing was the best the industry had at the time it was included in the ASME Code, the industry has known for a long time that it is a flawed testing methodology. It was also developed before MOV diagnostic equipment had been developed. In the September 1999 issue of the *Federal Register*, 10 CFR Part 50, Section 2.3.2.5 Modification, the NRC states:

> "Since 1989, it has been recognized that the quarterly stroke-time testing requirements for MOVs in the Code are not sufficient to provide assurance of MOV operability under design-basis conditions."

A few paragraphs later the NRC refers to "the present weakness in the information provided by quarterly MOV stroke-time testing."

Previously, in GL 96–05 (Periodic Testing), the NRC identified the OMN–1 Code Case as one approach to meeting the requirements of that GL. Based on the Code of Federal Regulations Section 2.3.2.5 and Section 2.5.3.1, the OMN–1 Code Case is now approved for use without a pre-approved valve testing relief request for changing a plant's ISTC testing program.

The MOV Program is the vehicle used by utilities to measure the health of a particular MOV and to ensure that the MOV population at a plant remains healthy over time. The IST testing has become an unneeded burden. The OMN–1 Code Case can result in a reduction in MOV testing, exercising and administrative overhead by eliminating the unnecessary parts of IST testing.

Background on the weakness of ISTC stroke-time testing

The flaw associated with the stroke-time testing strategy lies in the use of control board lights to monitor the time for a valve to stroke open and closed. The lights are controlled by limit switches in the MOV that are regularly adjusted and reset by the MOV maintenance personnel as part of the maintenance evolutions performed on the MOV. Setting or resetting the MOV limit switches is an integral part of many MOV maintenance tasks. The limit switch settings use bands from 5%–25% of valve stroke length depending upon the particular valve and its operating requirements and capabilities. Therefore, changes in stroke-time testing usually don't indicate valve degradation but instead reflect adjustments that the maintenance personnel have made to the MOVs limit switches as part of preventive maintenance, corrective maintenance or modification activities.

There is value to exercising an MOV to redistribute the MOV's lubricant. But quarterly exercising most MOVs is excessive and unnecessary, especially in light of the lubricants available today.

There is also a large administrative overhead associated with maintaining the MOV IST records. The IST Program is tasked with monitoring and trending the changes to stroke-time test results. This becomes an exercise in reviewing the MOV maintenance history to determine if the stroke-time change is a result of degradation or a result of a maintenance activity that changed the limit switch setpoint that operates the control board lights.

In today's world, the MOV Program, not the ISTC testing, is what is insuring the MOV is healthy, remains healthy and is ready to operate when needed.

Development of Code Case OMN–11

The ASME O&M work in the risk initiative world began as risk based, but early in the process changed to risk informed. Risk based testing is a deterministic approach that uses the risk evaluation, usually a probabilistic risk assessment, as another classification tool in assigning predetermined testing strategies. Risk informed testing, uses a component's risk determination as another factor in determining what testing is appropriate for that component or group of components.

As the ASME risk informed initiatives progressed in the 1990's, the OM–8

Working Group, submitted another Code Case to expand the risk initiative section of the OMN–1 Code Case. The second MOV code case has been recently approved by the Board of Nuclear Codes and Standards and has been identified as the OMN–11 Code Case.

This code case was written to apply to Code Case OMN–1, not ISTC. In order to use Code Case OMN–11, an owner must be using the OMN–1 Code Case in lieu of the testing requirements found in ISTC. The OMN–1 Code Case provides an owner with a significant savings over maintaining the dual testing requirements of the NRC mandated MOV Program and the testing required by ISTC.

Code Case OMN–11 allows the owner to relax the OMN–1 grouping criteria found in Section 3.5 of OMN–1 for those MOVs identified as LSSCs. Existing groups of MOVs can have the similar LSSCs associated with them for the purpose of reducing the testing requirements for LSSC MOVs. The reduction in testing requirements for the LSSCs provides an owner additional, but smaller, savings (than implementing just OMN–1).

Lack of OMN–1 Code Case Use

The OMN–1 Code Case has been implemented, at least in part, at two nuclear plants—Wolf Creek and Comanche Peak.

Palo Verde Nuclear Generating Station has an approved valve relief request and has begun implementing OMN–1. SONGS had applied for a valve relief. OMN–1 can now be implemented without an approved valve relief from the NRC due to the NRC's endorsement of the Code Case in 10CFR50.

There are several reasons the OM–8 Working Group has uncovered as to why the industry has been reluctant to implement the OMN–1 Code Case. Those reasons include:

- Regulatory responsibility shift between plant staff / organizations
- MOV population applicable to ISTC vs. MOVs applicable to NRC MOV GLs
- Lack of confidence in the NRC's response to using OMN–1 in lieu of ISTC
- Cost of changes needed in plant procedures and technical specifications
- How to handle the excerising requirement in OMN–1 that is not part of NRC mandated MOV Programs

The O&M Main Committee is currently working to incorporate the OMN–1 Code Case into ISTC as an appendix to replace the ISTC MOV testing requirements. This will hopefully make the OM Code more consistent and improve the Code by removing a flawed testing methodology.

OM–8 Working Group response to the lack of OMN–1 Code Case use

OM–8 response to possible reasons for not implementing the OMN–1 Code Case are as follows:

- Regulatory responsibility shift between plant staff / organizations

  Existing plant organizations are established and have matured in their understanding of their role and responsibilities. Change is always hard. The MOV Group has had responsibility to interact with the NRC and is used to dealing with those regulatory issues and pressures. The IST Group is used to interacting with the ASME Code. These groups are usually not under the same plant management and there is a reluctance to change based solely on the changes that the plant organizations would experience. Cost savings and the elimination of excessive/redundant testing requirements needs to be understood to drive through this perceived barrier.

- MOV population applicable to ISTC vs. MOVs applicable to NRC MOV GLs

  There are plants that have MOVs that with ISTC testing requirements that for various reasons are not part of their MOV Program. The concern here is the work required to produce the design basis review, perform the design basis testing and put in place the MOV Program requirements for MOVs not previously analyzed and controlled in this way. The OMN–1 Code Case provides the option of using engineering evaluations and justifications for adjusting and determining the type and frequency of testing needed for particular valves. All safety-related valves have an existing performance criteria that could be used for consideration under the OMN–1 Code Case. For MOVs that have not been part of the MOV Program, that information could be used in the determination of how to proceed with that particular MOV. The OM–8 Working Group is also available to help owners on particular valves through the ASME Code Inquiry process.

- Lack of confidence in the NRC's response to using OMN–1 in lieu of ISTC

  Any question concerning the NRC's confidence in the OMN–1 Code Case was clearly addressed in the September 1999 *Federal Register* of 10CFR50, as referenced earlier in this paper. This should no longer be an issue.

- Cost of changes needed in plant procedures and technical specifications

  Changes to plant procedures and technical specifications are real and need to be considered. However, it is our contention that the cost of maintaining the current ISTC testing program for MOVs far exceeds the cost of changes to plant program documentation.

- How to handle the exercising requirement in OMN–1 that is not part of NRC mandated MOV Programs

  Code Case OMN–1 requires that each safety-related MOV be exercised at least once during each fuel cycle. In most cases, the MOVs are operated during a plant startup, plant shutdown, safety train swap, or plant evolutions. Those operations would satisfy the OMN–1 Code Case exercising requirement. A single Surveillance Test or Preventive Maintenance task could be generated to be implemented during refueling to exercise the remaining few valves not normally operated as part of plant operations, such as the containment sump isolation valves.

The OM–8 Working Group would be glad to address any questions or inquiries related to how to implement the OMN–1 or OMN–11 Code Cases.

Conclusion

As OM–8 looks at the lack of use of this code case, it is apparent that the nuclear industry has not recognized the savings associated with implementing OMN–1. The utility plant staffs seem to be comfortable in maintaining the separate MOV testing programs under ASME ISTC and NRC GLs. Once this savings becomes more apparent the conversion from ISTC to the OMN–1 Code Case will occur. The use of OMN–11 will coincide with the use of the OMN–1 Code Case.

Regulatory Perspective on Risk-Informed Inservice Testing (RI–IST) of Pumps and Valves

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Abstract

The operational readiness of certain safety-related pumps and valves is vital to the safe operation of nuclear power plants. Inservice testing (IST) is one of the mechanisms used by licensees to ensure this readiness. In the past, the type and frequency of IST have been based on the collective best judgement of the industry and U.S. Nuclear Regulatory Commission (NRC) and requirements have been established through the American Society of Mechanical Engineer (ASME) Code consensus process and NRC rulemaking process. Furthermore, IST requirements have neither explicitly considered unique component and system designs nor contribution to overall plant risk. Because of the broad-based applicability of ASME Code test requirements and non-reliance on risk estimates, current IST requirements may under-emphasize testing those components that are more important to safety and may over-emphasize testing of less safety significant components. The development of risk-informed inservice testing (RI–IST) programs has the potential to improve the use of NRC and industry resources without having an adverse effect on safety.

In this presentation, the author will describe an alternative approach for defining the scope, type, and frequency of IST requirements for pumps and valves that is acceptable to the NRC staff. The presentation will summarize lessons learned from the review and implementation of RI–IST programs in the United States, identify potential revisions to the NRC staff's guidance in this area, and provide a status of ongoing regulatory activities related to RI–IST.

Introduction

The operational readiness of certain safety-related pumps and valves is vital to the safe operation of nuclear power plants. Inservice testing (IST) is one of the mechanisms used by licensees to ensure this readiness. In the past, the type and frequency of IST have been based on the collective best judgement of the industry and U.S. Nuclear Regulatory Commission (NRC) and requirements have been established through the American Society of Mechanical Engineers (ASME) Code consensus process and NRC rulemaking process. Furthermore, IST requirements have neither explicitly considered unique component and system designs nor contribution to overall plant risk. Because of the broad-based applicability of ASME Code test requirements and non-reliance on risk estimates, current IST requirements may under-emphasize testing those

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components that are more important to safety and may over-emphasize testing of less safety significant components. The development of risk-informed inservice testing (RI–IST) programs has the potential to improve the use of NRC and industry resources without having an adverse effect on safety.

Alternative Approach For Defining IST Requirements

NRC Regulatory Guide (RG) 1.175 describes an alternative approach for defining the scope, type, and frequency of IST requirements for pumps and valves that is acceptable to the NRC staff. [A companion Standard Review Plan (SRP) Section 3.9.7 provided review guidance for the NRC staff.] The acceptable alternative approach uses an integrated decision making process which considers probabilistic risk assessment (PRA) results in conjunction with more traditional engineering information. The approach is consistent with the defense-in-depth philosophy and ensures safety margins are maintained. The approach will ensure that any risk increases associated with the RI–IST program are small and consistent with the intent of the Commission's Safety Goal Policy Statement. The approach will also monitor the effects of changes to the licensee's IST program and correct any adverse conditions that might result from these changes.

More specifically, the process will *not* allow licensees to ignore components that are categorized as LSSC. There still must be a basis for concluding that these components will function when called upon.

- For RI–IST, PRA is used for two purposes: 1) to categorize components into high and low safety significant (HSSC and LSSC) categories and 2) to assess the overall change in Core Damage Frequency (CDF) and Large Early Release Frequency (LERF).
- LSSCs are candidates for reduced testing (e.g., less frequent than once every quarter). HSSCs are candidates for enhanced testing (e.g., diagnostic testing of motor-operated valves (MOVs) as opposed to merely stroke-time testing). Diagnostic testing would thus be part of their IST program and not simply a commitment in response to a generic letter.
- The testing and trending of *both* HSSCs and LSSCs must be such that there is reasonable confidence that the components will be operable the next time you test them.
- There is step-wise implementation or grouping/staggering to gain data on component performance as test intervals are extended for LSSCs.
- There is component-level monitoring (not only plant/system/train-level monitoring as required by the Maintenance Rule) to ensure that, if unexpected component performance or degradation (of either HSSCs or LSSCs) occurs, it will be promptly detected and corrected.
- There is feedback of performance information to both the test strategy of components (i.e., test intervals will be shortened if unacceptable component performance is detected at the extended test intervals) and to the component ranking process.
- In addition, non-Code components that are categorized as HSSCs by the

licensee's expert panel will be included in the RI–IST program.

- Components that are categorized as HSSCs by the licensee's expert panel may be tested using more stringent criteria than required by the current Code.

So why would a licensee want to adopt an RI–IST program?

- Most components in the licensee's current IST program would be categorized as LSSCs and would have their test intervals extended considerably (e.g., about 80% of the valves and about 40% of the pumps).
- The RI–IST program provides licensees with flexibility to categorize/re-categorize components consistent with the NRC-approved RI–IST *process* and ΔCDF/ LERF guidance provided in RG 1.174.
- The RI–IST program provides licensees with flexibility to change component test strategies based on performance data, consistent with the NRC-approved RI–IST *process*.

For example, implementation of an RI–IST program at Comanche Peak enhanced safety and cost effectiveness. It reduced the number of pump and valve surveillance tests from 1758 to 498 tests per cycle, per unit. This is a reduction of 1260 tests each cycle, which in turn reduces the number of work orders to be generated, reviewed, worked, evaluated, and vaulted. This corresponds to a cost savings of $1.5 million dollars per 18-month cycle for the Comanche Peak RI–IST program.[1]

[1]As reported in *Nuclear Plant Journal*, July–August 1999, Volume 17 No. 4.

Status of RI–IST Reviews

The NRC staff completed the Comanche Peak RI–IST pilot plant review and issued its safety evaluation report (SER) on August 14, 1998. The staff also completed the review of a limited-scope RI–IST application from the South Texas Project (STP) licensee which was intended to provide flexibility in the schedule for testing 24 containment isolation check valves in the safety injection and component cooling water systems. The STP SER was issued on July 23, 1999. The staff recently completed its review of a second full-scope RI–IST application from the San Onofre Nuclear Generating Station (SONGS) licensee. The SONGS RI–IST SER was issued on March 27, 2000.

While the Comanche Peak review was lengthy and resource intensive for both the licensee and the NRC staff, it was performed without the benefit of the RI–IST RG and SRP. Now that those guidance documents are available, the development of RI–IST submittals by licensees should be much simpler and the staff's review of those submittals should be much shorter. For example, the SONGS RI–IST program review was completed in a little over one year in contrast to the Comanche Peak RI–IST submittal which required almost three years to complete. I believe future RI–IST program reviews can be completed in about half of that time, or about six months, with much fewer resources expended.

A noteworthy aspect of the limited-scope RI–IST application from the STP licensee is that it used a bounding calculation of the change in risk (primarily ΔLERF from the inter-system loss of coolant accident analysis) as part of the justification for extending the test interval for the selected containment isolation check valves. While the NRC staff also relied on the licensee's proposed performance monitoring, feedback, and corrective action activities in authorizing this alternative, the bounding calculation provided the staff with a high degree of confidence that extending the test interval for these valves would not adversely affect safety.

The staff expects to receive a full-scope RI–IST submittal from the STP licensee in the summer of 2000. In addition, limited-scope RI–IST submittals are expected from the Sequoyah and Davis Besse licensees in the near future.

Lessons Learned from the Review and Implementation of RI–IST Programs and Potential Revisions to the NRC Staff's Guidance

Lessons learned from the NRC staff's review of the Comanche Peak, San Onofre, and South Texas submittals, and from the implementation of the RI–IST program at Comanche Peak, will be incorporated into revisions to the NRC's RI–IST guidance. Lessons learned from the withdrawal of Palo Verde as an RI–IST pilot plant will also be considered in making these revisions to the staff's RI–IST guidance.

For example, while the guidance in RG 1.175 indicates that testing and performance monitoring approaches for LSSCs may be less rigorous than for HSSCs, the staff's guidance on performance monitoring of LSSCs exceeds traditional IST program requirements and current Maintenance Rule requirements. The intent of the performance monitoring guidance was (1) to prevent insidious failure mechanisms that are related to the revised test strategies from altering the failure rates assumed in the justification of the RI–IST program changes; and (2) to ensure that adequate component capability (i.e., margin) exists above that required during design-basis conditions. This will ensure that component operating characteristics over time do not reach a point of insufficient margin before the next scheduled test activity. The staff recognizes that occasional random failures of individual LSSCs could be tolerated, but the staff believes that components, including LSSCs, should not routinely be allowed to be found in a failed state when the inservice test is performed.

However, when risk insights and/or other engineering results (supported by appropriate performance data) indicate that a particular component or group of components (e.g., LSSCs) does not contribute to plant risk, even when common cause failures are considered, the RG 1.175 guidance on performance monitoring and corrective actions may be unnecessarily conservative. More specific guidance in this area may be appropriate. Approaches currently being considered include a reduced "level of assurance" of operability over the IST interval for LSSCs and system/train level performance monitoring pursuant to the Maintenance Rule for LSSCs.

Other potential changes to RG 1.175 currently being considered include:

- The addition of a recommendation that each licensee's submittal should include an RI–IST Program Description that addresses each major area in RG 1.175. The RI–IST Program Description should adequately describe the process to be reviewed and approved by the NRC staff as an acceptable alternative pursuant to 10 CFR 50.55a(a)(3).
- The addition of a recommendation that licensees provide a description of the relief requests in their current IST program. The staff's guidance as

contained in RG 1.175 suggests that relief requests for components categorized as HSSCs be resubmitted to the NRC and reevaluated in light of the safety significance of the subject components. The licensees for the two full-scope RI–IST program reviews completed to date did not follow this approach. Rather, they provided a description of the relief requests in their current IST program. This description identified the nature of the relief and specified each component's categorization (e.g., HSSC or LSSC). In this manner, the staff was able to determine that the previously approved relief requests, for components categorized as HSSC, continued to be appropriate in light of the safety significance of the components. The staff's guidance documents may be revised to describe this as an acceptable approach.

- A recommendation may be added that licensees endorse certain risk-informed ASME Code Cases. This might also result in the elimination of some of the prescriptive detail in RG 1.175 related to the categorization and treatment of SSCs. Over the past several years, the ASME has been developing a series of risk-informed Code Cases addressing inservice testing of pumps and different classes of valves (e.g., motor-operated valves, air-operated valves, check valves). These Code Cases incorporate risk considerations to provide acceptable alternatives to the present ASME Code specified provisions for performing inservice testing. As these Code Cases are approved by the NRC staff, they will be referenced in the NRC's regulatory guidance documents or regulations as appropriate (e.g., ASME Code Case on Component Categorization (OMN–3)).
- The addition of a recommendation that licensees include a statement in their submittal that their corrective action program procedures will meet the acceptance criteria. Staff review guidance on evaluation of a licensee's feedback and corrective action program suggests that the reviewer should verify that the licensee's corrective action procedures meet the specified acceptance criteria. The staff has determined that it is unnecessary for the reviewer to review the actual procedures. A commitment from the licensee (e.g., in their RI–IST Program Description) that their corrective action program procedures will meet the acceptance criteria is adequate.

Based on information provided to the Commission, revisions to the IST guidance documents would begin in late calender year 2000 and approximately one year, including time for public comment and review by the Advisory Committee on Reactor Safeguards and the Committee to Review Generic Requirements.

Status of Ongoing Regulatory Activities Related to RI–IST

The NRC staff has developed and sent to the Commission a paper (SECY–99–256) which contains a rulemaking plan to risk-inform 10 CFR Part 50—Domestic Licensing of Production and Utilization Facilities. (Reference: SECY–98–300, dated December 23, 1998, and the corresponding staff requirements memorandum dated June 8, 1999.) This extensive effort is phased, and the initial focus is on the regulatory *scope* of structures, systems and components (SSCs) needing special treatment in such areas as quality assurance, environmental qualification, Technical Specifications, 10 CFR 50.59, and ASME Code. The approach is described in SECY–99–256 and involves the development of a new section to Part 50 (50.69) supported by a new appendix to Part 50. While this approach would incorporate into the regulations an alternative that offers licensees the flexibility to utilize a risk-informed categorization process to assess the need for special treatment, the current regulatory requirement for SSC functional capability would not be removed. Rather, the SSC functional capability requirements (for low risk important SSCs) would remain and the SSCs would be expected to perform their design function, but without additional margin, assurance or documentation associated with high safety significant SSCs.

Several public meetings with stake holders have been held with additional meetings scheduled to be held in the near future to identify the rules and the methods of integrating risk insights with balance between margins and defense-in-depth. Additionally, SECY–99–256 contains an Advanced Notice of Proposed Rulemaking (ANPR) that provides a description of, and requests public comment on, the staff's proposed regulatory approach. The ANPR for risk-informing the scope of 10 CFR Part 50 was published in the *Federal Register* on March 3, 2000, for a 75-day comment period that expired on May 17, 2000 (65 FR 43, 11488–11505).

It should be recognized that various aspects of Part 50 affect the criteria and considerations that involve the ASME Code, such as quality assurance, seismic design, construction, inspection, examination, testing, repair/replacement, etc. While risk-informed IST programs have been approved on a plant-specific basis using provisions incorporated within 10 CFR 50.55a, each required plant-specific approval. Under the new risk-informed rulemaking effort, the objective is to eliminate or minimize the need for plant-specific review and approval.

Related to the risk-informing Part 50 activity is the staff's review of STP's request submitted July 13, 1999, for an exemption from various sections of Part 50. This 14-part exemption request would exclude certain components from the scope of special treatment requirements imposed by various sections of the regulations, such as those which require seismic and environmental qualification and Appendix J containment leakage limitations.

Conclusion

Risk-informed initiatives in the area of inservice testing have matured over the past several years. Licensees and the NRC staff are becoming more efficient at submitting and reviewing RI–IST submittals. These submittals remain a high priority at the NRC, but are becoming more routine. In addition, the NRC staff is continuing to work with the various stakeholders (e.g., ASME Code Committees, Nuclear Energy Institute, and members of the public) to streamline the regulatory guidance related to RI–IST in a manner that maintains safety, reduces unnecessary regulatory burden, and enhances public confidence.

Session 2(B)

General Issues on Motor-Operated Valves

Session Chair
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Implementation Program of and Technical Approach to the Regulatory Recommendation on Safety-Related MOVs in Korea

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Abstract

Korea Electric Power Corporation (KEPCO) has developed a program recently to evaluate the operability of safety-related Motor Operated Valves (MOVs) and Power Operated Gate Valves (POGVs) for the following 5 years for 18 Nuclear Power Plants (NPPs). Operability of all safety-related MOVs and POGVs will be evaluated and the final results will be submitted to the Korean Government by the year 2005 and 2002, respectively. In preparation for an effective MOV program, twelve MOVs in Yonggwang (YGN) Unit 1 were selected and tested. In this paper, the implementation results of YGN Unit 1 pilot program and the details of the formalized Korean MOV program, including the grouping method, are described.

1. Introduction

MOVs have been a focus of attention for operators and regulatory bodies due to reported problems with operability in the past years. The operability assurance of safety-related MOVs and POGVs is one of the very important factors in the plant safety and reliability. U.S. NRC issued GL 89–10 and GL 95–07 to evaluate the MOV operability, to review the susceptibility to pressure locking and thermal binding, and to take corrective actions. The implementation of the MOV program in the U.S. has already been extensively addressed and many NPPs in the world have set up or are in the process of setting up MOV programs including maintenance programs to address MOV operability issues.

KEPCO has not experienced any significant problems with valves jeopardizing successful operation, but a few minor problems were reported in connection with the In-Service Test (IST) or operation. While the IST program basically consists of stroke and leakage tests according to ASME requirements, the program does not include integrity and operability evaluation of MOVs and POGVs.

The Ministry Of Science and Technology (MOST) determined a position to the MOV operability evaluation and issued 'Regulatory Recommendation' to take appropriate actions which are required for the evaluation of the operability of safety-related MOVs and POGVs. After reviewing the evaluation results of the operability of MOVs done by the U.S. utilities, KEPCO has agreed with the regulatory body to evaluate the operability of all safety-related power operated valves, particularly MOVs. However, it is not easy to implement the MOV program in Korea because of a variety of actuator/valve manufacturers: MOVs in KEPCO NPPs are provided by 8 actuator manufacturers and 29 valve manufacturers. Actuator manufacturers include Autotork, EIM, Limitorque, Rotork, Hopkinsons, ITT, Jamesbury, and Joucomatic. Valves are provided by Aloyco, B.I.F., Crane, Fisher, Posi-Seal, Valtec, Velan, Westinghouse, etc., including several domestic manufacturers.

To perform the program with the minimum cost and to avoid trial-and-error, it is necessary to optimize the methodology by developing the evaluation and diagnostic techniques in the pilot project. In June through August 1999, therefore, twelve MOVs in YGN Unit 1 were selected and tested for this purpose.

In this paper, the contents of Korean regulatory recommendations which are similar to the U.S. NRC's Generic Letters are introduced and implementation of the pilot MOV program and the details of the formalized Korean MOV program are described. The grouping method is also presented.

2. Regulatory and Utility Positions

A. Regulatory Recommendations by MOST

The MOST has issued regulatory recommendations to provide the general guideline, which has been basically derived from the U.S. NRC Generic Letters. These recommendations were issued on June 13, 1997. In the administrative measures, the MOST requires the Korean Utility (KEPCO) to take appropriate actions to address the problems which have been identified in NPPs in Korea and abroad.

The regulatory recommendations consist of 6 items as follows:

Item 1: Review the design-basis for all safety-related MOVs and take appropriate actions to address the results from the review. Each Valve's capability should be demonstrated by testing under design-basis conditions. Any alternative methods of demonstrating design-basis capability may be allowed with proper written justification.

Item 2: For all safety-related power-operated gate valves, review the susceptibility to Pressure Locking and Thermal Binding and take one of the following actions to ensure each valve's intended safety function: test, analysis, design change (modification), and operation procedure change.

Item 3: The utility should complete its MOV implementation program by June 12, 2005 for the units that are under construction (including YGN Units 5 & 6) and in operation. An overall implementation plan (for all safety-related MOVs and POGVs) should be submitted by June 12, 1999. For each MOV and POGV, design-basis review results should be submitted to the regulatory body at least two months prior to the scheduled testing date of the valves.

Item 4: The utility should complete its pressure locking and thermal binding implementation program for POGVs by June 12, 2002, for the units which are under construction (including YGN Units 5 & 6) and in operation.

Item 5: For the other units under construction or planned, the design basis capability issue of the MOVs and POGVs should be addressed in the component purchasing process. A written document should be also submitted before applying for an operating license to show that the design basis capabilities of MOVs and POGVs have been ensured.

Item 6: Any non-compliance with the completion date specified above should be notified to the regulatory body in writing and a new schedule should be approved.

Tables 1 and 2 show the schedule and details of the scheme of recommendation.

B. Basic Regulatory and Utility Positions

KINS, Korea Institute of Nuclear Safety, entrusted by MOST, is an organization of technical expertise which performs regulatory functions such as safety reviews, inspections, and development of regulatory technical standards for NPPs and radiation facilities.

The basic regulatory positions of KINS for the MOV program are:

(1) All safety concerns that have been raised in Korea and abroad should be addressed;

(2) Any methods or techniques used and approved by the U.S. NRC are acceptable;

(3) Any alternative method which has not been used before may be accepted if it is technically justified; and

(4) The valve data provided by the vendor are not necessarily required.

The positions of KEPCO are as follows:

(1) Develop a grouping method which consists of plant grouping and valve sub-grouping to reduce the cost and man-power;

(2) Measure the valve design data, if possible; and

(3) Verify the valve operability using EPRI PPM Code when an MOV can not be tested under conditions with flow and differential pressure.

3. Pilot MOV Program at YGN Unit 1

To develop an effective MOV program, YGN Unit 1 is selected for the pilot plant, which is a 14-year-old power plant. Five valve companies and two actuator companies were involved in the manufacture of the selected safety-related MOVs, as shown in Table 3. KEPCO and engineering groups of collaborative organizations cooperated in conducting the pilot program.

A. Objectives

The main purpose of this pilot program was to optimize and finalize the subsequent program through evaluation and diagnosis of the safety-related MOVs at YGN Unit 1. Additionally, it was necessary to cultivate an ability to perform for internal technical staffs of power plants and to develop procedures that are applicable to each work scope to meet the requirement effectively. Twelve representative safety-related MOVs were selected with the manufacture of body parts and driving units, design pressure ratings, operating temperature, valve size, system in which a valve is located, and installation location taken into consideration.

The pilot program was scheduled to include the following eight tasks:

(1) Implementation of the program to evaluate and diagnose 12 safety-related MOVs in the pilot plant (YGN unit 1);

(2) Collection and review of the material and cases which had been performed overseas;

(3) Selection and inspection of the MOV to be evaluated;

(4) Extraction and organization of the required information and development of the valve design and O&M related technique;

(5) Development of procedures for the test, design basis review and technical administration;

(6) Design basis review, diagnostic test and the final evaluation for the MOVs selected in the pilot program;

(7) Documentation of the evaluation results of the selected MOVs and the implementation program to submit to the regulatory body; and

(8) Development of the evaluation procedures, which are applicable to all nuclear power plants in Korea.

B. Test Results

The engineering analyses and diagnostic static tests were completed for 12 MOVs in June of 1999. Then, 7 of the 12 MOVs were successfully tested under dynamic DP (Differential Pressure) conditions. The results showed that these MOVs had a proper degree of margin (valves A ~ G in Table 3). The evaluation results including the engineering analysis results were submitted to KINS for regulatory review.

The other five valves (valves H ~ L in Table 3), which were not dynamically tested, had ample static baseline test margins, except two valves (valves K and L). Here, the static baseline test margin is defined as a difference between the torque switch trip thrust (or torque) and the adjusted required thrust (or torque) for torque switch controlled strokes. For limit switch controlled strokes (typically opening stroke), the margin is based on available actuator capability and the adjusted required thrust (or torque). In the margin evaluation, the uncertainties were also considered. As shown in Table 3, valves K and L showed negative static test margin. To improve the margin, the following are being considered: (1) re-calculation of the design basis DP using the appropriate two-phase model; (2) change of the operating procedure, if possible; or (3) increase of the valve actuator capabilities. However, the operability of these five valves will be finally evaluated under the current MOV program.

C. Contribution to Final MOV Program

To finalize the MOV program, the safety-related MOVs in 16 NPPs were selected as shown in Table 4 and the following 9 procedures were developed:

(1) System design basis analysis;

(2) Required thrust/torque analysis;

(3) Setpoint and margin analysis;

(4) Margin analysis in case PPM is used;

(5) Performance prediction using PPM code;

(6) Performance prediction for Westinghouse Flexible Wedge Gate Valves;

(7) Static diagnostic test;

(8) Dynamic diagnostic test; and

(9) Data procurement.

In addition, the KEPCO has improved the in-house technology related to the valve design, operation and maintenance through the implementation of the pilot MOV program and it was possible to minimize the entire cost by developing the evaluation and diagnostic methodology, which is appropriate for the KEPCO nuclear power plant.

4. Current MOV Program

A. Approach and Schedule

KEPCO has made a program to evaluate the operability of safety-related MOVs and POGVs for the following 5 years for 18 NPPs owned by the company. This program basically consists of engineering and diagnostic analyses of which methodology is being developed by KEPCO's engineering group. According to the current MOV program, all safety-related MOVs and POGVs will be evaluated, the evaluation results of each year will be submitted to the Korean Government by the end of every year, and the program should be completed by the year 2005 for MOVs and 2002 for POGVs.

The general implementation plan for Regulatory Recommendations was already submitted to the Government in June of 1999. In addition, the development of computational analysis and database management programs is scheduled. Appropriate procedures and methods for these activities have been developed or are under development.

B. Organization

To make the program effective and consistent, the head office of KEPCO supervises planning, policy-making, and overall management of the program. A separate organization is maintained at the Nuclear Power Generation Department (NPGD) of the head office and there are also teams that are exclusively in charge of the present program in each power plant. The task force team, which consists of the KEPCO R&D group, is operated to develop the procedures and evaluation methodology and to provide the technical support for the plant such as improving the technical ability of plant MOV engineers. In the meantime, outside organizations participate in the program under contract for the on-site diagnosis and engineering analysis.

C. Engineering and Diagnostic Analyses

KEPCO's current MOV program consists of 5 sub-activities including:

(1) Step 1: Grouping and prioritization for all safety-related MOVs;

(2) Step 2: Design basis review which consists of system design basis review, required torque and thrust calculation, weak link analysis, electric degradation analysis, actuator capability analysis, and set-point and margin analysis;

(3) Step 3: Static test;

(4) Step 4: Dynamic test if practicable; and

(5) Step 5: Valve operability evaluation.

a. Step 1 Grouping and Prioritization

KEPCO has 16 NPPs in operation and 4 Plants under construction. This situation requires a special program in the evaluation of operability of MOVs. That is, it is necessary to establish a proper program utilizing the statistical method in order to exclude unnecessary tests and to reduce expenses since there are many MOVs having identical standards for design in the sister plants. The grouping is, therefore, one of the very significant factors to set up an optimized plan from the view point of KEPCO's several 'Sister Plants' and a bunch of 'Similar Valves' among MOVs. This would reduce the program implementation effort as well as increase the easiness in the maintenance.

Table 4 shows that 1526 safety-related MOVs in 16 NPPs in operation will be evaluated. Eighteen power plants (including YGN 5&6) which are to be tested are classified into seven groups as shown in Table 5. These plant groups are further divided into several valve sub-groups according to whether they can share the results of review and analysis. The classification of the valve sub-groups is based on the following information:

(1) System in which the MOV is located;

(2) Function that the MOV performs;

(3) Valve drawing; and

(4) Model and size of the motor and actuator.

The valve sub-groups are divided into three categories such as the system group, dynamic test group, and weak-link group. The system group consists of several identical valves that are located in the same system and perform identical functions. MOVs in the same system group have, therefore, the same valve drawing and can share the engineering analysis results. MOVs with the same valve drawing are also included in the same dynamic test group and share the dynamic test results. To share the weak-link analysis results, the weak-link group is introduced to include essentially MOVs with the same valve drawing, and identical motor and actuator information. Through the detail design basis review (Step 2), these valve sub-groups may be changed but the change is expected to be minor.

Prioritizing MOVs is based on two criteria: namely, probabilistic and deterministic bases. As a probabilistic basis for prioritizing the schedule for testing MOVs, the relative ranking according to the F–V (Fusel-Vesely) importance value using the result of PSA evaluation is used. Another deterministic criterion is to determine the relative ranking using all of the following factors:

(1) Safety significance of the system in which each MOV is located;

(2) Operating mode;

(3) Safety class;

(4) Valve type;

(5) Valve size;

(6) Design pressure; and

(7) Design temperature.

As a result of the grouping and prioritizing, the total number of 259 MOVs in 13 NPPs were selected for the operability evaluation under design basis conditions in 2000. These MOVs are classified into 152 system groups, and a representative MOV within each system group is selected for engineering analysis of step 2 described below. Dynamic tests will be performed for 85 MOVs, although the number may be changed depending on the plant conditions and detailed results of review of the testability.

b. Step 2 Design Basis Review

Step 2–1 System Design Basis Review

This activity is related to an assessment to determine the design intent of each MOV by a careful review of design basis and postulated accident conditions. The main objectives of this activity are to provide a basic MOV and system operating information in which the MOV is installed and to calculate the differential pressure

(DP) when the MOV is actuated under the design basis conditions.

The DP is calculated for opening and closing strokes separately, if the operating conditions or scenarios are different. The simple Bernoulli equation or EPRI SFM (System Flow Model) is used to calculate the design basis DP for most MOVs, but other commercial flow analysis codes would be used for the optimum calculation to improve the margin.

Step 2–2 Required Thrust and Torque Calculation

In this step, the minimum required stem thrust to operate an MOV against the design basis conditions is calculated under the somewhat conservative assumptions. The assumed packing thrust will be adjusted from the static test result (Step 3).

The thrust due to differential pressure is assumed by the valve factor methods with valve factors of 0.5 and 1.1 for the gate and globe valves, respectively. The actual DP thrust will be obtained through the dynamic testing (Step 4) and the valve factor can be re-evaluated. If the dynamic testing for an MOV (gate or globe valve) is not applicable or an MOV is a butterfly valve, EPRI PPM Code is used to calculate the minimum required thrust or torque.

Step 2–3 Weak Link Analysis

The weak link analysis is performed to determine the structural loading limits. The method is based on the classical static force balancing equations using the worst expected load combinations. However, simplified plastic methods are employed for some MOV parts.

Step 2–4 Electric Degradation Analysis

It is an objective of this step to calculate the degraded voltage for each MOV. For the degraded voltage analysis, ELMS–AC code is used. Initially, an electrical system model for each NPP is developed and then, the model is modified for the design basis conditions for each MOV. The obtained result is used to obtain the maximum actuator output capability.

Step 2–5 Actuator Capability Analysis

Actuator torque capability by the electric motor is determined based on the torque capability, actuator rated torque, and maximum spring-pack setting. This step determines the maximum assured actuator output torque at its design basis voltage and temperature conditions. The maximum allowable actuator torque is based on the maximum assured actuator output torque and weak link analysis.

Step 2–6 Set-Point and Margin Analysis

Prior to the MOV testing, it is necessary to find the target window for the switch setting and appropriate switch set-point target. A target window is the thrust or torque range that is used to set up the MOV. Based on the values determined in steps 2–1 through 2–5, the limiting values are calculated with all uncertainties such as:

(1) Rate Of Loading (ROL);

(2) Spring Pack Relaxation (SPR);

(3) Stem Lubrication Degradation (SLD);

(4) Repeatability of torque switch trips; and

(5) Diagnostic equipment and sensor accuracy.

Structural integrity is already considered as a limiting factor in Step 2–5.

In this step, the operability margin for an MOV is expected based on the design basis review results prior to the diagnostic tests. Actual operability margin, however, will be evaluated using the test values after both static and dynamic testings are successfully completed.

c. Step 3 Static Test

The main objectives of the static test are to correct the valve degradation, to set up the control switch properly, and to measure the packing load, stem factor and actuator inertia. The static test and maintenance procedures are designed to achieve these objectives.

The static test is basically performed under no flow and no differential pressure conditions. Full drain condition is preferable but not required. Diagnostic equipment utilized in the static test (and in the dynamic test) is MOVATS–3500 or VOTES–100 depending on the NPPs.

d. Step 4 Dynamic Test

The basic approach is to perform the dynamic test for all safety-related MOVs if practicable, because it is possible to directly verify the operational capability of an MOV. KEPCO's position is, however, that the diagnostic dynamic testing should be based on the risk significance, operating conditions, and baseline static test margin though the actual margin estimation is possible through the dynamic testing.

The dynamic test will be performed under or near the design basis conditions for verification of the MOV operability. If such condition is not available, the tests will be made under 3 different conditions among 50% ~ 85% of the design basis DP so that the design basis DP thrusts and valve factors are estimated using the extrapolated test data.

As stated in the U.S. NRC Generic Letter 89–10 (Supplement 6), a minimum of 30% MOVs, with no less than 2 valves, should be tested within the same dynamic test group. In 2000, therefore, the dynamic test will be scheduled for 85 MOVs among 259 MOVs, although the number may be changed depending on the plant conditions and detailed review results of the testability.

The dynamic test procedures are also designed to verify that MOV performance will meet the criteria defined by the design basis review and regulatory recommendations.

e. Step 5 Valve Operability Evaluation

Based on all the results obtained from step 2 through step 4, the MOV operability and actual switch setting margin are evaluated. The minimum required thrusts are evaluated from the measured actual packing and DP thrusts. The highest valve factor for tested MOVs is applied to all other MOVs within the same dynamic test group.

All engineering and diagnostic test results, including structural integrity, statistics of valve factor and ROL, and correctness of valve degradation are evaluated and documented to verify design assumptions and design set-points. The data obtained from the tests will be also used to establish an operational baseline so that the condition of the MOV can be properly assessed and trended.

D. Perspective and Future Work

Since most of the manufacturers are foreign or even some manufacturers are merged into other companies, it is difficult to obtain accurate information on the valves/actuators. Sometimes, great expenses to obtain information on the valve/actuator are expected. Therefore, PPM inputs and the data necessary for weak-link analysis will be measured during the plant outage.

KEPCO will place emphasis on the following matters:

(1) Optimization of the procedures and methodology;

(2) Training of the staff to improve their ability;

(3) Extending the applicability of the EPRI PPM code or development of other advanced methods;

(4) Development of the computational analysis and database management programs;

(5) Development of the optimized periodic verification program; and

(6) Active cooperation with overseas organizations such as JOG, EMPUG and other utilities.

5. Conclusion

To optimize and finalize the MOV program in Korea, the Pilot MOV program at YGN Unit 1 was performed. The experiences and results obtained in the pilot program are summarized below:

(1) Control switch settings of some MOVs were incorrect and even actuator capability was insufficient for a few cases. But most MOVs evaluated had a proper degree of margin.

(2) EPRI PPM code predicted the required thrust conservatively and its applicability was limited.

(3) The procedures for design basis review and diagnostic testing for the entire NPPs in Korea were developed.

Based on the implementation of the pilot MOV program, the finalized MOV program was made and is being implemented. The main features of this program are as follows:

(1) To optimize the program, 18 NPPs are classified into seven plant groups that are further divided into three valve sub-groups.

(2) In the design basis review, appropriate conservatism is considered so that both the regulatory body and KEPCO may accept the results.

(3) The optimized periodic verification program and database management plan are under development.

References

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[4] EPRI, "EPRI MOV Performance Prediction Program: Motor-Operated Valve Margin Improvement Guide," EPRI TR–100449, Feb. 1992.

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[8] KEPRI, "The Development of Evaluation Program of Safety-Related Power Operated Valve in the Nuclear Power Plant," Feb. 1999 (In Korean).

Table 1 Regulatory Recommendations for MOV			
Plants	**Work Item**	**Due Date**	**Comments**
Under Operation or Construction	Design Basis Review and Evaluation Plan	06/12/99	- Submission of annual status - Submission of design basis review 2 months before the test(progressed to submit the report within 5 years)
	Closure Document	06/12/05	
Planning	Full Analysis Report	Prior to Operation License Issue	- Applicable to all plants after YGN 5 & 6

Table 2 Regulatory Recommendations for POGV			
Plants	**Work Item**	**Due Date**	**Comments**
Under Operation or Construction	Design Basis Review Plan for Pressure Locking and Thermal Binding	06/12/99	- Submission of annual status - Submission of design basis review and correction plan 2 months before the test
	Submission of Correction Report	06/12/02	
Planning	Full Analysis Report	Prior to Operation License Issue	- Applicable to all plants after YGN 5 & 6

Table 3 Test Results of Pilot MOV Program at YGN Unit 1							
Valve	**Valve Vendor**	**Actuator**	**Type**	**Min. Required Thrust or Torque**		**Test Margin (%)**	
				Open	**Close**	**Open**	**Close**
A	W.H.	SMB-1	Gate	5964.2 lbf	10245 lbf	151.6	16.5
B	W.H.	SMB-000	Gate	2156.9 lbf	N/A	181	N/A
C	W.H.	SMB-000	Gate	3186.4 lbf	3135.3 lbf	3.05	54.6
D	W.H.	SMB-00	Gate	1978.1 lbf	1364.6 lbf	311	284
E	W.H.	SMB-00	Gate	1047.6 lbf	N/A	183	N/A
F	Valtek	7NA1 (Rotork)	Globe	365.2 lbf	N/A	405	N/A
G	B.I.F	SMB-00	B.F	N/A	681.3 ft-lbf	N/A	187.2
H	W.H.	SB-00	Gate	5202.8 lbf	9785.4 lbf	134	24.4
I	Anch. Darling	SMB-00	Gate	5538.5 lbf	N/A	11.1	N/A
J	B.I.F	SMB-000	B.F	197.0 ft-lbf	N/A	67.6	N/A
K	B.I.F	SMB-00	B.F	9923.1 ft-lbf	N/A	-47.1	N/A
L	Velan	SMB-0	Gate	N/A	10085.5 lbf	N/A	-8.8

Table 4 Number of Safety-Related MOVs to be Evaluated

	Plant	Rx. Type	Capacity (MWe)	Rx. Design	No. of MOVs	No. of POGVs	Comm. Operation
1	Kori 1	PWR	587	W.H	47	46	1978
2	2	PWR	650	W.H	88	63	1983
3	3	PWR	950	W.H	102	109	1985
4	4	PWR	950	W.H	102	109	1986
5	YGN 1	PWR	950	W.H	112	109	1986
6	2	PWR	950	W.H	112	109	1987
7	3	PWR	1,000	C.E.	138	91	1995
8	4	PWR	1,000	C.E.	138	91	1996
9	UJN 1	PWR	950	Framatom	88	72	1988
10	2	PWR	950	Framatom	88	72	1989
11	3	PWR	1,000	Hanjoong	121	90	1998
12	4	PWR	1,000	Hanjoong	121	90	1999
13	WSN 1	PHWR	679	AECL	62	36	1983
14	2	PHWR	700	AECL	69	40	1997
15	3	PHWR	700	AECL	69	40	1998
16	4	PHWR	700	AECL	69	40	1999
Total		2	13,716	8	1,526	1,389	-

Table 5 Plant Grouping of NPPs for MOV Program Implementation

Plant Group	Rx. Design	Rx. Type and Capacity (MWe)	Plant Name	No. of MOV Subjected
A	Westinghouse	PWR(587)	Kori 1	47
B	Westinghouse	PWR(650)	Kori 2	88
C	Westinghouse	PWR(950)	Kori 3 & 4 YGN 1 & 2	436
D	AECL	CANDU(679)	Wolsung 1	62
E	AECL	CANDU(700)	Wolsung 2, 3 & 4	207
F	Framatome	PWR(950)	Uljin 1 & 2	176
G	KSNP	PWR(1000)	YGN 3, 4, 5 & 6 Uljin 3 & 4	752

Testing of dc Motor Actuators for Motor-Operated Valves

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Abstract

This paper presents the results of dc-powered motor-operated valve (MOV) research sponsored by the U.S. Nuclear Regulatory Commission (NRC) and conducted at the Idaho National Engineering and Environmental Laboratory (INEEL). The tests measured the capabilities of typical dc-powered valve actuators during operation at simulated loads and operating conditions. Using a test stand that simulates the stem load profiles a valve actuator would experience when closing a valve against flow and pressure, we tested four typical dc electric motors and two gearboxes at conditions a motor might experience in a power plant, including such off-normal conditions as operation at high temperature and reduced voltage. We also monitored the efficiency of the actuator gearbox and the efficiency of the actuator-torque/stem-thrust conversion at the valve-stem/stem-nut interface (stem nut coefficient of friction). The testing produced the following results:

- For both of the actuator gearboxes we tested, the actual running efficiencies were lower than the published running efficiencies. Below certain motor speeds, actual pullout efficiencies were lower than the published pullout efficiencies. Because of the decrease in gearbox efficiency at low-speed, high-torque operation, increases in motor torque at motor speeds lower than about 300 rpm failed to produce a corresponding increase in actuator output torque. Thus, in these MOV applications, a worm shaft speed threshold of about 250 rpm represents the lower limit for the production of usable output.

- For the motors we tested, estimates that anticipated linear reductions in both motor torque and motor speed fell very close to actual dc motor performance at reduced voltage. However, in some instances the actual and predicted performance fell below the motor speed threshold identified in the previous paragraph. The conventional linear method used in the industry for predicting reduced-voltage-related torque losses underestimated the actual torque losses; this comparison looked at the same motor speed in tests at different voltages.

- For all four motors, the actual motor torque losses due to elevated temperature conditions were significantly greater than losses indicated by the manufacturer's published data. The motor torque loss was approximately linear with the change in temperature.

- For all four motors, changes in running load had a significant effect on valve stroke times. Longer stroke times combined with operation at low speeds and high loads cause additional motor

heating and further degradation in motor performance.

- At normal voltages and temperatures, two motors produced torque at or above the torque indicated by the manufacturer's published torque/current and torque/speed curves. Two motors produced less torque than indicated by the manufacturer's curves.

- For all four motor/gearbox combinations, the high loads and slower speeds had little effect on the stem nut coefficient of friction.

Background

During the past several years, the U.S. Nuclear Regulatory Commission (NRC), Office of Nuclear Regulatory Research has supported research at the Idaho National Engineering and Environmental Laboratory (INEEL) addressing the performance of motor-operated valves. The research included tests and analyses to determine the capability of safety-related MOVs to perform their intended functions when subjected to their design-basis conditions. For some of these valves, the design-basis conditions include high flow and pressure loads, elevated temperature, and operation of the electric motors at reduced voltages. A detailed discussion of the testing, results, and conclusions can be found in Reference 1.

This paper presents the results of tests performed to address factors that affect the performance of dc-powered actuators for MOVs. Specifically, the testing addressed the following questions:

- What is the actual efficiency of the actuator gearbox, and how do high loadings and low speeds affect the actual efficiency? How does the actual efficiency compare with the manufacturer's published efficiency values?

- How does the output torque, current, and speed of the dc motors change as the voltage supplied to the motor decreases? How do these measured values compare with estimates produced by typical analytical predictions?

- How does the output torque, current, and speed of the dc motors change as the operating temperature of the motor increases? How do these measured values compare with estimates produced by typical analytical predictions?

- How does the output torque, current, and speed of the dc motors change as the motors heat up under high load conditions? How does stroke time change with increasing load?

- How does the actual output torque, current, and speed of the dc motors compare with the torque, current, and speed characteristics published by the manufacturer?

- What are the actual valve-stem/stem-nut efficiency and load sensitive behavior characteristics? How do the high loadings and low motor speeds affect these characteristics?

Test Equipment

The tests were conducted at the INEEL on the motor-operated valve load simulator (MOVLS) shown in Figure 1. The MOVLS is an instrumented test stand that provides dynamometer-type testing of valve actuators using load profiles that are very similar to the load profile a valve actuator would experience when closing a valve against a flow load. For these tests, we imposed a gradually increasing load on the valve actuator until the load caused the motor to stall, while taking continuous measurements of motor speed, motor voltage and current, motor torque, valve stem torque (gearbox output torque), motor temperature, and other parameters.

In this test program, we tested three combinations of actuator gearboxes and electric motors:

- An SMB−0 actuator equipped with a 10-ft-lb dc motor
- An SMB−0 actuator equipped with a 25-ft-lb dc motor
- An SMB−1 actuator equipped with a 40-ft-lb dc motor.

Our research also included analysis of data from previous testing of an SMB−1 actuator with an older 40-ft-lb dc motor (Reference 2). Table 1 summarizes the information provided by the motor manufacturers and actuator manufacturer about the three actuator combinations, including the gear ratios of the helical gear sets. Table 1 also includes information about the older 40-ft-lb motor and the actuator tested under the previous research effort. Table 2 shows the motor nameplate information for each motor.

The tests included normal stroke tests and stall tests, with baseline tests at room temperature and normal voltage, a series of tests at various stages of degraded voltage, a series of tests at various stages of elevated operating temperature, and tests at selected combinations of the two conditions. Continuous measurements of gearbox input torque (motor torque) and gearbox output torque (valve stem torque) allowed us to monitor the gearbox efficiency during the entire test. Likewise, continuous measurements of valve stem torque and valve stem thrust allowed us to monitor the stem nut coefficient of friction during the entire test.

Actuator Gearbox Results

Gearbox efficiency is part of the relationship between the input torque and the output torque of an actuator gearbox. The output torque can be represented by the following equation:

$$Tq_{output} = Tq_{input}(Eff_{gearbox}OAR) \qquad (1)$$

where

Tq_{output}	=	output torque
Tq_{input}	=	input torque
$Eff_{gearbox}$	=	efficiency of the gearbox
OAR	=	overall gear ratio.

The input torque consists of the torque delivered by the electric motor to the input side of the gearbox, and the output torque consists of the torque delivered to the stem nut by the worm gear. The overall gear ratio is the total gear reduction in the gearbox—the number of motor revolutions required for one revolution of the stem nut. Overall gear ratios for the actuators we tested (Table 1) range from about 35:1 to about 70:1. The gearbox efficiency accounts for losses to friction at the helical gear set, the worm/spline interface, the worm/worm-gear interface, and the associated bearings. Typical efficiency values for actuator gearboxes are in the range of 0.4 to 0.6. The more efficient the gearbox performance (the less the loss to friction), the higher the efficiency value. The gearbox efficiency value does not include motor effects or friction at the stem/stem-nut interface, which are separate calculations. The main drive train components of an actuator gearbox are shown in Figure 2.

Typical gearbox efficiencies are referred to as pullout efficiency, stall efficiency, and running efficiency. The pullout efficiency is the lowest of the three. This value applies when the motor is lugging at very low speed under a load or starting up against a load. The stall efficiency is higher than the others because it includes consideration of motor inertia during a sudden stall; it is typically used in evaluations of possible overload problems. The running efficiency is typically used to estimate the efficiency of the gearbox at normal motor speed and normal loads.

Our tests, conducted on the MOVLS, were designed to determine actual gearbox efficiencies with the gearboxes subjected to a full range of possible loads. By measuring the motor and the actuator output, and by accounting for the gear reduction, we were able to continuously monitor the efficiency of a gearbox at various loads. In Figure 3, the upper left plot shows the valve stem torque (output torque) measured during the baseline test (100% voltage, room temperature) of the SMB–0 actuator with the 10-ft-lb dc motor. The negative convention for this measurement indicates that the actuator was being operated in the closing direction. Note that the actuator output torque gradually increases in a manner representing valve closure against a high-flow load. The lower right plot in Figure 3 shows the motor torque measured during the same test. By plotting the output torque (valve stem torque) versus the input torque (motor torque), we can produce an XY plot of gearbox performance, as shown in the upper right plot on Figure 3. The format of this plot is based on Equation (1); the slope from the origin (0,0) to any point on one of the data traces represents the gearbox overall ratio times the *actual* gearbox efficiency for that data point. The two straight lines represent the overall gear ratio times (a) the published *running* efficiency, and (b) the published *pullout* efficiency. Figure 3 shows that for this actuator, the actual gearbox efficiency lies mostly between the published running efficiency and the published pullout efficiency. However, at higher loads the actual gearbox efficiency approaches and then crosses below the published pullout efficiency.

Figure 4 shows the gearbox performance data for the SMB–0–10 dc actuator for the reduced voltage tests (upper right), elevated temperature tests at 100% voltage (lower left), and elevated temperature tests at 80% voltage (lower right). In each of the tests, the measured efficiency is near the published running efficiency when the motor is near its normal speed (early in the stroke), but drops toward the pullout efficiency as the motor approach stall. Here, the results suggest that as the motor speed decreases, the efficiency of the gearbox decreases. In the 60% voltage test, the efficiency crosses the pullout value at a motor torque of 8 ft-lb, in the 70% test at 9.5 ft-lb, and so on. In each instance, the specific decline in efficiency, most evident on the tail of the trace near motor stall, corresponds more with the change in motor speed than with the change in motor torque. For any of the four low-voltage traces (90 to 60% voltage), the efficiency at the peak motor torque (stall) is notably lower than the efficiency indicated by higher-voltage traces at the same motor torque value but at higher motor speeds. This performance is consistent for all of the dc actuators tested.

Because the dc motor produces progressively higher torque at lower speed, the traces shown in Figure 4 for the various low voltages are more distinctly separate than in typical ac actuator tests (See Reference 2). In contrast to ac motors, which produce their rated torque at moderate speeds (typically at about 1200 rpm for 1800-rpm ac motors), dc motors produce their rated torque at much lower speeds. Thus, a dc-powered actuator will have a lower efficiency than an ac-powered actuator when operated under the conditions that demand the rated output torque from the motor.

Note that as the motor approaches stall, the trace becomes approximately horizontal, indicating that no increase in stem torque occurs, even though the motor torque continues to increase. Figure 5 provides a closer look at this phenomenon, by plotting actuator output torque versus worm shaft speed; these data are from the degraded voltage tests of the 10-ft-lb motor. Notice that this measurement is *valve stem torque*, not *motor* torque. These data traces show that there is a worm shaft speed threshold below which the actuator produces no additional output torque. This is the case even though the motor does in fact produce additional torque below the threshold speed. The worm shaft speed threshold ranges from about 130 to 200 rpm (an eyeball estimate), corresponding with motor speeds of about 250 to 370 rpm in this actuator.

For all of the dc actuators tested, the data show that a worm shaft speed of about 150 to 250 rpm is the threshold below which no additional valve stem torque can be expected. The relationship between worm shaft speed and motor speed is different for different actuators, because of differences in the ratios of the motor gear sets, as listed in Table 1.

DC Motor Results

Degraded Voltage Testing of dc Motors

Operation at degraded voltage is a design-basis condition for some dc-powered motor-operated valves. As such, our testing included operating the 125-volt dc motors at 60, 70, 80, 90, and 100% of the rated voltage to determine the actual torque produced at these voltages.

Analytical evaluations of MOV capability typically use the following formula to account for reduced dc motor output at degraded voltage conditions:

$$Tq_{\mathrm{act}} = Tq_{\mathrm{rat}} \left(\frac{V_{act}}{V_{rat}}\right). \qquad (2)$$

Tq_{act} = actual motor torque
Tq_{rat} = rated motor torque
V_{act} = actual voltage
V_{rat} = rated voltage.

This formula is identical to the voltage squared calculation used for ac motors, except that the exponent is 1 instead of 2. As part of our data analysis, we compared the results of the degraded voltage tests to estimates calculated from Equation (2).

Figure 6 shows data from the reduced voltage tests of the 10-ft-lb dc motor. The figure presents four data plots addressing the parameters of primary interest. The upper left plot shows the temperature of the motor series field as a function of motor torque for the five reduced voltage tests. (These temperature values are calculated from electrical resistance values derived from measurements of the series field voltage and current.) Note that each test began near 70°F and experienced a 30 to 50°F temperature rise by the end of the run. This temperature increase represents the motor heating that occurs during the run as the motor operates against a load.

The voltage plot (lower left) shows that although each test began at its assigned nominal voltage, a voltage drop occurred during the run. This voltage drop is due to line losses and to losses that occurred in the dc power supply during actuator operation. Similar motor heating, line losses, and voltage drops will occur during dc-powered valve operation in a plant.

The comparison shown in Figure 7, which looks at torque losses at a given speed, shows that the actual torque losses are greater than predicted by Equation (2). For example, at 871 rpm, a voltage reduction from 120 volts to 97 volts (the actual voltage at that moment during the 80% nominal test) causes a loss of 3.2 ft-lb, a loss notably greater than the 2.0 ft-lb loss predicted by Equation (2).

The comparison method shown in Figure 7 (comparing actual versus predicted torque output at a given motor speed) has merit, in that it acknowledges motor speed as an important operating characteristic. For some valves, operability denotes not only that the valve will successfully close (or open), but also that it will do so within a specified stroke time. Also, operation of the dc actuator motor at very high loads and very low speeds introduces other concerns related to motor heating and high friction in the gearbox. Nevertheless, the comparison method shown in Figure 7 fails to account for the lower speed at which the motor does in fact achieve the predicted torque output.

The visual pattern projected by the five traces in Figure 7 indicates step changes not only toward the left side of the plot, indicating reduced motor torque with reduced voltage, but also toward the bottom of the plot, indicating reduced motor speed. We inferred from this pattern, and from the inadequacy of the comparison method described in the previous paragraphs, that reduced voltage produces a linear shift in the curves for both motor torque and motor speed. We therefore applied a linear relationship [similar to the torque relationship in Equation (2)] to the motor speed, as follows:

$$S_{act} = S_{rat} \left(\frac{V_{act}}{V_{rat}}\right). \tag{3}$$

S_{act} = actual motor speed
S_{rat} = rated motor speed
V_{act} = actual voltage
V_{rat} = rated voltage.

Figure 8 shows the results of this calculation for the 10-ft-lb dc motor. The estimates shown in this figure predict that operation at reduced voltage causes a linear reduction in motor speed as well as a reduction in motor torque. These estimates are very close to the actual measurements over the full range of test conditions. Note, however, that in some instances, the predicted and actual motor torque values fall below the worm shaft speed threshold (at a motor speed of about 300 rpm) where high gearbox friction renders additional motor torque useless.

Elevated Temperature Testing of dc Motors

For some actuator motors, operation at elevated temperature is one of the design-basis conditions that must be considered in analytical evaluations of MOV capability. The output of the electric motor tends to degrade at higher temperature, mostly because of the increased resistance in the motor windings. This is the case regardless of whether the increase in the motor temperature is caused by ambient conditions or by motor operation for extended periods or at high loads.

The actuator manufacturer recognizes this effect, and for dc motors that are expected to operate at high ambient temperatures, the manufacturer recommends the use of environmentally qualified RH insulated dc motors. The RH insulated motors are qualified for operation at 340°F, and the manufacturer provides information regarding the maximum temperature at which the motor nameplate torque can be produced for each motor design (Reference 3). The manufacturer also provides a table (Reference 4) recommending adjustments to the rated torque value for sizing a nuclear qualified actuator. According to this table, a dc motor with a rated torque of 40 ft-lb expected to operate at 340°F would be treated as if it were a 39-ft-lb dc motor. (The adjustment is greater for larger motors; for example, the adjusted torque of a dc motor with a rated torque of 60 ft-lb would be 54 ft-lb).

Figures 9 and 10 provided the results for the 40-ft-lb dc motor. This is a Class B motor, so we heated it only to 250°F. In the 100% voltage tests, the increase from room temperature to 250°F reduced the motor's torque output by 10 ft-lb at the rated torque of 40 ft-lb at 245 rpm. For the 80% voltage test, at 245 rpm, elevated temperature reduced the output by 8 ft-lb.

A comparison between the guidance provided by Reference 4 and the observed response on all four dc motors shows that the guidance consistently overestimates the capability of the motor under elevated temperature conditions. (The guidance provides for no loss in capability for the two smaller motors, and a loss of only 1 ft-lb for the 40-ft-lb motors.) This result is based on an analysis that compares motor output torque at the same motor speed but at various temperature conditions. This analysis approach has merit, because it recognizes that some actuators are expected to close or open a valve within a specified stroke time. In instances where stroke time is not an issue, it might be possible to use a different analysis approach that recognizes motor output torque at lower speeds, for comparison with predicted values. However, such an approach would need to account for the motor speed threshold below which the motor produces no additional useful actuator output torque, an issue discussed earlier in this paper.

Based on our observation of the response of the motors during the elevated temperature tests, it was apparent that temperature has a linear effect on output torque, similar to the temperature effect on the resistance of copper wire. Also, the shift in the motor speed versus torque curves appears to be a horizontal shift (motor torque axis) only. For example, compare Figure 8 with Figure 9 and note the difference in the pattern projected by the traces in the lower right plots.

We also recognized that only the shunt field in a compound-wound dc motor is a strong function of resistance, and that treating the entire motor as resistance-dependent would result in overprediction of the temperature effect. We therefore applied a linear relationship to estimate the actual torque from the rated torque, based on the ratio of the change in temperature to the temperature above absolute zero. This relationship is

$$Tq_{act} = Tq_{rat}\left(1 - \frac{T_e - T_a}{TV_a - T_z}\right) (4)$$

where:

Tq_{act}	=	actual motor torque
Tq_{rat}	=	rated motor torque
T_e	=	elevated temperature
T_a	=	ambient temperature (room temperature of about 70°F)
T_z	=	absolute zero (−273.15°C or −459.67°F).

Figures 11 and 12 show the results of this calculation for the 40-ft-lb motor, for 100% and 80% voltage, respectively. We have arbitrarily selected 500 and 1,000 rpm as the motor speeds for which we perform the analysis. In these figures, Equation (4) has been applied to the motor torque to estimate the elevated temperature performance. This produces an estimate that is reasonably close to the actual measurements. In all cases, these estimates are based on the actual motor temperature at that point in time in the test, not the nominal temperature at which the test began. We also adjusted the estimates to account for any differences in voltage that might result from the voltage drop that occurs during the test, as discussed in the preceding section of this paper.

Effects of Load on Stroke Times and Motor Heating

Figure 13 shows the results of six tests of the SMB−0−10 dc-powered actuator. In these tests, the MOVLS was set up to create a fairly constant load for the entire stroke until the hydraulic cylinder bottomed out, simulating valve wedging. The top plot presents results from three tests with loads nominally designated low, medium, and high at 100% voltage, and the bottom plot presents results from tests at 80% voltage. At 100% voltage, the 3,000-lb running load produced an 8-sec stroke time; the stroke time was greater by 1 sec, or 12.5%, in the test with the 7,000-lb load. Combining this load increase with a voltage reduction to 80% yielded a 3.5-sec increase in stroke time, a 44% increase.

Performance Curves for dc Motors

The following discussion takes a close look at the performance of the motors at 100% voltage and room temperature (70 to 80°F) and compares the measured performance to the theoretical performance curves published by the actuator manufacturer.

The manufacturer's curve is a theoretical curve that does not account for the motor heat-up and voltage drop that occur during the run. For comparison purposes, we have included in the lower plot in Figure 14 a curve representing the actual speed versus torque data, adjusted for voltage drop and temperature. This adjustment is based on a linear voltage relationship for motor torque and motor speed and a linear temperature relationship based on the resistance in the series field. Our intention in this analysis is to create a *theoretical* performance curve derived from measurements, for comparison with the *theoretical* curve published by the manufacturer.

Figure 14 presents data plots showing motor current versus torque (upper plot) and motor speed versus torque (lower plot) from our baseline test of the 25-ft-lb dc motor. This motor required slightly more current than indicated by the manufacturer's curve: 56 amp at the 25 ft-lb rated torque, compared with 54 amp on the manufacturer's curve. The actual torque at a given speed is lower than indicated by the manufacturer's curve, reaching stall at 31 ft-lb, compared with 40 ft-lb on the manufacturer's curve. However, adjusting the test data to remove the effects of voltage drop and motor heatup during the run produces a curve that matches the manufacturer's curve.

The actual torque output of three of the four dc motors (torque at a given current) was lower than predicted by the manufacturer's curves. However, some line voltage drop occurred, and the motors were observed to heat up during the testing. With both motor speed and motor torque adjusted for voltage drop, and with motor torque adjusted for temperature, the torque output of the 10-ft-lb dc motor was well above that predicted by the manufacturer's curves, while the output of the 25-ft-lb dc motor closely matched the manufacturer's curves.

Stem/Stem Nut Results

In rising stem MOVs, the conversion of valve stem torque output to a stem thrust output occurs at the stem nut, as shown earlier on Figure 2. The ratio of valve stem torque to stem thrust is generally referred to as the stem factor. For a specific valve stem and stem nut, the only variable in the conversion of torque to thrust is the coefficient of friction, as shown in the following power screw equation.

$$\frac{Tq_{output}}{Th_{stem}} = \frac{d(0.96815 \tan a + \mu)}{24(0.96815 - \mu \tan a)} = \text{stem factor} \quad (5)$$

where

Tq_{output}	=	output torque of the valve actuator
Th_{stem}	=	valve stem thrust
d	=	$O.D._{Stem} - \frac{1}{2} Pitch$
$\tan a$	=	$\frac{Lead}{\pi d}$
μ	=	stem/stem-nut coefficient of friction.

The above equation (Reference 5) is written for U.S. Customary units, where torque is in foot-pounds, thrust is in pounds force, and stem diameter and thread pitch and lead are in inches. The pitch is the distance from the peak of one thread to the peak of an adjacent thread (inches/thread). The lead is the distance the stem travels in one revolution of the stem nut (inches/stem revolution). As an example, if the configuration consists of two threads spiraling the stem instead of one, the lead is different from the pitch. The diameter, pitch, and lead information for the stems we tested is listed in Table 1. The output torque consists of the torque delivered to the stem nut, which is equal to the torque reacted by the valve stem. The stem thrust is the thrust applied to the valve stem to move the stem and valve disc. The ratio of torque to thrust, shown in Equation (5), is the stem factor. The term *d* represents the mean diameter of the stem in terms of the thread contact area, and the term *tan a* is the slope of the thread. The *Pitch*, *Lead*, and *O.D.* for each stem are listed in Table 1. The behavior of the friction coefficient μ in the reduced voltage and elevated temperature tests deserves close examination, because dc-powered actuators slow down much more than ac-powered actuators do when subjected to high load, reduced voltage, and elevated temperature.

In Figure 15, the upper left plot shows the valve stem thrust measured during the 100% voltage test of the SMB–0 actuator with the 10-ft-lb dc motor. The negative convention for this measurement indicates that the actuator was being operated in the closing direction. Note that the valve stem thrust gradually increases in a manner representing a valve closure against a high-flow load. The lower right plot in Figure 15 shows the valve stem torque (output torque) measured during the same test. By plotting the valve stem thrust versus the actuator output torque, we can produce an XY-plot of stem/stem-nut efficiency, as shown in the upper right plot on Figure 15. The format of this plot is based on Equation (5); the slope from the origin (0,0) to any point on the data trace represents the inverse of the stem factor for that data point:

$$Th_{stem} = \frac{1}{SF} Tq_{output} \tag{6}$$

The three straight lines represent stem/stem-nut coefficients of friction of 0.12, 0.13, and 0.14. This format allows comparison of these fixed values with the actual stem nut performance over the entire operating range (in terms of torque load). For example, Figure 15 shows that for this actuator, the actual stem nut coefficient of friction starts out near 0.14; as the load increases, the friction improves to near 0.13. The data crosses 0.12 near the very end of the stroke, but because the motor is essentially at stall, this occurrence is not important for this analysis.

Conclusions

The conclusions of this research are summarized in the following paragraphs, with main topics indicated in italics.

Gearbox efficiency. Overall, the test results show that actual efficiencies can differ from those published by the actuator manufacturer. For the actuators we tested, the published running efficiency was generally not adequate for predicting actual performance of the gearboxes, especially at higher loads. The published pullout efficiency was adequate for predicting gearbox performance for some gearboxes and at some conditions (moderate loads), but at very low speeds, some of the efficiency data fell below the published pullout efficiency.

Each gearbox appears to have a minimum speed below which the pullout efficiency is no longer bounding. This friction effect resulted in a threshold motor speed below which additional motor torque produced little or no additional valve stem torque. For all four actuators we tested, this worm shaft speed threshold was at about 250 rpm. For these actuators, motor torque increases below this threshold cannot be relied upon to produce additional actuator output torque.

Degraded voltage. For the motors we tested, estimates that anticipated linear reductions in both motor torque and motor speed fell very close to actual dc motor performance at reduced voltage. Note, however, that in some instances, the actual and predicted performance fell below the worm shaft speed threshold discussed above. (High motor torque values that fall below the threshold cannot be relied upon to produce correspondingly high actuator output torque.) The conventional linear method used in industry for predicting reduced-voltage-related torque losses under-estimated the actual torque losses; this comparison looked at the same motor speed in tests at the same motor speed at different voltages.

Elevated temperature. In elevated temperature testing of the dc motors we tested, the adjustments recommended by the manufacturer for accounting for torque losses due to motor heating under-estimated the actual torque losses. Elevated temperature had an immediate effect on dc motor output torque; the motor torque reduction was approximately linear with the change in temperature.

Stroke times. Changes in running load had significant effects on valve stroke times.

The results suggest that longer stroke times combined with operation at low speeds and high loads can cause additional motor heating, which would further degrade motor performance.

Performance curves. The actual performance of three of the four dc motors (torque output at a given speed) was below that indicated by the manufacturer's generic curves. For example, the manufacturer's published curve for the 25-ft-lb motor indicated a torque of 40 ft-lb at motor stall, while the test data showed a torque of about 30 ft-lb. However, some line voltage drop and motor heating occurred during the run. With the motor speed data and the motor torque data adjusted for voltage drop, and with the motor torque data adjusted for temperature, the performance of the 10-ft-lb dc motor was well above that predicted by the manufacturer's curves, while the performance of the 25-ft-lb dc motor matched the manufacturer's curves very well.

The newer 40-ft-lb dc motor performed about the same as the older 40-ft-lb dc motor. The performance of both motors was below that predicted by the manufacturer's generic curves, even after adjustments for voltage and temperature.

Stem/stem-nut coefficient of friction. The high loads and slower speeds had little effect on the stem/stem-nut coefficient of friction in the actuators we tested. We found no additional load-sensitive-behavior concerns for dc motor actuators beyond those applicable to ac motor actuators.

References

1. NUREG/CR−6620, *Testing of dc-Powered Actuators for Motor-Operated Valves,* K. G. DeWall, J. C. Watkins, D. Bramwell, Idaho National Engineering and Environmental Laboratory, INEEL/EXT−99−00083, May 1999.

2. NUREG/CR−6478, *Motor-Operated Valve (MOV) Actuator Motor and Gearbox Testing,* K. G. DeWall, J. C. Watkins, D. Bramwell, Idaho National Engineering Laboratory, INEL−96/0219, July 1997.

3. Limitorque 10 CFR Part 21 Notification, November 3, 1998.

4. Limitorque SEL−5, November 9, 1988.

5. Limitorque SEL−10, March 1988.

Table 1. Actuator information.[a]

	SMB–1–40	SMB–0–25	SMB–0–10	Old SMB–1–40
Motor Rated Torque (ft-lb)	40	25	10	40
Motor Stall Torque (ft-lb)	62	40	16	63
Motor Speed (rpm)	1900	1900	1900	1750
Motor Gear Set Ratio	32:40	37:35	25:47	32:40
Worm Gear Ratio	34:1	37:1	37:1	34:1
Overall Ratio (OAR)	42.50	34.96	69.56	42.50
Running Efficiency	0.50	0.55	0.50	0.50
Pullout Efficiency	0.40	0.40	0.40	0.40
Stall Efficiency	0.50	0.55	0.50	0.50
Stem Diameter (in)	2.13	1.75	1.25	2.13
Stem Pitch/Lead	¼ / ½	¼ / ¼	¼ / ½	¼ / ½
Stem Speed (in/min)	22.4	13.6	13.7	22.4

a. Data provided by Peerless Electric Division, H. K. Porter Company, Inc. and Limitorque. Gearbox efficiency data from References 4 and 5.

Table 2. Motor nameplate information.

	40-ft-lb motor	25-ft-lb motor	10-ft-lb motor	Old 40-ft-lb motor
Manufacturer	Porter Peerless	Peerless-Winsmith	Peerless-Winsmith	Porter Peerless
Frame	D202G	DK56H	DG56D	D202G
Voltage (vdc)	125	125	125	125
Time rating	––	––	––	––
Duty	5 min	5 min	5 min	5 min
Serial number	XF64300	ZV47576	QW49138	HG50272
Torque (ft-lb)	40	25	10	40
HP	2.89	1.805	0.72	––
KW	––	––	––	––
Insulation type/class	B	RH	RH	B
Shunt Field Amps	––	––	––	––
RPM	1900	1900	1900	1750
Amps	24	14.5	6.5	––
Rise °C	115	115	115	––
Ambient °C	40	40	40	––
Type Winding	Comp. 175–34–0009–0	Comp. 176–18–0048–0	Comp. 176–18–0047–0	Comp. ––

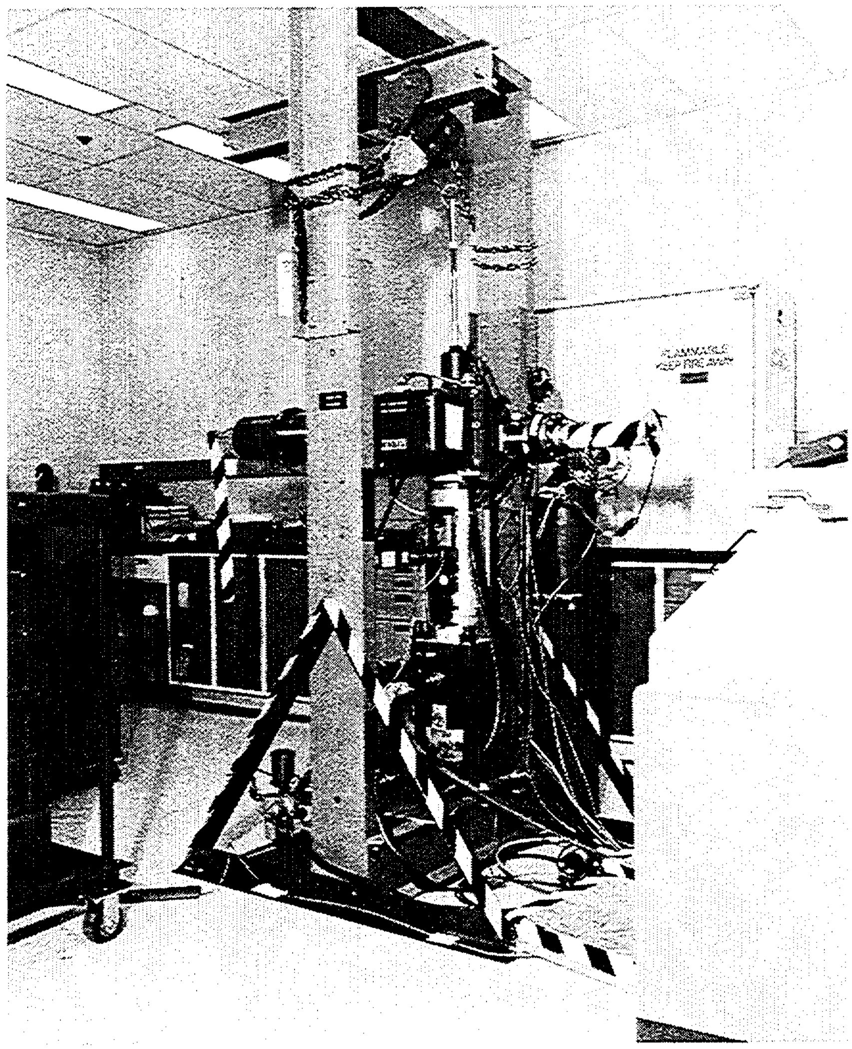

Figure 1. Photograph of the motor-operated valve load simulator (MOVLS).

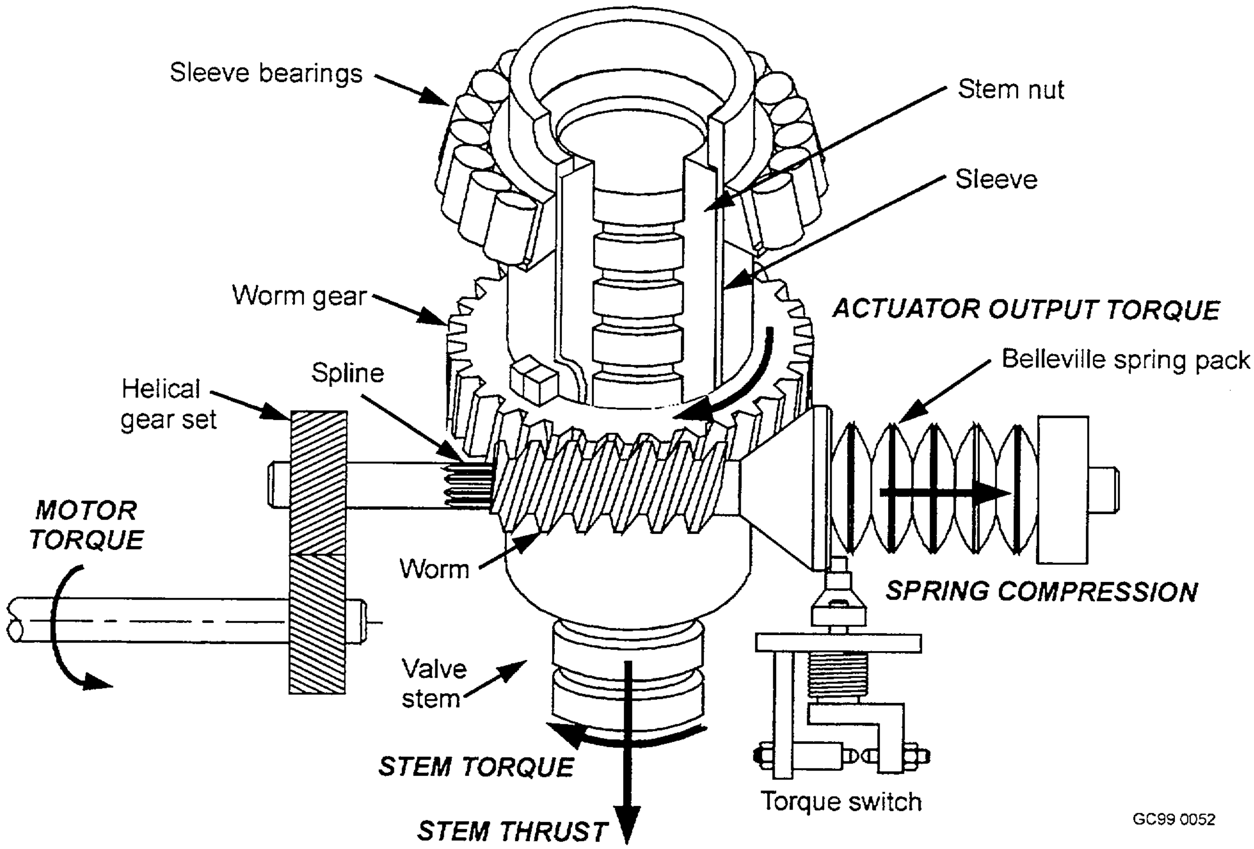

Figure 2. Diagram of the main components inside an actuator gearbox.

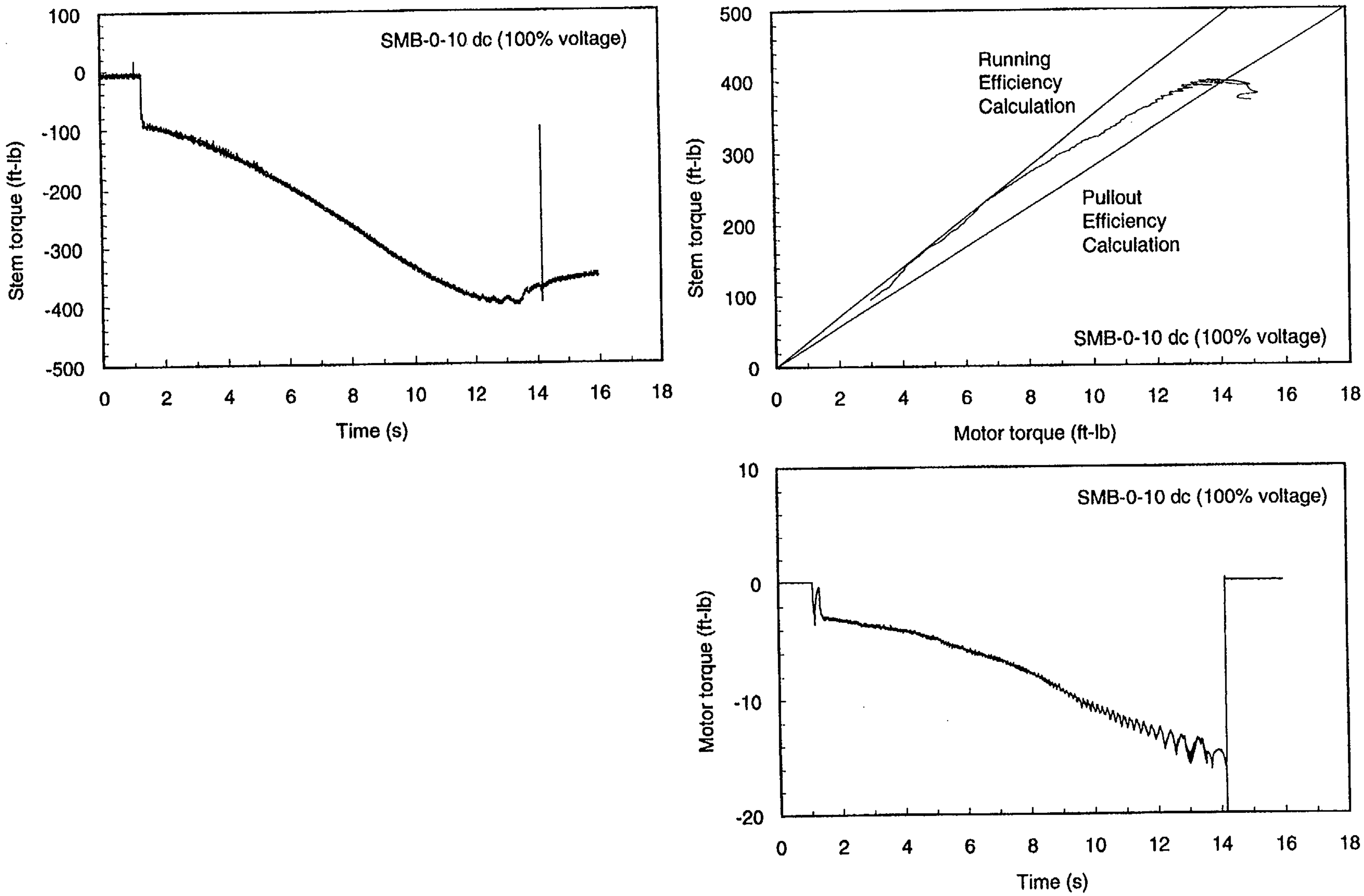

Figure 3. Actuator gearbox input torque (motor torque), output torque (stem torque), and efficiency calculations, derived from testing of the SMB–0–10 dc actuator.

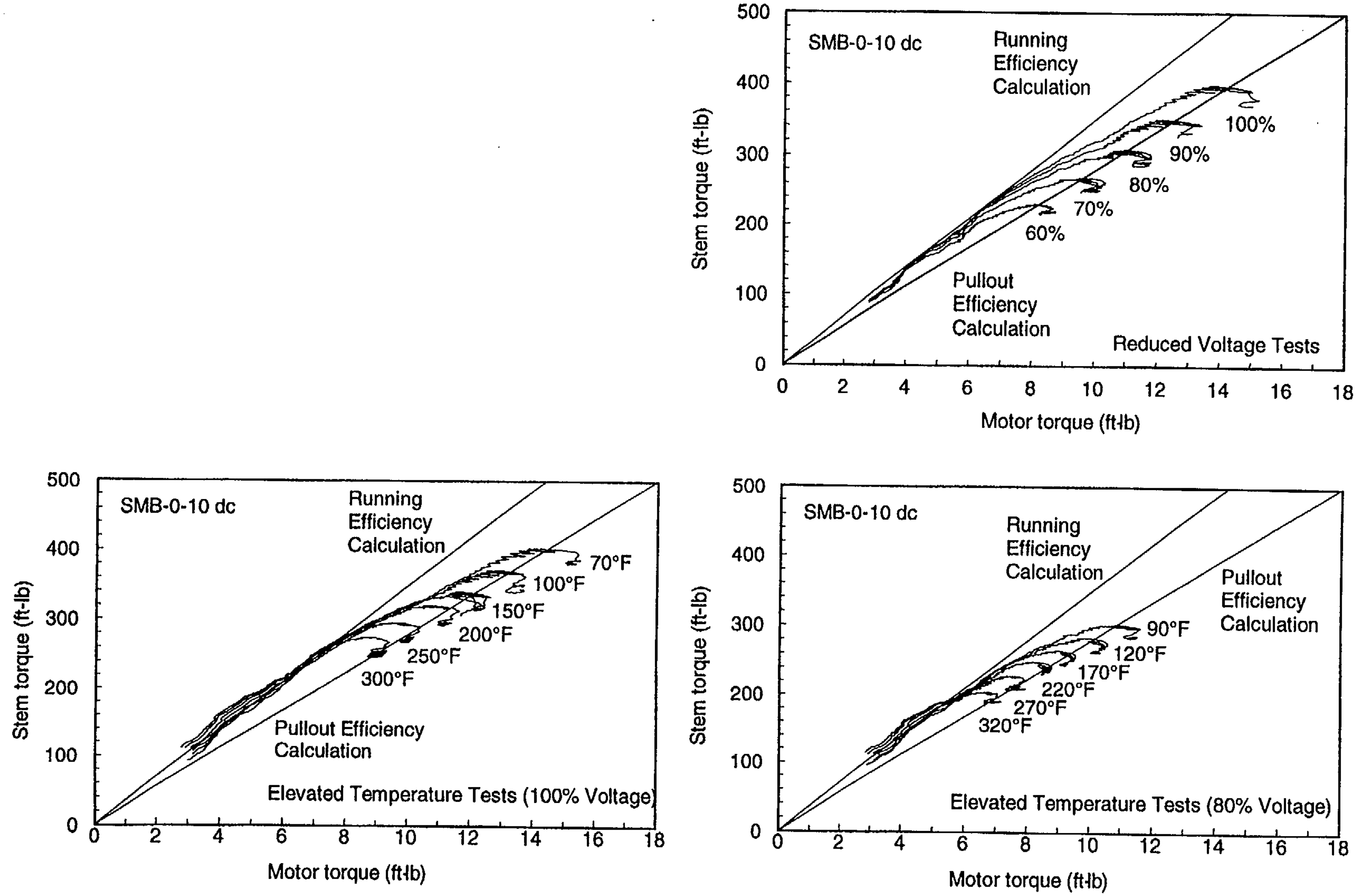

Figure 4. Gearbox efficiency calculations derived from testing of the SMB–0–10 dc actuator.

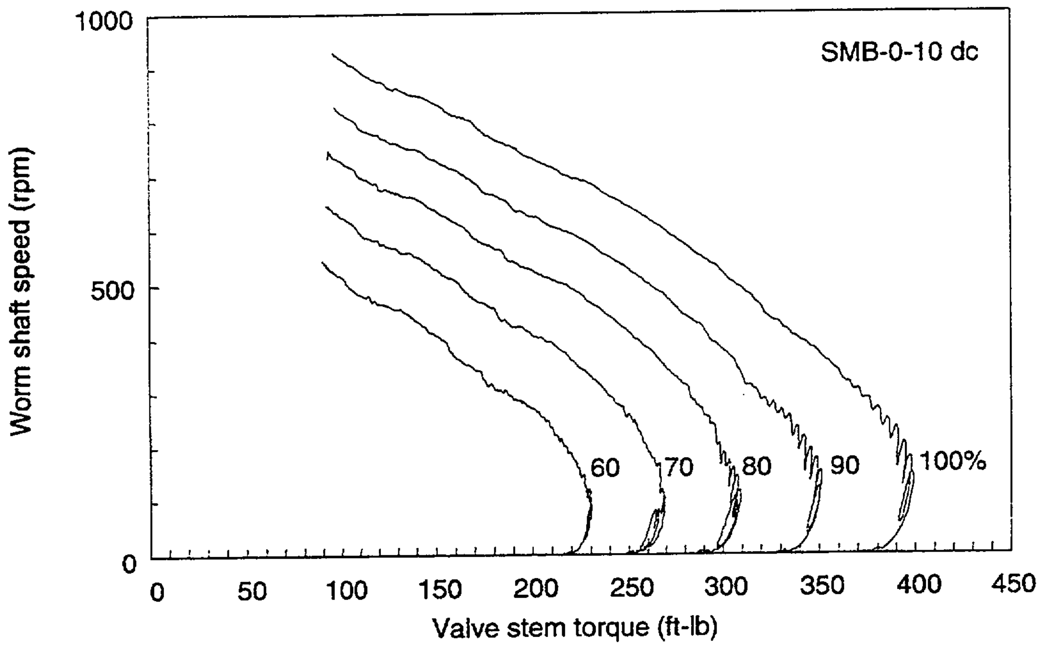

Figure 5. Worm shaft speed versus actuator output torque, derived from testing of the SMB–0–10 dc actuator at degraded voltage.

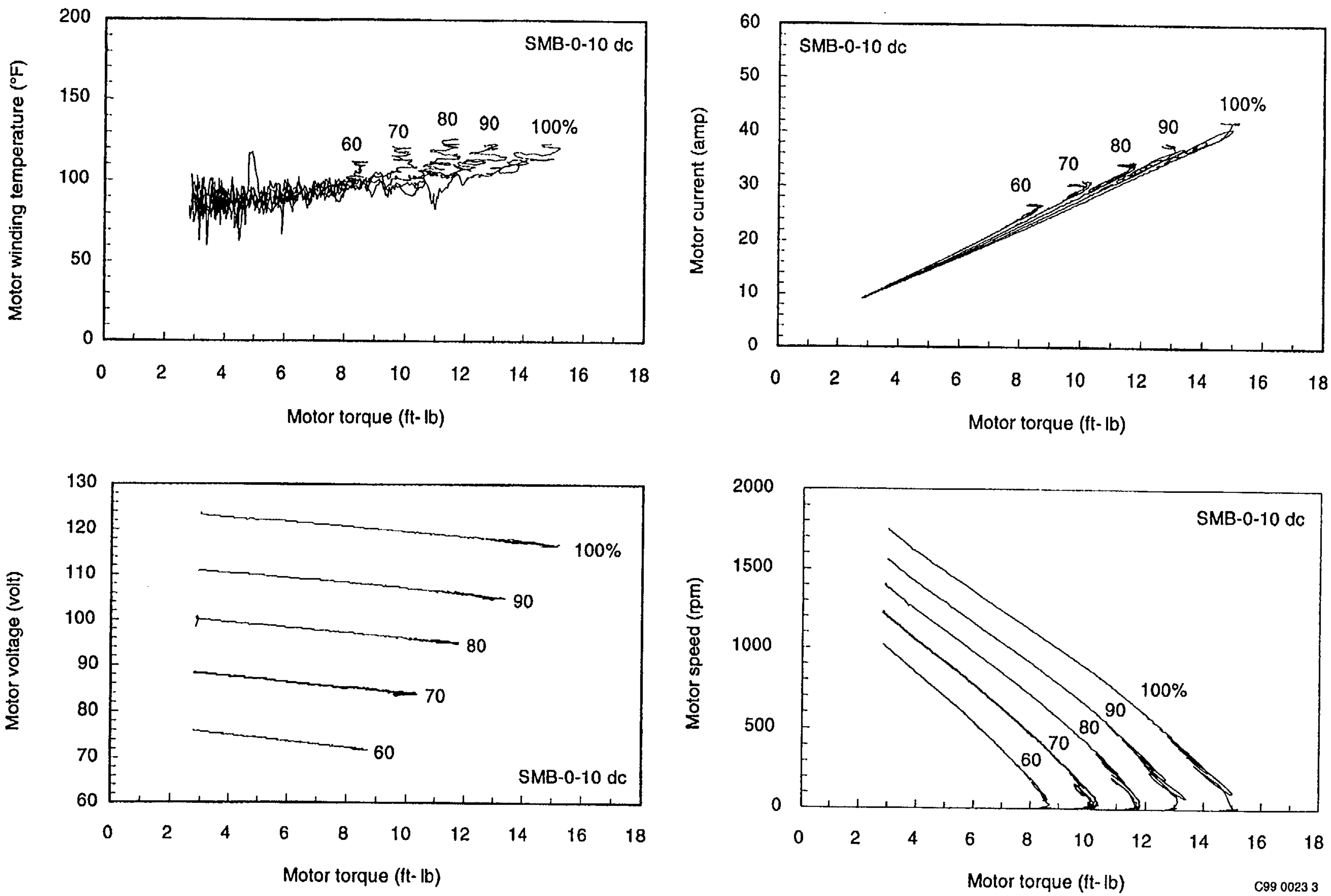

Figure 6. Motor temperature, current, voltage, and speed versus torque, derived from testing of the 10-ft-lb dc motor at degraded voltage.

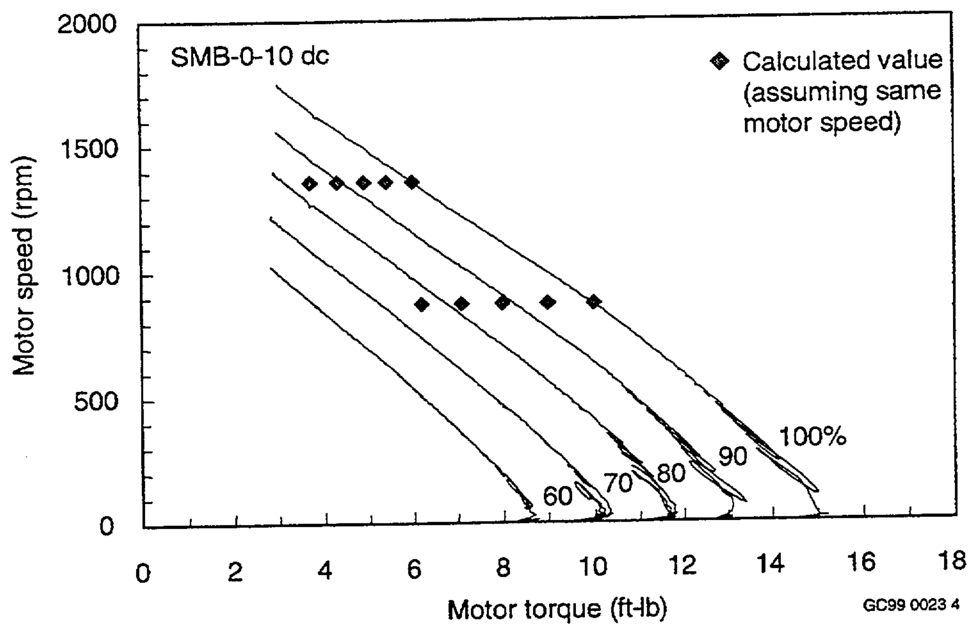

Figure 7. Motor speed versus torque, derived from testing of the 10-ft-lb dc motor at degraded voltage, with predictions of torque loss at a given speed.

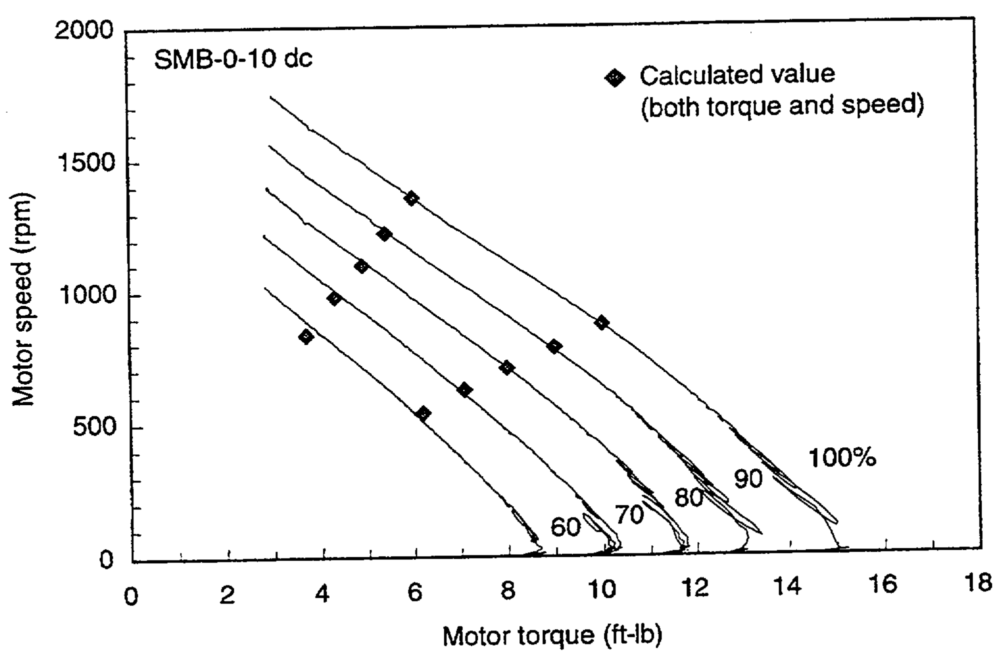

Figure 8. Motor speed versus torque, derived from testing of the 10-ft-lb dc motor at degraded voltage, with predictions based on the voltage ratio applied to both motor torque and motor speed.

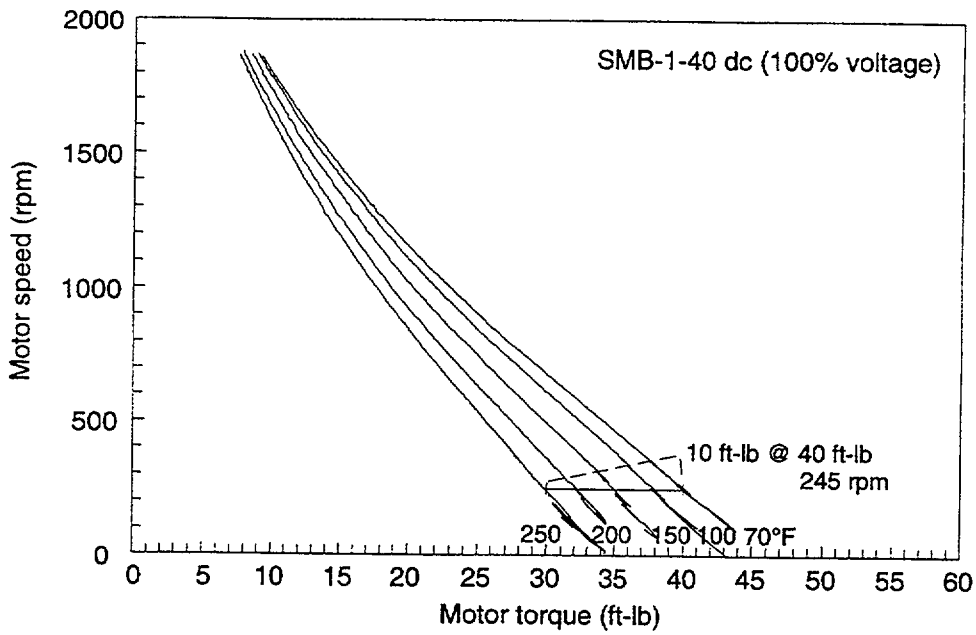

Figure 9. Motor speed versus torque, derived from testing of the 40-ft-lb dc motor at elevated temperature and 100% voltage .

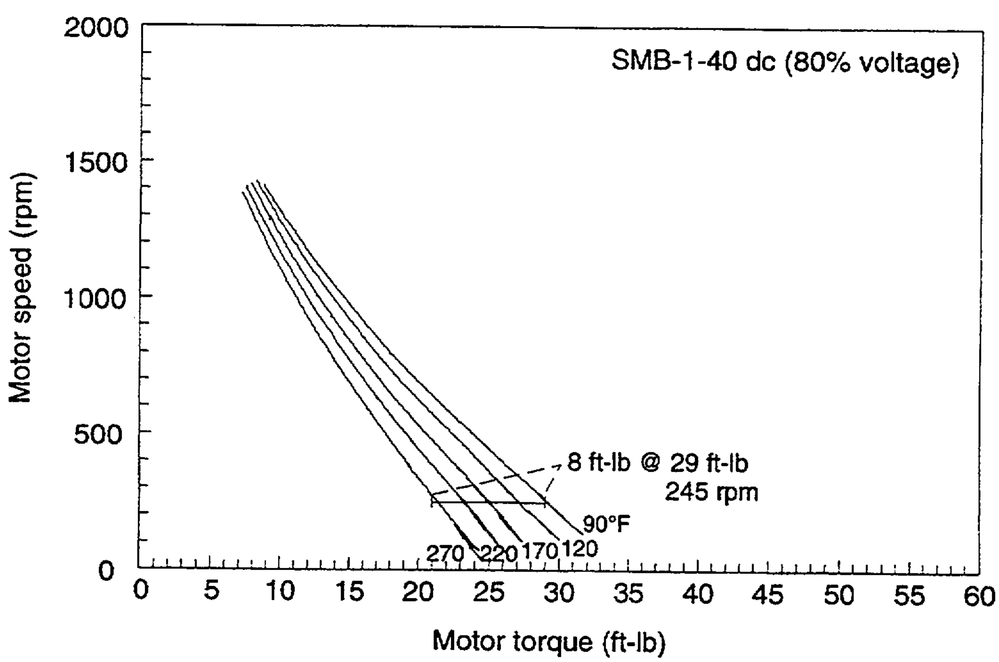

Figure 10. Motor speed versus torque, derived from testing of the 40-ft-lb dc motor at elevated temperature and 80% voltage .

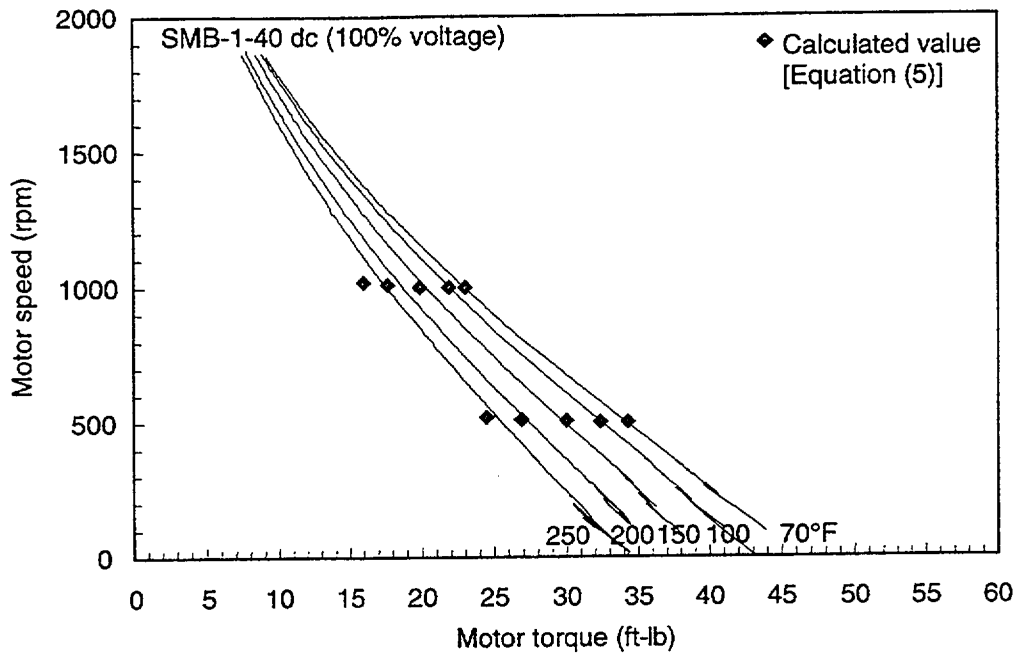

Figure 11. Motor speed versus torque, derived from testing of the 40-ft-lb dc motor at elevated temperature and 100% voltage, with predictions based on Equation (5).

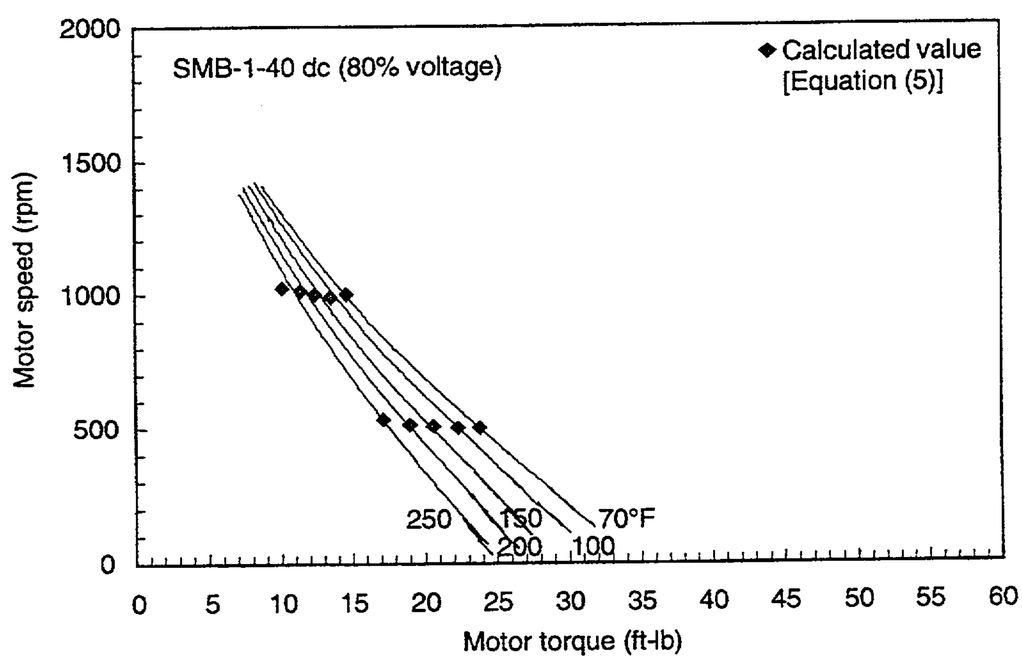

Figure 12. Motor speed versus torque, derived from testing of the 40-ft-lb dc motor at elevated temperature and 80% voltage, with predictions based on Equation (5).

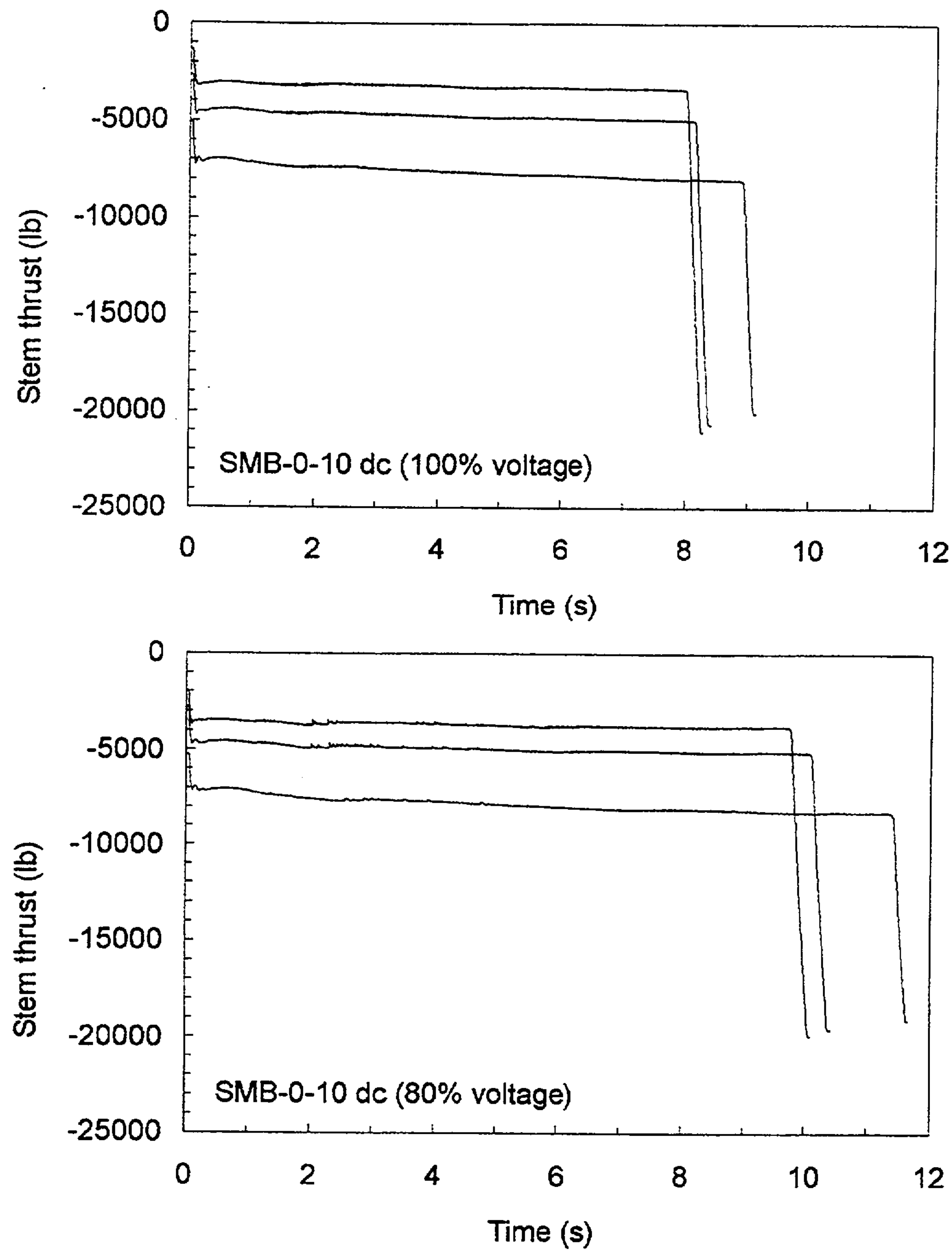

Figure 13. Valve stem thrust versus time, derived from stroke testing of the SMB–0–10 dc actuator at both 100% and 80% voltage.

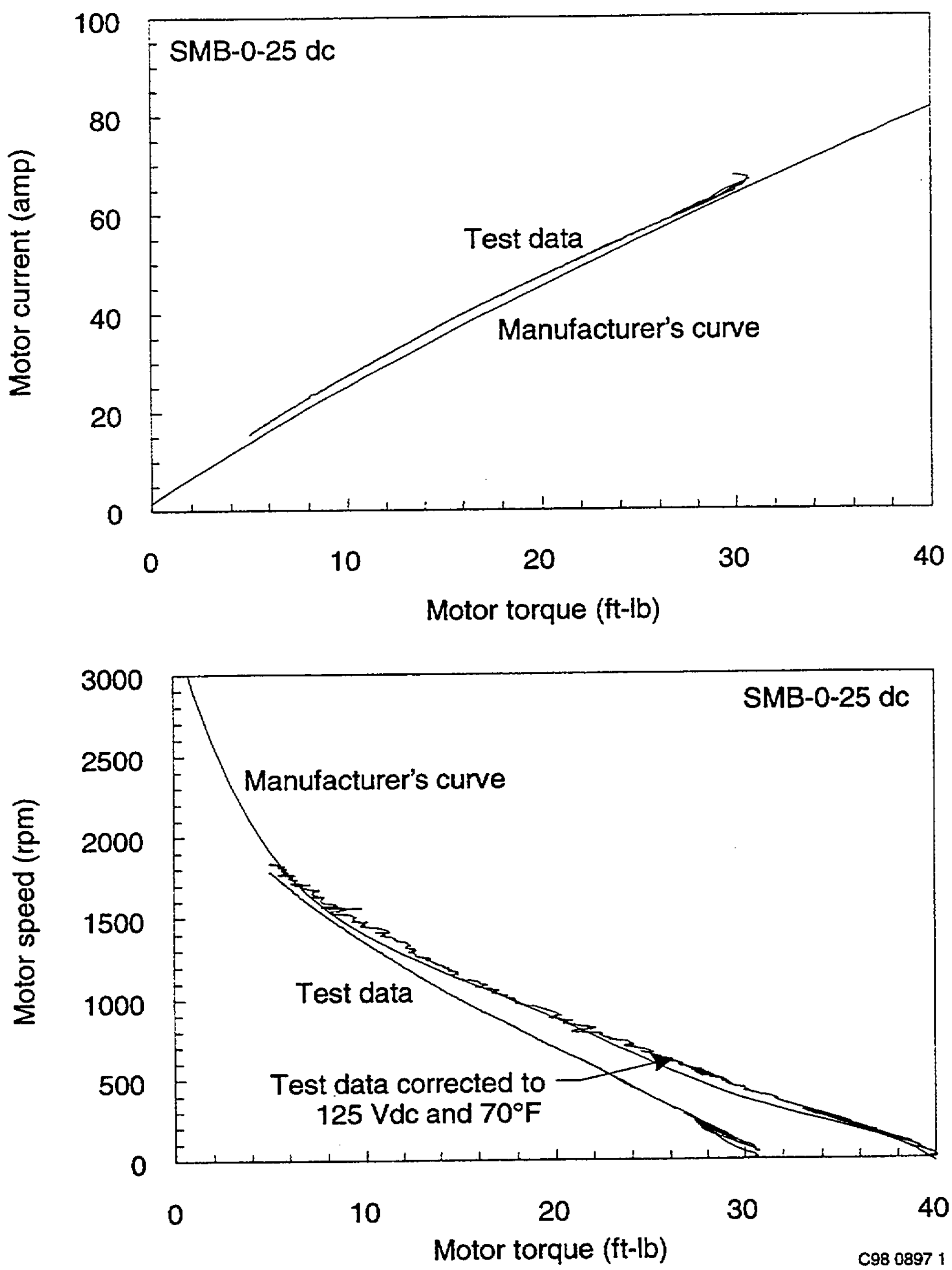

Figure 14. Motor performance curves derived from testing of the 25-ft-lb dc motor; manufacturer's published data are also shown.

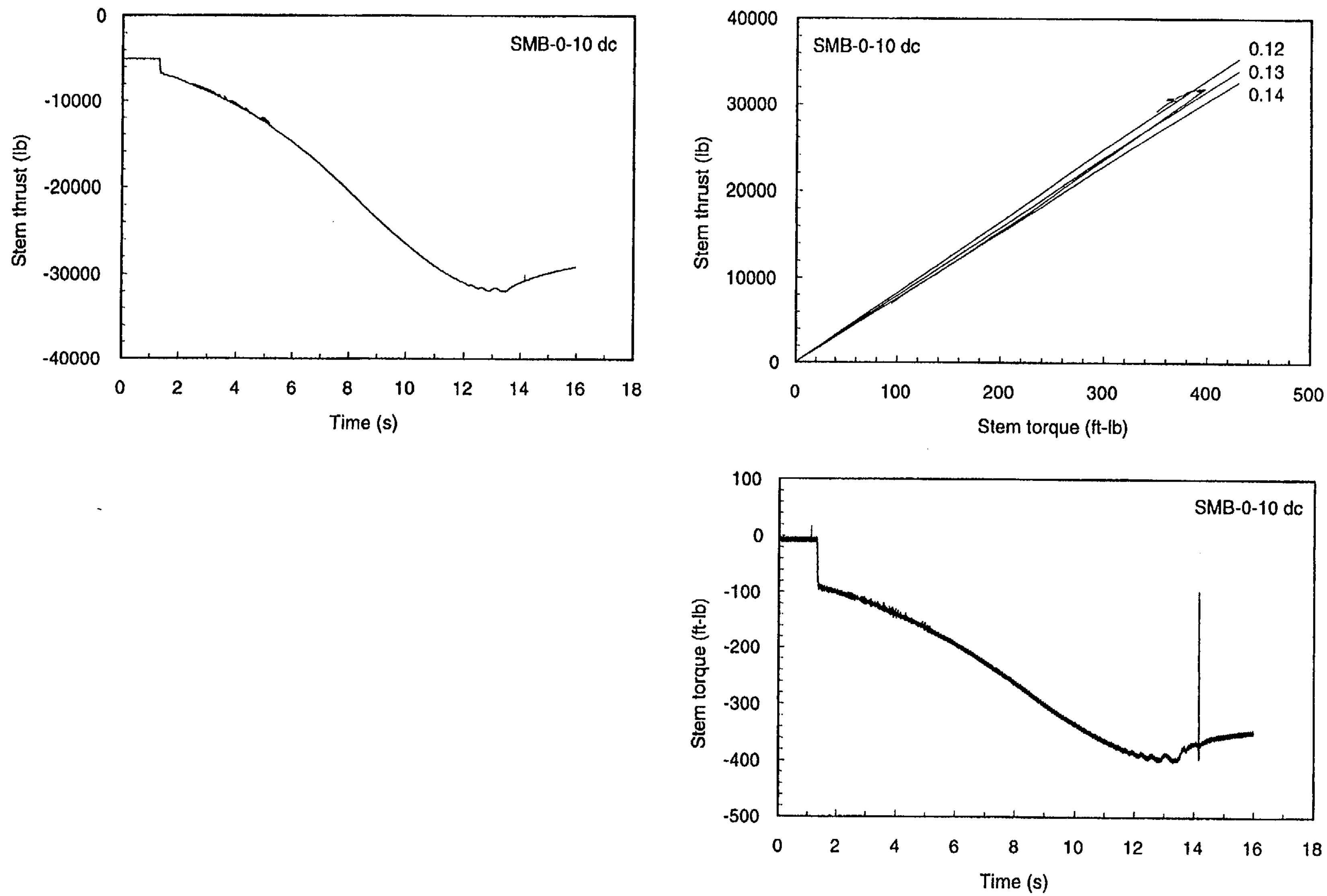

Figure 15. Valve stem thrust, stem torque, and stem factor calculations, derived from testing of the SMB–0–10 dc actuator.

Predicting Capability and Stroke Time In dc Motor-Operated Valves

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Abstract

The actuator performance and stroke time for valves powered by DC motor actuators are strongly affected by the load profile (which is affected by the fluid system conditions and the packing load), motor voltage and motor temperature. The actuator performance and stroke time measured during a test at nominal voltage and ambient temperature and with no flow or differential pressure in the pipe is not indicative of the valve's performance under design basis conditions.

This paper summarizes work sponsored by the BWR Owners' Group to develop a method for calculating the performance of DC MOVs. The method was developed based on first principles equations for DC motor performance and has been justified and validated using test data from a variety of sources, including motor dynamometer tests, actuator tests and in-plant MOV tests. As part of this justification and validation, the accuracy of the vendor performance curves for motors typically installed on MOVs in nuclear power plants was addressed, and recommendations were made for either using the vendor curves or alternate curves in the method for predicting motor performance.

The paper summarizes how the method was developed, justified and validated. The paper also describes how the method is implemented and provides an example method prediction.

Introduction

The actuator performance and stroke time for valves powered by DC motor actuators are strongly affected by the load profile (which is affected by the fluid system conditions and the packing load), motor voltage and motor temperature. The actuator performance and stroke time measured during a test at nominal voltage and ambient temperature and with no flow or differential pressure in the pipe is not indicative of the valve's performance under design basis conditions. Limitorque, who supplies most of the motor actuators installed on valves in nuclear plants, does not provide sufficient guidance for determining stroke time under design basis conditions and for evaluating the impact of motor heatup on actuator capability. Accordingly, a justified and validated method for calculating actuator thrust capability and stroke time of DC motor operated valves (MOVs) is needed.

To address this need, the BWROG has sponsored work to develop, justify and validate a DC motor performance method. This method includes procedures for predicting valve stroke time and actuator capability under design basis conditions. Specific features of the method have been justified using motor dynamometer and actuator test data, and the stroke time method has been validated against data from in-plant valve tests.

Scope and Applicability

The BWROG DC motor performance method can be used for any compound wound DC motor for which motor performance data are known and justified. DC motor types are identified by the nominal starting torque (e.g., 10 ft-lbs) and the nominal voltage (125 or 250 VDC). As part of justification and validation of the method, the accuracy of the vendor motor performance curves for motors typically used in nuclear power plant valves was evaluated. For some of the motors, the vendor motor performance curves were determined to be bounding and are recommended for use in the method. For other motors, the vendor motor performance curves were inadequate, and alternate motor performance curves, based on the test data, are recommended for use in the method. For motors that were not covered in justification and validation of the method, it is recommended that the vendor motor performance curves be used as "best available information" and that the user justify the data for input into the model. For example, users may perform motor dynamometer or actuator testing to justify motor performance data for use in the method.

The DC motor method includes a method for predicting the load profile, i.e., the stem thrust as a function of stroke position, based on the valve's design basis condition. The load profile method is applicable to solid wedge, flexible wedge, double disk and parallel disk gate valves with pumped or blowdown flow, and unbalanced disk globe valves with pumped flow. If the user specifies the stem thrust versus stroke position, there is no restriction related to valve type or flow type for use of the method.

Description of Method

The DC motor performance method includes three key elements—a load profile method, a stroke time prediction method and an actuator capability prediction method. Each of these methods is described below.

Load Profile Method

The valve load profile method is used in the DC motor performance method to determine the stem thrust as a function of stroke position. The method considers the valve stroke in discreet increments, and required valve stem thrust is determined at the beginning and end of each increment. Stroke increments were defined to emphasize portions of the valve stroke where the DP, and therefore the valve stem thrust and motor torque, are changing most rapidly. The stem thrust for each increment is calculated as the average of the stem thrust at the beginning and end of the increment.

The maximum required thrust is input by the user. If water inertia was considered in calculating the required thrust, the required thrust attributable to water inertia effects is also input. The required thrust at 0% open is set to the maximum required thrust (including the required thrust due to water inertia, for closing strokes only). For gate valves, the required thrust at negative stroke positions (after flow isolation for closing strokes and before flow initiation for opening strokes) is set to the maximum required thrust, not including the required thrust due to water inertia. The required thrust at fully open is calculated as the sum of the packing and stem rejection loads. The variation of required thrust from the value at fully open to 0% open is determined by the user from "load profile coefficients," which are based on the design basis conditions for the valve and provided in the method. Different coefficients are provided for wedge and parallel disk gate valves, for pumped and blowdown flow and for opening and closing strokes.

For gate valves with pumped flow, the load profile coefficients are determined by the user by calculating an overall resistance for the system in which the valve is installed and selecting the appropriate load profile coefficients. These coefficients were determined based on first principles modeling (freebody diagrams to determine the forces on the valve disk), computational fluid dynamics work from the EPRI MOV Performance Prediction Methodology (PPM) (to determine the horizontal and vertical loads on the valve disk as a function of DP) and the default PPM flow coefficients for gate valves (to determine the DP across the valve as a function of stroke position). Since the method does not predict when the disk transitions between the guides and the downstream seat, the load profile coefficients conservatively assume the worst case at each stroke position.

For gate valves with blowdown flow, the load profile coefficients are determined by the user by evaluating the equivalent length of piping upstream of the valve and selecting the appropriate load profile coefficients. These curves are based on predictions from the PPM. The load profiles predicted by this method for gate valves are expected to be similar to the load profiles predicted by the PPM.

Flow coefficients for globe valves can vary significantly, depending on design. The load profile method for globe valves assumes that the flow coefficient varies linearly with disk position, from 0 at fully closed to the maximum value at fully open. The valve flow coefficient at fully open is input by the user.

Stroke Time Method

Stroke time is calculated for each increment, and the predicted stroke time is the sum of the stroke times for all increments. Motor performance is evaluated based on a first principles model of a DC motor. This first principles model shows that the rotational speed of a DC motor at different voltages and winding temperatures can be related to the output torque using motor performance curves (i.e., torque and current versus speed) applicable under nominal voltage, ambient temperature conditions. DC motor or actuator vendors typically provide a motor performance curve that is applicable for nominal voltage and a reference temperature. The DC motor performance method uses these vendor-supplied motor performance curves (or alternate curves, where appropriate) to determine the motor speed and current from the motor torque.

The following steps are performed at each stroke increment.

- The average motor torque is calculated from the average stem thrust (from the load profile method) using the stem factor under design basis conditions, the actuator overall gear ratio and gearbox efficiency. Gearbox efficiency values (η) are included in the method and are calculated based on "f" factors, which are a function of the rotational speed of the worm, the stem thrust (as a function of the actuator rated thrust) and the run (η_R) and pullout (η_P)

efficiencies of the actuator. See the section below titled "Actuator Gearbox Efficiency" for a discussion of how the f-factors in the method were determined. The gearbox efficiency is calculated using the following equation.

$$\eta = \eta_P + f\,(\eta_R - \eta_P)$$

At low worm speeds (less than about 200 rpm) or low stem loads (less than about 6.5% of the actuator rated torque), values less than the pullout efficiency are used. The maximum value used in the method is pullout efficiency plus 70% of the difference between pullout and run efficiency.

- The motor current is determined from the torque using the motor performance curve. Per the first principles DC motor model, motor current is not affected by elevated temperature or degraded voltage. Therefore, adjustments to the current for voltage and temperature are not required. Motor terminal voltage is calculated using the Motor Control Center (MCC) voltage, current and resistance from the MCC to the MOV.

- The motor speed is determined as follows.

  - Adjusted motor torque is determined from the average motor torque by accounting for degraded voltage and motor temperature (the motor temperature at the end of the previous increment is used). This step is accomplished using a single equation that combines the two effects. Per the first-principles model, voltage has a linear effect on motor torque; therefore, the average motor torque is multiplied by the ratio of the nominal motor voltage to the calculated voltage at the motor terminals. Elevated temperature affects motor output torque because the resistance of the copper windings increases with temperature. Accordingly, the average motor torque is multiplied by the ratio of the resistance of copper at the calculated motor temperature to the resistance of copper at ambient temperature (25°C)

  - Adjusted motor speed is then determined from the motor performance curve, using the adjusted motor torque. Per the first-principles model, voltage has a linear effect on motor speed; therefore, the adjusted motor speed is multiplied by the ratio of the calculated voltage at the motor terminals to the nominal motor voltage.

- The time for the stroke increment is calculated based on the increment length, motor speed, stem lead and actuator overall ratio.

- The motor heatup is calculated using heatup rates provided by the motor manufacturer and added to the motor temperature at the beginning of the increment to determine motor temperature at the end of the increment. The motor or actuator manufacturer typically provides heatup rates at one or more reference torques. For other torques, the heatup rates are multiplied by the ratio of the square of the calculated motor torque to the reference torque.

- These steps are performed for the full stroke of the valve. The predicted stroke time of the valve is the sum of the times calculated for each stroke increment.

Actuator Capability and Margin

Two actuator capabilities and margins are evaluated—"instantaneous" actuator capability and margin, and "functional" actuator capability and margin. Maximum allowable thrust at torque switch trip (closing strokes) and at unwedging (opening strokes) are also evaluated. In all cases, the nominal motor torque capability is defined as the motor torque at 200 rpm under nominal temperature and voltage conditions.

The instantaneous actuator capability (thrust) is calculated at each stroke position by using the nominal motor torque, adjusted for degraded voltage, elevated motor temperature, and actuator gearbox efficiency (at that stroke position), along with the actuator overall gear ratio and stem factor. The instantaneous margin is calculated at each stroke position by subtracting the required stem thrust from the instantaneous actuator capability. The minimum instantaneous margin for a given valve stroke is an indication of the degree to which the actuator is challenged during the stroke.

The functional actuator capability is calculated by iteratively increasing the required stem thrust for the stroke until the nominal motor torque is reached at some point during the stroke. At this point, the instantaneous margin is zero. This step requires an iterative calculation. The functional actuator capability is an indication of the thrust capability of the actuator for that particular application, considering the effects of degraded voltage, elevated temperature and decreased efficiency as the motor slows down.

The minimum instantaneous margin will typically be higher than the functional margin; however, a valve with zero instantaneous margin will also have zero functional margin. Therefore, an MOV can perform its design basis function if the instantaneous capability exceeds the required thrust at all stroke positions or if the functional capability exceeds the maximum required thrust for the stroke. Therefore, either capability can be used to verify proper MOV design.

The maximum allowable thrust at torque switch trip is calculated for closing strokes only. For this calculation, the required thrust is assumed to increase instantaneously from the value at the last stroke position to a value corresponding to the nominal motor torque under nominal conditions. To determine the actual motor torque at this point, the nominal motor torque is adjusted for degraded voltage (calculated based on the current at nominal motor torque) and the temperature at the last stroke position. The gearbox efficiency is determined using an iterative procedure, based on the calculated motor speed at nominal motor torque and degraded voltage, and used with the nominal motor torque to calculate the maximum allowable thrust at torque switch trip.

The maximum allowable thrust at unwedging is calculated for opening strokes only. This calculation is analogous to the calculation for determining maximum allowable thrust at torque switch trip for closing strokes, except that the required thrust is assumed to increase instantaneously from the value at the first stroke position to a value corresponding to the nominal motor torque under nominal conditions.

Justification of Method

Specific features of the DC motor performance method have been justified using data from the following sources.

- Testing of three actuators with DC motors by the Idaho National Engineering and Environmental Laboratory (INEEL), sponsored by the Nuclear Regulatory Commission (NRC),
- Dynamometer and actuator testing of nine types of DC motors (covering 25 different motors) by Crane-MOVATS and Vermont Yankee (VY), and
- Dynamometer testing of two motors by a nuclear utility.

This data covers ten of the 24 motor types typically used on nuclear power plant DC MOVs and includes testing at nominal and reduced voltage. Justification of key model features is discussed in the sections below.

Use of Vendor Motor Performance Curves

Data from testing of all ten motor types was used to justify the method for adjusting the nominal motor performance data (e.g., the vendor curves) for degraded voltage and elevated temperature. The approach used was to adjust the measured motor torque and speed for each test for degraded voltage and elevated temperature using equations developed in the first-principles DC motor model. This adjusted test data was then plotted against the vendor motor performance curves. All data for a given motor type was shown on the same plot. Figure 1 shows results for the 15 ft-lb, 125 VDC motor. As shown in this figure, the vendor curve for this motor type bounds the test data. Figure 2 shows the results for the 100 ft-lb, 125 VDC motor. For this motor type, the vendor curve does not match the test data. For seven of the ten motors, the vendor motor curves were found to be bounding. For the other three motor types, the vendor curves used in the evaluation did not match the test data. For these three motor types, alternate motor performance data were determined from the test data and are recommended for use in the method, as shown in Figure 2 for the 100 ft-lb, 125 VDC motor. Limitorque is working with Peerless and the BWROG to address this inconsistency and to determine whether more appropriate motor performance curves exist for these motors. Note that motor heatup rates from the vendor curves are used for these motors. See the section below titled "Prediction of Motor Heatup and Use of Vendor Heatup Rates."

Actuator Gearbox Efficiency

As discussed previously, the DC motor performance method includes f-factors that are used to calculate gearbox efficiency. The f-factor values are a function of the rotational speed of the worm, the load on the stem (as a function of the actuator rated thrust) and the run and pullout efficiencies of the actuator. Values in the methods were determined from actuator testing performed by INEEL and sponsored by the NRC (Reference 1). This testing included SMB–0 and SMB–1 actuators. To justify the values used, f-factors calculated from the test data were compared to f-factors predicted by the model (based on the measured worm speed and stem load). Figure 3 shows a plot of measured versus predicted f-factor for the SMB–1 actuator with a 40 ft-lb, 125 VDC motor. The diagonal line in this plot represents the situation where the measured f-factor is equal to the predicted f-factor. Data above this line represents conservative f-factor predictions from the method. As shown, the predicted f-factors bound the measured f-factors for the entire stroke for this actuator. The f-factor plots for the other two actuator were similar to Figure 3.

Prediction of Motor Heatup and Use of Vendor Heatup Rates

There is limited data to justify the vendor motor heatup rates. The NRC/INEEL testing (Reference 1) provided data to justify the heatup rates for 10, 25 and 40 ft-lb motors. The DC motor performance method was implemented for the 60% and 100% voltage test of these motors to justify the motor heatup prediction. In general, the method predicts that the motors stall prior to the end of the test, indicating that the method provides conservative predictions of actuator output capability. The predicted temperature rises for these tests are the predicted temperature at the last stroke position before predicted motor stall minus the initial temperature. For one of the tests evaluated, the motor was not tested to stall, and the predicted temperature rise is the predicted temperature at the end of the stroke minus the initial temperature. For one of the tests, the piston (which provided resistance to the valve stem for the tests) bottomed out before the motor stalled. For this test, the method predicts that the motor stalls just as the piston bottoms out, and the predicted temperature rise is the predicted temperature at the final stroke position minus the initial temperature.

The table below summarizes the measured and predicted temperature rises. The motor speeds corresponding to the predicted temperatures rises are also listed.

Motor/Test	Maximum Measured Motor Torque, ft-lbs	Measured Temperature Rise, °F	Predicted Temperature Rise, °F
10 ft-lbs, 60% voltage	8.5	23	30 @ 74 rpm
10 ft-lbs, 100% voltage	15	40	44 @ 75 rpm
25 ft-lbs, 60% voltage	17	45	68 @ 103 rpm
25 ft-lbs, 100% voltage	30	80	97 @161 rpm
40 ft-lbs, 60% voltage	21	negligible	24 @ 236 rpm
40 ft-lbs, 100% voltage	40	40	55 @ 120 rpm

As shown above, the predicted temperature rises compare favorably to the measured temperature rises.

Validation of Method

To validate the DC motor performance method, test data from in-plant valve flow tests were obtained for seven MOVs from four utilities. The seven MOVs included wedge gate, double disk gate and globe valves, driven by six different actuator types and four different motor types (all Peerless). The test data cover 22 valve strokes (11 static strokes and 11 DP strokes). The DP strokes include pumped flow opening and closing strokes, steam flow opening and closing strokes and a hydrostatic opening stroke. All tests were performed at ambient temperature.

The predicted stroke times bound the measured stroke times for all strokes except two, a static closing stroke (40 ft-lb, 250 VDC motor) and a hydrostatic opening stroke (15 ft-lb, 125 VDC motor). In both cases, the predicted stroke times were within 2% of the measured stroke times.

For both strokes, the loads on the valve stem were small for the majority of the stroke. Therefore, these non-conservative results indicate that the motor speed may be slightly overpredicted at very low stem loads (at or near running load). The stroke time predictions for the other strokes of these valves matched or bounded the measured stroke times. In addition, another valve in the validation set has a 40 ft-lb/250 VDC motor, and the stroke predictions for that valve were bounding. There was ample data for 15 ft-lb, 125 VDC motors in the method justification to justify the vendor motor performance curve for these motors. Therefore, these slight stroke time underpredictions are considered acceptable.

Example

An Excel spreadsheet is provided with the DC motor performance method to assist in implementation of the method. Figures 4, 5 and 6 show an example implementation of the method using the spreadsheet. Figure 4 lists the inputs used for the example and the predicted stroke time (at the bottom of the figure). Figure 5 is a table that lists the following results (in columns) for each stroke increment.

- Stroke Position (%)
- Stem Thrust (pounds)
- Average Stem Thrust (pounds)
- Average Stem Torque (ft-pounds)
- Worm Speed (rpm)
- Gearbox Efficiency
- Average Motor Torque (ft-pounds)
- Motor Current (amps)
- Voltage at the Motor Control Center (V_{mcc}) (volts)
- Motor Voltage (voltage)
- Adjusted Motor Torque (ft-lbs)
- Motor Speed (rpm)
- Stroke Time Increment (seconds)
- Time (seconds)
- Motor Temperature (°C)
- Instant Torque Capability (pounds)
- Instantaneous Actuator Capability (pounds)
- Instantaneous Margin (pounds)

Figure 6 shows plots of stem thrust versus stroke position, motor speed and current versus motor torque and stem position, motor temperature, stem thrust and instantaneous margin versus time.

Summary

The BWROG DC motor performance method is a first-principles method for predicting the performance of DC motor operated valves. The method provides procedures for calculating stroke time and actuator capability under design basis conditions. Methods are also provided for determining the maximum allowable thrust at torque switch trip and at unwedging. The method has been justified and validated using motor, actuator and valve test data. The results show that the method provides bounding predictions of valve stroke time. Key aspects of the method, for example, use of the vendor motor performance curves, adjustment of the vendor curves for degraded voltage and elevated temperature and gearbox efficiency values, have been justified by the data. A spreadsheet is included with the method to facilitate the calculations.

Reference

1. NUREG/CR–6620, "Testing of dc-Powered Actuators for Motor-Operated Valves," May 1999

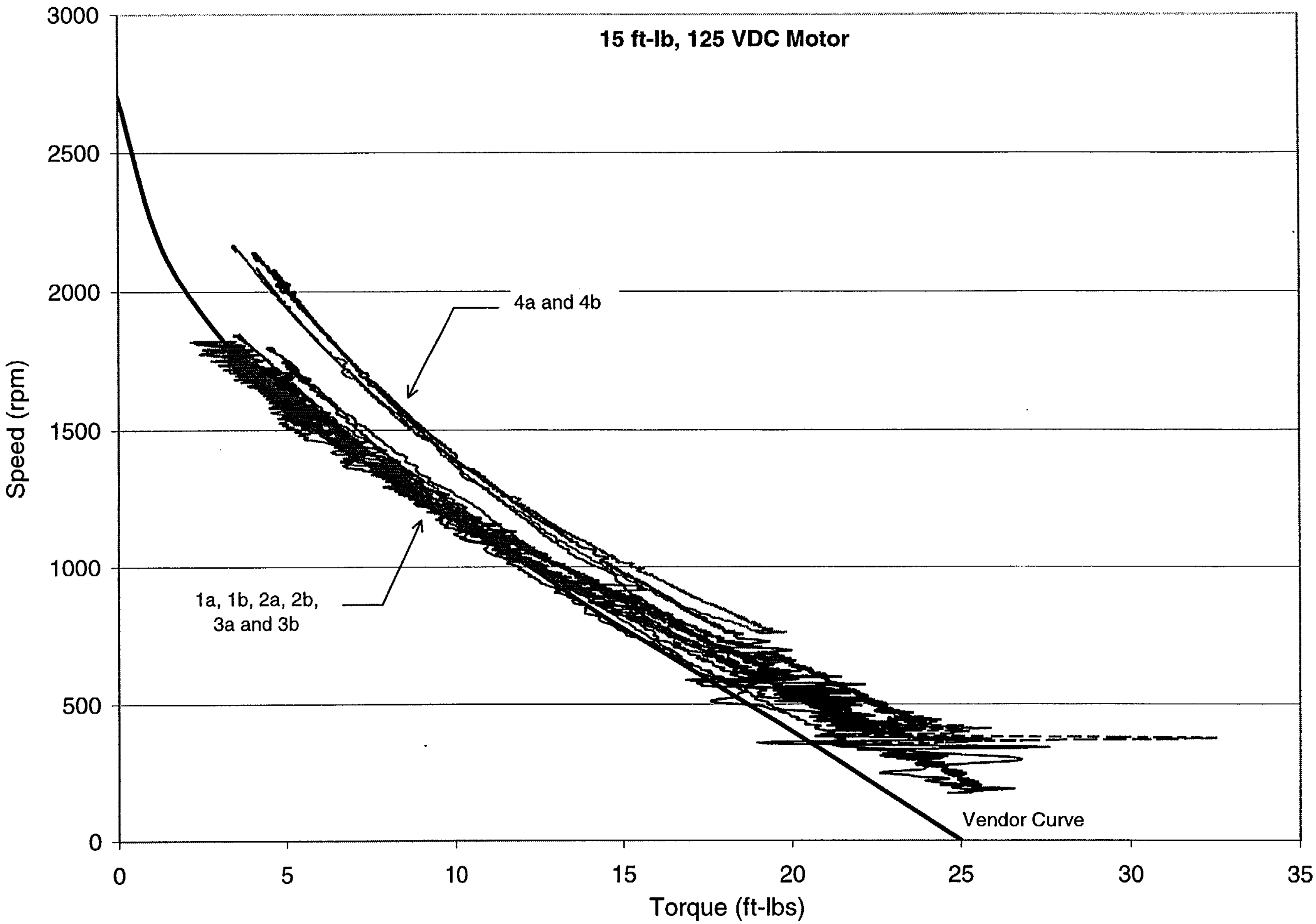

Figure 1. Comparison of Adjusted Speed versus Torque Curve to Vendor Motor Performance Curve for 15 ft-lb, 125 VDC Motor

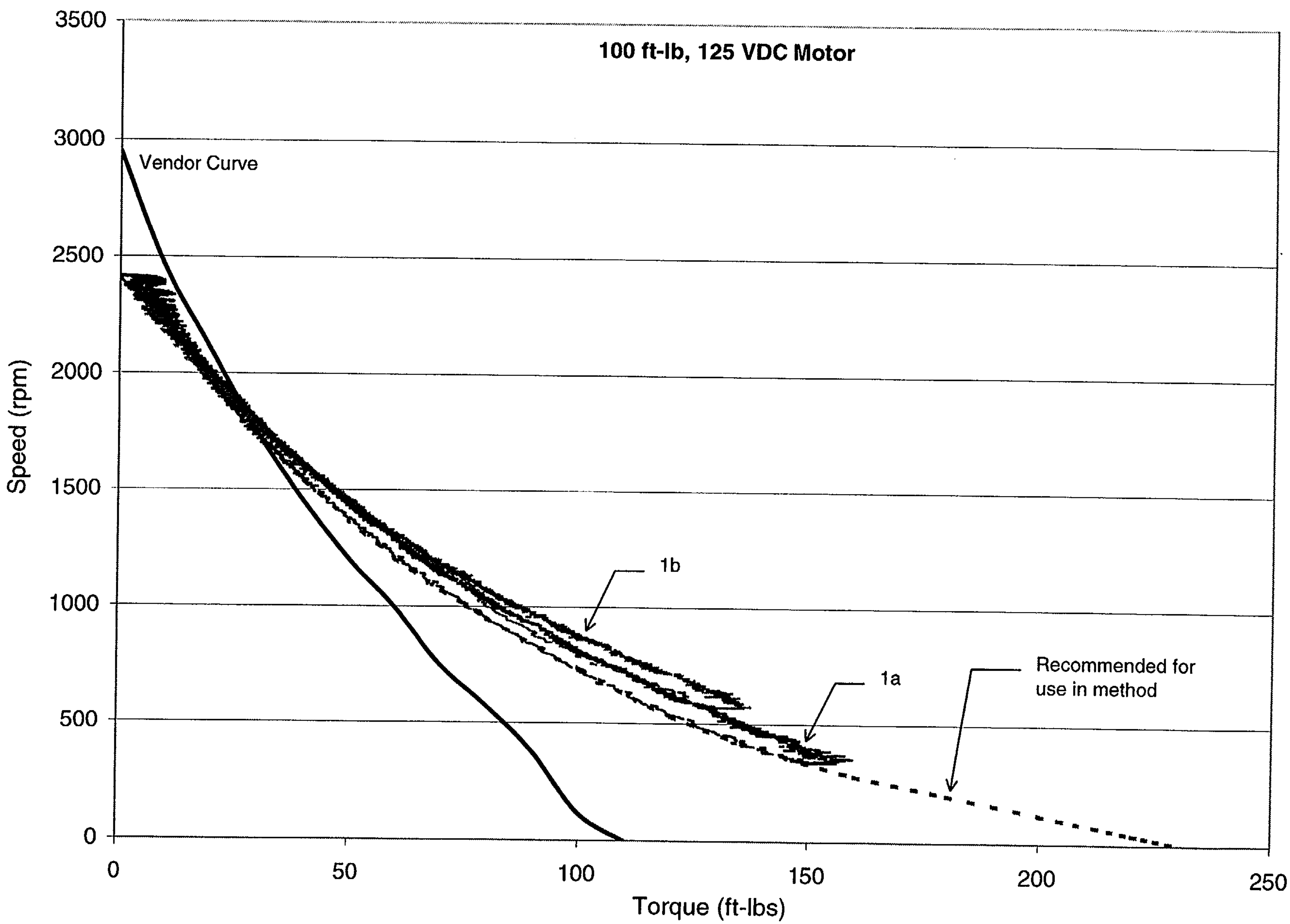

Figure 2. Comparison of Adjusted Speed versus Torque Curve to Vendor Motor Performance Curve for 100 ft-lb, 125 VDC Motor

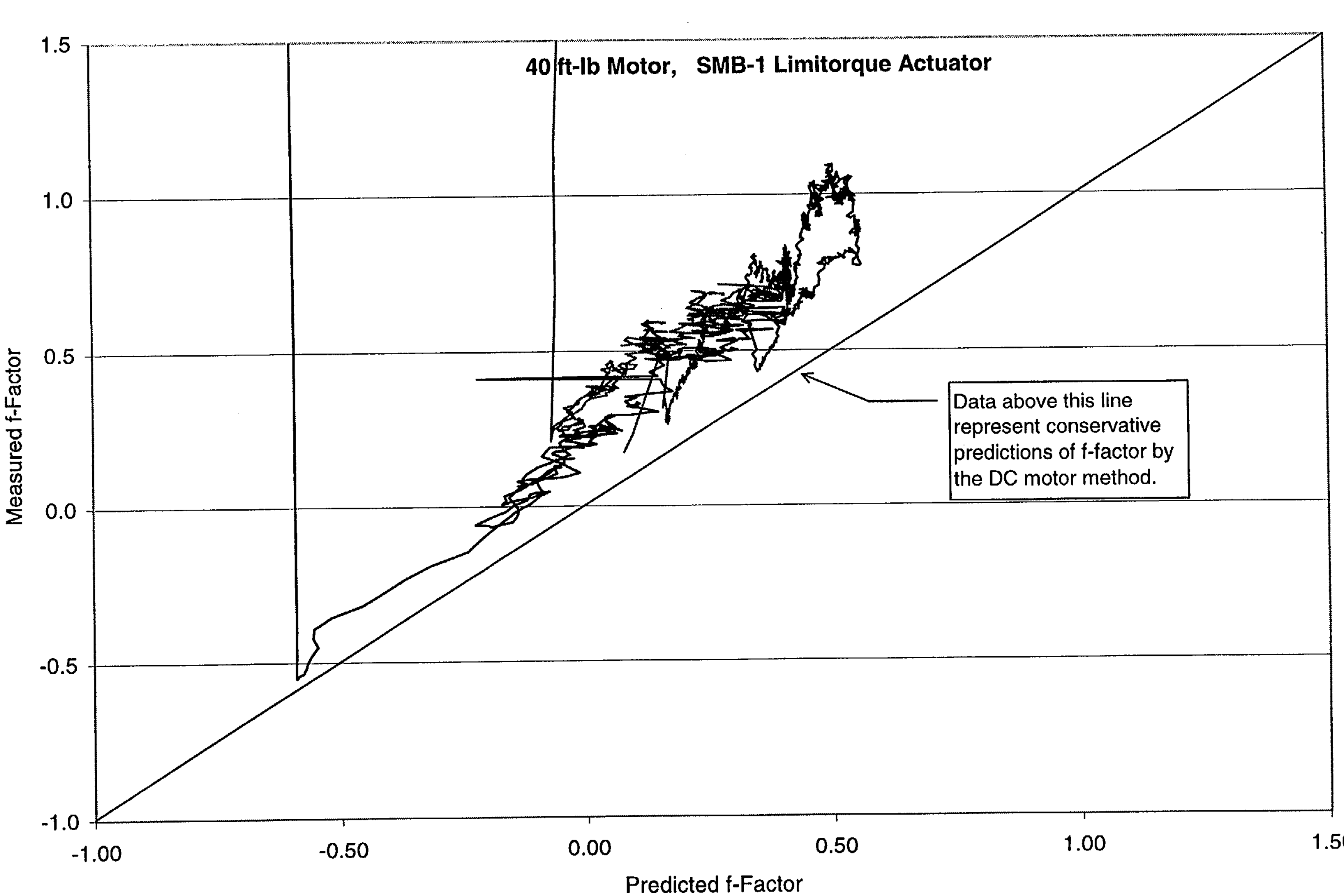

Figure 3. Measured versus Predicted f-factor for SMB–1 Actuator Tested by NRC/INEEL

Select Valve--> BWR Plant E, dynamic close

General Information	***Required?***	***Value***
Valve Type	yes	double disk
Flow type	yes	blowdown
Stroke direction	yes	close
Fluid (blowdown only)	yes	steam
Load profile method	yes	Use default
Valve and Actuator Information		
Stem diameter, D_{stem} (inches)	yes	2.250
Seat ring face ID, $D_{body-ID}$ (inches)	yes	9.581
Seat ring inner diameter, D_{sr} (inches)	no	0.000
Globe valve stroke length, D (inches)	no	0.000
Isolation-to-wedging travel, X (percent)	yes	10%
Packing load, F_{pack} (lbs)	yes	1,422
Required thrust (including water inertia), F_R (lbs)	yes	38,056
Required thrust due to water inertia, F_{WI} (lbs)	yes	0
Flow coefficient, C_V (gpm/psi$^{1/2}$)	no	0
Actuator overall ratio, OAR	yes	32.13
Motor gear set ratio, MGSR	yes	1.06
Actuator rated torque, τ_{rated}	yes	850
Stem factor, SF (ft-lbs/lb)	yes	0.0061
Voltage at MCC, V_{mcc} (volts)	yes	250.0
Cable resistance, R_{cable} (ohms)	yes	0.465
Thermal overload resistance, R_{tol} (ohms)	yes	0.018
Nominal voltage, V_{nom} (volts)	yes	250
Motor type	yes	40 ft-lb, 250 VDC
Valve stem lead, lead (inches)	yes	0.500
Pullout efficiency, η_p	yes	0.40
Run efficiency, η_r	yes	0.50
Nominal motor speed (rpm), ω_{nom}	optional	1,900
Design Basis Conditions		
Differential pressure, DP (psi)	no	0
Valve pressure at full open, P_B (psig)	yes	1025
Flow rate, Q_{max} (gpm)	no	0
Fluid density, ρ (lbs/ft^3)	no	0.000
Ambient temperature (C)	yes	32
Length of upstream piping, L_{up} (pipe diameters)	yes	10000
Calculated Values		
System resistance, K_{sys}	no	—
Fully open stem thrust, F_O (lbs)	yes	5,497
Nominal motor torque, τ_{nom} (ft-lbs)	yes	48.8
Maximum motor torque (ft-lbs)	yes	16.1
Maximum adjusted motor torque (ft-lbs)	yes	17.5
Minimum motor speed (rpm)	yes	1128
Minimum Instantaneous actuator capability (lbs)	yes	97,669
Minimum Instantaneous margin (lbs)	yes	68,157
Maximum torque switch setting	yes	89,691
Efficiency at minimum instantaneous capability	yes	0.393

Predicted Stroke Time (Seconds): 22.34

Figure 4. Input Sheet for DC Motor Performance Method Spreadsheet

Stroke	Position	Stem Thrust	Average Stem Thrust	Average Stem Torque	Worm Speed	Gearbox Efficiency	Average Motor Torque	Motor Current	Vmcc	Motor Voltage	Adjusted Motor Torque	Motor Speed	Stroke Time Increment	Time	Motor Temp	Instant Torque Cap	Instant Act Cap	Instant Margin
100%	9.58	5497												*0.00*	*32.0*			
90%	8.62	5647	5572	33.94		*0.400*	2.64	3.96	250.00	248	2.73	2336	1.58	1.58	32.0	607	99,597	94,024
80%	7.66	5985	5816	35.42	2476	0.393	2.81	4.21	250.00	248	2.91	2302	1.60	3.19	32.1	595	97,669	91,853
70%	6.71	6404	6194	37.72	2440	0.393	2.99	4.48	250.00	248	3.09	2266	1.63	4.82	32.1	596	97,807	91,613
60%	5.75	7089	6747	41.09	2402	0.395	3.24	4.86	250.00	248	3.36	2214	1.67	6.48	32.2	597	98,013	91,267
55%	5.27	7553	7321	44.59	2347	0.396	3.51	5.26	250.00	247	3.64	2161	0.85	7.34	32.2	598	98,225	90,903
50%	4.79	8031	7792	47.45	2290	0.397	3.72	5.58	250.00	247	3.87	2117	0.87	8.21	32.2	599	98,402	90,610
45%	4.31	8510	8270	50.37	2244	0.398	3.94	5.91	250.00	247	4.10	2081	0.89	9.10	32.3	600	98,580	90,310
40%	3.83	8989	8749	53.28	2206	0.399	4.16	6.20	250.00	247	4.33	2058	0.90	10.00	32.3	601	98,765	90,016
35%	3.35	9650	9319	56.76	2182	0.400	4.42	6.52	250.00	247	4.60	2031	0.91	10.91	32.4	603	98,989	89,670
30%	2.87	10429	10039	61.14	2153	0.402	4.74	6.92	250.00	247	4.94	1996	0.93	11.83	32.4	605	99,274	89,234
28%	2.68	10740	10585	64.46	2116	0.403	4.98	7.23	250.00	247	5.20	1970	0.37	12.21	32.5	606	99,481	88,897
26%	2.49	11588	11164	67.99	2089	0.404	5.24	7.55	250.00	246	5.47	1943	0.38	12.59	32.5	607	99,716	88,552
24%	2.30	12816	12202	74.31	2060	0.406	5.69	8.12	250.00	246	5.95	1895	0.39	12.98	32.5	610	100,143	87,941
22%	2.11	14044	13430	81.79	2008	0.409	6.23	8.78	250.00	246	6.52	1838	0.40	13.38	32.6	613	100,648	87,218
20%	1.92	15272	14658	89.27	1948	0.411	6.76	9.45	250.00	245	7.08	1781	0.41	13.79	32.6	615	101,061	86,403
18%	1.72	16500	15886	96.75	1888	0.413	7.29	10.11	250.00	245	7.65	1725	0.43	14.22	32.7	618	101,441	85,555
16%	1.53	17924	17212	104.82	1828	0.416	7.85	10.81	250.00	245	8.25	1672	0.44	14.66	32.8	620	101,839	84,627
14%	1.34	20454	19189	116.86	1772	0.419	8.68	11.68	250.00	244	9.14	1609	0.46	15.12	32.9	624	102,513	83,325
12%	1.15	22984	21719	132.27	1706	0.424	9.72	12.72	250.00	244	10.26	1531	0.48	15.60	33.0	629	103,363	81,644
10%	0.96	25798	24391	148.54	1623	0.428	10.80	13.80	250.00	243	11.44	1449	0.51	16.11	33.2	634	104,170	79,779
9%	0.86	27388	26593	161.95	1536	0.431	11.68	14.68	250.00	243	12.40	1387	0.27	16.38	33.3	638	104,715	78,122
8%	0.77	28597	27993	170.47	1470	0.433	12.24	15.21	250.00	243	13.02	1353	0.27	16.65	33.4	640	105,013	77,020
7%	0.67	29642	29120	177.34	1434	0.435	12.69	15.60	250.00	242	13.51	1326	0.28	16.93	33.6	641	105,285	76,165
6%	0.57	30917	30280	184.40	1405	0.437	13.14	16.00	250.00	242	14.01	1298	0.28	17.22	33.7	643	105,581	75,302
5%	0.48	32402	31659	192.81	1376	0.439	13.67	16.46	250.00	242	14.60	1266	0.29	17.51	33.9	645	105,947	74,288
4%	0.38	33745	33073	201.42	1342	0.441	14.21	16.94	250.00	242	15.20	1232	0.30	17.81	34.0	647	106,290	73,217
3%	0.29	34823	34284	208.79	1306	0.443	14.68	17.34	250.00	242	15.72	1204	0.31	18.12	34.2	649	106,535	72,252
2%	0.19	35900	35362	215.35	1276	0.444	15.09	17.70	250.00	241	16.18	1181	0.31	18.43	34.5	650	106,735	71,374
1%	0.10	36978	36439	221.92	1252	0.446	15.50	18.06	250.00	241	16.64	1162	0.32	18.75	34.7	651	106,947	70,508
0%	0.00	38056	37517	228.48	1232	0.447	15.90	18.41	250.00	241	17.10	1144	0.32	19.07	34.9	653	107,166	69,648
-0.1%	-0.01	38056	38056	231.76	1212	0.448	16.10	18.60	250.00	241	17.34	1134	0.03	19.10	34.9	653	107,177	69,121
-1%	-0.10	38056	38056	231.76	1202	0.448	16.11	18.61	250.00	241	17.35	1134	0.29	19.40	35.2	652	107,121	69,065
-2%	-0.19	38056	38056	231.76	1202	0.448	16.11	18.61	250.00	241	17.37	1133	0.33	19.72	35.4	652	107,030	68,974
-3%	-0.29	38056	38056	231.76	1201	0.448	16.11	18.61	250.00	241	17.38	1132	0.33	20.05	35.7	651	106,928	68,872
-4%	-0.38	38056	38056	231.76	1200	0.448	16.11	18.61	250.00	241	17.40	1132	0.33	20.37	35.9	651	106,825	68,769
-5%	-0.48	38056	38056	231.76	1200	0.448	16.11	18.61	250.00	241	17.42	1131	0.33	20.70	36.2	650	106,723	68,667
-6%	-0.57	38056	38056	231.76	1199	0.448	16.11	18.61	250.00	241	17.43	1130	0.33	21.03	36.4	649	106,621	68,565
-7%	-0.67	38056	38056	231.76	1198	0.448	16.11	18.61	250.00	241	17.45	1130	0.33	21.35	36.7	649	106,519	68,463
-8%	-0.77	38056	38056	231.76	1198	0.448	16.11	18.61	250.00	241	17.47	1129	0.33	21.68	36.9	648	106,417	68,361
-9%	-0.86	38056	38056	231.76	1197	0.448	16.11	18.61	250.00	241	17.48	1129	0.33	22.01	37.2	647	106,315	68,259
-10%	-0.96	38056	38056	231.76	1196	0.448	16.11	18.61	250.00	241	17.50	1128	0.33	22.34	37.4	647	106,213	68,157
TST	---	---	*89691*	*546.22*	*195*	*0.396*	*42.90*	41.17	250.00	230	*48.84*	184	---	---	---	*546*	89,691	0

Figure 5. Results Sheet for DC Motor Performance Method Spreadsheet

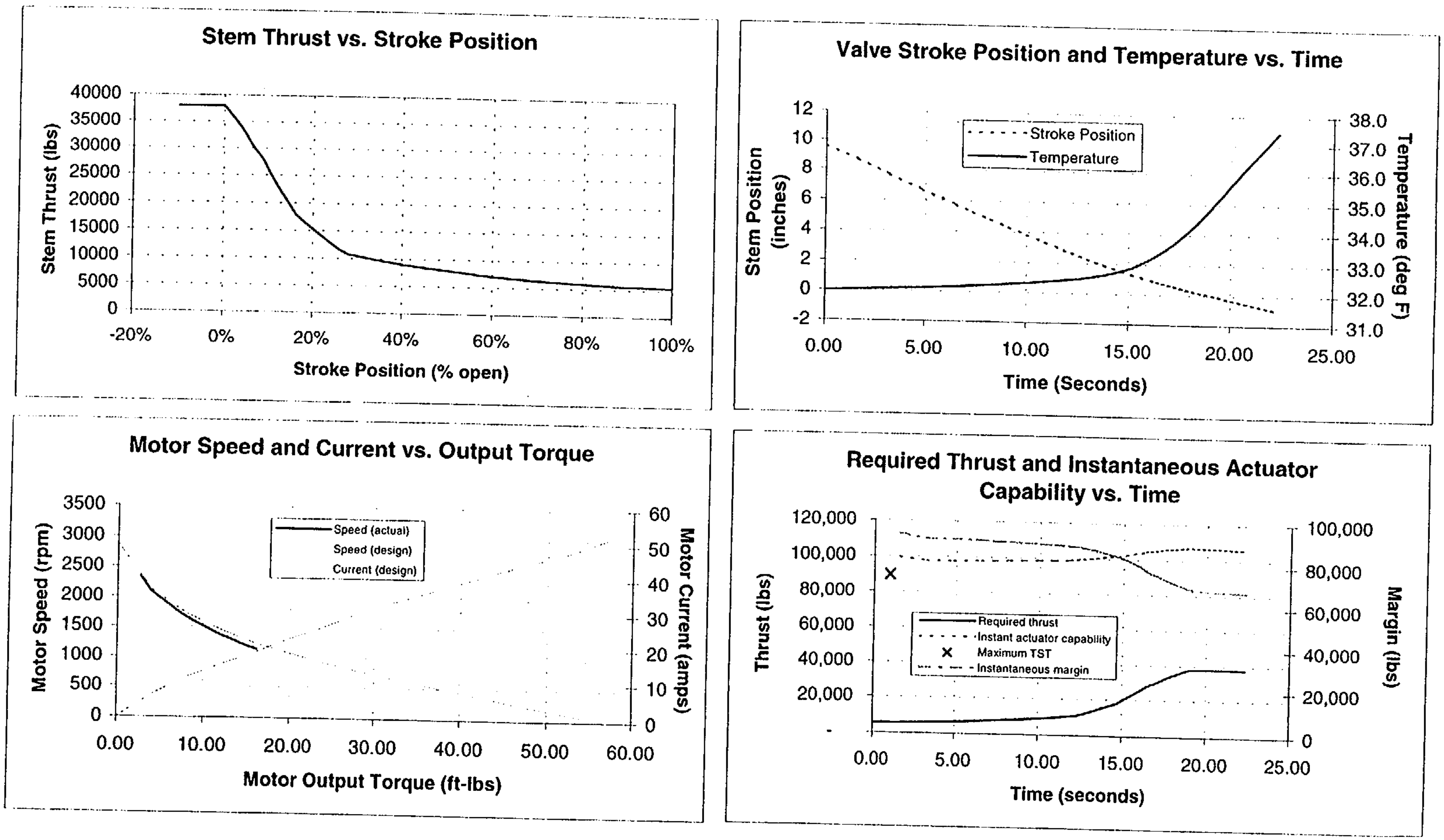

Figure 6. Output Plots for DC Motor Performance Method Spreadsheet

Selection of Greases for Motor-Operated Valve Stem/Stem Nut Lubrication

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Abstract

In Motor-Operated Valves (MOVs), the choice of a high-performance grease for stem/stem nut lubrication is required to guarantee long-term operability. Valve stem regreasing intervals and/or actuator limit switch resetting intervals of less than two years are no longer acceptable to most stations because of the large number of valves in service.

A joint MOV test program was launched in 1995 by Atomic Energy of Canada Limited (AECL), Electricité de France (EDF) and Ontario Power Generation (OPG) to select the best grease candidates available and address issues relative to valve reliability, increased maintenance costs, and environmental protection. The objective of this research was to identify candidates for 4 to 5 years in service without regreasing.

A thermal aging procedure was developed by OPG to simulate a five-year exposure to in-service temperature of a stem/stem nut arrangement. The grease candidates were initially screened using data from pin-on-disk wear tests and/or results of thermal aging tests (grease consistency variations, acidity build-up...). Samples thermally aged for the equivalent of five years in service were then used for mechanical tests in a full-scale MOV test rig developed by AECL and EDF at the Chalk River Laboratories (CRL). The assessment of the stem/stem nut lubrication condition was mainly based on stem/stem nut friction coefficient measurements (value and stability of this value with valve stroke number).

In this paper, the effect of grease thermal and mechanical aging on stem/stem nut lubrication is assessed, along with the effect of high temperature and high irradiation levels (accident conditions). The MOV test results are compared according to the type of base oil and thickener used in the grease. The effect of stem material and stem geometry on stem/stem nut lubrication is investigated. Results obtained for grease mixtures are also reported. Finally, grease specifications recommended to maintain adequate long-term MOV stem/stem nut lubrication are provided.

Introduction

Greases are used at the valve stem/stem nut location to protect both threaded parts in contact and to ensure that torque developed within the motor-operator is efficiently converted into stem thrust to operate the valve. This efficiency can be greatly affected by the type of grease used and by the effect of thermal aging, mechanical aging and other environmental parameters, such as temperature or radiation, on grease condition.

One of the many challenges facing nuclear power plant operators is the improvement of MOV lubrication conditions to maintain long-term component operability and reduce costs associated with MOV maintenance programs. Valve stem regreasing intervals or torque limit switch resetting intervals of less than one or two years are no longer acceptable to most stations, considering the large number of MOVs in service. The use of very stable, low-friction greases for stem/stem nut lubrication is the key to long-term valve operability and limited maintenance costs.

At the request of several nuclear power generating stations in Canada, an MOV grease test program was started at AECL, Chalk River, and at OPG, Toronto, in 1995. A collaboration with EDF was launched in 1996 and a full-scale MOV test rig dedicated to MOV grease selection was developed. A specific procedure including the testing of grease thermal and mechanical aging properties and post-test evaluation of grease characteristics was put in place. More than thirty grease candidates were tested and ranked in accordance with criteria and key parameters as defined through consultations with station valve and lubricant specialists.

This paper summarizes the results obtained for the various types of grease tested. The test results are compared according to grease composition (type of base oil, type of thickener, and additives used). The effect of thermal and mechanical aging on grease performance is studied along with the effect of high temperature and high irradiation levels. Based on this study, the grease characteristics required for adequate long-term stem/stem nut lubrication of motor-operated valves are specified.

Background

Stem/stem nut lubrication characteristics

In MOVs, valve stroking is generated by an electrical actuator that provides a torque for stem nut rotation. The rotation of the stem nut induces the vertical translation of a threaded stem/valve gate arrangement that results in flow isolation. The efficiency of the torque/thrust conversion is a function of stem/stem nut geometry and of friction properties at the stem/stem nut location (Equation 1).

$$F = \frac{Torque}{Thrust} = R\frac{\tan(\alpha) + \frac{\mu}{\cos(\beta)}}{1 - \tan(\alpha)\frac{\mu}{\cos(\beta)}} \qquad (1)$$

where

F = Stem Factor (in meters),
R = Average Thread Radius (in meters)= 0.5 x (Thread Diameter (in meters) − 0.5 x pitch),
β = 14.5° (ACME thread),
$\tan(\alpha)$ = lead/(2πR), and
lead = pitch x number of start threads (in meters).

For given loading conditions, the stem/stem nut friction coefficient depends, to some extent, upon the characteristics of bearing surfaces but is in fact mostly related to the properties of the lubricant used at the stem/stem nut location. Therefore, changes in lubricant condition as a result of mechanical aging or as a result of the effect of environmental parameters may have a significant effect on the thrust delivered for a given torque and have serious implications. On one hand, increasing stem/stem nut friction coefficients can result in stem thrusts generated at torque switch trip that are not sufficient to properly close the valve. A significant decrease of friction coefficient can, on the other hand, cause overloading of the valve. Also, design guidelines for proper actuator operation sometimes require that the friction coefficient remains below a pre-set value (i.e., 0.15 for EDF power plants).

Greases are preferred lubricants for open systems such as the stem/nut arrangement in MOVs, where the lubricant must maintain its original position in the mechanism. A grease consists of a base oil (80 to 95% of grease weight), a thickener, and additives. Grease lubricity is usually determined by base oil properties, especially base oil viscosity. Grease consistency and rheological properties are determined by the type of thickener used. Anti-oxidation, anti-wear, extreme pressure, anti-corrosion additives, solid lubricants, or soft metal particles are added to enhance grease performance.

Requirements from station end-users

If a grease used for stem/stem nut lubrication is not the best from an operability or from an environmental qualification point-of-view, the benefit of using a new grease must be clearly demonstrated prior to stem/stem nut grease replacement. The grease characteristics must satisfy the requirements associated with valve mechanical performance, valve environment, and economic benefit.

There are many requirements relative to valve mechanical performance. The stem/stem nut arrangement in MOVs is typical of a "high load/low speed" lubricated mechanism. Therefore, in order to maintain a film of lubricant between the loaded bearing surfaces, the use of Extreme Pressure (EP) greases (or products with similar characteristics) is required. As mentioned above, based on design guidelines, the friction coefficient must sometimes be kept below pre-set values (for EDF power plants). The emphasis is also put on friction coefficient stability to avoid frequent resetting of torque switch trip values, the possibility of improper closures (increasing friction coefficient), or valve overloading (decreasing friction coefficient). The overall mechanical performance of the stem/stem nut arrangement must remain quasi-unchanged throughout the regreasing interval chosen, whether the valve is actuated several times a day or only once a year. A small difference in thrusts developed at torque switch trip under static and dynamic loading is also preferred (i.e., low grease sensitivity to Rate-of-Loading (ROL) effects).

The impact of the nuclear reactor environment on grease performance must also be considered when selecting potential grease candidates. The grease must have antioxidation properties because of the relatively high in-service temperatures (45 to 60°C in many parts of the reactor building). The concentration of additives found in many EP or anti-wear greases (such as chlorides or sulfur) is often above the limits set by the industry regulators (usually 400 or as low as 200 ppm allowed). This drastically restricts the number of EP greases to be considered. The greases must also withstand in-service radiation levels (up to 10 Mrad over 5 years) and resist conditions resulting from Loss Of Coolant Accident (LOCA) or Major Steam Line Break (MSLB). Naturally, high-temperature greases would be preferred candidates to guarantee valve operability at peak temperatures of 165°C and accident steady temperatures of 110–120°C. The resistance of the grease to high irradiation levels must also be demonstrated (if adding in-service, LOCA, and post-LOCA irradiation). Resistance to water or steam washout and to chemical attack would also be an asset in the event of an accident.

A major issue is also the extensive cost associated with the maintenance and monitoring of the large population of MOVs in service at each plant. Valve stem regreasing intervals or torque limit switch resetting intervals of less than one or two years are no longer acceptable. Access to many valves is restricted and short regreasing intervals often mean increased radiological doses to workers and increased amounts of waste for disposal. Therefore, a reasonably priced, high-performance grease, suitable for 4 to 5 year regreasing intervals, is required.

Objectives

The stem/stem nut lubrication is a specific industrial application and there are many requirements to be satisfied in order to guarantee that the lubricant can safely be used in MOVs. For commercial reasons, information such as the nature of the additives, the fabrication process, or the thickener/oil ratio are rarely disclosed to the end users. Therefore, the selection of a grease candidate solely based on grease properties available from grease manufacturers is not recommended.

Several standard tests can be used to study grease characteristics (oxidation resistance, wear resistance, etc.) but they are not representative of MOV stem/stem nut lubrication conditions and would not provide the information relative to friction coefficient that is absolutely needed in this case.

Therefore, the main objective was to develop a test rig that closely reproduced the MOV operating conditions (both in-service and accident conditions). This would allow the best grease candidates to be selected according to the requirements outlined in the previous section.

However, accelerated mechanical testing in an MOV test rig cannot take into account the effect of long-term exposure to in-service reactor conditions. Therefore, another objective was to develop a test procedure for representative thermal aging and irradiation of the grease candidates prior to MOV testing.

Description of Test Facility and Grease Testing Methodology

Thirty-six grease candidates were initially considered by OPG (24) and EDF (12) for use at the stem/stem nut location. The initial list consisted of products already used at the various sites, products recommended by grease manufacturers, or products selected based on some of their basic properties (i.e., EP properties or high-temperature resistance).

Pin-on disk tests

Pin-on-disk testing was used by OPG to select 12 greases from the initial list. The pin-on-disk test is a standard test (ASTM G–99) that consisted of rotating a disk at constant speed against a pre-loaded, stationary pin for 60 minutes. The grease to be tested is applied onto the disk prior to the test. The Herguth Wear Rating (HWR) formula (Equation 2) is used to rank the grease candidates. The lower the rating, the better the grease from a lubrication and wear point-of-view.

$$HWR = (100f) + (1000V) + (4.7DR) \qquad (2)$$

where

f = friction coefficient,
V = pin wear volume, and
DR = visual disk rating from 1 to 10

This type of test is valuable for preliminary screening of a large number of grease candidates, but it may not be the most adequate selection tool. Some of the EDF greases with high HWRs appeared to be very successful candidates during the MOV test campaign while greases with very low HWRs did not necessarily do well in the MOV test rig. There are significant differences between a pin/disk contact and a stem/stem nut contact. Also, the HWR does not account for the stability of measured parameters with time. Therefore, pin-on-disk tests should not be used for MOV stem/stem nut grease qualification.

Thermal aging and irradiation of selected grease candidates

The objective was to thermally age and irradiate all grease candidates prior to MOV testing to simulate long-term exposure to in-service temperature and radiation (5 and even up to 8 years at 45–60°C and 100 kGy over 5 years).

Accelerated thermal aging can be obtained at elevated temperature in a relatively short period. The limiting factor is the sensitivity of additives to high temperature. An Arrhenius model of thermal aging (Equation 3) is generally applicable to greases [1]:

$$t = t_0 \exp\left(\frac{Q}{RT}\right) \qquad (3)$$

where

t = Exposure Duration,
t_0 = Constant,
Q = Activation Energy,
R = 8.314 kJ.K^{-1}.mol^{-1} (constant), and
T = Temperature in K.

Nowadays, EP additives used in premium extreme pressure lubricants can easily survive test temperatures of up to 135°C without premature decomposition. Thermal aging of the various grease candidates was performed in an oven either at 120°C for 8 weeks or at 130°C for 4 weeks. Similar aging temperatures and durations have been used by others [2, 3]. The 130°C aging condition was used for rapid product screening while the 120°C condition was used for aging "short listed" products prior to MOV mechanical testing. According to Equation 3, this is equivalent to five-years of aging at 77°C.

For thermal aging in the oven, the characteristics of the stem/stem nut arrangement (vertical open system using a thin layer of grease) must be taken into account. Therefore, all grease samples were applied to vertical steel panels. The grease on each panel was 2 mm thick, and was redistributed on the test panel at the equivalent of 7.5 months of ambient aging. This redistribution served two purposes: (1) it simulated occasional valve stroking, and (2) it ensured that all portions of the grease sample were regularly exposed to air. Good air exposure is required so that oxygen does not become a limiting reactant in grease oxidation reactions introducing non-Arrhenius effects.

The end-of-life criteria were chosen to be most relevant to the type of transformation that would affect grease performance at the stem/stem nut location. The end-of-life criteria adopted for grease aging resistance were: (1) a two grade change in grease consistency (a two grade loss would be a sign that the grease could run out of the stem/stem nut arrangement, while a two grade gain could result in nut jamming); (2) an 80% loss in acid buffering capacity or an absolute Total Acid Number (TAN) limit of 3 (which would indicate excessive grease oxidation); and (3) extreme slumping from vertical panels or other gross property changes during the aging process. Grease weight loss was also monitored but was not used as an end-of-life criterion.

At the end of the five-year aging period, all vertical panels were gamma (Cobalt 60) irradiated at room temperature to a dose of 100 kGy (10 Mrad) at a dose rate of approximately 1–3 kGy/h (0.1–0.3 Mrad/h). This exceeds a usual five year in-service dose and is intended to simulate a worst case service condition.

Mechanical testing in MOV test rig

The objective was to compare the grease performances using a test stand that is representative of Motor-Operated Valves used at the stations. To this end, a valve body and a valve actuator were purchased and assembled. The MOV grease test rig was then built around these two major components, and includes the following items:

- mechanical assembly for supporting the valve body
- Velan ANSI 150 valve body (8" diameter)
- Limitorque SMB0 actuator
- valve packing for static loading
- bronze nut
- stainless steel stem
- mechanical assembly for dynamic loading
- MOVATS cell for on-line measurement of torque, thrust, and friction coefficient
- sensor for on-line measurement of stem displacement
- unit for motor current measurement
- counter and relays for control-command
- PC and SNAP MASTER software for control-command and signal acquisition.

A schematic of the CRL grease test stand is shown in Figure 1.

A mechanical assembly was developed to generate side forces and simulate differential pressure effects from the process fluid on the valve gate during valve operation. These side forces result in stem dynamic loading through friction forces generated at the gate/valve body interface.

The core of the dynamic loading assembly consists of a main shaft and a set of Belleville washers of two different sizes. The shaft is connected to the lower half of the valve gate. It rotates around two axes and slides in a fixed bearing housing in order to accommodate the gate displacement. The shaft displacement results in Belleville spring compression and sideloading of the valve gate. The spring set is assembled to generate 53 kN in the horizontal direction in the closed position (equivalent ΔP of 1.13 MPa). This results in 20 kN of additional "dynamic" thrust on the stem. The washer arrangement is designed to generate low dynamic loading of the stem during most of the valve closing sequence and highly increased loads at the end of the closing sequence.

Figures 2 and 3 compare typical MOV closing sequences [4] and CRL MOV test rig closing sequences. Figures 4 and 5 compare typical MOV opening sequences [4] and CRL MOV test rig opening sequences. These graphs show that the mechanical assembly developed at CRL provides a good reproduction of a typical MOV dynamic load-time history for both opening and closing sequences.

The differential pressure simulation system offers a cost effective alternative to valve test rigs using flow generated by a circulation loop. Compared to "static" systems, it also has many advantages in view of MOV grease selection:

– the greases will be stressed using representative stem thrusts and side forces,

– the system allows the comparison of thrusts near Torque Switch Trip (TST) for dynamic strokes (lower loading rates, i.e., increasing stem thrust throughout closure cycle) and static strokes (higher loading rates, i.e., increasing stem thrust only at end of closure cycle). This will be useful to assess the grease sensitivity to rate-of-loading (ROL) effects,

– unlike vertical springs sometimes used to reproduce dynamic loading, the CRL mechanical assembly generates a sudden thrust variation at valve unseating which is representative of the stem compression/traction transition phase. This sudden thrust variation at the beginning of the opening stroke allows a representative brisk redistribution of the grease on the opposite side of the thread. This effect would not be simulated using a vertical spring system (i.e., the stem would remain loaded on the same side of the thread).

The preparation, performance, and analysis of each grease selection test consisted of the following steps:

1) A new bronze nut was used for each test. The bronze nuts were manufactured at CRL to ensure that consistent thread geometry was used. Both nut and stem were thoroughly cleaned (using chloride and sulfur free products) and were greased (25 ml of grease evenly spread on both parts) using the product to be tested

2) Stem and nut were installed in the valve, along with a new packing.

3) Valve yoke and actuator were assembled and 10 static closing and opening sequences for packing positioning and set up of packing load were performed.

4) The mechanical assembly to simulate differential pressure effects was attached to the valve.

5) A series of 240 dynamic opening and closing sequences were performed to assess the effect of valve stroking (i.e., mechanical aging) on stem/stem nut lubrication. A rest period of 3 min was allowed between each stroke.

6) After removal of the sideloading system, five static closing and opening sequences were performed.

7) The valve body, actuator, stem and nut were disassembled and grease samples were taken for post-test analysis.

8) The data were analyzed and the grease performance was assessed.

For each grease, several tests of this type were performed using a new formulation and thermally aged formulations (5 year and 8 year equivalent).

Grease rating criteria were selected in consultation with station valve and lubricant specialists. For the rating process, weighting factors were assigned based on the impact of each parameter on valve operability. From a mechanical point-of-view, a high-performance grease will show:

- low friction coefficients (especially for high thrusts reached while P is simulated or during valve seating)
- stable friction coefficient with increasing stem load and stroke number
- stable thrust at TST with increasing stroke number
- low sensitivity to thermal aging effects.

Curves in Figure 6 show that the MOV test rig results will discriminate between low-performance and high-performance greases. For example, the friction coefficient for Grease A is very unstable with increasing stroke number, while both tests with Grease B show stable results. Grease C exhibits the best characteristics with a low and relatively stable friction coefficient. Test results for a given grease candidate were repeatable within 5% (see Test #1 and Test #2 for Grease B). The accuracy of friction coefficients measured while P is simulated and during valve seating was 1.2 and 0.9%, respectively. The accuracy of thrust measured at 185 Nm was 0.8%.

The effect of the rest period duration between strokes was studied. Similar friction coefficients were measured after rest periods of 3, 20, 40, 80, or 1000 minutes as shown in Figure 7. However, rest periods as short as 1 minute were shown to affect friction coefficient stability. Therefore, a 3-minute rest period was chosen as the best compromise in order to generate relevant test results within reasonable time periods.

Post-test analysis of grease samples

Grease samples were collected from three parts of the stem (top, middle, and bottom). The amount of grease collected was usually very small, but sufficient for post-test analysis. Post-test analysis consisted of microscopic evaluation of wear debris, a deleterious particles test, trace element analysis, and penetration measurements using the Dynamic Mechanical Analyzer (DMA), which was developed to determine penetration on samples of less than 1 g.

The ASTM D1404 "Deleterious Particles" test was performed on the "post-MOV test" grease samples to determine if gross particulate was removed from the nut or stem during the MOV test. A small grease sample was loaded to 1.38 MPa (200 psi) between two plastic plates. The plates were then rotated relative to each other through one 30° arc. The plastic plates were then inspected for scratches. A finding of more than 40 scratches identified a "high" scratch risk, a finding of 10-40 scratches was classed as a "medium" scratch risk, and a finding of less than 10 scratches as a low risk.

Trace element analyses by atomic emission spectroscopy were carried out on the "post-MOV test" grease samples to determine if micron-sized particles or dissolved wear metals were present.

The DMA technique is a new method to assess the rheological condition of very small grease samples. A 0.5 g grease sample is held in a thin layer between horizontal and parallel metal plates. The upper plate is the driving oscillator and the lower plate picks up the response signals.

The measured force is used to calculate the "complex modulus" (G*), which is a vector combination of two other properties, the "viscous modulus" and the "elastic modulus." These are the "bouncing" and "damping" characteristics of the sample. The complex modulus can be related empirically to penetration values. Therefore, penetration values for small samples collected from the MOV stem could be derived using this technique.

Thermal Aging Results

Greases aged at 40°C during 1.4 years and aged the equivalent of 1.25 years at 120°C showed similar consistency after the test. Therefore, a thermal aging duration-temperature equivalence based on the Arrhenius model proved to be suitable for accelerating the thermal aging process.

Only nine out of twenty-four greases initially selected by OPG and EDF met all criteria for resistance to thermal aging during 5 years at 45 to 60°C. Some greases slumped from the vertical panels and many showed unacceptable changes in consistency and/or buffering capacity. However, some of the best greases withstood the equivalent of 8 years at in-service temperatures.

Most high-temperature greases (i.e., maximum service temperature recommended by grease manufacturers is above 150°C) showed good resistance to thermal aging, but not all.

MOV Test Rig Results

Effect of the type of grease (base oil, thickener, additives combination) on MOV lubrication characteristics

Friction coefficients measured during the application of side forces (simulation of P) and during valve seating and thrusts measured near TST are shown in Figure 8 for 9 different families of greases according to their composition (various base oil-thickener-additives combination). The range of friction coefficients obtained was relatively wide, from 0.10–0.11 up to 0.17. For most greases tested, the average friction coefficient was approximately 0.14.

One must be cautious when comparing results according to the grease type because the amount of information released by grease manufacturers is usually minimal and does not include the nature of additives or the proportions of base oil and thickener in the grease. However, some interesting trends were observed. The lowest friction coefficients were found for greases using a synthetic PAO base oil and a clay thickener (#1 and #2 in Figure 8). With mineral oils, the friction coefficients were usually around or above 0.14, except for the value of 0.13 obtained for the mineral-clay combination (#5 in Figure 8). Therefore, using clay as a thickener seems to decrease friction at the stem/stem nut location. The comparison of grease types 6a and 6b (Figure 8) shows that the presence of EP additives can also decrease friction. However, EP additives are not always required to obtain low friction coefficients at the stem/stem nut location (grease types #1, 2, and 5 do not contain EP additives yet exhibit the lowest friction coefficients).

For EDF applications, the actuator tripping point set up is based on equations that use a friction coefficient of 0.15. In that case, lower friction coefficients such as 0.10 obtained for the synthetic PAO–clay combination are preferred since the margin from the 0.15 design value becomes larger. For OPG applications, the actuator tripping point set up is based on the thrust value obtained for a given torque.

Therefore, the stem thrust developed at Torque Switch Trip and the stability of this thrust value with stroke number and time are key parameters.

Effect of mechanical aging on MOV lubrication characteristics

The variation of TST thrust and friction coefficient over 250 MOV opening and closure cycles is shown in Figure 9. For most greases, this variation does not exceed 5–10%. The type of grease recommended for good stability with increasing stroke number uses the mineral base oil-lithium complex thickener combination (#6a in Figure 9). One of the greases that contained a calcium complex thickener (#8 in Figure 9) proved very sensitive to mechanical aging (variations of parameters of up to 30–35% after 250 cycles).

Effect of thermal aging on MOV lubrication characteristics

Five-year equivalent thermally aged greases were tested in the MOV rig to assess the effect of thermal aging on stem/stem nut lubrication. Thermal aging resulted in either slight softening or hardening of the greases, depending on their composition. However, these changes in consistency could not be related to friction coefficient variations. Results for friction coefficients are shown in Figure 10. For most 5-year aged formulations, the friction coefficients are very similar to those obtained when using the unaged formulation. The difference between the two sets of results usually does not exceed 10%. This is not entirely surprising because most greases selected for MOV testing were amongst the best candidates from the thermal aging screening process described earlier in this paper.

Most 5-year aged formulations showed fair resistance to mechanical aging. The variation of friction coefficients and thrusts with increasing stroke number ranged from 10 to 20% over 250 cycles for most greases tested (Figure 11). The friction coefficients obtained for the 5-year aged formulations were usually less stable than those obtained for the unaged formulations. Several greases that used lithium complex as a thickener showed higher friction coefficient and thrust stability. The grease that contained a calcium complex or a polyurea thickener showed the most friction coefficient and thrust variations. The PAO + clay greases showed friction coefficient variations over 250 cycles in the order of 20% but the average friction coefficient for the 5 year-aged formulation was still low at 0.11–0.12.

The best candidates were aged for the equivalent of 8 years in service. The variation of friction coefficients and thrust between 5-year aged and 8-year aged formulations is shown in Figure 12 (comparison over 50 cycles). The best candidate showed only 2% variation (mineral oil-lithium complex-EP additives combination). However, some 8-year aged formulations showed significantly decreased performance from the 5-year aged case (average friction coefficients that are 25 to 30% higher, i.e., decreased resistance to mechanical aging).

Results for grease mixtures

Mixtures should be tested because greases are sometimes replaced by a new product without full cleaning of the stem prior to regreasing. MOV tests were performed using mixtures of some of the best greases previously tested. For these mixtures, the friction coefficient was usually between the two average friction coefficients initially derived for each pure product. This trend is shown in Figure 13. However, testing was performed for a very limited number of mixtures and, more importantly, the thermal aging behavior of these mixtures was not studied. Therefore, it is advisable to fully clean the stem when replacing one grease by another.

Effect of stem material and geometry

Stem material was 410 stainless steel for testing of the OPG greases and 17–4–PH for testing of the EDF greases. When testing similar greases with the two types of stem, it was shown that stem material had no significant effect on the stem/stem nut lubrication condition. On average, using these two different stems, the difference between friction coefficients obtained for the same grease was only about 5%. That is within the repeatability of test results for a given configuration.

For one of the greases, additional tests were performed to study the effect of stem/stem geometry on lubrication characteristics. For a given thrust, the results showed higher friction coefficient for increasing contact pressure at the thread location (i.e., lower stem diameter or lower pitch).

Results of Post-MOV Test Analysis

The deleterious particles test showed that none of the greases tested had scratch counts of 10 or greater. Therefore, no significant large particles were detected in any of the greases tested in the MOV test rig. No significant wear damage occurred during these MOV tests. This is consistent with visual inspections of both stem and nut after each test. Therefore, all greases tested in the MOV rig offered adequate wear protection to both stem and nut contacting surfaces. As a result, no microscope (optical or SEM) inspection of the grease samples was performed.

The results of trace element analysis show that all of the greases picked up zinc, copper, aluminium and iron as a result of MOV testing. This is attributed to slight wear of the stem nut, which is made of C86300 bronze (60% copper, 22–28% zinc, 5–7% aluminium, 2–4% iron and 2% manganese). For most grease candidates, trace element results are within the same order of magnitude after testing in the MOV test rig. Therefore small particle pick-up levels cannot be strongly related to grease performance at the stem/stem nut location.

DMA results were shown to be repeatable for grease samples collected following two MOV tests using the same grease. For most greases, the post-test consistency was usually lower than the consistency measured before the MOV test, but this could not be related to grease performance in the MOV test rig. However, for some of the greases tested, there seemed to be a correlation between TST thrust instability measured during the 500-stroke MOV test and significant grease hardening measured with the DMA technique.

Results for Accident Conditions

Effect of high temperature on MOV lubrication characteristics

The OPG nuclear stations requested MOV testing of the best 5-year aged grease candidates under Major Steam Line Break (MSLB) temperature conditions. A temperature profile to cover all OPG MSLB profiles was provided. It consists of increasing the stem/stem nut temperature from 20 to 165°C, maintaining 165°C for 0.5 h, then cooling to 110°C and maintaining that temperature for 7 days before cooling to room temperature. The effect of steam jet and grease washout on stem/stem nut lubrication was not assessed in this study.

A cartridge heater was inserted in the stem to provide heat at the stem/stem nut location. The advantage to using a localized heat source is that the MOVATS cell can be maintained around room-temperature while testing at 165°C at the stem/stem nut location. However, maintaining 165°C at the nut location resulted in extremely high temperatures (>300°C) on the protruding parts of the stem (above and below the nut) because of the greater heat dissipation at the stem/stem nut location. This is not representative of MSLB temperature conditions for which the protruding parts of the stem will only be exposed to a temperature of 165°C. Therefore, the temperature on the lubricated protruding part of the stem (above the nut in opened position, below the nut in closed position) was controlled at 165°C.

The effect of this temperature profile on friction coefficients and TST thrust is shown in Figure 14 for various 5-year aged high-temperature greases. The sensitivity to the temperature profile remained minimal for most 5-year aged greases. Variations of parameters were below 5% for the best case and 10% for most candidates. However there was up to 30% variation in friction coefficient for one grease even though it was a high-temperature grease.

Similar tests were performed for the EDF greases but with a somewhat different high-temperature profile: the greases were slowly heated up to 120°C, then stroked at 120°C before cooling down to room temperature. Increasing temperature, as well as stroking at high temperature, resulted in increased friction coefficients (> 0.15 for most greases tested). However, cooling down to room temperature usually resulted in the return to initial friction coefficient values. For most greases, the variation of friction coefficient was 5–10%.

Effect of high irradiation levels on MOV lubrication characteristics

The best five-year aged greases (best performers in the MOV test rig) were irradiated to a dose of 850 kGy (85 Mrad) to simulate in-service + LOCA + post-LOCA irradiation conditions.

Previous studies showed that, for some greases, properties can be modified by irradiation doses as low as 10 to 100 kGy (1 to 10 Mrad). However, most high quality greases can withstand irradiation doses of up to 1000 kGy (100 Mrad). Beyond 1000 kGy (100 Mrad), most greases are not useable and special products are required [5]. The irradiation level of 850 kGy is just below the threshold value beyond which grease performance can be severely decreased. Therefore, MOV testing of 850 kGy-irradiated greases was requested as part of the grease evaluation process.

The effect of irradiation on friction coefficient and thrust is shown in Figure 15 (comparison over 20 cycles). The change in friction coefficient or thrust as a result of irradiation increase from 100 to 850 kGy is on average 5 to 10%. However, some greases are more affected by irradiation than others (up to 30% variation for one of the greases tested).

Additional tests at high temperature using 850 kGy-irradiated greases showed that, for the best candidates, there was no negative combined effect of high temperature and high irradiation level on MOV grease performance.

Recommendations for the Selection of MOV Stem/Stem Nut Greases

MOV stem/stem nut lubrication is greatly affected by the type of grease used. The following is a set of recommendations to help in choosing suitable lubricants for this specific application. These guidelines are based on test results obtained for more than 30 grease candidates.

A first short list can be drawn up by restricting it to greases that have a suitable consistency for this type of application (usually penetration grades of 1 or 2) and to greases that can be used at temperatures above 150°C.

The resistance to thermal aging should be assessed: following exposure on oven panels at 120°C over 8 weeks, the grease should not have slumped, the change in grease consistency should be less than 2 grades, and the loss in acid buffering capacity should be less than 80% (or TAN limit number of 3).

The grease should be tested in an MOV test rig to derive friction coefficients and assess the resistance to mechanical aging. For best performance, the friction coefficients measured during valve seating for both the unaged and 5-year aged formulations of the grease should be less than 0.15. Grease types combining a synthetic PAO oil and a clay thickener are particularly suitable if a low friction coefficient is required.

Both unaged and 5-year aged formulations of the grease should show adequate resistance to mechanical aging. Ideally, the change in friction coefficients or TST thrust over 250 cycles should be less than 10%, especially if the actuator tripping point is defined using the developed thrust initially measured for a given torque. Grease types combining a mineral oil and a lithium thickener are particularly suitable if parameter stability is required.

Long-term exposure to in-service temperature should have a minimal effect on friction coefficients. For best performance, average friction coefficients measured for the 5-year aged formulation of a grease should not vary by more than 10% from the average value obtained for the unaged formulation. The grease should also be suitable for high temperature and high irradiation levels that would be typical of accident conditions. The use of high-temperature greases is required in order to guarantee valve operability at peak temperatures of 165°C and accident steady temperatures of 110–120°C. An indication of above average performance at these temperatures is a variation of friction coefficient and TST thrust from reference data measured at room temperature of less than 10%. Some greases combining PAO base oil and clay thickener or mineral oil and lithium complex thickener are suitable for valve stroking at accident temperatures.

Conclusion

Standard lubrication tests are not suitable to assess the lubrication performance of greases used at the MOV stem/stem nut location. Specific procedures have been developed by AECL, EDF, and OPG for thermal aging and MOV testing of grease candidates. The assessment also includes MOV grease testing for temperatures and irradiation levels that are typical of accident conditions.

Since a large number of greases were studied, general trends were observed according to the type of grease tested:

- The stem/stem nut friction coefficients measured for the various grease candidates ranged from 0.10 to 0.17. The lowest friction coefficients were found for greases that combine a PAO base oil and a clay thickener.

- The greases that offered the best resistance to mechanical aging (friction coefficient and TST thrust stability with stroke number) combine a mineral oil and a lithium complex thickener. Some of the greases that combine a mineral oil and a calcium complex and greases that contain a polyurea thickener showed poor resistance to mechanical aging.

- The use of EP additives in the grease was not detrimental but did not seem to be absolutely required for optimal stem/stem nut lubrication.

- The five-year aged formulations of the greases were less resistant to mechanical aging than the unaged formulations.

- Most high-temperature greases were suitable for short-term exposure at 165°C and for prolonged exposure at 110–120°C. Several irradiated greases were suitable for stem/stem nut lubrication, even for levels of up to 850 kGy.

Overall, the screening process proved to be successful. From a thermal stability point-of-view, only nine grease candidates out of an initial list of twenty-four were declared suitable for a 5 year regreasing interval. Following testing in the MOV rig, only five of these greases met the majority of requirements for optimal MOV stem/stem nut lubrication (mostly high-temperature greases). Only one grease met these requirements and was also suitable for specific low friction coefficient applications.

Acknowledgements

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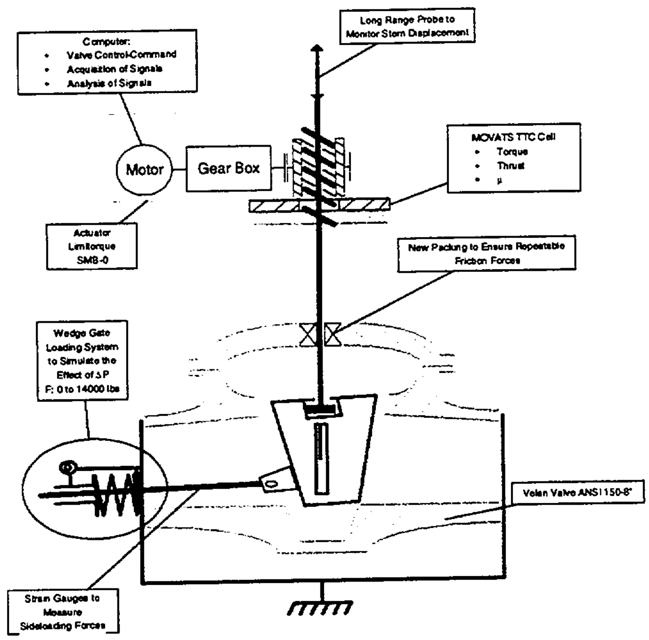

Figure 1: MOV Test Rig

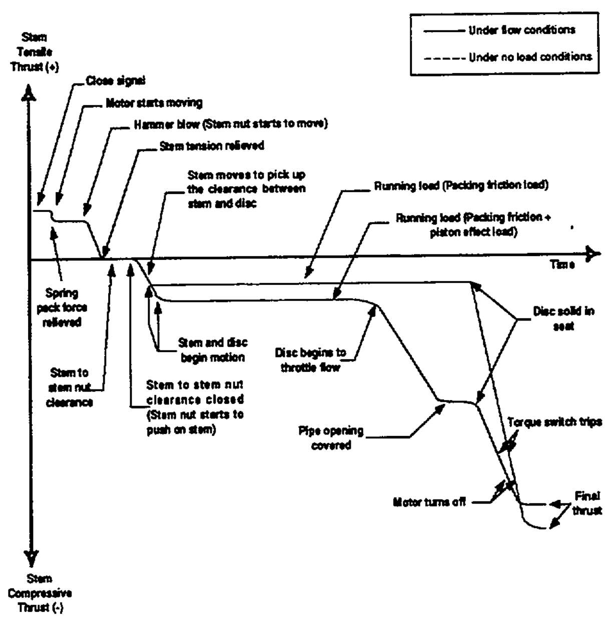

Figure 2: Thrust as a Function of Time for a Typical MOV Closing Sequence [4].

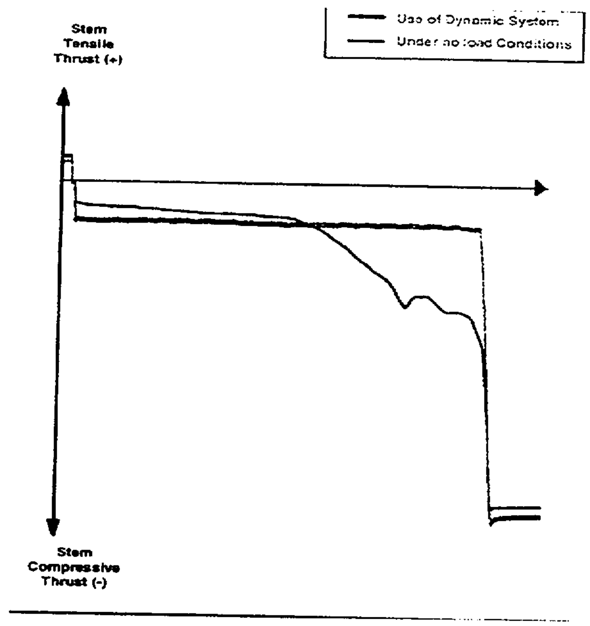

Figure 3: Thrust as a Function of Time for MOV Test Rig Closing Sequences

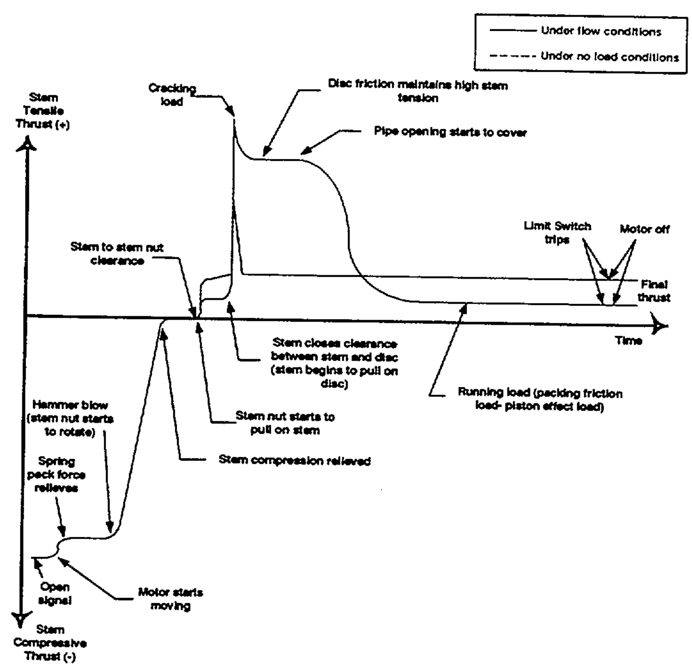

Figure 4: Thrust as a Function of Time for a Typical MOV Opening Sequence [5].

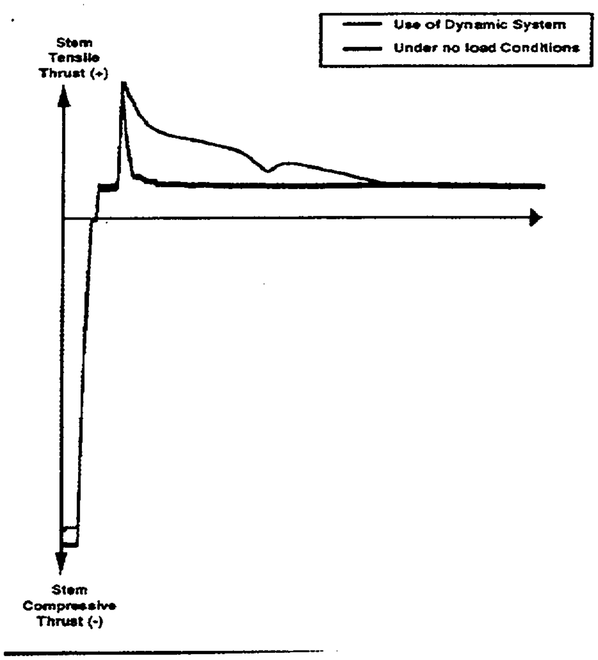

Figure 5: Thrust as a Function of Time for MOV Test Rig Opening Sequences

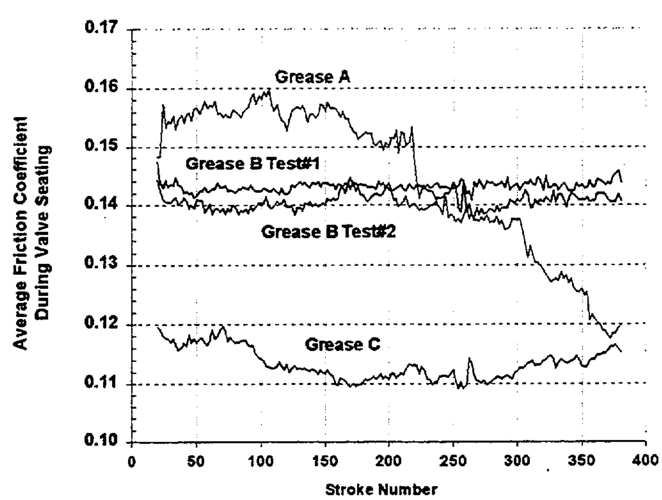

Figure 6: Comparison of Friction Coefficients Measured During Valve Seating for Three Different Greases.

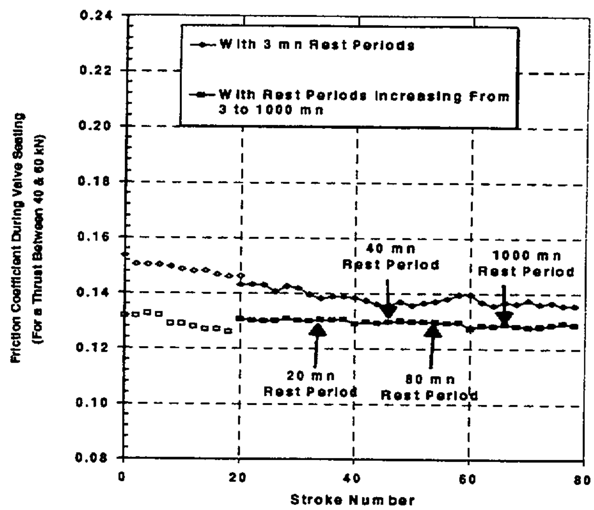

Figure 7: Effect of Rest Period Duration on Friction Coefficient Stability

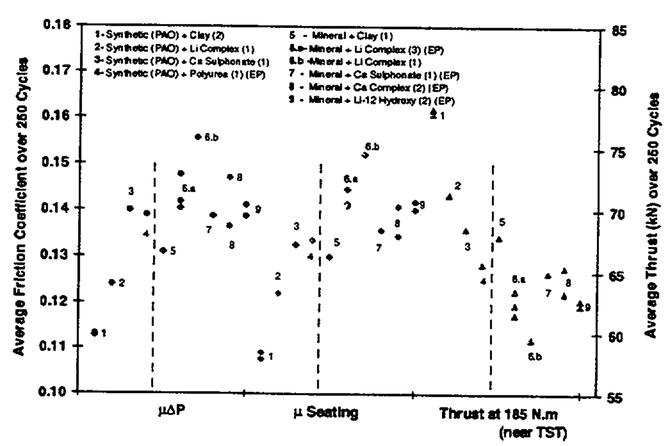

Figure 8: Comparison of Average Friction Coefficients and TST Thrusts for Various Types of Unaged Greases

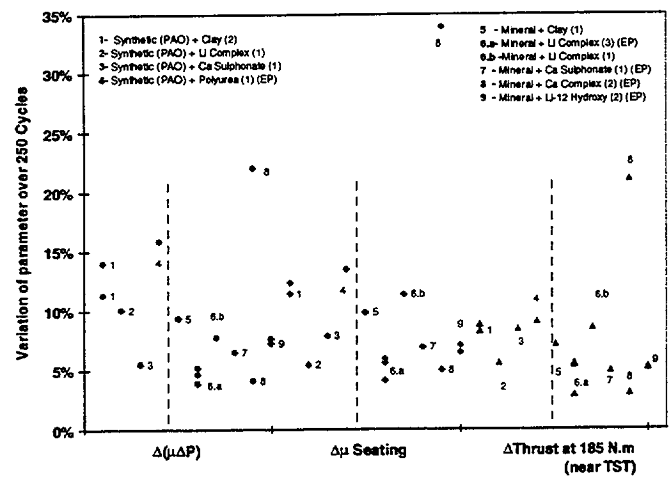

Figure 9: Comparison of Parameter Stability over 250 Cycles for Various Types of Unaged Greases

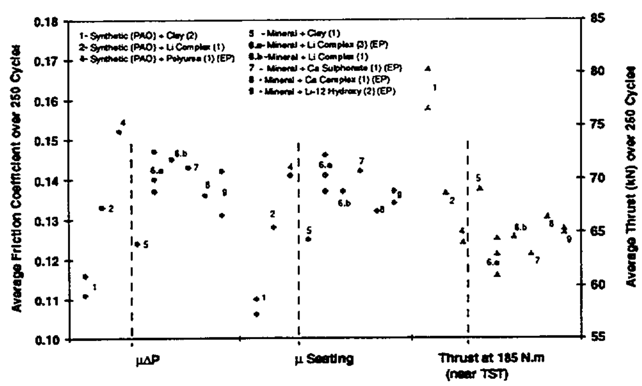

Figure 10: Comparison of Average Friction Coefficients and TST Thrusts for Various Types of Thermally Aged Greases (5 Year Equivalent)

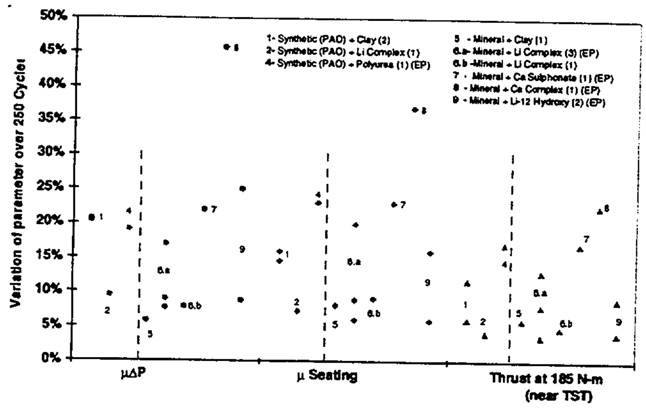

Figure 11: Comparison of Parameter Stability over 250 Cycles for Various Types of Thermally Aged Greases (5 Year Equivalent)

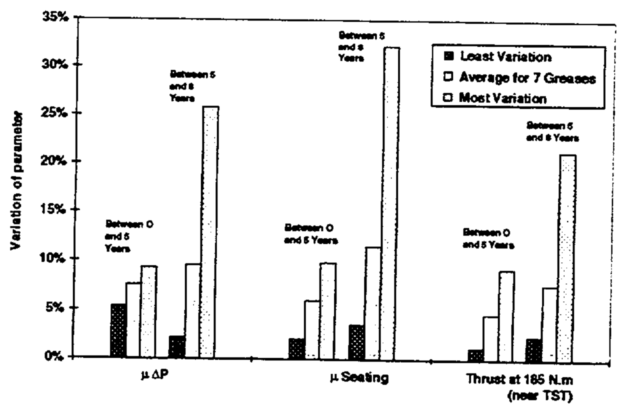

Figure 12: Variation of Average Friction Coefficient and TST Thrust between Unaged, 5-Year Aged, and 8-Year Aged formulations (Most, Least and Average Variation over 50 Cycles)

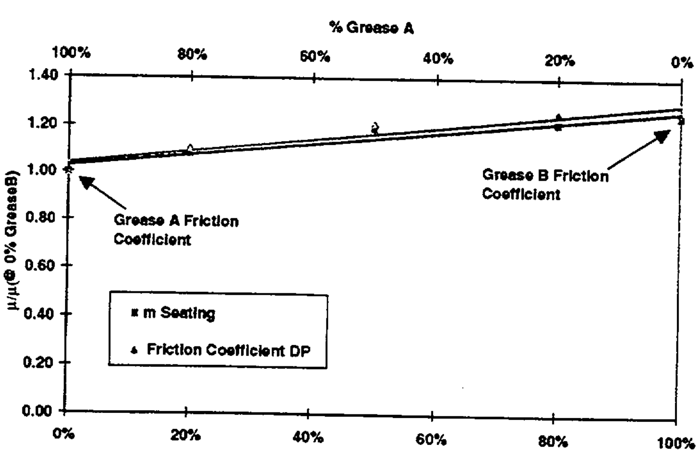

Figure 13: Comparison of Friction Coefficients Measured for Two Compatible Greases and for Various Mixtures of these Two Greases.

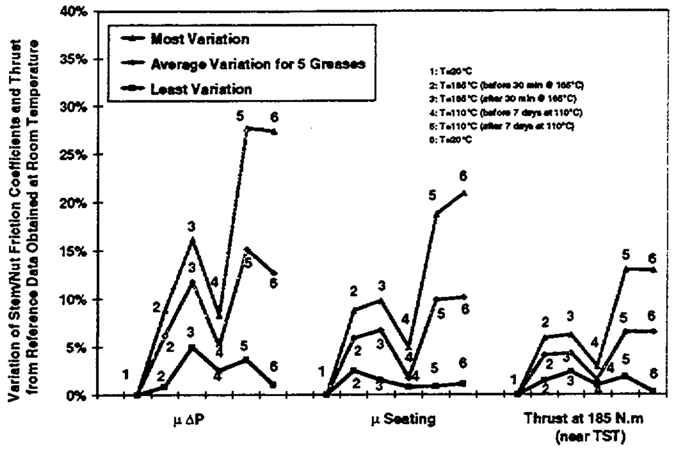

Figure 14: Effect of a Typical OPG MSLB Temperature Profile on Average Friction Coefficients and TST Thrusts

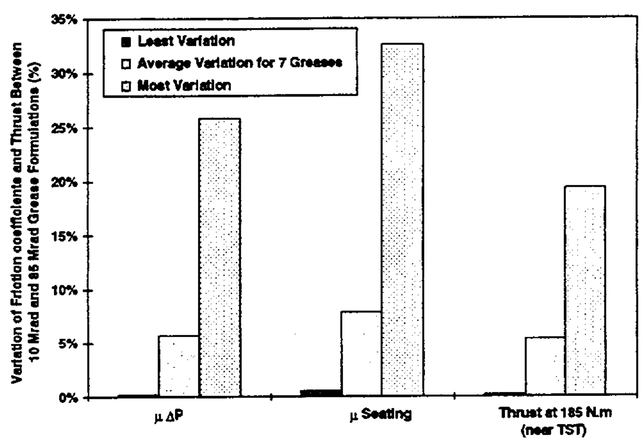

Figure 15: Comparison over 20 cycles of Average Friction Coefficients and TST Thrusts for 10 Mrad and 85 Mrad Grease Formulations

Use of MCC Based Testing Technology for Static MOV Periodic Verification Testing

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Abstract

Several commercial nuclear plant licensees have provided information to the U. S. Nuclear Regulatory Commission (USNRC) regarding use of motor control center (MCC) technology for periodic verification of motor-operated valve (MOV) operational readiness. The licensee information describes strategies ranging from use of MCC test data to extend at-the-valve test intervals to use of MCC testing alone for certain MOVs.

In late 1998, the MOV Users Group organized a committee of industry experts and commissioned development of an industry guidance document for members to use during development and implementation of MCC-based static testing procedures. The MOV Users Group guidance was issued earlier this year and is available for licensee use. The ASME MOV working group has considered one plant's inquiry on use of a specific MCC testing approach as a suitable alternative to at-the-valve in-service testing of MOVs under certain conditions and is exploring modification of Code Case OMN−1 to better accommodate MCC based approaches. Individual licensees have performed site specific studies to assess the effectiveness of MCC technology while others have utilized more extensive laboratory programs, combined with site specific results, as a basis for transitioning to MCC technology for periodic verification of MOV output capability.

At least four different technologies are currently available for MCC based MOV testing. This paper will discuss the applicability and limitations of each currently available technology and describe how licensees have worked to validate the various approaches and gain confidence in MCC testing as an alternative to at-the-valve testing for the purpose of evaluating MOV operational readiness.

Background

USNRC Generic Letter (GL) 89−10, *Safety-Related Motor-Operated Valve Testing and Surveillance*, GL 96−05, *Periodic Verification of Design-Basis Capability of Safety-Related Motor-Operated Valves* and the in-service testing guidance of ASME Code Case OMN−1, *Alternative Rules for Preservice and Inservice Testing of Certain Electric Motor-Operated Valve Assemblies in LWR Power Plants,* identify the need to monitor MOVs in a comprehensive programmatic fashion in order to ensure operational readiness. A comprehensive MOV periodic verification program employs design basis information, field procedures, grouping strategies, technical basis documents, performance testing and condition monitoring, evaluations and corrective actions, trending results and other technical information necessary to ensure the long term operational readiness of MOVs.

In addition to the above, periodic verification of MOV operational readiness requires maintenance and testing strategies that target potential age related degradation mechanisms. Specific testing strategies must be focused on those degradation mechanisms that; 1) tend to increase the thrust or torque required to operate a valve and 2) decrease the thrust or torque available from the actuator to operate the valve to its required safety position. As a consequence, individual licensees must employ a range of testing and preventive maintenance activities designed to detect and control the various degradations that affect overall MOV performance.

A comprehensive programmatic approach also requires ongoing assessment of critical assumptions used in MOV engineering processes. Valve thrust or torque requirements and actuator capability results make up the critical elements of the engineering process and resulting set-up acceptance criteria. Many licensees employ grouping strategies or "control Groups" in order to assess future change in these parameters. For example, a critical assumption used in the thrust calculation process for rising stem gate valves is valve factor. Though valve factor may be comprised of many components it is typically associated with the sliding friction between the valve seats and seat ring or guides and guide arms under dynamic conditions. Changes in valve factor over time can only be detected through periodic dynamic testing.

As a consequence of the above and to minimize program costs, a key element of many licensee MOV programs is ongoing participation in the joint NSSS owners group (JOG) MOV Periodic Verification Program. This collaborative effort includes a study of variations in valve factor performance over time. Participating plants have agreed to test certain MOVs under dynamic operating conditions over a five-year period and share the results with other participants. The valves have been grouped such that the program covers most designs used in nuclear safety-related applications. One objective of the program is to minimize the amount of insitu dynamic testing required by individual licensees yet identify those valve designs or applications where increases in the thrust or torque requirement may occur over time.

Licensees will make adjustments, as appropriate, to MOV calculations based on the JOG results. This approach of sharing dynamic test data and other information regarding potential increases in valve thrust requirements satisfies part of the periodic verification issue for many valves and is consistent with the grouping strategies recommended in OMN−1. Individual licensees may perform additional dynamic testing as required to evaluate valves not covered by the JOG program.

Actuator capability calculations rely on assumptions for actuator efficiency and stem friction. Control groups should also be used to monitor changes in actuator capability assumptions and the results factored back into the periodic verification program. Because of differences in maintenance and lubrication practices, actuator control groups are typically site specific. Data available from previous testing can often be used to verify these assumptions.

In addition to determining how thrust or torque capability may vary, each licensee must develop site-specific programs and procedures necessary to periodically verify output capability of safety-related MOVs. Some licensees will continue to use at-the-valve diagnostic test methods, similar to those used to establish margin during the initial GL 89–10 program effort. Others will employ a mix of static at-the-valve testing and MCC based testing to periodically assess margin. In some cases licensees will rely more heavily on MCC based approaches as an alternative to at-the-valve testing for certain MOVs. Regardless of the approach used, a comprehensive programmatic effort must include performance testing necessary to facilitate a periodic margin assessment and condition monitoring necessary to evaluate health and other trends that may affect design basis assumptions.

Licensees are employing a range of processes to establish intervals for periodic verification program activities. Factors used to establish the frequency of activities such as preventive maintenance and testing typically include risk importance measures, margin and operating environment. The JOG interim static test matrix is one example of a process used to establish test intervals. The JOG approach employs risk significance and margin in order to establish periodic static test frequencies. Plants that use this approach will test MOVs that fall into the low margin, high-risk category more frequently.

Because of the margin criteria used, low margin MOVs cannot be allowed to degrade over time. In fact, any future change in the performance characteristics of an MOV in this category is unacceptable. Licensees that plan to employ the traditional at-the-valve test approaches each fuel cycle for periodic verification testing must guard against the increased risk caused by manipulation of low margin, high safety significant MOVs during the testing process. Contributors to increased plant risk include potential alteration of the MOV's physical condition during the testing process, misadjustment due to calibration or analysis errors and the impact of MOV unavailability during the testing process.

One approach that may be more practical for low margin, high risk MOVs is to perform condition monitoring at an increased frequency (once per cycle) using MCC based technology. Though these MOVs will not typically have enough margin to employ a performance test acceptance criteria, MCC based condition-monitoring approaches are more effective at detecting subtle changes in MOV performance than at-the-valve methods. The MCC test methodology should be supported by control group results and backed up by periodic at-the-valve testing at an extended interval.

High margin, low safety significant MOVs are ideal candidates for MCC testing alone for periodic verification of actuator output capability. Either of the available MCC based performance test technologies could be employed for this purpose.

Though they are ideal MCC candidates, MOVs in the high margin-low risk category are also well suited for control group service. A limited number of these MOVs should be used for control group studies because there are fewer operational restrictions and the licensee can experiment with different maintenance intervals in order to evaluate the optimal PM frequency. Therefore, certain high margin, low risk MOVs should be marked for control group service and tested as appropriate to assess change over time.

Performance Testing and Condition Monitoring

NUREG/CR–6578, *A Methodology for Evaluation of Inservice Test Intervals for Pumps and Motor-Operated Valves,* looks at periodic verification of MOV operational readiness from the inservice testing perspective and considers the benefits realized from the range of activities commonly included in a comprehensive MOV program. NUREG/CR 6578 also emphasizes the differences between performance testing and condition monitoring. Both serve important roles in a periodic verification program.

As defined in NUREG/CR–6578 performance tests are "...go/no-go tests that seek to determine whether a component meets some performance criteria." The static at-the-valve MOV testing performed during implementation of the GL 89–10 criteria is one type of MOV performance test.

The typical at-the-valve performance testing process involves use of direct thrust or torque sensors that facilitate measurement of MOV output capability. Engineering processes are used to establish minimum and maximum acceptance criteria values for MOV output. The at-the-valve acceptance criteria can be modified and used for evaluation of MCC based data.

NUREG/CR–6578 also recognizes that certain MCC based diagnostic methods, such as MCSA, provide better information on motor capability than traditional at-the-valve techniques. A key element of this conclusion is the use of the MOV motor as a transducer versus the traditional stem measurement approach. The stem measurement approach requires an assumption relating to motor health and internal actuator gearing instead of actual data.

MOV Users Group Guidance Document

In late 1998 the Motor-Operated Valve (MOV) Users Group organized a committee of industry experts for the purpose of reviewing currently available MCC technologies and developing guidance for member utilities to use during development and implementation of MCC related periodic verification test approaches.

The committee maintained a close working relationship with ASME MOV working group (OM–8) and JOG MOV program core group participants during development of this guidance. The committee also solicited feedback from the expanded MOV Users Group MCC committee during the final stages of development.

The MOV Users Group MCC document titled, *Guidance on the Use of MCC-Based Technologies for Static MOV Performance Testing and Condition Monitoring,* provides detailed information on four specific MCC analysis technologies. The document provides detail on how these technologies work and reviews the critical assumptions, applicability/limitations, strengths and weaknesses of each.

The MOV Users group MCC committee followed the guidance of NUREG/CR–6578 and characterized MCC methods based on whether the technology was applied as a performance test or used in a condition–monitoring role.

The MOV Users Group guidance targets use of MCC based technology for static periodic verification testing and discusses a range of conditions important to adoption of MCC technology for periodic verification testing.

MCC Based MOV Technology

There are a number of MOV diagnostic systems commercially available that can be used for MCC based testing of MOVs. These systems capture time based motor supply voltage, motor current and switch actuation events at the remote, MCC location. This basic MCC information may be processed differently depending on the particular system being used.

These systems are similar, portable data acquisition computers that contain signal conditioning hardware and software necessary to acquire signals from motor voltage and current probes attached to the motor power circuit.

The following analysis technologies are currently available for evaluation of MCC data:

1. MPM Equivalent Thrust
2. MC^2 Motor Torque and Motor Torque Correlation
3. Motor Power Analysis
4. Motor Current Signature Analysis (MCSA)

Though each technology utilizes the same basic MCC data, application of the analysis process is significantly different. In some cases it may be appropriate to use a combination of MCC analysis techniques for confirmation and higher confidence in the results. The following discussion on each technology was extracted from the MOV Users Group guidance document:

MPM Equivalent Thrust

"The prior thrust trace is analyzed in order to determine the amount of time between the point indicating hard seat contact and rapid loading begins (C11) and the torque switch trip point (C14) where the motor is de-energized. The corresponding thrust/time relationship is established as well as the fixed running load (packing load) prior to hard seat contact.

At a later date, motor power data acquired at the MCC is analyzed in order to determine the amount of time between hard seat contact and torque switch trip point. The thrust/time relationship that was established during the prior thrust test plus the fixed running load from the prior test is used to determine a new thrust at torque switch trip value."

MC^2 Motor Torque

"The motor torque method employs the time based motor torque history that was generated from motor electrical data acquired during an MCC based test. The motor torque data is used with the Limitorque design performance equations to calculate actuator torque and thrust.

Based on the Limitorque sizing and selection procedures and reiterated in Limitorque Technical Update 98–01, the relationship between stem thrust and motor torque is best described by the following equations:

Actuator Torque = Motor Torque X Ratio X Efficiency*

Thrust = Actuator Torque/Stem Factor

Or

Thrust = Motor Torque X Ratio X (Efficiency/Stem Factor)

* Efficiency is inclusive of application factor"

"The motor torque correlation method requires a simultaneous MCC and at-the-valve test. Data from this test is used to create a linear curve fit of the relationship between motor torque and actuator output torque and/or stem thrust. A representative correlation coefficient is developed. The correlation coefficient is mathematically equal to the product of gear ratio, actuator efficiency and stem factor. These correlation coefficients can be applied to subsequent motor torque signature sets to generate correlated stem thrust and actuator output torque signatures. The correlation coefficient should be adjusted to account for expected degradation over time."

Motor Power

"The 3-phase motor supply voltage and motor current signals can be combined electronically (in a circuit or in software) to create time domain motor power signatures. Motor power analysis provides a qualitative indication of change between tests and can be used for trending."

MCSA

"The MOV motor current frequency spectrum is a product of Fast Fourier Transform (FFT) analysis of induced modulations occurring in the electric supply current to the motor. The motor and internal rotating actuator components create modulations in the 60-cycle line frequency. The FFT algorithm calculates the frequency of cyclic/repetitive events from the instantaneous current signature. The presence of an event/peak usually indicates that energy is being expended at that particular frequency. Efficient gear meshing and good bearings may not contribute observably to the frequency spectrum."

"The primary function of frequency domain analysis is to track changes in the frequency spectrum that have occurred since a baseline test. Shifts in frequency peaks are indications of load changes. Differences in the amplitude of certain peaks indicate changes in energy being expended at certain component frequencies, and differences in sidebanding around certain significant peaks can indicate changes in the actuator efficiency.

The ability to accurately track the condition of internal MOV components and to 'second check' time domain analysis results make frequency domain analysis an integral part of the MCC based periodic static test approach. Frequency domain analysis results, combined with either EQT or motor torque results can provide additional information helpful in extending the frequency of 'at-the-valve' testing."

MCC Technology Critical Assumptions and Applicability

The MCC testing approaches described above can be characterized as either a quantitative performance testing approach or a qualitative condition monitoring approach. The equivalent thrust and motor torque processes are considered performance test approaches because an engineering process closely linked to the original GL 89–10 acceptance criteria process can be used to evaluate the test data. Motor power and MCSA are qualitative condition monitoring approaches because operational readiness is not easily evaluated in watts or as a frequency event.

The critical assumptions, applicability, strengths and weaknesses of each technology are discussed in the following sections:

Equivalent Thrust

The Equivalent Thrust methodology requires an initial static MOV thrust signature to establish the thrust load rate and duration while the valve is seating. During future tests, motor power data acquired at the MCC is used to evaluate changes in the seating duration. The thrust load rate from the initial test provides critical information for the evaluation. In order for the EQT methodology to work correctly the thrust load rate between C11 (hard seat contact) and C14 (torque switch trip) must remain constant over time.

The actuator performance parameter most likely to influence the EQT methodology is packing load. Changes in packing load result in a slower or faster motor speed at C11 and C14 depending on the direction of the change. Other changes that influence load rate to a lesser extent are supply voltage, stem friction and actuator efficiency changes.

The equipment OEM performed a series of field tests on certain Limitorque® SMB actuators with AC motors to assess the accuracy of determining thrust at torque switch trip (C14) with the EQT methodology. The OEM has also conducted a separate effects laboratory test program to assess the magnitude of individual changes that potentially effect accuracy. Detailed information on accuracy can be obtained from the OEM.

An attractive feature of the EQT process is the ability to directly assess thrust at torque switch trip and compare values to MOV program acceptance criteria. Weaknesses include limitations of the methodology to close direction thrust at CST only. There are also limitations due to valve seating characteristics and signature features that are a result of problems identifying C11. Because of the concern over possible load rate changes due to speed alternate methods (such as MCSA) should be used to verify motor speed in parallel with the EQT process.

Motor Torque

The Motor Torque Method requires use of actual or conservative stem factor and efficiency values, which include allowance for expected degradation over time. When evaluating motor torque data, the user employs actuator efficiency and stem factor values for converting motor torque to actuator torque and thrust. Bounding assumptions similar to those used in actuator capability evaluations may be used but this approach unnecessarily consumes available margin. The optimal approach requires analysis of parallel motor torque, actuator torque and stem thrust test data to establish more representative values based on actual equipment condition and maintenance practices.

Though the overall process of using motor torque data to evaluate actuator capability is applicable to all MOVs, the OEM has only validated the accuracy of the MCC based motor torque measurement for certain Limitorque® Motors 60 Ft-Lbs. and less that are commonly used on SMB, SB and SBD Actuators.

A key feature of the motor torque method is that a simultaneous calibration with direct thrust or torque is not necessary.

This allows the user to go directly to the MCC and assess performance. However the use of control group data to establish representative efficiency and stem factor values and expected degradation is required in order to satisfy performance-testing criteria.

Motor Power

Changes in motor power signature events are usually the result of changes in the load on the motor. Increased power consumption (watts) during running indicates a higher load on the motor provided the supply voltage has not changed. Increases or decreases in seating power and other events are also indicative of higher thrust or torque provided the motor has not degraded.

Motor power analysis is typically employed as a qualitative, condition-monitoring tool for trending variations in power signature events over time. Individual licensees have developed site specific procedures for evaluation of motor power data.

Motor power data can be acquired easily with a number of off-the-shelf transducers. The output of these transducers can be configured for input into most commercially available diagnostic systems.

Supply voltage and other changes may effect power levels and complicate the analysis and trending process and limit power analysis to a qualitative, condition monitoring tool.

MCSA

The initial (baseline) frequency spectrum should be analyzed to assess mechanical condition and future tests should be compared to the baseline to assess change. It is imperative that the initial baseline represents a "healthy" mechanical and electrical condition.

MCSA can be applied to all electric motors and actuators however; gear train and motor design data must be available to evaluate the frequency spectrum.

MCSA is sensitive to changes in motor/actuator electrical & mechanical condition and provides confirming data for motor torque and EQT methodologies. However, because of the wide range of actuators and resulting differences in frequency response, MCSA requires significant classroom training and extensive OJT.

Plant Specific Approaches

The number of licensees currently attempting to transition from at-the-valve MOV testing to remote MCC based testing continues to increase. With few exceptions all domestic U.S. licensees own equipment and software necessary to perform MCC based testing. Many are in the early experimental stage and acquire MCC data in parallel with ongoing at-the valve testing. Several licensees have taken a more aggressive posture and have completed site specific and laboratory validation programs. The following discussion identifies how three different licensees have approached development of MCC based testing approaches:

Farley

MCC based testing is used at the Farley Nuclear Plant as a tool to extend the at-the-valve test interval for certain MOVs.

Validation Strategy:

Certain plant installed MOV motors and a number of spare warehouse motors were initially tested on a precision dynamometer at the Farley Nuclear Plant in order to assess the accuracy of MCC based motor torque measurements for specific 550/575 volt motors. Once the motor torque accuracy was validated, simultaneous MCC and at-the-valve test results were evaluated to explore feasibility of the test equipment OEM's recommended analysis process. The simultaneous test results were used to establish the relationship between input motor torque and actuator torque (actuator efficiency) and the relationship between actuator torque and stem thrust (stem factor). The relationship between input motor torque and stem thrust was also evaluated for certain MOV groups.

The licensee worked with the test equipment OEM to develop a technical basis document for use of the data and a process for future MCC based testing. The licensee continues to perform parallel MCC tests when at-the-valve testing is required. This data and other control group data are factored back into the technical basis and adjustments to the analysis process made as appropriate.

Fermi

MCC based testing is used at Fermi as a tool to extend at-the-valve test intervals for certain MOVs and to facilitate an increased frequency for others.

Validation Strategy

In addition to the test equipment OEM's validation, a similar on-site validation was performed. MCC data was gathered in parallel with at-the-valve data and the MCC data used to calculate equivalent thrust. The MCC based results were then compared to the at-the-valve results to assess accuracy.

The Fermi site validation is an ongoing process that is continually updated with parallel test data. Each cycle, a formal evaluation is performed to document any required adjustments to the program.

Sequoyah & Watts Bar

MCC based testing is used at Sequoyah and Watts Bar as the sole test strategy for certain quarter-turn butterfly valves.

Validation Strategy:

The population of safety-related MOVs at the Sequoyah and Watts Bar Nuclear Plants includes certain Henry Pratt butterfly valves with Limitorque HBC gearboxes. The yoke and valve to actuator connections block direct access to the valve stem and prevent installation of strain gages for direct torque measurements. These valves are also soft seated and as a result direct torque measurements (if feasible) only provide meaningful information under design basis flow and differential pressure test conditions. These MOVs are also limit seated without torque switch protection.

The licensee worked with their MCC test equipment OEM to develop and validate a test methodology for the specific Limitorque HBC gearboxes used on quarter-turn butterfly valves at Sequoyah and Watts Bar. Twelve groups of representative Limitorque SMB/HBC actuators were tested under simulated in-plant, static conditions and the relationships between input motor torque and output HBC torque established. The twelve groups cover all SMB/HBC configurations used. A report was developed that provides the technical basis for evaluation of future MCC test data for the specific SMB/HBC gearbox configurations used at Sequoyah and Watts Bar.

Future ASME MOV Working Group Activity

ASME Code Case OMN–1 provides alternatives for inservice testing of MOVs. OMN–1 describes a complete programmatic approach necessary to ensure the long-term operational readiness of certain safety-related MOVs as required by the ASME code.

The ASME MOV Working Group responsible for the technical content of OMN–1 has recently interpreted the role of certain MCC-based static test approaches for inservice testing as part of the code inquiry process. The MOV working group recognized that under certain conditions MCC based approaches could be used to demonstrate margin for MOVs provided uncertainty guidelines and other requirements of OMN–1 are met.

In response to industry activity in this area and other improvements in MOV technology, the ASME MOV working group has begun the process of revising this Code Case to better reflect the current state-of-the-art in MOV testing and surveillance. An expected element of this revision will be features or guidance necessary to employ MCC based technology to meet inservice testing requirements for MOVs.

Conclusions

MOVs in nuclear safety-related applications have received considerable attention since the late 1980s. These MOVs have been subjected to a process of verifying design conditions, verifying and upgrading actuator sizing methods, verifying the field setup through testing and focused preventive maintenance. At the conclusion of GL 89–10 program efforts, industry-wide confidence in the operational readiness of nuclear plant safety-related MOVs was greatly improved.

In today's environment, the question that each licensee must satisfactorily answer is whether MOV performance may have changed or degraded since the original GL 89–10 tests. Though differences exist among licensee programs, all employ three fundamental strategies as part of the overall process of ensuring MOV operational readiness.

One strategy being employed to minimize the potential for change in actuator capability is focused preventive maintenance. A number of MOV degradation mechanisms can be identified and corrected during the preventive maintenance and visual inspection process. Licensees have also repeated at-the-valve testing to evaluate the effectiveness of the preventive maintenance program.

With few exceptions licensees are participating in and closely monitoring the JOG valve factor results. Licensees will modify MOV programs as appropriate based on JOG results.

Licensees are also relying on ongoing performance testing and condition monitoring to evaluate MOV output capability. In many cases this testing should be performed with MCC technologies. With the right combination of MCC and at-the-valve testing, licensees can further improve confidence in MOV operational readiness while shedding much of the testing and schedule burden typically associated with GL 89–10 MOV programs.

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Chad Smith, Duke
Frank Winter, CECO

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The Joint Owners' Group MOV Periodic Verification Program—Forging Ahead with Testing and Learning

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Abstract

To address long-term motor operated valve (MOV) performance, the Babcock & Wilcox, BWR, Combustion Engineering and Westinghouse Owners' Groups teamed to form the Joint Owners' Group (JOG) MOV Periodic Verification (PV) Program. The key element of this program is repetitive testing of in-plant MOVs under differential pressure conditions, to identify potential degradation in required thrust (gate and globe valves) and torque (butterfly valves). Each participating nuclear unit is testing two MOVs. Each MOV is tested three times over a period of five years, with at least one year between tests. The JOG PV Program has been running for over two years, and data from over 140 MOVs have been obtained. For about 30% of these MOVs, the data cover the first and second tests, and provide preliminary insights about potential degradation. Gate valves in low temperature water systems tend to show a slight increase in valve factor when they are stroked repetitively during a single test (i.e., multiple test strokes on the same day). Hence, conducting the test tends to slightly increase the valve factor. When these valves are subsequently tested at least a year later, a mixture of increases, decreases and stable valve factors is observed. For several of the increases, the valve was disassembled and reassembled prior to its first test, which tended to reduce the valve factor. Hence, the subsequent valve factor increase tends to restore the earlier valve performance. Therefore, low valve factors measured in testing after valve internal maintenance may not remain low. The results for butterfly valves tend to show decreases in bearing friction coefficient with stroking on the same day. For strokes separated by a year or more, a mixture of decreases and increases is observed. For globe valves, the results show small changes in valve factor with time or stroking.

Background

Over 90% of the nuclear power plants in the U.S. are participating in the Joint Owners' Group (JOG) motor operated valve (MOV) Periodic Verification (PV) Program. A previous paper (Reference 1) described the technical bases, content and approach of the JOG PV Program. Briefly, the program helps U.S. nuclear power plants address long-term MOV performance and periodic verification, to satisfy NRC Generic Letter 96–05. As discussed in Reference 1, the program started in 1997 with three Owners' Groups teamed together: the Boiling Water Reactor, Combustion Engineering and Westinghouse Owners' Groups. During

1999, the Babcock & Wilcox Owners' Group joined the program. With this addition, all four of the U.S. nuclear plant Owners' Groups are included. This united approach has key benefits for the participating plants and for the regulator. Importantly, it conserves resources. Cost effectiveness is achieved by sharing the burden of valve testing among the participating plants. Also, because the program provides a uniform approach for all participating plants, the regulator's burden to individually inspect and approve multiple programs is alleviated. Accordingly, the plants can operate under a predictable regulatory expectation with high certainty of acceptance. Finally, because the program has over 90 participating units, an extensive set of MOV test data is being generated, collected and evaluated. These data, which are far more extensive than any single plant could expect to generate, provide the basis for a strong technical justification.

As mentioned above, a key element of the JOG PV Program is MOV testing at the participating plants. This testing is performed under conditions with flow and differential pressure (DP). Each participating unit is testing two valves under DP conditions. Each valve is tested three times over a spread of five years, with at least a one-year separation between tests. The test valves were selected so that, in aggregate, they cover the valve features and system conditions most commonly encountered in nuclear power plants.

A recent paper (Reference 2) described the early experience with in-plant valve testing and showed results from some of the early tests. Most of the test results in Reference 2 were from the first (baseline) test of the planned three-test series. Accordingly, the insights on potential valve degradation were limited. At the time the present paper was prepared, the amount of data has increased considerably, and there are now an appreciable number of tests covering the second and, in a few cases, the third tests in the series. The purpose of this paper is to update the tests results and insights gained in the program from these data. However, because repeat test results still remain to be obtained for a majority of valves in the program, the observations and insights in this paper are preliminary.

In-Plant DP Testing

The DP test program includes 190 valves, subdivided into 141 gate valves, 29 butterfly valves, 12 unbalanced disk globe valves, and 8 balanced disk globe valves. Each valve is tested three times under nominally identical DP conditions. Consecutive tests are separated by at least one year.

To ensure that data obtained from in-plant tests are satisfactory for use in the JOG PV Program, a test specification is used, which includes requirements for:

- Test valve maintenance and material conditions
- Test conditions
- Test instrumentation
- Test sequence
- Test data evaluation
- Test documentation

The goal of the standard test specification is to ensure that all valves and testing are properly controlled to achieve adequate consistency and quality from test results obtained from multiple plants. Importantly, the test specification requires that time-history data for stem thrust (or torque for butterfly valves) and DP be obtained. Further, the specification requires analyzing and summarizing the data in a prescribed manner. Finally, the specification requires a test sequence which includes both static and DP test strokes.

When a plant performs a test for the JOG Program, they prepare and submit a test data package in accordance with the test specification. The contents of the test data package include information about the valve, system, test procedure and instrumentation, as well as plots of the measured data for the multiple valve strokes carried out during the test. Also, a test analysis data sheet is completed for each stroke, which identifies key points during the stroke and summarizes measured results for those key points.

When a test data package is received in the JOG PV Program, it is formally reviewed against the requirements of the specification. If questions or deviations are observed, these are resolved with the preparers of the packages. Once all questions and comments are resolved, the package is accepted into the JOG Program. By using this approach, it ensures consistency between test data packages received from different plants and ensures a high overall level of quality.

At the time this paper was prepared, about 200 test data packages had been submitted by the participating plants to the JOG Program, and about 150 of those packages had been reviewed and accepted. In general the experience with testing has been positive, and the data packages have provided a good source of information. Some plants have had difficulty getting pressure taps close enough to the valve to satisfy the program requirements, and special analyses have been necessary to justify use of the data. Although the specification requires one static open and close stroke pair and one DP open and close stroke pair in the test, the specification recommends that two static stroke pairs and two DP stroke pairs be obtained. The majority of the tests to date have been able to obtain two static and two DP stroke pairs.

On a few repeat tests, the plants were not able to achieve the desired repeatability of DP in the test specification (± 10% of the average value). Typically, this deviation has occurred because the system is not being operated in precisely the same manner as in the first test. In most of these cases, the DP unrepeatability affects only the closing stroke and not the opening stroke. Accordingly, the Program has been able to capture some valid data from these tests.

In a few tests (baseline and subsequent) the instrumentation or test conditions have failed to meet the specification, and test data were rejected. Also, a few repeat tests have missed their required conditions and have been rejected. At this time, the rejection rate of data submitted to the program is about 5%.

Gate Valve Test Results

Each gate valve DP test is evaluated in a consistent manner so that the data from different valves can be meaningfully compared. During this evaluation, several key "stroke points" are identified, and the "valve factor" is calculated for each of these stroke points. Valve factor is defined as the ratio of the thrust required to move the valve disk (or "DP thrust") to the product of DP and the area based on the mean seat diameter ($Area_{ms}$).

$$Valve\ Factor = \frac{DP\ Thrust}{DP * Area_{ms}}$$

The stroke points identified and evaluated in gate valve tests include:

Closing Strokes

- Flow isolation
- Initial wedging
- Initial wedging—second point (if applicable)
- Maximum thrust up to the initial wedging point

Opening Strokes

- Just after unwedging
- Maximum thrust after unwedging
- Flow initiation

In Reference 2, which discussed early results from the JOG PV Program, it was noted that the valve factor tends to slightly increase from the first DP stroke to subsequent DP strokes, when the strokes are performed as part of the same test, i.e., same day. The majority of data continue to show this effect. As mentioned in Reference 2, this effect is attributed to the tendency of the friction coefficient between self-mated Stellite surfaces to increase with cumulative stroking up to a plateau level, as was shown in the EPRI MOV Program (Reference 3).

To examine potential degradation of gate valves as a result of being in service, the data from valves that have tests beyond their baseline test is examined. Further, these data are grouped in various ways to attempt to reveal the influences that might affect degradation. The key design features and the fluid conditions (temperature, fluid, type, etc.) are the simplest groupings. Other groupings might include, for example, the value of baseline valve factor, amount of in-service stroking, normal valve position, stem orientation, and time between static and DP strokes. A limited evaluation of these types of groups has been performed and is discussed below. However, since the data in each group are limited, the results are preliminary.

Figure 1 shows results from several gate valves with self-mated Stellite disks and seats, and self-mated stainless steel guides, used in treated water systems near room temperature (<100 °F). This graph presents the valve factors at initial wedging during the closing stroke. Although results from opening strokes or other points of closing strokes could also be plotted, the data on Figure 1 are typical and representative. Repeat tests from four gate valves are included. Three of these four valves show a significant valve factor increase from the baseline test to the second test. Based on reviewing the data, the reason for the increase is related to the valves' history. These valves had been disassembled for internal maintenance and then reassembled just prior to the baseline tests. The disks were replaced as part of this maintenance. The maintenance activity resulted in a decrease in valve factor, which was confirmed by comparing the baseline results to other tests of these valves performed before the JOG PV Program. The increase in valve factor that occurred between the baseline and second tests returned the valve factor approximately back to its pre-disassembly levels. Figure 2 shows a measured thrust overlay of pre-disassembly, baseline and second tests. The decrease and subsequent increase can be observed.

Figure 3 shows results of repeat tests from seven gate valves with self-mated Stellite disks and seats and self-mated carbon steel guides, used in treated water systems near room temperature (<100 °F). On four valves, the valve factor remains nearly constant between the baseline and second tests. On the other three, the valve factor increases. With one exception, the valve factor increases occur on valves that had been disassembled and reassembled prior to the baseline test, similar to the results discussed on Figure 1. In the case of the one exception, an anti-corrosion chemical treatment had been injected into the system just prior to the second test, and it is possible that this treatment contributed to the increase. More data are currently being sought on valves in systems that have a similar chemical treatment, to see if this effect can be isolated.

Figure 4 shows results from gate valves with self-mated Stellite disks and seats and Stellite disk guide surfaces, used in treated water systems near room temperature (<100 °F). All of the valves show a relatively stable valve factor.

Figure 5 shows results from gate valves that have a split wedge construction with a ball and socket joint. The configuration of these valves is described in Reference 4. The valves were tested in treated water systems near room temperature (<100 °F). One of the valves shows a significant increase from the start of the baseline test to the end of the second test. As discussed in Reference 2, the increase occurs primarily as a result of stroking the valve during tests. Data from a third test of this valve tend to show that further increases between tests are small.

Figure 6 shows results from gate valves with self-mated Stellite disks and seats, used in elevated temperature systems with water or steam flow. With one exception, the valves show stable or decreasing valve factors. The causes and contributors to the increase on the one valve are still being sought.

Butterfly Valve Test Results

Butterfly valve tests results are evaluated to determine the bearing friction coefficient, which is proportional to the bearing torque component. Although other torque components affect the total required torque, the other components have been judged not to be susceptible to degradation (hydrodynamic torque) or are capable of being evaluated during normal static, or zero DP, testing (seat torque). Bearing torque is determined by examining the differences in required torque between static and DP strokes run during a test.

Figures 7 and 8 show bearing friction coefficient results from butterfly valves with bronze and non-bronze bearings, respectively. All of the valves are tested in water systems near room temperature (<100 °F). Both graphs show similar trends. When repetitive strokes are performed as part of a single test, the bearing friction coefficient tends to decrease between strokes. For consecutive tests separated by a year or more, the bearing friction coefficient tends to remain nearly constant or decrease (comparing first stroke to first stroke).

Balanced Disk Globe Valve Test Results

Balanced disk globe valves tend to exhibit low values of DP thrust, due to their design. Test results from the JOG PV Program have verified this expectation. Typical measured valve factors are about 0.1. At this value, the required thrust tends to be dominated by packing thrust and stem rejection load; the DP component is minor. The DP load in a balanced disk globe valve is attributed to the small amount of pressure imbalance present in the disk and to friction between the disk and its cage. Only the second of these mechanisms is susceptible to degradation. As shown in Figure 9, very little apparent degradation has occurred in the valves tested to date. Typically the valve factor remains constant, although minor increases have been observed. These increases are less than the uncertainty of the measurements. In general, degradation in balanced disk globe valves is of little consequence (and is difficult to detect) unless it is severe.

Unbalanced Disk Globe Valve Test Results

As discussed in Reference 1, no degradation mechanisms were identified for unbalanced disk globe valves. Repetitive tests of several unbalanced disk globe valves have been included in the JOG PV Program to confirm that the required thrust does not change while the valve is in service. Repeat tests from a few of those globe valves have now been obtained. The results are shown in Figure 10. All of these results are from globe valves in water systems near room temperature (<100 °F). One of the valves has results from all three tests. As expected, the valve factor remained nearly constant throughout the tests. Two other valves that have first and second tests also showed nearly constant behavior, as expected. One valve that has a first and second test shows an increase of about 10% from the first to the second test. During the second test of this valve, the DP obtained at the point when the valve closed was significantly lower than the first test. Although the valve factor would be expected to be about the same regardless of the DP, the change in conditions offers a way for variations in instrument response (due to measurement inaccuracy) to affect the results. Therefore, it is unclear if the change in valve factor is attributed to a genuine change in valve performance or to an effect of changing conditions. A firm conclusion is not being made until the data from the third test in the series is obtained.

As discussed in Reference 2, one globe valve is tested in steam. The results of an initial test are presented in Reference 2, but no repeat data are available to discuss here.

Summary

1. The JOG PV Program is being used by the vast majority of US nuclear power plants to implement their periodic verification testing and to determine the potential degradation in required thrust or torque for gate, globe and butterfly valves.

2. A key component of the JOG PV Program is in-plant valve testing. The testing is well under way, and an appreciable amount of data from repeat (second and third) tests is now being obtained.

3. Gate valves that are stroked repetitively on the same day tend to show a slight increase in valve factor.

4. Gate valves that are disassembled and then reassembled tend to exhibit a reduced valve factor when they are tested after reassembly. The valve factor can subsequently increase while the valve is in service.

5. Results from a split wedge gate valve with a ball and socket connection showed an increase in valve factor over a three-year period from the first test to the second test. This increase was related to the fact that the valve had never been previously DP tested and had a low initial valve factor. The subsequent valve factor increase from the second test to the third test was very small.

6. Butterfly valves tend to show a decrease in bearing friction coefficient when they are stroked repetitively on the same day. When they are repeat-tested after at least one year, a

mixture of increases and decreases is observed.

7. Balanced disk globe valves exhibit low valve factors and low changes in valve factor.

8. Unbalanced disk globe valves exhibit valve factors near 1.0, as expected. Repeat tests indicate that the valve factor is stable, i.e., changes are small. A valve factor increase has been observed in a situation where the test was not repeated accurately. Further data are being sought to determine if the increase is related to valve degradation or is a result of the test changes.

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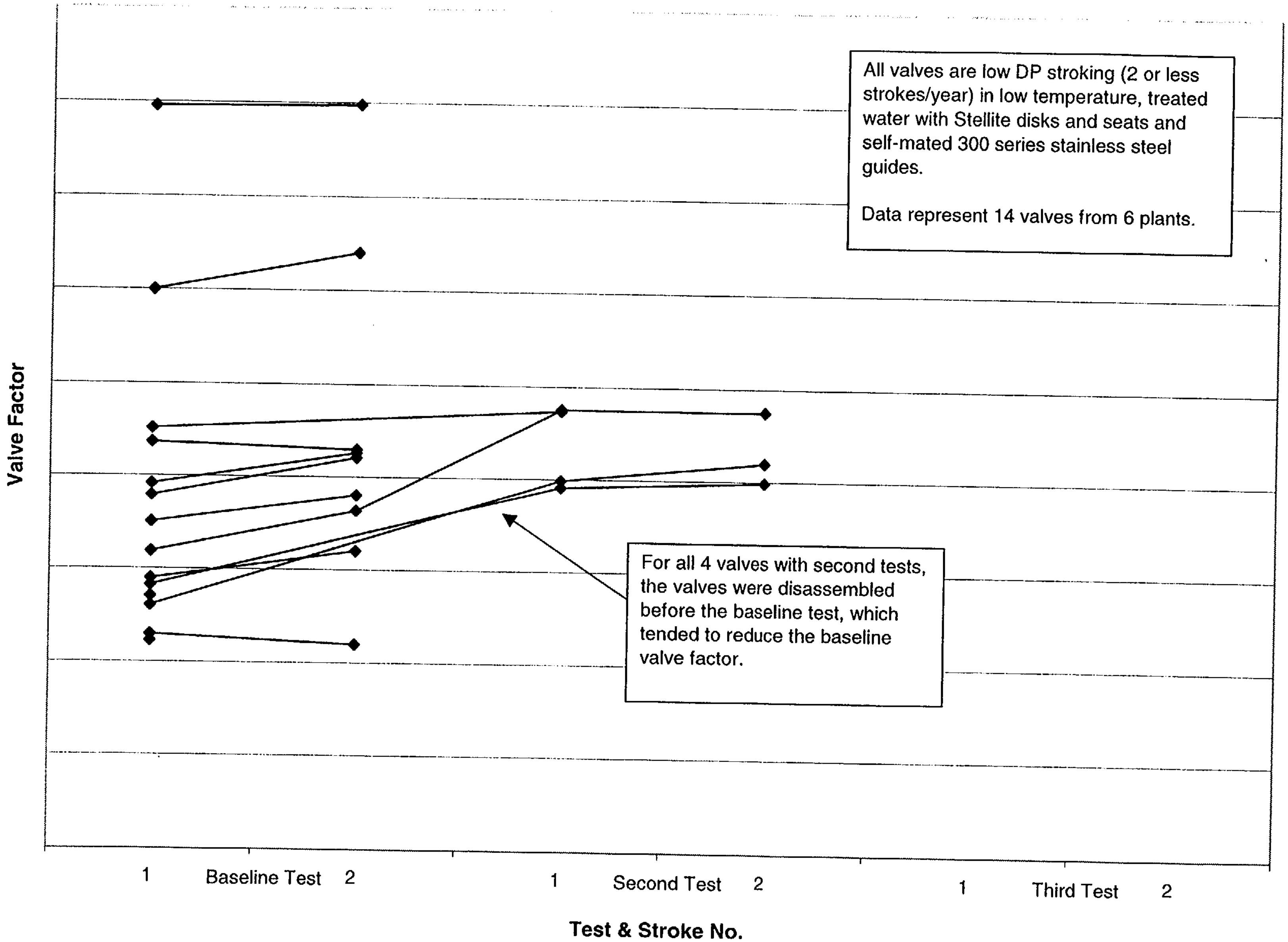

Figure 1. Progression of Closing Valve Factor at Initial Wedging for Gate Valves with Stainless Steel Guides in Treated Water Systems

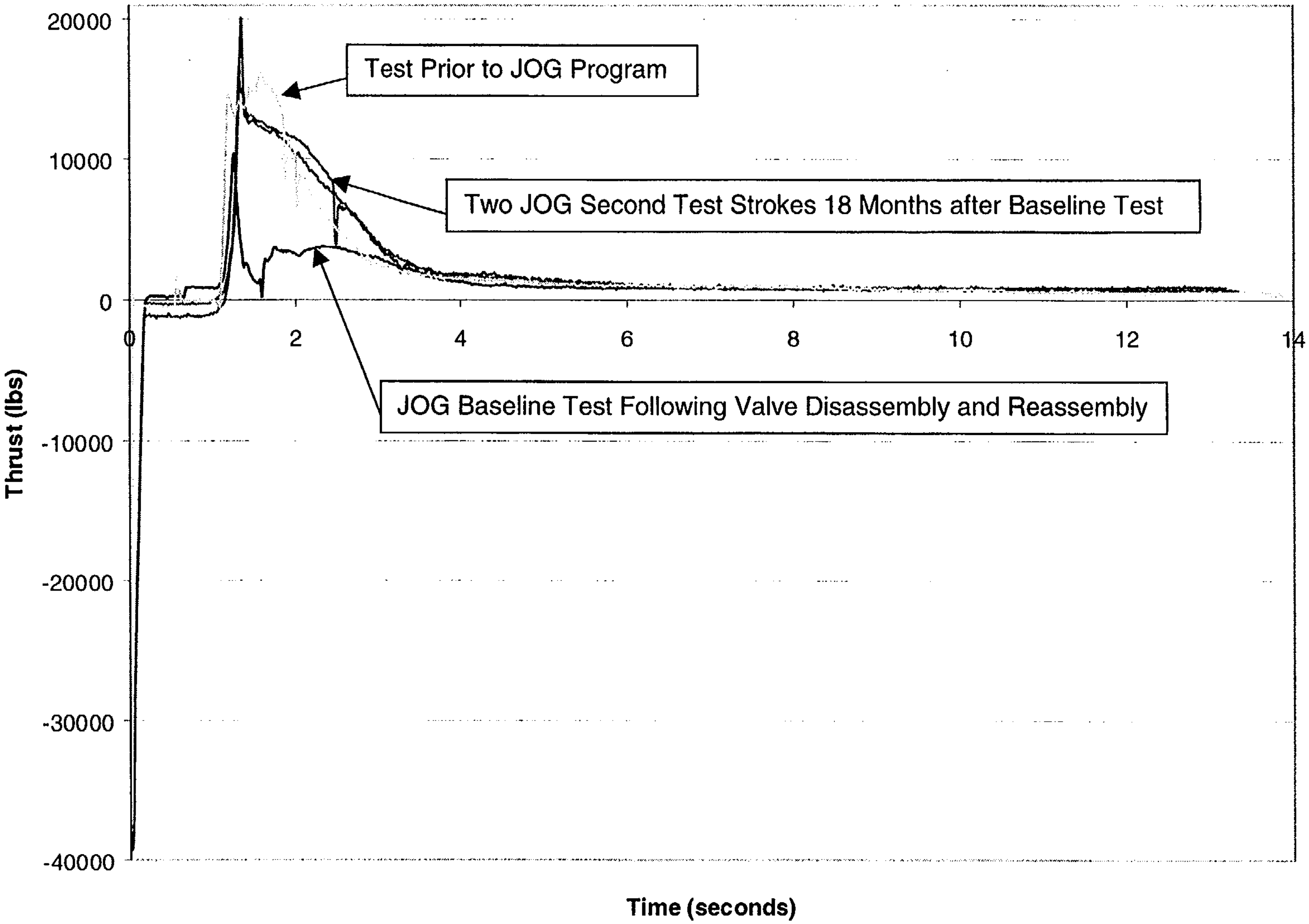

Figure 2. Overlay of Thrust Traces for Valve that Showed a Valve Factor Increase between the Baseline and Second Tests

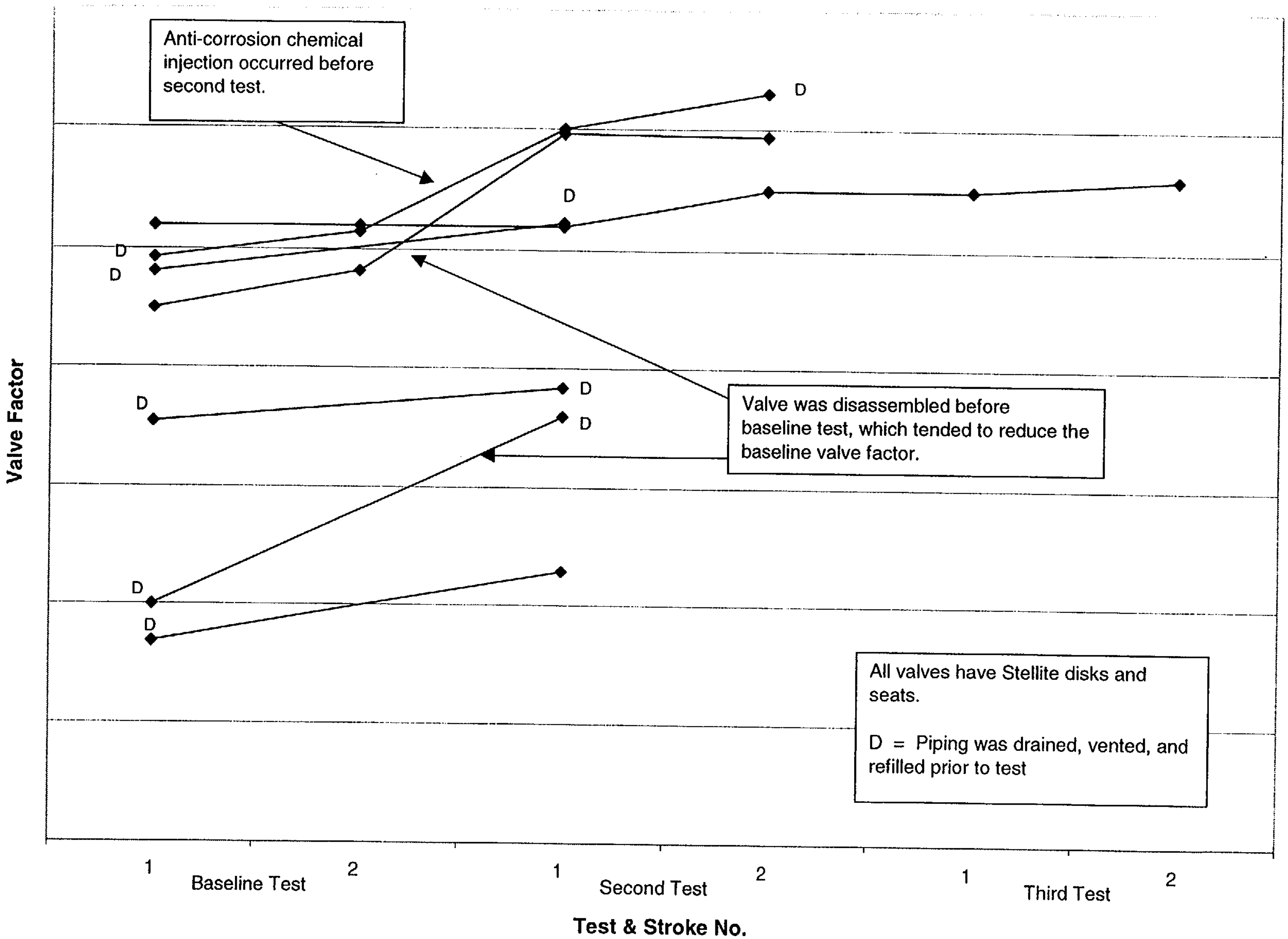

Figure 3. Progression of Closing Valve Factor at Initial Wedging for Gate Valves with Carbon Steel Guides in Treated Water Systems

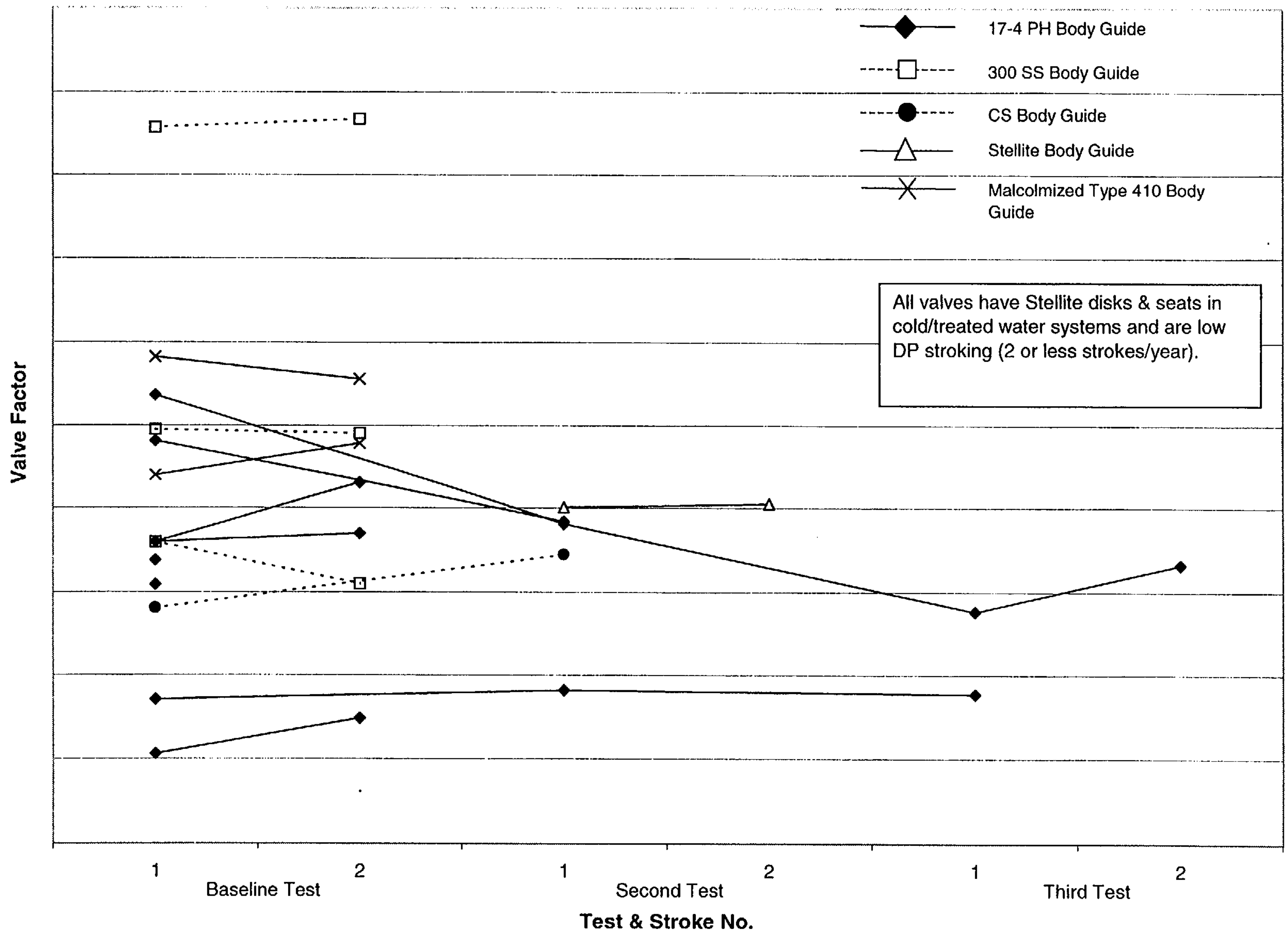

Figure 4. Progression of Closing Valve Factor at Initial Wedging for Gate Valves with Stellite Disk Guide Surfaces in Treated Water Systems

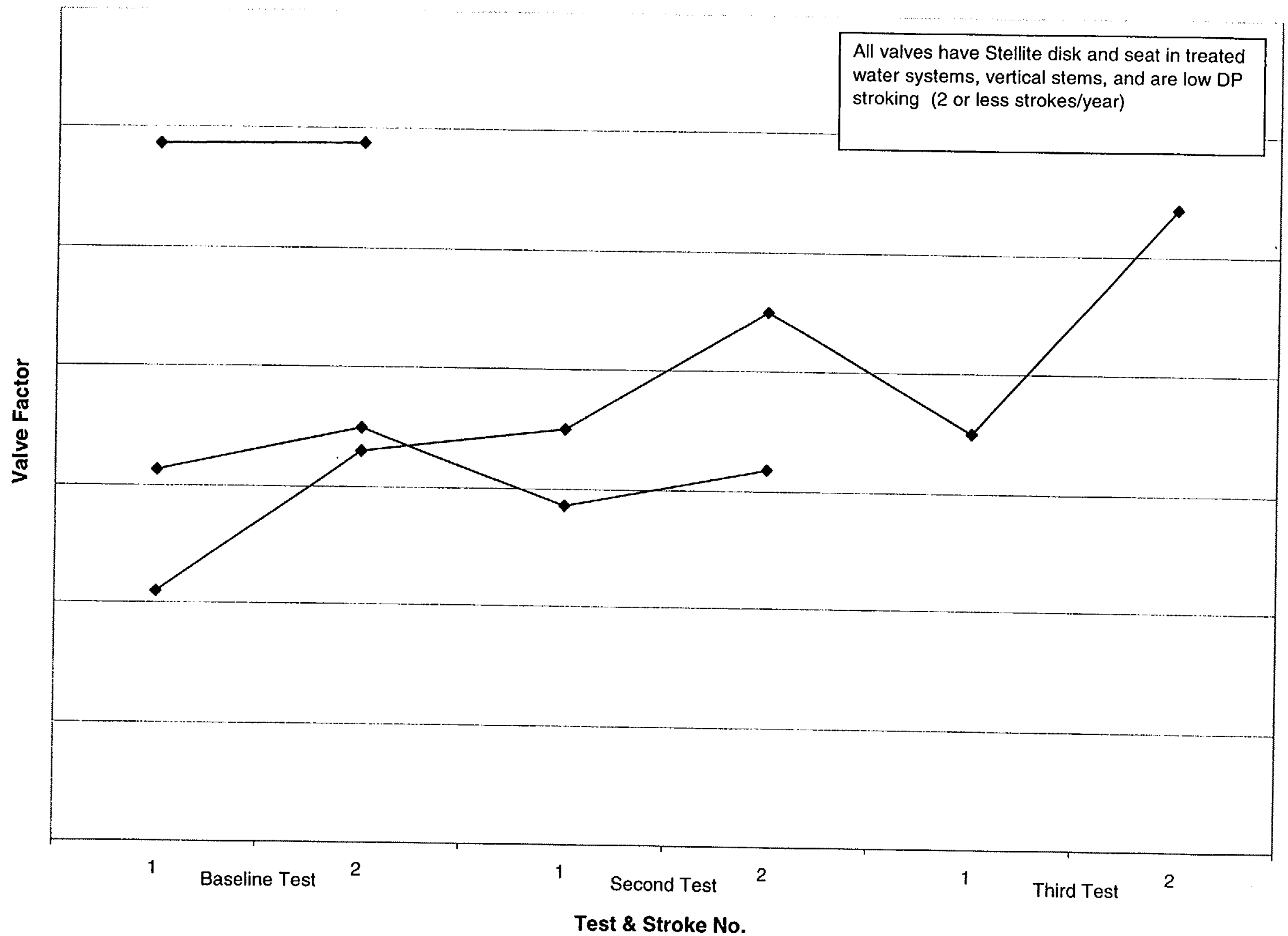

Figure 5. Progression of Closing Valve Factor at Initial Wedging for Split Wedge Gate Valves

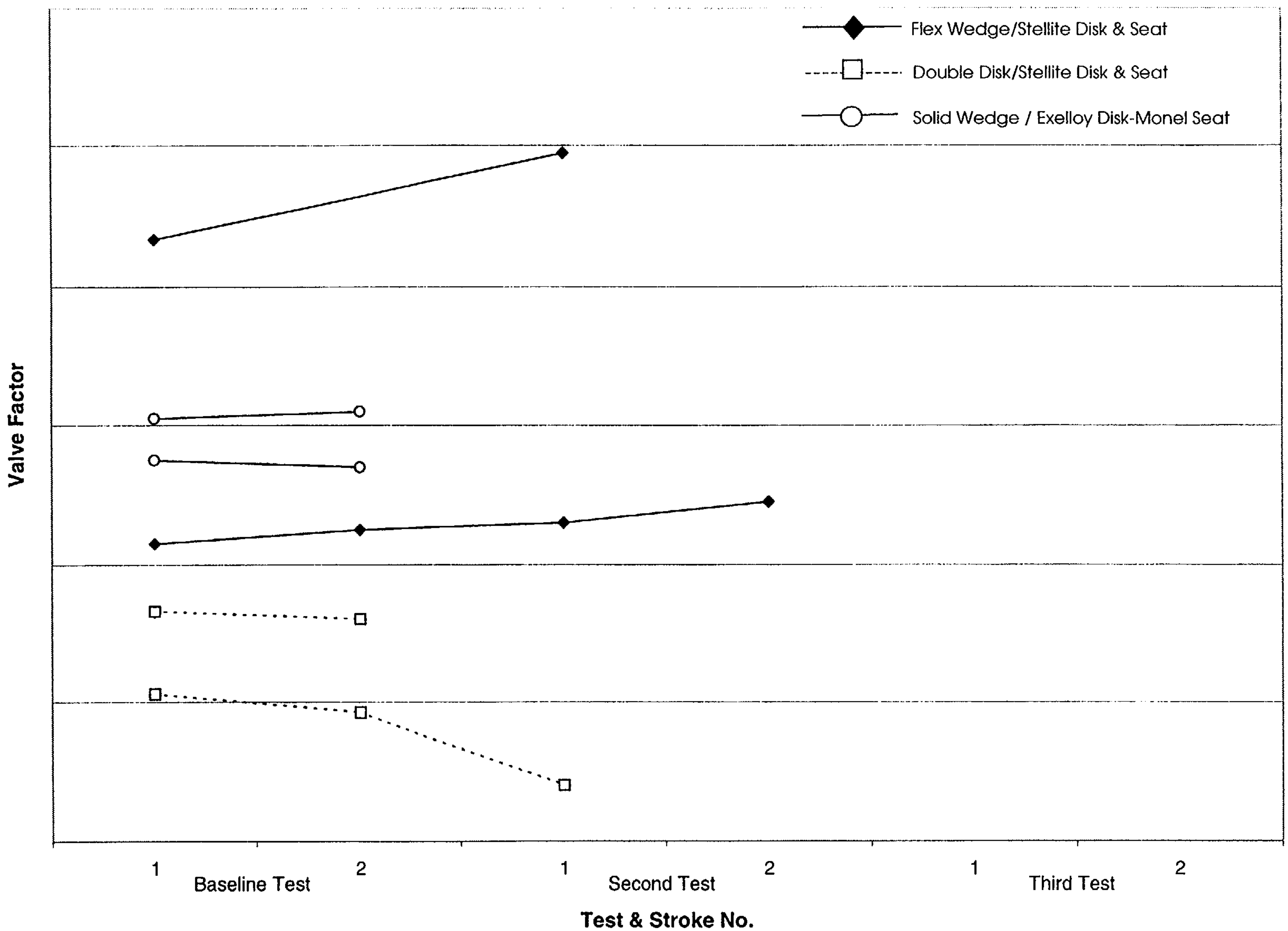

Figure 6. Progression of Closing Valve Factor at Initial Wedging for Gate Valves in Hot Water or Steam Systems

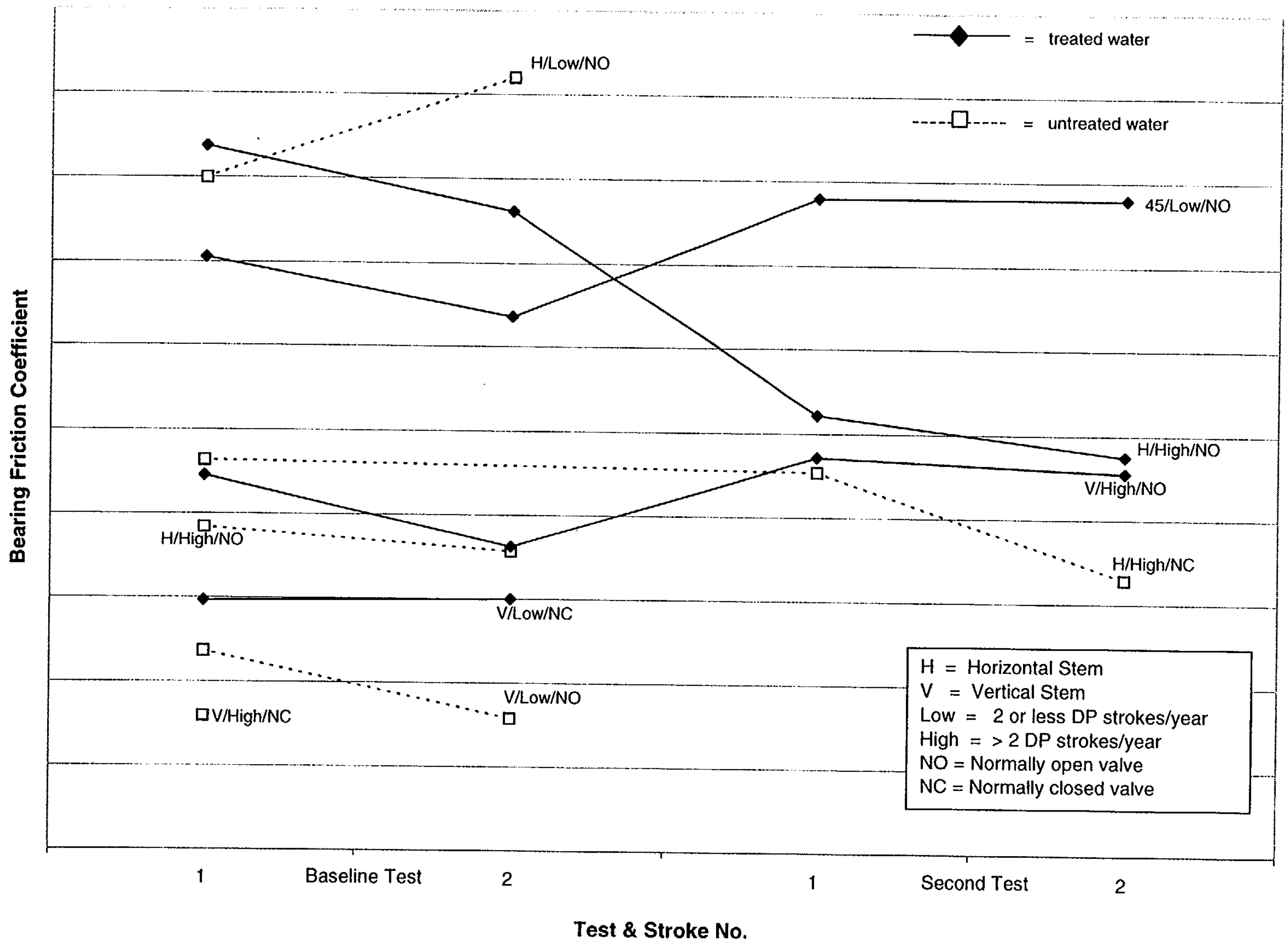

Figure 7. Progression of Bearing Friction Coefficient for Butterfly Valves with Bronze Bearings

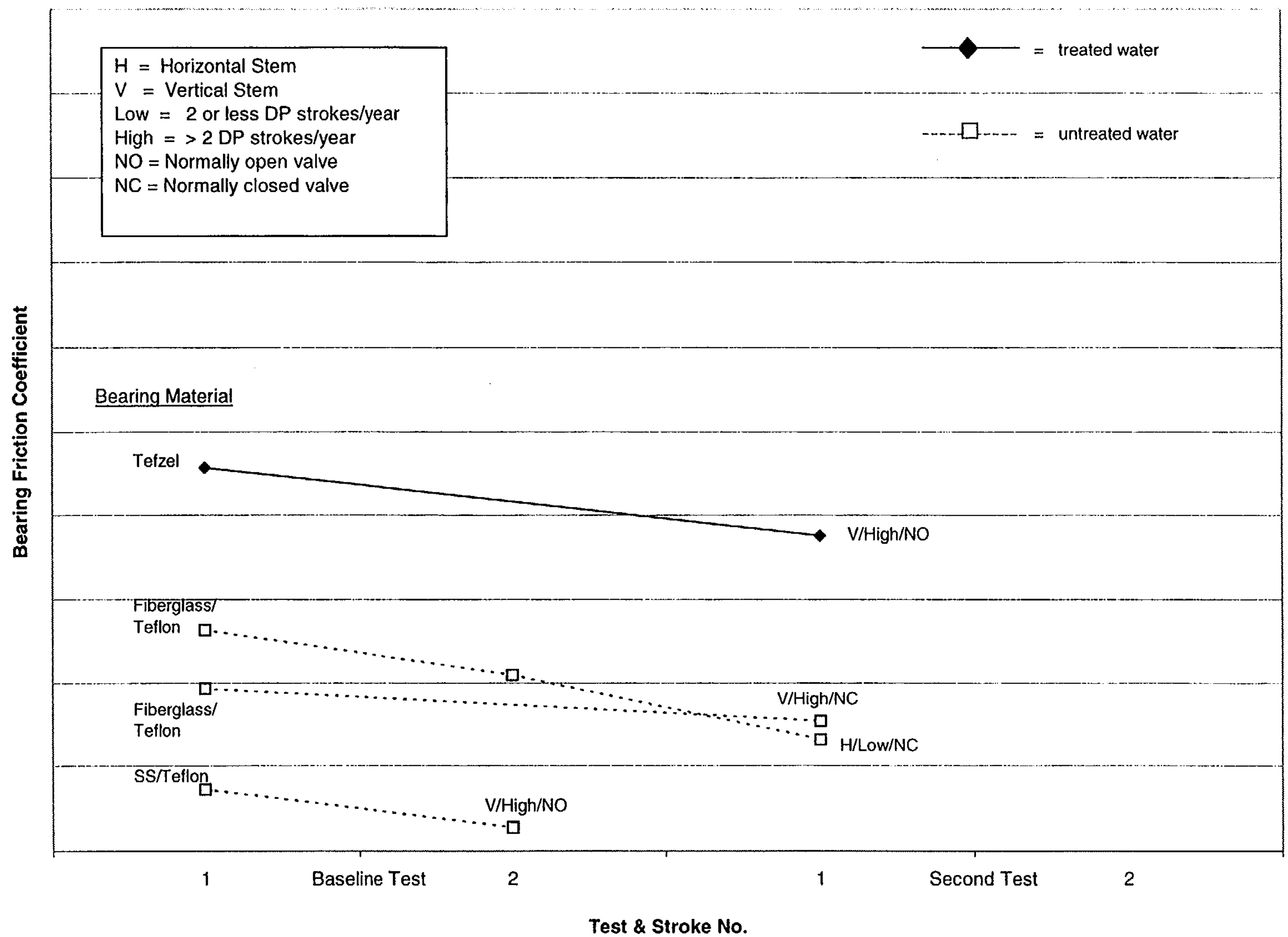

Figure 8. Progression of Bearing Friction Coefficient for Butterfly Valves with Non-Bronze Bearings

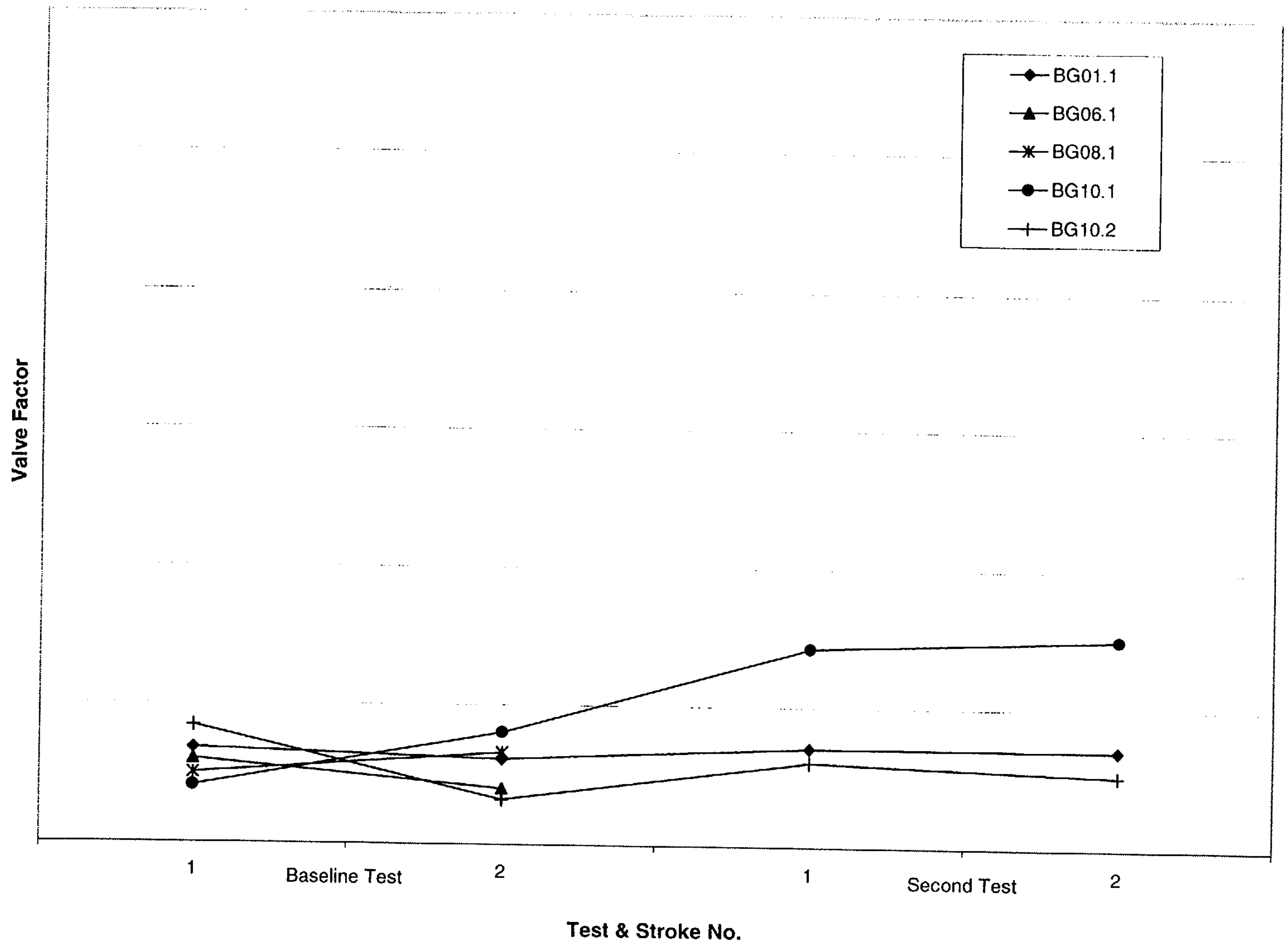

Figure 9. Progression of Closing Valve Factor at Seating for Balanced Disk Globe Valves

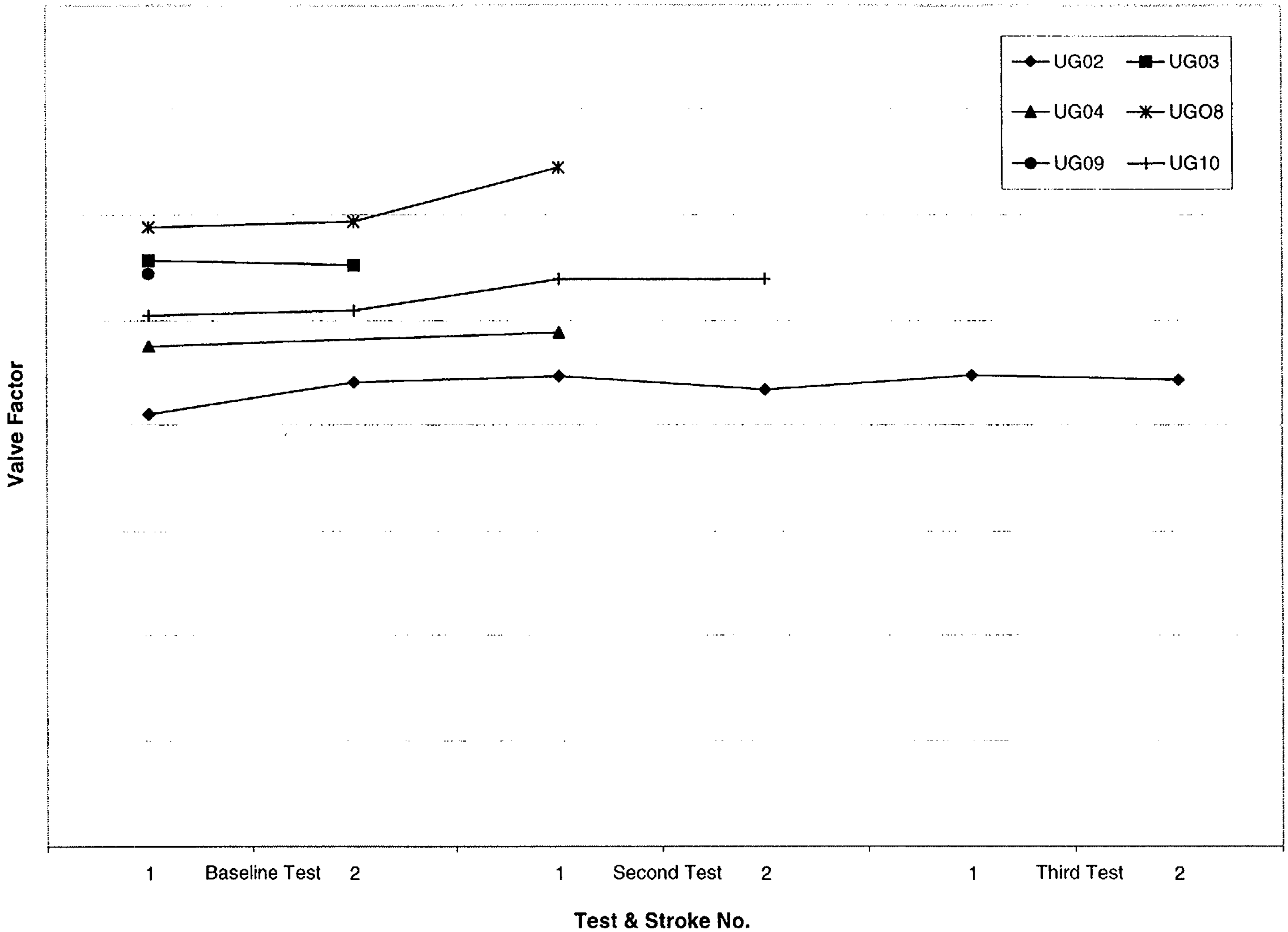

Figure 10. Progression of Closing Valve Factor at Seating for Unbalanced Disk Globe Valve