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

Held at Hyatt Regency Hotel
Washington, DC
July 21–23, 1992

Sponsored by
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of the American Society of Mechanical Engineers

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ABSTRACT

The 1992 Symposium on Pump and Valve Testing, jointly sponsored by the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the Nuclear Regulatory Commission, provides a forum for the discussion of current programs and methods for inservice testing and motor-operated valve testing at nuclear power plants. The symposium also provides an opportunity to discuss the need to improve that testing in order to help ensure the reliable performance of pumps and valves. The participation of industry representatives, regulators, and consultants results in the discussion of a broad spectrum of ideas and perspectives regarding the improvement of inservice testing of pumps and valves at nuclear power plants.

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ACKNOWLEDGMENTS

The editors acknowledge the efforts of the Session Chairs, authors, and panel members for their invaluable contribution to the success of the symposium. Special thanks is extended to Co-Chairmen John Allen and Edmund (Ted) Sullivan for their efforts in planning and conducting the symposium; to Forrest Rhodes, William Russell, Mark Sheehan, John Ferguson, and Steven Weinman for the opening presentations; to the foreign speakers and attendees; and to the symposium banquet guest speaker James J. Kilpatrick. Appreciation is expressed to the symposium steering committee: John Allen, Patricia Campbell, Joe Colaccino, R. Scott Hartley, Tom Hoyle, Joe Philipps, Mark Sheehan, Edmund Sullivan, and Gerald Weidenhamer, with a special acknowledgment of the efforts of Mark Sheehan in coordinating the logistics of the symposium. Appreciation is also expressed for the efforts of the session chairs in reviewing the papers: Kenneth Barry, Kevin DeWall, Gerald Dolney, Ivo Garza, Helmut Knoedler, Robert Parry, Chris Pendleton, Mark Pittman, Lawrence Sage, and Thomas G. Scarbrough. Gratitude is expressed to all the attendees, without whom the symposium would be meaningless.

DISCLAIMER AND EDITORIAL COMMENT

Statements and opinions advanced in papers presented at the Second NRC/ ASME Symposium on Pump and Valve Testing are to be understood as individual expressions of their authors and not those of the American Society of Mechanical Engineers nor 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.

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Opening Remarks

William T. Russell
U.S. Nuclear Regulatory Commission

NRC Goals for Improving the Performance of Pumps and Valves

William T. Russell, Associate Director
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Office of Nuclear Reactor Regulation
U.S. Nuclear Regulatory Commission

The U.S. Nuclear Regulatory Commission (NRC) receives almost daily reports on poor operating experience with the performance of pumps and valves in nuclear power plants. This operating history is the principal basis for the actions required by NRC to improve the overall performance of pumps and valves.

This symposium provides a unique opportunity for working level experts to exchange information, ideas, and suggestions to improve the performance of pumps and valves. Symposium participants include plant personnel involved with the day-to-day operation of pumps and valves, coordinators of pump and valve programs at nuclear plants, individuals who support the American Society of Mechanical Engineers (ASME) Operations and Maintenance Code for testing pumps and valves, and members of the NRC staff responsible for evaluating licensee activities to ensure the proper performance of pumps and valves.

The first Joint ASME/NRC Symposium on Pumps and Valves in 1989 was highly successful. It is encouraging to see this second pump and valve testing symposium taking place to build on the earlier accomplishments. Symposium participants should take full advantage of the opportunity offered by this symposium for the exchange of information and for frank discussions on the need to improve the performance of pumps and valves.

In recent years, the industry has seen improvements in plant safety and availability. Also, the number of unplanned plant trips and outages has decreased. However, the performance of pumps and valves has not improved to an acceptable level.

Licensees and regulators rely on probabilistic risk assessments (PRAs) to help make decisions regarding the safety significance of potential failures of pumps and valves to perform their design-basis functions. Those PRAs include a basic assumption that the pumps and valves have been designed and constructed to be capable of functioning in accordance with their design intent. With that assumption, the PRAs assume generic data for the failures of pumps and valves. If a pump or valve has not been designed and constructed to be capable of performing its safety functions, the PRA is not representative of the facility and will not provide accurate information on the risk associated with the failure of a pump or valve. Operating and test experience at nuclear plants has shown a significant variation in the performance of pumps and valves between apparently identical components at the same facility as well as at different facilities. For example, the licensee programs in response to Generic Letter (GL) 89-10 and NRC inspections of those programs have found a number of motor-operated valves (MOVs) to be incapable of performing their design-basis functions. The NRC staff issued Information Notice 92-17 to provide the results of those inspections of GL 89-10 programs. In addition to MOVs, operating events and NRC inspections have identified weaknesses in the design, installation, maintenance, and testing of check valves. Recently, the NRC staff prepared reports on problems with the performance of solenoid-operated valves (SOVs) and air-operated valves (AOVs).

In response to the growing concerns regarding the performance of pumps and valves, the industry has initiated efforts to improve their performance. These activities include the development of MOV programs at individual utilities, the tric Research Institute MOV Performance Prediction Program, the MOV Users Group of nuclear power plant licensees, improvements to inservice testing (IST) programs in response to GL 89-04, check valve programs in response to the Institute of Nuclear Power Operations (INPO) initiatives, the Nuclear Industry Check Valve Group, the recently-formed Air-Operated Valve Users Group, and the industry's efforts in the area of solenoid-operated valves. These specific programs are a vast improvement in the state of the industry compared to a few years ago. However, many of these efforts are only in their initial stages and need time to mature.

An integrated approach will be needed to resolve the concerns surrounding the performance of pumps and valves. Pump and valve vendors, licensees, and regulators must understand the design and engineering aspects of pump and valve performance. Licensees are responsible for demonstrating through test and analysis the capability of pumps and valves to function under design conditions. Following such demonstrations, licensees must maintain and periodically test the pumps and valves to ensure their reliability. Individually, none of these program elements are sufficient; but rather, all are needed to ensure the proper performance of pumps and valves in a nuclear power plant.

ASME and individual utilities need to develop appropriate testing criteria for the required functions of pumps and valves. ASME and individual utilities need to establish an appropriate frequency for particular pumps and valves given the test methodology and the service history. Licensees need to provide for the analysis, feedback, and trending of test results to demonstrate continued design-basis capability of pumps and valves. Such feedback of test results also relates to applicable pumps and valves for which testing under design conditions is not practicable. Licensees also need to establish adequate maintenance on appropriate intervals to detect degradation and to trend the results of the maintenance to allow them to anticipate pump and valve failures. Licensees need to perform adequate root cause analysis and corrective action for deficient components and to apply that information, where appropriate, to other components. The industry needs to develop diagnostic equipment that is capable of providing all necessary information for determining whether pumps and valves can perform their safety functions. Individual utilities will need to ensure that its personnel are properly trained on the use of such equipment and that the information provided by the equipment is applied appropriately. The industry and individual utilities need to develop a means to share knowledge and information on the performance of pumps and valves. The NRC staff has found during its MOV inspections that some licensees were unaware of testing under way at other facilities that would provide useful information. The industry needs to continue to work to resolve generic problems. A good example of the cooperation between utilities has been the industry study of the accuracy of MOV diagnostic equipment. The staff has found some utilities to have strengths in certain areas of pump and valve performance, but weaknesses in others. The industry and individual utilities need to examine each of these areas with attention given to discovered weaknesses.

The NRC staff has a number of programmatic and inspection-related activities under way aimed at improving the performance of pumps and valves. For example, in addition to reviewing IST programs and relief requests, the staff issued GL 89-04 to provide guidance for licensees to improve their inservice testing programs. The staff issued GL 89-10 to request licensees to help ensure the proper performance of MOVs by reviewing MOV design bases; establishing proper MOV sizing and switch setting procedures; conducting tests of MOVs under design-basis conditions in situ where practicable; developing adequate periodic verification methods for MOV capability; and improving the analysis of MOV problems, corrective action, and trending. The staff prepared Temporary Instructions (TIs) for the performance of NRC inspections of licensee activities in response to GL 89-04 and GL 89-10. The staff also prepared a TI for the performance of NRC inspections of licensee activities involving check valves. The staff has initiated inspections in each of these areas and will discuss the results of those inspections during this symposium. On a more broad scale, the staff is studying the need to modify the NRC regulations with respect to inservice testing to correct the weaknesses in the current methods of inservice testing. The staff informed Mr. Forrest Rhodes of the ASME Operations and Maintenance Committee of its planned rulemaking and requested that ASME address the weaknesses in the current IST methods such that the staff could reference subsequent improvements to the ASME Operations and Maintenance Code. The staff is an active participant in the committees and working groups that are developing this Code. In addition, the staff participates regularly at the meeting of the MOV Users Group, the Nuclear Industry Check Valve Group, and the AOV Users Group. Recently, the NRC promulgated a regulation which provides for the improvement of maintenance of components such as pumps and valves. The staff is working with the industry to develop new standard technical specifications and to improve surveillance requirements in technical specifications. The staff is also studying the potential impact of Individual Plant Evaluations on surveillance testing. In addition to these staff activities, the NRC's Office of Nuclear Regulatory Research and Office for the Analysis and Evaluation of Operational Data have ongoing studies of pump and valve performance.

The proper performance of pumps and valves is critical to the safe operation of nuclear power plants. Operating experience and NRC inspections have revealed significant concerns regarding the performance of these components. The nuclear industry, individual licensees, and the NRC staff have a number of activities under way to improve the performance of pumps and valves. This symposium provides an important opportunity to strengthen those activities.

Session 1A

MOV Research and Technical Considerations

Session Chair
Kevin DeWall
Idaho National Engineering Laboratory

Recent Experience with Testing of Parallel Disc Gate Valves Under Accident Flow Conditions

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J. K. Clayton, Chief Engineer
Hopkinsons Limited

ABSTRACT

This paper presents the nuclear valve industry's latest and most extensive valve qualification test program experience. The test program includes a variety of 25 different gate and globe valves. All the test valves are power operated using either air, electric, or gas/hydraulic operators. The valves are categorized in size and pressure class so as to form a group of appropriate parent valve assemblies. Parent valve assembly qualification is used as the basis for qualification of candidate valve assemblies. The parent and candidate valve assemblies are representative of a nuclear plant's safety-related valve applications. The test program was performed in accordance with ANSI B16.41-1983 "Functional Qualification Requirements for Power Operated Active Valve Assemblies for Nuclear Power Plants." The focus of this paper is on functional valve qualification test experience and specifically flow interruption testing to Annex G of the aforementioned test standard. Results of the flow test are summarized, including the coefficient of friction for each of the gate type valves reported. Information on valve size, pressure class, and actuator are given for all valves in the program. Although all valves performed extremely well, only selected test data are presented. The effects of the speed of operation and the effects of different fluid flow rates as they relate to the coefficient of friction between the valve disc and seat are discussed. The variation in the coefficient of friction based on other variables in the thrust equation, namely, differential pressure area is cited.

INTRODUCTION

The experiences discussed resulted from valve testing performed for the Sizewell Nuclear Power Station. Sizewell 'B' is a pressurized water reactor plant in Sizewell, Suffolk, England scheduled for commercial operation in 1994.

Three valves will be discussed: a size 30 Class 600 with a gas/hydraulic operator, a size 3 Class 1500 with electric motor operator, and a size 2 Class 600 with air operator. All three are gate valves, and represent three of the five valves that were flow interruption tested. In addition to the gate valves, two globe valves were also flow interruption tested. The size 30 Class 600 valve is to be used as the plant's main steam isolation valve (MSIV). The valve is a parallel disc gate valve. The size 30 MSIV was flow interruption tested at Siemens KWU, in Germany. The size 3 Class 1500 electric motor operated valve (parallel disc gate valve) and the size 2 Class 600 air operated valve (solid wedge gate valve) were flow interruption tested at Nuclear Electric's Marchwood Engineering Laboratory (MEL) in the south of England. The balance of valves were also subcontracted for flow interruption testing at the Marchwood Facility. Some additional testing services were subcontracted to the National Engineering Laboratory in Scotland, which conducted pipe reaction end loading tests and exploratory vibration tests (modal analysis).

The electric motor actuators are manufactured by Autotork. These actuators have undergone complete IEEE-382, IEEE-344, and IEEE-323 environment qualification for use as safety-related valve actuators. The gas/hydraulic actuators are manufactured by Commercial Hydraulics Keelavite. The gas/hydraulic actuator utilizes pressurized nitrogen as a stored energy source to generate high thrust and a fast closure time. These actuators are presently undergoing IEEE qualification.

As a result of industry concerns with valve performance, particular attention must be given to valve design. Valve design and manufacturing controls will affect valve performance. Variations in valve size may affect the geometric alignment of functional parts and also affect valve performance.

Valve testing is best used to determine the general suitability of a particular design to perform its intended safety-related function. In doing so, testing (parent valve assemblies) confirms that additional generic (candidate valve assemblies) valves built to the same set of design standards will perform as previously demonstrated. In Figure 1, the size 12 Class 600 valve is a qualified parent valve assembly and the size 10 Class 600 is a candidate in the same generic family.

BACKGROUND

In June 1989 the U.S. Nuclear Regulatory Commission (NRC) issued Generic Letter 89-10. In the wake of its issuance licensees initiated programs to provide for the testing, inspection, and maintenance of motor operated valves (MOVs) to provide the necessary assurances that the MOVs will function when subject to design-basis events. As a result, utilities performed in situ and some accident testing to demonstrate valve operability under conditions that would simulate design-basis events, including pressure, temperature, fluid medium, and fluid flow rate. However, some design-basis events, such as high energy line breaks (HELB), are difficult to simulate. This problem is complicated because during an HELB, an isolation valve is used to prevent system blowdown. This isolation valve function represents one of the most severe valve closure scenarios. In a line break isolation function the valve is subjected to very high blowdown fluid flow rates, high differential pressure and often a two-phase flow disturbance.

As a result of these difficulties, many questions related to valve performance are being investigated. A method of ensuring operability of existing, installed valves under design-basis events is under investigation. In some cases it is inherently obvious that a particular valve in a given application is cause for concern. In most applications where design-basis events are not the most severe and valve performance has never been suspect, valves warrant less concern.

Regardless of the valve or application, predicting the performance of a valve based on an existing test requires the qualification of the candidate valve, by analysis, to demonstrate its relationship to the tested parent valve. For valves addressed in a utility test program, qualification generally refers to the ability to perform satisfactorily under flow interruption capability testing. The test program was done in accordance with ANSI B16.41 and includes all test requirements of that standard.

TEST REQUIREMENTS

The valve test program was performed in accordance with ANSI B16.41, which requires each of the following tests to demonstrate complete functional qualification:

Annex A—Valve Leakage Test

Annex B—Cold Cyclic Test

Annex C—Hot Cyclic Test

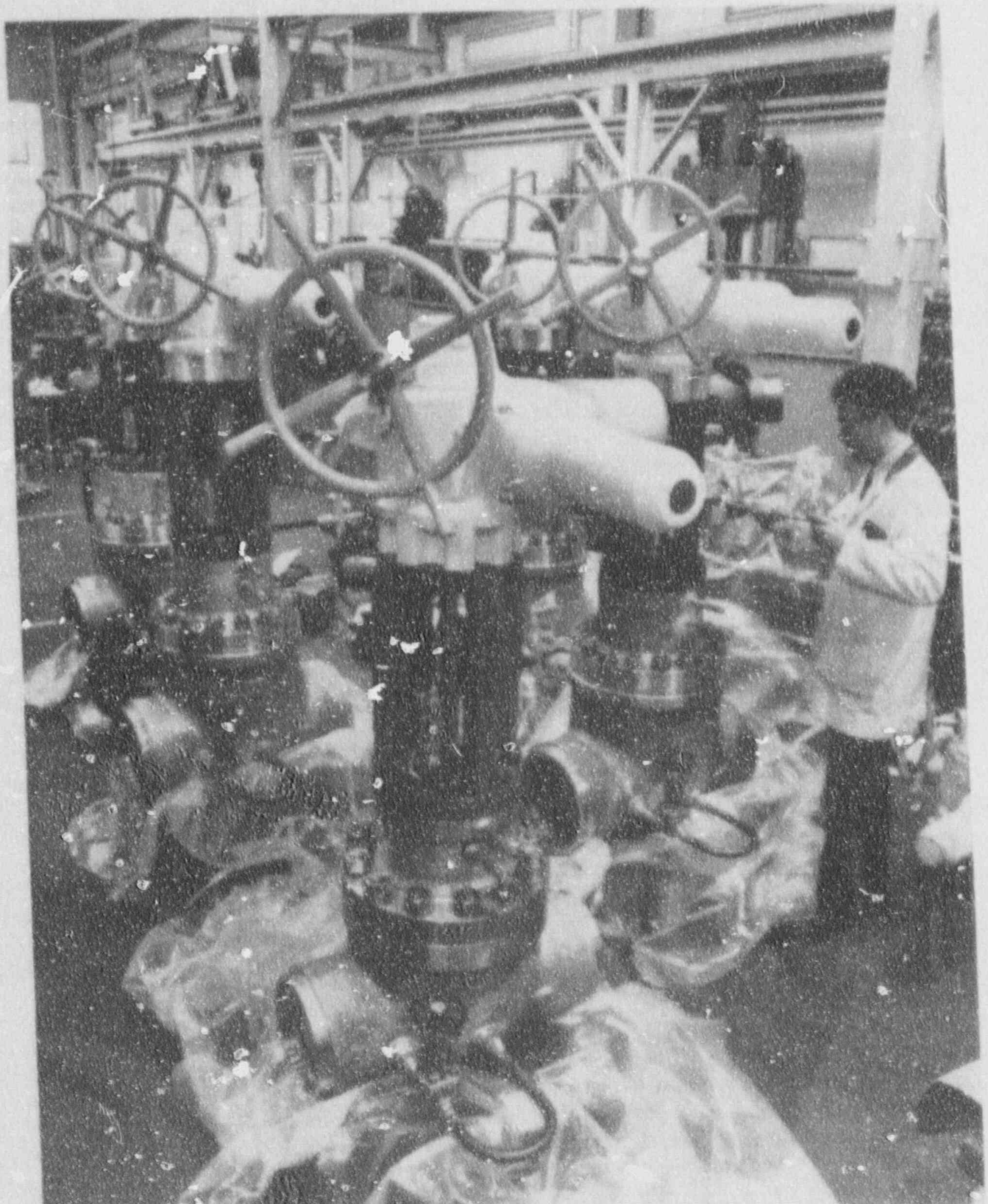

Figure 1. Size 10 and size 12 Class 600 electrically operated parallel disc valves for Sizewell 'B' Nuclear Power Station.

Annex D—Pipe Reaction End Loading Qualification Test

Annex E—Exploratory Vibration Test

Annex F—Seismic Loading Test

Annex G—Flow Interruption Capability Test

Annex H—Endurance Test

The test program included all of the above tests except the Annex H, Endurance Test. The Annex D, Pipe Reaction End Loading Qualification Test, and Annex F, Seismic Loading Test, were combined into one test with both loads applied simultaneously. The combination of Annex D and Annex F testing is very significant. The combined load test demonstrates operability under very severe pipe bending and upperstructure deflection. The Annex G, Flow Interruption Capability Test, were all full stroke tests; no partial closure tests were permitted even though allowed by ANSI B16.41. The Annex G test was performed only only valves that were required to close as a result of a pipe break.

TEST DESCRIPTION

Leakage Test

Seat, gland, and back seat leakage tests were performed to demonstrate the valves' sealing capability, and to establish baseline parameters.

Cold Cyclic Test

The valve is opened and closed under system pressure with normal, minimum, and maximum motive power to demonstrate the valves' cyclic capability. After valve closure, one side of the closure was depressurized to establish rated operating differential pressure in the most adverse direction and then the valve was opened. The test was conducted at room temperature using water as the fluid.

Hot Cyclic Test

The valve was opened and closed under system pressure and system temperature. The test was conducted with normal, minimum, and maximum motive power to demonstrate the valves' hot cyclic capability. After valve closure, one side of the closure was depressurized to establish rated operating differential pressure in the most adverse direction and then the valve was opened. On the final test run, the valve was closed under maximum motive power and allowed to cool to below 100°F. After the valve cooled, differential pressure was established in the most adverse direction, and then the valve was opened under minimum motive power. Refer to Figure 2.

Pipe Reaction and Seismic Load Test

The valve was opened and closed under system pressure with normal, minimum, and maximum motive power to demonstrate operability under maximum pipe reaction end-loading. With the valve open and operating pressure established, the test loading moment was applied to the valve. A closure cycle was then effected using minimum motive power. After valve closure, one side of the closure was depressurized to establish maximum closure differential pressure. With the valve closed and maximum pipe reaction end-loads and maximum closure differential pressure in place the seismic load test was superimposed. The addition of the seismic load test consisted of applying a static bending moment, simultaneously, to the valves' upperstructure equal to a seismic acceleration of 4.5g in the horizontal and the vertical plane. The resulting seismic load factor (SLF) was applied in the direction that caused the greatest deflection of the valves' upperstructure. This portion of the test demonstrated the valves' capability to operate during seismic and pipe end moment loadings. With all elements in place the valve was then opened under minimum motive power.

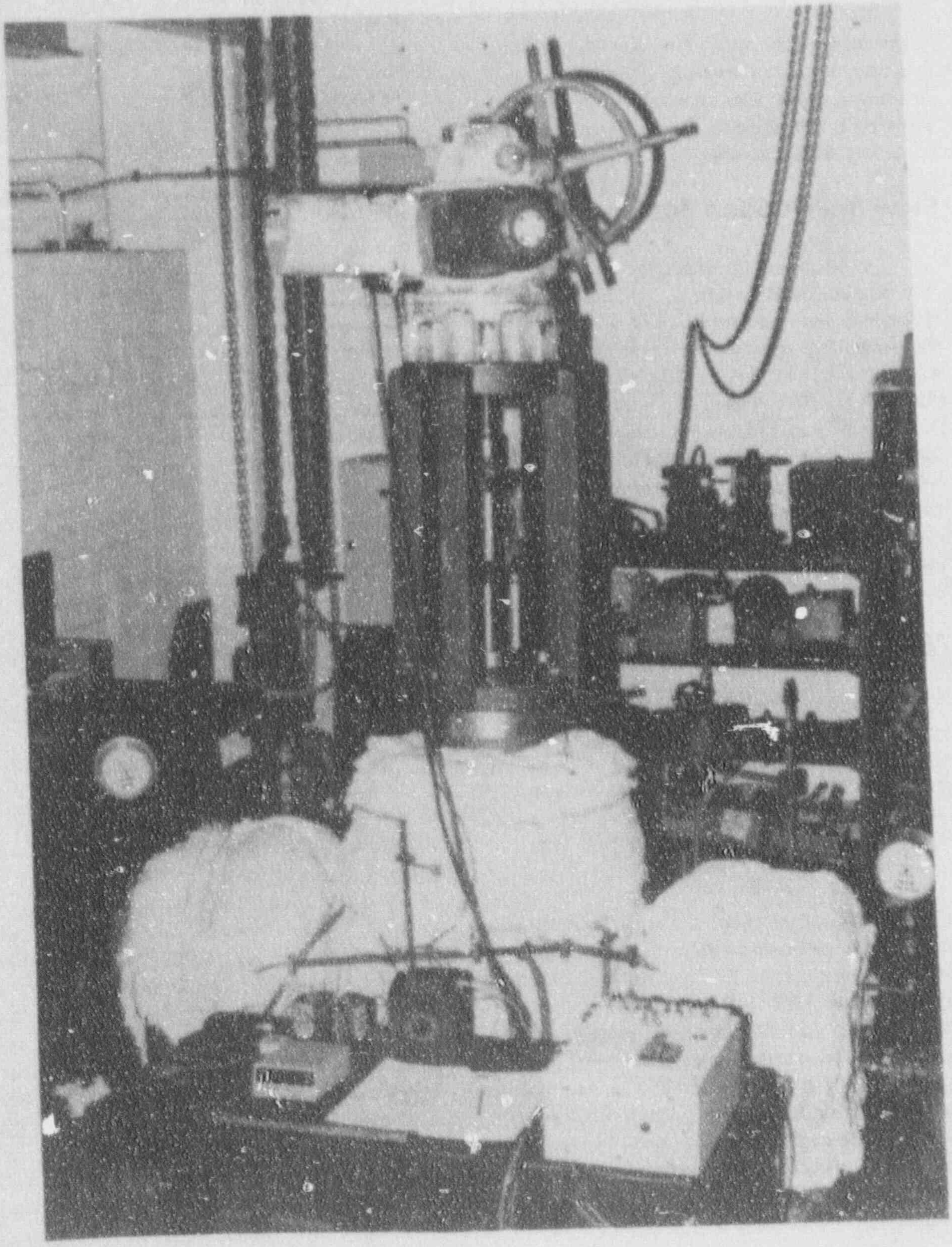

Figure 2. Size 12 Class 1642 parallel disc valve during ANSI B16.41 hot cyclic test.

Exploratory Vibration Test

The scope of the Annex E testing was extended to permit the use of alternative techniques, such as modal analysis. The exploratory vibration test was used to determine the fundamental frequency of the test valve assembly.

Flow Interruption Test

This test demonstrated the valves' capability to close against blowdown flow. No partial closure stroke tests were permitted. All flow interruption tests were full open to full closed stroke test. After initiation of the test fluid flow, the valve closure was effected using the minimum motive power qualification level for actuation. During conduct of flow interruption testing, the tested valves were instrumented with strain gauges to measure actual closing force. The actual closure force was used in the standard closure force equation to solve for a coefficient of friction based on testing.

TEST PROGRAM VALVES

The valves tested in this program are listed by type, size, and pressure class in Appendix A. A total of 25 valves were tested as part of the program. All valves were designed in accordance with ASME Section III Subsections NB, NC, and ND to the 1983 Edition with the Winter 1983 Addenda.

The largest valves in the test program were a size 14 Class 900 main feedwater isolation valve (MFIV), Figure 3; a size 30 Class 600 main steam isolation valve (MSIV), Figure 4; and a size 12 Class 600 and size 12 Class 1642 parallel disc gate valves. The design of these valves was such that they have floating discs and no thermal lock-up wedging loads when fully closed. These two factors mean that the design was ideally suited to be closed under hot conditions with maximum motive power and open under cooled conditions with minimum motive power. The floating discs take account of any thermal expansion. The cooling down of the discs cannot increase lock-up loads and cause binding.

VALVE DESIGN

Manufacturers' valve design standards and design practices will affect valve performance during testing. Geometric alignment of moving parts and parts along the disc closure path are of particular importance. Standards for design and tolerancing of the valve internals can ensure successful test results. One manufacturers' valve of a given size, pressure class, and type that meets a specific design code, like ASME Section III, will not necessarily perform the same as another manufacturers' valve. In fact, changes in design within one manufacturer can alter test results.

This later point raises a question of repeatability on the test valve, and repeatability of a valve "similar" to the test valve. The issue of repeatability of the test valve was addressed in the "Summary of Test Performance," where data are presented on six closures during flow interruption testing of the size 30 Class 600 MSIV. Repeatability between the test valve and a similar valve was derived through compliance with ANSI B16.41 on "Qualification of Parent and Candidate Valve Assemblies."

During the design phase great attention was placed on sizing actuators and designing internal components for parallel disc and wedge gate valve designs. For parallel disc gate valves, the disc was designed not to tilt during closure. The effects of flow-induced loads on the outlet, sealing discs were of great importance. The valve disc was designed not to tilt during closure, and run parallel and flat with the valve body seat. This design also provides for full guiding throughout the closure stroke. This point seemed to be of the utmost importance to valve performance.

All test valves were designed with stellite hard facing on the disc and the body seat.

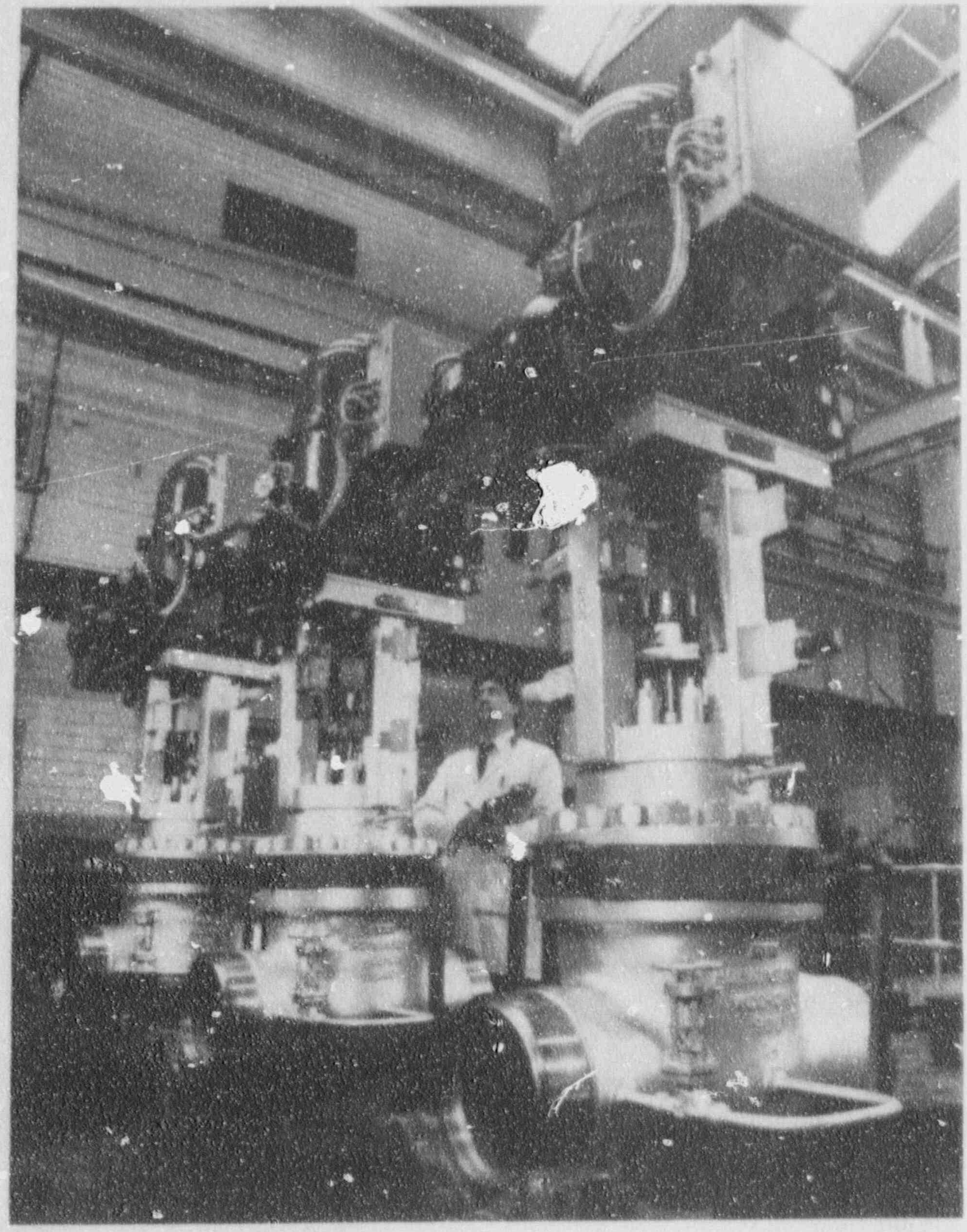

Figure 3. Size 14 Class 900 parallel disc main feedwater isolation valve at final inspection.

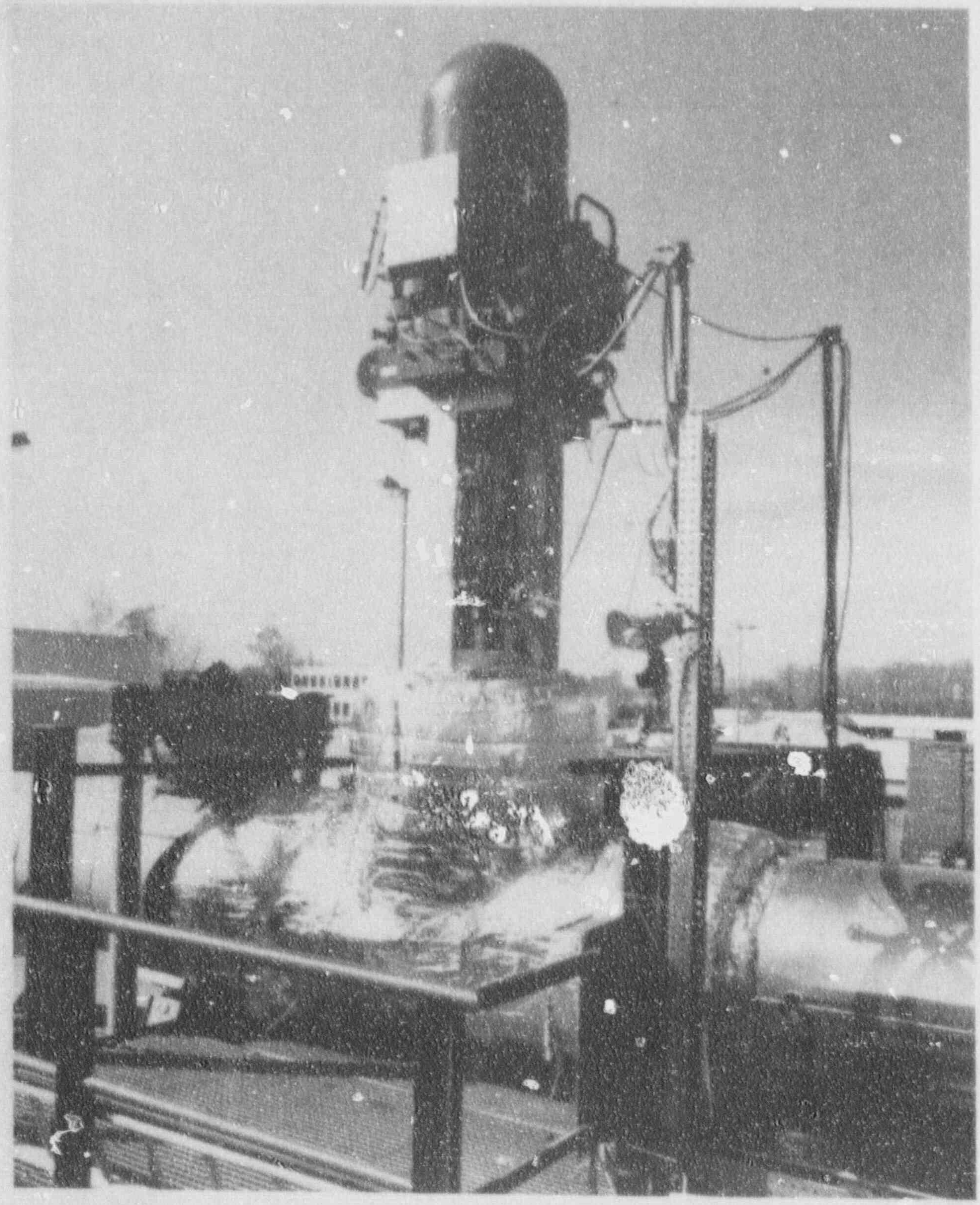

Figure 4. Size 30 Class 600 parallel disc main steam isolation valve in the large test fixture during ANSI B1641 flow interruption testing.

SUMMARY OF TEST PERFORMANCE

Leakage, Cold Cyclic, Hot Cyclic Tests

Leakage, cold cyclic, and hot cyclic tests were all performed in accordance with the ANSI Standard and yielded successful results. The basis for test acceptance was the maximum time to open and the maximum time to close the valve. Stroke times did not present a problem. Seat tightness and sealability were acceptable through each test.

Pipe Reaction and Seismic Load Test

Pipe reaction and seismic load tests were performed in accordance with the ANSI Standard and yielded successful results. Any resulting deflections at the seat location, as a result of pipe reaction and seismic loadings, did not affect the valves' performance characteristics.

Exploratory Vibration Test

Exploratory vibration tests were contracted to the National Engineering Laboratory at East Kilbride, UK. These tests were performed using the Modal Analysis Technique. The purpose of this testing was to determine the natural frequency of the valves' upperstructure. Each test valve was successfully tested with a natural frequency greater than 33 Hz, and was thereby determined rigid.

Flow Interruption Test

Flow interruption tests were contracted to Siemens KWU and the MEL.

At the writing of this paper four out of five of the gate valve flow interruption tests were complete. The size 4 Class 1500 wedge gate valve with an electric motor operator has not been tested. Flow interruption testing of this valve was scheduled for late May 1992. Testing will be conducted at an EDF facility in France. Of the four valves that have been tested, one valve has not been analyzed. The size 2 Class 600 wedge gate valve with air operator was tested, but the data have not yet been correlated from the test results. This valve passed all testing successfully, including the flow interruption test. The balance of three valves have been tested and fully analyzed. These valves include the size 30 Class 600 and size 3 Class 1500 parallel disc gate valves and the size 2 Class 1500 wedge gate valve. The size 30 valve was gas/hydraulic operated, the size 3 valve was electric motor operated and the size 2 valve was air operated. All flow interruption tests were performed successfully in accordance with ANSI B16.41.

During flow interruption testing the following average coefficient of friction between the valve disc and seat was measured for each of the test valves:

- Size 30 valve $\mu = 0.40$
- Size 3 valve $\mu = 0.33$
- Size 2 valve $\mu = 0.47$

The coefficient of friction was calculated based on actual maximum closure thrust as measured via strain gauge instrumentation. Packing loads were also measured which made it easy to correlate disc loads. An effective disc pressure area based on the disc outside diameter was used. The valve actuator sizing equation for determining maximum thrust requirement is:

$$F = A_D \cdot \Delta P \cdot \mu \pm A_S \cdot P + S$$

F = Total stem thrust, lbf

μ = Coefficient of friction between the disc and seat

A_D = Area of disc, in^2

A_S = Area of stem, in^2

P = Line pressure, lb/in^2

ΔP = Differential pressure across the disc, lb/in^2

S = Gland packing load, lbf.

The coefficient of friction, as calculated from the above equation, was dependent on several variables. The use of actual values for thrust, packing load, and pressure minimize inaccuracy. The coefficient of friction will, however, vary based on the effective disc area. A larger disc area (i.e., disc outside diameter) will yield a lower coefficient of friction and a smaller disc area (i.e., disc inside diameter) will yield a higher coefficient of friction in calculations from the same thrust value. Care should be taken to discuss the coefficient of friction in terms of a given pressure area. All the above quoted values of coefficient of friction are based on a disc area bounded by the disc outside diameter.

Size 30 Test Valve (MSIV)

Several flow interruption tests were performed on the size 30 Class 600 main steam isolation valve, as shown in Figure 4. The valve was closed with saturated steam and with a two-phase flow mixture. With the flow restrictor installed in the test fixture, four fast closures were achieved with two-phase flow. All four closures occurred in less than 3 seconds each. Actual closure times were between 2.6 to 2.9 seconds. The initial flow rate was approximately 11×10^6 lb/h increasing to approximately 26×10^6 lb/h for the two-phase flow mixture.

In addition to the two-phase flow test, two test runs were made with saturated steam as the fluid medium. One test run was done with the flow restrictor fitted and the other test run with the flow restrictor removed. The initial flow rate with the flow restrictor removed was 14×10^6 lb/h.

In conclusion, all flow interruption testing of the size 30 MSIV was successful. Four tests were conducted with two-phase fluid flow and two tests were conducted with saturated steam flow. All closure times were less than 3 seconds and were repeatable. Given the variation in fluid medium and flow rates, the calculated coefficient of friction based on actual measured thrust and disc outside diameter were virtually uneffected, $\mu = 0.39$ to 0.41 .

After flow interruption testing was completed, informational testing was conducted. These tests were conducted under hot conditions and with fluid flow. A total of two test runs were made in the closing direction over the final 3 in. of stroke. The first test run under slow closure conditions resulted in a coefficient of friction, $\mu = 0.20$, based on disc outside diameter. The second test run under fast closure conditions resulted in a coefficient of friction, $\mu = 0.40$, based on disc outside diameter. These two test runs appear to indicate that speed of operation has an effect on the value of the coefficient of friction. It was interesting to note that prior to flow interruption testing, the valve was slowly opened with a ΔP across the disc. Data from the test run showed the disc coefficient of friction, $\mu = 0.30$, based on disc outside diameter.

Size 3 Test Valve

Flow interruption testing of the size 3 Class 1500 valve was completed in accordance with ANSI B16.41 and yielded successful results. This valve was shown in Figure 5 during leakage testing, after completion of the flow interruption test. The seat leakage test demonstrated tight sealing even after severe testing. The calculated coefficient of friction was $\mu = 0.33$, based on the disc outside diameter. The lower coefficient of friction on this valve was particularly interesting as a result of the valves' speed of operation. The valves' closure time was 25 seconds, which constitutes a slow closure relative to high speed closing valves like the MSIV that closes in less than 3 seconds and was nominally 10 times as large. This testing would seem to indicate an important relationship between coefficient of friction and closure speed. Note that when the MSIV was closed on a partial stroke and opened at slower speeds, the coefficient of friction was $\mu = 0.20$ and $\mu = 0.30$, respectively. But when the MSIV was closed fast on a partial stroke the coefficient of friction was $\mu = 0.40$. A fast speed of operation yielded a coefficient of friction $\mu = 0.40$ for the MSIV at all tested flow rates.

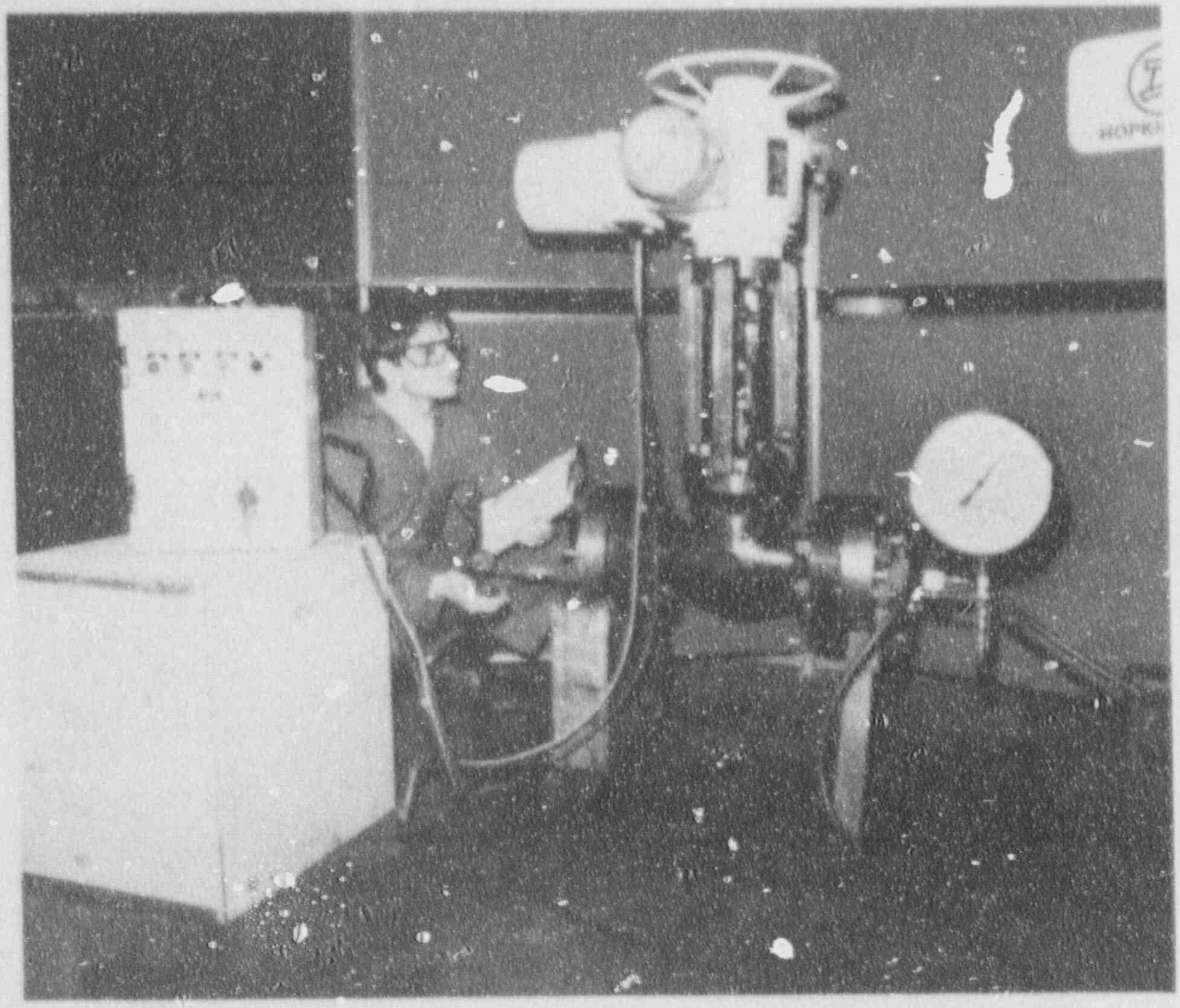

Figure 5. Size 3 Class 1500 parallel disc valve during ANSI B16.41 leakage test after completion of flow interruption testing.

The coefficient of friction was calculated based on actual closure thrust as measured via strain gauge instrumentation, the actual packing friction load as measured prior to initiation of pressure and fluid flow and an effective disc pressure area based on the disc outside diameter. During testing to determine the actual packing friction load, it was noted that the valve internal pressure did not effect the measured load. This raises a potentially interesting point that packing friction load may be independent of internal pressure. As a minimum, this area requires more testing and investigation. It also raises the question of speed of operation as it relates to measuring actual packing friction loads. Unfortunately no data have been established in this area; however, each actual packing friction load was measured at the test speed of operation.

Based on the actual data above, the coefficient of friction has been calculated for information and comparison using different effective pressure areas in the thrust equation. Based on the disc outside diameter, the coefficient of friction was μ = 0.33. Based on the disc/seat mean diameter, the coefficient of friction was μ = 0.49.

Refer to Figures 6 and 7 for additional performance data. Valve position vs. closing thrust and

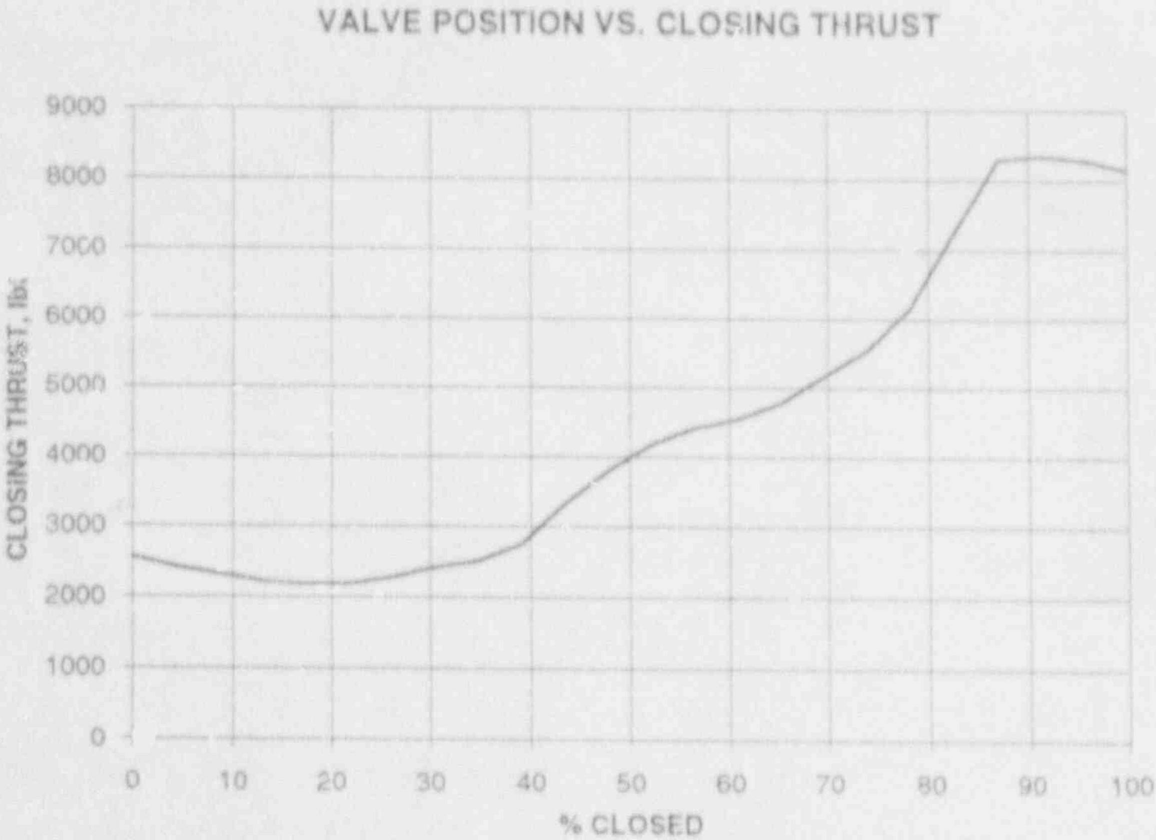

Figure 6. Size 3 Class 1500 parallel disc valve, valve position vs. closing thru.. test trace.

differential pressure are presented. Note the peak in closing thrust for the 3-in. valve. Maximum closing thrust was established before the end of stroke, at flow isolation on position seated valves, where no wedging act..n was present.

Size 2 Test Valve

Flow interruption of the size 2 Class 1500 valve was completed in accordance with ANSI B16.., and yielded successful results. The valve closure time was approximately 3 seconds. Again, while measuring the actual packing friction load, it was noted that the load was seemingly independent of internal pressure. The calculated coefficient of friction was $\mu = 0.47$, b..ed on the disc outside diameter, and $\mu = 0.61$, ..sed on the disc/seat mean diameter. These coeffic... of friction values are based on th.. ..ximum thrust required to isolate fluid flow. Note that these coefficients of friction do not consider the unbalanced vertical forces acting on the downstream disc that contribute to the maximum stem force. The final closure thrust, after the electric motor operator torque switch trips, was greater than the maximum thrust required to isolate fluid flow. The final closure thrust occurs after the disc wedges. This thrust was more a resul.. of actuator output or the actuator output margin, as set, above the maximum thrust requi..ed to isolate flow. After flow isolation occ..., the measured thrust increases very quickly just prior to actuator trip. This final closure thrust, when used in the thrust equation, will result in an increase in the coefficien.. of friction. Care should be taken in determining the appropriate thrust value for calc..ating the coefficient of friction.

SUMMARY OF FINDINGS

In conclusion, several major issues have been raised, and some of these issues warrant more investigation. It was hoped that the issues raised in this paper and summarized below bring additional information to the reader, and that this information will be helpful in the future.

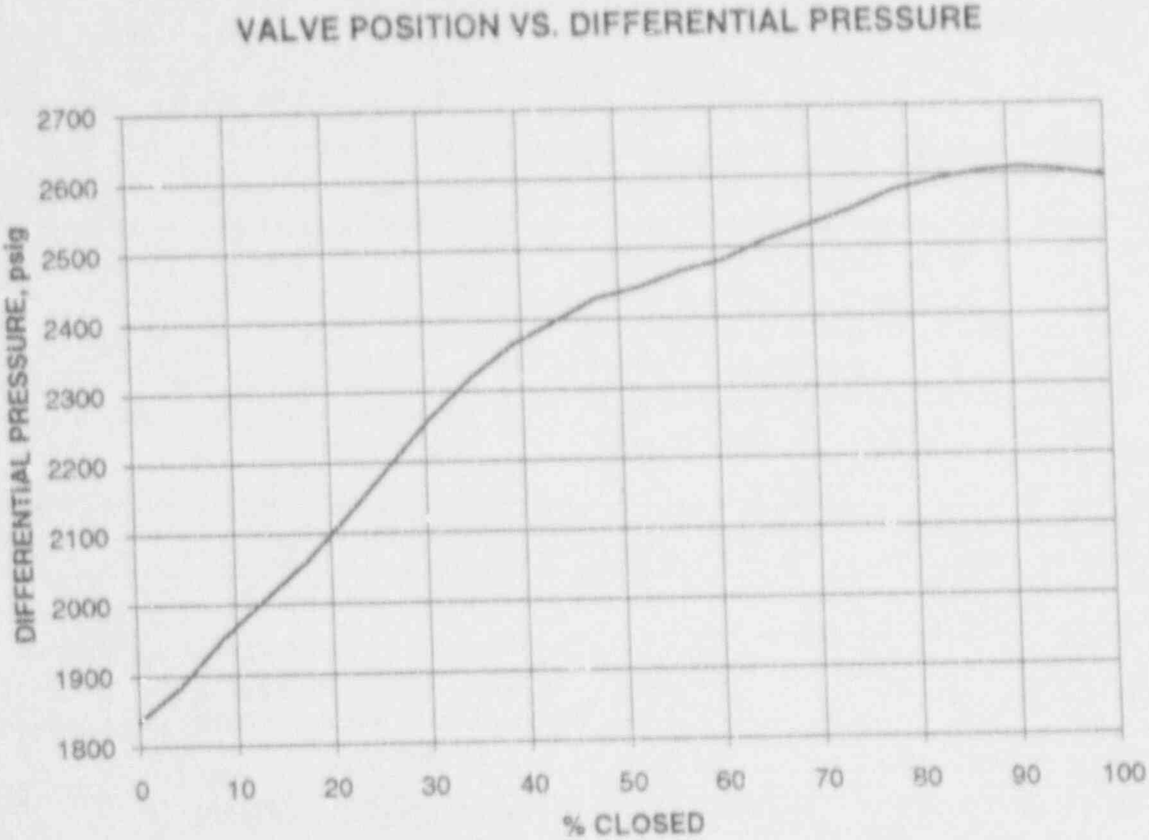

Figure 7. Size 3 Class 1500 parallel disc valve, valve position vs. differential pressure test trace.

There are three primary findings at this time:

1. The speed of operation, that was to say the stem travel rate in inches/minute, will effect the coefficient of friction. As the stem travel rate increases, the coefficient of friction increases. As the stem travel rate decreases, the coefficient of friction decreases. This area absolutely warrants more investigation to establish additional data and to determine more accurate guidelines on what constitutes a fast or slow closure. Initially, it seems that there was a definitive distinction between fast and slow closure, but there was a significant undefined region between the two closure speeds. In terms of rough numbers, a stem travel rate ≤ 20 in/min would be considered a slow closure from a flow interruption testing standpoint. A stem travel rate ≥ 40 in/min would tend to be considered a fast closure from a flow interruption testing standpoint.

2. The effects of the fluid flow rate were independent of the coefficient of friction, μ. In fact, the presence or absence of fluid flow did not vary the measured coefficient of friction. This area has the potential for very significant ramifications and absolutely warrants more investigation to establish additional data and to determine more accurate guidelines on the effects of fluid flow. This issue relates only to the effect of fluid flow when determining a seating or closing surface contact coefficient of friction. It does not relate to a test valves' ability to close against and isolate a given fluid flow.

   It should be noted that all closures were performed with the internal valve parts, including the valve seats, wetted.

3. The valve design relationship, that was to say the relationship of the test valve to other valves that will be evaluated based on test valve performance, is important. A viable

relationship must be established between a parent test valve assembly and a candidate valve assembly to insure their similarity.

Two secondary findings should be noted:

1. The effective disc pressure area should be related to the coefficient of friction. When using the thrust equation, a larger disc area, A_D, will yield a lower coefficient of friction and a smaller disc area will yield a larger coefficient of friction. Care should be taken to always discuss coefficient of friction in terms of effective pressure area.

2. Stem packing friction loads as used in the thrust equations should be based on actual measured values whenever possible. Although the differential pressure loads and unbalanced stem pressure loads are much more significant as they relate to thrust requirements, investigation into packing friction loads are warranted to establish the affect of pressure and speed of operation.

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The authors wish to thank the staff members at Atwood & Morrill Company and Hopkinsons Limited for their help in the preparation of this paper.

Appendix A

ANSI B16.41 Functional Qualification Parent Valves Tested for the Sizewell 'B' Nuclear Station

Valve Type	Size	Pressure Class	Actuator Type	Notes
Parallel Gate	30	600	Gas/Hydraulic	(1)
Parallel Gate	14	900 (950)	Gas/Hydraulic	
Parallel Gate	12	1500 (1642)	Electric	
Parallel Gate	12	600	Electric	
Parallel Gate	12	150	Electric	
Parallel Gate	6	1500 (1642)	Gas/Hydraulic	
Parallel Gate	6	900 (950)	Electric	
Parallel Gate	4	300	Air	
Parallel Gate	3	1500	Electric	(1)
Parallel Gate	3	900	Air	
Wedge Gate	12	150	Electric	
Wedge Gate	4	1500	Electric	(1)
Wedge Gate	2	1500 (2150)	Air	(1)
Wedge Gate	4	150	Electric	
Wedge Gate	2	600	Air	(1)
Wedge Gate	2	600	Electric	
Globe	6	150	Air	
Globe	4	900	Air	(1)
Globe	4	150	Electric	
Globe	3	900	Air	(1)
Globe	3	150	Air	
Globe	2	1500 (2150)	Electric	
Globe	2	600	Electric	
Globe	1	1500 (2150)	Air	
Globe	1	600	Air	

TOTAL: 25

1. Only valves required to isolate flow during a design-basis event will be Annex G flow interruption tested.

Load Sensitive Behavior in Motor-Operated Valves

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Idaho National Engineering Laboratory[a]
EG&G Idaho, Inc.

ABSTRACT

Motor-operated valves (MOVs) that produce less thrust under flow and pressure loadings than under low-load conditions may be experiencing MOV load-sensitive behavior, a phenomenon more commonly known as the rate-of-loading effect. When this phenomenon was first observed, it was thought to be produced by the response of the motor operator internals to the rate at which the operator was loaded. Recent testing at the Idaho National Engineering Laboratory (INEL) has shown that, while the response of operator internals might have a small effect, the first-order cause of this phenomenon is external to the operator. We have found that the stem factor, a measure of the conversion of torque to thrust in the MOV stem/stem-nut interface, changes depending on how the valve is loaded. If, during the running portion of the stroke, the valve is subjected to loads high enough to squeeze the grease out of the contact zone in the stem/stem-nut interface, the losses to friction at that interface will be higher at torque switch trip than they would be if the running load were small. These losses to friction reduce the efficiency of the conversion of torque to thrust. This loss of efficiency, represented as an increase in the stem factor, is important in determining the available performance margin in a given MOV. (Margin is the difference between what is required to perform and how much is available to perform.) This paper documents our analysis of MOV load-sensitive behavior, or the rate-of-loading effect.

INTRODUCTION

The Idaho National Engineering Laboratory (INEL), under the sponsorship of the U.S. Nuclear Regulatory Commission (USNRC), is performing research on the functional behavior of motor-operated valves. The results of full-scale testing at design basis conditions are documented in DeWall et al., 1989 and Steele et al., 1990. That testing identified two areas of concern: (a) valve opening and closing forces were not conservatively modeled by the industry's stem force equation (because of variations in the disc factor), and (b) the conversion of motor operator torque to stem force was not a constant, as had been assumed in the operator sizing equations (variations in the stem factor).

We have done a great deal of work on the disc factor concern. The most recent report documenting that work is Steele et al., (1991). This paper presents our most recent findings on the stem factor concern.

Recent regulatory initiatives have recommended in situ or prototypical testing to determine if a valve will perform its design basis function. In many of these tests, the measured

a. Work supported by the U.S. Nuclear Regulatory Commission, Office of Nuclear Regulatory Research, under DOE Idaho Field Office Contract DE-AC07-ID01570. G. H. Weidenhamer, NRC Program Manager. Computer-generated graphics by Geraldine S. Reilly; technical editing by Donovan Bramwell.

[illegible] significantly less than that [illegible] ing static or no-load test. This [illegible] non is what the industry has called the [illegible] ading effect. If the in situ or prototypical [illegible] was performed at design basis conditions and the valve seated with a lower stem thrust than obtained in the static test, the problem is less significant than if the valve failed to fully seat under the high load. However, some valves cannot be design basis tested, and that too presents a significant problem. These valves fall into two categories: those that can be tested, albeit at lower loads than their design basis loads, and those for which testing is not practical. Resolving the so-called rate-of-loading question will help to solve these significant issues.

The stem factor problem was observed in our first full-scale test program for the motor-operated gate valve (Figure 1), where a 6-in. Velan[b] valve closed successfully against differential pressures of 600 and 1000 psid, with stem thrusts of 18,100 and 18,600 lb at torque switch trip. However, the 1400-psid test achieved a thrust of only 16,500 at torque switch trip and did not fully close. The only difference between these tests was the differential pressure load. All operator parameters (torque switch setting, etc.) were the same. In each case, the test was the first loaded stroke run after several unloaded strokes,

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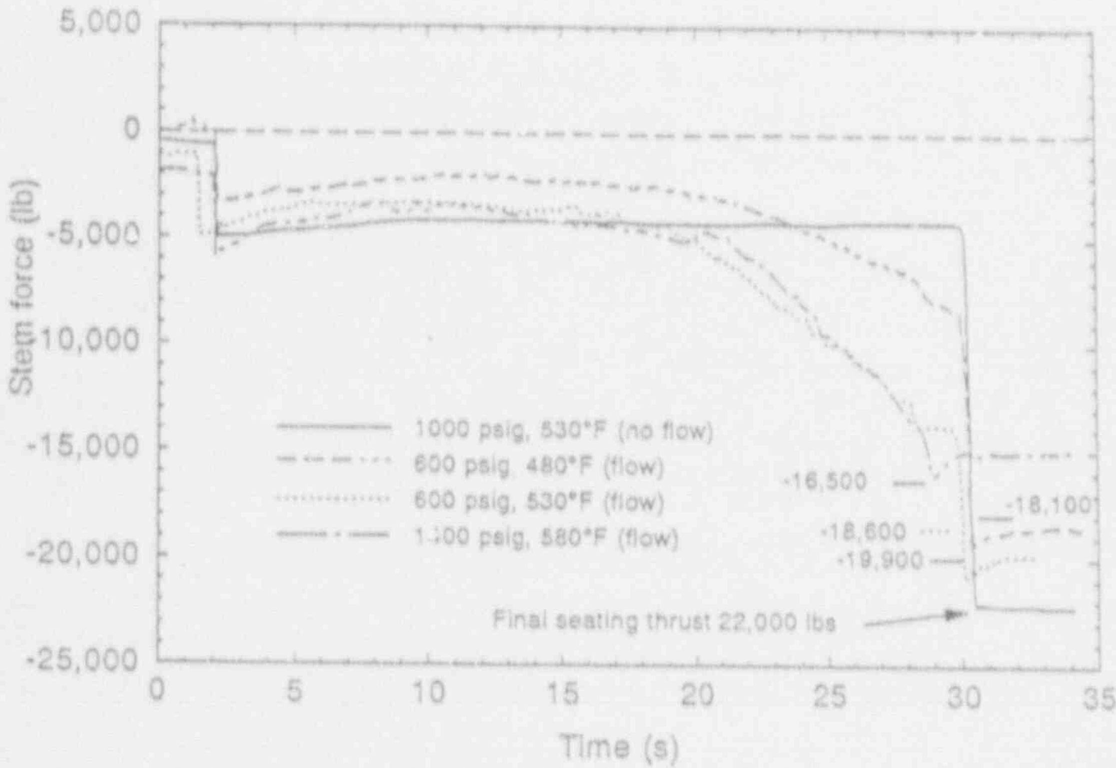

Figure 1. Stem thrust is affected by load history in the stem.

ensuring a well-lubricated stem. In the 1000 psid case, fresh lubricant (Lubriplate 930 AA) had been applied.

In our second full-scale test program, a 6-in. Anchor/Darling valve achieved 29,000 lb of measured thrust in static and lightly loaded tests but failed to close in an 800-psid test. The thrust at torque switch trip was 23,000 lb. Again the Anchor/Darling stem was well lubricated with Nebula EP1 grease. EP1 is Limitorgue's first choice for stem lubrication. The stems and the stem nuts for both the Anchor/Darling and the Velan valve were inspected prior to and after testing. There were no mechanical problems with either.

We began to get a better understanding of the cause of this phenomenon when we conducted tests on the INEL motor-operated valve load simulator (MOVLS) (Figure 2). We originally built the MOVLS as a simple device upon which we could perfect our measurement techniques and practice operating the data acquisition system in preparation for our Phase II full-scale gate valve test program (reported in Steele et al., 1990). After the Phase II test program, and during our analysis of the disc factor concern, we modified the MOVLS and used it to study the load-sensitive behavior observed in MOVs in the full-scale tests. The MOVLS provides realistic loads for this kind of study, and testing on the MOVLS is much more economical than full-scale valve testing.

The MOVLS is a laboratory tool built from a Limitorque motor operator and a Velan valve yoke bolted to a simulated valve bonnet. The valve stems and stem nuts used to date have been provided by nuclear valve suppliers. The valve stem is equipped with an instrumented arm that simultaneously measures torque and serves as an anti-rotation device. At the end of the valve stem is a thrust bearing mounted on a specially designed Lebow load cell that measures stem force. The load cell is attached to a hydraulic cylinder that discharges fluid to an accumulator as it is compressed during simulated valve closure. Different but typical valve stem load profiles are simulated with different initial amounts of water and gas overpressure in the accumulator. Comparing Figure 1 and Figure 4 shows how well the simulation loadings compare with full-scale differential pressure test loads.

All MOVLS measurements are recorded on a data acquisition system (DAS) consisting of a MEGADAC 2200C interfaced to an IBM System/2 personal computer. The MEGADAC is a high-speed data acquisition, signal-conditioning, data-recording system capable of a continuous sampling rate of 20,000 samples per second. The MEGADAC provides amplification, multiplexing, and analog-to-digital conversion of up to 128 channels of differential input. During the testing reported in this paper, all channels used were sampled at a rate of 1000 samples per second per channel. The IBM personal computer was used to control the MEGADAC, process the test data, and analyze the data for both quick-look plots and final plots. The MOVLS instrumentation and end-to-end accuracies (including the data acquisition system) are shown in Table 1.

INITIAL TESTS

We set up the MOVLS motor operator to simulate full-scale design basis testing of a 6-in. valve in a reactor water cleanup (RWCU) system. We set the torque switch in the SMB-0-25 motor operator at 2 and left it there for all 17 tests. The results of Tests 8 through 17 are shown in Figures 3 through 5. (Tests 1 through 7 were performed as setup tests.) These X-Y plots show stem thrust plotted against time. Each test was approximately 7 seconds in duration. For comparison purposes, several traces are shown on each plot. Tests 8 through 10 were performed at the same loading to demonstrate the repeatability of the results. The tests were set up to simulate closing a valve against a packing load and a 1000-psi pressure load without flow. (We refer to such tests as static tests.) Test 11 simulated moderate flow, and Tests 12 through 14 simulated design basis pipe break flow (Figure 4). Tests 13 and 14 were repeats of Test 12, with no changes in the load or the torque switch setting. In Tests 15 through 17

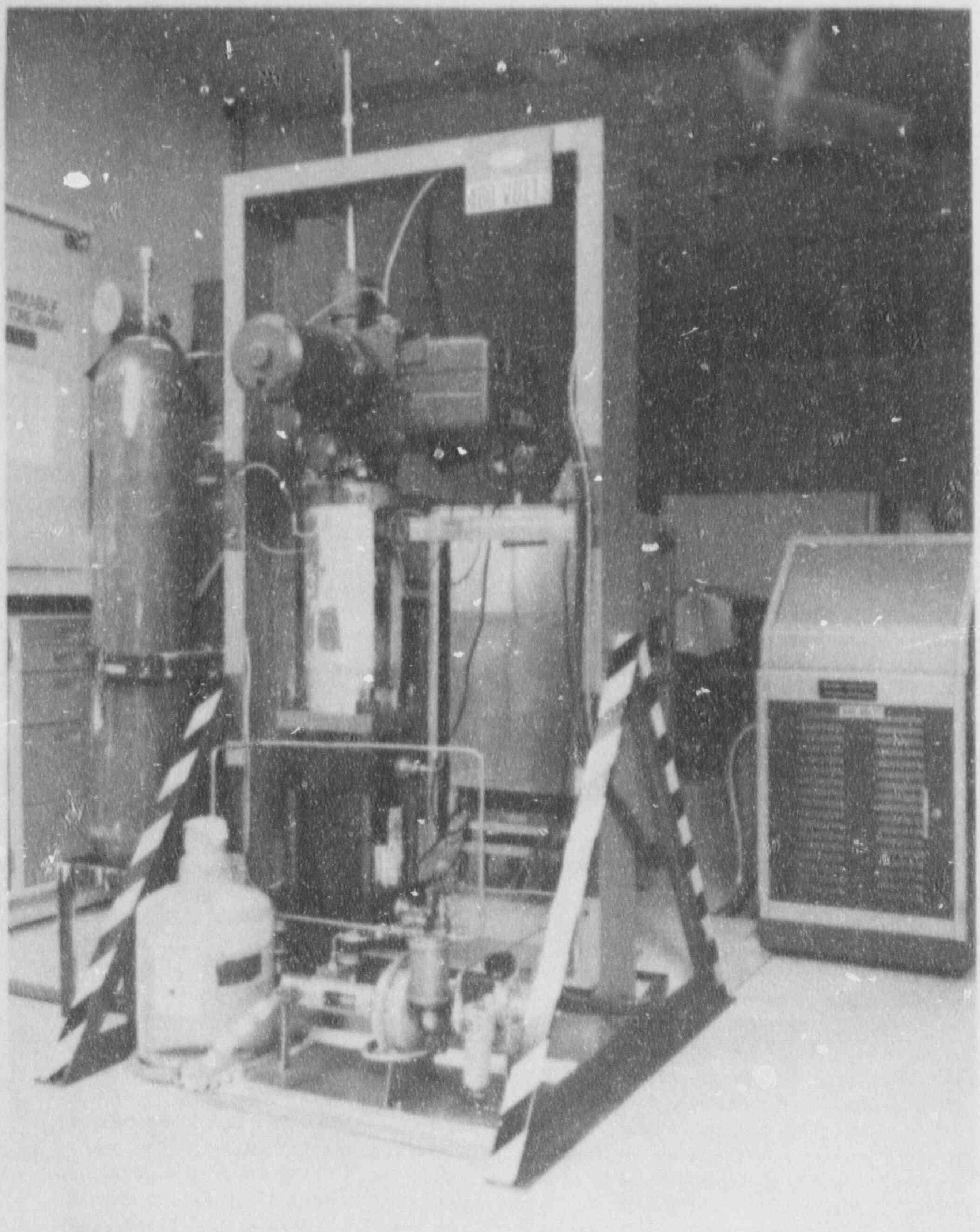

Figure 2. INEL motor-operated valve load simulator (MOVLS).

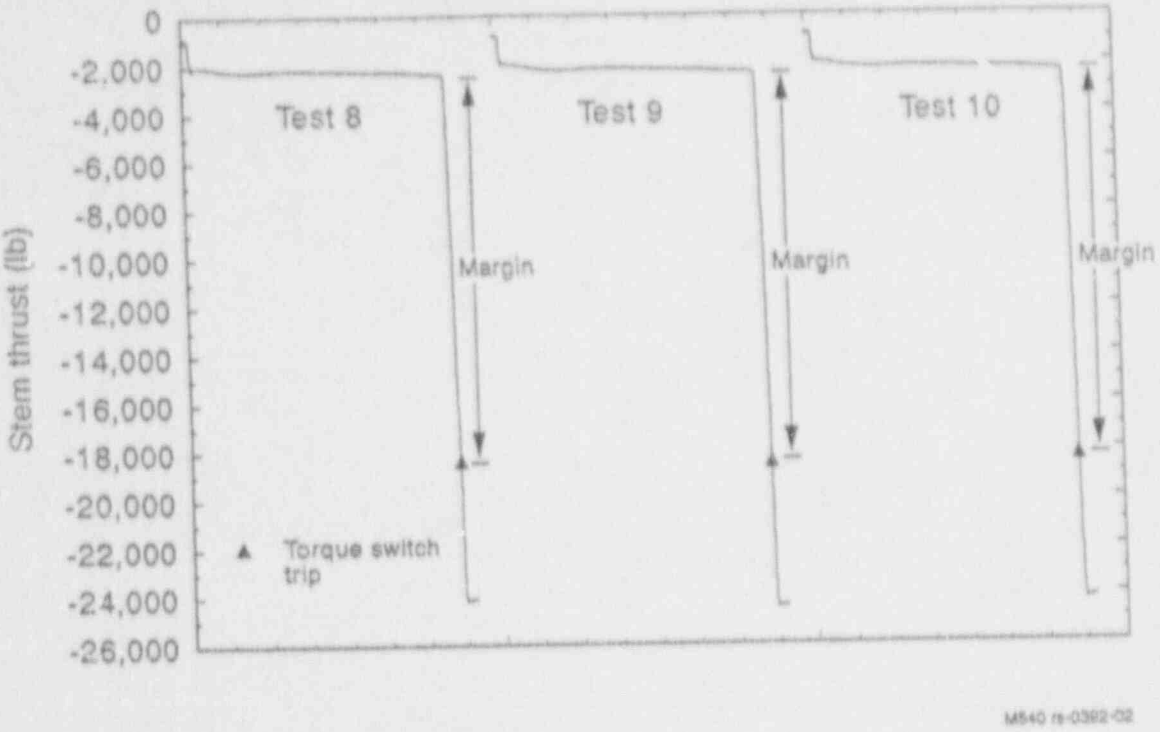

Figure 3. Lightly loaded stem thrust histories.

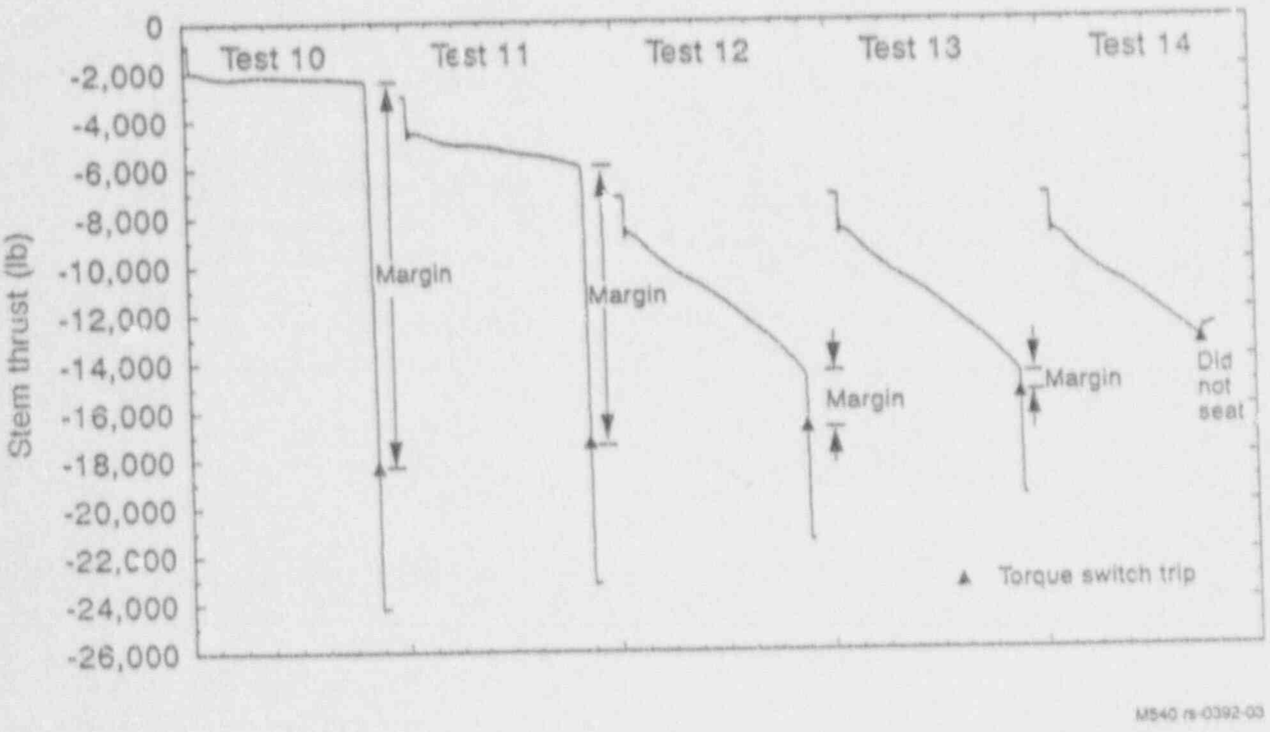

Figure 4. Stem thrust is affected by load history in the stem.

A possible cause outside of the operator is a change in the stem factor because of changes in the stem/stem-nut friction. The locations of the potential causes are shown in Figure 6.

Inside The Operator Analyses

A motor operator acts much like a planetary gear mechanism, as shown in Figure 7. The input motion is split between two output paths: motion of the stem nut and motion of the spring pack. There are no clutches. Each revolution of the motor prior to motor current cutoff must go into either turning the stem nut or compressing the spring pack. The motion takes the path of least resistance.

When the motor operator is lightly loaded, the input motion is transferred to the stem nut, and thus to the stem, until seating occurs. At that point, motion in the stem nut and the stem virtually ceases, and almost all the input motion is transferred to the spring pack. This transition from one motion path to another is very rapid in a lightly loaded valve. The worm then climbs the virtually static worm gear and compresses the spring pack. The operator goes through the torque switch trip point very quickly, allowing the motor controller dropout time to produce the maximum possible overcompression of the spring pack after torque switch trip. The operator momentum after motor controller dropout dissipates with further compression in the spring pack. Of course, overcompression of the spring pack produces additional output torque, which, in turn, produces additional stem thrust. A nuclear service, seismically qualified valve is a very stiff assembly when it is seated, and the stiffness of the valve assembly enhances the achievable thrust. Any friction in the worm-to-worm-shaft interface will reduce the force compressing the spring pack and increase the force producing the output torque. The effect, if any, of most of these phenomena would be to produce higher final stem thrusts in a valve when it is closed against a relatively light running load and a sudden seating load. Theoretically, any or all of these phenomena could contribute to the load-sensitive behavior.

When the motor operator is heavily loaded during the running portion of the closing stroke, the input motion is transferred exclusively to the stem nut only until the stem forces generate torque levels high enough to overcome spring pack preload. At that point, the motion is split between the two motion paths. As stem forces continue to increase because of valve flow and differential pressure loads, motion continues both in the worm gear and in spring pack compression through torque switch trip, unless seating occurs first. If torque switch trip occurs without hard seating, motion continues in both motion paths through contactor dropout and until the momentum is dissipated. If the valve is not fully seated, the assembly is not stiff; some of the input motion produces motion in the valve stem instead of compression in the stem. Thus, much of the input work is consumed in valve friction and stem\stem-nut friction loads. These phenomena may cause the resulting stem thrusts to be not as high as in the lightly loaded case.

Outside the Operator Analyses

One would expect the friction in the stem/stem nut interface to be higher with higher loads. However, if this friction is the cause of load-sensitive behavior in MOVs, one would have to explain why the friction is higher with a high running load than with a seating load of the same magnitude.

EXPERIMENTAL LOOK

The MOVLS is instrumented for a large number of measurements. Three of those measurements (spring pack force, stem torque, and stem thrust) proved very useful in isolating the first-order cause of load-sensitive MOV behavior. First, we calculated the ratio of spring pack force to stem torque for the tests shown in Figures 3 through 5. This ratio is proportional to the ratio of input torque to output torque for the operator, since the spring pack force times the effective worm gear radius (a constant) is the input torque. Figure 8 shows these relationships. This ratio, shown in Figures 9 through 11 did not vary significantly during the tests. This result basically eliminates the list (above) of potential causes inside the operator. Had any of those possible

A possible cause outside of the operator is a change in the stem factor because of changes in the stem/stem-nut friction. The locations of the potential causes are shown in Figure 6.

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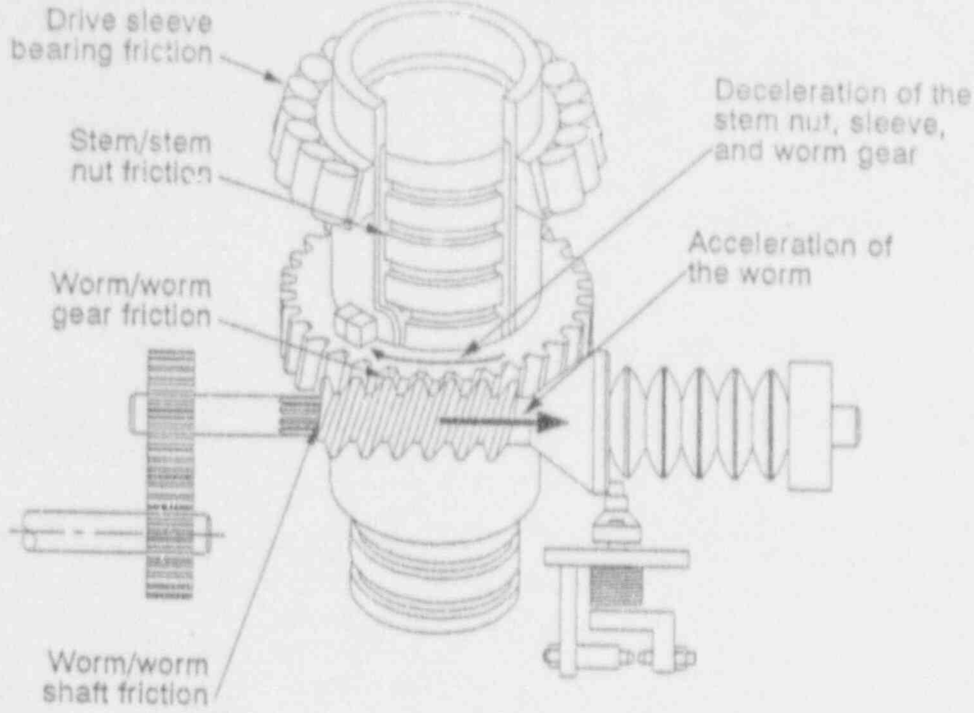
One would expect the friction in the stem/stem nut interface to be higher with higher loads. However, if this friction is the cause of load-sensitive behavior in MOVs, one would have to explain why the friction is higher with a high running load than with a seating load of the same magnitude.

EXPERIMENTAL LOOK

The MOVLS is instrumented for a large number of measurements. Three of those measurements (spring pack force, stem torque, and stem thrust) proved very useful in isolating the first-order cause of load-sensitive MOV behavior. First, we calculated the ratio of spring pack force to stem torque for the tests shown in Figures 3 through 5. This ratio is proportional to the ratio of input torque to output torque for the operator, since the spring pack force times the effective worm gear radius (a constant) is the input torque. Figure 8 shows these relationships. This ratio, shown in Figures 9 through 11 did not vary significantly during the tests. This result basically eliminates the list (above) of potential causes inside the operator. Had any of those possible



M540 rs-0392-06

Figure 6. Locations of potential load-sensitive behavior causes.

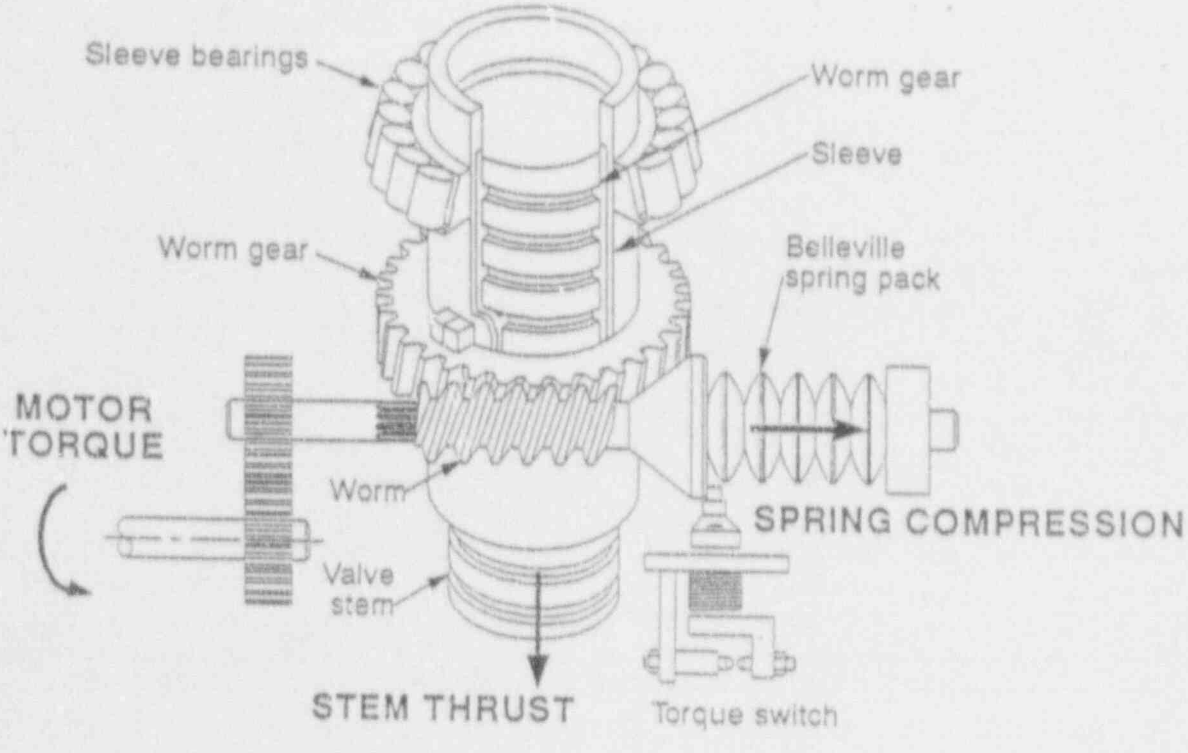


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Figure 7. Key components of motor operator and input-output paths.

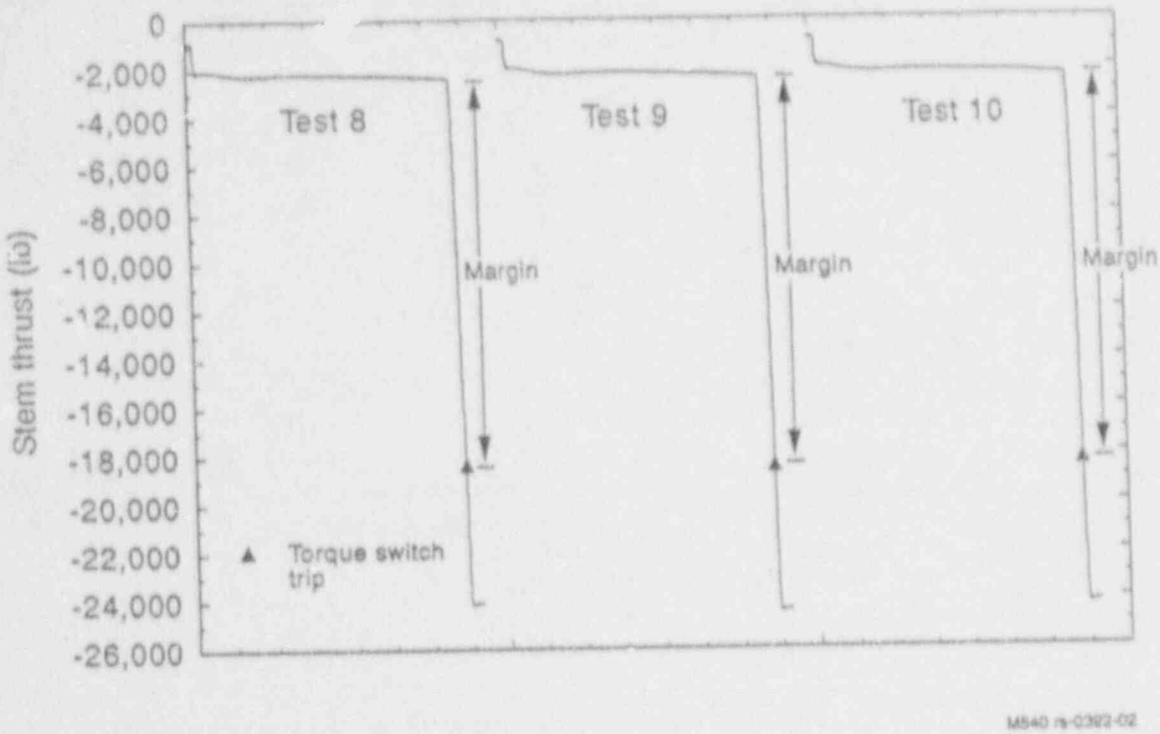

Figure 3. Lightly loaded stem thrust histories.

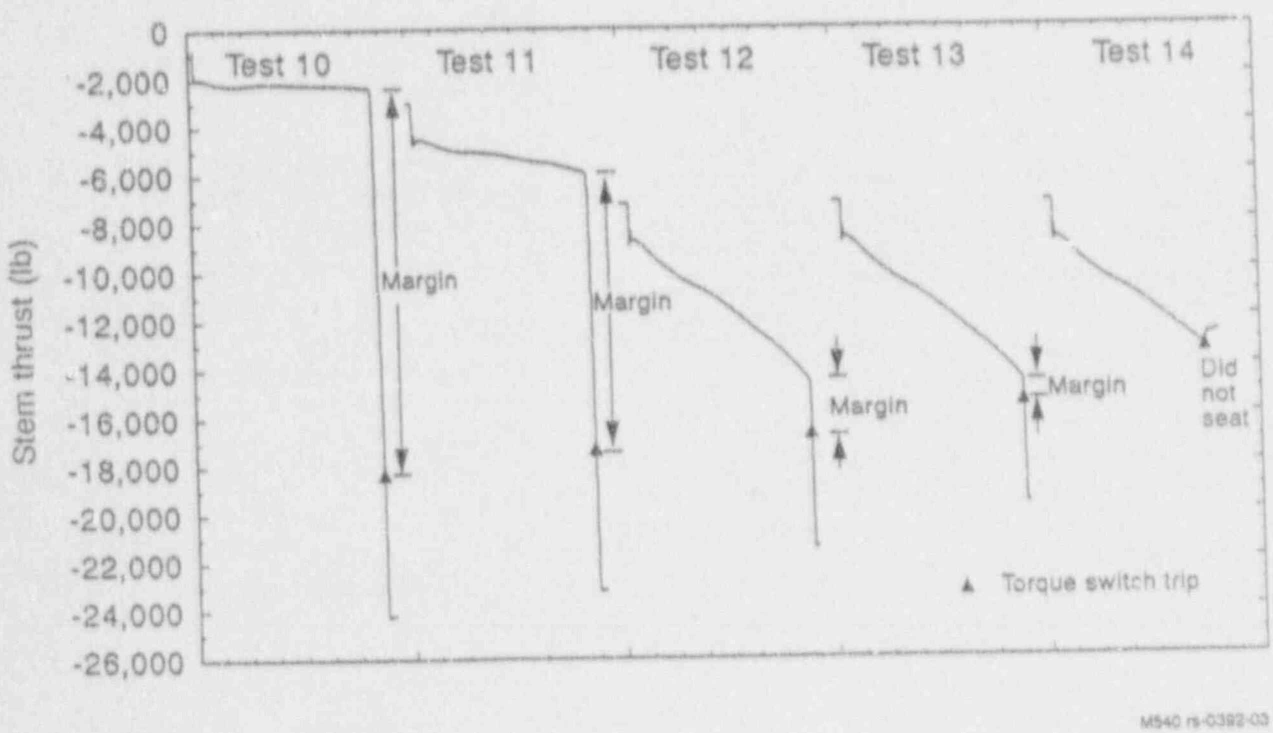

Figure 4. Stem thrust is affected by load history in the stem.

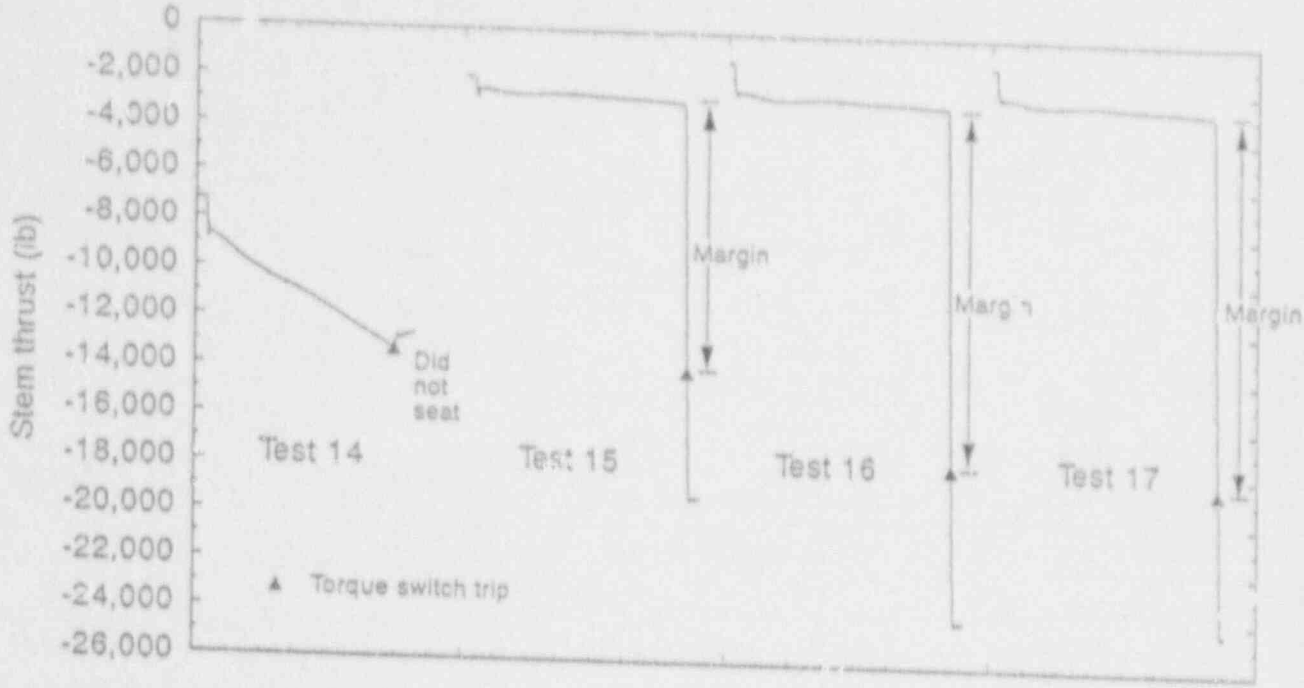

Figure 5. Stem force in Test 15 was affected by the high loads in Test 14.

(Figure 5), the load was returned to the same level as in Tests 8 through 10.

The test results show that we achieved load-sensitive behavior (the rate-of-loading effect) in the MOVLS by merely changing the simulated differential pressure (SΔP) load. Figure 4 shows how the 16,000-lb seating margin in Test 10 was reduced to an 11,000-lb margin in Test 11 and a 2300-lb margin in Test 12, with 1000-lb margin in Test 13, and a valve that did not seat in Test 14. Note also in Figure 4 how the measured thrust at torque switch trip varies with the SΔP load. Figure 5 shows how a high SΔP load in one test can affect the margin in a subsequent test. Six tests (Tests 8 through 10 and 15 through 17) were conducted at the same low SΔP load; all performed the same except Test 15, which is the only one of the six performed immediately after a test with a high SΔP load. Test 15 had less seating margin than the other five tests had.

THEORETICAL LOOK

Having successfully reproduced (on the MOVLS) the load-sensitive behavior we observed in the full-scale valve testing, we next conducted a thorough analysis to determine the cause of this behavior. We examined the kinematics and the kinetics of the motor operator. This examination included a close look at the original design basis relationships and at the inertial and specific friction effects. Elastic responses were not included.

From that analysis, we concluded that the potential causes of this load-sensitive behavior include: (a) changes in output stem torque as a result of inertial and frictional effects inside the operator, and (b) changes in stem thrust as a result of frictional changes outside the operator. Possible causes inside the operator include:

- Acceleration of the worm
- Worm-to-worm-shaft friction
- Friction in the drive sleeve bearing
- Deceleration of the worm-gear/drive-sleeve/stem-nut assembly.

(Figure 5), the load was returned to the same level as in Tests 8 through 10.

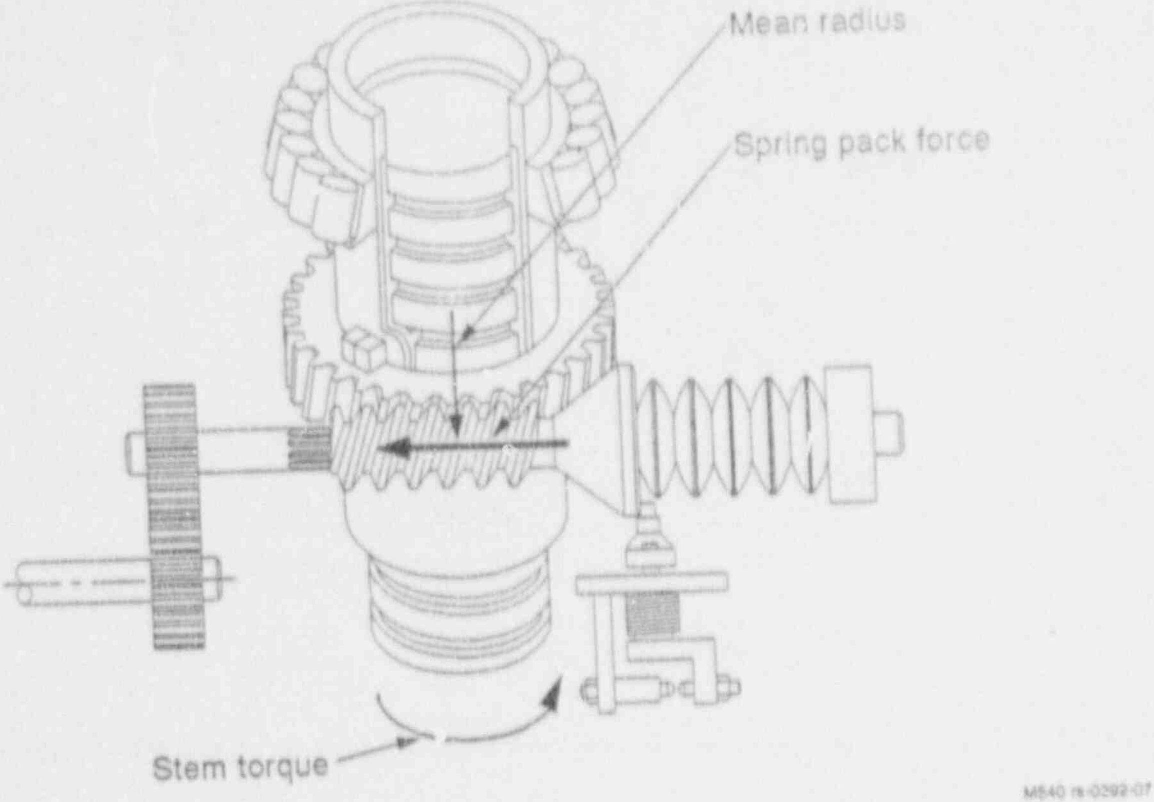

Figure 8. Motor operator moment arm.

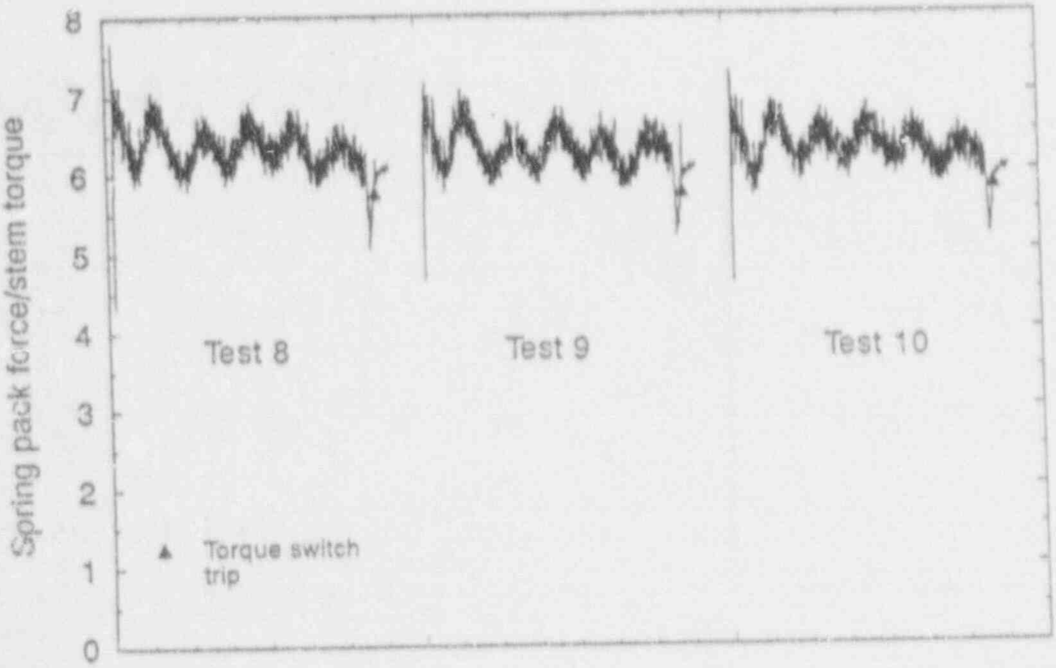

Figure 9. The ratio of spring pack force to stem torque remains constant in these lightly loaded tests.

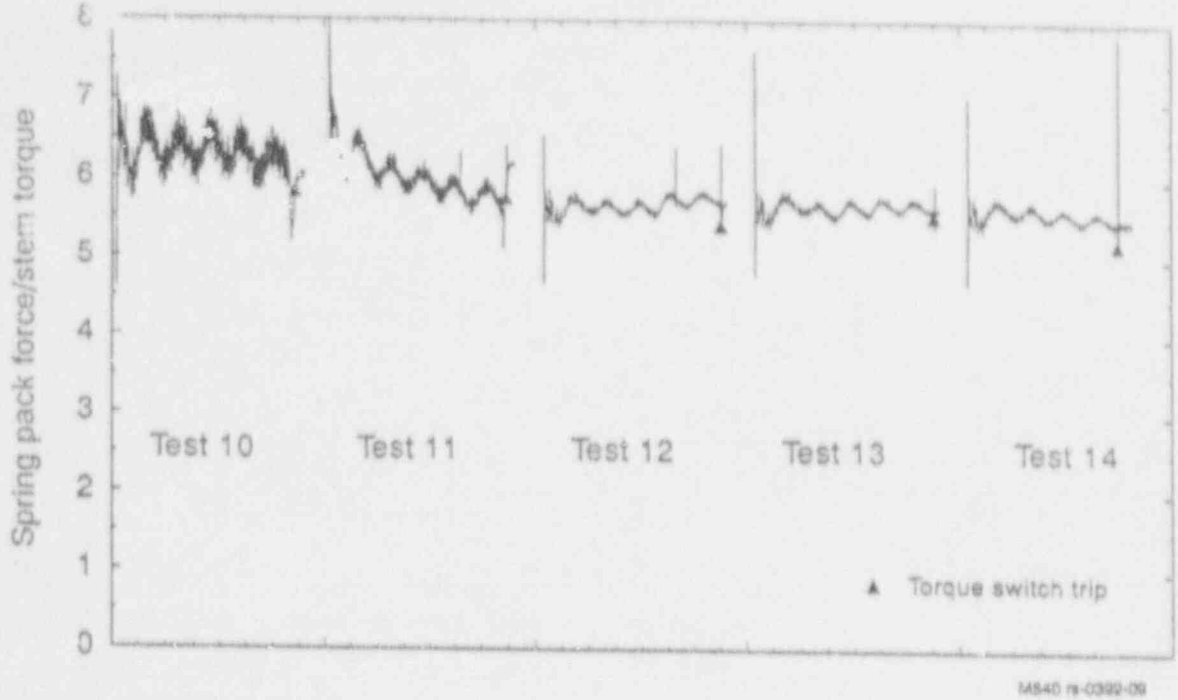

Figure 10. The ratio of spring pack force to stem torque remains relatively constant from lightly loaded to heavily loaded tests.

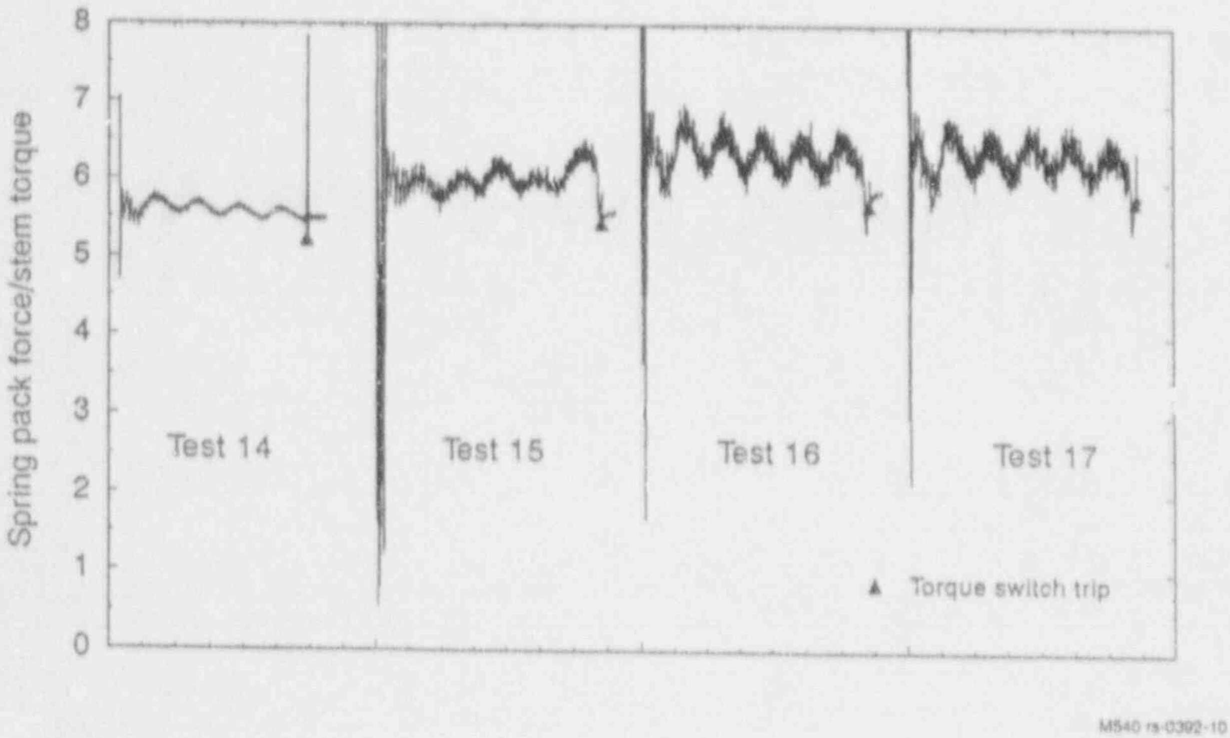

Figure 11. The ratio of spring pack force to stem torque remains relatively constant as the loading is reduced in these tests.

causes actually contributed to the behavior, the ratio of spring pack force to stem torque would have been different for the different tests.

Next we looked at the potential cause outside of the operator, the stem factor. Stem factor is stem torque divided by stem thrust. Solving for the stem factor was a straightforward calculation, as we measured both stem torque and stem thrust. Next we solved for the coefficient of friction, the only variable in the stem factor equation. We used the industry's power screw equation for modified ACME threads, as shown below, to solve for the coefficient of friction:

$$\mu = \frac{0.96815\,[24 \cdot SF - D \cdot \tan(\alpha)]}{D + 24 \cdot SF \cdot \tan(\alpha)}$$

where

μ = stem-to-stem-nut coefficient of friction

0.96815 = a constant that accounts for the face-to-face angle of the threads

SF = stem factor (measured)

D = stem diameter—1/2 pitch

$\tan(\alpha) = \frac{\text{lead of thread}}{\pi \cdot D}$

Figures 12 through 13 show the calculated coefficient of friction plots for the tests shown in Figures 3 and 4. (Our data reduction package, which has been verified and validated, allows us to input equations and the raw data from measurement channels and make real-time calculation plots.) As can be seen, the coefficient of friction was higher in tests with higher SΔP loads when all other parameters except load remained the same. These results show that the cause of the load-sensitive MOV behavior is the higher stem/stem-nut coefficient of friction in tests where the valve is subjected to significant SΔP loads before seating.

Tests 15, 16, and 17 were conducted at the same low loads as Tests 8, 9, and 10 (Figure 3). Figure 14 shows the result of those tests and provides insights into the root cause of the difference in stem nut coefficients of friction with different loads. Note that in Test 15, which was conducted immediately after a test with a high SΔP load, the stem nut coefficient of friction remained very high even though the SΔP load was very low. During the next two cycles (Tests 16 and 17) the coefficient of friction returned to the values derived in Tests 8, 9, and 10.

We believe that the cause of these results is as follows. During Tests 12, 13, and 14, the lubricant between the stem and stem nut was squeezed out of the contact zone when the stem nut turned under the high SΔP load. This resulted in a transition from boundary lubrication to dry (metal-to-metal) contact, with a large increase in the coefficient of friction. Test 15 remained at a high coefficient of friction level because the lubrication had not yet returned to the contact zone. With additional cycles at low SΔP loads (Tests 16 and 17), the adjacent grease relubricated the contact zone.

This lubrication squeeze-out does not occur on a low-load static test, because the first significant loading the valve stem experiences is the seat load. With this sudden seating load, rotation of the stem nut on the stem virtually ceases, and nearly all the motion goes into compressing the spring pack. The nearly static stem nut does not have the motion necessary to wipe the grease off of the contact zone, even though the load is very high.

CONCLUSIONS

We have found load-sensitive behavior on two occasions in the field: a 6-in. Anchor Darling valve and a 6-in. Velan valve. The Anchor Darling had a 1-1/2-in.-diameter stem and the Velan a 1-3/4-in.-diameter stem. Both stems had 1/4- by 1/4-in. pitch and lead. Two different lubricants recommended by manufacturers were used in the testing. Later theses two stem and stem nut combinations, plus a 6-in. Walworth stem and stem nut (1-1/4-in.-diameter, 1/4-in.-pitch, a[illegible] 1/2-in.-lead) were tested in the MOVLS a[illegible] included a third lubricant, an anti-seize formulation. All these MOVLS stem tests exhibited the

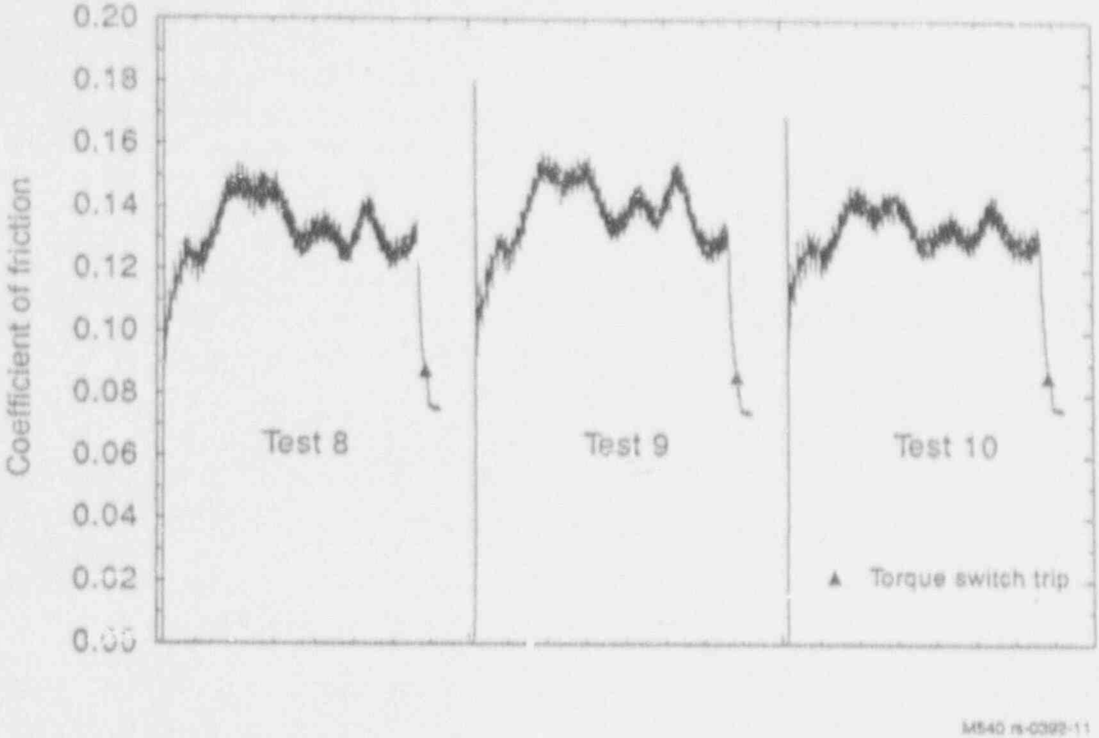

Figure 12. Stem-to-stem-nut coefficient of friction remains constant in these lightly loaded tests.

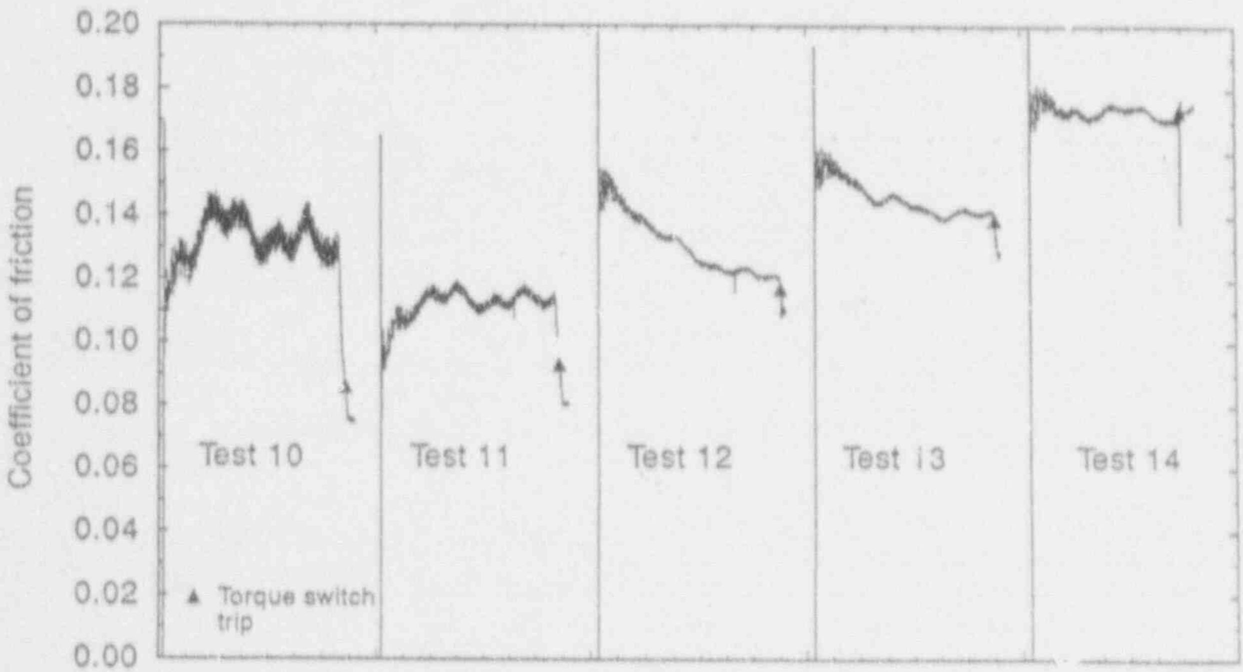

Figure 13. Stem-nut coefficient of friction increases with increased load (Tests 11 and 12) and continues to increase with repeated application of the same load (Tests 12–14).

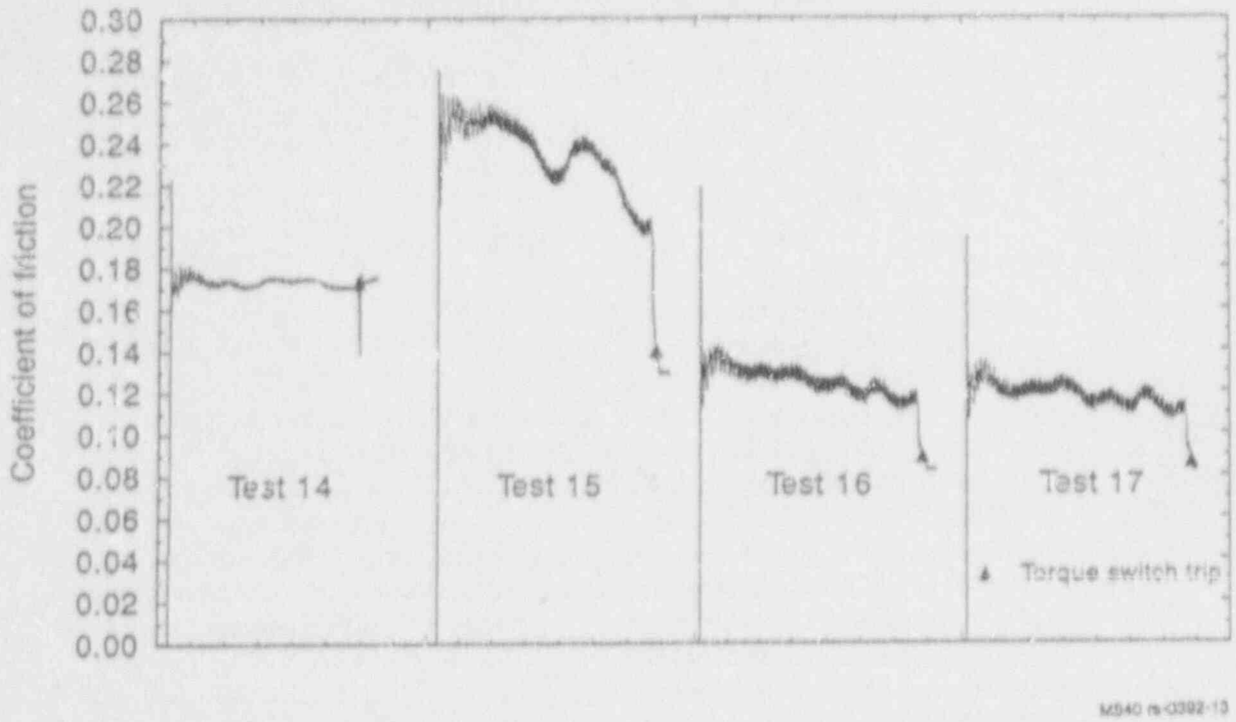

Figure 14. The coefficient of friction remains high (Test 15) in the first low-loaded test after a high-load test.

same behavior as the example shown in this paper. The higher stem/stem-nut coefficient of friction in tests with high SΔP loads is the first-order cause of MOV load-sensitive behavior. The higher coefficient of friction reduces the efficiency of the motor operator's conversion of torque to thrust. The result is a lower thrust for the same torque output in the motor operator. We believe the root cause is displacement of the lubrication from the contact zone in the stem/stem-nut interface.

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Application of Hydraulic Network Analysis to Motor Operated Butterfly Valves in Nuclear Power Plants

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Kalsi Engineering, Inc.

ABSTRACT

This paper presents the application of hydraulic network analysis to evaluate the performance of butterfly valves in nuclear power plant applications. Required actuation torque for butterfly valves in high-flow applications is often dictated by peak dynamic torque. The peak dynamic torque, which occurs at some intermediate disc position, requires accurate evaluation of valve flow rate and pressure drop throughout the valve stroke. Valve flow rate and pressure drop are significantly affected by the valve flow characteristics and the hydraulic system characteristics, such as pumping capability, piping resistances, single and parallel flow paths, system hydrostatic pressure, and the location of the motor-operated valve (MOV) within the system. A hydraulic network analysis methodology that addresses the effect of these parameters on the MOV performance is presented. The methodology is based on well-established engineering principles. The application of this methodology requires detailed characteristics of both the MOV and the hydraulic system in which it is installed. The valve characteristics for this analysis can be obtained by flow testing or from the valve manufacturer. Even though many valve users, valve manufacturers, and engineering standards have recognized the importance of performing these analyses, none has provided a detailed procedure for doing so.

A typical example from a nuclear power plant application is included to demonstrate the application of the hydraulic network analysis procedures under a number of operating conditions, including pipe rupture. The use of parallel line model in this example provided a reduction of approximately 40% in the torque predictions over the simplified single line model. The MOV safety function, either to open or to close, is taken into consideration when evaluating the analytical results. A combination of several parameters may significantly affect MOV performance under a given operating mode or a design basis condition.

INTRODUCTION

In 1989, the United States Nuclear Regulatory Committee issued Generic Letter 89-10 (GL 89-10), which recommends that all nuclear power plant licensees in the United States review the operability of motor operated valves (MOVs) in their plants. The letter specifically recommends that both normal operating conditions as well as design basis conditions be addressed. Although in situ testing may be used to demonstrate MOV operability under normal operating conditions, some design basis conditions, such as those involving downstream pipe rupture, do not lend themselves to such testing. Additionally, large-scale flow testing may not be feasible for several reasons, including plant operating constraints, cost, and inaccessibility of some MOVs. Analytical procedures in conjunction with judicious testing of some MOVs may be used to address the concerns of GL 89-10. One of the objectives of these procedures is to determine the actual valve flow rate and pressure drop against which the MOV is required to operate. In the absence of a well-established and structured methodology to calculate pressure drops and flow rates across the MOV at various disc positions, utility engineers often use overly conservative assumptions and analytical procedures. In many cases, torque and thrust predictions based on such assumptions may show that adequately sized MOVs are incapable of operating under the specified conditions. In other situations, the required actuation torque or thrust, based on such assumptions, may exceed the structural strength capability of the valve or the actuator. In these situations, the apparent problem is not in the MOV itself, but rather in the underlying assumptions or the use of simplified procedures.

This paper provides analytical procedures for predicting actual flow rates and pressure drops across MOVs. In the case of *incompressible* flow, the analytical procedures are presented for hydraulic systems that contain a single line or multiple branches. For *compressible* flow, the procedure is limited to single line systems. Flow characteristics of both the hydraulic system and the MOV are used to develop the mathematical models of the system using well-established engineering principles for calculating pressure drops and flow rates under unchoked or choked flow conditions. Even though many MOV users, valve vendors, and engineering standards have recognized the importance of performing these analyses, none has provided the detailed procedures for doing so.

It should be noted that butterfly valve torque requirements are particularly sensitive to actual valve flow rates and pressure drops as a function of disc angle. In high-flow applications, the required actuation torque is based on peak dynamic torque, which occurs at some intermediate disc position. The hydraulic network analysis methodology presented in this paper can be applied to determine the valve flow rates and pressure drops required to perform dynamic torque calculations at intermediate disc positions.

In this paper a general overview of the required actuation torque for butterfly valves is given first. This overview primarily applies to incompressible flow where the hydraulic network is most useful for nuclear power plant applications. The hydraulic network analysis is then presented for single line models and parallel line models. A typical example from an actual butterfly valve installation in a nuclear power plant is presented for several operating conditions including pipe rupture at valve discharge. This example demonstrates the effect of system modeling and simplifying assumptions on the required torque predictions.

Required Actuation Torque for Butterfly Valves

The torque required to actuate a butterfly valve is the larger of the seating/unseating torque or the peak dynamic torque. For *low-flow*, tight shut-off applications, the required actuation torque is generally bounded by the unseating torque. For *high-flow* applications the required actuation torque is typically dictated by the dynamic torque. The characteristics of required actuation torques and their components are discussed briefly in this section. It should be noted that flow and torque characteristics of butterfly valves depend on several factors, including disc shape, flow direction (for nonsymmetrical disc designs), and whether the fluid is compressible or incompressible. The primary focus of the butterfly valve overview presented here is on incompressible flow applications in which the hydraulic network analysis is most beneficial. More detailed discussions are found in many references, such as Kirik and Driskell (1991), ANSI/AWWA Standard C504-87 (1988), and *ISA Handbook of Control Valves* (1976).

Total Seating and Unseating Torque, T_{seat}

The total required torque to seat and unseat the disc of a butterfly valve is calculated as follows:

$$T_{seat} = T_s + T_b + T_p + T_{hyds} \quad . \qquad (1)$$

Most butterfly valve vendors provide seating/unseating torque requirements to meet specific operating conditions. In general, seating/

unseating torque, T_{seat}, is dominated by the seat torque component, T_s, and the bearing torque component, T_b. T_s depends on the particular seat design. Tight shut-off valves (except those with inflatable type seats) require a higher torque to unseat the disc than to seat the disc. This results from the difference between the static coefficients of friction (which are applicable during unseating) and dynamic coefficients of friction (which are applicable during seating). Furthermore, for rubber seated valves, the *break open* torque after the valve has been closed for several days may be as much as three times higher than that required when the valve is opened immediately after closing (*ISA Handbook of Control Valves*, 1976). Seat hardening, aging, and degradation may further increase seat torque requirements.

Valve bearings support the stem against the fluid-induced disc forces (hydrostatic force when the valve is fully closed and hydrodynamic force when the valve is partially or fully open). The combination of bearing force and friction results in bearing torque, which is calculated as follows (ANSI/AWWA Standard C504-87, 1988):

$$T_b = \frac{1}{12} \times \frac{\pi}{4} d^2 \times \Delta p_v \times \mu_b \times \left(\frac{d_s}{2}\right) \qquad (2)$$

A bearing coefficient of friction of 0.25 is generally assumed for metal type bearings (ANSI/AWWA Standard C504-87, 1988). Bearing torque peaks at the fully closed disc position where the valve pressure drop is maximum. Equation 2 is also used to calculate bearing torque contribution to dynamic torque (discussed later) under all flow conditions including post-choked flow.

Sealing the valve stem in butterfly valves (and quarter-turn valves in general) is easier than sealing rising stem valves, such as gates and globes. This tends to reduce packing torque requirements for quarter-turn valves to properly seal the stem. The stem seals most commonly used in butterfly valves are the standard V-type packing, the standard O-ring seals, and the stuffing box with pull-down packing gland. Required packing torque varies considerably among these types. For instance, the preload used in the packing gland could be made high enough to lock the stem in position. The actual packing torque may be determined using a torque wrench on the handwheel drive. The torque wrench reading, under no-flow condition, multiplied by the appropriate gear ratio and efficiency gives the actual packing torque.

In the closed disc position, the hydrostatic torque component, T_{hyds}, results from the difference in static head of the process fluid on both sides of the disc. It may assist or oppose the actuator in the seating direction and vice versa in the unseating direction. Hydrostatic torque is calculated as follows (ANSI/AWWA Standard C504-87, 1988):

$$T_{hyds} = 3.06\left(\frac{d}{12}\right)^4 \times \frac{\rho}{62.4} \qquad (3)$$

The hydrostatic torque may be neglected except for very large valves (typically 30 inches and larger). The hydrostatic torque becomes zero (or negligibly small) when (a) the valve stem is vertical; (b) liquid levels in both the upstream and downstream pipes are the same (either full, empty, or partially full); or (c) the process fluid is air, gas, or steam with insignificant head variation due to density.

In summary, the total torque required to seat or unseat the valve disc, T_{seat}, is the sum of four components: (a) seat torque, T_s; (b) bearing torque, T_b; (c) packing torque, T_p; and (d) hydrostatic torque, T_{hyds}.

Dynamic Torque, T_{dyn}

The dynamic torque at any disc angle is the required torque at the stem-to-actuator connection to rotate the disc through that angle, and is calculated as follows:

$$T_{dyn} = T_b + T_p \pm T_{hyd} \qquad (4)$$

The positive sign is used when calculating opening torque and the negative sign is used when calculating closing torque. The peak dynamic torque occurs in the opening direction when the hydrodynamic torque component is self-closing throughout disc stroke. The magnitude of the dynamic torque, T_{dyn}, varies with disc angle and reaches a peak value at some intermediate disc position. In high-flow applications, the peak dynamic torque exceeds the maximum required seating/unseating torque. To determine the peak dynamic torque, it is necessary to know the valve pressure drop and flow rate as functions of disc angle. Analytical procedures for calculating pressure drop and flow rate are presented in the Hydraulic Network Analysis section.

In general, the dynamic torque is dominated by the hydrodynamic torque component, T_{hyd}, and the bearing torque component, T_b. The nominal required packing torque, T_p, is relatively small. In the intermediate disc positions, the bearing torque and, to a much lesser degree the packing torque, are affected by the valve pressure drop and gage pressure at different disc angles.

The hydrodynamic torque component, T_{hyd}, is the product of the resultant hydrodynamic force induced by the flowing fluid and its moment arm to the center of disc rotation, Figure 1. T_{hyd} is a self-closing torque that tends to close the valve. Some exceptions do exist, however. For instance, at high disc opening angles, torque reversals (from self-closing to self opening) are encountered with single and double offset disc designs when the valve is oriented such that the flow is towards the flat face of the disc (valve shaft is downstream). Continuous operation with the disc at or near the torque reversal position can cause damage to both the valve and the actuator. In many cases the disc

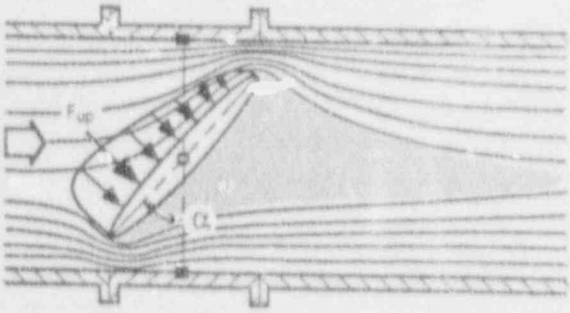


Figure 1. Flow through a symmetric disc butterfly valve.

travel is limited to a smaller disc angle (60 to 70 degrees open) to prevent such damage. Another exception to the self-closing characteristics of T_{hyd} was reported for compressible flow by Watkins et al. (1986). In their blowdown testing of containment purge valves, they found that the dynamic torque is self-opening throughout the valve stroke when the shaft is downstream. Butterfly valve *aerodynamic* torque in compressible flow has been recognized to be a complex phenomenon and continues to be a subject for further investigation (Morris and Dutton, 1989).

In general, the magnitudes of both the resultant hydrodynamic force and its moment arm depend on valve design-related parameters and installation-related parameters. The effects of these parameters are briefly discussed below.

Valve Design-Related Effects. Valve design-related parameters include valve size, valve pressure class, disc shape, disc-to-stem connection design, and the direction of flow through the valve. Major butterfly valve manufacturers develop flow and torque coefficients, C_v and C_t, as functions of disc angle for their product lines. However, only a few manufacturers publish their data. Knowing the torque coefficient, C_t, and differential pressure across the valve, ΔP_v, the hydrodynamic torque at any disc angle, α, can be calculated as follows (ANSI/AWWA Standard C504-87, 1988):

It should be noted that some manufacturers provide a specific hydrodynamic torque coefficient, C_t' (= $C_t d^3$), for each size, pressure class, and flow direction.

The hydrodynamic torque of butterfly valves with non-symmetric disc designs (such as single and double offset discs) depends on the flow direction. With the shaft upstream (often referred to as curved face forward or CFF), the hydrodynamic torque can be as high as twice its value with the shaft downstream (also referred to as flat face forward or FFF) in some valve designs. This results from the dependence of both the flow coefficient, Cv, and the hydrodynamic torque coefficient, C_t, on the flow direction through the valve. In the published literature, the dependence of C_t on the flow direction is well recognized. However, the dependence of C_v on the flow direction has not been addressed. Using the same flow coefficient for both flow directions may affect pressure drop, flow rate calculations, and ultimately the dynamic torque predictions. Thus, accurate predictions of the hydrodynamic torque component require that both ΔP_v (which depends on C_v) and C_t be known as functions of disc position. In choked flow operation, the hydrodynamic torque component should be calculated using the valve pressure drop at the onset of choking, ΔP_{choke} (Cohn, 1951; Sarpkaya, 1961). Using the total pressure drop across the valve, ΔP_v, to calculate T_{hyd} may result in unrealistically high torque predictions; the reason being that T_{hyd} depends on the actual mass flow rate rather than the valve pressure drop. In the choked flow region, the mass flow rate is independent of valve pressure drop.

$$T_{hyd} = \frac{C_t d^3 \Delta Q_v}{12} \tag{5}$$

Valve Installation Related Effects. Several installation-related parameters affect valve pressure drop, valve flow rate, and inlet velocity profile, which in turn affect T_{hyd}. Accurate determination of valve pressure drop and valve flow rate at various disc angles requires the evaluation of the entire hydraulic system in some nuclear power plant applications. The peak hydrodynamic torque under constant head applications (such as flow between two large reservoirs with relatively low piping resistance) occurs at large disc opening angles, typically in the range of 60 to 80 degrees open. In centrifugal pump applications (with relatively high piping resistance), the peak hydrodynamic torque occurs near the fully closed position, typically in the range of 10 to 30 degrees open. The effect of the pumping system on the critical disc angle where the peak hydrodynamic torque occurs may be summarized as follows (Dally, 1952): "As the ratio of valve pressure drop at zero flow ($Q = 0$) to valve pressure drop at maximum flow ($Q = Q_{max}$) increases, the disc opening angle for the peak dynamic torque decreases towards the fully closed position."

Upstream Flow Disturbance Related Effects. An important factor that affects the magnitude of T_{hyd} is the velocity distribution at valve inlet. The presence of upstream disturbance sources, such as elbows, in close proximity to the valve inlet introduces velocity skew which affects T_{hyd}. Several experimental investigations have been conducted to quantify these effects (Morris and Dutton, 1991; Watkins et al. 1986). Careful review of these experimental results leads to two important conclusions. First, the presence of an upstream disturbance may have a beneficial effect on the hydrodynamic torque requirement for butterfly MOVs. In general, when the perturbation velocity from a skew in the inlet velocity profile tends to open the valve, the presence of an upstream flow disturbance will reduce the hydrodynamic torque component and the total dynamic torque. This is contrary to the generally accepted notion that the presence of an upstream disturbance always has an adverse effect on butterfly valve torque requirements. Although credit should not be taken for this beneficial effect, this conclusion is useful in not penalizing MOVs for the presence of upstream disturbances with such orientations.

Second, as the flow rate decreases, the effect of upstream disturbances on the inlet velocity profile and T_{hyd} diminishes. Thus, at low disc opening angles, where the flow rate is small, the effect of upstream disturbances is small, and in many cases negligible. In centrifugal pump applications with high piping resistance, peak dynamic torque occurs at small disc opening angles. For these applications, the effect of upstream disturbance on the required torque is small and has been found to be negligible in several cases. The presence of upstream disturbances also affects the valve flow coefficient. This in turn affects the valve pressure drop as a function of disc angle. For instance, the effect of inlet/outlet concentric reducers on the valve flow coefficient at various disc angles is accounted for by introducing a piping factor, F_P, which is a reduction factor (ISA-S75.01, 1985).

Downstream Pipe Rupture Related Effects. Experimental results reported by several investigators and summarized by Cohn

(1951) and Sarpkaya (1961) show that the hydrodynamic torque coefficient, C_t, under free discharge flow conditions (pipe rupture immediately at valve discharge) is as low as 50 percent of its value under continuous flow conditions (no pipe rupture). This may be attributed to the effect of pipe rupture on the vena contracta pressure within the valve. Downstream pipe rupture in close proximity of the valve may not allow the vena contracta pressure to drop to the same level when the rupture occurs several pipe diameters from the valve. Thus, hydrodynamic torque predictions, assuming pipe rupture to occur at several pipe diameters from valve discharge, will envelop torque predictions with pipe rupture right at valve discharge. This is contrary to the generally accepted notion that a pipe rupture right at valve discharge represents the worst location for pipe rupture from the MOV operation standpoint.

Other Considerations. Self-closing dynamic torque has important implications in some applications. For instance, under design basis conditions involving pipe ruptures at valve discharge (blowdown), the valve is generally not required to open. In such applications the opening dynamic torque predictions need not be used to size the actuator motor or to set the torque switch limits. However, it should be ensured that the valve and actuator components are capable of withstanding the maximum predicted torque in either direction.

The magnitudes of the hydrodynamic torque and the bearing torque components in mid-travel depend on the actual valve flow rate and pressure drop at various disc angles that may be determined by in situ testing. Although in situ testing is the most reliable approach, it is not always possible to perform such tests under design basis conditions for some MOVs, especially those involving blowdown. Valve pressure drop as a function of disc angle may be calculated using analytical procedures that can be verified using available plant data or by testing under normal operating conditions. The results of these pressure drop calculations can vary considerably and depend on the assumptions used in the analysis. Simplified analytical procedures tend to provide upper bound pressure drop predictions that may be overly conservative in some cases. From the dynamic torque predictions based on these upper bound assumptions, one may erroneously conclude that some of the properly sized MOVs are incapable of operating under design basis conditions. For those applications, the detailed hydraulic network analysis, discussed in the next section, may be used to determine the margins between equipment capability and torque requirements under the limiting conditions. Typical examples from nuclear power plant applications follow the hydraulic network analysis discussion to demonstrate the quantitative differences in required operating torques under different assumptions.

HYDRAULIC NETWORK ANALYSIS

Hydraulic network analysis can be used to determine pressure drop across MOVs and the corresponding flow rate as functions of disc angle for quarter-turn valves, or valve travel for rising stem valves. Depending on the layout of the particular installation, the network may be analyzed using a single line model or a parallel line model. The single line model is easier to apply and may be used to bound valve pressure drop and flow rate for applications with parallel flow. In applications where additional margins between actuator output and required dynamic torque are required, a credit for the flow through parallel branches may be taken by using the parallel line model. These additional margins are problem dependent and may be as high as 40% in some applications. Similar margins may be obtained while evaluating the structural integrity of MOVs.

The hydraulic network analysis assumes that the flow and torque coefficients, as well as choking characteristics of the valve under consideration are known. These valve characteristics should be determined experimentally or obtained from the valve vendor for the specific valve design, disc orientation, and fluid media (incompressible or compressible).

Single Line Model

Figures 2 and 3 show a single line model of hydraulic network with a centrifugal pump and a constant head, respectively. The presence of a surge tank pressure (with P_o, [illegible] in Figure 2) has two effects. Under normal flow conditions (no pipe rupture), the presence of high P_o tends to retard liquid flashing and the onset of choking. On the other hand, under blowdown conditions the pump head is increased by P_o.

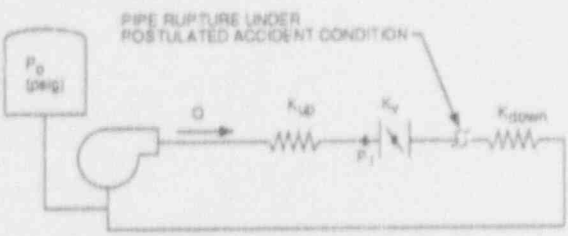


Figure 2a. Simplified model of a hydraulic system with a butterfly valve.

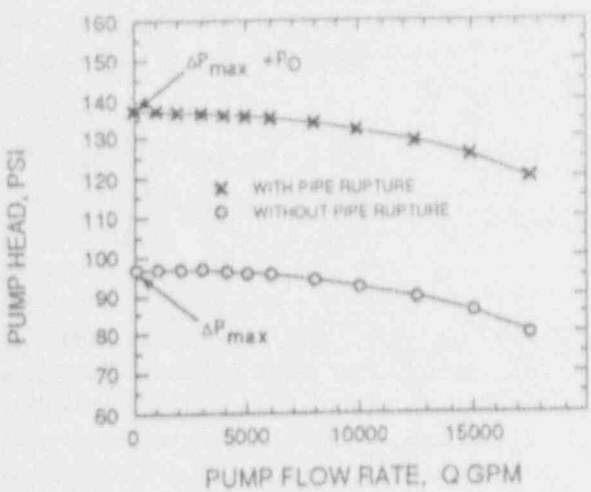


Figure 2b. Pumping characteristics with and without pipe rupture (NCWS pump in examples).

Figure 3a. Simplified model of a hydraulic system with a butterfly valve.

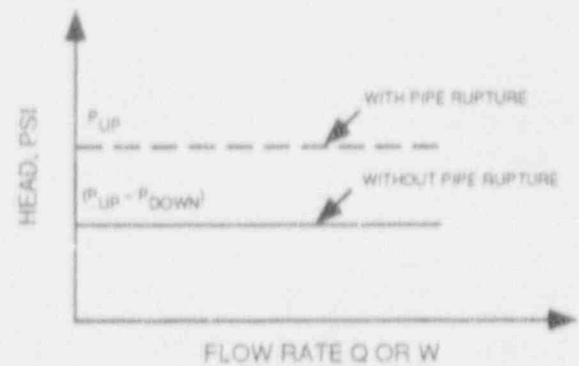


Figure 3b. Effective head with and without pipe rupture.

Resistance Coefficients. Piping resistance coefficients K_{up} and K_{down} may be calculated or estimated using K factors for the upstream and downstream piping components, respectively, using methods described in many fluid-dynamic references, such as Crane Paper No. 410 (1988). Underestimating K_{up} and K_{down} gives conservative ΔP_v predictions. For instance, neglecting the piping resistances ($K_{up} = K_{down} = 0$) is equivalent to forcing the MOV to operate against the full system head with no piping losses. Under blowdown conditions K_{down} is completely lost from the network and the MOV has to compensate for the absence of downstream head loss. With pipe rupture in close proximity of valve discharge, K_{down} is typically set to one for sudden expansion (Crane Paper No. 410, 1988).

Flow Rate and Valve Pressure Drop. At any disc angle, α, the valve resistance coefficient, K_v, is related to the valve flow coefficient, C_v, by the following relationship:

$$K_v = \frac{891 d^4}{C_v^{\,2}} \qquad (6)$$

The valve flow coefficient should be adjusted to account for the presence of inlet and outlet concentric reducers (if any) using the piping geometry factor, F_p, given in ISA-S75.01 Standard (1985). The corresponding system resistance coefficient is given by

$$K_{sys} = K_{up} + K_v + K_{down} \tag{7}$$

The system pressure drop and the corresponding flow rate are related by (Crane Paper No. 410, 1988)

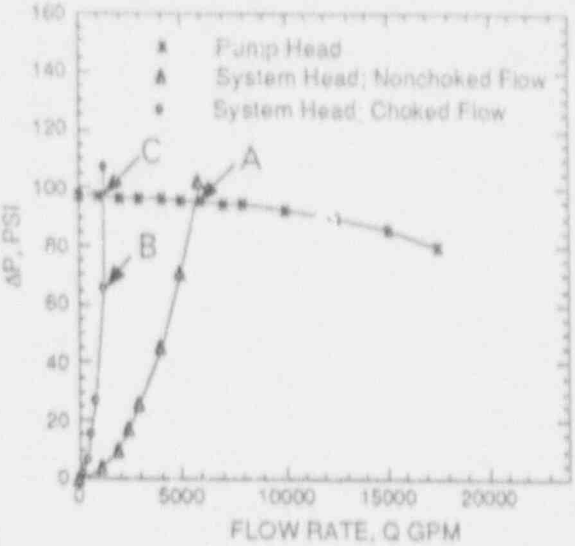
$$Q_{sys} = 236d^2 \sqrt{\frac{\Delta P_{sys}}{K_{sys}\rho}} \quad \text{(incompressible flow)} \tag{8a}$$

$$W_{sys} = 1{,}891Yd^2 \sqrt{\frac{\Delta P_{sys}}{K_{sys}V_1}} \quad \text{(compressible flow)} \tag{8b}$$

At any disc angle, α, the flow rate, Q_{sys} or W_{sys}, and system pressure drop, ΔP_{sys}, are obtained by finding the intersection of the system curve (Equation 8a or 8b) and the effective system head (Figure 2b or 3b). A typical graphical solution procedure is shown in Figure 4 with a centrifugal pump application under normal operating conditions without pipe rupture. In this figure the solution is depicted for a large disc opening angle (Curve O-A) with non-choked flow, and for a small disc angle (Curve O-B-C) with choked flow. The onset of valve choking is identified by Point B, beyond which the flow rate remains constant regardless of the pressure drop across the valve. The pressure drop across the upstream and downstream resistances, ΔP_{Kup} and ΔP_{Kdown}, respectively, are calculated using the flow rate found for that disc angle. The pressure drop across the valve, ΔP_v, is then found by subtracting the pressure drop across the upstream resistance, ΔP_{Kup}, and downstream resistance, ΔP_{Kdown}, from the system pressure drop, ΔP_{sys}.

$$\Delta P_V = \Delta P_{sys} - \left(\Delta d_{Kup} + \Delta P_{Kdown}\right) \tag{9}$$

As the flow rate increases with disc opening angle, the pressure drop across the valve



$$\Delta P_A = \Delta P_{Kup} + \Delta P_V + \Delta P_{Kdown}$$
$$\Delta P_B = \Delta P_{Kup} + \Delta P_{choke} + \Delta P_{Kdown}$$
$$\Delta P_C = \Delta P_{Kup} + \Delta P_V + \Delta P_{Kdown}$$

Figure 4. Graphical solution of a single line model under unchoked and choked flow condition.

decreases because of the increase in the pressure drop across the system (other than the valve) as given by $\Delta P_{Kup} + \Delta P_{Kdown}$. This procedure should be repeated at various disc positions. The objective is to determine the valve flow rate and pressure drop as functions of disc position throughout the valve stroke.

Valve choking may be accounted for by applying choked flow equations and the liquid pressure recovery factor, F_L, for t valve as outlined in ISA-S75.01 Standard (1985). For butterfly valves the pressure drop across the valve at the onset of choking, ΔP_{choke}, is used to calculate T_{hyd} in the post-choking region. The bearing torque is directly proportional to the total pressure drop across the valve regardless of whether the flow is choked or not choked.

The accuracy of the valve pressure drop predictions depends on how closely the single line model represents the actual network. If the valve is located in the main pipeline with small or no parallel flow through other branches, these predictions are quite accurate. In the presence of parallel branches with relatively small flow resistance, the pressure drop predictions based on single line assumptions may be too conservative, and more accurate predictions can be obtained by using the parallel line model described next.

Parallel Line Model

Figure 5 shows a parallel line model of hydraulic network in centrifugal pump applications. As mentioned above, taking credit for the flow through a parallel branch, K_3, will show that the valve pressure drop and flow rate are less than those predicted using the single line model. For instance, valve pressure drop when the disc is fully closed is less than the pump shut-off head the flow through K_3 and the resulting pressure drop across K_1 and K_4. The presence of surge tank pressure, P_o, has a similar effect, as described earlier in the discussion of the single line model.

Resistance Coefficients. Conservative valve pressure drop predictions are obtained by underestimating K_1, K_{2up}, K_{2down}, and K_4, for the same reason discussed in the single line model. Overestimating K_3 also gives conservative predictions

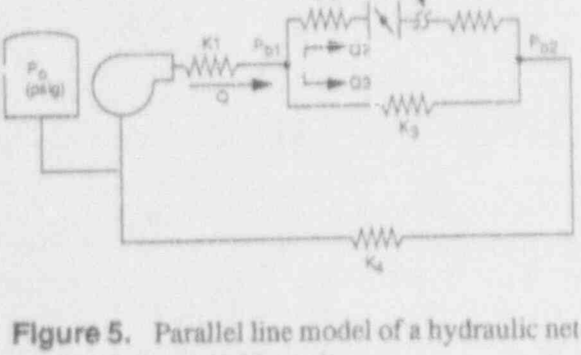


Figure 5. Parallel line model of a hydraulic network with a parallel branch.

because more flow would be forced through the valve branch, resulting in higher pressure drop across the valve.

A complex network may be reduced to a parallel line model as follows (Crane Paper No. 410, 1988):

1. Calculate the resistance coefficients (K factors) of the piping components.

2. Normalize the K factors with respect to the nominal size of the MOV.

3. Combine K factors for the different branches (K_{branch}) as follows:

Resistance in series:

$$K_{branch} = K_1 + K_2 + K_3 + \dots \tag{10a}$$

Resistances in parallel:

$$\frac{1}{\sqrt{K_{branch}}} = \frac{1}{\sqrt{K_2}} + \frac{1}{\sqrt{K_3}} + K \tag{10b}$$

Alternatively, if the flow rate (Q_{branch}) and pressure drop (ΔP_{branch}) across a pipe branch are known from plant performance/test data, Darcy's formula may be used to calculate the branch resistance coefficient (K_{branch}) as follows:

$$K_{branch}\left(\frac{236d^2}{Q_{branch}}\right)^2 \frac{\Delta P_{branch}}{\rho} \tag{11}$$

If ΔP_{branch} is measured at two locations with different pipe diameters, the difference in velocity head at these two locations must be accounted for.

Flow Rate and Valve Pressure Drop. At each disc angle, α, the valve flow rate, Q_2, and pressure drop, ΔP_v, can be determined using either analytical or graphical solution procedures. One such procedure is shown in Figure 6a for normal flow (no pipe rupture) and in Figure 6b for blowdown condition (pipe rupture at valve discharge). In these figures the line resistances K_1 and K_4 (identified in Figure 5) were set to zero for clarity. The flow through the pump is equal to the

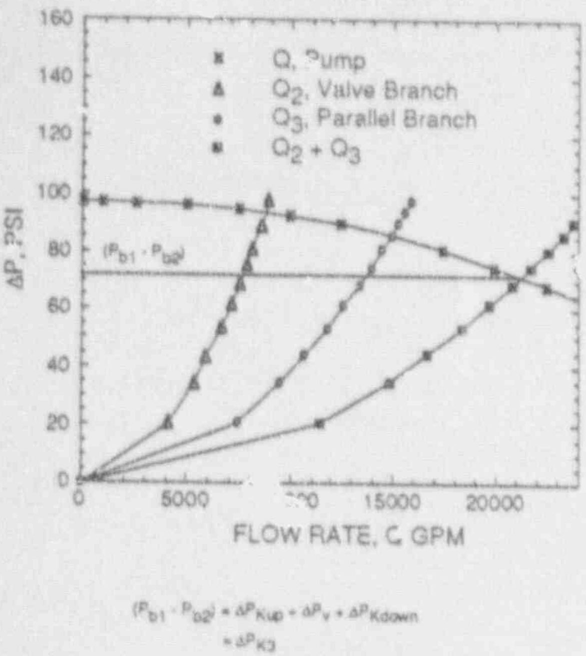


Figure 6a. Without pipe rupture.

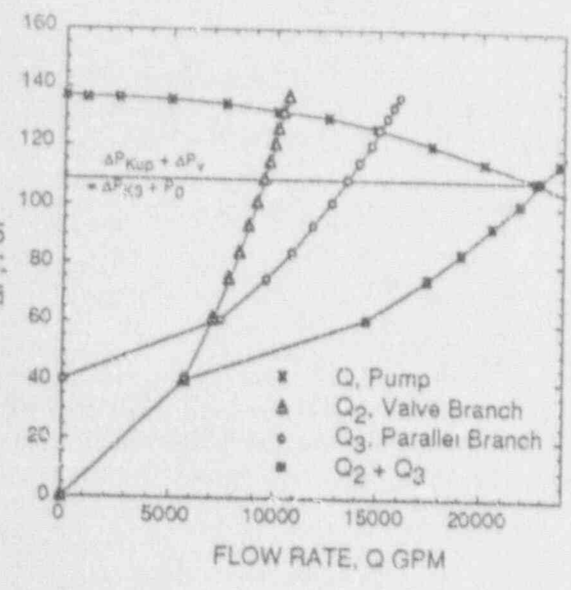


Figure 6b. With pipe rupture.

sum of the flows through the valve branch and the parallel branch. For instance, under normal flow conditions the pump flow rate is given by

$$Q_{pump} = Q_2 + Q_3$$

$$= 236d^2 \sqrt{\frac{(P_{b1} - P_{b2})}{(K_{2up} + K_v + K_{2down})}}$$

$$Q + 236d^2 \sqrt{\frac{(P_{b1} - P_{b2})}{K_{3Q}}} \qquad (12)$$

Under blowdown conditions, the pressure drops across the two parallel branches delivering flow rates of Q_2 and Q_3 are not the same (see Figure 6b). The solution under valve choked flow conditions becomes a little more involved and may require iterative solution. This procedure should be repeated at various disc angles to determine valve pressure drop and flow rate as functions of disc position.

The magnitude of K_3 as compared to $K_{2up} + K_{2down}$ is a key factor which affects the valve pressure drop and flow rate. Significant reductions in valve pressure drop predictions are obtained when K_3 is small as compared to the sum of $K_{2up} + K_{2down}$. On the other hand, as $K3 \rightarrow \infty$, the parallel line model approaches the single line model. Under blowdown conditions, the flow resistance coefficient, K_{2down}, is set equal to 1.0 for sudden expansion. The reverse flow on the other side of the ruptured pipe is typically prevented by check valves.

EXAMPLE

A typical example from an actual butterfly valve installation in a nuclear power plant is presented in this section. Required torque predictions are given for both single line and parallel line models discussed earlier. In what follows, torque predictions are calculated for normal operating conditions, as well as for blowdown conditions.

Problem Definition

A high performance butterfly valve is located in a nuclear cooling water system (NCWS). The valve is a 10-inch, ANSI Class 150, with a single offset disc. The valve is a line size valve without inlet or outlet reducers. The flow through the valve is such that the shaft is upstream into the flow (curved face forward). The valve flow coefficient, C_v, liquid pressure recovery factor, F_L, and torque coefficient, C_t', are provided by the valve vendor and are as shown in Figure 7a, 7b, and 7c, respectively. The following data are also provided by the valve vendor:

T_s = 116 ft-lb

T_p = 57 ft-lb

m_b = 0.15

d_s = 1.125 inch

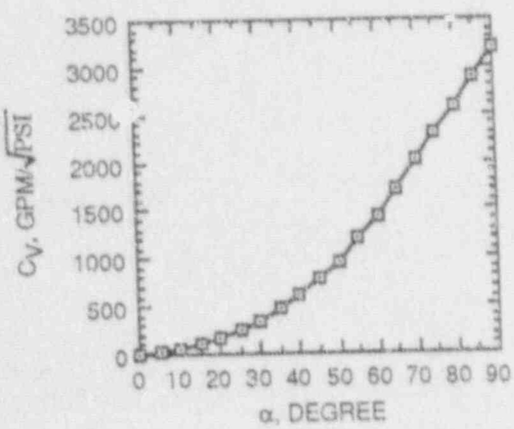


Figure 7a. Valve flow coefficient, CV, vs disc opening angle, α.

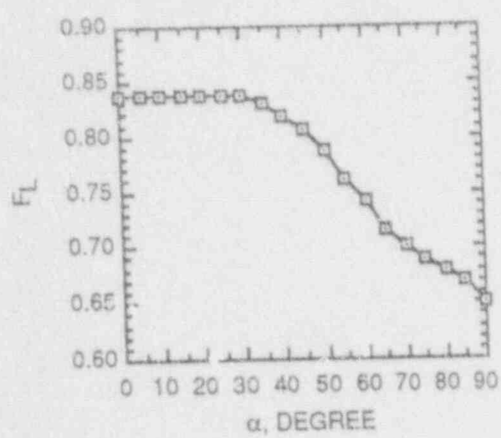


Figure 7b. Liquid pressure recovery factor, FL, vs disc opening angle, α.

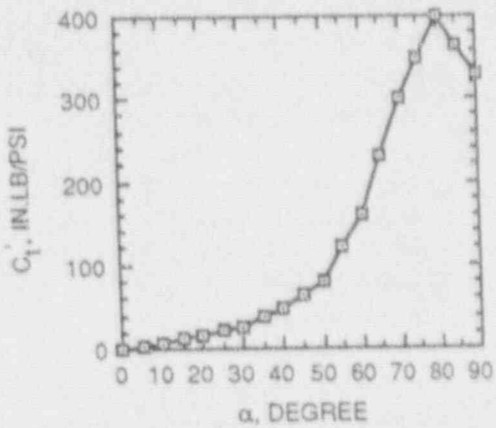


Figure 7c. Hydrodynamic torque coefficient, Ct', vs disc opening angle, α (shaft upstream).

The flow is sub-cooled water at 100°F. The NCWS pump characteristics are shown in Figure 2b, with a pump shutoff head of 97 psi. A surge tank increases the hydrostatic head in the system by 40 psi (P_o = 40 psig). The hydraulic resistances of the system depicted in Figure 5 (normalized with respect to the nominal valve size of 10 inches) are:

K_1 = 10

K_{2up} = 10

K_{2down} = 30

K_3 = 20

K_4 = 10

The valve is equipped with a Limitorque actuator with pullout torque of 462 ft-lb at full voltage and 296 ft-lb at a reduced voltage of 80%. The actuator consists of a Limitorque SMB-000 unit and a Limitorque H1BC gear reducer. The SMB unit has a 2 ft-lb—3,600 rpm motor and 62.5:1 overall gear ratio. The H1BC adaptor has a gear ratio of 35:1. The pullout efficiencies of the SMB-000 unit and the H1BC unit are 0.40 and 0.33, respectively.

Problem Requirement

1. Find the required actuation torque under (a) normal operating conditions, (b) postulated pipe rupture conditions with surge

tank pressure, and (c) postulated pipe rupture conditions with no surge tank pressure. Find these torque requirements using single line and parallel line models.

These conditions and modeling approaches result in the following six cases:

Case 1. Single line flow with P_o = 40 psig and without pipe rupture

Case 2. Single line flow with P_o = 0 psig and with pipe rupture at valve discharge

Case 3. Single line flow with P_o = 40 psig and with pipe rupture at valve discharge

Case 4. Parallel line flow with P_o = 40 psig and without pipe rupture

Case 5. Parallel line flow with P_o = 0 psig and with pipe rupture at valve discharge

Case 6. Parallel line flow with P_o = 40 psig and with pipe rupture at valve discharge

2. Evaluate the actuator torque output under the above six cases.

Solution

For each of the six cases, the hydraulic network analysis presented above is used to determine the valve pressure drops at various disc angles and the corresponding dynamic torque requirements. It should be noted that even though the analytical procedures described in the previous section are simple, the number of calculations involved is quite intensive to cover the entire range of disc angles. Under choked flow conditions, iterative procedures are typically needed to determine the actual flow and pressure drop across the valve. Because of the extensive calculations involved, a computer code was developed. Additionally, the code was used to perform sensitivity analyses to explore the effect of the resistance coefficients on the valve pressure drop, flow rate, and dynamic torque predictions for each of the six conditions defined above. The results of these analyses are presented in Figures 8 through 11. A summary of the peak dynamic torque results and the corresponding valve pressure drops and disc angles are given in Table 1. The seating and unseating torque for each case are also shown in Table 1.

Single Line Model

A single line model of this system is obtained by conservatively neglecting the flow through the

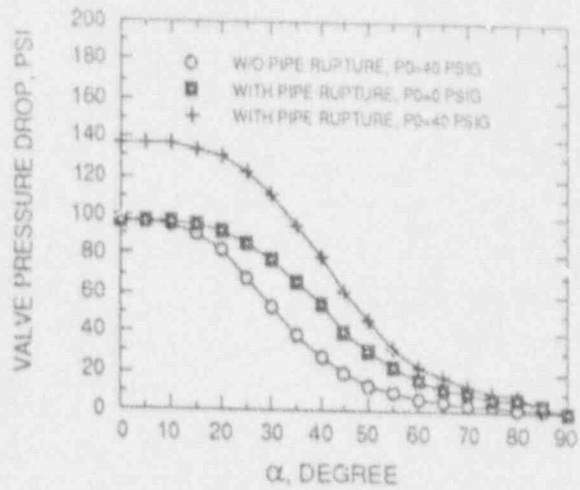


Figure 8. Valve pressure drop versus disc opening angle using single line model.

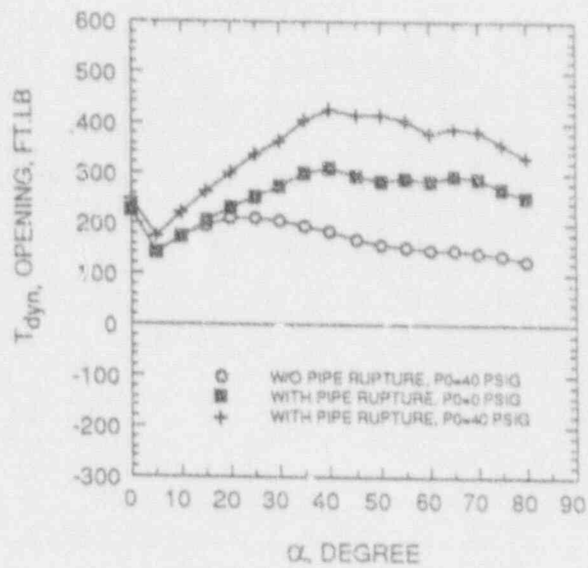


Figure 9a. Opening stroke.

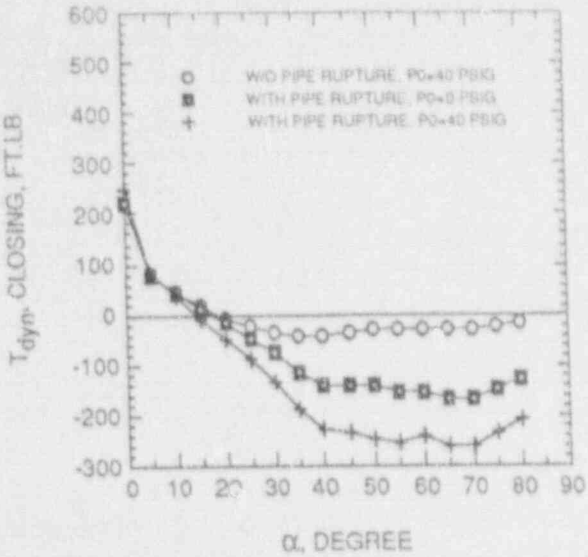


Figure 9b. Closing stroke.

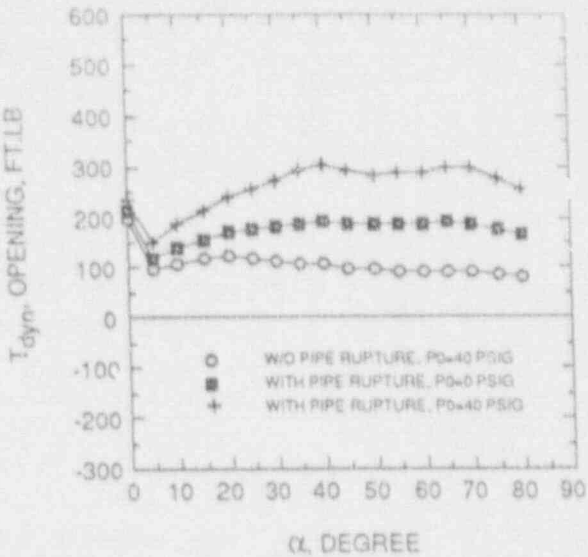


Figure 11a. Opening stroke.

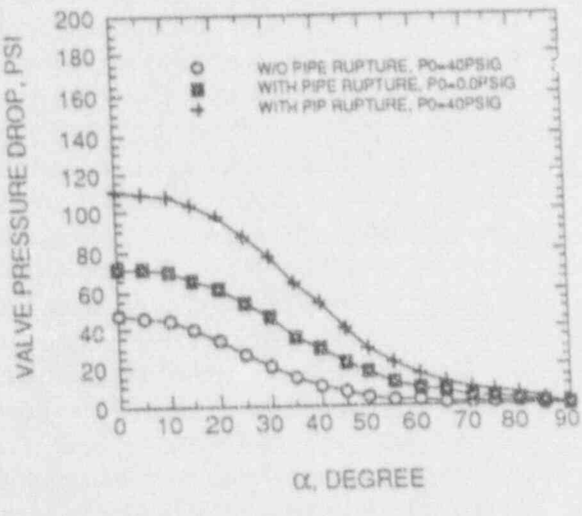


Figure 10. Valve pressure drop versus disc opening angle using parallel line analysis.

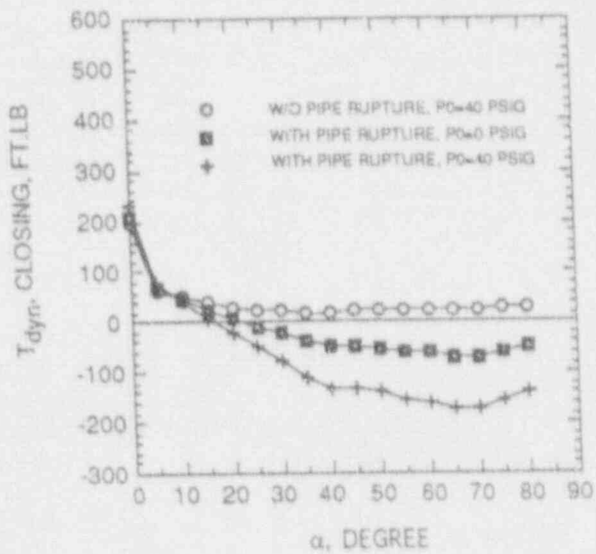


Figure 11b. Closing stroke.

Table 1. Summary of torque requirements for example problem.

	Peak Dynamic Torque Results						Seating and Unseating Results	
	Opening Stroke			Closing Stroke				
Analysis Conditions	Location of Peak Dynamic Torque, α, deg.	Valve Pressure Drop ΔP_v, psi	Peak Dynamic Torque T_{dyn}, ft · lb	Location of Peak Dynamic Torque, α, deg.	Valve Pressure Drop ΔP_v, psi	Peak Dynamic Torque T_{dyn}, ft · lb	Valve Shutoff Pressure Drop ΔP_v, psi	UTotal Seating/reseating Torque T_{seat}, ft · lb
Case 1	20	82.9	212	35	38.2	−42	97.0	225
Case 2	40	54.9	310	65	12.0	−168	97.0	225
Case 3	40	79.8	426	65	17.0	−261	137.0	247
Case 4	20	34.4	121	15	39.4	38	47.3	198
Case 5	40	29.3	192	70	5.1	−69	71.7	211
Case 6	40	53.2	303	70	9.4	−174	111.7	233

a. In this example negative values indicate that hydrouynamic forces induced by the flow are in disc closing directic. The actuator is not required to deliver this peak dynamic torque to close the valve. Note that the hydrodynamic torque in some disc designs and orientations are not self-closing.

parallel branch with line resistance K_3. Analytically, this is equivalent to setting the flow coefficient of the parallel branch to zero, or K_3 to infinity. The hydraulic resistances of the single line model as shown in Fig. 2a are as follows:

$$K_{up} = K_1 + K_{2up} = 10 + 10 = 20$$

For normal flow conditions (Case 1):

$$K_{down} = K_{2down} + K_4 = 30 + 10 = 40$$

For pipe rupture conditions (Cases 2 and 3):

$$K_{down} = 1$$

At any disc angle the flow coefficient is obtained from Figure 7a. The corresponding valve resistance coefficient factor, K_v, is calculated using Equation 6. The K factor of the system is then calculated using Equation 7. The intersection of the system curve given by Equation 8a, and the applicable pump curve (Figure 2b) gives the flow rate, Q, and the system pressure drop, ΔP_{sys}. The F_L value from Figure 7b is used to check the choking condition of the valve by applying the procedure in ISA-S75.01 Standard (1985). The valve pressure drop, ΔP_v, is calculated using Equation 9. The hydrodynamic torque component, T_{hyd}, and the bearing torque component, T_b, are calculated using Equations 5 and 2, respectively. The hydrodynamic torque coefficient for this valve, Ct' (= $C_t d^3$) is obtained from Figure 7c. The total dynamic torque in both the opening and closing directions are then calculated using Equation 4. This procedure is repeated at various disc positions using the corresponding valve flow coefficients from Figure 7a.

The total seating and unseating torque is calculated using Equation 1. In this equation, the seat torque, T_s, and the packing torque, T_p, are 116 and 57 ft·lb, respectively, as given in the problem definition. The bearing torque, T_b, is calculated using Equation 2, and shutoff head's of 97, 97, and 137 psi (see Figure 2b) for Cases 1, 2, and 3, respectively. Using Equation 3, the hydrostatic torque is calculated to be $T_{hyds} = 1.5$ ft·lb, which is negligible in this case.

The detailed results of the single line analyses (Cases 1, 2, and 3) are summarized in Figures 8, 9a, and 9b and Table 1.

Valve Pressure Drop Results. Examination of the valve pressure drop results at various disc angles (Figure 8) shows that

1. For disc positions between 40 and 90 degrees open, the valve pressure drops for Case 2 are more than double their values for Case 1. This increase in valve pressure drop has a significant effect on the dynamic torque (Figs. 9a and 9b).

2. For disc positions between 10 and 45 degrees open, the difference in valve pressure drops of Cases 3 and 1 are more than P_o (40 psi). The maximum difference of these pressure drops is about 58 psi at 30 degrees open, which corresponds to 145% of the surge tank pressure, P_o. Thus, over a certain part of the disc travel, the increase in valve pressure drop from a pipe rupture can be more than the system hydrostatic pressure imposed by surge tanks and accumulators.

Dynamic Torque Results. Figures 9a and 9b show the required actuation torque results for opening and closing strokes, respectively. Positive values indicate that the actuator has to supply the required torque to move the disc in the stated direction (Figure 9a); and negative values indicate that the actuator has to merely resist this torque while moving the disc in the stated direction of rotation (Figure 9b).

Opening Stroke: Examination of the opening stroke results (Figure 9a and Table 1) shows that

1. Under normal operating conditions (Case 1) the required actuation torque is 225 ft·lb, which is dictated by the total seating and unseating torque.

2. The required opening dynamic torque of 426 ft·lb for the case of pipe rupture with full tank head (Case 3) is about double its value under normal operating conditions (Case 1).

3. The required opening dynamic torques of 310 ft·lb and 426 ft·lb, respectively, in both pipe rupture cases (Cases 2 and 3) exceed the total seating and unseating torque.

4. The actuator pullout torque of 296 ft·lb at reduced voltage is less than the required opening dynamic torque for Cases 2 and 3.

5. The margin between the actuator pullout torque of 462 ft·lb at full voltage and the required opening dynamic torque of 426 ft·lb for Case 3 is less than 10%.

Closing Stroke: Examination of the closing stroke results (Figure 9b and Table 1) shows that the total seating and unseating torque of 225 to 247 ft·lb bounds the required closing torque for all three cases. Other butterfly valves with self-closing hydrodynamic torque characteristics will exhibit similar performance trends.

PARALLEL LINE MODEL

The parallel line model is shown in Figure 5. The resistance coefficients are given in Problem Definition. Using the solution procedures for parallel line model described earlier in this paper, the following results are obtained.

Pressure Drop Results. The valve pressure drop results at various disc angles for parallel line model are summarized in Figure 10 for Cases 4, 5, and 6. Valve pressure drops at shutoff and peak dynamic torque locations are summarized in Table 1. These results show that

1. Under normal operating conditions, the valve pressure drops using the parallel line model (Case 4) are less than 50% of the corresponding values using the single line model (Case 1). This has a significant effect on both the seating/unseating torque and the dynamic torque.

2. The difference between valve shutoff pressure drops for Cases 6 and 4 is 64 psi, which corresponds to 160% of the surge tank pressure, P_o. Thus, the increase in valve pressure drop from a pipe rupture is more than the system hydrostatic pressure imposed by surge tanks and accumulators. Furthermore, the increase in valve pressure drop from pipe ruptures in parallel flow models is higher than the corresponding increase in the single line model.

Dynamic Torque Results. At any disc angle the total dynamic torque is calculated using the valve pressure drop at that disc angle. The dynamic torque results are summarized in Figures 11a and 11b and Table 1 for opening and closing strokes, respectively.

Opening Stroke: The most significant advantage of using the parallel line model can be seen by comparing the governing torque prediction obtained by the parallel line model (Case 6) to that obtained by the single line model (Case 3). The use of parallel line model results in a peak dynamic torque requirement of 303 ft·lb (Case 6, Table 1). The corresponding result for the single line assumption is 426 ft·lb (Case 3, Table 1). This is an increase of 40% in the predicted torque requirements caused by simplifying assumptions.

Several other conclusions can be drawn by comparing parallel line results against the corresponding single line results. For instance, under normal operating condition (Case 4), the peak dynamic opening torque using a parallel line model is about 57% of that predicted using a single line model (Case 1). The required actuation torque in both cases, however, is based on the total seating and unseating torque. The use of a parallel line model provides a 12% reduction in the required actuation torque predictions as compared to the single line model.

Closing Stroke: Figure 11b shows that the actuation torque is dictated by the total seating and unseating torque for all three cases considered. Under pipe rupture conditions, the dynamic torque is self-closing for most of the valve stroke. Comparison of results for Cases 4 and 6 shows that an 18% increase in the total required seating torque is predicted as a result of pipe rupture at the valve discharge.

CONCLUSIONS

The analytical procedures for performing hydraulic network analysis to evaluate the performance of butterfly MOVs are presented. Both single line and parallel line models described herein can be used to determine valve pressure drops, flow rates, and torque requirements. An example problem from a typical installation in a nuclear cooling water system is used to demonstrate the application of the analysis procedures. A matrix of three different analyses performed using single line and parallel line models shows a significant reduction in torque requirements when the more realistic parallel line model is used. Detailed results are presented for each case. For the example problem, torque requirements for the parallel line model under the limiting operating conditions were found to be 40% less than that obtained for the single line model.

In evaluating the analysis results, the safety functions of the subject MOV should be taken into account. For instance, MOVs that are not required to open under postulated pipe rupture conditions should not be penalized for high opening torque or thrust. However, pipe rupture may expose the valve and actuator components to these high loads. Thus, although the postulated accident conditions may not challenge the actuator output capability, the structural integrity of the MOV should be evaluated under such high loads.

The butterfly valve dynamic torque predictions at large disc opening angles ($\alpha > 45$ degrees) are more sensitive to the resistance coefficients of the hydraulic network. At large disc angles the valve resistance coefficient, K_v, is relatively small, as compared with the system resistance coefficients. Thus, the network analysis is dominated by the system resistance coefficients. For this reason, higher margins should be imposed on torque predictions at high disc opening angles.

ACKNOWLEDGMENTS

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NOMENCLATURE

Symbol	Description
C_b	Effective bearing torque coefficient,in.lb/psi
C_t	Hydrodynamic/aerodynamic torque coefficient, dimensionless
C_t'	Hydrodynamic/aerodynamic torque coefficient for a specific valve size, pressure class, and flow direction, ($=C_t\, d^3$), in.lb/psi
Cv	Valve flow coefficient at any disc angle, α, gpm
F_L	Liquid pressure recovery factor of a valve without attached fittings, dimensionless

F_p	Piping geometry factor, dimensionless
K	Head loss coefficient of a device, dimensionless
P_1	Upstream absolute static pressure, measured two nominal pipe diameters upstream of valve or valve-fitting assembly, psia
Q	Flow rate, gpm
Q_{choked}	Valve flow rate under choked flow conditions, gpm
Q_{max}	Maximum flow rate when the valve is fully open, gpm
T_b	Bearing torque, ft·lb
T_{dyn}	Total dynamic torque at disc angle α, ft·lb
T_{hyd}	Hydrodynamic/aerodynamic torque, ft·lb
T_{hyds}	Hydrostatic torque, ft·lb
T_p	Packing torque, ft·lb
T_s	Seat torque, ft·lb
T_{seat}	Total seating/unseating torque, ft·lb
$\overline{V}_1$	Specific volume of fluid at inlet conditions, ft^3/lb
W	Flow rate, lb/hr
W_{choked}	Valve flow rate under choked flow conditions, lb/hr
W_{max}	Maximum flow rate when the valve is fully open, lb/hr
Y	Expansion factor, ratio of flow coefficient for a gas to that for a liquid at the same Reynolds number,dimensionless
d	Nominal valve size, inch
d_s	Stem diameter, inch
α	Disc opening angle, deg.
μb	Bearing coefficient of friction, dimensionless
ρ	Density, lb/ft^3
ΔP_{choked}	Pressure differential at the onset of valve choking, psi
P_{max}	Maximum pressure differential at shut-off, psi
ΔP	Pressure differential across the valve or valve-fitting assembly, psi

Operability Testing of Electrical Motor Operated Valves

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ABSTRACT

OM-10 "In-Service Testing of Valves" defines operational *readiness* as the ability of a motor operated valve (MOV) to perform its intended function. OM-10 confirms readiness through measurement of a single parameter, strike time. An MOV would be declared inoperable only when the referenced stroke time has not been met. The retest interval under OM-10 guidelines is a function of the category of the valve's service.

MOV testing under the guidelines of NRC Generic Letter 89-10, "Safety-Related Motor Operated Valve Testing and Surveillance," has demonstrated that MOVs that are not capable of performing their intended design function can meet OM-10 operational readiness criteria. These experiences support the need for an alternative criteria that confirms operability based on parameters directly defined by the intended design function.

The OM-10 method is an indirect test of operability using a single parameter and acceptance criterion. MOV operability testing requires direct measurement and acceptance criteria based on design requirements. Such methods have been developed in response to Generic Letter 89-10; however, the recognition that these methods should supersede OM-10-based operability determinations is not generally accepted. This paper proposes a method of using parametric testing, based on design requirements, to determine MOV operability.

BACKGROUND

Since the issuance of Generic Letter 89-10, Licensees have increasingly used parametric test equipment. Licensees are verifying actuator design capability through testing that includes measuring multiple parameters such as direct stem thrust, stem torque, limit switch activation, and motor current. This capability testing more adequately quantifies MOV operability than the requirements of OM-10.

PROPOSED METHODOLOGY

The minimum required measurements when verifying MOV operability are actuator torque, stem thrust, packing drag, motor current, stroke time, and torque switch bypass time. The acceptance criteria for each of the six parameters are established in a set-point calculation (example attached) which is based on design bases operability requirements. The set-points are maintained through a set-point verification testing procedure that establishes a pass/fail acceptance criteria for determining MOV operability. Failure to meet the acceptance criteria is adequate evidence to declare the MOV inoperable.

The six parameter test (capability test) is intrusive and man-power intensive and, therefore, cannot be performed as frequently as stroke time testing. Acceptance criteria must account for variations in the MOV's performance over time to justify less frequent operability testing. Of the six parameters, stem thrust is the most time dependant as a result of changes in stem factor caused by wear and lubricant depletion. Therefore, it is conservative to establish a primary time interval between operability tests that is based on stem factor degradation. A secondary interval can be established to address changes in actuator efficiency. In order to account for variations in MOV performance margin over time, a predetermined operating period must be established and validated over time by capability testing.

Thrust Performance Margin

In order to defend a predetermined operating period after a baseline capability test (with static or dynamic system conditions), a performance margin must be determined that accounts for output thrust degradation. The operating period is defined as the performance margin divided by the margin degradation rate, and can be expressed as

$$OP = \frac{PM}{PMDR} \qquad (1)$$

where

OP = operating period

PM = performance margin

PMDR = performance margin degradation rate.

The performance margin is the sum of the assumed stem factor degradation, packing drag degradation, and actuator efficiency degradation (see Figure 1). The operating period can be restated as

$$OP = \frac{(SF + PD + AE)}{PMDR} \qquad (2)$$

where

OP = operating period

PD = packing drag degradation

AE = actuator efficiency degradation

SF = stem factor degradation

PMDR = performance margin degradation rate.

Defining operating period, performance margin, and performance margin degradation rate in these terms, we can now discuss ways to control the factors involved, defend the concept of a predetermined operating period after baseline capability testing, and discuss operability determinations when as-tested outputs are less than the required minimum output at the beginning of the operating period.

Margin degradation rate can be bound to a predictable value by applying programmatic controls to the elements affecting it. Control over actuator efficiencies can be obtained through a scheduled actuator maintenance program. Stem factor variations can be bound through scheduled stem lubrications. Packing drag variations can be bound by setting design limits on maximum packing drag, and requiring measurement of packing drag after valve packing adjustments or replacements.

Thrust Correction Factors

The use of higher valve factors and more conservative design base differential pressures has resulted in higher required minimum output thrust and torque values. Because of these higher applied values, total available margin has decreased. With lower total margin available, the effort of identifying, quantifying, and controlling the variables that affect required minimum output gains greater significance. Those factors that directly affect required minimum output are design minimum output, correction factor for rate of loading, correction factor for control switch repeatability, and correction factor for test equipment accuracy.

One way to lower the required minimum output is by using the most accurate test equipment available. Accurate test equipment works to decrease the required minimum output in two ways: (a) when valve factors are calculated from dynamic test data, values are not inflated by

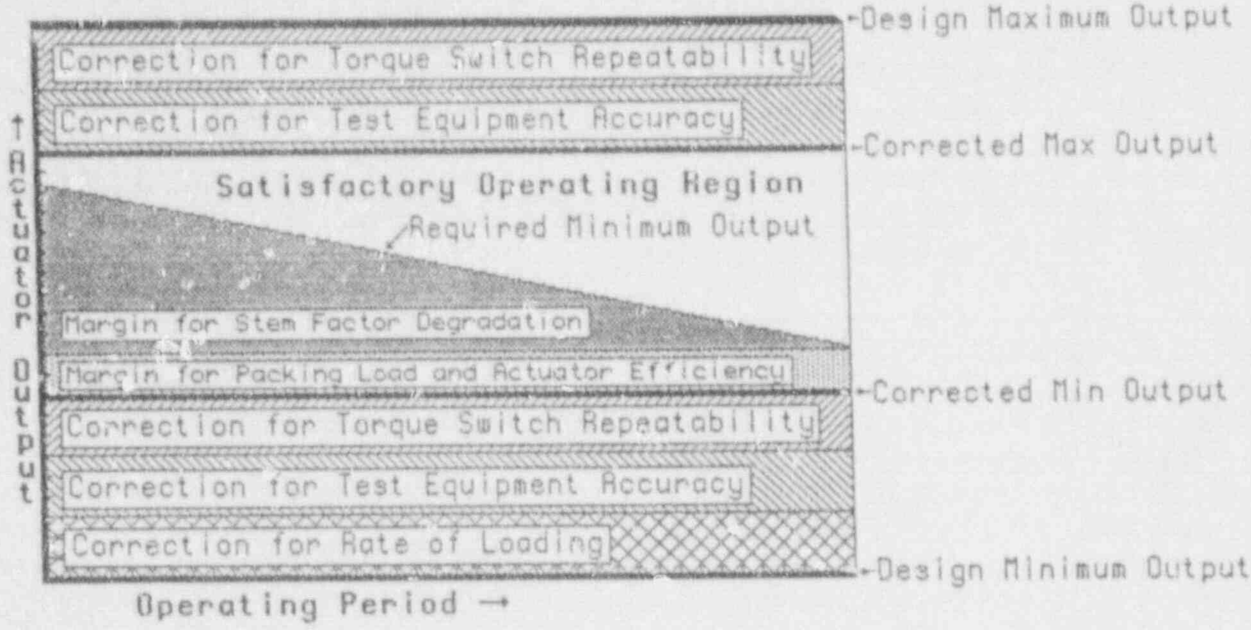

Figure 1.

corrections for inaccurate test equipment and (b) when actuator capability test acceptance criteria are established, limits do not have to be reduced by inflated corrections to account for inaccurate test equipment. Accurate test equipment is essential when dealing with marginal MOVs. Greater instrument accuracy means reducing required minimum output, and extending the operating period; extending the operating period leads to reduced cost. Stem-mounted, field-calibrated strain gauges are one of the more accurate test method for measuring stem thrust and torque.

Control switch repeatability is established by the actuator manufacturer and should not be lowered without adequate justification.

Rate of loading must be accounted for when using any test ΔP less than design basis ΔP to account for operability under dynamic conditions. Limited testing on a single actuator at Portland General Electric shows that the effect is proportional to the difference in the loading rates between the static test and the dynamic test. Therefore, rate of loading must always be taken into account when considering performance margin. Those MOVs with excessive performance margin may not appear to be affected by rate of loading until margin is diminished. Maximizing the operating period involves diminishing the performance margin throughout the operating period to its lowest level without compromising MOV operability. Therefore, to safely maximize operating period, rate of loading must be accounted for.

Operating Margin

Operating margin is the excess output above the required minimum output as measured at the beginning of the operating period.

CONCLUSION

The authors have found that present OM-10 stroke time testing does not accurately assess an MOV's "readiness" to complete its design function. The root cause of its deficiency lies in the test simplicity. A more rigorous, less frequently conducted test, which meets the guidance presented in this paper, may prove a practical alternative.

The following parameters were found to be essential when establishing and verifying MOV operability: actuator torque, stem thrust, packing drag, motor current, stroke time, and torque switch bypass time. Of the factors that impact actuator operability, stem factor, valve factor, actuator efficiency, and packing drag are functions of time and, therefore, should be enveloped by a performance margin. The performance margin should be evaluated under the guidelines of the design setpoint calculation, to ensure operability of the MOV over the defined operating period.

As a minimum the MOV should be tested at the beginning and end of the operating period so that the acceptance criteria for all six parameters, as established through design calculation, are maintained. The test verifies operability through the demonstration of the MOVs' capabilities to operate within the acceptance criteria. The acceptance criteria includes correction factors and performance margin that compensate for instrument dependent effects (torque switch repeatability, instrument accuracy), test condition effects (rate of loading), and time dependent effects (MOV degradation) for a fixed operating period. Satisfactory testing performed at the end of the operating period defends the established period. Unsatisfactory testing performed at the end of the operating period identifies the need to redefine the operating period or establish higher performance margins.

An MOV tested under static conditions should not be declared operable unless instrumented for a minimum set of parameters, which together demonstrate adequate performance margin for the required operating period. Capability testing performed under static conditions will have to apply acceptance criteria that account for the decrease in MOV output under dynamic conditions. Static test-acceptance criteria shall count for rate of loading, test equipment accuracy, control switch repeatability, and performance margin degradation.

An MOV tested under dynamic conditions should not be declared operable unless instrumented for a minimum set of parameters that together demonstrate adequate performance margin for the required operating period. If dynamic testing is performed without instrumentation ("blind testing"), a performance margin cannot be defined. Without a quantified performance margin, MOV operability is demonstrated for an undetermined period of time.

Portland General Electric uses the following correction factors:

- Test equipment accuracy = 10%
- Control switch repeatability
  = 10%, when output is ≥ 50 ft-lbf
  = 5%, when output is <50 ft-lbf
- Rate of loading = 20%.

In addition Portland General Electric uses the following performance factors:

- Stem factor degradation = ratio of $\frac{\text{Stem Factor } w/\mu = 0.2}{\text{Stem Factor } w/\mu = 0.15}$
- Actuator efficiency degradation = 0%
- Fixed operating period for stem factor = 3 years
- Fixed operating period for actuator efficiency = 6 years.

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SAMPLE CALCULATION

1. SAMPLE CALCULATION AND VERIFICATION OF PERFORMANCE MARGIN ACCEPTANCE CRITERIA FOR MOVS ON GATE AND GLOBE VALVES

The objective of this calculation is to establish torque, thrust, packing drag, motor current, stroke time, and percent torque switch by-pass setpoints for safety-related MOVs. This calculation establishes setpoints that will ensure operability of the MOV under maximum design basis conditions for a predetermined operating period. This is accomplished by calculation, testing and analysis in four steps, as follows:

1.1 Valve Minimum Thrust Required

First, calculate the minimum thrust required to cycle the valve during design base conditions.

1.2 Limiting Torque and Thrust Values

Second, evaluate the maximum output capability of the MOV (actuator, valve) assembly.

1.3 Torque Switch Displacements

Third, calculate the maximum torque switch setting (displacement). This setting shall result in an actuator output that does not exceed the maximum torque capability.

1.4 Actuator Testing and Analysis

Fourth, verify that actuator performance meets or exceeds performance margin using data from in-situ capability testing.

2. ACCEPTANCE CRITERIA

2.1 The maximum torque switch displacement, or less, that will allow the MOV to achieve the thrust (torque) required to fully stroke under design base conditions throughout the predefined operating period without exceeding the maximum torque or thrust allowable for the installation.

2.2 The maximum normalized motor current will not exceed 120% of the nameplate FLA during the running portion of the valve stroke.

2.3 The maximum stroke time will meet the design requirements.

2.4 The open torque switch by pass will not be less than 20% of stroke time after valve unseating.

2.5 The packing drag will be less than the shaft diameter multiplied by 1000, expressed in lbf.

3. ASSUMPTIONS

3.1 All torque switches addressed in this calculation have been calibrated to their design specification and are assumed to meet repeatability values stated by the manufacturer.

3.2 The margin applied to account for ROL during static testing is 20%. The basis for this assumption is presented in Kim (1990) and Kelly (1991).

3.3 The maintenance cycle is shorter than the established operating period. This assumption is based on accumulated maintenance history and the as-found condition of actuators and lubricants undergoing refurbishment.

3.4 Margin degradation factors:

3.4.1 The degradation of stem factor (SF) is conservatively bound by the ratio of

$$\frac{\text{Stem Factor } w/\mu = 0.2}{\text{Stem Factor } w/\mu = 0.15}$$

The valve and actuator are tested following maintenance, when the friction coefficient associated with the stem to stem nut interface is at its lowest. This ratio accounts for the degradation that occurs over the operating period. The basis for this assumption is taken from Limitorque Selection Procedures.

3.4.2 The degradation of the actuators efficiency is bound throughout the defined operating period by manufacturer's applications factors. The valve and actuator are tested following maintenance, when the friction coefficient associated with the worm, worm gear, and worm gear spline are at their lowest. This accounts for the degradation that occurs over the operating period.

3.4.3 The packing drag will remain below the design assumed value of 1,000 lb per inch of stem diameter. Throughout the defined operating period measuring packing drag is required following pack adjustment and replacement.

3.4.4 The valve factor will remain below the assumed value. The valve factor for gate valves is bound by 0.5 and for globe valves is bound by 1.1 (EPRI; Bethesda Licensing Office). When test data reveals valve performance not bound by these Valve Factors a new F_v shall be calculated and applied to that specific valve.

3.5 The Limitorque Selection Criteria is an accurate standard to use with regard to actuator specific criteria such as application factors, pullout efficiency, thrust and torque ratings, and gear rating service factors.

3.6 Spring packs are installed in motor actuators with the vendor specified preload.

4. METHOD AND EQUATION SUMMARY

4.1 Valve Minimum Thrust Required

The minimum valve thrust required to stroke during design base conditions is calculated using an equation from SEL 1 (Limitorque Selection Procedures), as follows:

$$V_{minTHr} = (F_V \times V_{PA} \times \Delta P_{mx}) + (S_A \times P_L) + (1000 \times S_D)$$

where

V_{minTHr} = Valve minimum thrust required = the minimum thrust needed to stroke a valve during design base conditions, in lbf.

F_V = Valve factor = the factor used to represent the losses within the valve during stroking. For globe valves this number is 1.1 per Limitorque Selection Procedures. For gate valves 0.5 is used per Kelly (1990) in lieu of 0.3 per the Limitorque Procedures. Industry experience has shown the 0.3 F_v may result in a non conservative calculated minimum thrust.

V_{PA} = Valve port area = the area of the valve port (area inside the seat ring), calculated by multiplying the square of the port radius by π, in square inches.

ΔP_{mx} = Maximum differential pressure = the maximum differential pressure that the valve must be capable of operating against, in either the open or closed direction, in psi.

S_A = Area of the stem = the cross-sectional area of the stem in square inches.

P_L = Line pressure = in psig.

S_D = Stem diameter = the diameter of the stem that passes through the stuffing box, in inches.

1000 = Conversion factor, this multiplier is used to determine the limiting stuffing box load, in lbf.

4.2 Limiting Torque and Thrust Values

The actuator and valve installation limits with regard to torque and thrust must be established. Operation beyond these limits may result in damage to, or disfunction of, the actuator and valve installation.

4.2.1 MOV Maximum Torque Limiting Value. The MOV torque limiting maximum value (MOV_{TQlmx}) is the maximum torque output capability of the actuator and valve assembly. This value is determined by comparing valve maximum torque capability, spring pack maximum torque capability, maximum actuator allowable torque capability and the maximum actuator TQ at minimum voltage. The lowest of these values is determined to be the MOV_{TQlmx}.

4.2.2 MOV Maximum Thrust Allowable Value. MOV maximum thrust allowable value (MOV_{mxTHa}) is the limiting maximum thrust as established by the actuator or valve manufacturer whichever is lowest.

4.2.3 MOV maximum Thrust Value With Inertia. The MOV maximum thrust value with inertia (MOV_{mxTHai}) is the limiting maximum thrust that can be measured at the MOV. This value is determined by the following equation:

$$MOV_{mxTHai} = MOV_{mxTHa} \times 1.1$$

1.1 = Conversion factor for developing the limiting inertial thrust as established by Limitorque (1988).

4.3 Torque Switch Displacements

4.3.1 Maximum Torque Switch Displacement. In order to prevent the actuator exceeding MOV_{TQlmx} a maximum setpoint for the actuator torque switch must be developed. The torque switch setting is determined by locating the MOV_{TQlmx} intersect on the appropriate spring pack curve (Limitorque Spring Pack Curves).

This torque switch setting correlates to the displacement of the switch arm required to open the switch contacts and deenergize the operator. Field experience has shown that two identical torque switches set to the same value will not have identical switch arm displacements. Therefore, to eliminate the error between torque switches, the maximum allowable displacement (Dsp_{mxa}) is developed.

The Dsp_{mxa} is based on the torque switch setting (TSS) at the MOV_{TQlmx}, determined by the applicable spring pack curve, and a regression analysis of the calibration curve for the applicable torque switch. The torque switch is then placed in a B&W Torque Switch Tester (Babcock & Wilcox) and the setting that corresponds to Dsp_{mxa} is marked.

4.4 Actuator Testing and Analysis

The MOV is now tested in situ. This testing is performed for all MOVs under static conditions. Where practical, dynamic system stroke testing is also performed. Testing must be performed after preventive maintenance is completed: the actuator is freshly refurbished and the stem to stem nut interface is clean and lubricated. The purpose of this testing is to

- Verify the measured actuator thrust output at torque switch trip is greater than or equal to the setpoint "minimum valve thrust required at torque switch trip" (V_{THmin}@TST)
- Verify the actuator torque output at torque switch trip does not exceed the setpoint "MOV maximum torque limiting value" (MOV_{TQlmx})
- Verify the actuator thrust output at torque switch trip does not exceed the setpoint "MOV maximum thrust limiting value" (MOV_{THlmx})
- Verify the actuator thrust output does not exceed the setpoint "thrust limiting maximum value with inertia" (MOV_{THlmxi})
- Assess the performance margin between the V_{THmin}@TST and the actual thrust at torque switch trip
- When possible, calculate the actual valve factor (F_{Vact}) based on test data. This F_{Vact} will then be used to calculate an actual valve minimum thrust ($V_{minTHact}$). Finally a performance margin assessment will be performed between the $V_{minTHact}$ and the actual thrust measured at torque switch trip.

Performance margin assessments will be used to determine the adequacy of the actuator capabilities. Actuators with a positive performance margin will be considered operable for a defined operating period without further evaluation. Actuators with negative margins will be evaluated for operability. It would be possible for an actuator to have a negative margin and still be operable for a reduced operating period.

4.4.1 Valve Minimum Thrust at Torque Switch Trip Setpoint. Valve Minimum Thrust at Torque Switch Trip (V_{THmin}@TST) is verified during static closure by measuring direct stem thrust at the point the torque switch contacts open. V_{THmin}@TST is calculated as follows:

$$V_{THmin}@TST = (V_{minTHr} \times (F_{S.2cof}/F_{S.15cof})) \times e_{min}$$

Where

V_{THmin}@TST	=	Valve Thrust Minimum at Torque Switch Trip = the minimum thrust setpoint value that must be measured at the valve under static conditions. This ensures the V_{minTHr} will be achieved under design basis conditions. The calculation of this value assumes that at the time of the test, the F_S is at its lowest (i.e., the stem is freshly lubricated), and the F_S increases to its maximum value at the time the valve must operate under design basis conditions for the predetermined Operating Period.
V_{minTHr}	=	Valve Minimum Thrust Required = in lbf. (see Step 5.1).
$F_{S.2cof}/F_{S.15cof}$	=	Performance margin ratio. This represents the potential degradation in stem factor prior to design limiting stroke of the actuator.

e_{min} = Minimum error factor = the value used to account for; Rate of Loading (r), Torque Switch Repeatability (t) (Limitorque, 1986), and test equipment accuracy (e_{Eqpt}). The error factor, a unit less term is calculated from these values by taking the square root of the sum of the squares of the two random errors (t and e_{Eqpt}) and adding to r, as follows:

$$e_{min} = 1 + r + \left[\sqrt{(e_{Eqpt})^2 + (t)^2}\right]$$

Once calculated, V_{minTHr}@TST represents the minimum acceptable thrust setpoint for static testing that will maintain MOV operability for a predetermined Operability Period. Actuator output greater than, or equal, to this setpoint is verified during static testing by measuring stem thrust.

4.5.2 Limiting Torque and Thrust Setpoints; Maximum Values at Torque Switch Trip

4.5.2.1 MOV Torque Limiting maximum at Torque Switch Trip. Torque less than MOV_{TQlmx} is assured prior to cycling the MOV by verifying that the torque switch setting is less than or equal to the $TQSW_{mx}$. In this way the DSP_{mx} cannot be exceeded and therefore, the MOV_{TQlmx} will not be exceeded.

This setpoint if verified during the static close stroke by measuring springpack displacement at Torque Switch Trip.

4.5.2.2 MOV Thrust Limiting Maximum Value at Torque Switch Trip. MOV Thrust Maximum Limiting Value at Torque Switch Trip (MOV_{THlmx}@TST) is the limiting maximum thrust that should be applied to the actuator at torque switch trip. MOV_{THlmx}@TST is calculated as follows:

$$MOV_{THlmx}@TST = MOV_{mxTHa} \times e_{max}$$

Where

MOV_{mxTHa} = MOV Maximum Thrust Allowable (see Step 5.2.2).

e_{max} = Maximum Error Factor = is the value used in this equation to account for Torque Switch Repeatability (t) and diagnostic equipment accuracy (e). The error factor is calculated from these values by taking the square root of the sum of the squares as follows:

$$Error\ factor = 1 - \left[\sqrt{(e)^2 + (t)^2}\right]$$

4.5.2.3 MOV Thrust Limiting Maximum Value With Inertia. The MOV Thrust Limiting Maximum Value With Inertia (MOV_{THlmxi}) is verified during closure by measuring the maximum stem compression. A_{THlmxi} is calculated as follows:

$$MOV_{THlmxi} = MOV_{mxTHai} \times e_{max}$$

Where

MOV_{mxTHai} = Actuator Maximum Thrust Allowable with Inertia. (see step 5.2.3)

e_{max} = Maximum Error Factor

4.5.3 Performance Margins: Calculated vs. Measured. Performance margins are calculated to assess the adequacy of the actuator as tested. These margins may be calculated using static test data with the assumed F_V or by using static test data with the measured $F_{V_{meas}}$. Whenever dynamic ΔP data is available the latter method shall be used.

4.5.3.1 Performance margin for actuators that have completed static testing is developed using the following calculation:

$$PM = V_{THmeas} \frac{@TST - V_{THmin}}{V_{THmin@TST}} @TST \times 100$$

Where

PM = Performance Margin = a value, in units of percent, that expresses the difference between the calculated minimum thrust required to close the valve and the measured thrust at Torque Switch Trip. A positive percentage indicates that the measured thrust is greater then the calculated thrust required

V_{THmeas}@TST = The measured thrust recorded at torque switch trip on a static close stroke.

V_{THmin}@TST = Valve Thrust Minimum at Torque Switch Trip = the minimum thrust that must be measured under static conditions.

4.5.3.2 Performance Margin for an actuator that has completed dynamic stroke testing with ΔP data available shall be calculated using the following method.

4.5.3.2.1 When dynamic stroke testing recorded the pressure drop across the valve during the test stroke it is possible to solve for $F_{V_{meas}}$ as follows:

Open Stroke:

$$F_{V_{meas}} = \left(V_{THmeas} \frac{- Rng\ Load + (SA \times P_{meas})}{(V_{PA} \times \Delta P_{meas})} \right)$$

Close Stroke:

$$F_{V_{meas}} = \left(V_{THmeas} \frac{- \mathrm{Rng\ Load} + (\mathrm{SA} \times P_{meas})}{(V_{PA} \times \Delta P_{meas})} \right)$$

Where

$F_{V_{meas}}$ = Valve Factor, Measured = the test derived factor used to represent the losses within the valve during stroking, a unitless term. Based on a solution from applied data.

V_{THmeas} = Valve Thrust measured = the thrust attributable to the force exerted on the disc or wedge as it is exposed ΔP during stroke, in lbf.

V_{PA} = Valve port area.

ΔP_{meas} = Differential pressure measure = the differential pressure in of psi, recorded at the corresponding point of the thrust trace identified as the V_{THmeas}.

P_{meas}	=	Pressure measure = the pressure in psi, recorded at the corresponding point of the thrust trace identified as the V_{THmeas}.
Rng Load	=	Running load = A mean average representing the running load exerted on the stem, during a static stroke, in units of lbf. This value is subtracted to isolate that amount of force required due to the differential pressure associated with the dynamic stroke.

4.5.3.2.2 The measured required minimum thrust ($V_{minTHmeas}$) is calculated as follows:

$$V_{minTHmeas} = ((F_{Vmeas} \times V_{PA} \times \Delta P_{mx}) + (S_A \times P_L) + (1000 \times S_D^2)) \times (F_{S.2cof}/F_{S.15cof}) \times e_{min}$$

Where

V_{minTHm}	=	The minimum valve thrust required at torque switch trip based on a measured valve factor. This term includes the conservatism factors for rate of loading, torque switch repeatability, diagnostic equipment accuracy and stem factor degradation as are found within the setpoint V_{THmin}@TST.
F_{Vmeas}	=	as defined above.
V_{PA}	=	Valve Port Area
ΔP_{mx}	=	Maximum Differential Pressure
S_A	=	Area of the stem
P_L	=	Line pressure
S_D	=	Stem diameter
1000	=	Conversion factor
e_{min}	=	Minimum error factor
$F_{S.2cof}/F_{S.1cof}$	=	Stem Factor degradation ratio.

4.5.3.2.3 Performance Margin for actuators that have completed dynamic testing with, ΔP data is developed using the following calculation:

Where

PM	=	Performance Margin = a value, in units of percent, that expresses the difference between the calculated minimum thrust required to close the valve and the actual thrust at Torque Switch Trip. A positive percentage indicates that the actual thrust is greater than the calculated thrust required.
V_{THmeas}@TST	=	The measured thrust recorded a torque switch trip on a static close stroke.
$V_{minTHmeas}$	=	as defined above.

Tennessee Valley Authority Motor-Operated Valve Methodology for Design Basis Review and Thrust Calculations

I. L. Beltz

ABSTRACT

The Tennessee Valley Authority (TVA) has developed a program plan to address the issues in the U.S. Nuclear Regulatory Commission's Generic Letter 89-10 "Safety Related Motor-Operated Valve Testing and Surveillance." As part of the program, TVA has developed standards for preparing design basis reviews and thrust calculations. These standards were developed by using applicable parts of the Nuclear Maintenance Application Center's (NMAC) document NP-6660-D, "Application Guide for Motor-Operated Valves in Nuclear Power Plants." TVA also used Seimens Nuclear Power Services as an engineering consultant in the preparation of its standards.

The standard for determining thrust requirements contains equations, safety factors, and friction factors that were based on the best information available at the time the standard was written. TVA will review the actual test results obtained from testing at its Sequoyah and Browns Ferry plants and compare these results to calculated values to verify the equations and factors and revise the calculation methodology where necessary. This paper will discuss the standards that TVA uses and present a comparison of data from Sequoyah Nuclear Plant differential pressure testing with the values obtained using the calculation methodology.

BACKGROUND

In 1989 the Nuclear Regulator Commission (NRC) issued Generic Letter 89-10, which provides recommended actions for licensees to follow in the development of a program to ensure that their safety-related motor-operated valves are capable of performing their safety-related function during a design basis event.

PURPOSE

The purpose of this paper is to describe the Tennessee Valley Authority's (TVA) methodology for design basis reviews and thrust calculations which were prepared in response to GL 89-10. This paper will describe the procedure TVA uses to develop the functional requirements of each motor operated valve, the calculation methodology used, the procedure used to set torque switches, and the comparison of differential pressure thrust values obtained at TVA's Sequoyah Plant versus the calculated values.

METHODOLOGY

Design Basis

The first step in TVA's methodology is to ensure that an understanding is obtained of the full range of design bases under which the valve is required to operate and the conditions that have an effect on the ability of the valve to function during these events. This understanding is required to provide input to the calculation methodology.

The procedure TVA uses to evaluate the design basis requires engineers to examine the valve from an electrical, mechanical, seismic and functional standpoint. The understanding of a valve's required function under all anticipated events is the first step in TVA's review and perhaps the most difficult and important. This review requires an experienced system engineer to document the internal fluid pressure, temperature, flow, and operating time requirements. During this design basis review TVA also confirms specific characteristics of the motor operated valve assembly, such as motor size, gear ratio, and valve manufacturer.

In parallel with the mechanical system review the electrical system is reviewed by an electrical engineer to document the voltage which will be available during these events. This review considers reduced voltages due to various conditions such as elevated ambient temperatures and their effects on cable and motors. The review will also consider the voltage available at the time in the diesel generator loading sequence that the valve is required to function.

The thrust loads, which are calculated as described in the following section, are used by the Civil Engineering department to evaluate the valve from a seismic analysis standpoint. The new thrust loads are coupled with other design loads, including seismic loads where appropriate, to determine if the valve assembly is capable of performing its safety function when subjected to these loads. The combined loads are used to evaluate the valve assembly from a weak link standpoint which ensures all parts of the valve are structurally capable of withstanding this load combination. These weak link evaluations have required several modifications including replacement of valve yokes and yoke bolts.

Calculations

The output from the design basis review is used as input to the TVA calculation methodology. This paper will describe the equations TVA uses for determining thrusts for gate and globe valves. Several methodologies have been developed in the industry for butterfly valves; however, they require input from the valve manufacturer, and the more elaborate methodologies require the user to make several unverifiable assumptions. TVA has opted to perform a design basis review on the butterfly valves and contract with the valve manufacturer to perform the torque calculation. TVA will perform an acceptance review of the calculations prepared by the butterfly valve manufacturer.

The equations and factors TVA uses for determining thrusts for gate and globe valves were developed in conjunction with Seimens Nuclear Power Services, TVA's technical consultant for GL 89-10 issues and use the information contained in the NMAC document NP6660, "Application Guide for Motor Operated Valves in Nuclear Power Plants."

Gate Valve Thrust Calculations

The thrust calculation methodology that TVA uses for gate valves is significantly different from that used by the valve or actuator manufacturer during the original sizing of the actuator. Many of the valves installed in the Sequoyah and Browns Ferry plants were purchased through a principal piping contractor with a minimum of input from a system standpoint. In many instances only the line pressure and temperature were considered when sizing the actuator. The equation used by most valve vendors to calculate thrust was in the format of $Fr = \mu dp + Fp + \text{Piston}$ and is used in both the opening and closing direction. The important differences between the valve vendors' methodology and the TVA methodology are how the μdp is calculated and the addition of a wedging force Fw. The TVA equation takes into account the angle of the wedge and uses a valve safety factor of 1.2 and a friction factor of 0.4. The methodology used by TVA for determining gate valve thrust requirements is in the following format:

$$F_r = F_{pack} + F_{dp} + F_p + F_w$$

F_r = total required thrust

where F_{pack} = packing friction load. The packing load is the force required to slide thestem through the packing. The packing friction loads are computed by multiplying the stem diameter by 1,000 lb.

F_{dp} = differential pressure force acting on the disc and calculated as follows:

$$F_{dp} = dP \times A_o \times F_v \frac{\mu}{Cos\ \theta \pm \mu\ Sin\ \theta}$$

where:

dP = differential pressure

A_o = orifice area of the valve $\pi D^2/4$

D = seat diameter at point of sealing, i.e., midpoint of seat ring

F_v = safety factor (1.2). This is a safety factor that adds conservatism to account for effects such as torque reaction, varying seat friction, degraded seat condition, rate of loading, and other effects which have not been fully quantified.

μ = friction factor seat/wedge (0.4)

θ = wedge angle (degrees from stem axis)

F_p = piston effect load. This is the force from the internal line pressure acting on the stem. It assists the actuator in the opening direction and resists the actuator in the closing direction. This force is computed as follows:

$F_p = P \times A$

P = line pressure

A = stem area

F_w = wedge load required to pull the wedge out of the seat. This load is applied for solid and flex wedge gate valves during the open stroke.

The wedging force is calculated as follows:

$$F_w = F_k \times f_w \times F_v \frac{(1 - \mu\ tan\ \theta)}{Sin\ \theta + \mu\ Cos\ \theta}$$

$$\times \frac{\mu}{Cos\ \theta + \mu\ Sin\ \theta}$$

where

$F_k = F_{rmax} - F_{pack} - F_p - F_{dp}$ (closing values)

F_k = thrust delivered to the seat

Frmax = max stem thrust (closing), i.e., 1.15 F_r - This is to account for inertia effects and contactor drop time.

F_{pack} = packing friction load

F_p = piston load

F_{dp} = differential pressure load

f_w = wedge factor. This is dependent on the flexibility of the wedge, i.e., 0.6 for flex wedge and 1.0 for solid wedge.

Flex wedge valves will relieve some sealing force where solid wedge valves retain a significant portion of the sealing force. Test data is used if it is shown these factors are not appropriate for specific applications.

F_v = safety factor (1.2)

μ = friction factor (0.4)

θ = wedge angle

As part of our effort to upgrade our methodology, we compared results obtained from the Idaho National Engineering Laboratory (INEL) methodology for determining thrust requirements for 5-degree flex wedge gate valves in the closing direction, which was presented in a paper at the 19th Water Reactor Safety Information Meeting (Watkins et al., 1991) with results obtained from the TVA methodology. By using the valve dimensions and fluid pressures, from the example in the INEL paper, in our equation we determined the thrusts to be within 2%. The thrust calculated in the INEL paper was 12,924 lb, and the thrust calculated using the TVA methodology was 12,716 lb.

Globe Valve Calculations

The globe valve methodology utilizes the following format:

$F_r = F_{pack} + F_{dp} + F_p + F_s$ (closing)

$F_t = F_{pack} + F_{dp}$ (opening)

where

F_{pack} = packing friction load (1,000 lb × stem dia.)

$F_{dp} = dP \times A_o \times F_v$

dP = differential pressure. When the pressure is above the disc in closing and below the disc in opening, dP shall be zero

A_o = orifice area of valve $\pi(D^2 - d^2)/4$

D = seat diameter at midpoint

d = stem diameter

F_v = valve safety factor—1.2 opening and 1.0 closing

F_p = piston effect

$F_p = P \times \pi d^2/4$

P = line pressure

d = stem diameter

F_s = seating force

$F_s = R_r \sin\theta + \mu R_r \cos\theta$

R_r = sealing force between seat and disc

$R_r = S_s A_s$

S_s = 4000 psi where dP = 0–500 psi
6000 psi where dP = 500–1000 psi
8000 psi where dP = 1000–2500 psi

A_s = area of the seat, i.e., circumference × seat width. For valves with line contact the width is assumed to be 0.0625 in.

μ = coefficient of friction between seat and disc (0.4)

θ = seat angle

Switch Settings

The thrust values obtained from these equations are transmitted via engineering output documents from the site engineering organization to the maintenance organization to be used for switch settings. TVA has an engineering specification that provides criteria for the setting and maintaining of torque and limit switches. For safety related valves, the torque switch is bypassed during the opening stroke and is bypassed for 95 to 98% of the stroke during closing. This provides greater assurance that the valve will be able to perform its primary safety related function.

TEST DATA

During the Unit 1 1991 and Unit 2 1992 outages at SQN, TVA set torque switches on 76 gate valves based on this calculation methodology. Of these 76 valves TVA was able to test 36 at differential pressure conditions. TVA used the MOVATS torque thrust cell where feasible to measure thrust and torque.

There were several cases where the valves did not wedge completely into the seats before torque switch trip; however, all valves closed the orifice and isolated flow.

Evaluation of actual stem nut friction factors for static and differential pressure, motor-operated valve testing, based on data recorded through the use of the MOVATS torque thrust cell, indicated that the 0.15 coefficient factor used in the calculation methodology is justified.

The SQN testing showed that inertia frequently results in an overthrusting and made it difficult in some cases to set up the torque switch in the range specified by engineering output documents. In future testing engineering will specify thrust ranges as large as possible to accommodate this phenomena.

Parallel Disc Gate Valve

The first table and series of plots are labeled 1-FCV-63-25 and 2-FCV-63-25 and are for a

1500 lb Anchor Darling 4-in. parallel disc gate valve with a SMB-0 actuator. The valve has a mean seat diameter of 3.875 in. and a stem diameter of 1.625 in. These valves are installed in Sequoyah Units 1 and 2 and are used as isolation valves between the reactor coolant system cold leg and the centrifugal charging pump. The maximum calculated differential pressure during opening and closing was 2676 psig. The maximum test pressure was 2545 psig. The tests were performed on Unit 1 valves during the 1991 outage and on Unit 2 valves during the 1992 outage. Table 1 is a comparison between calculated and measured thrusts during the opening and closing strokes.

The graphs for 1-FCV-63-25 are Figures 1 and 2 and were plotted using a load cell. The corresponding graphs for 2-FCV-63-25 are Figures 3 and 4 and were plotted using a direct thrust measuring device. The measured torque at torque switch trip in the closing direction for 2-FCV-63-25 was 263 ft·lb. Torque values were not available for 1-FCV-63-25.

The calculated steam factor was 0.0187, and the measured stem factor at torque switch trip was 0.0148 for 2-FCV-63-25.

Wedge Gate Valve

The second table and series of plots (Figures 5 and 6), labeled 1-FCV-63-152, 2-FCV-63-152, are for a 900 lb 4-in. Velan flexible wedge gate valve with a SMB-00 actuator. The valve has a mean seat diameter of 3.5 in. and a stem diameter of 1.375 in. These valves are also installed in Sequoyah Units 1 and 2 and are used to isolate flow at the safety injection pump outlet. The maximum calculated differential pressure was 1520 psig and the maximum test pressure was 1650 psig. Table 2 is a comparison between calculated and measured thrusts during the opening and closing strokes. These tests were also performed during the 1991 and 1992 outages.

The corresponding graphs for 1-FCV-63-152 and 2-FCV-63-152 are Figures 5, 6, 7, and 8, respectively. These were also plotted using a direct thrust measuring device.

The calculated maximum required torque was 207 ft·lb, and the measured torque at torque switch trip in the closing direction for 1-FCV-63-152 was 199 ft·lb, and the measured torque for 2-FCV-63-152 at torque switch was 142 ft·lb.

The calculated stem factor was 0.0171, and the measured stem factors were 0.0191 for 1-FCV-63-152 at torque switch trip and 0.0131 for 2-FCV-39-152.

CONCLUSIONS

These two examples showed that the methodology used by TVA to provide thrust values that were used to set the valves did provide assurance that they would fulfill their safety functions. The results also show that additional refinement to the methodology is required to predict specific aspects of the overall methodology, especially in computing the thrust required to open the valve. It appears that requiring the wedging load to be added to the differential pressure load produces overly conservative results.

REFERENCES

Application Guide for Motor-Operated Valves in Nuclear Power Plants, NP-6660-D.

Watkins, J. D., Steele, R., Jr., and DeWall, K. G., 1991, "NRC Test Results and Operations Experience Provide Insights for a New Gate Valve Stem Force Correlation," *19th Water Reactor Safety Meeting*.

Table 1. Evaluation of differential pressure test results parallel disc gate valves.

CLOSING DIRECTION

	Calculated (3)	1-FCV-63-25 (Measured)[1]	2-FCV-63-25 (Measured)
Packing	1,625		
Running Load (Packing plus Piston Effect)		(2)	5,296
Differential Pressure	11,361	13,211 (4)	11,028 (4)
Piston Effect	5,550		
Total Req'd. Thrust	18,536		16,324
Torque Switch Trip	N/A	16,291	17,766
Total Delivered Thrust	N/A	18,262	18,839

NOTES:

1. The actual measured thrusts for this valve were determined using the MOVATS TMD/Load Cell. The accuracy of this device is under question and must be factored into the results depicted in the table. TVA is currently evaluating this issue for impact on the GL 89-10 program.
2. Running load could not be quantified since it was less than spring pack preload.
3. These values were calculated using a friction factor of 0.3.
4. This includes piston effect and differential loads.

OPENING DIRECTION

	Calculated	1-FCV-63-25 (Measured)[1]	2-FCV-63-25 (Measured)
Packing	1,625		
Differential Pressure	11,361		
Piston Effect	-5,550		
Total Req'd. Thrust	7,436	6,374	5,849

NOTES:

1. The data obtained for this valve was with the MOVATS TMD/Load Cell.

Table 2. Evaluation of differential pressure test results wedge gate valves.

CLOSING DIRECTION

	Calculated	1-FCV-63-152 (Measured)	2-FCV-63-152 (Measured)
Packing	1,375		
Running Load (Packing plus Piston Effect)		2,018 (1)	
Differential Pressure	7,302	7,212 (2)	5,077 (2)
Piston Effect	2,257		
Total Req'd. Thrust	10,934	9,230 (1)	7,107
Torque Switch Trip	N/A	11,553 (1)	10,837
Total Delivered Thrust	N/A	13,894 (1)	14,252

NOTES:

1. These were increased by 1150 to account for initial piston load.
2. This includes differential and piston effect loads.

OPENING DIRECTION

	Calculated	1-FCV-63-152 (Measured)	2-FCV-63-152 (Measured)
Packing	1,375		
Differential Pressure	6,808		
Piston Effect	−2,257		
Wedging	6,215		
Total Req'd. Thrust	12,141	4,644	4,744

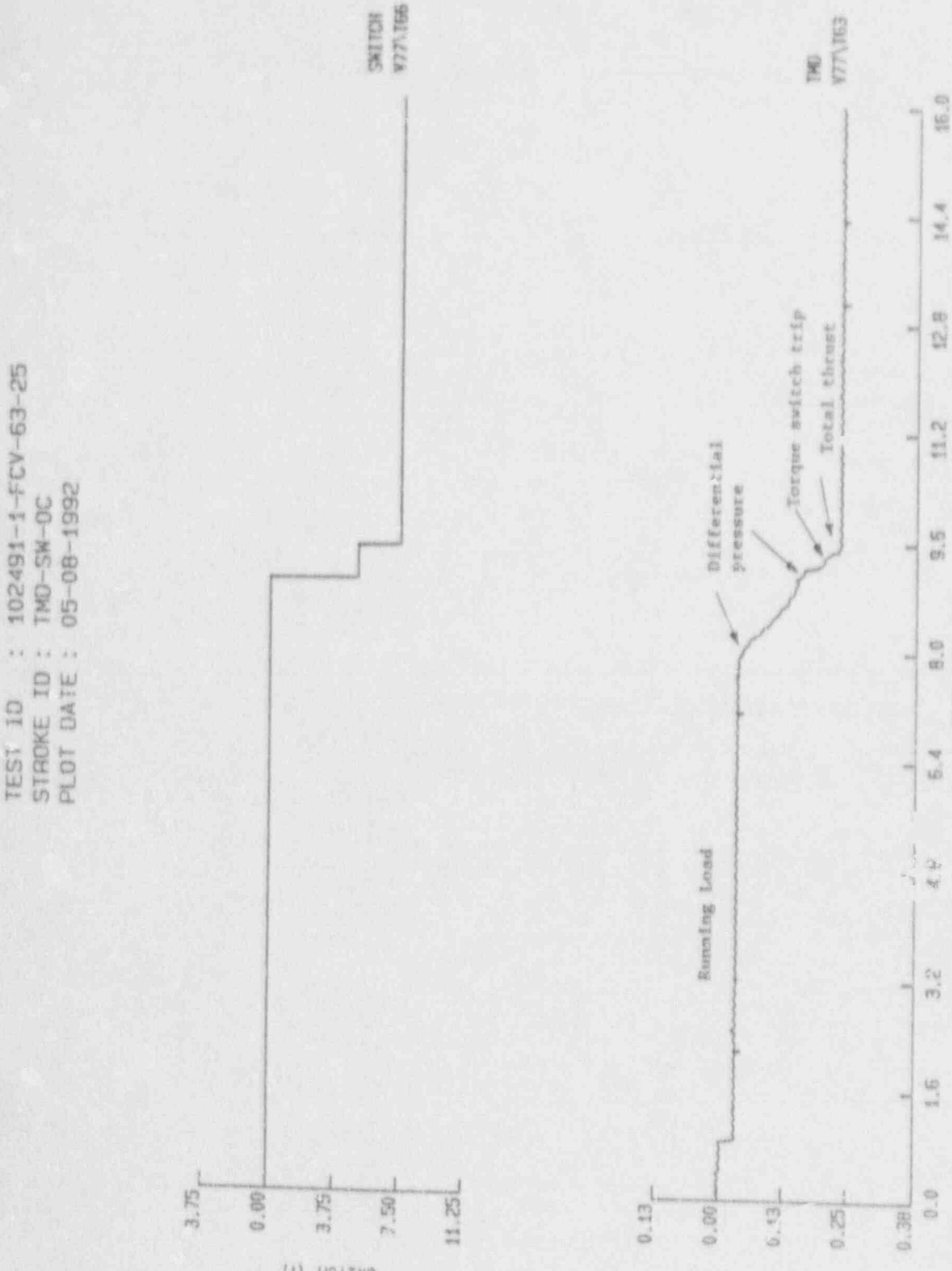

TEST ID : 102491-1-FCV-63-25
STROKE ID : TMD-SW-OC
PLOT DATE : 05-08-1992
SWITCH
V77\T66
SWITCH (V)
3.75
0.00
3.75
7.50
11.25
TMD
V77\T63
Disp. (I)
0.13
0.00
0.13
0.25
0.38
Running Load
Differential pressure
Torque switch trip
Total thrust
0.0
1.6
3.2
4.8
6.4
8.0
9.5
11.2
12.8
14.4
16.0
TIME (Seconds)

Figure 1.

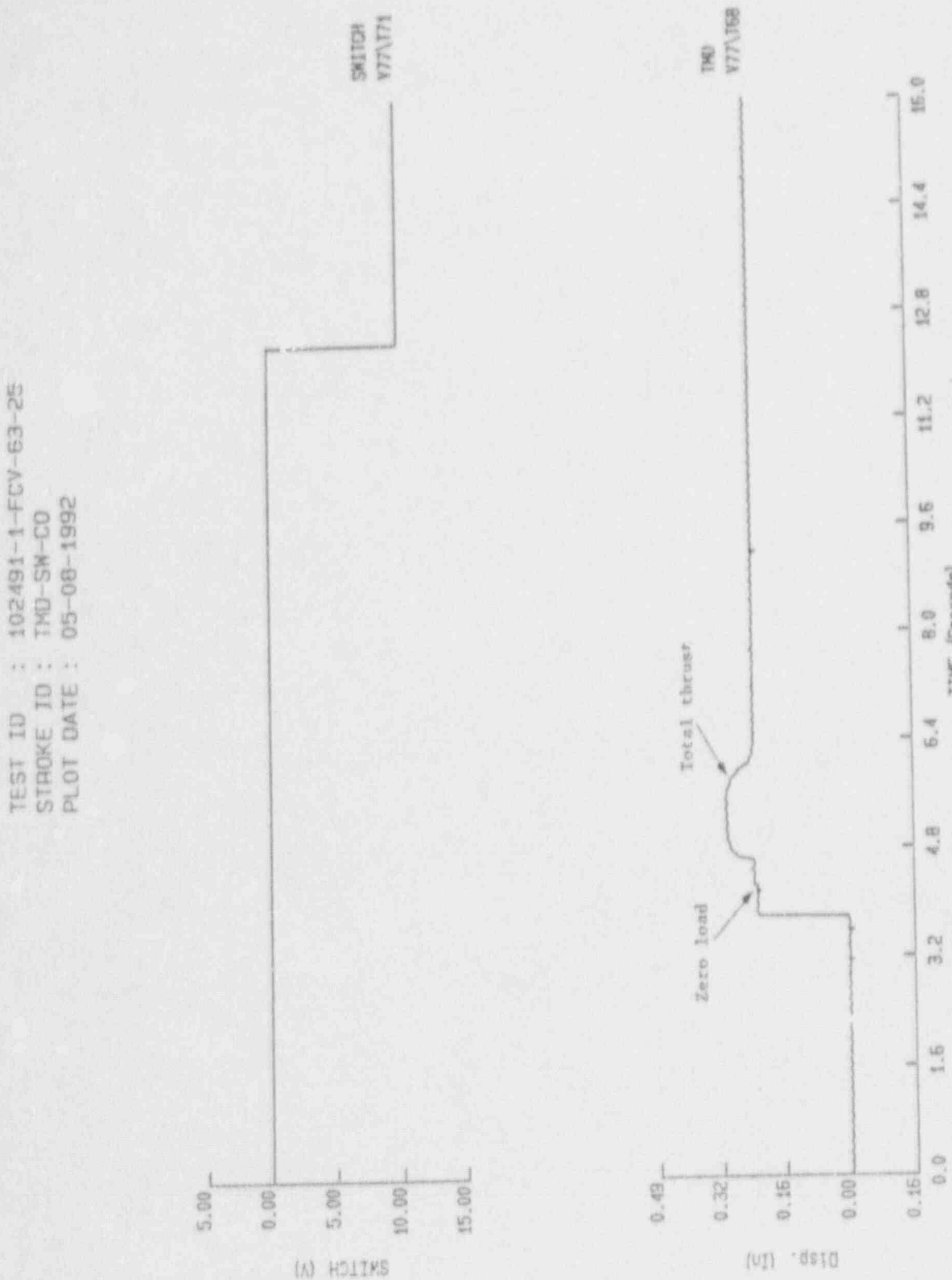

TEST ID : 102491-1-FCV-63-25
STROKE ID : TMD-SW-CO
PLOT DATE : 05-08-1992
SWITCH
V77\T71
SWITCH (V)
TMD
V77\T68
Total thrust
Zero load
Disp. (In)
TIME (Seconds)

Figure 2.

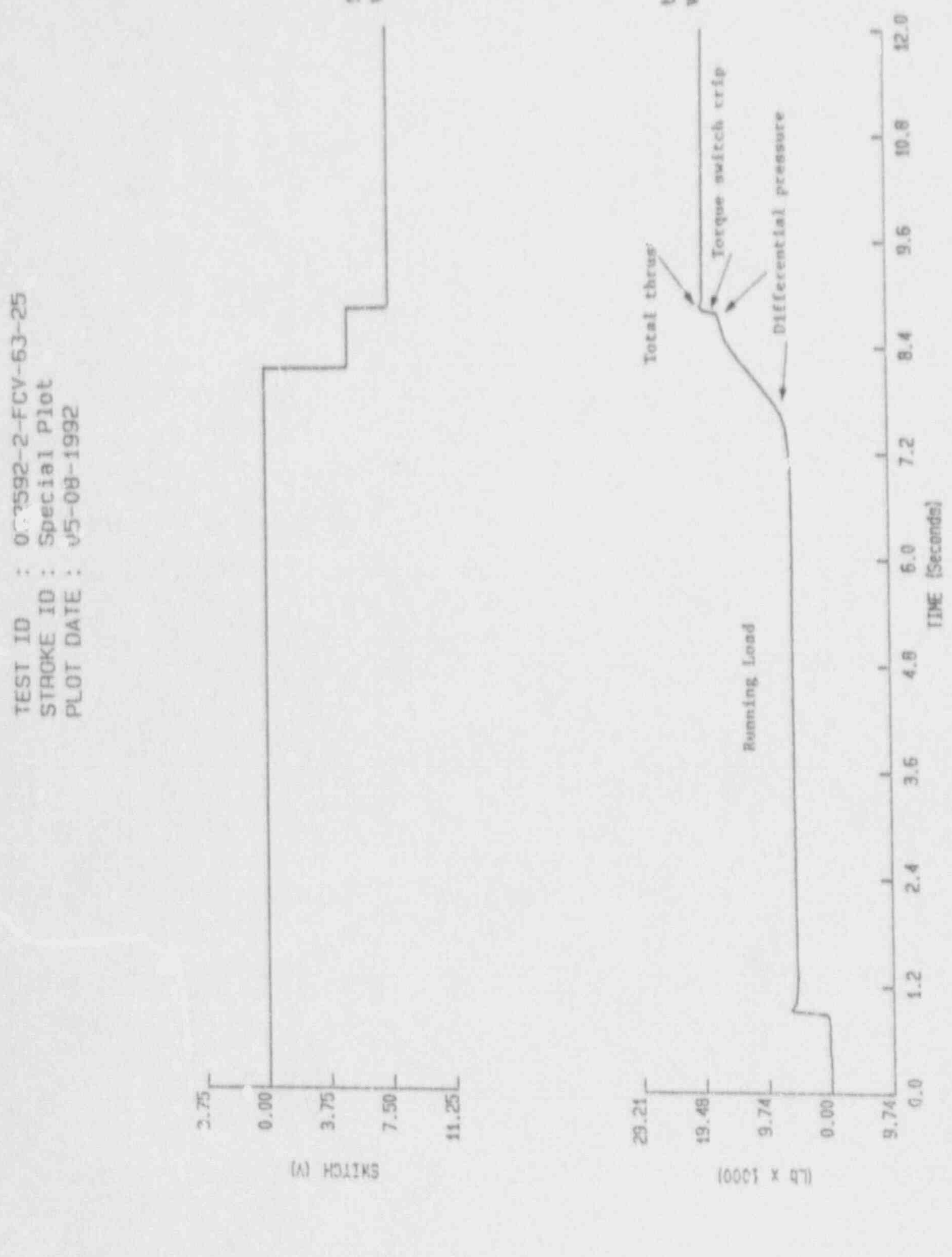

TEST ID : 592-2-FCV-63-25
STROKE ID : Special Plot
PLOT DATE : 05-08-1992
SWITCH
V12\T33
SWITCH (V)
thrust
V12\T31
(LB x 1000)
Total thrust
Torque switch trip
Differential pressure
Running Load
TIME (Seconds)

Figure 3.

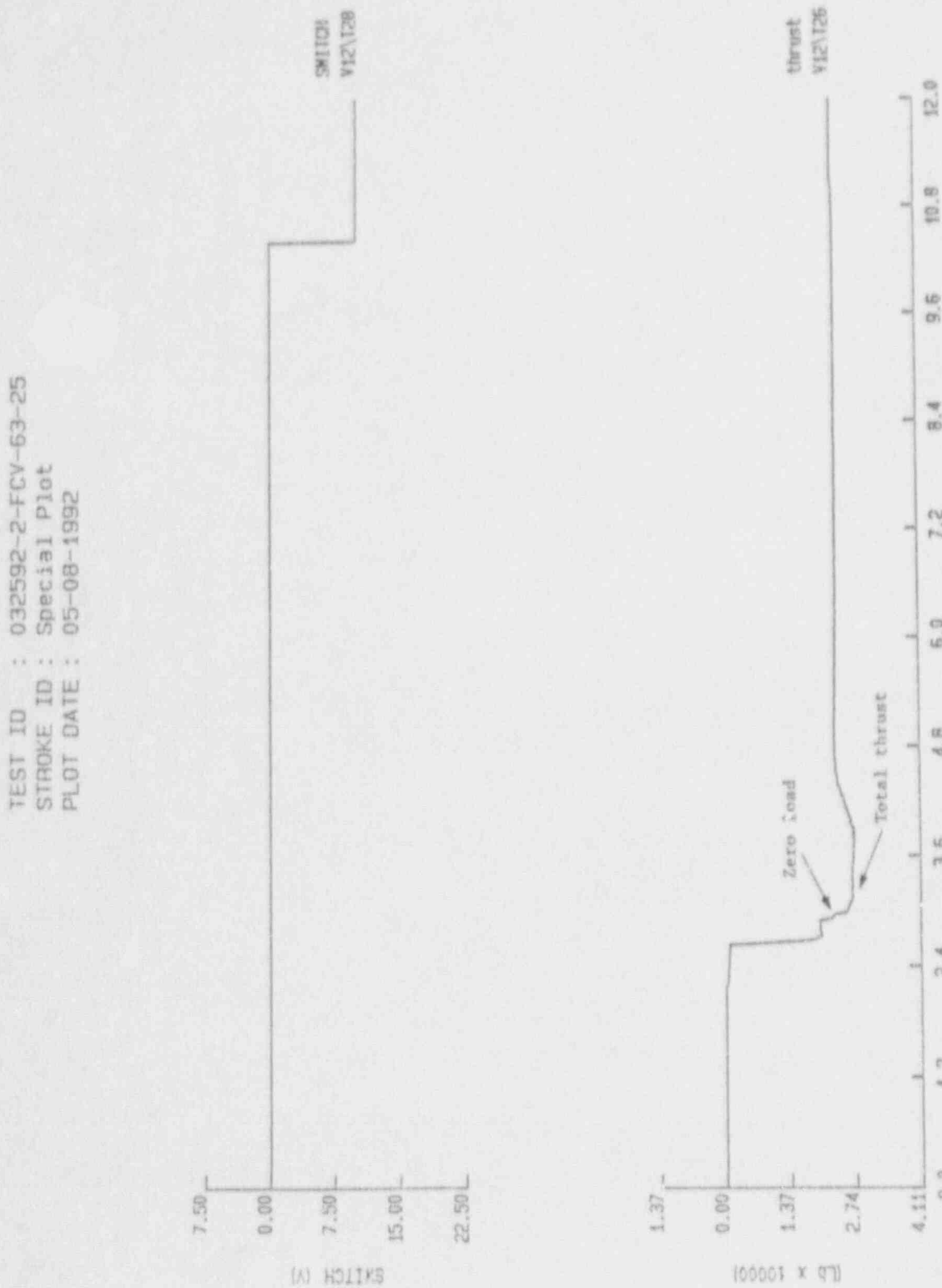

TEST ID : 032592-2-FCV-63-25
STROKE ID : Special Plot
PLOT DATE : 05-08-1992
SWITCH
V12\T28
SWITCH (V)
7.50
0.00
7.50
15.00
22.50
thrust
V12\T26
Zero load
Total thrust
(LB x 1000)
1.37
0.00
1.37
2.74
4.11
0.0
1.2
2.4
3.6
4.8
6.0
7.2
8.4
9.6
10.8
12.0
TIME (Seconds)

Figure 4.

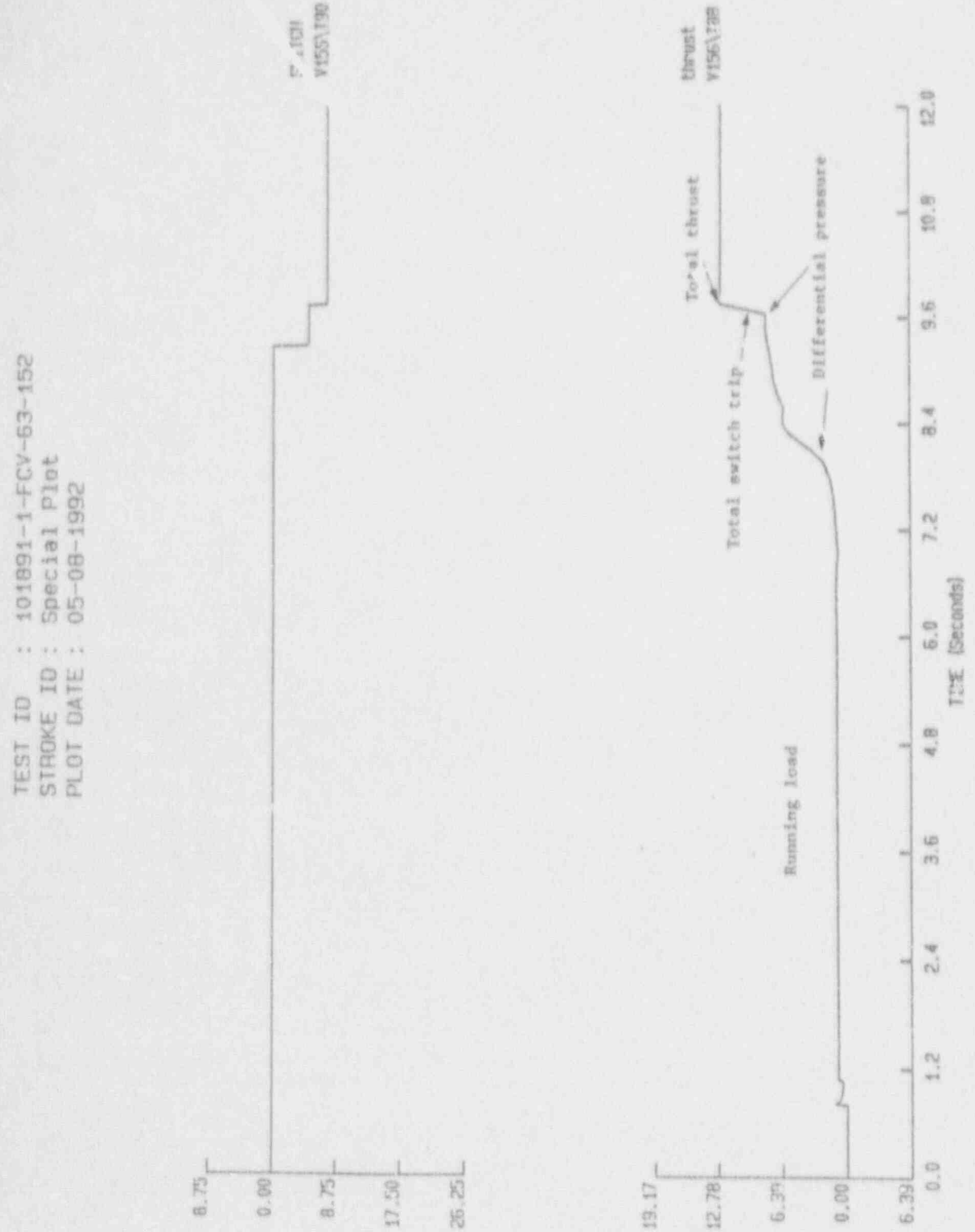
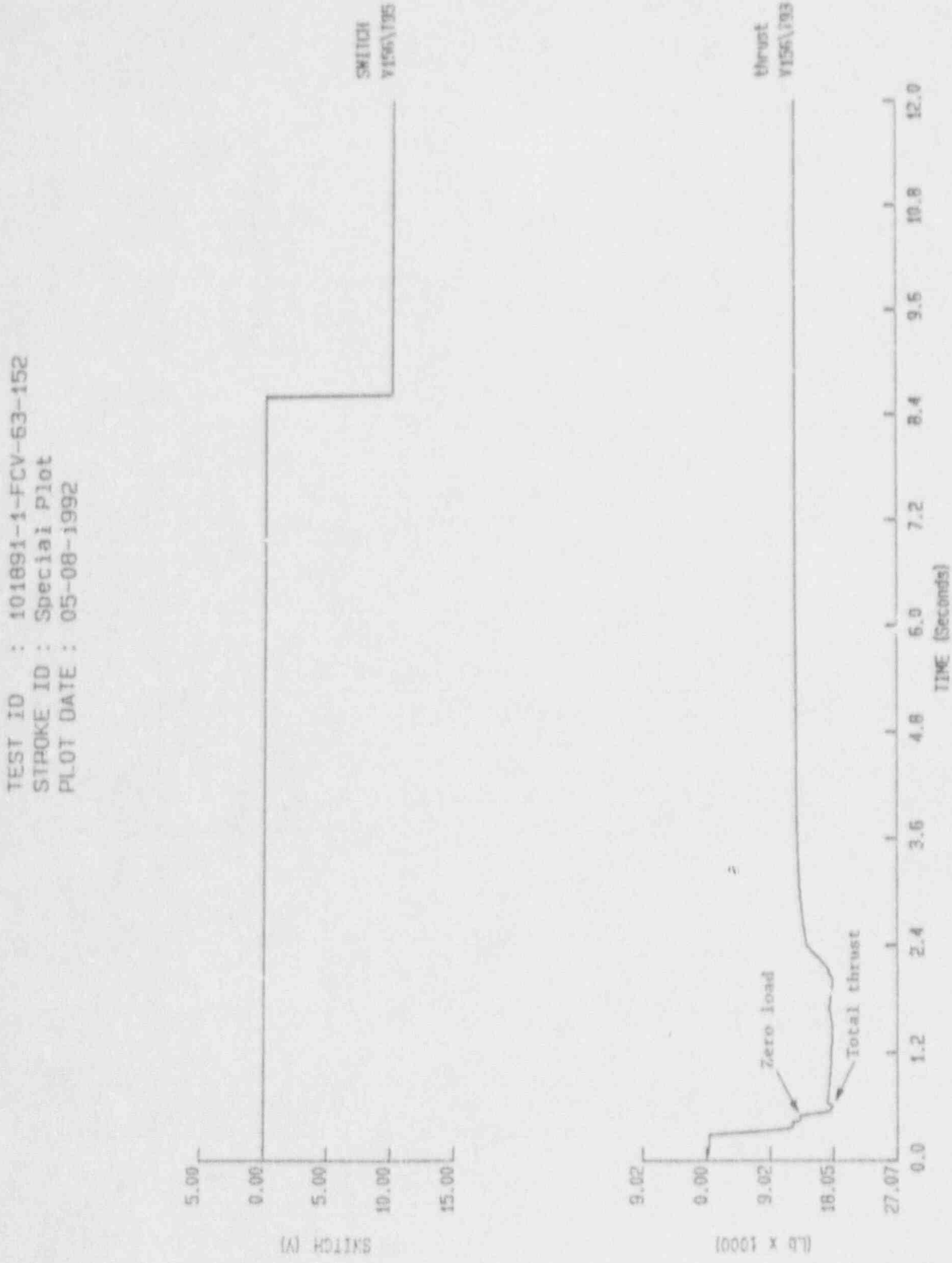

TEST ID : 101891-1-FCV-63-152
STROKE ID : Special Plot
PLOT DATE : 05-08-1992
SWITCH (V)
(LB X 1000)
TIME (Seconds)
Running load
Total switch trip
Differential pressure
Total thrust
thrust

Figure 5.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STROKE ID : Special Plot
PLOT DATE : 05-08-1992
SWITCH (V)
SWITCH
thrust
LB x 1000
Zero load
Total thrust
TIME (Seconds)

Figure 6.

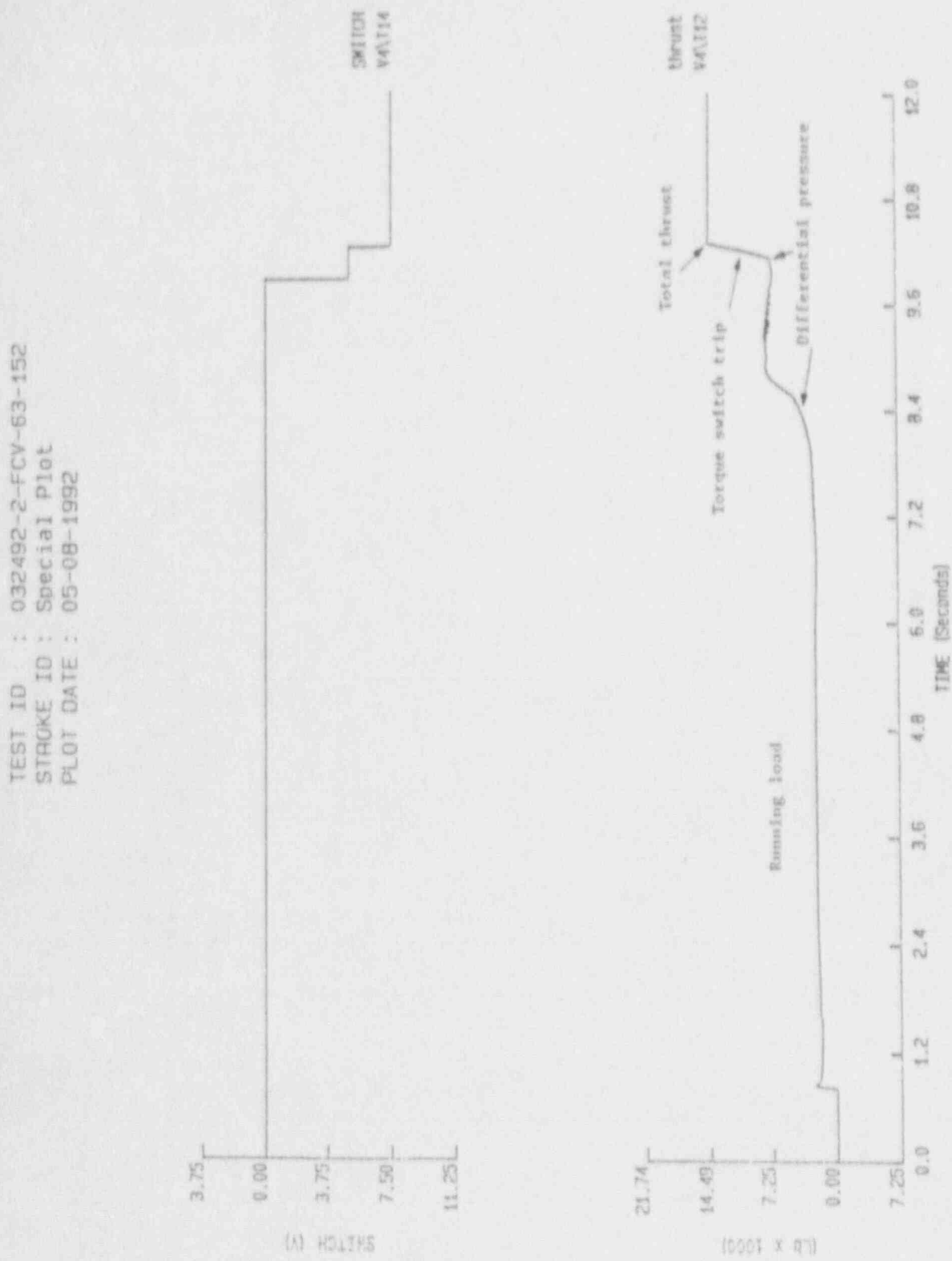
TEST ID : 032492-2-FCV-63-152
STROKE ID : Special Plot
PLOT DATE : 05-08-1992
SWITCH (V)
SWITCH
V4\T14
LB X 1000
thrust
V4\T12
Total thrust
Torque switch trip
Differential pressure
Running load
TIME (Seconds)
0.0
1.2
2.4
3.6
4.8
6.0
7.2
8.4
9.6
10.8
12.0
21.74
14.49
7.25
0.00
3.75
7.50
11.25

Figure 7.

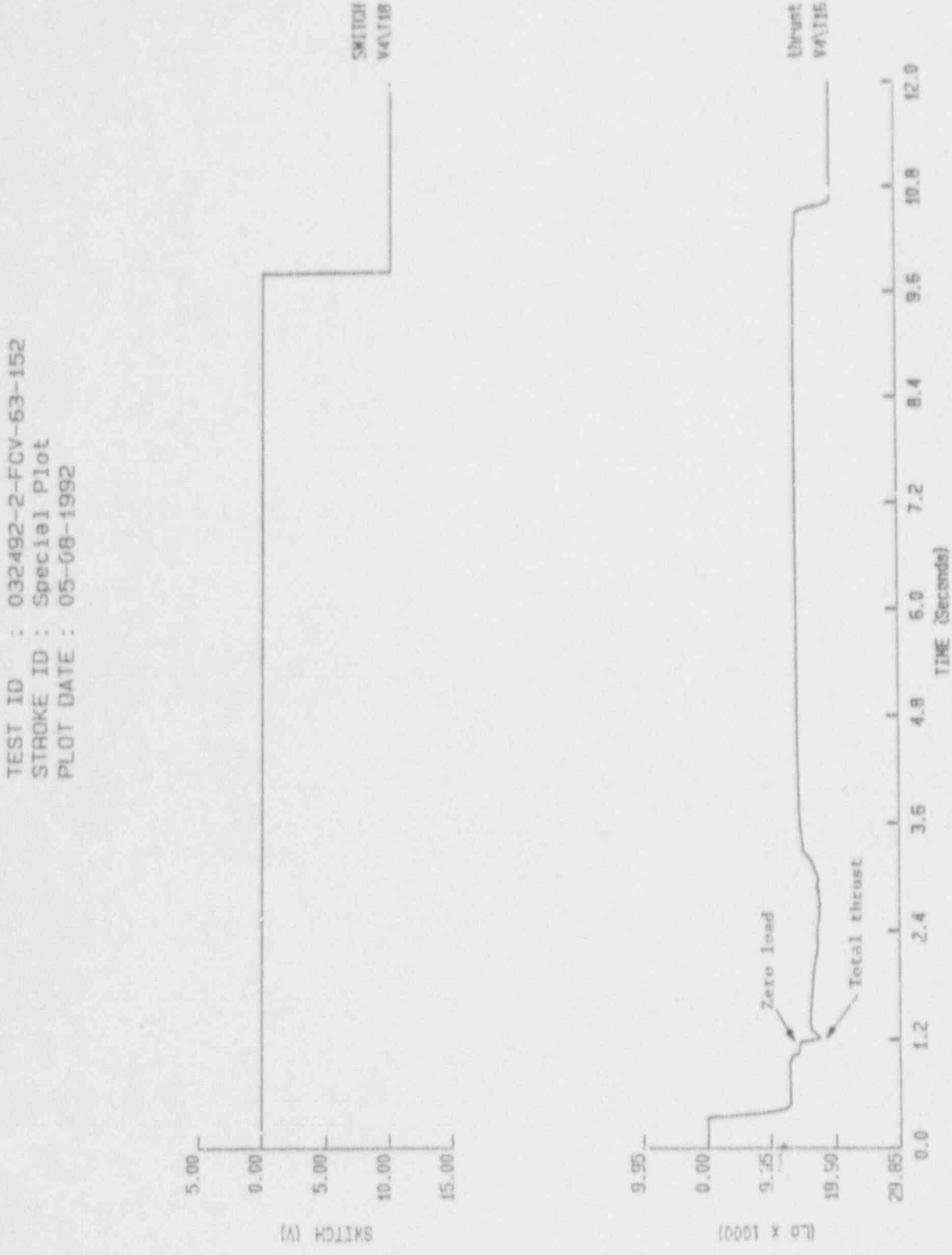
TEST ID : 032492-2-FCV-63-152
STROKE ID : Special Plot
PLOT DATE : 05-08-1992
SWITCH
V4\T18
SWITCH (V)
thrust
V4\T15
Zero load
Total thrust
TIME (Seconds)

Figure 8.

Session 1B

Pump Performance and Testing

Session Chair
Chris Pendleton
Pacific Gas & Electric

Pump Testing—Comparison of Factory vs. Field Test of Centrifugal Pumps

Rudolf Fehlau, BW/IP International, Inc.

ABSTRACT

Testing of pumps in situ, i.e., as installed in a system, will typically yield somewhat different performance results from the original manufacturer's factory test. This paper discusses some of the reasons for these variations. It shows that the factory test curves can be used for evaluation of initial acceptance tests but not for reference in normal inservice testing (IST). This is the basis for reference values used in American Society of Mechanical Engineers (ASME) Section XI specifications and the revised ASME Code 1990.

HISTORY

Historically, the "Reference Value or Values" for inservice testing of pumps was established in situ, after the pump was known to be operating satisfactorily. It was relatively easy to repeat during following ISTs, which were performed according to ASME Section XI, Operating and Maintenance (OM)-6 specifications, by checking the hydraulic performance and mechanical operation at one operating point, i.e., one head/capacity condition.

With time it became more and more difficult to evaluate deviations. At the same time more stringent compliance and survey requirements were imposed by regulatory agencies.

This led to the rewrite of OM-6, incorporating improvements to the standard. The rewrite, now known as ASME OM Code 1990, Subsection ISTB, retained the concept of the "Reference Value" as previously established, i.e., an operating condition that was easily obtainable in the existing installation, using in most instances the existing instrumentation. Concerns about the accuracy of the instrumentation and instrument loop readings resulted in establishment of tolerances for instrumentation and deviations of the measured parameters.

The proposed tabulation was still questioned by some members of the Review Board. One of the suggested solutions was to use the "Manufacturer's Factory Performance Test Curve" as the "Reference Value." This would require taking a number of field performance points of developed head vs. capacity at operating speed, and comparing these values with the original "Factory Test Curve."

This "Comprehensive Pump Test Specification" is now being drafted. The use of manufacturer's "Factory Test Curve" for comparison has been omitted in favor of an in situ test curve obtained when the pump was known to be operating satisfactorily, i.e., using the results of the preservice test.

The intent of this paper is to clarify the relationship between original "Manufacturer's Factory Tests" and results of "Field Tests," i.e., values obtained using actual piping configurations with permanent or temporary instrumentation.

HYDRAULIC CONSIDERATIONS

First, a few basic physical and hydraulic factors affecting centrifugal pump performance should be reviewed. They form the basis for further evaluations and correlations of the test results.

1. The relationship of pump flow, expressed as capacity (Q) in gpm, and the developed differential pressure, expressed as total head

(H) in feet of water, is fixed as shown on the H/Q curve for any individual pump unit operating at a constant speed (Figure 1).

2. A deviation from this H/Q curve is possible only if speed or impeller diameter is changed. This assumes that no physical or mechanical damage to the pump or excessive wear has occurred (Figure 2).

3. The actual point on this curve at which the pump is operating is a function of the system curve, i.e., the actual head/capacity value is a function of the pump/system interaction. It is the point where the system curve intersects the pump head/capacity curve at any given time (Figure 3).

Expressed in different terms, the pump cannot operate at any other H/Q point than the one dictated by the system resistance. The system resistance curve can change with a change in static head, resulting from change in liquid level or from change of pressure in the discharge vessel. A change of system friction, such as brought about by throttling a valve, will also change the system curve. The operating point is always at the intersection of the system resistance curve with the pump head/capacity curve. The pump head/capacity curve remains unchanged for constant speed operation, assuming no wear within the pump (Figure 4).

Now let's get back to the differences and correlations between "Factory" and "Field" tests.

FACTORY TEST

1. A factory pump test is equivalent to a laboratory test. The test setup conforms to the requirements of the Hydraulic Institute Standards for Centrifugal Pump Testing. All conditions are optimized to obtain the most accurate evaluation of the pump performance. Required straight pipe runs before instrument locations ensure stable and normalized flow conditions for correct metering of all data (Figures 5 and 6).

2. All instrumentation complies with the accuracy, repeatability, and reliability requirements of the Hydraulic Institute Standards. Therefore, it satisfies requirements of the ASME Test Code and the AWWA Code, as well as other applicable laboratory standards.

3. The instrument pressure tap connections are prepared to minimize flow disturbance, thus minimizing test instrument reading fluctuations. This results in optimum accuracy (Figure 7).

4. All instrumentation is connected at points with minimum impact from system losses, thus producing a true indication of the actual pump performance.

5. All measured test values are corrected for velocity head, elevation in relation to datum elevation and instrument calibration variations, and the H/Q curve is plotted using H/Q values converted to a constant speed.

The resulting "Manufacturer's Factory Performance Test Curve" is an accurate representation of how the individual pump will perform, i.e., discharge liquid under specific conditions.

Compare this to a given installation of this individual pump.

FIELD TEST

1. Pipe configurations are designed to maximize the use of available space, and in most cases to minimize the effects of thermal expansion and resulting loads and/or stresses. In most cases this results in pipe runs with many elbows, resulting in turbulent and unbalanced flows at instrument locations, which affects the accuracy of the readings. The location, orientation, and type of regulating valves, isolating valves, and check valves also affect accuracy of the readings because they affect the flow condition at the instrument measurement location. In addition, sidestream leakage flows may not be accounted for in the test.

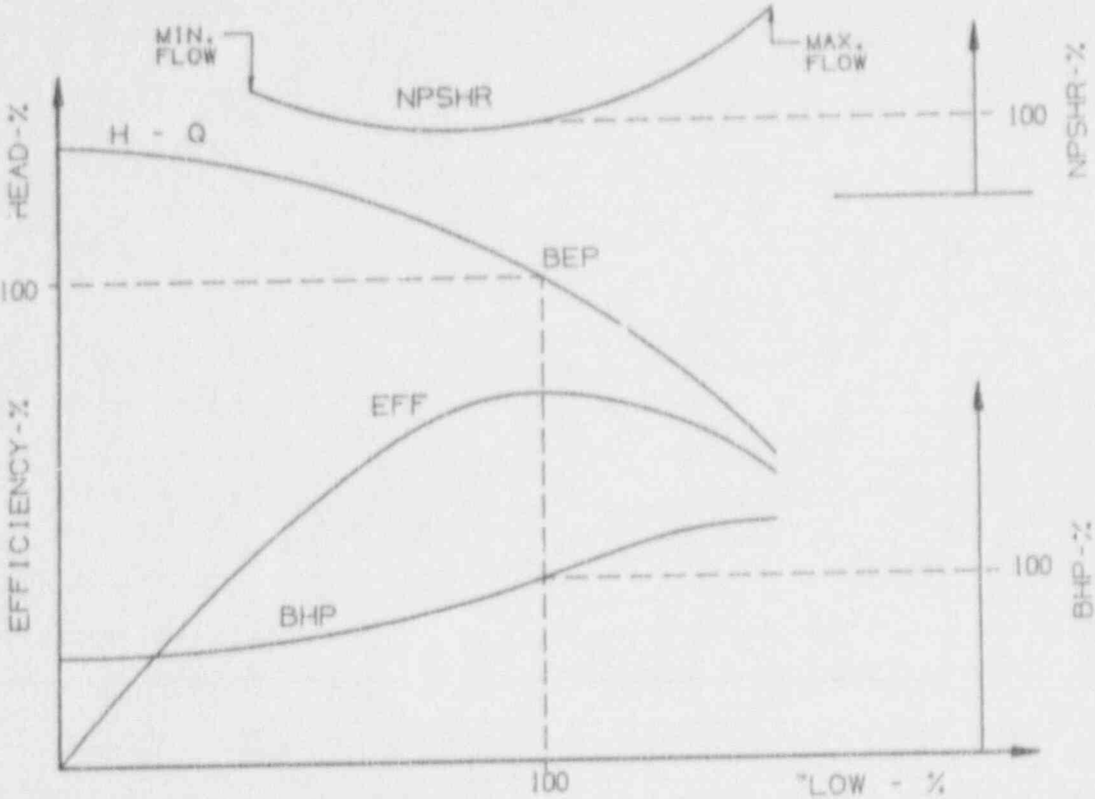

Figure 1. Typical pump performance curve.

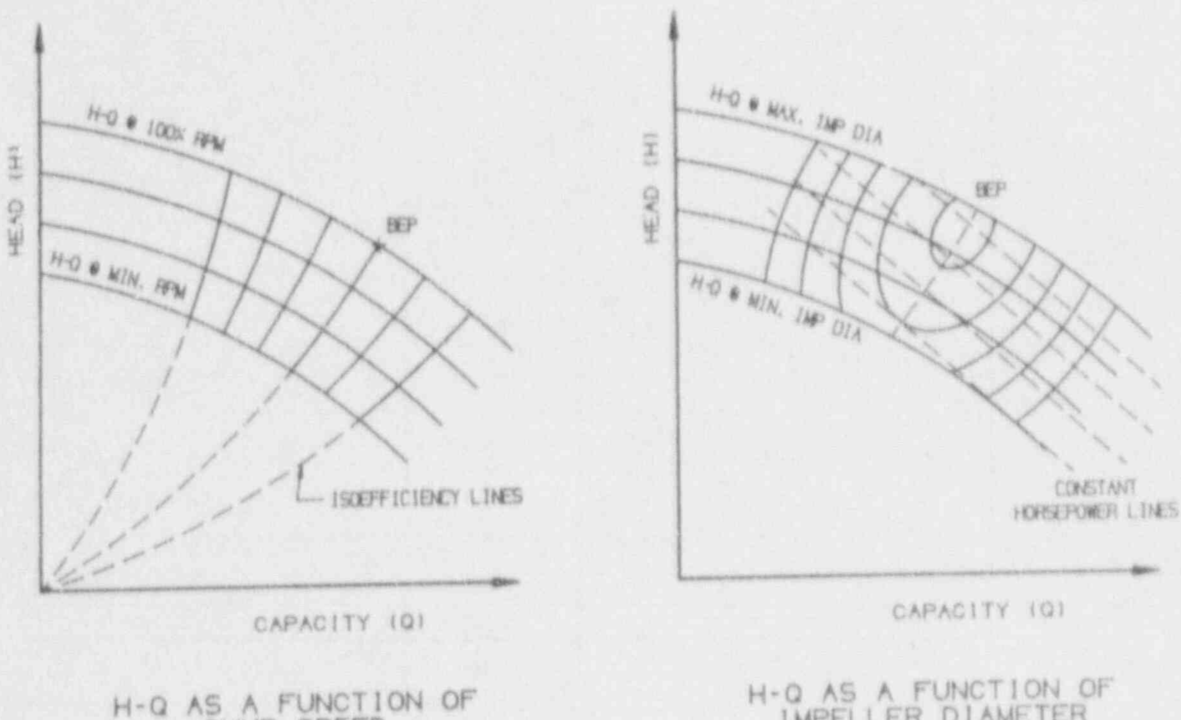

Figure 2. H/Q curve showing differences with speed and impeller diameter.

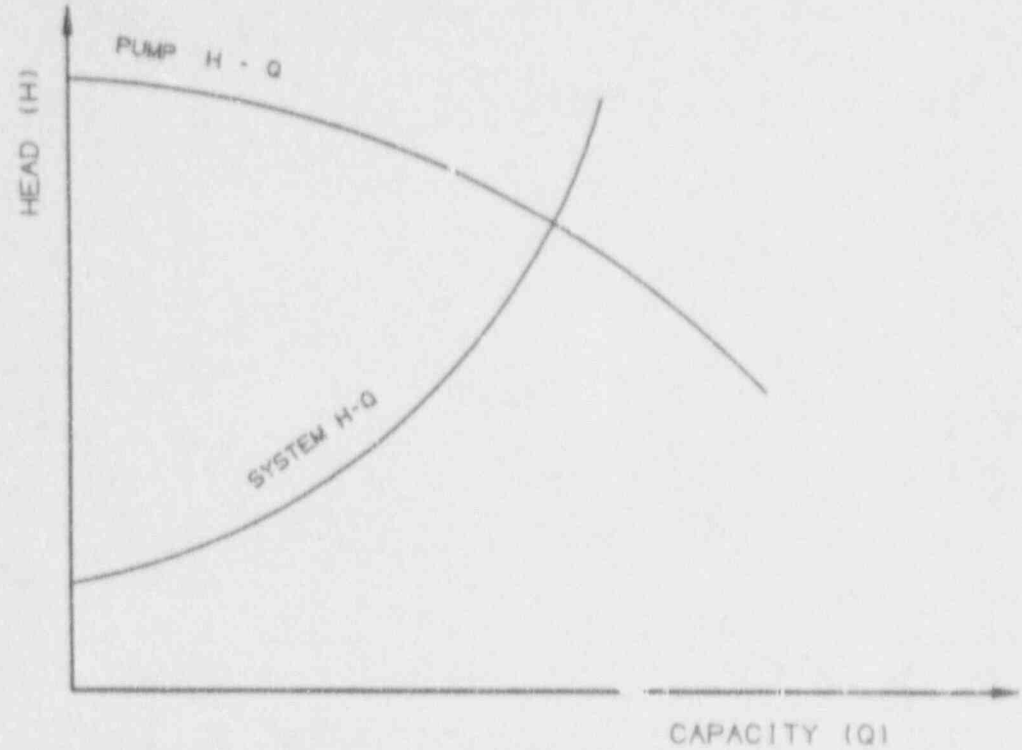

Figure 3. Typical system head curve.

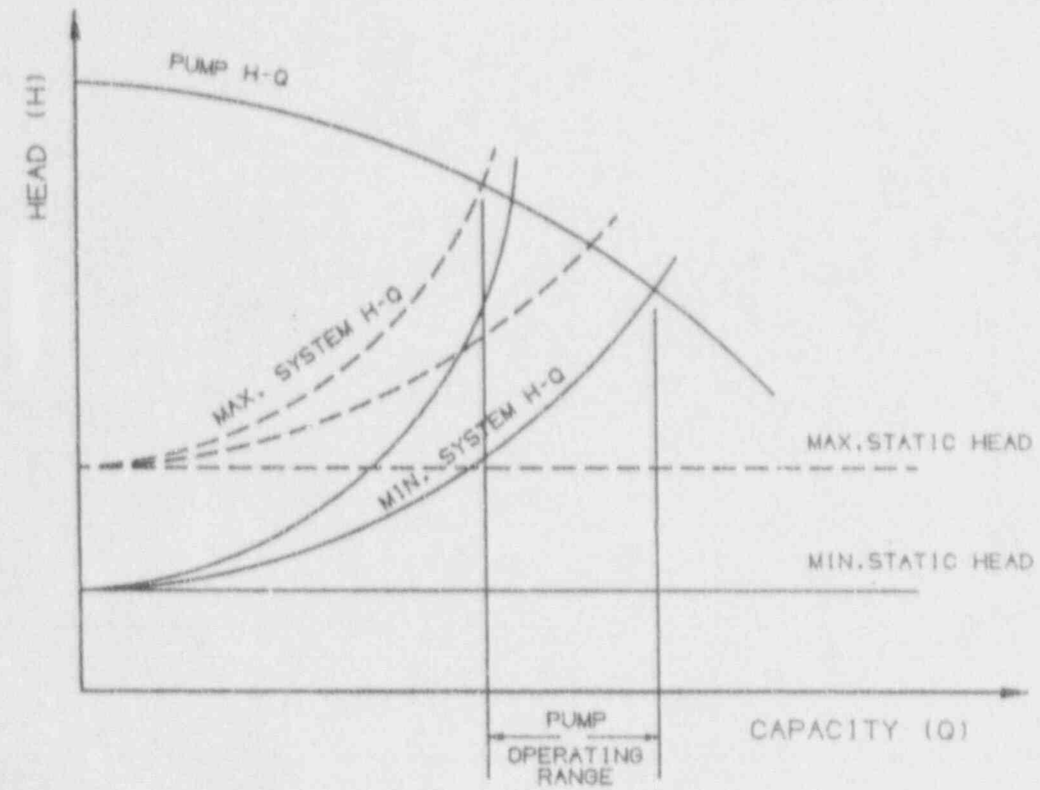

Figure 4. Pump operating range as determined by pump and system curves.

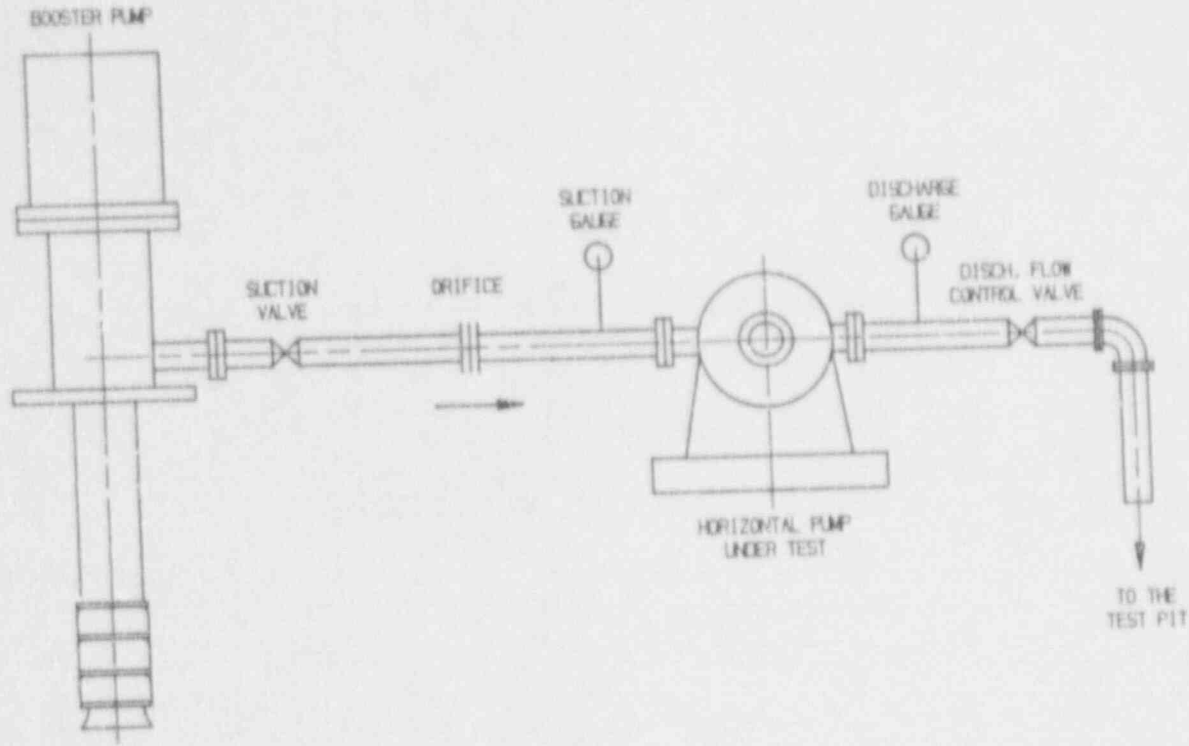

Figure 5. Horizontal pump open pit test loop.

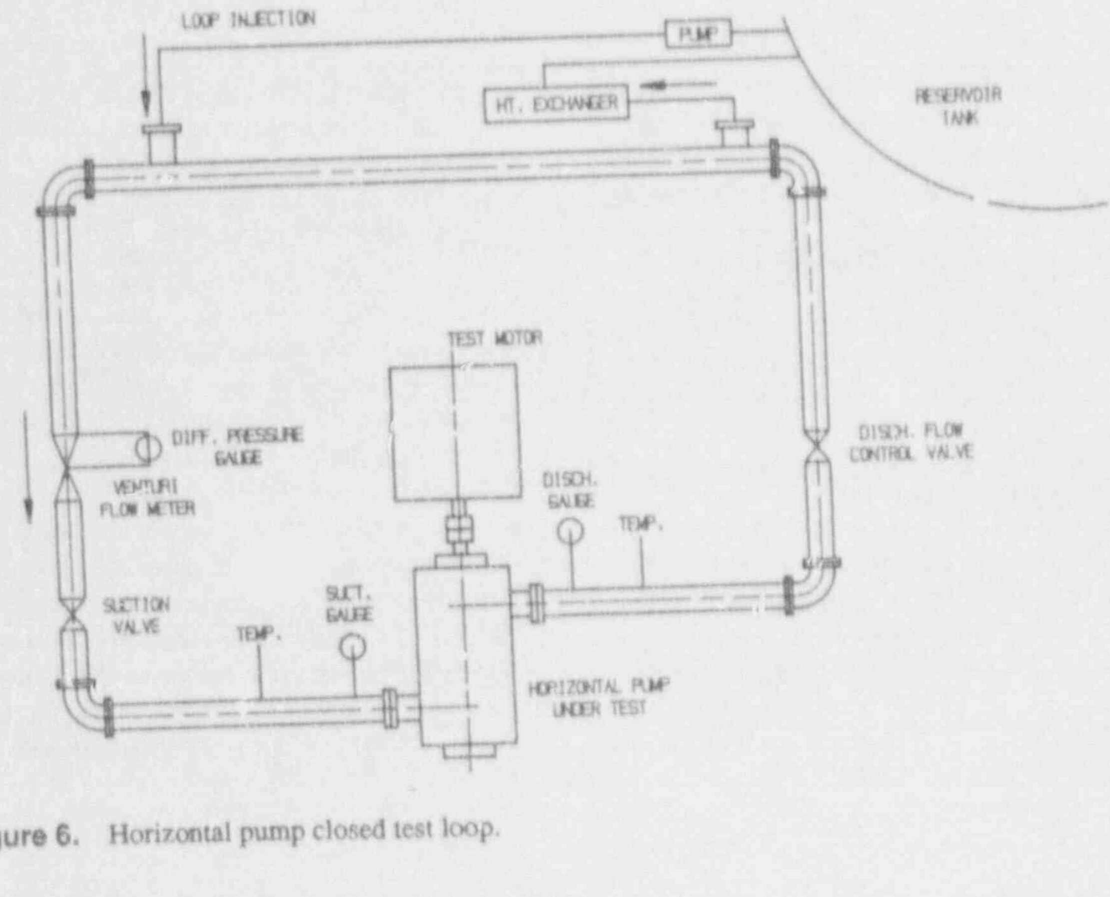

Figure 6. Horizontal pump closed test loop.

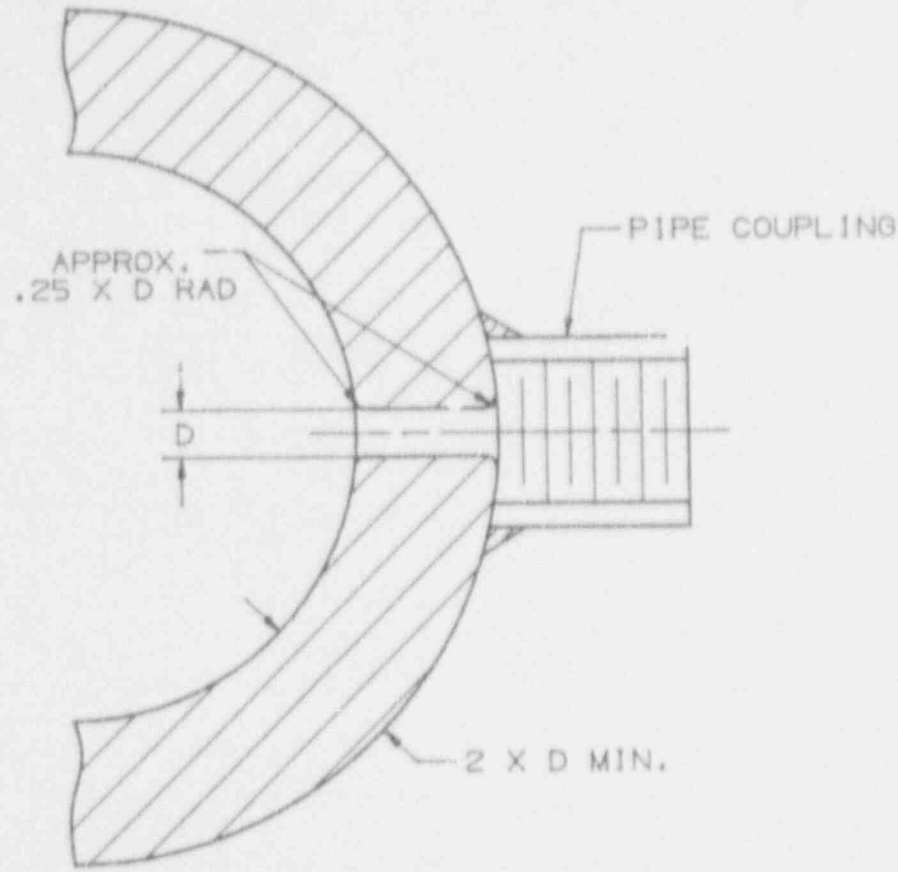

Figure 7. Welded on pressure tap opening.

2. Normally, permanently installed instruments are not of laboratory accuracy and/or within the laboratory required ranges. They fully satisfy system control requirements, but are not necessarily correct in the absolute values of the readings.

3. Measuring instrument connections are located at the best available points; however, in most cases the minimum required straight run or flow straightening devices cannot be provided, which affects the accuracy of the readings.

4. Velocity head and datum elevation head corrections are not usually applied to system and field test values, nor are the values converted to a constant speed condition.

5. An accurate determination of flow rate is usually the most difficult measurement to obtain in a field test. All flowmeters require adequate lengths of straight pipe runs, both upstream and downstream. Unless these runs are provided in the system design, accurate flow measurement is not possible.

Considering the variations between "Factory" and "Field" test arrangements and provisions, clearly differences and variations will exist between the two. An evaluation of these variations must be made before the two can be compared.

In some cases field tests are specified as acceptance tests, with the prearranged condition that field installation incorporates provisions for accurate measurements. However, this is not the case with the majority of pumps subject to IST under ASME OM-6 or the present standard, ASME OM Code 1990, Subsection ISTB.

CONCLUSION

What is the best approach to improved IST of safety-related pumps in nuclear power plants? The initial acceptance must be determined by what constitutes acceptable operation of the individual pump unit in its specific system, i.e., "as installed." This should be established, or should have been established, during initial plant commissioning and the pre-service test. At that time the "Reference Value" for IST was or should have been established. If expanded IST surveillance will be initiated later, it must be based on a tangible condition at the time the expanded criteria were established. In other words, if the pump is now known to be operating satisfactorily, based on the presently known "Reference Value," i.e., operation at one designated H/Q point, then a multipoint H/Q curve can be established by test and designated as the "Class A Test Reference." This can then be used for future testing of the pump and evaluation of degradation of pump operation from wear, corrosion, erosion, or other effects.

The presently approved draft for comprehensive pump testing uses this method. It will provide increased data for IST evaluation.

Comparison of the "Field Test Curve" to the "Factory Test Curve" can be a only one time engineering action, and not a continued point of review and controversy.

REFERENCES

ASME OM Code 1990.

Hydraulic Institute Standards, 14th Edition.

GLOSSARY OF TERMS

Acceptance Test	-	Factory or field test, depending on applicable purchase specification
AWWA	-	American Water Works Association
BEP	-	Best efficiency point
Comprehensive Test	-	Expanded IST test performed at several H/Q points
Datum	-	Reference elevation determined by pump type
Factory Test	-	Pump test performed at manufacturer's test facility
Field Test	-	Pump test performed after pump is installed in its operating position
H/Q Curve	-	Head (H) developed by the pump as function of flow (Q), plotted for full range of flow
H/Q Point	-	Specific value of head (H) at a specific value of flow (Q), see Reference Value
In situ Test	-	Same as Field Test
IST	-	Inservice test performed at specified intervals according to ASME OM Code 1990
OM	-	Operating and Maintenance
Performance Curve	-	Graphic representation of pump operating characteristics
Pre-Service Test	-	Initial field test of pump in its system, after having been accepted per acceptance test (Can be the acceptance test)
Reference Value	-	A specific H/Q operating point established during pre-service test
Velocity Head	-	Kinetic energy of the liquid at a given cross section ($hv = \frac{V^2}{2g}$)

Pump Monitoring and Analysis

Kevin R. Guy, President
C J Analytical Engineering, Inc.

ABSTRACT

The paper describes how to set up a periodic vibration monitoring program implemented with electronic data loggers. Acquired data will be analyzed and evaluated to determine pump condition. Periodic measuring frequency, reporting procedures, and conditions of mechanical components will be discussed in detail based on the actual case study.

INTRODUCTION

The technological advances of data collectors over the last 3 years have strengthened predictive maintenance techniques. Good vibration data is still the backbone of a solid predictive maintenance program; however, there is more to predictive maintenance. Predictive maintenance is vibration data collection, vibration analysis, testing, engineering, operations, maintenance and quality control all wrapped up into one program. The growth of the data collector and its analysis ability have allowed most of the vibration analysis required of the trend data and further testing to be done by one piece of equipment. The goal of this program is to increase the unit output, availability and reliability through monitoring and analysis while lowering the maintenance cost.

The one item everyone needs to be aware of is that the predictive maintenance program does not happen overnight; it occurs over time. The normal gestation time in the past has been 3 to 5 years. Present-generation data collectors typically speed this process by one year, so now you can have an effective program in two to three years. The first several years are going to be filled with eliminating all the little equipment problems no one has taken the time to solve. During this time period, a vast amount of learning will take place. The biggest investment management has to make is their commitment to the program. Management and corporate commitment make a solid program.

The initial stages of the program will be costly with the purchase of equipment, training of personnel, and the increase in maintenance costs. The misconception encountered with the birth of a predictive maintenance program is that it will save money immediately. The program will initially increase the cost of plant maintenance while you eliminate problems found during the early stages of your program. Management and maintenance people are usually shocked by this. It usually results in someone saying, "This program doesn't work," or "We never had this problem until we started predictive maintenance." The best advice is to forewarn management personnel of the possible consequences involved with starting programs. You have to spend money to make money.

The engineer responsible for predictive maintenance should review the maintenance history on the plant pumps and their drivers for 5 years. This will allow viewing the trends in equipment maintenance and problems. Looking at 5 years of history will give the engineer enough data to see if there are any repeat problems.

Once the investigation is complete and the past maintenance history of these pumps has been researched, you are now able to move on to the program setup. One thing never to forget while performing predictive maintenance: Don't expect the diagnostic equipment to find all your problems and analyze them for you, especially if you are using expert systems. The diagnostic equipment will help you keep tabs on the equipment health; but, you still need to understand how the equipment works and what its function is. The initial problems will be equipment related. However, later in the program many vibration problems are operational, system-related or a result of engineering design. A lack of understanding in this area will lead to many unsolved problems.

SETTING UP A PUMP MONITORING PROGRAM

Equipment File Packages

The first item of business in setting up a program is to learn as much about the equipment as you can. This will entail putting together an equipment file package. The equipment file package will be the first step in the learning process on how the equipment operates and what its function is in the system. Appendix A contains the section on pumps from the master information sheet. This section of the sheet will contain the pertinent information required for the analysis of pump vibration problems. The information required for this sheet may not be readily available in the early stages, so you will be constantly adding to it as information becomes available. Examples of this would be the critical speed and balancing information that will become available after balancing is performed on the equipment and bode plots are generated. Alignment tolerances may also fall into this category if the equipment OEM is trying to sell his services. All other information required to complete these sheets should be obtained while setting up the program.

The equipment file package should also contain prints of the equipment layout (Figures 1 and 2), driver and driven, along with foundation information. These prints should be reduced down into a basic scheme as in Figure 3. Also needed is a pump curve (Figure 4). This should complete the basic information required to do a basic analysis should a vibration problem arises. The equipment prints are needed to ensure that you develop the cor   t data collection points.

EQUIPMENT CATEGORY BREAKDOWN

The next priority is to divide your pumps into categories corresponding to their mounting. These groups should be vertically and horizontally mounted pumps. The horizontally mounted pumps should be further broken down into center-hung or overhung.

This breakdown should be in addition to the normal group breakdown corresponding to their importance to the plant operation. Group 1 will be any capability reducing pumps (i.e., boiler feed pumps). These pumps should be monitored every 2 to 4 weeks. Group 2 will be any pumps that have a redundant backup (i.e., main condensate pumps). These pumps should be monitored every 4 to 8 weeks. Finally, Group 3 is any miscellaneous pumps that don't fall into the first two groups [i.e., low-pressure service water (LPSW) backwash pumps]. These pumps should be monitored every 8 to 12 weeks.

Knowing how equipment is mounted will give some insight into possible problems encountered when troubleshooting your pumps. It will also allow you to start a case history file on pumping problems encountered with different types of mounting schemes.

MONITORING EQUIPMENT

This can be a very difficult part of your program, if you make it so. If you have completed the equipment file package, you should have developed a feel for past pump problems. Knowing what the past problems have been will give you insight into conditions to look for. Armed with this information, you should now be able to look into different types of monitoring programs.

If this is the first predictive maintenance program you or your plant have been involved in, be very careful when choosing the type of equipment you will use. I do not recommend building an online monitoring system to do the job unless you have had several years of vibration data with which to design your system. Also, continuous monitoring systems DO NOT replace the need to

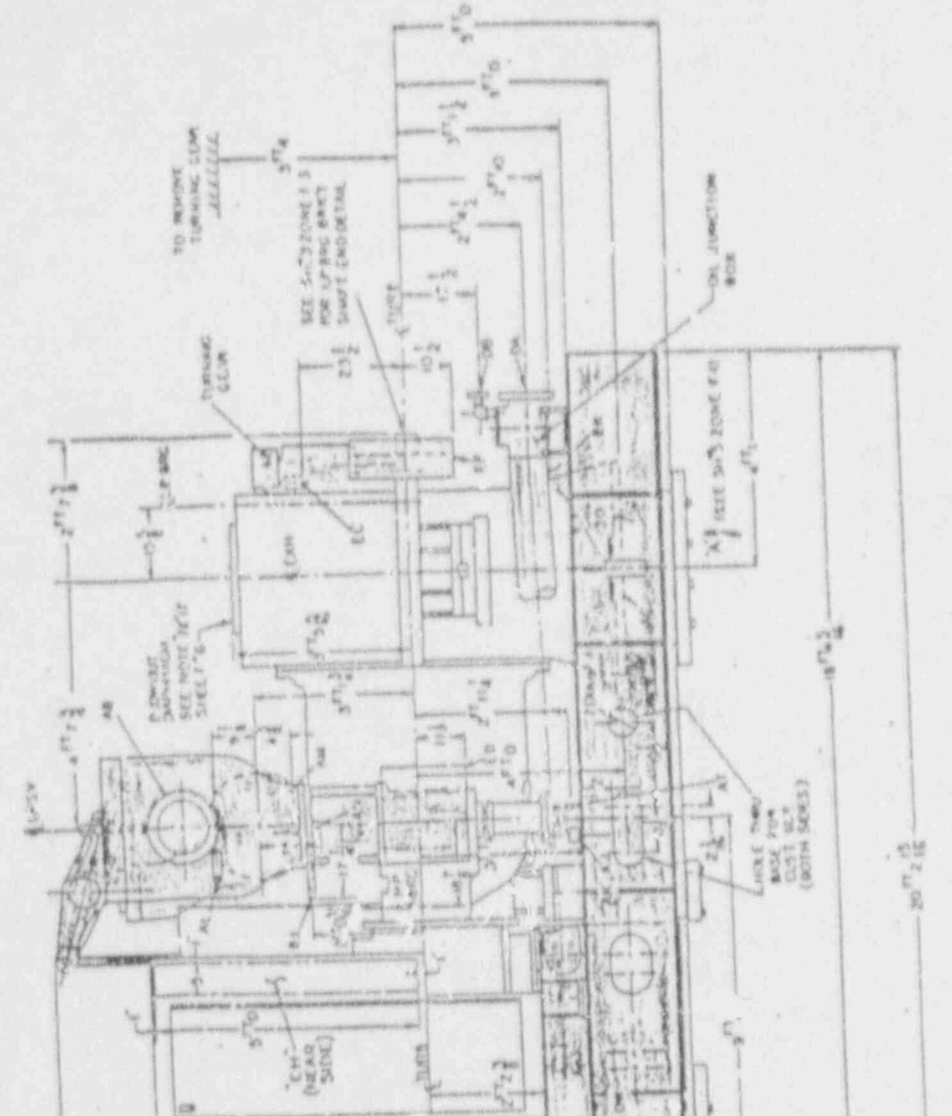

Figure 1.

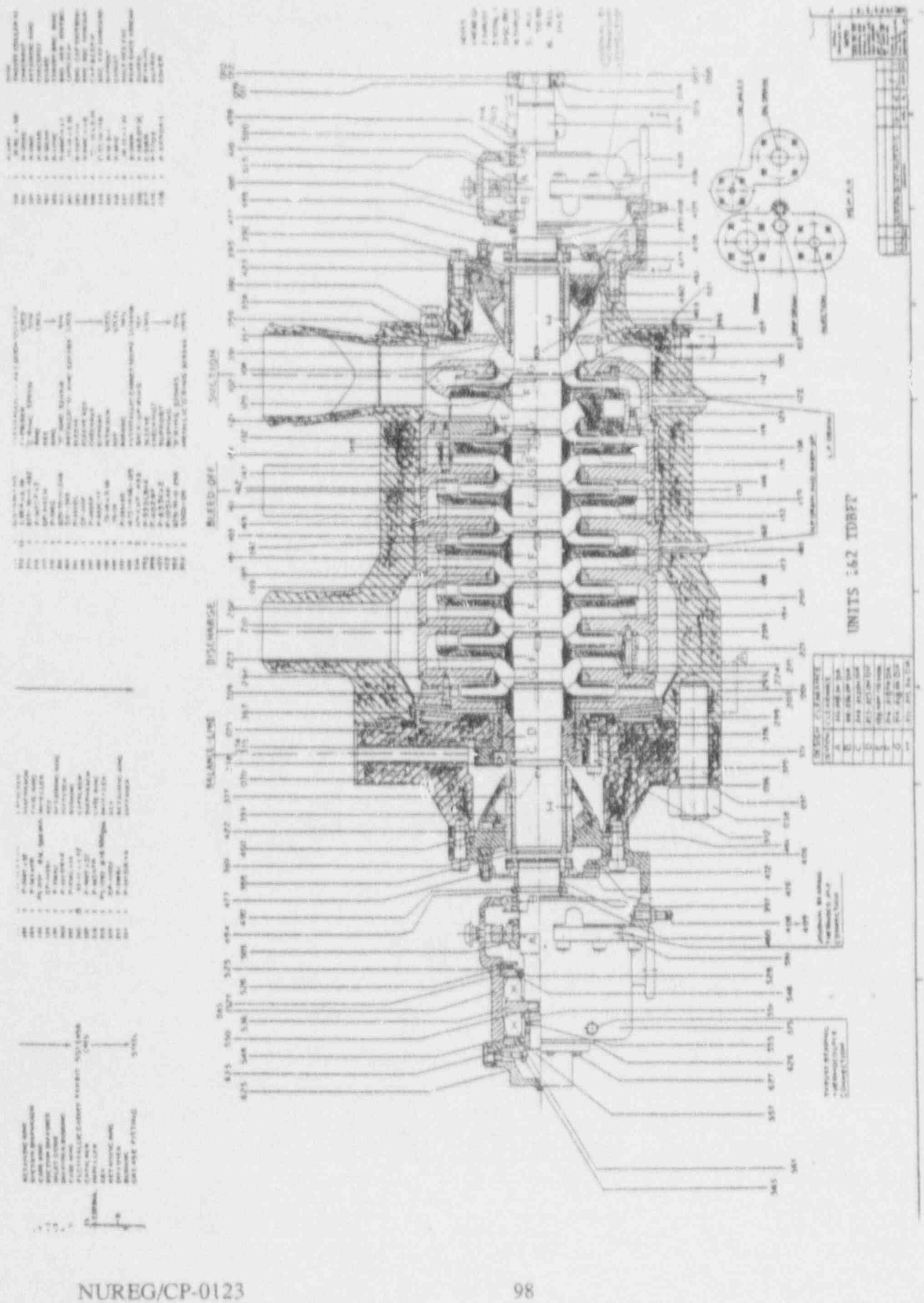

Figure 2.

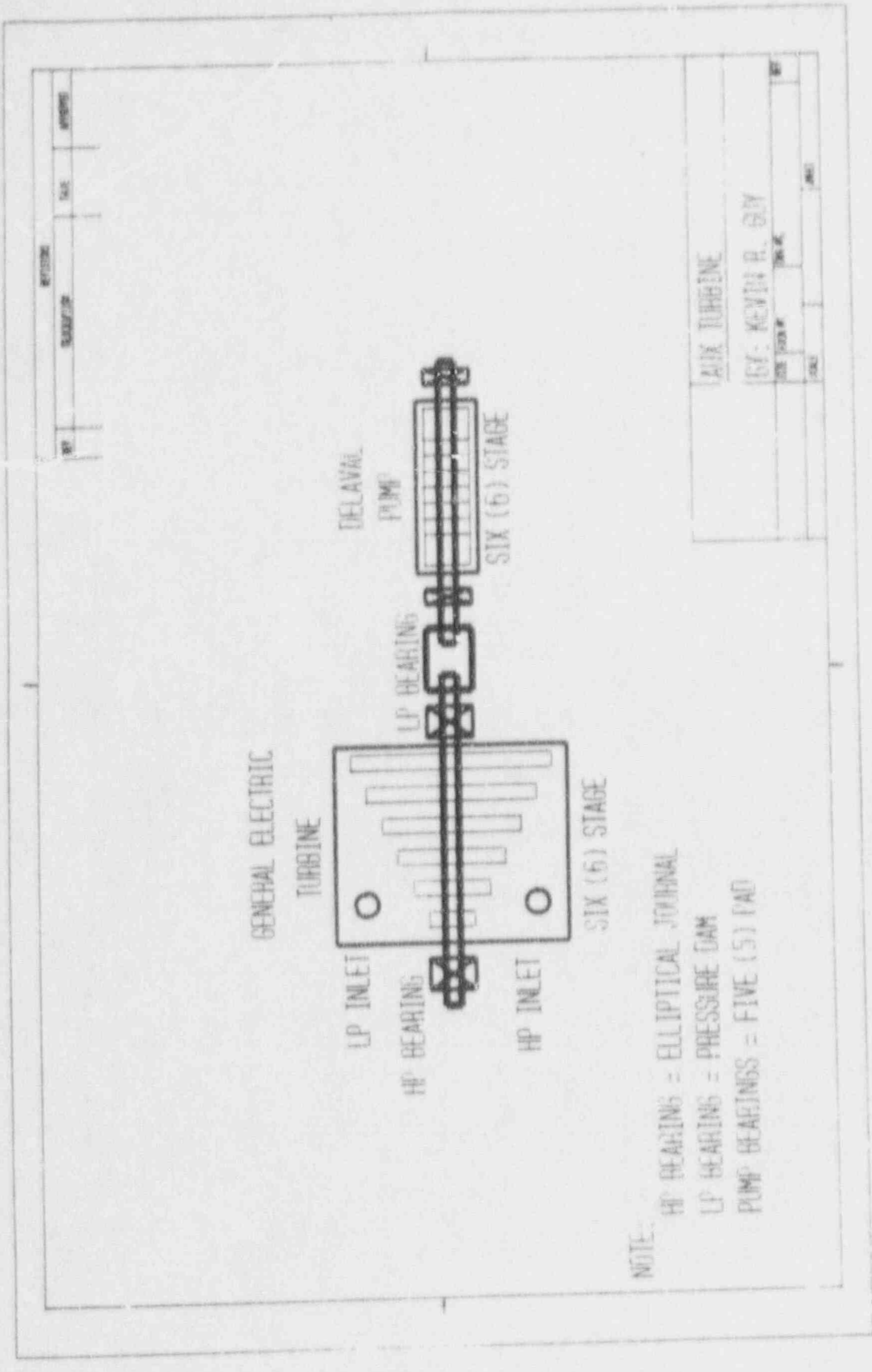
GENERAL ELECTRIC
TURBINE
LP INLET
HP BEARING
HP INLET
SIX (6) STAGE
LP BEARING
DELAVAL
PUMP
SIX (6) STAGE
NOTE:
HP BEARING = ELLIPTICAL JOURNAL
LP BEARING = PRESSURE DAM
PUMP BEARINGS = FIVE (5) PAD
AUX TURBINE
BY: KEVIN R. GUY

Figure 3.

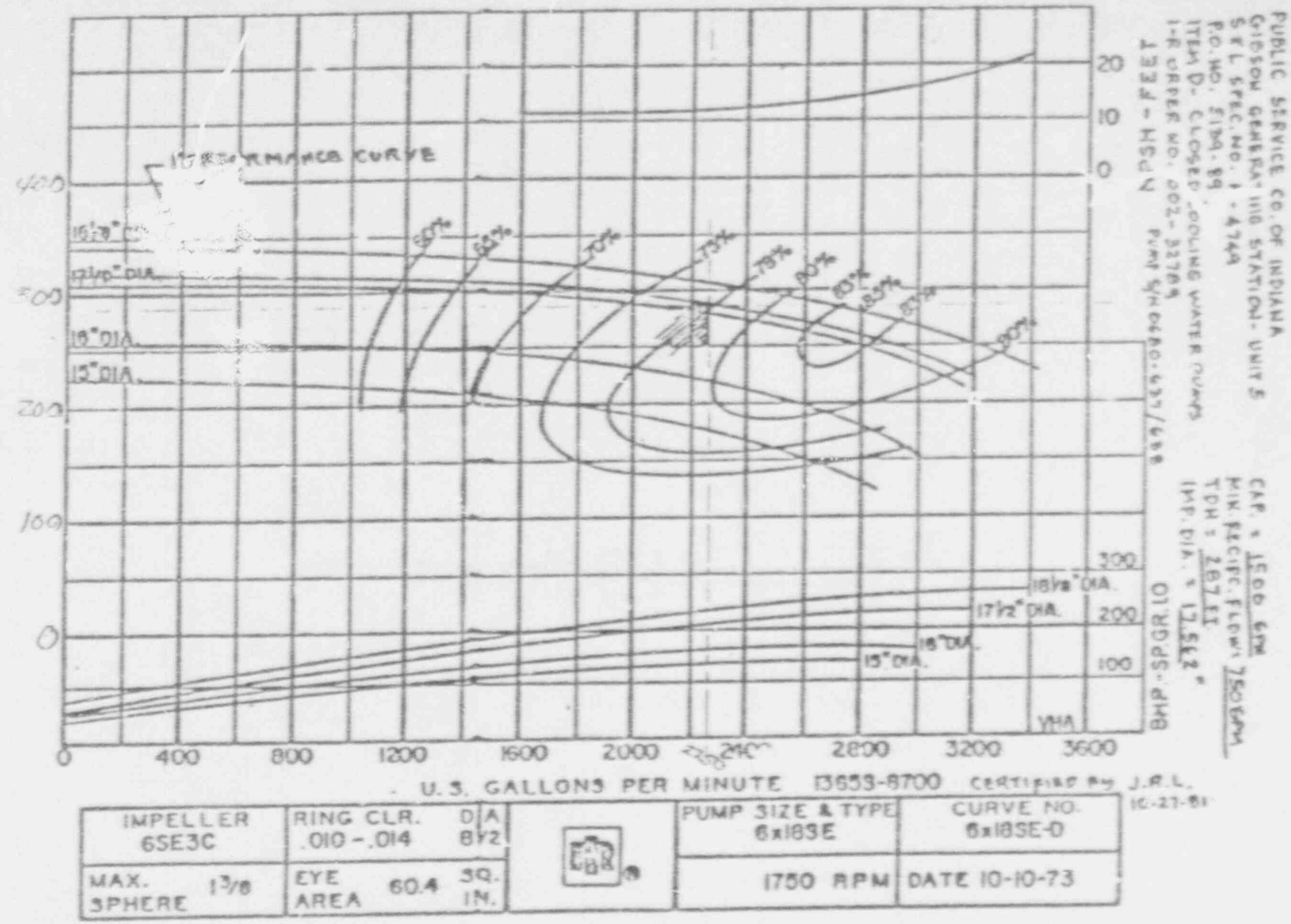
PUBLIC SERVICE CO. OF INDIANA
GIBSON GENERATING STATION - UNIT 5
CAP. = 1500 GPM
TDH = 287 FT
U. S. GALLONS PER MINUTE
NPSH - FEET
18" DIA.
15" DIA.
17½" DIA.
IMPELLER
6SE3C
RING CLR.
.010 - .014
MAX. SPHERE
EYE AREA
60.4
SQ. IN.
PUMP SIZE & TYPE
CURVE NO.
1750 RPM
DATE 10-10-73

Figure 4.

monitor the data on a periodic basis. Continuous monitoring systems raise a flag for problems that occur suddenly.

Data loggers are excellent for taking periodic data, and the software that accompanies the loggers allow you to trend vibration data and other equipment parameters. They will allow you to take the overall readings, bandwidth readings along with the time and spectrum plots.

The analysis capabilities of the data loggers in the past, in my opinion, were minimal or very poor at best. The present-generation data collectors have excellent analysis capabilities and should be able to handle the majority of the equipment problems that the vibration analyst encounters (Guy, 1992).

DATA POINTS

There are many vibration equipment manufacturers who say to trend only a few points (2 to 3 points) on each monitored equipment train. The key to choosing the correct number of points rests with the type of program that your are trying to establish. During the initial stages of any vibration monitoring program, 6 to 8 points should be taken to establish a starting trend. Figures 5 to 9 show data points that should be used for initial monitoring. It is important that on belt-driven equipment and equipment with rolling element bearings the data be taken in the load zone. Normally, during the first data collection, trend data is taken along with timewave and spectrum plots. Once this data is analyzed for problems, spectrum and time plots are taken only when alarms are triggered. Then the analyst moves into the analysis phase (Guy, 1992).

During the initial stages of a vibration monitoring program, the analyst is trying to get a feel for what the problems are and what the normal vibration amplitudes should be for the overall levels and the bandwidth levels. In the early stages of the program, databases are normally set up from the generic sets provided by the data collector and software manufacturers. Once the initial monitoring phase is complete, the analyst should build new databases or add to the present databases based on the experiences observed during the monitoring. The creation of new databases or additions to the present databases allows for movement into the predictive maintenance mode or monitoring and analysis part of the monitoring program.

Using the present-generation data collector and software will take approximately 2 to 3 years to allow entry into a monitoring and analysis program.

Predictive maintenance, as stated earlier, does not happen overnight. You don't just jump in and start predictive maintenance. Predictive maintenance is the culmination of a minimum of 2 years' work in vibration monitoring. Vibration monitoring will allow you to find the normal operation characteristics of your equipment. Without performing the initial vibration monitoring, you will have no baseline to judge your data. Many predictive programs fail because people don't allow enough time to gather initial trends and baseline plots, and don't develop a good database. During this vibration monitoring period, many equipment problems will be found that need to be corrected before a reliable predictive maintenance program can begin. The problems you will encounter are alignment, balance and mechanical looseness. These problems can be cured easily if addressed correctly. When troubleshooting for mechanical looseness, don't just think of looseness as loose bolting. The majority of time, mechanical looseness is poor bearing fits and oversized bearing clearances. Likewise, misalignment is not always across the coupling. Many misalignment problems are present because of bearing-to-bearing misalignment across a pump. Balance and looseness will be the easiest to cure with alignment being the toughest. The 3 years you devote to vibration monitoring will allow these problems to be addressed and corrected.

When vibration monitoring is being performed, be aware of the impact you will have on other peoples' jobs. Many people will look upon this type of program as a way to check their work. All personnel in the plant should be made aware

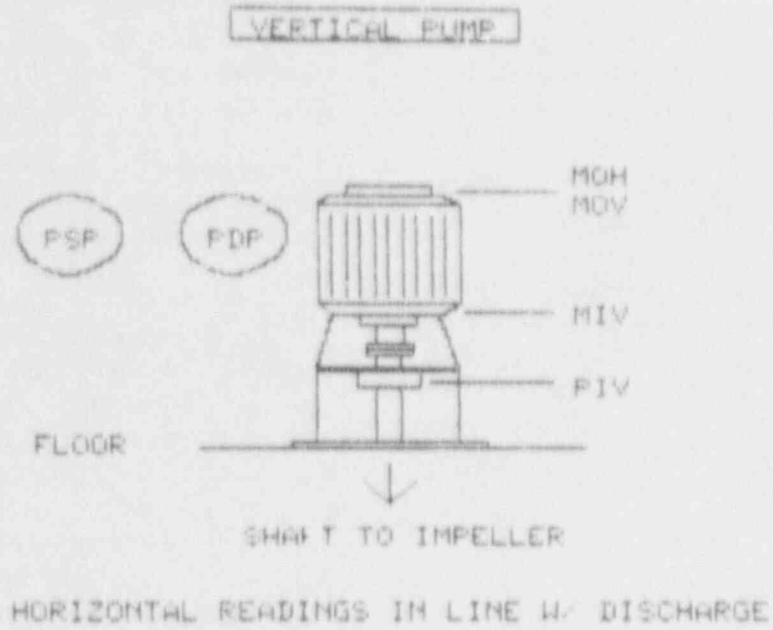

Figure 5.

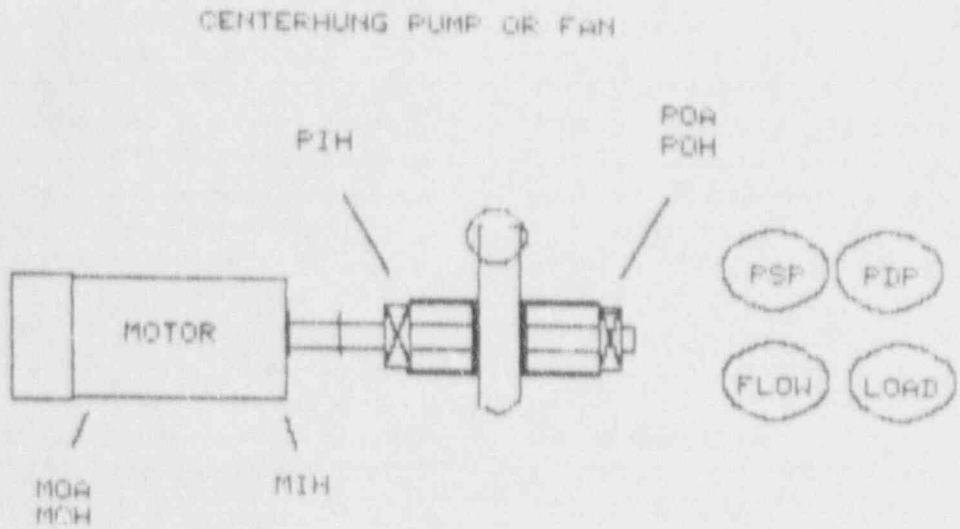

Figure 6.

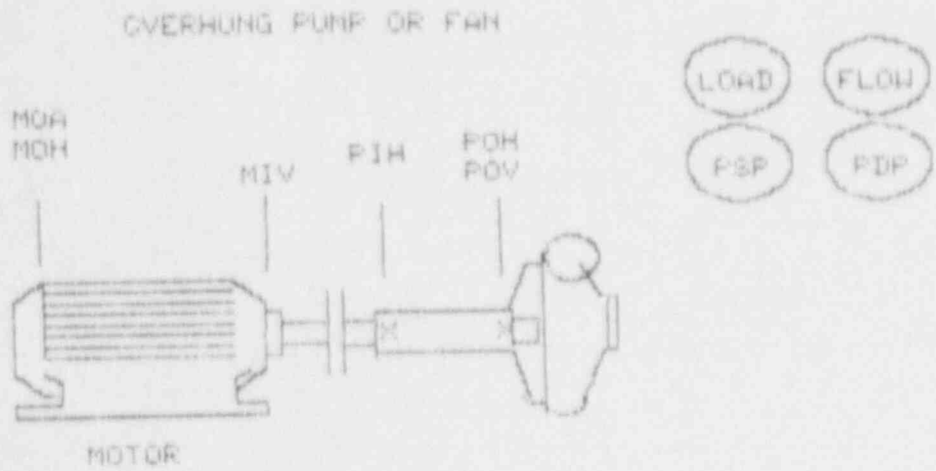

Figure 7.

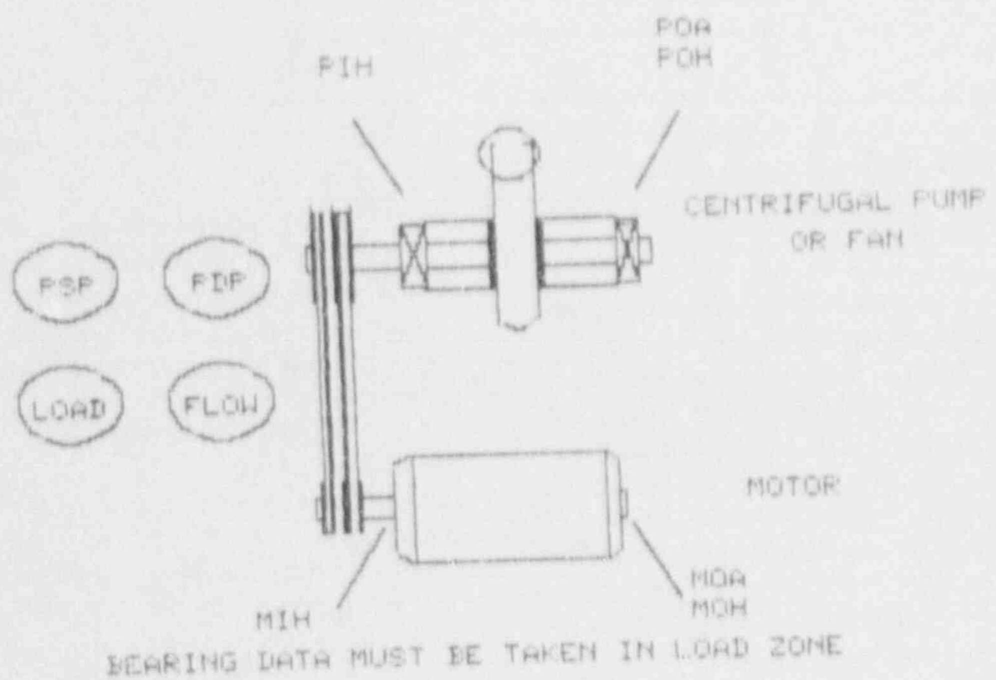

Figure 8.

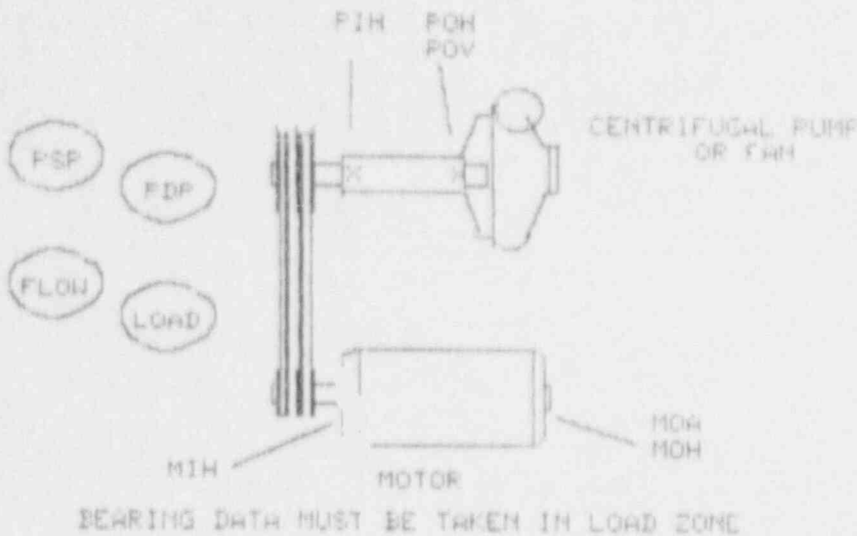

Figure 9.

of what the program entails and how it will benefit them. Benefits to them will include less rework of jobs, useful training, reduced overtime, and less overhaul work. Since the equipment is being monitored, you can perform overhauls on an as-needed basis instead of a schedule basis. Don't forget you still have to do the routine preventive maintenance (i.e., oil changes and packing adjustments).

Once the above steps are completed, you should be ready to start your monitoring and analysis data collection for predictive maintenance trending. The difference between monitoring and monitoring and analysis is the number of data points that you process each collection period. Also, when collecting data on each point you will be collecting timewave and spectrum plots. The other data collection points that need to be addressed for a complete predictive maintenance program are oil analysis and performance indices.

PERFORMANCE

The next area to set up for monitoring is the performance parameters. Many people have different ideas on what should be done in this area. Testing can range from complete ASME pump tests with calibrated test instrumentation to pump tests involving plant instrumentation. If the plant instrumentation is calibrated and kept in good condition, there is nothing wrong with using it. What you are trying to accomplish is to find what the normal operating parameters are and when the equipment falls outside its best operating range.

The easiest method to use is to set up static points for the routine route data collection. These points should be a key pad entry that is set as an out-of-window alarm. This means that if the key pad data is out of a certain range, the operational data is not consistent with past collection data. You want to take data at approximately the same operational setting for every collection period. The easiest way to set up these points is to review the pump curve and find the best efficiency point (BEP). Considering pump inlet and discharge pressure, the alarm for the high end should be at 120% of the BEP to indicate the onset of cavitation, and the low end alarm should be set at 65% of BEP because of the beginning of hydraulic instability. Figure 10 is a spectrum plot showing broad band vibration being excited by an abnormal pump flow condition. The broad band vibration spikes are at identified rolling element bearing frequencies, giving the appearance of bad bearings. Furthermore, if a bearing condition reading (i.e., spike energy, HFD, HFB, VHFB)

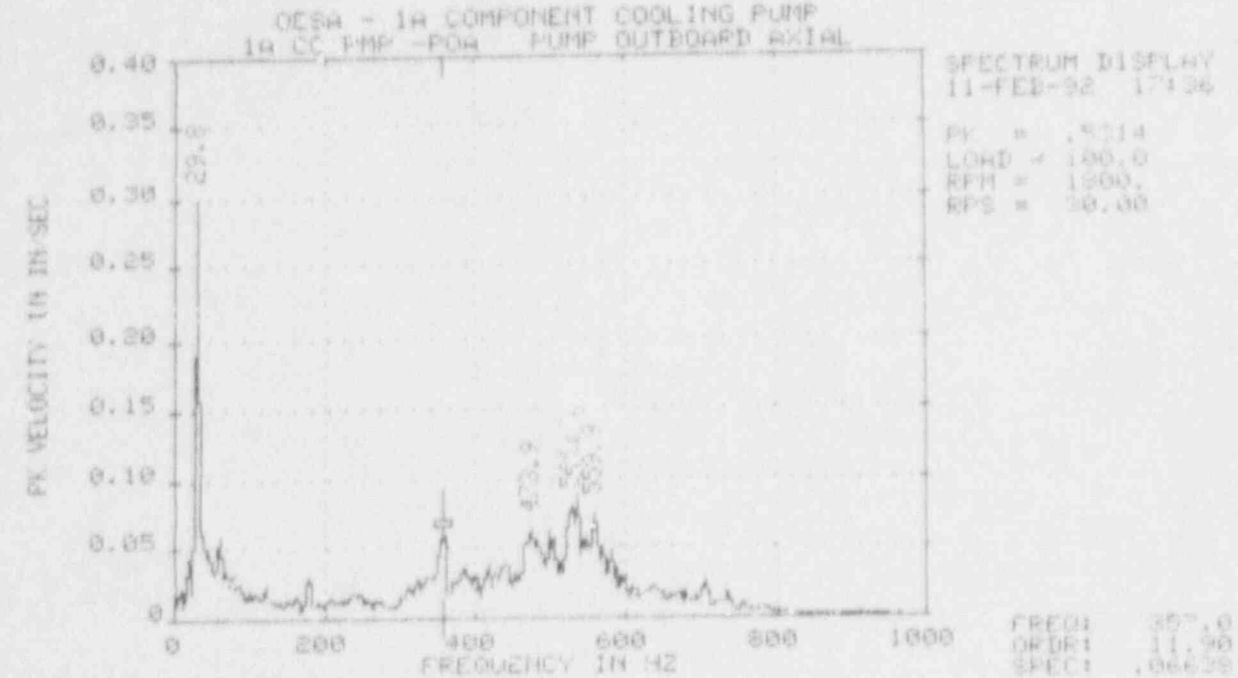

Figure 10.

were taken, the resulting bearing information would indicate a bearing change out is required.

The analyst must be aware that abnormal flow conditions excite bearing frequencies. This is one of the reasons that operation data, such as suction and discharge pressures, must be recorded when taking vibration data. At this stage in the program there are more vibration problems caused by operational conditions then mechanical problems. This is especially true in nuclear power plants where emergency pumps are tested quarterly at flow well below normal operation conditions.

Figure 11 shows data from a residual heat removal pump that has a problem with low flow coupled with a blade passing frequency. This condition is abnormal and was most likely caused by the low flow condition. This data can be stored in the database and compared to future data.

Since emergency equipment is run quarterly for tests, it is difficult to get data after the pump has been shut down to investigate problems that may have gone undetected during th[illegible] data collection. I recommend that for data [illegible] on all emergency equipment or equipment that cannot easily be put in service, have the data tape recorded. The recording of data from the test locations does not take any longer then normal data collection with the data collector. When the data tape is brought back to the lab, it can be run into the data collector and stored in the database. If problems are then found, the analyst can play back the tape and run the data through the data collector with different frequency ranges and different lines of resolution to analyze the data for the cause of the problem. This allows the analyst more time to analyze problems because normal test runs may not last long enough to find or analyze problems.

The analyst should not rely totally on one pickup. Vibration monitoring and analysis requires specific pickups for different conditions. Know the roll off frequencies of the pickups, and use the correct pickup. Also, don't rely on overall reading to trigger your vibration warning alarms.

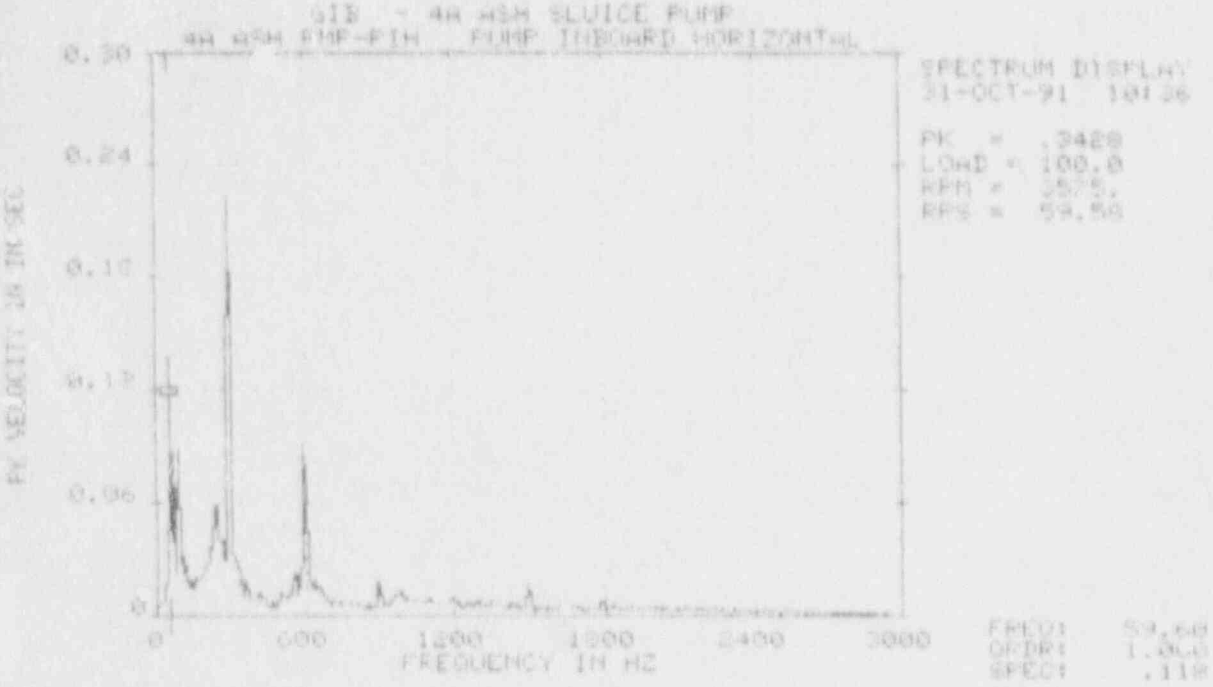

Figure 11.

These should also be triggered by the higher order bandwidth alarms. These bandwidth alarms will provide the analyst with insight into pending problems. Overall readings may or may not indicate problems. Watch the bandwidth alarms; these are the key to an early warning of what will be the problem frequencies.

ANALYSIS PARAMETERS

The development of the FFT analysis parameters is not an easy task unless you have complete knowledge of the machinery's past history and the equipment components. If the information sheets were filled out correctly, the analyst should have all that is needed to set up the correct FFT ranges for the data collector to take spectrum and timewave plots during routine data collection. These data can then be used for the analysis of pump problems. Tables 1 through 4 provide rule-of-thumb analysis parameters for rolling element and babbitt bearings that can be used if only a minimal amount of information is known initially about a piece of equipment.

The analysis of the spectrum plots is made much easier when fault frequencies are established in the database. These fault frequencies are identified vibration frequencies that will be excited when the following specific problems occur:

1. Running speed
2. Two times running speed
3. Multiples of running speed
4. Blade pass frequencies
5. Gear mesh frequencies
6. Rolling element bearing frequencies
7. Natural frequencies.

Figure 11 shows a plot without the use of fault frequency identification. Figures 12 through 16 show the same spectrum plot with the identifying fault frequencies.

```
Computational Systems Incorporated        MODIFY ANALYSIS PARAMETER SET
MasterTrend v2.14   Copyright 1985        DBASE    : DE06

        Analysis Parameter Set   6 - Primary Spectra/Waveform Setup

          SET DESCRIPTION:  Generic Roller Brg
          SPECTRAL FREQUENCY SETUP                       :  Order
          LOW FREQUENCY SIGNAL CONDITIONING LIMIT  (Hz):  2.0

FFT ANALYSIS . . .                         USE PRE-CONDITIONING UNIT?  No
  UPPER FREQ  (Hz/ORDER) :  50.0             FILTER SETTING        :  0
  LOWER FREQ  (Hz/ORDER) :  0.0              ENVELOPE DEMODULATOR:  Off
  NUMBER OF LINES        :  400
  NUMBER OF AVERAGES     :  6              OBTAIN SPECIAL TIME WAVEFORM?  No
  SPECTRAL AVERAGING MODE:  0                FMAX  (Hz OR ORDER):  0.0
  WINDOW TYPE            :  Hanning          DATA UNITS          :  0
  SPECTRAL WEIGHTING     :  None             NUMBER OF POINTS    :  64
  PERFORM 1/3 OCTAVE ANALYSIS?  No           TRIGGER             :  None

NUMBER OF ANALYSIS PARAMETERS:  6
                                                                  13:48:19
F1=Accept       F2          F3=Add/Edit    F4=Backup      F5=Help
F6=Device       F7          F8             F9             F0=Exit
```

Table 1.

```
Computational Systems Incorporated        MODIFY ANALYSIS PARAMETER SET
MasterTrend v2.14   Copyright 1985        DBASE    : DE08

                      Set ID   8:  Generic Sleeve Brg

                 Specify the Setup of the Trend Parameters...

                    Parameter     Type of       Lower       Upper
 Description        Units Type    Parameter     Frequency   Frequency
 ---------------    ----------    ---------     ---------   ---------
 SUBHARMONICS           0             2            0.0         0.8
 1xRPM                  0             2            0.8         1.5
 2xRPM                  0             2            1.5         2.5
 3-4xRPM                0             2            2.5         4.5
 5-15xRPM               0             2            4.5        15.0

                                                                  13:30:07
 Accept         F2          F3=Add/Edit    F4=Backup      F5=Help
 Device         F7          F8             F9             F0=Exit
```

Table 2.

```
Computational Systems Incorporated          MODIFY ANALYSIS PARAMETER SET
MasterTrend v2.14   Copyright 1985          DBASE    : DE06

        Analysis Parameter Set   6 - Primary Spectra/Waveform Setup

          SET DESCRIPTION:  Generic Roller Brg
          SPECTRAL FREQUENCY SETUP                          :  Order
          LOW FREQUENCY SIGNAL CONDITIONING LIMIT  (Hz):  2.0

FFT ANALYSIS . . .                          USE PRE-CONDITIONING UNIT?  No
  UPPER FREQ  (Hz/ORDER) :  50.0              FILTER SETTING         :  0
  LOWER FREQ  (Hz/ORDER) :  0.0               ENVELOPE DEMODULATOR:  Off
  NUMBER OF LINES        :  400
  NUMBER OF AVERAGES     :  6               OBTAIN SPECIAL TIME WAVEFORM?  No
  SPECTRAL AVERAGING MODE:  0                 FMAX   (Hz OR ORDER):  0.0
  WINDOW TYPE            :  Hanning           DATA UNITS           :  0
  SPECTRAL WEIGHTING     :  None              NUMBER OF POINTS     :  64
  PERFORM 1/3 OCTAVE ANALYSIS?  No            TRIGGER              :  None

NUMBER OF ANALYSIS PARAMETERS:  6

Press the SPACE bar to toggle selection.                                     13:33:44
F1=Accept        F2              F3=Add/Edit      F4=Backup       F5=Help
F6=Device        F7              F8               F9              F0=Exit
```

Table 3.

```
Computational Systems Incorporated          MODIFY ANALYSIS PARAMETER SET
MasterTrend v2.14   Copyright 1985          DBASE    : DE08

                       Set ID   6:  Generic Roller Brg

                 Specify the Setup of the Trend Parameters...

                  Parameter    Type of       Lower        Upper
 Description      Units Type   Parameter     Frequency    Frequency
 -------------    ----------   ---------     ---------    ---------
 SUB & 1xRPM          0            2          0.0          1.5
 2xRPM                0            2          1.5          2.5
 3-4xRPM              0            2          2.5          4.5
 5-20xRPM             0            2          4.5          20.5
 21-50xRPM            0            2          20.5         50.0
 1. - 20. kHz         3            4          1000.0       20000.0

                                                                             13:37:52
F1=Accept        F2              F3=Add/Edit      F4=Backup       F5=Help
F6=Device        F7              F8               F9              F0=Exit
```

Table 4.

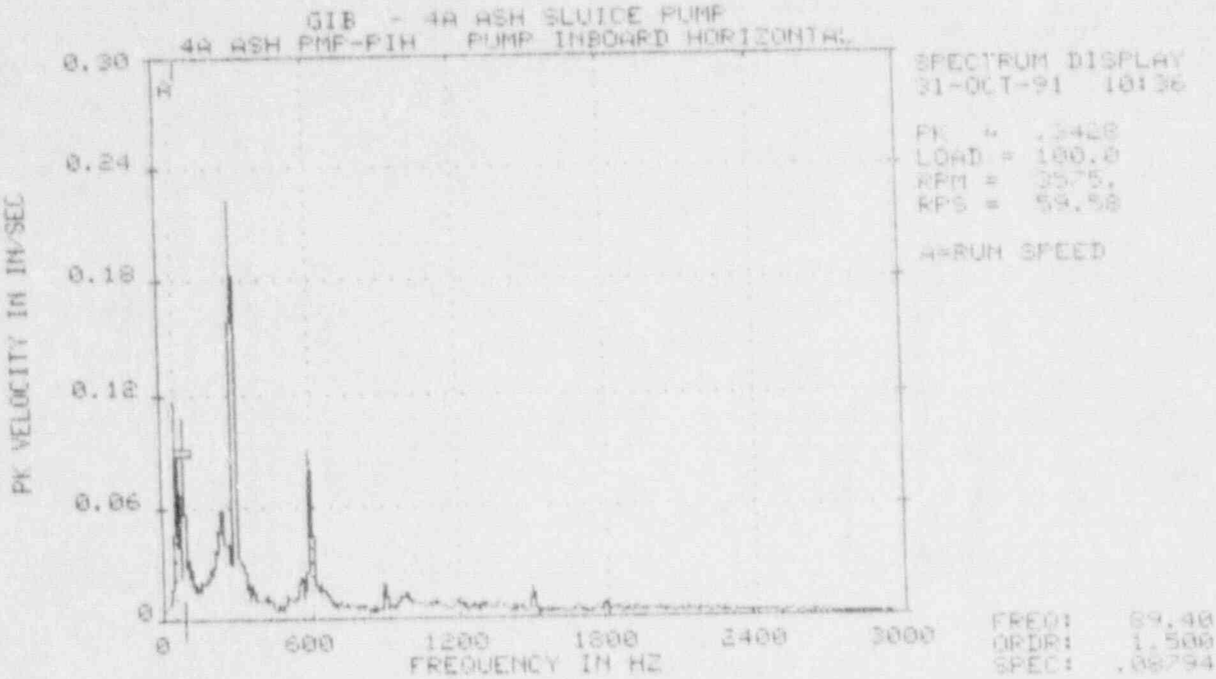

Figure 12.

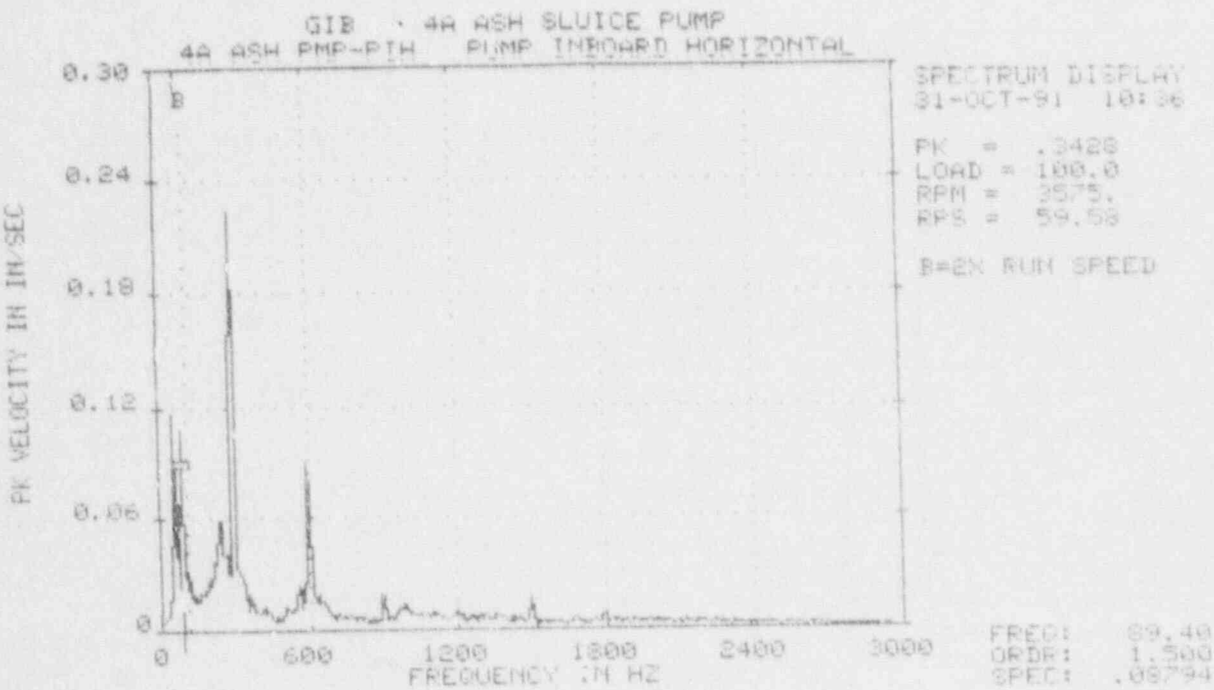

Figure 13.

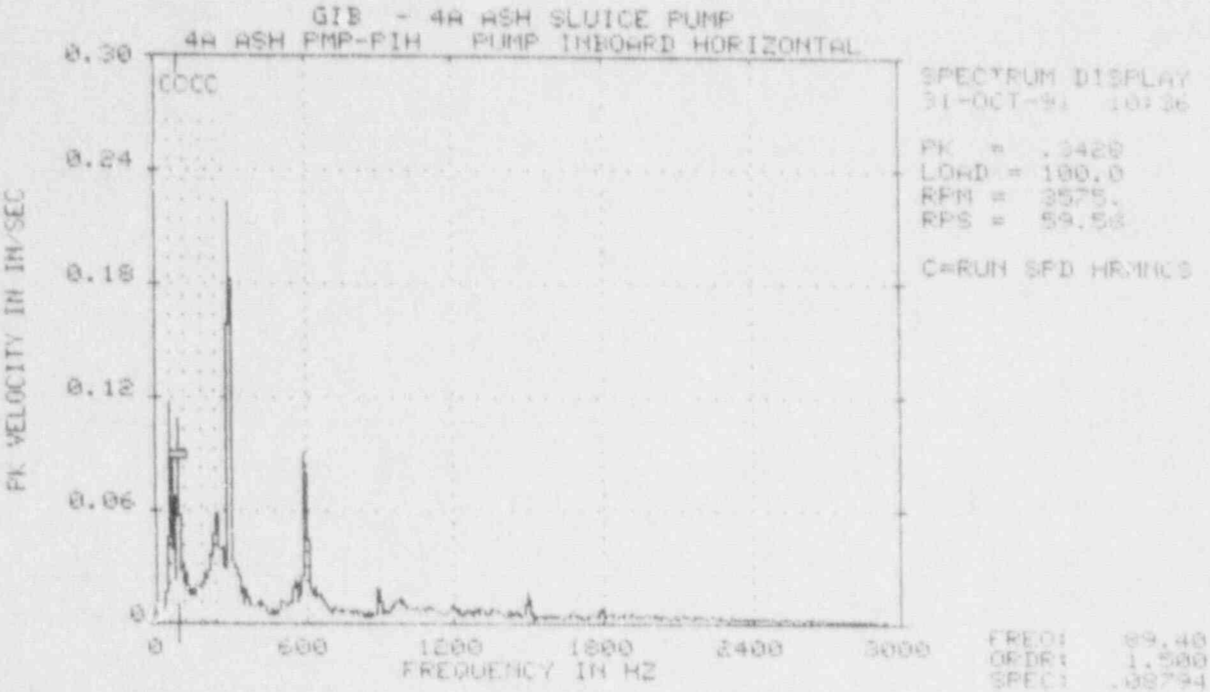

Figure 14.

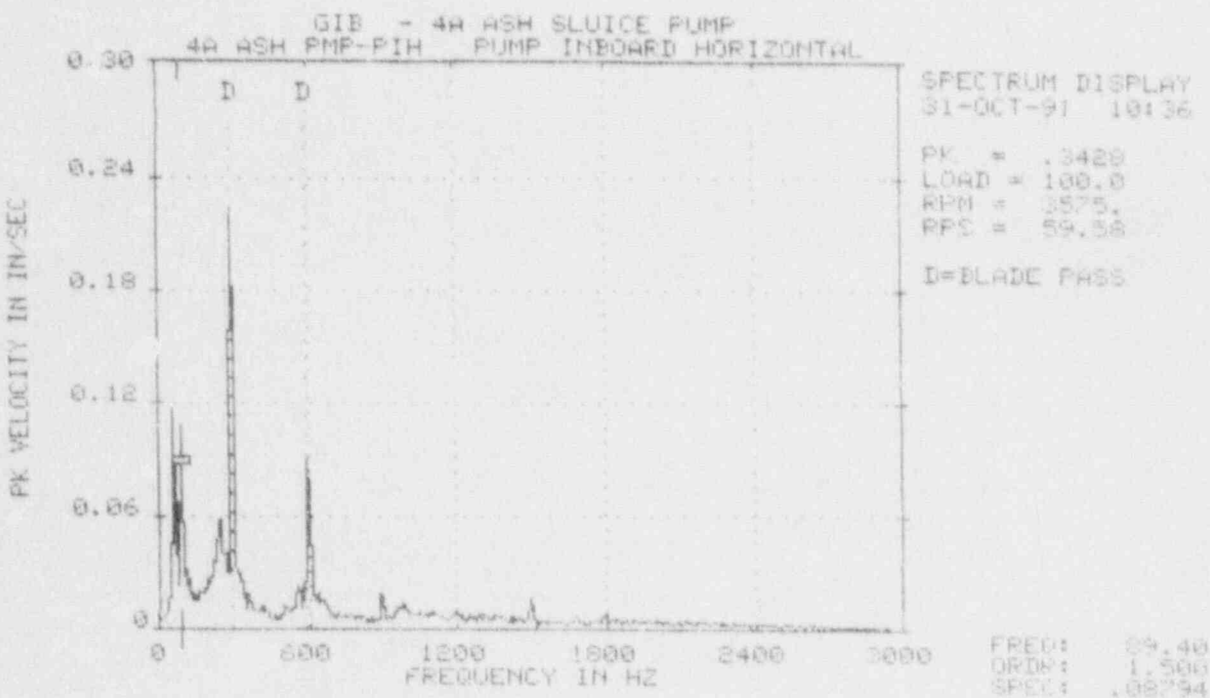

Figure 15.

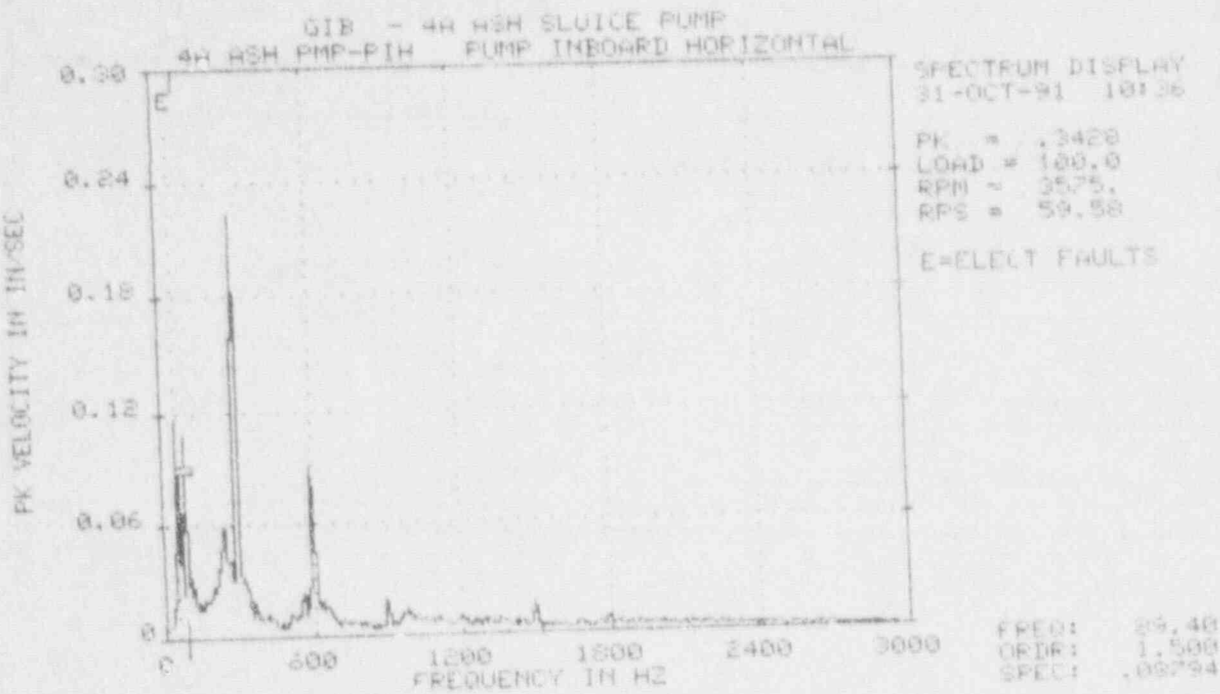

Figure 16.

OIL ANALYSIS

Some of the top vibration monitoring software packages have oil trending packages for their software. The procedure is to take oil samples, send them to the lab, who sends the results back via a modem to the software. Alarm reports are then run to find the problem samples. This information is kept in the same database as the vibration data. If the software you are using does not have a oil package, many of the oil labs now provide a service that will send oil reports back via modem into a spread sheet program so you may trend the oil parameters.

Table 5 is an example of how a database is set up for an oil parameter data point. Figure 17 is an example report of how the data is returned via modem into a trend table. The data from this table can then be put in a trend plot for analysis purposes.

CASE HISTORIES

Documentation of equipment history is essential for analysis of problems through records of past solved problems. These reports are also good for showing plant management what you have accomplished. During the initial period of your program, you will be highly visible. However, after a period of time when the initial problems are solved, your visibility will drop drastically. There is another more important factor to be considered: training. People don't tend to stay in one job all their life. They are promoted or leave for other reasons. Something is needed to train newer employees and assist the analyst when troubleshooting. The case history will fill this need. The case history should be dived into six parts:

1. Problem: Discuss how the problem was found and include a description of the equipment.

2. Symptoms: Describe the initial findings and any abnormal external visual conditions.

```
Computational Systems Incorporated        DEFINE NEW MEASUREMENT POINT
MasterTrend v2.50   Copyright 1985        DBASE   : DE04

       New Machine                     (Mch Code   ) - Point Number:  6
     Specify the following Measurement Point definition parameters:

        MEASUREMENT POINT ID/ORIENTATION : OIL
        MEASUREMENT POINT DESCRIPTION : Oil Point 1
        OIL DESCRIPTION : Crude Oil
        UNIT ID : XLE-1A
OIL ID CODE              : EX 1345     ANALYSIS PARAMETER SET ID : 1
OIL FLUID - SERVICE      : 1 - 1       ALARM LIMIT SET ID        : 1
OIL VISCOSITY CODE       : 1           FIRST FILTER DESCRIPTION  : STANDARD
UNIT TYPE - BEARING      : 1 - 0       SECOND FILTER DESCRIPTION : NONE
OIL CAPACITY(gals.)      : 350.0       THIRD FILTER DESCRIPTION  : NONE
USAGE(MILES/HOURS)       : HOURS       PRESSURE (lbs/in²)        : 1[illegible].8
MONITORING SCHEDULE      : 30.0        FLOW RATE (gal/min.)      : 2.5

        NUMBER OF YEARS TO STORE LAB DIAGNOSTICS : 1.0
        NUMBER OF DATA VALUES IN STATISTICAL CALCULATIONS : 12
                                                                  10:39:33
F1=Accept     F2=OTHER/OIL     F3=Add/Edit     F4=Backup     F5=Help
F6=Device     F7=Mark          F8=Save         F9=Recall     F0=Exit
```

Table 5.

```
COMPANY NAME          :PLANT 54
UNIT IDENTICATION     :XLE-1A
OIL DESCRIPTION       :Crude Oil
```

SAMPLE DATA		WEAR METALS AND ADDITIVES (in parts per million)										
		Al	Sb	Ba	B	Cd	Ca	Cr	Cu	Fe	Pb	Mg
Date: 12-JUN-90	1	0.	0.	77.	0.	0.	35.	0.	1.	2.	100.	0.
Date: 16-JUL-90	2	0.	0.	73.	0.	0.	33.	0.	0.	1.	101.	0.
Date: 15-AUG-90	3	0.	0.	0.	0.	0.	0.	0.	0.	0.	29.	0.
Date: 21-SEP-90	4	0.	0.	15.	0.	0.	7.	0.	0.	0.	66.	0.
Date: 11-OCT-90	5	0.	0.	0.	0.	0.	0.	0.	1.	3.	71.	0.

	WEAR METALS AND ADDITIVES (continued)										
	Mo	Ni	P	K	Si	Ag	Na	Sn	Ti	V	Zn
1	0.	0.	19.	0.	0.	0.	0.	0.	0.	0.	19.
2	0.	0.	23.	0.	1.	0.	0.	0.	0.	0.	17.
3	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
4	0.	0.	230.	0.	2.	0.	0.	0.	0.	0.	3.
5	0.	0.	0.	0.	2.	0.	1.	3.	1.	0.	9.

```
= Line Up      End = Bottom          Esc = Exit
```

Figure 17.

3. Test Data and Observations: Tell what you are tying to accomplish with your test and what the priorities are.

4. Corrective Action: In detail, describe the findings along with the recommendations for repair of the problem.

5. Results: Briefly discuss the repair findings.

6. Conclusion: Describe the solution, how well the problem was diagnosed, and what can be done to improve the diagnostic techniques. This is also the place to recommend improvements in analysis equipment and maintenance repair practices.

Many of the software programs have a section to keep case histories, including cost savings. This is an excellent way to keep track of your "SAVES." They can then be used to justify the program or improvements/upgrades to the program equipment.

CONCLUSION

Implementing a predictive maintenance program with the present generation data collectors and software packages is not difficult if it is done in logical order. The main thing is to lay a solid foundation on which to build your program. The keys to a quality program is discipline and dedication; however, no program will succeed without MANAGEMENT COMMITMENT. If the management of your plant and company does not believe in the program it will not prosper.

Use the monthly reports and solved case histories to generate management backing for your program. Include your misses, as well as your successes. Constantly sell the program. For every program that is beginning, there are five or six being eliminated or having manning reduced. Constantly justify the merits and savings of the program. Don't let management forget that you exist to help them do their job better.

NOTE: Turbine driven boiler feed pumps should be addressed as if they were turbine generators. Each month temperatures, pressures, and steam flows, along with filtered and unfiltered vibration, should be taken. Phase angles will also be needed to trend vibration location so balance shots can be calculated. If the pumps have prox probes installed, the data collector can be hooked directly into the probes, and the vibration parameters, gap voltages, and phase angles can automatically be collected and trended.

REFERENCES

Guy, Kevin R., "Monitoring and Analysis with Electronic Data Collectors," *Vibration Institute 1992 Annual Meeting*, Mini Course Notes.

Appendix 1

MOTOR:
NAME PLATE DATA:
VENDOR: ____________
MOTOR TYPE: ____________
HP: ____________ S.F.: ____________
RPM: ____________ FRAME: ____________
AMPS: ____________ POLES: ____________
ROTOR BARS: ____________ STATOR SLOTS: ____________
END SHIELD SUPPORTED BEARINGS: ____________
BEARING HOUSING OVERHUNG: ____________
VENTILATION: ____________
TYPE OF ENCLOSURE: ____________
BEARINGS: LUBRICATION:
OUTBOARD: ____________ OUTBOARD: ____________
INBOARD: ____________ INBOARD: ____________
PUMP:
TYPE: ____________ (DIFFUSER/VOLUTE)
MOUNTING: ____________
FLUID: ____________
IMPELLER DIAMETERS:
OUTSIDE DIAMETER: ____________
INSIDE DIAMETER: ____________
DIFFUSER/VOLUTE DIAMETERS:
OUTSIDE DIAMETER: ____________
INSIDE DIAMETER: ____________
BEARINGS OUTBOARD: ____________
INBOARD: ____________
LUBRICATION:
OUTBOARD: ____________
INBOARD: ____________
BLADES:
INLET:
IMPELLERS: ____________
DIFFUSER: ____________
OUTLET:
IMPELLERS: ____________
DIFFUSER: ____________
STAGES: ____________
NPSH: ____________
SUCTION PRESSURE: ____________ FLOW: ____________
FLUID TEMPERATURE: ____________
PIPING ATTACHMENT: ____________
SNUBBERS IN DISCHARGE PIPE: ____________
BELTS:
NUMBER: ____________
PULLEY SIZE:
DRIVE: ____________
DRIVEN: ____________
C-LINE DIST: ____________

Considerations for Reference Pump Curves

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ABSTRACT

This paper examines problems associated with inservice testing (IST) of pumps to assess their hydraulic performance using reference pump curves to establish acceptance criteria. Safety-related pumps at nuclear power plants are tested under the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (the Code), Section XI. The Code requires testing pumps at specific reference points of differential pressure or flow rate that can be readily duplicated during subsequent tests. There are many cases where test conditions cannot be duplicated. For some pumps, such as service water or component cooling pumps, the flow rate at any time depends on plant conditions and the arrangement of multiple independent and constantly changing loads. System conditions cannot be controlled to duplicate a specific reference value. In these cases, utilities frequently request to use pump curves for comparison of test data for acceptance. There is no prescribed method for developing a pump reference curve. The methods vary and may yield substantially different results. Some results are conservative when compared to the Code requirements; some are not. The errors associated with different curve testing techniques should be understood and controlled within reasonable bounds. Manufacturer s pump curves, in general, are not sufficiently accurate to use as reference pump curves for IST. Testing using reference curves generated with polynomial least squares fits over limited ranges of pump operation, cubic spline interpolation, or cubic spline least squares fits can provide a measure of pump hydraulic performance that is at least as accurate as the Code required method. Regardless of the test method, error can be reduced by using more accurate instruments, by correcting for systematic errors, by increasing the number of data points, and by taking repetitive measurements at each data point.

INTRODUCTION

The Code requires that fixed reference values for differential pressure and flow rate be established at points of pump operation that can be readily duplicated during subsequent tests by varying system resistance. The Code provides a measure of hydraulic performance based on the comparison of differential pressure and flow rate measurements to their reference values.

An increasing number of utilities are proposing the use of reference curves, representing the functional relationship between pump flow rate and differential pressure, in lieu of a fixed reference point(s). The problems with duplicating a fixed reference point, as required by the Code, fall into two broad categories.

Either system design is such that its resistance cannot be varied, or varied precisely, or flow rate

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is dependent on plant and/or climatic conditions. In cases such as these, it may not be possible to duplicate test conditions. Additionally, some utilities are proposing the use of reference curves for pumps even when the Code test method is not impractical. The question is whether the use of reference pump curves as reference values provides a measure of pump hydraulic performance that is equivalent to the Code testing method. Before one can make a reasonable comparison of this alternative to the Code, the limitations and uncertainties of the Code method must be examined.

THE CODE TESTING METHOD: UNCERTAINTIES AND LIMITATIONS

With the Code testing method, the system resistance is varied until either the pump flow rate or differential pressure (the independent variable) matches a fixed reference value. The remaining test quantity (the dependent variable) is then measured and compared to its fixed reference value. The difference between the reference point and the subsequent test point represents the pump's hydraulic degradation, expressed as a percentage of the reference value. Because test measurements and results are stated as fixed numbers, we tend to think of them as being absolute. However, all physical measurements are, to some degree, uncertain. The Code instrument accuracy and range requirements limit the uncertainty of the measurements and determine the error in the test results. The Code testing method does not provide an absolute measure of hydraulic degradation because the test results are based on the comparison of two uncertain numbers.

If the functional relationship, f(Q), represented by the pump curve was known precisely, the uncertainty in the reference value of the dependent variable, ΔP, based on the value of the independent variable, Q could be computed (Taylor, 1982) by:

$$\delta \Delta P = \frac{\partial f}{\partial Q} \delta Q \qquad (1)$$

However, the true functional relationship is unknown, and there is uncertainty in the measured values of flow rate and differential pressure. The unknown functional relationship between pump head and flow rate is fixed by the physical characteristics of the pump.

The variation in the measurement of the dependent variable is amplified by the uncertainty in the independent variable. If the curve defining this functional relationship is not flat, only one true value of pump head corresponds to a given true flow rate. As shown in Figure 1, for any indicated setting of flow rate, the true value lies within a specified range of measurement uncertainty. The true value may be greater than, less than, or equal to the indicated flow rate.

For this range of possible true flow rates, there is a corresponding range of possible true differential pressures. Additionally, the measured differential pressures may vary within the range of instrument error. If the independent variable is consistently set to the same *indicated* value and the measurement errors were completely random, we could see a significant amount of scatter in our test data. The possible variation could be greater than the instrument error associated with the dependent variable. The greater the slope of the curve within the range of uncertainty of the true flow rate, the greater the variation will be in the measurement of the dependent variable.

Instrument error, however, is not completely random (Bentley, 1983). As shown in Figure 2, systematic errors due to hysteresis, non-linearity, and perhaps variations in environmental conditions such as temperature cause instruments to indicate consistently in one direction. Random errors affect the repeatability of an instrument and result in measurements that are distributed about the true value. The error associated with almost all physical measurements will be a combination of random and systematic errors. Instrument manufacturers lump all these errors together and define performance in terms of error bands.

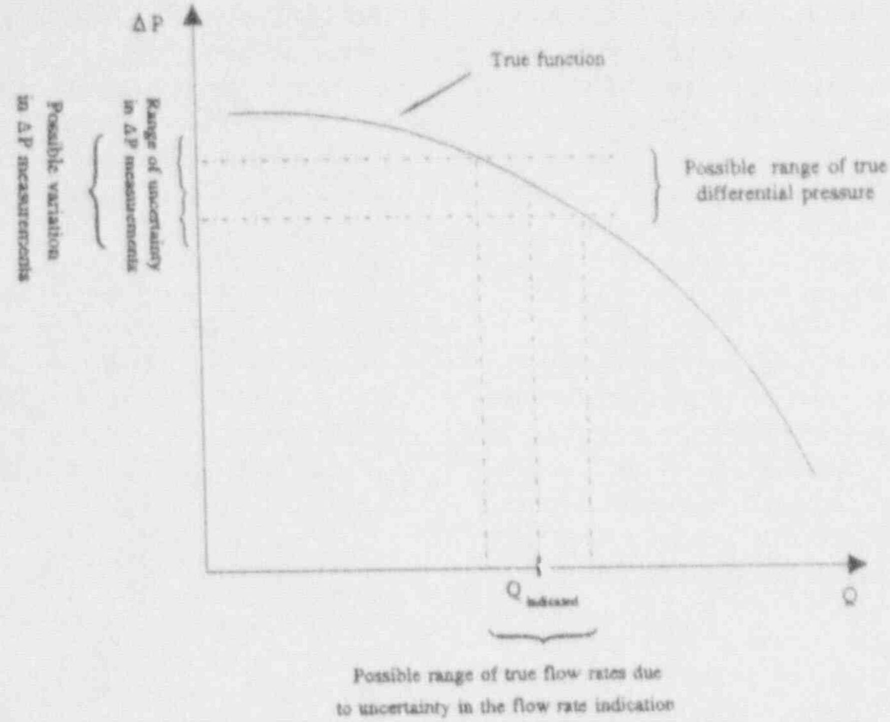

Figure 1. The possible variation in a measured test quantity due to uncertainties.

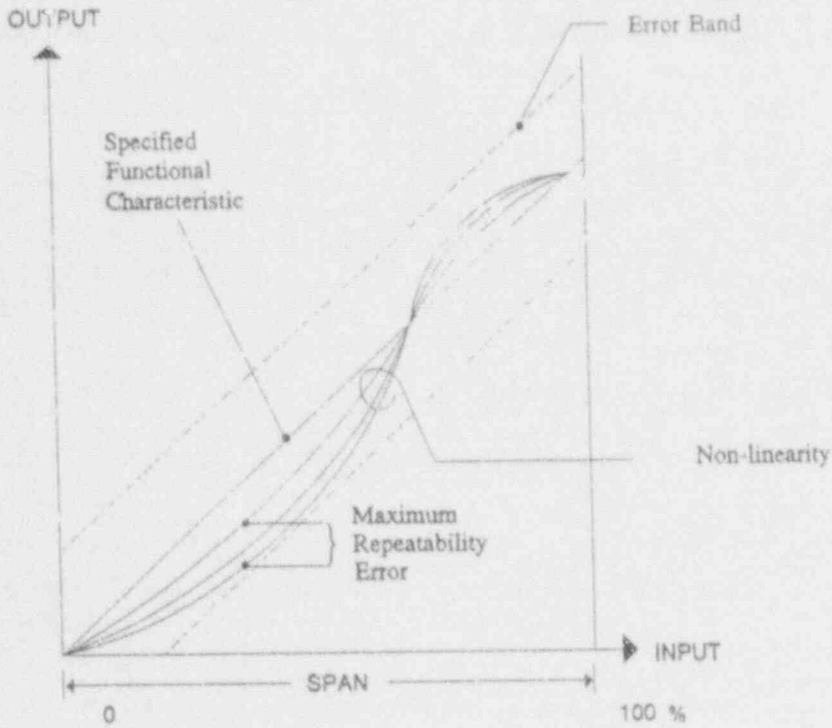

Figure 2. Calibration curve showing systematic and random errors.

Systematic errors may be reduced or characterized by calibration. Corrections can be made for systematic errors once it is known that an instrument reads consistently in one direction at a fixed point under specified conditions. Since similar instruments with equivalent accuracies may have differences in systematic errors, running a test with different instruments may yield data that are consistently different.

If the pump differential pressure is computed as the difference between the measured suction and discharge pressures, yet more uncertainty is introduced. The error bound for the difference or sum of two uncertain measurements is the sum of the errors. If the errors associated with each of these measurements are random and independent (i.e., systematic errors have been eliminated or reduced to a negligible level), a better estimate of the uncertainty of the difference (Taylor, 1982) is obtained by adding the errors in quadrature:

$$\delta P = \sqrt{\delta P_s^2 + \delta P_d^2} \quad . \tag{2}$$

Even after accounting for uncertainties, the Code testing method does not ensure a quality test. The Code does not specify the range of pump operation where testing must be performed. Pump testing is frequently performed at low flow. However, most pump degradation mechanisms do not result in a consistent reduction in hydraulic performance across the entire range of operation (Yedidiah, 1977a, 1977b, 1977c). Typically, degradation causes the pump curve to droop, with a significant decrease in performance at high flow rates and little or no change at shut-off head, as shown in Figure 3. Low flow test data may be within the acceptable range of the Code, while degradation in the range of required pump operation may be outside the Code limits.

Besides providing little useful data, pump testing at low flows may accelerate degradation (Tinney). Pump operation away from the best efficiency point (BEP) generates hydraulic anomalies within the casing and impeller. This may cause recirculation cavitation and uneven radial thrust on the impeller leading to shaft deflections, reduced bearing and mechanical seal life, and accelerated wear. Additionally, mixing occurs at low flow rates between the liquid that has already entered the impeller and the liquid at the pump

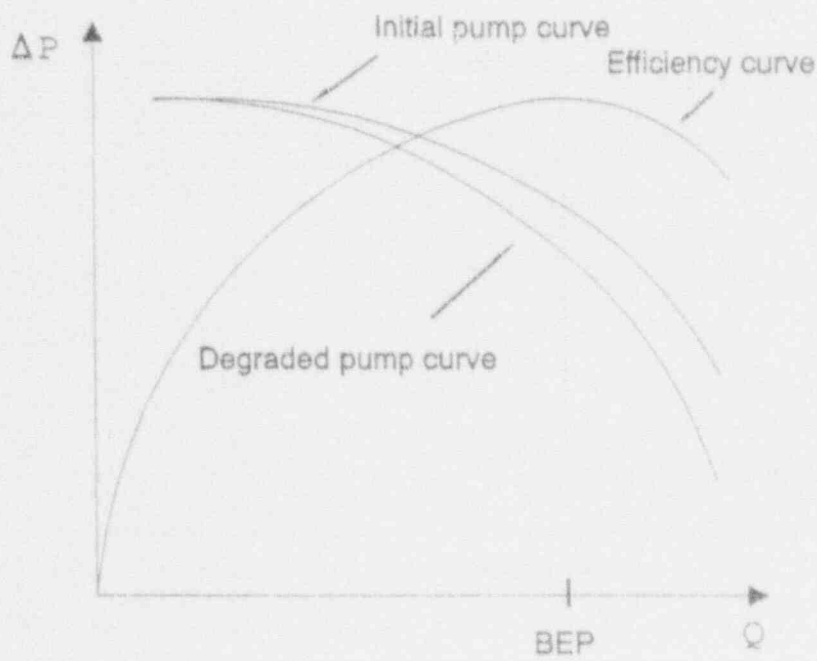

Figure 3. Reduction in hydraulic performance due to typical degradation mechanisms.

suction. This mixing imparts a rotation on the flow in the suction line. The energy of prerotational flows can cause the indicated suction pressure to be higher than actual, affecting the calculated or measured value of pump differential pressure.

The Code testing method does, however, provide consistency because the testing is required to be performed at repeatable point(s) of pump operation. Paragraph IWP-3112 allows the establishment of an additional set of reference values; however, the reasons for doing so must be justified and documented, and the pump must first be satisfactorily tested at the previous reference point.

In summary, the degradation calculated using the Code IST method is not absolute. The actual pump degradation may be more or less than that indicated by a test. The uncertainty in the pump test results depends on the measurement error for differential pressure and flow rate and the functional relationship between the two. The data scatter may be larger than expected if only the measurement error of the dependent variable is considered. However. much of this variation in the data can be accounted for by analysis. While not perfect, the Code testing method does provide consistency for trending test results. Finally, testing at low flow rates provides little useful data, accelerates degradation, and may lead to a false sense of operational readiness.

USING REFERENCE CURVES

Pump testing using reference pump curves in lieu of fixed reference values can provide a measure of pump hydraulic performance that is at least as accurate as the Code testing method. However, there are no guidelines for testing using reference pump curves. A number of different methods have been proposed and implemented. Utilities frequently propose using a manufacturer's pump curve in lieu of generating a curve. A number of different methods could be used to generate curves from data. One should keep in mind that a reference curve is only an approximation of the true functional relationship between flow rate and differential pressure. It is important that the reference curve is developed, or the manufacturer's curve is verified, when the pump is known to be in good operating condition, and the curve is based on, or validated by, a sufficient number of data points. The greater the number of data points, the less the error of the approximation is likely to be. Based on the variance in the data used to verify or generate the curve, and our knowledge of the possible errors, we can estimate the uncertainty in this approximation. To obtain an equivalent measure of hydraulic performance, the total uncertainty from testing with reference pump curves should be no greater than that allowed by the Code method. However, additional limitations and sources of error associated with using reference curves should be considered.

The instruments used for pump testing should meet the accuracy and range requirements of the Code regardless of the test method. However, because the precision of any plot is limited, error is also introduced when a reference value is read from a curve. The combination of the error due to the readability of the plot and the instrument error should be less than or equal to that allowed by the Code. Clearly, a plot of a pump curve that cannot be read with a precision of ±2% of full-scale will not meet the Code requirements, regardless of the accuracy of the instruments used for testing.

Using a reference curve implies that pump testing may be performed over a wide range of flow rates and differential pressures. Since the levels of vibration may vary over the range of pump operation, a method of assigning vibration reference values should be developed also. By testing at significantly different points of pump operation, we lose the consistency from test to test obtained by testing at fixed points of pump operation. Since the indicated percentage of hydraulic degradation may be less at lower flow rates, it may be possible to avoid corrective action simply by testing at lower flow rates. However, if using reference pump curves enables testing at higher flow rates using instrumentation in the main flow path, it can be superior to testing on a low flow test loop using the Code testing method.

Because pump hydraulic performance does not improve with age and use, upper limits are specified in Table IWP-3100-2 of the Code, the Allowable Ranges of Test Quantities. The NRC staff views these upper limits as aids in detecting instrumentation problems and ensuring that the reference values have not been set too low. However, considering the possible variation in the dependent variable, the upper alert and required action limits of 1.02 and 1.03 times the dependent reference variable, respectively, are probably too restrictive. Since pumps degrade from the reference values with use, test results above the upper range limits become less of a problem as time passes. Paragraph IWP-3210 of the Code allows utilities to specify expanded range limits if the limits of Table IWP-3100-2 cannot be met. The NRC staff interpretation is that licensees should demonstrate that the Code ranges cannot be met on a pump-specific basis and that expanded ranges are adequate for determining that significant degradation has occurred. Many utilities are requesting, or specifying, expanded range limits in conjunction with relief requests to allow the use of reference pump curves. The same quality instrumentation is used whether a fixed reference point or a reference curve is used. However, based on my experience with the review of IST programs, test results exceeding the upper range limits appears to be more of a problem when testing with reference pump curves than with the Code testing method. This problem arises most frequently when using manufacturer's pump curves. Test results that routinely exceed reasonably established upper range limits when using pump curves as reference values are indicative of accuracy problems with the reference curve.

Manufacturer's Pump Curves

Each point of pump operation has uncertainty in two directions from the flow rate and differential pressure measurements. The cumulative errors for each point of pump operation can be represented by bands of uncertainty above and below the pump curve. If all data points fall within these bands, then the data is consistent with the hypothesis that the manufacturer's curve represents the true pump curve. However, this consistency does not mean that the manufacturer's curve is the best representation of the true pump curve.

If the measured points are randomly distributed above and below the curve, the maximum deviation is less than the measurement error, and the mean deviation is small, we may conclude that the manufacturer's curve is a reasonable representation of the true reference values for the range of pump operation. The more measurement points that are used to verify the curve, the more confidence we may place in the results. However, if the data points are skewed in one direction above or below the curve or there is a discernable trend in the deviations, then either the data are affected by systematic errors or the manufacturer's curve is not the best representation of the true pump curve.

Manufacturer's curves are easy to use and require little sophistication to implement. However, there are some significant disadvantages. First, these curves are generic in nature. They apply to all pumps of the same design. There may be significant deviations in the performance of pumps of the same design due to minor differences in casting, machining, and installation (Buse, 1977). Second, as previously discussed, significant error may be introduced from reading the curve. Also, considering the possible variation in the dependent variable, it is understandable why many utilities using manufacturer's pump curves are having problems with the allowable ranges of the Code. Given the pitfalls of using a manufacturer's curve, generating a curve for use as reference values is likely to be more accurate. Additionally, the function of the generated curve is known and reference values can be computed. Therefore, error from reading a plot can be eliminated.

Generating a Pump Curve

Interpolation methods fit a curve through all data points with no deviations between the curve and the data. The most basic method of curve fitting is by polynomial interpolation, which yields a unique polynomial function of degree N though N+1 data points. Polynomial interpolants are most easily generated using Newton's method of divided differences. For data points $f(x_0)...f(x_n)$, the interpolating polynomial, $p_n(x)$, that agrees with the function $f(x)$ at these points may be expressed in Newton form:

$$p_n(x) = a_0 + a_1(x - x_0)$$
$$+ a_2(x - x_0)\ (x - x_1) + \ldots$$
$$+ a_n\ (x - x_0) \cdots (x - x_{n-1})\ . \qquad (3)$$

The co-efficients $a_0 \ldots a_n$, are the divided differences:

$$a_0 = f[x_0] = f(x_0)$$

$$a_1 = f[x_0, x_1] = \frac{f(x_1) - f(x_0)}{x_1 - x_0}$$

$$f[x_1, x_2] = \frac{f(x_2) - f(x_1)}{x_2 - x_1}$$

$$a_2 = f[x_0, x_1, x_2] = \frac{f[x_1, x_2] - f[x_0, x_1]}{x_2 - x_0}$$

. . . etc.

The interpolating polynomial is, therefore,

$$p_n(x) = f[x_0] + f[x_0, x_1](x - x_0)$$
$$+ f[x_0, x_1, x_2](x - x_0)(x - x_1) + \ldots$$
$$+ f[x_0, \ldots, x_n]\ (x - x_0) \cdots (x - x_{n-1})\ . \qquad (4)$$

Polynomial interpolation is not well suited to our problem for many reasons. Curves generated with polynomial interpolants will have wiggles, which increase in frequency and amplitude as N increases. These wiggles can be suppressed over a limited range by careful selection of the data points, but cannot be eliminated. Interpolation with polynomials of degree greater than 3 will produce a plot with excessive wiggles. If the polynomial is of degree greater than 10, the interpolation scheme will be numerically unstable.

To some degree, these problems are overcome by interpolating with piece-wise polynomials. Piece-wise polynomials are generated by interpolating in segments with polynomials of low degree. Requiring the function values at the ends of the segments (or knots) to match produces a continuous function.

The simplest of piece-wise interpolation schemes uses linear functions. Adjacent data points are connected with straight lines, as shown in Figure 4, which are generated from the interpolating polynomial:

$$p_2(x) = f[x_i] + f[x_i, x_{i+1}](x - x_i) \qquad (5)$$

It is shown in de Boor (1978) that the error bound from interpolating the function f(x) with this method is:

$$\| f(x) - p_2(x) \| \leq \frac{1}{8}(\max \Delta x_i)^2 \| f''(x) \|$$

$$\Delta x_i = (x_{i+1} - x_i)\ . \qquad (6)$$

Not surprisingly, the error in the approximation of f(x) goes to zero as the distance between data points goes to zero. However, for pump curves that are shaped concave down, the error is non-conservative because the piece-wise polynomial always lies at or below f(x).

The piece-wise function is smoothed, and the error of the interpolation is reduced, by using cubic polynomials and requiring the function and derivative values to agree at the knots. The four pieces of data used to generated the cubic functions connecting adjacent data points are ($f(x_i)$, $f'(x_i)$, $f(x_{i+1})$, and $f'(x_{i+1})$). In general, the divided difference is:

$$f[x_i, x_{i+1}, \ldots, x_{i+k}] = \frac{f^k(x_i)}{k!} \qquad (7)$$

if

$$x_i = x_{i+1} = \ldots = x_{i+k}\ . \qquad (8)$$

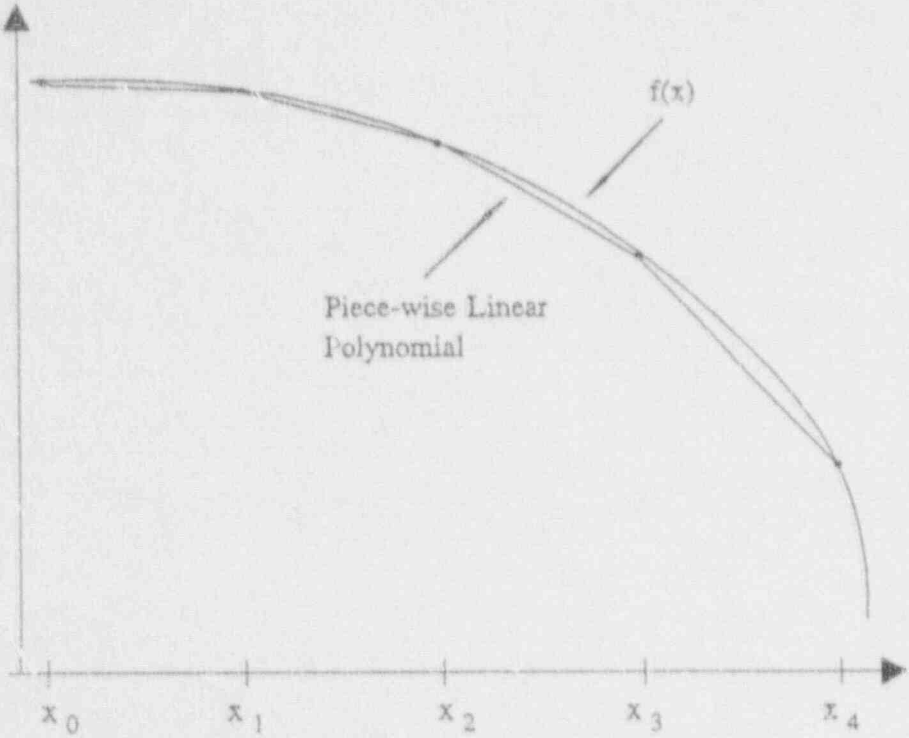

Figure 4. Piece-wise linear interpolation.

Therefore, the divided differences for the cubic function are:

$$f[x_i] = f(x_i)$$

$$f[x_i, x_i] = f'(x_i)$$

$$f[x_i, x_i, x_{i+1}] = \frac{f[x_i, x_{i+1}] - f[x_i, x_i]}{x_{i+1} - x_i}$$

$$f[x_i, x_{i+1}, x_{i+1}] = \frac{f[x_{i+1}, x_{i+1}] - f[x_i, x_{i+1}]}{x_{i+1} - x_i}$$

$$f[x_i, x_i, x_{i+1}, x_{i+1}] = \frac{f[x_i, x_{i+1}, x_{i+1}] - f[x_i, x_i, x_{i+1}]}{x_{i+1} - x_i} \quad . \quad (9)$$

And the interpolating cubic function for each segment is:

$$p_i(x) = f[x_i] + f[x_i, x_i](x - x_i) + f[x_i, x_i, x_{i+1}](x - x_i)^2 + f[x_i, x_i, x_{i+1}, x_{i+1}] (x - x_i)^2(x - x_{i+1}) \quad . \quad (10)$$

The obvious drawback of using these functions to generate the polynomial, known as a Hermite cubic piece-wise polynomial, is that the derivative values at each data point must be known.

As it turns out, imposing the additional restriction that the second derivatives also match at the knots enables us to compute the piece-wise cubic polynomial with only knowledge of the function values. The resulting function, called a cubic spline, is not only smooth but also uniquely minimizes curvature. The error bound due to interpolation with cubic splines, as shown in de Boor (1978), is:

$$\| f(x)-p_4(x) \| \leq \frac{1}{16}(\max \Delta x_i)^4 \| f^4(x) \| \qquad (11)$$

Compared to the linear piece-wise polynomial scheme, the error associated with cubic spline interpolation is much less. If the fourth derivative of the function f(x) is bounded, we can make the error as small as we wish by decreasing the distance between data points.

Interpolation methods provide an effective means of generating a reference curve if the function values are sufficiently accurate. Without high confidence in the data, other methods such as least squares approximations should be used. Unlike interpolation, least squares methods allow deviations between the data points, f_i, and the curve to account for the uncertainty in the data and to obtain a simpler function. Based on our knowledge of what the function should look like, it is logical to perform the least squares fit using polynomials. With the method of least squares, we minimize the sum of the squared deviations:

$$E(p) = \sum [p(x_i) - f_i]^2 \qquad (12)$$

where

$$p(x_i) = a_0 + a_1(x_i) + \ldots + a_m(x_i)^m \quad . \qquad (13)$$

Taking the partial derivative of E with respect to each coefficient of the polynomial, $p(x_i)$, yields a set of m + 1 equations with m + 1 unknowns, as shown in Figure 5.

These equations can be solved to obtain the coefficients of the fitted polynomial. When using polynomial functions, the equations are linear and the matrix is guaranteed to be invertible. Therefore, a solution is guaranteed, and there is a unique polynomial for which the functional E(p) is minimized.

With the least squares method, the data can be fitted to a polynomial of any degree. However, solving the normal equations can become cumbersome with polynomials of high degree and the fitted curve will develop the wiggles characteristic of these polynomials. Also, solving the matrix for this system of linear equations becomes numerically ill-conditioned, and the solutions may be subject to significant roundoff error as the degree of the polynomial (and hence the size of the matrix) increases.

$$
\begin{aligned}
a_0 \sum_{i=0}^{N} x_i^{\,0} + a_1 \sum_{i=0}^{N} x_i^{\,1} + \ldots + a_m \sum_{i=0}^{N} x_i^{\,m} &= \sum_{i=0}^{N} f_i \\
a_0 \sum_{i=0}^{N} x_i^{\,1} + a_1 \sum_{i=0}^{N} x_i^{\,2} + \ldots + a_m \sum_{i=0}^{N} x_i^{\,m+1} &= \sum_{i=0}^{N} x_i f_i \\
&\vdots \\
a_0 \sum_{i=0}^{N} x_i^{\,m} + a_1 \sum_{i=0}^{N} x_i^{\,m+1} + \ldots + a_m \sum_{i=0}^{N} x_i^{\,2m} &= \sum_{i=0}^{N} x_i^{\,m} f_i
\end{aligned}
$$

Figure 5. Equation set for m + 1 equations with m + 1 unknowns.

If the data points are normally distributed about the true values (i.e., the errors are random), then the deviations $[y_i - p(x)]$ are normally distributed, all with the same central value (0) and the same width (σ). We can compute the variance which leads to an estimation of the error of our fitted curve:

$$\sigma^2 = \frac{1}{N - m} \sum_{i=0}^{N} (f_i - p(x_i))^2 \quad . \tag{14}$$

The best choice for the degree of the polynomial is the lowest m that makes the variance reasonably small (Lancaster, Salkauskas, 1986). With pump curves a third degree polynomial provides a good fit.

A more sophisticated procedure for generating a curve without high confidence in the data is with a cubic spline least squares fit. This is accomplished by performing a least squares approximation over each segment of a piece-wise cubic spline, with multiple data points between the knots. In general, this method is more accurate, with the improvement in the error most significant when approximating functions with complicated shapes. However, because pump curves are generally well behaved and considering the extra degree of complexity of cubic spline least squares fits, a polynomial least squares fit is sufficient. Over limited ranges of pump operation, a polynomial least squares fit is capable of providing a reference curve that enables the determination of hydraulic degradation as well as the Code required method. There are, however, some assumptions regarding least squares methods that should be understood.

With least squares approximations, it is assumed that there is uncertainty only in the dependent variable, which is not true. However, because the variation in the measurements of one variable are amplified by the uncertainties in the other, we can account for this to some degree. Also, it is assumed that the measurements are governed by a normal random distribution, which may not be entirely true. Depending on the instruments and methods of collecting data, this assumption may or may not be trivial. Without a normal random distribution, the method of least squares may still provide the best fit. However, statistical significance should not be attached to the results.

Ideally, what we want is a total least squares fit because we have uncertainty in both variables. This is an intense mathematical problem into which research is continuing. As is often the case, in the real world we must settle for something less than the theoretically desirable. There are ways, however, to reduce error and improve the approximation of the real function.

Reducing Error

The error in the use of reference curves may be reduced by using more accurate instruments, by correcting for systematic errors, by increasing the number of data points, and by taking repetitive measurements at each data point.

Regardless of whether piece-wise polynomial interpolation or some least squares method is used, the accuracy of the curve fit will be improved by increasing the number of data points. The mean of a number of measurements is always a better estimate of the true value than a single measurement (Barford, 1985). When repetitive measurements of the dependent variable are taken at the same *indicated* value of the independent variable, data are distributed about a mean value as a result of random errors. As shown in Figure 6, the estimated mean may be displaced from the true mean by systematic error.

As the number of measurements increases, the estimated mean $(\overline{x})$ converges to the true mean (μ). Based on the standard deviation (σ) of the data distribution, the standard deviation of the estimated mean $(\overline{\sigma})$ can be computed:

$$\overline{\sigma} = \frac{\sigma}{\sqrt{N}} \quad . \tag{15}$$

With this equation, the number of measurements (N) necessary to yield a desired accuracy in the estimation of the true mean can be calculated. The standard error, or the error in the approximation of the true mean is expressed as:

$$\mu = \overline{x} \pm n\overline{\sigma} \tag{16}$$

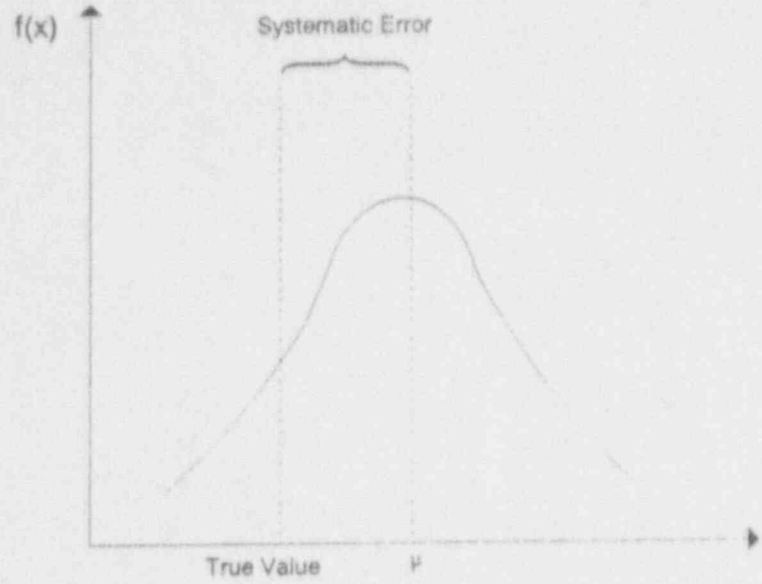

Figure 6. The mean of a number of measurements.

Two standard deviations (n = 2) provides 95% confidence in the estimate.

As shown in Figure 7, higher precision instruments have narrower distributions and smaller standard deviations. Therefore, with more accurate instruments, the standard error may be reduced to a specified tolerance with fewer measurements. Because the improvement in the estimation of the mean of the dependent variable for each data point goes as $N^{-1/2}$, beyond a certain point the improvement in accuracy would be best achieved by using higher precision instruments. However, regardless of the instrument precision, using the mean of a number of measurements, corrected for systematic error, in lieu of single measurement yields greater accuracy.

CONCLUSIONS

The Code testing method does not provide an absolute measure of hydraulic degradation because the test results are based on the comparison of uncertain numbers. The uncertainty in the pump test results depends on the measurement error for differential pressure and flow rate and the functional relationship between the two. The Code does not specify the range of pump operation where IST should be performed. Testing at low flow rates provides little useful data, may accelerate pump degradation, and may lead to a false sense of operational readiness.

Inservice pump testing using reference curves generated with polynomial least squares fits over a limited range of pump operation, cubic spline interpolation, or cubic spline least squares fits can provide a measure of pump hydraulic performance that is at least as accurate as the Code required method. If using reference pump curves enables testing at higher flow rates using instrumentation in the main flow path, it can be superior to testing on a low flow test loop using the Code method. However, there are limitations particular to testing with reference curves.

By testing at significantly different points of pump operation, we lose the consistency from test to test obtained by testing at fixed points of pump operation. Additionally, since the indicated percentage of hydraulic degradation may be less at lower flow rates, it may be possible to avoid corrective action simply by testing at lower flow rates.

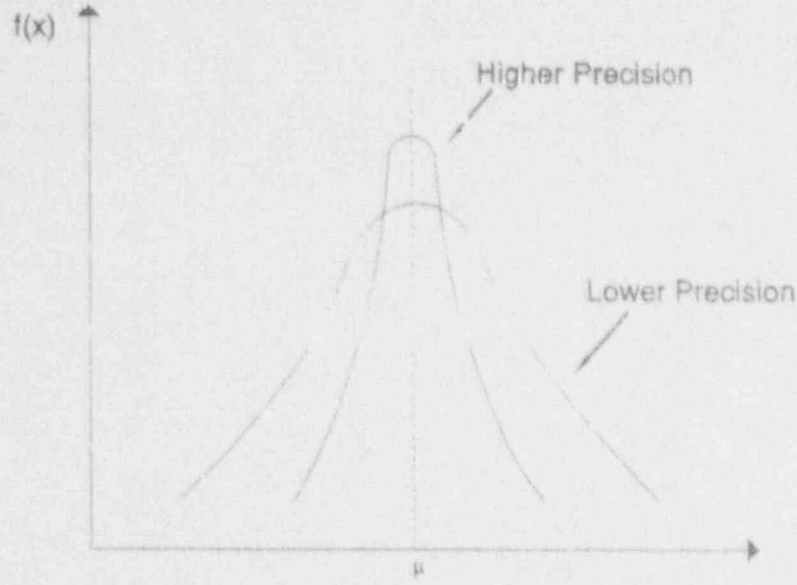

Figure 7. Distributions of data for instruments of different accuracies.

If a plot of the reference curve is used to obtain a reference value, additional error is introduced. Since the levels of vibration may vary over the range of pump operation, a method of assigning vibration reference values should be developed. Test results that frequently exceed reasonably established upper range limits indicate that the curve used for pump testing may not be sufficiently accurate. Because manufacturer's pump curves are generic, it is not likely that they will be acceptable for use as reference curves. Finally, regardless of the method, the error in the use of reference curves may be reduced by using more accurate instruments, by correcting for systematic errors, by increasing the number of data points, and by taking repetitive measurements at each data point.

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Computer-Based Pump Inservice Test Program at San Onofre Nuclear Generating Station

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ABSTRACT

Pump inservice test (IST) data is intended to provide documentation of pressure boundary integrity and to assess operational readiness during its service life. In most cases, the official paper records of the final, single value test results are the only documents retained. This "snapshot" approach focuses on the immediate results and all but ignores evaluation of equipment condition trends. This system satisfies regulatory requirements and is of limited usefulness for engineering analysis and equipment condition monitoring. The use of computers, in conjunction with specialized software and digital data collectors, to perform ISTs and archive data greatly enhances the information collected from these tests. In addition to improving both the quantity and quality of engineering data obtained, it drastically reduces the potential for human error.

INTRODUCTION

The San Onofre Nuclear Generating Station (SONGS) spends approximately 1500 engineering man-hours per year testing 90 pumps in Units 1, 2, and 3. This high number of man-hours made it highly desirable to obtain more useful information from this manpower investment. After field testing a number of commercially available digital data collectors and contacting several vendors, the SKF Palomar MICROLOG CMVA10 Data Collector and the VISION software program from Halliburton NUS Environmental Corporation were selected for this purpose.

The MICROLOG instrument possesses all of the functions of an industry standard data collector and vibration analyzer, as well as alphanumeric keyboard input. Its analytic capabilities provide useful information regarding a machine's mechanical condition. However, the real power behind the instrument is its capability to directly interface with more sophisticated software residing in network computers. Once the instrument is downloaded with pre-established routes for one or more IST pump procedures, the instrument's menu-driven program permits engineering personnel to conduct enhanced testing with only minimal training. Any conditions, hydraulic or mechanical parameters, that exceed preset limits are immediately flagged during the test (any parameter that requires calculations of several data points is processed by the software program). Following completion of the test, the data are uploaded to the VISION program via any networked computer in the plant.

The VISION software program is a highly secure, quality affecting program that produces the completed IST hardcopy record for engineer/supervisor signatures and official IST program retention. Automated functions of the program (such as trends, statistical parameters, comparisons to similar equipment, and spectrum displays) allow the engineer to immediately analyze the pump's condition. The trend displays are visually coordinated to differentiate the NORMAL, ALERT, and REQUIRED ACTION ranges in order to simplify the engineer's evaluation of the equipment. The program's network capability allows several individuals to view the same, or other, IST data simultaneously.

SONGS IST PROGRAM FEATURES

The SONGS computer-based pump IST program meets Section XI's instrumentation accuracy and the Nuclear Regulatory Commission's IST record retention requirement. The MICROLOG instrument is tested and certified on a periodic basis through the Measuring and Test Equipment (M&TE) program and the VISION program—certified as Quality Affecting software through a Validation and Verification (V&V) process—produces the official IST record.

The use of MICROLOG and VISION increases the quantity and quality of data collected and also reduces the potential for human error. The digital format of data collection allows the cognizant engineer to significantly increase the amount of data gathered without increasing manpower time spent. The instrument automatically collects vibration spectrum data, along with manual keyboard input of other parameters, and provides instant feedback if a parameter exceeds preset limits. This approach sensitizes the engineer to be more aware of the condition of the equipment.

All data are transferred and stored digitally. This method reduces human interface by eliminating transcription of data and minimizing calculation errors. Once the information is entered into the database, the VISION program performs the required calculations and makes them available for immediate review. All test data from current and past ISTs are available for comparison and analysis. In addition, comparison of data from other pumps may also be used to enhance the evaluation process. If required, several individuals may review the same, or other, information by accessing VISION through the SONGS Local Area Network (SLAN).

The review process takes place at the field, during data upload, initial acceptance at the cognizant engineer level, and finally, through final acceptance at the engineering supervisor level. If the IST satisfies each of these steps, the official hard copy of the test is automatically generated with all the appropriate fields filled in. The report is then printed, signed, and routed to all appropriate parties.

EQUIPMENT AND SOFTWARE SELECTION PROCESS

The SONGS's Performance Monitoring Group (PMG) based its selection of the MICROLOG instrument on its following capabilities:

1. Menu driven
2. Alpha-numeric keyboard input
3. Automatic spectrum data collection
4. High reliance on single key method of data collection
5. Minimal amount of training required to operate it
6. Easy to read screen (Pertinent information, such as point identification (ID), current and past readings, and alarm conditions, are clearly identified.)
7. Network compatible.

All of these features keep the data collection method relatively simple and enhance the quality of data available for analysis.

The selection of a specialized IST program was more difficult because there were no vendors that had one readily available or were willing to modify existing software to meet SONGS's needs. Predictive Maintenance Applications (PdMA) was contacted and agreed to modify their VISION program to meet the SONGS requirements outlined below:

1. Secured access (limited access based on assigned rights)
2. Network capability
3. Automatic data upload, sort, and archive

4. Manual data entry (keyboard)

5. Formula calculation based on several parameters

6. Data review

7. Spectral analysis

8. Trending

9. Procedure management

10. Miscellaneous report generation (scheduled tests, rejected test comments).

In addition, the program flags IST discrepancies, generates, stores, and routes the completed IST report for cognizant engineer and engineering supervisor approval. With VISION residing on the SLAN, IST data are accessible wherever a networked personal computer (PC) exists.

SONGS IST PROCESS

Figure 1 provides a simplified flow path of the 1st process. The performance of an IST requires the following steps be performed in sequence:

1. VISION Access

2. Route Download

3. Data Collection

4. Field Review

5. Route Upload

6. Data Review

7. Cognizant Engineer Approval

8. Engineering Supervisor Approval

9. IST Hardcopy Report

10. Signatures/Distribution.

All of the steps, with the exception of field review and signatures/distribution, require specific VISION access rights assigned by the IST program administrator, otherwise VISION will not allow access to them. The field review step is controlled through the MICROLOG's menu driven system, and the signature/distribution step is controlled through SONGS administrative procedures. Completion of this sequence provides information to plot trends/vibration spectrums, display alert or action levels, identify and display statistical parameters, build and compare IST machine histories, identify baselines, and generate IST reports.

VISION Access

All users have a network ID, along with their social security number, assigned to them during initial access update to the VISION program. Once an individual accesses the program, he or she will be requested to add a password known only to them. This security measure is required in order to prevent somebody else accessing VISION through that user's ID (VISION tracks and records all users accessing the program). VISION performs a security check by requesting either the network ID/password or social security number/password before allowing a user to enter the program (Figure 2).

Adding Machines to the VISION Database

Predefined IST routes are manually entered and then stored in the VISION database (Figures 3 and 4). This information is used to identify and track the IST due dates for all machines. Once this information is available in the database, it is then used to perform the IST.

Route Download

Any number of machines (route) are easily selected for download by using the PC insert key, or through the Mlog/Procedure function (Figure 5). When the MICROLOG is connected to the computer and the Mlog/Download function is selected, the program requests which instrument is being used to perform the test. This is necessary

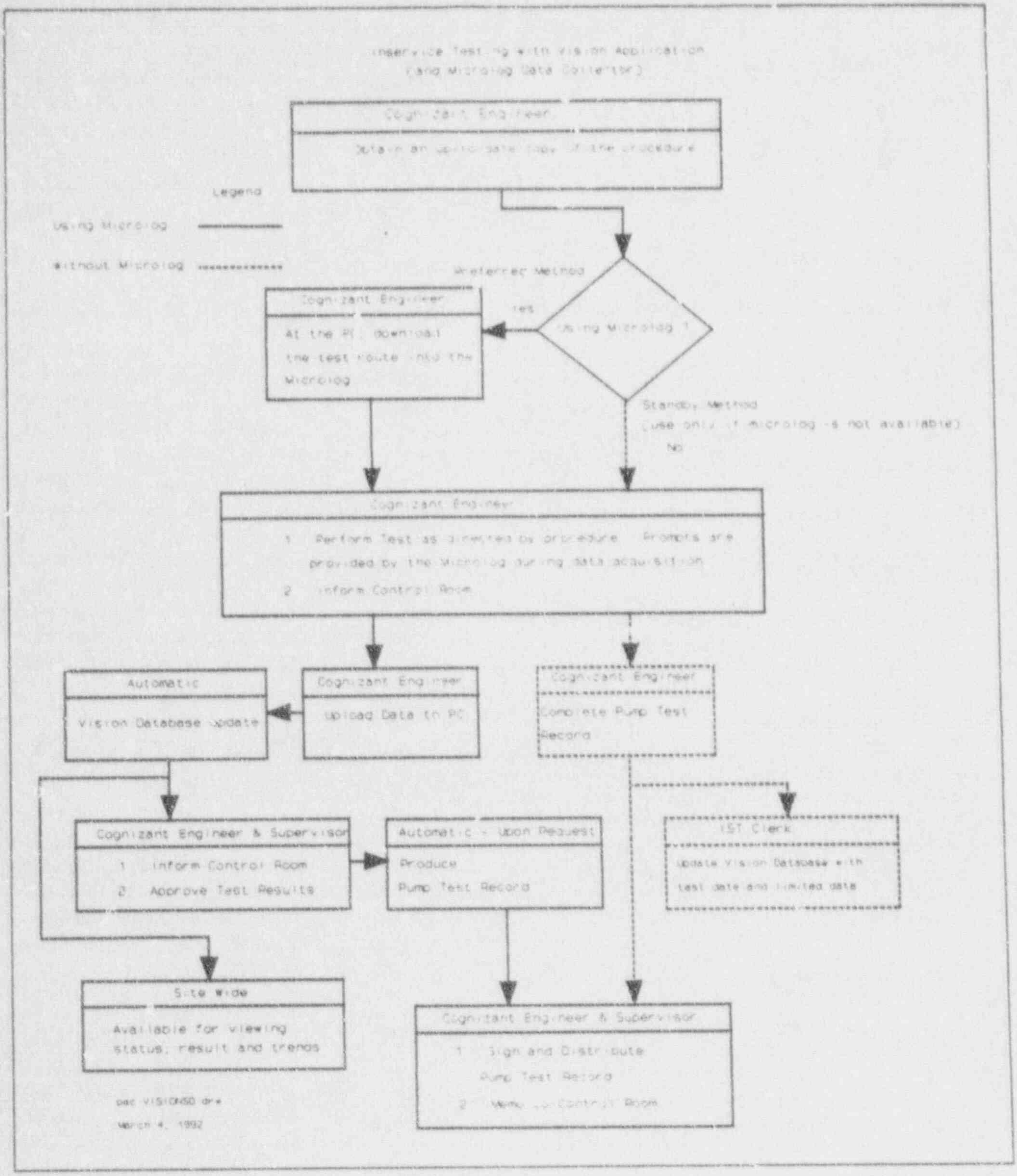


Figure 1. SONGS IST flow path.

```
* * V i s i o n * *

         ┌──────────Security Check──────────────┐
         │                                      │
         │ Name:......[John Doe            ]    │
         │ Password...[****** ]                 │
         │                                      │
         └──────────────────────────────────────┘

Enter your password.
```

Figure 2. This is one example of the security checks that appear on the screen. As the individual's password is entered, it is automatically masked by asterisks.

```
F2-Display F3-Report F4-DataBase F5-Mlog F6-Accept F7-Utility  F1-Help

            ┌──────────────┤ Edit Machine ├──────────────┐
┌──────────┤ Name...........[S21413MP112      ]
SONGS     │ Description....[SaltWater Cooling Pump      ]
├─«TEST S│ Supervisor:
├─«TEST D│  First Name..[John       ]
├─»SONGS │  Last Name...[Smith        ]
│  ├──S1-├──────────────────────────────────────────────────┤
│  ├──S1-│ Comments
│  ├──S1-│ [3/3/92: This machine's IST data is going to be  ]
│  ├──S1 │ [used as an example for the ASME/NRC symposium   ]
│  ├──S1-│ [paper on the VISION program. Most of the data will]
│  ├──S1-│ [be entered manually from old IST records and one ]
│  ├──S1-│ [set from a MICROLOG test setup. V. Herrera      ]
│  ├──S1-├──────────────────────────────────────────────────┤
│  ├──S1-│ Procedure......[SO23-V-3.4.8    ]
│  ├──S1-│ Data Sheet.....[SCE EG(123)53 Rev. 2]
│  ├──S1-│ Frequency......[84  ]
├─»SONGS │ Days +.........[31  ]  Days -.........[0   ]
│  ├──S21│ Outage.........[NO ]   Outage Mode....[0   ]
│  ├──S21│ Name Plate Information....[Allis-Chalmers    ]
└────<↑>─└──────────────────────────────────────────────────┘
Edit Machine  <Tab=Next Field> <Shift+Tab=Prev Field> <Esc=Exit Edit>
```

Figure 3. VISION's initial screen for adding a machine to the database. Several of the fields have pick list to choose from.

```
F2-Display F3-Report F4-DataBase F5-Mlog F6-Accept F7-Utility  F1-Help

┌──┤ S21413MP112 ├──┐┌────────────┤ Modify ├──────────────┐
│ Plant Power        ││ Name: [Pt. 1 Horiz. (0) D ]   Var Name: [ ] │
│ Run Time Before Tst││                                    │
│ Prestart Suction   ││ Description: [Motor Inboard Horzntl West-Mils ] │
│ Discharge Pressure ││                                    │
│ Running Suction    ││ Prism Ref: [          ]  Field #: [75 ] │
│ Differential Press ││                                    │
│ Motor Current      ││ Units: [Mils   ]      Down Load Y/N [Y] │
│ Flow Rate          ││                                    │
│ Water Head (Feet)  ││ Point Type: [Displacement /Vel  ]  │
│ Info.: L=          │├──────────┤ Frequency ├──────────┤
│ Calc.: Pi=Pa=      ││                                    │
│ Calc.: DP=         ││ Full Scale: [10   ]  Detection: [Peak To Peak] │
│ Pt. 1 Horiz. (0) D ││                                    │
│ Pt. 1 Horiz. (0) V ││ Input Sensitivity: [100  ]  Low Cut Off: [3 ] │
│ Pt. 1 Vert. (90) D ││                                    │
│ Pt. 1 Vert. (90) V ││ Alarm Type: [Level Alarm ]   # Averages: [8 ] │
│ Pt. 1 Axial   D    ││                                    │
│ Pt. 1 Axial   V    ││ High Frequency: [1000 ]      [Tolerances] │
│ Stuffing Box   D   ││                                    │
└────────────────────┘└────────────────────────────────────┘
 Tolerance Window  <↑=Change Field> <Esc=Exit> <Space=Tolerance Window>
```

Figure 4. This screen appears when a new point is being added or edited. All pertinent information associated with the point is displayed on the Modify window.

```
 Display   Report   DataBase   Mlog   Accept   Utility   Help
                   ┌──────────────────────┐
                   │ Procedure            │
┌──────────────────┤ SONGS ├──────────────┼──────────────┐
│SONGS             │ Download             │
│├─«SONGS Unit 1       │ Upload               │
││  ├──S1-AFW-G-10     │ Initialize Microlog │
││  ├──S1-AFW-G-10S    └─────────────────────┘
││  ├──S1-AFW-G-10W
││  ├──S1-CCW-G-15A
││  ├──S1-FWS-G-3A
││  ├──S1-FWS-G-3B
││  ├──S1-SHA-G-200A
││  ├──S1-SWC-G-13A
││  ├──S1-SWC-G-13B
│├─»SONGS Unit 2
││  ├──S21413MP112
││  ├──S21305MP504
└──────<↑>───<↓>───<←>───<→>───<Tab>────┘
```

Figure 5. Any of the machines listed on this screen can be selected for downloading through the use of the PC's Insert key or Mlog/Procedure function.

for tracking instruments used in the performance of the IST and to prevent unauthorized instruments from being used. Once the required instrument identification is entered, the route is stored into the MICROLOG and it is ready for use. The route contains all the necessary parameters (data points) to perform the IST.

Data Collection

All machines are listed on the MICROLOG's screen (Figure 6), and any one of them can be selected for testing through the use of the instrument's cursor arrows and ENTER keys. The first point on each machine requests the social security number of the individual performing the test. VISION uses this number to identify the cognizant engineer with a specific IST test record and retains it in its history files. All other points are specific parameters necessary to complete the test.

Field Review

During the performance of the IST, the MICROLOG prompts the cognizant engineer through each parameter. Each parameter is clearly described on the instrument's screen, i.e., Discharge Pressure—Pressure from 2PI6230. Initial evaluation is performed as each value is entered, and the engineer compares it against the previous test value (displayed on the screen). In addition, a separate field on the screen identifies any parameter in an alarm condition (Figure 7). When the pump test is complete, control room personnel are informed of the preliminary test results

Route Upload

When the MICROLOG is re-connected to the computer and the Mlog/Upload function is selected, the program retrieves all the data and stores them in a temporary file. The Cognizant Engineer performs another review of the data through the use of the Accept/Process Upload function (Figure 8). This gives him an opportunity to assign a unique IST record number, re-verify all data and verify/assign specific test equipment or local instrumentation used in the performance of the test. Upon completion of the Process Upload, VISION sorts the data and performs any necessary calculations (some parameters require multiple data points for the final test value). The data are then stored in a "hold" file and tagged for cognizant engineer approval. However, they are ready for immediate review by anybody who has VISION Review rights.

Data Review

The cognizant engineer accesses the Display/History Chart function to review all the current and previous data for the machine of interest (Figure 9). VISION displays any parameter in the ALERT or REQUIRED ACTION range by highlighting it in yellow or red. This method provides instant recognition that a parameter has exceeded its limits. In addition, statistical parameters are also displayed and are used to determine unusual patterns in the data. The cognizant engineer also has the option of reviewing the data through the use of graphical displays (Figure 10). He can view individual point/multiple points trend plots or single/waterfall spectrums upon request. Any number of people can review this information simultaneously.

Cognizant Engineer Approval

After reviewing the data through the History Chart or Graph functions, the cognizant engineer Approves/Rejects the test by using the Accept/Engineer function (Figure 11). If the test is rejected, the "hold" file is erased and the test has to be performed again. If the test is accepted, the "hold" file is now tagged for supervisor review. In either case, comments may be entered to document items of interest (maintenance order numbers, nonconformance reports, or general information).

Engineering Supervisor Approval

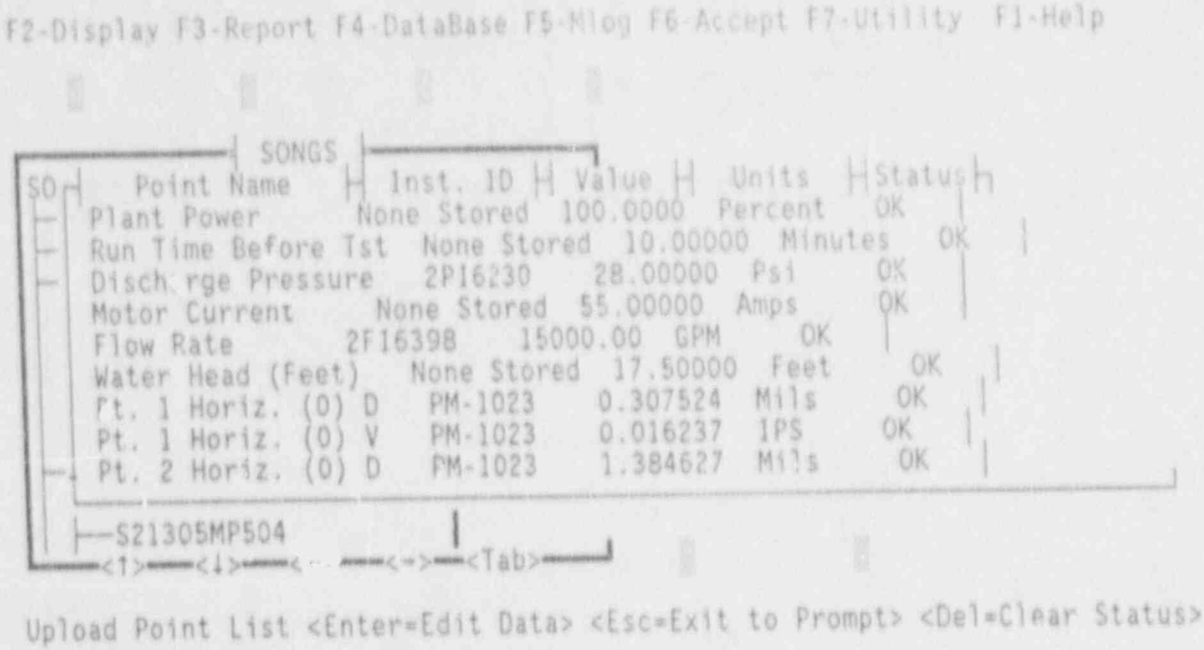
The engineering supervisor reviews the test data through the use of the History Chart or Graph functions. After reviewing it, he Approves/Rejects the test by using the Accept/Supervisor

```
12-11-91   98%          13:24:35
---------------- ROUTE ----------------
N NONROUTE
N S023-V-3.4.8
N   S21413' 9112
D     User ID
N     Plant Power
D     Run Time Before Tst
N     Discharge Pressure
D     Motor Current
N     Flow Rate
N     Water Head (Feet)
N     Pt. 1 Horiz. (0) D
N     Pt. 1 Horiz. (0) V

ENTER-start   MENU-options   ESC-exit
```

Figure 6. The MICROLOG's oper route screen is similar to this. All data points are listed for easy retrieval through the use of the instrument's cursor arrows and ENTER key. As each data point is collected, the "N" changes to "D," indicating that data have been collected.

```
12-11-91   98%          13:24:35
  DYNAMIC DATA POINT

 ID: Pt. 1 Horiz. (0) D        OS
 Motor Inboard Horzntal West-Mils
 UNIT  Mils

 ---THIS---   ---LAST---   ---ALARMS---
 0.3512       0.3208

 AVG
 SPECTRUM STORED

 0               0.350
ENTER-start   MENU-options   ESC-exit
```

Figure 7. The MICROLOG's screen is similar to this. All relevant information about the test point is displayed.



F2-Display F3-Report F4-DataBase F5-Mlog F6-Accept F7-Utility F1-Help

SONGS

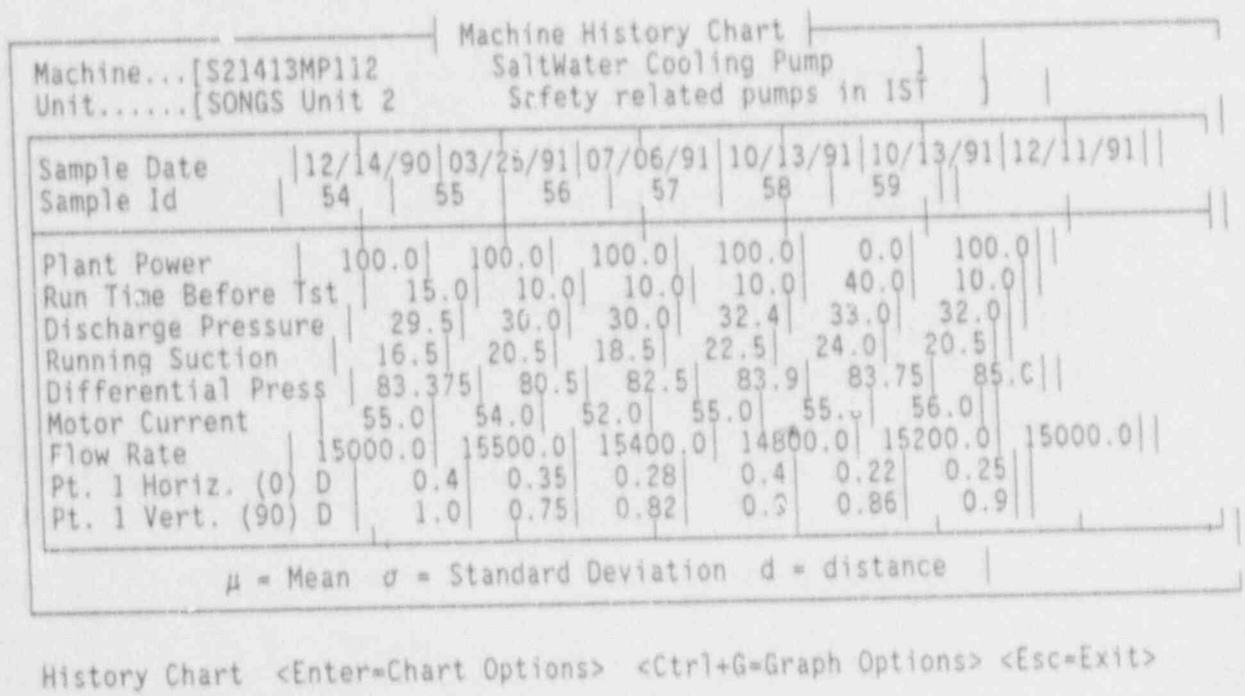
Point Name	Inst. ID	Value	Units	Status
Plant Power	None Stored	100.0000	Percent	OK
Run Time Before Tst	None Stored	10.00000	Minutes	OK
Disch rge Pressure	2PI6230	28.00000	Psi	OK
Motor Current	None Stored	55.00000	Amps	OK
Flow Rate	2FI6398	15000.00	GPM	OK
Water Head (Feet)	None Stored	17.50000	Feet	OK
Pt. 1 Horiz. (0) D	PM-1023	0.307524	Mils	OK
Pt. 1 Horiz. (0) V	PM-1023	0.016237	IPS	OK
Pt. 2 Horiz. (0) D	PM-1023	1.384627	Mils	OK

S21305MP504

<↑> <↓> <←→> <Tab>

Upload Point List <Enter=Edit Data> <Esc=Exit to Prompt> <Del=Clear Status>

Figure 8. VISION's Process Upload screen. Instruments are selected through the use of a pick list accessed by pressing the space bar. Any parameters out of tolerance are indicated in the Status section.



Machine History Chart

Machine...[S21413MP112 SaltWater Cooling Pump]
Unit......[SONGS Unit 2 Scfety related pumps in IST]

Sample Date	12/14/90	03/25/91	07/06/91	10/13/91	10/13/91	12/11/91
Sample Id	54	55	56	57	58	59
Plant Power	100.0	100.0	100.0	100.0	0.0	100.0
Run Time Before Tst	15.0	10.0	10.0	10.0	40.0	10.0
Discharge Pressure	29.5	30.0	30.0	32.4	33.0	32.0
Running Suction	16.5	20.5	18.5	22.5	24.0	20.5
Differential Press	83.375	80.5	82.5	83.9	83.75	85.C
Motor Current	55.0	54.0	52.0	55.0	55.[illegible]	56.0
Flow Rate	15000.0	15500.0	15400.0	14800.0	15200.0	15000.0
Pt. 1 Horiz. (0) D	0.4	0.35	0.28	0.4	0.22	0.25
Pt. 1 Vert. (90) D	1.0	0.75	0.82	0.[illegible]	0.86	0.9

μ = Mean σ = Standard Deviation d = distance

History Chart <Enter=Chart Options> <Ctrl+G=Graph Options> <Esc=Exit>

Figure 9. The cognizant engineer uses this screen to review present and past history. Miscellaneous information and graphs are available by pressing the proper keys.

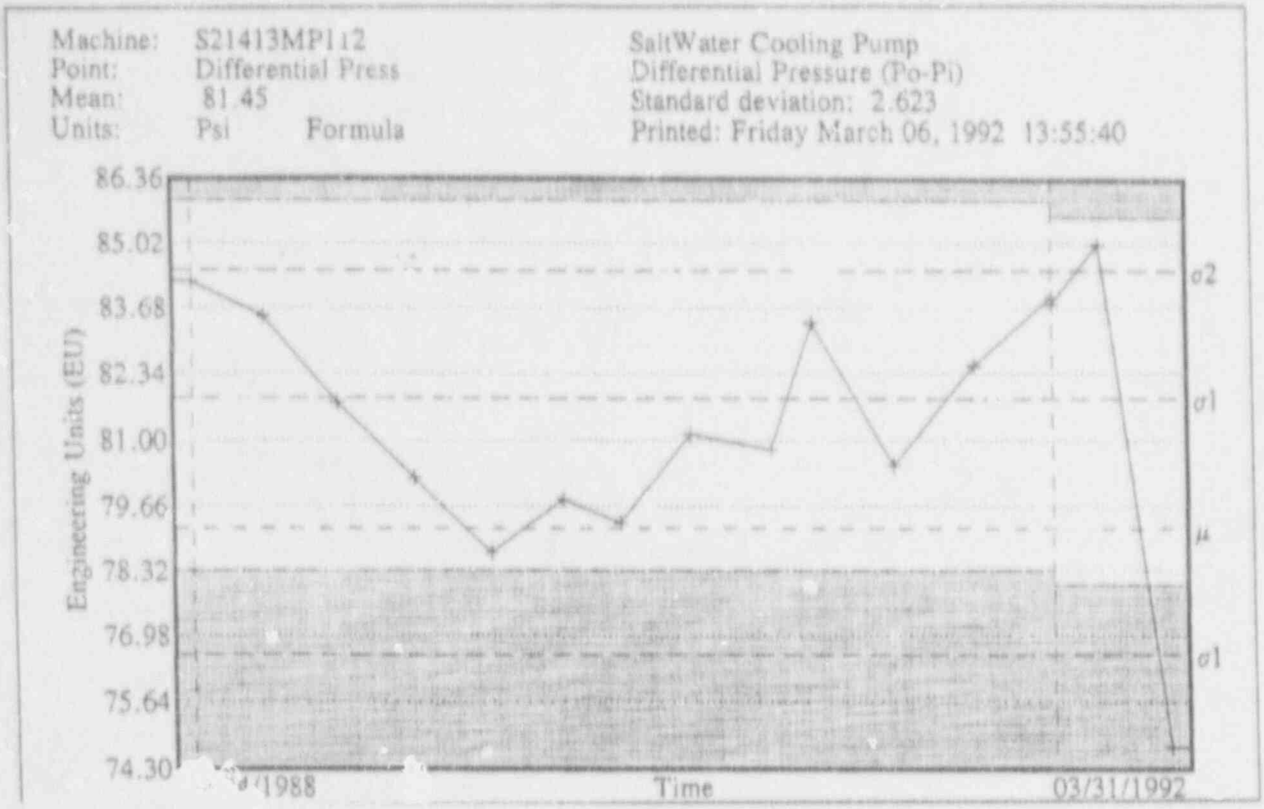


Figure 10. Example of trend available for review. Miscellaneous information is clearly labeled along with statistical parameters. Tolerances are represented by shaded regions and new baselines are shown at each vertical dotted line.

```
F2-Display F3-Report F4-DataBase F5-Mlog F6-Accept F7-Utility  F1-Help
              Engineer Accept/Reject Sample

  User Name........[Victor M. Herrera         ]
  Machine Name.....[S21413MP112     ]
  Record Number....[2P112-10-91 ]
  Accept (Y/N).....[YES]
  NCR Number and/or =
  M.O. Number......[MO# 91100376000           ]
  Required Action..[None          ]

  Enter Notes for Sample:
  [Evaluating VISION Accept/Reject process.          ]
  [                            ]
  [                            ]
  [                            ]

Edit Sample Info <Tab=Next Field> <Shift+Tab=Prev Field> <Esc=Exit Edit>
```

Figure 11. At this screen, the cognizant engineer accepts or rejects the IST. Miscellaneous information is entered to document any actions taken.

function. The same steps outlined in Cognizant Engineer Approval apply at this level of review. However, upon acceptance, the file becomes permanent and cannot be erased. At this time, the program automatically generates a IST report file ready to be printed upon request.

IST Hardcopy Report

The cognizant engineer or engineering supervisor produces a hard copy of the completed IST data record, Figure 12, through the Report/Print function. The VISION program duplicates Southern California Edison's Inservice Pump Test Record, SCE EG (123)53 Rev. 2, with all parameters (reference values, test values, acceptable limits, instrument IDs/calibration due dates, and miscellaneous information) filled in and ready for signature approval.

Signatures/Distribution

The cognizant engineer and engineering supervisor sign the Inservice Pump Test Record, distribute the results to all interested parties, and provide a memo to control room personnel officially notifying them of the IST results.

PLANNED DATA COLLECTION ENHANCEMENT

The Performance Monitoring Group at SONGS has taken steps to further simplify the data collection process by having vibration monitoring test pads installed at the appropriate sample locations. The test pads are one-turn stainless steel attachments that are stud mounted or epoxied in place. A mating connector attached to the vibration detector completes the arrangement. This setup has been in place on the secondary system since April 1991 and has shown that it is easy to use and has increased the reliability of readings taken. There is no longer a question as to how and where data were taken when different individuals are involved. Consistency of readings has been the key benefit to this approach and is the main factor for proceeding with the installation of test pads on safety related equipment.

SUMMARY

The use of networked computers, special IST software, and digital data collectors reduce the potential for human error and dramatically increase the amount and quality of data available for equipment evaluation. The four-tiered data review process—Field, Upload Process, Engineer, and Supervisor—all but eliminates the human error factor and increases the quality of data collected. The "finger tip" availability of current and past IST data to all interested parties provides for immediate evaluation of potential equipment problems. The network compatibility of the system allows individuals with different areas of expertise to review the data simultaneously and make group decisions on the best course of action to take.

ACKNOWLEDGMENTS

The SONGS Computer-Based IST Program would not have been possible without the full cooperation of Halliburton NUS Environmental Corporation's PdMA department and Mr. Dale E. Mountain, Manager of Software Services. His expertise in software and understanding of SONGS's requirements have produced a very powerful IST program.

NOMENCLATURE

IST	Inservice Testing of pumps according to the Section XI and OM-6 velocity vibration limits.
MICROLOG	SKF Palomar Technologies microprocessor-based data collector.
M&TE	Measuring and Test Equipment program used at SONGS to calibrate and document all required test [illegible] each piece of test equipment. The program also controls issuance of test equipment.

INSERVICE PUMP TEST RECORD

CDM Invoice No. ______

Test Procedure Used: SO23-V-3.4.8 Test Date: 10-13-1991

Unit: SONGS Unit 2 Report No. 2P112-10-91

1 Pump Tag No. S21413MP112	2 Pump Name (System) Saltwater Cooling Pump
3 Tested By ISTUSER	4 Reason for Test Retest/IST; 3P114 is now 2P112
5 [illegible] OR	6 Reference IST Report No./Date 3P114-10-89 10-15-1991
7 Test Frequency 84 Days 8 Date Last Tested 10-13-1991	9 Run Time Before Test 60

HYDRAULIC DATA	INSTRUMENT ID. (ENTER "C" FOR CALC'D VALUE)	CALIBRATION DUE DATE	UNITS	SET REF.	REFERENCE VALUE	TEST VALUE	ACCEPTABLE RANGE
10 Prestart Suction Pressure (Ps)	C	N/A	Psi		22.5	24	> 2.5
11 Speed (N)	N/A	N/A	N/A	N/A	N/A	N/A	N/A
12 Discharge Pressure (Pd)	2P16230	10-29-1991	Psi		32.4	33	N/A
13 Running Suction Pressure (Pi)	C	N/A	Psi		22.5	24	> 2.5
14 Differential Pressure (Pd - Pi)	C	N/A	Psi		83.9	83.75	78.03-85.58
15 Motor Current (A)	N/A	N/A	Amp		55	55	< 60
16 Flow Rate (Q)	2FI6398	12-12-1991	GPM		14800	15200	> 14800

Calculations

L = 14.5'
Pi = Pa + 38.5 - L = 24
DP = 2.25 * Pa + L - 5 = 2.25 * 33 + 14.5 - 5 = 83.75

MECHANICAL DATA Vibration Instrument ID 12-7839 \ 12-7839A Calibration Due Date 12-24-1991 \ 12-26-1991

		Vibration Axis	Displacement (Mils): REFERENCE VALUE	TEST VALUE	ACCEPTABLE RANGE	Velocity (IPS): REFERENCE VALUE	TEST VALUE	ACCEPTABLE RANGE
17	Point No. 1	Horiz. (0 Deg.)	.4	.22	< 1	.035	.02	< .088
18		Vert. (90 Deg.)	.9	.86	< 1.8	.05	.04	< .125
19		Axial	.35	.44	< 1	.025	.04	< .063
20	Point No. 2	Horiz. (0 Deg.)	1.3	1.1	< 2.6	.09	.06	< .225
21		Vert. (90 Deg.)	.9	1.2	< 1.8	.07	.06	< .175
22		Axial	.8	.3	< 1.6	.06	.02	< .15

23 Lubrication Level/Pressure Bulls Eye X Oil Feeder ___ Other ___ Sat X Under ___

24 BEARING TEMPERATURES

Instrument ID. N/A
Cal. Due Date N/A
Date Last Taken 10-13-1991
Reference Date Record No. N/A
Reference Temperature N/A
Max. Allowable Temperature N/A

	Point No. 1 TIME	TEMP	% CHANGE	Point No. 2 TIME	TEMP	% CHANGE
1	N/A	N/A		N/A	N/A	
2	N/A	N/A	N/A	N/A	N/A	N/A
3	N/A	N/A	N/A	N/A	N/A	N/A
4	N/A	N/A	N/A	N/A	N/A	N/A

CORRECTIVE ACTION/REVIEW RESULTS

25 REQUIRED ACTION: None	26 NCR NO. [illegible]: NBM 91100376000
27 ENGINEER PERFORMING OPERABILITY ANALYSIS: Victor M. Herrera	28 SUPERVISING ENGINEER OR DESIGNEE: Generic Supervisor DATE 03-06-1992

Figure 12. Inservice pump test record.

Network ID A unique name given to all individuals that use SLAN. The name is used by VISION as the first level of security prior to accessing the program.

OM-6 ASME/ANSI OM-1987, Operation and Maintenance of Nuclear Power Plants, Part 6, Inservice Testing of Pumps in Light-Water Reactor Power Plants, with Addenda through May 31, 1989.

PCs Personal Computers

PdMA The Predictive Maintenance Applications department of Halliburton NUS Environmental Corporation.

PMG SONGS Station Technical Performance Monitoring Group

Section XI American Society of Mechanical Engineers (ASME), Boiler and Pressure Vessel Code, Section XI, 1977 edition including Addenda through Summer of 1979.

SLAN SONGS Local Area Network

SONGS San Onofre Nuclear Generating Station operated by Southern California Edison

VISION Specialized pump IST software program developed by PdMA.

V&V Verification and Validation plan used on VISION to document the tests performed on it, which qualifies VISION as a Quality Affecting program. The V&V was written in accordance with the Topical Quality Assurance Manual, Chapter 1-J, Electronic Data Processing Controls, and SONGS' Nuclear Information Services procedure S0123-XVII-10.15, Control of Computer Based Systems.

Pump Testing in the Nuclear Industry: The Comprehensive Test and Other Considerations

Thomas F. Hoyle
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ABSTRACT

The American Society of Mechanical Engineers Operations and Maintenance Working Group on Pumps and Valves is working on a revision to their pump testing Code, ISTB-1990. This revision will change the basic philosophy of pump testing in the nuclear industry. Currently, all pumps are required to be tested quarterly, except those installed in dry sumps. In the future standby pumps will receive only a start test quarterly to ensure the pump comes up to speed and pressure or flow. Then, on a biennial basis all pumps would receive a more extensive test. This comprehensive test would require high accuracy test gauges to be used, and the pumps would be required to be tested near pump design flow. Testing on minimum flow loops would not be permitted except in rare cases. Additionally, during the comprehensive test, measurements of vibration, flow, and pressure would all be taken. The OM-6 standard (ISTB Code) will also require that reference values of flow rate and differential pressure be taken at several points instead of just one point, which is current practice.

The comprehensive test is just one step in ensuring the adequacy of pump testing in the nuclear industry. This paper also addresses other concerns and makes recommendations for increased quality of testing of certain critical pumps and recommendations for less stringent or no tests on less critical pumps.

INTRODUCTION

Inservice testing (IST) of pumps in the nuclear industry has been conducted essentially the same since the introduction of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code Section XI in 1973. Recently, these rules were revised in the ASME O&M Code, Subsection ISTB. This paper will focus on changes to ISTB that are currently under review by O&M. These proposed changes to ISTB will require more comprehensive testing of safety-related pumps and will reduce the extent of periodic testing for standby pumps. In addition, this paper makes some additional recommendations for actions to be taken by ASME, the U.S. Nuclear Regulation Commission (NRC), and owners of nuclear power plants to increase the effectiveness and minimize the impact of inservice testing.

BACKGROUND

Subsection IWP of the Winter 1973 Addenda of Section XI included for the first time rules for IST of pumps. Section XI prescribed hydraulic and vibration testing of ASME Class 1, 2, and 3 pumps that were provided with an emergency power source and that had a specific function in shutting down a reactor, or in mitigating the consequences of an accident. The basic philosophy of Section XI was to detect change by measuring and evaluating hydraulic and mechanical (vibration) parameters. Once a significant change was detected, it was necessary for the owner to determine the cause of the change and take appropriate corrective action. Section XI contains specific corrective action requirements and specific "alert" and "required action" ranges (Table 1) that may be absolute or based on reference values. The Section XI pump testing, as previously described, is currently being performed by commercial nuclear plants in the U.S. Most of the requirements contained in the Winter 1973 Addenda of IWP, except the monthly test frequency, are still in effect today.

When the transition from IWP to ISTB is made, a significant change in the pump testing methodology will be noted. The Pump and Valve Working Group's philosophy in the development of ISTB was that the mechanical condition of a pump, as detected by vibration measurement, would be an earlier and more reliable predictor of pump degradation. A significant improvement in the ability to assess a pump's mechanical condition, introduced in OM-6, is the improved method for measuring vibration. OM-6 provides requirements for measuring and assessing vibration velocity measurement, which is a more sensitive indicator of pump condition. Subsection ISTB allows a wider range of acceptance criteria for hydraulic parameters, but at the same time requires more vibration measurements. In addition, the vibration acceptance criteria has been made more conservative (Table 2). That is, an upper limit of vibration is now specified for centrifugal pumps. Many utilities are currently using vibration detection methods and acceptance criteria specified by ISTB (OM-6).

Table 1. Allowable ranges of test quantities.

Test quantity	Acceptable range	Alert range		Required action range	
		Low	High	Low	High
P (positive displacement pumps)	0.93 to $1.10P_r$	0.90 to $<0.93P_r$	...	$<0.90P_r$	$>1.10P_r$
ΔP (vertical line shaft pumps)	0.95 to $1.10\Delta P_r$	0.93 to $<0.95\Delta P_r$	...	$<0.93\Delta P_r$	$>1.10\Delta P_r$
Q (positive displacement and vertical line shaft pumps)	0.95 to $1.100Q_r$	0.93 to $<0.95Q_r$	...	$<0.93Q_r$	$>1.10Q_r$
ΔP (centrifugal pumps)	0.90 to $1.10\Delta P_r$	...	...	$<0.90\Delta P_r$	$>1.10\Delta P_r$
Q (centrifugal pumps)	0.90 to $1.10Q_r$	...	...	$<0.90Q_r$	$>1.10Q_r$

GENERAL NOTE: The subscript r denotes reference value.

Source: ASME O&M Working Group on Pumps and Valves-Comprehensive Pump Test, Revision 8, 12/4/91.

Table 2. Ranges for test parameters.[a]

Pump type	Pump speed	Test parameter	Acceptable range[a]	Alert range	Required action range
Centrifugal and vertical line shaft[b]	<600 rpm	V_d or V_v	$\leq 2.5\ V_r$	>2.5 V_r to 6 V_r or >10.5 mils	>6 V_r or >22 mils
Centrifugal and vertical line shaft[b]	$\geq$ 600 rpm	V_v or V_d	$\geq 2.5\ V_r$	>2.5 V_r to 6 V_r or >0.325 in./sec	>6 V_r or >0.7 in./sec
Reciprocating	—	V_d or V_v	$\geq 2.5\ V_r$	>2.5 V_r to 6 V_r	>6 V_r

a. Vibratim parameter is per Table ISTB5.2-1. V_r is vibratim reference value in the selected unit throughout the table.

b. Refer to Figure ISTB5.2-1 (ASME OM Code, 1990) to establish displacement limits for pumps with speeds ≥ 600 rpm or velocity limits for pumps with speeds < 600 rpm.

Source: ASME OM Code, Subsection ISTB, 1990 Edition.

A meeting between the NRC and the O&M committee was held in April of 1989 to discuss NRC acceptance of OM-6. At that time, the NRC expressed their concern about accepting OM-6 because of the expanded acceptance criteria for pump hydraulic parameters (Table 3). The O&M representatives proposed the idea of a comprehensive pump test as a resolution to the NRC's concern. The comprehensive pump test would supplement and, in some cases, replace the current quarterly test. It would use more accurate instrumentation to obtain hydraulic parameters at higher flow rates that could be used to accurately detect degradation. The NRC agreed to accept OM-6, as written, if the working group would start working on a comprehensive pump test at once. The working group agreed. The remainder of this paper describes the changes in pump testing methodology for the comprehensive pump test and the basis used to develop this methodology.

PUMP TESTING

The working group set out to develop a test method that would meet the NRC objective of ensuring that significant pump degradation is detected and acted on. The working group also wanted a test that would be beneficial in providing additional operational data to help support a conclusion that a slightly degraded pump was still operable. To achieve these goals, some of the instruments used to assess pump condition would have to be more accurate. And, the test would need to be practical, and thus able to be performed at plant refueling outages.

The working group also recognized that resources at every plant are limited. Therefore, if a more difficult test were required, it would be appropriate to specify the test at a lesser frequency if possible.

NEW PUMP CATEGORIES AND TEST FREQUENCIES

The first task was to categorize pumps found at commercial nuclear plants. Initially, the working group considered three categories; however, these were difficult to define clearly. The group then decided to define two pump categories: Group A pumps and Group B pumps. Group A pumps are those that operate continuously or routinely during normal operation, cold shutdown, or refueling. These pumps are required to receive a

Table 3. Acceptance criteria—Group A test.

Test parameter	Acceptable range	Alert range	Required action range	
			Low	High
P (positive displacement pumps)	0.93 to $1.10P_r$	0.90 to $<0.95\Delta P_r$	$<0.90P_r$	$>1.10P_r$
ΔP (vertical line shaft pumps	0.95 to $1.10\Delta P_r$	0.93 to $<0.95\Delta P_r$	$<0.93\Delta P_r$	$>1.10\Delta P_r$
Q (positive displacement and vertical line shaft pumps)	0.95 to $1.10Q_r$	0.93 to $<0.95Q_r$	$<0.93Q_r$	$>1.10Q_r$
ΔP (centrifugal pumps)	0.90 to $1.10\Delta P_r$	...	$<0.90\Delta P_r$	$>1.10\Delta P_r$
Q (centrifugal pumps)	0.90 to $1.10Q_r$	...	$<0.90Q_r$	$>1.10Q_r$

GENERAL NOTE: The subscript r denotes reference value.

Source: ASME O&M Working Group on Pumps and Valves—Comprehensive Pump Test, Revision 9, March 24, 1992.

Group A test quarterly. Examples of Group A are residual heat removal (RHR), service water, and component cooling pumps. Group B pumps are those installed in standby systems that are not routinely operated, except for testing or maintenance. These pumps are required to receive a Group B test quarterly. Examples of Group B are containment spray, core spray, and standby liquid control (SLC) pumps. As noted in Table 4, all pumps receive a comprehensive test biennially. Table 5 shows the test parameters required for each test.

As the above objectives were developed, three types of inservice tests (Table 4) evolved: Group A test, Group B test, and comprehensive test. The preservice test was also changed to obtain more baseline performance data.

Group A Test

The Group A test (Tables 3 and 6) is similar to the current quarterly Section XI and OM-6 type pump tests that all nuclear utilities currently perform. The OM-6 test differs somewhat from its predecessor, the Section XI test, in that the "required action" range high is now 110% of the reference value for hydraulic parameters instead of 103%, as specified by Section XI (Table 3). Also, there is no "alert range" for hydraulic parameters. A more important change between Section XI and OM-6 is the vibration acceptance criteria (Table 2). Note that the vibration criteria contains an absolute limit for both the "alert" and "required action" ranges in addition to a limit based on a multiple of a reference value.

Group B Test

The simplest test is the Group B or start test (Table 7). This test requires the pump to be successfully started. A successful start is one in which the discharge pressure, or flow rate, is within 10% of its reference value. Note that for variable speed pumps, the speed must be varied to within 1% of its reference value. For a positive displacement pump, flow rate is the parameter that must be compared to its reference value at a reference point of discharge pressure.

Table 4. Inservice test frequency.

Pump group	Group A test	Group B test	Comprehensive test
Group A	Quarterly	N/A	Biennially
Group B	N/A	Quarterly	Biennially

a. A comprehensive test may be substituted for a Group A test.

b. A comprehensive or Group A test may be substituted for a Group B test.

Source: ASME O&M Working Group on Pumps and Valves—Comprehensive Pump Test. Revision 8, December 4, 1991.

Table 5. Inservice test parameters.

Quantity	Preservice test	Group A test	Group B test	Comprehensive test	Remarks
Speed: N	X	X	X	X	If variable speed
Differential pressure: ΔP	X	X	X[a]	X	Centrifugal pumps, including vertical line shaft pumps
Discharge pressure: P	X	X		X	Positive displacement pumps
Flow rate: Q	X	X	X[a]	X	—
Vibration:	X	X		X	Measure either V_d or V_v
Displacement, V_d	—	—	—	—	Peak-to-peak
Velocity, V_v	—	—	—	—	Peak

a. For positive displacement pumps flow rate shall be measured or determined; for all other pumps differential pressure or flow rate shall be measured or determined.

Source: ASME O&M Working Group on Pumps and Valves—Comprehensive Pump Test, Revision 9, March 24, 1992.

Comprehensive Test

The comprehensive test (Tables 6 and 8) requires pressure instrumentation with greater accuracy (Table 9), and it places tighter limits on both hydraulic and mechanical acceptance criteria. That is, the hydraulic criteria now have a high "required action" range of 103% of the reference value.

The intent of the comprehensive test is two fold: first, to achieve testing that results in more accurate data and, second, to achieve testing at higher flow rates. The data can then be trended

Table 6. Acceptance criteria—Group A and comprehensive test criteria.[a]

Pump type	Pump speed	Test parameter	Acceptable range	Alert range	Required action range
Centrifugal and vertical line shaft[b,c]	<600 rpm	V_d or V_v	$\leq 2.5\ V_r$	$>2.5\ V_r$ to $6\ V_r$ or >10.5 to 22 mils	$>6\ V_r$ or >22 mils
Centrifugal and vertical line shaft[b]	≥ 600 rpm	V_v or V_d	$\geq 2.5\ V_r$	$>2.5\ V_r$ to $6\ V_r$ or >0.325 to 0.7 in./sec	$>6\ V_r$ or >0.7
in/sec shaft[b,c]					
Reciprocating	—	V_d or V_v	$\geq 2.5\ V_r$	$>2.5\ V_r$ to $6\ V_r$	$>6\ V_r$

a. Vibratim parameter is per Table 2 ISTB5.4.1-1. V_r is vibratim reference value in the selected units.

b. Including positive displacement pumps except reciprocating.

c. Refer to Figure ISTB5.2-1 to establish displacement limits for pumps with speeds ≥ 600 rpm or velocity limits for pumps with speeds <600 rpm.

GENERAL NOTES: The subscript r denotes reference value.

Source: ASME O&M Working Group on Pumps and Valves—Comprehensive Pump Test, Revision 9, March 24, 1992.

Table 7. Acceptance criteria—Group B test.

Test parameter	Acceptable range	Alert range		Required action range	
		Low	High	Low	High
ΔP (centrifugal pumps including vertical line shaft pumps) or	0.90 to $1.10\Delta P_r$	...	...	$<0.90\Delta P_r$	$>1.10\Delta P_r$
Q (all pump types)[a]	0.90 to $1.10Q_r$	...	...	$<0.90Q_r$	$>1.10Q_r$

a. Measure Q for positive displacement pumps.

GENERAL NOTES: The subscript r denotes reference value.

Source: ASME O&M Working Group on Pumps and Valves—Comprehensive Pump Test, Revision 9, March 24, 1992.

Table 8. Acceptance criteria—comprehensive test.

Test parameter	Acceptable range	Alert range	Required action range	
			Low	High
P (positive displacement pumps	0.93 to $1.03P_r$	0.90 to $<0.93P_r$	$<0.90P_r$	$>1.03P_r$
ΔP (vertical line shaft pumps)	0.95 to $1.03\Delta P_r$	0.93 to $<0.95\Delta P_r$	$<0.93\Delta P_r$	$>1.03\Delta P_r$
Q (positive displacement and vertical line shaft pumps)	0.95 to $1.03Q_r$	0.93 to $<0.95Q_r$	$<0.93Q_r$	$>1.03Q_r$
ΔP (centrifugal pumps)	0.93 to $1.03\Delta P_r$	0.90 to $<0.93\Delta P_r$	$<0.90\Delta P_r$	$>1.03\Delta P_r$
Q (centrifugal pumps)	0.94 to $1.03Q_r$	0.90 to $<0.94Q_r$	$<0.90Q_r$	$>1.03Q_r$

GENERAL NOTE: The subscript r denotes reference value.

Source: ASME O&M Working Group on Pumps and Valves—Comprehensive Pump Test, Revision 9, March 24, 1992.

Table 9. Required instrument accuracy.

Quantity	Group A and B tests	Comprehensive and preservice tests
Pressure	± 2	± 1/2
Flow rate	± 2	± 2
Speed	± 2	± 2
Vibration	± 5	± 5
Differential pressure	± 2	± 1/2

Source: ASME O&M Working Group on Pumps and Valves—Comprehensive Pump Test, Revision 8, December 4, 1991.

more meaningfully, and changes in pump performance characteristics can be determined more accurately. By specifying more accurate instruments, the measurement uncertainties for both the baseline and subsequent inservice tests will be minimized.

To compensate for the extra effort required to install more accurate test instruments and test at full flow, the comprehensive test is required to be run only once every 2 years. This requirement should have a minimal impact on safety because many of these pumps are run only for testing purposes. The pumps that are run routinely will continue to receive a quarterly test equivalent to the type of testing currently required by OM-6.

More Extensive Preservice Testing

An essential part of the O&M IST methodology is establishing a set of reference values for a particular pump and then comparing future test results to these reference values to detect changes in pump condition. The revised preservice testing requirements require a minimum of five points of pump flow rate and head to be measured for developing a pump curve for each centrifugal pump installed in a variable resistance system (Figure 1). The preservice test is performed using more accurate instruments, as is the comprehensive test. One of the advantages of using a pump curve instead of a single reference point is that an additional set of reference values can be established based on the curve. The pump curve, which provides additional higher accuracy data above that taken for Section XI tests, should also prove beneficial in evaluating future pump performance problems.

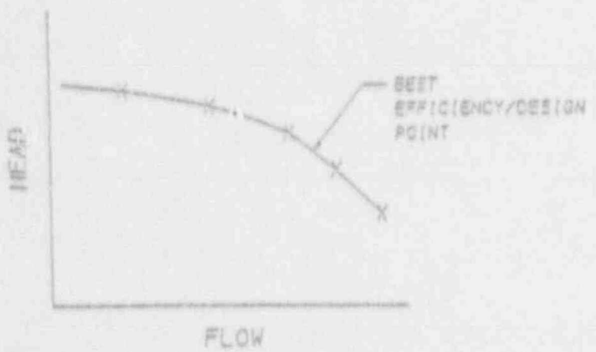

Figure 1. Sample pump curve.

Another significant change to the requirements for preservice testing is that reference values for assessing the results of future tests are required to be taken at or near the pump design flow rate (± 20% for centrifugal pumps and at or near design pressure for positive displacement pumps). The bases for these requirements are to acquire pump hydraulic data at points readily duplicated and that provide assurance that significant degradation, if present, will be detected. This change precludes establishing reference values on pump minimum flow circuits for the comprehensive test, as is currently allowed by Section XI or OM-6.

FUTURE CONSIDERATIONS FOR PUMP TESTING

Some critical pumps are required by plant Technical Specifications to deliver a specified flow rate at a given differential pressure. The testing is usually specified to be done in accordance with ASME Code requirements. ASME Section XI requires all Class 1, 2, or 3 safety-related pumps, including some Technical Specification pumps, to be tested in accordance with the ASME Code on a periodic basis and after maintenance has been performed that could affect pump performance.

Anytime a question arises concerning the operability of a safety-related pump, the pump is declared operable or inoperable based on the results of a surveillance test. A surveillance test is a Section XI Code test that measures flow rate and differential pressure for comparison with minimum requirements, if so specified in the Technical Specifications or by the utility.

Reliance on the surveillance test to demonstrate pump operability is normally an acceptable practice, but in certain instances the results can be misleading. For example, if a pump can only be tested at minimum flow conditions, a significantly degraded pump may not be identified by the Code test. This is especially true if the test loop does not contain a flow measuring device. In contrast the comprehensive test will give a much higher level of assurance that a pump can perform its intended function.

Potential Problems with Design Basis Testing

A current priority of the NRC is to require design basis testing of all pumps and valves that have an active safety function. The obvious merits of this philosophy are that fewer assumptions are necessary and that the test shows that the pump will operate at design basis conditions. However, a significant drawback with design basis testing is that uncertainties associated with the testing must be taken into account. For example, if the measurement of flow is accurate to within 4% when gauge accuracies and temperature are taken into account, then an additional 4% margin would be needed. This can be a problem for some pumps that operate near their design point. A pump, unlike most mechanical equipment, does not operate better at lower flow rates. It is free from vibration and cavitation at its design point. Therefore, it may not be possible to demonstrate that a well-designed pump that operates at its design point is adequately sized after measurement uncertainties are subtracted. A full flow test may be more desirable for operating plants.

Pump Testing Program

Section XI and O&M require a pump to be in good operating condition when it is initially tested and when reference values are established. Any significant change from reference conditions requires an increased testing frequency or corrective action. These requirements should be viewed as a basis to build an effective IST program, which will help to ensure detection of degradation and, consequently, increase pump reliability and safety. All testing programs should be structured such that degrading performance is identified and corrective action can be scheduled prior to component failure.

Many test parameters, which are not required by the Code, may be valuable to observe during pump testing. For example, pump bearing temperature may be an excellent parameter to monitor if vibration readings suggest possible bearing degradation. Likewise, pump suction conditions may give important clues when differential pressure indicates degrading hydraulic performance. Pump seals should be inspected specifically at each pump test for excessive leakage. Baseline readings of additional parameters (such as bearing temperature, inlet pressure, motor bearing vibration, and motor power) are also recommended. These parameters may prove very valuable for subsequent analysis when a Code-required test parameter exceeds acceptance criteria and requires corrective action. Also, when a test program is established, one should recognize and consider the types of failures that may not be identified by Code testing. For example pump shaft failure may not be determined until the failure occurs. Note that failures of this type are infrequent and usually identified as applicable to a problem with a specific pump design. Another consideration is that on any pumped system, the system plays an important, if not dominant, role. Therefore, understanding the specific application is critical in understanding or predicting failure. A test engineer should review the pump in its installed condition and determine if it is operating as expected. An overall program for ensuring increased reliability should also incorporate good maintenance practices in addition to IST.

Another potential shortcoming of the O&M Code and its predecessor, Section XI, is that pump drivers are not required to be tested. However, the motor or turbine is critical to the operation of a pump. Parameters that monitor degradation of the motor should be specified as part of an effective testing program.

Scope of Pump Testing

Currently all pumps that perform an active safety-related function require testing in accordance with ASME Section XI. Some pumps are less important than others, and rigorous testing as specified currently by Section XI and the O&M codes may not be warranted. For example, diesel fuel oil transfer pumps are included in many IST programs. These pumps normally supply a much larger volume of fuel oil than required to operate the diesel. Even a severely degraded pump will likely meet its system requirements. Many other pumps, such as keep fill or control room habitability system pumps, perform a safety-related function, but are of lesser importance than emergency core cooling system or other safety-related pumps. Given the ever-increasing cost of testing and the difficulty of obtaining resources, it seems logical and necessary that we focus these resources as effectively as possible to increase the safety of the plant operation through a graded approach to IST based on component importance. The NRC and Code writing organizations should reassess the need for the same level of testing of all components.

Recommendations

The following changes should be considered to increase the quality of testing of critical safety-related pumps and at the same time minimize the impact of pump testing on the industry.

1. The NRC should consider mandatory implementation of the new O&M comprehensive testing philosophy as soon as possible. These requirements should be implemented within [illegible] years of publication and not at the start of the next 10-year IST interval.

2. Where specific acceptance criteria for pumps is not established in the plant Technical Specifications, the owner should be allowed the option of setting acceptance criteria based on system considerations. A comprehensive test is recommended, but should be optional.

3. Design basis testing should be required only for those critical pumps where specific credit is taken in safety analysis, e.g., RHR, HPCS, and other pumps that have flow rate and pressure requirements specified in the plant Technical Specifications.

4. Pumps that have minimal safety impact or are tested as part of a system, e.g., a skid-mounted pump, should be specifically excluded from a formal IST program. However these pumps should be tested with the skid-mounted equipment.

SUMMARY

When OM-6 was published several years ago, the working group recognized that more technical work was needed on this standard to produce pump testing results that would not only fulfill a Code requirement, but would provide meaningful results for the industry to use in assessing pump performance. The comprehensive test is one step in the direction of more meaningful pump testing, and thereby improving plant safety. Through the use of more accurate instruments, real pump problems can be distinguished from instrument errors more easily. This improved pump testing will ensure that pumps which are degrading are appropriately identified. This testing will not falsely require pumps to be declared inoperable because of instrument inaccuracies. Other changes are also needed to eliminate or reduce testing requirements for pumps that are less critical to plant safety. Resources should be directed towards those areas of pump testing where maximum benefit will be realized.

ACKNOWLEDGMENTS

The author would like to express sincere thanks and appreciation to R. Scott Hartley for his help in developing the comprehensive pump test and for his editing of this paper.

Appendix A
O&M Code, Standards and Guides

Throughout this paper I have referred to the pump testing Code as ISTB-1990. I would now like to give a short summary of the changes in the O&M documents in the past 3 years (Figure A-1). O&M standards were initially issued as individual standards.

ASME/ANSI OM-1987

In 1987 the O&M Committee consolidated all O&M documents in ASME/ANSI OM-1987 "Operation and Maintenance of Nuclear Power Plants." Part 6 of OM-1987 is technically the same as OM-6.

OM CODE-1990/OM-S/G-1990

In 1990 the ASME/ANSI OM-1987 document described above was split into two documents: a) ASME OM Code-1990 "Code for Operation and Maintenance of Nuclear Power Plants" (Figure A-2) and (b) ASME OM-S/G-1990, "Standards and Guides for Operation and Maintenance of Nuclear Power Plants." (Figure A-3) Pump testing (OM-6) is now contained in the O&M Code and is redesignated as ISTB "Inservice Testing of Pumps in Light-Water Reactor Power Plants."

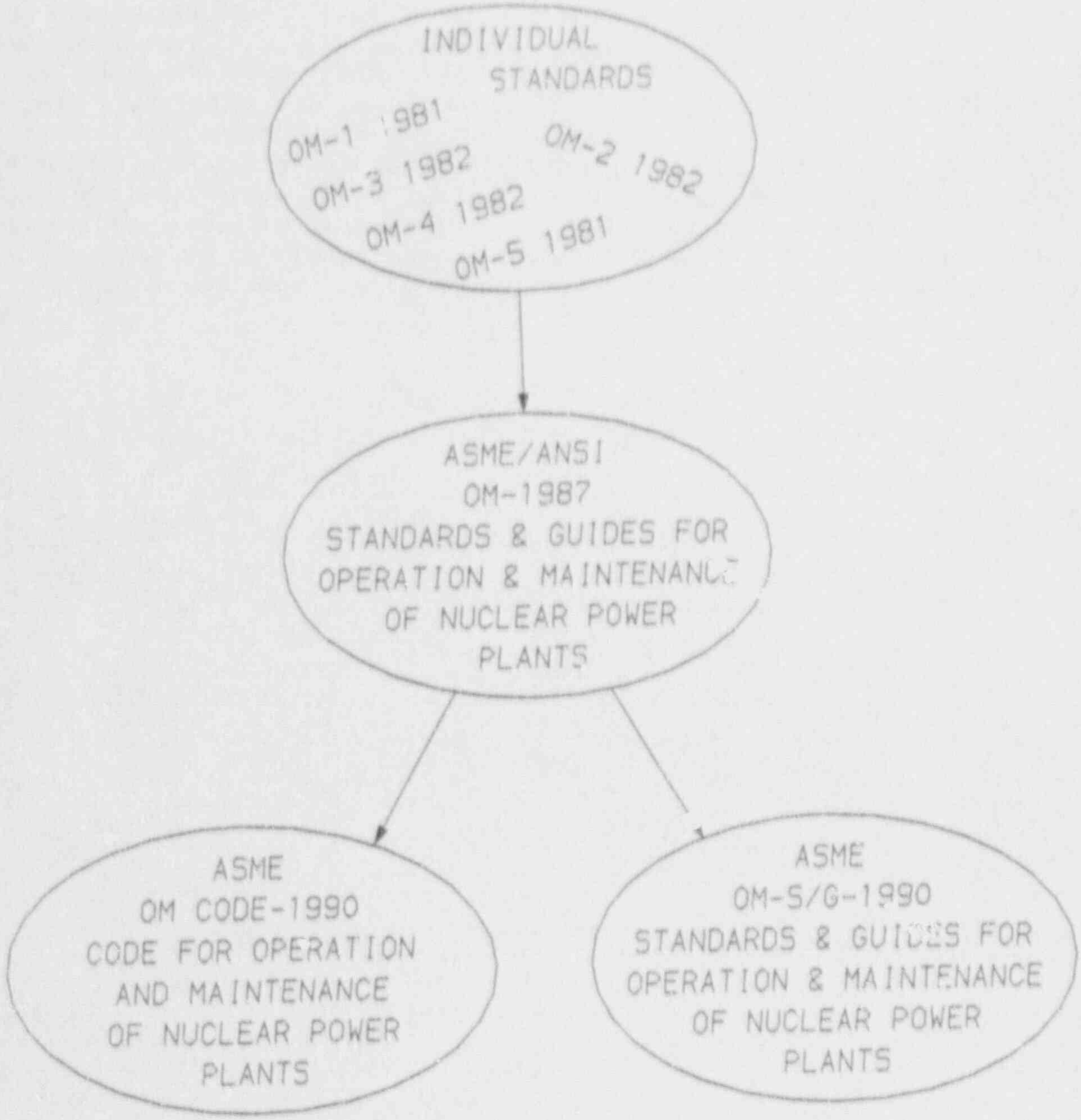

Figure A-1. O&M standards.

SECTION IST

RULES FOR INSERVICE TESTING
OF LIGHT-WATER REACTOR POWER PLANTS

CONTENTS

Subsection ISTA	General Requirements
Subsection ISTB	Inservice Testing of Pumps in Light-Water Reactor Power Plants
Subsection ISTC	Inservice Testing of Valves in Light-Water Reactor Power Plants
Subsection ISTD	Inservice Testing of Dynamic Restraints (Snubbers) in Light-Water Reactor Power Plants

MANDATORY APPENDIX

I	Inservice Testing of Pressure Relief Devices in Light-Water Reactor Power Plants

NONMANDATORY APPENDICES

A	Preparation of Test Plants
B	Dynamic Restraint Examination Checklist Items
C	Dynamic Restraint Design and Operating Information
D	Comparison of Sampling Plans for Inservice Testing of Dynamic Restraints

Figure A-2. ASME OM Code-1990, code for operation and maintenance of nuclear power plants.

STANDARDS

Part 2	Requirements for Performance Testing of Nuclear Power Plant Closed Cooling Water Systems
Part 3	Requirements for Preoperational and Initial Start-Up Vibration Testing of Nuclear Power Plant Piping Systems
Part 12	Loose Part Monitoring in Light-Water Reactor Power Plants
Part 13	Requirements for Periodic Performance Testing and Monitoring of Power-Operated Relief Valve Assemblies
Part 16	Inservice Testing and Maintenance of Diesel Drives in Nuclear Power Plants

GUIDES

Part 5	Inservice Monitoring of Core Support Barrel Axial Preload in Pressurized Water Reactors
Part 7	Requirements for Thermal Expansion Testing of Nuclear Power Plant Piping Systems
Part 8	Start-Up and Periodic Performance Testing of Electric Motor Operators on Valve Assemblies in Nuclear Power Plants
Part 11	Vibration Testing and Assessment of Heat Exchangers

Figure A-3. ASME OM-S/G-1990, standards and guides for operation and maintenance of nuclear power plants.

Session 1C
IST Programmatic Issues

Session Chair
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Gulf States Utilities

Test Quality

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ABSTRACT

Inservice testing of safety-related components at nuclear power plants is performed under the American Society of Mechanical Engineers Boiler and Pressure Vessel Code (the Code). Subsections IWP and IWV of Section XI of the Code state test method and frequency requirements for pumps and valves, respectively. Tests vary greatly in quality and frequency. This paper explores the concept of test quality and its relationship with operational readiness and preventive maintenance. This paper also considers the frequencies of component testing. Test quality is related to a test's ability to detect degradation that can cause component failure. The quality of the test depends on several factors, including specific parameters measured, system or component conditions, and instrument accuracy. The quality of some currently required tests for check valves, motor-operated valves, and pumps is also discussed. Suggestions are made to improve test quality by measuring different parameters, testing valves under load, and testing positive displacement pumps at high pressure and centrifugal pumps at high flow rate conditions. These suggestions can help to improve the level of assurance of component operational readiness gained from testing.

INTRODUCTION

Operators of nuclear power plants (NPPs) are responsible for ensuring that certain pumps and valves will function properly, if needed, to prevent or mitigate the consequences of an accident. Inservice testing (IST) of these components during their service lifetimes gives a measure of assurance that they will perform their intended function(s).

Section XI of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (the Code) states various test requirements and defines operational readiness as "the capability of a pump (or valve) to fulfill its function." The Code tests assess in varying degrees the operational readiness of pumps and valves. However, the Code tests are written in a general fashion to cover a great variety of components and are, therefore, not optimal for every

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b. This presentation was prepared (in whole or in part) by an employee of the United States Nuclear Regulatory Commission. It presents information that does not currently represent an agreed-upon staff position. The NRC has neither approved nor disapproved its technical matter.

component. An operator may test a component to the letter of the Code requirements and get little information about the condition of that component.

The Nuclear Regulatory Commission (NRC) recognizes many of the deficiencies of Code testing and has expressed their concerns in many ways, including a letter to the ASME Operations and Maintenance (OM) Main Committee (Richardson, 1991). That letter expressed serious concerns about the state of the ASME Code test requirements.

There is no perfect test for predicting a component's condition relative to failure. Instead, we test components and achieve degrees of knowledge about their condition, or capacity, that are far from perfect. Ideally, component testing would identify, in a timely manner, any form of degradation that could pose a threat to a specific component's continued ability to function properly. Certain forms of degradation, such as a valve's exterior finish or the clarity of a pump's gearbox lubricant, would not affect the component's operation and have no value to the process of assessing component condition. The results of an ideal test would be a powerful tool to a variety of disciplines. These results would help plant operators ensure that continued operation is safe. The plant maintenance department could use the information to schedule preventive maintenance more effectively, which could lead to improved plant availability. The results of ideal tests would also accurately indicate the failure rates of components and their various parts. Accurate failure rates might be useful in determining appropriate frequencies for testing components. These failure rates are also important to analysts performing plant risk studies, such as probabilistic risk assessments (PRAs). PRAs can indicate the higher risk components at a facility and may be used as tools for determining how to allocate plant resources. The effectiveness of a PRA, used to allocate resources, is no better than the quality of the information used in that study, which in turn is affected by test quality.

This paper discusses the testing regulations and codes that guide testing at NPPs. It then defines test quality and discusses its relationship with component operational readiness. Additionally, the paper identifies several factors that affect the quality of testing and discusses the relationship between test quality and preventive maintenance.

The paper then discusses test frequencies and their relationship with test quality. It shows that frequencies that are dependant on plant conditions can be extended randomly.

The paper closes with a discussion of the quality of tests that are often performed on motor-operated valves (MOVs), check valves, and pumps, which can be low, even if the tests are performed according to the Code. We then make suggestions for improving the tests of those components.

REGULATIONS AND TEST CODES

Regulations and test codes set the stage for testing at NPPs. This section discusses some of the regulations applicable to IST programs at licensed commercial NPPs. It also addresses the Code of Federal Regulations (CFR, 1991) provisions for granting relief from the testing requirements and some of the codes and standards applicable to testing.

Regulations

Title 10, CFR 50 requires testing of safety-related pumps and valves. With certain exceptions, these components must be tested according to the requirements of Section XI of the ASME Code (ASME, 1980), Subsections IWP and IWV, for pumps and valves respectively. The Code subsections specify several test plan attributes, including test methods and frequencies. The testing is intended to assess operational readiness of components as defined in the Code. Standard Technical Specification 4.0.5 and other plant technical specifications state that inservice testing of pumps and valves must be performed in accordance with Section XI of the Code and applicable addenda identified by Section 10 CFR 50.55a(g).

Component testing at NPPs can be difficult. Often design constraints or operating conditions can preclude testing certain components according to the Code test method and/or frequency requirements. Alternatives to Code requirements may be used when authorized by the NRC. 10 CFR 50 states the "Provisions for Relief from Test Requirements" and allows the regulatory authority, the NRC, to grant relief, allowing modified testing or an extension of the test interval. 10 CFR 50.55a, Paragraphs (a)(3)(i), (a)(3)(ii), and (g)(6)(i) state the following criteria for granting relief. Relief may be granted if

- "The proposed alternatives would provide an acceptable level of quality and safety"
- "Compliance [with the Code requirement] would result in hardship or unusual difficulty without a compensating increase in the level of quality and safety."

The third provision is that, "The Commission will evaluate determinations . . . that Code requirements are impractical. The Commission may grant such relief and may impose alternative requirements as it determines is authorized by law . . . giving due consideration to the burden upon the licensee if the requirements were imposed on the facility."

The regulations allow the NRC to grant relief for an extended interval between tests, particularly if the Code-specified interval is impractical and the Code-specified test method will be used at the extended frequency [according to (g)(6)(i)]. When licensees request relief from testing in accordance with the Code-specified test methods using either of the other relief provisions of the regulations, the quality of the alternate test is even more important and should be considered carefully.

Test Codes

Test quality is driven by the quality of the test Codes used by the industry. The following discussion identifies the currently required test Code and provides the status of rules for using newer codes. Most plants test in accordance with the ASME Code, Section XI, Subsections IWP and IWV, as required by 10 CFR 50. The NRC is in the final stages of rulemaking to reference the 1988 addenda to the 1986 edition of ASME Section XI. That edition and addenda refers to Parts 6 and 10 of the ASME Operation and Maintenance (O&M) Standard (ASME, 1987) for pump and valve test requirements. Subsection IWP of the ASME Code states that "Pump testing shall be performed in accordance with the requirements stated in ASME/ANSI OM (Part 6)." According to Subsection IWV, Part 10 is similarly applicable for valves. That Standard was rewritten and approved by the Board on Nuclear Codes and Standards in the fall of 1990 as the ASME OM Code-1990 (ASME, 1990). The NRC is currently working to reference the OM Code for pump and valve test requirements.

The OM test codes are constantly being improved. Several committees are working to develop more specific and better methods of testing components. The ASME OM Committee has formed working groups and task groups and chartered them to improve particular areas of the OM Code. These committees are composed of skilled technical people. Their work will enhance the quality of the OM Code. Licensees should share their knowledge and resources with these committees and learn what they can from them to improve the quality of testing at their plants.

EFFECTS OF TEST QUALITY ON OPERATIONAL READINESS, TESTING, AND PREVENTIVE MAINTENANCE

Test quality is essentially a measure of the level of knowledge about a component's condition that is gained from the results of testing done on the component. The quality of the testing is vital to the assessment of the component's ability to perform its intended function. Test quality is related to, or has an effect on, many aspects of plant operation. Test quality affects the level of component operational readiness assured through testing. Preventive maintenance and test quality are closely related. This section describes these relationships.

Test Quality and Operational Readiness

The Code requires testing to assess the operational readiness of each covered component. The applicable tests vary depending on the component type and function. There is a close relationship between test quality and the degree of assurance of operational readiness achieved through the testing. On the component level, the total test quality is essentially the sum of the quality of the various tests conducted on the subject component. When carefully analyzed, some of the testing performed according to the Code is of low quality (as discussed in the section of this paper on tests of MOVs, check valves, and pumps) and allows a limited assessment of operational readiness. On the other hand, some testing done according to the Code can yield a very high level of assurance of operational readiness. From discussions with licensees over the years, it is evident that the limited assessment of operational readiness afforded by some of the Code tests is not widely recognized. We must, therefore, closely examine test quality even when testing according to the Code. The total test quality for a component should be understood by those performing the tests and analyzing the results. They should strive for high test quality and recognize that data quality and validity can vary, even if all the testing is performed according to the Code.

Testing and Test Quality

This section discusses aspects of testing that greatly impact test quality. The process of testing a component may consist of several tests, each having its own level of quality to contribute to the overall testing. Many factors complicate the testing process and affect test quality. Types of component degradation vary greatly, as do the techniques employed to detect them. Often changes in component condition are detected through parametric measurements, such as flow rate or vibration levels for pumps, or stroke times for valves. Factors that affect test quality include test conditions, parameters measured, measurement inaccuracies, the applicability of the acceptance criteria, and human factors influences. Note that certain types of degradation might be tolerable in one application, but not in another, for essentially identical components. For instance, slight valve seat degradation on a service water pump discharge check valve could be acceptable, whereas the same degradation on an identical check valve at a primary containment vessel boundary would not.

An important consideration affecting test quality is the state of the component or the system containing it. For instance, is the system at low or high temperature or pressure? Is the valve being tested under load? Is the pump being tested at a high flow rate? These conditions have been shown to be very important for both pumps and valves. The importance of loading for some valves must be addressed to achieve a high-quality test. Otherwise, the effects of frictional forces on the sliding surfaces, which can affect operational readiness, cannot be assessed.

The parameters that are measured to assess a component's condition play a large part in the test quality. The measured parameter should be appropriate to the phenomena of interest. For instance, the stroke time (time from the open to closed position or vice versa) of an electric MOV can be measured and assessed. That parameter can be compared to a reference value (baseline) stroke time that was taken when the valve was newly installed or refurbished. Stroke time is one of the tests specified in the Code for MOVs. That test, however, is widely considered to be of little value for assessing the condition of most MOVs. This is partly because the test is generally done under light loads. Severe valve degradation would be necessary to produce a significant change in the measured stroke time under these conditions. As will be discussed later in this paper, valve stem thrust is a much better parameter than stroke time for assessing the condition of some MOVs.

Another concern is the accuracy or inaccuracy of parametric measurements. Certain parameters are sensitive to changes in a component's condition. But the accuracy of the measurement is the telling factor for determining if too much change has occurred and whether the component no longer meets the criteria for operational readiness. The accuracy of the measurement may also challenge the applicability of the acceptance criteria. If the measurements are inaccurate, a component in good condition might be declared inoperable unnecessarily, resulting in unneeded maintenance, or vice versa. Higher accuracy instruments allow more accurate baselining and give a better indication of changes in condition. The improved data are also easier to analyze for trends, since data scatter caused by inaccuracies is reduced. A proposed revision to the OM Code for pump testing employs pressure instruments of higher accuracy for pump testing. When feasible, licensees should test using high accuracy instruments, particularly for component performance baselining.

There are also many human factors, such as the levels of ambient lighting, quality of readout interfaces, test procedures, training, and experience, that have a significant effect on test quality. These factors should be considered to help ensure high test quality.

Another important factor affecting test quality is the attitude of the persons involved in the testing. Some test primarily to comply with the test Codes. Others perform the tests that comply with the Code, but that is secondary to their intent of obtaining as much information as possible about the condition of the tested components. The latter philosophy is obviously preferred by the NRC. Another factor that affects the quality of a testing program is its priority. Utilities should demonstrate their commitment to high quality testing by giving high priority to testing programs.

This section discussed several aspects of testing that impact test quality. These aspects should each be considered as part of the component testing process to help ensure high test quality. The mechanical aspects of testing, such as parameters measured, component loading, and instrument accuracies, and the less tangible aspects, such as attitude and experience, all play a part.

Test Quality and Preventive Maintenance

Preventive maintenance can reduce the need for tests and is, therefore, interrelated with test quality, as we will see in the following. The goal of preventive maintenance is to maintain, repair, or replace components or their parts before they fail. This maintenance helps ensure plant safety by increasing component reliability. A good program of preventive maintenance also helps to reduce the costs resulting from unscheduled downtime and replacing failed components or other damaged equipment. The importance of testing to detect degradation can be lessened by conducting timely preventive maintenance activities. For instance, if part of a component can withstand only a certain number of cycles because of the effects of fatigue or erosion, and maintenance is performed on that part so that the part is refurbished or replaced before it could cause the component to fail, testing to determine the condition of that part is of little value. This can be particularly important if it is difficult or expensive to test the part to assess its condition. However, much of the maintenance at NPPs has been done in response to test results or failures, and since failures of standby components may be latent, there may be no maintenance called for or done. This is true particularly if testing was of low quality and failed to indicate the need to perform preventive maintenance.

The NRC has expressed significant concerns about the effectiveness of maintenance at NPPs. These concerns culminated in what is known as the "Maintenance Rule." On July 10, 1991 the NRC amended the 10 CFR 50 to incorporate the rule (10 CFR 50.65), "Monitoring the Effectiveness of Maintenance at Nuclear Power Plants." This rule has been followed by a draft Regulatory Guide. The Maintenance Rule addresses NRC's concern and takes a performance-based approach to improving the maintenance performed at NPPs. These improvements in maintenance can affect the testing needed at NPPs.

TEST FREQUENCIES

The frequency of testing also affects test quality. Infrequent testing may not detect problems in a timely manner for scheduling maintenance or repairs. Also, frequent testing can consume resources that could be more effectively directed toward other safety issues. This section briefly discusses the Code test frequencies. The Code specifies several frequencies for pump and valve testing. The Code writers recognize the general nature of some Code-specified frequencies and the need to optimize test frequencies. The frequency that applies depends upon the component type and function. Pumps require quarterly testing. Most valves must be tested either quarterly or during cold shutdowns. Some valve tests, such as position indicator checks and leak tests, are specified to be done once every 2 years. As written, 10 CFR 50 allows the NRC to consider and allow testing at frequencies other than those specified in the Code. Generally, when licensees request relief from the Code frequency, the proposed alternate frequency is to test during refueling outages. As you will see here, the interval between tests whose frequency is based on plant conditions, can vary.

Cold-Shutdown Testing

Cold-shutdown testing is primarily a consideration for valves. Many valves cannot be tested quarterly during plant operation and are, therefore, identified to be tested when the plant enters the cold-shutdown condition. Section XI of the Code allows testing valves during cold shutdowns rather than quarterly if "such operation is not practical during plant operation." The newer OM Code allows the frequency to be further reduced to refueling outages, "if exercising is not practicable during plant operation or cold shutdowns." There is also a provision in the OM Code that cold-shutdown testing need not be started unless the shutdown is longer than 48 hours, and testing may be stopped when the plant is ready for startup. Therefore, the testing of some of the valves identified to be tested during cold shutdown may be deferred to a later time. This would result in a longer interval between tests.

There is generally no set interval associated with cold shutdown testing. With the exception of planned mid-cycle outages, most cold shutdowns are unplanned. The duration of the shutdown is kept as short as possible so that the plant can be put back into power production. For short-duration cold shutdowns, few valves may be tested. Therefore, depending on the plant operating schedule, many of the valves identified to be tested during cold shutdowns may be tested only during refueling outages.

Refueling Outage Testing

Many valves and a few pumps cannot be tested quarterly during plant operation or during the cold-shutdown condition. These components are identified in requests for relief and are to be tested when the plant is in the refueling condition. Plants enter that condition on a variety of schedules, based on their fuel cycle. Nominal fuel cycles are 12, 18, and 24 months. There is no explicit cap on the interval between refueling outage testing. An extended cold shutdown could cause an indefinite extension of testing done during refueling. In this case, the interval extension would not be subject to NRC evaluation or approval.

TESTS OF MOVS, CHECK VALVES, AND PUMPS

This section discusses some of the tests conducted on MOVs, check valves, and pumps. Also discussed are some of the problems with current tests and recommendations that can enhance the tests. The recommendations relate to changes in testing methodology, such as measuring different parameters, testing valves under load, or testing pumps at a high flow rate or pressure. Information on testing methods being developed by Code writing committees is also provided.

Testing Motor-Operated Valves

The Code has several tests that may apply to MOVs. One is the exercise test, where the valve is exercised to the position required for it to fulfill its safety function. During the test, an appropriate indicator (such as position indicator light) is observed, to ensure that the valve disk has changed position. Another test is the stroke time. The time of the valve stroke is measured and compared to the owner-assigned or technical specification limit and to the previous stroke time, to determine if corrective action is needed. The stroke time test is intended to be the primary test for assessing the condition of the valve. Another test that applies if the valve performs a leakage restriction function is a leak rate test. The leak test is effective for indicating the condition of the valve's seating surface. The Code also requires that valves with remote position indicators be observed at least once every 2 years to verify that valve operation is accurately indicated. The Code does not require testing under load, nor specify that valves must be tested under repeatable, or reference, conditions (as it does for pumps).

MOV research conducted at the Idaho National Engineering Laboratory (INEL) for the NRC (NRC, 1991a) established the importance of evaluating MOVs under load. Without adequate loading, the frictional forces on the sliding surfaces of the MOVs (and other phenomena that affect the valve's ability to perform its function) cannot be measured, and their effect on the operational readiness of the subject MOVs can not be assessed. NRC Generic Letter No. 89-10 (NRC, 1989a) on testing and surveillance of MOVs expressed many of the NRC's concerns in this area and requested that licensees help ensure the functional capability of safety-related MOVs by reviewing design bases, verifying switch settings, and testing the valves under design-basis loading conditions (where practical), among others.

In addition to the loading, it is important to take the correct parametric measurements. For example, stroke timing of alternating current (ac) powered MOVs is ineffective for assessing their condition. It gives little assurance of operational readiness. Though difficult to measure, a telling feature regarding performance capabilities of ac MOVs is the valve assembly's conversion of operator torque to stem thrust. Motor operators are torque output devices. They control motor torque, not stem thrust. The conversion to thrust occurs across the stem-to-stem-nut interface, which is outside the control of the motor operator. At the stem-nut interface, the torque from the motor and drive train is converted into stem thrust. Measuring the operator torque provides information on operator capabilities. A better assessment of the valve's operational readiness can be made by measuring the stem thrust and comparing that with appropriate acceptance criteria based on the thrust required to meet the design-basis functional requirements. Measurement of parameters such as operator torque and stem thrust, or appropriate alternate indicators of those quantities under expected differential pressure and flow, would be important inputs in obtaining higher test quality for MOVs.

The industry is now actively improving the procedures for assessing the condition of MOVs. An ASME Code development committee, the Working Group on Motor-Operated Valves (OM-8) and other industry groups are devoting significant efforts to improving the quality of MOV testing. Licensees should consider the results of the OM-8 efforts to help improve the quality of their tests for MOVs.

Testing Check Valves

The Code specifies several tests that may apply to check valves. One is the exercise test to the open or closed position, or to both positions. The Code specifies test requirements for "normally open valves" and for "normally closed valves." As with the MOV, the valve is exercised to the position required to fulfill its function (this is called a full-stroke exercise). The Code says that during the open test, an appropriate indicator must be observed to ensure that the valve ". . . . disk moves promptly away from the seat when the closing pressure differential is removed and flow through the valve is initiated. . . ." The requirement reads similarly for the closure test. Most check valves do not have position indication. The most common method used to full-stroke exercise check valves open is to pass the maximum required accident flow rate through the valve. If the valve has a power actuator, a stroke time, and possibly a fail-safe test (depending on the actuator function), could be required. The fail-safe test is essentially removing power and verifying that the valve travels to the "safe" position. Another test that may apply is a leak rate test. That test applies if the valve performs a leakage restriction function.

The NRC has expressed significant concerns regarding the testing of check valves. A large portion of NRC's Generic Letter No. 89-04 (NRC, 1989b) was devoted to check valve testing and alternatives. The NRC has also prepared a Temporary Instruction for inspectors to use when assessing licensee check valve testing programs (NRC, 1991b). A critical assessment of the condition of many of the safety-related check valves at NPPs is difficult, since most are totally enclosed, with no position indication. Therefore, determining that the disk moves promptly to or from the seat can be difficult.

Some check valves are normally closed and have only an open safety function. They require the testing specified for normally closed valves, which translates to a full-stroke exercise open with flow. Passing the accident flow rate through the check valve to full-stroke exercise it open ensures that the flow passage is not obstructed at the time of the test, but gives little other information about the valve's condition. The valve disk could be severely degraded, or missing, but that condition would not be detected by the personnel performing the full-flow test or reviewing the test results. The quality of that test alone would, therefore, be very low. The full-flow test could be augmented with a reverse flow test to verify that the disk (or obturator) is still attached and is free to move into the flow channel. Even better would be a leak rate test with strict acceptance criteria. That would indicate that the disk moved into the flow channel (closed) and could indicate the presence of misalignment resulting from degradation of valve internals and the condition of the seating surface. Some NPPs have implemented a program of performing "alternate position tests," such as the closure test just described, for their normally closed valves.

Significant research has been done over the past few years by groups, such as the Nuclear Industry Check Valve Group (NIC) and researchers at the Oak Ridge National Laboratory (ORNL), to develop new methods for nonintrusively assessing the condition of check valves. These methods employ technologies, such as magnetics, acoustics, radiography, and ultrasonics, to assess changes in the condition of check valve internals. The results of this research are promising and informative regarding assessing valve condition. Employing some of these emerging techniques may allow better assessment of operational readiness, and effect better test quality.

The industry is improving the procedures for assessing the condition of check valves. The ASME Working Group on Check Valves (OM-22), and other industry groups are working to improve the quality of check valve testing. Changes to the OM Code test methods developed by those groups are in the OM Committee approval process. Licensees should consider the results of those efforts, as well as augmenting the required tests, to help improve test quality.

Testing Pumps

This part talks briefly about pump testing and changes to the testing Codes, and suggests ways to improve the quality of pump tests. The Code has several tests that apply to pumps. These tests are required to be done at a repeatable point of pump operation. Reference (baseline) values of test parameters are taken at a point of operation when the pump is new or has been refurbished and is operating acceptably. These parameter values are compared to values measured during subsequent tests to determine if the pump has "operational readiness." The pump operational parameters that must be measured or observed are pump speed, inlet and differential pressure, bearing vibration displacement, flow rate, bearing temperature, and lubricant level or pressure. The Code does not specify that testing must be done at a high or low flow rate, which is primarily a concern for centrifugal pumps, or if the testing should be done at high or low pressure, which may be a concern for positive displacement pumps.

The NRC has expressed concerns regarding the quality of testing of certain pumps. Most pumps are equipped with a low-flow or mini-flow path to prevent overheating in case they are run against a dead head in their main flow path. Many pumps are tested with flow through the mini-flow path. It is widely recognized (Yedidiah, 1977) that testing centrifugal pumps at low flow rates, such as the rates achievable on mini-flow lines, is of low quality. That test gives little assurance of operational readiness. Significant hydraulic degradation is manifested only slightly in differential pressure measurements. Additionally, the test may cause damage to the pump in some cases. There is a similar problem regarding test quality when testing positive displacement pumps at low pressure. Leakage past the seals, which indicates degradation, is less evident at low pressure. However, the Code allows tests of positive displacement pumps at low pressure and centrifugal pumps at low flow.

This discussion has focused primarily on the hydraulic performance (the pump's ability to pump fluid) aspects of pump tests. Another important consideration is mechanical performance. The Code requires measurement of pump bearing vibration in units of displacement (measured in inches) to assess a pump's mechanical condition. The OM Code-1990 provides a program for measuring pump bearing vibration in units of velocity (inches per second). That test is widely recognized by the industry to be a much higher quality test for assessing the mechanical condition of the pump. Many licensees recognize the improvement and are testing according to the OM Code program.

The OM Code represents a significant improvement to the pump testing requirements, particularly for vibration. A proposed revision to that Code, called the "Comprehensive Pump Test" represents another significant improvement to the requirements. That revision will be discussed in detail at another session of this conference. Among other things, if approved, that revision will require testing centrifugal pumps at high flow rates, and testing positive displacement pumps at high pressures. Therefore, to improve the quality of testing of most centrifugal pumps, the authors recommend testing them at higher flow rates that are at or near the pump design flow rate. Additionally, positive displacement pumps should be tested at or near design basis pressures.

CONCLUSIONS

The quality of component testing at NPPs needs improvement. It is largely driven by the applicable industry codes and standards. Those codes and standards are being revised and improved to help ensure that better test quality is achieved.

Operational readiness is gained in various degrees from testing and is a product of test quality. Test quality is affected by several factors, such as the parameters measured, component loading, instrument accuracy, and operator experience. A large part of test quality depends on the attitude of the persons performing the tests and reviewing the results.

Test frequencies are related to test quality. The frequencies that are dependant on plant conditions can be extended randomly.

The quality of tests performed on MOVs, check valves, and pumps can be very low even if the testing is performed according to the Code. The quality of these tests can be improved as follows: by testing MOVs under load and assessing stem thrust, by augmenting check valve open tests with a closure test, by testing centrifugal pumps at relatively high flow rates, and by testing positive displacement pumps at higher discharge pressures.

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Guidelines Document for Inservice Testing Programs

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INSERVICE TESTING PROGRAM REVIEWS

Prior to Generic Letter (GL) 89-04, "Guidance on Developing Acceptable Inservice Testing Programs," the U.S. Nuclear Regulatory Commission (NRC) reviewed inservice testing (IST) programs extensively. The reviewer attempted to verify that all applicable components were part of the program and that the testing established for the components met the Code requirements. This resulted in long review periods and contributed to a large backlog of programs without safety evaluations. Meetings with licensees were routine elements of the review, allowing for discussion on the content of the programs.

This level of review is not required by 10 Code of Federal Regulations (CFR) 50.55a, "Codes and Standards." NRC approval of relief requests is required by the regulation. Shortcomings in performing a total program content review became evident, including the following:

- The licensees, in some cases, relied on the NRC reviewers to identify areas of weakness in the inservice testing programs.
- Following discussions with licensees, specific comments were resolved, but other areas may not have received licensee attention as needed because they were not discussed
- Because of the specifics of each plant's safety analysis and design basis, without a thorough design basis review, NRC review cannot be responsible for ensuring that all pumps and valves are included in the program.

Plant personnel are responsible for identifying components within the scope of American Society of Mechanical Engineers (ASME) Section XI and the applicable test requirements for these components. NRC will question the content of the programs during inspection activities, and may identify concerns when performing reviews of relief requests. Enforcement action may be taken if components within the scope of ASME Section XI are not included in the program.

NRC reviewers are not responsible for justifying the impracticality of a Code-required test, but this was (is) often necessary because of the lack of details in relief requests. The licensees have the knowledge of other pertinent information, but often do not include it in the relief request because they do not realize that it would be useful to the NRC staff or contractor reviewer.

NRC resources are limited and must focus on review of relief requests to meet regulatory requirements and plant Technical Specifications. When the necessary information is not provided in a relief request, additional time and effort are required for the reviewer to investigate the issue in more detail, resulting not only in delays, but in additional charge to the licensees. Improvements in the licensees' preparation of relief requests could eliminate this additional effort.

The current reviews concentrate on relief requests required by the regulations. The results

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of the reviews are provided in an NRC Safety Evaluation (SE). Cold Shutdown Justifications written to comply with IWV-3412 or IWV-3512 are reviewed, but are not evaluated in the SE (Minutes of the Public Meetings on Generic Letter (GL) 89-04, Question 102). Currently, no specific attempt is made to verify that all applicable pumps and valves are included in the program or to verify all testing requirements. However, a concern is identified during the review of the program content, an anomaly may be included in the SE. Additionally, if relief requests are submitted as approved by GL 89-04 positions, a review will be performed to establish general agreement with the licensee. The SE will not generally include an evaluation, although it may identify concerns related to the relief request. Implementation of all such Cold Shutdown Justifications and Relief Requests is subject to NRC inspection, and if it is determined that compliance is not assured, enforcement action may be taken.

Similarly, the licensee is responsible for the basis upon which relief is granted. If the basis is determined to be incorrect during an inspection, enforcement action may be taken. If the licensee is not complying with the relief as granted, based on the information provided in the relief request, the NRC approval of the relief may be invalidated. This could result in a violation or a deviation, depending on the circumstances.

Nobody knows the plant better than the licensee. The expertise for an outstanding inservice testing (IST) program may not be in one individual (the IST Program Coordinator) but collectively in the design engineers, system engineers, operations personnel, and other technical staff. NRC reviewers cannot bring the same diversity of expertise into a program review. The elements should be integrated into the program before it is submitted to the NRC. The program should describe how the requirements are implemented, how components are selected for inclusion, how modifications are reviewed for impact on the IST program, how the licensee determines when relief is necessary, and how administrative aspects are handled such as test results and trending reports.

Many plants are performing design bases reviews of plant systems. During this process, components will be identified that are not in the inservice testing program, but should be, and conversely. When components are identified that fall within the scope of Section XI and have not been tested previously, an operability determination must be made in accordance with plant Technical Specifications. The requirements of 10 CFR 50.72, "Immediate Notification Requirements for Operating Nuclear Power Reactors," and 10 CFR 50.73," Licensee Event Report System," may also apply. If testing cannot be done in the near-term, a justification for the continued operability of the affected component may be required (GL 91-18, "Information to Licensees Regarding Two NRC Inspection Manual Sections on Resolution of Degraded and Nonconforming Conditions and on Operability"). NRC involvement may be necessary to resolve nonconforming issues. We are currently considering issuing a generic letter that provides guidance on actions specific to IST nonconformances. The proposed generic letter would be published in the Federal Register for public comment prior to issuance, according to new NRC guidelines.

Guidelines Document

To address topics that are frequently encountered in implementing and regulating IST, a guidelines document is being developed that will identify various positions relative to implementation of the requirements of Section XI and the guidance of GL 89-04. The purpose of the document is to provide information on common relief requests or interpretations being made by licensees, as observed in program submittals and NRC inspection reports. It will (a) provide guidance on what information should be included in relief requests, (b) describe what should be included in IST programs, and (c) address programmatic issues related to inservice testing. Examples of topics to be included in the guidelines document are listed in Table 1.

The guidelines document will be issued attached to a generic letter. The generic letter will provide the recommendations for using the guidelines document. The guidelines document will improve IST programs and preapproving a number of relief requests beyond those addressed in GL 89-04 and will provide a better understanding of the NRC's position on the topics included in the document. It will not impose additional requirements. The appendix provides some discussions from the proposed guidelines documents.

Table 1. Guidelines document topics.

Inservice Testing Topic	Description
Inservice Test Frequencies	• Frequencies and extensions • Delay of startup from cold shutdown as delineated in O&M, Part 10 • Deinerting containment of BWRs to allow cold shutdown testing of pumps and valves • Stopping reactor cooling pumps for cold shutdown valve testing - RCP seal water supply valves
Startup with Component or System Inoperative	• Case where a cold shutdown valve exercise or stroke time test fails and starts up with the affected component or system "out-of-service" allowed by Technical Specifications (TS)
Starting Point for Time Period in TS ACTION Statements	• GL 89-04, Position 8 • Time allowance for review by cognizant individual responsible for declaring a component inoperable, such as within the shift when the testing is performed
Qualifications of Safety and Relief Valve Testing Supervisor	• Applies to PTC-25.3, 1976 • OM-1 does not specify test supervisor qualifications • Acceptable alternatives
Emergency Diesel Generator Systems	• Skid-mounted components • IST relation to revision of Regulatory Guide on testing diesels • Code versus non-Code components
Trending	• Identifying degrading conditions • Comparison of test results to previous test data • Prediction of impending failure of a component

Table 1. (continued).

Inservice Testing Topic	Description
Systems Out-of-Service Which Cannot Be Tested Prior to Startup	• Components which require steam to test such as turbine- driven pumps, safety/relief valve setpoint testing in-place • Components which require a differential pressure to test, such as verifying a check valve is closed • Testing of main steam isolation valves in Mode 4
Positive Displacement Pumps	• Inlet pressure measurement when no instrumentation is installed • Evaluation of discharge pressure in lieu of differential pressure as in O&M, Part 6
Pump Flow and Differential Pressure Instrumentation	• Instrument accuracies • Instrument ranges, particularly where test inlet pressure is lower than pressures expected during normal operations, or accident conditions • Replacement of instrumentation to meet later editions of Code • Installation of instruments to meet Code requirements
Use of Non-Permanent Instruments	• Ultrasonic flow rate measurement repeatability concerns • Range requirements for vibration instruments • General guidelines
Calculated Parameter Values	• Accuracy of calculational methods for determining values of differential pressure, pump inlet pressure, flow rate
Observation of Pump Lubrication Level or Pressure	• Acceptable actions
Monitoring Pump Vibration Velocity in Lieu of Displacement	• Acceptable per O&M, Part 6 if all requirements related to vibration are met
Pump Testing Using Mini-Flow Return Line With or Without Flow Measuring Devices, GL 89-04, Position 9	• Acceptable alternatives • Extension of complete testing to cold shutdown or refueling outage interval
Pump Bearing Temperature Measurements	• Eliminated in O&M, Part 6
Hydraulic Acceptance Criteria	• Overdesigned pumps - what is the required action limit? • Expanded ranges
Vibration Reference Values for Smoothly Running Pumps	• Limits for acceptance criteria
Vertical Line Shaft Pumps	• Measurements at locations other than as specified in Code • Appropriate acceptance criteria for "noisy" pumps

Table 1. (continued).

Inservice Testing Topic	Description
Use of Variable Reference Values for Flow Rate and Differential Pressure for Pump Testing	• Validation of Manufacturer's pump curves • New curves developed with pumps operating acceptable • Points taken for developing curve using instruments at least as accurate as Code requires for single reference value • Curves based on an adequate number of points, with a minimum of five • Points beyond flat portion (low flow rates) of curve • Acceptance criteria does not conflict with TS or Safety Analysis criteria • Vibration "regions" may need to be identified if vibration varies over the range of the curve
Frequencies for Part-Stroke Exercising Check Valves	• Importance of part-stroke exercising quarterly, at cold shutdown, refueling outage, and following reassembly
Increased Frequency of Testing for Valve Tested During Cold Shutdown	• Requirements of IWV-3417(a) apply for increased frequency • Monthly (increased frequency) testing cannot be delayed to the next cold shutdown • Corrective action should be taken prior to startup to preclude plant shutdown to do increased frequency testing
Valve Position Indication	• Verification of both open and close indication • Category B passive, including manual valves with position indication, require verification
Check Valve Issues	• Dual function PIV/CIV must meet requirements for both functions • Disassembly and inspection activities must be performed in accordance with GL 89-04, Attachment 1, Position 1 • Valves tested in series as a pair are to both be repaired if test results are unacceptable • When plant operations demonstrate functionally adequate seat tightness • Stop check valve guidance • Vacuum breaker check valves

Table 1. (continued).

Inservice Testing Topic	Description
Containment Isolation Valves	• Testing as a group can only be acceptable when individual valve testing can not be performed • PIV function, if applicable, must be demonstrated • Closure verification by leakage testing at refueling outage frequency • Nonintrusive methods are recommended as preferable to disassembly and inspection
Power Operated Valves	• Valves which cannot be stroke-time tested must be monitored for degradation by some means • ADS valves are required to be stroke-time tested or an acceptable alternative must be provided to monitor for degradation • Use of reference values may be advantageous • Solenoid valves are required to be stroke-time tested or an acceptable method of monitoring for degradation must be provided • Diagnostics can provided a more accurate measurement of valve stroke time
Safety and Relief Valves	• 1986 Edition of ASME Section XI references OM-1-1981 • Scope of 1986 Edition of ASME Section XI, IWV-1100 includes ". . .or in providing overpressure protection." • Frequency of testing Class 2 safety valves is extended to once per 10 years • Testing PORV/ERV, GL 90-06
Control/Regulating Valves	• Valves that have a fail-safe position, even if the normal function is as a control or regulating valve, are required to be tested for the fail-safe function and stroke-time tested to monitor for degradation

Appendix

EXAMPLE OF DISCUSSIONS FROM THE GUIDELINES DOCUMENT

Components that Cannot be Tested Prior to Startup

The Code requires testing of components that have had maintenance performed or are in systems out of service before the component or system is returned to service (Section XI, Paragraphs IWV-3200 and -3416). Frequently the applicable system or component is required by plant Technical Specifications to be operable prior to plant startup. In these cases the testing must be performed during cold shutdown. Many nuclear facilities have pumps and valves that cannot be practically tested during the cold shutdown mode because necessary conditions (e.g., high temperature or pressure) cannot be established. In these cases, licensees generally request relief to delay testing until the requisite plant conditions can be established.

It is impractical to test some pumps and valves when the plant is shutdown. If shutdown, it is necessary to return to power before these components can be tested. For example, a valve in a system supplied by a turbine driven pump cannot be exercised open with flow until the plant is operating at power and steam is available to drive the turbine. The inability to test a component during shutdown is of particular concern in cases where maintenance has been performed which requires that the component receive a post maintenance test prior to it being returned to the operable condition and Technical Specifications require the component or its system to be operable before plant startup. Plant startup cannot be performed until the component is operable, the component cannot be declared operable until it has been tested, and the testing cannot occur until the plant is operating at power. A similar situation can occur for components that cannot be tested during shutdown located in a system that is declared inoperable during an outage. These components must be tested within 30 days prior to returning the system to the operable status, however, the system may be required to be operable before plant startup can begin.

Code Requirements. Section XI, Paragraph IWV-3200 states: "When a valve or its control system has been replaced or repaired or undergone maintenance that could affect its performance, and prior to the time it is returned to service, it shall be tested. . . ."

Section XI, Paragraph IWV-3416 states: "For a valve in a system declared inoperable or not required to be operable, the exercising test schedule need not be followed. Within 30 days prior to the return of the system to operable status, the valves shall be exercised and the schedule resumed in accordance with requirements of this Article."

Staff Position. If all of the following conditions are met, the Code required testing of the affected components may be delayed until after startup when system conditions can be established to accommodate the testing.

1. The component is required to be tested before plant startup.

2. The licensee adequately demonstrates the impracticality of testing the component during cold shutdown.

3. The licensee implements any practical measures that could minimize the consequences of failure of the affected component.

When component testing is delayed until after startup per this position, the plant conditions required for testing should be established as soon as practical following startup. The components should be tested as soon as the prerequisite conditions are established.

Basis for Position. Testing of certain components is impractical during the cold shutdown mode. Therefore, a plant in a shutdown mode would have to be heated up and possibly returned to power operation to permit testing of these components. If the plant is shutdown and the Code requires one of these components to be tested before it can be returned to the operable status (to comply with Section XI, Paragraph IWV-3200 or -3416), the licensee would be stymied if Technical Specifications required the component or its system to be operable prior to plant startup. Unless relief were obtained from the Section XI requirement, this situation would result in a Technical Specification violation. Since compliance with the Code would necessitate violating a Technical Specification in this situation, compliance is considered to be impractical and burdensome.

Because the affected components have either been subjected to maintenance or repairs that could affect their performance or had their normal test frequency suspended because the system was declared inoperable, their operational readiness would be questionable if they are not tested prior to plant startup. This is especially true of the components that have undergone maintenance or repairs because they may have been substantially disturbed or modified which increases the chance of error by the individual(s) performing the maintenance. Because these components have a reduced operational readiness, the likelihood of failure is increased until they are tested. Therefore, testing should be performed as soon as reasonably possible following plant startup.

Section XI, Paragraph IWV-3416, appears to require testing of valves declared inoperable for any length of time even if the valve test schedule is not extended. The ASME OM Code-1990, Paragraph ISTC 4.5.5, clarifies this point by requiring valves in systems out of service to be tested to return them to service only "if the test schedule is not followed." Both Section XI and the OM Code require valve testing prior to return to service if any repairs or maintenance are performed that could affect their performance. If the test schedule is maintained for a valve in a system out of service, it need not be tested to return it to service unless it has undergone repairs or maintenance that could affect its performance.

The number of components that cannot be practically tested during a shutdown mode is relatively small. Most of these components either rely on high pressure steam or coolant to provide their motive force [i.e., the turbine driven auxiliary feedwater pumps in pressurized water reactors (PWRs) and the automatic depressurization system (ADS) valves in boiling water reactors (BWRs)] or are exercised by flow produced by high pressure steam (i.e., the pump discharge check valves for the turbine driven auxiliary feedwater pumps). Of this number, not all are required to be operable prior to plant startup. This position does not apply to components not required to be operable prior to startup because they can remain inoperable until after startup and then be tested and returned to operable status. Since the number of affected valves is small, conditions that require testing to be delayed because of this position, should occur infrequently. Therefore, if the provisions of this position are followed, it should provide a reasonable level of quality and safety.

Recommendations. Where the licensee demonstrates that components meet the criteria of this position, relief should be granted to allow testing to be delayed until after startup. Any testing delayed by this provision should be performed as soon as practicable after startup.

Use of Portable Instrumentation for IST of Pumps

Many pumps in IST programs do not have installed instruments for required test measurements or the installed instruments do not meet the Code accuracy or range requirements. This lack of suitable installed instrumentation can be the result of several different factors. Many plants were designed and/or built prior to the implementation of the ASME B&PV Code, Section XI, for commercial nuclear power plants. Some systems lack instrumentation because they were not understood to be within the scope of Section XI. In some cases the omission was simply an oversight. Some pump designs and configurations are not conducive to the installation of instrumentation for required measurements (i.e., inlet pressure measurements for submerged vertical line shaft pumps).

The lack of suitable installed instrumentation has necessitated relief from taking the IST measurements or resulted in the use of alternate testing methods or portable instruments. In some cases plants are calculating the Code required parameters based on measurement of other related parameters (i.e., calculating inlet pressure based on the water level above the pump suction). Use of portable instrumentation and calculations based on other measured parameters are not specifically permitted or addressed by the Code, therefore, it is necessary to request and obtain relief prior to implementing these methods. The NRC has determined that these alternatives can provide information that is adequate for evaluating pump condition and detecting degradation and, therefore, can be found to be acceptable. However, to produce acceptable results, these alternatives must be properly implemented. The general lack of guidelines for these methods has resulted in the need to specify some general criteria for their acceptable use. The staff position for the use of portable instruments is presented below. When pressure instruments other than the installed instruments are used for testing, it normally involves connecting test gages to existing gage lines, therefore, they are not portable instruments and are not specifically addressed by this position. This position primarily deals with portable flow rate instruments used for IST. The use of calculations from other parameters will be discussed in the guidelines document, but is not addressed in this paper.

Code Requirements. It appears that Section XI requires the use of installed direct reading instruments for IST flow rate measurements. IWP-4600 states: "Flow rate shall be measured using a rate or quantity meter installed in the pump test circuit." Paragraphs IWP-4110 and -4120 establish requirements for the accuracy and full-scale range of these instruments. Relief is normally requested from one or more of these Code requirements when portable flow rate instruments are used for IST.

Staff Position. In situations where no flow rate instruments are installed in the pump test circuit or where the installed instruments do not meet the Code accuracy or range requirements, portable instruments may be used for IST of pumps provided they consistently yield measurements at least as accurate as would be provided by installed instruments that meet the Code accuracy requirements.

In situations where there are installed instruments that meet the Code requirements, portable instruments should be used only if the licensee demonstrates that using the portable instruments would not result in a decrease in reading accuracy or repeatability. Where practical, the same instrument should be used for IST to permit meaningful trending of test results and allow detection of degradation.

When using portable instruments for IST, the licensee should implement procedures and policies that provide the greatest accuracy and repeatability reasonably obtainable from the instruments.

Basis for Position. The purpose of the instrument requirements of Section XI, IWP-4100 is to ensure that pump test measurements are sufficiently accurate and repeatable to permit evaluation of pump condition and detection of degradation. Although it is not defined in the Code, repeatability is extremely important when monitoring for component degradation. Test measurements that have poor repeatability (i.e., the readings vary randomly from test to test when there are no variations in actual flow rates) inhibit the ability of testing to detect degradation. A high level of repeatability is necessary detect a trend that is indicative of pump deteriorization and predict possible failure prior to entering the required action range. Portable instruments may provide indication that is less repeatable than measurements made with installed instruments, which may reduce the ability to monitor pump condition and detect degradation.

Installed flow elements (e.g., flow orifices and venturis) are welded or securely fastened in the system. The response of these elements should not change from test to test because the mechanisms that can affect the differential pressure across them (e.g., erosion, corrosion, and fouling) should occur at a relatively slow rate. Therefore, installed flow elements should have a high level of repeatability. The remainder of the instrument system (i.e., transmitter, electronics, meter) is subject to drift and systematic errors, however, because it is a fixed system made up of installed components, the occurrence of random errors should be minimized. Therefore, if an installed flow instrument is routinely calibrated, it should provide reasonably repeatable flow rate indications.

The sensing probes of portable flow rate instruments are attached to the system piping each time the instruments are used for IST measurements. There are many variables associated with probe installation that can affect the flow rate measurement. The location, alignment, mounting pressure, and application of conducting gel, can all play a factor in the repeatability of the indication. Most portable flow instruments do not directly measure flow rate, they measure the average velocity of the fluid passing through a pipe. The fluid velocity must be converted to a flow rate by calculations based on the cross-sectional area of the pipe and the flow characteristics. Slight differences in pipe area or characteristics where the probes are attached could cause variations in flow rate measurements. Additionally, due to the portable nature of the equipment, it is more likely that the components used for IST from one test to another will be different. Because each particular component may introduce its unique systematic error, using different instrument components from test to test could produce variations in readings unrelated to pump condition.

For the above reasons, installed instruments are generally inherently more repeatable than portable instruments. Therefore, all other things being equal, installed instruments are preferred over portable instruments for IST measurements. If there is an installed flow rate instrument that meets the Code requirements that can practically be used for IST measurements, a portable instrument should not be used unless it can be demonstrated that the portable instrument provides at least equivalent reading accuracy and repeatability. Test quality (the ability of the test to determine pump condition and detect degradation) should not be compromised or reduced solely for convenience. In fact, every reasonable effort should be made to increase test quality and improve the ability to detect degradation and anticipate pump failure.

When portable instruments are to be used for IST, relief is frequently requested from the Code instrument accuracy and full-scale range requirements. Most portable flow rate instruments have digital readouts and can indicate flow rate with the same accuracy over an extremely large range. The full-scale range requirements of the Code are intended to help define instrument accuracy and to insure adequate gage or meter readability. Since the accuracy and readability of digital instruments are independent of the instrument range over its usable span, the full-scale range requirements are meaningless for these instruments. The usable range of portable flow instruments is normally sufficiently large to bound all safety related pump applications.

The accuracy of portable instruments is generally based on the measured reading and not on the instrument range. The accuracies claimed for portable flow instruments vary widely from site to site. Some utilities have been capable of achieving high levels of accuracy from portable flow instruments, however, this involves a great deal of rigor when calibrating the instruments. Others claim accuracies that exceed the Code requirements and have had to expand the Code acceptance criteria for the applicable pumps. Where instrument accuracy and/or repeatability are so poor that the Code allowable ranges cannot be met, it is highly unlikely that the test permits detection of pump degradation. This situation should be avoided where practical by improving the test method, the instrument accuracy, and/or the measurement repeatability. Where there are no installed flow rate instruments or where the installed instruments do not meet the Code requirements, portable instruments may be used, however, the instruments should provide measurements at least as accurate as would be obtained from installed instruments that meet the Code requirements.

The accuracy and repeatability of portable instruments can be affected by many factors. Identifying all of the possible factors for the various instrument types and models would be a large task and is beyond the scope of this document. The instrument manufacturers should provide technical manuals that contain instructions and guidelines for the use of the instruments. The licensee should follow these instructions and any other reasonable practices that would produce measurements that are more accurate and repeatable.

Recommendations. Neither the B&PV Code, Section XI, nor the ASME OM Code-1990 provides adequate direction on the use of portable flow rate instruments. The O&M Working Group on Pumps and Valves should prepare changes to the ASME OM Code-1990, Subsection ISTB, that specifically addresses the use of portable flow rate instruments for IST. The changes should establish requirements that insure that portable instruments provide measurements that are adequate to evaluate pump condition and detect hydraulic degradation.

The following guidelines should be considered where portable instruments are utilized:

1. Follow all appropriate instructions and guidelines provided by the instrument manufacturer.

2. Provide detailed procedures and a means to ensure consistent attachment of the instrument probe from test to test.

3. Use the same instrument components during each test of a particular pump, if practical.

Toeing the Line—Meeting Minimum Compliance Regulations

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ABSTRACT

This paper discusses the history of pump and valve testing regulation in the United States and identifies the result of decades of testing under those rules. While the current regulations for pump and valve testing under O&M rules are an improvement over the previous Section XI requirements, there is an essential need for a regulatory distinction between licensees that meet minimum compliance with current rules and licensees that exceed the requirements of existing regulation. In addition, case histories of problems in pump and valve testing under the current set of rules will be presented.

Reviews of ASME Section XI Pump and Valve Relief Requests Post Generic Letter 89-04

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ABSTRACT

This paper will present a discussion of American Society of Mechanical Engineers (ASME) Section XI Pump and Valve Inservice Testing relief request reviews by the U.S. Nuclear Regulatory Commission (NRC) and their contractors. Topics that will be discussed include the scope of NRC reviews in Technical Evaluation Reports and Safety Evaluations, including the basis for granting relief requests, the status of relief requests in Inservice Testing (IST) Program updates, and the Generic Letter 89-04 approval process; and the level of technical detail required in submitted programs. This presentation is based on the experiences of Brookhaven National Laboratory in reviewing IST Programs for the Mechanical Engineering Branch of the NRC.

INTRODUCTION

Experience has shown that there are a number of common problems in the preparation of relief requests that make them difficult to review and grant their approval. Generally, these fall in the categories of (a) providing insufficient justification for not performing the Code required tests, (b) not providing adequate information on the alternate tests, (c) numerous "typographical errors," (d) not providing the requests' status, and (e) a lack of understanding of the safety significance of the components.

In order to assist the industry in understanding the approval process, this paper will address the following: the basis for granting relief, relief request status, approval via Generic Letter 89-04, the recommended contents and format of relief requests, the use of ASME Code Cases and later editions of the Code, acceptable alternatives to the Code, and cold shutdown justifications. The focus of this paper will be what information should be included in the relief requests from the perspective of a reviewer for the NRC. A discussion of the various means of obtaining relief and the elements necessary to be included in a relief request or cold shutdown justification is provided so that the relief requests and cold shutdown justifications can be reviewed promptly and efficiently with a minimum number of iterations.

BACKGROUND

Utilities are required by 10 CFR 50.55a to perform inservice testing of ASME Class 1, 2, and 3 pumps and valves in accordance with ASME Section XI. Alternatives to the requirements of Section XI may be used when authorized by the NRC staff. The regulations allow alternatives to be used when the "proposed alternatives would provide an acceptable level of quality and safety" [10 CFR 50.55a(a)(3)(i)], when "compliance would result in hardship or unusual difficulty without a compensating increase in the level of quality and safety" [10 CFR 50.55a(a)(3)(ii)], or

a. Work performed under the auspices of the U.S. Nuclear Regulatory Commission.

when "the Code requirements are impracticable. The Commission may grant relief and may impose alternate requirements as it determines is authorized by law...giving due consideration to the burden upon the licensee if the requirements were imposed on the facility" [10 CFR 50.55a(g)(6)(i)]. The NRC staff has also provided generic relief in Generic Letter 89-04 "Guidance on Developing Acceptable Inservice Testing Programs." The relief requests submitted to the staff are reviewed and approved pursuant to the regulations and the Generic Letter. Licensees should ensure that the request can be approved by either the regulations or the Generic Letter. Relief may be granted unconditionally, for an interim period, or with provisions, or it may be denied.

Since the NRC is no longer conducting detailed reviews of the IST Programs as part of the Safety Evaluations (SEs), and there are generally no meetings with the licensee to discuss the relief requests, it is imperative that the relief requests be self supporting and complete in order to expedite their review and approval. In accordance with the Standard Technical Specifications 4.0.5, testing shall be performed in accordance with Section XI, unless specific relief has been granted. Therefore, relief must be granted prior to implementation.

STATUS OF RELIEF REQUESTS

Unless the licensee has stated in the IST Program or relief request submittal that the request is "grandfathered" by the Generic Letter (i.e., that the request was submitted by a licensee not listed on Table 1 or 2 of the Generic Letter before April 3, 1989) or is approved by a position in the Generic Letter, the NRC staff has performed a detailed evaluation of the request. For future submittals, the licensee should provide a status of all the requests, including the revision and NRC status. Additionally, the licensee should identify each relief request uniquely and maintain the same numbering system, even if some of the requests have been deleted. All requests and documentation of the alternate positions allowed by the Generic Letter are reviewed to ensure that the proposed testing is acceptable; however, detailed evaluations are not included in the Technical Evaluation Report (TER)/SE. Additionally, all relief requests submitted with the next 10-year interval update are reviewed and evaluated for consistency with the current NRC regulatory positions. These requests are no longer considered grandfathered.

Licensees often include components in the IST Program that are not within the scope of ASME Section XI, paragraphs IWP-1100 and IWV-1100, and the regulations. The Section XI IST Program provides an acceptable program for monitoring component operational readiness for components subject to periodic testing in accordance with 10 CFR 50, Appendices A and B. However, relief is not required to be approved by the staff in accordance with the regulations. The licensee should, nevertheless, ensure that appropriate documentation exists for these components. The NRC staff will review and approve all the relief requests submitted, unless the licensee has stated in the relief request that the components are outside the scope of Section XI. Licensee's IST Programs will be inspected in accordance with NRC Inspection Manual, Temporary Instruction 2515/114, *Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs*, to ensure all ASME Class 1, 2, and 3 pumps and valves with safety related functions are in the program, and that relief requests have been submitted when the requirements of the Code or the Generic Letter cannot be met.

RELIEF GRANTED BY GENERIC LETTER 89-04

The most expeditious way to conduct alternate testing is to implement one of the positions contained in Generic Letter 89-04. The NRC staff has determined that generic relief is granted to follow the alternate testing delineated in Position 1 (Full-stroke Check Valve Testing), Position 2 (Alternates to Full-Flow Testing), Position 6 (Rapid Acting Valves), Position 7 (BWR Control Rod Scram Valves), Position 9 (Minimum-Flow Pump Testing), and Position 10

(CIV Testing) pursuant to 10 CFR 50.55a (g)(6)(i).

The licensee should ensure that all the criteria contained in the Generic Letter's positions are met and are adequately documented in the IST Program. Specific relief requests are then not required. The relief request format, however, provides a suitable, and the preferred, method for documentation. The staff will review the documentation and any deviations from the positions will be identified as an anomaly in the TER/SE. Relief is only preapproved if the licensee complies with all the recommendations. Reviews of recent submittals have identified a number of problem areas related to the Generic Letter.

Position 1 allows check valves to be full-stroke exercised with flow rates less than the maximum required accident condition flow rates, provided that the licensee documents a number of items. Many licensees do not address and document each item in the relief request, including the method and results of the qualification program. The qualification program should ensure that the alternate test method is quantifiable and repeatable, and the documentation should be available for review by the NRC inspectors if it is not included in the request.

Position 2 contains numerous criteria for an acceptable disassembly and inspection program to be used as a means of determining that a valve's disk will full-stroke exercise open, or of verifying closure capability. The NRC staff's position is that a disassembly and inspection program may be instituted as a means of demonstrating the full open and closure capability of the valve, provided there are no other means of verification possible. Although such a program is acceptable for verifying valve closure, it is considered by the staff as a maintenance procedure with inherent risks, and only limited information on the valve's ability to seat promptly upon flow reversal or cessation is gained. It is generally recommended that licensees investigate the use of other testing techniques, such as nonintrusive methods (e.g., acoustics or radiography) and implement those that are demonstrated effective.

When proposing a disassembly and inspection program, the documentation should clearly identify the sample grouping(s) of valves, as determined in accordance with the Generic Letter, and provide information on why other means are not possible or practical. The licensee should also perform partial valve stroking or provide a justification for not performing it quarterly or during cold shutdowns, or after reassembly. A valve should be inspected each refueling outage and each valve in the group should be inspected at least once every 6 years. If the licensee is proposing an alternate schedule to this, extreme hardship should be documented and the following information should be developed: (a) document each valve in the grouping's condition and capability to be full-stroked, (b) review industry experience, and (c) review the installation of each valve for problematic locations. It is up to the licensee to establish a hardship case.

The licensee should explicitly state that the Generic Letter is being utilized. Often times, licensees propose a disassembly and inspection program without directly referring to Position 2.

Position 10 applies only to containment isolation valves (CIVs). Generic relief has not been approved for pressure isolation valves (PIVs), including CIVs that are also PIVs. The licensee should ensure that relief from paragraph IWV-3427(b) is applied only to CIVs 6 inches and larger.

CONTENTS OF RELIEF REQUESTS

NUREG-0800, Standard Review Plan Section 3.9.6, *Inservice Testing of Pumps and Valves*, details the information required for NRC review of relief requests. This guidance had previously been issued to operating plants in 1976. ASME Section XI, 1987 Addenda and later editions and addenda, also provides guidance on preparing justification of substitute ex. inations or tests (i.e., relief requests) in a non-mandatory appendix, Appendix F. The following basic format should be followed when preparing relief requests:

1. *Component Identification:* Identify the valve or pump name and component identification number, unit (if a multiple unit site), ASME Code Class, safety function, valve category, and reference drawings, including drawing coordinates. The drawings should be submitted with the relief request or IST Program submittal and should include system instrumentation and test connections. If the drawings are larger than 8-$^1/_2$ X 11 inches, the licensee should submit one copy directly to the NRC Project Manager. All relief requests should be referenced in the pump and valve program tables. Licensees should ensure that there are no typographical errors in the relief requests. Wrong valve numbers, valve categories, or references, to name only a few of the errors found, affect the length of the review time, and the lack of attention and quality may be noted in the SALP Report.

2. *Section XI Code Requirement:* Specifically identify the code paragraph and requirement from which relief is requested. For example, if relief from check valve exercising in accordance with Section XI, paragraphs IWV-3521 and 3522, is requested, the request should state if relief is required for exercising the valve open or closed or both. Do not identify all the test requirements for that component, only those from which relief is requested and ensure that the alternate test and basis address each of these test requirements. A discussion of the tests performed may, however, be appropriate for the Alternate Test or Basis sections.

3. *Alternate Tests:* State what alternatives will be performed, the acceptance criteria that will be applied, the test frequency, and the schedule for implementation. Most recently submitted requests do not provide an implementation schedule. Alternate testing should provide a means of determining each component's condition and measuring degradation. The condition of individual components should be assured when there is redundant equipment (e.g., two check valves in series). Alternate tests should not be simply a reference to the Technical Specifications or a procedure number. A complete, detailed description of the test should be provided. If proposing to monitor system parameters, the parameters must be capable of detecting degradation and quantitative acceptance criteria should be provided.

4. *Basis:* Document either why the Section XI test requirements are impractical or would result in hardship or unusual burden, or how the proposed alternatives provide an acceptable level of quality and safety (i.e., provide equivalent protection as provided by the Code). When documenting the impracticality, burden, or unusual hardship, the licensee should provide a detailed description of the problems associated with performing the test. Factors such as personnel hazards, radiation exposure (including the radiation levels, estimated man-rems to perform the testing, possible means to reduce the exposure), high costs, impact on plant startup, operation and safety, manpower required, length of time the component is out of service to perform the testing, and any potential damage to equipment may be discussed. Each relief request should specifically discuss the impracticality or burden of performing tests quarterly and at cold shutdowns, when proposing testing at refueling outages. Entering a Technical Specification limiting condition of operation (LCO) by itself, is not sufficient reason not to perform the Code required tests. If the length of time required to perform the testing is less than the allowable outage time (AOT) of the Technical Specification action statement, the testing should be performed. If the testing removes a train or system from service and places the plant in a condition such that the design basis function cannot be met, the testing may be postponed to cold shutdowns. The licensee should directly

quote the Technical Specifications and any other documents, or attach copies with the submittal for the reviewer to use. Additionally, as documented in the Generic Letter and the Minutes to the Letter, the addition of instrumentation is not generally considered by the staff to be impractical.

The Basis should also include a discussion of the function and safety importance of the component. The facility safety analysis report (FSAR), emergency operating procedures, and probabilistic risk assessments (PRAs) may be consulted. Additionally, NRC generic communications, such as bulletins and information notices, may assist in determining safety functions. Some recently submitted NRC information notices that should be considered in developing the IST Program include 91-56, *Potential Radioactive Leakage to Tank Vented to Atmosphere*, 90-78, *Previously Unidentified Release Path from BWR CRD Hydraulic Units, and* 89-32, *Supplement 1, Surveillance Testing of LTOPS*.

A number of relief requests submitted employ the plant's PRA to justify not performing testing in accordance with the Code. Relief can be granted in accordance with the regulations if the Code requirements are impractical, burdensome, or if the proposed alternates provide an acceptable level of quality and safety. As discussed above, the staff considers alternates that provide an acceptable level of quality and safety to be those that provide an equivalent level of protection, as provided in the Code. Licensees should, therefore, provide a discussion of the burden or impracticality in addition to the risk impact of not performing the tests (e.g., a cost/benefit or value/impact analysis).

There is no standardized method of performing PRAs, and they are not part of the plant's design basis. Although Generic Letter 88-20 required utilities to complete an Individual Plant Evaluation (IPE) or PRA, the NRC currently does not perform a detailed review or approve them. Many PRAs are developed based on generic industry component and system data as opposed to plant specific data. The level of detail, assumptions, system interactions, human-reliability analysis, and treatment of passive components, for only a few examples, varies from plant to plant. Additionally, systems' and components' importance to safety and risk varies, if based on plant core melt frequency (such as the results of a Level 1 PRA), or on containment failure/fission product release (Level 2 PRA), or public health risk and consequences (Level 3 PRA). There are substantial uncertainties contained in the absolute core melt or containment failure or fatality probabilities. PRAs are useful tools to determine the relative risk significance. If tests are proposed to be deleted based on the low risk significance, then the highly risk significant components should be reviewed to ensure that testing and maintenance activities are adequate. This concept is discussed in the new "Maintenance Rule," 10 CFR 50.65. Assessments of maintenance (which includes testing in this context) effectiveness should be based on actual component and system reliability/availability and failure histories. There should be a feedback mechanism to revise maintenance (i.e., testing) techniques and frequencies based on the plant and industry operating data. Additionally, PRA configuration control is necessary. It may be acceptable to delete testing of Valve A or Valve B; however, the increase in risk may not be acceptable for deleting both Valves A and B. The PRAs must also be kept current to reflect plant operations, testing, maintenance, and design. PRA techniques, such as failure modes and effects analysis (FMEAs), are useful tools to determine efficient and effective test methods and frequencies.

There is a precedent of using PRA to revise Technical Specification allowable outage times, and the NRC staff has evaluated the risk-based relief requests on a case-by-case basis. The industry should, however, consider developing risk-based testing guidelines. An ASME Research Task Group has prepared general risk-based inspection guidelines (ASME Document CRTD-Vol.20-1) and is working on a supplemental guideline specific to nuclear plants. The Section XI Code Committee now has a task group on

ISI optimization and will be evaluating changes to the Code as a result of the Research Task Group on Risk-Based Inspection Guidelines' recommendations.

USE OF CODE CASES AND LATER EDITIONS OF THE CODE IN RELIEF REQUESTS

Many utilities have submitted relief requests to utilize the ASME/ANSI-1988 Operation and Maintenance Standards, OM Part 6. OM Part 6 has been approved for use by the staff via Regulatory Guide 1.147. All ASME Code cases related to ISI and IST that are acceptable to the NRC, unconditionally and with provisions, are included in this Regulatory Guide. These Code cases may be used without prior NRC approval provided that they are used in their entirety and are documented in the IST Program. If only portions of the Code cases are proposed to be used, a specific relief request is required [for instance, if only the pump vibration requirements of OM-1988, Part 6 are to be used (Code Case N-465) or only the test supervisor requirements of OM-1-1981 are used (Code Case N-415)]. Code Cases other than those described in Regulatory Guide 1.147 may be used, provided a relief request is submitted and approved (see Footnote 6 of 10 CFR 50.55a).

Additionally, utilities may use later editions and addenda of Section XI which are incorporated by reference in 10 CFR 50.55a, subject to NRC approval (i.e., a relief request is required). If portions of these editions and addenda are used, all the related requirements must also be used. For example, in the case of utilizing the OM Part 6 vibration velocity requirements, all requirements related to vibration, including measurement location, acceptance/alert/required action ranges, and test methods, must be used as well as the pump hydraulic requirements for vertical line shaft pumps and positive displacement pumps. The hydraulic parameters for these pumps were made more stringent to compensate for the less stringent vibration alert and required action ranges.

SPECIFIC RELIEF REQUESTS

A number of testing issues are generally acceptable to the NRC. However, specific relief is required and is often not requested. Examples include the following:

- Calculating pump inlet pressure from tank/intake structure levels in lieu of measuring pressure directly, as required by IWP-3100. In the absence of installed instrumentation, this may be acceptable provided the licensee properly proceduralizes the calculation and the calculated pressure accuracy meets the requirements of the Code.

- Testing safety valve and relief valve setpoints in accordance with ASME PTC 25.3-1976 (as required by IWV-3512) requires supervisors to be degreed engineers with at least 2 years experience in fluid-flow measurement. Qualifying personnel in accordance with the owner's quality assurance program may be acceptable. Additionally, licensees are able to implement OM Part 1, which does not include this requirement, in its entirely without relief. ASME Code Case N-415 allows the use of ANSI/ASME OM-1-1981, and is included in Regulatory Guide 1.147.

- Using reference pump curves in lieu of multiple reference values may be acceptable for pumps that have variable system resistance, e.g., service water or component cooling water pumps. The licensee should develop curves or validate manufacturer's curves when the pumps are known to be operating acceptably. The curves should be based on an adequate number of points. The curves should be revalidated after any maintenance or repair that might affect the reference curve. A method of assigning alert and required action ranges must be developed and should not conflict with the Technical Specification or FSAR operability criteria. Additionally, if vibration levels vary significantly over the range of pump conditions, a method for assigning vibration acceptance criteria should be developed.

- Leak rate testing groups of valves (i.e., valves in parallel) may be acceptable when individual leak testing is impractical because of a lack of installed test connections. The maximum assigned group leak rate should be based on the smallest valve in the group.

- Section XI specifies testing intervals without any extensions. However, the standard Technical Specifications allow extensions of the surveillance intervals (weekly, monthly, quarterly, semiannually, every 9 months, and yearly), not to exceed 25% of the specified surveillance intervals. The staff's position is that relief may not be granted to apply the 25% extension to safety and relief valve testing frequencies (i.e., once every 5 years).

COLD SHUTDOWN AND REFUELING OUTAGE JUSTIFICATIONS

Section XI paragraphs IWV-3412 and 3522 allow utilities to delay quarterly valve testing to cold shutdowns when testing during operations is impractical. OMa-1988, Part 10, also allows valve testing to be deferred to refueling outages when exercising is impractical during operation or cold shutdowns. The staff reviews the cold shutdown and refueling outage justifications for acceptability. These justifications should follow the same format as discussed above for relief requests and should be detailed enough so that it is evident that testing during power operation or cold shutdowns is impractical. Testing inconvenience is not sufficient justification.

OMa-1988, Part 10, provides an acceptable alternative to testing all cold shutdown valves every cold shutdown, regardless of length. The licensee should document the use of OMa-1988, Part 10, paragraphs 4.2.1.2(f) and (g) and 4.3.2.2(f) and (g) in the IST program. If valves cannot be tested during any cold shutdown and can only be tested during certain cold shutdowns, for example only when the reactor coolant pumps are not operating, specific relief is required.

Section XI, paragraph IWV-3417(a) requires monthly testing for power operated valves that experience a 25% increase or more in stroke times, where the stroke time is greater than 10 seconds, or 50% or more for valves with a stroke time less than or equal to 10 seconds. Although paragraph IWV-3412 allows valves that cannot be exercised during plant operation to be tested at cold shutdowns, the licensee should ensure the valve's ability to perform its safety function prior to startup, otherwise relief is required to postpone the monthly testing to cold shutdowns.

CONCLUSION

Generic Letter 89-04 was written to help expedite the IST relief request review process. Licensees can assist in the process by writing requests that clearly communicate the basis and the alternate testing, and by submitting reference materials and drawings with the relief requests. The status of relief request approvals should also be provided to assist the reviewers.

The quality of the relief requests greatly affects the approval process. Incorrect valve numbers, drawings, and references to Code requirements; incomplete relief bases which fail to address all the Code requirements from which relief is requested; and abbreviated discussions of alternate testing all contribute to a longer review cycle and possible denial of the request based on insufficient information or justification. Numerous requests have been denied as a result of too little information—none have been denied as a result of too much!

A Critical Review of Valve Categories

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ABSTRACT

From the first publication of Subsection IWV in the Summer 1973 Addenda of Section XI to the present Subsection ISTC in the OM Code, there has been a requirement to categorize the valves included in the inservice testing (IST) program. The only major changes in all this time have been the deletion of Category E and the addition of "passive" valves. Either the Code Committee got valve categories "right" the first time or they are long overdue for a change.

This paper reviews the current valve categories in the light of almost 20 years of IST program implementation. The paper will explore the logic behind the current categories; examine how the valve categories are currently implemented, including some anomalies; and present finally, some conclusions and recommendations for changes to be made.

INTRODUCTION

Inservice testing (IST) requirements for valves first appeared with the addition of Subsection IWV to Section XI in the Summer 1973 Addenda. Prior to this, the only IST requirements were those in the plant Technical Specifications. One requirement in Subsection IWV was to categorize, in accordance with the rules of IWV, all the valves which were included in the plant's IST program.

The requirement for categorizing the valves in the IST program have been maintained, with only two changes, in all later editions and addenda of Section XI. The changes, dropping Category E and adding provisions for "passive" valves, occurred in the Winter 1977 Addenda. The descriptions of the valve categories have remained the same.

When Part 10 of the Operation and Maintenance (O&M) Standards, the successor document to IWV, was issued, the requirement to categorize the valves in the IST program and the valve categories were retained. The present OM Code also contains the same valve categorization requirements. The descriptions of the categories are essentially identical to the earlier codes and standards.

REVIEW OF VALVE CATEGORIES

What can account for the remarkable longevity of the valve categories in a Code where almost every other aspect has undergone major revisions? There can only be two answers—either the valve categories, as originally developed, were so good that no changes have been needed, or the categories are in need of a long overdue critical review. Since the first answer seems very unlikely, a critical review of valve categories is needed.

A good way of reviewing the valve categories is to look at the testing requirements. Table 1 compares the testing requirements for the different categories of valves, as required by different codes and standards.

Categories A and B

Category A valves are defined as "valves for which seat leakage is limited to a specific maximum amount in the closed position for fulfillment of their (safety) function."

Table 1. Comparison of valve test requirements.

	Testing Requirement[a]				
Valve Category	Sec. XI S '73	Sec. XI W '77	Sec. XI W '85	OM-10 ISTC	OM Code
A					
active	IWV-3410	IWV-3410	IWV-3410	4.2.1	ISTC 4.2
	IWV-3420	IWV-3420	IWV-3420	4.2.2	ISTC 4.3
passive	NA	IWV-3420	IWV-3420	4.2.2	ISTC 4.3
B					
active	IWV-3410	IWV-3410	IWV-3410	4.2.1	ISTC 4.2
passive	NA	None	None	None	None
C (Relief)					
active	IWV-3510	IWV-3510	IWV-3510	4.3.1	ISTC 4.4
	PTC 25.2	PTC 25.3	OM-1	OM-1	App. I
passive	NA	None	None	None	App. I
A-C (Relief)					
active	IWV-3510	IWV-3510	IWV-3510	4.3.1	ISTC 4.4
	PTC 25.2	PTC 25.3	OM-1	OM-1	App. I
	IWV-3420	IWV-3420	4.2.2	ISTC 4.3	
passive	NA	IWV-3420	IWV-3420	4.2.2	ISTC 4.3
C (Check)					
active	IWV-3520	IWV-3520	IWV-3520	4.3.2	ISTC 4.5
passive	NA	None	None	None	None
A-C (Check)					
active	IWV-3520	IWV-3520	IWV-3520	4.3.2	ISTC 4.5
	IWV-3420	IWV-3420	IWV-3420	4.2.2	ISTC 4.3
passive	NA	IWV-3420	IWV-3420	4.2.2	ISTC 4.3
D (Valve)	IWV-3610	IWV-3610	IWV-3610	4.4.1	ISTC 4.6
D (R Disk)	IWV-3620	IWV-3620	IWV-3620	4.4.1	ISTC 4.7
			OM-1	OM-1	App. I
E	IWV-3700	NA	NA	NA	NA

a. All valves with remote position indication require position indicator verification.

Category B valves are defined as "valves for which seat leakage in the closed position is inconsequential for fulfillment of their (safety) function."

Using a literal interpretation of these category definitions, all valves that perform a safety function in the closed position must be either Category A or B. Either a valve must limit leakage or leakage is inconsequential to fulfilling its safety function. In addition, if leakage in the closed position is inconsequential to fulfilling a valve's safety function, why have a valve at all? A piece of pipe would do the same thing! Using this logic, the only Category B valves would be valves that have a safety function only in the open position.

In practice, the category definitions are not taken that literally. Table 1 shows that the testing requirements, except for leakage testing, for Category A and B valves are the same. A review of the test requirements for Category A and B valves clearly shows that these categories were meant only to apply to power-operated valves (e.g., ac electric motor-operated or air-operated). Even at that, the difference between the various types of motor-operated valves has led to a differentiation of the exercise testing requirements for different types of valve operators in OM-10 and the OM Code.

In practice, the difference between Categories A and B is based on whether the seat leakage must be limited to a relatively small amount (e.g., containment isolation valves) for the valve to perform its safety function in the closed position. If so, the valve is Category A. Seat leakage is not inconsequential to a Category B valve fulfilling its safety function in the closed position. Some leakage can be tolerated, but gross leakage cannot.

Category A valves, generally, are required to be leak tested by the plant Technical Specifications. This also applies to many valves that are not power-operated valves. The use of combined categories, such as A-C, is an attempt to make the valve categories fit circumstances for which they were not designed.

Category C

Category C valves are defined as "valves which are self-actuating in response to some system characteristic, such as pressure (relief valves) or flow direction (check valves)."

The definition itself divides Category C valves into two groups: relief valves and check valves. A look at Table 1 confirms that there are two completely different sets of test requirements. In fact there is no similarity between the test requirements for the two types of valves. Testing requirements for relief valves are in referenced documents while requirements for check valves are in the base document.

Category C makes no provision for any type of valves other than relief valves and check valves. There is also no provision for Category C valves requiring a leak test. Section XI (for example) states "When more than one distinguishing category characteristic is applicable, all requirements of each of the individual categories are applicable...." This would give us a Category A-C valve. It is also stated that "...repetition of common testing requirements is not necessary." The exercise test required by Category A and Category C are not "...common testing requirements..." as the exercising test requirements are very different. It was not the intent to require both sets of exercising test requirements to be applied. Only the leak test requirements of Category A are required.

Category D

Category D valves are defined as: "valves which are actuated by an energy source capable of only one operation, such as rupture disks or explosive-actuated valves."

The situation in this case is much like that of Category C. The definition itself divides Category D into two groups: rupture disks and explosive-actuated valves. Again, Table 1 shows that there are two completely different sets of test requirements. Since the adoption of OM-1, testing requirements for rupture disks are found in OM-1 (later Appendix I of the OM Code).

Category D makes no provision for any type of valves other than rupture disks and explosive-actuated valves. Their is also no provision for Category D valves requiring a leak test. The use of a Category A-D is even more problematical than Category A-C. Nether rupture disks nor explosive-actuated valves can, in any way, be exercise tested, as this would render them unfit for further service.

Category E

Category E valves are defined as "valves which are normally locked (or sealed) open or locked (or sealed) closed to fulfill their (safety) function."

Although Category E was deleted by the Winter 1977 Addenda, it is interesting to look at some of the problems that this category engendered. The biggest problem is what does "locked (or sealed)" mean.

A chain with a padlock obviously qualifies, but does a wire and lead seal? The wire and lead seal will not stop any deliberate tampering, but will discourage accidental tampering and will make any tampering evident. What about the case of a remotely operated valve that has a keylocked switch in the control room, but also has a local handwheel that is unlocked? What about the converse, a valve with the local handwheel locked, but a plain hand switch in the control room? Is a switch cover on the control panel a seal? What about valves that have administrative controls, such as being in a locked room, but do not have a lock? These and other questions would have needed answers if Category E had not been deleted.

Active and Passive Valves

At the same time Category E was deleted, the concept of active and passive valves was introduced. Active valves are defined as "valves which are required to change position to accomplish a specific (safety) function." Passive valves are defined as "valves which are not required to change position to accomplish a specific (safety) function."

In simple terms an active valve is one that has to move from its normal position (open or closed) to a different position to fulfill its safety function, or where the valve may be either open or closed during normal operation. A passive valve is one that needs to stay in its normal position, either open or closed, to fulfill its safety function.

Table IWV-3700-1 of Section XI lists Category A passive valves as the only type of passive valve. Category A passive valves only require leak testing and do not require any exercise test. Table ISTC 3.6-1 of ISTC of the OM Code (Table 1 of OM-10) adds Category B passive valve. Category B passive valves only require position indicator verification. (This requirement was also added for Category A passive valves.) Although this solves most of the problems of Category E, other problems come up. One problem is that Category B passive valves are included in the IST program but have no test requirements. Another problem is the application of the term "passive" to Category C check valves. This use of the term "passive" was never intended.

Anomalies

As a result of some of the problems discussed above and plant-specific configurations, a number of anomalies have been encountered during preparation and review of IST programs. A number of these highlight the problems with the current categories and will be discussed here.

Category A-C and A-D Valves. As discussed above, the convention of using Category A with either Category C or Category D to indicate Category C or D valves requiring leak testing has become routine. Obviously, all the testing requirements for Category A valves cannot be met by the Category C or D valves. Although almost everyone knows how this is interpreted, it is still a misapplication of the valve categories.

Power Operated Relief Valves. Power operated relief valves (PORVs) are generally found in pressurized water reactors (PWRs) attached to the pressurizer and main steam lines. Some PORVs are power operated block valves which open and close in response to pressure upstream of the valve. The valve itself is a power operated valve and, based on that, is usually Category B. The valve functions as a relief valve. Mandatory Appendix I of the OM Code (OM-1) has provisions for testing PORVs. An O&M standard (OM-13) also addresses PORVs. These valves are not taken credit for on the Section III overpressure protection report and do not perform a safety function, but this type of valve was a significant contributor to the TMI accident. How to test it is a very confusing subject and is currently under discussion at several O&M working groups.

BWR ADS Valves. The automatic depresserization system (ADS) valves of boiling water reactors (BWRs) are similar to the PORVs discussed above. The ADS valves perform two different functions: one as a normal safety valve and one as a remotely operated, power-operated (air or nitrogen) valve using an auxiliary actuating device. These valves are normally designated Category B-C. They require a test using the external actuating device in accordance with Category B requirements and a test of the safety valve in accordance with Category C requirements.

Closed Cooling Water Supply Valves. Figure 1 shows the arrangement of valves in a closed cooling water (CCW) system. The CCW system is a safety related, two train system which supplies cooling water to a number of safety related loads such as the diesel generator heat exchanger and safety related pump coolers. Each load can be supplied by either train of the CCW system. A stop check valve is located in each supply line to prevent backflow. The handwheel of the stop check valve is up on the train which is supplying the load. The handwheel is down on the train which is not supplying the load to prevent flow in that line. The stop check valve with the handwheel up is classified as a Category C check valve and is tested as such. The stop check valve with the handwheel down is categorized as Category B-C passive valve. If the load is transferred to the other train of CCW, the categorization is reversed.

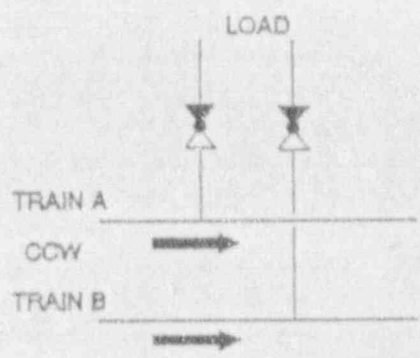

Figure 1. CCW system.

Rupture Disks. Currently, rupture disks are categorized as Category D. Rupture disks are tested in accordance with the rules for nonreclosing pressure relief devices in Appendix I (OM-1). If a rupture disk is a pressure relief device, as stated in Appendix I, it should be Category C. On the other hand, it is specifically called out as Category D. This, obviously, needs some change.

CONCLUSIONS AND RECOMMENDATIONS

The current valve categories obviously do not fit all situations. Category A has become, for other than power operated valves, a designator for the need for a leak test. In the case of Category C and Category D, they should not combine two very different types of valves. In general, the valve categories, as presently structured, do not seem to provide any benefit, but tend to confuse matters. This situation has been perpetuated in the new OM Code and should be corrected.

Subsection ISTC of the OM Code would be better served by deleting the current valve categories. In general, valves should be classified based on their distinguishing characteristics. Table 2 shows an outline of one possible valve classification scheme. In this classification scheme, valves are classified based on the most distinguishing valve and/or valve operator characteristic.

The defining characteristic of a power-operated valve is the fact that it is power operated. This allows tests such as stroke timing to be performed. The specifics of valve type (gate, globe, ball, etc.) can be looked at as secondary characteristics. Power-operated valves are subdivided based upon the type of operator, as different types of operators have different characteristics and would have, to some extent, different test requirements. Solenoid-operated valves are treated separately because of their fast operating time and lack of external position indication.

Check valves are characterized by the valve itself and the type of service it performs, i.e., preventing backflow. These valves are subdivided by the testing method used. Safety and relief valves are again characterized by the valve itself and the type of service it performs, i.e., pressure relief. These valves are subdivided by the type of plant to conform to the division of Appendix I. Explosively actuated valves are completely different in that any test of the valve itself makes the valve unfit for further service. In this case the explosive charge must be tested separately from the valve. Finally, leak testing requirements may be added to any type of valve when required.

The proposed restructuring of ISTC is only one of a number of possible configurations. Any change should build on the experience gained in almost 20 years of inservice testing implementation. Just because "that's the way it has always been done" is no justification for continuing to perpetuate confusing requirements.

ACKNOWLEDGMENT

The author would like to thank Tom Hoyle, Chairman of the Subgroup on Pumps and Valves. The original idea for the restructuring of ISTC proposed in this paper came from the work of Mr. Hoyle, and this paper could not have been written without his contribution.

Table 2. Valve classification for ISTC.

1. Power Operated Valves
   - 1.1 AC Motor
   - 1.2 DC Motor
   - 1.3 Pneumatic
   - 1.4 Hydraulic
2. Solenoid Operated Valves
3. Check Valves (including Swing Check, Stop Check, and Spring Closing Check Valves)
   - 3.1 Exercise Tested
   - 3.2 Tested using nonintrusive techniques
   - 3.3 Disassembled
4. Safety and Relief Valves (including Nonreclosing Pressure Relief Devices - Rupture Disks)
   - 4.1 BWR Safety and Relief Valves
   - 4.2 PWR Safety and Relief Valves
5. Explosively Actuated Valves
6. Leak Test Requirements

Session 2A

Solenoid and Air-Operated Valve Performance and Testing

Session Chair
Ivo Garza
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Recent Solenoid-Operated Valve Experiences Involving Maintenance and Testing Deficiencies

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ABSTRACT

The paper presents recent solenoid operated valve operating experience. It describes common-mode failures at the Salem Unit 2 and Peach Bottom Units 2 and 3 plants. At the Salem plant, a catastrophic failure of turbine equipment resulted in an outage of about 6 months. The Peach Bottom event involved repetitive failures of safety systems with a shorter outage. The lessons learned from both events are that prudent preventive maintenance and surveillance testing are needed to enhance safe economic plant operation.

INTRODUCTION

In February 1991, The U.S. Nuclear Regulatory Commission (NRC) issued an AEOD Case Study, "Solenoid-Operated Valve Problems at U.S. Light Water Reactors," NUREG-1275, Volume 6 (1991a). That report presented information on about four dozen events in which solenoid-operated valves (SOVs) had failed or were degraded so that the safety margins of plants were reduced below the levels assumed in plant safety analyses. Many of the events were attributed to inadequate maintenance resulting from the fact that the SOVs were "unrecognized piece-parts" of other equipment or systems. In many instances, inadequate maintenance (or no maintenance) was performed because the Licensees were unaware of the SOVs' maintenance or life-cycle requirements. About one-third of the common-mode events presented in the case study were found during surveillance testing. This paper presents information on recent common-mode SOV failures. It demonstrates the need for prudent maintenance and surveillance testing.

DISCUSSION

It is the author's view that prudent maintenance and surveillance testing can help minimize the likelihood for common-mode SOV failures. However, maintenance and surveillance testing are not substitutes for good engineering and should not be relied upon to overcome all design and application errors. [As noted above, about one-third of the common-mode SOV events presented in the AEOD case study (NRC, 1991a) were found by surveillance testing.]

Although it did not involve any "safety-related" systems, one of the most costly common-mode SOV events in the United States was the turbine overspeed event that occurred at Salem Unit 2 on November 9, 1991. That event involved the failure on demand of three solenoid-operated valves in the turbogenerator's overspeed protection control system. Those three SOV failures resulted in major damage to the turbine and generator, condenser failures, lube oil and hydrogen fires, and a hydrogen explosion. The turbine speed was estimated to be 2900 rpm (vs. 1800 rpm rated speed). Turbine missiles

a. This presentation was prepared (in whole, or in part) by an employee of the United States Nuclear Regulatory Commission. It presents information that does not currently represent an agreed-upon staff position. NRC has neither approved nor disapproved its technical matter.

penetrated the turbine casing, and some missiles traveled over 100 yards. Figures 1 and 2 show some of the damage. The event resulted in an extended outage expected to be about 6 months with a cost estimated at over $100 million.

About 3 weeks before that event, the Licensee was performing a test of two SOVs in the main turbine's overspeed trip system (Public Service Electric and Gas Company, 1991; NRC, 1992). The surveillance test being conducted was not capable of revealing the malfunction or degradation of one of two parallel overspeed protection controller SOVs (OPC 20-1 or OPC 20-2 shown in Figure 3). A successful test would confirm that at least one of those two SOVs was operating properly. An unsuccessful test would indicate failures of both SOVs. The operators performing the test in October were puzzled when the surveillance test showed that both SOVs were inoperable. They repeated the test a second time and had the same results. The operators discussed the test failures with other station personnel, and they collectively concluded that the SOVs could not both have failed and that something must have been wrong with the test procedure. It is highly probable that both SOVs did not fail at once in October 1991. It is quite likely that one SOV had failed earlier (and had been undetected), and that the second SOV had undetected degradation and failed at the time of the surveillance test. Three weeks later, on November 9, 1991, the main turbine overspeed protection system was tested. However, in addition to the two undetected failed SOVs, a third SOV being tested also failed (SOV ET-20 shown in Figure 3). Had either of the first two SOVs been operable (OPC 20-1 or OPC 20-2), one of them would have actuated upon failure of the third SOV (ET-20) and a simple turbine trip would have occurred without any damage.

The AEOD case study (NRC, 1991a) noted many other previous events in which inadequate surveillance testing was responsible for not detecting common-mode SOV failures or degradations which compromised safety systems at other plants. An example of inadequate surveillance testing that did not detect individually failed SOVs in a similar parallel arrangement was observed in Liebstadt's emergency diesel generators (EDGs) (NRC 1991a). On a visit to the Waterford Unit 3 nuclear power plant in March 1992, the author learned that after the Salem 2 event, the Waterford Licensee conducted a test of its turbine overspeed system, using a revised testing procedure in order to determine the operability of each of the two parallel SOVs, OPC 20-1 and OPC 20-2. Previous testing at Waterford, like the testing at Salem, was incapable of detecting a single failed SOV. Waterford's first test (Waterford, 1992), which was performed on February 21, 1991, revealed a failed SOV (Parker Hannefin Model No. MRFN16MX0834—the same model valve as the ones that failed at Salem). The Waterford staff proceeded to test the second Parker Hannefin MRFN16MX0834 SOV with great hopes that it would work satisfactorily—otherwise, they feared that they would have been in a situation similar to that at Salem Unit 2 (i.e., performing a new test, finding both SOVs failed, and suspecting that the SOVs were really operable and assuming that the surveillance testing procedure was flawed). Fortunately, the surveillance test of the second SOV at Waterford found that it did operate satisfactorily, thereby confirming that the new surveillance testing procedure was not flawed, and that the first SOV tested had truly failed.

A less dramatic, but more safety-significant example of inadequate SOV surveillance testing occurred at the Peach Bottom Unit 3 facility in August 1991. At that time, the Peach Bottom station experienced widespread degradation and multiple failures of SOVs affecting many safety-related systems. As noted in NRC 1991a and 1991c, SOVs piloting air-operated valves (AOVs) controlling emergency service water (ESW) for the HPCI and RCIC room coolers had been sticking during surveillance tests. Similar events involving valves controlling ESW to other safety-related equipment were reported in an initial notification report to the NRC Operations

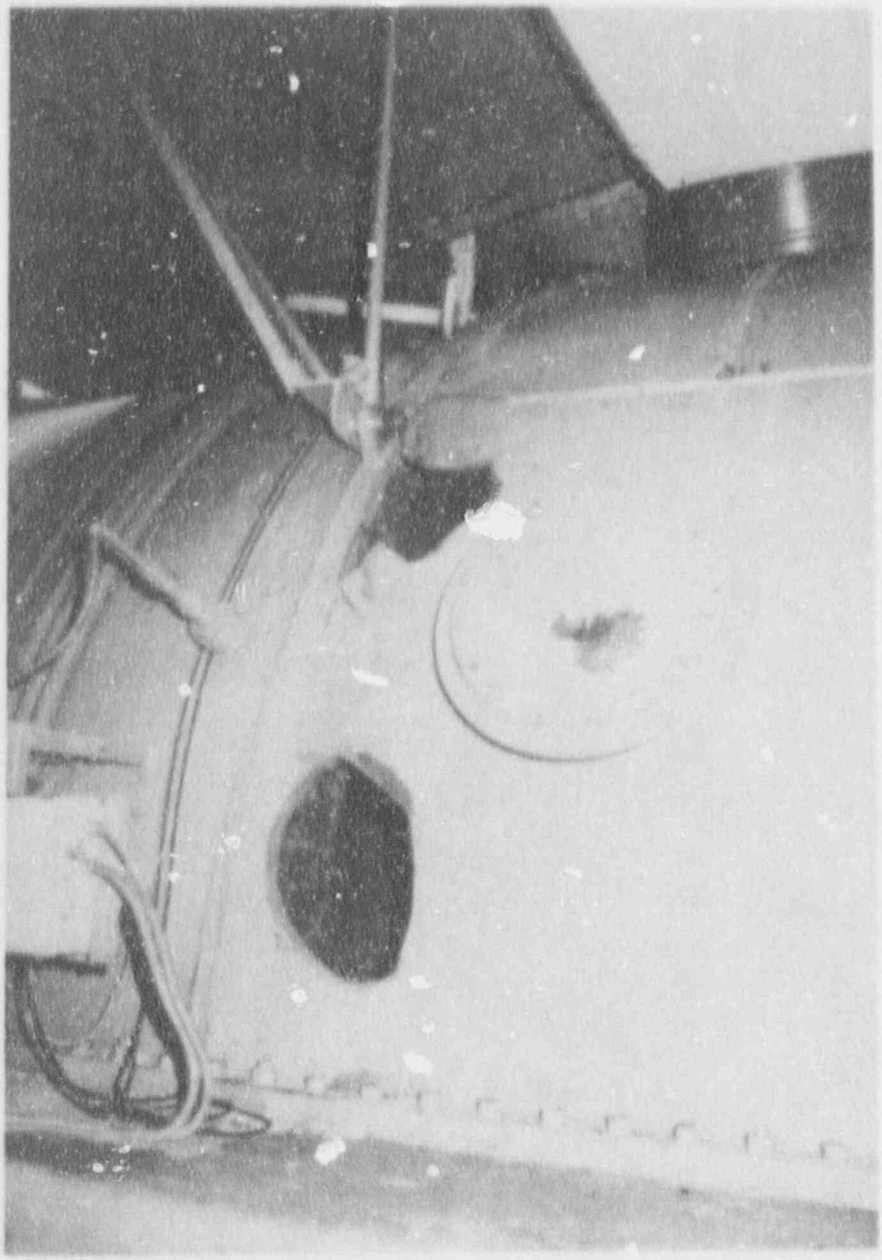

Figure 1. Turbine casing Salem Unit 2.

Figure 2. Low pressure turbine Salem Unit 2.

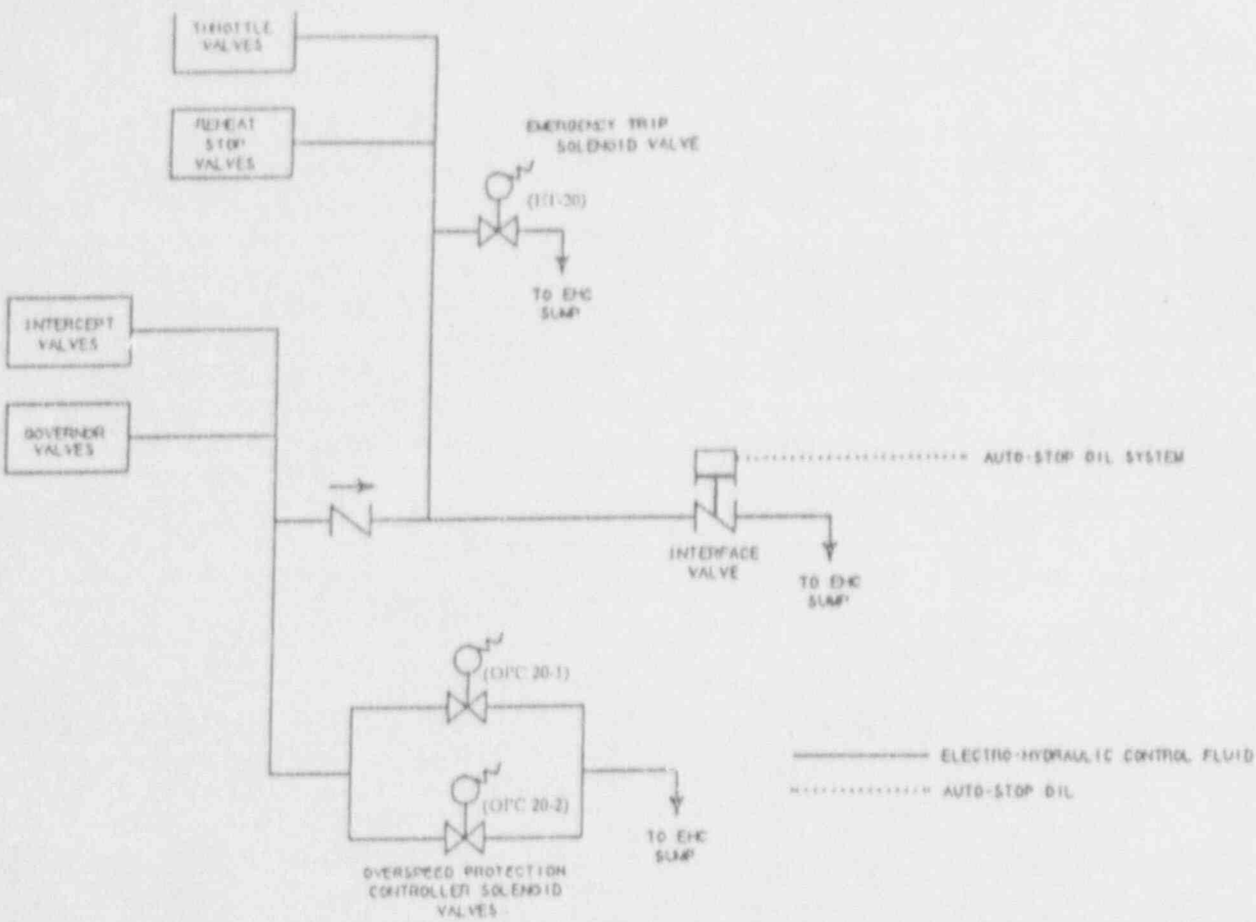

Figure 3. Turbine electro-hydraulic control system Salem Unit 2.

Center (NRC 1991b). That report noted that on August 25, 1991, two SOVs controlling ESW to HPCI and RCIC room coolers were found to be mechanically bound. With regard to RCIC room coolers, the Licensee "immediately agitated the valve enough so it would operate properly." As a result, the Licensee considered the room cooler "fully operable." In addition, the Licensee said that the plant had several problems with AOVs failing to operate in the past on other systems. However, the Licensee noted that the station had "always been able to mechanically agitate these valves so they became operational again." The NRC (1991b) report noted that the Licensee also said that for the August 25, 1991, event, "they never declared HPCI or RCIC inoperable because the room cooler problems were corrected immediately upon discovery." Subsequently, since redundant HPCI and RCIC room coolers were already isolated, Peach Bottom management declared the HPCI and RCIC systems inoperable, and the unit was shutdown. A subsequent NRC inspection (1991c) concluded that the issue of unreported SOV failures noted on August 25, 1991, appeared to have been an isolated case.

Subsequently, a review of SOV applications found that each Peach Bottom Unit 44 of the same model (ASCO 206-832) SOVs as the ones that had failed on August 25, 1991. Forty of those SOVs controlled room cooling for all the RHR and core spray pumps, as well as room cooling for the HPCI and RCIC systems at each Peach Bottom unit. The other four SOVs were located in the ESW return lines from each of the station's four EDGs. The NRC (1991c) report indicates that 22 reported failures of those SOVs had occurred prior to the August 25, 1991, event.

In response to some of the previous SOV failures, the Licensee did extensive root cause failure analysis. Contaminants from the instrument air system and valve lubricants were believed to have caused some of those failures. However, prior to the August 25, 1991, event, the Licensee had not taken any systematic steps to preclude common-mode failures of the SOVs in multiple safety systems at both units. [It is interesting to note that prior to installing these SOVs in the late 1980s, the Licensee conducted Nuclear Plant Reliability Data System (NPRDS) searches on ASCO 206-832 valves and only found two entries, both of which involved installation errors with no operating failures reported.] None of Peach Bottom's 22 "internally" reported failures were reported to NPRDS or the NRC's Licensee Event Report (LER) system before August 1991.

After the August 25, 1991, event the Licensee embarked on an aggressive program to prevent similar common-mode SOV failures. In addition to performing detailed plant walkdowns with verification of SOV applications (temperature, orientation, MOPD, and voltage life cycle), the Licensee has implemented frequent stroke testing of the SOVs of concern (weekly and in some cases, semi-weekly testing). For the longer term, the Licensee is planning to implement staggered maintenance, staggered surveillance testing, and SOV diversity (the use of different SOV models in alternate ECCS trains).

CONCLUSIONS

The events described above at Salem Unit 2 and Peach Bottom Units 2 and 3 are exemplary of situations where less than adequate surveillance testing and maintenance of SOVs resulted in the reduction of plant safety margins and significant financial burden. In recognition of the fact that highly reliable, nonobtrusive SOV diagnostic/monitoring equipment is not available, prudent preventive maintenance and surveillance testing should be used to minimize the likelihood for common-mode SOV failures, thereby enhancing reactor safety and possibly avoiding major down times.

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Use of Ultrasonics and Acoustics in Measurement of Solenoid Valve Stroke Time at Hope Creek Generating Station

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ABSTRACT

All light water reactors in the United States use solenoid valves in some safety related application. Many of these applications are classified as Category A or B under Section XI, IWV-3400, of the American Society of Mechanical Engineers Code. The requirements contained in IWV-3413(b) present a problem when valves with stroke times of <2 seconds (rapid-acting) are tested. Conventional stop watches are difficult to use when measuring these values. The physical configuration of the valves is such that they are completely enclosed and possess no electrical or mechanical position indication system. Exploration with various techniques resulted in the selection of a technique that is non-intrusive, capable of detecting movement in hundredths of a second, and provides hard-copy results in graphic and tabular format.

INTRODUCTION

"Rules for Inservice Inspection of Nuclear Power Plant Component," Section XI, Paragraph IWV-3413(b) of the American Society for Mechanical Engineers (ASME) code, requires that all power operated valves be periodically stroke timed to the nearest second. Generic Letter (GL) 89-04, "Guidance on Developing Acceptable Inservice Testing Programs," Positions 5 and 6 state that the intent of the Code is to verify operability and detect degradation of the valve over a period of time. Additionally, Position 6 of GL 89-04 states that valves with a stroke time of 2 seconds or less are classified as "rapid acting valves." Relief may be granted from Paragraph IWV-3417(a), provided the Licensee assigns a maximum value of full-stroke time of 2 seconds to these valves and, upon exceeding this limit, declares the valve inoperable and takes corrective action in accordance with IWV-3417(b). Testing these valves using a stopwatch requires operator training to ensure that reaction time is quick enough not to affect operability. This problem was compounded at Hope Creek because these valves had no electrical or mechanical position indicating system. The initial proposed solution was to request relief from ASME Section XI, Paragraph IWV-3413 stroke time requirements with the alternate of full stroke exercising and fail-safe testing quarterly. Unfortunately, this option did not provide a long-term solution to evaluating changes in valve condition. The NRC suggested we explore other technologies in this application.

These technologies included installation of position indicating lights, system modification to detect flow, and local position indication devices. A total of 13 safety related valves in this specific population were in diesel generator lube oil, station auxiliary cooling, and residual heat removal systems. A novel approach to solve this problem was attempted. A diagnostic system was currently being used for Institute of Nuclear Power Operations Significant Operating Experience Report 86-03 (check valve failures or degradation). This system employed ultrasonics and acoustics to assess the condition of check valves. At the time it was thought that the same ultrasonic signal used to detect check valve disc location could also detect globe valve disc location. Additionally, the acoustics mode would detect the solenoid valve movement. This method was proposed to the NRC and met the intent of Paragraph IWV-3413.

INVESTIGATION

Technician training for check valve testing was currently in progress during the investigation. At that time the alternate test method was proposed to the equipment manufacturer for validation. It was determined that the test configuration was similar to that of check valve stroke testing and within the ability of the equipment. A full size mock-up was set up for testing and training. The solenoid power supply was handswitch controlled, and the valve body filled with demineralized water to simulate system fluid (Figure 1).

The valves were 1-in., nominal pipe size, 600-lb, socket welded end, normally closed (NC) solenoid valves. The valve body configuration (square block like in nature) easily lent itself to being scanned with a transducer. The ultrasonic transducer was placed at the bottom of the valve and positioned to focus upward into the valve seat (Figure 2). Because of the angle of incidence, which caused sound beam refraction, a 20-degree lucite wedge had to be attached to the transducer to correct for the sound velocity differences of steel and water (Figure 3). This wedge focused the beam at the valve seat. The detection of the disc was displayed on the ultrasonic screen in the following manner. The valve was placed in the closed position, and demineralized water filled the waterways. A 2.25 MHz signal was focused at the bottom of the valve toward the disc. Once the signal peak was located on the screen of the oscilloscope, the "gate" was positioned to cover the entire horizontal range of motion of the signal peak by adjusting the "start" and "length" control knobs. Once the peak was located, the gate level was raised or lowered as necessary until the peak triggered the gate. This was accomplished by setting the "horn" switch to the "on" position and listening for the steady tone (or watching for the corresponding light) when the spike intersected the gate (Figure 4). Full disc contact was assumed when the ultrasonic signal peaked off the screen at the approximate linear distance from the valve outer surface to the inserted valve disc. Verification was performed when the valves' disc was opened a second time and the signal disappeared.

In order to corroborate the stroke time measurements made with the ultrasonics mode, a second test method was used. This method used microdot style accelerometers mounted to a stud attached by epoxy resin to the valve body and operator. The epoxy was a phenolic resin with analyzed halogen content of <200 ppm. The mounting stud was plastic with the same halogen requirements as the phenolic resin. The mounting stud was small, (0.500 in. diameter × 0.125 in. thick) with a No. 10 x 32 stud 0.125 in. long on one side centered in middle. The total weight is <0.500 grams. The minimal weight does not present a problem from a seismic standpoint. Testing indicated that these valves would stroke in a range of 200 to 300 milliseconds, which was well within the 2-second requirement.

EQUIPMENT

The test equipment comprised various subcomponents. The following is a list of the major components:

- Model 3386SX series SAM (signature analysis module) with automated software including a digital signal process card and accelerometer control card
- EGA/VGA monitor with flat top screen
- Accelerometers with cables
- Transducers with cables
- Ultrasonic Oscilloscope.

Used to compile the raw acoustic and ultrasonic data, the SAM was an IBM compatible personal computer with a 80386SX central processing unit, a 40-megabyte hard drive, and a 1.44-megabyte, 3.5-in. floppy disc drive. The monitor could be a flip up amber screen or a color EGA.

The SAM was used in the field to acquire data from the ultrasonic scope and acoustic functions. The data were then saved either on the hard drive or the floppy disc, and analysis could be performed at the valve, or the unit could be removed to an office area. An analog to digital card was used to convert the analog input from the Stavely QC 400 scope (ultrasonics) into a digital signal to be used within the software. The resultant data were presented in a graphic format with time display on a linear chart (Figure 5). The data could also be produced in a tabular format to record the time between spikes and height of spikes. The acoustic analysis was based on sound vibrations received from piezoelectric accelerometers. The piezoelectric accelerometer converted sound from mechanical motion into an electric signal. That signal was displayed on the monitor (Figure 6). The spikes displayed on the screen represented the noise generated when the solenoid energizes and degenergizes. As noted in Figure 6, the intensity of the spikes varied. This is caused in part by the exercising of the valve. The more frequent the exercising, the less noise the solenoid will produce when energized and degenergized. The true determination of the valve stroke was made by measuring the distance from peak to peak. At the bottom of the page, a linear time graph was divided into an 8-second band. By transferring the peaks to the time graph, the stroke time can be determined. The accelerometers were connected to the SAM via microdot cable with insulated bulkhead connectors.

The digital signal processing board analyzed the acoustic signals, provided the analog to digital conversion, acted as the central processing unit for the acoustic data, and stored signature to the hard drive of the computer. The main functions of the SAM are to control the accelerometer transducers, adjust the gain for acquired signatures, and magnify the amplitude, as necessary to view the individual impacts.

RESULTS OF TESTING

Once this method provided repeatable data for the stroke time testing, procedures were written to use this method on in-plant equipment. Since this method was relatively new to plant personnel, the credibility of the results were called into question by various organizations. The testing provided information that verified the subject valves operated consistently within a 200- to 300-millisecond band. The testing proved itself to be accurate when one valve was very slow in response time. This diesel generator lube oil makeup valve had a nominal stroke time of 1.50 seconds, which is quite a bit longer than average, but still within the acceptance range of GL 89-04. A work request was written to disassemble and inspect. Upon disassembly, an accumulation of fibrous material was found on the valve seat. The source of the material was unknown, but it appears to have been a piece of rag from construction. (These lines now have suction filters to prevent introduction of foreign material into the system.)

This method is currently being used on 13 valves at the Hope Creek Generating Station with excellent results. The relative ease of operation and the storage of data on the computer have removed the subjectivity of data collection. With the areas identified to reattach the probes to the valves, repeatability of the test has proven to be excellent. This method is well accepted at Hope Creek, and its use will be expanded to troubleshoot valves in both safety and nonsafety systems as a first choice rather than disassembly.

ACKNOWLEDGMENTS

This paper represents the combined effort of numerous individuals and organizations. Appreciation is extended to all for their contributions. Mr. Howard Hiles was the maintenance supervisor who served as test coordinator and provided day-to-day coordination to keep testing on schedule. Appreciation is extended to ITI Movats for their cheerfulness in responding to questions and their support this project. Many hours were spent trying to resolve basic questions on the system adaptability and usefulness of data. Finally, we appreciate the support of station management for their belief in the system and the concept.

Inservice Diagnostics for Solenoid Operated Valves

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ABSTRACT

Solenoid-operated valves (SOVs) were studied at Oak Ridge National Laboratory as part of the USNRC Nuclear Plant Aging Research (NPAR) Program. The primary objective of the study was to identify, evaluate, and recommend methods for inspection, surveillance, monitoring, and maintenance of SOVs that can help ensure their operational readiness-that is, their ability to perform required safety functions under all anticipated operating conditions, since failure of one of these small and relatively inexpensive devices could have serious consequences under certain circumstances.

An earlier (Phase I) NPAR program study described SOV failure modes and causes and identified measurable parameters thought to be linked to the progression of ever-present degradation mechanisms that may ultimately result in functional failure of the valve. Using this earlier work as a guide, the present (Phase II) study focused on devising and then demonstrating the effectiveness of techniques and equipment with which to measure performance parameters that show promise for detecting the presence and trending the progress of such degradations before they reach a critical stage.

Intrusive techniques requiring the addition of magnetic or acoustic sensors or the application of special test signals were investigated briefly, but major emphasis was placed on the examination of condition-indicating techniques that can be applied with minimal cost and impact on plant operation. These include monitoring coil mean temperature remotely by means of coil dc resistance or ac impedance, determining valve plunger position by means of coil ac impedance, verifying unrestricted SOV plunger movement by measuring current and voltage at their critical bistable (pull-in and drop-out) values, and detecting the presence of shorted turns or insulation breakdown within the solenoid coil using interrupted-current test methods

Experimental results are presented that demonstrate the technical feasibility and practicality of the monitoring techniques assessed in the study, and recommendations for further work are provided.

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b. This presentation was prepared (in whole, or in part) by an employee of the United States Nuclear Regulatory Commission. It presents information that does not currently represent an agreed-upon staff position. NRC has neither approved nor disapproved its technical matter.

BACKGROUND

In the context of this report, "aging" is defined as degradation that occurs with the passage of time. This degradation is associated with the alteration of physical properties brought about by the action of environmental and operational stressors. Aging affects all reactor structures, systems, and components to some degree and has a potential to increase the risk to the public health and safety if its effects are not recognized and controlled. Therefore, to ensure continuous safe operation of a nuclear power plant as it ages, measures must be taken to monitor these systems, components and interfaces in order to detect the presence of degradation and, if necessary, to restore integrity through effective maintenance, repair, or replacement. The research will be documented in *Aging and Service Wear of Solenoid-Operated Valves Used in Safety Systems of Nuclear Power Plants, Vol. 2: Evaluation of Monitoring Methods*, NUREG/CR-4819 by R.C. Kryter.

The USNRC Nuclear Plant Aging Research (NPAR) Program was established in response to just such concerns, and seeks to help resolve technical safety issues related to the aging of electrical and mechanical components, safety systems, support systems, and civil structures used in commercial nuclear power plants. Solenoid-operated valves (SOVs) were studied at Oak Ridge National Laboratory as a part of this NPAR program.

A solenoid-operated valve is one that is opened and closed by means of an electrically actuated solenoid coil that, in most designs, lifts a plunger to open or close the valve port(s). The process fluid that is thus controlled is most often instrument air, but nitrogen or process water may be encountered in some plant systems. SOVs may be direct-acting (i.e., the solenoid coil provides the motive force for the opening and closing of the valve) or may be pilot-assisted (where the solenoid coil causes the opening of a pilot orifice, thereby allowing the process fluid to force the main orifice open). Only the direct-acting variety was employed in this study.

Solenoid-operated valves are available in a variety of sizes and constructions, both with and without nuclear qualification, from a number of different manufacturers and are found throughout nuclear power plants (Table 1) in relatively large numbers,[c] oftentimes being a subcomponent of larger, more complex, and clearly safety-related systems such as containment isolation valve actuators, BWR control rod scram systems, and PWR safety injection systems. They are relatively simple devices (see Figure 1), with a long history of satisfactory operation in a variety of both nuclear and nonnuclear industrial applications. However, their presence in systems important to safety requires an especially high degree of assurance that they are ready to perform their required function under all anticipated operating conditions, since failure of one of these small and relatively inexpensive devices could have serious consequences under certain circumstances (Verna, 1991).

IDENTIFICATION OF THE SOV AS A COMPONENT FOR STUDY

In accordance with NPAR Program strategy, a component, system, or structure is identified for study by considering information from several sources. Criteria used in the selection process include (a) the potential contribution to risk from failures of components, systems, and structures; (b) experience obtained from operating plants; (c) surveys of expert opinion; and (d) user needs (including the resolution of generic issues, plant performance indicators, and plant maintenance and surveillance).

c. Bacanskas et al. estimate that the population of SOVs used in safety-related systems at U.S. LWRs lies between 1,000 and 3,000 per plant, with BWRs generally having a greater number than PWRs.

Table 1. A partial list of safety- and non-safety-related systems which use SOVs at U.S. LWRs (Ornstein, 1991a).

1. BWR scram
2. Reactor coolant pump seal
3. Safety-injection
4. Auxiliary feedwater
5. Primary containment isolation
6. High pressure coolant injection/reactor core isolation cooling
7. High pressure injection
8. Automatic depressurization
9. Emergency diesel generator
10. Instrument air
11. Chemical volume control/charging and letdown/boration
12. Pressurizer control
13. Steam generator relief (power-operated relief valves, atmospheric dump valves)
14. Low-temperature overpressurization protection
15. Decay heat removal/residual heat removal
16. Component cooling water
17. Service water
18. Reactor head vent
19. Reactor cavity/spent fuel/fuel handling
20. Torus and drywell/vent and vacuum
21. Emergency dc power
22. Main steam (main steam isolation valves/auxiliary boiler)
23. Reactor building/auxiliary building (ventilation and isolation)
24. Main feedwater
25. Condensate

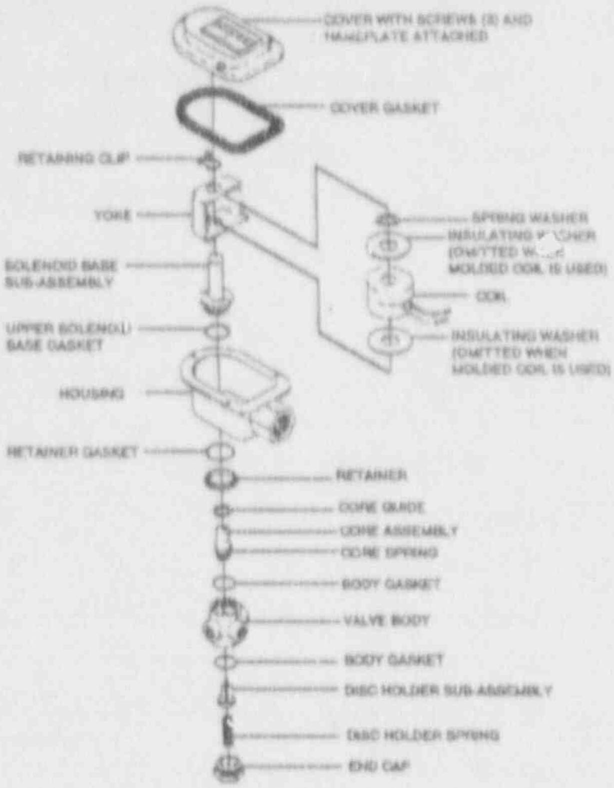

Figure 1. Exploded view of a typical enclosed type, direct-acting three-way SOV.

Information relevant to the selection of the solenoid-operated valve as a component meriting study on the basis of these criteria was developed and is documented in an NPAR Phase I report. (Bacanskas et al., 1987) In addition, the NRC Office for Analysis and Evaluation of Operational Data (AEOD) has recently documented its assessment of the vulnerability of safety-related equipment to common-mode failures or degradations of SOVs in an operating experience feedback report (Ornstein, 1991a). This study cites over twenty representative operational events in which SOV failures or degradations affected—or had the potential to affect—multiple safety systems or multiple trains of individual safety systems. Although such common-mode SOV failures and degradations are often beyond the conditions analyzed in plant final safety analysis reports and are not ordinarily modeled explicitly in present-day probabilistic risk assessments, operating experience indicates that such failures and degradations have indeed compromised front-line safety systems in the past (Ornstein, 1991a) and will likely continue to do so in the future. Interestingly, most plant Technical Specifications do not require periodic testing of SOVs per se, although the valves' performance may come into play in the regular exercise of systems and devices covered by Technical Specification testing requirements.

The AEOD report concludes that ". . . SOV problems represent a significant safety concern," and that ". . . the SOV problems outlined in this report need to be addressed to ensure that the margins of safety for U.S. LWRs remain at the levels perceived during original plant licensing. Generic and plant-specific actions are needed to correct the SOV problems in order to restore the plants' safety margins to their original perceived values," (Ornstein, 1991a).

The findings and conclusions summarized above—plus continuing occurrences at plants (Verna, 1991), resulting in the issuance over the past few years of some 36 NRC communications alerting licensees to generic problems with SOVs—justify the inclusion of solenoid-operated valves in the list of components to be studied by the NPAR Program.

EVALUATION OF DIAGNOSTIC METHODS APPLICABLE TO SOVS

General Considerations; Hardware Studied

Any proposal for implementation of surveillance and diagnostics in nuclear power plants (NPPs) must address the issues of practicality and cost effectiveness. For example, it must be recognized that the SOVs installed in present-day NPPs are uninstrumented and that backfitting them with instrumentation would likely be quite expensive. It is also necessary to understand that many SOVs important to plant safety are inaccessible during plant operation, and that some needing verification of operational readiness will change state only rarely during normal operations, therefore offering little opportunity for measuring SOV dynamic performance parameters. In view of these circumstances, many utilities have, in fact, elected to periodically replace degradable components or subassemblies within SOVs that are embodied in safety-related systems (Bacanskas et al., 1987); (i.e., use preventive maintenance) rather than attempt to practice predictive maintenance using one or more of the approaches described in Bacanskas et al., (1987). In the extreme, some utilities replace the entire SOV as a precautionary measure at predetermined intervals based on lifetime expectations derived from environmental qualification (EQ) tests, even though no malfunction has been observed (Bacanskas et al., 1987; Ornstein, 1991).

In carrying out these Phase II studies, we viewed the above as challenges rather than insurmountable obstacles. Hence, while acknowledging that implementation of diagnostic capabilities will have to be cost effective relative to the alternative of SOV replacement, we continued to search for techniques and/or equipment with which to measure some of the performance parameters identified in the Phase I study and thereby detect and trend the progress of any degradation that might eventually compromise the ability of a SOV to perform its intended function. In recognition of the cost-effectiveness issue, attention was focused on remotely applied, completely nonintrusive techniques (i.e., ones that do not require physical access to the SOV, the addition of sensors or signal wires, the removal of power to the solenoid, or application of a special test signal). However, we found it necessary to depart from these restrictions to detect some well-recognized modes of SOV degradation. Clearly, tradeoffs must be made between disruptiveness to plant operations and the amount of information obtained.

Nine small enclosed type solenoid-operated valves from various manufacturers were obtained for use in the study (see Table 2). Three were nuclear-grade valves; one was used and the others were new. Five of the remaining six valves were new. The valve pressure ratings (60-2200 psi) are indicative of valve wall thickness and material of construction (brass or stainless steel), while electrical ratings (115-120 Vac or 125 Vdc) are established by solenoid coil construction (wire gage and number of turns). The nominal power consumption for all nine valves was in the range 10-20 W at rated voltage.

The Phase II study began with measurement of every conceivably useful electrical property of an SOV because, in many nuclear plant applications, the solenoid lead wires provide the only available link to the outside world where measurements can be performed. Details of these fundamental electrical measurements, along with the conclusions and possibilities for monitoring schemes that resulted from thoughtful consideration of each measurable property, will be available in the Phase II study final report (Kryter), but are not treated here for lack of space. However, the surveillance or diagnostic techniques studied (summarized in Table 3) and described in the following subsections were a natural outgrowth of the preceding investigations of fundamental electrical properties of SOVs. Each technique was, in fact, proposed to be specifically responsive to one or more of the proximate or root causes of SOV failure/malfunction cited in Bacanskas et al., (1987). The strengths and weaknesses of each technique as means for detecting and/or tracking the progression of age-related degradation were then evaluated.

Table 2. Solenoid-operated valves used in this study.

Manufacturer and Valve Identification	Design Power	Comments
ASCO NP8320A 185V S/N S14	125 Vdc	Nuclear-grade, rated for 115 psi
ASCO NP8320A 185V S/N S15	125 Vdc	Nuclear-grade, rated for 115 psi
ASCO NP8314C29E S/N K-62	125 Vdc	Used; nuclear; rated for 60 psi air
Skinner V5H30650 CTN	120 Vac	Rated for 150 psi
Skinner V5H38880	120 Vac	Rated for 100 psi
Skinner (no type no. avail.)	120 Vac	
ASCO 8262A214 S/N S94804	115 Vac	Rated for 2200 psi
ASCO 8210B26 S/N 91634S	115 Vac	Rated for 350 psi air
ASCO HTX831429 S/N 2445A	125 Vdc	Used; rated for 70 psi air

Table 3. An overview of SOV monitoring methods evaluated in this study.

Method	Degradation(s) or malfunction(s) addressed	Attributes	Promise for in-plant use
Measurement of SOV temperature, via coil resistance or impedance	Electrical failure of coil and degradation of elastomers resulting from prolonged operation at excessively high temperatures	• Nonperturbative to plant operations • No new sensors or signal cables are required • No permanent instrumentation required; can be applied as needed from a remote location • Applicable to ac- and dc-powered SOVs	High; ready for immediate use
Indication of valve position and change of state upon application of power, via change in coil impedance	Mechanical binding, sluggishness, or failure to shift as a result of worn or improper parts or the presence of foreign materials inside the valve	• No need for add-on sensors or signal cables • Valve position readout from a remote location • Static method does not disturb SOV	High; some additional development work required

Table 3. (continued).

Method	Degradation(s) or malfunction(s) addressed	Attributes	Promise for in-plant use
Indication of mechanical binding, by tracking changes in current and voltage at SOV pull-in and drop-out points	Mechanical binding and sluggish shifting caused by worn, swollen, or improper parts or the presence of foreign materials inside the valve	• Detects simultaneously degradation of magnetic or spring forces, and increase in frictional forces • No need for add-on sensors or cables or access to SOV • Applicable to ac- and dc-powered SOVs	Medium; further testing needed to ascertain cause of poor repeatability of test results
Indication of shorted coil turns or insulation breakdown, based on characteristics of electrical transient generated upon deenergizing a dc SOV	Electrical failure of solenoid coil, caused by high-voltage turn-off transients in combination with insulation weakened by prolonged operation at high temperatures	• Detects presence of defects within coil that cannot be revealed by other means	Low; useful for laboratory post-mortem tests
Indication of mechanical binding, by analyzing the time-varying characteristics of the inrush current accompanying application of electrical power to the SOV	Mechanical binding and sluggish shifting caused by worn, swollen, or improper parts or the presence of foreign materials inside the valve	• No need for add-on sensors, signal cables, or access to SOV • Information could be obtained as a result of everyday valve operation	Minimal; investigation of method abandoned early in the study
Indication of mechanical looseness within ac-powered valves, via electrical detection of humming or chattering of the plunger assembly (frequency decomposition of steady-state coil current)	Wear of internal valve parts, improper assembly, or replacement with incorrect parts	• No need for add-on sensors, signal cables, or access to SOV • Nonperturbative to plant operations	Minimal; investigation of method abandoned early in the study. Addition of miniature acoustic sensor to SOV might prove worthwhile

The first four monitoring methods listed in Table 3 received the majority of attention, and three of the four are described briefly in the pages to follow. For additional detail, the interested reader is referred to Kryter.

The final two monitoring techniques listed in Table 3 were examined briefly, then abandoned because initial results showed little promise for eventual development into effective degradation detection/diagnosis tools. These two monitoring techniques are not described here, but will be documented in Kryter.

Nonintrusive Measurement of Solenoid Coil Operating Temperature

As a result of the strong temperature dependence of chemical reactions [Arrhenius theory (Arrhenius, 1899)] that are constantly in progress and tend to break down organic compounds, excessive operating temperature can be expected to shorten the service life of both the solenoid coil insulation and critical elastomeric parts (O-ring seals and valve seats, in particular) within the valve. Arrhenius theory is widely used in extrapolating results obtained in equipment qualification tests performed to industry standards (IEEE, 1974; IEEE, 1985) to more realistic (as well as abnormal) plant conditions in order to obtain an estimate of qualified service life for the component. To be meaningful, the Arrhenius calculations must employ correct time/temperature profiles, especially the temperatures actually encountered during plant operation.

Measurement of true coil temperature using the method described here may help to improve the accuracy of such qualified service life predictions.

Operating temperatures higher than desirable may be caused by a number of circumstances such as application of higher than normal voltage to the solenoid coil (as when station batteries are being charged), restricted air flow or an elevated ambient air temperature at the valve location, elevated temperature of the fluid being controlled by the valve, or insulation breakdown within the solenoid coil winding (resulting in shorted turns, hence lowered electrical resistance and increased power consumption). Thus, one major cause of premature valve failure might be vastly curtailed if a simple, reliable, and inexpensive means were available to monitor (at will) the actual operating temperatures[d] of critical SOVs that may be inaccessible and thereby unmonitorable by traditional methods such as infrared pyrometry.

The test results presented below indicate that a promising means of fulfilling this need is to use he copper winding of the solenoid as a self-indicating, permanently available resistance thermometer. To do so requires only *in situ* measurement of coil dc resistance (or ac impedance), combined with prior knowledge of the temperature coefficient of resistivity for the copper winding and the coil resistance (or impedance) at a single known temperature (most conveniently, normal room temperature).[e] The dc resistance or ac impedance may be obtained, nonperturbatively, by measuring voltage applied to and current drawn by the solenoid coil, the former via a voltmeter and the latter via a clamp-on current transformer or a Hall effect probe (neither of which requires disturbance of the valve control circuitry). The resistance of the electrical leads connecting the SOV to the power source (and the possible variation of this lead resistance with changes in ambient temperature) is normally so small relative to the resistance of the solenoid coil that it can be ignored, but it could be measured

d. Monitoring actual SOV operating temperature is, of course, only one approach to increasing life; others are to increase heat dissipation capability (e.g., by the addition of cooling fins) or to curtail heat generation within the solenoid. Devices are available to automatically reduce solenoid operating voltage to a "holding" level (~70 V) once the solenoid has achieved pull-in on full voltage (125 V), the reduction taking place ~5 s after valve actuation.

e. It should be recognized that the temperature inferred by this method will be an *average* value for the volume occupied by the copper winding rather than the hottest spot within the solenoid coil. However, unless the coil has a localized fault (such as a shorted turn) this difference may not be very large: measurements by Bacanskas et al., (1988) indicate the "hot spot increment" is ~3 to 5°C.

separately and subtracted from the field-acquired data if such correction is thought to be warranted.

Principles of Operation

To serve as an accurate and remote sensor of local temperature, some electrical property of the copper solenoid coil must be demonstrated to have a stable dependence on temperature. Though not essential, it is desirable that the relationship of the electrical property and temperature be linear. It is also desirable that the relationship not be affected by changes in other conditions that may not be controllable in a real plant environment. Data to be presented in Kryter illustrate that several candidate parameters exist, each having advantages and disadvantages both in theory and in practice. Because SOVs are designed for two fundamentally distinct electrical environments-alternating current (ac) and direct current (dc)-the two types are treated separately below.

Dc-powered SOVs

Coil dc resistance is an easily measured quantity, satisfying the criteria given in the foregoing paragraph. Similar to platinum (a metal widely used as a resistance thermometer), copper has a stable and sufficiently large (~0.2 to 0.3% of value per °C, depending on copper purity) temperature coefficient of resistance (Handbook, 1955) to permit its use as the resistance element of a practical industrial thermometer. Figure 2 shows the resistance/temperature relationships obtained for two SOV coils having quite different dc resistances resulting from differences in wire size and the number of turns employed in their construction. These data, which are typical of results measured on nine separate valves designed for both ac and dc operation and produced by different manufacturers, were obtained by placing all nine valves in a thermostatically controlled oven and measuring their dc resistances at selected elevated temperatures. For each of the nine valves an extremely linear relationship (correlation coefficient >0.9997) was obtained over the temperature range of interest (20 to 170°C). It must be stressed that, regardless of the actual numerical value of the coil resistance, the temperature coefficient of resistance (expressed as a percentage of value) is sufficiently large (~0.3% per °C) to permit temperature measurement with better than ± 10°C accuracy using resistance measurement equipment of only modest (2 to 3%) accuracy. Temperature measurement accuracy of this order is surely adequate for indicating coil temperatures that exceed accepted operating limits established by qualification tests (IEEE, 1974; IEEE, 1985) or service life prediction curves based on Arrhenius reaction rate theory.

Figure 3 provides a laboratory demonstration of this technique. When the 125-Vdc SOV was first energized at the start of the test (ambient temperature was known to be 25°C), a coil resistance of 793.9 Ω was established by means of Ohm's Law from applied electrical potential and current readings. This single calibration point, in conjunction with the empirically determined slope of the resistance vs. temperature relationship for the copper wire used in this particular solenoid (3.41 Ω/°C), allows establishment of a temperature scale for the right-hand ordinate of the graph that exactly matches the directly measured resistance scale on the left-hand ordinate, namely,

$$T(°C) = \frac{R(\Omega) - 793.9}{3.41\ \Omega/°C} + 25.0$$

$$= -207.8 + 0.293R\ (\Omega)\ . \quad (1)$$

Once established, this linear scaling relationship permits direct interpretation of changes in SOV coil resistance accompanying altered test conditions in terms of their temperature equivalents. (Such dual scales are used in figures throughout this section as a reminder that resistance or impedance is the quantity directly measured.)

Placed inside a 2-ft^3 open-ended enclosure with only natural convection for cooling and no instrument air flowing through it, the SOV was first allowed to approach thermal equilibrium at a dc excitation voltage of 117 V. The coil temperature inferred from periodic measurement of dc resistance (o's in Figure 3) reached about 113°C (due to the deposition of about 14 W of resistive heat

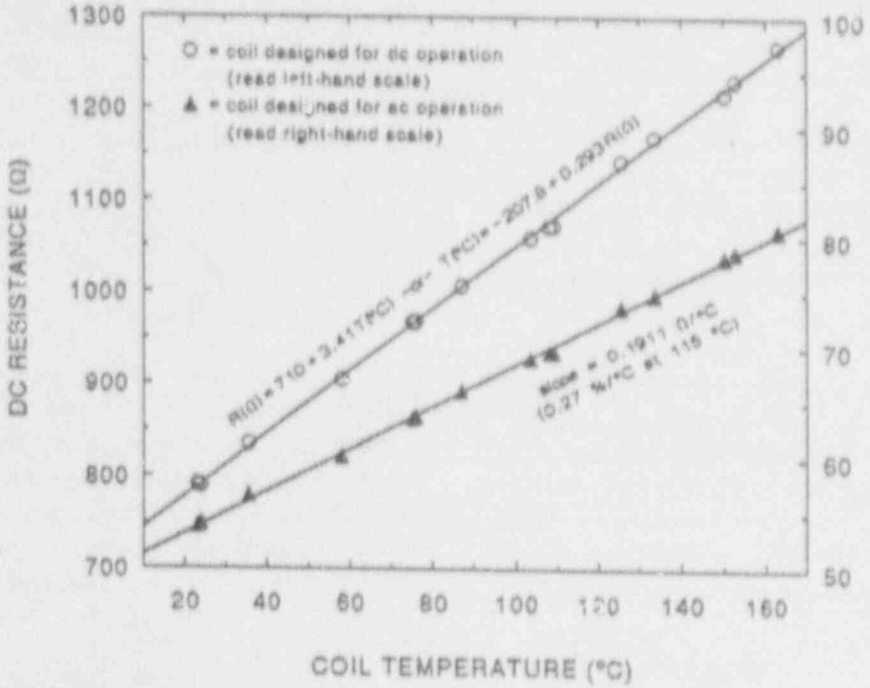

Figure 2. Linear variation of coil dc resistance with temperature for two different solenoid coils.

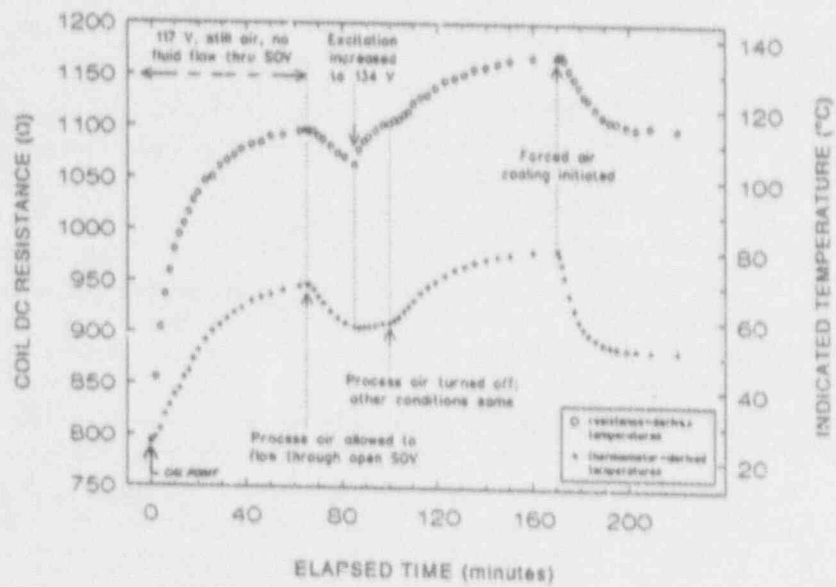

Figure 3. Changes in solenoid coil temperature, as inferred from *dc resistance*, brought about by altered electrical excitation, fluid flow, and environmental conditions.

within the coil) 65 min after initial power-up, whereas the temperature indicated at this time by a mercury-in-glass stem thermometer (+'s in Figure 3) placed in contact with the periphery of the potted solenoid coil was only ~70°C. This rather large (>40°C) difference is *not* indicative of a calibration problem, but, as has been shown by data acquired under more nearly isothermal conditions, results from a combination of high thermal resistance at the thermometer/coil contact point and the existence of a large temperature gradient between the center and the outer periphery of the coil, which is encapsulated in material having poor thermal conductivity. The point here is that, despite the considerable differences in the *absolute* temperatures indicated by the coil resistance and the mercury bulb, the two curves in Figure 3 clearly track each other in detail throughout the 220 minute duration of the test.

At t = 65 minutes, instrument air was allowed to flow through the already-open SOV. The cooling effect of the room-temperature air flowing through the brass valve body is evident in both curves, somewhat more so for the mercury bulb than for the coil, which is understandable because the bulb was positioned closer to the valve body than is the bulk of the coil. At t = 85 minutes, the dc supply voltage driving the solenoid was increased to 134 V to simulate a condition that might be encountered on a nominal 125-V dc power bus during times when station batteries are on charge. Note that the resulting temperature increase is immediate in the coil, but slower to appear at the mercury bulb; this is because the additional heat produced by the increased excitation voltage is generated instantaneously within the wire of the coil, whereas it must be transported to the coil periphery (where the mercury thermometer was located) by the relatively slow process of thermal conduction.

The 20-psi instrument air was turned off 100 minutes into the test, but the SOV remained powered at 134 Vdc and cooled only by natural convection within its enclosure. Over the next 70 min the indicated coil temperature rose to an asymptotic value of about 135°C,[f] and a commensurate rise was recorded on the mercury bulb thermometer. At t = 170 minutes a muffin fan positioned below the enclosure surrounding the SOV was turned on to draw air down past the valve at low velocity. The effect of this forced-air cooling is seen to be prompt reduction of both indicated temperatures-somewhat more rapid for the bulb thermometer than for the coil, which is explainable by the same reasoning (time lag due to conductive heat transport) offered for the difference in time response of the two curves when the electrical excitation was increased at t = 70 min. However, now the effect is in the opposite direction because heat is being removed from the outside of the coil rather than added from the inside.

The results of the test described above illustrate that using the copper winding of a dc-powered SOV as a resistance thermometer can provide meaningful real-time indication of altered excitation, environmental, and fluid flow conditions likely to occur in power plants from time to time that could result in unacceptably high SOV operating temperatures and hence shortened service life.

Ac-powered SOVs

Valves powered by alternating current offer many more electrical measurement possibilities, some of which may be useful for the inference of coil temperature. The list of candidates includes inductance, quality factor, (Henney, 1959) and impedance (expressed either as a vector magnitude or in terms of its real and imaginary orthogonal components). However, some measurable parameters prove more suitable than others, to be

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f. It should be noted that even this relatively high temperature, representing an increase relative to ambient of about 110°C, is still well below what is considered acceptable for the NEMA Class H coil insulation that has been used in recent years in nuclear-grade SOVs. (Class H insulation is rated for continuous operation at 160°C; the newer Class N for 175°C.) Had the ambient been 50°C, however-a condition that might be encountered in some areas within containment---Class H insulation would be at its operational limit.

described in detail in Kryter. As a result of this finding, temperature inference via inductance and quality factor was not pursued further; instead, attention was turned to the obvious analog of dc resistance: ac impedance.

By analogy to the measurement procedure previously illustrated for a dc-powered SOV, the inference of operating temperature for an ac-powered SOV would follow the same path except that impedance (measured at the power line frequency—60 Hz in the U.S., but often 50 Hz abroad) would replace resistance as the quantity obtained by applying Ohm's Law to the root-mean-square (rms) voltage and current measured at the SOV's electrical leads. However, two complicating factors arise that were not troublesome when dealing with dc valves: (a) diminished temperature coefficient of impedance and (b) variation of impedance with level of excitation. While neither poses an insurmountable obstacle to temperature inference, these troublesome factors do indeed diminish the accuracy of the temperature inference, as is evident from a comparison of the impedance-derived data of Figure 4 with the resistance-derived data of Figure 3. Space limitations do not permit further explanation of the ac phenomena responsible for these inaccuracies or the corrective measures available; the interested reader should consult Kryter. Impedance-derived temperature measurements are practical for ac-powered SOVs—as will be illustrated in the following section—although the results are likely to be less accurate than for dc-powered valves.

Practical Field Application

To illustrate the workability of this coil resistance/impedance method of SOV operating temperature inference in a real field environment, the rms voltage impressed upon and the rms current drawn by an ac SOV controlling the flow of refrigerant in a large chilled-water air-conditioning system were recorded at 100-s intervals over a 55-h period of system operation. The recorded voltages were divided, point by point, by the recorded currents to yield a curve of the absolute value of the coil's electrical impedance |Z| vs. time. Knowing the 60-Hz |Z| of the solenoid at a known temperature (593.9 Ω at 25.5°C) and assuming a value of 0.115%/°C (the median value for the five ac-powered SOVs tested) for the temperature coefficient of impedance, an equivalent temperature scale was affixed to the data plot, namely,

$$T(°C) = \frac{|Z|(\Omega) - 593.9}{(0.00115)(593.9)} + 25.5$$

$$= -844.1 + 1.464\,|Z|(\Omega) \,. \qquad (2)$$

The test results, split into two roughly 27-h time periods for clarity, are presented in Figures 5 and 6. The entire 55-h time period encompassed by these data was characterized by generally rising outside air temperatures (necessitating continuous compressor operation to maintain the chilled water temperature setpoint during the initial 28 h of the test) as well as rising ambient temperature at the SOV location (which is near the compressor, the essentially continuous operation of which caused the entire equipment room in which it is located to heat up). These weather and local environment temperature trends explain the generally rising SOV temperatures seen in the two figures. An environmental perturbation was introduced 24-h into the test just to see its effect: a blower that had been aiding cooling of the finned SOV was turned off. This change resulted in a prompt rise in SOV coil temperature (about 10°C), followed by a slow fall as nighttime brought cooler temperatures to the equipment room.

The second 27-h of operation (Figure 6) illustrate a new data feature, namely, cyclic compressor operation. At first glance, the inferred temperatures during these periods appear to have a great deal of scatter (main portion of figure), but when examined on an expanded time scale (upper left inset) what had appeared to be data scatter is revealed as repeated SOV heat-up during each cycle of compressor operation, followed by cool-down after each compressor shutoff.

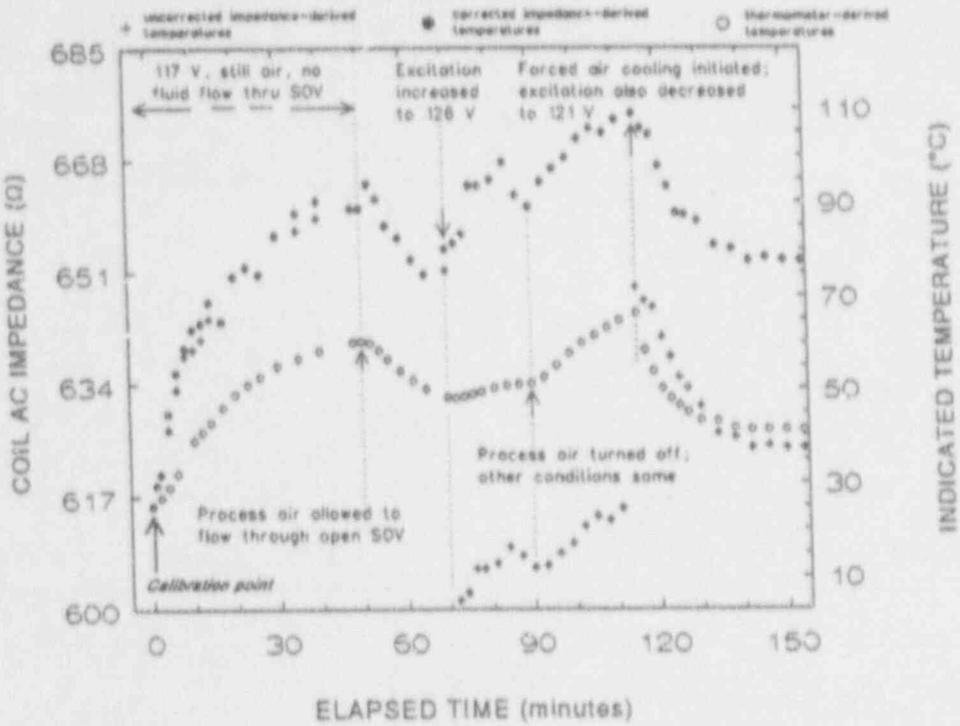


Figure 4. Changes in solenoid coil temperature, as inferred from *ac impedance*, brought about by altered electrical excitation, fluid flow, and environmental conditions.

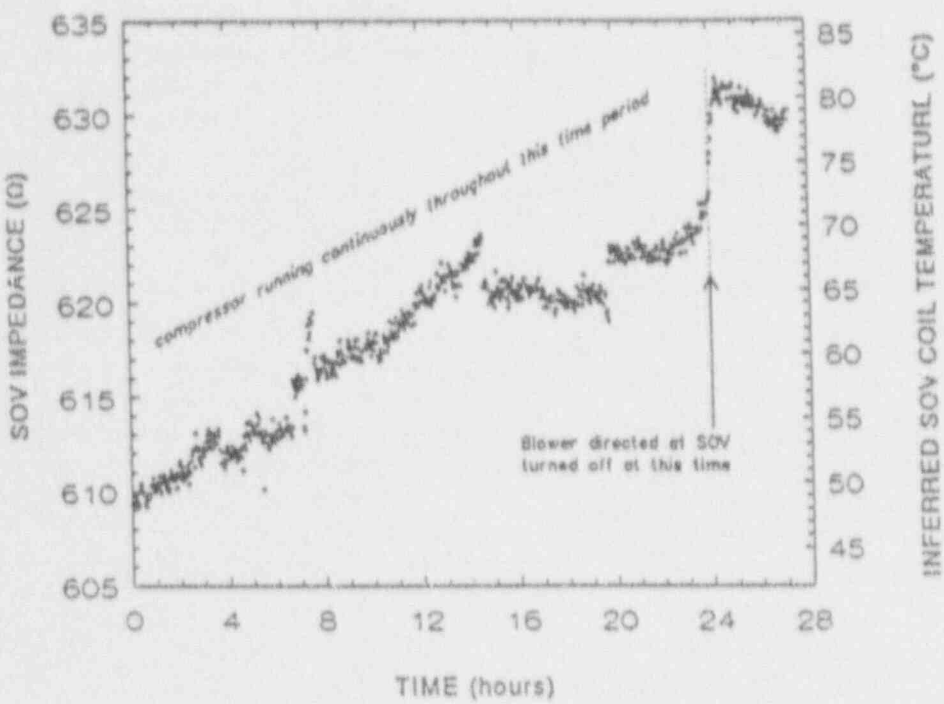


Figure 5. Weekend performance of an ac-powered SOV in a refrigeration system-first 27 hours.

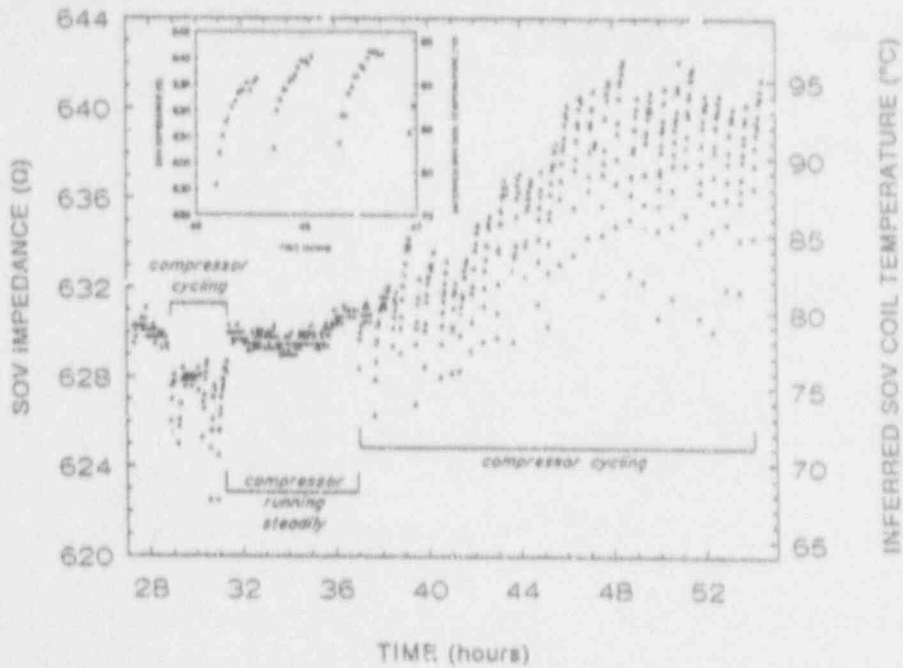


Figure 6. Weekend performance of an ac-powered SOV in a refrigeration system-second 27 hours.

The temperatures inferred for the SOV installed on the chilled-water system during the monitored period were never so high as to pose a threat to its continued operation. However, the data illustrate that, had some operational anomaly occurred that would have resulted in a dangerous rise in SOV temperature, it almost certainly would have been detected by this relatively simple, nonintrusive temperature measurement technique, which required only the connection of a voltmeter and a clamp-on current transformer.

Conclusions

Laboratory- and field-acquired test data illustrate that the true operating temperature of a solenoid-operated valve can be inferred simply, nondisruptively, and with satisfactory accuracy to detect temperature conditions that exceed accepted operating limits by using the copper winding of the solenoid coil as a self-indicating, permanently available resistance thermometer. This approach to monitoring the temperatures actually "seen" by SOVs during the course of plant operation and the temperature changes introduced by altered environmental or process conditions might be used to advantage by those concerned with prediction of qualified life. The method has a number of merits, including (a) there is no need for an add-on temperature sensor, (b) the true volume-averaged temperature of a critical—and likely the hottest—part of the SOV (namely, the electrical coil) is measured directly, (c) temperature readout can be provided at any location at which the SOV electrical lead wires are accessible (even though remote from the valve), (d) the SOV need not be disturbed (whether normally energized or deenergized) to measure its temperature *in situ*, and (e) the method is applicable to all types of SOVs, large and small, ac- and dc-powered. From a standpoint of prediction of qualified life, the principal shortcoming of the method is that the coil's *volume-averaged* temperature—rather than its *peak* temperature—is the quantity measured, although this difference may not be substantial (Bacanskas et al., 1988).

This method is useable in its present form, although additional development work could improve the accuracy of temperature indications derived from impedance measurements on ac-powered SOVs. The hardware implementation would depend on plant needs, but could take the form of a permanently installed, computerized data logger or hand-held, walk-around instrumentation for use on an as-needed basis.

Indication of Valve Position, Change of State, and Mechanical Binding

The technique presented in the preceding section addresses the aging issue of reduced solenoid coil and elastomer life resulting from prolonged operation at excessive temperature, an acknowledged problem in the nuclear industry. Here we address another well-known problem, that of the solenoid plunger (core) "frozen" in position so that it refuses to move when the SOV is called upon to change states, whether normally energized or normally deenergized. Such a condition may arise from age-related changes in the physical properties of the elastomeric seats (the seat material may become sticky, thus holding the plunger fast), thermally degraded and deformed shaft seal O-rings, faulty valve assembly, the use of incorrect replacement parts, or the presence of internal contaminants such as metal chips, dirt, paint, thread sealant, desiccant, and hardened lubricant (Verna, 1991; Bacanskas et al., 1987; Ornstein, 1991b) that can interfere with free movement of the core within the core tube. Depending on the nature of the information to be obtained, this general technique can be implemented either as a nonintrusive static measurement of coil impedance or as a dynamic measurement, the latter requiring application of a special ramped-voltage profile to the SOV. Only the dynamic measurement will be treated here. (Kryter will describe those and the static tests as well).

Dynamic Tests (Intrusive to Plant Operations)

For *ac-powered* SOVs, dynamic tests that verify both the presence of plunger movement upon application of power and also the absence of binding throughout the transition from unpowered to powered state are also possible if one is able to temporarily remove the 117-V, 60-Hz ac power applied to the SOV and substitute an ac power supply capable of ramping up and down more or less uniformly with time. To see how this is possible, Figure 7 shows the manner in which the current drawn by the solenoid changes with time when the voltage applied to the coil leads is slowly ramped down in a linear fashion from about 135 Vrms to 5 Vrms over a period of 50 s.[g] The nonlinear behavior of the current over the initial 33-s reflects the variation of impedance caused by eddy current and hysteresis losses, whereas the behavior over the final 15-s (t = 35 to 50-s) reflects the approximately constant and much lower impedance produced by the plunger being somewhat outside the solenoid coil.

The shift of the plunger in its guide tube is clearly seen at about t = 33 s as an abrupt change in the current drawn by the SOV as a result of the change in impedance that accompanies plunger movement. The current/voltage (i.e., impedance) characteristics obtained during this brief time of transition reveal not only the overall movement of the solenoid core but also any tendency it may have to bind within its guide tube during the valve state transition, as will be illustrated later. Before doing so, it is instructive to show that the ramp rate can be increased considerably in order to improve time definition and thereby the capture of detail. If, for example, we take an ac-powered SOV in good condition and make provision to ramp its excitation voltage over the full operational range (0 to 135 V) in a rather short time interval (200 ms), as shown in Figures 8 and 9, then a marked rise in impedance, |Z|, and a corresponding fall in current will be evident when the valve changes state (t = 125 ms in Figure 8). If, on the other hand, the plunger is jammed in position and cannot move (the condition purposely created in Figure 9), the *lack* of impedance change and the continued monotonic rise of current throughout the ramp-up will clearly signal a lack of plunger movement.

g. The downward-going voltage ramp was produced by driving the shaft of a Variac autotransformer at a constant rotational speed (1 rpm/min) with the aid of a gear motor, thus producing a ramp rate of ~2.75 V/s. The gear motor's direction of rotation could be electrically reversed to produce an upward-going voltage ramp having the same rate of change with time.

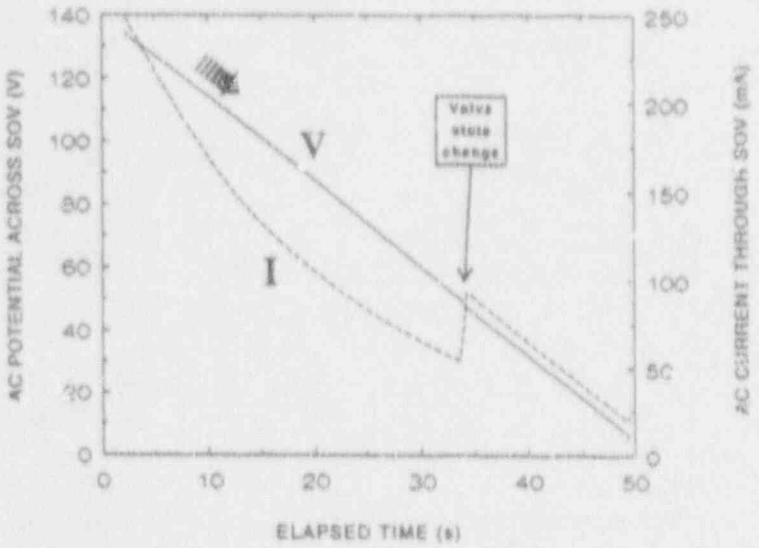


Figure 7. The nonlinear variation of SOV current during a linear ramp-down of voltage over a 50-s period is a reflection of the solenoid's nonconstant impedance except at low levels of excitation.

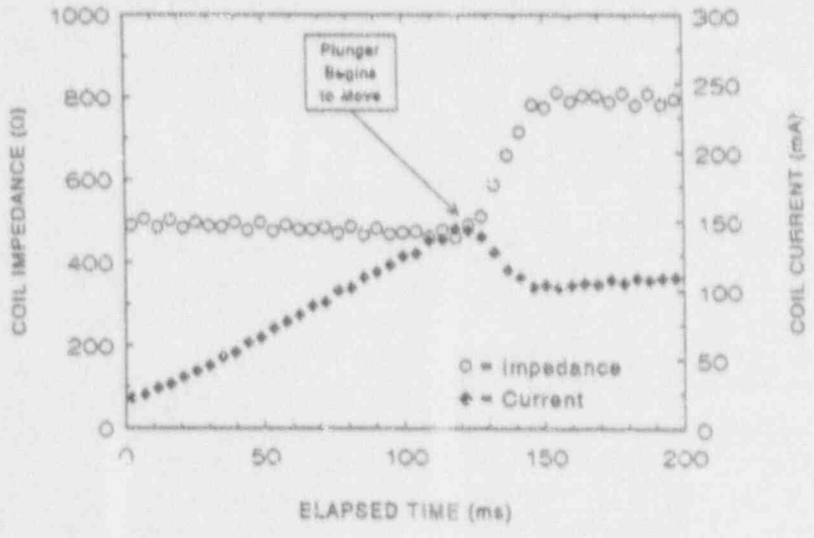


Figure 8. Current and impedance discontinuities occurring during linear voltage ramp-up over an interval of 0.2 s verify plunger movement in a normally operating ac-powered SOV.

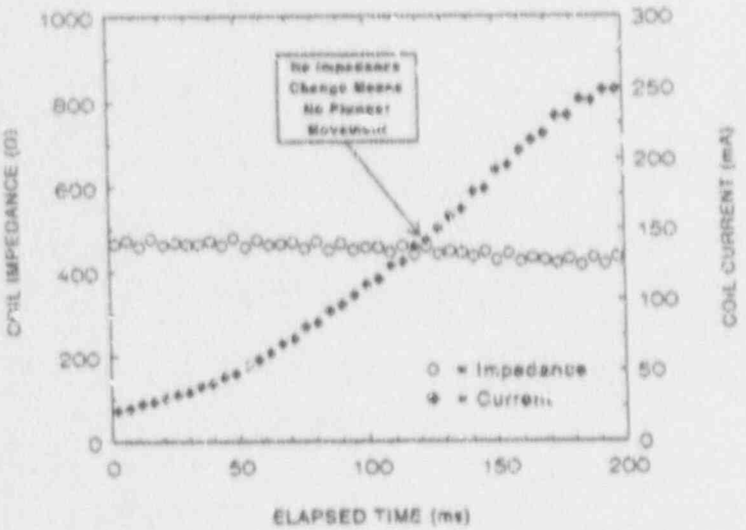


Figure 9. Lack of impedance change and current discontinuity during linear voltage ramp-up over 0.2 s interval warn of no plunger movement in an ac-powered SOV in which the plunger was purposely jammed.

As mentioned, the ramp-up technique just described is also useful in diagnosing mechanical abnormalities less severe than a completely immobile plunger (e.g., a plunger whose movement within its guide tube is impeded by the presence of foreign substances such as dirt or oil) (Verna, 1991 and Ornstein, 1991b). Figure 10 illustrates the current/voltage relationship obtained for an ac-powered SOV when its excitation is ramped up over its full voltage range in a time period of a few tens of seconds. Curve A (solid) was obtained with a clean, normally assembled valve, whereas curve B (dashed) was obtained from a valve of identical construction except that the core and the interior of its guide tube had been coated with a thin layer of thickened, sticky liquid shellac during assembly to simulate the mechanical binding which has been experienced from polymerized lubricants or contamination from use of excessive thread sealant or unfiltered air supplies (Verna, 1991; Bacanskas et al., 1987; Ornstein, 1991b). The curves are seen to track each other extremely well in the low- and high-excitation regions that represent static plunger positions (i.e., the purely electrical properties are quite reproducible), but they differ markedly in the middle region of the graph at which the valve shifts state. The "normal" SOV is seen not only to shift at a lower level of excitation than its "gummy" counterpart but also to exhibit some instability as it begins to execute the change of state (the valve was audibly buzzing at this time during the ramp-up), whereas the SOV having the internal contamination exhibits a smoother (damped) transition once sufficient magnetic force is developed to overcome the resistance to movement produced by the sticky shellac.

Similar results are obtained if the excitation is ramped in a *de*creasing rather than *in*creasing direction, namely, a pronounced difference between the voltage/current transition points for the normal and sluggish SOVs—in addition to lower V and I critical values than were obtained for the upward-going ramp. Performance of both upward and downward ramp tests thus establishes a set of four pull-in and drop-out critical points—identified by "TP" (trendable parameter) symbols in Figure 10—which can be trended over time to provide early indication of mechanical binding, should it occur.

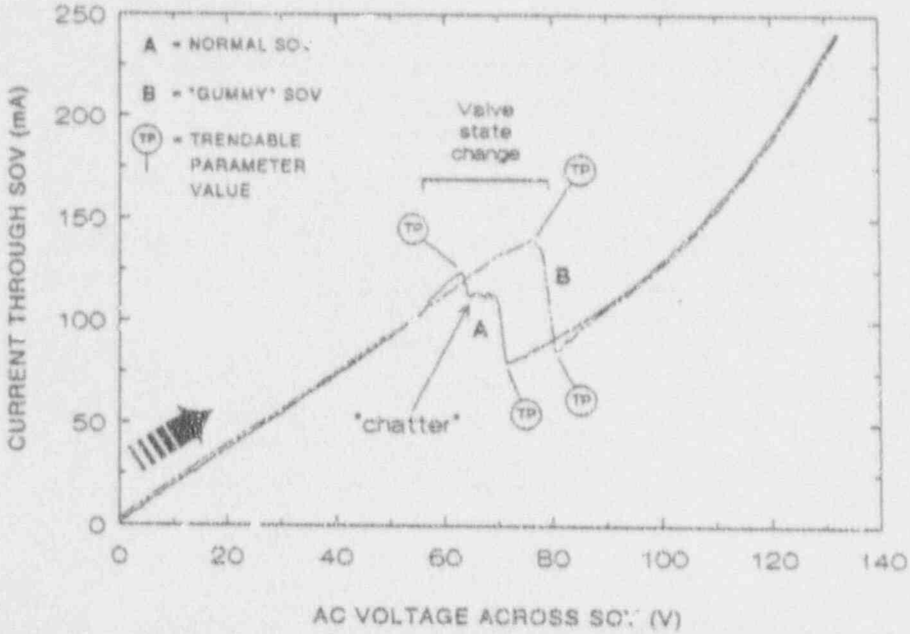


Figure 10. Ac current/voltage characteristics obtained during a slow excitation ramp-up, showing sensitivity to the presence of binding.

Conclusions

Positive indication of current SOV state and an ability to change state (dynamic response to an "open" or "close" command) is often possible using either static or dynamic ac measurement techniques. The merits are similar to those cited for temperature inference via coil resistance/impedance, namely, (a) there is no need for an add-on sensor, (b) valve state readout can be provided at any location at which the SOV electrical lead wires are accessible (even though remote from the valve), and (c) the SOV need not be disturbed (whether normally energized or deenergized) if the static method is used. The shortcomings are that (a) the method is applicable only to *ac-powered* SOVs and (b) arrangements must be made to power the valve from a special ramp-voltage power supply if the dynamic method is used. The methods, while useable in their present form, would need additional development work to provide a means of obtaining desired dynamic information from instantaneous switch-on tests rather than the 200-ms ramps described here. In any case, the preferred hardware implementation would depend on plant needs.

Indication of Shorted Turns/Insulation Breakdown Within the Coil

On the basis of recorded operational experience and opinions cited in the Phase I study, (Bacauskas et al., 1987) a considerable number of coil open- and short-circuit SOV failures may be attributable to high-voltage transients generated by the abrupt collapse of a dc solenoid's magnetic field as a result of circuit interruption. SOVs operating in elevated-temperature, high-humidity environments are particularly susceptible to failure from this stressor because the insulating properties of the coil varnish, encapsulating material, and electrical lead wires are weakened under these severe environmental conditions. The phenomenon is quite similar to the generation of the high-voltage pulses of short duration feeding the spark plugs in an automobile's ignition system, which is composed of an ignition coil whose primary winding current is turned on and off rapidly as a result of periodic closing and opening of the breaker points within the distributor. In both the automobile and the SOV, energy that is stored in the magnetic field of an inductor (the coil) as a result of being energized with dc current is converted at the instant of circuit interruption to an electrostatic potential (voltage) that appears across the distributed capacitance of the coil (and, in the case of the automobile, across the secondary winding).

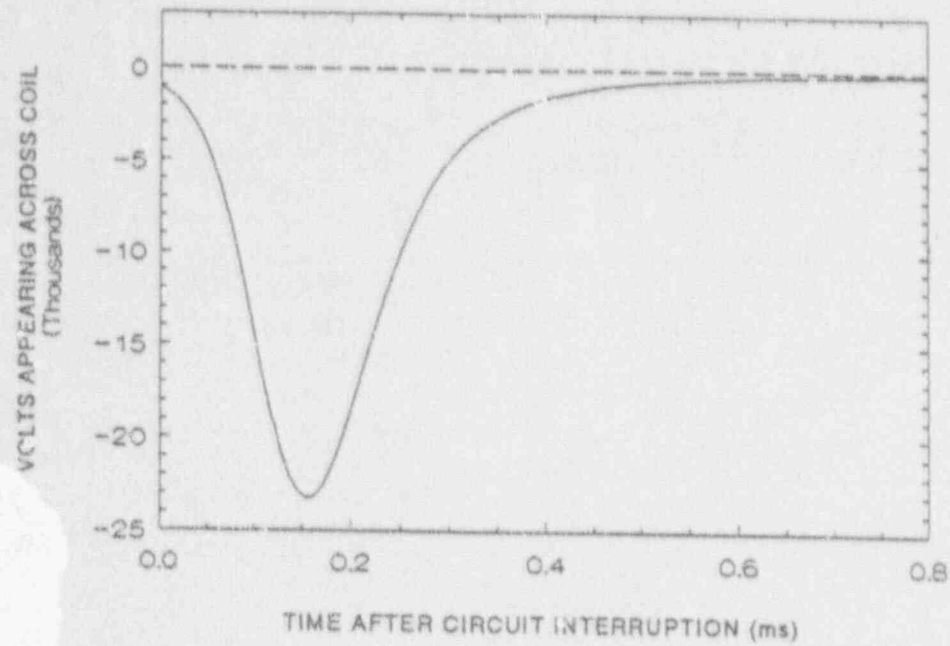
It stands to reason that such high-voltage transients within the solenoid coil may produce damage by puncturing the insulation covering the copper wire of the winding at any weak points that may exist. Once this happens, the area surrounding the point of puncture becomes carbonized, thereby lowering the insulation resistance and so assuring that the same spot will be the site of electrical arcs on future valve deexcitations. Once started, this process produces rapid degradation of the insulation, which will ultimately burn away and leave a permanent turn-to-turn or layer-to-layer short circuit. According to industry sources the consequences of such a short circuit within the coil are not especially serious for dc solenoids, which will continue to function normally with a considerable number of shorted turns, but ac solenoids are not so tolerant in this regard since, in effect, a step-down transformer is created by the shorted turn. The result is large current flow through the shorted turn, intense localized heating, and eventual thermal (followed by electrical) destruction of adjacent portions of the coil ("coil burnout").

Figure 11 illustrates that very high voltages can indeed be generated at the coil terminals of a dc-powered SOV as a result of power turn-off (in this example, almost 25 kV for a duration of ~100 µs). However, we do not believe that inductive surge is, or need be, a major problem in connection with SOV service life for the following reasons:

- *Coil insulation is tough.* Hundreds of transients of the magnitude shown in Figure 11 were produced during the course of this study but they failed to cause any shorted turns in a variety of coil types having NEMA Class H insulation. (It must be acknowledged, however, that our tests were performed in a mild operating environment rather than under LOCA conditions.) The new Class N insulation is claimed to have electrical properties superior to Class H materials, particularly under high humidity conditions.[h]

- *Transient suppression devices are readily available.* Inexpensive metal-oxide varistors (MOVs) can be placed across the coil leads to effectively absorb the energy released by the sudden collapse of the solenoid's magnetic field upon turn-off. (See Figure 12 in comparison to Figure 11.) The magnitude of the high-voltage transient produced upon deenergizing the SOV is thereby greatly reduced (~600 V rather than ~3,000 V), although the transient's duration is correspondingly extended (~20 ms rather than 0.8 ms). The positiveness and rapidity of solenoid release do not, however, appear to be noticeably affected by the presence of the transient-suppressing device (tests demonstrated an ability to cycle an MOV-protected SOV at a rate of at least 50 actuations per seimd). Hence, no major change in SOV operating characteristics would be expected by the addition of such a protective device.

h. Telephone discussion, John Shank and Frank Fry (Automatic Switch Company) with Robert Kryter (ORNL), November 17, 1988.



Large "flyback" voltages generated when current to a 125-V dc SOV was interrupted abruptly by means of a mercury-wetted contact relay.

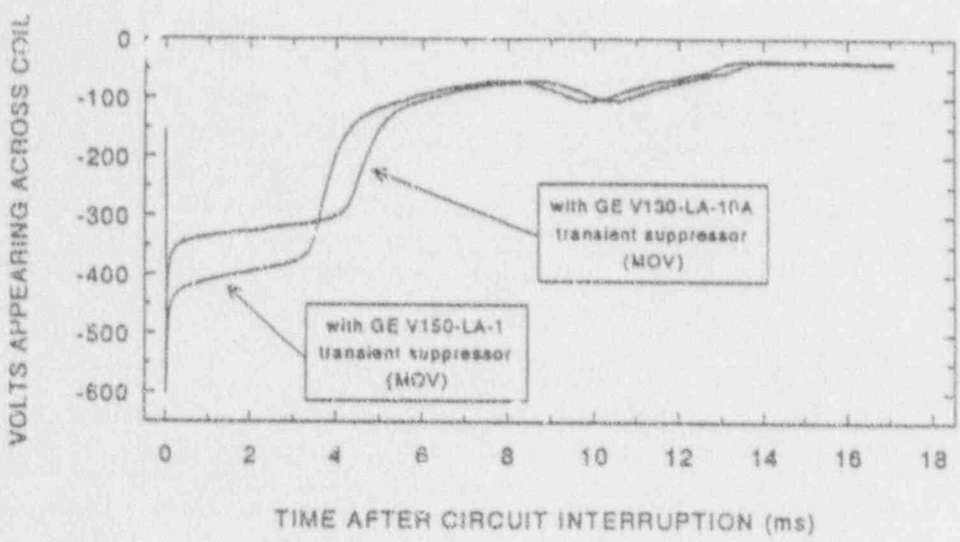


Figure 12. High voltage transient suppression provided by two varieties of metal-oxide varistor placed across the SOV terminals. Same valve and experimental setup as Figure 11.

- *Potentially damaging voltage differences are not as great as they might at first appear.* The voltage differences appearing between adjacent turns and even between successive coil layers are only a small fraction of the total inductive surge voltage appearing across the coil terminals as a result of magnetic field collapse. Industry sources[g] state that while turn-to-turn short circuits can occur as a result of imperfections in the lacquer insulation coating the copper coil wire it is quite rare that this causes any problem because the turn-to-turn voltages are too small to jump the interconductor air gap. It is also stated[i] that layer-to-layer short circuits are practically unheard of, since most coil manufacturers place additional dielectric material between successive coil layers during the winding process.

- *Circuit interruption is likely to be slower in practice.* The manual switch or relay employed in a real plant will probably not break the circuit as rapidly as the mercury-wetted contact relay used in these laboratory tests; hence the flyback voltage generated in practice is unlikely to be as high as that shown in Figure 11.

- *Dc SOVs are tolerant of shorted turns.* Industry sources cited previously state that shorted turns present a serious operational problem only for ac-powered SOVs, whereas the production of a large flyback voltage upon circuit interruption is a characteristic of dc-powered SOVs.

CONCLUSIONS

Additional research will be needed to advance the understanding of the various electrical phenomena that occur in coils having defective windings before a practical SOV coil diagnostic system can be devised, should it be decided that one is truly needed. Results presented in Kryter also show that it may be impractical to perform coil diagnosis *in situ* from a location distant from the SOV using any sort of measurement technique that relies on free oscillation upon deexcitation, owing to the fact that the capacitance of the power leads connecting the valve to its power source is likely to be several orders of magnitude larger than the distributed capacitance of the coil itself, thereby clouding interpretation of the measurement. If, on the other hand, the SOV coil power leads can be accessed directly, the flyback transient characteristics may provide an indication of coil insulation breakdown and shorted turns that is far more sensitive than simple change in dc resistance.

CLOSING REMARKS AND RECOMMENDATIONS

Since our review of the technical literature did not reveal any degradation-monitoring techniques-either presently in widespread industrial use or under development-that are oriented specifically to SOVs, this Phase II study was necessarily more inventive than evaluative in nature. The results of the work, however, have revealed a number of SOV monitoring methods based on electrical properties that are potentially useful for ascertaining operational readiness through the measurement of degradation-sensitive performance parameters that may be trended over component life. In so doing, early indications of malperformance that may foreshadow more serious malfunctions or failures of these devices can be obtained, thus aiding in the scheduling of maintenance activities or timely replacement of the entire valve.

The attributes of the monitoring techniques receiving the majority of attention during this study are listed in Table 2 at the beginning of the paper. The *strengths* shared by most of these monitoring techniques include the following:

- The parameters measured have been demonstrated to be sensitive to historically important SOV failure modes and causes

i. Telephone discussion, John Shank and Frank Fry (Automatic Switch Company) with Robert Kryter (ORNL), November 17, 1988.

- The measurement techniques employed are, in the main, minimally disruptive to plant operations
- The diagnostic tests can be performed at a location remote from the SOV with no attendant degradation in sensitivity
- The SOV can often be made to serve as its own sensor, thereby eliminating the need for additional instrumentation and signal wires.

Weaknesses of the monitoring techniques examined include the following:

- Some of the techniques are disruptive to plant operations (requiring, for example, momentary change of valve state or short-term substitution of a special electrical power supply)
- The monitoring techniques described may not cover all conceivable types of aging-related degradation and may not be universally applicable to SOVs of all type, size, and construction.

Also, as a result of programmatic priorities and limited resources, some planned tests whose results might have yielded additional insights were never performed, but are recommended for serious consideration in any future investigation in this area. Among these are

- Further study of the reproducibility of SOV pull-in and drop-out critical points and their sensitivity to realistic, progressively worsening problems
- Further study of the coil impedance change accompanying plunger motion.

Kryter will suggest particular tests that merit consideration.

A major criticism that can be leveled at the overall investigation is that it doesn't go far enough-that is, that the monitoring techniques studied were demonstrated only in a controlled laboratory environment, using a small population of unaged SOVs of substantially similar construction and a limited number of implanted (rather than naturally occurring) defects. Accordingly, it is recommended that in a future study

- The performance monitoring techniques be field tested using a larger population of both new and naturally aged SOVs that would be likely to display one or more varieties of degraded performance;
- The techniques be refined and adapted as necessary to permit their use in a real plant environment. (For example, devise a means for applying a dc interrogation signal to an ac-powered SOV so that its temperature could be measured accurately even in a normally energized state. Or devise a means for ascertaining free plunger movement upon SOV turn-on without need for a special ramped-voltage power supply.)

Finally, there would be added benefit if the findings of this SOV investigation could be applied and utilized creatively, wherever practical, in surveillance and testing programs directed at safety-related electrical components of nuclear power plants other than solenoid-operated valves.

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Toledo Edison Program and Diagnostic Experience with Pneumatic Operated Valves

John H. Hayes
Toledo Edison

PROGRAM ELEMENTS

The key elements to establishing and maintaining a successful air-operated (pneumatic) valve testing program are management commitment, plant personnel support, and adequate procedures. Additionally, the program is most effective if as much of the testing, corrective maintenance, and preventive maintenance as possible be performed during power operations. This prevents these activities from being critical path during outages.

PROGRAM HISTORY

The Toledo Edison air-operated valve (AOV) task force was formed March 1988. The responsibilities of the task force were assigned as follows:

1. Support resolution of immediate AOV problems
2. Analyze and evaluate AOV problems
3. Recommend program improvements.

These tasks were segregated into short-term and long-term projects.

SHORT-TERM ACTIONS

The evaluation of known valve problems was given a high priority. Examples of the types of problems and the actions taken are as follows:

- Turbine Bypass Valves

  These valves had experienced a number of problems. With the implementation of an AOV diagnostic testing program, several actions were taken to improve the performance of these valves. Specific maintenance and calibration procedures were developed. Spare par[illegible] upport was established to ensu[illegible] ct replacement parts. Valves with incorrect parts installed were rebuilt with correct parts, such as when an improper positioner CAM was identified for these valves. It was also identified that these valves were improperly sized. Functional testing and preventive maintenance were developed and initiated.

- Component Cooling Temperature Control Valves

  These valves experienced sticking and would not maintain a fail-safe position. The root cause was determined to be related to the design of the valves. Both the internals and the valve controls were redesigned, and a study was conducted to identify replacement valves more appropriate to the application.

- Main Feed Startup and Control Valves

  For these valves, specific maintenance and calibration procedures were developed. The controls were rebuilt and functional testing and preventive maintenance were established, based on the results of the diagnostic testing.

LONG-TERM ACTIONS

Several action items were identified to enhance and improve the operation of AOVs. An AOV data base was established to identify the valves included in the AOV program. A file including each individual valve's operating and maintenance history was created. The AOVs were categorized based on the importance of the valve to the operation of the plant. A design review of valves experiencing chronic failures was initiated. Preventive maintenance activities on non-critical valves were suspended. A diagnostic testing system was developed. "Time directed" preventive maintenance was established. Quick disconnects were installed to allow ease of diagnostic testing. Predictive maintenance using diagnostics was initiated and is being incorporated into the inservice testing of the AOVs. Personnel training and qualification programs have been established.

VALVE GROUP

A technical "Valve Group" was established in Performance Engineering to provide a uniform approach to all issues related to valves. This group is responsible for operation, testing, and maintenance related concerns for AOVs, solenoid-operated valves, motor-operated valves, check valves, manual valves, and relief valves.

USES OF DIAGNOSTIC TESTING

The AOV diagnostic testing equipment has been used for various purposes, other than the main use of diagnosing AOV operating problems. The uses include the following:

- Surge line deflection measurement
- Steam generator pressure transducer diagnostics
- Solenoid valve testing
- Main feedwater control valve testing at 100% power
- Heater drain valve tuning
- Inservice testing
- Design verification
- Calibration
- Predictive maintenance
- Investigation of check valve slamming
- Other, as creatively applied by various users.

RESULTS

The development of the AOV task force has led to improvements in the operation and reliability of AOVs and has had very positive results for Toledo Edison.

An Integral Part of an Integrated Preventive Maintenance Program

Brian J. Ferguson,
Bruce Nuclear Generating Station 'A'

ABSTRACT

This paper outlines the evolution of the Integrated Preventive Maintenance Program from the Reliability Centered Maintenance pilot program at Bruce Nuclear Generating Station (NGS) 'A.' The Integrated Preventive Maintenance Program provides the means for taking control of the Bruce NGS 'A' Preventive Maintenance Program.

In addition, the development of the Bruce NGS 'A' Air-Operated Valve Preventive Maintenance is reviewed.

INTRODUCTION

Until the late 1980s performance of Ontario Hydro's nuclear units met or exceeded our standards and compared favorable with world standards. Since that time we have seen a gradual increase in our average nuclear incapability factor, as illustrated by Figure 1, a plot of average nuclear incapability factor versus year for all of our nuclear units.

In response Ontario Hydro has developed a Maintenance Strategy to improve the conduct of maintenance at all of its nuclear facilities. The strategy focuses our improvement efforts in four main areas:

- Resource management
- Human performance
- Outage management
- Equipment performance.

Bruce Nuclear Generating Station (NGS) 'A' has taken the lead in equipment performance improvement for the Corporation by developing and implementing a Preventive Maintenance Optimization Program. This program has two components:

1. Reliability Centered Maintenance (RCM) analysis
2. Integrated Preventive Maintenance Program (IPMP).

Together they will provide effective control over our existing Preventive Maintenance (PM) program.

This paper will describe the Integrated Preventive Maintenance Program and the development of an Air-Operated Valve (AOV) Preventive Maintenance Program within the IPMP.

INTEGRATED PREVENTIVE MAINTENANCE PROGRAM

Improving equipment performance requires us to move away from performing "breakdown" or corrective maintenance and to focus more maintenance effort on preventive maintenance, where we can detect degrading performance and repair or replace critical components before failure occurs.

The state of Preventive Maintenance in our plant (and I suspect many others) prior to this program can be described diagrammatically by Figure 2, i.e., the PM program is inconsistent, lacks a technical basis, and many PM tasks are neither applicable nor effective.

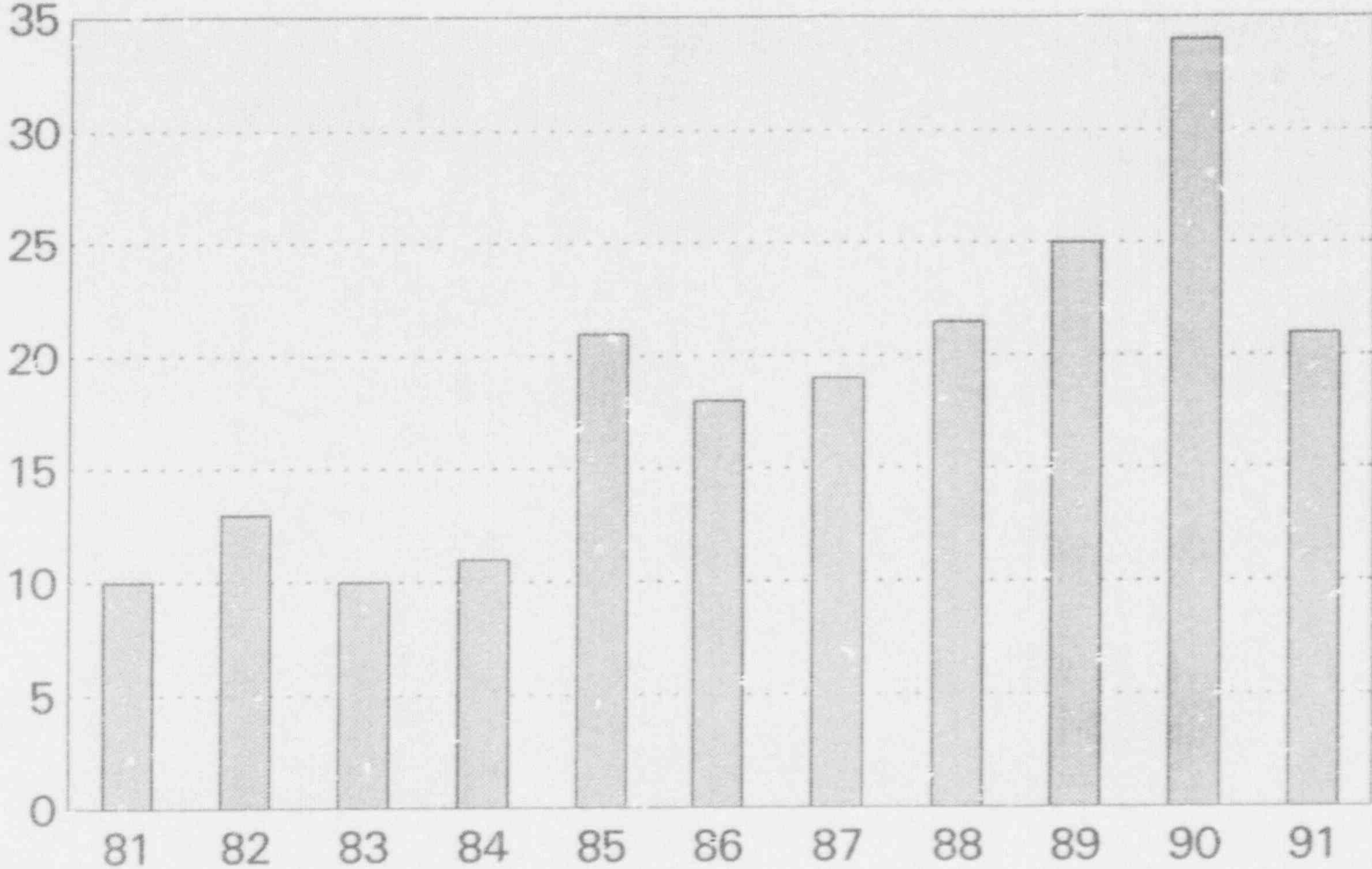

Figure 1. Nuclear incapability factor (ICbF%) (does not include retubing).

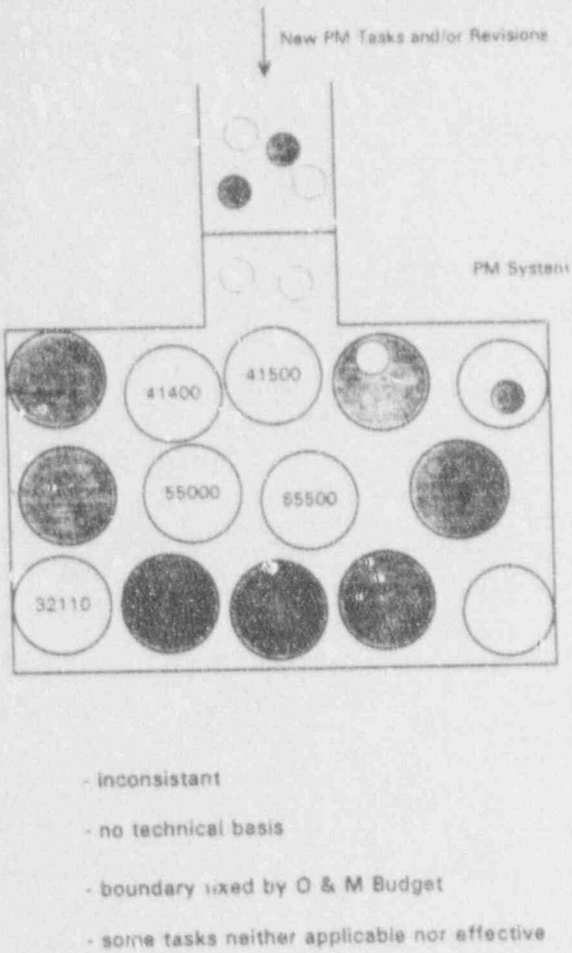

Figure 2. Current state of PM program.

The PM improvement plan began as a pilot to evaluate the use of Reliability Centered Maintenance (RCM) as a PM program optimization methodology for four station systems (main moderator, moderator auxiliaries, dc power, and moisture separation/reheat). All analysis was carried out using the EPRI "RCM Work Station." This software was made available to Ontario Hydro as a "Beta" test version and is now in general release to EPRI members.

A short time after the project started, it became apparent to us that RCM analysis alone was not going to accomplish the desired PM program optimization because we did not have proper control of our existing PM program, as illustrated in Figure 2. Consequently, it was necessary for us to develop the concept of the Integrated PM Program in order to provide a framework within which PM program improvement could occur. This integrated approach was necessary to ensure successful implementation of RCM analysis and to maintain the program over the life of the Station.

The IPMP has two components:

1. A procedural framework as shown in Figure 3
2. The PM Engineering Organization as illustrated in Figure 4.

The five procedures completely describe how the program will operate throughout the Station's life cycle by defining program strategy, staff roles and responsibilities, RCM methodology, current plant predictive maintenance capabilities and feedback and trending of PM program activities.

The PM Engineering Organization establishes single point responsibility for the total PM program. This group has responsibility for RCM analysis, PM task administration and feedback, trending and analysis of equipment performance. Performance trending is the key to maintaining a "living" PM program.

Figure 4 shows the total resource requirements for plant wide implementation of the Integrated Preventive Maintenance Program. The net result of the IPMP will be an improved PM program with the three enhancement recirculation loops as shown in Figure 5 (the Khan model). This will force all PM tasks within the program to undergo constant review to ensure their applicability and effectiveness. The pilot phase of the IPMP (i.e., analysis and implementation of the four previously noted systems) is expected to be completed by July 1992. Full plant implementation will be completed by the end of 1994.

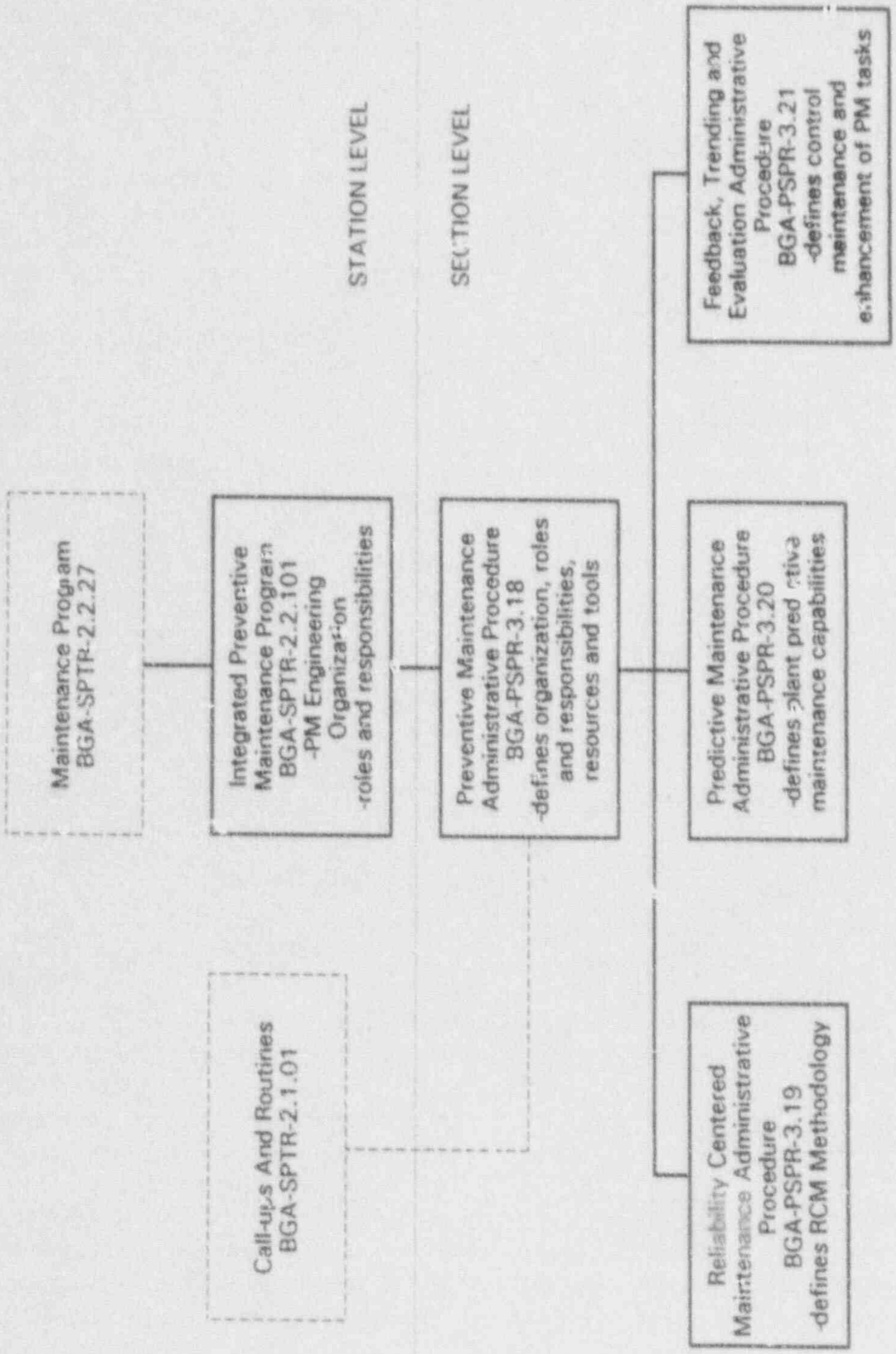

Figure 3. Integrated Preventative Maintenance Program procedure hierarchy.

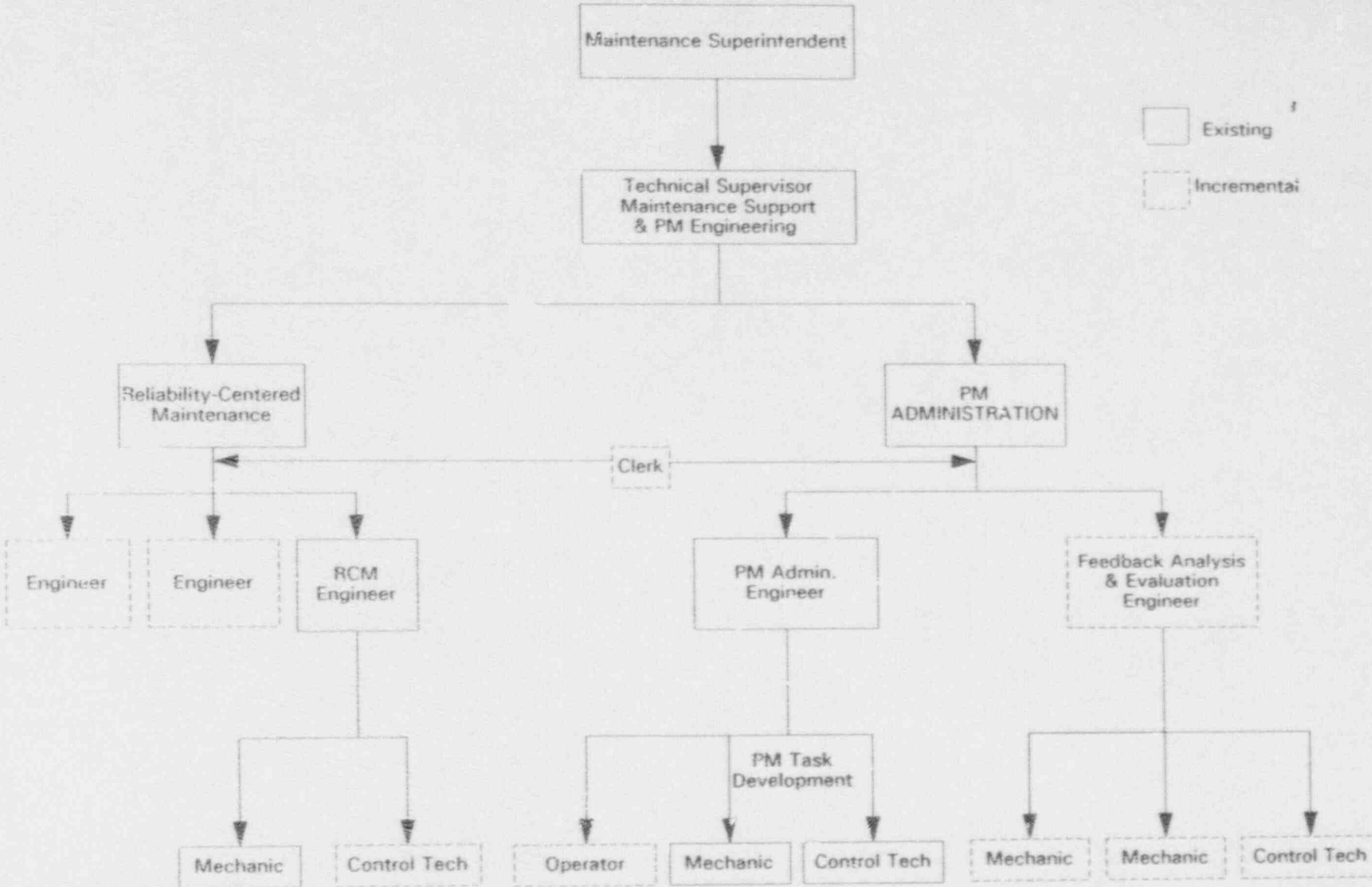

Figure 4. Proposed Bruce NGS 'A' PM Engineering Organization.

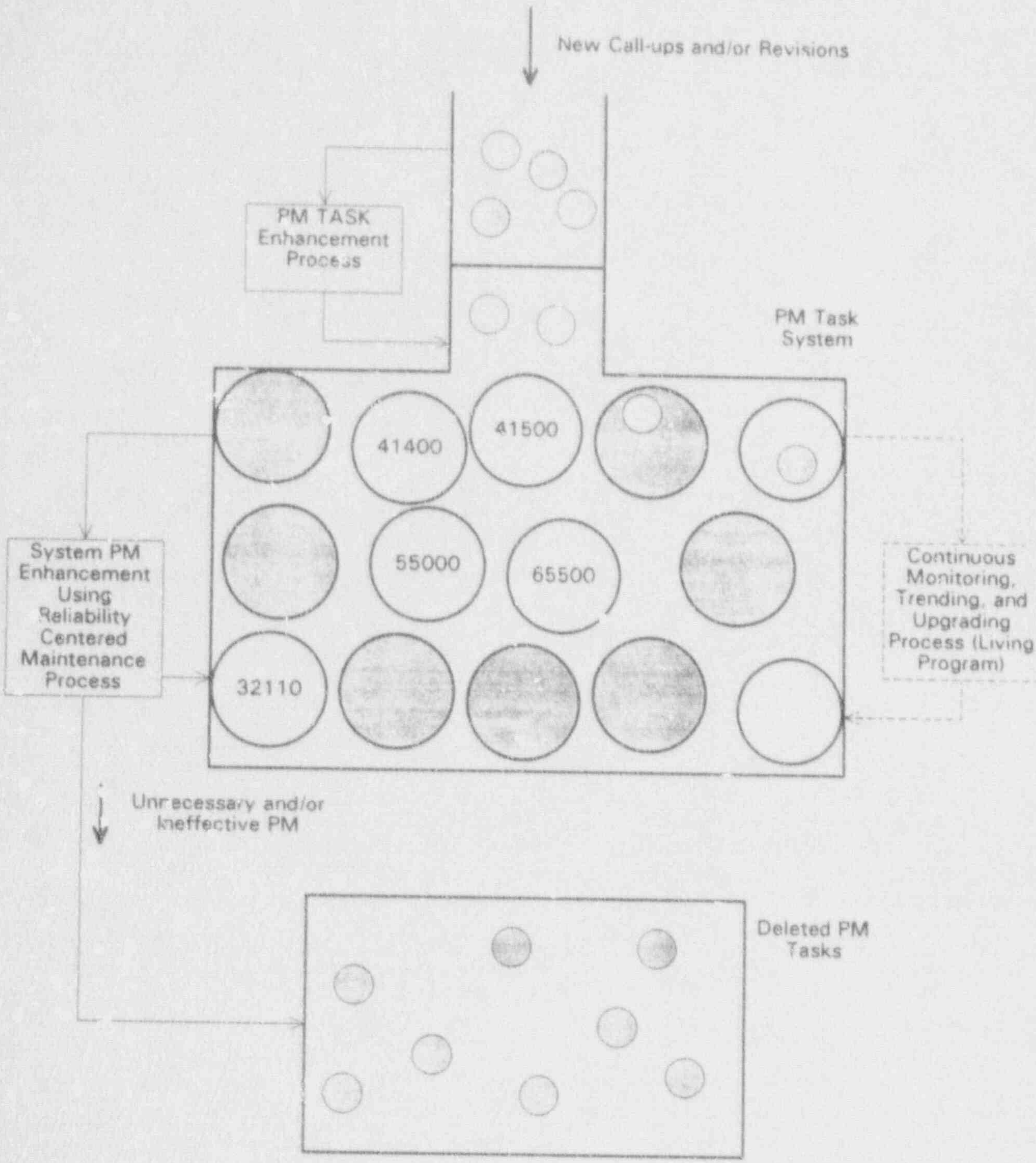

Figure 5. Khan model for PM program improvement.

AIR-OPERATED VALVE PREVENTIVE MAINTENANCE PROGRAM

Introduction

Traditionally, air-operated valves (AOV) have received little attention by our maintenance staff. Valve maintenance has been predominantly corrective and calibration/setup of AOVs has been looked upon as an art, not a science.

Recent developments in diagnostic analysis of AOV performance by at least three vendors (Fisher Controls, ABB Combustion Engineering, and ITI MOVATS) have provided us with a new diagnostic technique for determining the condition of AOVs (Fisher). An AOV PM program based on the use of AOV diagnostics is now being incorporated into the Bruce NGS 'A' Integrated Preventive Maintenance Program. The AOV PM program has been in evolution for approximately two years. A previous paper (Walker, 1991) describes our early experience with AOV diagnostics.

Diagnostic System

The diagnostic system in use at Bruce NGS 'A' measures a number of characteristic parameters (or signature) of an AOV's response to an instrument input signal during a full stroke cycle of the valve. The parameters include valve stem travel, instrument air pressure to the positioner, supply air pressure, and actuator air pressure.

The system can indicate

- Bench set and actuator spring rate
- Valve stroke length and friction
- Condition of bearing and guide surfaces
- Valve stroke time
- Seat load and seat condition
- Stem nut adjustment
- Air supply pressure
- Positioner calibration, linearity and hysteresis
- I/P calibration, linearity and hysteresis.

Figure 6 shows a typical signature that the diagnostic system records for an AOV. Assessment of valve condition requires both analysis of the signature trace and software generated parameters such as stroke length, valve friction, and seat load. Comparison of subsequent diagnostic signatures with the initial (or baseline) signature valve condition permits trending of valve performance and condition.

Full Program Evolution

The AOV diagnostic pilot program (Ferguson and Cooper, 1991) mentioned previously, examined a population of about 50 valve/actuator combinations (valves: Posiseal, Fisher, Masoneilan, Rockwell, Hattersly-Newman-Hender; actuators:Bettis, Masoneilan, and Fisher) and successfully demonstrated that

1. AOV diagnostics can non-intrusively assess condition of a complete valve/actuator assembly and as a consequence is a useful predictive maintenance tool
2. AOV diagnostic equipment can dynamically calibrate and setup AOVs
3. Station maintenance staff can quickly develop skills in AOV diagnostic analysis.

As a result of these positive findings we proceeded with development of an AOV PM.

Three key elements of the program are as follows:

1. Organization
2. Valve priority ranking
3. Preventive and corrective maintenance work flow.

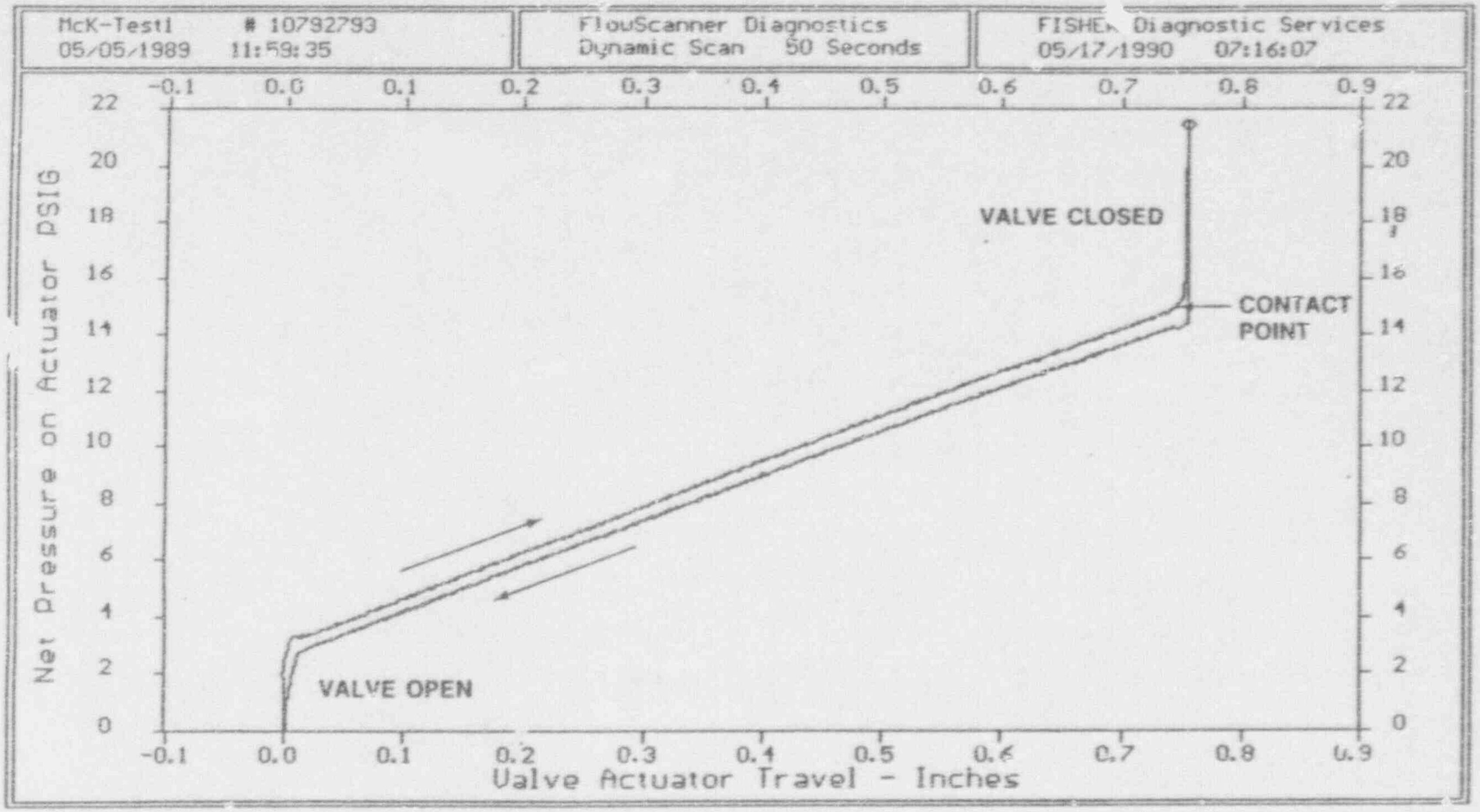

Figure 6. Typical valve signature.

Organization. A full time "valve diagnostic team" has been dedicated to this program. Its structure is shown in Figure 7. This team is responsible for both the AOV and MOV PM programs and is a specialist "day only" crew composed of both control (electrical/I&C) and mechanical maintainers. Their duties include

1. Collection, interpretation, and storage of valve signature data
2. Repair, calibration, and setup of valves in the program.

Program management and engineering support is provided by the PM Engineering Unit and Technical Section.

PM Priorities. Currently there are approximately 60 valves (per unit) in the PM program. They have been selected by the following criteria:

1. Valves inside reactor building
2. Problem valves identified by operators or engineering
3. Valves identified by RCM analysis.

Priority ranking of valves within the program will be provided by RCM analysis.

Inspection frequency has been set at once per year. However, as RCM analysis proceeds and trend data accumulates, frequencies will be refined to reflect failure history and service conditions.

Preventive and Corrective Maintenance Work Flow. The flow of AOV diagnostics and repair work within the maintenance department is described by a flowchart in Figure 8. It addresses two paths:

1. Corrective
2. Preventive maintenance (or call-ups).

In the case of corrective maintenance it is important to note that

> (a) AOV deficiencies can be initiated by maintenance or operations staff and (b) AOV work is usually assigned to the valve diagnostic crew, but in the case of emergency, duty shift maintenance personnel can perform troubleshooting and repairs. However, an "as left" signature of the valve must be taken by the valve diagnostic team.

All preventive maintenance is performed by the valve diagnostic team. This includes performing "as found" and "as left" diagnostics, repair, and calibration for each valve.

Results

To date, a population of about 100 valves has been tested (Bruce NGS 'A' has approximately 400 AOVs per unit). Figure 9 is a plot of the percent of occurrence versus fault type found with the valve population tested using the diagnostic equipment. The bar chart shows both initial and followup calibration results. The followup calibration represents the second diagnostic visit to the valve population and indicates a trend to improved valve performance except in the case of the I/P transducers where we have seen an increase in faults primarily due to "zero drift" problems.

It must be stressed that the deficiencies listed are of varying degrees. Not all require immediate action of repair.

Three specific examples of valve signatures where valve performance deficiencies have been identified:

- Bleed and Relief Circuit

  CV 22 and 23 are identical valves in over pressure relief service on the Unit 1 Primary Heat Transport Pressurizer. Figure 10 (actuator pressure vs stem travel) shows the effect of a loose stem inside the split coupling. Figure 11 (stem travel vs I/P output

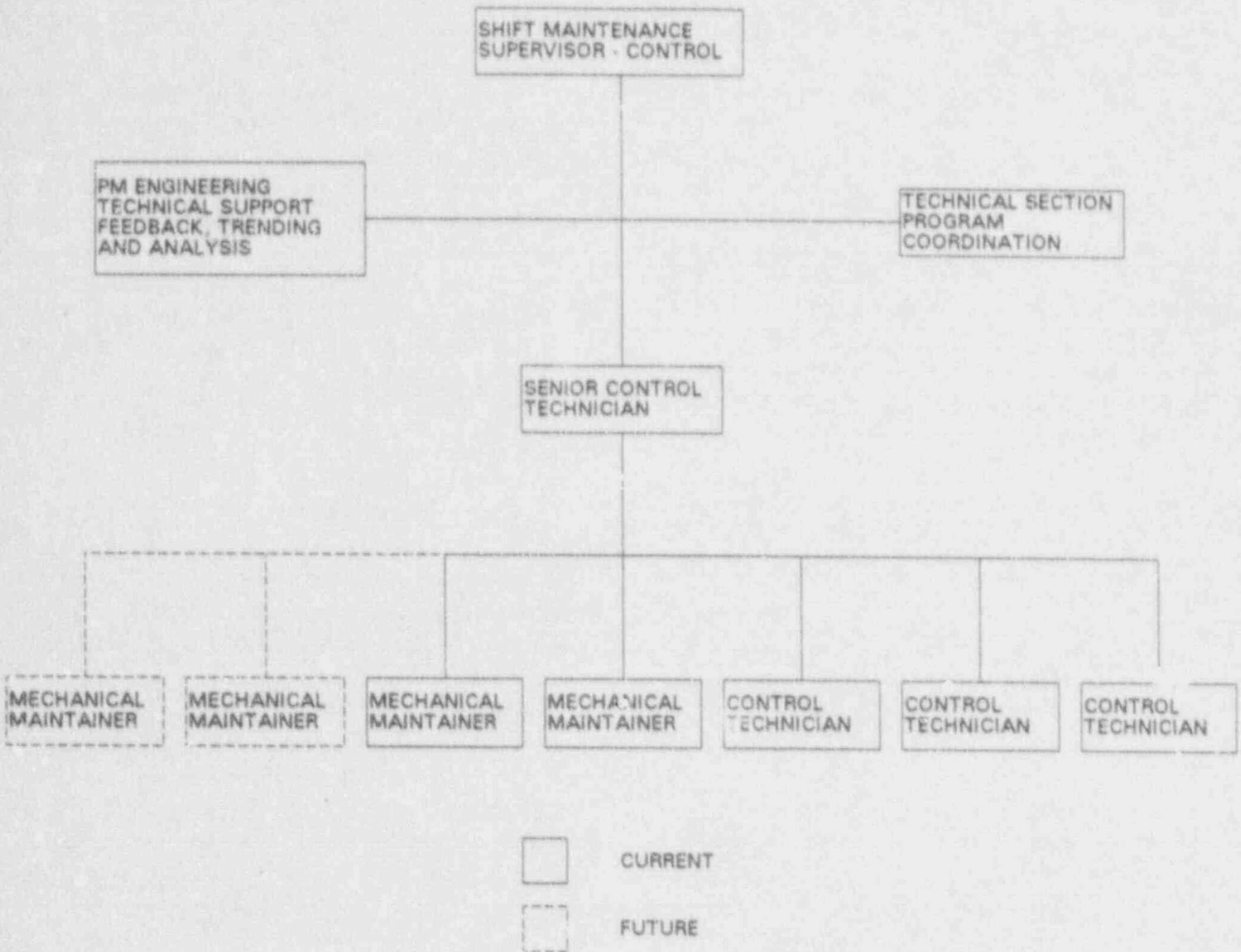

Figure 7. Valve diagnostic team organization.

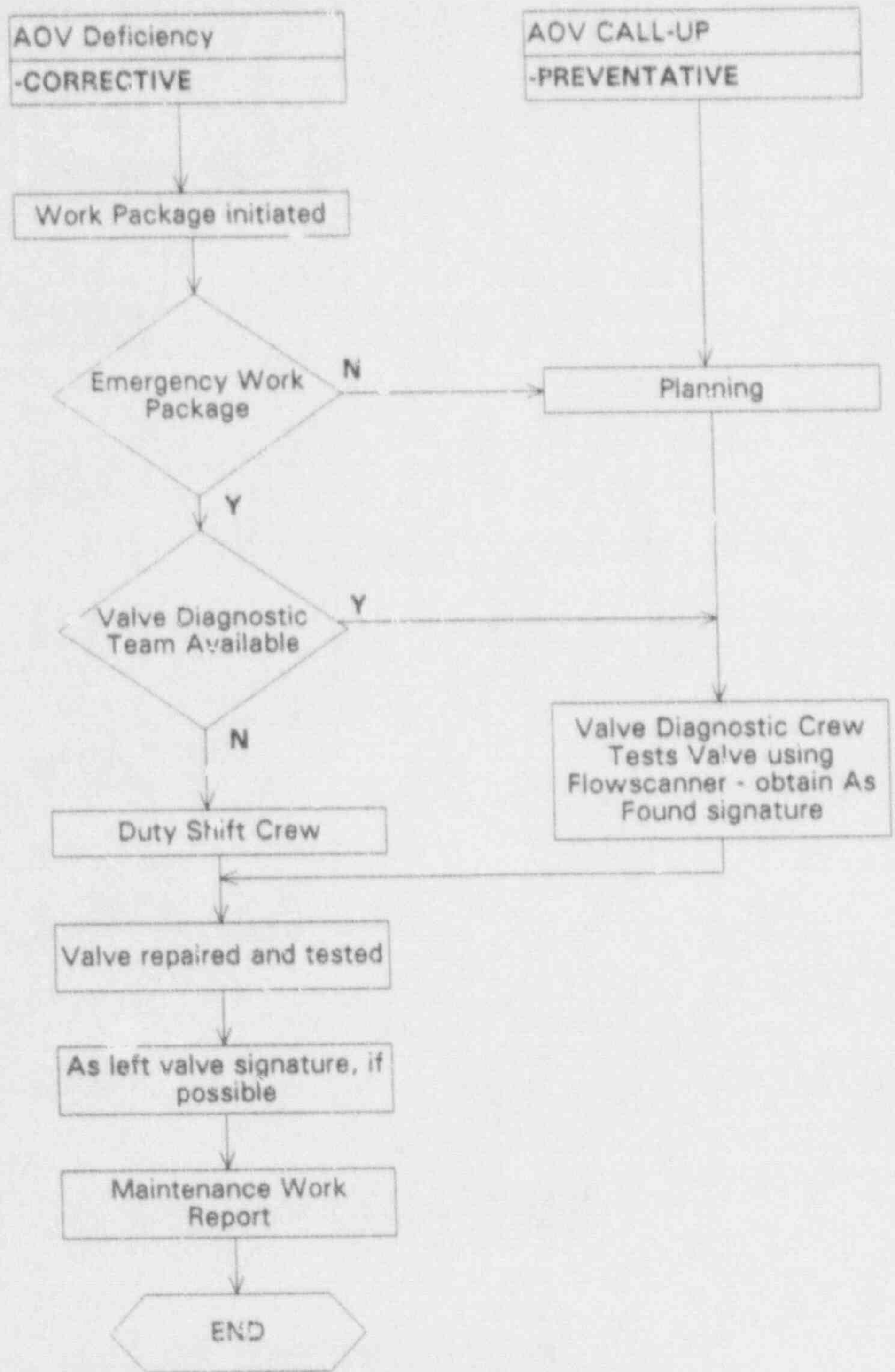

Figure 8. Process flow diagram for AOVs.

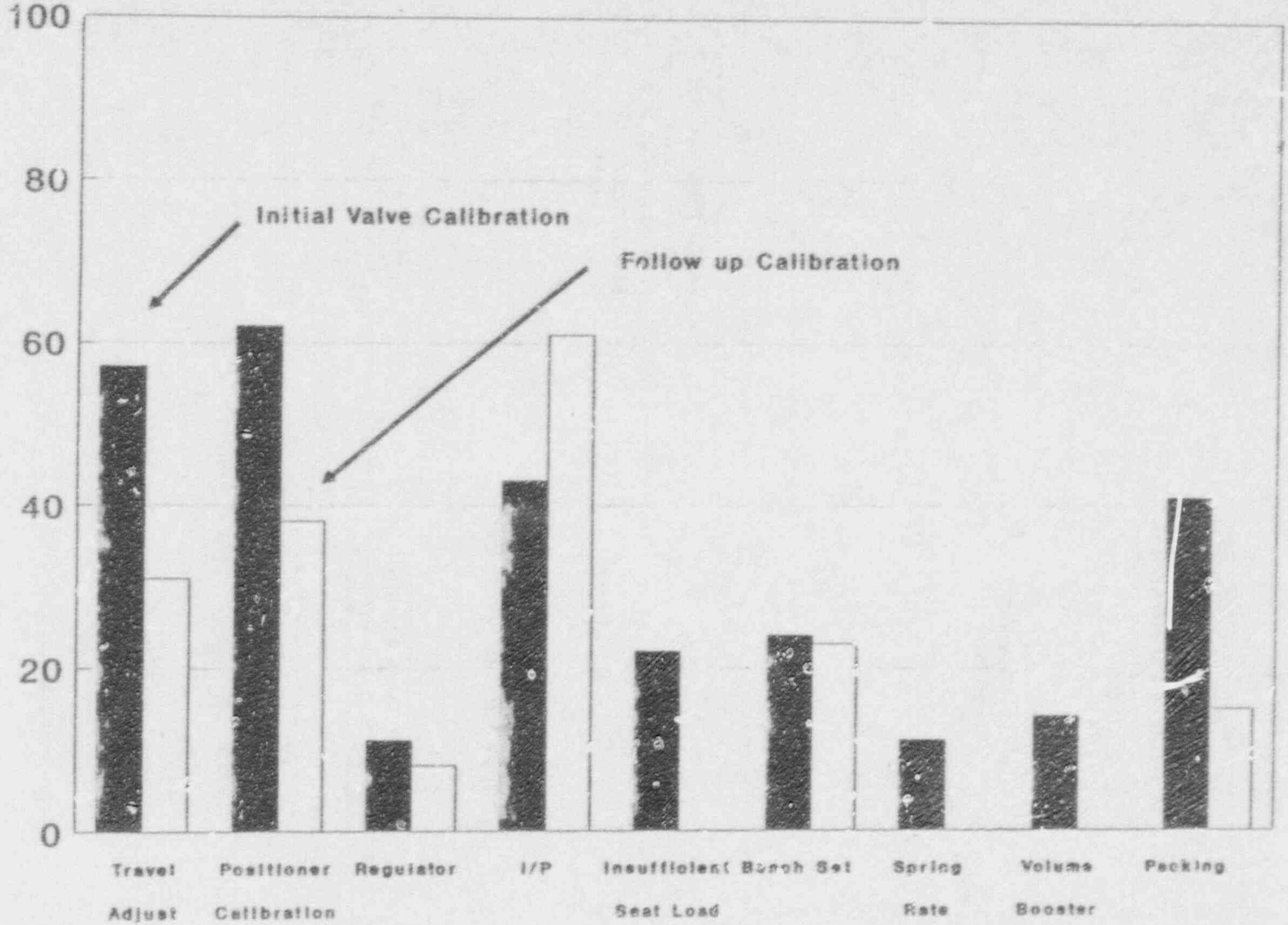

Figure 9. Deficiency frequency distribution.

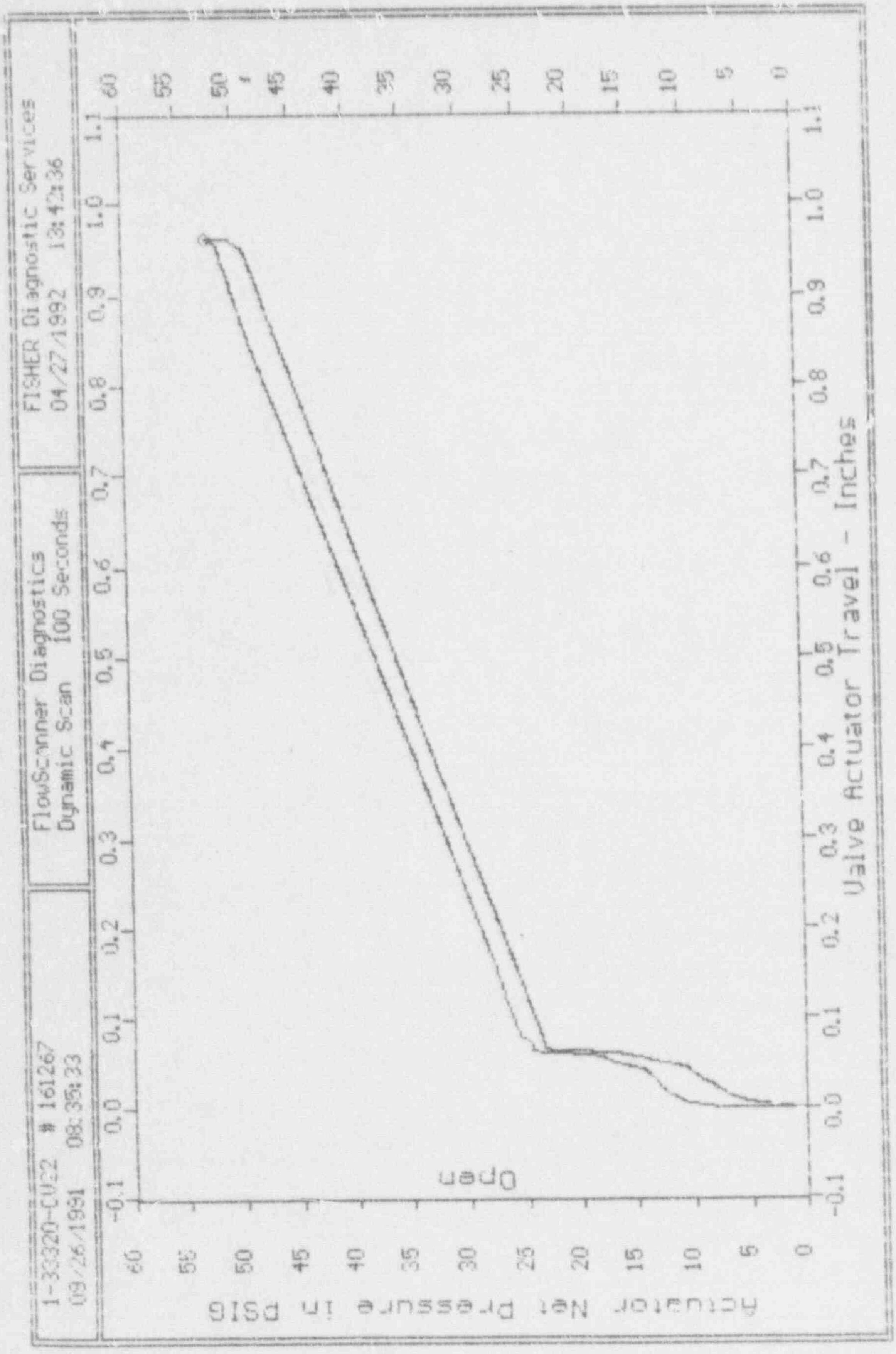

Figure 10. Actuator pressure versus stem travel for CV22 and CV23.

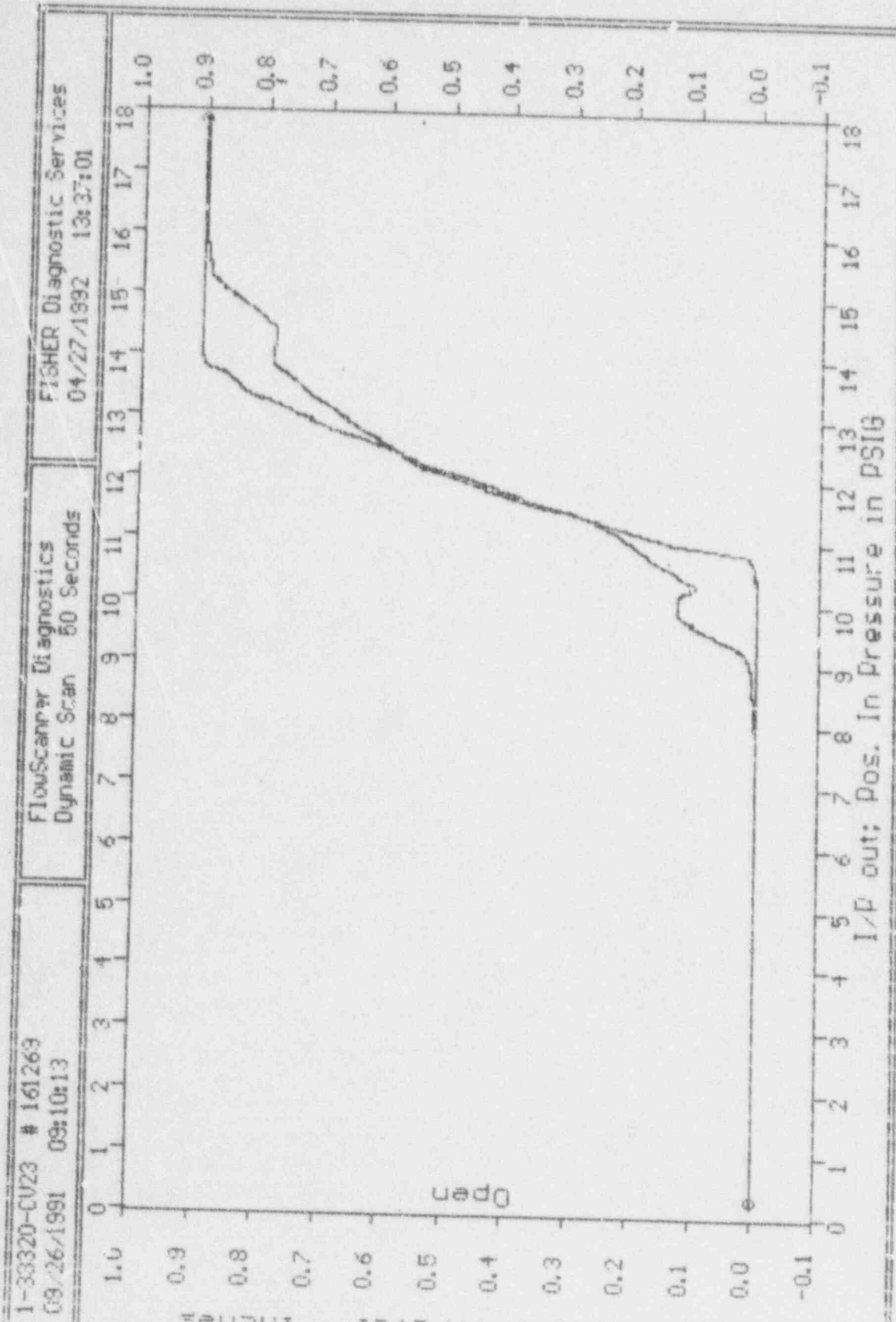

Figure 11. Stem travel versus I/P output pressure for CV22 and CV23.

pressure) illustrates a positioner fault. Subsequent inspection showed that the signal bellows/beam was loose.

- Auxiliary Boiler Feedwater System

  FCV 2 is an auxiliary boiler feed pump recirculation valve. Operators reported flow problems but the valve appeared to be functioning properly. In Figure 12 (actuator pressure vs stem travel) the peaks after opening and before seating are indicative of a plug/stem separation.

CONCLUSIONS

Development and implementation of the Integrated Preventive Maintenance Program has enabled us to begin to take control of our Preventive Maintenance Program at Bruce NGS 'A' and, with the use of RCM analysis, we have begun to develop a technical basis for all PM tasks. Full implementation is expected to have a major positive impact on safe, reliable, cost effective operation of Bruce NGS 'A.'

We have also developed and implemented an AOV preventive maintenance program based on the successful results of the Bruce NGS 'A' AOV diagnostic pilot program.

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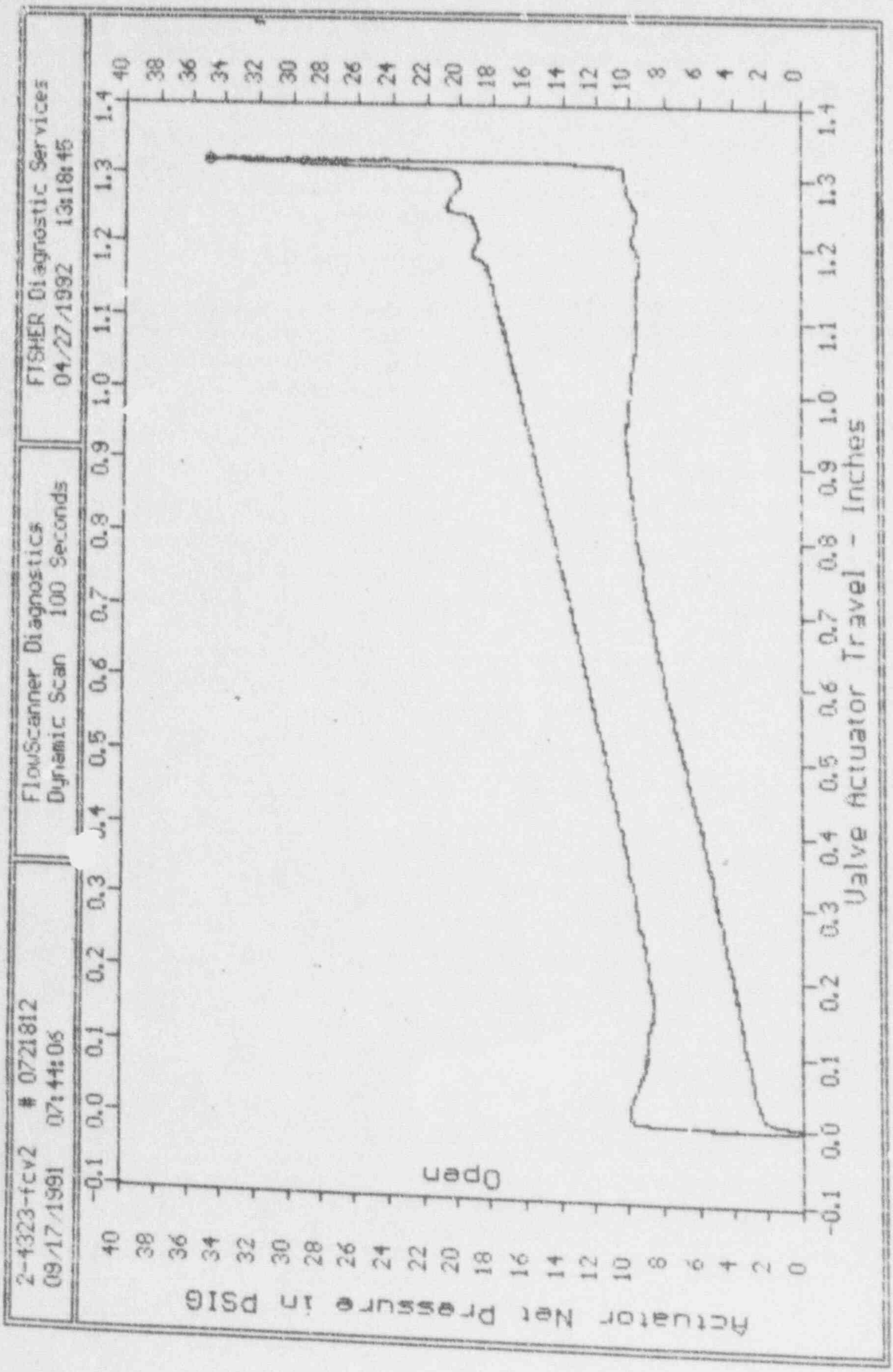

Figure 12. Actuator pressure versus stem travel for FCV2.

Session 2B

IST Technical Issues

Session Chair
Lawrence Sage
Illincis Department of Nuclear Safety

The Future of Inservice Testing Pump and Valve Programs at Nuclear Power Facilities

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ABSTRACT

The American Association of Mechanical Engineers/American Institute of Standards Operation and Maintenance (OM) Standards and Guidelines that were developed over the last 12 years are now the *Code for Operation and Maintenance of Nuclear Power Plants, ASME OM Code-1990*. This new code provides the rules and requirements for inservice testing of components in light water cooled nuclear power plants. This Code, when adopted by the Nuclear Regulatory Commission, will replace the existing ASME Section XI, Subsections IWP and IWV, as well as OM-1987, Parts 6 and 10.

This paper provides insight and background for nuclear power plants that are preparing future inservice testing (IST) program plans.

BACKGROUND

Recent changes in the nuclear industry and new regulatory requirements have compounded the problems for utilities to prepare and submit their next 10-year interval IST program plan. Nuclear power plants are required by federal regulation [10 Code of Federal Regulations (CFR) 50.55a(g)] to test certain ASME Code Class 1, 2, and 3 safety-related pumps and valves in accordance with ASME Section XI of the Boiler and Pressure Vessel Code. The Board on Nuclear Codes and Standards (BNCS) decided 12 years ago that inservice testing of pumps and valves should be performed in accordance with an ASME/ANSI Operation and Maintenance (OM) Standard. With that direction in mind, the Operations and Maintenance Main Committee formed subcommittees, subgroups, working groups, and task groups to develop new standards and guidelines to accomplish this task. In 1990, a significant change took place. The BNCS published the *Code for Operation and Maintenance of Nuclear Power Plants, ASME OM Code-1990*. In addition, the Nuclear Regulatory Commission (NRC) has recently issued two generic letters: GL 89-04, "Guidance on Developing Acceptable Inservice Testing Programs," and GL 89-10, "Safety Related Motor-Operated Valve Testing and Surveillance." These have added additional requirements to the IST program plan.

IST PROGRAM SCOPE

Nuclear utilities are required to submit IST programs to the NRC at the start of each 10-year interval. Prior to 1988, ASME Section XI, Sections IWP-1100 and IWV-1100 defined the scope of pumps and valves to be included as follows:

- Pumps. Class 1, 2, and 3 centrifugal and displacement pumps that are required to perform a specific function in shutting down a reactor or in mitigating the consequences of an accident, and that are provided with an emergency power source
- Valves. Certain Class 1, 2, and 3 valves that are required to perform a specific function in shutting down a reactor to the cold shutdown condition or in mitigating the consequences of an accident.

New changes have occurred since the CFR was published. These new documents require inclusion of more than what Section XI subsections IWP and IWV originally required. These documents must be taken into account when developing a 10-year interval program plan.

Newly published, but not yet endorsed by the NRC OM ASME Code-1990, Subsections ISTB and ISTC give a broad approach to require that all pumps and valves that perform a safety function be included in the program, not just ASME Class 1, 2, and 3 pumps and valves.

GL 89-04 explains that 10 CFR 50.55a(g) should be viewed as one part of a broad effort to ensure operational readiness of pumps and valves rather than viewed in the narrow sense as compliance to 10 CFR 50.55a(g) and the ASME Section XI Code.

Appendix A of 10 CFR 50, "General Design Criteria for Nuclear Power Plants," GDC-1, and Appendix B, "Quality Assurance Criteria for Nuclear Power Plants," Criterion XI, require that all pumps and valves necessary for safe operation are to be tested to demonstrate that they will perform satisfactorily in service. Appendix J, "Primary Reactor Containment Leakage Testing," requires preoperational and periodic verification by tests of leak-tight integrity of the primary containment, and systems and components that penetrate containment. All containment isolation valves (CIVs) that are required to be tested by Appendix J (Technical Specification–Primary Containment Integrity) should be included in the IST program.

In addition, other Licensee basis documents may define additional pump and valve testing requirements. Plant Technical Specifications (4.0.5) and other surveillance requirements state testing requirements and limiting conditions for operation for certain pumps and valves that must be fulfilled for the pump or valve to be considered operable. These requirements should be incorporated into the IST program to eliminate redundant testing and conflicts in acceptance criteria.

Plant Final Safety Analysis Reports (FSAR) or Updated Safety Analysis Reports (USAR) may delineate additional requirements or provisions made in the original submittal for construction and operating licenses.

NRC Information and Enforcement (I&E) Bulletins provide guidance or recommendations on test requirements for pumps and valves.

NRC Information Notices (INs) are intended to alert holders of operating licenses to an NRC evaluation. Note, however, that suggestions contained in these INs do not constitute NRC requirements; therefore, no specific action is necessary, in most cases.

Institute of Nuclear Power Operations (INPO) Good Practices, SOERs, SERs, and Assessments/Findings have specific directions or guidance in performance of testing or trending of data for performance monitoring.

Licensing/Site Administrative commitments to regulatory/state agencies that affect IST components or testing should be incorporated into the IST program. This is a result of internal response to Condition Reports, Deficiency Reports, Licensee Event Reports, and Operating Procedures that make a specific change based on an event or operational issue experienced at the facility.

IST PROGRAM DOCUMENT

The IST Program document should be divided into two sections: program text and program appendices.

The text should define the basis for the document, list and explain specific exemptions, selection criteria for pumps and valves, thus defining the scope and extent of IST testing requirements. The appendices should include pump and valve IST program piping and instrumentation drawings or other drawings used, pump tables, valve tables, pump relief requests, pump cold shutdown testing justifications, valve relief requests, and valve shutdown testing justifications.

INSERVICE TESTING BOUNDARIES

Pump IST Program

The scope of a pump IST program shall establish the requirements for preservice and inservice testing to assess the operational readiness of certain safety-related centrifugal and positive displacement pumps that are provided with an emergency power source and that function to

- Shut down a reactor to safe shutdown condition
- Maintain the reactor in the safe shutdown condition
- Mitigate the consequences of an accident.

Excluded from the program are (a) pumps that are supplied with an emergency power source, but used solely for operating convenience, and (b) drivers, except where the pump and driver form an integral unit, or where the motor bearings are the only accessible bearings to assess operational readiness of the pump as a unit.

Valve IST Program

The scope of a valve IST program shall establish the requirements for preservice and inservice testing to assess the operational readiness of certain safety related valves and pressure relief devices. The active or passive valves are those that are requi l to perform a specific function in shutting down a reactor to the safe shutdown condition, in maintaining the safe shutdown condition or in mitigating the consequences of an accident.

Exempted from the program are

- Valves used only for operating convenience, such as vent, drain, instrument, and test valves
- Maintenance valves used only to isolate components or perform maintenance
- Valves used only for system control, such as pressure or flow control, and manual throttle valves
- Valves used in external control and protection systems re, onsible for sensing plant conditions and providing signals for valve operation.

Pump and Valve Selection

Pumps and valves that are within the scope of the IST Program shall be tested in accordance with the Section XI Code Editions, or ASME OM Code-1990, when approved. Tables listing the pur , and valves, including all applicable testing requirements and frequencies, are contained in Attachments 1, 2, and 3, respectively. Pump and valve table descriptions are found in Attachments 2 and 3. When valve testing must be deferred to cold shutdown, cold shutdown test justifications are to be provided deferring testing until the plant is placed in cold shutdown. When the pump or valve testing requirements of the applicable Code cannot be met for any reason, relief requests are to be written requesting NRC approval for alternative test methodology to verify component operational readiness.

Code Requirement Positions

This section indicates the positions, such as NRC GL 89-04 and 89-10, used to specify new or clarify existing Code requirements, as they apply for the formulation and the implementation of this program.

Passive Valves

Passive valves are valves that are not required to change position to accomplish a specific function. For this program, passive valves are further defined as those valves where the normal and safety position are the same, and the valve is not required to change position during any normal plant operating condition.

Containment Isolation Valves

All Containment Isolation Valves (CIVs) that are included in an Appendix J program should be included in the IST program as Category A or A/C valves. In this case, utilities have specific leak test procedures and requirements for containment isolation valves specified by 10 CFR 50, Appendix J, which are equivalent to the requirements of ASME Section XI IWV-3421 through 3425, or ASME OM-1990 ISTC 4.3.1 through 4.3.3.

In addition, utilities must comply with the Analysis of Leakage Rates and Corrective Action requirements of IWV-3426 and 3427(a), or ASME OM-1991 ISTC 4.3.3(e) and (f).

Cold Shutdown Testing

For valves in which testing is deferred to cold shutdown but not less than every 92 days, testing shall commence within 48 hours after cold shutdown is achieved, and will continue until all tests are complete or the plant is ready to return to power. Any testing not completed at one cold shutdown will be performed during any subsequent cold shutdowns. For planned cold shutdowns in which the unit will have sufficient time to complete the testing of all valves identified to be tested at cold shutdown, exception may be taken to the 48 hour start time. As a minimum, all cold shutdown valves will be tested during each refueling outage; however, valve testing does not need to be performed at a frequency greater than quarterly (92 days).

Check Valve Full/Partial Stroke

In most cases, full design flow through a check valve requires less than full mechanical valve movement. As used in this program, with the exception of testable check valves, the term full stroke refers to the ability of the valve to pass design flow, and not the full mechanical stroking. Forward flow stroke operability testing will be by any method that verifies the valve is capable of passing design flow. Any test that verifies less than full design flow capability is considered as a partial stroke test.

Valve Position Indicator Verification

Verification of proper indication of remote position indication will normally be accomplished by locally observing the position of the valve and comparing it with the remote indication. However, certain valves, such as solenoid operated and excess flow check valves, are not equipped with a local means to verify position. Therefore, position should be verified by the observation of system parameters such as flow or pressure during valve cycling.

ALTERNATIVES TO CODE REQUIREMENTS

Pump Vibration Testing

ASME OM Code, Subsection ISTB, specifies vibration to be taken in a plane approximately perpendicular to the rotating shaft in two orthogonal directions, and in the axial direction on each accessible pump bearing, for measuring pump vibration on all pumps. Justification should be provided in a relief request based on the conclusion that vibration velocity provides a significant improvement in the predictive capability to detect degradation that will result in an earlier correction of potential problems.

Pump Bearing Temperature

Pump bearing temperature monitoring need not be performed if the plant uses an enhanced vibration program. Justification should be provided in a relief request based on the conclusion that the ability of improved vibration testing to detect very small changes in bearing condition would preclude any bearing condition severe enough to cause an abnormal rise in the temperature on the bearing housing when performed on an annual basis.

Pump Proper Lubricant Level or Pressure

Where practical, pump lubricant levels or pressure will be observed in accordance with

IWP-3100-1. Certain pumps do not allow observation if their level or pressure. In this case, justification for relief may be requested. A station maintenance procedure or preventative maintenance (PM) program for lubrication of plant equipment, which is performed on a frequent basis, may be used in justification.

Increased Testing Frequencies

The testing frequencies for certain Category C and D valves should be increased, if needed for compliance with Technical Specification or FSAR commitments. The specific frequency designations and the associated references are to be contained in the test frequency section of the pump and valve program tables. See Attachments 2 and 3.

COLD SHUTDOWN TEST JUSTIFICATION DISCUSSION

The Code permits valve testing that is impractical during operation to be delayed and performed during cold shutdown. Justification for this delay of testing is provided by the cold shutdown test justifications. Each cold shutdown test justification should be formatted to contain the following information:

- Cold shutdown test justification number
- System
- Valve identification number
- Category
- Function
- Quarterly test requirements
- Cold shutdown test justification
- Quarterly partial stroke testing
- Cold shutdown testing.

RELIEF REQUESTS DISCUSSION

Written relief requests under 10 CFR 50.55a(g)(5)(iii) may be submitted for pumps and valves to provide justification for the following:

- Performance of testing requirements on pumps and valves that are impractical during both operation and cold shutdown. The Licensee must show that conformance to the Code would cause unreasonable hardship without a compensating increase in safety.
- The use of alternative testing methods, when Code requirements are impractical, or where the alternative methods provide an acceptable level of quality and safety, or equal to or greater assurance of pump and valve operability.

Each relief request should be formatted to include the following information:

- Relief request number
- System (specific relief requests only)
- Pump or valve identification number
- Category (valves only)
- Class
- Function (specific relief requests only)
- Testing requirement
- Basis for relief
- Alternate testing.

All IST programs contain requests for relief from various Code requirements. In addition, the surveillance requirements of Technical Specification 4.0.5 for most plants state that this testing of pumps and valves must be performed in accordance with ASME Section XI except where specific written relief has been granted by the NRC. As stated in excerpts from the NRC GL 89-04, the general nature of the IST sections of the ASME

Code do not consider plant specific designs. With the resulting difficulty in complying with all the ASME Code and other requirements, utilities frequently revise their programs as more experience with IST is acquired. Programs at most plants are revised several times during the 10-year interval as program updates. However, should a utility be unable to comply with one of these positions because of design considerations or personnel hazard, as opposed to inconvenience, any alternative testing must fulfill the basic test objective of detecting component degradation. Alternative testing should be evaluated individually by the utility and the utility's plant safety review committee or equivalent. When evaluating testing, the utility should address the following:

1. Maintenance history of the individual (specific) component

2. Maintenance history of related components in a similar environment

3. Component vendor records of degradation at other facilities

4. Records on degradation of the same or like component from other utilities.

Plants may find good justification through using in-plant records, INPO Component Failure Analysis Reports (CFAR), Nuclear Plant Reliability Data System, and other referenced sources to compile data to address the above four areas. A lack of service experience, [illegible]ue factors, or potential test failure are not sufficient to justify the alternative test or not performing the IST requirement.

The alternative test is not considered acceptable unless the above data are sufficient to justify its adequacy for detecting degradation and ensuring continued operability. Justification for the alternative test should be documented and retained in the IST program.

Changes to the IST programs as a result of the NRC GL 89-04 should be submitted to the NRC.

CONCLUSION

Although the requirements for inservice testing have been in effect for 15 years, there are still many unanswered questions, problems, and areas needing improvement. Even after the issuance of GL 89-04, many IST programs are submitted to the NRC for review in which safety related pumps and valves are excluded.

In addition, when specific requests for relief are submitted, the technical information and justification for cold shutdown testing, or requests for relief, are incomplete or inadequate. Another problem related is the lack of published guidance on the preparation of IST programs.

Utilities are concerned with the interpretation of the codes and regulatory requirements, their implementation, and surveillance test requirements for the pumps and valves. As indicated in GL 89-04, the NRC will be more active in the utilities' performance of their IST program, with less emphasis on the IST program plan itself. In order to have a good test program, the utility must have a good IST program. I hope this technical paper will give guidance in this area.

ACKNOWLEDGMENTS

I would like to acknowledge certain people for their help to make this paper possible with their contributions: Mr. R. S. Hartley, Ms. L. G. Holland, Mr. C. D. Jones, Mr. R. H. Martin, Mr. R. J. Nyberg, and Mr. C. W. Walling.

REFERENCES

ASME Boiler and Pressure Vessel Code, 1983, Section XI.

Nuclear Regulatory Commission, "Guidance on Developing Acceptable Inservice Testing Programs," Generic Letter 89-04.

ATTACHMENT 1
SYSTEMS FOR INCORPORATION IN AN IST PROGRAM PLAN

Pressurized Water Reactors (PWR)

- Reactor Coolant System (RCS) and any proposed flow path for establishing natural circulation
- Portions of Main Steam System or Supply (MSS)
- High Pressure Injection System (HPCI)
- Low Pressure Injection System (LPCI)
- Accumulator System
- Containment Spray System
- Primary and Secondary System Safety and Relief Valves and Atmospheric Relief Valves
- Portions of Main Feedwater System
- Auxiliary Feedwater System
- Residual Heat Removal System (Shutdown Cooling)
- Service Water Systems
- Containment Isolation Valves (CIVs) that are required to change position on a containment isolation signal
- Component Cooling Water System
- Chemical and Volume Control System
- Emergency Diesel Engine Air Start System
- Emergency Diesel Engine Fuel Oil Storage and Transfer System
- Instrument Air System required to support safety system functions
- Ventilation Systems that perform a safety function
- Boric Acid Transfer System
- Spent Fuel Pool Cooling System
- Containment Combustible Gas Control System
- Post Accident Sampling System

Boiling Water Reactors (BWR)

- Reactor Coolant Recirculation System (RCS)
- Portions of Main Steam Supply System (MSS)
- High Pressure Injection/Core Spray Systems (HPCI) (HPCS)
- Low Pressure Injection System (LPCI)
- Low Pressure Core Spray System (LPCS)
- Residual Heat Removal System (Steam Condensing, Shutdown Cooling, Suppression Pool Cooling)

- Safety, Relief and Safety/Relief Valves of the Reactor Coolant Boundary and Secondary Systems
- Containment Cooling (Spray) System
- Containment Isolation Valves (CIVs) that are required to change position on a containment isolation signal
- Reactor Core Isolation Cooling System (RCIC)
- Standby Liquid Control System (SLC)
- Automatic Depressurization System
- Control Rod Drive Hydraulic System
- Active Valves in Service, Standby or Back-up Water Service, Closed Cooling Water, Fire Water and Well Water Systems
- Emergency Diesel Engine Fuel Oil Storage and Transfer System
- Emergency Diesel Engine Air Start System
- Portions of Main Feedwater System
- Instrument Air Systems that are required to support safety system functions
- Spent Fuel Pool Cooling System
- Containment and Drywell Equipment and Floor Drain Systems (DFR) (DER)
- Control Building Cooling and Chilled Water Systems
- Containment and Drywell Hydrogen Mixing Systems
- Transverse Incore Probe System
- Post Accident Sampling System

ATTACHMENT 2
PUMP IST PROGRAM TABLE DESCRIPTIONS

Pump Table Summary. The Pump IST Program Table should contain an alpha-numeric listing of all the pumps included in the Pump IST Program. The data contained in these tables identify the IST parameters to be measured, the applicable relief requests, and any applicable remarks. Proper lubricant level or pressures are included in the table because they must be observed for all pump operating and testing procedures. A column for *testing frequency* should be included if pump testing is performed other than QUARTERLY.

Pump Program Table Format. The Pump IST Program Tables must be organized to provide the following information:

1. SYSTEM NAME OR NUMBER: The system identification name or number.

2. PUMP MARK NUMBER: The pump identification number.

3. CODE CLASS: ASME Code Class.

4. P&ID: Piping and Instrumentation Drawing showing where pump is located.

5. COORD. Location coordinates of the pump on the P&ID.

6. PARAMETERS: List of the applicable testing parameters that will be measured. The parameters listed are those required by the code, unless alternative testing is provided by relief request. The following is a description of applicable parameters:

   a. SPEED—Pump speed (only required for variable speed pumps).
   b. INLET PRESS—Pump suction pressure.
   c. DIFF PRESS—Pump differential or discharge pressure.
   d. VIB VEL—Pump vibration velocity.
   e. FLOW—Pump flow rate.
   f. Lubricant Level or Pressure.

7. RELIEF: Indicates whether or not there is a relief request applicable, where Y = Yes and N = No. The specific relief request number is found in the remarks column.

8. REMARKS: Any additional pertinent information, such as the applicable Relief Request Number, FSAR or Technical Specification reference, is provided in this space.

ATTACHMENT 3
VALVE IST PROGRAM TABLE DESCRIPTIONS

Valve Table Summary. The Valve Program Tables should provide a tabulation of all safety-related valves included in the Valve IST Program. The tables are arranged by system and the valves in each system are listed in alpha-numeric sequence.

Valve Program Table Format. Each valve program table needs to be organized to provide the following information.

1. SYSTEM: The system to which the particular table applies.

2. VALVE NUMBER: The valve identification mark number.

3. CLASS: ASME Code Class.

4. P&ID: Piping and Instrumentation Drawing showing where the valve is located.

5. COORD: Location coordinates of the valve on the P&ID.

6. VCAT: Valve category as identified by IWV-2200 or ISTC 1.4.

7. A/P: Classification according to IWV-2100 or ISTC 1.4 and Table ISTC 3.6-1, where A = Active and P = Passive.

8. SIZE: Valve size in inches.

9. VALVE TYPE: Valve design type.

10. ACTUAT TYPE: Type of actuator used to change position of valve.

11. VALVE POSITION:

    a. NRM - Position during normal plant operation.
    b. SAF - Position to fulfill safety function.
    c. FAL - Position valve fails to or loss of electrical power.

12. TYPE C: Indicates whether valve requires Appendix J, Type C Leak Test, where Y = Yes and N = No.

13. TEST, FREQ, (DIR): The testing requirement, frequency the test will be performed, and, in parentheses, either the direction(s) stroke time should be measured for power operated valves, or the direction(s) the valve should be exercised for check valves. This column identifies the alternative Code requirements, unless testing is provided by relief request or cold shutdown test justification. Format for the field is as follows:

    Test Code - Frequency Code (stroke time or exercise direction), for example:

    FE-Q(F&R) = Full stroke exercise quarterly in the forward and reverse flow direction

ST-CS(O) = Measure valve stroke time during cold shutdowns in the open direction

FE-R(R) = Full stroke exercise during a refueling outage in the reverse direction.

14. RELIEF: Indicates whether or not there is a relief request or cold shutdown test justification applicable, where Y = Yes and N = No. The specific relief request or cold shutdown test justification number is found in the remarks column.

15. REMARKS: Any additional pertinent information, such as the applicable Cold Shutdown Test Justifications numbers or Relief Request numbers, is provided in this space.

Inservice Testing Bases Program

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ABSTRACT

Inservice testing (IST) programs generally address those components necessary for safe operation as specified in Subsections IWP-1100 and IWV-1100 of Section XI (American Society of Mechanical Engineers Boiler and Pressure Vessel Code) and Position 11 of the U.S. Nuclear Regulatory Commission Generic Letter 89-04. Program tables, generated manually or from computer databases, are frequently used to describe those components and testing requirements contained in IST programs. Table entries are usually derived from review of system drawings, accident analysis, and other technical documents; however, the bases for such entries are not always documented. Because IST programs are dynamic, the continual evaluation of program entries may be inhibited if bases for program entries are unclear. When IST programs have been developed by architect-engineers, nuclear steam supply system suppliers, or other contractors, the bases for IST program contents become even more nebulous to the program users. The presentation will provide a discussion of IST bases programs used by Entergy Operations' four nuclear plants, and the benefits derived from their use.

An Update to Inplace Testing of Safety/Relief Valves Utilizing Lift Assist Technology

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ABSTRACT

Inplace testing of safety and relief valves with lift-assist devices has received mixed reviews from nuclear power plant testing personnel. While many plants use the technology, most limit its use to testing main steam safety valves (even though both OM-1-1981 and PTC 25.3-1976 allow its use for several different service applications). Test coordinator concerns regarding the technology range from lift set point accuracy and repeatability to the quality of the test result output. In addition, OM-1-1981 and PTC 25.3-1976 differ in their approach to the technology.

PTC 25.3-1976 allows the use of lift assist devices to test any compatible valve, regardless of the system service. OM-1-1981 allows the use of lift assist devices for compressible fluid systems. However, it recommends not using such devices for liquid service systems. The reasons for the differences between PTC 25.3-1976 and OM-1-1981 will be discussed along with additional considerations applicable to the use of the technology in testing liquid service valves.

Advances in assist devices over the last few years have improved the accuracy, repeatability, and quality of test result output. This paper will show that lift assist technology is capable of determining lift set points within the accuracy requirements of OM-1 and PTC 25.3. It will also demonstrate that the technology should not be limited to compressible service systems. Also, improvements in test repeatability and output quality will be discussed as a function of the assist device design used and valve characteristics.

Lift assist testing is often preferred over inplace testing that uses direct system pressure. It is often more cost efficient than bench testing because it does not require removal of critical systems from service and transportation of components. Also, duplicating system temperatures and other environmental factors is not an issue during inplace testing. Valve testing that once required an outage and maintenance period can now be conducted prior to such periods. This approach minimizes the possibility of failures becoming critical path limiting items.

In summary, the full potential of lift-assist technology is not widely understood. As a result, the technology is underused in most plants, despite its many advantages.

Alternative Method for Full Flow Stroke Testing of Safety Injection Tank Check Valves at Fort Calhoun Nuclear Station

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ABSTRACT

This paper presents the evolution that led to the use of a reduced pressure safety injection tank (SIT) dump as the preferred test method for full-stroke testing the SIT check valves at Fort Calhoun Station. This discussion includes the following:

1. An interpretation of the Code requirements for full-stroke testing of check valves
2. Problems encountered with various test methods
3. The analysis technique used to relate the reduced-pressure flow test results to safety analysis flow requirements
4. A description of the test method and results to date
5. Summary of resource expenditures while evaluating the test methods.

Fort Calhoun Station found that a reduced-pressure SIT dump was the most cost-effective method of SIT check valve testing that would yield credible test results.

INTRODUCTION

Fort Calhoun Station (FCS) is a 485 MWe pressurized water reactor located on the Missouri River about 17 miles north of Omaha, Nebraska. FCS has four safety injection tanks (SIT) that each have one motor operated valve (MOV) and two check valves in 12-in. nominal diameter piping separating the SIT from the reactor coolant loops. The eight check valves are identified in the FCS Inservice Inspection (ISI) Program as Category C valves requiring full-stroke testing under Paragraph IWV-3520 of ASME Boiler and Pressure Vessel Code, Section XI (1980). These check valves are 12 in., 1500#, weld end, Duo-Chek check valves manufactured by Mission Manufacturing Company.

Because of the difficulty of verifying full-stroke open operability of these valves, FCS obtained relief from the Nuclear Regulatory Commission (NRC) in 1988 to perform a sample disassembly of two valves at refueling frequency to satisfy IWV-3520. Because of valve design and the inaccessible location, the sample disassembly of these valves proved to be so difficult and time consuming that an alternative method of full-flow testing was pursued. This led to a trial performance of a reduced-pressure SIT dump test in the spring of 1990 that proved to be practical to perform and that yielded credible test results.

In January 1991, FCS received interim approval from the NRC on an ISI Program that adopted the reduced-pressure SIT dump test for full-flow testing of the SIT check valves. In February 1992, the test was performed with acceptable results on all four SITs.

PROBLEM STATEMENT

The ISI Coordinator at FCS interpreted Paragraph IWV-3522 and NRC Generic Letter 89-04 (NRC) to say that the full-flow requirement for a check valve could be verified in three ways:

1. By verifying that the design flow rate would pass through the valve when the design differential pressure was imposed across the valve in the flow direction

2. By verifying that the valve disc will move to its full open position when a force of the appropriate magnitude is applied to the disc

3. By sample disassembly to verify the condition of valves that cannot practically be tested by 1 or 2, above.

Performing a test to pass the design flow rate through the SIT check valves at the design differential pressure proved to be impractical for the following reasons (see Figures 1 and 2):

- The 12 in. motor-operated gate valve takes about 54 seconds to fully open

- At the design differential pressure (~240 psig) and level (~60% or 6,000 gallons), the SIT will be empty well before the MOV is completely open, and the design flow rate (~15,000 gpm) will never be achieved

- If the SIT is dumped at refueling through the open reactor vessel (RV), expansion of the SIT nitrogen (N_2) blanket bubble under design conditions will result in releasing the N_2 to the containment atmosphere through the RV in a violent manner that would cause substantial airborne contamination.

The mechanical exercising of the SIT check valve disc requires disassembly of the check valve because the valve is constructed so that no nonintrusive mechanism is available to move or observe the valve disc position. Relief was obtained from the NRC to satisfy the SIT check valve full-flow test requirement by sample disassembly in 1988. Disassembly was performed in situ by removing an access port that was sealed with a silver plated, soft iron seal.

Sample disassembly and exercising of the SIT check valves were performed for the first time at FCS in the Fall of 1988. The valves were found to be in excellent condition, but many craft resources were expended in providing access to the valve and in getting the inspection port to seal after the disassembly. The access port was slightly out-of-round, and required machining before it would reseal. The disassembly of these valves is performed while the reactor coolant level is at mid-loop, so the potential for critical path delays is high. The same difficulties were encountered on the valves disassembled for inspection during the 1990 refueling outage. The resources expended in performing the disassembly and inspection of two SIT check valves include:

- $4,000 in materials

- 800 hours of craft/engineering time

- 6.5 man-rem of radiation exposure

- Radiological waste from 155 containment entries.

Thus, the per-operating-cycle cost of performing the SIT check valve sample disassembly is estimated to be around $90,000. In 1990, FCS performed a reduced pressure SIT dump (pilot test)[a] in an effort to qualify the procedures capability to satisfy the requirements of IWV-3520 for the SIT check valves. The dump test was performed on the SIT that dumped through valves that had just been disassembled and inspected. Thus, as the valves were known to be in good condition, the test would provide credible baseline data.

a. Omaha Public Power District Special Procedure, SP-SI-7, "Safety Injection Tank SI-6C Dump Test," April 2, 1990.

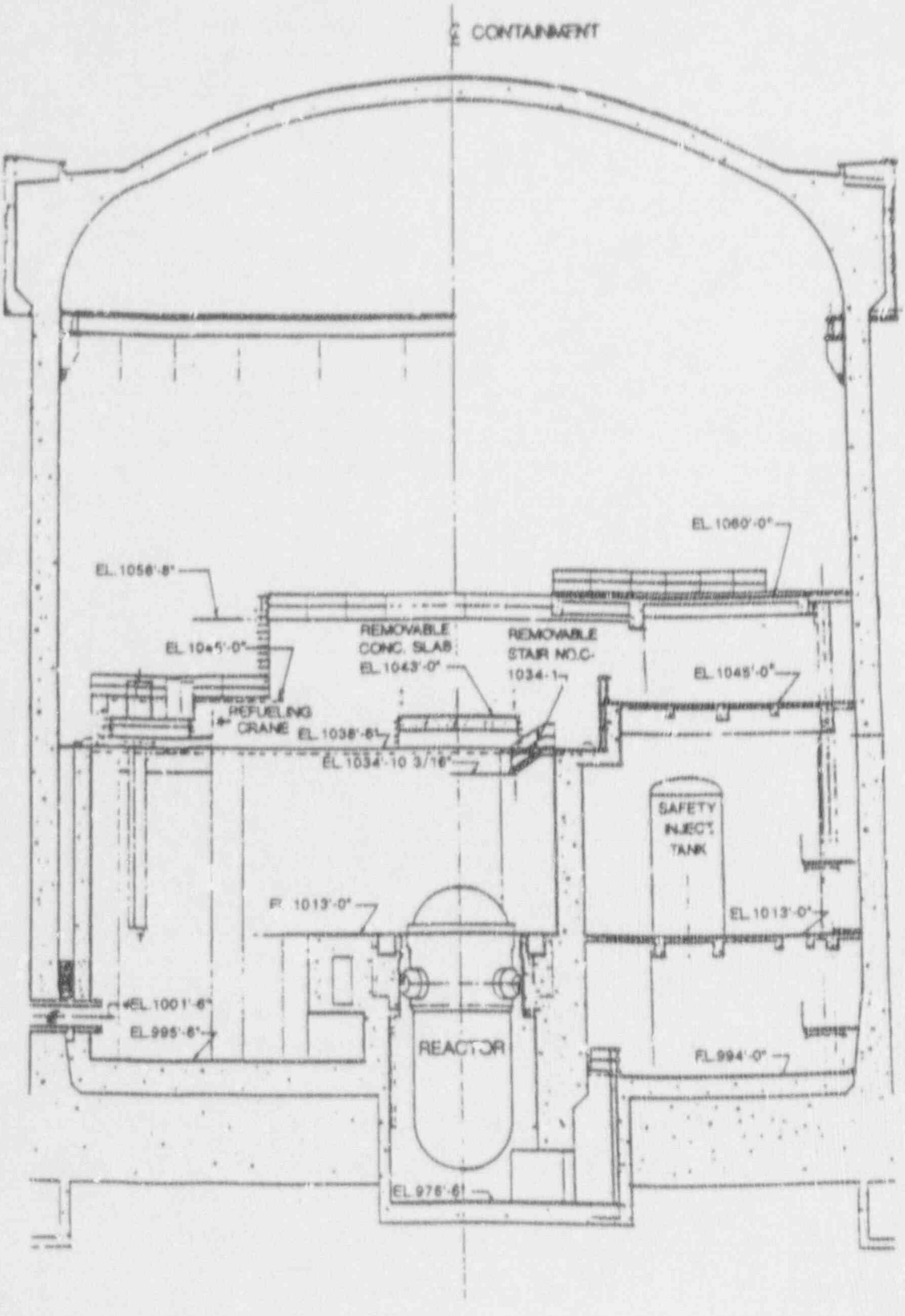

Figure 1. Plant layout.

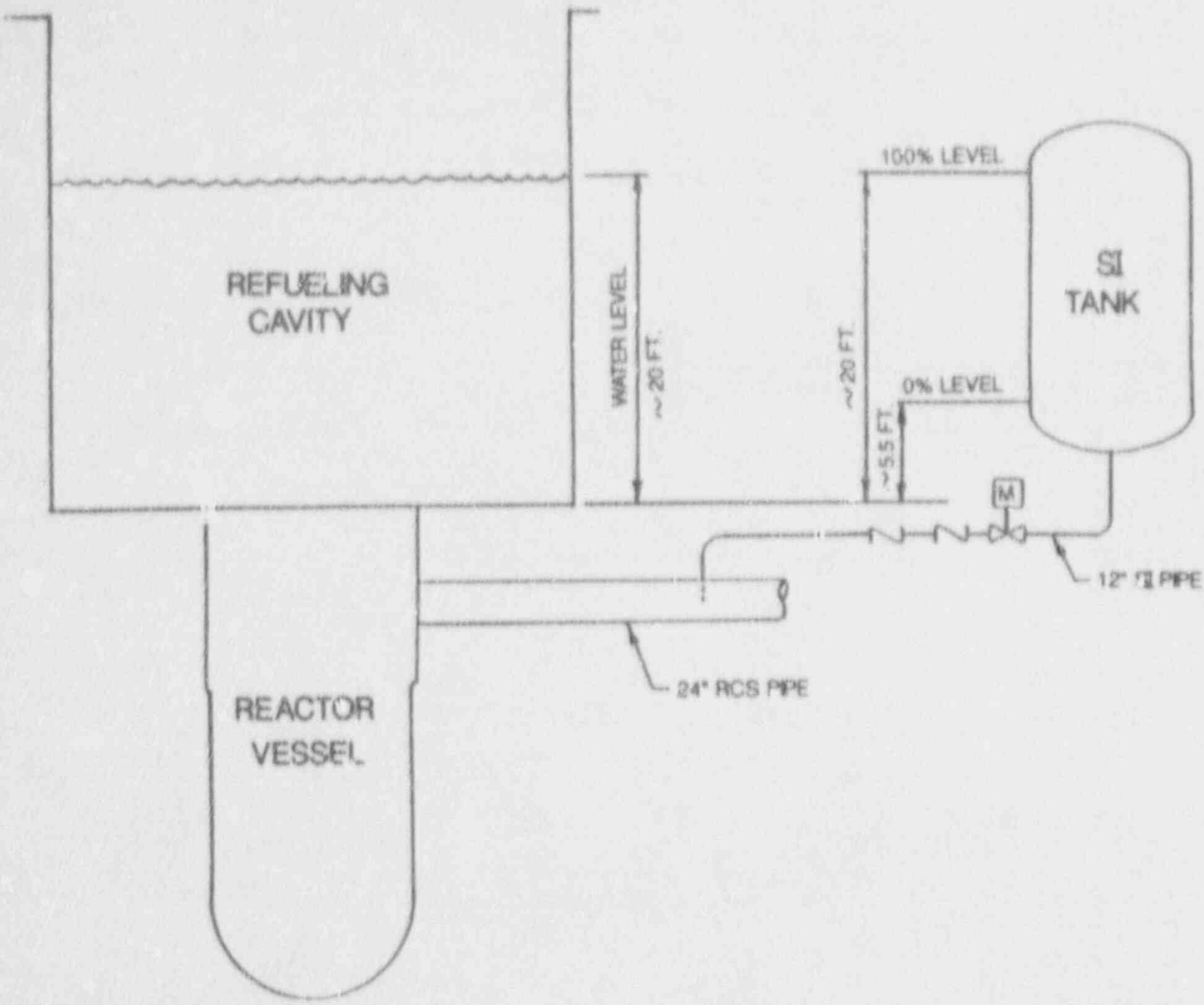

Figure 2. SIT dump schematic.

CONCEPTUAL BASIS FOR REDUCED PRESSURE DUMP TEST

Using the reduced-pressure SIT dump test to verify full-flow capability of the SIT check valves is based on the concept that flow rate (Q) through a piping system is proportional to the square root of the differential pressure (ΔP) across the piping system.

$$Q \propto \sqrt{\Delta P} \quad . \tag{1}$$

In Crane, *"Flow of Fluids,"* (1985), Equation 2.7, the same concept is stated:

$$Q = C_v \sqrt{\Delta P(62.4/\rho)} \tag{2}$$

where ρ = fluid density and C_v is a flow coefficient that is dependent only on the mechanical configuration of the flow path. For the purpose of the SIT dump, ρ is assumed to equal 62.4 lb/ft^3, which reduces Equation B to

$$Q = C_v 1 \sqrt{\Delta P} \quad . \tag{3}$$

With this concept in mind, the goal of the SIT dump test is to establish a C_v that is adequate to satisfy the safety analysis SIT flow requirement, and then to perform a dump test to measure the C_v using manageable test parameters. If the measured C_v is equal to or greater than the safety analysis C_v, the SIT check valves have fulfilled the full flow test requirement.

TEST SETUP/DEFINITION

The SIT dump is performed by establishing adequate initial conditions of water level and nitrogen pressure in the SIT while the MOV in the SIT discharge piping is closed. Any initial conditions that will fully open the check valve and not inject nitrogen into the reactor coolant piping when the MOV is opened are considered adequate. Then SIT pressure and level versus time are recorded as the MOV is opened to release the water to the refueling cavity through the SIT check valves.

The most practical test setup for FCS was to dump the SITs to the reactor vessel when the vessel head was removed, the core was offloaded, and about 20 ft of water was in the refueling cavity. Figure 2 shows a schematic of the test arrangement.

The initial conditions in the SIT (i.e., water level and nitrogen blanket pressure) were determined based on the desire to have the equilibrium water level in the SIT of 0% after an isothermal expansion of the nitrogen blanket expelled the water from the SIT to the refueling cavity. This is desirable to prevent nitrogen from being injected into the reactor coolant loop. The increase in water level in the refueling cavity (~5 in.) as the SIT dumps was neglected. The initial conditions chosen for the test included an SIT level of 90% and an SIT pressure of 104 psig. These compare to normal values of 60% and 240 psig, respectively.

The test was performed by recording the SIT level and pressure on a strip chart recorder while the motor operated block valve was opened fully and then closed. The flow (Q) and differential pressure (ΔP) were calculated from the rate of change of the SIT level and the SIT pressure adjusted by the fluid level. The ΔP and Q were then used to calculate C_v in accordance with Equation (3).

In order to register the maximum C_v from the test, the flow rate after the MOV is fully opened (54 seconds after test initiation) must still be high enough (about 3,500 gpm if the valves are in new condition) to fully open the SIT check valves.

TEST RESULTS/ CALCULATIONS

The pilot test to qualify the SIT dump procedure was performed on one SIT in the spring of 1990. Figure 3 shows plots of some of the critical test parameters versus time. The calculated[b] flow rate through the check valves at the point when the MOV was 100% open was 4,462 gpm. This is above the 3,500 gpm (as stated by the manufacturer) required to fully open the check valves. The ΔP calculated at this point was 12.58 psig.

$$\therefore\ C_v\ (MEAS.) = \frac{Q}{\sqrt{\Delta P}} = \frac{4462}{\sqrt{12.58}} = 1,258\ . \qquad (4)$$

An uncertainty analysis was performed based on instrument accuracy that indicated the calculated C_v has an uncertainty $\pm 3.4\%$. This conservatively indicates that measured $C_v = 1,215$.

The acceptance criteria for this test [C_v (LOCA)] was determined by using the nuclear steam supply system (NSSS) vendor's loss of coolant accident (LOCA) analysis (Combustion Engineering, 1974). The NSSS vendor developed flow resistance coefficients (K) and effective flow area (A) that can be used in Crane's (1985) Equation 2.6.

$$C_v = \frac{29.9\ d^2}{\sqrt{K}}\ . \qquad (5)$$

When the NSSS vendor's values of K = 7.34 (for piping from SIT SI-6C) and A = 0.5592 ft^2 for the piping from the tank being dumped are inserted into Equation (5), the acceptable C_v is obtained.

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b. Omaha Public Power District, Calculation #FC-05428, "Valves SI-207/208 Full Open Stroke Data Analysis for SP-SI-7, April 1990 Performance."

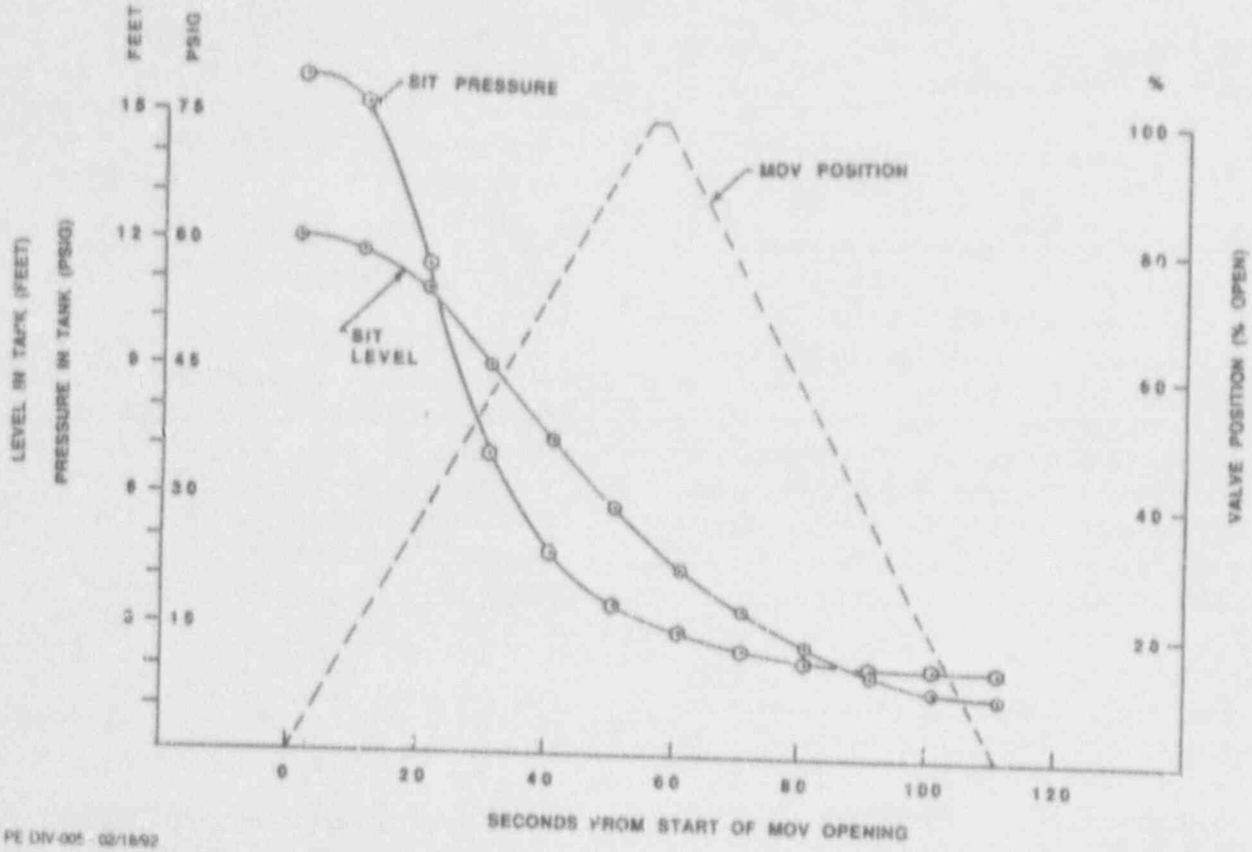

Figure 3. Results of pilot test.

$$d^2 = \frac{4A}{\pi} = \frac{4}{\pi}(.5592\ ft^2)\ (144\ in^2/ft^2)$$

$$= 102.5\ in^2 \qquad (6)$$

$$C_v = \frac{29.9 \times 102.5}{\sqrt{7.34}} = 1,131 \qquad (7)$$

The acceptance value C_v(LOCA) = 1,131 (SI-6C) is the reference value that must be exceeded to verify the NSSS vendor's calculations. The pilot test showed a C_v margin of 7.3%, demonstrating acceptability of the tested SIT check valves and the pilot test methodology.

Although the acceptance criteria was based on the NSSS vendor's LOCA analysis, it should be remembered that the intent of the ASME Section XI Pump and Valve Program is to detect component degradation. The baseline data for valves tested in the Pilot Test is C_v = 1258. This baseline value of C_v is a credible indicator of valve condition regardless of how it compares to the LOCA Analysis, because the check valves were disassembled and verified to be in satisfactory condition prior to performing the flow test.

A cost estimate on the effort required to perform the pilot test was developed to project the cost of dumping all four SITs to test the eight SIT check valves. The performance of this test is expected to cost about $2,500 with negligible impact on outage radiation exposure. It is planned to dump all four SITs for this test each refueling outage.

All four SITs were dumped for the first time during a refueling outage in February 1992.[c] The test results showed that all of the SIT check valves satisfied their respective acceptance criteria (see Table 1, tes ult summary).

Some interesting observations may be drawn from the test results (for example):

- The C_v values obtained from the test indicate that the test technique is dependable and capable of producing consistent results
- The N_2 bubble in the SIT approximated an isothermal expansion as the water was dumped from the SIT
- The apparent discrepancy between the C_v values obtained for SI-6C in 1990 as compared to 1992 appears to be the result of different initial conditions rather than a negative performance trend, but since both results meet the acceptance criteria, further investigation will be postponed until the next SIT dump.

The test is considered successful in that it demonstrated the viability of the test technique, it established credible baseline information that may be used to evaluate future test results (performance trending), and it demonstrated that the check valves performed as predicted by the NSSS designer within a reasonable tolerance.

SUMMARY

The test method described in this paper for full-flow test verification of the SIT check valves by dumping the SIT has been demonstrated to be a viable, cost-effective alternative to sample disassembly. The sample disassembly of two valves each refueling outage is estimated to cost $90,000.00, compared to an estimated cost of $2,500 for testing all eight SIT check valves by dumping the four SITs.

FCS has received interim approval from the NRC for the reduced-pressure SIT dump methodology and plans to continue to satisfy the full-flow test requirement for the SIT check valves by performing the reduced pressure SIT dump test. All eight valves will be tested by this mechanism each refueling outage.

c. Omaha Public Power District, Surveillance Test Procedure SS-ST-SI-3015, "Safety Injection Tank Discharge Check Valve Test," February 22, 1992.

Table 1. SIT dump test results summary.

Tank (Valves)	Acceptance Criteria (C_v)	1990 Pilot Results Flow	1990 Pilot Results C_v	1992 Test Results Flow	1992 Test Results C_v
SI-6A					
(SI-219 & 220)	1,189	—	—	3,563 gpm	1,206
SI-6B					
(SI-215 & 216)	1,164	—	—	4,126 gpm	1,201
SI-6C					
(SI-211 & 212)	1,131	4,424 gpm	1,258	3,897 gpm	1,184
SI-6D					
(SI-207 & 208)	1,159	—	—	4,241 gpm	1,229

NOMENCLATURE

A = Cross sectional area of pipe (ft^2)

C_v = Flow coefficient (gpm with 1.0 psi pressure drop)

K = Resistance coefficient (dimensionless)

ΔP = Differential pressure ($lb/in.^2$)

Q = Rate of flow (gpm)

d = Internal diameter of pipe (in.)

ρ = Weight density of fluid (lb/ft^3)

ACKNOWLEDGMENTS

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ASME Boiler and Pressure Vessel Code, Section XI, 1980 Edition, Winter 1980 Addendum, Subsec a IWV.

Combustion Engineering, 1974, "*LPSI Pump, HPSI Pump and Safety Injection Tank Data for New ECCS Evaluation Model*," O-PD-113 Dated March 15.

Crane, 1985, "*Flow of Fluids Through Valves, Fittings, and Pipe.*"

Nuclear Regulatory Commission, Generic Letter 89-04 "*Guidance on Developing Acceptable Inservice Testing Programs,*" Attachment 1, Item 1, "*Full Flow Testing of Check Valves.*"

Recommendations on Frequently Encountered Relief Requests

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ABSTRACT

This paper is based on the review of a large database of requests for relief from inservice testing (IST) requirements for pumps and valves. From the review, the paper identifies areas where enhancements to either the relief request process or the applicable test codes can improve IST of pumps and valves. Certain types of requests occur frequently. The paper examines some frequent requests and considers possible changes to the requirements to determine if the frequent requests can be eliminated. Recommended changes and their bases will be discussed. IST of safety-related pumps and valves at commercial nuclear power plants is done according to the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (the Code), Section XI. Because of the design and function of some safety systems in nuclear plants, performing Code testing of certain pumps and valves is impractical or a hardship without a compensating increase in the level of safety. Deviations from the Code are allowed by law, as reviewed and approved by the United States Nuclear Regulatory Commission (NRC), through the relief request process. Because of similarities in plant design and system function, many problems encountered in testing components are similar from plant to plant. Likewise, there are often common problems associated with test methods or equipment. Therefore, many relief requests received by the NRC from various plants are similar. Identifying and addressing the root causes for these common requests will greatly improve IST.

INTRODUCTION

Commercial nuclear power plants are large and complicated, containing many important fluid systems. These systems contain numerous pumps and valves. Plant operators (licensees) periodically test components, including important (or safety-related) pumps and valves, to help ensure that they are available to perform their intended functions. The Code of Federal Regulations (CFR), 10 CFR 50, gives specific directions regarding inservice testing (IST) of safety-related pumps and valves. The U.S. Nuclear Regulatory Commission (NRC) is tasked with ensuring that licensees comply with the applicable testing requirements. In certain circumstances, licensees may request relief from strict compliance with the test requirements when components are inaccessible or otherwise impractical to test. The NRC evaluates these requests and may grant relief, as authorized by the CFR.

This paper addresses the process of requesting and granting relief from the test requirements. Licensees request relief from Code requirements

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for many reasons and many component types. Some reasons are particular to a problem with a specific instrument or component installation. Other reasons are more general in nature, affecting similar components that are found at a number of plants. This paper focuses most on requests of a broad based and general nature, as determined from the authors' combined experience (16 years) in reviewing IST programs and relief requests for the NRC. Recommendations in this paper are based on this experience and are intended to aid in the review and approval process.

This paper describes many of the regulations and requirements for IST. It discusses general relief request problem areas and identifies ways to present information in requests to enhance the likelihood of having relief granted. The paper also looks at several specific areas where relief is requested from pump testing requirements, such as requests to test using reference curves instead of specific reference values or to measure pump bearing vibration in units of velocity rather than displacement. These and other pump issues are discussed and recommendations and insights are given that should help licensees prepare requests. Several valve issues are also addressed, such as the relationship between testing to assess the containment isolation or pressure isolation capability of valves, stroke time testing considerations for certain power-operated valves, and alternate testing of check valves.

REGULATIONS AND TEST CODES

The NRC is working to improve the quality of IST. This is evident in their high level of support and participation in ASME Code Committee activities, as well as their co-sponsoring of this symposium. The NRC is making great efforts to grant relief for each case where it is requested and is allowed by the CFR. As part of that effort, the NRC issued Generic Letter 89-04 (GL 89-04), "Guidance on Developing Acceptable Inservice Testing Programs." The letter contained NRC Positions on 11 frequently encountered issues. The NRC determined that relief is granted according to Paragraph (g)(6)(i) of 10 CFR 50.55a to follow the alternative testing, as delineated in Positions 1, 2, 6, 7, 9, and 10 of the letter. That effort helped to streamline the relief process.

Regulations

This section discusses the regulations applicable to IST programs at licensed commercial nuclear power plants. It also describes the process of granting relief from the testing requirements. As written, 10 CFR 50, requires testing of certain safety-related pumps and valves. With certain exceptions, these components must be tested according to the requirements of Section XI of the ASME Code, Subsections IWP for pumps and IWV, for valves. The Code subsections specify many test plan attributes, including test methods and frequencies. The testing is intended to assess operational readiness of components. Standard Technical Specification 4.0.5 and other plant Technical Specifications (TS) state that IST of ASME Code Class 1, 2, and 3 pumps and valves shall be performed in accordance with Section XI of the Code and applicable addenda identified by 10 CFR 50.55a(g).

Nuclear power plants are dynamic and complicated. Often, design constraints or operating conditions can preclude testing certain components according to the Code test method and/or frequency requirements. Alternatives to Code requirements may be used when authorized by the NRC. "Provisions for Relief from Test Requirements," as stated in 10 CFR 50, provides for the regulatory authority, NRC, to grant relief allowing modified testing or extending the test interval. Section 10 CFR 50.55a gives the following criteria for granting relief:

- Paragraph 10 CFR 50.55a(a)(3)(i) states that "the proposed alternatives would provide an acceptable level of quality and safety." Relief under this provision is generally limited to cases where the licensee proposes a method of testing that is equivalent to or better than the Code method.

- Paragraph 10 CFR 50.55a(a)(3)(ii) states that "compliance [with the Code requirement] would result in hardship or unusual difficulty without a compensating increase in the level of quality and safety." Relief is granted under this provision in cases where the proposed testing is not equivalent to or better than the Code method and the difficulty of compliance with the Code requirement would not be compensated for by an increase in safety.

- Paragraph 10 CFR 50.55a(g)(6)(i) states that "The Commission will evaluate determinations . . . that Code requirements are impractical. The Commission may grant such relief and may impose alternative requirements as it determines is authorized by law . . . giving due consideration to the burden upon the licensee if the requirements were imposed on the facility." Relief is granted under this provision in cases where the Code requirement is demonstrated to be impractical and a reasonable alternative method is proposed or imposed by the NRC. The burden on the licensee is also considered as part of the evaluation.

Test Codes

This following section discusses some of the codes and standards applicable to testing at commercial nuclear power plants. Most plants test in accordance with Subsections IWP and IWV of the ASME Code. These subsections are being replaced in the regulations (10 CFR 50) by Parts 6 and 10 of the ASME Operation and Maintenance (O&M) Standard, "Operation and Maintenance of Nuclear Power Plants, ASME/ANSI OM-1987." That Standard was rewritten and approved by the Board on Nuclear Codes and Standards in the fall of 1990 as the "Code for Operation and Maintenance of Nuclear Power Plants, ASME OM Code-1990." There are also other codes and standards that might be used for testing. Code Cases determined by the NRC to be suitable for use are identified in NRC Regulatory Guide 1.147, "Inservice Inspection Code Case Acceptability-ASME Section XI, Division 1." That guide specifically allows the use of newer Code versions in certain instances, such as Code Case N-416, which allows the use of OM-1 for testing safety and relief valves, or Code Case N-465, which allows the use of OMa-1988, Part 6, for testing pumps.

The process of writing and revising testing codes and standards is long, difficult, and complex. So is the process of writing and gaining approval for regulations to use them. Many problems and shortcomings with Section XI have been addressed in the revisions. This paper addresses problems licensees have implementing Section XI requirements and suggests ways to confront those problems. Where the authors feel problems are effectively addressed by the ASME OM Code-1990, Subsection ISTB or ISTC, for testing of pumps and valves respectively, mention will be made. Discussions will generally be limited to comparisons between the current ASME OM Code-1990 and the 1980 edition of Section XI.

The NRC recently issued the NRC Inspection Manual, Temporary Instruction 2515/114, "Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs." Site IST inspections will be done according to the temporary instruction to assess program implementation relative to the test code requirements and other licensee commitments.

GENERAL PROBLEMS WITH RELIEF REQUESTS

This section describes some of the general problems associated with relief requests. The NRC reviews and evaluates relief requests and grants relief according to the regulations. The decision to grant or deny relief from the Code requirements depends on the quality of the technical information provided in a request. The information should include the basis that justifies granting relief. Inconvenience alone is not a suitable justification. The request should thoroughly and clearly specify the alternate method of testing and frequency for each test.

Attention to detail and careful checking can greatly enhance the likelihood of getting relief granted without additional discussions, revisions, and resubmittals. Licensees can do some very simple things to enhance the request process. First, ensure the accuracy and completeness of information in each request. Simple mistakes, such as typographical errors (for component identification), can make it difficult to evaluate a request and grant relief. Second, address only related Code requirements in a request. Complicated requests addressing diverse requirements are difficult to prepare and even more difficult to grant. Often, complex requests must be broken into manageable pieces and evaluated, possibly with a different outcome than that desired by the submitter. And third, each request should deal only with components with related concerns. Stroke time testing should not be lumped with check valve testing. Generally, if the basis is not equally applicable to all the components, the request is too broad in scope.

Licensee's Basis for Requesting Relief

The basis is a very important part of a relief request and an area where many problems occur. The basis states the need (according to the regulations) for relief. Problems fit into three categories: the basis is either not complete and clear, not entirely applicable, or does not establish impracticality, burden, or hardship. The plant operator knows the plant and equipment capabilities and limitations best. The specific reasons that support the granting of relief should be thoroughly and clearly stated in requests. The basis should follow a logical sequence to ensure that all essential elements are included. It should not assume an in-depth plant or system knowledge by the reviewer, but should clearly explain why the Code testing cannot, or should not, be performed as specified. Another problem is applicability of the basis. Lumping many components, with different safety functions, in a single request makes the review process difficult and often results in additional discussions and submittals. Lastly, relief can be granted by NRC in three ways, as previously discussed, via 10 CFR 50.55a(a)(3)(i), (a)(3)(ii), or (g)(6)(i). An adequate basis must be provided to enable NRC to grant relief under one of those three provisions. The licensee should demonstrate the equivalency of proposed testing, or the impracticality, burden, or hardship (as applicable) associated with compliance with the Code.

Licensee's Proposed Alternate Testing and Frequency

The licensee's proposed alternate testing and frequency is an area where many problems occur. Often, some aspect of the proposal is either not stated, unsatisfactory or unclear. The NRC evaluates the proposed testing to determine if it is equivalent to or better than the Code-required testing, or if it is reasonable for cases where the Code-required testing is impractical or a hardship. If the proposed testing is not adequately described, it cannot be fully assessed according to the regulations. In many cases, relief cannot be granted based on the proposal. In cases where the Code test is shown to be impractical and no testing is proposed or the testing that is proposed is unsatisfactory, the NRC may impose alternative requirements according to Paragraph 10 CFR 50.55a(g)(6)(i).

Sometimes, the frequency of testing is not specified. We have seen many requests that proposed to perform a test, say a full-stroke open exercise, at a frequency such as, "when the system configuration is conducive to the test." This statement neither defines nor limits the time interval between tests. The proposed frequency cannot be determined and compared to the Code requirements. Relief cannot be granted according to the regulations to test at an undefined frequency.

Another common problem is not addressing the frequency of part-stroke exercising. This is most often a problem with check valves. Relief requests address only an extended frequency for full-stroke exercising, but fail to mention the frequency for performing part-stroke exercising. Also, the part-stroke test and its frequency are not identified in the program. The part-stroke exercise may be a very important aspect of the testing done on a valve. For check valves that cannot be full-stroke exercised with flow and that must be disassembled and inspected, it may be the only test done. Generally, when it appears that a licensee can part-stroke exercise the valve without much difficulty and the test is not addressed, the request will be granted with the provision that a part-stroke exercise be performed. Also, relief is often granted for disassembly and inspection with the provision that part-stroke exercising be done after valve reassembly.

Part-stroke exercising should be done as near the Code test frequency as practical.

For some requests, the proposed alternate testing frequency is considered unsatisfactory. A licensee may propose to full-stroke exercise and stroke time a power-operated valve every other refueling outage, yet the test could be done easily when the plant is in the cold shutdown condition. These kinds of requests are usually accompanied with a weak basis. In this case, the proposed frequency would be considered unsatisfactory, and relief should not be granted to extend the interval between tests.

Then, there is the issue of alternate testing. Again, the operator knows the plant and equipment. The licensee is ideally suited to develop alternate proposals for assessing component operational readiness when the Code requirements cannot be met. Licensees have developed some very innovative approaches to problems with component testing. For this reason, the NRC is very reluctant to prescribe alternate tests. Whenever possible, the NRC will put the burden on a licensee to develop a test method that adequately assesses operational readiness and provides a reasonable alternative to the Code.

For some requests, the proposed alternate testing or acceptance criteria is considered unsatisfactory. Sometimes, licensees propose low quality tests that do not allow an adequate assessment of operational readiness. Many of these tests rely on subjective judgements, such as an operator observing a valve's stroke for smoothness rather than timing it. Other proposals might measure parameters that are marginal or useless in their ability to indicate degradation, short of total failure. Still others might measure only part of the required parameters during a poor test, such as testing a large centrifugal pump at shutoff head and measuring and assessing only discharge pressure. In other cases, the proposed acceptance criteria might be wholly inappropriate, such as assigning a lower required action range of 6 gpm for a pump with a normal capacity of 30 gpm, or a stroke time limit of 120 seconds for a fast-acting solenoid valve.

For requests with no proposed testing, relief is usually denied or a provision is made to perform some test in lieu of the Code test. In a few cases, a test has very little value for a specific component application.

Summary

Licensees should pay attention to detail when preparing relief requests, especially the basis and proposed alternate testing sections. The Code specifies testing to assess component operational readiness. The questions that licensees should apply to each request for relief are (a) "Are the listed components appropriate and correctly identified?" (b) "Does the request correctly indicate the specific Code requirement(s) from which relief is requested?" (c) "Does the request provide the NRC an adequate basis to fully understand the issue and evaluate the issues?" and (d) "Does the proposed test ensure an adequate assessment of operational readiness and provide a reasonable alternative to the Code requirement(s)?"

PUMP-RELATED RELIEF REQUESTS

This section addresses several topics related to pump testing that are frequently seen in relief requests. The discussion of these topics identifies the affected Code requirement, problems licensees have complying with the requirement, typical licensee proposals, and issues of concern for granting relief. Some of these issues are addressed by subsequent editions of the Code or are being considered by Code Committees.

Testing Pumps Using Reference Curves

Code Paragraph IWP-3100 requires that pump flow rate or differential pressure be set at a specific reference value for testing. The quantities of Table IWP-3100-1 are then measured or observed and compared to the corresponding reference value. This comparison monitors pump condition and assesses its operational readiness. Many plants have pump systems that operate under a variety of conditions. These conditions typically result from multiple independent changes, such as cycling of flow control valves, in system cooling demand. Also, the overall system demand may vary significantly from season to season. This can make it extremely difficult or impossible to return to a specific reference value of flow rate or differential pressure for testing. Multiple reference points could be established according to the Code, but it would be impossible to obtain reference values at every possible point, even over a small range. Licensees often request to test centrifugal pumps using acceptance criteria based on a reference curve supplied by the pump manufacturer or one that is derived from points of flow rate and differential pressure measured in situ, instead of at a specified reference value(s).

Generally, the level of detail in these requests does not describe the proposed testing adequately. Many variables can affect the quality of reference curve testing. For instance, the higher the inaccuracy of the instruments used to develop a reference curve or to validate the manufacturer's curve, the higher will be the uncertainty associated with the curve and the adequacy of the acceptance criteria. The region of the pump curve in which testing is done can largely affect the quality of the test. As a general rule, testing done at very low flow rates (relative to the pump's design flow rate) is less meaningful for evaluating changes in condition than tests conducted near the pump's design flow rate. The range of the curve and the number of test points taken to establish or verify it can also have a large effect on the uncertainty associated with the reference curve and the acceptance criteria. The technique used to fit the curve to measured points should also be considered. Other questions regard establishment of complementary curves for analysis of changes in vibration. Many factors can have a negative impact on the quality of curve testing. Therefore, these factors should be understood by licensees and described in requests for relief.

The NRC generally grants relief for reference curve testing when certain conditions are met. Where it is impractical to test at a reference value of flow rate and differential pressure during normal power operation, testing in the "as-found" condition and comparing values to an established reference curve may be an acceptable alternative. Pump curves represent an infinite set of reference points of flow rate and differential pressure. Establishing a reference curve for a pump when it is known to be operating acceptably, and basing the acceptance criteria on this curve, can permit evaluation of pump condition and detection of degradation, though not in accordance with IWP. However, there may be a higher degree of uncertainty associated with using a curve to assess operational readiness. Therefore, the development of the reference curve should be as accurate as practicable. Additionally, when using reference curves, it may be more difficult to identify instrument drift.

Because it is impractical to vary the flow rate of certain pumps during normal plant operating conditions, the use of a reference curve and acceptance criteria based on the curve are an acceptable alternative to the requirements of IWP for those pumps if the following seven elements are used in the IST program when developing and implementing the reference curve(s):

1. Curves will be developed, or manufacturer's pump curves will be validated, when the pumps are known to be operating acceptably

2. The reference points used to develop or validate the curve will be measured using instruments at least as accurate as those required by the Code

3. Curves will be based on an adequate number of points, with a minimum of five

4. Points will be beyond the flat portion (low flow rates) of the curves in a range, which includes or is as close as practicable to design basis flow rates

5. Acceptance criteria based on the curves should not conflict with Technical Specifications or Facility Safety Analysis Report operability criteria for flow rate and differential pressure for the affected pumps

6. If vibration levels vary significantly over the range of pump conditions, a method for assigning appropriate vibration acceptance

criteria should be developed for regions of the pump curve.

7. When the reference curve may have been affected by repair, replacement, or routine service, a new reference curve shall be determined or the previous curve revalidated by an inservice test.

The NRC believes that the use of reference curves is not equivalent to testing at fixed reference values under IWP and that relief should be granted for these cases pursuant to 10 CFR 50.55a(g)(6)(i), based on the impracticality of varying the pump's operating condition in order to test at a specific point.

Where it is practical to test at reference values of flow rate and differential pressure, testing in the "as-found" condition and comparing values to an established reference curve is not an acceptable alternative.

The issue of reference curve testing is being considered by the ASME O&M Working Group on Pumps and Valves. The authors believe that a Code procedure for using pump curves is needed to help ensure that curve testing is of consistently high quality.

Measurement of Inlet Pressure

The Code, Paragraph IWP-3100, requires measurement of pump inlet pressure. Table IWP-4110-1 specifies an instrument accuracy of ± 2% (of full scale) for pressure. However, many pumps (such as vertical line-shaft and some positive displacement pumps) have no installed instruments. This results in many requests for relief. The proposed alternatives vary widely, from doing nothing to calculating inlet pressure from the height of liquid above the pump suction. Quite often the accuracy of the calculated inlet pressure value is not mentioned. The Code does not specify accuracy requirements for calculated parameter values. It does not specify range or accuracy requirements for level gauges, yardsticks, or the other devices that might be used to measure the fluid level above the pump suction point.

Relief is generally granted in these cases to calculate inlet pressure, provided the calculation meets the accuracy requirement of Table IWP-4110-1 for direct measurement of inlet pressure. The ASME O&M Code Committee is aware of this problem. A proposed change to ISTB states: "If a parameter is determined by analytical methods instead of measurement, then the determination shall meet the parameter accuracy requirement of Table ISTB 4.6.1-1. . . ." The authors believe that the accuracy requirements for calculated parameter values are adequately addressed in the proposed revision to ISTB.

Positive Displacement Pump Relief Requests

The Code, Paragraph IWP-3100, requires measurement of pump differential pressure for all pumps. However, many positive displacement pumps are not equipped with inlet pressure gauges. In other cases, the installed inlet pressure instrument does not meet the accuracy requirement of Table IWP-4110-1 for pressure, ±2% of full scale. Licensees frequently request relief from either the instrument accuracy requirements or from measuring inlet pressure, differential pressure, or both. The requests generally propose to evaluate the pump condition by assessing changes in discharge pressure.

Hydraulic degradation of positive displacement pumps is evident by evaluating discharge pressure rather than differential pressure, as required by the Code. The discharge pressure of a positive displacement pump depends on the pressure of the system into which it is pumping. It is not significantly affected by either flow rate or inlet pressure as long as adequate net positive suction head exists. Inlet and differential pressure are not meaningful parameters to determine if hydraulic degradation is occurring. For positive displacement pumps, measurement of pump discharge pressure allows an assessment of operational readiness that is essentially equivalent to that of the Code. In virtually all cases, relief is granted to assess positive displacement pumps using discharge pressure in lieu of differential pressure. Also, ISTB does not require measurement of pump inlet pressure. Paragraph ISTB 5.2 and Table ISTB 5.2-1 specify the use of, and acceptance criteria for, discharge pressure for positive displacement pumps. Therefore, the authors believe this issue is adequately addressed in that version of the Code.

Vibration Measurement Method

The Code, Paragraph IWP-4510, states requirements for measuring pump vibration amplitude. The OM Code offers an improved method of assessing the mechanical condition of pumps that incorporates vibration velocity measurements and other attributes. The NRC and many in the industry believe that the OM Code method is much better than the method in IWP. NRC Regulatory Guide 1.147 approves of the use of Code Case N-465, which specifically allows the use of OM-1988, Part 6. Part 6 contains essentially the same vibration monitoring program as ISTB Many licensees request relief from measuring pump bearing vibration under IWP-4500 and propose to measure vibration in either displacement or velocity units according to a program that incorporates certain aspects of the OM Code.

There are distinct advantages of using a program that measures vibration velocity to evaluate the mechanical condition of certain pumps. For pumps with a high rotating shaft speed, measurement of vibration velocity can allow a better assessment of a pump's mechanical condition than displacement. This is widely acknowledged in the industry. A pump testing program using vibration velocity readings taken over a wide frequency range, such as that in ISTB, can provide more information about pump mechanical condition than could be obtained from vibration displacement readings.

Proposals may be similar to the program described in ISTB, but differ in the approach or lack certain elements, such as the location of measurements, acceptance criteria, or the frequency response of the instrument for measuring vibration. In most cases, no justification is provided for the differences. This is not to say that the differences cannot be justified, but only that no information is provided to justify the difference. Sometimes, only the acceptance criteria are provided. In most cases, some additional information is needed to evaluate and approve the proposed program with its differences from the ISTB program. If specific requirements of ISTB are deemed impracticable, the licensee should develop and propose alternates that adequately assess pump condition.

A pump vibration monitoring program performed in accordance with all the applicable vibration testing requirements of OMa-1988, Part 6, or a newer version through ISTB-1990, is acceptable to the NRC. It gives an adequate level of quality and safety and provides a reasonable alternative to the Code vibration measurement requirements.

Vibration Measurements for Submerged, Vertical Line-Shaft Pumps

The Code, Paragraph IWP-4510, specifies that pump vibration displacement measurements be taken on a pump bearing housing or its structural support. However, many submerged, vertical line-shaft pumps have their lower pump coupling bearings located below the pump support structure level, often submerged. The bearings are inaccessible during testing; therefore, direct measurement of vibration at the Code-specified location is impractical. The OM Code, Paragraph ISTB 4.6.4, also states requirements for the location of vibration measurements. It allows vibration measurements to be taken on the upper motor bearing housing of the pump for vertical line shaft pumps. In some cases, licensees state that taking measurements at the upper motor bearing housing is impractical. In these cases licensees request relief from the location of measurement requirement and submit proposals to address this problem. Licensees also request relief from the vibration acceptance criteria for measurements taken at the upper motor housing because of high motor vibration levels.

Pump-related vibration levels measured at the upper motor thrust bearing housing can be significantly attenuated by intermediate materials. Pump vibration can also be masked by motor vibrations. These effects make it difficult to evaluate pump mechanical condition based on vibration readings taken at the upper motor bearing housing. Yet, this is generally considered the best available location for remotely measuring vibration to assess the mechanical condition of submerged pumps. ISTB allows measurement on the upper housing and specifies the number and location of measurements. It also specifies acceptance criteria for vibration displacement and velocity, instrument accuracies, and other program aspects. The NRC approves the use of the ISTB vibration testing requirements in their entirety.

However, there are some valid concerns about the ability to assess a submerged pump's mechanical condition based on vibration readings taken on the motor. These concerns are underscored when licensees propose to relax the acceptance criteria. An Electric Power Research Institute (EPRI) report, *On-Line Vibration Monitoring for Submerged Vertical Shaft Pumps*, EPRI NP-5704M, was issued in March 1988. It discusses problems associated with assessing pump condition based on vibration measurements taken at remote locations. The report suggests methods for monitoring vibration on submerged pumps, including attaching motion sensors on the submerged pump housing.

Again, the upper motor thrust bearing housing is usually the best location for remotely measuring vibration of submerged pumps. However, in some cases that location may not be accessible for vibration measurement. In those cases, something needs to be done to adequately assess pump condition and determine operational readiness. Some installations might provide another location on the motor with an equivalent or better indication of pump condition. However, relief requests generally do not provide data to support this case. If an adequate assessment of pump condition cannot be made by vibration measurements taken at an alternate location, the licensee should consider other methods, such as those described in the EPRI report.

Regarding relaxed acceptance criteria applied to vibration measurements taken on the motor housing, the measured levels attributable to the pump are likely to be lower (possibly much lower) than the actual pump levels. Because of the reduced levels attributable to the pumps, the Code-specified vibration alert and required action levels might be of marginal value to detect pump degradation. It might be much more appropriate to reduce the acceptance ranges because of attenuation of pump-related vibration. We often see requests to increase the acceptance criteria (allow higher multiples and absolute values). Upon closer inspection, the problems with meeting the specified ranges usually stem from high vibration levels of the motors, whether resulting from a degraded condition, poor mounting, or vibration of nearby or associated equipment. Licensees should look closely at problems with vibration measurements for submerged pumps and ensure that the acceptance criteria are established to assess pump operational readiness and not solely to ensure measurements in the acceptable range, irrespective of pump condition.

Licensees should ensure that the proposed alternate measurement location and acceptance criteria will adequately assess pump condition. This implies that the correlation of measured vibration data with actual pump conditions is known and that the acceptance criteria will require corrective action prior to pump failure. If the relationship between these is unknown, then the proposed location and acceptance criteria might not be justified (or suitable) for the application. In these cases, licensees should do one of the following: (a) measure vibration on the upper motor thrust bearing housing in accordance with all the applicable requirements of ISTB for vibration measurement or (b) justify an alternate measurement location, method, and acceptance criteria that adequately assess operational readiness of the affected pumps.

The authors believe that alternate methods of assessing the mechanical condition of submerged pumps, such as attaching accelerometer or displacement probes to the pumps (as described in the EPRI report), are needed to adequately assess the operational readiness of many of these pumps. This problem should be considered and addressed by the ASME Code Committees.

Hydraulic Acceptance Criteria

The Code, Paragraph IWP-3100, specifies acceptance criteria for a pump's hydraulic performance. In many cases, a pump's hydraulic capacity greatly exceeds the system design requirements. This is often the case for pumps in the emergency diesel generator fuel oil transfer system and sometimes the case for pumps in other systems, such as the cooling system for the spent fuel pool.

In one case, a pump with a capacity of approximately 28 gpm was installed in a system with a design requirement of only 3.3 gpm. There were additional complications in this case, such as instrument accuracy and repeatability, and the ability to duplicate the system test configuration (for flow rate). The licensee stated that the pump could meet its system design function with greater than 75% degradation and that the pump would function reliably with 25% hydraulic degradation, provided that bearing vibration was not excessive. The licensee proposed to measure flow with a survey flow meter with an approximate ±5% accuracy and to assign corrective action ranges of 15% flow rate degradation for alert and 25% flow rate degradation for the required action. The authors believe that expanded hydraulic performance acceptance ranges may be appropriate and justifiable for certain cases where there is a high level of confidence that the pump remains capable of performing its required function at the degraded level. However, the relaxed acceptance criteria should consider all uncertainty (i.e., instrument inaccuracy) associated with the measured parameter values. Otherwise, the pump may degrade and fail before the condition is detected and acted upon.

In the case just described, the information provided did not show that the proposal was equivalent to or a reasonable alternative to the Code. The licensee was asked to evaluate methods for more accurately measuring or determining pump flow rate and to reconsider and justify the alternate allowable ranges to ensure that they require corrective action before the pump's condition degrades to the point where there is an increased possibility of failure. In cases where relief is determined to be necessary and alternate ranges are proposed, licensees should consider the instrument inaccuracies and other uncertainties when setting the alternate limits. In no case should acceptance criteria be set below the point where operational readiness cannot be assured. It is the authors' opinion that the pump manufacturer should be consulted or a detailed analysis be performed any time the Code acceptance criteria are relaxed. Consulting the manufacturer will help to ensure the proposed alternate limits are appropriate for the specific application of that equipment.

Summary

Testing pumps in the "as-found" condition using reference curves may be acceptable in cases where it is impractical to establish a reference flow rate or differential pressure. When using this method, the licensee should follow the seven elements listed earlier in this section. For pump installations without inlet pressure instruments, the pressure may be determined from the level of fluid above the pump suction. In these cases, the calculated pressures should be determined as accurately as the Code requires. Pump inlet and differential pressures are not indicative of pump hydraulic condition for positive displacement pumps. Relief can be obtained from measuring these parameters if pump flow rate and discharge pressure are measured and evaluated against appropriate acceptance criteria.

The vibration monitoring program of OMa-1988, Part 6, or any newer version through ISTB-1990, is equivalent to or better than the vibration program of Section XI. Licensees may use these requirements for vibration measurements, provided they use them in their entirety except for cases where this is shown to be impractical. In these cases, the licensee should develop and propose alternatives that adequately assess pump operational readiness. Vibration measurements on submerged, vertical line-shaft pumps should be taken in accordance with the vibration measurement requirements of OMa-1988, Part 6, or any newer version through ISTB-1990, if practicable. If licensees deviate from these requirements, they should ensure that the proposed alternative (i.e., measurement location and acceptance criteria) will adequately assess pump operational readiness.

When pump capabilities greatly exceed the system design requirements, expanded hydraulic performance acceptance ranges may be appropriate and justifiable for cases where there is a high level of confidence that the pump remains capable of performing its required function at the degraded level. However, the relaxed acceptance criteria should consider all uncertainty (i.e., instrument inaccuracy) associated with the measured parameter values.

VALVE-RELATED RELIEF REQUESTS

This section addresses several topics related to valve testing that are frequently seen in relief requests. The discussion of these topics identifies the affected Code requirement, problems licensees have complying with the requirement, typical proposals, and issues of concern for granting relief. Additionally, some of these issues are addressed by subsequent editions of the Code or are being considered by Code Committees.

General Requests

Containment Isolation Valves/Pressure Isolation Valves. The relationship between the leak rate testing requirements of Section XI of the Code and 10 CFR 50, Appendix J, is frequently not understood. The Code, Paragraph IWV-3420, states requirements for leak rate testing Category A valves. Appendix J, states requirements for leak rate testing containment isolation valves (CIVs). The NRC staff determined that the leak test procedures and requirements for CIVs specified in Appendix J are equivalent to the requirements of IWV-3421 through -3425. In addition, the licensee must comply with the Analysis of Leakage Rates and Corrective Action requirements of Paragraphs IWV-3426 and -3427(a). Many licensees request relief from the test method requirements of Paragraph IWV-3427(b) for various Category A valves and propose to test them according to GL 89-04 Position 10, "Containment Isolation Valve Testing." The valves for which relief is requested are usually CIVs. But, the valves often perform other important leakage restriction functions. Testing under the Code, or Appendix J and Paragraphs IWV-3426 and -3427(a), as specified in Position 10, gives adequate assurance of the operational readiness of CIVs for performing the containment isolation function. However, that testing may not adequately assess their ability to perform other important leakage restriction functions, such as pressure isolation. There have been several cases recently where licensees have misunderstood this position.

The requirements of Paragraph IWV-3427(b) are applicable to Category A valves that perform a leakage restriction function other than or in addition to containment isolation. Valves in this group include pressure isolation valves, even if they also perform a containment isolation function. If Paragraph IWV-3427(b) is not followed, the testing does not provide a reasonable alternative to the Code for these valves. Relief is granted to test the containment isolation function under GL 89-04, Position 10. Relief is limited, however, to assessing the containment isolation capability. Relief from Paragraph IWV-3427(b) applies only to testing of the containment isolation function. For Category A valves that perform any other leakage restriction function in addition to or other than containment isolation, the requirements of Paragraph IWV-3427(b) should be met.

Verification of Valve Position Indication Accuracy. The Code, Paragraph IWV-3300, states requirements to verify that valve position indications are accurate. Many valves, such as sealed solenoid-actuated valves, are totally enclosed. They cannot be observed to verify that their operation is accurately indicated. Their correct positioning can be very important to the safe operation of the plant. Though it may be difficult to do, valve position indication should be verified periodically to ensure that it accurately reflects valve position.

Licensees propose alternative tests that vary from doing nothing, to observing system operational parameters, such as pressure differential or flow, to verify the accuracy of the indication. Even though the accuracy of valve position indication cannot be verified by visual observation, it should be done by some positive means, such as observation of pressure or flow, using appropriate objective acceptance criteria. The OM Code-1990, Paragraph ISTC 4.1, allows use of other indications to verify the accuracy of valve position indication. The authors believe this issue is adequately addressed in ISTC.

There is also some confusion as to whether Note 1 to Table IWV-3700-1 excludes Code Category B passive valves from the requirements of Paragraph IWV-3300 for verification of valve position indication accuracy. In the 1974 edition of the Code, all valves with remote position indication, except those specifically excluded from the scope by Paragraph IWV-1300, were required to be checked. In ASME Standard OMa-1988, Part 10, and all subsequent editions through the OM Code-1990, Subsection ISTC, Category B passive valves with remote position indication must be checked. Additionally, incorrect position indication on a Category B passive valve could have disastrous effects on system operability during an accident. The NRC believes, as do the authors, that the requirements of Paragraph IWV-3300 apply to Category B passive valves with remote position indication.

Power-Operated Valve Stroke Time Testing

The Code, Paragraph IWV-3413, states requirements for stroke time testing of power-operated valves. Many valves are difficult to stroke time as required. Some are not equipped with position indication or manual control switches. Others receive control signals from pressure switch/pressure programmers or level controllers, without either a manual control switch or a suitable valve position indicator. Licensees frequently request relief from the stroke time measurement requirements because of these sorts of problems. Proposed testing varies from observing valve operation, e.g., observation of appropriate system responses, to no test at all.

In some cases, the valve's stroke can be timed, but that test is not practical on a quarterly basis. For example, it may be impractical to quarterly test a valve whose actuation is controlled by changes in a system parameter, such as pressure. However, the associated pressure instrument is calibrated at some frequency, and during calibration it may be feasible to input a signal that demands full opening (or closing) of the valve. The full-stroke time could probably be measured as required by the Code during that calibration, but at a reduced frequency. However, because this testing involves instrument response time and delays from other control elements, it may be difficult to obtain accurate stroke times for many of the valves in this group.

In many cases, it is practical to verify stroking of valves that are not equipped with position indication by observing changes in system pressure or flow. However, because this testing could involve delays from instrument loop response time or system dynamic response, it may be difficult to obtain accurate stroke time measurements for many of the valves in this group. Although it is difficult, methods and acceptance criteria are needed to ensure that severely degraded valves are declared inoperable and repaired or replaced, as needed. Though stroke timing is specified in the Code to assess degradation, other methods might be just as effective. Recently, ultrasonics, magnetics, and acoustics have shown promise for their utility in stroke timing and monitoring the condition of power-operated valves. Licensees submitting these types of requests should evaluate alternate methods for assessing valve operational readiness. The authors believe this issue should be considered by the ASME Code Committees.

Check Valves

The most frequent topic of relief requests relates to the testing of check valves. Some requests are to use closure testing for series check valves in pairs. Most of the requests are to perform disassembly and inspection in lieu of full-stroke exercising open with flow. A small percentage propose to verify closure capability by disassembly and inspection. Most of these requests (for exercising open) are granted by the provisions of GL 89-04, Position 2, and need not be specifically evaluated. In many cases the requests appear to conflict with Position 2, or propose an alternative to the generic letter position. Some propose an extension of the inspection interval, but address only part of the issues raised in the position.

Series Check Valve Tests. The Code, Paragraph IWV-3522, states the method requirements for testing check valves. Many situations involve check valves located in series with other check valves with no provisions, such as test taps, for verifying closure individually. Many of the valves in this situation are in systems that supply fluid makeup to injection systems of boiling water reactors. They are usually simple check valves that are not equipped with position indication or external operators. Their safety function is to close to prevent loss of the injection system fluid. A demineralized water system or a dedicated keep-fill, or jockey, pump usually supplies the flow through these valves. Licensees generally request relief to test these series valves as a single unit.

Individually verifying the reverse closure capability of each of these series valves with differential pressure is impractical at any frequency. Yet, in many of the cases covered by this discussion, the closure capability of the series pair can be verified by leak testing the pair. Often, that test can be done quarterly when the associated injection system is tested. Testing the pair closed as near as practical to the Code frequency may be the best that can be done without performing system redesign and modification. However, where the pair test is done in lieu of the Code test (individual valve closure testing), both valves in the pair should be declared inoperable and repaired or replaced if excessive leakage is noted through the pair. This closure test should be performed quarterly or during cold shutdowns, as practicable. A (pair) test of this type is preferred over valve disassembly and inspection, which is sometimes proposed.

Summary

Testing per 10 CFR 50, Appendix J, and Section XI, Paragraphs IWV-3426 and -3427(a), as specified in Position 10 of GL 89-04, gives adequate assurance of the operational readiness of CIVs for performing the containment isolation function. However, the requirements of Paragraph IWV-3427(b) are applicable to Category A valves that perform a leakage restriction function other than or in addition to containment isolation. The accuracy of valve position indication should be verified periodically for all valves. The accuracy of the remote position indication for Category B passive valves should be checked under the Code.

Many valves are difficult to stroke time as required by the Code. However, licensees should develop methods to ensure that severely degraded valves are declared inoperable and repaired or replaced, as needed. Check valves located in series with no provisions, for verifying closure individually can be tested in series in some cases. That test should be done as near as practical to the Code frequency.

ADDITIONAL RELIEF REQUEST TOPICS

This paper has discussed several common relief requests in detail. The following section briefly discusses additional issues. The section presents the topics and usual problems, and indicates some of the items that should be considered and addressed by licensees requesting relief in these areas.

General Relief Requests

- Licensees request relief from performing tests [illegible] may result in entry into a Technical Specification [illegible] (TS) Limiting Condition for Operation (LCO) [illegible]on Statement.

- Licensees request to defer component testing stating only that the test will cause entry into an LCO Action Statement.

- The NRC grants relief in cases where testing could cause personnel harm, damage equipment, unnecessarily challenge a safety system, or cause a plant trip. Usually relief is not granted when the sole justification is that testing could cause entry into an LCO Action Statement. The licensee should address any applicable concerns as discussed above. If a case exists where entry into an LCO causes an extreme hardship or burden, that case should be fully described in the request.

Pump-Related Relief Requests

Licensees request not to measure pump bearing temperature (IWP-3100).

- Bearing temperature is not considered a good indicator of bearing condition, especially when measured only once per year. That measurement can be affected by a great variety of factors not related to bearing condition.

- Requests should indicate the proposed alternate testing. They should state, for instance, whether the vibration testing program of OM-6 will be used at the plant. That program represents an improvement in the technique for assessing pump bearing mechanical condition and is a suitable alternative to measuring bearing temperature. Generally, the case can be made that taking the measurement is a hardship not compensated by an increase in the level of quality or safety.

Licensees request relief from the Code instrument range requirements for ultrasonic flow rate or vibration measuring instruments (IWP-3210).

- Digital instruments, often have many selectable or automatically selected ranges. The accuracy of the instrument is generally independent of range.

- Requests of this sort should describe the instrument accuracy and the repeatability for each specific application (grouping may be appropriate in certain cases). The requests should also describe the steps taken to improve the repeatability of these instruments.

Licensees request relief from the required duration of pump tests (IWP-3500).

- The suction sources or discharge collectors for some pump systems cannot contain the quantity of fluid needed to meet the run-time requirements of either IWP-3500(a) or (b).

- Licensees should describe the method used to test the pump and show that it is impractical, if that is the case, to use other methods to meet the duration-of-test requirement. Additionally, if information is available from the manufacturer, or it can be shown that the pump can reach stable conditions in the short time available for the test, this should be presented to help justify the granting of relief.

Licensees request relief from the vibration instrument frequency response requirements for slow-speed pumps IWP-4520(b).

- Licensees with certain slow speed pumps (rotational speeds near 180 rpm) cannot calibrate their instruments to meet the accuracy specified in the Code.

- The licensee should show that compliance with the Code instrument frequency response requirements are impractical or show that meeting the Code frequency response at low frequencies would be a hardship without a compensating increase in the level of quality and safety. For the latter case, an instrument may be very close to meeting the response requirement and the replacement would be very expensive. Another possible alternate, would be to propose a different test method that adequately indicates pump bearing mechanical condition. Still another possibility, would be to show that the phenomenon of interest (e.g., oil sling in journal bearings) is not applicable to the particular pump design.

Valve-Related Relief Requests

Licensees request to use plant TS to determine operability instead of IWV-3417(b). The plants feel that their TS adequately define operability requirements.

- IWV-3417(b) states that valves that fail the stroke time test or fail to exhibit the required change in stem or disk position shall be declared inoperative and the condition corrected before startup. Some of these valves may not be needed for plant operation. It may not be feasible to correct the condition prior to startup.

- A valve that is declared inoperable in accordance with IWV-3417(b) need not be repaired or replaced prior to plant startup if the following items are true: (a) the valve and its associated system are not required to be operable by plant TS, and (b) the valve or its associated system is specifically addressed by the TS operability criteria.

If a safety-related valve and its associated system are not specifically covered by TS operability requirements, the valve should be repaired or replaced before startup unless the licensee specifically justifies doing otherwise.

Licensees interpret the Code to require the performance of a stringent fail-safe test. This test is considered to be different from a normal test, which may actually cause a loss of actuator power.

- IWV-3415 states that valves with fail-safe actuators shall be tested by observing the operation of the valves upon loss of actuator power. Licensees request relief to perform this test while exercising valves using normal valve operating controls.

- Licensees should determine whether actions taken to stroke valves during periodic IST meet the Code requirement to cause a "loss of actuator power." If so, no special test is needed. For instance, the operation of the position control switch for solenoid valves typically results in disruption of actuator power to those valves and satisfies the fail-safe test requirement. Similarly, venting the air from the actuator of an air-operated valve typically causes a loss of actuator power.

Licensees request relief from test supervisor qualifications per PTC-25.3-1976.

- Licensees have problems meeting the engineering degree and/or 2 years experience in fluid-flow requirements of PTC-25.3. They do not think that supervising this testing requires this level of training and experience.

- Periodic surveillance testing is performed to verify the set pressure of relief valves and not to qualify their relief capacity. Therefore, the requirements for 2 years experience in fluid flow may constitute a hardship without a compensating increase in the level and quality of safety. Since the test supervisor evaluates test data and makes determinations of component operability, the other requirements for people in this position are considered to be appropriate.

Licensees request relief from cold shutdown testing under OM-10, Paragraph 4.2.1.2(g) or OM Code-1990, Paragraph ISTC 4.2.2(g).

- Many cold shutdowns are of short duration. They may be planned or the result of equipment problems that can be fixed in a short time after the plant reaches the cold shutdown condition. Testing all the valves that are identified to be tested in the cold shutdown condition during each cold shutdown can present a significant financial burden on utilities.

- Often, problems occur when licensees request this relief and add additional conditions to the cold shutdown testing discussion, for example, during cold shutdowns "when conditions permit," or "when the system is not needed." The OM-10 and OM Code-1990 requirements state that, "Valve exercising during cold shutdown shall commence within 48 hr of achieving cold shutdown, and continue until all testing is complete or the plant is ready to return to power. For extended outages, testing need not be commenced in 48 hr provided all valves required to be tested during cold shutdown will be tested before plant startup. However, it is not the intent of this Subsection to keep the plant in cold shutdown in order to complete cold shutdown testing." Any deviation from this provision should be identified and justified.

Licensees request relief from the requirement to measure the stroke time (IWV-3413) of system control valves that perform a fail-safe function.

- Most power-operated valves in this category perform system control functions, such as pressure, temperature, or flow control. Many of these valves are not equipped with position control switches and are difficult to accurately stroke time. Licensees interpret the Code to require only fail-safe testing and not stroke timing as for other power-operated valves.

- Power-operated valves that have a required fail-safe function and that perform a system control function must be fail-safe tested. They should also be stroke time tested according to the Code. Derivations from the Code requirements should be justified in a relief request.

Licensees submit discussion in their IST program as relief requests however many of them are actually cold shutdown justifications.

- Often licensees submit requests for relief and propose to test valves during cold shutdowns.

- IWV-3412(a) and -3522 provide for testing valves at cold shutdowns when it is impractical to test them quarterly during power operation. The Code requires this testing to be specifically identified by the owner. Since the proposed testing in these relief requests is in accordance with the Code, these tests should not be included as relief requests, rather, they should be identified as Cold Shutdown Justifications in the IST program.

Licensees request relief to use Appendix J leak rate testing to verify closure capability of check valves.

- IWV-3522 (a) states: "Confirmation that the disk is on its seat shall be by visual observation, by an electrical signal initiated by a position indicating device, by observation of appropriate pressure indications in the system, or by other positive means. These relief requests generally deal with the frequency of testing.

- Verifying check valve closure by Appendix J leak rate testing meets the criteria in IWV-3522(a) and is generally found to be acceptable. The main problem with these requests is with the frequency of the proposed alternate testing. Appendix J testing is generally performed during refueling outages while check valves are required to be exercised quarterly or during cold shutdowns. The fact that Appendix J testing is performed during refueling outages by itself, is not a sufficient justification not to test check valves at the Code specified frequencies. The licensee should meet the Code or provide an adequate basis demonstrating that the only practical way to test the valves is leak rate testing and that testing quarterly or during cold shutdowns is either impractical or a hardship without a compensating increase in the level of quality and safety.

CONCLUSION

The NRC grants relief based on the specific situation and on the merit of the licensee's proposal. There are many differences between nuclear power plants in design, construction, and operating conditions. Because of these differences, similar relief requests can be evaluated with a different outcome at different plants. However, by considering the guidance in this paper and addressing these and other pertinent issues, the licensee is more likely to receive a favorable review of requests for relief from the Code requirements stated in 10 CFR 50.

Session 2C

MOV Diagnostic Application

Session Chair
Mark Pittman
Virginia Power

Lessons Learned From Validation Testing of Diagnostic Systems for Motor-Operated Valves

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ABSTRACT

Nuclear power plant utilities are responding to Generic Letter 89-10 and Generic Issue 87 by accelerating efforts to perform diagnostic in-plant testing of motor-operated valves (MOVs). Various vendors are developing diagnostic equipment for such testing and supplying it to the utilities. During the spring of 1991, the Idaho National Engineering Laboratory participated, at the request of the MOV Users Group and most of the vendors of MOV diagnostic equipment, in a test program to demonstrate the vendors' ability to meet their stated accuracy claims. Among other things, the program served to document the vendors' accuracy claims in a single report for easy reference, and it allowed the vendors to check their claims on a calibrated test device and revise them if necessary. This paper presents an overview of the lessons learned during the test program, including calibration and zeroing instruments, data sample rates, the conversion of spring pack displacement (or force) to stem torque, the conversion of stem torque to stem thrust, and error margins.

INTRODUCTION

The Idaho National Engineering Laboratory (INEL) is performing motor-operated valve (MOV) research in support of U.S. Nuclear Regulatory Commission (NRC) efforts regarding Generic Issue 87 (GI-87), "Failure of HPCI (high-pressure coolant injection) Steam Line Without Isolation," and Generic Letter 89-10 (GL-89-10), "Safety-Related Motor-Operated Valve Testing and Surveillance." The research includes a recent test program conducted at the INEL at the request of NRC and in cooperation with the nuclear industry's MOV Users Group (MUG). The program consisted of testing commercially available MOV diagnostic equipment on an instrumented, calibrated test stand capable of simulating valve closure. Measurements taken from the instrumented test stand served as a standard against which the measurements taken by the commercial diagnostic equipment could be compared.

The test stand, built by the INEL, is called the motor-operated valve load simulator (MOVLS). The MOVLS can simulate the loadings imposed on a 900-lb class, 6-in. valve closing against differential pressure loadings up to 1000 psid. The MOVLS, shown in Figure 1, is powered by a

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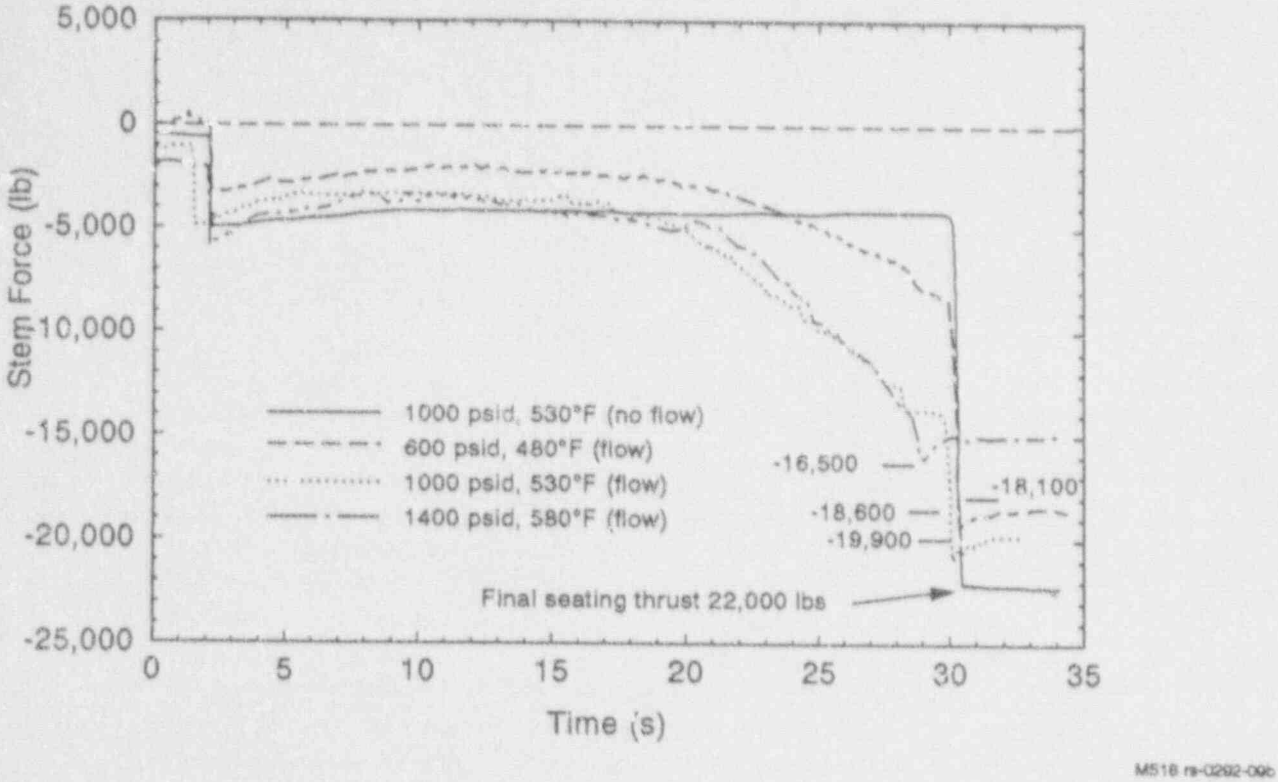

Figure 1. The INEL motor-operated valve load simulator.

Limitorque[b] SMB-0-25 motor operator mounted on the yoke of a 6-in. Velan reactor water cleanup (RWCU) system containment isolation valve. The valve stem for this program was a 1-3/4-in.-diameter, 1/4-in.-pitch-and-lead, ACME thread stem, also manufactured by Velan.

The valve stem is equipped with an instrumented arm that simultaneously measures torque and serves as an anti-rotation device. At the end of the valve stem is a thrust bearing mounted on a specially designed Lebow load cell that measures stem force. The load cell is attached to a hydraulic cylinder that discharges fluid to an accumulator as it is compressed during simulated valve closure. Different but typical valve stem load profiles can be simulated with different initial amounts of water and gas overpressure in the accumulator. This configuration is capable of a 6-in. stroke with a stem speed of 12 in. per minute. For this test program, however, the last part of the stroke would provide the important data, so the stroke length was limited to about 2 in. to reduce the amount of data that would have to be collected and processed.

The MOVLS is instrumented to provide direct measurements of every parameter currently measured either directly or indirectly by the commercially available MOV diagnostic systems. All MOVLS measurements are recorded on a data acquisition system (DAS) consisting of a MEGADAC 2200C interfaced to an IBM System/2 personal computer. The MEGADAC is a high-speed data-acquisition, signal-conditioning, data-recording system, capable of a continuous sampling rate of 20,000 samples per second. The MEGADAC provides amplification, multiplexing, and analog-to-digital conversion of up to 128 channels of differential input. During the validation testing, all the channels that were used

b. Mention of specific products or manufacturers in this document implies neither endorsement, preference, nor disapproval by the U.S. Government, any of its agencies, or EG&G Idaho, Inc., of the use of a specific product for any purpose.

were sampled at a rate of 1000 samples per second per channel. The IBM personal computer was used to control the MEGADAC, process the test data, and analyze the data for both quick-look plots and final plots. The MOVLS instrumentation and end-to-end accuracies (including the data-acquisition system) are shown in Table 1.

Each of the six major vendors of MOV diagnostic equipment came in series to the INEL for one week to test their equipment on the MOVLS and to demonstrate their ability to meet their own accuracy claims. The participating vendors were (in alphabetical order) ABB Impell (OATIS), ITI MOVATS, Liberty Technologies (VOTES), Siemens KWU, Teledyne Engineering, and Wyle Laboratories (AVMODS). The MOV Users Group and the vendors agreed to a standardized test plan, which was administered by a test coordinator and a utility host, both of whom were utility personnel familiar with in-plant testing. At the request of MUG the INEL's role was support and service. This effort was funded by the NRC, with the understanding that the MUG final test report would be public. The report would provide the industry and the NRC with insights into the accuracies of the vendors' diagnostic equipment. These accuracies would be incorporated in the utilities' GL-89-10 programs.

Table 1. End-to-end accuracies of MOVLS instrumentation.

Tag	Description	Full Scale	Accuracy (%)	Error	Units
CHAN16	Motor current I1, rms	20	0.141	0.0282	amps
CHAN17	Motor current I2, rms	20	0.094	0.0189	amps
CHAN18	Motor current I3, rms	20	0.119	0.0237	amps
CHAN19	Motor voltage L1-2, rms	575	0.090	0.516	volts
CHAN20	Motor voltage L2-3, rms	575	0.095	0.546	volts
CHAN32	Stem force	40,000	0.361	145	lb
CHAN33	Torque spring force	5,000	0.113	5.67	lb
CHAN34	Motor current I1, abs	400	0.215	0.86	amps
CHAN35	Motor current I2, abs	400	0.513	2.05	amps
CHAN36	Motor current I3, abs	100	1.056	1.06	amps
CHAN48	Stem torque #7	400	0.926	3.70	ft-lb
CHAN48	Stem torque #8	400	1.016	4.06	ft-lb
CHAN57	Open switch	—	—	—	—
CHAN59	Close switch	—	—	—	—
CHAN64	Valve stem position	5	0.318	0.0159	inch
CHAN66	Torque switch	—	—	—	—
CHAN67	Motor power, 3-phase	40	0.206	0.0825	kW
CHAN68	Motor power factor	1	1.009	0.0101	—
CHAN69	Torque spring position	1	0.161	0.00161	inch
CHAN75	Motor speed	1,700	0.108	1.83	rpm

Additional information about this test program is documented in the final report, *MUG Validation Testing as Performed at Idaho National Engineering Laboratory* (published February 3, 1992, four volumes), which is available in the NRC Public Document Room.

LESSONS LEARNED

Calibration

Many of the diagnostic systems tested on the MOVLS used linear devices that responded well to change, but the accuracy of the measurements was sensitive to the initial calibration of the devices. In fact, in situ calibration was probably the most significant single problem encountered. Most of the devices used in diagnostic testing are calibrated by the manufacturer for accuracy of elongation, radial expansion, or other measurements, depending on the device. However, once installed, the devices require in situ calibration to relate the measurement to the actual MOV parameter of interest (e.g., to relate elongation to stem force). This requires a zero-balancing step and possibly some form of in situ calibration.

Zero-balancing a transducer installed on an MOV is not always an easy undertaking. For thrust and torque measurements, test personnel may need to rotate the handwheel and try to "feel" for zero load, or they may cycle the valve several times and look for the zero load steps in data traces at the transition between opening and closing. The success of this method depends almost completely on the training and experience of the test personnel and on their ability to distinguish very minor disturbances in the instrument traces.

We can improve the success of the zero balance and calibration steps by performing some simple checks. Different diagnostic instrument measurements can be compared to each other using the standard Limitorque relationships found in the selection guides. For example, the stem force should relate to the output torque of the operator with normal values of stem/stem-nut coefficient of friction. Deviations from these normal values should be investigated. Similarly, the output torque of the operator can be compared to the output torque of the motor using motor power or motor current and voltage measurements.

For those devices measuring the stem thrust directly, the procedures developed at New Hampshire Yankee and Portland General Electric represent an improved method for instrument calibration. With those procedures, the operator stem nut is loosened, a strongback is installed above the operator, and a hydraulic ram with a load cell is installed. The ram compresses the stem, and the load cell measures the compression load. The thrust-measuring device on the stem or yoke is then calibrated to the known load.

Equipment Problems

The participants, including the utilities, vendors, and the INEL, gained new insights on the function of diagnostic equipment and methodologies, and on the function of the Limitorque motor operator under load. Part of these lessons learned came from the vendors who experienced equipment problems. For example, two diagnostic devices experienced internal shorts. The shorts did not keep the devices from working, but they affected their accuracy. One vendor has since developed a system to detect the internal short; the other may be pursuing a similar solution. (We do not have knowledge of the details.)

One vendor had problems because the upper bearing housing on the operator bound on the gear box. This vendor has since developed a procedure to check for this problem. Another vendor had an intermittent computer problem that caused offsets in his measurements when the electric motor was running. The problem, being intermittent, disappeared after the first day of testing. This problem was investigated but never resolved. The same problem appeared later on the same computer during diagnostic testing in the field. The vendor was able to recognize this later appearance of the problem because of the earlier experience during testing on the MOVLS.

The MOVLS, too, experienced a mechanical problem. The thrust bearing installed between the lower end of the stem and the load cell became degraded with use, enough to affect the accuracy of the stem torque measurements taken during some tests.

Strain gauges and other devices installed on the smooth portion of the stem were more successful at measuring thrust than those installed on the threaded portion. In some cases, newly developed methods for taking measurements on the threaded portion of the stem were withdrawn from the test program. The configuration of the threaded portion of the stem makes reliable measurements difficult to obtain.

Sample Rate

Diagnostic systems sample rates varied from a fixed total number of samples per test to 1,000 samples per second over the entire duration of the test. A sample rate of 1,000 samples per second yields a 1-ms resolution, which is more than adequate for MOV diagnostic work. However, those diagnostic systems that spread a fixed number of samples over the entire test (full valve stroke) may yield a very coarse time resolution; the resulting torque switch trip time may have an uncertainty of 4 to 10 ms or greater.

During actual in-plant diagnostic testing, one might reduce the effect of the limited sampling capability by collecting data for gate and globe valves near the end of the valve stroke, thereby concentrating the available number of samples on the portion of the valve stroke of most interest. We understand that this procedure is recommended by at least one vendor, but we do not know if it is widely used.

The data sample rate is critical in determining the correct torque and thrust values corresponding to torque switch trip. On systems with slow sample rates, the analyst might have no choice but to use the data point immediately before torque switch trip, when the torque and thrust are lower. Additional conservatism would result, compensating for the lower readings.

Conversion of Spring Pack Displacement to Stem Torque

If the spring pack has been calibrated, the spring pack displacement can be used to estimate operator torque with fairly good accuracy. Any worm-to-spline friction would tend to make the estimated torque slightly lower than the actual torque, thus making the estimate more conservative. Manufacturers' tolerances affecting the operator moment arm length (worm gear effective radius) can go either way, making the estimate either more or less conservative. Testing the operator on a dynamometer can quantify the relationship between spring pack displacement and operator torque. This could reduce the error in the torque estimate from about 7 to 10% to about 3 to 4%.

Conversion of Stem Torque to Stem Thrust

Estimating the conversion of torque to thrust requires the use of a well proven power thread equation. However, this equation has an unknown, the coefficient of friction between the stem and stem nut. To estimate the stem thrust from a measurement or an estimate of the stem torque, one must estimate the coefficient of friction in the stem/stem-nut interface. This is most typically accomplished by mounting a load cell on the operator upper bearing housing and testing the operator in the opening direction, simulating a backseating load (static test). The spring pack deflection is then compared to the measurement taken by the load cell to create a deflection-versus-stem-force relationship. Embedded in this relationship are the stem factor and the corresponding stem/stem-nut coefficient of friction.

Unfortunately, this methodology is less than reliable, because the stem/stem-nut coefficient of friction can be lower in a static test (with little running load) than in a differential pressure closure. (This phenomenon is known in the industry as the rate-of-loading effect.) The following discussion explains why a stem/stem-nut coefficient of friction determined from a static test should not be used to estimate the stem thrust expected in a high-flow valve closure. The test results described here are typical of several stem/stem-nut combinations and lubricants that we have tested on the MOVLS. During this testing, we used a 1.25-in.-diameter, 1/4-in. pitch, and 1/2-in. lead (double lead) valve stem and EP1 lubricant. The torque switch setting remained the same during these tests.

Figure 2 shows stem force traces for three valve-closing tests. The first trace (labeled zero test) simulates a calibration stroke (static test), where the stem is stroked hard into the seat with little or no running load. The stem force is small during the running portion of the stroke, then increases rapidly during seating and through torque switch trip at 20,700 lb to a final stem force in excess of 28,000 lb. Test 4, run with a longer stroke time, is a moderately loaded test, equivalent to closing a pressurized valve with packing and stem rejection loads but little flow. Here the valve fully closes and solidly seats. The stem force at torque switch trip is 15,300 lb, and Test 4, Test 6 shows higher stem loads and a the final stem force is over 23,000 lb. The Test 6 trace represents a valve closing against a significant differential pressure load. In contrast to failure of the valve to fully close. The torque switch tripped at 14,900 lb, a drop of 5,800 lb from the zero test. The earlier apparent margin disappeared because of the higher loads imposed during the running portion of the valve stroke.

Figure 3 shows the stem/stem-nut coefficient of friction for the same three tests. In the zero test with little or no running load (only a seating load), the coefficient of friction is about 0.08 during seating when the stem is being loaded. This is the value one might expect to derive from the simple static opening tests described above. The coefficient of friction in Test 4 starts out at

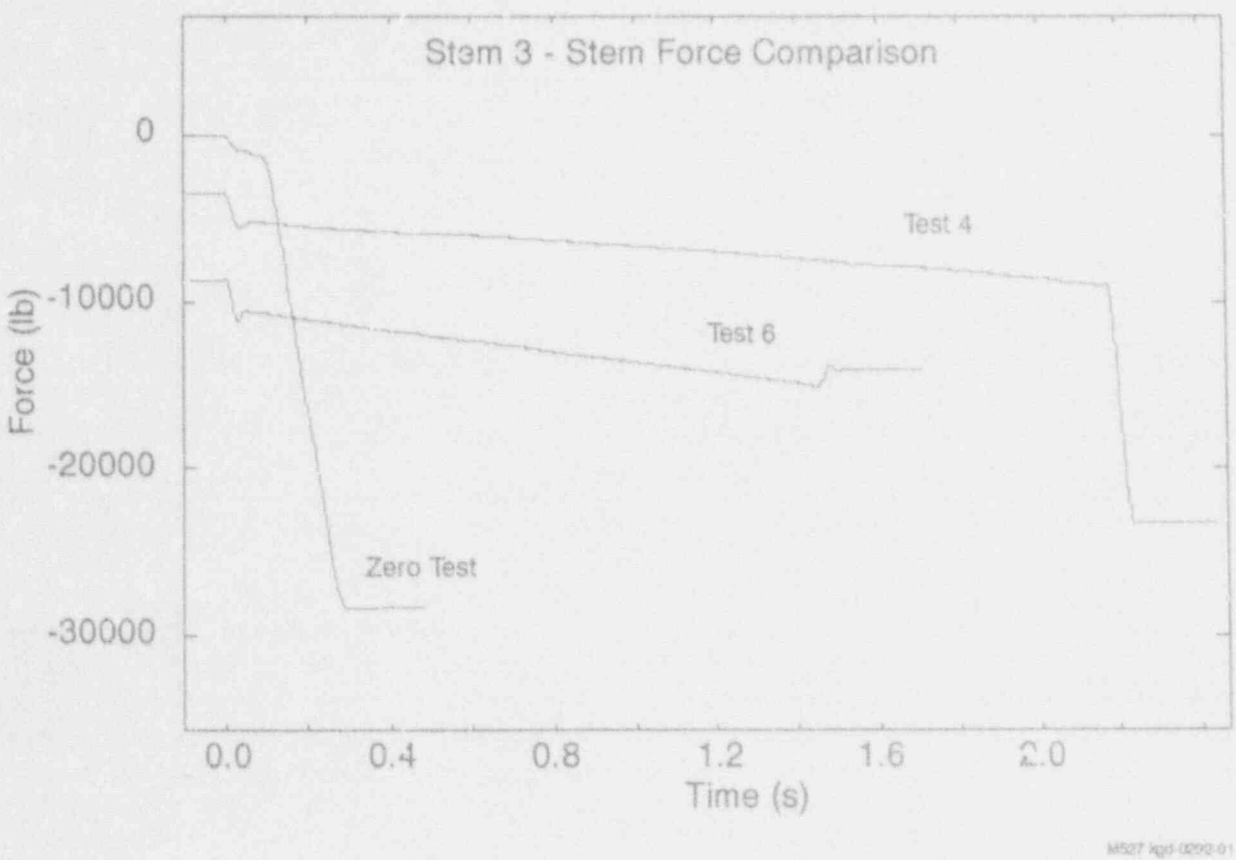

Figure 2. Three stem force traces from testing on the MOVLS: no running load (zero test), moderate running load (Test 4), and design basis load (Test 6). Note the effect of the higher running load on the final thrust.

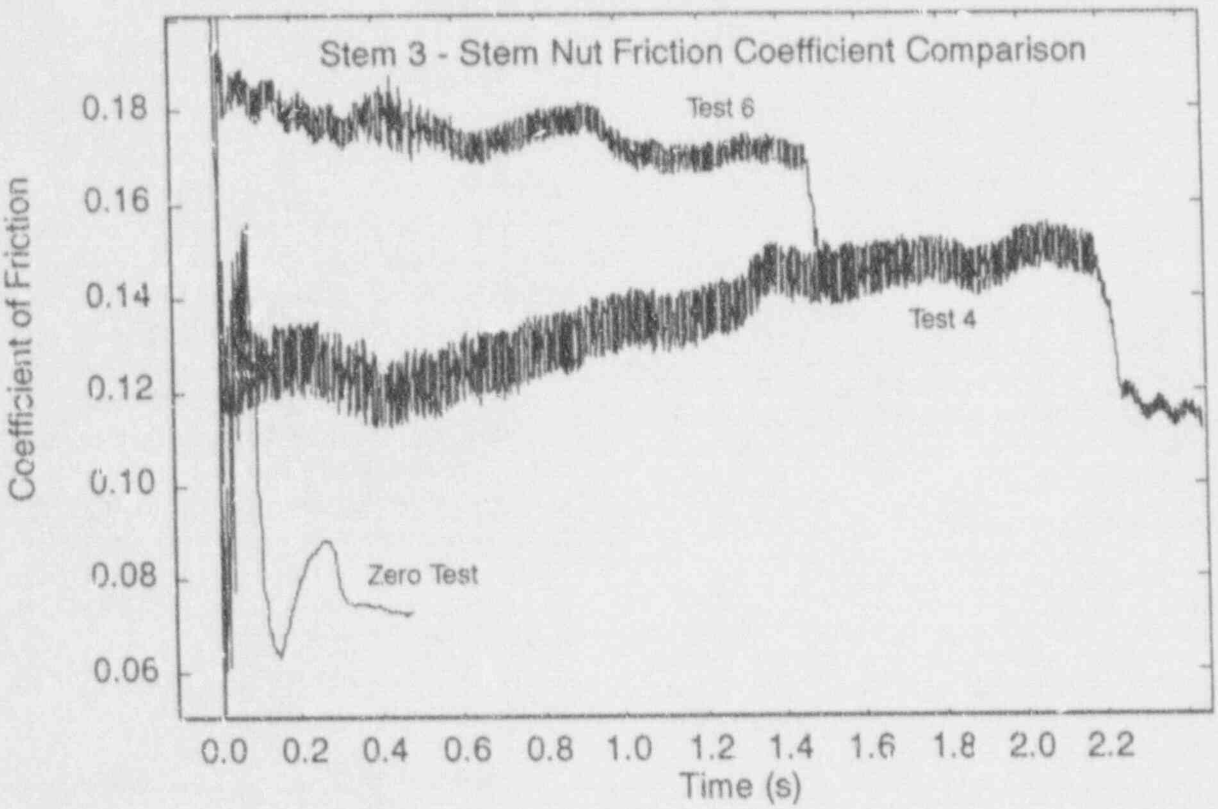

Figure 3. Stem/stem-nut coefficients of friction derived from the tests shown in Figure 2. Note that the friction coefficients are higher with higher loads.

slightly less than 0.12 and increases to about 0.135 at seating. The coefficient of friction under the high-flow and pressure loads simulated in Test 6 is even higher, at about 0.16. The coefficient of friction obtained in the static test is significantly lower than those obtained when the operator is subjected to a running load. Thus, coefficients of friction derived from static tests are not conservative, and the resulting thrust calculated from this process would be underestimated.

Accuracy Claims

Accuracy claims for the diagnostic equipment were as varied as the number of transducers. Some accuracy claims were based on percent of reading, while others were based on percent of full scale, percent of reading plus a percent of full scale, and even percent of reading plus a percent of range (twice full scale). Taken at face value, these claims can be confusing. For example, an error of 5% of full scale may appear to be acceptable, but full scale for a stem force measurement device may be 100,000 lb, so the error would be 5,000 lb. Since a 6-in. valve may only see 20,000 lb thrust, a 5,000-lb error translates to a 25% uncertainty. An additional 25% margin for measurement uncertainty may not be acceptable for this valve assembly.

Determining the margin needed to account for measurement uncertainty can be confusing using typical vendor accuracy claims. An accuracy claim may represent full end-to-end accuracy (transducer, signal conditioning, and computer error combined) or transducer error alone. The analyst may not know whether the claim refers to a 99% confidence interval (one sample in 100 exceeds claim) or a 68% confidence interval (eight samples out of 25 exceed claim). This information may have a significant impact on the analysis. Accuracy claims can refer to precision errors, bias errors, or both. Simply stated, precision errors represent the variation of the measurement about the true value, while bias errors represent the floating of the zero value. This precision-versus-bias error may not be included in the vendor claims, but it may be very important to the analyst.

Because of the possible confusion associated with vendor accuracy claims, additional information will be needed in many applications. MUG may wish to develop a consistent set of questions, similar to those listed below, to properly address instrument uncertainty margins in MOV evaluations.

- Does the accuracy claim include the accuracy claims of the calibration device?
- Does the accuracy claim include end-to-end uncertainties from transducer to final engineering unit output? If not, what margin must be added to cover the other uncertainties in the system (propagation of errors)?
- How will the system sample rate and the response time of the torque switch sensing circuit affect the overall accuracy?
- If I read 100 data points from a plot or 100 data values from a list, how many measurements will be within the accuracy claims?
- What additional margin must I include to cover the uncertainty associated with balancing (zeroing) the device?
- If the accuracy claim is given as a percent of scale or percent of range, how will the error compare with the expected value of the reading? 25%? [For example, a 5,000-lb error (5% of 100,000 lb) at an expected reading of 20,000 lb might not be acceptable.]

The answers to these questions can help provide the information utilities and regulators may need to address measurement uncertainty and justify the margins they use when assessing valve performance.

CONCLUSIONS

When evaluating MOV performance, the MOV analyst must consider the unique characteristics of the diagnostic device used. Design basis calculations will provide an estimate of valve requirements, while diagnostic testing provides an estimate of motor operator capabilities. A comparison of the two provides some insights on available margins. As motor operator sizing margins decrease, the need for more accurate diagnostic equipment will increase. The end-to-end accuracy of the diagnostic equipment is affected by in situ calibration, transducer uncertainty, equipment response time, and data acquisition system uncertainty. Those using one vendor's transducer with another vendor's data acquisition system may need to compute their own end-to-end accuracies.

While spring pack measurements can be used for estimating operator output torque, a very important parameter itself, it is not practical to estimate stem thrust from spring pack force or displacement measurements, because of the very large error bands needed to cover stem factor uncertainty.

All participants were repeatedly exposed to the rate-of-loading phenomenon. The utility hosts, the test coordinators, and at least some of the vendors learned a great deal about its cause.

One must look at the MUG program for validating diagnostic equipment as a demonstration program. We can show that some of the diagnostic equipment worked well in the laboratory with SMB-0-25 motor operator and a very stiff yoke, but this program does not provide guaranteed calibration of any vendor's equipment. It only shows that, under the conditions tested, some equipment performed well while other equipment did not. Each vendor will have to provide accuracy criteria for each of the intended end uses. The program also showed that some methodologies do not work, even under the best circumstances.

We believe the MUG program was an overall success. The validation program was a valid learning experience for everyone who participated; we learned the importance of asking better, more informed questions related to diagnostic equipment. If all concerned parties will carefully read the MUG validation report, the conclusions will be obvious.

Diagnosing MOV Problems Using Comparative Trace Analysis

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ABSTRACT

The paper presents the concept of comparative trace analysis and shows it to be very effective in diagnosing motor operated valve (MOV) problems. Comparative trace analysis is simply the process of interpreting simultaneously gathered traces, each presenting a different prospective on the same series of events. The opening and closing of a motor operated valve is such a series of events. The simultaneous traces are obtained using Liberty Technologies' VOTES® system. The traces include stem thrust, motor current, motor power factor, motor power, switch actuations, vibration in three different frequency bands, spring pack displacement, and spring pack force. Spare and auxiliary channels enable additional key parameters to be measured, such as differential pressure and stem displacement. Though not specifically illustrated in this paper, the VOTES system also provides for FFT analysis on all traces except switches.

Liberty Technologies' VOTES System, which stands for Valve Operation Test and Evaluation System, is used in nuclear power plants to verify the ability of safety related MOVs to properly open or close when called upon to do so.

Many channels of data are sampled simultaneously during the time that the valve is stroked through an opening and closing cycle. The data traces include stem force, motor current, limit switch positions, torque switch positions, motor current envelope, motor power factor, motor active power, low frequency actuator vibrations, high frequency actuator vibration, total actuator vibration, and two additional free channels for spring pack displacement, differential pressure, or any other relevant parameters.

A gate valve whose function is to close against a certain differential pressure must demonstrate that it can do so with a sufficient thrust margin. This is the main purpose of the stem force measurement. For most MOVs, it is not possible to put a thrust sensor on the stem because the exposed section of the stem moves up into the actuator or down into the valve during part of its stroke. For that reason, VOTES uses a special sensor mounted on the yoke to measure thrust. The yoke sees the same thrust as the stem, but with opposite sense. Figure 1a shows a typical gate valve closing, with the associated compressive force on the stem and tensile force on the yoke. Figure 1b shows the intermediate position with little or no stem or yoke force. Figure 1c shows how the forces reverse for backseating or for pulling on the gate when it is wedged in the seat. The validity of measuring stem force from the yoke is demonstrated in Figure 2, where a trace from a VOTES Force Sensor mounted on the yoke is compared with a trace from a strain gauge mounted on the stem for a valve first opening, then closing against differential pressure.

The basic diagnostic concept of VOTES is to compare simultaneous trace outputs at the mechanical events known to be taking place in the actuator and the valve as it is opening and closing. Abnormal times, abnormal values, or abnormal traces or trace relationships can be related directly to what is wrong in the motor operated valve. Obviously, a complete understanding of the MOV is necessary to be able to make judgements about what is and is not normal.

Several traces and trace comparisons will now be shown to illustrate how the process works.

Figure 3 shows a gate valve cycling first open, then closed under 1500 psi differential pressure. The top trace shows the stem force as obtained from the yoke mounted VOTES sensor. The bottom trace shows the envelope of the high frequency vibration seen by an accelerometer mounted on the actuator. The high-frequency acceleration envelope is an excellent indicator of metal impacts and rubs. The two large peaks in the lower trace show regions of severe rubbing. These line up with the points where the disc (or gate) transitions from the seat surface back to the guides on opening, and then from the guides back to the seat on closing. The severe rubbing is caused probably by too great a clearance in the guides, thereby allowing the disc to cock over during its transition.

Figure 4a shows a different gate valve, again cycling open then closed, but under zero differential pressure, a so-called "static" test. The top trace is the output of the VOTES Force Sensor; the bottom traces are the control switches. The motor is turned off by the torque switch in the closing direction, as is typical, and turned off by the torque switch in the opening direction, which is not typical. Thus, the valve is allowed to backseat and as a result, it builds up tensile force in the stem to 11,400 lb. Also, the packing forces are shown to be excessive, causing an average 6,400 lb of stem tension on opening and an average 5,950 lb of stem compression on closing. Packing forces for this size stem (1.25 in.) should be under 2,000 lb. The high packing force leaves only about 7,000 lb to overcome differential pressure forces and seat the valve.

In addition, Figure 4a shows a low-level oscillation in the closing running load trace. This region is expanded in time and amplitude in Figure 4b, along with the corresponding motor power factor trace. Power factor is especially sensitive to variations in motor load at low torque levels. The force oscillations are seen to be approximately sinusoidal with a period of 1.800 seconds. This matches precisely to the rotational speed of the drive sleeve within the actuator. The lack of a corresponding oscillatory load on the motor pinpoints the problem to the inner race of the upper drive sleeve bearing being slightly cocked with respect to the axis of the drive sleeve. The oscillation is caused by the resulting off-center reaction force at the top of the actuator rotating around with the drive sleeve. This sinusoidally loads the yoke mounted sensor, but has no affect on motor load. An offset worm gear or drive sleeve by contrast would have affected motor load as well. This example points out how conclusive the technique of comparative trace analysis can be.

Figure 5 expands the force trace of Figure 4a in three key regions, and displays them along with the motor current envelope trace. The unseating region at the beginning is shown in Figure 5a, the backseating and restart-to-close region is shown in Figure 5b, and the seating region is shown in Figure 5c. The minimal amount of current buildup upon backseating as compared to the current buildup with seating shows that readjusting (lowering) the open torque switch setting would do little to lower the backseating force, and that controlling on the open limit switch would be the only practical way to accomplish this.

The traces of Figure 5, because of their expansion, are useful in seeing the major aspects of the operation of the MOV. The valve starts out in Figure 5a fully seated with a stem compressive force of about 12,000 lb. The motor is then started in order to open the valve. The inrush motor current peak (typically about six times the running current) signals the actual initial turning of the motor. The worm immediately turns and relieves the torque it exerts on the worm gear, drive sleeve, and stem nut. This slightly relieves some of the stem force, but most of this force is retained since the stem and stem nut remain locked together by friction. A lost motion area allows the worm to turn the worm gear, but not the drive sleeve so the motor can easily come up to speed. At the end of the lost motion area the worm gear impacts the drive sleeve (hammerblow), and starts it and the stem nut turning to relieve the stem compression. When all the compression is relieved (zero force), the stem nut continues to turn, moving through the thread clearance until it starts to pull up rather than push down on the stem. As the stem moves upward, it is dragged through the tight packing causing a tensile force of about 6,000 lb on the stem. This force remains fairly constant showing that the stem is not yet pulling on the disc, but is instead moving through the clearance of its T-block attachment. When this clearance is taken up, the bottom of the stem starts to pull on the disc causing the stem force to increase rapidly, but the disc has not yet moved. Finally, at 10,100 lb of stem tension, the disc pops free from the seat and the stem force drops back down to the previous packing force value as the stem continues to be dragged up through the packing.

Figure 5b picks up the action about 10 seconds later with the valve continuing to open. Nearly 13 seconds into the stroke the valve runs into its backseat, restraining the stem at the bottom causing the tensile force in the stem to again rise very rapidly. Finally, the open torque switch trips to stop the motor. Notice in Figure 4a that the position-controlled open torque bypass switch has opened just one second earlier, thus arming the open torque switch. Also, notice that the bypass switch was closed during the time of disc pullout, which is normal, so that even if 15,000 lb would have been needed to unseat the disc, the torque switch opening would not have shut off the motor. Returning to Figure 5b, at motor restart-to-close, a very small portion of the stem force is again relieved, as it was in the other direction. As before, no other change occurs until the end of the lost motion area where hammerblow starts the stem nut turning to relieve the tensile stem force. Also, as before, when all tension is relieved zero force is maintained during the period that the stem nut turns through the thread clearance to push down, rather than pull up, on the stem. Now, pushing the stem down through the packing requires a compressive force of about 6,000 lb.

Figure 5c picks up the action again 9 seconds later. About 4 seconds into the trace, the stem starts to wedge the disc into the seat. The wedging forces the stem compression to increase rapidly, causing the close torque switch to open and trip the motor. Though not shown here, the force is about 12,000 lb at torque switch trip, 12,500 lb at motor trip (12 millisecond contactor dropout time), with 500 lb of inertial thrust. This brings the maximum compression to 13,000 lb, which then relaxes back to 12,500 lb. Clearly the traces contain information not only about force and current magnitudes, but timing information as well. This timing information, accurate to a millisecond, can be used to determine clearances opening up as result of wear or the timing of events being altered by misadjustment. The actual clearances or adjustment distances affecting the timing of events are themselves determined by gear ratios and other known geometries of the actuator, motor, and valve.

To illustrate, the motor current envelope and power factor for the combined valve stroke of 4a are shown in Figure 6a. A half second of data about the cursor in the opening portion of the stroke is expanded in amplitude and time in Figure 6b. The cyclic pattern that appears in the upper and lower traces results from a severe wear condition of the worm gear teeth purposely machined in to see if it could be picked up. The period in the pattern is 53 milliseconds, which corresponds exactly to the time it takes for each worm gear tooth to be in contact with the worm that drives it (1.800 seconds/34 teeth). Had there been no tooth wear, there would be little cyclic loading of the motor as the gear tooth contact would be fairly uniform. But with the wear condition, the load on the motor varies through the period of contact of each of the 34 gear teeth, each tooth itself, of course, being slightly different. Notice how the motor current envelope, which is purely the amplitude variation of the motor current, is extremely similar to the power factor which is purely based on the phase variation between the motor current and the line voltage. Also, notice how the special mathematical process used by VOTES to compute these traces allows motor load variations even faster than line frequency to be seen.

Figure 7a shows another event on a different valve made more clear by timing. The upper and lower (half second) plots show the actuator vibration associated with the relaxation of the spring pack following a motor start to reclose the valve.

The start of the inrush current is indicated by C0, and its peak by C1. At this point the motor and the worm begin to turn. The spring pack compression gets relieved until the outer thrust washer impacts the locking ring. If the elastic stop nut at the end of the spring pack is adjusted just right to fit within the space allotted for it, the worm will impact the inner thrust washer at the same time the outer thrust washer impacts the locking ring, and only a single impact will be noted. But if this adjustment is not right, either too short or too long, a so-called spring pack gap results, which causes the two impacts to be separated in time. The gap distance can be determined from this time separation multiplied by the speed of the worm. In this case the gap has been determined to be 0.299 inches (0.029 sec x 10.3 in/sec). As expected, the impact separation on reopening is virtually identical as shown in Figure 7b.

Figure 8a shows still another valve, first closing then reopening. The valve closing and opening is controlled by limit switches. The valve never seats while under power (no current buildup) and does not coast into the seat either (no force buildup). In addition, both the tensile and compressive forces relieve slightly after the motor turns off. All these problems aside, still another problem is evident from Figure 8a and the associated zoomed traces in Figures 8b and 8c. In 8b, after the motor is turned on to reclose, the worm gear goes through its lost motion area as is normal, and then the stem goes through a zero force plateau, also normal, before picking up the compressive packing force. What is not normal is the amount of time spent in the zero force plateau. This plateau is supposed to be due to the stem nut rotating through its thread clearance with the stem, typically 10 to 20 thousandths of an inch. But the 0.580-second duration at the known stem rate of 0.307 in/sec would indicate an impossible thread clearance of 0.178 inches. At least 0.150 inches of the 0.178 inches would have to be due to a loose stem nut locknut allowing the stem nut to move up 0.150 before it stops and pushes down on the stem. Figure 8b confirms the stem nut to locknut clearance, as the stem nut is seen to have to move the same distance back down to the shoulder of the drive sleeve before pulling up on the stem when reopening. The ability of the VOTES sensor to sense forces in both directions, including zero force, is what makes this diagnosis possible.

Still another valve is pictured in Figure 9. Here the closing force is plotted along with the motor power for a good lubrication case (stem to stem nut) in Figure 9a, and a bad lubrication case in Figure 9b. The ratio of stem force to motor power at the torque switch trip point explains the situation. The ratio reduces from its good lubrication value of 6.25 lb/watt to the bad lubrication value of 5.98 lb/watt. One could have been deceived by looking at motor power alone, expecting that the poorer lubrication would have put more load on the motor. The explanation for the opposite result is that the poorer lubrication caused the torque switch to trip out earlier, resulting in less motor power. The lubrication degradation is obvious only by looking at the thrust trace, where an even greater reduction in stem force clearly indicates the lubrication problem.

These examples show the diagnostic power of comparative trace analysis. The technique combines an understanding of the machine and its function with multiple independent simultaneous traces describing that function. By providing this for motor operated valves, VOTES is a very powerful MOV diagnostic tool. It helps to determine conclusively what *is* wrong, not just what *might be* wrong.

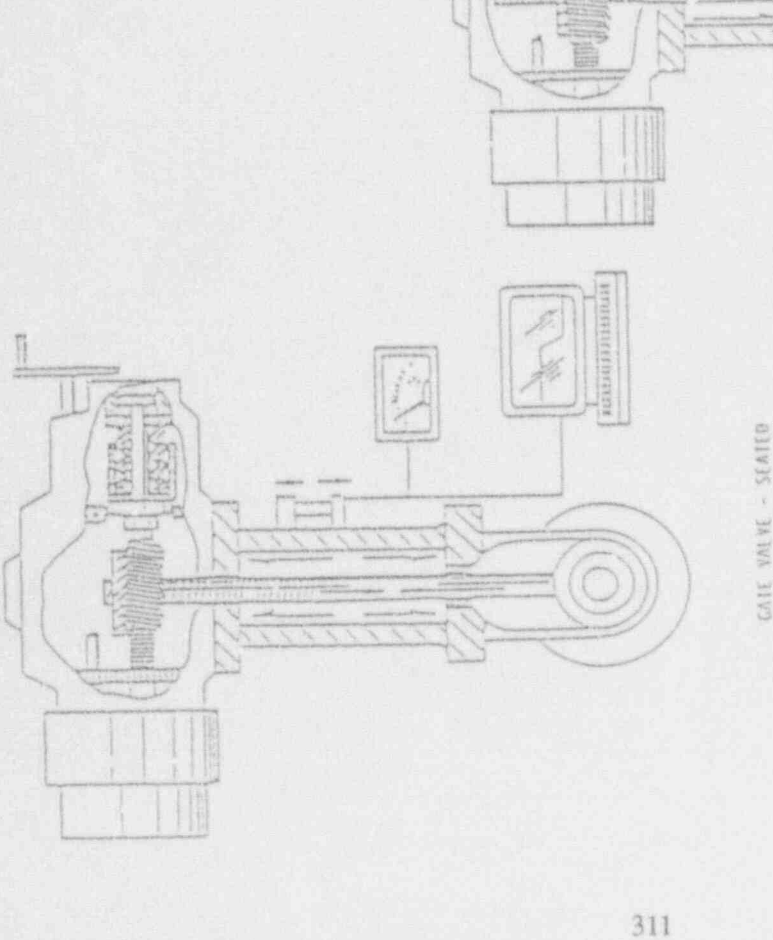

Figure 1a.

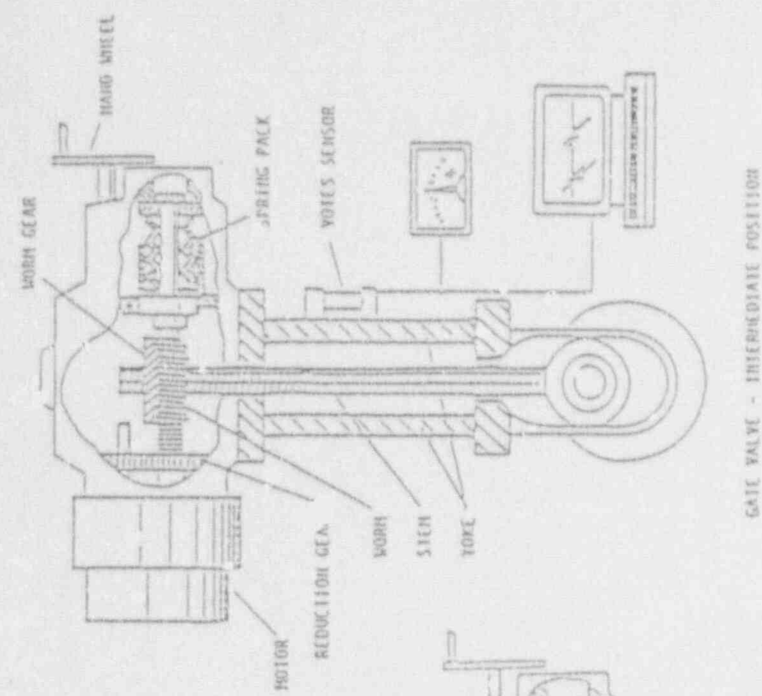

Figure 1b.

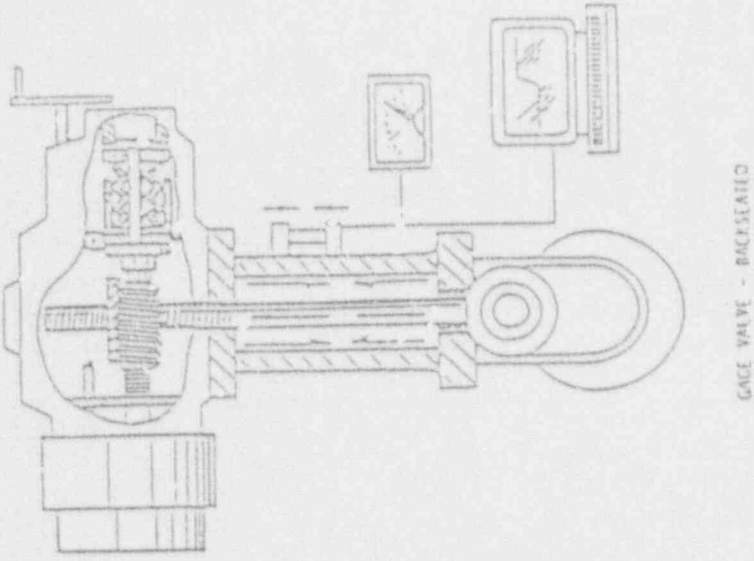

Figure 1c.

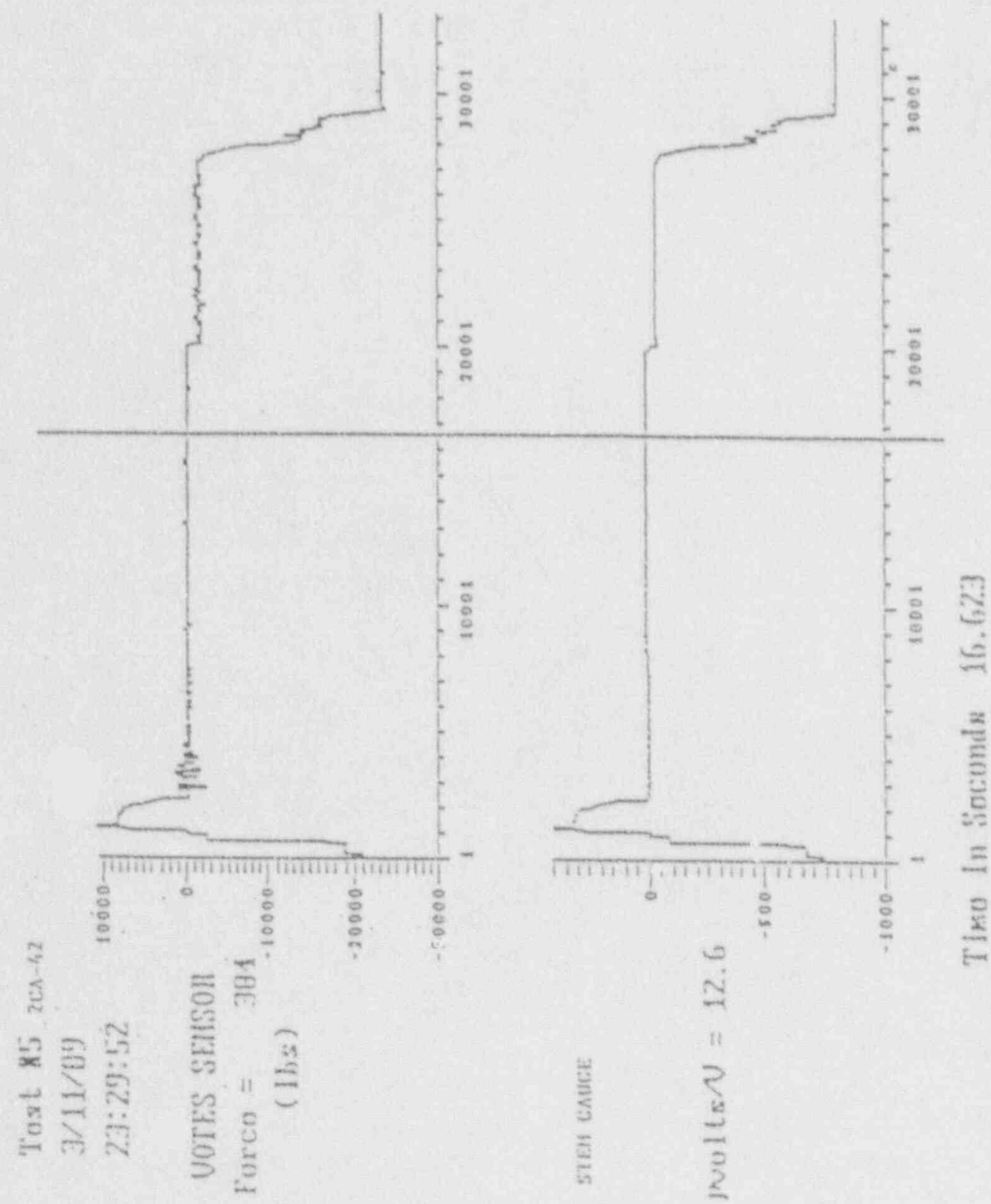

Figure 2. Yoke mounted VOTES force sensor vs. stem strain gage opening and closing with differential pressure.

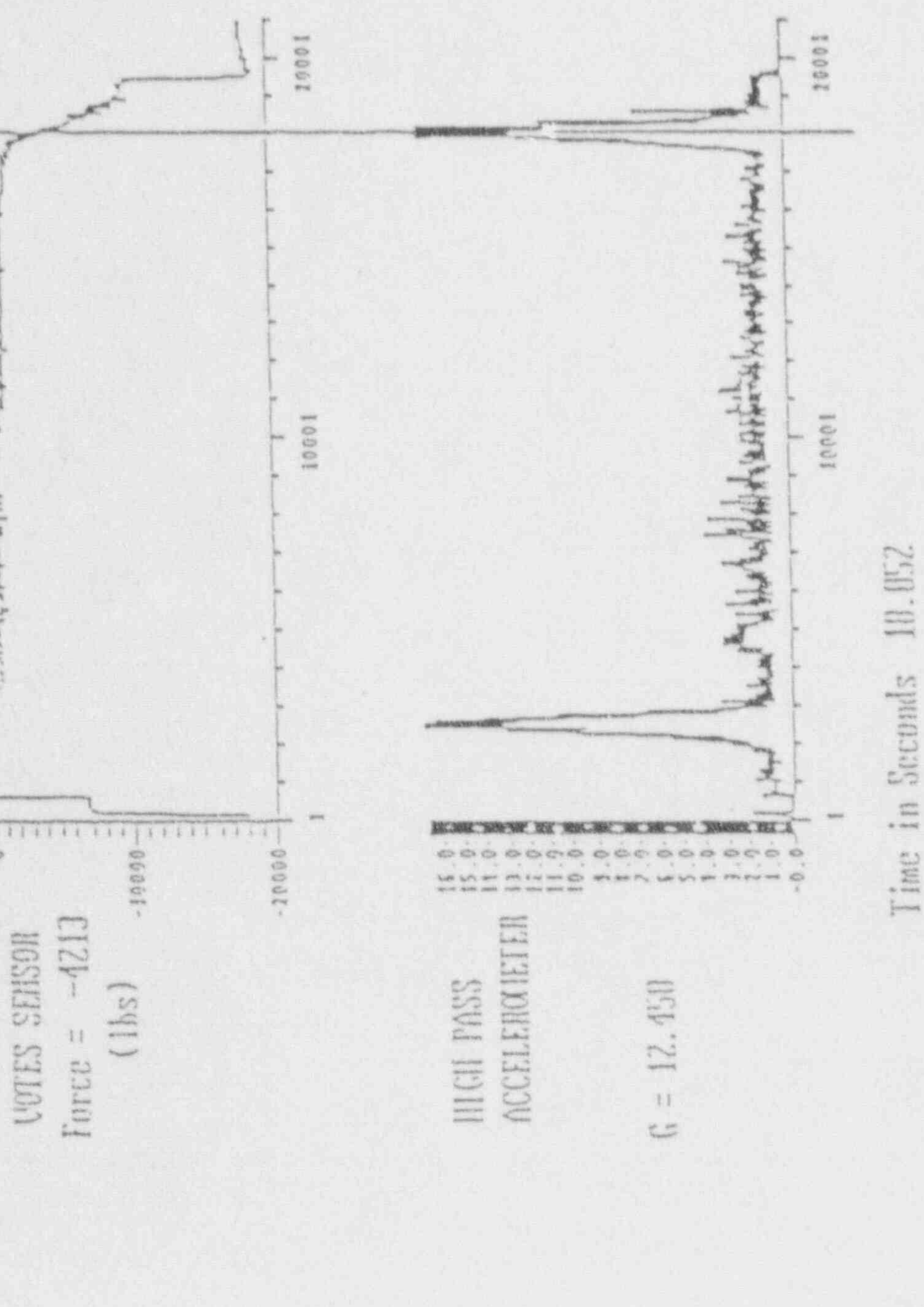

Figure 3. Gate rubbing while transitioning to and from the guides.

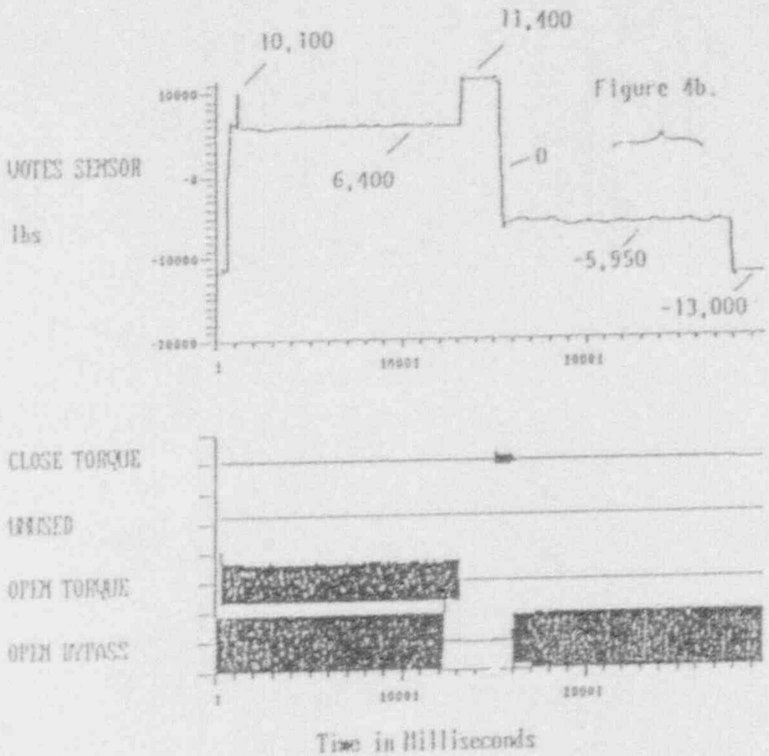

Figure 4a. Combined open/close stroke (static). High packing, high backseating force.

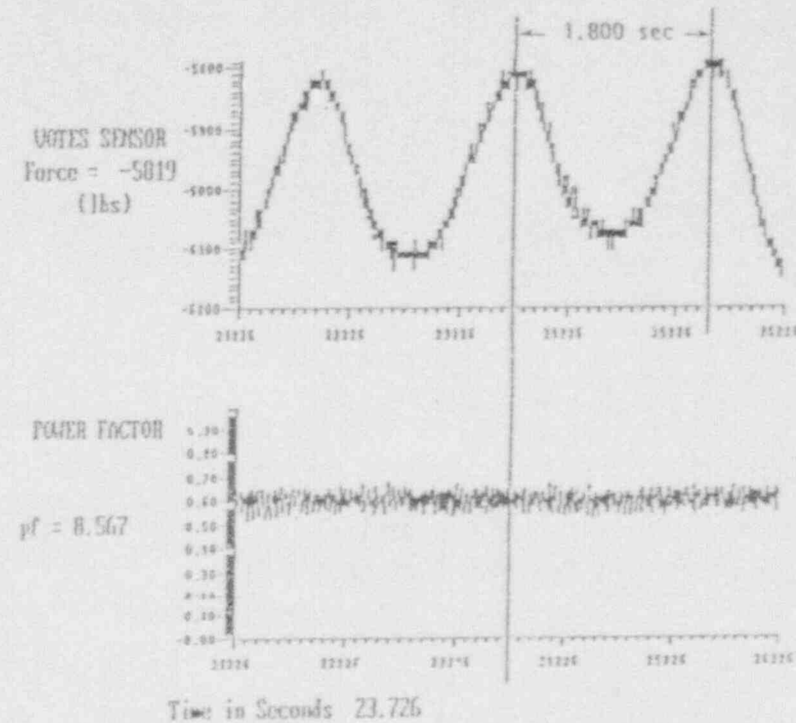

Figure 4b. Oscillation at drive sleeve rate. Inner race of upper guide bearing is off axis.

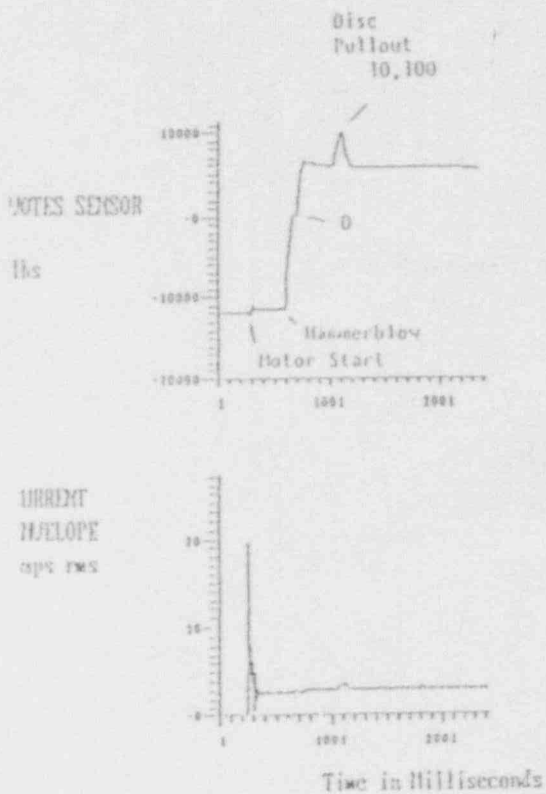

Figure 5a. Unseating.

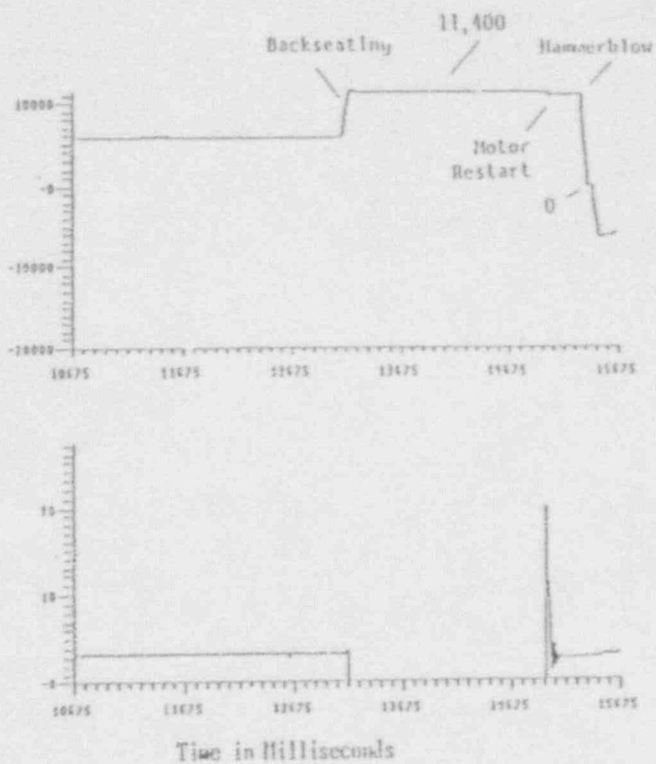

Figure 5b. Backseating & restart-to-close.

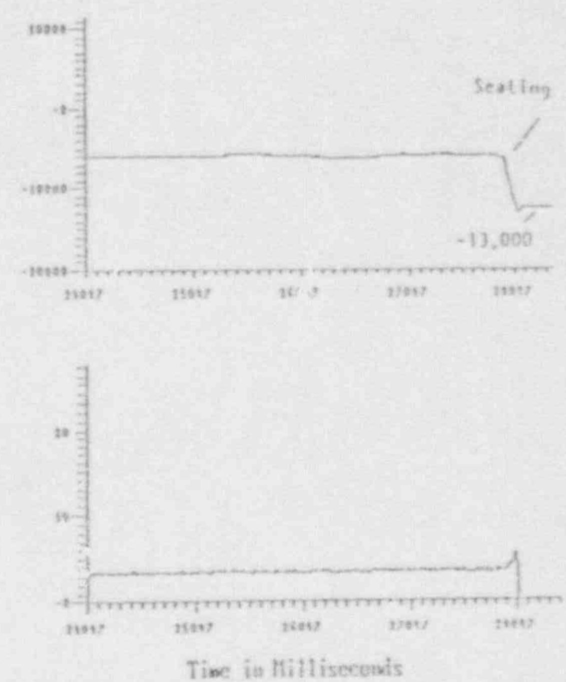

Figure 5c. Seating.

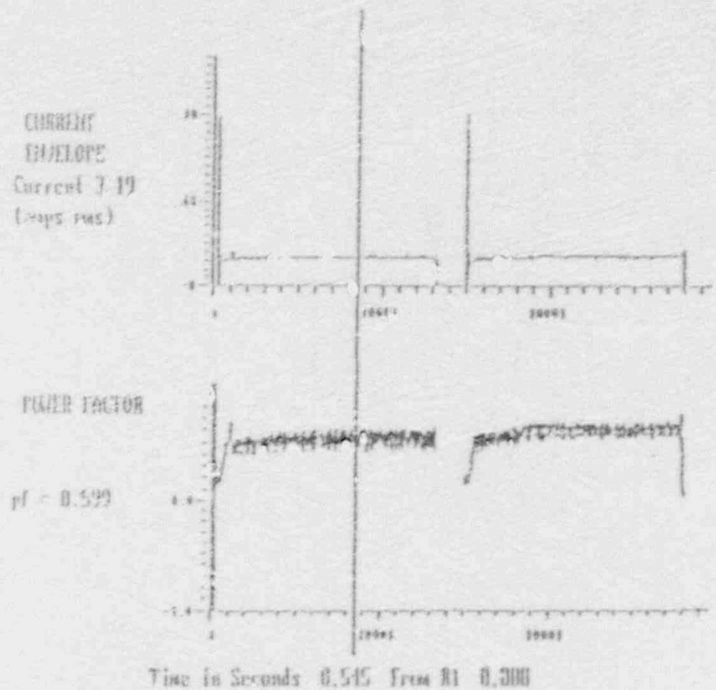

Figure 6a. Combined open/close stroke (static). Motor current envelope and power factor.

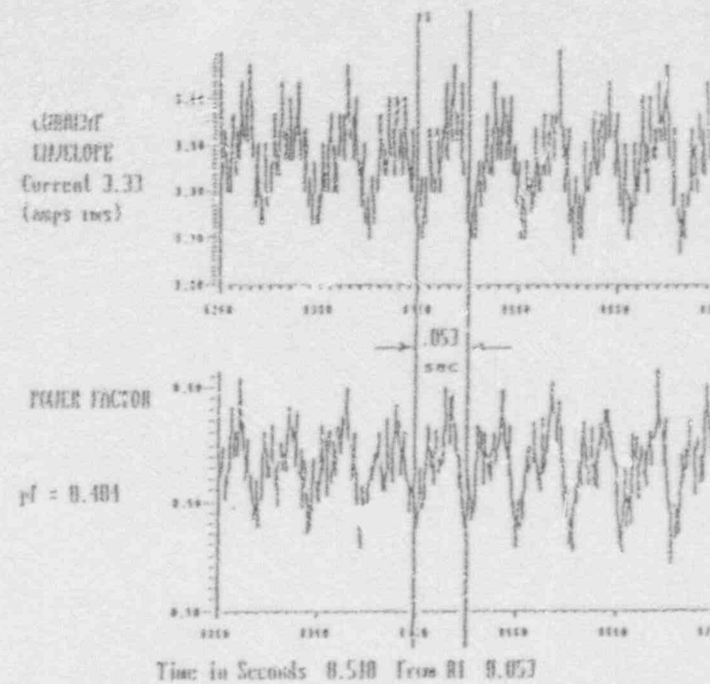

Figure 6b. Expansion about cursor location shows worn worm gear teeth.

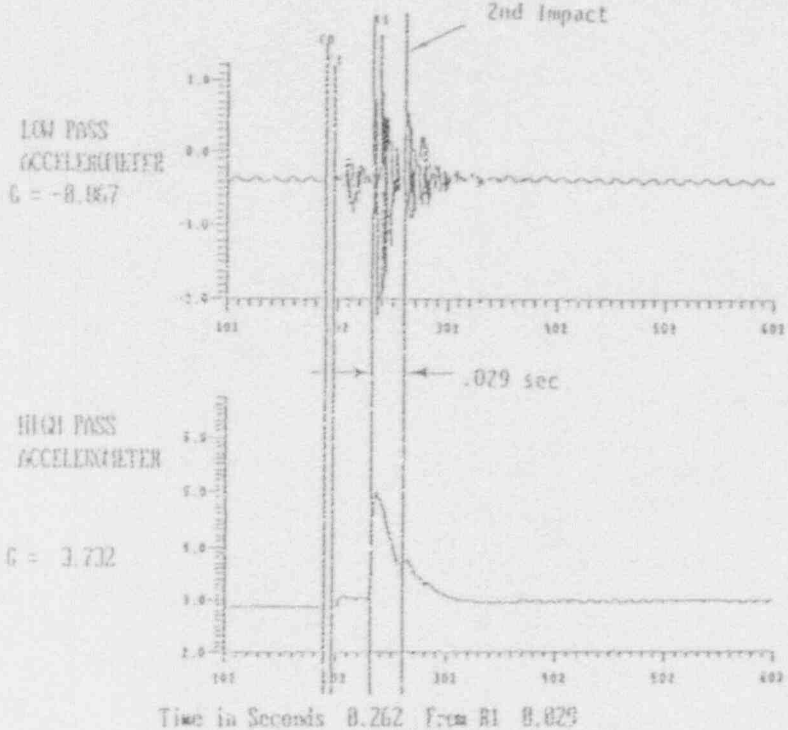

Figure 7a. Dual impact at start of closing stroke shows spring pack gap

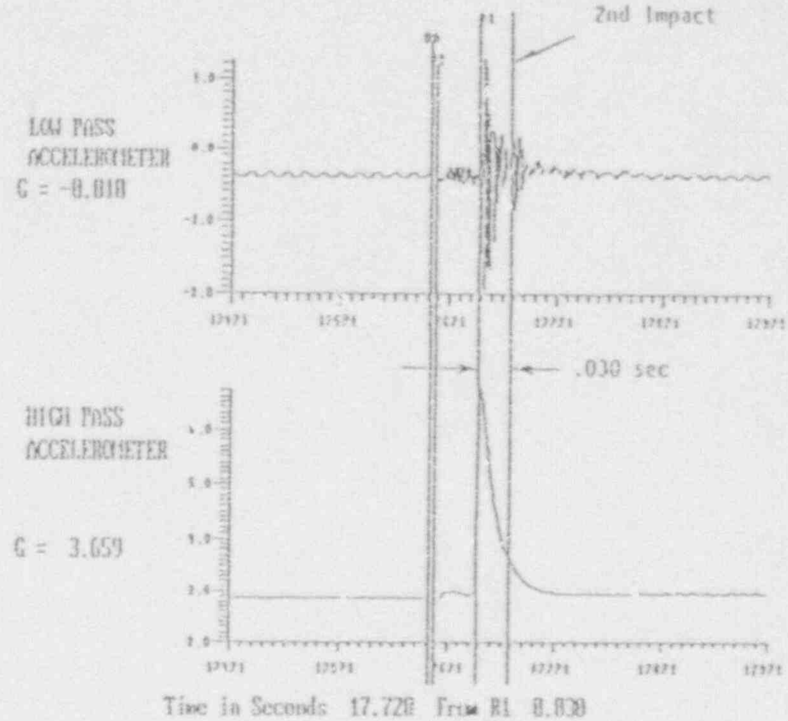

Figure 7b. Dual impact a start of opening stroke shows same spring pack gap.

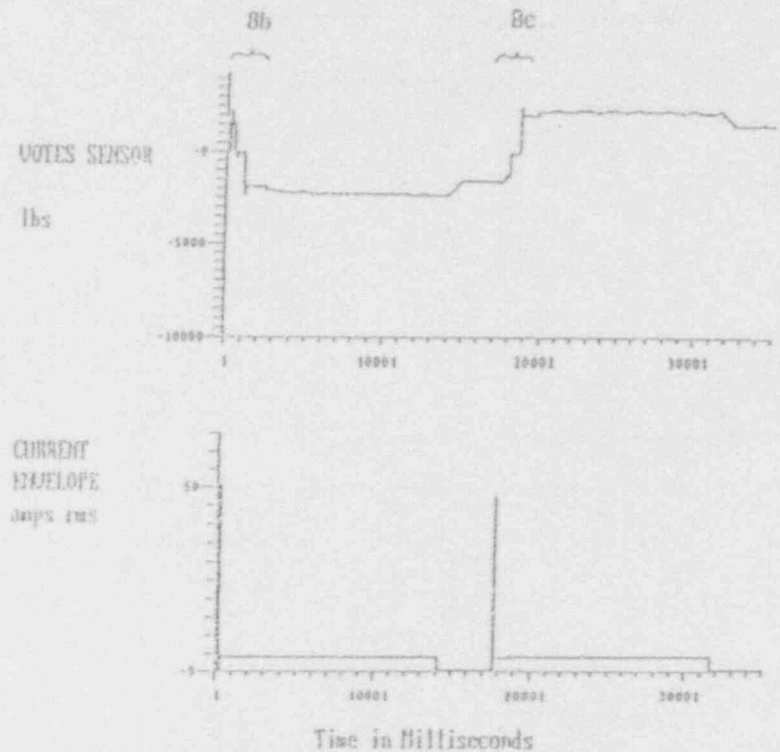

Figure 8a. Combined close/open stroke (static. High packing force, no seating.

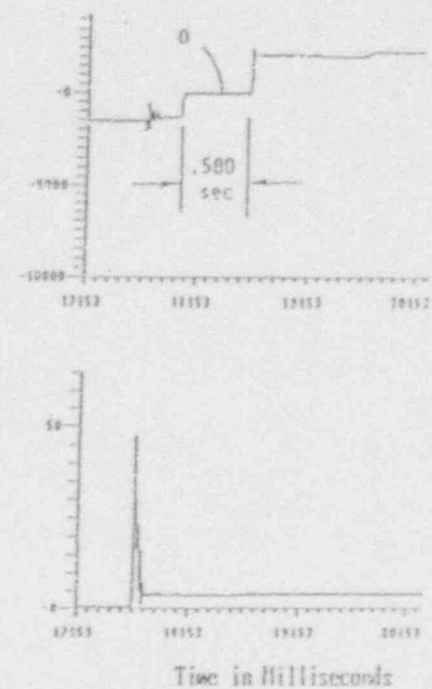

Figure 8b. Loose locknut extends zero force plateau on closing.

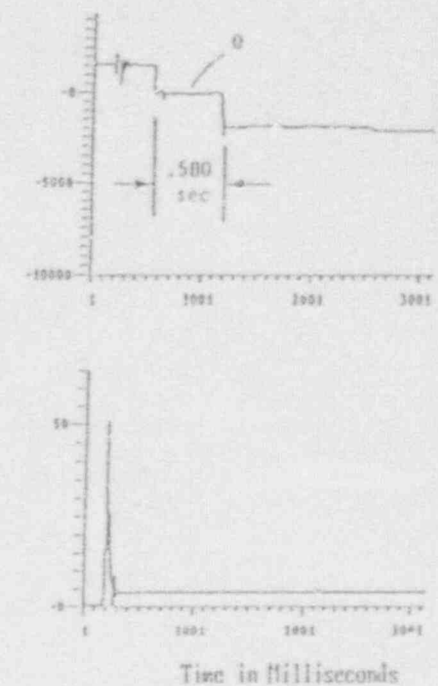

Figure 8c. Loose locknut extends zero force plateau on re-opening.

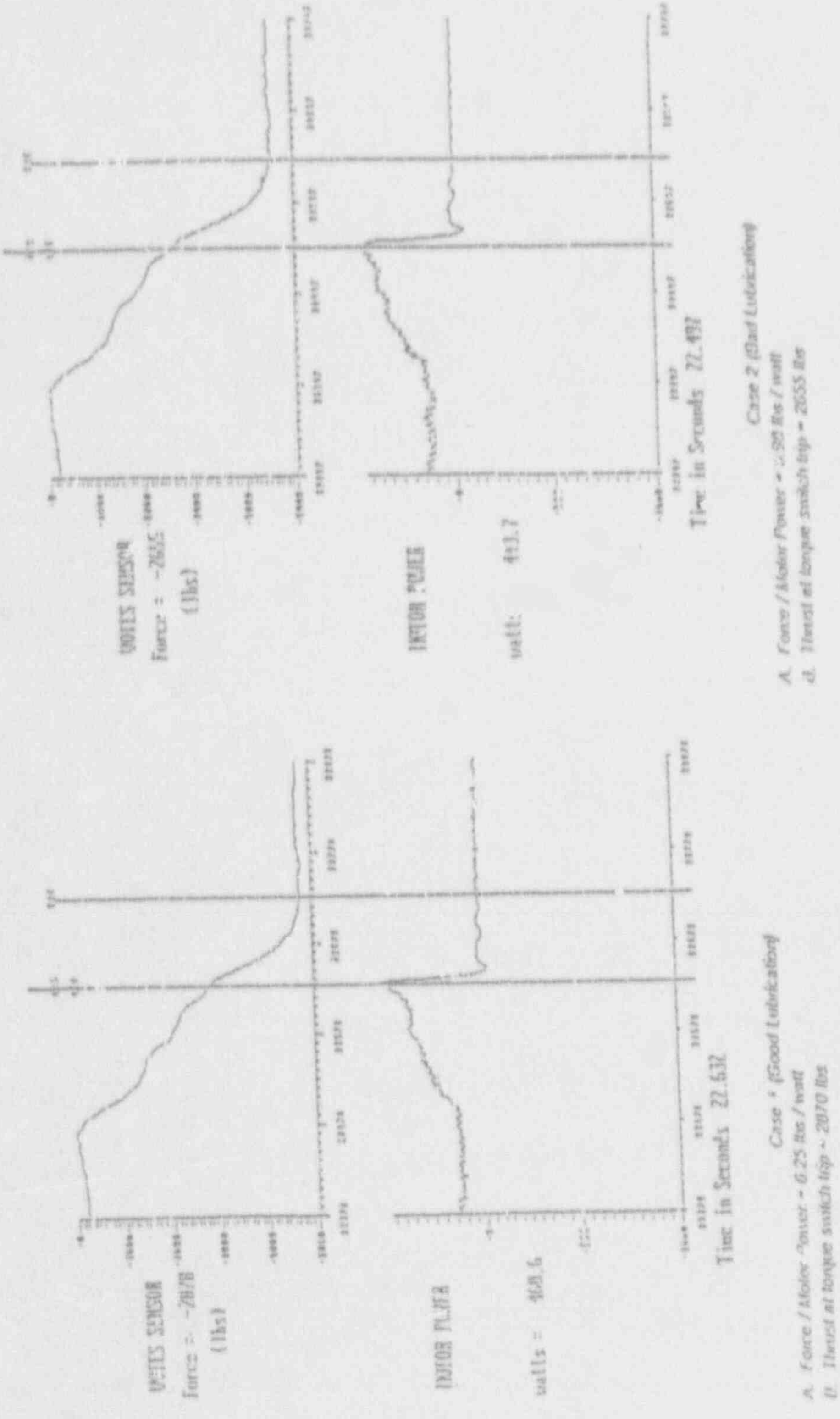

Figure 9. Ratio of force/motor power shows lubrication degradation.

Using Torque Switch Settings and Spring Pack Characteristics to Determine Actuator Output Torques

Bill R. Black
TU Electric

ABSTRACT

Actuator output torque of motor operated valves is often a performance parameter of interest. It is not always possible to directly measure this torque. Torque spring pack deflection directly reflects actuator output torque and can be directly measured on most actuators.

The torque spring pack may be removed from the actuator and tested to determine its unique force-deflection relationship. Or, a representative force-deflection relationship for the particular spring pack model may be available. With either relationship, measurements of torque spring pack deflection may then be correlated to corresponding forces. If the effective length of the moment arm within the actuator is known, actuator output torque can then be determined. The output torque is simply the product of the effective moment arm length and the spring pack force.

This paper presents the reliability of this technique as indicated by testing. TU Electric is evaluating this technique for potential use in the future. Results presented in this paper should be considered preliminary. Applicability of these results may be limited to actuators and their components in a condition similar to those for which test data have been examined.

INTRODUCTION

TU Electric is testing motor operated valve (MOV) actuators as part of a program to comply with Generic Letter 89-10 issued by the United State Nuclear Regulatory Commission. This document presents analysis of some of the test results.

Refurbished actuators of rising stem valves were cycled at several torque switch settings in both open and close directions on test stands that exert a resisting torque without producing axial load on the actuator's drive sleeve. For actuators of the same size with balanced torque switches, the same torque switch setting has been observed to produce a significant range of torque spring pack deflections at torque switch trip. For actuators of the same size, with balanced torque switches and the same model torque spring pack, test data indicate there can be significant variation in actuator output torque at torque switch trip.

During refurbishment of the actuators, the torque spring packs were verified to be properly configured. Then the spring packs were compressed to obtain the force-deflection (preload and stiffness) characteristics. Tests of several nominally identical spring packs show that variations in spring pack force-deflection characteristics may be significant as a result of preload force differences that occur despite efforts to ensure that the preload nut is screwed onto the shaft the vendor-specified distance.

Following torque stand testing, the actuators were then installed and tested on valves having rising non-rotating stems. Strain gauges were installed on the stems of most valves and calibrated for thrust and torque. Several of these valves have been tested both with and without fluid flow through and differential pressure across the valve body. During testing of the refurbished actuator, both on the torque test stand and installed on the valve, actuator output torques and torque spring pack deflections were simultaneously measured and digitally recorded by calibrated test equipment. This paper concentrates on the results of torque stand testing.

The length of the effective moment arm that relates spring pack force to actuator output torque has been observed to vary during a stroke, from stroke to stroke, and from actuator to actuator. Nevertheless, the range for effective moment arm length variation reveals a consistency for size 00 actuators. Overall, there is a consistent relationship between spring pack force and actuator output torque over a load range up to the actuator's torque rating. It has been postulated that an axial stem load may alter the relationship between torque spring pack deflection and actuator output torque as a result of additional friction loads caused when the drive sleeve is subjected to axial loading. Additional study of test data will be required to address this possibility.

TORQUE SWITCH SETTING VERSUS TORQUE SPRING PACK DEFLECTION AND ACTUATOR OUTPUT TORQUE

Each torque spring pack was verified to be properly installed in its actuator prior to collecting torque stand or in situ test data. Spring pack gap was reduced to below 0.010 in. (typically below 0.002 in.). The torque switch was then balanced in the actuator, typically at open and close torque switch settings of 1.0. Each actuator was tested with the open and close torque switches at several settings.

Figure 1 presents the correlation between torque switch settings and torque spring pack deflections obtained for 29 size 00 actuators with a variety of gear ratios and torque spring pack models. Figure 1 provides a best fit line through

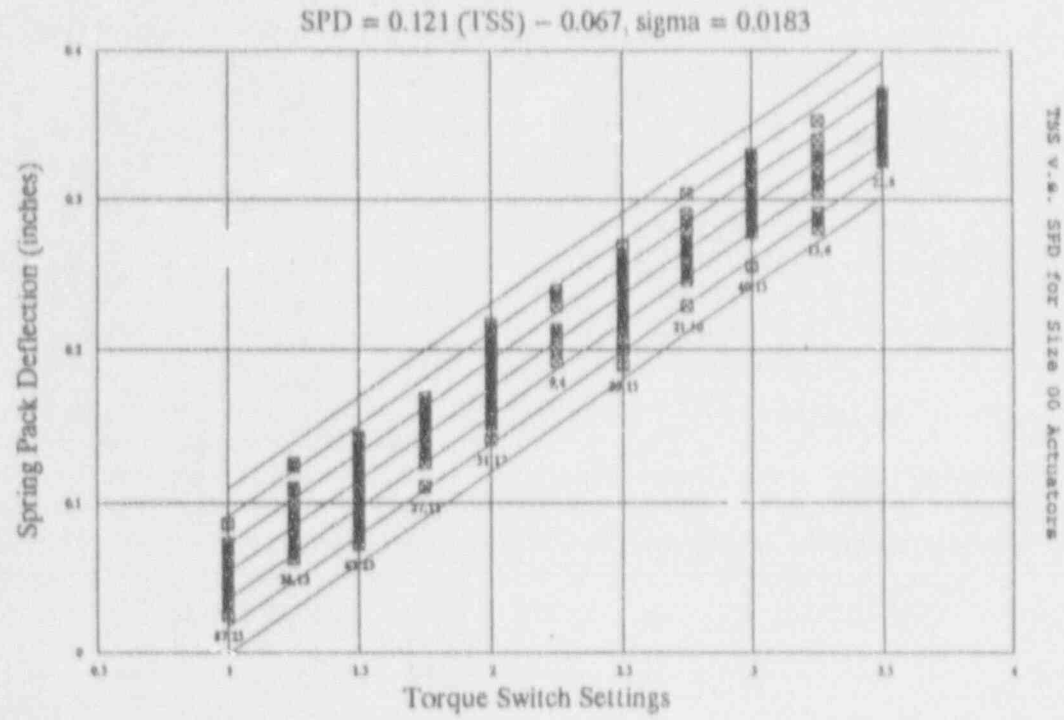

Figure 1. TSS vs. SPD for Size 00 Actuators.

the test data, and shows parallel lines at one, two, and three sample standard deviations above and below the best fit line. One sample standard deviation (one sigma) is 0.0183 in. for this data. Shown on Figure 1 are the number of open and close direction data points (411 total) and the number of actuators for which these data were collected at each torque switch setting from 1 to 3.5. The twenty-nine MOVs include twelve SMB-00, eleven SB-00, four SBD-00, and two SB-00S actuators, with the following torque spring packs: five 021, four 044, three 047, seven 048, four 049, three 050, and three 051.

Based on Figure 1 data, at a torque switch setting of 2 about 99.9% (Three sample standard deviations) of size 00 actuators have torque spring pack deflections of 0.175 ± 0.055 in. About 2% of nominally identical actuators with a torque switch setting of 1.25 or 2.75, and about 50% with a setting of 1.5 or 2.5, will also produce spring pack deflections within the range observed for a setting of 2.

These results illustrate part of the reason why utilities may observe substantial variation in actuator output torques for actuators of the same size, the same model torque spring pack, and the same torque switch setting. Figure 2 illustrates torque switch setting versus actuator output torque for six nominally identical MOVs. Torque switch setting versus actuator output torque data determined from the actuator manufacturer's design information is shown in Figure 2 by darkened squares connected by a line. The results reflect the findings shown in Figure 1.

Based on Figure 2 data, at a torque switch setting of 2, 99.9% of size 00 actuators with 048 torque spring packs will have actuator output torques between 45 ft·lb and 120 ft·lb. It is of interest to note that these data indicate about 10% of nominally identical actuators with a torque switch setting of 1 or 3 and 80% with a torque switch setting of 1.5 or 2.5 will also produce actuator output torques within the range observed for a setting of 2.

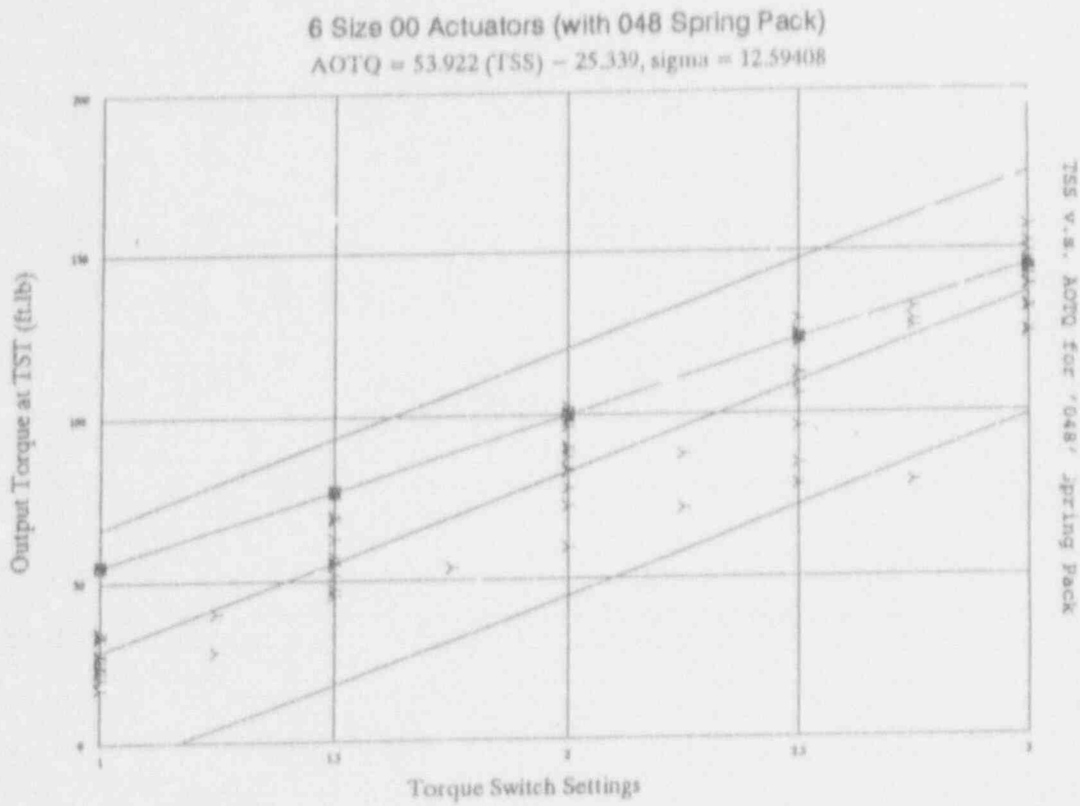

Figure 2. TSS vs. AOTQ for 048 Spring Pack.

Figures 3a and 3b, respectively, show torque switch setting versus deflection, and torque switch setting versus output torque results for one actuator with an 021 spring pack assembly. A total of 19 data points are plotted. A best-fit line and lines at ± 3 sigma are drawn through the data points. Based on Figure 3a results, a setting of 2 may yield a deflection, which on another occasion was produced at a setting of 1.5 or at a setting of 2.5. Based on Figure 3b results, a setting of 2 may yield a torque, which on another occasion was produced at a setting of 1 or at a setting of 3.

It would appear that even the effort of torque stand testing each individual actuator may reduce by only a small amount the uncertainty associated with future adjustments of the torque switch to a particular setting, ensuring that the torque switch trips at a desired spring pack deflection or within an acceptable torque range. Torque switch setting versus actuator output torque data determined from the actuator manufacturer's design information are shown in Figure 3b by darkened squares connected by a curve.

The preceding suggests that practice of controlling actuator output torque by selecting a torque switch setting from a representative setting-versus-torque design curve, or even a curve generated for a specific actuator by torque stand testing at several torque switch settings, may need to be reviewed to ensure it is appropriate for specific nuclear industry applications. If representative design curves are conservative from the standpoint of ensuring enough torque is produced to operate the MOV, there may be a lack of conservatism in ensuring design load limits and motor capabilities will not be exceeded prior to torque switch trip.

SPRING PACK DEFLECTION VERSUS SPRING PACK FORCE

Over seven different torque spring pack standard configurations have been available for size 00 actuators. Each spring pack model has a different nominal preload and nominal stiffness. Thus,

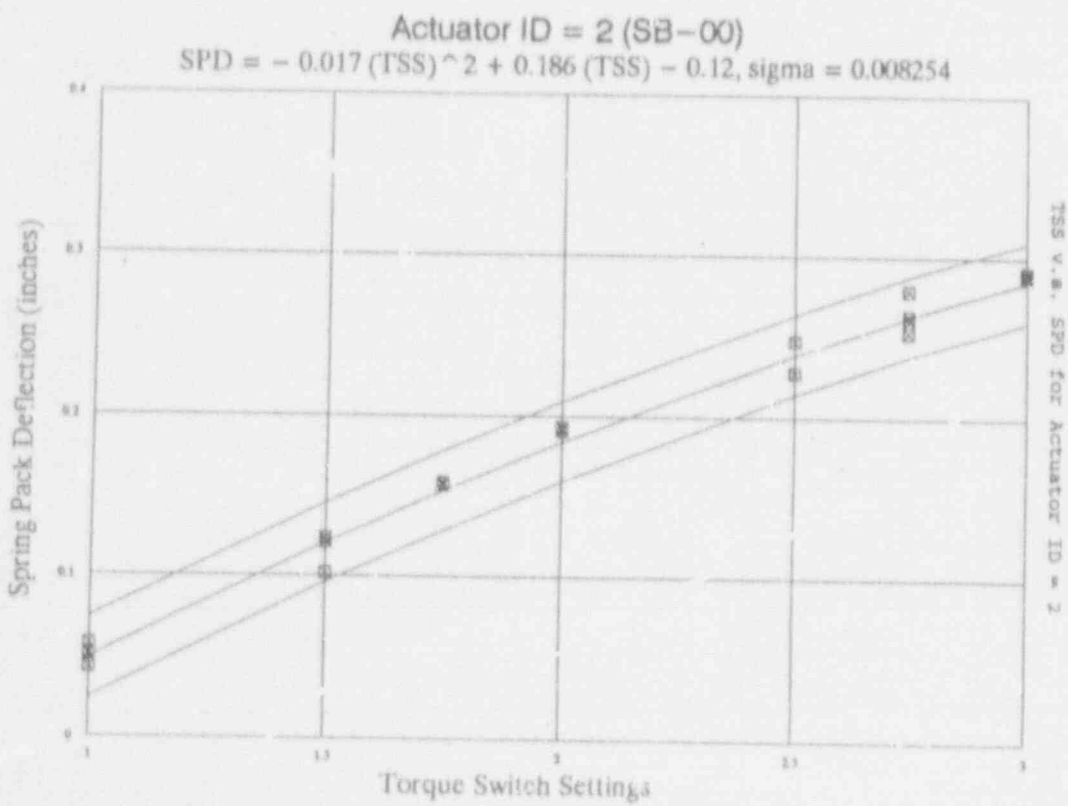

Figure 3a. TSS vs. SPD for Actuator ID = 2.

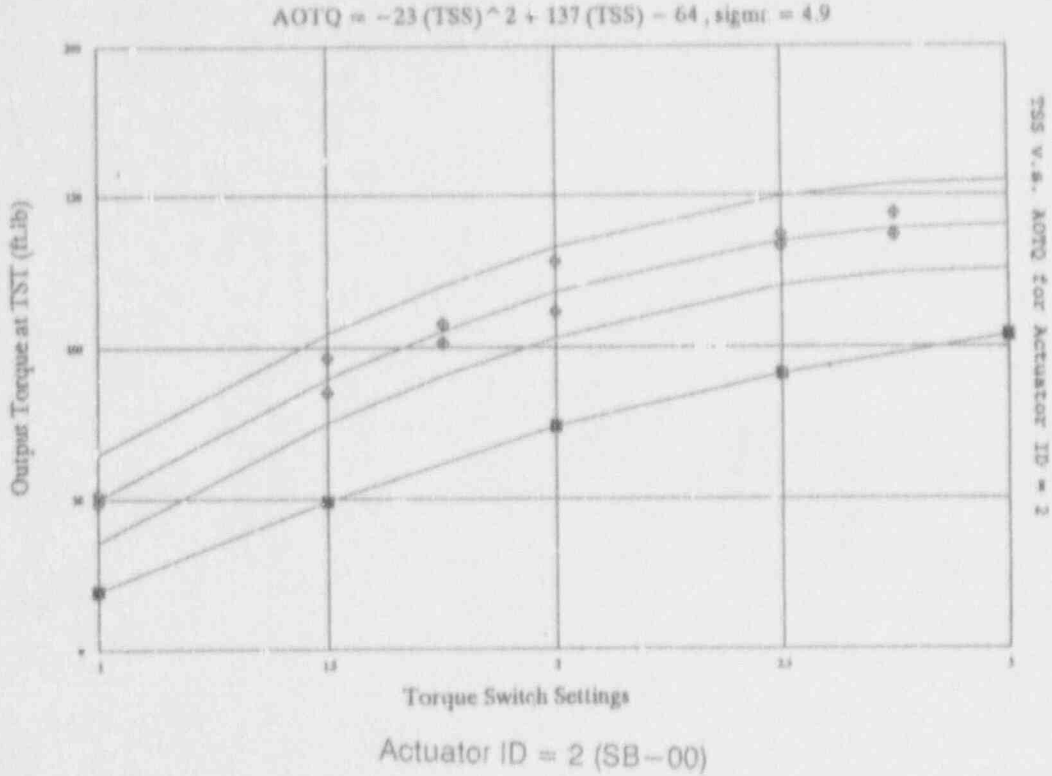

Figure 3b. TSS vs. AOTQ for Actuator ID = 2.

a specific spring pack deflection value is produced by a different typical force for each different spring pack model.

This paper focuses attention on a torque spring pack deflection range of 0.05 to 0.30 in. As Figure 1 shows, this deflection range typically corresponds to torque switch settings from 1 to 3 for size 00 actuators.

All force-deflection data reported here were obtained by compressing torque spring packs in a fixture while measuring and digitally recording the changing force and the resulting deflections. Prior to testing, each spring pack had been disassembled, cleaned, and reassembled to the configuration specified by the manufacturer. For nominally identical torque spring packs, these test results confirm that variations are to be expected in the spring pack force-deflection relationships determined for nominally identical spring packs.

Figure 4 shows 26 data points for each of twelve nominally identical model 021 torque spring packs (312 total data points). Each set of 26 data points lies on the best-fit second order curve through the force-deflection digital test data for each torque spring pack. A best-fit curve has been used in each case to eliminate the effect on data scatter of a cyclic noise pattern apparent in the deflection data. Figure 4 shows the best fit second order curve through the 312 total data points, and curves at ± 3 sigma.

Figure 4 indicates that there is a probability of 99.9% that a particular torque spring pack deflection will correspond to within ± 129 lb (± 3sigma) of a "typical" value for the 021 spring packs tested.

Figure 5 is similar to Figure 4, but provides data for four nominally identical model 049 torque spring packs. For the model 049 data, a 99.9% confidence level corresponds to ± 109 lb. It appears that the stiffness was very repeatable and that the biggest source of differences in performance was the preload. This variation was encountered, although a specific set of instructions for correctly configuring and preloading the

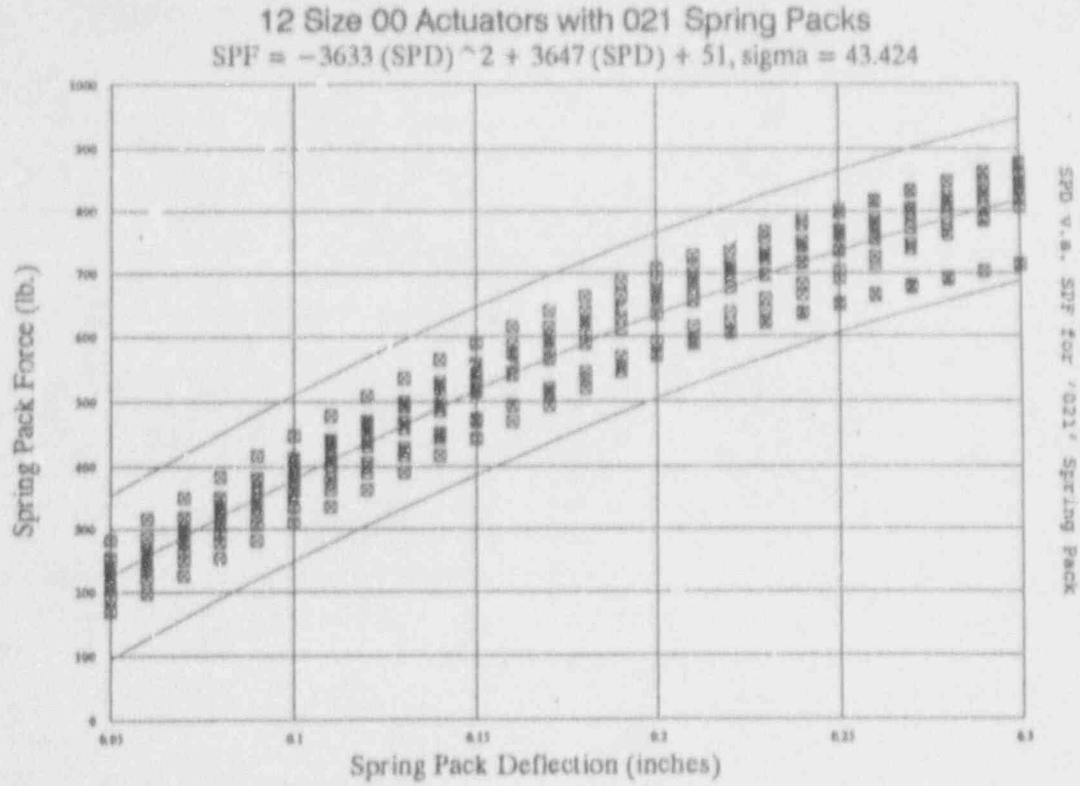

Figure 4. SPD vs. SPF for 021 Spring Pack.

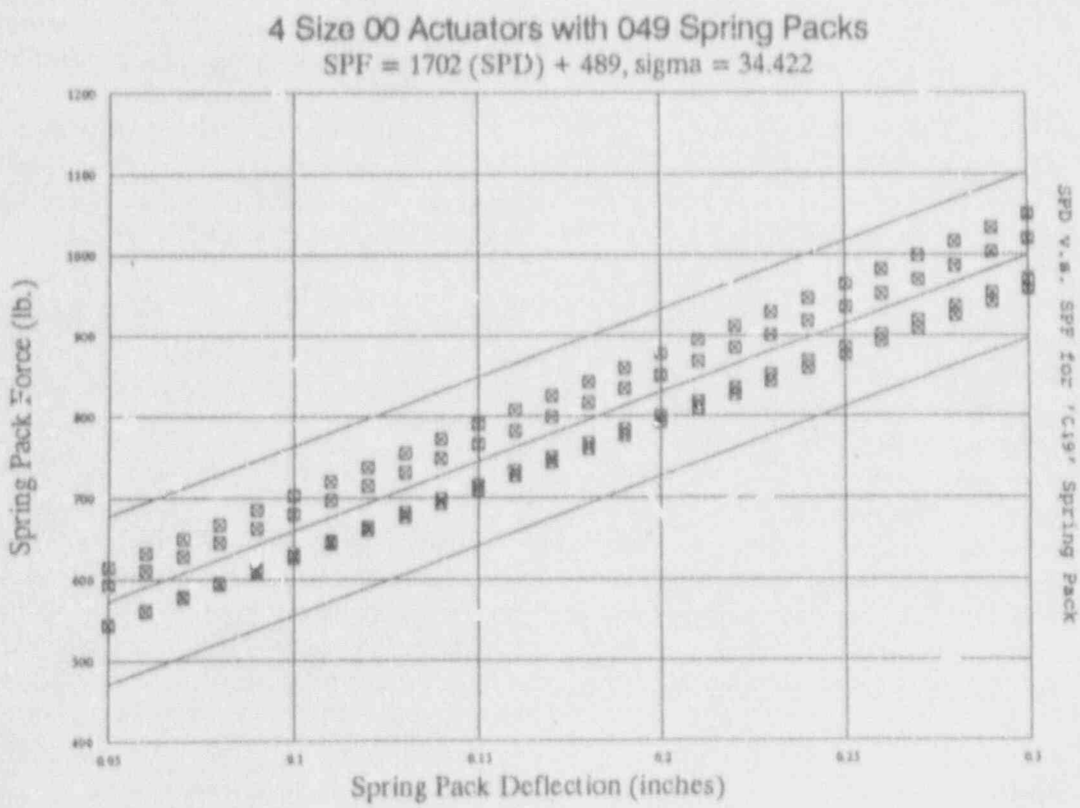

Figure 5. SPD vs. SPF for 049 Spring Pack.

spring pack was used. If adjustments to the spring pack and tests were repeated several times, the iterations could reduce the magnitude of differences observed.

It must be noted that contributors to the variations observed in Figures 4 and 5 are variations in the actual Belleville washer properties, the precision with which the spring packs were configured as specified by the actuator manufacturer, the repeatability of spring pack performance from one compression to another, and the accuracy of the test instruments and curve fitting analyses.

The variations in spring pack performance shown in Figures 4 and 5 demonstrate that a "representative" relationship of the torque spring pack force-deflection may differ by as much as 200 lb from the actual force on a 021 or 049 spring pack at a specific deflection. For stiffer spring packs the difference in force is likely to be even greater. This demonstrates that using a representative force-deflection relationship instead of the actual force-deflection relationship of the spring pack installed in the actuator introduces substantial uncertainty into the determination of actuator output torque from spring pack characteristics.

RELATING SPRING PACK DEFLECTION AND FORCE TO ACTUATOR OUTPUT TORQUE

When evaluating spring pack deflection measurements of the spring pack after it is installed in the actuator, evaluators of spring pack deflection have excluded any displacements that result at very low loads as a result of spring pack gap. In this way, the deflections measured for the installed spring pack may be more correctly correlated to deflections measured on the bench test fixture and to the forces that produced those deflections. At very small deflections, deflection measurement and force-deflection correlation errors become significant. Consequently, this paper focuses attention only on deflections greater than 0.05 in., approximately the deflection corresponding to a torque switch setting on 1 for the tested size 00 actuators.

The force that compresses the torque spring pack is approximately equal to the force imparted to the worm by the worm gear. Friction losses account for the difference between these forces. The force transfer between the worm and the worm gear occurs at approximately the theoretical pitch circle of the worm gear. Wear, manufacturing tolerances, and deflection under load can slightly move the actual location of load transfer. Consequently, the actuator output torque is approximately equal to the product of the pitch radius of the worm gear and the force that compresses the torque spring pack. Friction loads that oppose rotation of the worm gear and drive sleeve can cause additional difference between the actual actuator output torque and the value calculated using spring pack force and the worm gear pitch radius.

It is helpful to introduce a term, "effective moment arm" (MARM), which accounts for all friction effects and allows actuator output torque to be practicably calculated:

$$AOTQ = (SPF)(MARM) \quad (1)$$

The effective length of the actuator moment arm is determined by correlating a torque spring pack deflection measurement to the corresponding spring pack force, and dividing this force into the measured actuator output torque. (Some proprietary data on effective moment arm lengths have been developed by using a load cell to directly measure the spring pack force. These proprietary length values are then used with load cell measurements of spring pack force in other actuators to determine actuator output torques.)

Figure 6 plots actuator output torque against torque spring pack force for tests performed on five SB-00 and five SMB-00 actuators with four different model torque spring packs. This plot shows that actuator output torque can be determined fairly reliably as long as the spring pack force and the relationship between the spring pack force and actuator output torque are known. (During testing with the stiffest of the four spring pack models, the torque switch maximum setting was 1.75. At this setting the torque spring pack deflection was 0.145 in. and the actuator output torque at torque switch trip was 244.3 ft-lb in the close stroke on the torque test stand. Figures 6, 7, and 8 show output torques greater than 250 ft-lb only because the best-fit equations through the test data have been evaluated for deflections up to 0.3 in. for all four spring pack models. If the fictitious data points above the 250-ft-lb actuator torque rating were excluded, the second order equation for the best-fit curve and the calculated sample standard deviation would be slightly different from that presented in Figure 6.)

The best-fit second order curve and curves at ±3 sigma are shown in Figure 6. For the test data presented, the ±3 sigma uncertainty equals ±14 ft-lb. Other methods of presenting the same test data are shown by Figure 7 (AOTQ vs. SPD), Figure 8 (MARM vs. SPF), and Figure 9 (MARM vs. SPD).

Figure 7 clearly depicts the different actuator output torque ranges for these four spring pack models over a range of deflections from 0.05 to 0.30 in.

Figure 8 shows that the effective length of the moment arm generally increases with greater values of spring pack force. Contributors to this could be movement of the worm centerline away from the worm gear centerline and a lesser relative effect of friction losses. For each spring pack model, the range of variation in length of the effective moment arm is nearly the same. One actuator appears to have a larger than normal and fairly constant magnitude torque loss, which causes the value of MARM to be significantly reduced at the lower end of its load range. This is reflected, but not as clearly, by the Figure 6 data point at about (x,y) = (250 lb, 25 ft-lb).

Figure 8 tends to segregate the test data along the x axis, data for less stiff spring packs being grouped toward the left of data for stiffer spring packs. Figure 9 eliminates this segregation by plotting spring pack deflection along the x axis. Data for all tested spring packs is thus represented all along the x axis (including extrapolated data for the stiffest model as explained earlier).

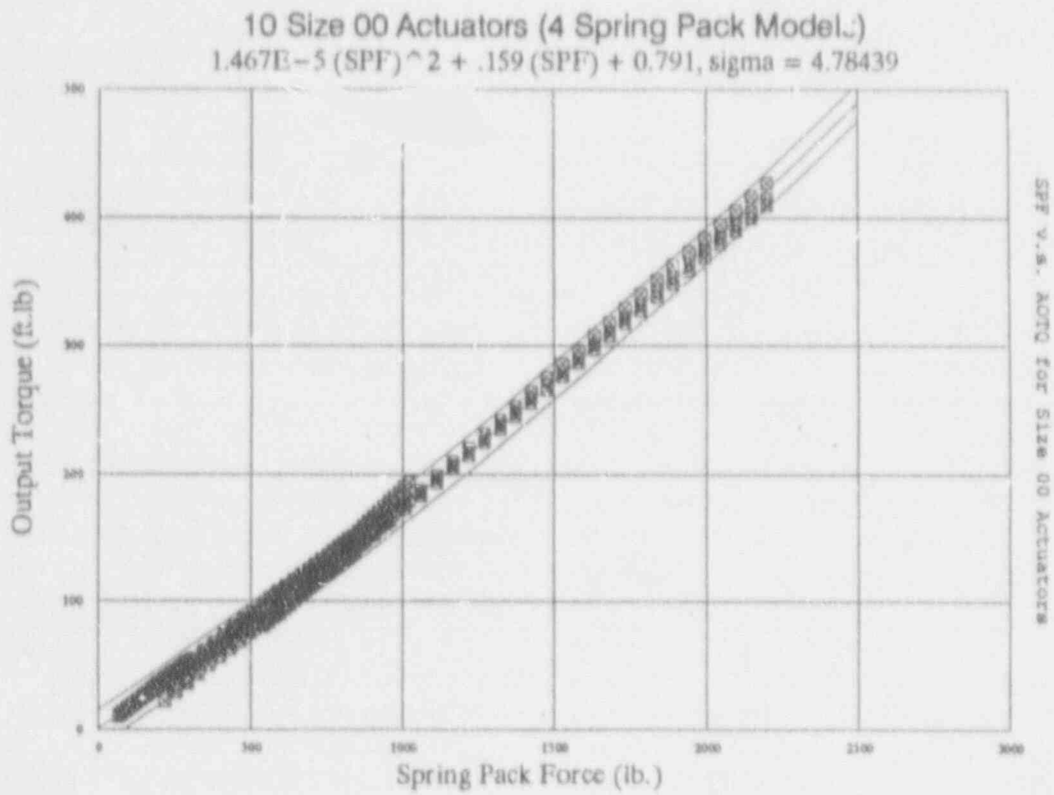

Figure 6. SPF vs. AOTQ for Size 00 Actuators.

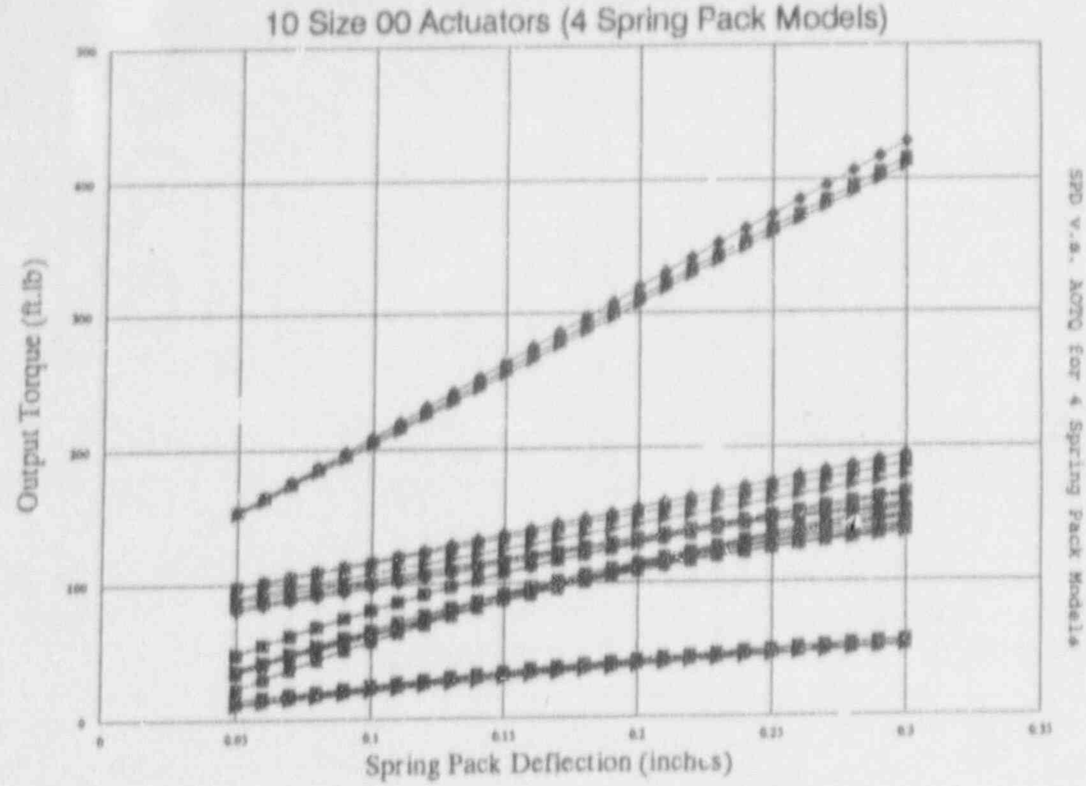

Figure 7. SPD vs. AOTQ for four Spring Pack Models.

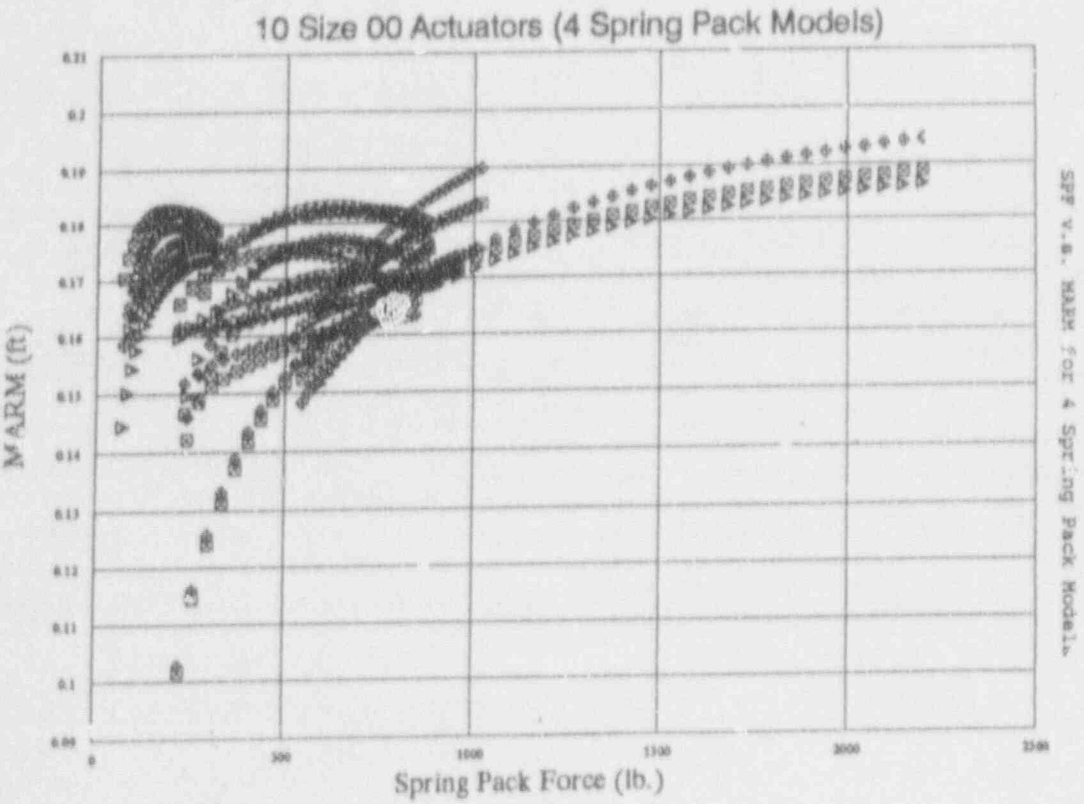

Figure 8. SPF vs. MARM for four Spring Pack Models.

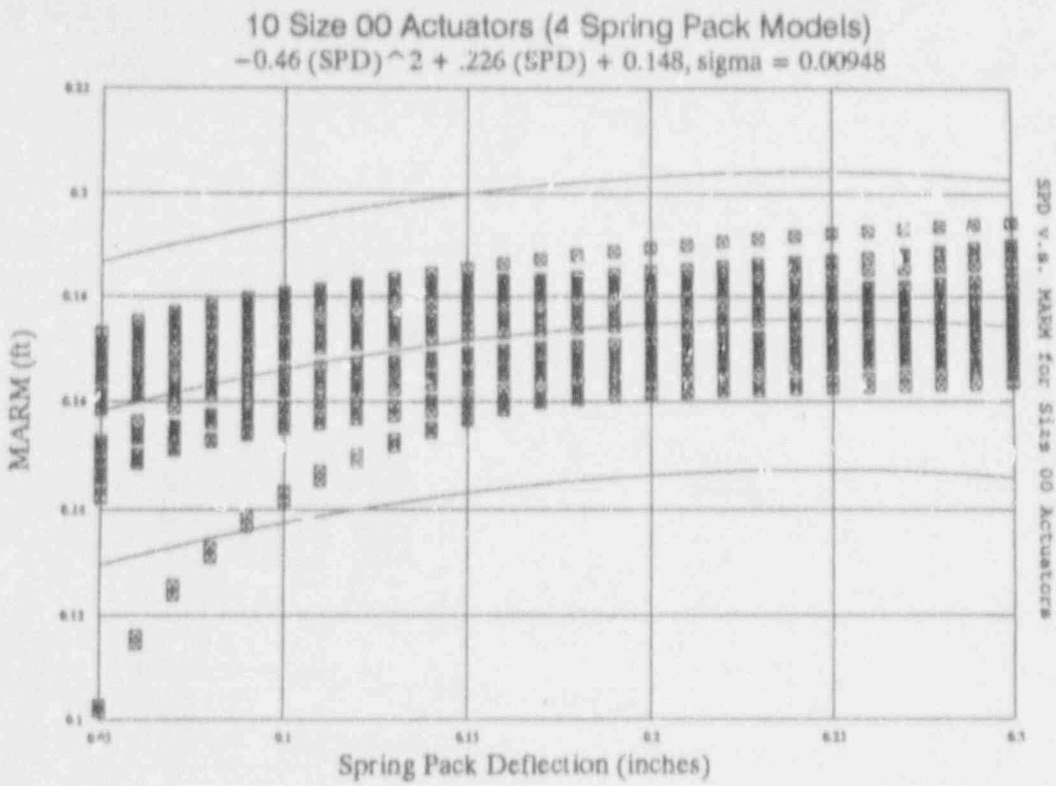

Figure 9. SPD vs. MARM for Size 00 Actuators.

Figure 9 shows that the length of the effective moment arm is generally consistent for the 10 actuators tested. For example, at a spring pack deflection of 0.2 in. the test data indicate the most typical moment arm length is 0.175 ft, with a 2-sigma deviation of 0.019 ft (10.8% of 0.175 ft) bounding the expected range of moment arm lengths for 97.7% of size 00 actuators. Figures 8 and 9 suggest that the value of MARM is affected not only by the magnitude of the spring pack force, but also by the magnitude of spring pack deflection (worm displacement).

Letting the Figure 9 data for ten size 00 actuators and four spring pack models serve as a sample of the size 00 actuator population, the following provides a means to calculate MARM as a function of SPD and to calculate the uncertainty (error) of MARM as a function of SPD for size 00 actuators.

$$\mathrm{MARM} = (-0.446\ \mathrm{ft{\cdot}in^2})(\mathrm{SPD})^2 + (0.226\ \mathrm{ft{\cdot}in})(\mathrm{SPD}) + 0.148\ \mathrm{ft} \qquad (2)$$

$$\mathrm{error} = (n)(\mathrm{sigma})/\mathrm{MARM} \qquad (3)$$

sigma = 0.00948 ft

n = 1, 2, or 3 for various confidence levels (see definition of "sigma")

Thus, if magnitudes of actuator output torque are needed at points in the valve stroke where the torque spring pack of Figure 10 has deflected various amounts, the torques may be calculated as shown in Table 1. Figure 11 shows the results of the Table 1 calculation along with test data (squares) of measured actuator output torques at various deflections of the Figure 10 spring pack.

The error for the value of MOTQ calculated as in Table 1 must be combined with other sources of error, including those of the instruments that collected the test data used to determine Figure 9 data points.

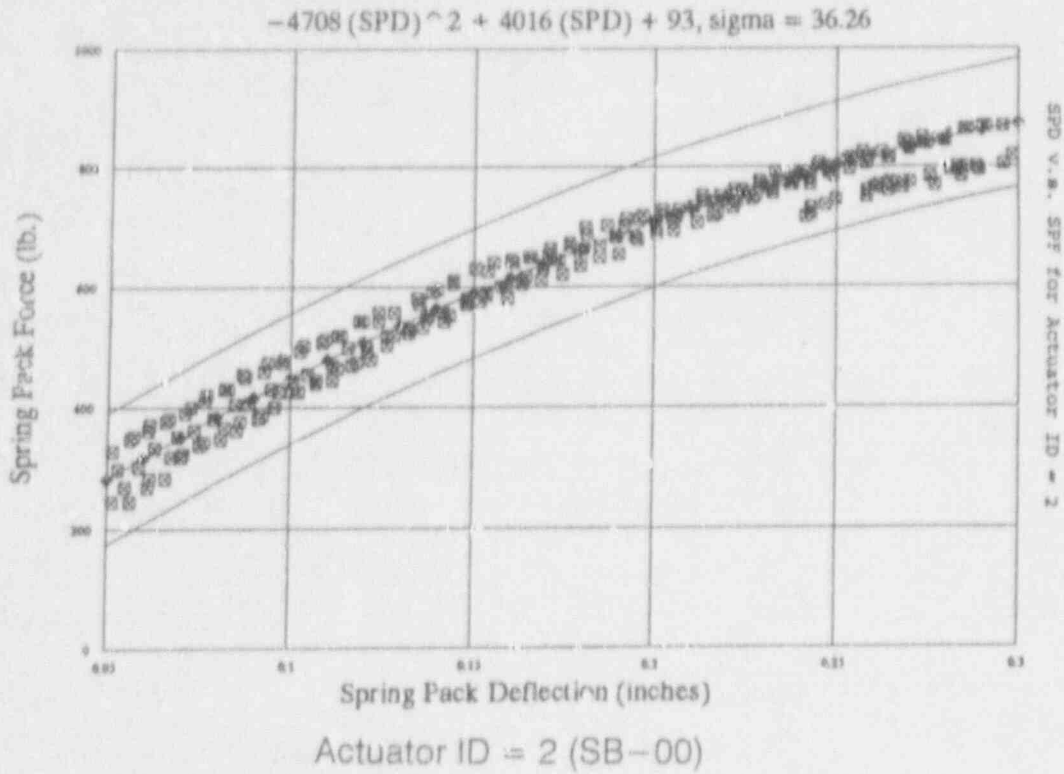

Figure 10. SPD vs. SPF for Actuator ID = 2.

Table 1. Using SPD, SPF, and MARM to calculate AOTQ.

SPD (in.)	SPF (lb) [Fig. 10]	MARM (ft) [Eqn(2)]	AOTQ (ft·lb) [Eqn(1)]	Error (ft·lb)	AOTQmin (ft·lb)	AOTQmax (ft·lb)
0.05	282	0.1582	44.6	8.0	36.6	52.6
0.10	448	0.1660	74.4	12.7	61.7	87.1
0.15	589	0.1716	101	17	84	118
0.20	708	0.1748	124	20	104	144
0.25	803	0.1758	141	23	118	164
0.30	874	0.1744	152	25	127	177

error = (3)(sigma)(AOTQ)/MARM
AOTQmin = AOTQ - error
AOTQmax = AOTQ + error

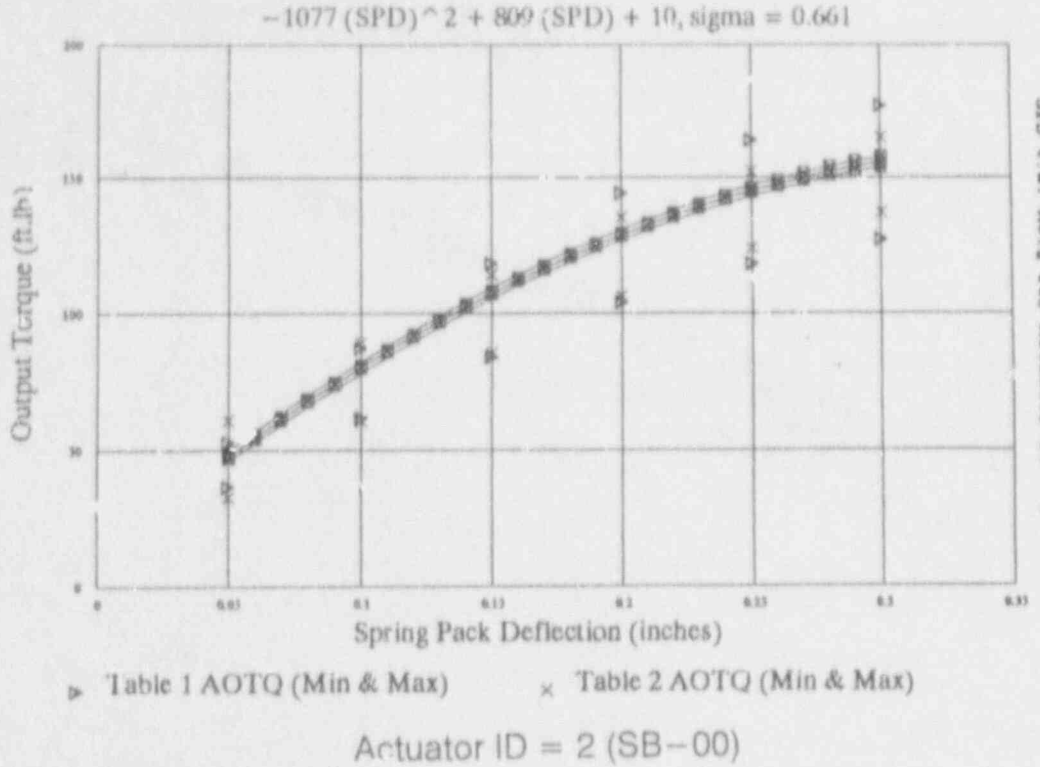

Figure 11. SPD vs. AOTQ for Actuator ID

Figure 11 shows that without considering the uncertainty of the effective moment arm length, Equations (1) and (2) yield calculated actuator output torque valves that are very close to the actual measured test data torque values. Measured torque values throughout the 0.05-to 0.30-in. deflection range are bounded by the torque values at ±3 sigma.

This example demonstrates that torque spring pack deflection measurements in the actuator can be used along with:

1. Force-deflection data obtained by testing that spring pack outside the actuator

2. An equation for the "typical" effective moment arm length (MARM) for that size actuator

3. The uncertainty of the "typical" MARM equation to calculate with very good reliability the actuator output torque that caused the torque spring pack to deflect the amount measured.

It should be noted that the ±3 sigma uncertainty determined by the method shown in Figure 6 for the same test data is only ±14 ft·lb. In contrast, the calculation results plotted in Figure 11 have an uncertainty of 8 ft·lb at 44.6 ft·lb actuator output torque (0.05-in. deflection) and 25 ft·lb at 152 ft·lb actuator output torque (0.30-in. deflection). Thus, it may be that an alternate and equally reliable method of determining AOTQ for a particular SPD is to

1. Obtain the corresponding SPF from spring pack force-deflection test data

2. Directly calculate AOTQ by means of an equation such as is presented in Figure 6

3. Add or subtract as appropriate the uncertainty of the calculated AOTQ.

Table 3 shows that this later method actually involves fewer steps:

Table 2. Using SPD and SPF to calculate AOTQ.

SPD (in.)	SPF (lb), Figure 10	AOTQ (ft·lb)	AOTQ min (ft·lb)	AOT Qmax (ft·lb)
0.05	282	46.8	32.4	61.1
0.10	448	75.0	60.6	89.3
0.15	589	99.5	85.2	114
0.20	708	121	106	135
0.25	803	138	124	152
0.30	874	151	137	165

AOTQ = (1.467E-5 ft·lb/lb^2)(SPF)^2 + (0.159 ft·lb/lb)(SPF) + 0.791 ft·lb [from Figure 6]
AOTQmin = AOTQ -3(sigma)
AOTQmax = AOTQ +3(sigma)
sigma = 4.784 ft·lbf

Figure 11 shows the results of this alternate Table 2 AOTQ calculation for comparison with the results of the Table 1 AOTQ calculation and actuator output torque test data (squares). Each of the two methods presented for calculating actuator output torque from spring pack deflection measurements and spring pack characteristics may provide torque values of greater certainty than the other method for particular ranges of spring pack deflection.

Figure 11 shows that for actuator ID = 2 with an 021 spring pack, the first calculational method (Table 1) provides greater accuracy at lesser AOTQ values; but the second calculational method (Table 2) provides greater accuracy at greater AOTQ values. For actuators with stiffer spring packs, all Tables 1 and 2 SPF values and AOTQ values will be greater. Table 1 calculated torque error values will increase with increasing SPF values. Table 2 torque error values do not increase. Consequently, it appears that for all AOTQ values greater than about 80 ft·lb, the Table 2 (Figure 6) methodology may provide a more certain determination of AOTQ.

SUMMARY

1. For size 00 actuators, while a balanceable-style torque switch may be balanced at one torque switch setting, it may not be balanced at other settings.

2. For size 00 actuators with balanced torque switches, equal torque switch settings (1, 1.5, 2, etc.) have been observed to produce a significant range of torque spring pack deflections at torque switch trip.

3. Tests of several nominally identical torque spring packs show that variations in spring

pack force-deflection characteristics may be significant as a result of preload differences that occur despite the use by utility personnel of standard assembly instructions.

4. For size 00 actuators with balanced torque switches and the same model torque spring pack, equal torque switch settings (1, 1.5, 2, etc.) have been observed to produce a significant range of actuator output torques at torque switch trip. The variations in actuator output torques for the tested actuators appear to result from (a) equal torque switch settings producing different torque spring pack deflections at torque switch trip, (b) equal torque spring pack deflections corresponding to different torque spring pack forces, and (c) equal torque spring pack forces producing different actuator output torques.

5. The force on the spring pack when it is installed in the actuator can often be reliably determined using deflection measurements of the spring pack in the actuator along with the spring pack force-deflection characteristic determined by testing the torque spring pack outside the actuator. Using a "representative" force-deflection characteristic instead of the unique characteristic for each torque spring pack will introduce uncertainty into the determination of spring pack force and into any subsequent determination of actuator output torque.

6. With some exceptions, the relationship between torque spring pack force and output torque for size 00 actuators is very consistent for loads up to the 250 ft-lb torque rating. Thus, actuator output torque can often be reliably calculated as a function of the force on the torque spring pack. The degree of uncertainty with the resulting calculated actuator output torque value may be dependent upon the methods of relating spring pack deflection to spring pack force, and spring pack force to actuator output torque.

7. After a torque switch has been adjusted from one setting to another, and readjusted back to the original setting, the deflection at torque switch trip is unlikely to be the same as it originally was. The scatter in the deflections that may result upon doing this several times for one actuator may approach the scatter in deflections observed for several actuators all with the same torque switch setting. Consequently, even for a specific actuator, torque stand test results that indicate a particular torque switch setting produced a particular actuator output torque may not be a reliable indicator of the torques that will result from later adjustments of the torque switch to the same settings.

8. Measured spring pack deflections and forces can be much more reliably correlated to actuator output torque than can torque switch settings, and are unaffected by torque switch setting adjustments. For an individual actuator, the uncertainty involved with relating spring pack force or deflection to actuator output torque can be minimized by a torque stand test wherein these parameters are measured with appropriate accuracy.

9. Additional evaluation of test data is required to ascertain whether axial stem load reacted by the actuator drive sleeve may cause a change from the relationship between spring pack force and actuator output torque as determined on a torque test stand.

10. The practice of controlling actuator output torques by selecting torque switch settings from "representative" setting-versus-torque design data, or even from setting-versus-torque test results of each actuator, may need to be reviewed to ensure it is appropriate for specific nuclear industry applications.

ACKNOWLEDGMENTS

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NOMENCLATURE

Acronyms

AOTQ	actuator output torque
MARM	effective length of actuator moment arm; the length that when multiplied by the force on the spring pack results in a value equal to the actual actuator output torque.
MOV	motor operated valve
SPD	torque spring pack deflection
SPF	torque spring pack force
TSS	torque switch setting

Terminology

Balanced Torque Switch	If the torque switch is "balanced," equal torque switch settings in the open and close directions will correspond to equal axial displacements of the worm and equal deflections of the torque spring pack. In practice while the torque switch may be balanced within 0.010 in. at one setting, it is often not balanced within 0.010 in. at other settings. A common practice is to balance the torque switch at the least torque switch setting, which permits actuator operation in both directions (typically, a setting of "1").
DP Test	A test during which the MOV in a piping system is stroked while there is a differential pressure across the valve seats and fluid flow through the valve seats.
sigma	One standard deviation of a sample taken from a population. Assuming a normal distribution, the probability that an event will occur is 84.1%, 97.7%, and 99.9% for one, two, and three standard deviations, respectively. When analyzing data herein, bounds which provide a probability of 99.9% are used. Such a high level of assurance is often commensurate with considerations affecting the safety of nuclear power plants. (These statistical analyses are not intended to be rigorous, but are intended to provide in a simple manner a consideration of uncertainties.)

Spring Pack Gap	If the spring pack adjustment collar is either too tight or too loose, then the spring pack will move along its axis under very little load when the actuator stroke direction is changed. The total distance the spring pack may displace in this manner is called the spring pack gap.
Static Test	A test during which the MOV in a piping system is stroked while there is negligible or zero pressure (psig) in the valve body.
Torque Spring Pack	A spring assembly that balances the axial load applied to the worm by the worm gear.
Torque Spring Pack Deflection	When the axial load imparted by the worm to the torque spring pack exceeds the preload force on the torque spring pack, the worm *displaces* as the torque spring pack *deflects* (compresses). Torque spring pack deflection is commonly (and imprecisely) called "spring pack displacement." To clarify, spring pack displacement occurs only if there is a spring pack gap, and ceases to displace when it has moved an amount equal to that gap.
Torque Spring Pack Force	The compressive force that causes torque spring pack deflection and that (when multiplied by the length of the effective moment arm of the actuator) produces the actuator output torque.

Special Considerations for Testing Rising Rotating Stem MOVs

Anthony Moffa, Liberty Technologies

ABSTRACT

Rising stem gate and globe valves have one plane of motion: linear. The stem is either pushed or pulled into position. For rising and rotating stems however, there are two planes of motion: linear and rotational. The stem is twisted in addition to being pushed or pulled into position. Typical motor operated valve (MOV) sizing equations account only for the linear requirements of the valve to open or close. Theoretical calculations performed for a two-dimensional system predict that in the running load region, rotational torque requirements far exceed the linear requirements. To validate the theoretical model, torque testing of rising rotating stem valves was preformed, using Liberty Technologies Valve Operation Test and Evaluation System (VOTES). Theoretical and empirical data have produced a new perspective for operational requirements and a guideline for testing rising rotating stem valves.

INTRODUCTION

Background

A motor operated valve (MOV) is a mechanical assembly consisting of a valve and an electric-(ac or dc) actuated gear box. The gear box is designed to generate torque, which is then converted to thrust and used to open or close a valve. The actuator torque output is determined by the actuator size, spring pack size, motor size, gearing, and control switch settings. The torque to thrust conversion is a function of the valve stem thread geometry, which is generally determined in terms of the expected operating requirements, such as closing thrust and stroke time.

The process of sizing an actuator for a given valve involves a series of calculations covering stem thrust, gearing, and motor applications. Most utilities have an actuator manufacturers selection guide that they use as a reference. The thrust calculations in these publications use known values, such as stem geometry and system pressure, and statistical values for the various coefficients of friction to produce the total stem thrust required for the valve to operate against system conditions. Once stem thrust is calculated, it is converted to torque. The output torque requirements are used to select actuator size, motor size and gear ratios.

The stem thrust calculation can be divided into three parts:

1. Differential Pressure (DP)
2. Stem Rejection (SR)
3. Packing Friction (PF).

The sum of these three loads is the total stem thrust required to operate the valve against system pressure.

Differential Pressure (DP)

System pressure creates a normal load (perpendicular to disk face) on the valve disk. For a gate valve, the DP load is the product of this normal load and the valve factor. The valve factor is the coefficient of friction between the valve disk and seat; it varies from valve to valve. However, a reasonable estimate for a solid wedge gate valve is 0.3. Equation 1 is used to calculate DP stem load for a gate valve. This load is a bi-directional force acting parallel to the stem centerline. It will always oppose stem motion and is independent of flow direction.

$$DP_{Load} = \pi \frac{d_{SR}^2}{4} P_{System} \mu_{VF} \qquad (1)$$

where

DP_{Load}	=	differential pressure load
P_{System}	=	maximum system operating pressure (psi)
d_{SR}	=	diameter of seat ring (in.^{2})
μ_{VF}	=	valve factor.

In the case of a globe valve, the valve factor (μ_{VF}) will default to a value of 1.1 in the previous equation. Additionally, the force is now uni-directional and dependent upon flow direction: it opposes stem motion in the open direction for flow over the disk, and in the closing direction for flow under the disk (most common).

Stem Rejection Load

The stem rejection load, or piston effect, is a uni-directional force that also acts parallel to the stem centerline. System pressure creates a load that attempts to push the valve stem out of the actuator. It works against the actuator as it tries to close the valve and with it as it opens the valve. The rejection load is the product of the stem cross sectional area and the system pressure. It is independent of flow direction for both gate and globe valves. Equation 2 can be used to calculate the stem rejection load.

$$SR_{Load} = \pi \frac{d_{Stem}^2}{4} P_{System} \qquad (2)$$

where

SR_{Load}	=	stem rejection load (lb)
d_{Stem}	=	stem major diameter (in.).

Packing Friction

The third load, packing friction, is the most complex axial load. In has been common practice for the packing friction force to be estimated, 1,000 pounds per inch of stem diameter. Packing friction is a function of the stem and gland diameters the stem and packing contact area, packing bolt preload, and the coefficients of friction between the stem and packing and the packing nut and stud. Typically, valve and packing manufacturers have a desired gland pressure they want to achieve. Using the gland pressure, packing cross sectional area, and the number of packing bolts, they calculate the necessary bolt preload force, then convert it to bolt torque. For the end user, only the bolt torque is available. Equation 3 (Shigley, 1983) can be used to calculate bolt preload from bolt torque.

$$F_B = \frac{T_{Nut}}{\frac{d_{mP}}{2}\,\frac{(l + \pi \mu_{PB} d_{mP} \sec \alpha)}{(\pi d_{mP} - \mu_{PB}\, l \sec \alpha)} + \frac{\mu_c d_c}{2}} \qquad (3)$$

where

F_B	=	Force on one packing bolt (lb)
T_{Nut}	=	Torque on one packing bolt (in·lb)
α	=	1/2 the thread angle (30 degree)
d_{mP}	=	Packing bolt pitch diameter
l	=	thread Lead (pitch x thread starts) (in)
μ_{PB}	=	coefficient of friction between the Packing Bolt and Nut
μ_c	=	coefficient of friction between the packing nut and collar
d_c	=	diameter of packing nut and collar contact area.

Using the bolt preload, stem and packing dimensions, and the coefficient of friction between the stem and packing, the packing friction load can be calculated. Packing friction is a bi-directional force acting parallel to the stem centerline.

$$PF_{Load} = \frac{4NF_{Bolt}\, d_{Stem}\, H_P}{\left(D_G^2 - d_{Stem}^2\right)}\, \mu_{SP} \qquad (4)$$

where

PF_{Load}	=	packing friction load (lb)
N	=	number of packing bolts
H_P	=	height of packing (in.)
D_G	=	gland diameter (in.)
μ_{SP}	=	coefficient of friction stem to packing.

The sum of the forces calculated in Equations 1, 2, and 4 is the total required stem thrust to operate against system pressure. One final equation needs to be employed to produce actuator torque from stem thrust. The equation is referred to as the stem factor equation. It is derived from the Acme Power Screw equation (Shigley, 1983) for raising a load. The equation presumes that an applied rotational torque will be converted into an axial load.

$$FS = \frac{d_{ms}}{2}\, \frac{(l + \pi\, \mu_{SSN}\, d_{ms}\, \sec\alpha)}{(\pi\, d_{ms} - \mu_{SSN}\, l\, \sec\alpha)} \qquad (5)$$

where

FS	=	stem factor
d_{ms}	=	stem pitch diameter (ft)
l	=	Thread lead (pitch x thread starts) (ft)
μ_{SSN}	=	Coefficient of friction stem and stem nut.

Note: Due to the uncertainties associated with the friction coefficients, μ_{SSN}, μ_{PB}, μ_{PS} and μ_{VF}, the sizing equation should only be viewed as an approximation. Actual stem thrust and actuator torque values must be verified through some form of diagnostic testing or functional test.

With the required thrust and torque identified, the sizing process would continue with the selection of the actuator size, spring pack size, motor size, and internal gear sets. Even with all the estimated friction coefficients, the sizing equation can be fairly accurate, although there are some limitations. A review of all the equations and resultant forces indicates that these sizing equations are useful for calculating *axial loads only*. All motion is limited to one axis. For rising stem gate and globe valves, this is not a concern. The stem only moves along the axis of the stem centerline. Rising and rotating stem valves present a different picture: two planes of motion, axial and rotational.

RISING ROTATING STEM VALVES

A rising rotating stem valve is very different from a normal rising stem valve. Figure 1 shows the side view of an Edward's Y globe. There are two distinct differences between this valve and a rising stem valve: the actuator drive sleeve interface and location of the stem nut.

In a rising stem valve, the stem threads into the stem nut, which is mated to the drive sleeve. Rotation of the drive sleeve results in rotation of the stem nut and axial motion of the stem. In the rising rotating stem valve, two separate components are used to perform the same function as the rising stem stem nut: the spline and threaded bushing. The spline is a grooved metal cylinder that is bolted to the end of the stem. The groves on the spline mate with grooves on the inside of the actuator drive sleeve. The function of the spline is to transmit actuator torque to the stem while allowing for axial motion of the stem. The torque to thrust conversion is performed by the threaded bushing mounted in the top of the valve yoke.

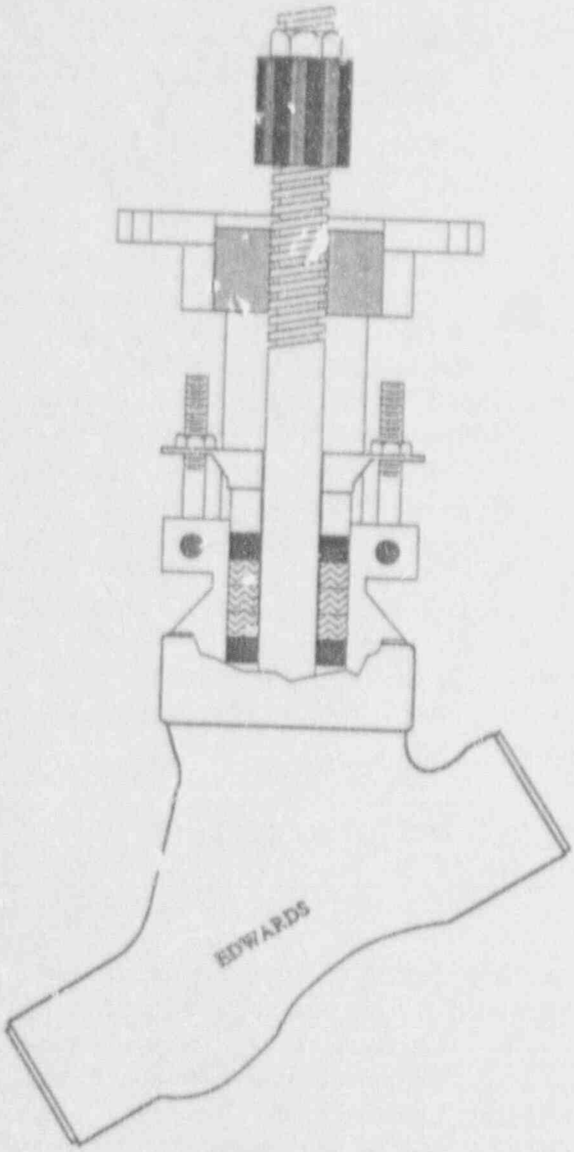

Figure 1. Edwards globe valve.

Packing Friction

The rising rotating stem valve is subject to the same axial loads as the rising stem valve. However, in addition to the axial loads, the stem is subject to radial loads that have not been accounted for in the previous sizing equations.[a]

a. Representatives from Edwards indicated their sizing equations are proprietary, but do account for additional radial loads, unlike the Limitorque and EPRI publications.

Of the three loads, packing friction is the only force component that places additional loading on the valve stem when rotation is induced. The DP_{LOAD} and SR_{LOAD} are constants (functions of the disk and stem diameter, valve factor, and system pressure), producing no additional torque or force requirements due to the rotational motion of the stem.

Figure 2a is the side view of the stem and packing cross section. F_{PN} is the packing force normal to the stem diameter and PF_{Load} is the resultant force resisting stem motion. In order to move the stem down, a force must be generated that exceeds PF_{Load}. The actuator torque required to overcome packing friction is calculated using the following equation

$$T_{Axial} = PF_{Load} \times FS \quad . \tag{6}$$

Figure 2b is the top view (Section A-A) of the same stem. In the event stem motion was limited to rotation, there would be little or no force in the axial direction, only torsional resistance. Clockwise rotation of the stem would result in counterclockwise torsional resistance from PF_{Load}. The actuator torque required to rotate the stem is a function of packing friction and the stem radius

$$T_{Radial} = PF_{Load} \times r_{Stem} \quad . \tag{7}$$

In the case of the rising rotating stem valve, both axial and rotational resistance is experienced, thus Equations 6 and 7 need to be combined in order to determine the total torque required to move the stem against Packing Friction.

$$T_P = PF_{Load} (FS + r_{Stem}) \quad . \tag{8}$$

Note: When using conventional stem factor tables supplied by valve and actuator manufacturers, the stem radius in this equation, should be in feet, not inches.

Equation 8 is the total torque required to push/pull and rotate the stem through packing. Comparison of the two components shows the radial torque requirement to be significantly higher than

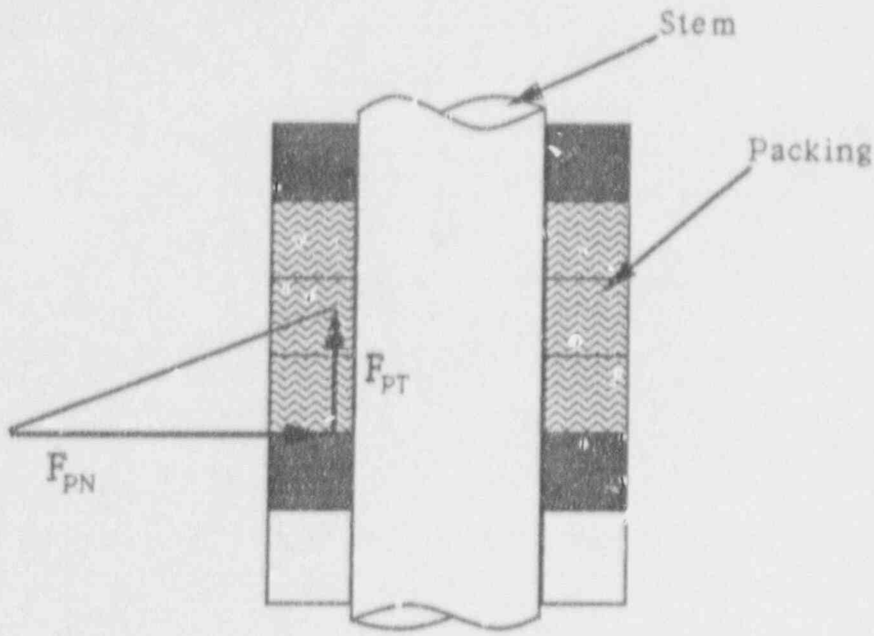

Figure 2a. Stem and packing cross section (side view).

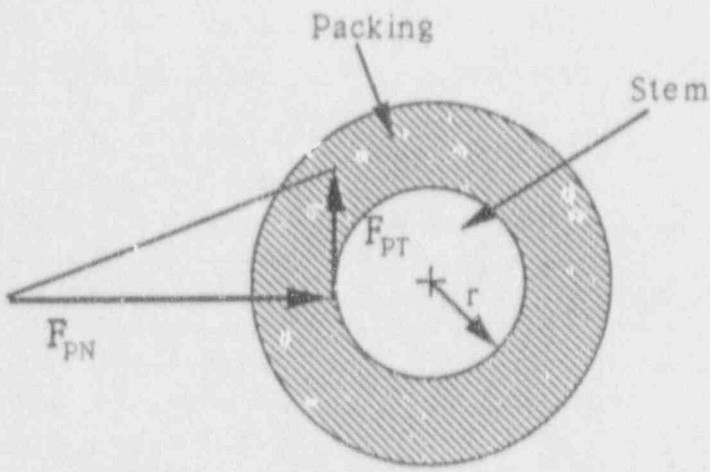

Figure 2b. Stem and packing cross section (top view).

the axial torque requirement. Dividing Equation 7, (T_{Radial}), by Equation 8, (T_P) produces a ratio of radial torque requirements to total torque requirements. Figure 3 is a graphical representation of this ratio for a 1.000 in. diameter, 6 thread per inch (TPI) stem. The graph shows that for low coefficients of friction between the stem and stem nut, the rotational running torque comprises more than 90% of the total running torque. Even at a 0.3 coefficient of friction, rotational loads account for more than 77% of the total running torque. It must be stressed that all these comparisons are based on running loads only, just moving the stem against packing, no differential pressure, stem rejection, or seating loads are being considered at this time.

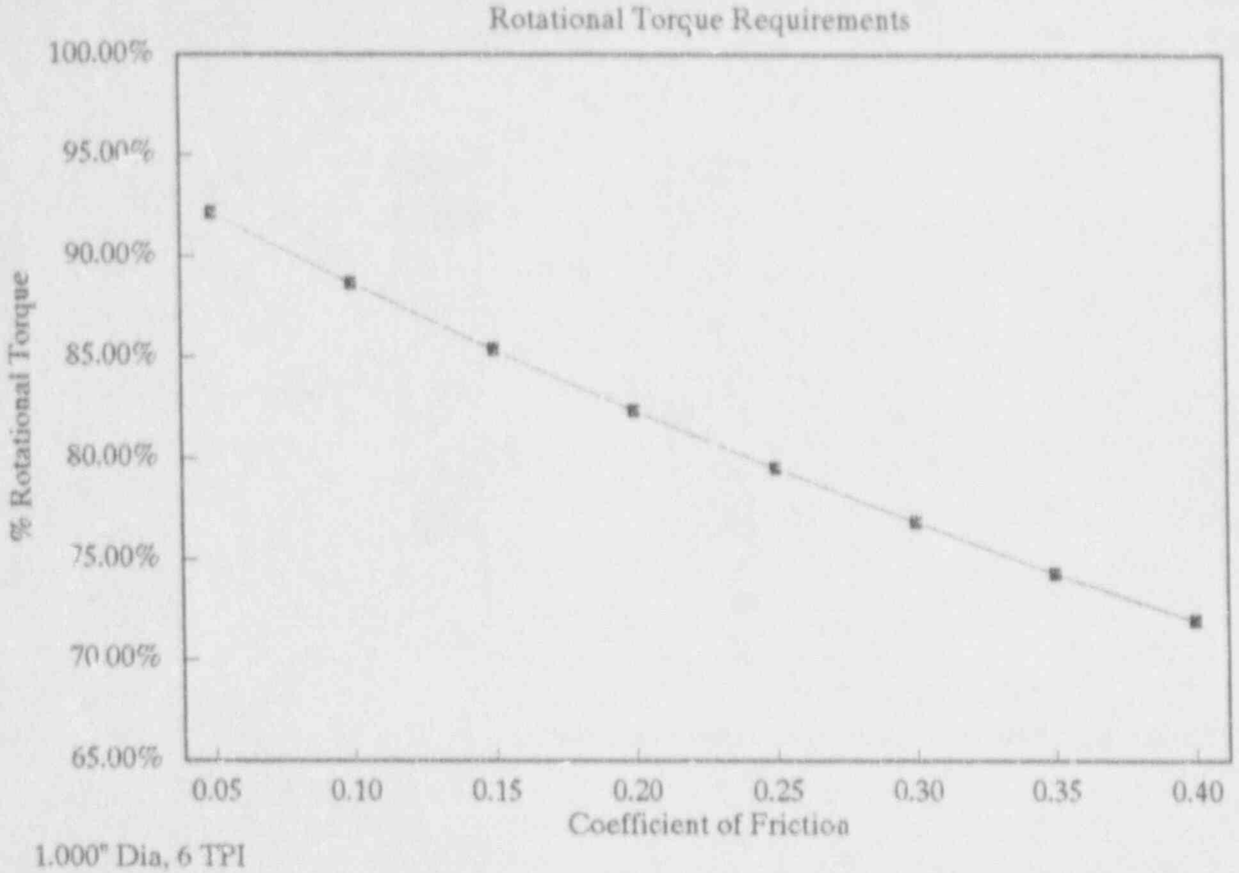

Figure 3. Rotational torque requirements.

Additional Areas of Concern

Spline. The stem-to-actuator spline has two functions: transmit actuator torque to the stem and allow axial motion of the stem. If the spline has a rough surface, a gouge, or is not lubricated, the axial motion of the stem will be impeded. Torque transmission, however, will not be affected. The axial restriction would require additional actuator torque to move the stem. As a precaution, the spline should be checked for visual flaws, when possible, and should be lubricated on a regular basis.

Threaded Yoke Bushing. The stem nut, or in this case, threaded yoke bushing, is accounted for in the stem factor equation. As with all the sizing equations listed here, there are limitations to their application. In the case of the rising rotating stem valve, the physical layout of the valve makes it prone to experience cyclic loading. This results primarily from the alignment of the spline, bushing, and packing gland. Each of the components can be represented as a cylinder. In a rising stem valve, two cylinders need to be aligned: the stem nut and packing gland. Alignment of the two cylinders is aided by the appreciable distance between the two and only one of the cylinders is "fixed" (stem nut). For the rising rotating stem valve, three cylinders need to be aligned: the spline, stem nut, and packing gland. Maintaining cylindricity between the three components is difficult. The largest portion of the misalignment comes from the stem nut. Typically, it is threaded or pressed into the top of the yoke. The combination of the thread and machining tolerances results in poor alignment of the three cylinders, and cyclic loading that occurs at the drive sleeve output speed. This misalignment potentially places large radial loads on the bushing, resulting in it wearing at an abnormal rate. Realignment is difficult, the only action that can be taken is to lubricate the bushing regularly.

EMPIRICAL DATA

Test Objective

The test objective was to validate the theory by acquiring running torque profiles from rising rotating stem valves set at incremental packing friction loads.

Test Procedure

Test equipment included three Edwards 2-in. rising rotating stem globe valves packed with three rings of Chesterton's Style 5300 (GTPI) - Die-Formed Inhibited Graphite Rings; one Walworth 4-in. rising stem globe valve packed with three rings of ARGO's 6300 J Flexible Graphite packing; two Limitorque SMB-000 Actuators; a VOTES System, a Mini-C Clamp, Portable Strain Meter, and VOTES Torque Cartridge. One of the Limitorque actuators was used to run the Walworth valve, and the other was shared between the three Edwards valves. Both actuators have the same gear ratio. The Edwards valves all have a 1.000-in. diameter, 6 TPI, stainless steel stem. The Walworth stem is 1.125-in. diameter, 4 TPI, and is made of naval brass.

The Edward's valves were tested first. Data was acquired at 0 ft·lb torque on the packing nuts for a base line. Then the packing nuts were torqued to 5.0 ft·lb and running load data was acquired. Bolt torque values were increased in 2.5 ft·lb increments up to 17.5 ft·lb, with data acquired at each level.

In addition to monitoring the bolt torque, bolt stress was monitored. The bolt stress was measured using a diametral strain measuring device (Mini C-Clamp see Figure 4), mounted to the solid diameter of the packing stud, and a portable strain meter (Measurements [illegible]500). As the packing nuts were torqued down, the packing bolts began to get longer and thinner. The Mini-C Clamp produced a calibrated output proportional to the bolt diametral reduction. This diametral deflection was converted to thrust using the following equation:

$$F_{Bolt} = \frac{E}{\nu} \frac{\pi D}{4} \frac{\Delta\ Mini-C\ \ OUTPUT}{CAL\ SENSITIVITY} \qquad (9)$$

where

E	=	modulus of elasticity - bolt
ν	=	Poisson's ratio - bolt
D	=	bolt diameter (solid section).

The same sequence was followed for the Walworth valve, zero packing load data were acquired for a baseline. The packing nuts were torqued to xx ft·lb and running load data were acquired. Data were acquired at four additional torque levels.

DISCUSSION

Baseline

The baseline data from both valve types were consistent; the average, no load running torque was 0.3 ft·lb. This gave relatively good assurance that the two actuators were in similar operating condition, and that the two valves were producing minimal resistance.

Packing Bolt Friction

Using Equation 3 to calculate the bolt preload requires that the user assume two coefficients of friction: one between the packing stud and nut the other between the nut and the collar. Friction studies conducted by Orthwein (1981) identified that the range of friction coefficients for dry metal to metal surfaces varies from 0.4 to 0.8. In cases where the surfaces were lubricated, the range dropped to 0.005 to 0.20. Most machine design and mechanics handbooks use 0.15 as the coefficient of friction for lubricated bolts. The actual value depends on surface finish, tolerance interference, and the type and amount of lubrication. As a rule, the better the surface finish and appearance, the lower the coefficient of friction.

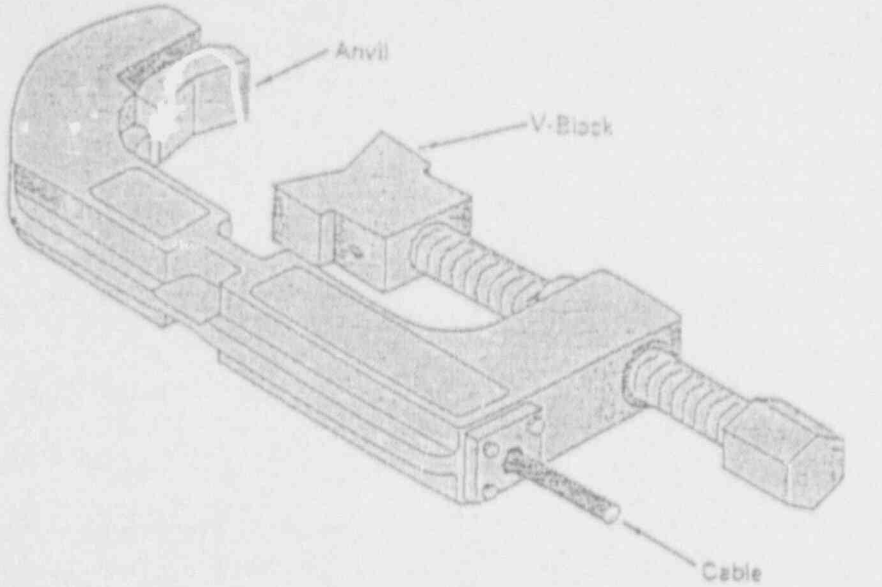

Figure 4. The mini C-clamp. This instrument is a strain gauge type sensor that measures the diametral growth of a stem or bolt (placed between the anvil and V-block) under loading.

Bolt torque and preload experiments conducted by Blake and Kurtz (1965), showed bolt preload, for 20 tests of 1/2-20 UNF bolts torqued to 800 in·lb (dry torque), ranged from 5.3 to 9.6 kips. The average was 7.7 kips, and the standard deviation was 1.107 kips. Thus indicating that the coefficient of friction varied from 0.09 to 0.19 for the two mating parts. Lubrication of the bolts did not affect the mean, but did cut the standard deviation to 0.681 kip.

For both valves, the collar was a semismooth casting, surface finish in the 250 µin. range. The packing bolts were smooth, better than 125 µin. finish. Due to the difference in surface finishes, the coefficient of friction between the nut and the collar is expected to be higher than the coefficient between the nut and bolt. Taking into account no lubrication on either surface, the surface finish, and the absence of rust, burrs and gouges, the coefficients could be estimated as $\mu_{PB} = 0.2$ and $\mu_c = 0.3$. These numbers are lower than expected for an unlubricated surface, but they reflect reductions in the coefficients because of the excellent surface conditions. According to Equation 3, the higher the coefficient of friction, the lower the bolt force for a given torque. The lower the bolt force, the lower the estimated packing friction. Underestimating the packing friction could lead to inadequate sizing. In order to be conservative, the friction coefficients should be lower, possibly $\mu_{PB} = 0.15$ and $\mu_c = 0.25$.

Both bolt preload and bolt torque were measured during the tests. Limited to one equation with two unknowns, there is a way to solve equation 3 for μ: assume both friction coefficients are equal ($\mu_c = \mu_{PB}$). The coefficient of friction can be solved for, using Equation 3, the measured bolt preload and bolt torque. Solving equation 3 for μ yields an average coefficient of friction of 0.20 for the Edwards valve. Table 1 lists measured bolt preloads for the Edwards globe valve vs. the calculated bolt preloads using equation 3 and assumed coefficients of friction $\mu_{PB} = 0.15$ and $\mu_c = 0.25$.

The measured preload vs. the theoretical preload produces an average error of -1.07%, which is due solely to the difference in the actual and estimated coefficients of friction. If the coefficients of friction were assumed to be 0.15 and 0.25 respectively, the packing estimates would have been relatively close. However, under the circumstances, with a dry bolt and collar, selecting 0.2 and 0.3 would not be out of the question. As an

Table 1. Edwards bolt preload.

Bolt torque	Measured bolt preload	Equation 3 preload	Error
5.0	372	366	-1.6
7.5	554	550	-0.7
10.0	741	733	-1.1
12.5	927	916	-1.1
15.0	1108	1099	-0.8
17.5	1297	1283	-1.1

Note: All preload values are in lb.

additional test, the nut, bolt, and collar were lubricated with a teflon base gel. After lubrication, the coefficient of friction dropped 28% to 0.14.

Packing Friction

The Walworth valve had no problem operating against any of the packing friction loads. Even at 22.5 ft·lb of torque, which is in excess of 3,000 psi gland pressure, the actuator running torque only averaged 8.7 ft·lb. Table 2 lists the average running torque data for the Walworth rising stem valve.

High packing gland loads had a more noticeable effect on the Edward's valves (Table 3). The manufacturers suggested bolt torque was 30 ft·lb. The maximum achievable bolt torque was only 17.5 ft·lb DRY. At 17.5 ft·lb, the torque switch tripped at the onset of stem motion. Using the coefficient of friction calculated above, $\mu_{PB} = 0.2$, gland pressure was only at 227 psi. In addition, the running torque for the Edwards valve was on the order of 5 times higher than the running torque for the Walworth valve at the same normal packing load F_{PN}.

Theoretically, if the coefficients of friction between the packing stud and nut, the nut and collar, and the stem and packing were to vary ±25% between rising rotating and rising stem; then the packing friction should vary on the order of ±45%; rising rotating versus rising stem. This variation would be seen directly in the running torque. Comparing the running torque for the rising rotating stem to the rising stem shows in excess of 500% (±250%) between the two. Even without solving for the coefficients of friction, the 500% variation in running torque indicates more than a variation in the coefficient of friction.

Packing manufacturers estimate the coefficient of friction between the stem and packing. Their values are based on in-house test programs and theory. Typically for a new stem $\mu_{SP} = 0.05$. The low coefficient is due to the extremely smooth, polished surface finish of the stem. As the stem is used, the surface finish changes. Cuts and gouges from packing extraction tools and wrenches scar

Table 2. Walworth test data.

Bolt torque	Running torque
2.0	0.60
3.5	1.05
11.5	4.45
17.5	6.65
22.5	8.70

Note: All torque values are in ft·lb.

Table 3. Edwards test data.

Bolt torque	Running torque	Ratio
5.0	2.85	0.57
7.5	5.65	0.75
10.0	8.30	0.83
12.5	12.05	0.96
15.0	16.65	1.11
17.5	20.10	1.15

Note: All torque values are in ft·lb.

the finish and increase resistance between the stem and packing. Using Equation 8 and the Edwards bolt preload data, the average static coefficient of friction between the stem and packing, μ_{SP}, is equal to 0.07, the dynamic value is slightly lower, 0.06.

Running Torque

Equations 6, 7, and 8 explain the theory behind the torque requirements for the rising rotating stem valve. Data acquired from the Edward's valves support this theory. For example, at 2.0 ft·lb of bolt torque, the rising stem running torque is 0.6 ft·lb. For the rising rotating stem, the same normal packing force is achieved at 5 ft·lb of bolt torque resulting in 2.85 ft·lb running torque. This indicates the rising rotating stem requires 476% more torque to move the stem against the same normal packing force. At another common force level; 3.5 ft·lb rising, 7.5 ft·lb rising rotating, the running torques are 1.05 and 5.65 ft·lb respectively—now indicating 526% more torque required to move the stem. The difference is too large to attribute to variations in the coefficients of friction.

Using the running torque data for the two valve types, use the rising stem running torque as the axial component, and the rising rotating torque as the total torque. The ratio of the axial to the total for 1.05 and 5.65 ft·lb is 0.19 or 19%. Therefore the rotational component comprises 81%. Figure 3 is a graphical representation of Equation 8. At 81% rotational torque, the curve shows a coefficient of friction of 0.22 between the stem and stem nut.

Using the measured bolt load, the measured coefficient of friction between the stem and packing, and Equations 4, 5, and 8, the static and running torque requirements can be calculated for the rising rotating stem. Table 4 lists the rising rotating theoretical and actual, and the rising stem theoretical torque values.

Table 4. Torque requirements.

Bolt torque	RRS theoretical	RRS actual	RS theoretical
5.0	4.4	4.2	0.76
7.5	9.66	7.8	1.14
10.0	12.89	11.6	1.52
12.5	16.09	18.10	1.90
15.0	19.31	22.35	2.28
17.5	22.53	24.00	2.67

Note: All values are in ft·lb.

Adding Equation 8 to the sizing equations brought the theoretical values much closer to the actual. The remaining difference is probably due to variations in friction coefficients.

CONCLUSIONS

For the rising rotating stem valve, the packing acts like an externally constricting brake which restricts radial motion in addition to axial motion. The present sizing equations need to be revised to account for the additional radial component. The following equation combines the traditional sizing equation with the radial torque calculation from equation 7.

$$T = (DP_{Load} + SR_{Load} + PF_{Load})\, FS + (PF_{Load}\, r_{Stem}) \tag{10}$$

This equation may overestimate the required torque, but in comparison to the axial equation, it is significantly closer to the actual.

For situations where diagnostic testing involves monitoring torque only, then calculating thrust using the torque values and a constant stem factor, the calculations will overestimate the axial component of packing friction. In order to accurately measure packing friction, axial loads must be measured below the stem nut.

Proper maintenance of rising rotating stem valves, particularly lubrication of the spline and yoke bushing, will help to eliminate additional radial and axial loads, and improve overall performance.

Installation of a flat washer between the packing nut and the collar will effectively reduce the coefficient of friction between the collar and the nut, thus producing a higher bolt preload for a given torque.

The test program revealed there is a running torque requirement for the rising rotating stem valve that far exceeds the rising stem valve. Testing of the rising rotating stem valves will continue with plans to instrument several valve stems and packing bolts with strain gauges.

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Session 3A

Emerging Technologies for Valves

Session Chair
Kenneth Barry
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Pump And Valve Research at the Oak Ridge National Laboratory

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ABSTRACT

Over the last several years, the Oak Ridge National Laboratory (ORNL) has carried out several aging assessments on pumps and valves under the NRC's Nuclear Plant Aging Research (NPAR) Program. In addition, ORNL has established an Advanced Diagnostic Engineering Research and Development Center (ADEC) in order to play a key role in the field of diagnostic engineering. Initial ADEC research projects have addressed problems that were identified, at least in part, by the NPAR and other NRC-sponsored programs.

Results from these research activities have included the identification and evaluation of existing monitoring methods for pumps and valves, and the development of several new diagnostic techniques. These developments include nonintrusive magnetic monitoring methods for valves and motor current signature analysis (MCSA) techniques for remote testing of electrically-powered equipment, including MOVs and motor-driven pumps. These developments have been successfully demonstrated in the laboratory, in local field installations, and in operating nuclear power plants. They provide useful diagnostic capabilities when used alone and when used in conjunction with other available monitoring equipment.

This paper summarizes the pump and valve related research that has been done at ORNL and describes in more detail several diagnostic techniques developed at ORNL that are now commercially available.

INTRODUCTION

Nuclear Plant Aging Research Program

The Oak Ridge National Laboratory (ORNL) has become familiar with pump- and valve-related issues largely as a result of work performed in support of the U. S. Nuclear Regulatory Commission (NRC) Nuclear Plant Aging Research (NPAR) Program. The NPAR Program was established in 1985 primarily as a means to resolve technical safety issues related to the aging of electrical and mechanical components, safety systems, support systems, and civil structures used in commercial nuclear power plants (NRC, 1991)

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The objectives for a comprehensive NPAR aging assessment of a component, system, or structure include:

The objectives for a comprehensive NPAR aging assessment of a component, system, or structure include:

1. Identify and characterize aging and wear effects

2. Identify failure modes and causes attributable to aging

3. Identify measurable performance parameters including functional indicators

4. Perform in-depth engineering studies and aging assessments based on in-situ measurements

5. Identify improved methods for inspection, surveillance, and monitoring, or for evaluating residual life

6. Perform post-service examinations and tests of naturally aged/degraded components

7. Make recommendations for utilizing research results in the regulatory process.

The results from an NPAR aging assessment may form the basis for implementation of improved inspection, surveillance, maintenance, and monitoring methods; modifying present codes and standards; developing guidelines and review procedures for plant life extension; and resolving generic safety issues.

Advanced Diagnostics Engineering R&D Center

ORNL has established the Advanced Diagnostic Engineering Research and Development Center (ADEC) in order to play a key role in the field of diagnostic engineering. ADEC has an organized multidisciplinary diagnostics research program that brings together experts in many fields in order to develop and apply new advanced diagnostic technologies having broad applications in the electric power, manufacturing, and defense industries. ADEC activities comprise the following four areas: (a) Diagnostic Sensor Research, (b) Signal Processing Research, (c) Data Analysis Research, and (d) System and Application Testing. Funding for this work has been provided initially by the ORNL Director's Discretionary Fund. Long-term funding is expected to be provided partially by industrial partners that are participating with ORNL in cooperative research programs.

A substantial portion of the ADEC research projects to date have focused on solving problems that were identified by the NPAR and other NRC-sponsored programs. In particular, several ADEC research tasks have concentrated on the development and demonstration of nonintrusive monitoring methods for valves and other equipment. Descriptions of a few of these developments are provided in this paper.

Description of Selected Pump and Valve Research Projects

As part of the NPAR program, aging assessments have been performed by ORNL on several components and systems involving pumps and valves, including

- Auxiliary feedwater pumps (AUXFPs)
- Auxiliary feedwater (AFW) system
- Power-operated relief valves (PORVs)
- Solenoid-operated valves (SOVs)
- Motor-operated valves (MOVs)
- Check valves (CVs).

ORNL NPAR research activities have also included a study of valve body erosion (Casada, 1991a), a review of industry responses to NRC Bulletin 88-04 "Potential Safety-Related Pump Loss" (Casada, 1991b), an evaluation of proposed inservice testing procedures for check valves (Moyers and Casada, 1992), a study of gate valve internal clearances and their effect on disk-seat interference during valve closure (Moyers and Eissenberg, 1990), and aging assessments of other power plant components such as heat exchangers (Moyers, 1992), air compressors and dryers (Moyers, 1990), BWR control rod drives,[b] turbine generator and controls,[c] instrumentation and protection components (Gehl and Kryter), and core internals.[d]

Key research results from the first six assessments listed above were extracted (in some cases verbatim) from their final reports and technical papers and are included in the following sections. Research activities on MOVs and CVs are discussed in more detail, including a description of nonintrusive monitoring methods developed by ORNL to monitor these components.

AUXILIARY FEEDWATER PUMPS (AUXFPS)

In 1986, ORNL published the results of an aging assessment of auxiliary feedwater pumps (AUXFPs) used in PWR nuclear power plants (Adams and Makay, 1986). AUXFPs are multistage (normally 5 to 9 stages) high-head centrifugal pumps, normally driven by motors or turbines. The function of these pumps is to deliver water from either a condensate storage tank or, as a backup, from the emergency service water system, to the steam generators. The pumps are automatically started in response to several emergency conditions, such as low steam generator level, a safety injection signal, and emergency bus undervoltage; however, many plants also use AUXFPs in support of normal shutdown and startup sequences, since the main feedwater system pump capacity greatly exceeds demand during these conditions. Much of the operation of AUXFPs is at low-flow (minimum-flow) conditions. AUXFPs are normally tested under minimum flow conditions, and much of the normal startup and shutdown support operations are at low flow.

One of the major findings from the aging assessment was that operation at low flow results in accelerated wear of the pumps as a result of the hydraulically unstable conditions (Casada and Adams, 1991). The wear can result in impeller or diffuser breakage, thrust bearing and/or balance device failure due to excessive loading, cavitation damage on suction stage impellers, increased seal leakage, seal injection piping failure, shaft or coupling breakage, and rotating element seizure.

In addition to recommending further investigations aimed at determining if present operating practices (especially low-flow operations) are a significant contributor to wear and aging of AUXFPs, the assessment suggests that tighter specifications of certain materials of construction and fabrication methods could potentially provide marked improvements in AUXFP durability and thus higher reliability. Furthermore, the application of state-of-the-art monitoring techniques should be studied in regard to its value in assessing wear and aging factors in AUXFPs.

AUXILIARY FEEDWATER (AFW) SYSTEM

In 1990, ORNL published the results of a study of the PWR Auxiliary Feedwater System (Casada, 1990). The study reviewed historical failure data for AFW system components and provided a detailed review of the AFW system design and operation practices at a plant owned by a cooperating utility. Failure data were compiled from three sources: Licensee Event Reports (LERs), the Institute for Nuclear Power Operations (INPO) Nuclear Reliability Data System (NPRDS), and Stoller Power, Inc.'s Nuclear Power Experience (NPE). Each record from the three databases was reviewed and combined to form a single ORNL database, thereby avoiding

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b. R. H. Greene, *Aging Assessment of BWR Control Rod Drive Systems*, NRC Report NUREG/CR-5699, to be published.

c. D. F. Cox, *Aging of Turbine Drives for Safety Related Pumps in Nuclear Power Plants*, NRC Report NUREG/CR-5857, to be published.

d. K. Luk, *Pressurized Water Reactor Internals Aging Degradation Study—A Phase 1 Report*, NRC Report, to be published.

redundant entries while establishing a more thorough set of failure records.

Components were classified into five groups for this study: pump drivers, valve operators, valves, pumps, and other. Information was extracted from the compiled ORNL database for each failure, including the method of detection, the subsystem affected, and the extent to which the system was degraded as a result of the failure. The data were then analyzed for trends and comparative evaluation.

The single largest source of AFW degradation, based upon the historical failure data review, is the turbine drive for AFW pumps. In addition, the failures of valve motor and air operators combined were found to have resulted in approximately the same level of degradation of the AFW system as the turbine drives alone. Pump and check valve failures were also significant contributors to system degradation.

In the review of the reference plant procedures, it was found that testing frequencies for system components varied substantially. For example, the trip & throttle (T&T) valve for the AFW turbine was found to be stroked over 40 times a year in conjunction with AFW system testing, while the AFW turbine's I&C/governor control system is checked only once every 18 months, at best. The testing frequencies are dictated by the plant's implementation of their Technical Specifications.

The review thus identified a need for enhanced testing requirements that would reduce excessive testing of certain components while at the same time ensuring that thorough performance verification is conducted periodically.

POWER-OPERATED RELIEF VALVES

In 1987, ORNL published the results of a review of nuclear power plant operating events from 1971 to 1986 involving failures of power-operated relief valves (PORVs) and associated block valves (BVs) (Murphy and Clether, 1987). This review was largely based on information obtained from LERs, NPRDS records, NRC Foreign Event Files, NPE reports, and interviews with four PORV manufacturers.

The review was conducted with the understanding that PORVs and their BVs were not designed as safety-related components but are, in fact, relied upon to mitigate certain design-basis accidents. The acceptability of relying on non-safety-grade PORVs to mitigate a design-basis accident is the subject of NRC Generic Issue 70: "PORV and Block Valve Reliability." Information resulting from the ORNL review should help support the resolution of GI-70.

Of the 230 events identified by the review, 101 involved PORV mechanical failure, 91 were attributable to PORV control failure, 6 events involved design or fabrication of the PORVs, and 32 events involved BV failures. The most common mechanical failure mechanism for PORVs was degradation of the seat/disc interface or other internal parts by high-pressure steam and/or water. Most BV failures involved torque switch failure or mis-adjustment.

Based on this review, it was concluded that the greatest safety benefit could be achieved by using PORV designs that are resistant to sticking open. New PORV designs were identified by the review that may provide higher reliability, but they had not been in service long enough to provide long-term operating experience. The review also concluded that reductions in PORV and BV failures might result by upgrading the PORVs and BVs to safety-grade status, where more rigorous testing, diagnostics, and maintenance are required.

SOLENOID-OPERATED VALVES

Solenoid-operated valves (SOVs) are found throughout nuclear power plant safety-related systems in relatively large numbers (between 1,000 and 3,000 per plant) and are often a subcomponent of larger, more complex systems. Their presence in systems important to safety requires an especially high degree of assurance that they are ready to perform their required function under all anticipated operating conditions, since failure of one of these small and relatively inexpensive devices could have serious consequences under certain circumstances. Thus, a comprehensive aging assessment of solenoid-operated valves was carried out by ORNL as part of the NPAR Program (Bacanskas et al.; Kryter, 1992). The assessment reviewed SOV failure modes and causes and identified measurable parameters thought to be linked to degradation that may ultimately result in the functional failure of the valve.

A major focus of the assessment was the identification and demonstration of monitoring methods that are useful in measuring SOV performance parameters that can be used to detect the presence of and trend the progress of SOV degradations. Intrusive techniques requiring the addition of magnetic or acoustic sensors or the application of special test signals were examined briefly, but major emphasis was placed on the examination of condition-indicating techniques that can be applied with minimal cost and impact on plant operation. SOV monitoring methods evaluated are summarized in Table 1.

The study recommended that the performance monitoring techniques developed during the assessment be field tested using a larger population of both new and naturally aged SOVs that would be likely to display one or more varieties of degraded performance. In addition, the study recommended that these techniques be refined and adapted as necessary to permit their use in a real plant environment.

MOTOR-OPERATED VALVES

NPAR Aging Assessment

Motor-operated valves (MOVs) can be found in almost all nuclear power plant fluid systems. Their failures have resulted in significant maintenance efforts and, on occasion, have led to the loss of operational readiness of safety-related systems. For these and other reasons, ORNL carried out a comprehensive aging assessment of MOVs (Greenstreet et al., 1985; Haynes, 1989), during 1985-1989 in support of the NPAR Program.

At the time the aging assessment was carried out, only one MOV monitoring system was commercially available. In addition to evaluating this system in depth (Crowley and Eissenberg, 1986), the diagnostic information available from many MOV measurable parameters was determined by ORNL using MOVs that were mounted on test stands (see Figure 1). Those parameters included

- Valve stem position
- Valve stem velocity
- Valve stem strain
- Torque and limit switch actuations
- Internal and external motor temperatures
- Vibration
- Torque switch angular position
- Motor current.

These evaluations led to the conclusion that the single most informative MOV measurable parameter was also the one which was most easily acquired, namely, the motor current. Motor current signature analysis (MCSA)[e] was found to provide detailed information related to the condition of the motor, motor operator, and valve across a wide range of levels from mean values and gross variations during a valve operation to information which characterizes transients and periodic occurrences.

Motor Current Signature Analysis

Basic Principles. MCSA is based on the recognition that a conventional electric motor (ac or dc) driving a mechanical load acts as an efficient and permanently available transducer, detecting both large and small time-dependent motor load variations generated anywhere within the mechanical load and converting them into electric-current noise signals that flow along the power cable (Kryter and Haynes, 1989a; 1989b).

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e. U.S. Patent Number 4,965,513 "Motor current Signature Analysis for Diagnosing Motor Operated Devices."

Table 1. An overview of SOV monitoring methods.

Method	Degradation(s) or malfunction(s) addressed	Attributes	Promise for in-plant use
Measurement of SOV temperature, via coil resistance or impedance	Electrical failure of coil and degradation of elastomers resulting from prolonged operation at excessively high temperatures	• Nonperturbative to plant operations • No new sensors or signal cables are required • No permanent instrumentation required; can be applied as needed from a remote location • Applicable to ac- and dc-powered SOVs	High; ready for immediate use
Indication of valve position and change of state upon application of power, via change in coil impedance	Mechanical binding, sluggishness, or failure to shift as a result of worn or improper parts or the presence of foreign materials inside the valve	• No need for add-on sensors or signal cables • Valve position readout from a remote location • Static method does not disturb SOV	High; some additional development work required
Indication of mechanical binding, by tracking changes in current and voltage at SOV pull-in and drop-out points	Mechanical binding and sluggish shifting caused by worn, swollen, or improper parts or the presence of foreign materials inside the valve	• Detects simultaneously degradation of magnetic or spring forces, and increase in frictional forces • No need for add-on sensors or cables or access to SOV • Applicable to ac- and dc-powered SOVs	Medium; further testing needed to ascertain cause of poor repeatability of test results
Indication of shorted coil turns or insulation breakdown, based on characteristics of electrical transient generated upon deenergizing a dc SOV	Electrical failure of solenoid coil, caused by high-voltage turn-off transients in combination with insulation weakened by prolonged operation at high temperatures	• Detects presence of defects within coil that cannot be revealed by other means	Low; useful for laboratory post-mortem tests
Indication of mechanical binding, by analyzing the time-varying characteristics of the inrush current accompanying application of electrical power to the SOV	Mechanical binding and sluggish shifting caused by worn, swollen, or improper parts or the presence of foreign materials inside the valve	• No need for add-on sensors, signal cables, or access to SOV • Information could be obtained as a result of everyday valve operation	Minimal; investigation of method abandoned early in the study
Indication of mechanical looseness within ac-powered valves, via electrical detection of humming or chattering of the plunger assembly (frequency decomposition of steady-state coil current)	Wear of internal valve parts, improper assembly, or replacement with incorrect parts	• No need for add-on sensors, signal cables, or access to SOV • Nonperturbative to plant operations	Minimal; investigation of method abandoned early in the study. Addition of miniature acoustic sensor to SOV might prove worthwhile

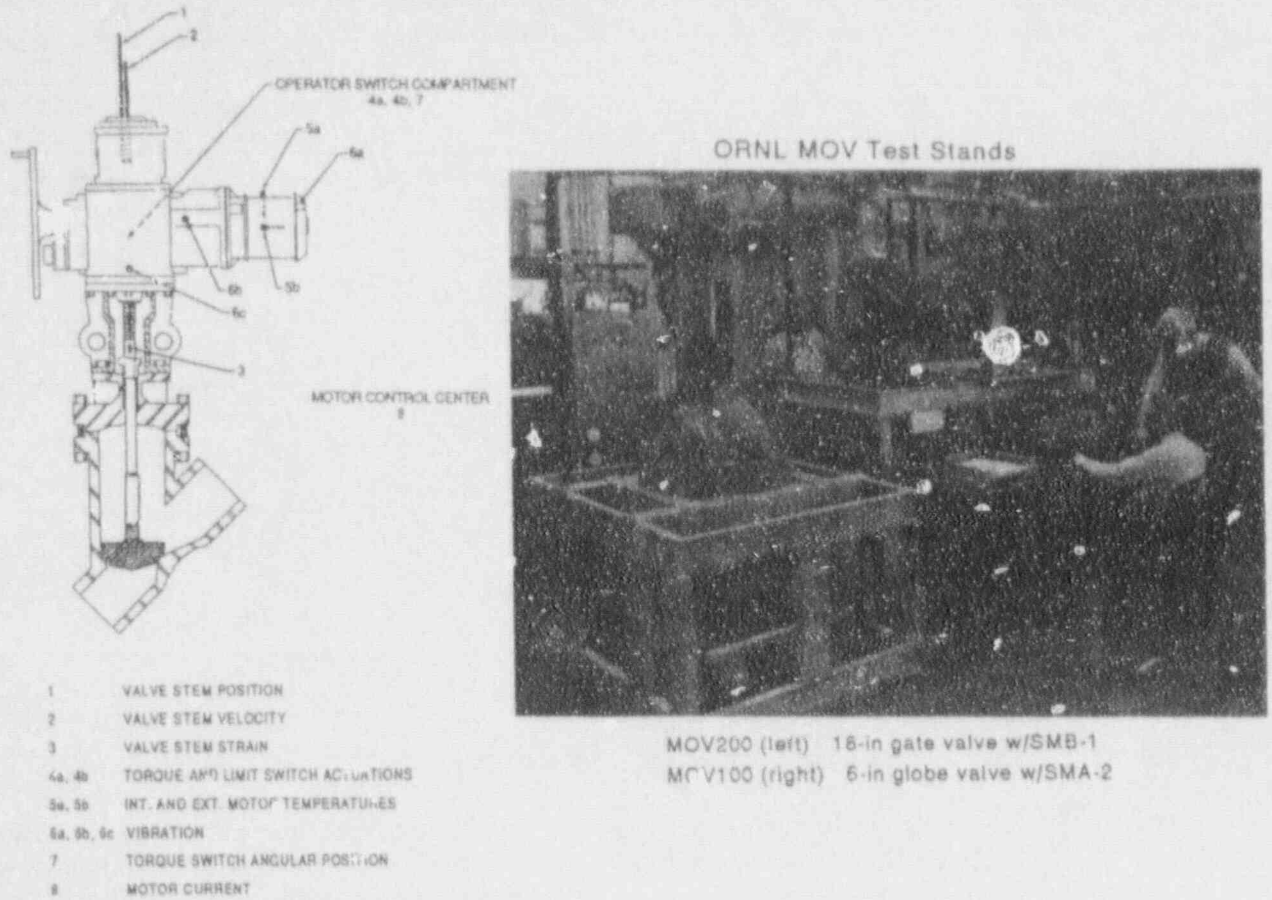

Figure 1. MOV measurable parameters evaluated by ORNL during the NPAR aging assessment.

As illustrated in Figure 2, MOV motor current signals can be obtained remotely (e.g., at a motor-control center, which may be several hundred feet from the equipment to be monitored). By using a clamp-on current probe to acquire raw motor current signals, no electrical connections need to be made or broken; thus, equipment operation is not interrupted and shock hazard is minimal.

Specially developed signal conditioning electronics were developed by ORNL to transform the raw current signal provided by the probe into two diagnostic signals: one optimized for time-domain analysis, and the other optimized for frequency-domain analysis.

The basic objective of the signal conditioning is maximizing dynamic range in the subsequent data analysis process. This is accomplished in part by demodulation of the raw current signal, followed by selective filtration and amplification. The resultant processed signals provide MOV condition indicators (within both time and frequency domains) that may be trended over time. MCSA has a number of inherent strengths, the most notable being that it

- Provides nonintrusive monitoring capability at a location remote from the equipment
- Provides diagnostic information comparable to conventional instrumentation but without the attendant disadvantages of added sensors and signal cables
- Offers high sensitivity to a variety of mechanical disorders
- Offers means for separating one form of disorder from another (selectivity)

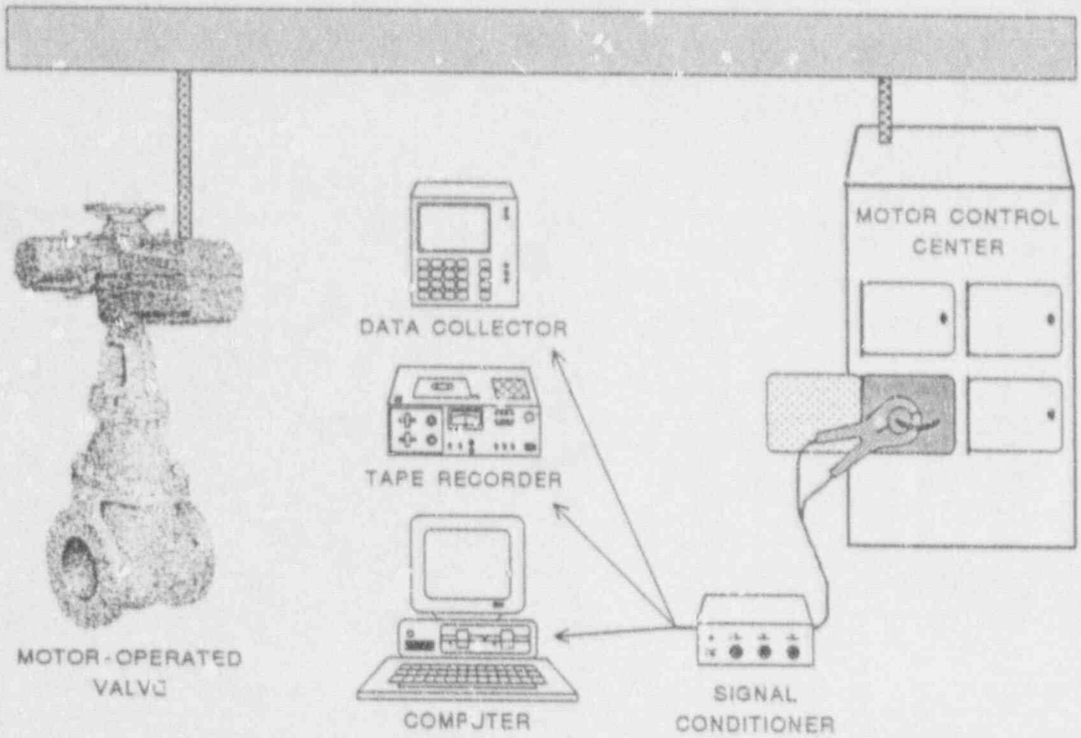

Figure 2. MOV motor current monitoring at a remote location.

- Can be performed rapidly and as frequently as desired by relatively unskilled personnel using portable, inexpensive equipment.
- Is applicable to high-powered and low-powered machines, driven by either ac or dc motors.

Time Waveform Analysis. Figure 3 presents a motor current time waveform for a close-to-open stroke of an 18-inch motor-operated gate valve operated at ambient conditions. This signature includes features that reflect normal gate valve operations such as the relatively large motor inrush current generated during motor starting and the motor current peak associated with valve unseating. Additional quantifiable features identified in this figure include the valve stroke time and the average running current.

Figure 4 presents the initial 3.5 seconds of the motor current time waveform shown in Figure 3, but is replotted with an expanded amplitude scale in order to better illustrate the signature details which are generally seen during the beginning of a close-to-open valve stroke. In addition to the large valve unseating peak, several other pre-unseating events are observed, including the motor operator hammerblow and the indication of initial valve stem movement. The increase in motor running current observed when the valve stem begins to move reflects the increase in motor running torque required to overcome the friction between the valve stem and the stem packing gland.

Both the amplitudes and the times of occurrences of these features provide useful condition indicators which may be trended over time. For example, the time differential between the hammerblow and initial stem movement generally reflects the clearance between the stem nut and stem thread surfaces. Likewise, the time between initial stem movement and gate unseating largely reflects the clearance between the gate and stem coupling surfaces. Thus, an increase in either (or both) of these time measurements provides an early indication of wear in these regions.

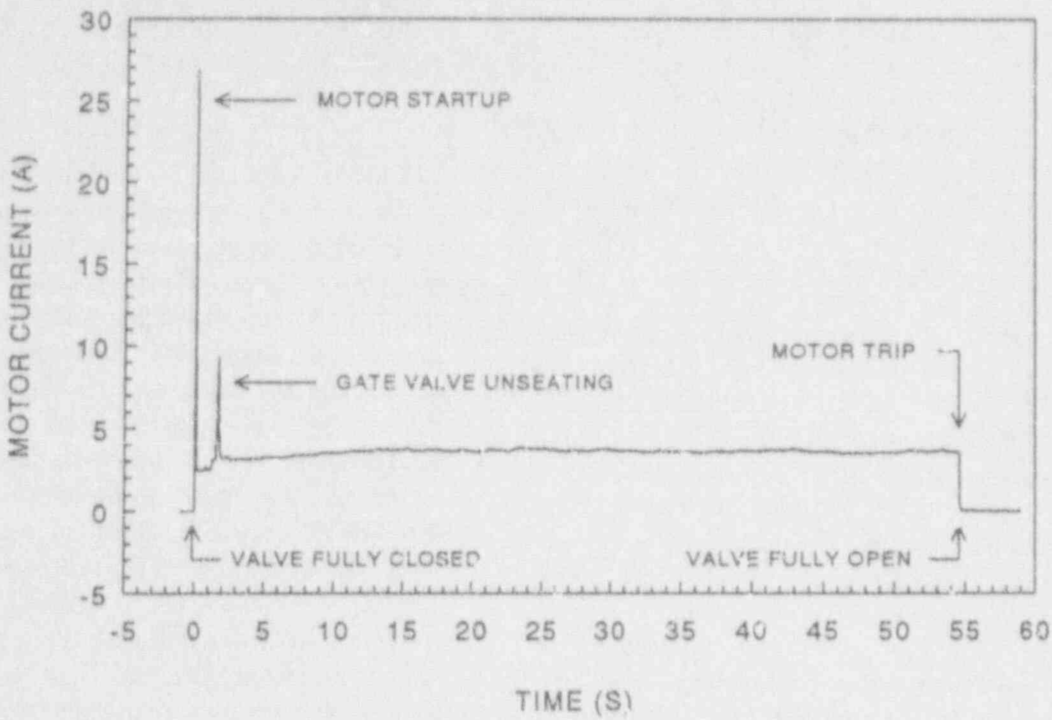

Figure 3. Typical motor current time waveform for an 18-inch MOV (close-to-open stroke).

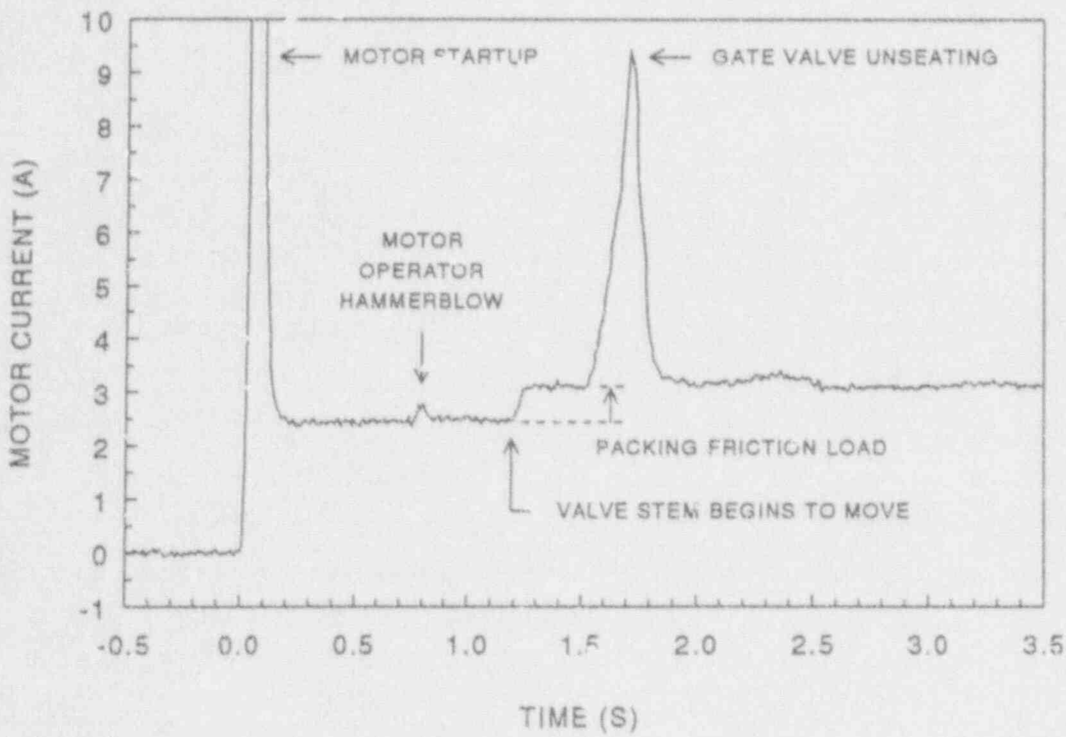

Figure 4. The initial 3.5 seconds of the close-to-open MOV motor current signature shown in Figure 3, plotted with an expanded vertical scale.

Only an abbreviated description of an MOV motor current time waveform for the close-to-open valve stroke has been presented in this paper. A full time waveform analysis (including both valve stroke directions) can provide many useful signature features which provide a means of determining and quantifying MOV performance and condition (Haynes, 1989).

Frequency Spectral Analysis. Early in the motor current signature assessments, it was recognized that, if properly pre-conditioned (e.g., demodulated, filtered, and amplified), motor current signals could be effectively examined for frequency content using standard spectrum analysis equipment. Figure 5 illustrates a motor current frequency spectrum for the same 18-inch MOV described earlier. Included in the frequency spectrum are two peaks which provide direct motor speed indication. Besides a frequency component at the true motor shaft speed, a peak identified as the slip frequency is also seen. The slip frequency is related to the motor shaft speed by the relation:

slip frequency = (synchronous speed-motor shaft speed) x (no. of motor poles).

Since the number of motor poles is typically 2 to 6, the slip frequency provides a sensitive means of detecting otherwise subtle changes in motor speed that could provide initial indication of running load changes within the valve or operator. A more detailed characterization of running loads is accomplished by an examination of the remaining spectral peaks. A major frequency component in this and other MOV motor current spectra is the worm gear tooth meshing frequency. The existence of this peak indicates that a significant motor load component is associated with the meshing of the worm and worm gear. In addition to the fundamental worm gear tooth meshing frequency, its second harmonic was also observed along with worm gear rotational sidebands, providing further MOV condition indication related to the worm gear drive.

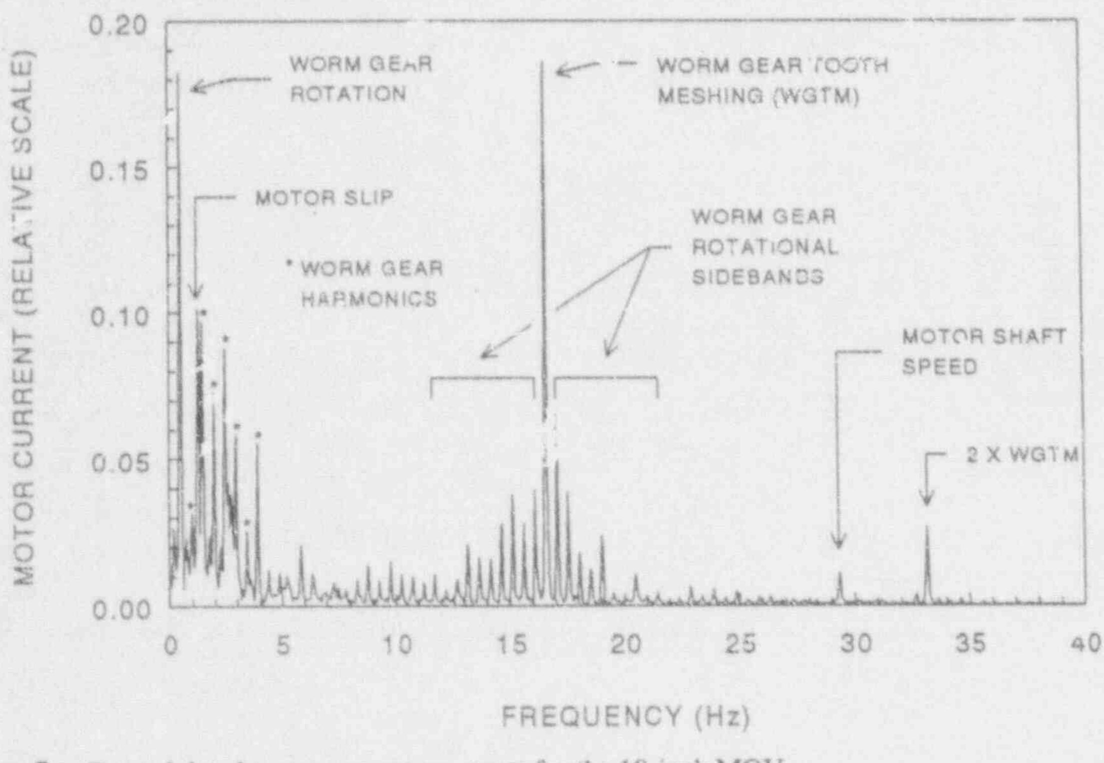

Figure 5. Demodulated motor current spectrum for the 18-inch MOV.

Additional MCSA Analysis Techniques for MOVs. As mentioned above, the MCSA method offers high sensitivity and selectivity for monitoring MOV operational characteristics. These benefits are further exemplified through the use of the selective waveform inspection method (SWIM). By selectively filtering the demodulated motor current noise signal, a unique time waveform is obtained which reflects the amplitude modulations of a specific periodic load component. Thus, if the worm gear tooth meshing frequency component is "singled out" using this technique, a tooth-by-tooth gear meshing profile can be produced, as shown by Figure 6. As shown in this figure, the signature exhibits a basic repetitive pattern consisting of a fixed number of peaks equal to the number of teeth on the worm gear (in this case, 34 teeth).

Reproducibility of this pattern throughout a valve stroke is generally observed; however, some slight modifications may be seen during a valve stroke as a result of the worm sliding axially (along the worm shaft in response to changing running loads) which results in slight variations in the worm and worm gear meshing surfaces.

Other MCSA techniques have been developed for MOVs, such as

- Estimating motor voltage (at the MOV) from motor current amplitude and noise frequency information acquired at the motor control center
- Determining motor operator gear ratios from motor current noise spectra
- Estimating valve stem travel from motor current time and frequency signatures.

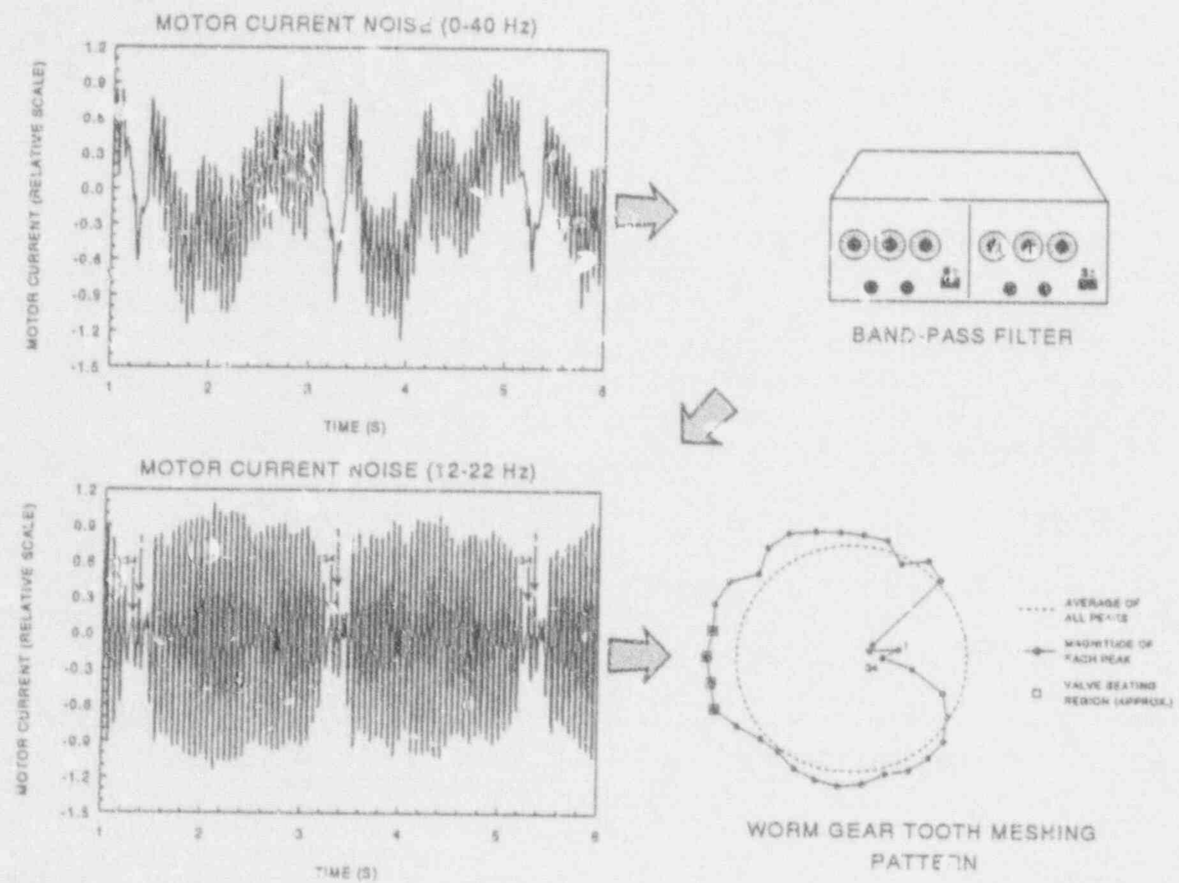

Figure 6. Application of Selective Waveform Inspection Method (SWIM) to the demodulated motor current signal from an 18-inch MOV whose worm gear has 34 teeth.

Further information on these and other techniques may be found in Haynes (1989).

Summary of MCSA Capabilities for MOVs. It has been demonstrated that numerous performance indicators are extractable from MOV motor current time- and frequency-domain signatures which may be quantified, documented, and trended over time. These include the following:

- Mechanical and frictional loads such as gear train friction, packing gland friction and gate/guide friction
- Initiation time, duration, and magnitude of transients including hammerblow, valve seating, unseating, backseating, and any unusual transient events
- Worm gear tooth meshing waveforms on a tooth-by-tooth basis using the selective waveform inspection method (SWIM)
- Periodic load variations within the MOV drive train such as worm gear tooth meshing, stem nut and worm gear rotation, motor shaft speed, motor slip, etc.

MOV Testing. Several tests were carried out by ORNL to investigate the capabilities of monitoring methods (especially MCSA) for detecting, differentiating, and tracking the progress of the following MOV abnormalities:

- Degraded valve stem lubrication
- Obstructions in valve seat area
- Disengagement of motor pinion gear
- Stem packing degradation or tightness changes
- Incorrect torque and limit switch settings
- Abnormal line voltage
- Worm gear tooth wear
- Stem nut thread wear
- Valve stem taper
- Degraded gearcase lubrication.

Haynes (1989) discusses all abnormalities listed above and describes their effect on MOV performance and on a variety of diagnostic measurements.

In-situ signature analysis tests were performed by ORNL on a total of 20 aged MOVs at a nuclear power plant. Five of these MOVs were later retested after they were refurbished. In all tests, MOV motor current signals were acquired at the motor control center with a clamp-on current transformer, demodulated and further processed by battery-powered signal conditioning electronics, and recorded on a portable tape recorder for off-site analyses. Selected results from those tests are described in Haynes (1989) and illustrate differences in motor current signatures from similar MOVs that reflect control switch setting variations and suggest differences in component wear. The influences of refurbishing and inactivity on MOV operations were clearly seen in motor current signatures as well.

ORNL also participated in the Gate Valve Flow Interruption Blowdown (GVFIB) tests (Haynes, 1990a) carried out in Huntsville, Alabama, during April-June, 1988. These tests were intended primarily to determine the behavior of motor-operated gate valves under the temperature, pressure, and flow conditions expected to be experienced by isolation valves in Boiling Water Reactors (BWRs) during a high-energy line break (blowdown) outside of containment. In addition, the tests provided an excellent opportunity to evaluate signature analysis methods for determining the operational readiness of the MOVs under those accident conditions. Results from those tests are described in Haynes (1989).

ADEC RESEARCH RESULTS

ADEC research activities have resulted in the development and/or demonstration of several MCSA-based monitoring methods that are applicable to MOVs and other motor-driven equipment. A short description of these developments are provided below.

On-Line MOV Monitoring System

In 1990, ORNL installed and demonstrated an on-line, automated motor current data acquisition system for monitoring the long-term effects of aging and service wear on the performance of eight critical MOVs located in a turbine steam extraction system in Unit 2 of the Philadelphia Electric Company's Eddystone Power Plant. Motor current data were acquired by the on-line system for over 1200 valve actuations and converted into a database that is compatible with a commercially available data analysis and plotting package. Motor current signature analyses were then carried out in both time and frequency domains.

The use of MCSA at the Eddystone Plant on MOVs and other equipment is described in a paper that was presented at the EPRI-sponsored 4th Incipient Failure Detection Conference (Haynes et al., 1990) in 1990.

Detection of Broken Rotor Bars in an Induction Motor Using MCSA

Electric current signals were acquired by ORNL on a specially designed test rig comprised of two motors: one in "good" condition and one in "bad" (defective) condition, each connected to an electric generator (providing a means of loading each motor) by a belt of identical length. The bad motor was identical to the good motor with one exception: four rotor bars were purposely cut (broken) in order to simulate one type of naturally occurring motor defect.

It is recognized that the use of motor current analysis for detecting broken rotor bars and other motor degradation has been well documented by others and consists primarily of examining the amplitude of motor slip sidebands observed around the power line frequency in the "raw" motor current noise spectrum. Broken rotor bars are known to increase the induced currents in the stator windings, resulting in increases in slip sideband amplitudes (Reason, 1987).

ORNL tests demonstrated that the use of demodulation provides enhanced sensitivity (increased dynamic range) for acquiring motor current diagnostic information that would be undetectable in the raw motor current signal. Figure 7 illustrates demodulated motor current spectra for both motors, operating under no load and while fully loaded by the electric generator.

The application of MCSA to detect rotor degradation in MOV motors was investigated in a series of tests carried out by a nuclear utility and a commercial supplier of MCSA technology (Kueck et al., 1992). The purpose of these tests was to evaluate the effectiveness of MCSA techniques in detecting open-circuited rotor bars in valve actuator motors and to determine the maximum number of rotor bars that can be broken before the motor torque output drops below its rated torque value. These tests confirmed that MCSA can be used not only to detect motor rotor bar faults, but can identify seriously degraded motors in need of repair or replacement.

Seating Detector and Switch for Motor-Operated Valves

ORNL has carried out proof-of-principle tests of a device for de-energizing the MOV's electric motor during valve seating based on a special algorithm that utilizes the measured instantaneous motor current. The device de-energizes the MOV motor only when the slope, duration, and amplitude of a motor current rise exceed predetermined criteria.

Preliminary tests of the device have been carried out using ORNL MOV test stands and using recorded motor current data from the gate valve blowdown test described earlier. These tests indicate that the device avoids unwarranted midstroke tripping due to spurious motor current fluctuations of small amplitude or short duration, such as may occur due to roughness of valve guide surfaces, for example

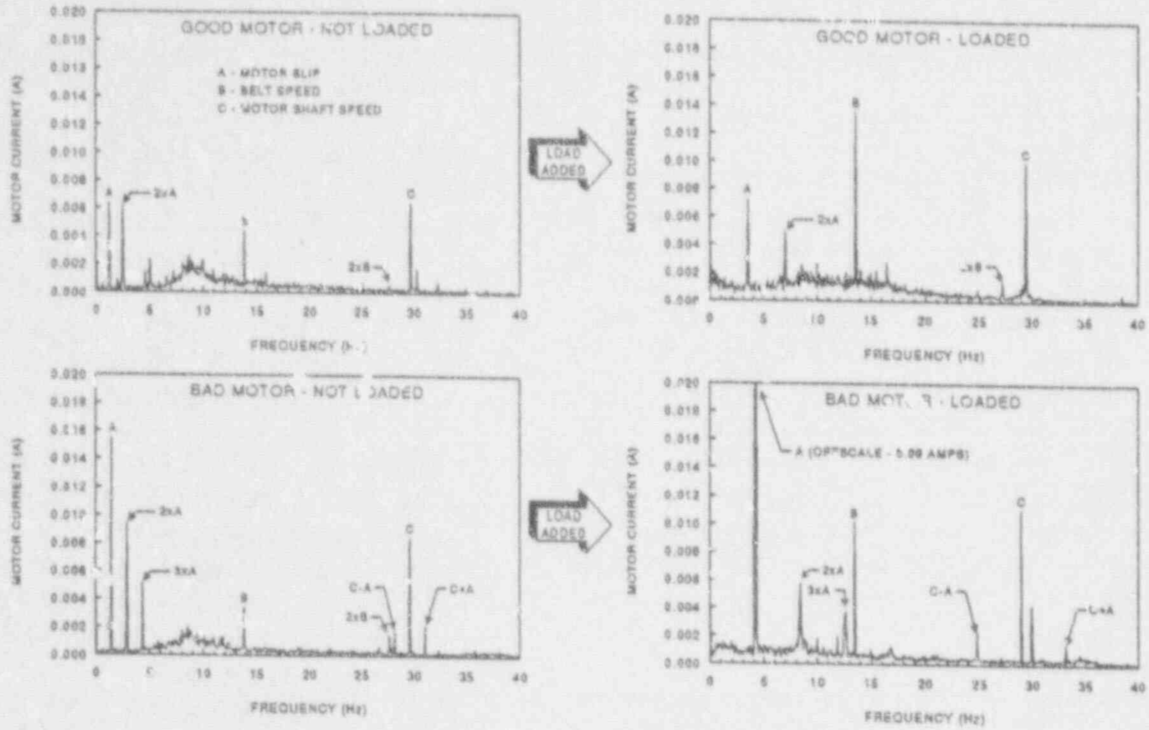

Figure 7. Demodulated motor current spectra for a good motor and one with 4 broken rotor bars, under no load and fully loaded by an electric generator.

This device may be useful in a variety of applications but has particular promise as a seating detector and switch for MOVs or other motor-driven devices where it is desired to trip the device motor according to a special algorithm which detects a large, sustained, and abrupt rise in the measured instantaneous motor current relative to recent running current history. By changing the circuit time constants, the characteristics of the device can be adapted to a wide variety of motor-driven devices and operating needs.

Application of MCSA to other equipment

ORNL is also conducting research aimed at further developing and improving MCSA technology and demonstrating MCSA on other equipment. For example, ORNL has carried out experiments that have demonstrated the applicability of MCSA technology for a wide variety of motor-driven equipment, including

- Nuclear power plant motor-operated valves
- Gaseous diffusion plant axial-flow compressors
- Air conditioning systems (residential heat pump, room a/c units)
- Fossil plant equipment (Oak Ridge facilities, Eddystone power plant): motor-operated valves, induced draft fans, overfire fan, coal pulverizers, boiler feed pumps, condensate feed pump, air compressors
- ORNL centrifugal chillers
- Fans (various sizes)
- Water pumps (various sizes, including Navy firepump)
- Laboratory vacuum pump
- Miscellaneous home appliances.

The interested reader is encouraged to contact the author for more information on MCSA as applied to these and other components.

CHECK VALVES

NPAR Aging Assessment

Check valves are used extensively in nuclear plant safety systems and balance-of-plant (BOP) systems. The failures of these valves have resulted in significant maintenance efforts and, on occasion, have resulted in water hammer, overpressurization of low-pressure systems, and damage to flow system components (NRC, 1986). Many check valve failures have been attributed to severe degradation of internal parts (e.g., hinge pins, hinge arms, discs, and disc nut pins) resulting from instability (flutter) of these parts under normal plant operating conditions. Check valve instability may be a result of misapplication (e.g., using oversized valves) and exacerbated by low flow conditions and/or upstream flow disturbances (MPR and Kalsi, 1988).

For these and other reasons, ORNL carried out a comprehensive aging assessment of check valves (Greenstreet et al., 1985; Haynes, 1991a) during 1985-1991. Research efforts were focused on identifying and evaluating potentially useful signature analysis methods for determining the operational readiness of check valves. As part of the NPAR aging assessment, ORNL carried out an evaluation of several check valve monitoring methods; in particular, those based on measurements of acoustic emission, ultrasonics, and magnetic flux. The evaluations were focused on determining the capability of each method to provide diagnostic information useful in determining check valve aging and service wear effects (degradation), check valve failures, and undesirable operating modes.

Two monitoring methods developed by others, acoustic emission and ultrasonic inspection, are briefly described in the following section. More detailed descriptions are then provided of check valve monitoring methods developed by ORNL that are based on the use of internal- and external-magnetic fields. Finally a comparison is made between monitoring technologies in order to emphasize the strengths and weaknesses of each method.

The descriptions of check valve monitoring methods in this paper refer to their use on swing check valves; however, all monitoring methods described herein may be applied to other check valve types (e.g., piston-lift, ball, stop-check, and duo-check designs).

Acoustic Emission Monitoring. Acoustic emissions (pressure waves) can be generated in a variety of ways. Of particular interest are those generated either when solids contact each other or when liquids or gases flow through pipes and fittings. Acoustic emissions are detected by sensors, such as piezoelectric-type accelerometers or microphones, which respond to pressure waves over a wide range of frequencies. Signal-conditioning electronics can be used to amplify selected acoustic signals while attenuating others, e.g., unwanted environmental background noise. Analyses of acoustic emission signals obtained from check valves can be used to monitor check valve internal impacts as well as fluid flow and/or leakage through the valve.

Acoustic emission monitoring has been used for many years to detect check valve disc movement (Suslick, 1986). Acoustic emission tests of check valves have also been performed under controlled flow loop conditions and with the introduction of various implanted defects that simulated severe aging and service wear (Suslick, 1988). By using accelerometers attached to the body of a check valve, the tapping of a valve disc against its backstop may be easily detected and distinguished from background flow noise. In addition, by using two (or more) valve-mounted acoustic sensors, the source of the tapping can be determined based on a comparison of the "time of arrival" of the acoustic signals acquired from the two sensors.

Acoustic emission techniques have also long been used to detect fluid leaking through a valve (McElroy, 1987). Through the acquisition of two sets of acoustic emission readings, one while the valve is unpressurized and one with a pressure differential across the (closed) disc, the noise associated with a leaking valve may then be determined on the basis of the difference in readings.

The primary strength of the acoustic emission technique is that it provides a means of detecting leakage, flow noise, and internal impacts that occur when the check valve is stroked open, stroked closed or when the valve is operating under flow conditions that result in impacts between internal parts. One should recognize, however, that the detection of flow noise without the presence of impact noise is no guarantee that the check valve is fully open since the valve disc may be oscillating without tapping in midstroke, may have fallen off, or may be stuck in a position that prevents it from impacting the valve body at any location. A minor limitation of this method is the necessity of using multiple sensors to determine the location of a tapping event.

Ultrasonic Inspection. Ultrasonic inspection involves the introduction of high-frequency sound waves into a part being examined and an analysis of the characteristics of the reflected beam. Typically, one (pulse-echo) or two (pitch-catch) ultrasonic transducers are used which provide both transmission and receiving (sensing) capabilities. The ultrasonic signal is injected from outside the valve by the transmitting transducer and passes through the valve body, where it is reflected by an internal part (e.g., disc or hinge arm) back toward the receiving transducer. (Note: When one transducer is used in a pulse-echo mode, it provides both transmitting and receiving capabilities.) By knowing the time required for transmission of the ultrasonic signal from the transmitting transducer and back to the receiving transducer, the transducer location(s), and other valve geometries, the instantaneous position of a check valve internal part may be determined. In general, signal processing circuitry must be used to filter out undesirable ultrasonic signal reflections present in the raw received signal so that the resultant processed signal provides a more easily interpreted valve disc position signature.

In addition to determining disc position, ultrasonic signatures can be used to detect missing and stuck discs, loose hinge arm/disc connections, and worn hinge pins. For example, if the disc is missing, no signal will be returned (reflected) from the disc region; however, if the hinge arm remains on the valve, its position can be verified by ultrasonic techniques. Furthermore, disc stud wear can be detected by monitoring the motion of both the disc and hinge arm using two pulse-echo transducers, one sensing movement of the disc and the other sensing hinge arm movement. Increased clearance between the disc stud and the hinge arm can result in increased movement of the disc, relative to the hinge arm.

In general, an ultrasonic time waveform can best be used to determine instantaneous position and movement of check valve internal parts. Detection of disc tapping (e.g., on the backstop or seat) is less obvious, since tapping is observed as a momentary cessation of movement and does not generate an abrupt and predominate transient signature feature, as is the case with acoustic emission. Furthermore, this technique can not differentiate between a fully closed valve that is leaking from one that is not leaking. Ultrasonic inspection, using a single transducer installed at a fixed position also may not provide valve disc position information throughout the entire valve stroke due to the limited viewing angle of the transducer. Furthermore, a low density fluid, such as steam and air, results in severe attenuation of transmitted and reflected signals and, ultimately, poor transducer response.

Magnetic Flux Monitoring. Research carried out by ORNL as part of the NPAR check valve aging assessment led to the identification of a new check valve diagnostic technique, magnetic flux signature analysis (MFSA) (Haynes and Eissenberg, 1989). MFSA is based on correlating the magnetic field strength variations monitored on the outside of a check valve with the position of a permanent magnet placed on a moving part inside the check valve (Figure 8). MFSA thus provides the ability to monitor disc position through an entire valve stroke using one externally mounted sensor.

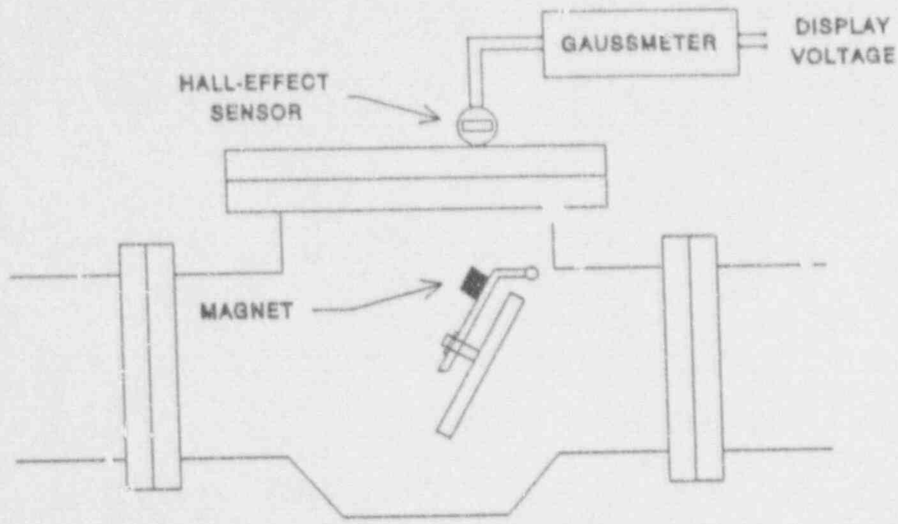

Figure 8. A simplified depiction of the magnetic flux signature analysis (MFSA) technique.

In proof-of-principle tests, a Hall-effect gaussmeter probe was used outside the check valve to detect the magnitude of the magnetic field produced by a small permanent magnet attached to the hinge arm. The Hall-effect probe detected both constant and varying magnetic fields and thus continuously monitored both the instantaneous position and the motion of the check valve disc. This was demonstrated by tests carried out by ORNL on a 2-in. swing check valve that was installed in a water flow loop. The acquired magnetic flux signatures (see Figure 9) showed that at a low flow rate (insufficient to open the valve fully), the disc fluttered considerably in mid-stroke, whereas at a higher flow rate, the same valve achieved a fully open and stable condition.

Experiments carried out at ORNL have shown that MFSA techniques can be used to detect small diameter (worn) hinge pins (Haynes, 1990b). Increased clearances between the hinge pin and hinge arm was observed as increased rocking motion of the hinge arm and disc assembly during flow testing and was detected using two externally-mounted magnetic field sensors (see Haynes, 1990b for details).

MFSA requires the installation of a permanent magnet inside the valve and thus, the method is not totally nonintrusive. As a result, the successful application of this method may be hindered by the following limitations:

1. Impacts between the valve disc and valve body may result in a demagnetization of the attached magnet.

2. The internal magnet may attract and hold small metallic particles that may build up and affect the magnetic field dispersion pattern and possibly the operation of the check valve.

3. If the magnet (and/or magnet assembly) detaches from the check valve and reattaches somewhere else, it may present a significant problem.

4. Certain magnetic flux signature features may be difficult to observe under field conditions due to the presence of relatively strong ambient magnetic fields (e.g., from nearby motors).

ADEC Research Results

External Magnetic Monitoring. As part of ADEC, two novel nonintrusive methods have been developed for monitoring the position and motion of equipment internal parts. These methods are based on the use of externally-applied

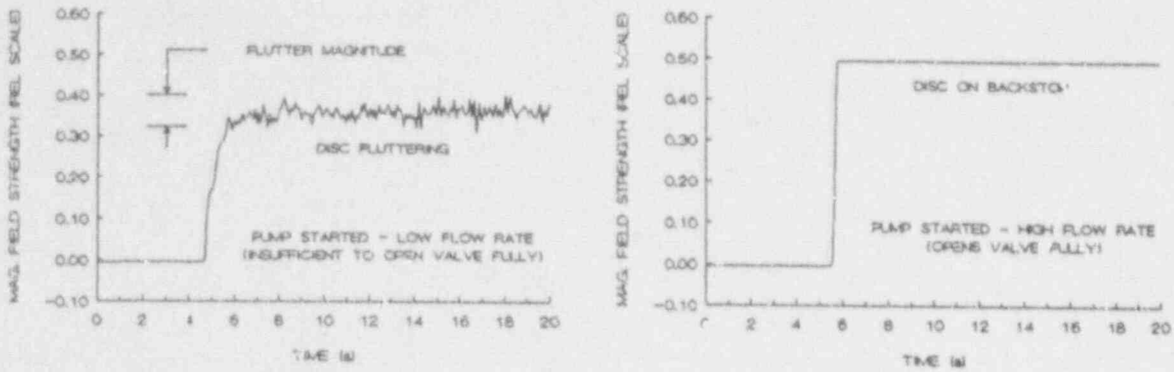

Figure 9. Use of MFSA to detect disc instability (flutter).

magnetic fields from permanent magnets and from electromagnet coils driven by either alternating or direct current (Haynes, 1991b). External magnetic monitoring techniques were initially disclosed and demonstrated at the NRC-sponsored 18th Water Reactor Safety Information Meeting in October, 1990. Laboratory and field tests have demonstrated that the position and motion of a swing check valve disc assembly can be monitored in real time and on a continuous basis by using these methods as described below.

External AC Magnet Method. A commonly tested embodiment of the external AC magnet method (see Figure 10) utilizes two coils of wire which are either wrapped around or attached (e.g., bolted or strapped) to different locations on the body of the check valve.

One coil (transmitter coil) is connected to a source of electric current at a fixed, selected frequency and thus produces a magnetic field whose amplitude and direction varies according to the source frequency. A second coil (receiver coil) senses the magnetic field which has been transmitted through the check valve and warped by both the body and internals of the valve. The local magnetic field present at the receiver coil induces a current in that coil which can then be displayed and measured. Specially developed signal conditioning electronics are used to increase the sensitivity of the receiver coil to the selected magnetic field frequency, and to provide a more easily interpreted signal. Since the position of the valve body is fixed relative to the two coils, the alteration of the transmitted magnetic field due to the valve body alone is also fixed and can be offset electronically. Changes in the position of the check valve internals produce variations in the receiver coil signal which may be monitored, quantified, and trended over time.

The external AC magnet method has been used to monitor disc position and motion of several swing check valves having different sizes, body materials, and fluid media (air and water). For example, Figure 11 illustrates an application of this method on a 3-inch stainless steel swing check valve installed in a water flow loop at Oak Ridge. Using one transmitter coil and one receiver coil, the position and motion of the valve internals were monitored across the full range of disc travel and under both stable (full open and full closed) and unstable (mid-stroke fluttering) operations. The AC system has also been demonstrated on a swing check valve that was installed in an active flow system at an operating nuclear power plant.

External DC Magnet Method. Another nonintrusive method for monitoring the position and motion of check valve internals makes use of one or more externally-applied dc magnetic fields supplied either by permanent magnets or by coils carrying dc current. The dc magnetic fields are transmitted through the check valve and detected externally at one or more locations by a magnetic field sensor such as a gaussmeter that employs a Hall-effect probe.

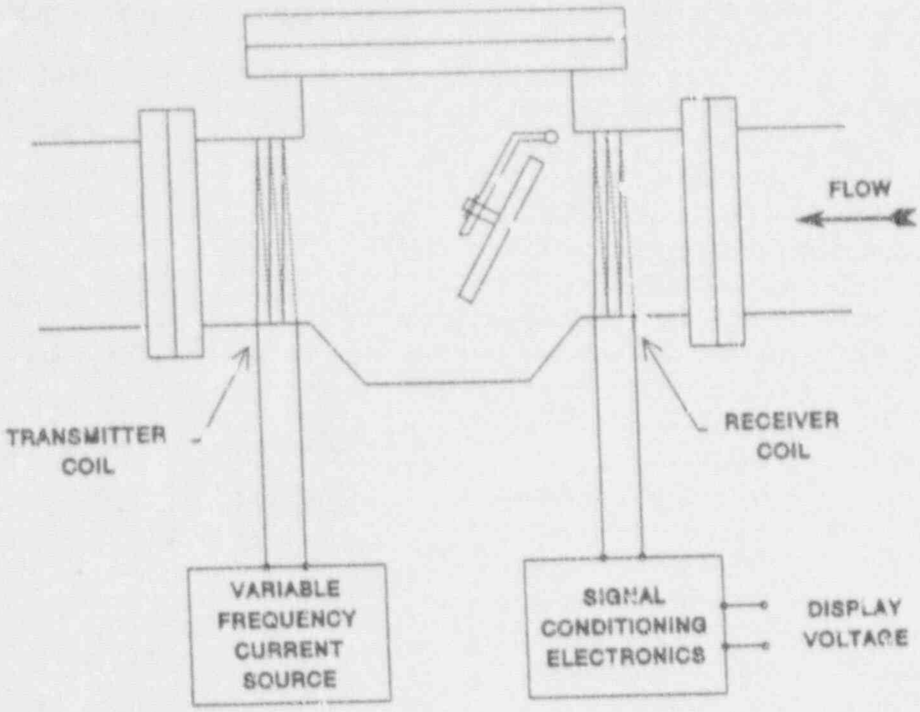

Figure 10. Simplified depiction of the external ac magnet check valve monitoring method developed by ORNL.

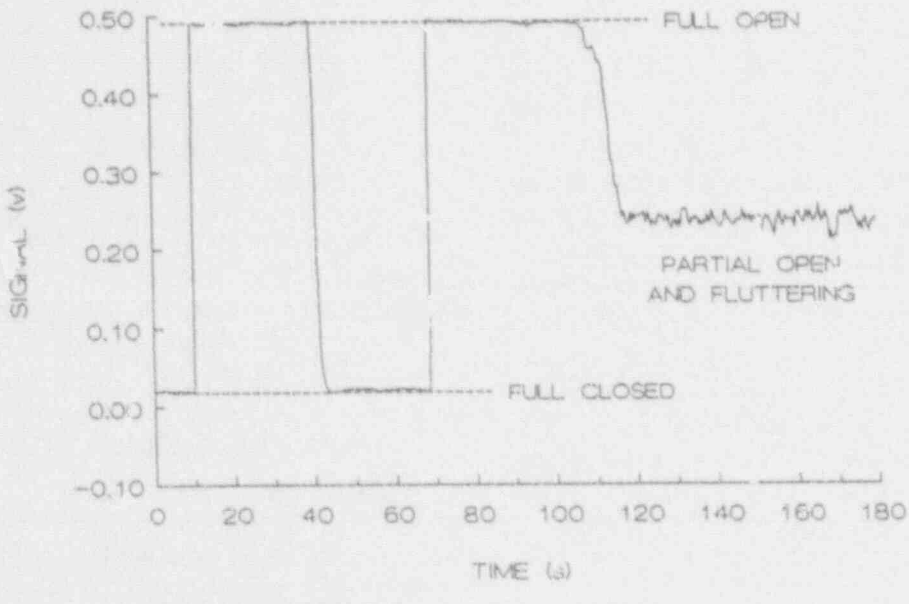

Figure 11. Application of the external ac magnet method to monitor disc position and motion of a 3-in. check valve installed in a flow loop at Oak Ridge.

This method has some similarity to MFSA in that it uses a magnetic field (e.g., Hall-effect) sensor installed externally to detect the position and motion of the check valve internal parts; however, the use of external dc magnetic fields overcomes the major deficiency of MFSA, that is, the necessity to open the check valve and install a permanent magnet on an internal part. In addition, the external magnet method provides greater flexibility since neither magnet size, strength, location, etc., are limited as in MFSA (e.g., what can fit in the valve and not adversely affect the performance of the valve). A commonly tested embodiment of the external dc magnet method (see Figure 12) utilizes two permanent magnets, one installed near the valve seat and one installed near the valve backstop. A single Hall-effect probe is installed near the hinge pin area and detects changes in local magnetic field strength resulting from changes in the position of the valve's internals.

This method has been used to monitor many check valves having different sizes, body materials, and fluid media (air and water). For example, Figure 13 illustrates an application to a 10-inch carbon steel valve at Oak Ridge. The dc system has also been demonstrated on two swing check valves that were installed in two active flow systems at an operating nuclear power plant.

Improvements In System Performance. Initial investigations of external magnetic monitoring techniques for check valves identified many parameters that, when optimally selected, resulted in significant improvements in system performance (e.g., sensitivity, signal-to-noise ratio, and reliability). Additional research was carried out to understand the effect of these parameters and how to select them so that the system's performance could be maximized. These parameters include the following for ac and dc systems

ac System:

- coil type (circular, semicircular, pancake, or solenoid valve type)
- coil size (large, small, long, short)
- number of coil turns, wire gauge
- core type (air, solid, laminated)
- installation location (on the valve, on the adjacent piping)
- installation method (permanent, portable)
- excitation signal (one or more discrete frequencies, random noise)
- excitation signal amplitude (input power)
- signal conditioning (amplifiers, filters, demodulation).

dc System:

- magnet strength and installation area size (local flux density)
- magnet locations (near the seat, backstop, hinge pin)
- magnet polarity (north field, south field)
- magnetic sensor location (near the seat, backstop, hinge pin)
- magnetic flux control techniques (focusing, direction)
- signal conditioning electronics (amplifiers, filters).

Techniques were then developed that provided major improvements in the ability of both ac and dc systems to monitor valve position. Descriptions of several of these techniques are described in Haynes (1991b).

Comparison Between Monitoring Methods

The check valve monitoring methods described above can provide diagnostic information useful in determining the condition of the valve (e.g., integrity of internal parts), and its operating state (stable or unstable). These methods utilize different transducers and principles of operation; hence, they provide different capabilities

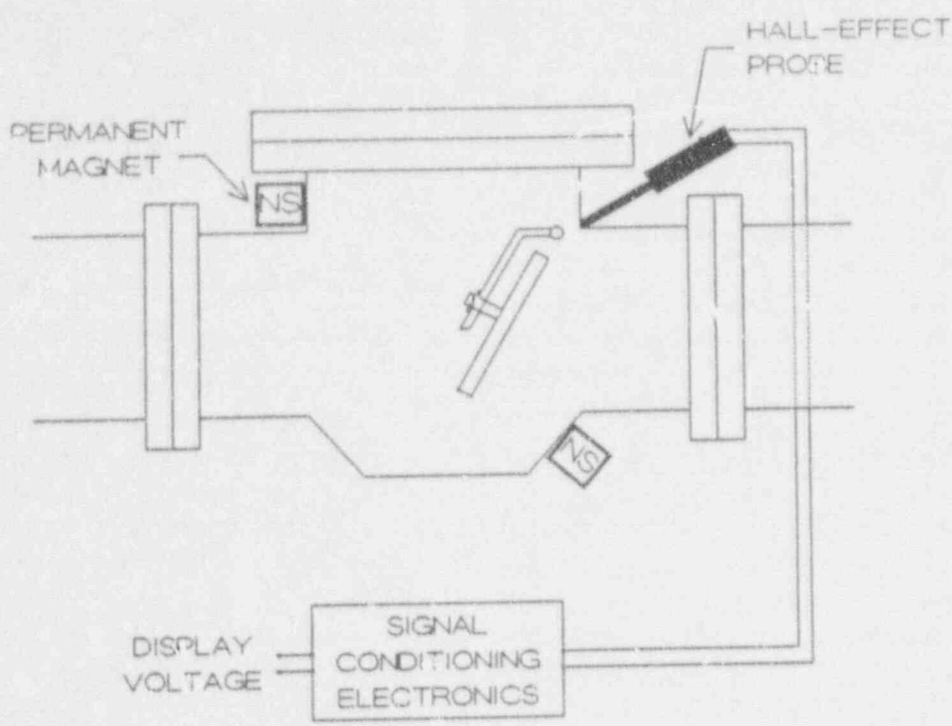

Figure 12. Simplified depiction of the external dc magnet check valve monitoring method developed by ORNL (note magnet polarities).

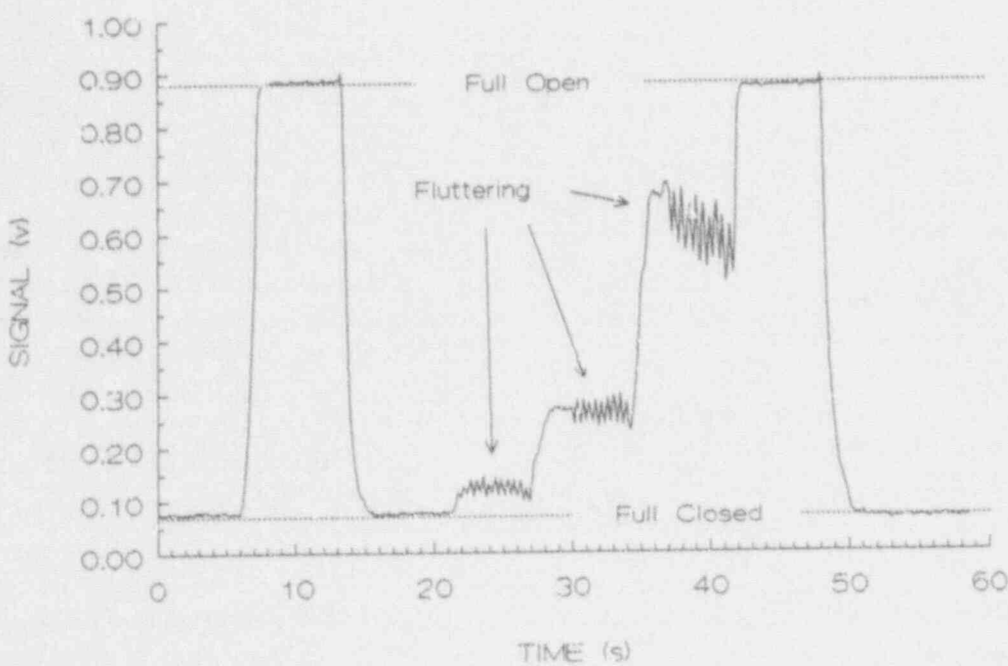

Figure 13. Application of the external dc magnet method to monitor disc position and motion of a 10-in. check valve at Oak Ridge.

suffer from different limitations. These methods are summarized in Table 2 along with selected diagnostic capabilities and limitations.

Combination of Methods

None of the methods described above can, by themselves, monitor the position and motion of valve internals and valve leakage; however, the combination of acoustic emission with either of the other methods yields a monitoring system that succeeds in providing the means to determine vital check valve operational information.

Both acoustic/ultrasonic and acoustic/magnetic combinations have been tested. For example, the combination of acoustic emission and MFSA was tested by ORNL on a check valve whose disc was moved manually to simulate disc fluttering at different disc positions. As shown in Figure 14, the acoustic signature did not provide direct indication of disc position when the valve's disc was stationary in the fully-open and fully-closed positions, nor did it detect the slowly moving disc or disc flutter in mid-stroke.

In all three tapping modes (seat tapping, backstop tapping, and hinge arm rocking), the acoustic signature detected the tapping but not its location. The magnetic signature did not unambiguously detect the tapping, but, in conjunction with the acoustic signature, identified its location. The combination of MFSA and acoustic emission monitoring was first demonstrated by ORNL at an EPRI check valve workshop held in January, 1989.

Check valve monitoring systems are now commercially available that are based on the combined use of acoustic emission with internal magnetics (MFSA), external magnetics, and ultrasonic inspection.

DISSEMINATION OF RESEARCH RESULTS

Interactions with Outside Organizations

A good indicator of the significance, relevance, and visibility of the pump and valve research at ORNL is the continued interest shown by electric utilities, private companies, government agencies, universities, and other national laboratories. For example, ORNL has corresponded with over 180 organizations on MOV and check valve related topics (Haynes, 1992a; 1992b).

ORNL Reports, Papers, and Articles

ORNL has made and continues to make a major effort to disseminate technical information to others via presentations at meetings and conferences and through numerous technical reports, papers, and articles. For example, valve monitoring technologies have been demonstrated to others on more than 50 occasions at Oak Ridge and other sites. In addition to those documents referenced in earlier sections of this paper, technical articles have appeared in several magazines including *Power Engineering* (Moyers and Eissenberg, 1990; Kryter and Haynes, 1989a), *Sound and Vibration* (Kryter and Haynes, 1989b), *Mechanical Engineering* (Haynes, 1991b), and other magazines and technical journals.

On March 24-27, 1992, the NRC held an Aging Research Information Conference in Rockville, Maryland. In addition to presenting eight papers at the conference, ORNL researchers hosted a suite which included videos of the effects of low-flow operation of centrifugal pumps and on the maintenance of BWR control rod drive mechanisms. Demonstrations of monitoring methods for MOVs, CVs, and SOVs were also given.

Commercialization of ORNL-Developed Technologies

Five private companies (Table 3) are presently marketing valve monitoring technologies originally developed at ORNL under non-exclusive licensing agreements with Martin Marietta Energy Systems Inc.

Table 2. Selected diagnostic capabilities and limitations of check valve monitoring methods.[a]

Method	Detects valve internal leakage	Detects internal impacts	Detects fluttering (no impacts)	Nonintrusive	Sensitivity to ambient conditions[b]	Monitors disc position throughout the full range of disc travel	Works with all fluids
Acoustic emission	Yes	Yes	No	Yes	Sensitive to externally generated noise/ vibration	No	Yes
Ultrasonic inspection	No	Yes (indirectly)	Yes	Yes	Unknown	Not in all cases—because of limited viewing angle of transducer	No - low density fluid (e.g., air or steam) results in severe attenuation of signals
Internal Permanent Magnet Techniques	No	Yes (indirectly)	Yes	No—requires initial installation of permanent magnet inside the valve	Sensitive to nearby external magnetic fields (e.g., from motors)	Yes	Yes
External AC and DC Magnetic Techniques	No	Yes (indirectly)	Yes	Yes	DC Method- Sensitive to nearby external magnetic fields (e.g., from motors)	Yes	Yes

a. Radiology and pressure noise analysis methods are not summarized in this table. This table does not reflect other attributes such as cost, east of use, etc.

b. Temperature and radiation effects are unknown.

CONCLUSIONS

ORNL has conducted research on pumps and valves under the NRC's NPAR Program. In addition, ORNL has continued to address issues that were identified, at least in part, by the NPAR and other NRC-sponsored programs by carrying out R&D activities at the ADEC. Results from this research have included the identification and evaluation of existing monitoring methods for pumps and valves, and the development of several new diagnostic techniques.

This paper has summarized several pump and valve related research tasks and has provided additional detailed information on the MOV and check valve projects. This paper also identified ORNL research activities in other areas that have

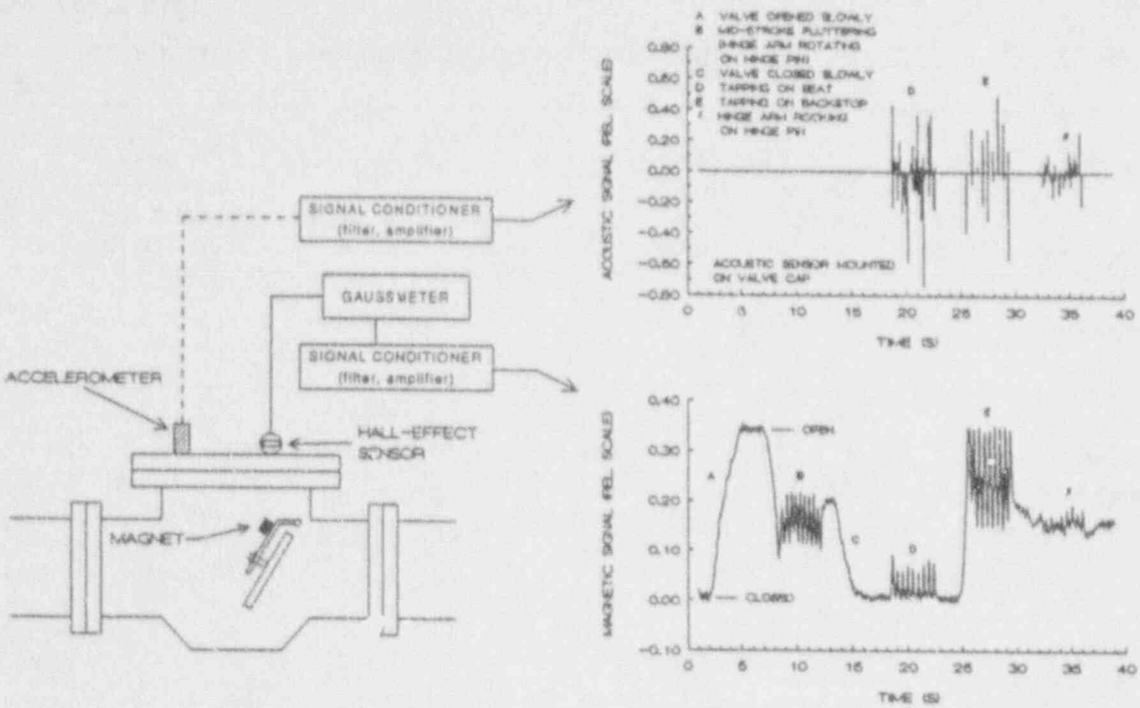

Figure 14. Magnetic flux and acoustic signatures for a check valve under several simulated operational conditions.

Table 3. Companies marketing monitoring technologies.

Technology	Company	Phone Number
MCSA	Predictive Maintenance Inspection, Inc.	(205) 464-9679
MCSA	Performance Technologies, Inc.	(804) 237-2583
MCSA	Spectrum Technologies USA, Inc.	(518) 382-0056
External Magnetics	Valvision, Inc.	(518) 854-3986
External Magnetics	ITI Movats, Inc.	(404) 424-6343

either been completed or are still underway in continued support of the NPAR program. The interested reader is encouraged to contact the author for more information in these areas and for more details on ORNL Diagnostic Center activities.

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Assessment of Valve Actuator Motor Rotor Degradation by Fourier Analysis of Current Waveform

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ABSTRACT

This paper presents a test report of a motor diagnostic system that uses Fourier Analysis of the motor current waveform to detect broken rotor bars in the motor or defects in the driven equipment. The test was conducted on a valve actuator motor driving a valve actuator that was in turn driving a dynamometer to measure the actuator torque output. The motor was gradually degraded by open circuiting rotor bars. The test confirmed the efficacy of the waveform analysis method for assessing motor rotor degradation and also provided data regarding the change in waveform characteristic as motor rotors are gradually degraded to failure.

INTRODUCTION

As a result of the number of past problems with valve actuator motors and the high priority that must be placed on ensuring that they are operable, it was decided to perform a test of a new motor diagnostic method to determine its value in assessing motor rotor degradtion from a remote location, that is, the motor control center. Past problems with valve actuator motors containing magnesium rotors have included failures from galvanic corrosion between the magnesium alloy end ring conductor and the iron core, and melting of the magnesium from extended locked rotor current in excess of the manufacturer's allowable locked rotor time.

Merely testing the motor by performance of stroke time tests at zero differential pressure has been determined not to be effective in determining motor degradation. We have observed motors where one end ring had completely melted off the motor rotor, yet at zero differential pressure, the motor was still stroking the valve within the acceptable stroke time tolerance.

A method developed to examine magnesium motor rotors for degradation is the use of a bore scope to physically examine the inside of the motor and investigate the end ring for such symptoms as cracks, discoloration, or magnesium hydroxide powder. These symptoms are indicative of galvanic corrosion. Significant problems are associated with the bore scope inspection, including radiation dose from examination of valves in a high radiation field and the clearance (tagout) on the motor, which is necessary before the motor can be examined.

An alternate method of motor diagnosis has recently been developed, in part by Oak Ridge National Laboratories, based on the principal that an electric motor acts as a transducer. When broken rotor bars are present, harmonic fluxes are produced in the air gap, which induce harmonic components in the motor current waveform (Thompson, 1988). The motor current waveform can be readily converted from a time domain to a frequency domain using Fast Fourier analysis and the amplitude of each of the component frequencies can be evaluated to determine problems both in the motor and in the driven equipment.

A major advantage of this method of diagnosis is that it can be done remotely. The current signal can be recorded using a clamp-on current transformer at the motor control center. The recorded current waveform can then be processed with a personal computer in a more convenient location with the computer performing the digital signal processing.

The specific goal of this test was to determine the number of broken rotor bars before the motor locked rotor torque output would fall below the rated torque value, and to determine whether the Fourier Analysis of the current waveform would provide positive indication that the motor had reached a failed state at the time when the locked rotor torque had dropped to its rated value. The resultant frequency spectra would provide guidance for future testing to assess the degradation of similar valve actuator motors. The test was successful in that after four rotor bars were broken, the motor torque output did drop to a value near its rating and the Fourier Analysis was showing clear indications of a motor failure. Progressive failure indications were noted as successive bars were broken.

DISCUSSION OF VALVE ACTUATOR MOTOR PROBLEM

Larger sizes of valve actuators often use motors that are required to produce a relatively high torque yet maintain a minimal inertia. For this application a magnesium alloy is often used for the rotor bar and end ring conductor material because of its higher resistivity and light weight. This motor has unfortunately had a troubling history of failures (Kueck, 1988). The standard electrode potential between magnesium and iron is 1.9 volts. This is ample voltage to drive a galvanic corrosion mechanism. The magnesium end ring has been known to corrode to the extent where it cracks and falls off during normal operating ambient conditions. In addition, the alloy will begin to melt at temperatures of 860°F and become completely molten at temperatures above 1100°F. These temperatures may be reached in 15 to 20 seconds of locked rotor current.

Because these motors are used in safety-related applications, it was necessary to develop some method for inspecting the rotors to determine whether corrosion or melting had occurred. Because there is a T drain or pipe plug in the end bell of the motor, access to the motor is usually available using a bore scope. The bore scope is used to check the end ring for evidence of a white powder on the surface, globular deposits of magnesium appearing anywhere on the surface, or cracks or darkening on the surface of the end ring. If any of these indications are noted, the motor is declared inoperable. Not only is this a difficult inspection from the stand point of motor clearance and radiation dose, but there is the potential for declaring motors inoperable based on the misinterpretation of a visual condition when the rotor is actually intact. We have noted that when the end rings do crack, they start to crack in one location, and adjacent rotor bars are successively open circuited.

The high levels of torque produced by the valve actuator motor are achieved by increasing the flux level in the motor, by using special rotor designs, or by a combination of these methods (Rebbapragadda, 1990). Open circuiting a rotor bar will result in an increase in the rotor resistance, a decrease in locked rotor current, and a decrease in torque output. Interestingly enough, because these motors are designed to operate with the flux density driven well into the saturated condition, when we first began to open-circuit the rotor bars, the locked rotor current decreased, but there was no noticeable decrease in locked rotor torque. This result is discussed further under test results below. Locked rotor torque is the key parameter for these motors and is usually the parameter used to size the motor for its application (Rebbapragadda, 1990). If the motor torque output is reduced by corrosion or melting of the rotor bars, the motor may have inadequate torque to perform its function at design basis conditions.

DEVELOPMENT OF TEST STAND

With the increased emphasis placed on motor operated valve (MOV) operability, the need arose to assess the condition of MOVs more accurately. Several diagnostic systems and methods have been developed that determine the condition of MOVs by measuring the output torque of the actuator or the output thrust supplied to the valve by the actuator. These systems/methods test both the actuator and the valve as a unit, requiring the valve and actuator to be installed before testing begins. This presented a concern, in that it was sometimes difficult to determine if the problems found were with the actuator or the valve. Also, any required output torque adjustments to the actuator had to be performed in the installed location. In an effort to more accurately assess the condition of actuators and to allow initial output torque adjustments to be made in a more favorable environment, it was determined that a torque test stand for MOVs was needed.

With this need in mind, a test stand was designed that would allow Limitorque SMB actuators to be run in the shop with a variable load applied while the output torque was being measured. The test stand was patterned on the concept used for initial set up of actuators at the factory. The test stand allows actuators to be mounted to the stand and have ac or dc power applied to the motor along with control power for the torque switch and limit switches. The actuator is then coupled to an instrumented shaft through a specially machined stem nut. The shaft is instrumented with two strain gages calibrated for 0-600 and 0-6000 ft · lb. The strain gages are coupled through a slip-ring arrangement to a transducer and a digital indicator designed to read out directly in ft · lb of torque. The end of the shaft opposite from the actuator mounting is coupled to an air operated disc brake assembly that allows the test technician to control the amount of resistance applied to the shaft as the actuator rotates it. The amount of torque produced by the actuator as it rotates the shaft is read directly on the digital display. This information can be used initially to set up the torque switch to provide the proper actuator output torque for a particular application before the actuator is installed on a valve.

DISCUSSION OF MOTOR CURRENT WAVEFORM ANALYSIS

Motor current waveform analysis has been studied as a means for determining the effects of aging and service wear for a broad range of electric motor driven rotating machinery (Haynes, 199?; Kliman and Stein, 1990; Reason, 1987; Thompson et al., 1983). Motor current signatures, obtained in both time and frequency domains, provide equipment condition indicators that provide early indication of degradation. When broken rotor bars exist, harmonic fluxes are produced in the air gap that result in harmonic components in the motor line current. The use of motor current analysis for detecting broken rotor bars has traditionally consisted of an examination of a Fast Fourier Transform (FFT) of the motor line current waveform (Kliman and Stein, 1990). The resulting frequency analysis, (Figure 1), shows a central peak at the line current frequency (60 Hz), and sidebands around this frequency. The sidebands are displaced from the line frequency by multiples of slip frequency (Reason, 1987). The amplitude of these sidebands is indicative of the number of broken rotor bars in the rotor.

Recently, an enhancement to this method of analysis has been developed (Haynes, 199?). This enhancement is the use of a demodulating, filtering, and amplifying circuit to remove the line current frequency and its harmonics from the motor current signal waveform.

The signal conditioning circuitry consists of two active components (Figure 2). These components perform the following operations on the input signal, assumed to be the output of a 1000:1 current transformer for an ac signal, and a 100 A = 1 V Hall effect probe for a dc signal:

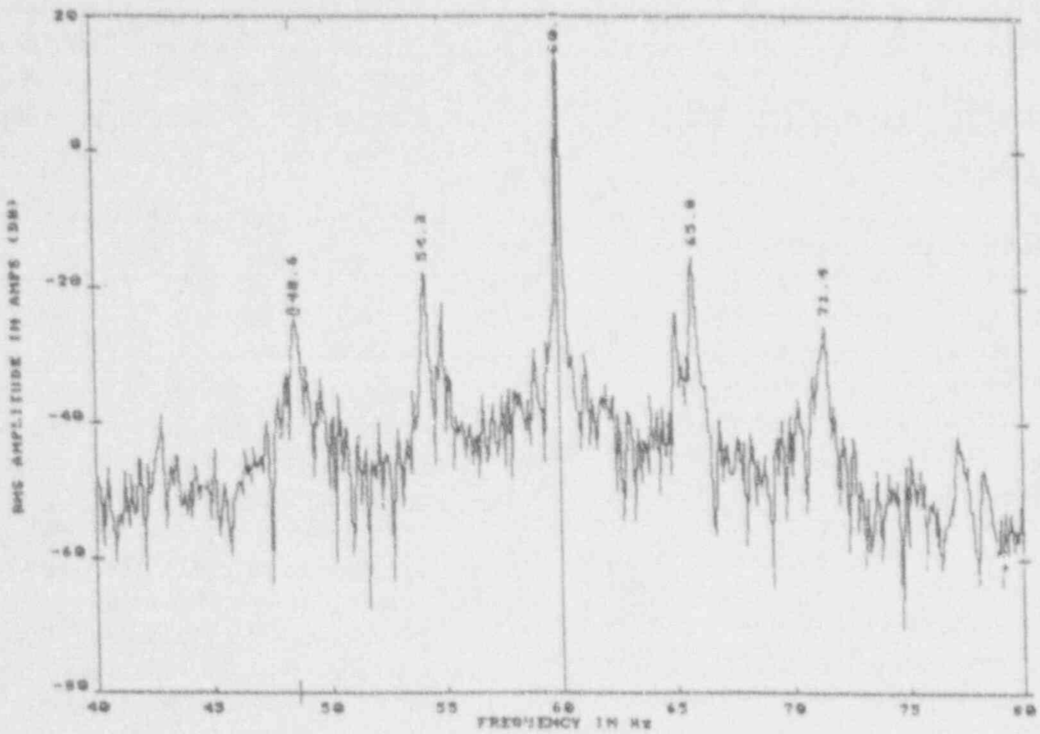

Figure 1. Run 5 FFT of line signal.

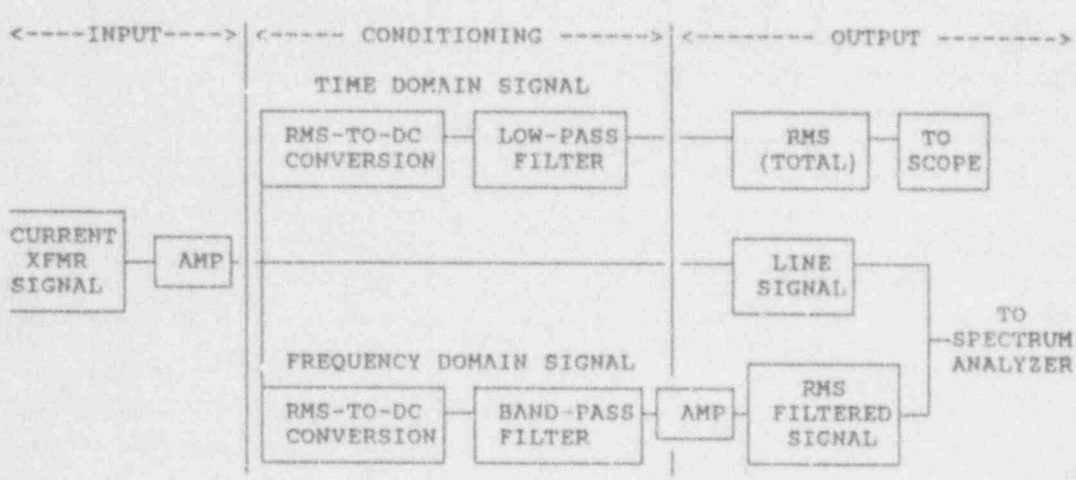

Figure 2. Signal analysis functional diagram.

1. Provide correct terminating (shunt) impedance to convert the input current signal to a more easily manipulated voltage signal

2. Provide amplification necessary to accommodate the desired range of input currents

3. Provide a low impedance output signal that is capable of driving the next stage; of being observed on an oscilloscope; or as input to a frequency (spectrum) analyzer

4. Compute the true root-mean-square (rms) value of the time-dependent, 60-Hz current transformer signal, using an averaging time appropriate to the application, as a means of demodulating the information embodied in the 60-Hz carrier signal

5. Band Pass filter the short term averaged rms signal to eliminate, or strongly attenuate, the ripple components at frequencies higher than the range of interest, e.g., 120 Hz, produced by harmonics of the 60-Hz input waveform.

The first three functions are performed by operational amplifiers that have been selected for their low input supply current requirements. The latter two functions are performed by true rms-to-dc converters with their associated network of circuit components. Figure 2 provides a functional diagram of the signal conditioning process.

The significance of the conditioning on the time waveform and ac-only output signals can be seen in the plots of the actual test results. The line signal output is not conditioned, but is amplified in accordance with the settings for the conditioned portion of the input in order to be compatible with the conditioned output.

The total output waveform appears as a *normal* scope trace, but with the low pass filter set between 4 and 7 Hz. The *noise* is extracted in order to view the true dc characteristics. Initiation time, duration, and magnitude of transients, including operator hammerblow, valve seating, unseating, and back-seating, and unusual transient events can be observed. See Figure 3 for an example of a total output waveform.

The rms filtered signal output provides the proper conditioning to effectively analyze the motor current signals for frequency content with standard spectrum analysis equipment. See Figure 4 for a frequency spectrum of a *conditioned* signal.

Included in the spectrum of the *conditioned* signal is a predominant peak identified as the slip frequency of the motor. The slip frequency is defined as the difference between the motor synchronous speed and the actual running speed multiplied by the actual number of poles of the motor. The significance of this ability to measure the true slip frequency will become readily apparent as the results of the subject tests are presented. The actual motor running speed is, of course, also seen in the spectrum, as well as the worm gear tooth mesh (WGTM) frequency and sidebands spaced from the WGTM at multiples of the worm-gear rotation speed.

CRITERIA FOR ASSESSING ROTOR BAR INTEGRITY

The following four criteria are used to evaluate the extent of rotor bar degradation. Both the FFT of the "LINE" signal and the FFT of the "CONDITIONED" signal are used.

1. The slip frequency is identified. It can be established by subtracting the indicated motor speed from the synchronous speed and multiplying by the number of poles. See Figure 4. The running frequency of 28.9 subtracted from 30, times the number of poles (4), yields the slip frequency of 4.4. The slip frequency is examined to determine its amplitude, relative to the total energy content of the spectrum, and for the presence of harmonics. Under normal conditions, with a rotor that is not damaged, the slip peak will generally be less than 25% of the total for the spectrum. As the rotor degrades, this contribution increases to over 50% for a rotor with several cracked or

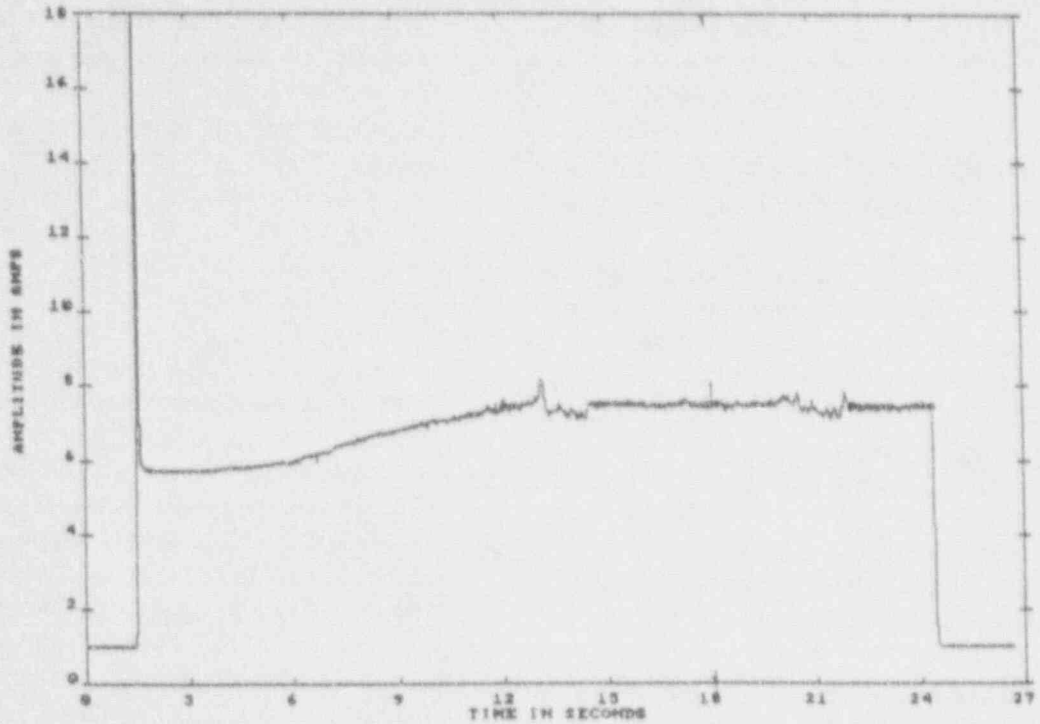

Figure 3. Run 1 RMS waveform.

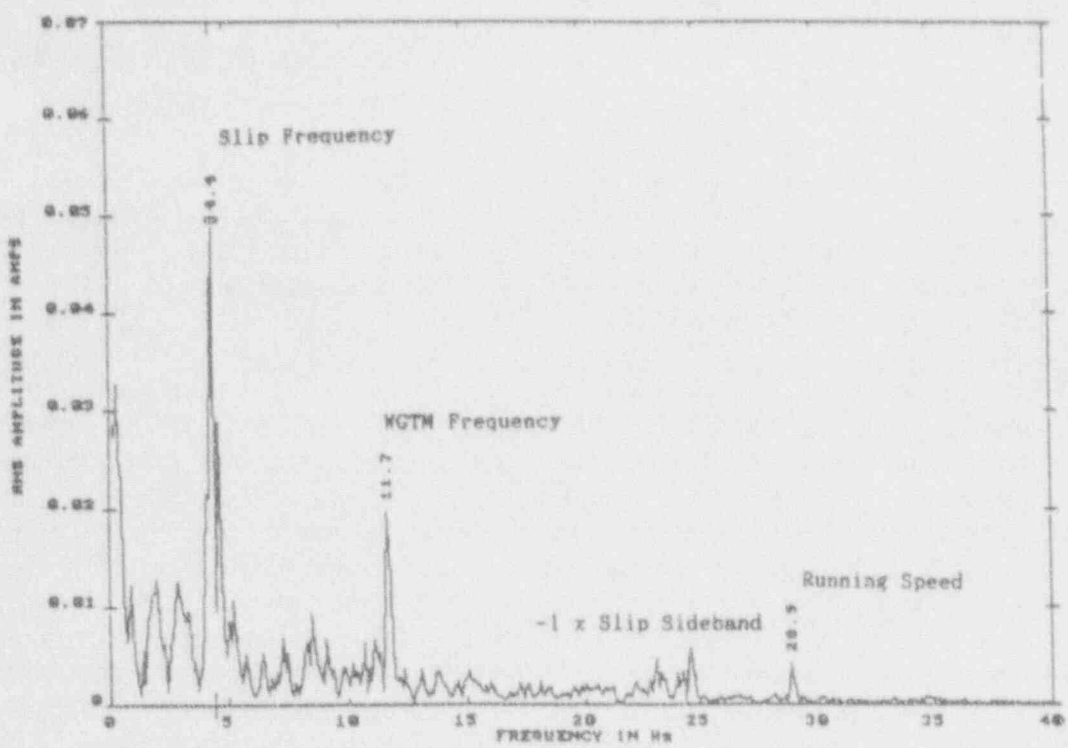

Figure 4. Run 3 FFT of filtered signal.

broken bars. In the case of Figure 4, the slip amplitude of 0.05 divided by the total energy content of 0.0976 yields 51%, indicating several broken bars.

2. There are no slip harmonics with an undamaged rotor. At the onset of rotor damage the presence of slip harmonics appears and may extend to several multiples of the slip. At the same time, sidebands about the running speed appear, spaced by the slip frequency, and multiples, thereof. See Figure 5, where the first harmonic is clearly visible and a second is just beginning, indicating several broken bars.

3. As a further confirmation of rotor deterioration, the power line spectrum (FFT of line signal) is investigated for the presence of slip sidebands as well as the level of the sidebands, relative to the power line peak.

   On a motor is good condition, these sidebands are typically more than 50-60 dB below the amplitude of the supply line peak. As the condition of the rotor deteriorates, the difference between the peaks decreases. Definite rotor bar damage is present when this difference is less than 45 dB. An estimate of the number of broken bars is then made based on the ratio of the sideband peaks to the supply line peak. See Figure 1, where the difference is less than 40 db, indicating several broken bars.

4. At the same time as the increase in slip peak amplitude is occurring, other key frequencies are decreasing in value, such as the WGTM. This happens due to the increased electrical *load* placed on the motor as a result of the damage to the rotor, rather than as a decrease in the mechanical loads being carried by the worm gear or other intermediate gears. See Figure 5, where the amplitude of the WGTM frequency, at 11.4, is overshadowed by the second slip harmonic. Note also the slip sidebands above and below the running speed at 34.4 and 22.8 Hz.

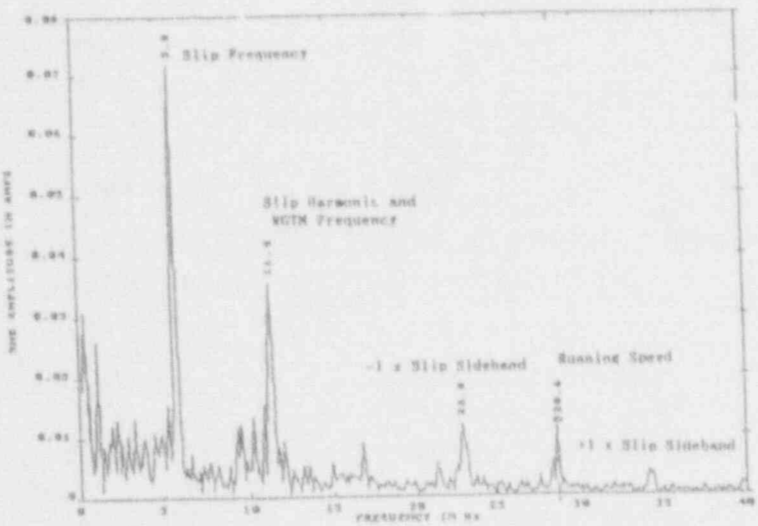

Figure 5. Run 5 FFT of filtered signal.

TEST PURPOSE AND METHOD

Purpose

The purposes of this test were as follows:

1. To evaluate the use of Fourier analysis of motor current waveform as a valid technique for remotely assessing degradation of valve actuator motors

2. To provide data regarding the number of rotor bars that must be open circuited before locked rotor torque drops to its rated value, and the reduction of locked rotor current associated with this rotor degradation

3. To provide data for the four Fourier analysis criteria to assess rotor bar integrity discussed above as a rotor is degraded from its normal condition to a failed condition.

Method

A valve actuator was obtained in good working order and was equipped with a magnesium rotor motor. The intent was to install this actuator on the test stand and perform both locked rotor testing and normal stroke testing on the test stand while monitoring the motor current with the current wave form analysis system. However, because of the extremely high unit ratio of this particular actuator, the locked rotor torque produced by the actuator when the motor was in a stalled condition would have exceeded the maximum torque rating of the actuator. The locked rotor testing had to be done with the motor separated from the actuator.

After each locked rotor current and normal stroke test, the motor rotor was removed and one more rotor bar was opened circuited. Adjacent rotor bars were opened circuited, as these motors typically fail in service with a single crack propagating around the circumference of the end ring.

The purpose of installing the motor on the test stand was to allow a load to be provided on the motor so that the motor would simulate its normal operating condition, and to allow the motor to drive the load through the valve actuator so that the frequencies associated with the actuator gears would be noted on the frequency spectrum. The motor was to be considered as failed when the locked rotor torque fell to the point at or below the motor rating.

To obtain base-line documentation, before the rotor was damaged in any way, the motor was meggered and bridged, and a locked rotor torque test was made. The motor was then installed on the actuator and operated near its rated running current while the actuator was stroked in the closed and open directions. First, 10-second stroke times were made and then 30-second stroke times were made. The 10-second stroke times were made because they are representative of the stroke times of the faster acting valves. The 30-second stroke times were made because the frequency spectra resolution is much better with a longer stroke time and we wished to determine whether a longer stroke time would provide significantly better resolution.

After the stroke tests were made, the motor was removed, the rotor removed from the motor, and one rotor bar was open circuited. The rotor bar was open circuited by drilling it out. The rotor core punchings were first removed with a chisel exposing the rotor bar and then the rotor bar was drilled out. After the rotor was open circuited the locked rotor test and stroke tests were performed again. The stroke tests were monitored by the current wave form analysis equipment.

The motor was removed again and another rotor bar was opened circuited; this process was continued until the motor locked rotor output dropped to 60 ft·lb, which was the rated torque. The torque output of the motor was then compared with the results of the frequency analysis for each rotor condition to evaluate the indicated motor health against the actual motor torque output. The locked rotor amps were compared with the number of broken rotor bars and torque for each rotor condition to evaluate the expected reduction in locked rotor current with the open circuiting of the motor rotor. Because each of the motor run tests were performed at conditions near rated current, the motor was operating at a slip near the rated values.

At the end of the test it was decided that it would be appropriate to do two more motor runs at a light load condition or a very small slip condition to ensure that the continued effectiveness of the diagnostic method. This consideration was made because valve actuators that are operating under normal condition at zero differential pressure may be driving valves that provide very low loads. These two runs were made at shaft loads of 100 and 200 ft · lb, and did indeed result in reduced motor current and slip. (Earlier runs were made at loads of approximately 800 ft · lb.)

TEST CONFIGURATION

Actuator

The actuator used for the performance of the test was a Limitorque Model SMB Size 2 with an overall unit ratio of 212.5:1. The 212.5:1 is accomplished with a motor pinion with 20 teeth mated to a wormshaft gear with 50 teeth and using an 85:1 worm set ratio. The actuator was a unit that had previously been removed from service and was now in use as a training aid. The actuator was disassembled and inspected before the test and determined to be in operable condition. The actuator was reassembled and lubricated to the manufacturer's recommendations using Exxon Nebula EP-1 grease.

Motor

The test was performed using a Limitorque actuator supplied with a 60 ft · lb rated motor manufactured by Reliance Electric with the following nameplate information:

Start 60 ft · lb	Run 12 ft · lb	3.9 hp
Frame 184 R 2	Duty 15 min.	3 Phase
Form P	RPM 1720	230/460V
60 cycles	CODE M	Rise 75°C
12.2/6.1 A	Identification	Y227636418
		Reliance Electric

Before the test began, the motor was disassembled and visually inspected. The visual inspection included looking for cracks on the surface of the end ring, inspecting for globular deposits of magnesium appearing on the surface of the end ring, darkening of the light green color of the end ring, and inspecting for deposits between the end ring and the first punching.

As a result of the visual inspection and the electrical resistance checks of the windings and insulation, it was determined the motor was in satisfactory condition for the testing.

The motor was run before the beginning of the test to determine if it was capable of providing the 60-ft·lb rated starting torque. This was accomplished by removing the motor from the actuator and affixing a calibrated torque wrench to the output shaft of the motor through a specially machined nut and socket arrangement. The motor and torque wrench were then placed in a jig designed to hold the motor and support the handle portion of the wrench. Rated voltage was then applied to the motor and the amount of torque being produced by the motor at locked rotor could then be obtained from the torque wrench. Motor current and voltage were also monitored during the locked rotor test. The first locked rotor test was performed before open circuiting any rotor bars, and the results showed that the motor was capable of producing greater than its rated locked rotor torque. During this first test the motor produced 80-ft · lb of output torque at 498 V with a locked rotor current of 62 A. Motor torque output is discussed further under TEST RESULTS, below.

TEST RESULTS

Torque and Current as Rotor Bars Were Opened

The motor nameplate torque rating was 60 ft · lb. Because the motor terminal voltage was

495 v, and the nameplate voltage rating was 460, it was anticipated that the locked rotor torque would be higher than rated torque by the factor of unit terminal voltage squared (Rebbapragadda, 1990). This yields 69.5 ft · lb. The motor actually produced 80 ft · lb locked rotor torque before it was damaged, and continued to produce 80 ft · lb after three rotor bars were opened. See plot of motor torque and current versus the number of open rotor bars in Figure 6. After the fourth bar was opened, the torque dropped to the rated value, 60 ft · lb.

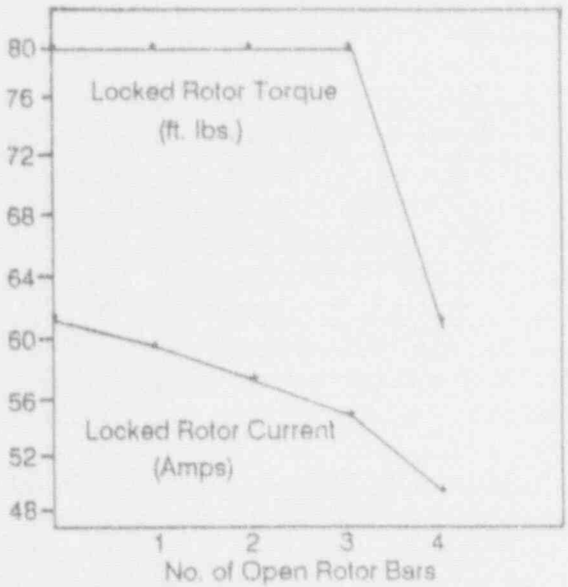

Figure 6. LR torque, current vs. number of open rotor bars.

This interesting result may be explained by the fact that for valve actuator motors, the load current does not have the most significant influence on total current. As stated in Rebbapragadda, "The total current is equal to the vector sum of the magnetizing current (reactive) and the load current (primarily resistive). . . . In the case of a valve actuator motor, the magnetizing current is dominant as a result of the high levels of flux, or saturation, used in the motors to obtain a high torque to inertia ratio." As the rotor bars were open circuited, there definitely was an effect on the total current, as can be seen by Figure 6. The load current, however, must have remained fairly constant for the torque output to remain constant. The reduction in total input current on Figure 6 must have been primarily produced by the reduction in magnetizing current as the rotor bars were opened. At the time the fourth bar was opened, the flux density probably fell below saturation as the magnetizing current was reduced below the knee of the magnetization curve. The resulting decrease in flux density produced the reduction in torque output at this point. The fact that the motor actually produced 80 ft · lb, while the rated torque is 60 ft · lb, and the voltage corrected torque is 69.5 ft · lb, is probably because the motor was conservatively rated.

FINDINGS OF FREQUENCY ANALYSIS

Results were obtained at the stroke durations of 10, 20, and 30 seconds to determine the sensitivity of the MCSA system. The 10-second period was also considered to be the shortest time that an actuator in the plant would be required to stroke. The results discussed here are all based on the 10-second stroke time, which yields a frequency resolution of 0.1 Hz which was seen to be adequate to detect and determine the degree of actuator/rotor degradation.

CONCLUSIONS

The test results have shown the ability of motor current waveform analysis as an effective means to detect broken rotor bars in a valve actuator motor. Cases examined included an undamaged rotor followed by successive open circuiting of the rotor circuit by drilling holes in the rotor to simulate broken bars. Up to four bars were *broken* in this manner; the actuator was run in both the opening and closing direction, and was operated from full load to 10% of full load. After four rotor bars were broken, locked rotor torque fell to near its rated value.

The following observations about changes in the performance parameters are noteworthy:

1. Time Waveform

a. Motor inrush current is not appreciably affected by rotor condition

b. Average running current increases with rotor degradation and thus appears as an increase in mechanical load, while the mechanical load is constant

c. Average running current can be monitored for changes and rotor degradation are suspected when there is no known change in load

d. Patterns of slip frequency are observed as the electrical load increases with a degraded rotor.

2. Spectral Analysis of Filtered Signal

a. Slip is a small component of the total energy content for a *healthy* rotor

b. Slip exactly identifies the true running speed

c. Energy associated with slip increases greatly as a rotor degrades from 15-20% for an undamaged rotor to over 90% for a severely damaged one

d. Slip harmonics appear as rotor degradation occurs

e. Slip sidebands appear about the running speed as rotor degradation occurs

f. The WGTM frequency, and its harmonics, predominate as the mechanical load increases, but decreases as the electrical load (slip) increases for a degraded rotor condition

g. The WGR (worm gear rotation) frequency associated with the worm gear-drive sleeve components decreases in magnitude as the slip increases for a degraded rotor

h. Running speed is readily observed.

3. Spectral Analysis of Line Current Signal

a. Used to confirm results of conditioned (filtered) signal

b. Slip sidebands about power line peak appear as the rotor degrades

c. Difference between magnitude of slip sidebands and power line peak decreases from 50-60 dB to less than 45 dB as rotor condition degrades

d. Multiple slip sidebands appear about power line peak as rotor condition severely degrades.

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Nonintrusive Testing of Check Valves

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ABSTRACT

This paper describes techniques for nonintrusive testing of check valves using permanent magnet and alternating current and summarizes the status of Public Service Electric and Gas Company of New Jersey's program with Valvision, Inc.

INTRODUCTION

Programs to ensure the operational readiness of safety related check valves in nuclear power plants can benefit from the use of nonintrusive diagnostics (NID). Nonintrusive diagnostics means obtaining information regarding valve operational readiness without penetrating or otherwise disturbing the check valve system boundary.

Determining operational readiness of a check valve can be broken down into whether a valve meets two criteria:

- Is the valve currently operable, i.e., does it meet current inservice testing (IST) performance requirements, as defined in ASME Code and as required by plant Technical Specifications?

- If the valve is currently operable, will it remain operable at least until its next IST, i.e., is it degrading such that it will fail to meet performance requirements when challenged in the future?

The first criterion—that of operability—basically requires a demonstration of proper check valve actuation, that is, that the valve obturator moves when subjected to fluid pressure in the direction(s) required to fulfill the valve's role in preventing or mitigating an accident. In addition, for some valves, there is a requirement that the valve not leak excessively when pressurized in the flow blocking direction.

At present, nuclear power plants generally meet the first criterion indirectly by flow testing. This usually involves subjecting the valve to fluid pressure in one or both directions. In the open direction, the criterion is met if flow occurs—in some instances without a direct measure of flow rate. In the flow blocking direction, the criterion is no bulk flow and, in some instances, an acceptable leak rate. Because of the indirect nature of the test, there have been instances where severely degraded or even failed valves have passed such a test successfully.

The second criterion requires a demonstration that the functioning parts of the valve have not degraded significantly as a result of aging such that further degradation may lead to valve failure.

The primary approach to meeting the second criterion is disassembly and inspection. The valve is periodically opened up and the functional parts inspected for signs of aging degradation, e.g., dimensional changes in the hinge pin. This direct method for determining degradation provides a clear indication of the state of the valve functional parts. However, it is intrusive, requiring precautions to ensure that the valve has been returned to its operating status following the disassembly and reassembly. In addition, there may be significant radiation exposure resulting from the inspection of some valves, and in some cases, outages may have to be extended. As a result, the cost of this approach in time and money is high.

Thus, the current procedures for insuring operational readiness, although capable of providing assurance of safety, clearly can be improved. The Nuclear Regulatory Commission (NRC) and The Institute of Nuclear Power Operations (INPO) have both encouraged the nuclear utilities to seek improved procedures. The improved procedures will require development of new NID techniques which provide the needed information regarding valve operability and state of degradation.

OBJECTIVES

The objectives of nonintrusive testing include the following:

- An improved ability to comply with existing and proposed ASME OM Code in meeting nuclear plant Technical Specifications for In Service Testing (IST) of safety related valves, including a more direct indication of obturator movement during the test.

- The elimination or reduction in frequency of periodic disassembly and inspections of check valves which may be needed to comply with NRC and INPO recommendations and/or regulations aimed at detecting and trending degradation of check valves.

- In addition to the above, the use of nonintrusive diagnostics can be made a part of a predictive maintenance program for improving overall plant availability by preventing unanticipated check valve failures anywhere in the plant which could result in unscheduled outages.

SELECTION OF TECHNIQUES

As part of the NRC Nuclear Plant Aging Research Program, researchers at Oak Ridge National Laboratory (ORNL) have been investigating the aging of check valves (including swing, vertical, and horizontal piston lift) and stop checks. Their first report consisted of a review of operating experience (Greenstreet et al; 1985), in which they identified the common failure modes and causes and the currently available methods for detecting them. The report made the case that nonintrusive diagnostics were needed for check valves.

Last year, ORNL issued a second report (Haynes, 1991a) evaluating various potential nonintrusive diagnostics techniques as to their ability to identify and trend the failure causes of check valves. They evaluated a total of five monitoring techniques:

- Acoustic emission

- Ultrasonic testing

- Magnetic flux

- Radiography

- Pressure noise.

Two of these methods, pressure noise monitoring and magnetic flux monitoring, were developed by ORNL specifically for this application.

Of the five techniques, ORNL recommended either magnetic flux monitoring or ultrasonic testing combined with acoustic emission monitoring as providing the sensitivity to detect all major check valve conditions. The magnetic flux monitoring method as described in that report, although very attractive from the standpoint of simplicity and cost, had one major limitation—it was not sufficiently nonintrusive, i.e., it required the initial attachment of a permanent magnet to the hinge arm or disk of the valve.

Recognizing that limitation, ORNL researchers subsequently developed two new external, magnetic-flux based methods that required no use of internal magnets, but which appeared to provide the same level of detail. These two NID methods, referred to here as permanent magnet

(PM) and alternating current (AC) magnetics, have been the subject of further development and testing as part of a joint R&D project between Public Service Electric and Gas Company of New Jersey (PSE&G) and Valvision, Inc., of Cambridge, NY., a licensee of the ORNL external magnetics technology.

The joint effort uses PSE&G's Energy Technology Development Center in Somerville, NJ. A flow loop has been constructed there for testing 4 and 6-in. check valves over a wide range of flow conditions. The goal of the effort is to develop and demonstrate the combination of external magnetics and acoustics in a practical, commercially available NID system that can be used for all types of check valves and that will provide improved IST, as well as degradation monitoring capability.

This paper describes the techniques and summarizes the status of this joint project.

DESCRIPTION OF TECHNIQUES

The external AC magnetics technique is illustrated by Figure 1, (Haynes, 1991b). Two coils are used. In the illustration, they are wrapped around the valve body. One coil, the transmitter, is connected to a source of electricity at a selected frequency. The second coil, the receiver, is connected to signal processing equipment and then stored or displayed for analysis. The mechanism of detection of the disk appears to be the effect that the disk has on the efficiency of AC magnetic flux transmission from the transmitter to the receiver coil, specifically by its effect on eddy current losses. Detection of the disk position and motion are detected by measuring the amplitude of the current flowing in the receiver coil.

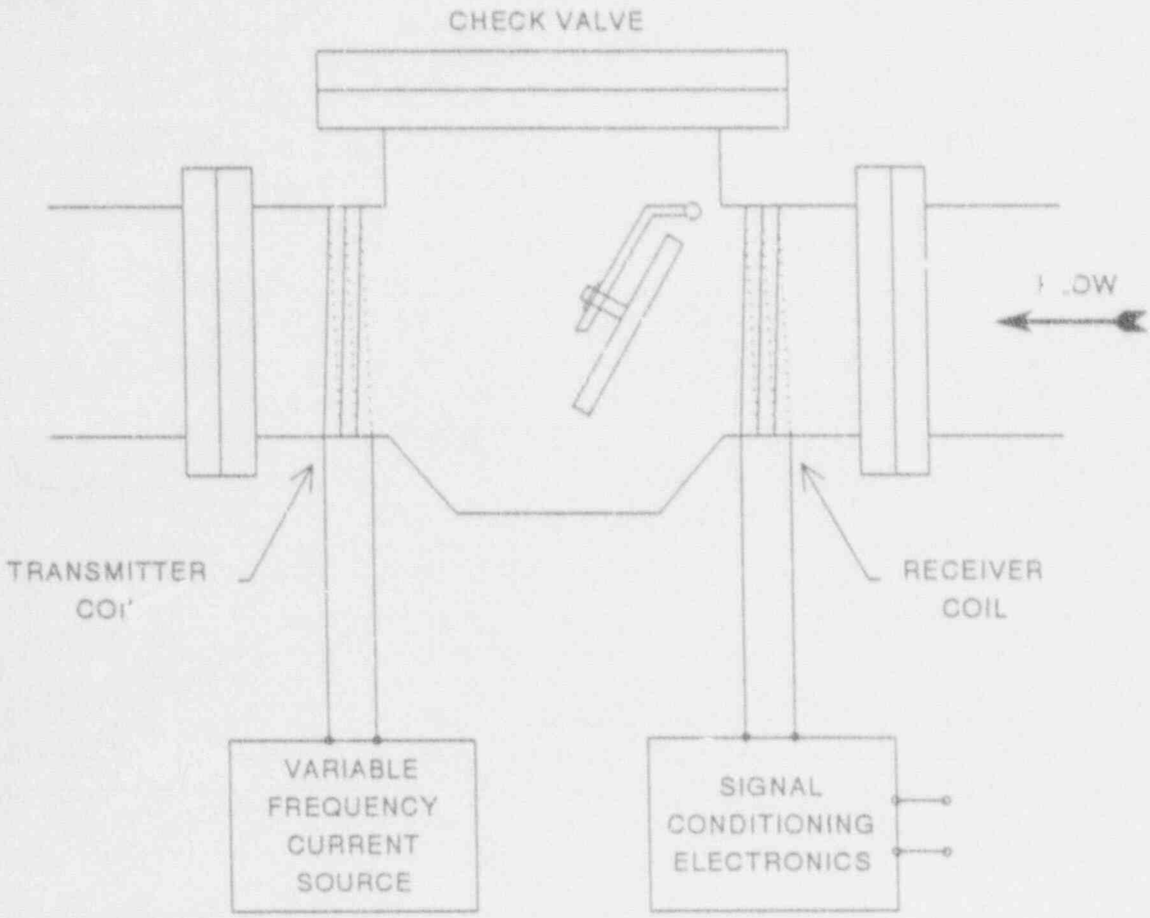

Figure 1. Simplified drawing of external AC magnetics technique.

Parameters that affect the performance of the AC magnetics system include the power input to the transmitter coil, the locations and design characteristics of the coils, the frequency of transmission of the magnetic signal, and the type of signal processing used with the receiver coil output. The AC technique has been found by ORNL to be more effective with 300 series (paramagnetic) stainless steel (SS) and brass valves than with carbon steel (CS) or 400 series (ferromagnetic) valves.

The external PM magnetics technique is illustrated in Figure 2 (Haynes, 1991b). In this arrangement, two magnets are attached to the valve body at locations near the seat and backseat. The effect of the magnets is to induce a stationary magnetic field within the valve body. The flux distribution of that field reflects the placement of the external magnets as well as the location of magnetic masses within the valve. Motion of the disk alters the flux distribution of the field.

The disk position and motion are detected by a probe incorporating a Hall-effect sensor to detect changes in the magnetic flux at the location of the sensor. The Hall effect signal is processed using a gaussmeter and other signal conditioning equipment.

Parameters that affect the success of this technique include the placement, design, and strength of the magnet assemblies; the location and design of the probe incorporating the Hall-effect sensor; and the type of signal processing. This technique has been found effective primarily with CS and other ferro magnetic valves.

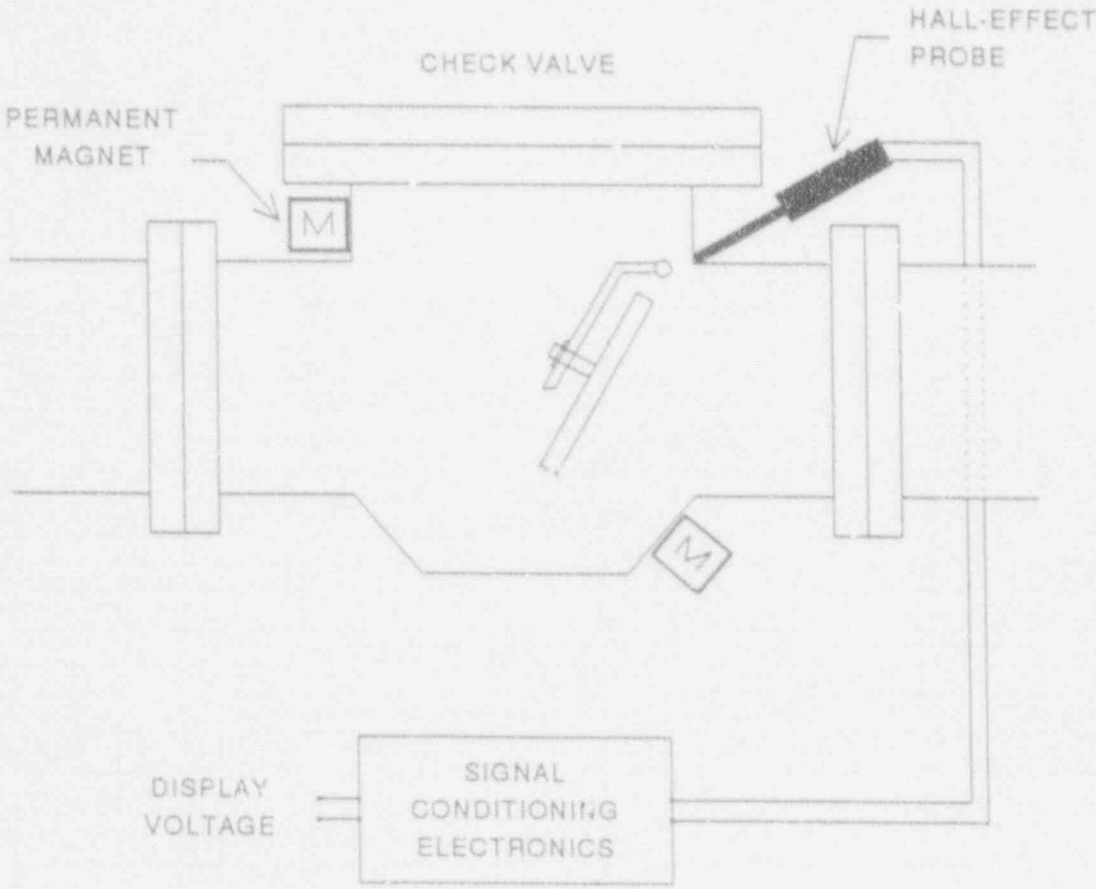

Figure 2. Simplified drawing of external PM magnetics technique.

The two magnetic flux monitoring techniques provide the same type of information. They locate the instantaneous position of the obturator (disk or piston) as a function of time over its full range of travel. They are both equally successful when the flowing medium is water, steam, or air. In each case, the resulting signature consists of a time trace of the position of the valve—both during an opening or closing transient (i.e., when the flow changes) or during steady state, when the flow is constant. The difference in the two techniques is in their relative effectiveness with different materials of construction.

Each magnetic flux monitoring technique is capable of detecting small movements of the obturator associated with its normal reaction to turbulent flow (flutter). As degradation increases clearances between moving parts, the resulting additional movements (wobbling) can also be detected.

The acoustic monitoring technique provides different, complementary information from that obtained from the external magnetics. It utilizes commercially available accelerometers and amplifiers. One or more accelerometers are attached to the valve body close to where discrete acoustic emissions would be expected to be generated by metal-to-metal contact of valve internals, both during opening and closing transients and when there is steady-state flow. The acoustic emissions generated during opening and closing transients indicate when the disk has either seated or backseated. During steady state, acoustic emissions associated with the small movements of the obturator can be detected.

In addition to detecting discrete emissions from the valve during transients or when there is steady state flow, the accelerometers provide an indication of steady state leakage through the check valve when the pressure gradient is reversed (flow is blocked).

Parameters that affect the success of acoustics include sensitivity and placement of the accelerometer(s), and the software needed to display and characterize the acoustic signals received from the sensor(s).

APPLICATION TO INSERVICE TESTING

Inservice testing of check valves in nuclear power plants is carried out in accordance with the relevant American Society of Mechanical Engineers (ASME) Code. Changes that supercede the ASME Boiler and Pressure Vessel Code, Section XI, Subsection IWV, are making their way through the relevant Code committees. However, the basic requirements of Section XI are not expected to be changed significantly. These include demonstrating that the valve obturator moves correctly under the influence of a change in fluid pressure in one or both directions.

Magnetic flux monitoring coupled with acoustic emission monitoring provides an unambiguous demonstration of obturator movement. For example, when the valve is subjected to a sudden application of fluid pressure in the open direction (i.e., from a pump turned on or a control valve opened), the magnetic flux monitor, either AC or PM, will generate a characteristic signature that resembles a plot of obturator position versus time (Figure 3). The signature provides a measure of the time of opening as well as an indication of the final position of the obturator at steady state. If the final position is not full open (e.g., not hard against the backstop for a swing check valve), the magnetic flux monitor will then generate a characteristic steady state signature which corresponds to the obturator flutter and/or wobble amplitude and frequency.

The acoustic monitor attached to the same valve will detect the corresponding acoustic emissions due to the obturator leaving its seat and then subsequently striking the backseat. The time interval between acoustic emissions will correspond to the time interval of the magnetic signature. The acoustic monitor can also detect that a swing check valve obturator is not fully open by sensing the metal to metal contacts in the hinge pin associated with the flutter and or wobble at steady state.

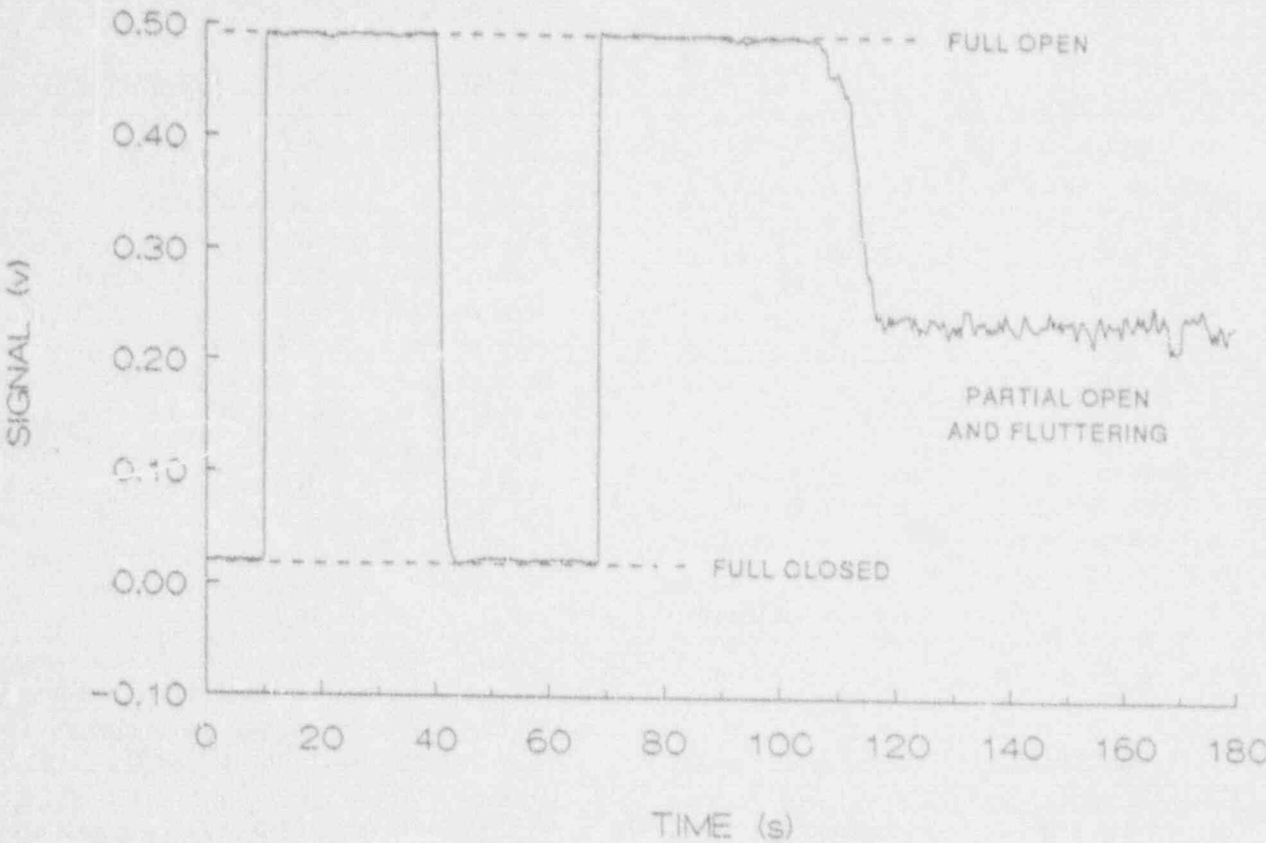

Figure 3. Characteristic signature of obturator position versus time generated by magnetic flux monitor.

The magnetic and acoustic signatures have been demonstrated to be reproducible from test to test provided the external equipment (accelerometers, coils for AC, magnets and probe for PM) are attached at the same positions and adjusted in the same manner for each test.

APPLICATION TO DEGRADATION MONITORING

Degradation monitoring of check valves is addressed both by the NRC and by INPO. The current (or fall-back) approach to degradation monitoring is periodic disassembly and inspection. The combined system of magnetic flux monitoring and acoustic emission monitoring is being investigated to determine its capabilities to provide a nonintrusive substitute.

The primary mechanism of check valve degradation is gradual erosion/corrosion of the moving parts, i.e., of the hinge pin-to-hinge arm bearing surface for a swing check valve, as a result of flutter at steady state.

The result of the wear is increased clearances and the introduction of additional degrees of freedom of motion of the obturator. These degrees of freedom are expected to be detected both during the transient actuation of the valve as a result of changing fluid pressure and also during steady state. In each case, under certain conditions, the additional degrees of freedom can be detected both in the magnetic flux signature and the acoustic emission signature.

Magnetic flux monitoring of the changes that occur as a result of hinge pin wear of swing check valves is currently being investigated as part of the Valvision-PSE&G project. Initial results have been positive, with the techniques capable of differentiating normal disk flutter from disk wobble.

STATUS AND LIMITATIONS

The initial application of combined magnetic and acoustic monitoring is for IST. At present, a prototype system based on PM magnetics and acoustics monitoring has been built and tested. It consists of a portable battery powered magnetics and acoustics data acquisition package and a data analysis and display package. The data acquisition package consists of self attaching magnet assemblies, probe assemblies including universal attachment hardware, battery powered gaussmeter and other signal processing equipment, and a battery powered digital audio tape recorder. The data analysis and display package consists of a customized Macintosh-based data reduction and analysis software program and a waveform plotting program.

The prototype system has been used successfully to monitor the disk position of 4-in. class 125 and 6-in. class 300 CS swing check valves in the PSE&G flow loop. It has also been successfully used to monitor disk position of 6-in. class 600 and 8-in. class 150 nuclear grade CS check valves located at a PSE&G warehouse at the Salem Plant.

The use of PM magnetics for ferromagnetic swing check valves had been successfully demonstrated earlier at ORNL using several large valves, including a 10-, 12-, and 16-in. class 125 and class 150 CS valves in an operating cooling water system. The method also has been applied to a vertical lift valve. The PM method has not yet been tested on large high-pressure (greater than class 600) check valves, or on the other types of check valves.

The application of AC magnetics monitoring to check valves has been demonstrated at ORNL using brass and paramagnetic SS valves. The maximum size tested was 3 in. In addition, ORNL has demonstrated the AC method using 6-in. and 10-in. class 125 CS swing check valves, although careful placement of the coils was found to be critical in achieving success. Valvision and PSE&G expect to begin loop testing of the AC system starting with a 4-in. paramagnetic SS swing check valve in the near future.

The application of AC or PM magnetics to all the types and sizes of check valves found in nuclear plants thus has not been fully demonstrated. This will be the focus of project activities over the next several months. However, there has been no reason to suspect that the methods will not be effective provided the design parameters needed to obtain useful signals are understood.

Meanwhile, initial field testing of the prototype PM system on the smaller (less than 8-in.) CS valves at the Salem Nuclear Plant is expected to begin very shortly.

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Toledo Edison Review of Thermographic Analysis of Valve Operations

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ABSTRACT

Davis-Besse began using infrared thermography to inspect electrical equipment in 1987 and currently has a well established Predictive Maintenance (PDM) program for electrical equipment. Additionally, Davis-Besse has used infrared thermography on a periodic, as applicable, basis to augment other existing PDM programs for valve reliability and thermal performance since late 1989. This paper will discuss those initial applications to detect valve positions in our component cooling water (CCW) system and steam trap bypass valves. It will also discuss recent attempts to acquire thermographic baseline data on the pressurizer safety relief valves. These data were acquired in an attempt to obtain actual thermal gradients across the valves. This may permit actual plant conditions to be recreated in lab testing.

The purpose of the paper will be to identify possible economic and safety benefits that can result from applying infrared thermography to valve applications. The paper should also enhance the interest to promote further application, development, and sharing of information on this topic.

INTRODUCTION

The use of infrared thermography to assist in identifying whether a valve is open or closed is not a new application for this technology. This paper is intended to reveal other potential applications. Specifically, those who have the infrared thermography technology in-house may find valve applications to be beneficial.

Before appreciating the capabilities of this technology as applied to valve applications, a basic understanding of infrared theory is required. The dictionary definition for infrared is as follows: "lying outside the visible spectrum at its end—use of thermal radiation of wavelengths longer than those of visible light." This definition does not identify how it can be applied to this type of application. To understand how this technology can be implemented, some basic concepts will be reviewed. This is a very brief review of infrared theory, intended to highlight some benefits and concerns of infrared monitoring. Most infrared equipment manufacturers are designing equipment to be most sensitive in the 3 to 5 or 8 to 12 micrometer, or micron, wavelengths. A micron is 10^{-6} meters. For a comparison, the average person can see in the 0.4 to 0.7 micron range. Therefore, it is obvious that the infrared equipment is using a much longer wavelength than is visible. The reason for various wavelength instruments results from the physical behavior of radiated energy in the infrared wavelengths. Certain applications needed the 3- to 5-micron instrument. However, for the applications encountered most in the power production industry, the 8- to 12-micron instrument has its advantages.

All objects above -273° C naturally emit electromagnetic energy, resulting from the movement of the molecules within the object. The behavior of an object in the infrared spectrum is dependent on its molecular structure. As the temperature of the object increases or decreases, the molecules within it increase or decrease in speed. Each molecule carries its own electrical charge from the atomic structure. The movement or oscillation of this electric charge causes the emitted electromagnetic energy. This is the source of energy that can be detected with the proper infrared instrumentation.

The energy an infrared detector receives is dependent on many factors. The operator has the most input on the proper use of the infrared equipment. Each manufacturer of infrared equipment has designed the equipment to meet certain specifications. Only a properly qualified person or thermographer will constantly be able to produce results within the stated specifications.

To understand the most basic concepts dealing with infrared thermography, the following principles or terms need to be understood:

- RADIOSITY is the total infrared energy coming from an object.
- EMITTANCE is the ratio of energy at a given wavelength emitted by a material at a certain temperature to that of a perfect radiator at the same temperature.
- REFLECTANCE is the fraction of energy reflected by a material at a given wavelength to that falling on it.
- TRANSMITTANCE is the fraction of energy transmitted through a material at a given wavelength to that falling on it.

The radiosity is always equal to the sum of the emitted energy, reflected energy, and transmitted energy. The sum of the emittance, reflectance, and transmittance of an object must equal one. As a general guide for predictive maintenance applications, the transmittance of an object will be equal to zero, or be so small it can be ignored. This makes the emittance and reflectance complements of each other. Most emittance tables do not identify the wavelength or temperature range that is applicable. The geometry of the object being monitored also affects the emittance. A rough or irregular shaped object will normally have a higher emittance than a smooth, flat object of the same material. As a guide, dielectrics or organic compounds have a high emittance, typically 0.75 to 0.98. Oxidized metals have a lower emittance, typically 0.30 to 0.70. The shiny metals often found in an industrial environment have typical emittance values of 0.05 to 0.20.

Infrared thermography is a specialized form of infrared monitoring, which is the process of displaying or printing a black and white or color image from an infrared imaging system. The image produced is identified as a thermogram. The thermogram is similar to a photograph of the image or the target's apparent radiosity within the equipment's wavelength.

One of the goals of applying infrared thermography to valves was to use the equipment currently used in the thermography program at Davis-Besse. This was done to increase the cost effectiveness of both programs. Our equipment is standard, commercially available hardware and software, which consists of two imaging radiometers with a peak sensitivity in the 8- to 12-micron spectrum and the associated personal computer-based analysis system.

COMPONENT COOLING WATER SYSTEM

The CCW system is a closed-loop, single-phase, low-temperature water system. The system has three pumps, three heat exchangers and two trains. Two 20-in. Mission duo-disk check valves on the heat exchanger discharge provide the train or system isolation. Because of the configuration of the system, these are difficult valves to test functionally. For this reason the nonintrusive thermography inspection method is being assessed.

The following items or concerns have been found to be applicable to all valve monitoring applications to permit this technique to be useful.

- A temperature differential across the valve is required in one of its operating modes; this differential is normally noted in the closed position.
- A static thermal mode of operation is desirable in at least one of the valves operating modes. This assists in temperature stability of the valve.
- The valve must have a clear sight path that ideally will include the valve and a minimum of three pipe diameters on each side. This will be a factor of the optics of the thermography equipment. Most equipment will have the option of changing the optics to assist in optimizing the view.
- The valve should not be insulated.
- The process will be simplified if the surface emittance is consistent.
- Reliability will be enhanced if the surface emittance is high.

The CCW valves were ideal for this type of analysis. When the valves are closed, they have a static condition downstream and the upstream side is affected by flow circulation from the other pump. The isolated water leg will attempt to obtain ambient temperature, which is slightly cooler than the process temperature. The entire system is not insulated and is painted with a organic-based paint that has a relatively consistent and high emittance. The configuration of the valves permit an adequate view, as seen by the infrared imager.

The temperature differential across the valves in the field was not very noticeable. However, after performing various techniques on the thermograms while in the image analysis system, the signature of the open valve was distinctively different than its signature when closed. This process has been repeatable and does not use the quantitative value of the upstream and downstream temperatures. The temperature profile appears to be the key to the technique with this type of application.

STEAM SYSTEM VALVES

Valves, such as a steam trap bypass valve, may not be simple to identify as open or closed. The temperature differential across a valve in a steam system will be dependent on the pressure drop across the valve. The potential for a very large temperature differential is possible. This difference can cause heat transfer from the valve to the pipe, which will complicate the analysis. Also these lines may be insulated and are frequently not painted, or are painted with a metallic based paint. Because of the high temperatures, the lines and valves are subjected too. A good working knowledge of the system and the valve is essential to obtain repeatable results.

If the valve is insulated, some of it must be removed to permit the surface of the valve and connecting lines to be scanned. If required, the emittance of the lines and valve can be made consistent and likely increased by coating or painting with a high-temperature material or paint. The color of the material does effect the emittance. Do not use a metallic base paint for this purpose, however. The paint should be tested to verify that its emittance is adequate before field application.

The evaluation of field data from valves in a two-phase or steam application will not be based solely on the thermal gradient across the valve. However, this is an important factor and does require careful analysis. The approximate or expected temperature for the low-pressure side of the valve should be known before inspection. This knowledge will enable a verification that the low-pressure or downstream side of the valve has an acceptable characteristic. For steam trap bypass valves, this will normally be the saturation temperature for the condensate or low-pressure side of the valve. Any one of a number of conditions can raise the condensate header pressure which will raise its associated temperature.

TEMPERATURE PROFILING

The ambient conditions and the thermal gradients or thermal profile across a safety or relief valve are approximated to permit the valve to be tested and the set point reset. This information is used at the test lab to approximate the service in which the valve will operate.

We are reviewing the use of infrared thermography to obtain an accurate thermal profile of the valve while it is in service. After verifying that the thermal profile is repeatable over multiple time frames, these data may be used to achieve a setup condition at the lab which will closely match the condition in which the valve actually operates. This should permit a increase in the accuracy of the set point of the valve.

The valves currently under study are the pressurizer relief valves. Two of these valves are mounted on the top of the pressurizer. The background for these valves, which contributes to the reflective component, consists of other very hot equipment. The surface of these valves was coated with a material to increase the emittance, reducing the reflected component.

The results of the first set of data show the two valves have different thermal profiles. This is most likely a result of the location of the valves and the effects of convective heating on each.

CONCLUSIONS

At Davis-Besse the results of this program appear cost effective and applicable for many valves. Implementation of this technique is intended as a program enhancement only. The main cost savings may be on valves in the balance of plant systems that can be inspected and maintained more effectively as a result of this program. Finding valves in steam applications that are not functioning properly and are passing steam when assumed closed can be a large expense. The safety benefit of this program will be two phased. First, this technique uses an inspection method that is non-contact and nonintrusive and can frequently be performed from convenient locations. These characteristics may permit the inspection of valves in the overhead without having a person climb to it or having to erect a scaffold. Second, the valve programs will be augmented with another tool to assist in proper valve operational assessment.

The use of infrared thermography to assist in identifying the position of a valve is worth the study effort. Not all valves can be inspected with this technology. Infrared thermography will not verify if a valve is leak tight or has zero leakage. However, in certain cases it may identify a valve that is not fully closed.

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Live-Loading Improvements

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ABSTRACT

This paper describes some of the developments and improvements in live-loading valves for the power industry. Live-loading was developed by AECL in the early 1970s and is specified for all heavy water valves in CANDU[a] nuclear generating stations (NGS). More recently, the loss of asbestos-based packing materials required changes to specified live-loading procedures. The new packings replacing asbestos use lower gland loads and are less forgiving of errors in design and installation, but do provide improved functional performance.

Improvements in packing materials from the valve packing manufacturers have focused on combination graphite packing configurations. However, the biggest gains come from implementing a detailed retrofit plan. This includes specifications, documentation, training, and R&D support. This paper provides field examples as well as recommended packing arrangements for some problem valves.

INTRODUCTION

There is a growing concern across industry regarding the consequences of valve stem packing leakage. Environmental concerns and clean air legislation are demanding improvements in valve sealing performance. In contrast, the cost of shutting down revenue producing equipment to repack key valves is high. In normally inaccessible or hazardous areas, the cost of valve maintenance and repair in terms of human exposure and safety is even higher. The demand for quality plant operation requires changes in the way industry approaches valve maintenance.

Live-Loading

In the early 1970s, AECL responded to requests to address the problem of heavy water losses related to stem packing at the Douglas Point NGS. AECL Research's Mechanical Equipment (MED) Branch at Chalk River Laboratories launched an extensive research and development (R&D) program into packing materials, stuffing box designs, service conditions, and valve operation. These early R&D efforts resulted in a significant improved understanding of the behavior of asbestos-based valve packing (Doubt, 1976). One of the more noted developments was live-loading, a method of stem packing that reduces valve leakage to zero under optimum conditions, to near zero under most severe services, and significantly prolongs packing life. In simple terms, live-loading is the application of spring loading (disc springs) to the gland follower (Pothier, 1976). Figure 1 illustrates the classic live-loading arrangement for double-packed valves using asbestos. Various other methods of applying the spring load to the gland follower have been used [i.e., Fisher Controls Live-Loaded Packing Systems, where the spring force is applied between the gland follower and the gland plate (Figure 2)].

Another important product of the R&D work was live-loading specifications (Ontario Hydro, L-942M-81). The specifications describe how to live-load, provide specifications for stuffing/stem

a. CANada Deuterium Uranium. Registered trademark.

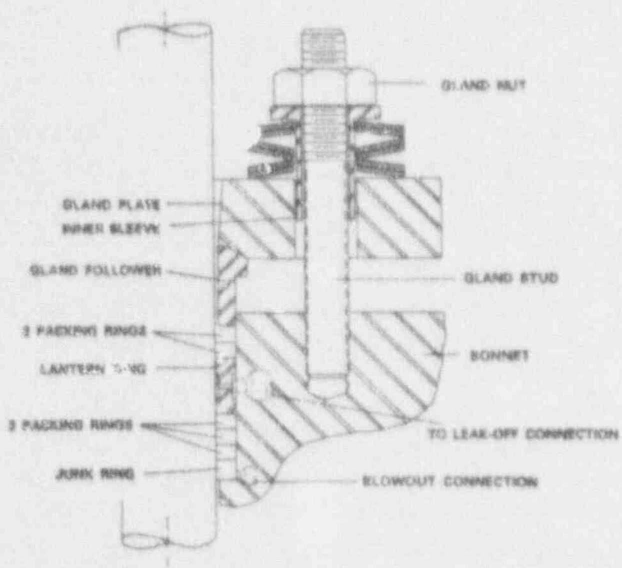


Figure 1. Sketch of double-packed valve with live-loading applied through gland bolts.

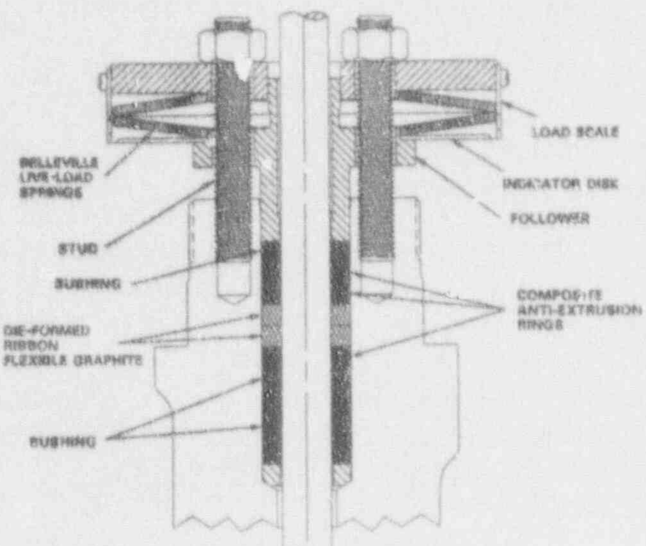


Figure 2. Sketch of Fisher Control live-loading applied between gland follower and gland plate.

design and identify packing materials. The recommended packing arrangement of three packing rings below the lantern ring and two above of braided asbestos JC 1871 (Figure 1) has been very successful, realizing estimated savings of $3M/year/unit by avoiding heavy water losses and radiation exposure to personnel.

When to Live-load. Live-loading is a means of reducing the risk of fluid loss for critical applications. The majority of the valves in a nuclear or fossil-fired station do not require live-loading. Conventionally-loaded, five-ring combination, graphite configurations will provide good, reliable sealing for most power plant valve applications.

When deciding to live-load, look at those systems that are critical to operation of the plant, where failures are likely to cause significant economic penalties (i.e., a forced outage or derating). For a CANDU nuclear plant, a minimum list would include main steam, boiler feed, all nuclear systems, and heavy water systems. All large heavy water valves in the primary heat transport (PHT) circuit are live-loaded for economic reasons.

This includes control valves and gate valves. Valves on less critical systems that are stroked frequently and have a history of packing leakage may benefit from live-loading.

The normal requirement for elasticity in packing can be relaxed somewhat for live-loaded packing because elasticity is provided outside the stuffing box. This means that more of the porosity in gland packings can be filled with blocking agents. A corollary to this is that packings that are inferior without live-loading, because of poor elasticity, may turn into superior products with live-loading [i.e., woven-polytetrafluoroethylene (PFTE) fibre, or Teflon[b]].

However, the design and application of disc springs alone will not provide effective, leak-free valves. The simple concept of live-loading does not end here.

Figure 3 illustrates some of the parameters and relationships that must be in place for a successful live-loading program. This paper will not walk through the steps of live-loading designs, since they are well documented. What is not common practice is the proper set-up of a quality repacking or live-loading program.

SPECIFICATIONS

The lack of specifications in stuffing box design and packing qualifications has been a problem in the valve industry for years. Fortunately, this has been recognized by the valve and packing industry, and new specifications are being drafted by the Manufacturers Standardization Society (MSS).[c]

Revised live-loading specifications for combination graphite packing have been issued in Canada by Ontario Hydro (OH) (M-724-91) and AECL CANDU (30830-TS-003). The revised specifications reflect a desire to ensure performance and plant availability of existing and new stations. The specifications are the result of a joint

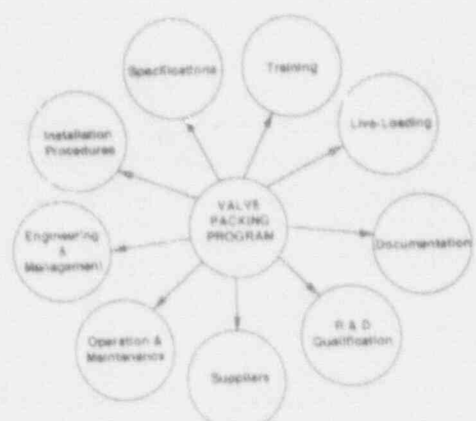


Figure 3. Integrated program for valve packing retrofitting.

b. Teflon is a Dupont trade name for PFTE.

c. Private Communications with E.A. Blake, Edwards Valve, July 1990.

effort among design, maintenance, R&D, purchasing, and the suppliers. It is beyond the scope of this paper to provide all the details from the specifications, but a few points are worth mentioning.

Disc Springs. Disc springs are the key component for a live-loaded valve. Unfortunately, there is not an ASME specification similar to DIN specifications 2092 and 2093 for disc springs. Although good manufacturing practices are followed by disc spring manufacturers, catalogue spring-loading information is not reliable enough for good live-loading.

To address this, the revised AECL/OH specifications call for fully-machined disc springs. All springs should be deburred to a surface finish of 1.6 μm arithmetic average (63 micro inches CLA) and compressed completely flat. The height-to-thickness ratio of disc springs should not exceed 1.4 to 1. The recommended spring material is ASTM A564, Type 631 stainless steel (17-7 PH) in the TH1050 condition or engineering equivalent. It is strongly recommended that a load deflection curve be provided with each lot of springs. When installing a spring set, the lubricant on the springs should be molybdenum disulphide free.

When installed, the spring stack should not be loaded to flat or to a stop. On large valves, the induced thermal taper on the stem is sufficient to generate a significant volume change in the stuffing box. On the return stroke or valve opening, this volume change translates into an increased load on the packing. If the elasticity of the packing is poor, packing fracture can occur. Loading the spring set flat does not allow the gland follower to effectively accommodate the volume and load change; premature stem packing failure could result.

Stem Material

Packing materials have various additives and blocking agents. These additives can cause stem corrosion with 400 series stainless steel. To reduce the risk, all stem materials for new valves and replacement stems for CANDU NGS [operating conditions 10 MPa (1,450 psi) and 290°C (563°F)] call for ASTM A564, Type 630, H1075 or H1100 (17-4 PH stainless steel). There are other suitable materials, and corrosion resistance is not the only requirement. The use of 17-4 PH stainless steel is not recommended for applications above 345°C, due to toughness limitations. The stem finish is important to reduced wear. The AECL/OH specifications call for 0.4 μm arithmetic average (16 micro inches CLA) or better.

Stuffing Box Internal Components

The newer valve packing products are much less forgiving of errors and sloppy machining of the stuffing box and related internal components than braided asbestos packing. Diametral clearances must be tighter for today's packings to avoid extrusion. As a general rule, to avoid stem galling, the diametral clearance between the stem and those components not in contact with the packing should be 1.15–1.25 mm (0.045-0.050 inches). Those in contact with the packing sets should have a diametral of 0.15–0.20 mm (0.006–0.008 inches) on the outside diameter and 0.55–0.65 mm (0.020–0.025 inches) on the inside diameter.

Packing Material Qualification

Valve packing qualification can be viewed as a first step towards leak-free valves. Qualification is needed because packing recommendations by many packing manufacturers tend to be unreliable, and there are no standards for comparison of stated manufacturers' claims.

To address this need, AECL and OH have included functional performance specifications for valve stem packing materials; they include chemistry, acceptable ring densities, consolidation, leakage and friction. Any packing used for critical service in CANDU NGSs must be qualified to these standards. The qualification testing includes a bake test, load testing, chemical analysis and short- and long-term stroke testing. The qualification specification used for the CANDU

NGS are at simulated PHT operating conditions. A broader packing qualification procedure has been drafted by the MSS that qualifies packing by valve classification. The fact that the manufacturers are regulating their own industry is most encouraging.

VALVE PACKING MATERIALS

The valve packing is the heart of the stem/stuffing box seal. Some new products that have been developed as asbestos replacement have been beneficial for industrial users. Several reports describe the development of these products (McKillop, 1989; Foster-Miller, 1988; Aikin et al., 1990). New legislation and a desire by industry to improve plant efficiency has resulted in a need for improved understanding of available products.

Braided Non-asbestos Materials

AECL has attempted to qualify more than 20 different braided non-asbestos valve packing products for use in the PHT systems. None of the products tested fully meets the requirements to maintain and improve CANDU NGS availability. The main problems with the packings concern blocking agents and fibre size. Since the fibre size is larger than asbestos, voids between the fibres are larger. These voids are filled with various blocking agents, such as polytetrafluoroethylene (PTFE), which tend to extrude or burn off at PHT operating temperatures. The other concern with large fibres is that they fracture at relatively low gland loads 27.6 MPa (>4,000 psi) and with relatively low numbers of stem cycles.

At this time, few non-asbestos braided products have been recommended, and only for low temperature systems <150°C (300°F). These must be live-loaded because of their inherent high consolidation, must not use glands loads >26.2 MPa (3,800 psi), and must have an anti-extrusion ring at the bottom of the packing set (Aikin and Doubt, 1991).

Combination Die-formed Graphite

Combination die-formed graphite for valve stem packing has been used since the early to mid-1970s. It was not widely used because of early stem corrosion problems and the availability of other suitable products. The loss of asbestos has resulted in the wide acceptance of graphite as a replacement valve packing. The five-ring combination graphite packing set (Figure 4) is rapidly becoming the most commonly used arrangement in the power industry.

This packing arrangement has an excellent sealing record. However, braided anti-extrusion rings increase stem friction, contribute little to the sealing, and provide poor load transfer to the bottom rings.

In an ongoing effort to improve the functional performance of valve stem sealing, the concept of the "composite end ring" was developed (US Patent 4,826,181). The composite ring is a combination of flexible graphite particles and amorphous carbon, pressed and sintered to form a ring with a density of about 1.6 g/cm^3 (100 lb/ft^3). The live-loaded, combination, die-formed graphite set

Figure 4. Standard five-ring combination graphite braid packing set 1kN = 224.81 lbf.

with composite end rings and carbon bushings showed improvement over braided rings (Aikin, 1990). The rings are particularly attractive to the Canadian nuclear industry because they can be manufactured in rectangular cross-sections (i.e., not square). This provides more design flexibility in fitting packing sets to available space.

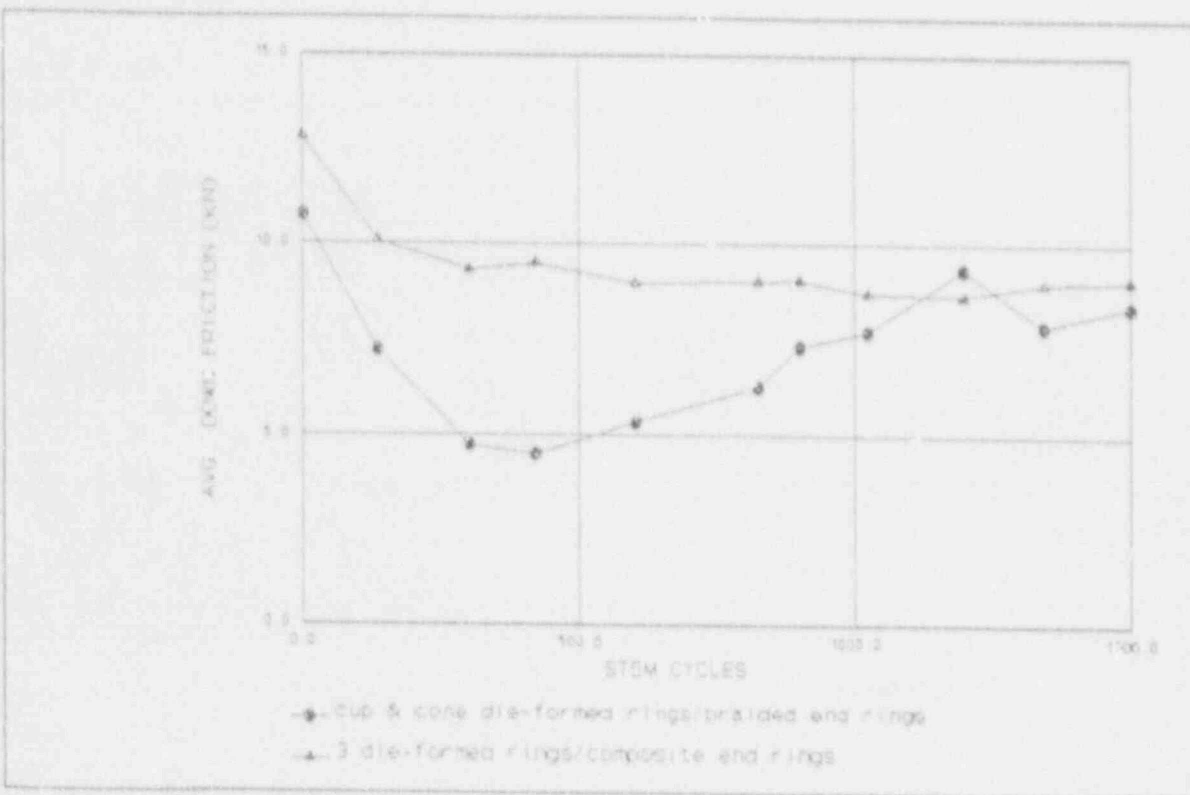
Another notable improvement available to the market is a "cup and cone" design. This packing set has been tested by the AECL; it is leak-free at a gland load of 26.2 MPa (3,800 psi) and has stem packing friction very comparable to that of combination graphite with composite rings (Figure 5). The friction was measured using a load cell connected to a 76.2 cm (3 in.) diameter stem and subtracting hydraulic forces from system pressure. All packing configurations used a gland load of 26.2 MPa (3,800 psi) and were tested at 10 MPa (1,450 psi) and 290°C (563°F).

Polytetrafluoroethylene (PTFE)

The appropriate use of polytetrafluoroethylene (PTFE) or Teflon in nuclear stations has been misunderstood and, in some cases, has resulted in the product being banned. This action has generated unnecessary problems with respect to valve actuation and sealing. Teflon does have advantages, and with proper maintenance documentation, it can provide effective and reliable service that improves plant availability (Aikin and Doubt, 1991).

The advantages of Teflon woven-fibre packings over the popular five-ring die-formed/braided end ring packing sets are

1. 10 to 100 times lower leakage at equal gland pressures
2. Half to one-third the stem friction
3. Lower risk of stem corrosion.



1kN = 224.81 lbf

Figure 5. Average down-stroke friction vs stem cycles for selected packing configurations.

The disadvantages of Teflon are that it tends to extrude through stuffing box clearances around the stem more than other materials, and decomposes in ionizing radiation. The recommended radiation limit for PTFE products is 1.7×10^4 Rads gamma. Woven or braided Teflon is generally not suitable for high-temperature applications, because of the special provisions required to prevent extrusion and accommodate high thermal expansion of the Teflon material. Extrusion of Teflon packings at temperatures <150°C can be prevented with a braided-carbon or composite anti-extrusion ring at the bottom of the stuffing box. Corrosion of stainless steel from Teflon products has been documented; however, the corrosion risk from Teflon is lower than with graphite.

TRAINING

Effective installation of the newer packing materials by maintenance teams requires proper training and documentation (Darlington-A NGS, 1987). The new materials handle differently and are much less forgiving than asbestos. This will often require close cooperation with the packing supplier, station maintenance, operation and engineering. This can be done. Bridges can be built. A good example is the packing program at Ontario Hydro's Bruce-B NGS. The Bruce-B maintenance support group worked closely with the packing supplier and AECL Research, Chalk River, from the start resulting in only three minor leakers out of 2,500 valves repacked. A recent peer audit of the plant stated that ". . . the packing program at Bruce-B is good enough that people should come from all over the world to look at it."

R&D SUPPORT TO FIELD PROBLEMS

With any emerging technology, R&D support is critical to provide a degree of confidence that the directions taken are correct. AECL Research, Chalk River, have worked very closely with the Bruce-B valve program, providing the required R&D support to address their concerns.

The atmospheric steam dump valves (ASDV) at Bruce-B had a history of chronic leakage during operation. Several different packing arrangements had been tried, with limited success. The valves were repacked with combination graphite/triple composite sets, as recommended by the packing supplier. After installation, the valves were checked using available diagnostic equipment and the friction was lower than expected. This raised the question, "Was the gland load of 26.2 MPa (3,800 psi) too light or was the friction being measured correct?" AECL responded by testing the triple composite, confirming that the friction values measured were realistic and that the valve should not develop sealing problems.

Table 1 summarizes the results of the friction study. For a 25.4 μm (1 inch) stem, a reduction of 2.5 times in stem friction is available using triple composites instead of braided end rings, without a loss in leakage performance. However, care must be exercised when using triple composites during installation, due to the limited sealing area of the packing set, (composite end rings do not provide stem sealing and should be live-loaded in control valve applications).

DOCUMENTATION

The lack of proper documentation on valve packing and valve datasheets is a problem for maintenance support groups. This is being addressed using various databases to input and document what is in the valve. These programs are an integral part of a live-loaded or conventionally-loaded valve. The database tracks such things as change-outs, actuator setup, packing arrangement, purchase lot number, the valve part numbers and major design dimensions (Williams, 1979). Figure 6 shows a typical datasheet processed by Bruce-B maintenance support. If something does go wrong, the information can be quickly accessed and corrective action taken. Good documentation provides constant methodologies and maintenance practices.

Table 1. Summary of temperature effects on packing friction for selected combination graphite packing sets.

PKG SET (top to bottom)	25.4 mm stem Temperature (°C)	25.4 mm stem Friction kN (lbf)	50.8 mm stem Temperature (°C)	50.8 mm stem Friction kN (lbf)
yarn x 3 die-formed x yarn	62	6.89 (1550)	59	11.60 (2607)
	297	5.88 (1321)	298	9.32 (2096)
comp x 3 die-formed x comp	58	5.98 (1344)	56	11.43 (2570)
	299	3.91 (880)	298	7.55 (1697)
2 comp x 2 die-formed x comp	58	3.10 (696)	51	7.13 (1602)
	298	2.28 (513)	298	5.58 (1254)

1 kN = 224.81 lbf

Applied gland load is 26.2 MPa (3,800 psi), live-loaded, operating pressure 8.62 MPa (1,250 psi) at 35°C to 300°C (average cooldown temperature is shown).

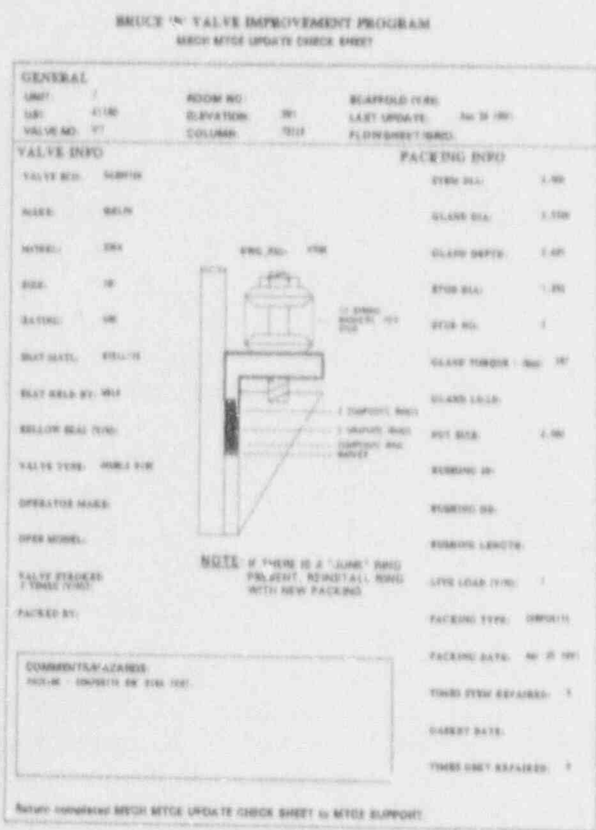


Figure 6. Valve datasheet Bruce-B NGS.

Documentation of valves also includes diagnostics of valve performance. Commercially available valve actuator diagnostic equipment provides unique opportunities for the valve engineer to improve plant availability. Another quality check is to perform a pressure check of the packing material for critical service valves (Aikin et al., 1991). The pressure fluid should be incompressible and compatible with system conditions.

ANGLE AND HORIZONTAL VALVES

Live-loading angle and horizontally installed valves (>6 in.) present some potential problems. The main problem is that die-formed packing does not have the radial stability or strength to support the weight of the stem and gate, plus forces from disc tilt. This causes the packing to form an oval shape, creating a leak path and potential rapid loss of packing material. This was less of a problem with asbestos because the asbestos rings, when loaded and subjected to temperature and pressure, formed a very hard bearing. The old valve designs depended on this bearing for stem support.

To address this problem, for new and old valves, where practical the clearances at the stuffing shoulder and gland follower should be increased to reduce the risk of stem contact and galling. The junk rings should be machined of ASTM A494 Grade CY5SnBiM (Waukasha 88) or engineering equivalent material. Waukasha 88 is a high nickel-based alloy that has demonstrated excellent anti-galling properties (Wensel, 1979). The carbon bushings should be close-fitting to both the stem outside and stuffing box inside diameter. A recommended graphite packing arrangement for a large, horizontally-installed, double-packed gate valve is shown below (reference from the bottom of the stuffing box):

1. One Waukasha 88 ring
2. One composite or braided end ring
3. Two die-formed rings (three if there is room)
4. One composite or braided end ring
5. One lantern ring
6. One composite or braided end ring
7. One die-formed end ring
8. One composite or braided end ring
9. One Waukasha 88 ring.

Note: Carbon bushings can tolerate very little side loading. A cracked carbon bushing will not damage the stem, but can act as an effective wedge during stem actuation and stall the actuator. Carbon bushings can be used instead of Waukasha 88 rings, for valves <6 in. However, care must be taken during measuring and installation. With proper care and good installation practice, the carbon bushings will provide long service.

REMARKS

Successful application of live-loading to improve valve leakage comes not from the calculations, but from

- Good training of maintenance crews, leading to increased ownership
- Clear product and performance specification, leading to improved plant availability
- A combination of good maintenance, procurement, and documentation
- A commitment from management to improve the quality of plant operation
- A commitment from the valve and packing industry to regulate their own industry by specifying performance qualifications.

Live-loading and packing improvements have evolved through cooperation among the valve suppliers, packing manufacturers, maintenance teams and support groups. The net result is better plant performance and a realization that teamwork is an effective ingredient in developing constructive solutions.

ACKNOWLEDGMENT

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Session 3B

Check Valve Performance and Testing

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Fermi II Check Valve Program

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ABSTRACT

In response to Nuclear Regulatory Commission and Institute of Nuclear Power Operations concerns in the area of check valve degradation, Fermi 2 Engineering and Production personnel have formulated and implemented a comprehensive check valve program. Prior to the issuance of Significant Operating Experience Report (SOER) 86-03, check valve testing and inspections were implemented through the inservice testing (IST) program. Upon issuance of this SOER, the check valve program was expanded beyond the original IST scope with an extensive review of check valves susceptible to degradation or known as poor performers. This program was originally based around the guidelines of Electric Power Research Institute Report NP-5479 "Application Guideline for Check Valves in Nuclear Power Plants."

A Design Review was performed which encompassed a review of Nuclear Plant Reliability Data System data, operational experiences, maintenance history, valve failures and design application concerns.

The results of this review were incorporated into the existing Inservice Inspection and Performance Monitoring program. This program is a combination of established performance testing, and visual internal examination with historical photographic documentation. These methodologies have enabled optimization of inspection cycles by lengthening inspection intervals based on sound technical documentation.

The check valve program is an ongoing, living program based on factoring in current industry events, in house performance, and the potential use of nonintrusive inspection technologies.

FERMI'S REVIEW OF INDUSTRY ISSUES

INPO SOER 86-03

The FERMI check valve program evolved beyond the IST Program through a review using the EPRI Application Guideline for Check Valves in Nuclear Power Plants. This initial review of the check valves involved included a design review in addition to a review of Nuclear Plant Reliability Data System (NPRDS), maintenance and operational history. The Design review was divided into those NPRDS systems shown to be experiencing the majority of failures according to SOER 85-03. The number of failures identified in the industry was shown to be small when compared to the total population of check valves in the respective systems. However, FERMI looked at all the systems applicable to its boiling water reactor (BWR) design, as recommended in the SOER. The approach to this design review was divided into two phases, as shown in Table 1.

Table 1. Design review outline.

NPRDS System	Reactor Type	Valve Failures	FERMI 2 System / Description	Review Phase
Main Steam	ALL	39	B21 / Nuclear Boiler Systems	I
Nuclear Service Water	ALL	26	P44 /Emergency Equipment Cooling Water	II
			P45 / Emergency Equipment Service Water	II
Auxiliary Feedwater	PWR	19	n/a	n/a
Diesel Air Start	ALL	8	R30 / Emergency Diesel Generators (Air Start and Cooling Water)	II
Suppression Pool Support	BWR	8	E1151 / Residual Heat Removal Service Water	I
Chemical and Volume Control	PWR	8	n/a	n/a
Feedwater	ALL	6	B21 / Nuclear Boiler Systems	I
Residual Heat Removal and Low Pressure Injection Systems	ALL	6	E11 / Residual Heat Removal System	I
			E21 / Core Spray	II
In Addition to the Above Identified Systems			E41 / High Pressure Core Injection	I
			E51 / Reactor Core Isolation Cooling	II

The original comparison of Fermi's BWR plant design to other units showed the possibility of between 100 to 150 valves susceptible to degradation based on the design review. Both phases of the design review were completed in 1989 and resulted in 94 valves falling into this category (39 in Phase I, 55 in Phase II). In addition to the valves identified as failures under the NPRDS system, other identified problem check valves as contained in Fermi's Licensing Event Report and Deviation Event Report processes were categorized by failure mode, number of failures and actions required/taken to prevent recurrence were reviewed. These valves were then factored into the program accordingly. Check valves used for primary containment or required for leak-tightness were identified and their performance reviewed for additional degradation concerns with respect to further monitoring. Additionally, review of past design modifications which had been implemented to improve check valve reliability was factored into the Design Review. A review for high maintenance check valves was also performed which made use of maintenance work history. Subsequently, surveillance testing history was reviewed for difficulties encountered in the testing, and this information was also factored into the program. The Engineering aspects factored into the review consisted of those critical parameters as identified in the Applications Guideline, such as the following:

1. Minimum Flow
2. Orientation
3. Flow turbulence
4. Erosion
5. Retention and locking of internal parts
6. Other design considerations.

NRC Generic Letter 89-04

When the NRC "Guidance on Developing Acceptable Inservice Testing Programs" (Generic Letter 89-04) was issued, FERMI's IST Program was reviewed in light of the guidance contained in the Generic Letter. Those portions that addressed check valves were reviewed relative to the current check valve program and testing/inspections were performed. As a result of this review, two sets of check valves were identified that used partial stroke actuators to satisfy full stroke testing requirements. This resulted in the current program requirements being expanded for both sets of check valves.

PROGRAMMATIC ACTIONS

Nuclear Engineering

In 1987, a flow loop was established at the Detroit Edison's Engineering Research Department that contained check valves and motor operated valves. Flow loop testing of nonintrusive diagnostic testing systems for check valves was started in early 1988. At the conclusion of the flow loop testing, it was found that the tested diagnostic equipment was prone to technical difficulties and unable to perform as promoted by the vendors. Further, the use of these diagnostic systems required a high level of expertise for conducting and interpreting the testing results.

Various actions were taken upon the implementation of the check valve program. A revision of the valve purchase specifications to include application guideline recommendations was implemented. Design and modification packages were generated by Engineering which resulted in enhancements for improving the performance of check valves and precluding degradation where implemented. To further coordinate site awareness and participation in the Check Valve Program, a program booklet that included history and pertinent overview documentation of the Fermi Check Valve Program was assembled and issued to the various site organizations.

Fermi is credited as being one of the founding utilities of the Nuclear Industry Check Valve Group, "NIC." By participation in NIC, Fermi has provided direct input into the update of the EPRI/INPO guidelines, Generic Letter 89-04, and industry flow loop testing of nonintrusive check valve diagnostic systems. Additionally, Fermi has been the focal point for the development of an industry maintenance guideline.

Nuclear Production Technical Engineering

The administration of the Check Valve Program currently falls under the Inservie Inspection/Plant Performance Evaluation Program (ISI/PEP) Engineering organization, which coordinates testing and physical inspection activities. Where check valves are not disassembled for inspection on a regular basis, as required by SOER 86-03 or IST Programs, comprehensive testing is performed to ensure IST Program check valves can perform their required function. This testing includes; leak rate testing, the verification of design flow capability, and the ability to prevent back flow. Should any valve exhibit degradation in these areas, a visual inspection may be performed to evaluate the extent of the degradation.

The results from internal inspection required by either degradation identified in testing or by scheduled internal inspections are trended by the ISI/PEP organization to quantify and/or qualify any degradation of the valves internals. Should degradation be identified, corrective action would be taken in accordance with the Fermi corrective action process.

The ISI/PEP organization is responsible for obtaining, correlating, reviewing and trending check valve condition. For further enhancement in the ability to perform these activities, the use of photographs of disassembled check valve internals has been pursued. This use of pictures to document the condition of check valve internals is being employed to fine tune the inspection intervals. For those valves in which degradation is not evident, the inspection interval will be relaxed. For those valves in which degradation is evident, inspections may become more focused with the net result being better system performance and reliability.

Nuclear Production Maintenance

From the Program's onset, critical parameters were identified for use during the maintenance of check valves. As part of a joint effort between Engineering and Maintenance personnel this information was placed into the check valve maintenance procedures. This information, coupled with maintenance training on known potential causes of check valve degradation, has allowed for the effective disassembly and performance of check valve inspections. This inspection effort is an augmentation of the inspections performed by ISI/PEP personnel.

In an effort to pinpoint the root cause for identified check valve degradations, Fermi actively participates in and supports the inclusion of *all* check valve deficiencies into the NPRDS data base. While the number of identified failures contained in the data base is comprehensive, the actual number of faulty check valves has been minimal.

PROGRAM IMPLEMENTATION

Implementing Procedures

Where check valve inspections are required, the performance of these inspections are controlled through the use of quantitative *and* qualitative procedures. Portions of the inspections performed by Maintenance personnel include the documentation of both internal dimensions and direct observation of check valve internal condition. The portion of the inspection performed by ISI/PEP personnel is performed with a separate procedure which relies more on the qualitative condition of the valves. The use of both procedures has provided overlap in the information which is used by Nuclear Engineering and ISI/PEP personnel to fine tune the check valve program. One insight garnered by obtaining valve internal dimensions is that the usefulness of the resultant dimensions is of limited value.

Inspection Performance

Check valve inspections are performed to ascertain the degree of degradation of the valves internals where other means are either not possible or reliable. During RF02, additional inspection documentation was gathered through the use of a still video camera. This allowed closer examination of the valve internals in addition to a semi-permanent record for each individual valve photographed.

The use of pictures to document the condition of check valve internals is being used to fine tune the inspection intervals. The usefulness of pictorial documentation has been demonstrated both during and since RF02. The pictures have been particularly useful when discussing potential design modifications, reviewing valve degradation modes and reviewing the condition of check valves with onsite organizations and vendor personnel.

By concentrating the physical inspections of these valves among ISI/PEP personnel, a common thread exists which enables these personnel to better evaluate problems (either potential or actual) with check valves. This practice also allows for a consistent contact for questions on check valve inspections. This is especially helpful to field questions from Nuclear Engineering, Maintenance, Planning and Scheduling or System Engineering.

Design Modifications

Incremental performance enhancing design modifications have been implemented in past outages and their performance is continually being evaluated through inservice testing, LLRTs and valve internal inspections. The evaluation of these modifications has been the catalyst to more advanced upgraded design improvements. This evolution in the design process has allowed us to address the root causes of certain check valve degradations in a systematic design approach to provide optimal solutions. By having one design engineer overseeing this part of the check valve program, consistency in design philosophy, industry involvement and corrective measures being used in the evolution of this program has been possible.

The inspections performed and design modifications implemented as part of this ongoing effort have been effective. Examples of check valve inspection scope and modifications performed are contained below in the following Refueling Outage chronology:

I. Refueling Outage 1—September 1989

A. Inspections

1. 18 valves originally scheduled for inspection
2. 35 total valves inspected due to scheduled scope increase, LLRT performance and surveillance testing.

B. Design Modifications

1. Removed unneeded counter weight on 24-in. LPCI check valve
2. Added antirotation lugs to same 24-in. LPCI check valve to mitigate soft seat wear.

II. Refueling Outage 2 - March 1991

A. Inspections

1. 22 valves originally scheduled for inspection
2. 31 total valves inspected due to scheduled scope, LLRT performance and surveillance testing.

B. Design Modifications

1. Added antirotation lugs to opposite train 24-in. LPCI check valve to mitigate soft seat wear
2. Full stroke actuator modification to both 24-in. LPCI check valves to mitigate soft seat wear experienced during system warmup
3. Existing soft seat retaining ring disc modified on the 4-in. Reactor Water Cleanup System Primary Containment Isolation Check Valve
4. Actuator Shaft indication removed on same 4-in. Reactor Water Cleanup System Primary Containment Isolation Check Valve
5. Actuator Shaft indication removed on 20-in. Feedwater Primary Containment Isolation Check Valves.

III. Refueling Outage 3 - September 1992 (Scheduled)

A. Inspections

1. 30 valves originally scheduled for inspection

2. Reduced original scope to 20 valves scheduled for inspection based on review of previous inspection documentation. This includes a scheduled scope increase of 6 valves that were added due to RF02 inspection results.

B. Design Modifications

1. Contingency design for operational logic change for 24-in. LPCI check valves

2. Disc shaft modification to enhance disc indication for the 4-in. Reactor Water Cleanup System Primary Containment Isolation Check Valve

3. Disc material change for service water check valves to mitigate corrosion degradation.

CONCLUSION

The Fermi Check Valve Program is an iterative process which constantly factors back into the program industry experience, inspection and equipment performance test results, and the review and evaluation of new diagnostic techniques. Further, our continued proactive participation in industry groups (such as NIC) serves to strengthen our program.

This overall program approach to date has proven to be very cost effective. Program success has been accomplished and assured by Fermi 2 Engineering and Production personnel formulating and implementing a comprehensive check valve program which continues to evolve through the execution of the program by onsite organizations.

Alternate Position Testing of Check Valves

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ABSTRACT

This paper addresses several different methods of testing currently used on check valves. The capabilities and limitations of each test method are discussed. It demonstrates that verifying a full-stroke exercise to both the open position and the closed position substantially increases the information about the condition of a check valve over what could be obtained by exercising it to just one position. The merits of periodic check valve disassembly and examination and/or nonintrusive diagnostic examinations are also discussed. Inservice testing (IST) of safety related pumps and valves at commercial nuclear power plants is performed in accordance with the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section XI. The purpose of IST is to assess the operational readiness of pumps and valves that perform a safety function. Assessing operational readiness involves determining if a component is capable of performing its intended function. However, performing a function once provides little confidence that the component will be capable of performing the function in the future. Therefore, the testing should also monitor component condition to detect degradation that may lead to future failure. Because the testing currently performed on most check valves in plant IST programs is incapable of detecting some failure modes, a failed valve could go undetected for a period of time. The testing is also inadequate for detecting check valve degradation mechanisms that could lead to failure. Therefore, the testing cannot adequately assess operational readiness for these valves and the test requirements should be augmented.

INTRODUCTION

The purpose of inservice testing (IST) at commercial nuclear facilities is to assess the operational readiness of pumps and valves that perform a specific function in shutting down a reactor to the safe shutdown condition, in maintaining the safe shutdown condition, or in mitigating the consequences of an accident. To assess operational readiness, IST primarily verifies that each component is capable of performing its intended function(s) at the time of the test. If the component is incapable of performing its function(s) during the test, it is in a failed condition and should be repaired or replaced as appropriate. However, performing a function during a test provides little confidence that the component will remain capable of performing the function at a later date if called upon to do so. Therefore, testing should also monitor component condition to detect degradation that may prevent or hinder subsequent operation.

The IST of check valves performed in accordance with the requirements of Section XI of The American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel (B&PV) Code may be incapable of detecting certain failure modes. This inability could permit continued plant operation with valves that have suffered some type of failure. The testing is also incapable of detecting most check valve degradation mechanisms that could lead to failure. Therefore, the testing cannot adequately assess operational readiness for these valves and the test requirements should be augmented.

This paper addresses the testing currently required for check valves. It identifies the failure modes that can be detected by the required testing and discusses how augmented testing can detect additional failure modes. The paper examines the detection of valve degradation and describes which testing methods can detect the various types of degradation before valve failure occurs.

REQUIRED TESTING OF CHECK VALVES

Section XI, Paragraph IWV-3522(a) states in part: "Valves that are normally open during plant operation and whose function is to prevent reversed flow shall be tested in a manner that proves that the disk travels to the seat promptly on cessation or reversal of flow." Paragraph IWV-3522(b) states: "Valves that are normally closed during plant operation and whose function is to open on reversal of pressure differential shall be tested by proving that the disk moves promptly away from the seat. . . ." These Code testing requirements are somewhat ambiguous in that it is not clear if a valve is required to be verified in its normal position prior to exercising. For example, is it necessary to perform a test to verify that a normally closed valve is in fact closed prior to exercising it open by passing flow through the valve? The common practice is to assume, rather than verify, that a valve is in its normal position immediately prior to exercising it to the other position. In most cases this assumption is justified for normally open valves because these check valves are maintained in the open position by system flow. There may or not be a differential pressure across normally closed check valves, and they could flutter open or be stuck in the open position. Therefore, passing flow through normally closed check valves to exercise them open may or may not verify that the disk moves away from the seat as required by Paragraph IWV-3522(b). Unless these valves are verified in the closed position by testing or observation of position indication or system parameter prior to exercising, there is no assurance that the valves are actually exercised, and the testing provides little or no information about valve condition.

Paragraphs IWV-3522(a) and -3522(b) both use the word "promptly" when defining the required exercise of check valves. The use of this term implies observing or verifying free unhindered disk motion during the check valve exercise. This suggests a test concerned with valve condition rather than just verifying that the valve is capable of performing its function one time.

Another area of confusion is whether the exercise test identified in Paragraphs IWV-3522(a) or -3522(b) is the only exercise test required for a valve even if it performs a safety function in both the open and closed positions. Paragraph IWV-3522 states: "Check valves shall be exercised to the position required to fulfill their function. . . ." which implies that a valve's function should determine the required testing and not solely its normal position. The NRC indicated their position on this issue in Generic Letter 89-04, "Guidance on Developing Acceptable Inservice Testing Programs" (GL 89-04), which states that check valves are to be exercised to the positions in which they perform their safety functions. ASME OM Code-1990, Subsection ISTC, Paragraph ISTC 4.5.2(a) requires each check valve to be exercised or examined in a manner that verifies obturator travel to the closed, full-open, or partially open position required to fulfill its function.

Based on the above requirements and interpretations, check valves should be tested as indicated in Table 1. Table 1 lists passive check valves because they are mentioned in Note 1 of Section XI, Table IWV-3700-1. However, few, if any, check valves can meet the definition of passive valves. Table 1 indicates that no testing is required for normally closed passive valves, however, it is the author's opinion that they should be periodically verified in the closed position by testing or observation.

DETECTING CHECK VALVE FAILURES

Failure modes for check valves are the possible ways a valve can fail to perform its functions. Due to their relatively simple manner of operation and construction, check valves are normally subject to only a few basic failure modes. These modes are plugged, failure to open, failure to

Table 1. Required testing of check valves.

Normal posit[illegible]	Safety position	Active or passive	Required testing
Open	Open	Active	Full-stroke exercise open
Open	Closed Cat. C	Active	Verify reverse flow closure
Open	Closed Cat. A	Active	Verify closure & leak test
Open	Open & Closed Cat. C	Active	Full-stroke exercise open & verify reverse flow closure
Open	Open & Closed Cat. A	Active	Full-stroke exercise open & verify closure & leak test
Closed	Open	Active	Full-stroke exercise open
Closed	Closed Cat. C	Active	Verify reverse flow closure
Closed	Closed Cat. C	Passive	None
Closed	Closed Cat. A	Active	Verify closure & leak test
Closed	Closed Cat. A	Passive	Leak rate test
Closed	Open & Closed Cat. C	Active	Full-stroke exercise open & verify reverse flow closure
Closed	Closed Cat A	Active	Full-stroke exercise open & verify closure & leak test

GENERAL NOTES:

1. Valve safety position is Closed Category A for valves whose leakage rate is limited to a specific amount to meet its closed safety function.
2. Valve safety position is Closed Category C for valves whose leakage rate is not limited to a specific amount to meet its closed safety function.
3. Valve full-stroke exercise involves verifying the valve fully open or verifying maximum required accident flow per GL 89-04, Attachment 1, Position 1.

close, internal leakage, and external leakage. There are several possible mechanisms that could cause each of these failure modes. The failure mechanisms will be identified and discussed later. Since external leakage falls under the purview of the inservice inspection (ISI) program, it will not be examined in this paper. Also, plugging will not be examined in this paper since it occurs very infrequently.[a] The failure to open will be examined in two levels of severity: (a) the inability to pass flow and (b) the inability to full-stroke open with flow. A full-stroke is defined as either

a. Mean failure rate of 5.0E-09/H (Eide, Chmielewski, and Swantz, 1990).

exercised to the open stop or opened sufficiently to allow the maximum required accident flow rate. The failure to close will also be examined in two levels of severity, (a) the inability to prevent gross back flow and (b) excessive seat leakage.

Each of the test or evaluation methods commonly employed for check valves at nuclear facilities can detect one or more of these failure modes. However, no single method is capable of detecting all of these modes. Table 2 lists each general test and evaluation method and identifies its ability to detect the four check valve failure modes being discussed.

Because none of the individual test methods can detect all four of the check valve failure modes being considered, a valve tested using only one of these tests could have one or more undetected failures. For example, a valve that only receives a full-stroke exercise test could have the valve disk separate from the swing arm, and this failure could go undetected for extended periods of plant operation. This type of failure has occurred at several reactor plants (e.g., the San Onofre Unit 1 feed and condensate system valves, the Dresden Unit 1 diesel raw water cooling check valves, and the Sequoyah Unit 1 main steam isolation check valves). Another example of undetected failure would be passive check valves that are only leak rate tested. The disk of one of these valves could stick in the closed position, and this failure could remain undetected until there is an attempt to establish flow through the valve.

Subjecting a valve to a full-stroke exercise open and one or more test in the closed direction can detect all pertinent failure modes. Therefore, performing alternate position testing in conjunction with exercising check valves to their safety function position provides reasonable assurance that there is no latent valve failure. This testing may also detect some types of degradation.

Alternate position testing would be additional testing for most licensees of commercial nuclear facilities. Testing methods and procedures would have to be developed for valves that are currently tested or verified in only one position. Additional tests would involve normally closed check valves whose only safety function is to open and normally open valves whose only safety function is to remain open. Several utilities already test check valves to both positions. This practice eliminates the need for engineering evaluations to determine the safety function position(s) of each check valve in the IST program. Other advantages to alternate position testing may partially offset the expense of the additional testing. Because one failure mode can indicate degradation that could cause or contribute to other failures, identifying a valve failure is important even if the valve can still perform its safety function(s). Detecting the non-fatal failure can allow the licensee to perform corrective maintenance on the valve at a convenient time before it degrades to where it cannot perform its safety function. Further, some failures result in loose valve parts and components in the system. Recognizing this failure early could permit corrective action before these items damage other system components. A valve that is so degraded that it cannot pass alternate position testing should not be relied on to perform its function in its safety-related position.

Based on the above reasons, alternate position testing is highly recommended for all active check valves in IST programs. Passive check valves (a very small population because very few check valves meet the criteria for passive valves) should not be affected by alternate position testing because they are periodically verified in their safety position and are not moved from that position during any operating mode where they are required to accomplish their safety function. Valves performing safety functions in both positions would not be affected by this recommendation because they must be tested to both positions under GL 89-04 and the ASME OM Code-1990, Subsection ISTC. Therefore, a Code change to implement alternate position testing should not be overly burdensome to licensees and would greatly enhance their ability to detect degradation and latent failure of the affected valves.

Table 2. Effectiveness of test methods at detecting check valve failure modes.

	Stuck closed	Full-stroke is restricted	Fails to block reverse flow	Excessive seat leakage
Part-stroke open with flow	Yes	No	No	No
Full-stroke open with flow	Yes	Yes	No	No
Verify reverse closure	No	No	Yes	No
Leak rate test valve	No	No	Yes	Yes
Disassemble and Inspect[a]	Yes	Probably	Probably	No
Nonintrusive Diagnostics[b]	Yes	Probably	Yes	Possible
External Mechanical Stroke[c]	Yes	Yes	Probably	No

NOTES:

a. The capabilities of disassembly, inspection, and manual exercise are affected by the thoroughness and rigor of the inspection and manual exercise. The seating surfaces can be examined for corrosion, damage, and wear, but leakage rates cannot be determined.

b. Diagnostics vary in their abilities to detect failure modes depending on the particular valve, system, equipment calibration, accuracy of baseline data, and technique employed. In some cases it may produce results better than indicated above (i.e., some techniques can identify leakage past a valve seat, however, the leakage rate cannot be quantified accurately).

c. It is assumed that force or torque is measured as required for the external mechanical stroke test.

DETECTING FAILURE MECHANISMS

Testing that detects all possible valve failure modes should greatly improve the reliability of check valves used in nuclear facilities. However, this testing may not provide the ability to anticipate or predict valve failure so that a valve could be repaired before actual failure occurs. To anticipate valve failure, it is necessary to use predictive testing or maintenance that detects and monitors for valve failure mechanisms or degradation. This testing or maintenance requires measuring and trending parameters that indicate valve condition and assigning acceptance criteria that are based on appropriate margins to assure valve operability. To evaluate the effectiveness of testing methods to monitor for valve degradation, it is necessary to determine which of the possible failure mechanisms each test can detect.

Because of the diverse types, designs, and manufacturers of check valves and the variety of applications in which they are used, it would be difficult to make a comprehensive list of check valve failure mechanisms. Some of the mechanisms that could cause a check valve to fail are listed below. Because the mechanisms would be very similar for the open failure modes, the "open" failure mechanisms are grouped together. Likewise, the failure mechanisms for the closed failure modes are grouped together. The severity of the degradation, the affected valve part, and the type and location of blockage would determine which of the failure modes would result. For example, minor corrosion of a valve disk or seat could result in excessive seat leakage, while more extensive corrosion of the valve internals could freeze the valve in its normal position and prevent it from changing position in response to system pressure or flow.

OPEN FAILURE MECHANISMS

1. Mechanical blockage of moving parts
2. Corrosion of valve disk or moving parts
3. Excessive wear of moving parts
4. Improper assembly of the valve
5. Fouling of valve internals by marine growth
6. Chemical precipitate buildup in or around the valve
7. Metal fatigue failure of valve internals
8. Stress corrosion failure of valve internals
9. Excessive erosion of valve moving parts.

CLOSED FAILURE MECHANISMS

1. Mechanical blockage
2. Corrosion of valve internals
3. Excessive wear of valve moving parts
4. Incorrect assembly of the valve
5. Fouling of valve internals by marine growth
6. Chemical precipitate buildup in the valve
7. Metal fatigue failure of valve internals
8. Stress corrosion failure of valve internals
9. Erosion of valve seating surfaces or moving parts
10. Impact damage to valve seating surfaces.

One or more of these mechanisms can be present in a valve for a period of time before the valve degrades to the point that it cannot perform its safety function(s) and is considered failed. This period can vary from almost no time to dozens of years depending on the type of degradation and the rate of progression. To anticipate valve failure, it would be necessary to monitor and trend the rate of valve degradation. It would then be necessary to determine how much of the various forms of degradation can occur in a valve before its ability to perform its intended function(s) becomes questionable. Quantifying valve degradation, trending its progression, and assigning appropriate acceptance criteria could result in corrective maintenance of a valve before the valve fails or degrades to the point where failure is imminent.

Many of the used test methods cannot detect valve degradation from the above mechanisms. Even when a seriously degraded valve, whose failure is imminent, is subjected to Code testing, the tests may not indicate the valve's impending failure. A forward flow exercise gives little or no information about valve closure failure mechanisms. For example, a valve's disk could be loosely attached to the swing arm and the valve would pass a forward flow exercise test with ease; however, misalignment might prevent the valve from closing to prevent reverse flow. Likewise, reverse flow closure testing by itself provides little or no information about degradation affecting a valve's open function. A leak rate test with tight limits and trending of test data provides a more objective indication of valve condition and can detect degradation from most closed mechanisms and some open mechanisms.

DETECTING PRE-FAILURE DEGRADATION AND DETERMINING MARGIN TO FAILURE

Valve degradation normally occurs gradually over time. In its early stages most degradation does not affect valve operation. However, as it worsens, it will impair valve operation and eventually result in failure. Small changes in a valve's ability to stroke open or closed may not be detected by valve surveillance testing as long as the test acceptance criteria are satisfied. This is because the full-stroke exercise open and leakage rate tests are essentially go/no-go tests that are not sensitive to degradation (i.e., the flow/leakage rate is either above or below the specified criteria and the results are generally not trended). Additionally, specific acceptance criteria are generally not identified for the part-stroke exercise open and reverse flow closure tests. Therefore, valve degradation would continue undetected until the valve is declared inoperable as a result of not passing a test or failing to perform its function when called upon to do so. Permitting degradation to remain undetected in safety-related check valves is undesirable. Steps should be taken to detect degradation before valve failure.

Tables 3 and 4 present information on the capability of various test and evaluation methods to detect valve degradation before actual failure. These tables indicate the ability of these tests to detect each of the open and closed failure mechanisms listed in the previous section. Check valve diagnostics is a relatively new field that is rapidly evolving. The capabilities of existing techniques are improving, and new techniques are being developed. Therefore, diagnostics may now or in the near future detect more than indicated in these tables.

As shown in Tables 3 and 4, exercising with flow and reverse flow closure verification does not allow detection of degradation from most of the identified mechanisms. If degradation cannot be detected and the margin cannot be determined between the valve's condition and the point where the valve can no longer perform its safety function(s), this testing cannot provide any assurance of the valve's future operability. As such, the testing normally performed to meet IST requirements does not allow detection of valve degradation and, therefore, does not provide confidence in a valve's continued ability to perform its function. These tests could be effective in determining the likelihood of future valve operability only if additional measurements are made in conjunction with the flow test (i.e., measure and trend the differential pressure across the valve with a repeatable flow rate through it).

Several tests and evaluation techniques considered in the table can detect certain degradation mechanisms and, to some extent, assess the margin between valve condition and where failure is likely. The various test methods and their strengths and weaknesses are discussed below.

Reverse Flow Closure Test

The reverse flow closure test generally cannot detect check valve degradation. Even though this test is similar to the leakage rate test, degradation may not be detected and monitored because the test does not usually involve actual measurements or specific acceptance criteria. The acceptance criteria are generally very subjective and left to the judgement of the individuals performing the test. The reverse flow closure verification test procedures normally use criteria such as "observe little or no flow," "observe less than a solid stream of water," and "verify that the valve checks flow." These subjective observations and criteria not only prevent detection of valve degradation, they may permit a severely degraded valve to remain in service (i.e., a valve at San Onofre Unit 1 passed a reverse flow closure test with the disk totally separated from the swing arm). Further, these observations and criteria may be interpreted differently by the various individuals performing the testing. The reverse flow closure verification need not determine a leakage rate or involve a great deal of rigor, however, unless this is done, it is unlikely that check valve degradation can be detected using this test method.

Leakage Rate Testing

Leakage rate testing performed in accordance with the requirements of Section XI, Paragraphs IWV-3421 through -3427 and/or 10 CFR 50, Appendix J, Type C [including Section XI, Paragraphs IWV-3426 and -3427(a)], can effectively detect many of the open and all of the closed failure mechanisms. However, for this method to detect degradation, either the acceptance criteria would have to be fairly restrictive or the test data carefully trended.

Disassembly and Inspection

This method is normally used in lieu of testing in cases where it is impractical to full-stroke exercise a valve with flow because sufficient flow through the valve either cannot be achieved or

Table 3. Effectiveness of test methods at detecting check valve open failure mechanisms.

Failure mechanism	Open 1	Open 2	Open 3	Open 4	Open 5	Open 6	Open 7	Open 8	Open 9
Part-stroke exercise open	N	N	N	N	N	N	N	N	N
Full-stroke exercise open	N	N	N	N	N	N	N	N	N
Verify reverse flow closure	N	N	N	N	N	N	N	N	N
Perform leakage rate test	N	N	P	P	P	P	P	P	P
Disassemble and inspect	Y	Y	Y	Y	Y	Y	Y	Y	Y
Nonintrusive diagnostics	P	P	Y	P	P	P	P	P	Y
External mechanical stroke	P	P	P	P	P	P	P	P	P
Examine using borescope	Y	Y	P	P	Y	Y	P	P	P

Table 4. Effectiveness of test methods at detecting check closed valve failure mechanisms.

Failure mechanism	Closed 1	Closed 2	Closed 3	Closed 4	Closed 5	Closed 6	Closed 7	Closed 8	Closed 9	Closed 10
Part-stroke exercise open	N	N	N	N	N	N	N	N	N	N
Full-stroke exercise open	N	N	N	N	N	N	N	N	N	N
Verify reverse flow closure	P	P	P	P	P	P	P	P	P	P
Perform leakage rate test	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y
Disassemble and inspect	Y	Y	Y	Y	Y	Y	Y	Y	Y	Y
Nonintrusive diagnostics	P	N	Y	P	N	N	P	P	P	N
External mechanical stroke	P	N	N	P	P	P	P	P	N	N
Examine using boroscope	Y	Y	P	P	Y	Y	P	P	P	N

KEY:

1. "Y" indicates the test or evaluation method should allow detection of degradation before valve failure.
   "N" indicates that it is not likely that the degradation would be detected before valve failure.
   "P" indicates that degradation may be detected depending on the quality and rigor of the test; however, there is not a high level of assurance of detection by the test or evaluation method.

verified. As shown in Tables 3 and 4, disassembly and inspection can be very effective at detecting failure mechanisms in check valves. This method involves partial valve disassembly; inspection of the valve internals for corrosion, erosion, wear, blockage, and structural soundness; and a manual exercise of the disk to verify free movement over its travel. Further, although it is not required, this method allows measurement of valve internal parts to monitor for wear and erosion. These measurements can be trended and evaluated to predict when a high level of confidence no longer exists that the valve can perform its safety function(s).

Because of the effectiveness of disassembly and inspection in determining valve condition and detecting degradation, GL 89-04 identifies this technique as an alternative to the full-stroke exercise required by the Code when testing with flow and mechanical exerciser is determined to be impractical. The Generic Letter also permits use of a sampling technique for nearly identical valves in similar applications. Subsection ISTC of the OM Code-1990 identifies disassembly and examination as an alternative to testing with flow or with a mechanical exerciser. A change to ISTC that permits the use of a sampling program is currently in the final approval process. Disassembly and inspection of check valves is also recommended in the Institute of Nuclear Power Operations (INPO) significant operating experience report (SOER), number 86-03, as one of the methods of examining high-risk check valves in selected nuclear plant safety systems.

In spite of its many capabilities, disassembly and inspection of check valves has drawbacks that may make routine use as a substitute for testing undesirable. After a valve is disassembled for inspection and manual exercise, it must be reassembled. Check valve reassembly presents an opportunity for the introduction of errors, such as misalignment or binding, which could hinder or prevent subsequent valve operation. Since it is impractical to full- or part-stroke exercise many valves following reassembly, an improperly reassembled valve could go undetected. Additionally, certain applications seal weld the caps of check valves to prevent leakage. The seal weld must be ground off to disassemble and inspect the valve. Grinding off the seal weld damages the valve and can be performed only a limited number of times before the valve has to be replaced. Another concern is that for some check valve installations reactor vessel level must be significantly reduced to permit disassembly and inspection. Because of the reduction in the reactor vessel level and coolant inventory, all fuel must be off-loaded from the core before disassembly of these valves. Further, to disassemble and inspect check valves, plant personnel must often work in high radiation areas for extended periods, which results in increased radiation exposures. For these and other similar reasons, the NRC does not encourage disassembly and inspection in lieu of Section XI testing when other methods are practical. However, they do encourage the periodic use of this or other techniques that detect degradation in predictive maintenance programs for check valves.

Nonintrusive Diagnostic Techniques

In the last several years a great deal of attention has been given to nonintrusive check valve diagnostic techniques. This is primarily because these techniques can be used to verify a full-stroke exercise for check valves that previously could only be verified by disassembly and inspection and because the nonintrusive techniques do not have the disadvantages of disassembly and inspection (e.g., the possibility that the valve will be inoperable as the result of improper reassembly). The Nuclear Industry Check Valve Group (NIC) and the Electric Power Research Institute, Inc. (EPRI) have sponsored research and testing of some of these techniques to investigate their ability to study the condition of check valves. Some results of the NIC testing have been published in their report *Evaluation of Nonintrusive Diagnostic Technologies For Check Valves (NIC-01)* (Utah State University Foundation, 1991).

These nonintrusive techniques can indicate check valve condition and detect many of the check valve failure mechanisms. The initial NIC testing involved three different nonintrusive technologies: (a) acoustics, (b) magnetics, and

(c) ultrasonics. Section 1.3 of the NIC report states:

> All three technologies demonstrated the ability to provide useful information about the condition of the internals of check valves. They are all a viable alternative to disassembly and visual inspection for certain aspects of check valve performance and condition when used properly. With baseline data available on a new valve (undegraded) the technologies in general were able to distinguish between a new valve and a valve with degraded internals. They could usually identify the area of degradation, and in a few cases were able to distinguish between a 15 and 30 percent degradation. They could determine if the disc was missing, stuck, or operating normally through its entire stroke. The ultrasonic and magnetic techniques were able to determine the mean disc position and identify the magnitude and frequency of disc flutter. The three technologies were able to detect seat and backstop tapping and movement of the internal parts.

Many other vendors and several new technologies are available besides those tested by NIC. This is an area of intense competition and activity; therefore, the industry can expect continued development and enhancement of the capabilities of nonintrusive techniques.

The NRC frequently recommends the use of nonintrusive diagnostic techniques to demonstrate a full-stroke exercise open of check valves in lieu of disassembly and inspection. The use of these techniques for predictive maintenance is also proposed and generally encouraged by EPRI, NIC, INPO, and the NRC. The ASME O&M Working Group on Check Valves, OM-22, is considering changes to the ISTC Code that will permit the use of nonintrusive diagnostics on a sampling basis in lieu of full-stroke exercise testing. This wide spread acceptance and support is primarily because these techniques provide many check valve testing and diagnostic capabilities without the problems associated with disassembly and examination. However, nonintrusive check valve diagnostics are not a panacea, and there are concerns associated with its implementation.

The following problems may be encountered when implementing nonintrusive diagnostics. The purchase and use of a nonintrusive diagnostic system can be costly and may not yield the desired results. Baseline data must be taken when a valve is in a new or undegraded condition to detect subsequent degradation with a reasonable level of confidence. The personnel who use the equipment must be trained and experienced in using the equipment before satisfactory results can be obtained. The vendor personnel employed in the NIC testing were highly trained and experienced individuals, and they were still unable to access all encountered conditions accurately. Some techniques have limitations and are not effective on certain process mediums, valve types, or valves fabricated out of particular materials. One of the diagnostic technologies requires that each applicable valve be disassembled and modified before the technique can be implemented. Other concerns involve equipment calibration and determining appropriate acceptance criteria or flag levels that indicate impending failure so corrective action can be performed before actual failure. The most effective diagnostic results were obtained when two of the tested technologies were used together; therefore, a facility may wish to purchase or develop a hybrid system to provide enhanced capabilities. Each licensee should consider these concerns when investigating check valve nonintrusive diagnostics and should take adequate steps to ensure that the program they implement will yield the desired results.

External Mechanical Exercise

This test method can detect many of the listed mechanisms. However, since it can be used on only those valves that have external operator connections, its application is limited to an extremely small population of check valves at most facilities. To be effective, it is necessary to measure and trend the force or torque values, both break away and running, required to stroke the valve. The ASME OM Code-19909, Subsection ISTC, currently only requires measurement of the breakaway force or torque. Measurement of the force or torque needed to maintain disk movement would be necessary to detect some failure mechanisms.

Examine Using a Boroscope

This method is very unusual and is included only because it has been considered by at least one licensee for use in certain applications. Examining the valve internals by boroscope may detect many of the listed failure mechanisms. However, unlike disassembly and inspection, it does not allow manual exercise of the disk to verify free movement. Further, this method does not facilitate the measurement of valve internal parts to monitor for wear and erosion. Conversely, since the valve is not disassembled, improper reassembly is not a concern. Another consideration is that this method may require the installation of special examination ports on or near the valve body.

CONCLUSIONS

The Code required testing cannot detect many of the possible check valve failure modes and cannot detect degradation from most of the identified failure mechanisms. Performing alternate position testing of all check valves in the IST program would greatly enhance a licensee's ability to detect check valve failures and determine valve condition. The ASME OM Code-1990 and NRC GL 89-04 require testing of check valves to all safety related positions; therefore, valves that perform both an open and a closed safety function should already be tested to their alternate positions. Normally open check valves that are verified closed during testing are essentially verified in both positions by the currently required testing. Therefore, alternate position testing would mainly affect normally closed valves whose only safety function is to open. Performing a closure verification test on these valves could detect valve degradation or failure that could inhibit or prevent valve closure.

Testing check valves to both positions solely by observing pressures and flow cannot detect many of the check valve degradation mechanisms. Performing other types of testing, such as using a mechanical exerciser, disassembly and inspection, and using nonintrusive techniques, may permit detection of most of these mechanisms. However, to detect degradation and accurately forecast valve failure, these methods must be performed with a high degree of rigor and expertise. This would involve taking accurate measurements and trending data. For nonintrusive techniques it may be necessary to purchase expensive equipment, obtain special training, and take baseline data on valves when they are not degraded. Where these test methods are employed because testing with pressure and flow is impractical, the inclusion of the necessary additional rigor should be considered because of the increased ability to detect degradation and preclude valve failure. Predictive maintenance techniques (i.e., disassembly and inspection and nonintrusive check valve diagnostics) should be incorporated into the maintenance program for all other safety related check valves. The predictive maintenance should be used at an appropriate frequency based on the system, valve type, service conditions, installation, maintenance history, and risk associated with valve failure.

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A Review of Historical Check Valve Failure Data

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ABSTRACT

Check valve operating problems in recent years have resulted in significant operating transients, increased cost, and decreased system availability. In response, additional attention has been given to check valves by utilities, as well as the U.S. Nuclear Regulatory Commission (NRC) and the American Society of Mechanical Engineers Operation and Maintenance Committee. All these organizations have the fundamental goal of ensuring reliable operation of check valves.

A key ingredient of an engineering-oriented reliability improvement effort is a thorough understanding of relevant historical experience. Oak Ridge National Laboratory is currently conducting a detailed review of historical failure data available through the Institute of Nuclear Power Operation's Nuclear Plant Reliability Data System. The focus of the review is on check valve failures that have involved significant degradation of the valve internal parts. A variety of parameters are being considered during the review, including size, age, system of service, method of failure discovery, the affected valve parts, attributed causes, and corrective actions.

This work is being carried out under the auspices of the NRC's Nuclear Plant Aging Research program. At this time, the study is approximately 50% complete. All failure records have been reviewed and categorized, and preliminary tabulation and correlation of data is underway. The bulk of the tabulation and correlation portion of the work is expected to be completed by the end of June 1992. A report draft is expected in the fall of 1992.

INTRODUCTION

Oak Ridge National Laboratory (ORNL) is carrying out a review of historical check valve failure data under the sponsorship of the Nuclear Regulatory Commission's (NRC's) Nuclear Plant Aging Research Program. The study involves the review and characterization of failure records from the Nuclear Plant Reliability Data System (NPRDS) data base. Failures in which significant internals degradation was detected are being characterized in detail. Parameters that are being considered include the age of the plant when the failure occurred, valve size, manufacturer, system of service, method of discovery, affected valve

a. Research sponsored by the Office of Nuclear Regulatory Research, U.S. Nuclear Regulatory Commission under Interagency Agreement DOE 1886-8082-8B with the U.S. Department of Energy under contract No. DE-AC05-840R21400 with the Martin Marietta Energy Systems, Inc.

parts, attributed failure causes, and corrective actions.[b]

BACKGROUND

The American Society of Mechanical Engineers (ASME) Committee on Operation and Maintenance (OM) of nuclear power plants has established a Working Group (WG) on check valves, OM-22, which is chartered with developing check valve performance test requirements. The WG met for the first time in June 1990.

Early on, the OM-22 membership recognized that a thorough understanding of historical failure patterns was critical to several aspects of the code development activities being pursued. A literature search found that while some historical failure data studies had been completed and documented, the studies were normally not oriented toward providing the kinds of information needed in code development activities.

One study that was initially selected by the WG as a basis for consideration in the development of disassembly and examination requirements (note that these requirements would apply only to valves that could not be properly tested) was a paper presented by M. L. Scott at the EPRI Power Plant Valves Symposium II (Scott, 1989). Scott reviewed NPRDS failure records for events occurring from 1985 to 1987. Moderate seat leakage and external leakage events were then eliminated from the data. Failure rate vs. valve size, valve service time, and plant system were discussed. One of the conclusions drawn by Scott was that there was a tendency for a large number of failures relatively soon after installation, followed by a period of fewer failures during the 4- to 9-year service period, and then subsequently followed by a sharp increase in failure occurrences. The sharp increase was attributed to wear-out of the check valves.

As OM-22 deliberated on the establishment of appropriate disassembly and examination intervals, the conclusion in the Scott study regarding the sharp increase in failures beginning at about 9 years was noted. The WG used this study as the basis for formulating requirements for 8-year disassembly and examination limitations for those valves that could not be properly tested.

During WG consideration of the paper and its application to code development, some questions arose concerning the technical validity of the WG's basis. As a result, ORNL was asked to conduct a preliminary review of failure data. This review was conducted by nonqualitatively tabulating NPRDS reported failures and valve populations during the years 1985-1987 (the years of the Scott study), as well as the years 1984-1990.

The preliminary review indicated that the age-related aspects of the study used as the WG's basis appear to have been heavily influenced by the age of plants in operation during the years considered. Figures 1 and 2 illustrate the basis for this observation.

Figure 1 provides comparative plots of the number of valves in service during the period of the study and the failure data from the WG basis study. The similarity of the traces indicates that the failures vs. age trend noted in the WG basis study is strongly affected by the valve population in existence during the study period.

Figure 2 shows comparisons of all check valve failures (regardless of failure nature) and population during same period. It provides further indication of the importance of the valve population to overall valve failure rate.

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b. Author's Note: Some pertinent background behind this study and the study approach are discussed below. Because the characterization of the data is not complete at the time this paper is being written (April 1992), the results cannot be included. It is expected that a significant portion of the characterization will be completed before the NRC/ASME Symposium. The results then available will be presented at the Symposium; updated copies of this paper that reflect our findings will also be made available. For additional information, address inquiries to:

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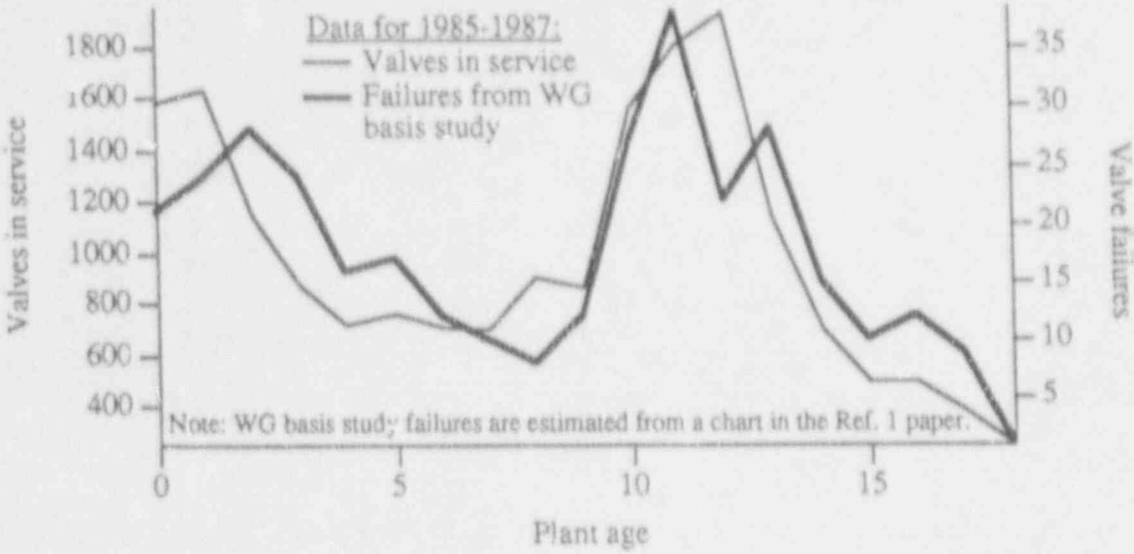

Figure 1. Comparison of failures from Scott paper and valves in service.

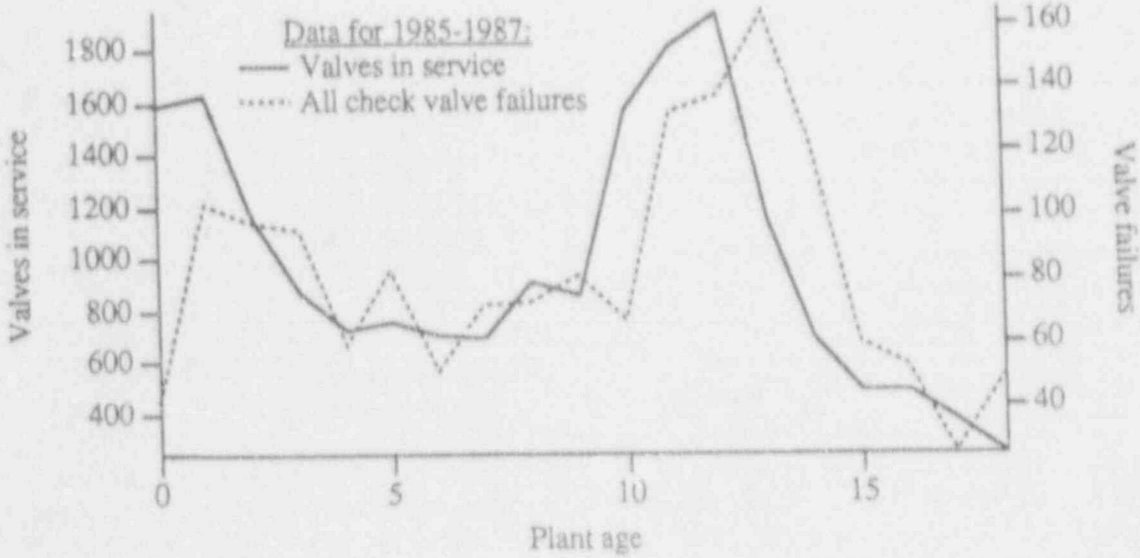

Figure 2. Comparison of all check valve failures and valves in service.

In order to provide a preliminary indication of the nonpopulation influenced valve failure-age relationship, normalized plots of the WG basis study and all check valve failures from 1985 to 1987 are provided in Figure 3. There do not appear to be strong, conclusive trends from the data shown, based on preliminary review. It appears, based on our review of the 1985-87 data and the WG basis study data, that just over 20% of the failures were deemed by the WG basis study to have involved problems other than moderate seat leakage or external leakage.

The results of the preliminary review substantiated concerns about the use of the WG's basis study conclusions for further use in code development activities. The NRC's Nuclear Plant Aging Research program asked ORNL to conduct a more thorough assessment of the historical failure data. The study was initiated in early 1992.

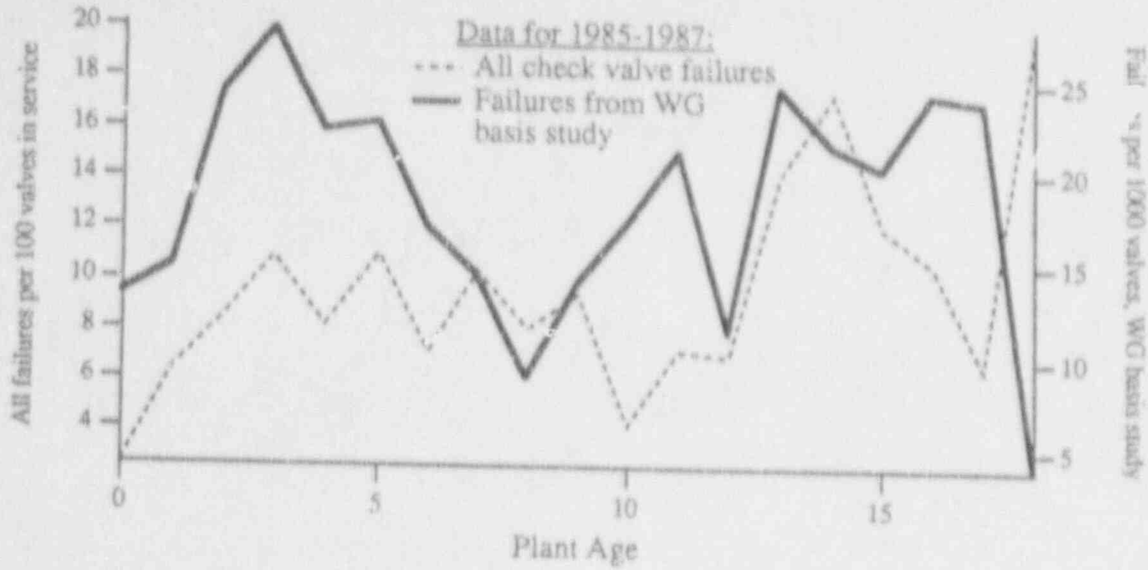

Figure 3. Comparison of normalized failure data.

STUDY METHODOLOGY

A primary study goal is to identify any apparent correlations of valve failure rates with age, size, system of service, and manufacturer. In addition, the study will categorize, to the extent possible, the affected parts of the valve, the method of failure detection, and corrective actions taken.

All failure data were acquired from the NPRDS system. Narratives and other pertinent information for all check valve failures, regardless of size or system of service, were initially downloaded. The data were then filtered to eliminate failures that did not involve significant internals degradation. Minor seat leakage and external leakage events (if these were the only degradations noted) were eliminated from further consideration. The failures that were then further analyzed were those that appeared to involve significant internals degradation. It should be noted that some failures records have minimal information about the nature of the failure, noting only that certain parts were replaced.

It should also be noted that some of the eliminated failures may have made certain valves technically inoperable. For example, minor seat leakage may have been discovered during a containment isolation valve leak test that made the valve technically inoperable. Alternatively, significant external leakage may render certain valves inoperable. However, the primary area of interest in this study is the assessment of check valve failures that involve significant wear or other degradation of valve internal parts. Further, in the cases where minor failures render equipment technically inoperable, the problems could reasonably be expected to be routinely detected by current means (i.e., visual observation of external leakage and seat leakage measurement testing).

After eliminating the nonsignificant failures, it was decided to consider only failures that occurred between 1984 and 1990, inclusively. Failure reporting to NPRDS improved dramatically beginning in 1984, and it appeared that use of prior years' data would not reflect the reporting practices employed thereafter. Failure events occurring in 1991 and afterward were not considered because, at the time the data was downloaded, all failure reports for 1991 were not filed.

The initial data downloaded contained 4,680 failure records, which reflected all check valve failure records for all years (including part of 1991); 3,761 of these failures were detected during 1984-1990. After the preliminary review of the individual narratives and elimination of those that did not involve significant internals degradation, 1,239 failures, or about 33% of the overall failures occurring during the period, remain. This compares with our estimate of slightly over 20% of all failures deemed to have been significant by Scott.

The results of the ORNL study are expected to be published as a NUREG/CR report. A draft of the report is expected in the fall of 1992.

REFERENCES

Scott, M. L., 1989, "Check Valve Failure Trends in the Nuclear Industry, Charlotte, NC, July" *EPRI Power Plant Valves Symposium II.*

Comanche Peak Check Valve Reliability Program

Diane Stewart
Texas Utilities Electric Co.

ABSTRACT

The Comanche Peak Unit 1 Check Valve Reliability Program was implemented in spring 1991, prior to 1RF01, in response to the Institute of Nuclear Power Operations SOER 86-03. The program emphasizes condition-directed disassembly. Candidates are selected by nonintrusive test results, primarily acoustic emission (AE) monitoring. The AE test frequency, initially based on recommendations from the SOER design review, will be refined by feedback from actual test or examination results. To date, 83 valves have been AE tested (baseline of all program valves to be completed by the end of 1RFO2). Out of an original 1RF01 disassembly scope of five valves, three required refurbishment. One necessitated sample expansion to two like valves (which also required refurbishment). The 1RF01 disassembly results provided (a) a caution to plant programs relying only on design review-based sample disassembly (without subsequent refinement), and (b) an early validation of the ability of AE to detect prime disassembly candidates.

INTRODUCTION

Current functional testing [inservice testing (IST)] establishes check valve operational readiness, but not necessarily *reliability*. (Reliability encompasses and extends the timeline of operational readiness.) Recurrent and longstanding industry check valve problems have been due, in part, to the inadequacy of IST to detect internal degradation.[a] Although IST ensures check valve operational readiness under test conditions during a specific snapshot of time, it does not ensure valve actuation under subsequent operating conditions. The intent of Comanche Peak's SOER 86-03 predictive maintenance program (CVRP) is to establish check valve *reliability*.

a. Nuclear Regulatory Commission (NRC) letter to J. DeWease, Chairman, American Society of Mechanical Engineers (ASME) Operation and Maintenance Committee, August 26, 1990.

DESIGN REVIEW BASIS

Kalsi Engineering (Kalsi Engineering, 1989) was contracted by CPSES to perform an SOER 86-03 "Applicability Evaluation" (the second SOER recommendation) of all check valves within the following eleven systems (eight systems recommended by the SOER and three suggested by industry experience, Unit 1 and Common), for a total of 278 check valves:

Auxiliary Feedwater	Main Feedwater
Component Cooling Water	Main Stream
Condensate	Residual Heat
Chemical Volume and Control	Safety Injection
Diesel Generator Air Start	Service Water

Low-use (less than 5% of the plant operating cycle) and small (less than 2 inches in size), non-nuclear safety-related check valves were not included in the complete analysis; therefore, 149 valves out of 278 were analyzed.

The foundation of the CPSES PM program was a design review with the following functions:

- Calculate the analyzed population's potential for wear and fatigue; address misapplication
- Supply results as "sample disassembly" program that groups valves in like service (as defined by IST Generic Letter 89-04)
- Contain preoperational valve use estimates (implying need for reevaluation, especially where severe service is expected).

Of the above 149 valves, *all but 25* were considered suitable for long-term operation: 15 were identified as accelerated wear/fatigue concerns and 10 as disc seating concerns (auxiliary feedwater system). In addition, 18 feedwater check valves were recommended for inspection because of erosion/corrosion susceptibility. (These 18 check valves are indirectly monitored via the site corrosion monitoring program.) Therefore, a total of 43 initial priority valves were identified by the design report.

CVRP POPULATION

The 149 valves analyzed in the design review were adopted as the program population, with emphasis on the 43 identified concerns. Check valves may be added to or subtracted from the program based on nonintrusive testing and disassembly examination results or site maintenance history (to date none have been added or deleted). The program may be used to fulfill IST requirements on check valves outside its population. (See attachment for a list of valves tested to date.)

Three *initial* priority categories were developed, based on Kalsi Engineering wear/fatigue index[b] assessments and other comments provided in the design review, as shown in Table 1.

TESTING/EXAMINATION METHODS

The CVRP is composed of two parts, nonintrusive testing and disassembly examination.

Nonintrusive Testing ("Fault-Finding PMs")

The primary method is acoustic emission (Table 2). The secondary methods are pressure, ultrasonics (UT), and magnetic flux.

b. The wear/fatigue index is a summary form of Kalsi Engineering's CVAP (Check Valve Analysis and Prioritization) software calculations for check valve wear and fatigue. The wear index represents the estimated hinge pin wear of a swing or tilt disc valve. The fatigue index represents the calculated fatigue life of the disc stud, for a swing check valve. The degree to which the wear or fatigue life is consumed is categorized by assigning an index number to each: from 1 = very low (0 - 7% life usage, limit to 14 plant cycles) to 5 = very high (>56% usage, limit to <2 plant cycles).

Table 1. CVRP program priority.

Design review results		CVRP priority (initial)	Subtotals for priority
Wear/fatigue index	Other comments		
4,5	43 Recommended exam priorities	A	43
2,3	Otherwise lower priority valves that were identified as possibly tapping at normal flow	B	20
1		C	86

Table 2. CVRP AE equipment.

CVRP data acquisition equipment

- TEAC RD-101T (or equivalent) PCM Data Recorder, 4-channels
- PCB Piezotronics 6-channel amplifying power unit, P/N 483A11, with continuous adjustable gain 0-100 (use of gain equivalency charts from QUICKCHECK User's Manual assures compatibility of data collection method with QUICKCHECK software)
- Accelerometers:

  VibraMetrics 6036 Century

  VibraMetrics 6022

  1/4-28 stud, to fit sensor mounts

  BNC-connector, to match power supply

  For standard temperature applications, mounts are affixed semipermanently with epoxy (approved consumable) to external surface of valve body.

  For high temperature applications (>350F), ceramic stand-offs (over extended studs) and accelerometer boots boost the temperature rating of these accelerometers to 500°F. VibraMetrics "super" magnetic mounts have been used to date, due to lack of an approved high temperature epoxy.

CVRP data evaluation equipment

- Liberty Technologies QUICKCHECK DAU with "full-channel download design modification," for signal conditioning hardware
- Liberty Technologies QUICKCHECK software, for analysis (including FFTs)

- Pressure. Simultaneous download of pressure information, during AE testing, has been of particular benefit in testing small piston check valves in air systems (Figure 1).
- Ultrasonics (UT). CPSES has been pursuing the simultaneous download of UT data to obtain disc position information during AE testing. The intent is to apply the technique on a case-by-case basis only (e.g., auxiliary feedwater steam generator suction check valves).

Test equipment components were chosen from among those currently available onsite (used in inservice inspection applications). Components are assembled in the following order, transducer (shear wave)[c] → [d]→ DAT recorder

c. The choice of transducer varies, based on valve size, wall material, and thickness.

d. Chosen unit must permit download of y-axis only.

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ATTACHMENT 8.C
PAGE 8 OF 14

CURRENT CVRP POPULATION

FEEDWATER

TAG	DESCRIPTION	MFR/TYPE	SCAFF	INSUL	ISI	TEMP/MATL	PRIORITY
1FW-0006	MAIN FW PUMP DISCHARGE	CRANE TILT	N	Y(B3)	NO	H/1	A*
1FW-0013	MAIN FW PUMP DISCHARGE	CRANE TILT	N	Y(B3)	NO	H/1	A*
1FW-0070	MAIN FW CONTAINMENT ISOLATION (SG#3)	ROCKWELL TILT	N	Y(B4)	CV/CS 123 EGT-741A, CV/CS 123 PPT-S1-8062	H/1	A*
1FW-0076	MAIN FW CONTAINMENT ISOLATION (SG#2)	ROCKWELL TILT	N	Y(B4)	CV/CS 123 EGT-741A, CV/CS 123 PPT-S1-8061	H/1	A*
1FW-0082	MAIN FW CONTAINMENT ISOLATION (SG#1)	ROCKWELL TILT	N	Y(B4)	CV/CS 123 EGT-741A, CV/CS 123 PPT-S1-8060	H/1	A*
1FW-0088	MAIN FW CONTAINMENT ISOLATION (SG#4)	ROCKWELL TILT	N	Y(B4)	CV/CS 123 EGT-741A CV/CS 123 PPT-S1-8063	H/1	A*
1FW-0191	BYPASS LINE CONTAINMENT ISOLATION	BW SWING	Y(11)	Y(C4)	NONE 123	H/1	A*
1FW-0192	BYPASS LINE CONTAINMENT ISOLATION	BW SWING	Y(11)	Y(C4)	NONE 123	H/1	A*
1FW-0193	BYPASS LINE CONTAINMENT ISOLATION	BW SWING	Y(11)	Y(C4)	NONE 123	H/1	A*
1FW-0194	BYPASS LINE CONTAINMENT ISOLATION	BW SWING	Y(11)	Y(C4)	NONE 123	H/1	A*
1FW-0195	TEMPERING LINE BACKFLOW (SG#4)	BW SWING	Y(33)	Y(C4)	CV/CS 123 OPT-506A	H	A*
1FW-0196	TEMPERING LINE BACKFLOW (SG#1)	BW SWING	Y(33)	Y(C4)	CV/CS 123 OPT-506A	H	A*
1FW-0197	TEMPERING LINE BACKFLOW (SG#2)	BW SWING	Y(33)	Y(C4)	CV/CS 123 OPT-506A	H	A*
1FW-0198	TEMPERING LINE BACKFLOW (SG#3)	BW SWING	Y(33)	Y(C4)	CV/CS 123 OPT-50[illegible]	H	A*
1FW-0199	SG BACKFLOW (SG#4)	BW SWING		Y(C4)	CV/QR 123 OPT-506A	H	A*
1FW-0200	SG BACKFLOW (SG#1)	BW SWING		Y(C4)	CV/QR 123 OPT-506A	H	A*
1FW-0201	SG BACKFLOW (SG#2)	BW SWING		Y(C4)	CV/QR 123 OPT-506A	H	A*
1FW-0202	SG BACKFLOW (SG#3)	BW SWING		Y(C4)	CV/QR 123 OPT-506A	H	A*

Figure 1. Population matrix within site-specific SOER 86-03 program procedure (selected page).

Limitations, other than the fact UT is not useable on steam or air valves, include the fact that use of a scope (not to mention complications of applying the technique to a variety of valve types) requires test personnel with full UT Level II certification.

- Magnetic Flux (Exterior). CPSES plans to team exterior magnetic flux (AC method) testing with acoustic emission this fall. Like UT, magnetic flux contributes disc position information. The test apparatus is still in the prototype stage.

Supportive methods for the testing and examinations include radiography, fiber optic/ boroscope probes, and thermography (to indicate valve leakage).

Disassembly Examination ("Condition-Directed PMs")

CVRP emphasizes condition-directed disassembly, that is, disassembly done when the examination results will confirm some condition predetermined by another (e.g., nonintrusive) technique. Therefore only the real problem valves are identified for disassembly (no random selection of one valve from a "like group"). Population disassembly candidates are selected primarily by nonintrusive test results and site maintenance history.

No valves will be opened purely to substantiate AE data. This stance is based on the predicted achievement of an AE baseline for all program valves (each manufacturer model) by the end of 1RF02.

Specific disassembly examination instructions have been incorporated into appropriate mechanical maintenance check valve disassembly procedures or work documents.

ACOUSTIC EMISSION TESTING

Technique Validation

CPSES was one of the sponsors of the NIC (Nuclear Industry Check Valve Group) Testing, (Utah State University Foundation, 1991) and has subsequently adopted the Phase 1 (liquid flow) results as validation of the AE technique for check valve reliability monitoring.

Software Quality Assurance

Since data evaluation cannot be performed in the field, requiring download into a computer for analysis (due in part to the sampling rate and filtering requirements), the "technique" involves software as well as hardware. Because CVRP AE data are now used to periodically assess operational readiness (IST) of 20 Unit 1 check valves, program software must meet certain site-specific verification requirements.

Measuring and Test Equipment (M&TE) Considerations

The data download equipment (filtering hardware) is classified as M&TE, on a recurrent 6-month calibration frequency. This appears justified based on "as found" calibration records at CPSES to date.

Among field data acquisition equipment (Table 2), only the power supply, with adjustable gain, is considered to be M&TE. Neither the DAT (digital audio tape) recorder nor the accelerometers are M&TE.

Classification of the accelerometers is specific for use under this program (not to be confused with vibration monitoring applications), that is, operational readiness assessments using AE data are only performed during IST open/close verifications. For this application, only the AE waveform shape and timing (time-of-arrival comparison between traces), and not the signal magnitude, are of importance. AE data used in the SOER program (condition monitoring) are used to identify disassembly examination candidates; therefore, operational readiness is determined during the disassembly exam (see Table 3).

Basically, disassembly candidates identified by the design report are tested acoustically in advance of the report's recommended disassembly examination outage to allow for prioritization

Table 3. AE test frequency for priority categories.

Priority	AE test frequency refueling cycles (minimum)
A	1
B	2
C	4

of candidate valves and a re-evaluation of the necessity for disassembly.

IST Check Valves. (Figure 2) AE data are acquired during specific, scheduled, full-stroke IST. This avoids redundant equipment actuation and capitalizes on scheduled test opportunities. Appropriate steps or notes are included in the applicable Surveillance Work Order instructions.

Sometimes surveillance test procedures unsuitable for AE testing can be modified to create a mutually beneficial test. This requires the involvement of appropriate operations and test personnel, IST Coordinator, and System Engineer. All parties must understand the basic requirements for a satisfactory AE test and ensure that a proposed test does not adversely affect other equipment (Figure 3).

Of CVRP valves, 72% are also IST valves. Note that although we may use IST test opportunities, we do not impose the associated IST frequency on these valves.

Non-IST Check Valves. (Figure 4) AE data are acquired either in accordance with a PM-generated work order or without a dedicated work order (using, as necessary, generic sensor mount installation, scaffolding erection, and insulation removal work orders). Scheduled system excursions should be considered as opportunities for obtaining data.

Test Definition

One of the more time consuming aspects of the program has been definition of the optimum steady-state/stroke test opportunity for each valve, with the subsequent tie to appropriate plant events. Refueling or cold shutdown testing was identified first, because of the planning time involved (and the fact that the implementation year of the program included two outages).

Determination of the optimum test event for an individual valve required an evaluation of the *quality* of test possible. The following items were considered:

1. Full stroke versus partial stroke (determine which is most representative of a 'normal' operating state; is an open/closure signature desirable).
2. Flow initiation quality—rapid versus gradual (throttling of a downstream valve following a pump start would be expected to prevent a good open signature). (Figures 5 and 6)
3. Back pressure, for closure testing (some degree of back pressure is usually necessary to obtain a close signature).
4. Filled pipe downstream of the check valve—if not, expect data to be adversely affected, regardless of the opening force.
5. Background noise (Figures 5 and 6) to investigate potential sources prior to the test. Where noise could be a problem, (a) adjust timing of test (e.g., during a reactor fueling outage, when reactor cooling pumps are not running), or (b) optimize test situation, when (a) is not possible (swap running pumps, to reduce effect of noise). Limitations of AE equipment/software must be known.

Data Trending

Initial trending efforts at CPSES have centered around the following items:

1. Steady state impact levels (under identical test conditions)—using the maximum (or often the "y range") value over a representative 20-second trace (Figures 7–10)
2. Fast Fourier Transform (FFT)—monitor change in frequency of interest (especially translational natural frequency of valve body, inclusive of internals) (Figures 7–11)
3. Flutter rate, in cycles/sec (of occasional interest, especially when test includes flow zones causing disc instability)—in these cases, total time spent in such a flow zone might be of greater interest than actual flutter rate.

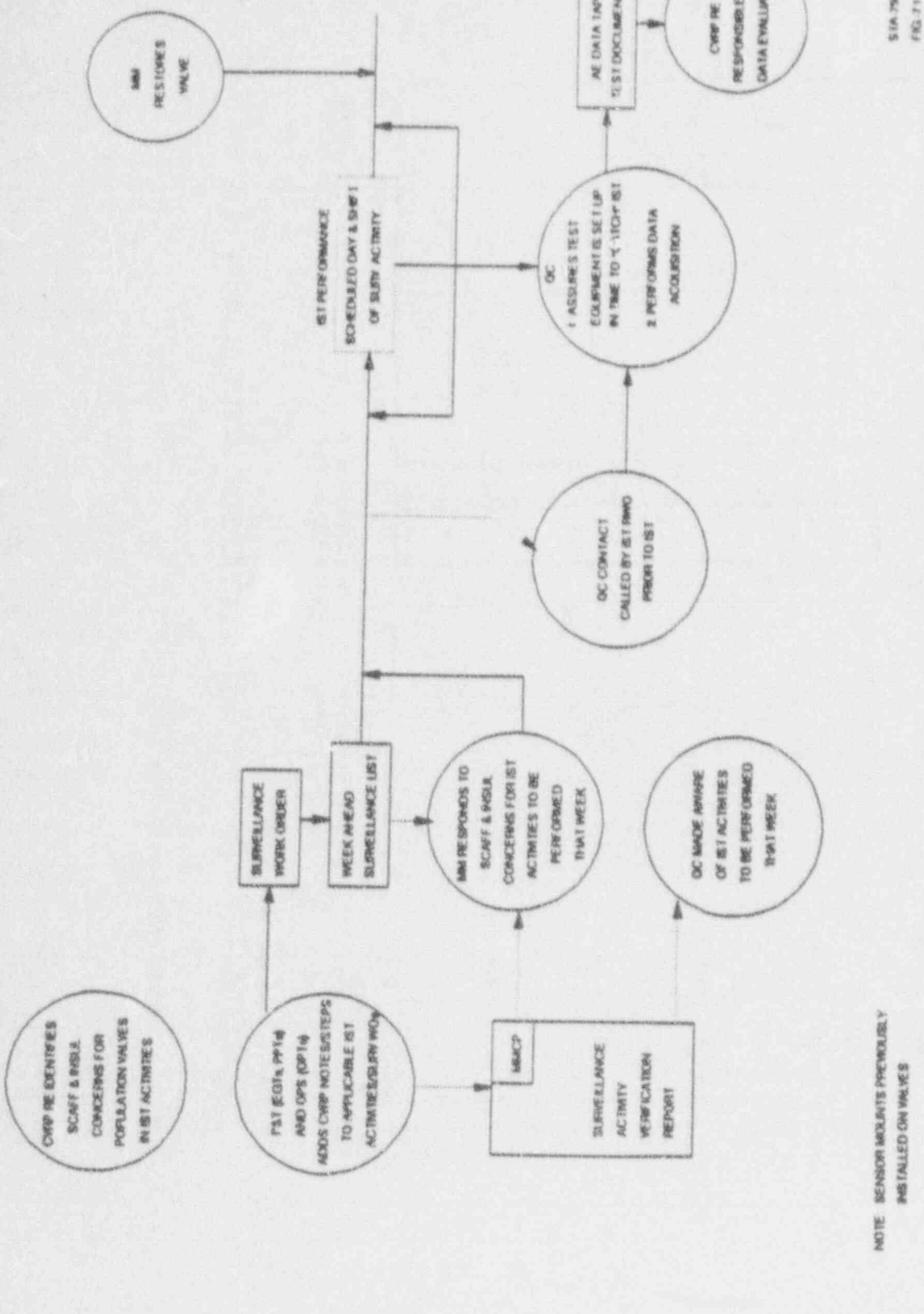


Figure 2. CVRP AE test performance flowchart: IST check valves.

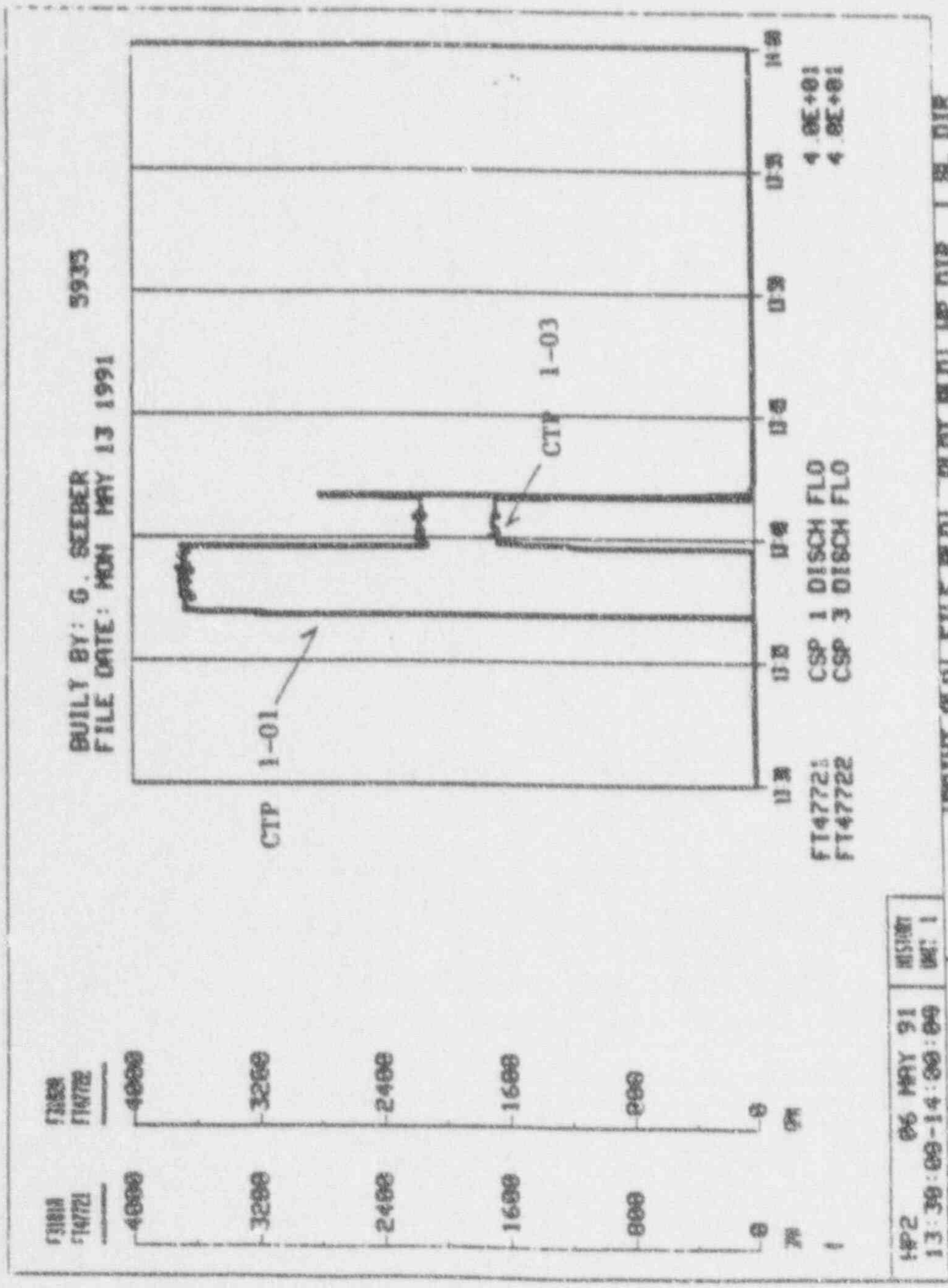


Figure 3. Flow data coincident with staggered two pump start.

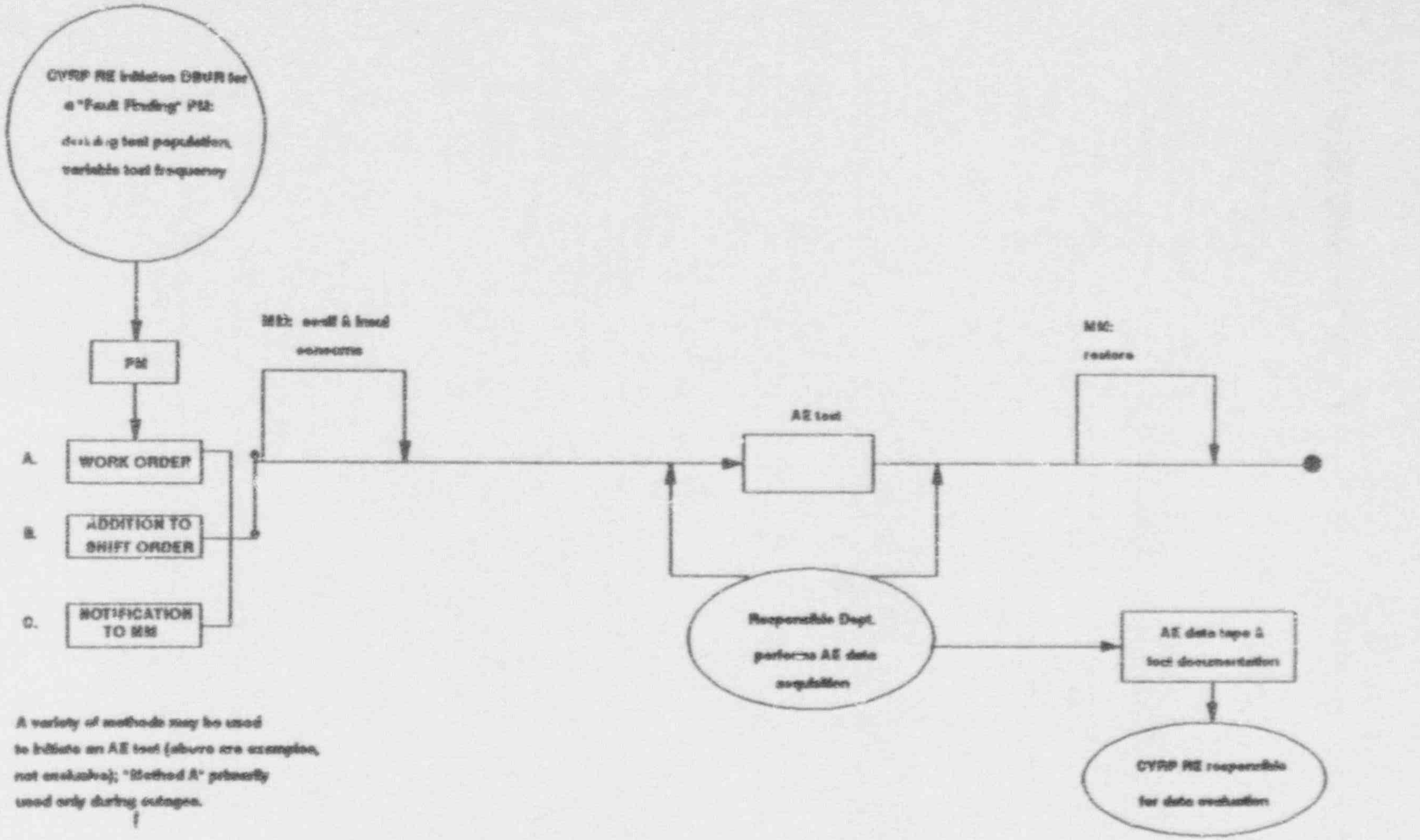
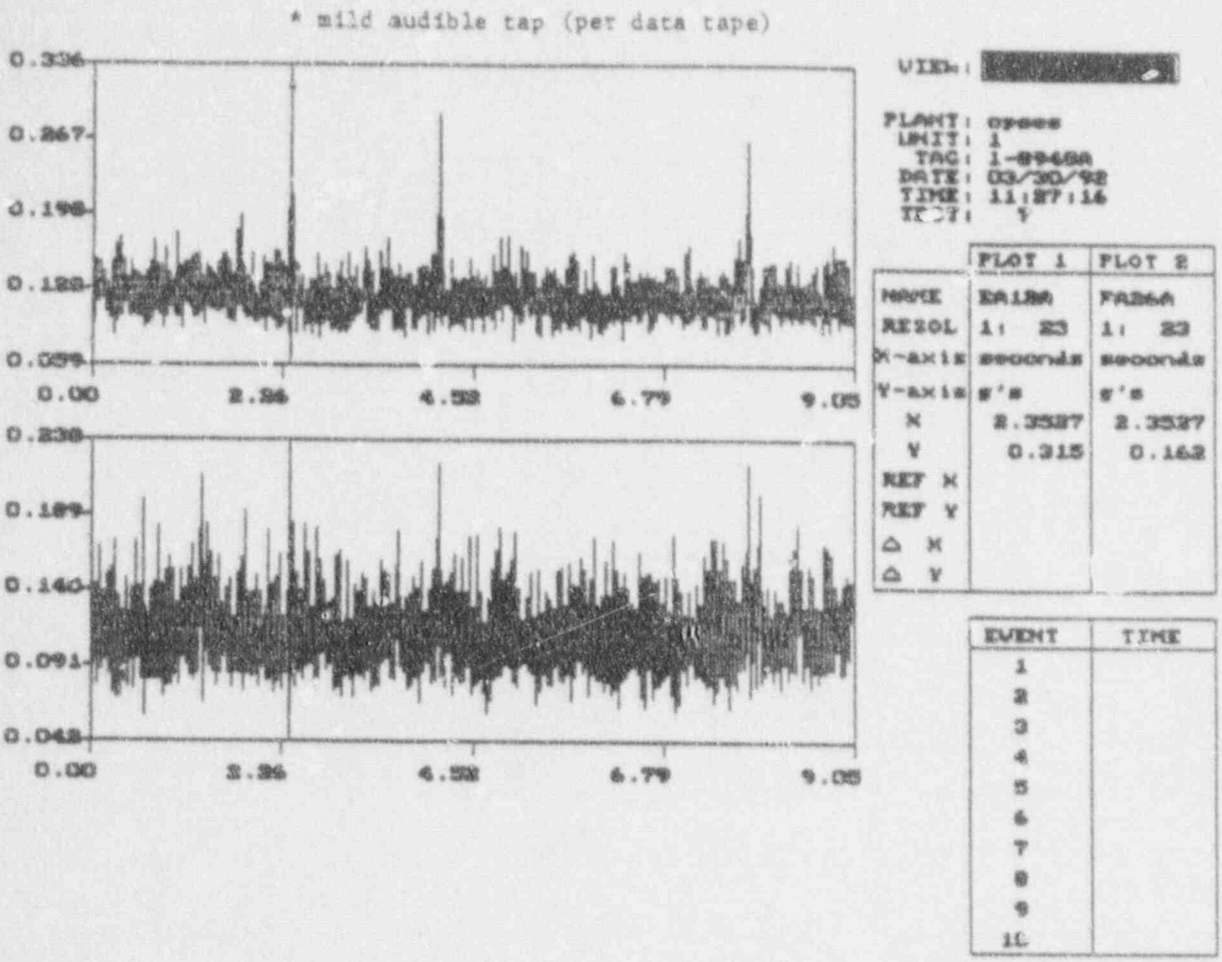


Figure 4. CVRP AE test performance flowchart: non-IST check valves.

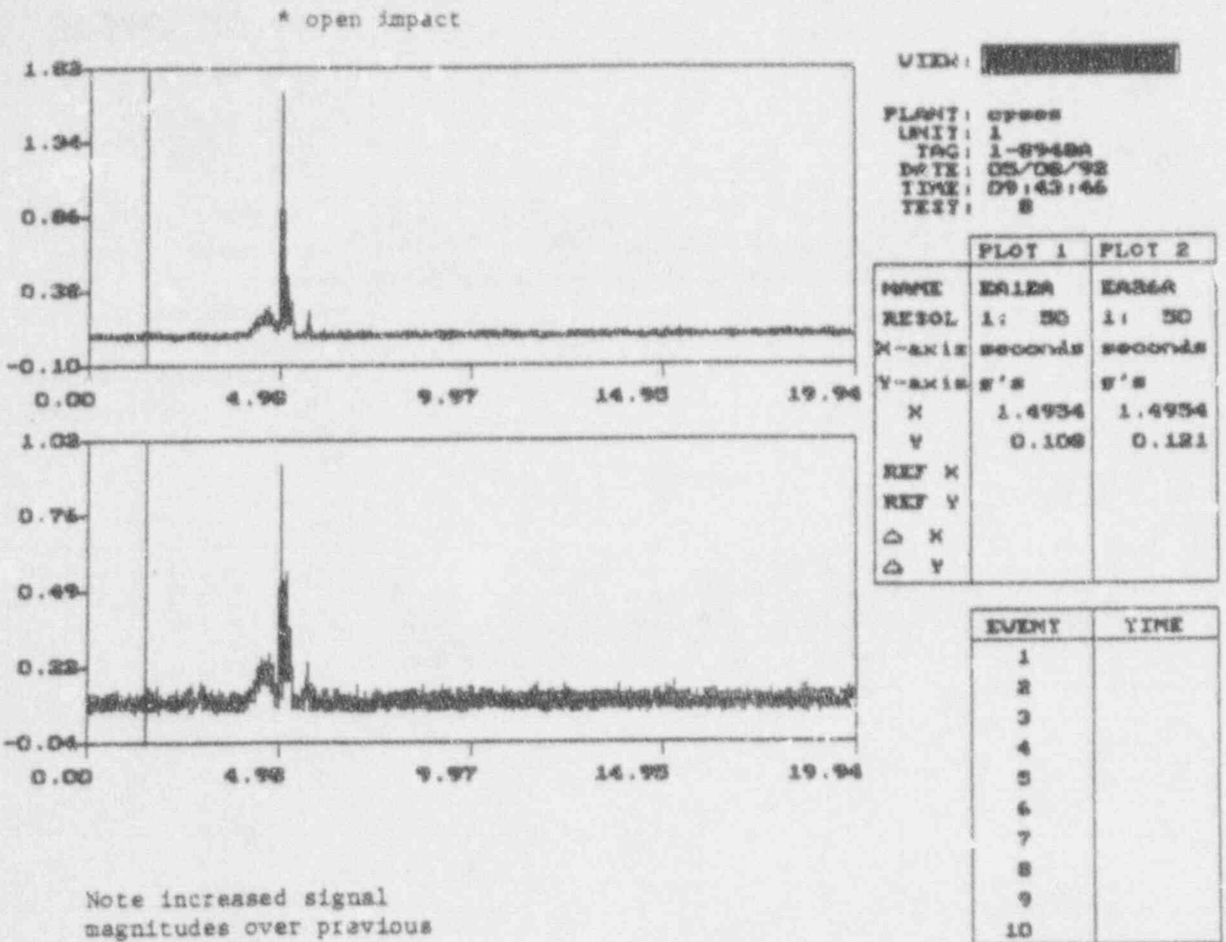


Plot 1: AE, envelope
(30g, ± 2V)
bonnet

Plot 2: AE, envelope
(30g, ± 2V)
neck

Test Scenario:
o RCPs 3 & 4 running
o RHR A pump start
(11/28/91, 1RF01)

Figure 5. Unacceptable "Open Verification" (IST) AE test.



Note increased signal magnitudes over previous unacceptable test, as well as better definition and shape of waveform

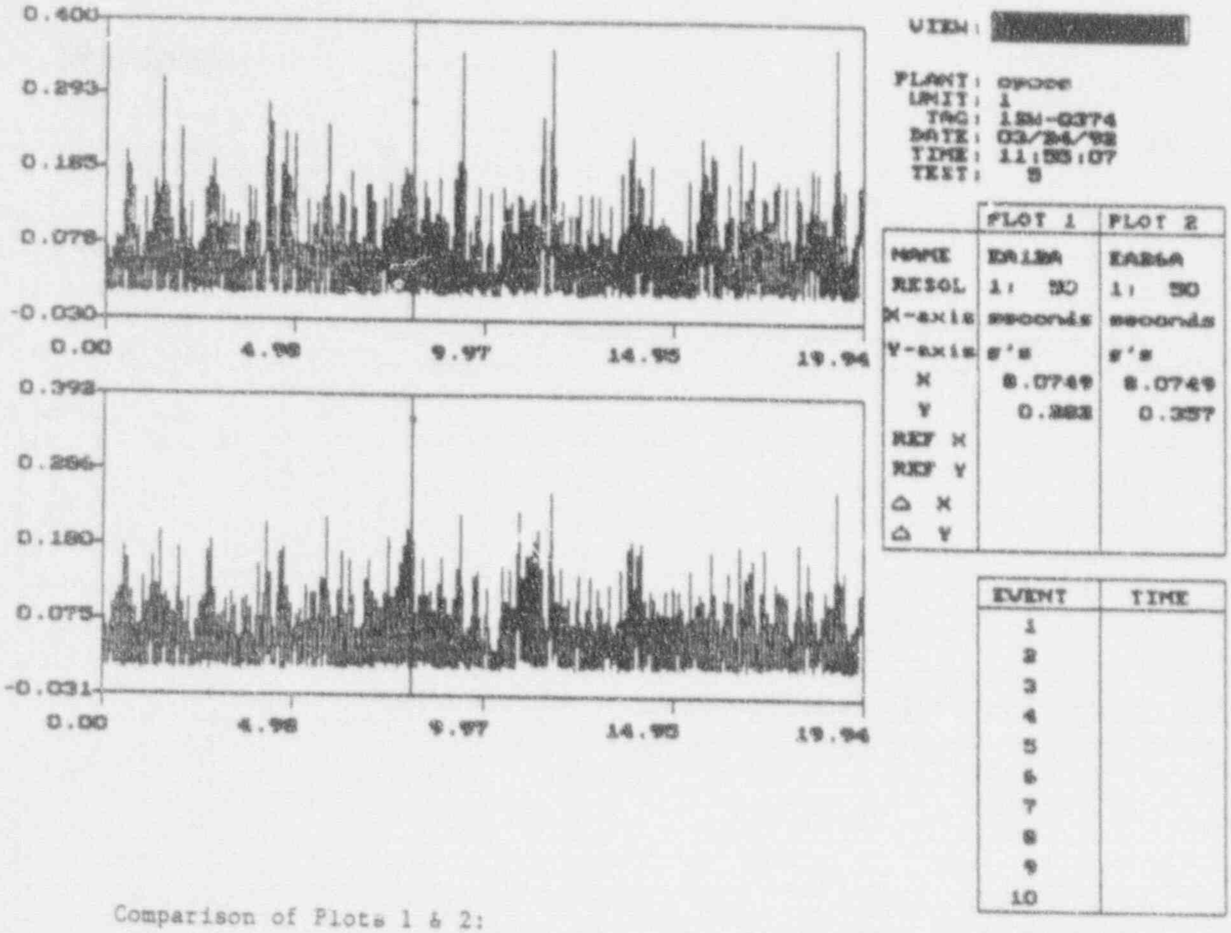
Plot 1: AE, envelope (30g, ± 2V) bonnet

Plot 2: AE, envelope (30g, ± 2V) neck

Test Scenario:
- o RCPs 3 & 4 running
- o cycle of MOV 1-8809A, after RHR A pump start

(11/30/91, 1RF01)

Figure 6. Improved "Open Verification" (IST) AE test.



Comparison of Plots 1 & 2:

- o magnitudes are close
- o many coincident peaks

Plot 1: AE, envelope
(30g, $\pm$ 2V)
hinge pin, top

Plot 2: AE, envelope
(30g, $\pm$ 2V)
hinge pin, bottom

Test Scenario:
steady state flow, 3/24/92
(virtually identical test
flow conditions and AE
equipment set-up as 1SW-0373
Test 2, Attachment 9)

Figure 7. Steady state data on Train A service water pump discharge check valve (24-in. TRW mission dual disc).

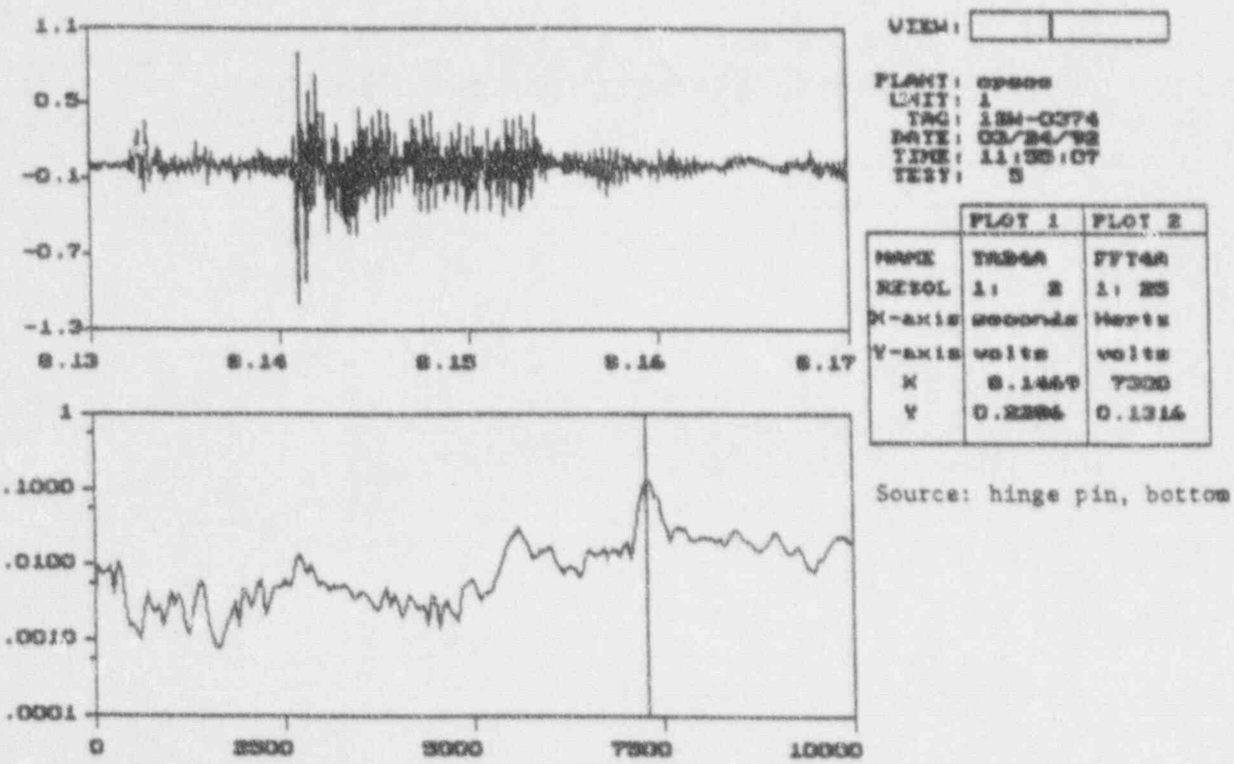
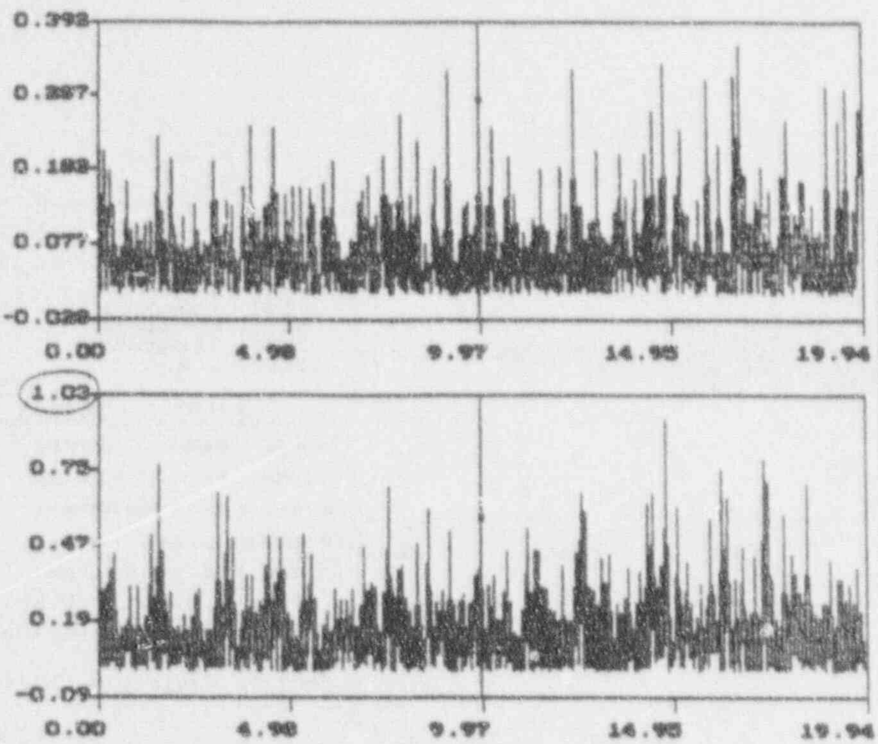


Figure 8. FFT of an isolated steady state impact, from Figure 7.



VIEW:

PLANT: opeee
UNIT: 1
TAG: 1SW-0373
DATE: 03/23/93
TIME: 18:06:38
TEST: 2

	PLOT 1	PLOT 2
NAME	EA1BA	EA26A
RESOL	1: 50	1: 50
X-axis	seconds	seconds
Y-axis	g's	g's
X	9.9192	9.9192
Y	0.278	0.576
REF X		
REF Y		
△ X		
△ Y		

EVENT	TIME
1	
2	
3	
4	
5	
6	
7	
8	
9	
10	

Impacting reaches 1g level in 'hinge pin, bottom' sensor.

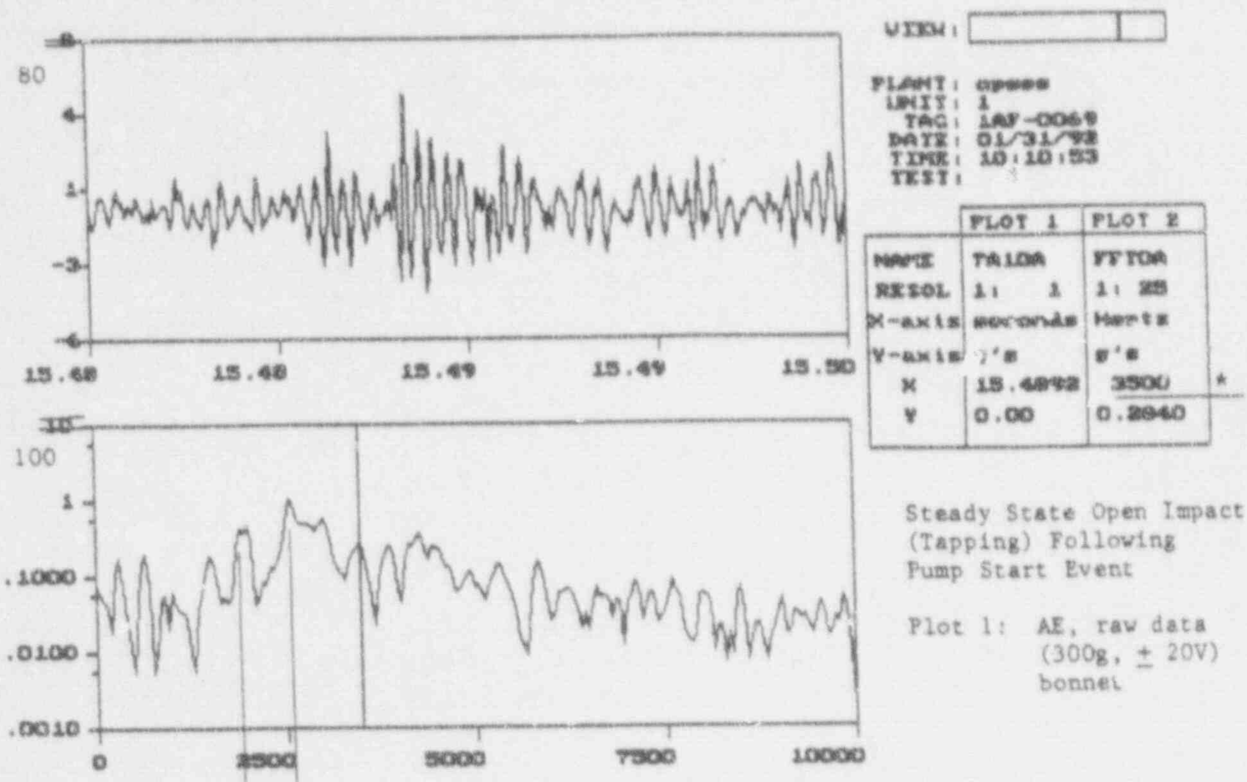
This valve is a 1RF02 disassembly candidate.

Plot 1: AE, envelope (30g, $\pm$ 2V) hinge pin, top

Plot 2: AE, envelope (30g, $\pm$ 2V) hinge pin, bottom

Test Scenario: steady state flow, approximately 17.5 kgm, 3/6/92

Figure 9. Steady state data on Train B service water pump discharge check valve (24-in. TRW mission dual disc).



** 1925
*** 2550-2600

per BWIP seismic analysis:

for valve body (inc. bonnet and internals),

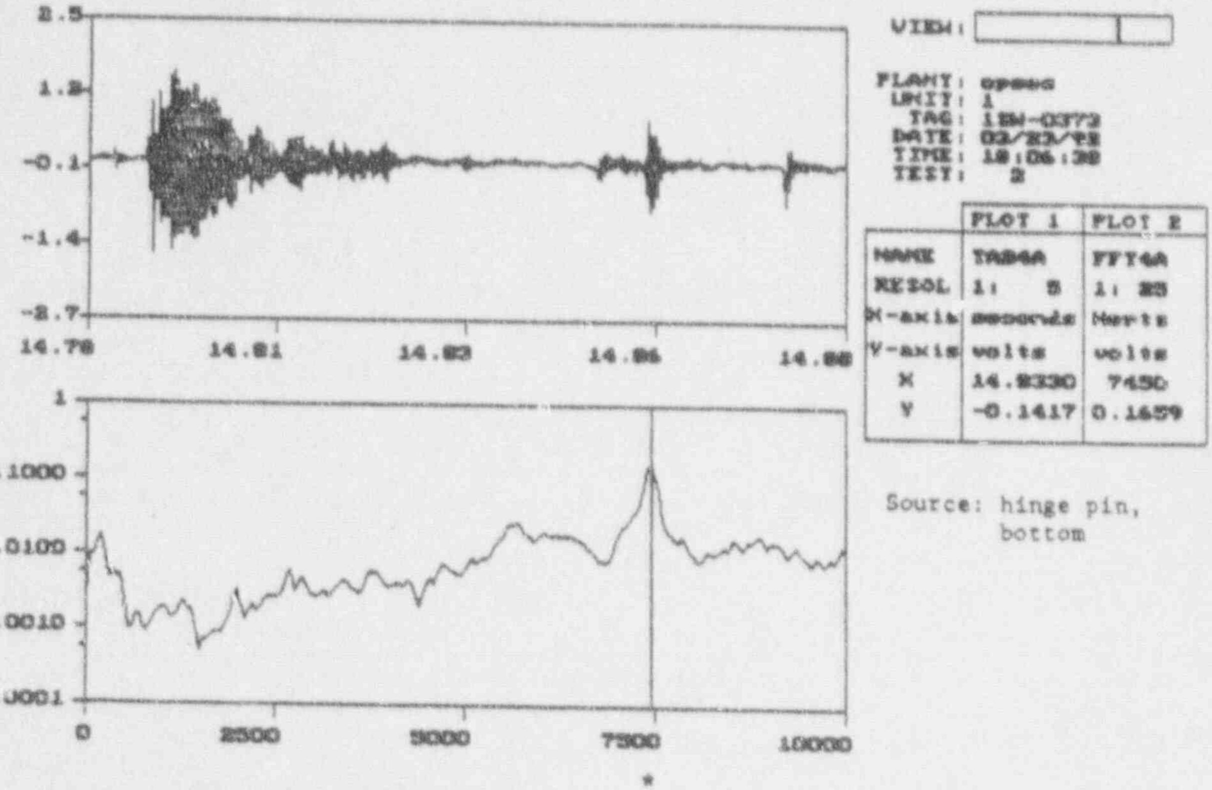
$$f_{n\ \text{transl}} = \frac{4}{2\pi}\sqrt{\frac{3EI_g}{ML^3}}$$

* f_n transl = 3444 Hz

** f_n tors = 1922 Hz

*** potentially, hinge pin f_n
Per Reference 6: estimated hinge pin frequency <u>in air</u> would be 3710 Hz (for this size/rating of valve); "in water" value predicted to be 15% lower

Figure 10. FFT analysis of steady state impacting on auxiliary feedwater pump miniflow check valve (3-in. BWIP swing, pressure sealed bonnet).



* Comparison with 1SW-0374 (Attachment 8B) shows a shift of +150 Hz in this dominant frequency

Figure 11. FFT of an isolated steady state impact, from Figure 9.

Items 1 and 2 were used in a trending application (comparison with like valve or past data) to select a valve for 1RF02 disassembly. The most notable benefits, however, are not expected for several years (once more definite trends are established).

Influence of Proper Signal Conditioning

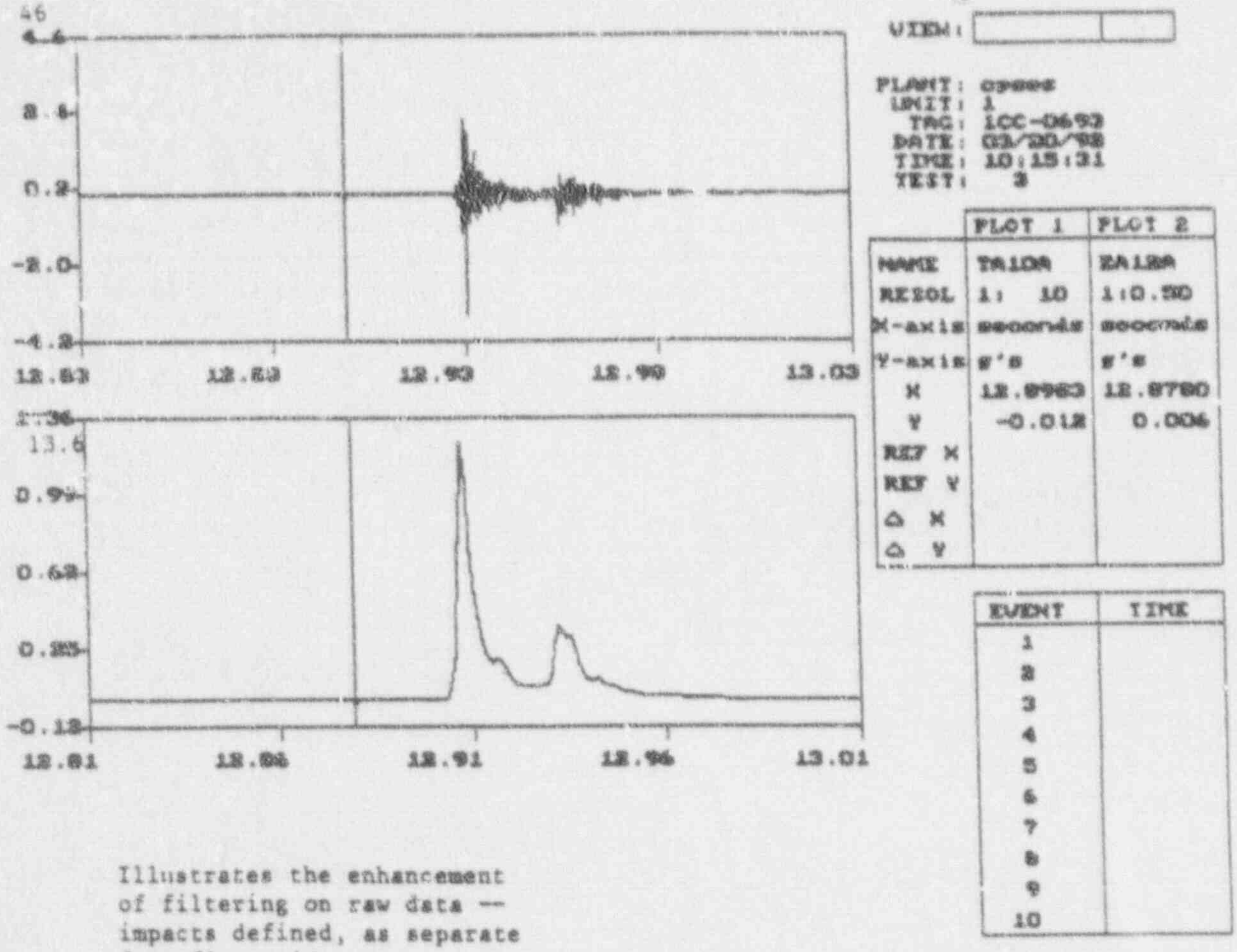
The choice of signal conditioning hardware and software can determine success at trending, or even at data evaluation as a whole (Figures 12, 13, and 14).

1RF01 RESULTS

The tests included two main feedwater pump discharge check valves, which are a 20-inch Crane tilt disc, Model 20-7109. There is one associated with each pump.

Train A Disassembly Results Versus Design Review

The Train A valve (1FW-0006), during its 1RF01 disassembly exam, exhibited significant wear on the plunger assembly (plunger, housing,



Illustrates the enhancement of filtering on raw data -- impacts defined, as separate from flow noise

* Note that gain setting is incapable of providing the degree of resolution necessary to show flow change

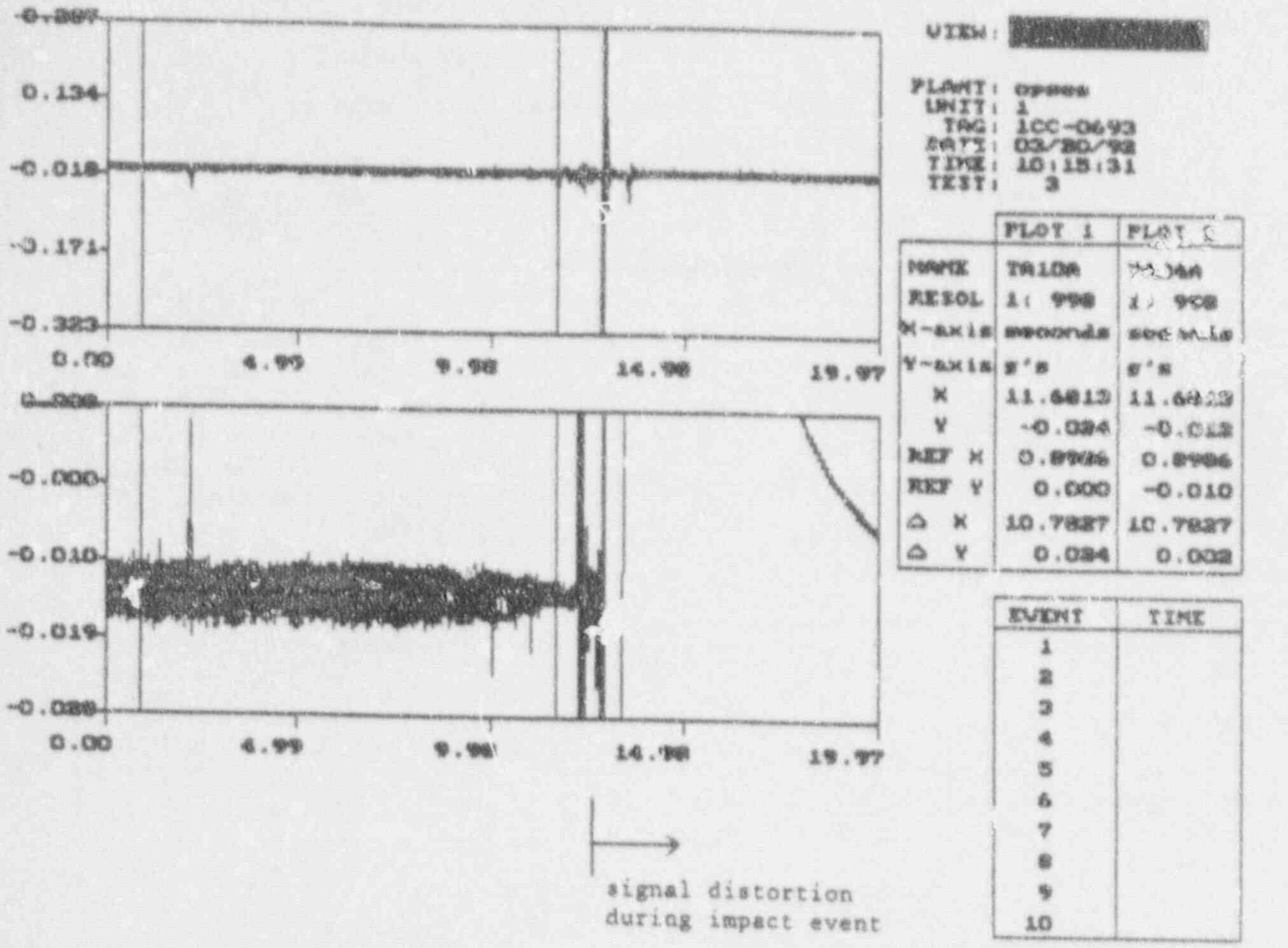
** Some early check valve AE data collected by utilities was similar in nature to this (unfiltered)

Closure Impact

** Plot 1: AE, raw data with x-axis zoom (10g, ± 20V) * bonnet

Plot 2: AE, high pass filtered data (envelope) of raw data in Plot 1 (x-axis zoom)

Figure 12. Closure event in 4-in. BWIP swing check valve (effect of filtering raw data).



Above illustrates that
even with appropriate
software (and filtering),
must have knowledge of the
effect of gain setting

Plot 1: AE, raw data with y-axis zoom (10g, ± 20V) bonnet

Plot 2: AE, raw data with y-axis zoom (3g, ± 20V)
higher gain shows subtle flow change--identifies event more clearly

Figure 13. Closure event in 4-in. BWIP swing check valve (effect of gain setting).

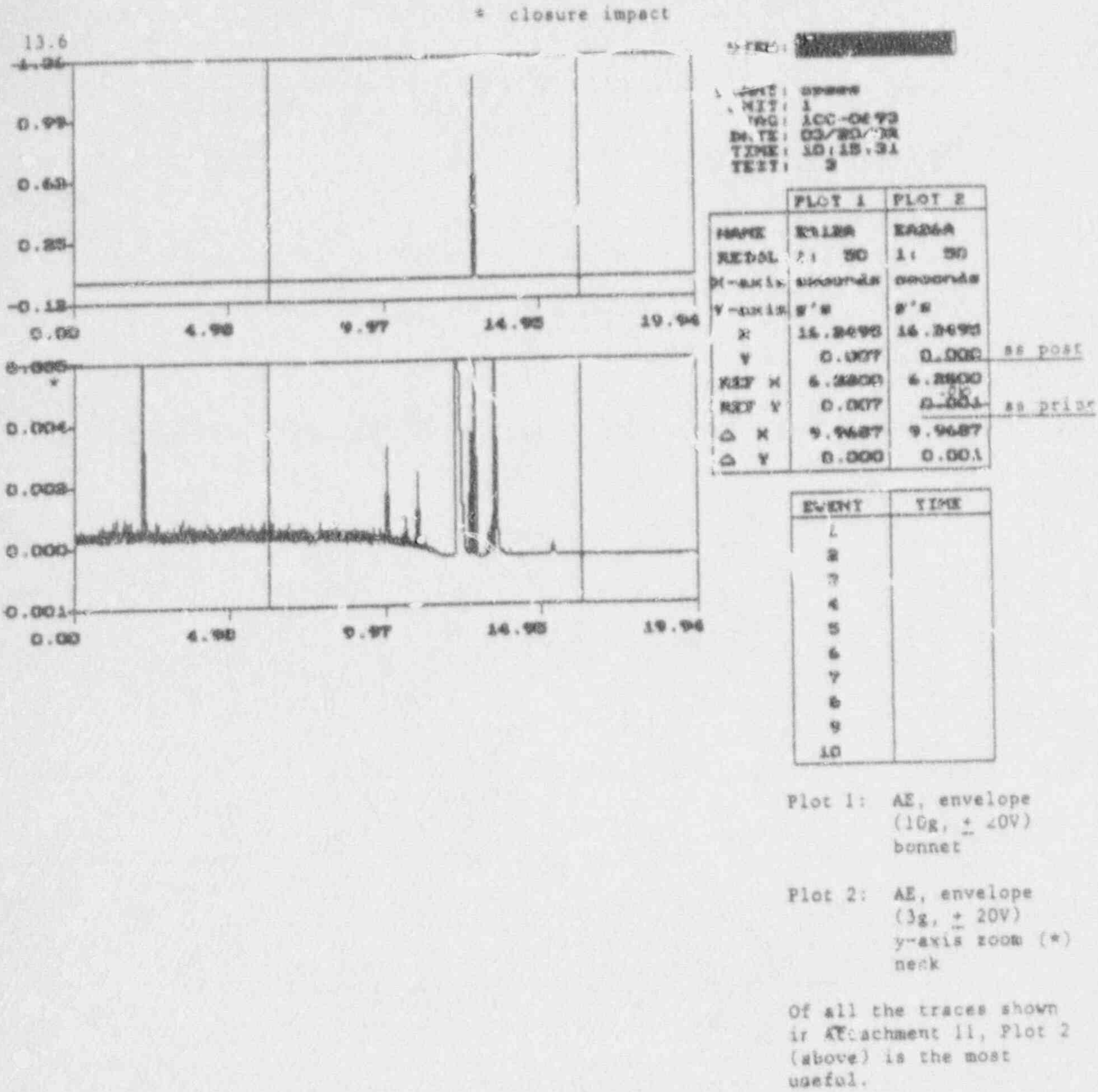


Figure 14. Closure event in 4-in. BWIP swing check valve (combined effect of filtering and proper gain setting).

and spring) (Figure 15). This evidence, at first, appears to be at odds with the original (pre-operational) SOER 86-03 Design Report, which concluded that this valve was not a wear problem (wear index = 1, fatigue index = 1) and that expected operating conditions provided a flow velocity 44% greater than the valve's minimum velocity requirements, resulting in a fully open disc. However, it must be realized that the CVAP analysis is oriented to detect only hinge pin wear and disc stud fatigue. With that definition of wear, a wear index of 1 is consistent with the 1RF01 disassembly results (Figure 16).

The design report (which includes CVAP results and consideration of valve internal geometry and corrosion concerns) did consider the valve

COMPARISON OF NEW VERSUS OLD INTERNAL CLEARANCES

Clearance	Old Side 1	Old Side 2	New (Spare)
Pin-to-bushing	0.0155	0.014	0.0135
Pin-to-fixed bushing (disc)[a]	0.0135	0.015	0.005
Bushing-to-seat	0.0025	0.001	0.002

a. Retaining pin intact; this apparently represents the as-installed condition.

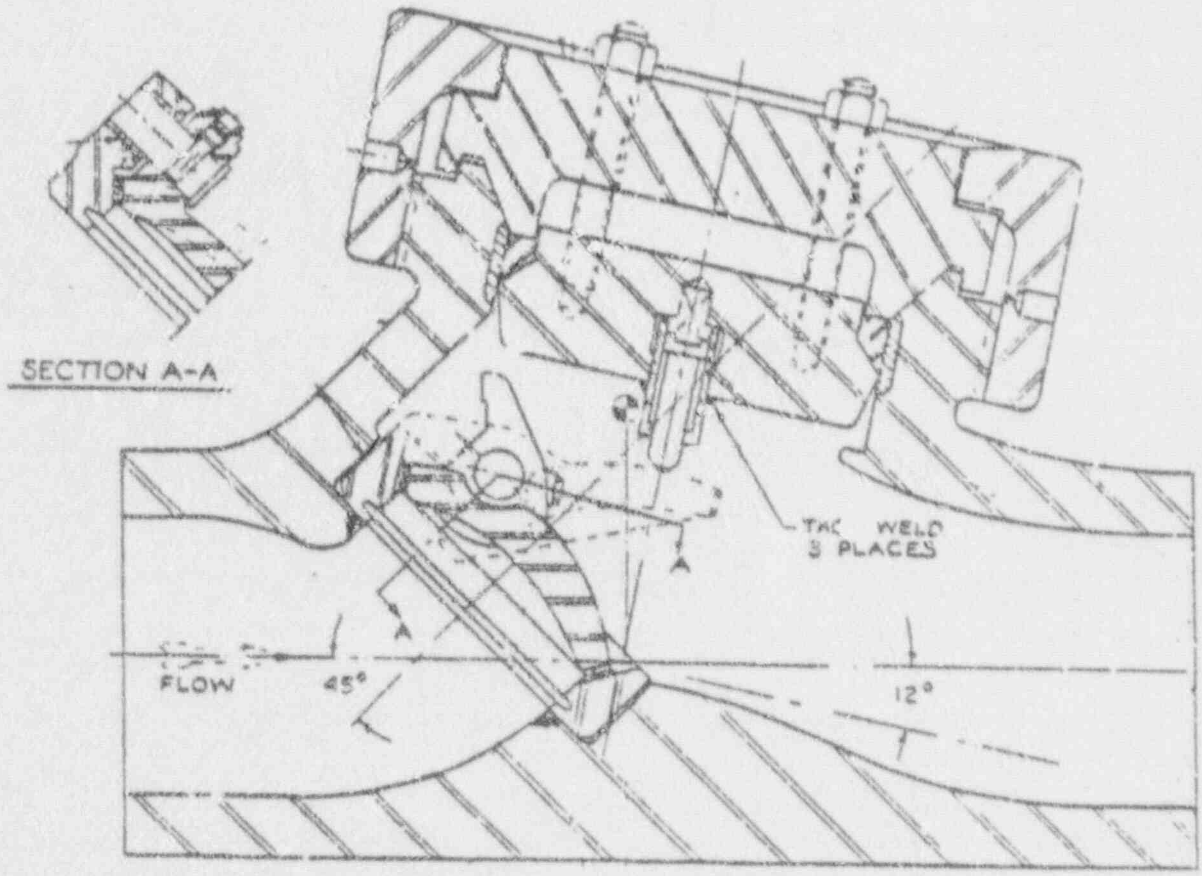


Figure 15. 1RF01 disassembly examination summary of Train A main feedwater pump discharge check valve—bushing wear.

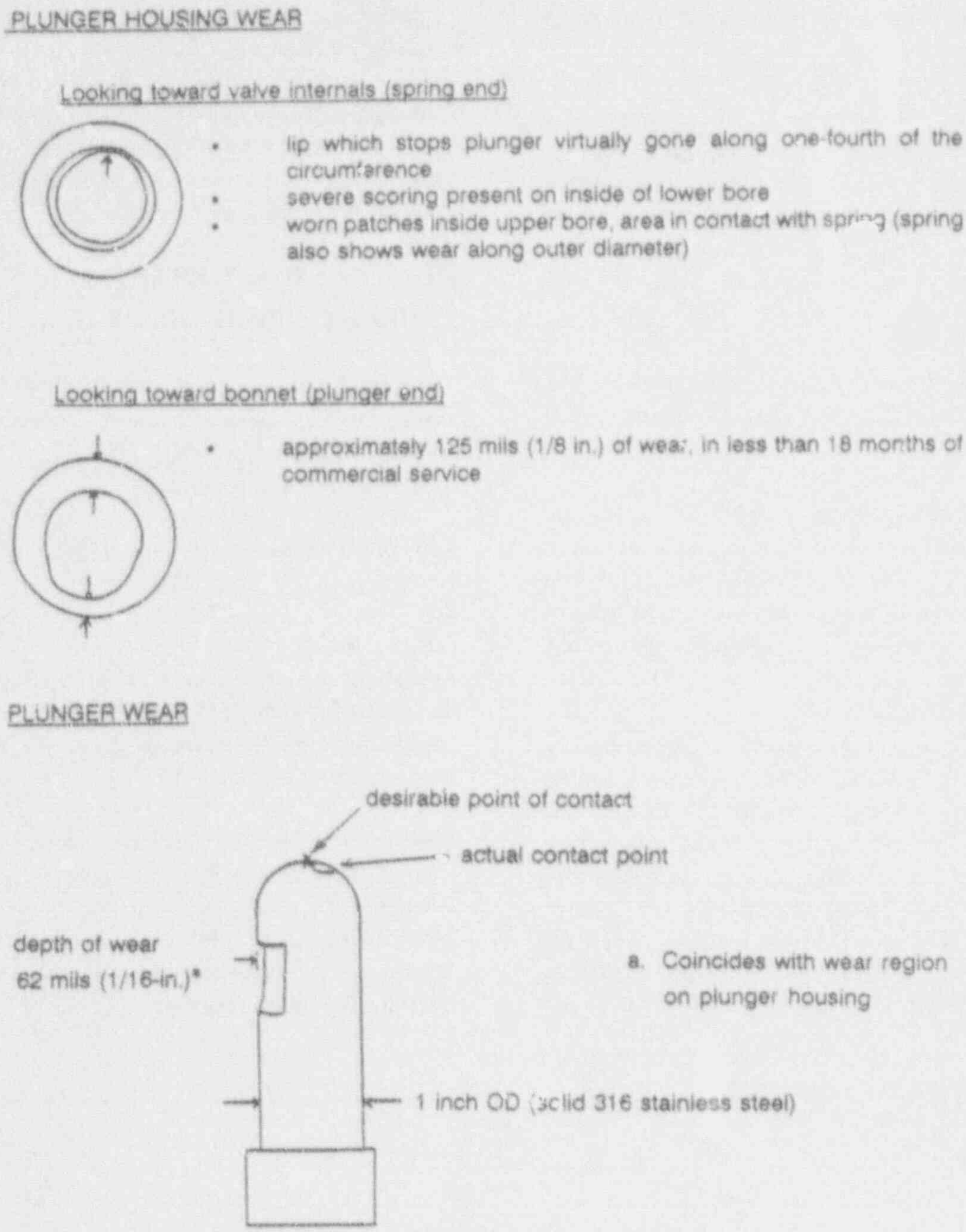


Figure 16. 1RF01 disassembly examination summary of Train A main feedwater pump discharge check valve—plunger assembly wear.

body and disc to be susceptible to erosion/corrosion, recommending yearly exams for this reason. However, it did not identify the plunger assembly as a candidate for excessive "wear."

Refinement of Design Review. Original inputs to the analysis were estimates provided by the utility. A recent informal re-analysis using more accurate values, as supplied by the valve manufacturer, for five specific parameters (disc weight, disc diameter, seat diameter, impingement angle, bushing length) showed that the valve was a potential wear concern. The revised values of three parameters were notably different from original estimates and proved to be quite influential in the analysis (Table 4).

Operational History. A review of plant computer data revealed a history of frequent cycling (approximately two to three times per month), leveling off at 16+ kgpm (greater than original design report Qmin of 12.3 kgpm), with a sometimes sporadic nature to the flow. This, however, is considered normal feedwater operation.

Corrective Action. The plunger assembly (plunger, spring, housing) and bushings were replaced.

Train B Disassembly Results. The Train B valve (1FW-0013), the only other of this model in Unit 1 or Common, was disassembled during the spring 1991 mid-cycle outage and found to have broken tack welds at the plunger-to-bonnet interface.

AE Data. AE data were first collected during 1RF01 (after the disassembly examination of 1FW-0006). At approximately 14 kgpm flow through valve, the AE data revealed consistent, though low-magnitude, steady-state disc-to-plunger activity (approximately 0.100 g; magnetic mounts used) with frequent backstop tapping (to 0.890 g in 1FW-0006, 0.400 in 1FW-0013). Also evident is the fact that the plunger is serving as the main initial backstop, as expected according the manufacturer.

Planned Action. Both valves have been scheduled for disassembly at 1RF02. Further action (e.g., design modification) will be based on disassembly examination results.

Auxiliary Feedwater Pump Miniflow Check Valve

Tests included three 3-inch Borg Warner swing, pressure sealed bonnet, Model 75510 valves, one associated with each pump (one turbine driven, two motor driven).

1AF-0057 Disassembly Results. 1AF-0057 was selected for disassembly based on previous AE data results (Figure 17). Disassembly revealed evidence of disc stud nut contact with the valve body wall in the neck region. (However, there were indications that the disc stud was completing the open trip to the backstop, when in fact the stop was mushroomed.)

Sample Expansion. Previous AE data (available on only one of the like valves) supported existence of the same problem, but to a somewhat lesser degree, in 1AF-0069. The 1AF-0057 disassembly results, along with this AE data, led to the disassembly of the other two like valves.

Table 4. Results of refinements.

	Value		
Parameter	Refined	Original	Reason for change
Disc weight	200 lb	123 lb	Casting weight supplied by vendor versus rough calculation by utility
Impingement angle	12 degrees	18 degrees	Rough estimate based on knowledge of tilt disc performance
Bushing length	4.1 in.	2.5 in.	Original estimate was conservative

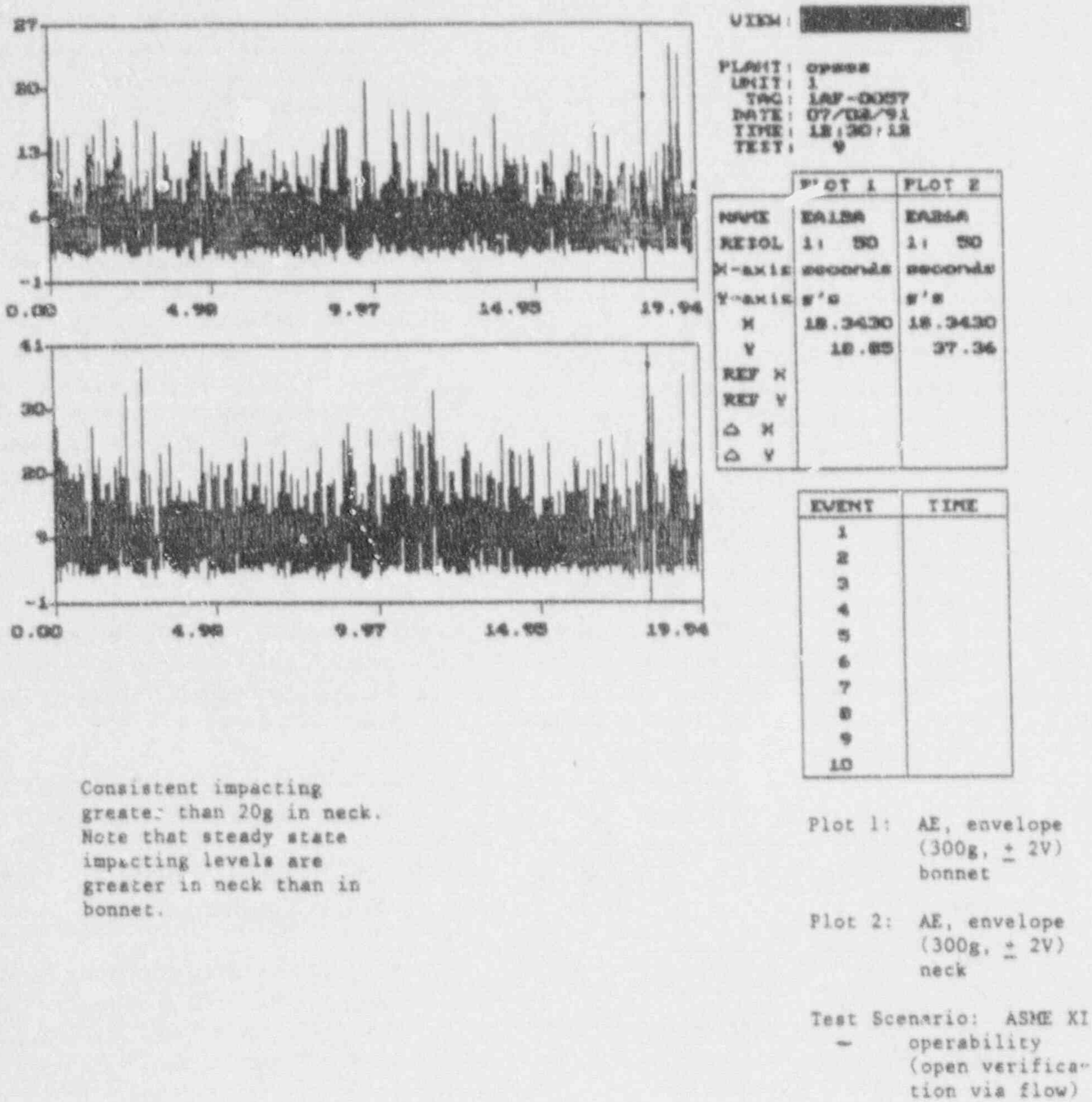


Figure 17. Steady state impacting in auxiliary feedwater pump miniflow check valve (3-in. BWIP swing, pressure sealed bonnet).

- 1AF-0069 showed (a) evidence of disc stud/nut contact with valve body wall, but to a lesser degree than shown in 1AF-0057 (evidence existed to show stud was hitting stop) and (b) disc rubbing valve body wall at 10:00 and 2:00 positions, but to a lesser degree than shown in 1AF-0045

- 1AF-0045 showed evidence of (a) same disc stud/nut contact, but most minor of the three; wear was evident on nut (no evidence of stud contact with stop); (b) cracked ball bushing; (c) stud out of alignment (angled from disc joint); and (d) evidence of disc rubbing valve body wall.

Corrective Action Taken. The following corrective actions were taken for the three valves:

- 1AF-0057. Honed valve neck to allow for greater internal clearance (wall kept within manufacturer's minimum requirements).
- 1AF-0069. Honed valve neck and body wall to allow greater clearance for movement of internals.
- 1AF-0045. (a) Replaced ball bushing and stud; (b) increased stop length to conform to manufacturer's recommendations (done earlier in 1AF-0057 and 1AF-0069); and (c) honed valve neck and body wall.
- Subsequent Issues. The attempted replacement of the swing arm (in 1AF-0057) with a manufacturer-endorsed "improved equivalent" part led to generation of a 10 CFR 21 Report by the utility. The utility's written report to the NRC, dated February 28, 1992, concluded that the "improved" part was not completely interchangeable in our application, leading to a potential failure of the valve to close. It also stated that there was, in our application, no safety significant consequence associated with a failure to close (only an open safety function is recognized).

CONCLUSIONS

- Though early in plant operation, CPSES Unit 1 has successfully used AE to identify valves requiring refurbishment.
- Post baseline, no design review analysis is more effective than properly used nonintrusive test data results and site maintenance history. In fact, these two factors basically revalidate or modify the design review.

AFTERTHOUGHT

Nonintrusive monitoring should be used only in the operational readiness assessment of "IST hardship valves," those that cannot be tested at maximum accident flow (or for which such is impractical), that is, valves facing disassembly inspection as an only recourse.

For IST efforts, the nonintrusive test is of interest only to prove that an open or closed position has been reached. When evaluated "with blinders on," the nonintrusive test data become no more useful than traditional flow testing (Figure 18). Only when consideration, in the form of an increased test frequency (or even a greater time allowance by Operations personnel for completion of data evaluation), is given for the more complete test should nonintrusive techniques be more thoroughly pursued for IST take-credit situations.

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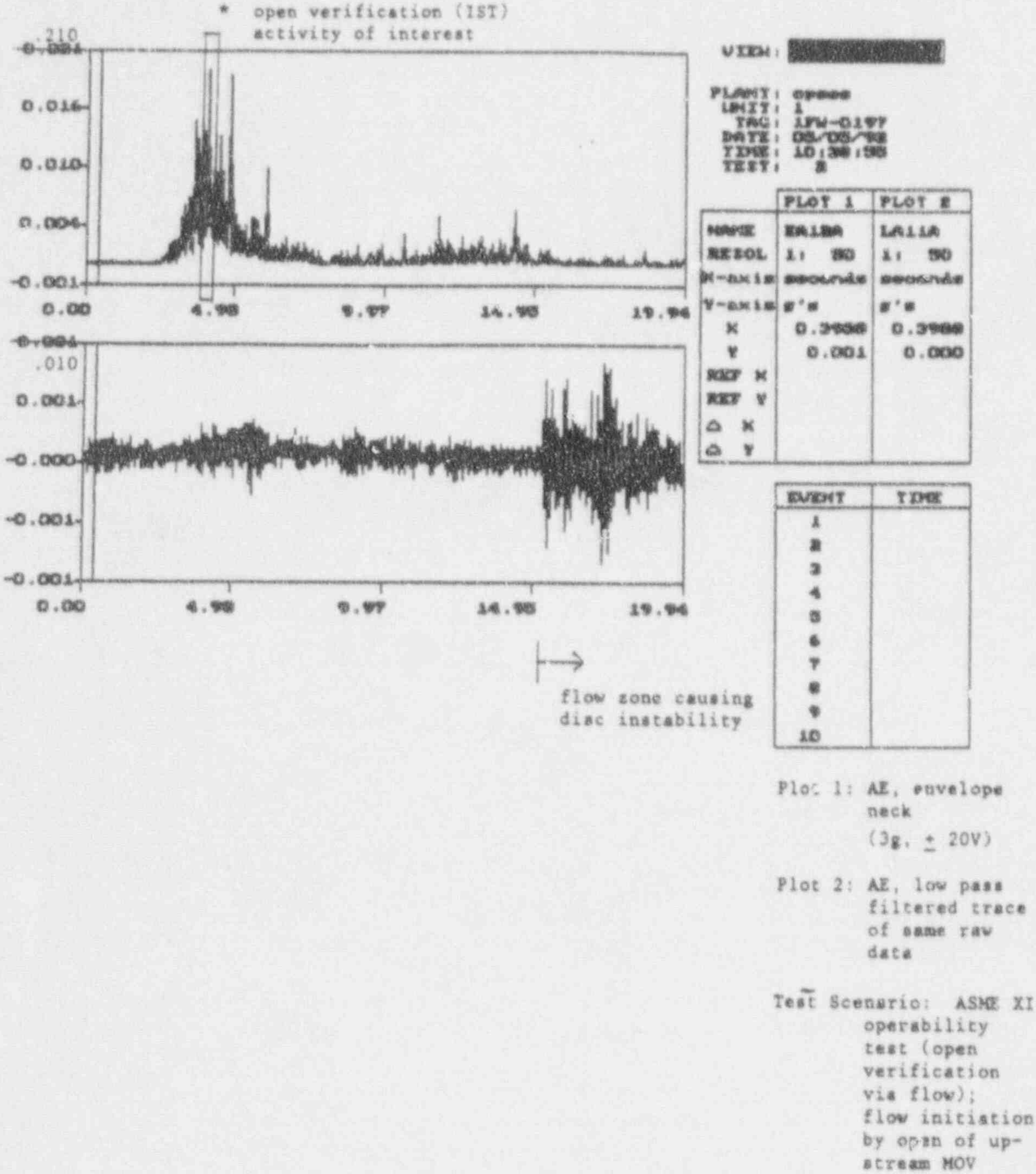

Figure 18. Open verification versus condition monitoring (6-in. BWIP Swing Check Valve in FWPB line).

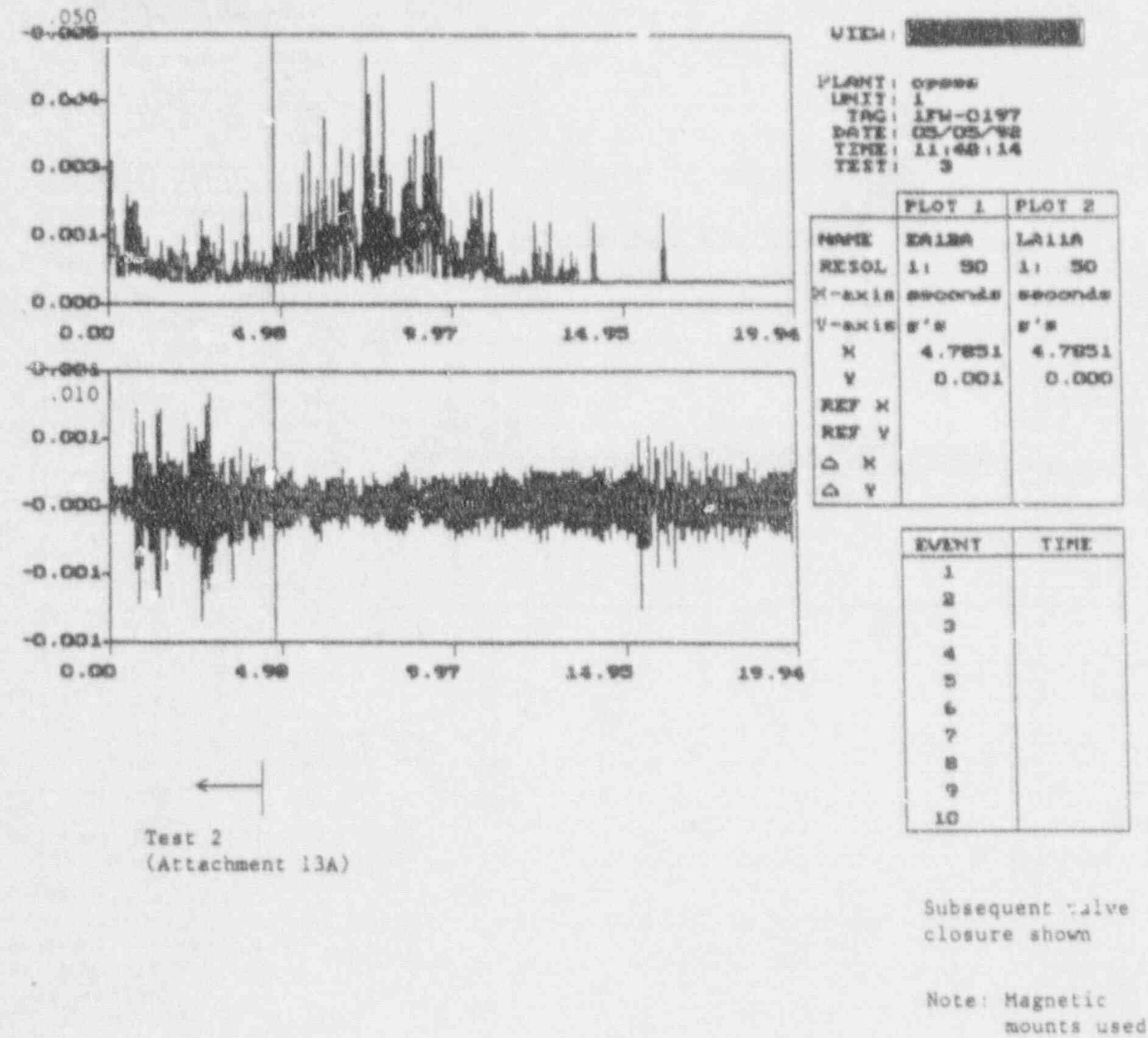

Figure 18. (continued).

Attachment 1. Summary of CPSES Unit 1 check valves monitored to date.

Manufacturer	Model	Description	System	Quanity
TRW Mission	24CC302WA	24-in. dual disc (SSWP disch)	SW	2
Westinghouse	04000CS88	4-in. swing (centrif chg pmp disch)	CS	2
	04000CS87	4-in. swing (SI pmp disch)	SI	2
	06000CS88	6-in. swing (RHR Hx to hot leg) (cold leg inj isol, 1-8818)	SI SI	2 4
	08000CS84	8-in. swing (RHR Hx to chg pmp suct)	SI	2
	08000CS82	8-in. swing (RWST to chg pmp suct)	CI	1 1
	10000CS84	10-In. swing (RCS cold leg inj, 1-8948; accum disch, 1-8956)	SI	5
	14000CS84	14-in. swing (RHRP suct)	RH	2
Rockwell	1-1/2-838YTI 2-D3664T1	1-1/2-in. piston (DG air start)	DO	8
	2-D3664T1	2-in. stop (piston) (RCP thermal barrier inlet)	CC	4
	2-B36164T3	(same)	CC	4
	18-970BQTY	18-in. t[illegible]	FW	4
Borg Warner	75560	4-in. swing pressure sealed bonnet (SG suct)	AF	8
	75810	16-in. swing bolted bonnet (CTP suct)	CT	2
	75510	3-in. swing pressure sealed bonnet (AFWP miniflow)	AF	3
	75640	6-in. swing pressure sealed bonnet (AFWP disch)	AF	3
	454KAB11	6-in. swing	FW	6
	75580	4-in. swing return from RCPs)	CC	4
	76360	10-in. swing	CC	1
Crane	20-323B	20-in. tilt (CO pmp disch)	CO	2
	20-7109	20-in. tilt (FWP disch)	FW	2
	3-147	3-in. swing 1/2 XU	AF	1
Atwood & Morrill	14308-01	14-in. swing (Turbine to FWH 1/A/B)	EX	2
	14308-02	14-in. swing (...2A/B)	EX	2
	14308-03	24-in. swing (...3A/B)	EX	2
	14308-04	30-in. swing (...4A/B)	EX	2
TOTAL				83

Above valves were all tested at least once since May 1991 when CVRP was procedurally put in place. Some information-only testing was conducted August 1990 through April 1991, but it is not reflected in above table.

Evaluation of Nonintrusive Examination Methods

Dr. Paul Tullis, Utah State University Foundation
Michael Lind, Philadelphia Electric Company

ABSTRACT

Check valves are used extensively in nuclear power plants. Studies conducted by the Institute for Nuclear Power Operations (INPO), the Electric Power Research Institute (EPRI), and the U.S. Nuclear Regulatory Commission (NRC) point out that many of these valves are located in safety-related systems and are not functioning properly. As a result, INPO issued a Significant Operating Experience Report (SOER) 86-03. This was followed by the NRC issuing Generic Letter 89-04, which required that all safety-related check valves in nuclear power plants have a regular monitoring program.

To minimize the impact of these directives on plant operations, the utilities need nonintrusive diagnostic techniques to replace or supplement disassembly and inspection procedures. To pursue this need, a group of utilities organized the Nuclear Industry Check Valve Group (NIC) in 1989. This parent committee subsequently organized a subcommittee, Nonintrusive Examination Committee (NEC), to investigate existing technologies for nonintrusively examining check valves. Technologies considered for evaluation included acoustic signature analysis, magnetic signature analysis, ultrasonic inspection, and fiber optic, radiographic, and thermographic inspection.

In cooperation with EPRI, the NIC committee developed two research programs to investigate acoustic, magnetic, and ultrasonic techniques for check valves operating in water and pneumatic systems. These two research programs were completed under contract with the Utah Water Research Laboratory. The phase 1 (water) testing was completed in 1990 and the phase 2 (pneumatic) in 199[illegible]. Three vendors representing three different technologies participated in phase 1 test program. Two of the vendors participated in phase 2 test program.

In addition to the testing program sponsored by the NIC committee, independ[illegible] testing was carried out at the Utah Water Research Laboratory by a fourth venc[illegible] to evaluate their acoustic/ultrasonic instrumentation system.

This paper summarizes the results of those three testing programs and describes the capabilities of the three technologies. The paper is not intended to be an evaluation or endorsement of any vendor, but an assessment of technology.

The results of the testing program show that the three technologies do provide viable alternatives to disassembly and visual inspection of check valves. With baseline data available on a new valve (undegraded), the technologies in general were able to distinguish between a new valve and a valve with degraded internals. They could usually locate the area of the degradation, and in a few cases, were able to distinguish quantitatively between degradations. The technologies could determine

if the disc was stuck, missing, or operating normally through its entire stroke. The ultrasonic and magnetic techniques were able to determine mean disc position, and identify magnitude and frequency of disc flutter in water systems. The magnetic technique was also able to do this in air systems. The three technologies were able to detect seat and backstop tapping, plus movement of the internal parts.

During the first phase of the NEC testing, internal magnetics were used. During the second phase (pneumatic) external acoustics and magnetics were used. During the pneumatic testing, no ultrasonic instrumentation was used.

INTRODUCTION

Check valves are flow-activated devices designed to fully open and operate firmly backseated under normal forward flow conditions and close quickly to prevent reversal of flow. If the valves are improperly designed, improperly selected, or operated at low discharges, the disc may float freely and not be fully backseated. If such a valve is located downstream from a disturbance, such as an elbow, control valve, or pump, the turbulent flow conditions will cause exaggerated disc motion and accelerated wear. If such valves are not properly monitored and maintained, they can malfunction.

As an alternative to disassembly and visual inspection, nonintrusive diagnostic techniques have been developed. Such techniques include acoustic signature analysis, magnetic signature analysis, ultrasonic inspection, and radiography.

In April 1989, a group of utilities formed the Nuclear Industry Check Valve Group (NIC) to investigate these technologies in detail. The committee sponsored two research programs at the Utah Water Research Laboratory to investigate the three technologies. The research program included tests on swing check, tilt disc check, lift check valves, and duo-check valves in sizes ranging from 1.5-inch (3.81 cm) through 24-inch (61.0 cm). The valves represented a variety of pressure classes, valve styles, and valve materials (carbon, stainless, and brass). Testing was conducted using water and air as the flow medium.

Participation in the NIC-sponsored testing program was open to all vendors: three accepted the invitation: the CANUS Corporation, who used acoustic signature analysis; Henze-Movats (ITT Movats) who used ultrasonics, and Liberty Technologies who used acoustics and electromagnetics. All three companies participated in the water phase of the NIC testing, and Henze-Movats and Liberty Technologies participated in the pneumatic phase of the testing.

In addition to the NIC sponsored testing programs, independent testing was carried out at the Utah Water Research Laboratory by B&W Nuclear Service Company on their acoustic/ultrasonic system.

This paper describes the capabilities of the three technologies represented by the four vendors.

EXPERIMENTAL PROGRAM

NIC Phase 1 Testing—Water

A total of eleven valves were tested with water during the phase 1 NIC sponsored testing. Part A of the testing program included tests on the following five valves:

- 10-inch (25.4 cm) Crane Swing Check
- 12-inch (30.48 cm) Valmatic Tilt Disc
- 4-inch (10.16 cm) Velan Swing Check
- 10-inch (25.4 cm) Mission Duo-Check
- 10-inch (25.4 cm) Velan Swing Check.

For these valves, the vendors were informed of all test conditions prior to testing and all details regarding internal conditions of the valves including degradations. Part A was designed to allow the vendors to gain additional experience on interpreting their results by knowing beforehand what degradation condition existed in the valves.

Part B of phase 1 included tests on the following six valves:

- 16-inch (40.6 cm) Rockwell Tilt Disc
- 24-inch (61.0 cm) Valmatic Tilt Disc
- 24-inch (61.0 cm) Atwood & Morrill Swing Check
- 20-inch (50.8 cm) Atwood & Morrill Swing Check
- 6-inch (15.2 cm) Powell Swing Check
- 6-inch (15.2 cm) Crane Tilt Disc.

For these valves, the vendors were not informed about the degradations being tested. The vendors were excluded from the test area when the valves were disassembled to make changes.

The vendors were required to submit a preliminary qualitative evaluation of the valve degradation within two days after completion of the tests. Upon receipt of their preliminary evaluations, the vendors were told what the degradation actually was, and were then allowed additional time for detailed analysis before submitting their final reports.

The independent testing performed by B&W Nuclear Service Company was done on the following four valves:

- 10-inch (25.4 cm) Crane Swing Check
- 20-inch (50.8 cm) Atwood & Morrill Swing Check
- 4-inch (10.16 cm) Velan Swing Check
- 4-inch (10.16 cm) Valmatic Tilt Disc.

These research programs were designed to determine if the technologies are capable of measuring disc motion, including:

1. Mean disc position
2. Magnitude and frequency of disc motion
3. Backstop tapping
4. Seat tapping
5. Movement of disc arm on hinge pin
6. Full stroke of disc
7. Movement of disc on stud pin.

Degradations tested in the valves varied with the specific type of valve, but included the following:

1. Hinge pin wear
2. Stud pin wear
3. Wear of the disc (for lift-checks)
4. Stuck disc and missing disc
5. Missing or broken spring
6. Seat leakage
7. Combined stud and hinge pin wear
8. Uneven hinge pin wear
9. New valve repeated (repeatability test, not a degradation).

Tests were conducted initially on a new valve to determine the minimum velocity required to fully open and backseat the valve under uniform flow conditions and with disturbed approach conditions. Following these primary tests, the valve was operated under the following conditions:

1. Valve closed with no flow

2. Valve closed with seat leakage induced (new valves only)

3. Establish sufficient flow to fully back seat (valve fully open) the valve without turbulence

4. Reduce the valve opening to induce backstop tapping without artificial turbulence

5. Induce turbulence and obtain backstop tapping

6. Reduce flow to set the disc at approximately 75% open without turbulence

7. Induce turbulence to increase disc motion at approximately 75%

8. Reduce flow to approximately 30% without turbulence

9. Induce turbulence at 30% disc position

10. Attempt to induce seat tapping (valve closed)

11. Simulate a pump trip/start test by rapidly stopping and then re-establishing the flow.

The independent testing carried out by B&W Nuclear Service Company at the Utah Water Research Laboratory closely followed the procedure of the phase 1 NIC testing.

NIC Phase 2 Testing—Air

A total of five valves were tested with air during the phase 2 NIC sponsored testing:

- 4-inch (10.16 cm) Velan Swing Check
- 3-inch (7.62 cm) Fairbanks Swing Check (Brass Body, Brass Internals)
- 3-inch (7.62 cm) Fairbanks Swing Check (Brass Body, Steel Internals)
- 1.5-inch (3.81 cm) Anchor Darling Lift Check
- 1.5-inch (3.81 cm) Vogt Lift Check.

Only Liberty Technologies and Henze-Movats participated in the phase 2 testing. The instrumentation evaluated during the phase 2 testing were acoustics and external magnetics.

No practice valves were included in phase 2. All of the valves were tested with unknown degradations.

The research program and the degradations tested were essentially the same as for the phase 1 water tests.

DESCRIPTION OF TECHNOLOGIES

Acoustics

The instrumentation package for acoustic monitoring consists of piezoelectric sensors (accelerometers) with associated electronics for signal conditioning, recording, and analysis. Analysis of the accelerometer data makes it possible to detect the condition of check valves by analyzing structure-borne noise created by moving parts or by leaking fluids.

The number and location of the sensors varies with the type of valve. Usually, a minimum of two sensors are used: one placed near the seat and one near the backstop. Additional accelerometers can be located if cross correlations are desired to further pinpoint the exact source of the degradation.

Numerous techniques are available to maximize the data obtained from the accelerometers. The simplest level of analysis is to observe the raw output from the accelerometer on a scope and observe those spikes that are above a selected threshold. While interpreting the raw signal on an oscilloscope, the operator should listen to the accelerometer noise to help identify the threshold level of the impact noises.

A variety of filtering techniques can be used to eliminate electrical noise, flow disturbances, or cavitation. This makes it easier to identify those spikes caused by loose internals and impacting of the disc against the seat or backstop.

The signal can also be plotted on an expanded scale to observe the form and the duration of the event. This makes it possible to distinguish between cavitation noise, tapping, and grinding sounds from the valve. With proper filtering and analysis techniques, reliable data can be obtained even in systems with a high level of background turbulence including cavitation. As the cavitation increases, it becomes more difficult to distinguish between valve noises and flow noises. During the test program, the background turbulence and cavitation were generated by throttling a valve located 2 to 5 diameters upstream from the check valve. If a control valve upstream from the check valve is cavitating heavily, it makes it impossible to use ultrasonics, makes the use of acoustics questionable, will greatly increase disc instability and wear, and possibly will cause erosion damage due to cavitation collapse.

More sophisticated analysis can be carried out using Fast Fourier Transform spectrum analysis techniques and cross-correlations to help pinpoint the exact location from which the noise is generated. This is helpful in interpreting the type of degradation.

The acoustic technique can detect disc movement and tapping, but cannot locate the exact position of the disc nor quantify the amount or frequency of disc flutter.

Baseline data on a valve in a new condition are helpful for proper interpretation of the accelerometer data.

Ultrasonic Signature Analysis

The ultrasonic transducers are devices that both transmit and receive ultrasonic information. The ultrasonic transmitter transforms electric energy into ultrasonic energy. The receiver converts ultrasonic energy back to electric energy. The transducers operate in the megahertz range, and send and receive directional ultrasonic pulses. The system only works in liquid mediums. It will not work with gas, steam, or in liquid systems where heavy cavitation results in two-phase flow.

The placement and orientation of the directional transducer is critical because the signal must be reflected off the proper surface. With check valves, the signal is normally reflected from the disc, the hinge arm, or the hinge. The system detects distance by measuring the time between the transmission and the reception of the reflected signal. The transducer must be positioned so that it can obtain a good reflected signal from the disc over the full stroke of disc motion. By first obtaining baseline data with the disc fully closed and then fully open, the position of the disc at an intermediate value can be determined. The high frequency response of the system also enables the magnitude and frequency of the disc motion to be determined.

To apply this technology properly, it is helpful to have detailed information on the geometry of the valve internals. This enables the transducer to be mounted properly and aligned to focus on the appropriate internal component.

Magnetic Signature Analysis

Two magnetic signature analysis systems were investigated by Liberty. For the phase 1 testing, a permanent magnet was installed on the disc arm and a Hall effect probe attached externally. The Hall effect probe is a magnetic flux density transducer that produces an electrical signal proportional to the magnetic flux. The magnet assembly consists of a samarium-cobalt magnet hermetically sealed in a stainless steel housing with a cone-shaped flux concentrator and a stainless steel base. Motion of the disc arm changes the magnetic field produced by the permanent magnet. This effect is picked up by the Hall probe and interpreted to provide information on disc position.

The second technique used by Liberty was external magnetics, where ac magnetic coils are attached externally to the valve and positioned so that the movement of the valve disc creates maximum interruption of the magnetic field. The interruption of the magnetic field by the movement of the disc is processed to provide information on disc position and motion.

ITT Movats also used an eddy current technique during phase 2 for measuring disc movement.

EXPERIMENTAL RESULTS

Determination of Mean Disc Position

The standard used to evaluate the accuracy of the ultrasonic and magnetic techniques in measuring mean disc position was a potentiometer attached to a cable wrapped around the hinge arm. The potentiometer data were taken independently by the laboratory. Tables 1 and 2 compare a number of test runs showing the comparison between the laboratory disc opening measurement and that measured by the ultrasonic and the internal magnetic instrumentation. In almost all cases, there is less than 5% difference between the laboratory reading and the vendors' readings. The uncertainty in the lab data is about 2 to 3%. The ultrasonic and magnetic techniques are most accurate near the seat and the full open position because the instrumentation is calibrated there.

The external magnetics have a resolution of a fraction of a percent near the full open position. During one test run when the valve was supposed

Table 1. Determination of disc position using Ultrasonics.

Valve number	Run number	USUF data disc open%	Ultrasonics disc open%
Part A			
1	8	73.2	73.4
	9	33.1	36.7
2	101	75.0	68.4
	103	30.4	34.9
3	146	76.2	81.3
	148	33.3	30.2
4	210	78.5	83.4
	212	52.3	48.3
5	275	74.0	82.2
	277	29.3	37.3
Part B			
6	345	73.7	69.4
	347	29.9	35.4
7	392	78.6	69.3
	394	35.7	30.0
8	443	63.3	55.4
	445	25.8	27.6
9	482	75.2	78.9
	484	29.5	29.4
10	534	74.0	79.4
	536	30.1	32.9
11	606	76.5	70.3
	608	29.6	29.6

Table 2. Determination of disc position using Magnetics.

Valve number	Run number	USUF data disc open%	Magnetics disc open%
Part A			
1	8	73.2	74.0
	9	33.1	31.0
2	101	75.0	78.0
	103	30.4	25.0
3	146	76.2	76.0
	148	33.3	33.0
4	210	78.5	86.0
	212	52.3	40.0
5	275	74.0	82.0
	277	29.3	33.0
Part B			
6	345	73.7	75.0
	347	29.9	30.0
7	392	78.6	79.0
	394	35.7	35.0
8	443	63.3	60.0
	445	25.8	30.0
9	482	75.2	70.0
	484	29.5	40.0
10	534	74.0	73.0
	536	30.1	32.0
11	606	76.5	78.0
	608	29.6	29.0

to be fully open, the magnetic sensor was able to detect that it was 99.6% open. Upon further investigation by the laboratory, it was determined that the valve was indeed not yet fully backseated.

Ultrasonic and magnetic techniques can both follow the disc motion during a pump trip and restart test. Figure 1 shows the output of the ultrasonic instrumentation for rapid closing and reopening of the check valve. Figure 2 is a similar plot from the magnetics. The data in Figures 1 and 2 were taken on valves operating with water as the fluid medium. For one of the test valves, the ultrasonic probe could not be positioned so it could measure the full stroke of the disc. After about 50% closure, the disc moved out of the ultrasonic beam.

Laboratory data on magnitude and frequency of disc flutter were compared with the ultrasonic and magnetic information. It was found that both techniques could accurately measure both magnitude and frequency of disc flutter.

Disc Impact

Impacts of the disc against the seat or valve body can be detected by the acoustic sensors. If ultrasonics or magnetics are used in conjunction with the acoustic sensors, seat tapping or backstop tapping can easily be detected because the

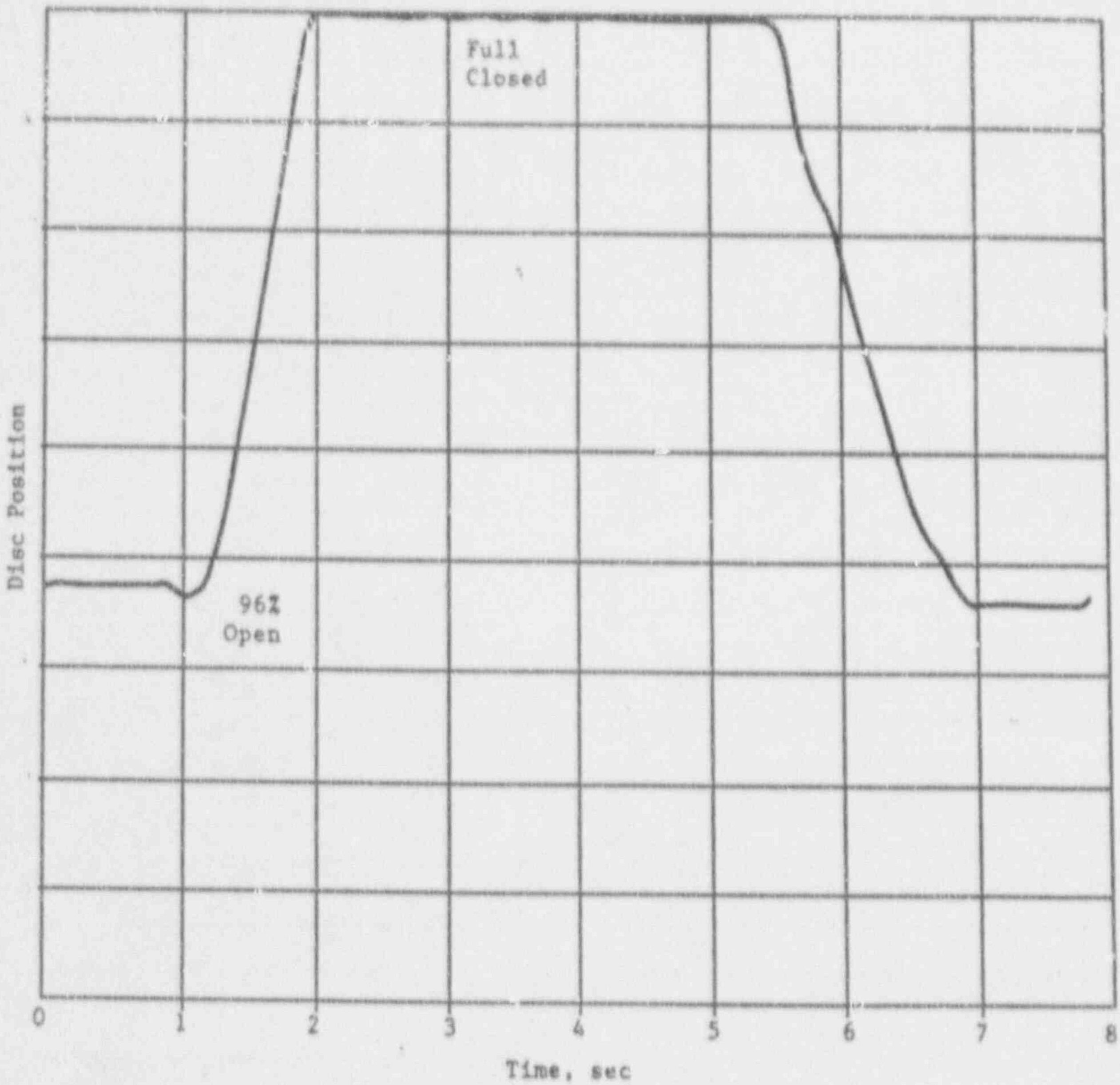

Figure 1. Ultrasonic output for pump trip/start test on 12-inch tilt disc (new).

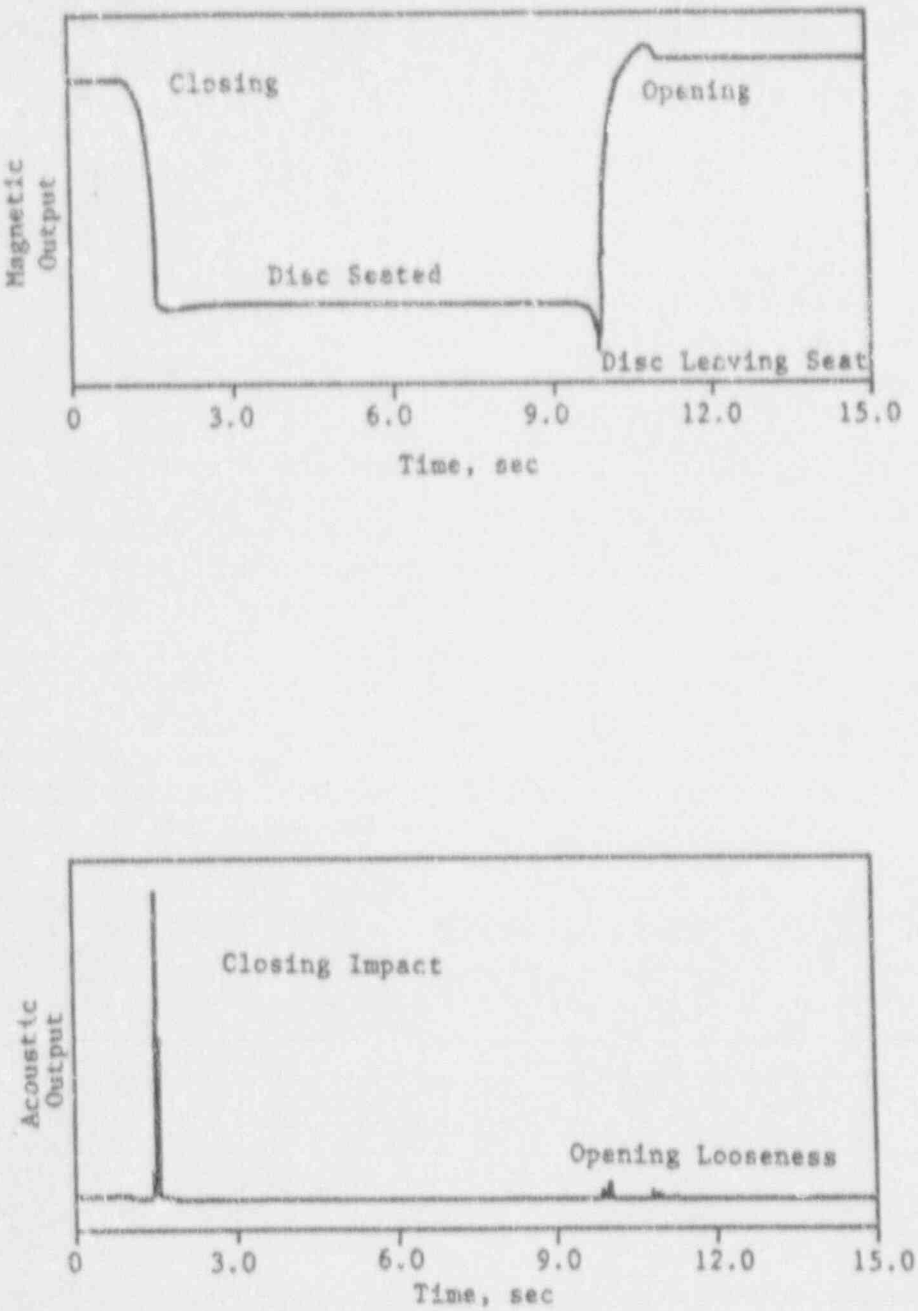

Figure 2. Magnetic and acoustic output for pump trip/start test for 12-inch tilt disc.

mean position of the disc can be measured. The ultrasonics and magnetics are also useful in setting the threshold level to distinguish between those excursions on the accelerometer output that are caused by flow noise and those that are caused by tapping.

If the ultrasonics are used by themselves, seat tappings can be distinguished from backstop tapping if the flow rate is known or by using cross-correlation techniques that compare the output from multiple transducers located at different locations on the valve.

Since the acoustic sensors detect flow noise as well as structural noise, filtering or spectrum analysis to separate the disc noises from the flow noises is sometimes necessary. Filtering can also be used to eliminate the effect of cavitation on the accelerometers signal.

The test data also indicate that a difference in the duration and wave form of the disturbances is caused by cavitation, disc impacting, and loose rubbing parts. The cavitation event usually is a high amplitude and very short event (approximately 5 msec), a disc impact is typically around 30 msec and has a sharp rise with an exponential decay. Rubbing or movement of loose parts typically approximately 100 msec, with a more random wave form.

Maximum benefit can be obtained from the nonintrusive diagnostic techniques if baseline data on a new undegraded valve are available. Periodically testing the valve will produce trending data that will help qualify the amount of wear as a function of time.

The acoustic monitoring system can provide useful information about the instability of valve internals. When combined with either the ultrasonic or magnetic instrumentation, the acoustic monitoring system gives a fairly complete picture of the movement of valve internals. The location and degree of degradation can then be determined.

Detecting Valve Degradations

One of the most useful tests for detecting valve degradation is the pump trip/start test. Figure 2 is a typical plot of such a test, recording acoustic and magnetic information simultaneously. The magnetic (or ultrasonic transducer data) tracks the movement of the disc during the closure and opening. The accelerometer data detects the impact of the disc against the seat during closure and the impact of the disc against the backstop during opening.

The key to detecting the degradation is proper analysis of the accelerometer data at impact. For example, for a new valve, the impact of the disc against the seat during a pump trip usually generates a single, clean spike in the accelerometer data. If the valve contains degraded internals, typically the impact consists of a number of small impacts plus one large impact. Figure 3 is a blow-up of multiple closure impacts caused by 15% hinge pin wear on a 12-inch (30.5 cm) tilt disc check valve.

Degraded internals can also be detected from the magnetic signal from a pump trip test. A degraded valve normally produces a rippling effect in the Hall effects probe as the disc lifts off the seat. For a new valve, the signal is clean as the disc departs from the seat.

Degraded internals can be diagnosed by comparing the magnitude at different frequencies and the spectrum of the accelerometer data for the degraded valve compared to a new valve operating under similar flow conditions. If data are not available for the valve in a new condition, the analysis is less precise. Each vendor has developed different techniques of filtering and performing spectrum analyses and cross-correlations to maximize the data obtained from the accelerometers.

Another method of obtaining hinge pin wear is by use of an ultrasonics transducer focused on the hinge pin. If baseline data are available from the valve when it was in a new condition, the ultrasonic transducer can detect the amount of wear by measuring the drop of the hinge pin from its original position. A similar technique could be employed using magnetics to detect the dropping of the hinge arm.

Degradation such as a stuck or missing disc is easily detected by any of the three techniques. A stuck disc obviously produces a constant output from either the ultrasonic or the magnetic sensor. If the disc is missing, the ultrasonic transducer will fail to locate the disc, and will focus on the valve body, which again will have no motion. If the transducer is focused on the hinge arm (as with a swing check valve), the position of the arm

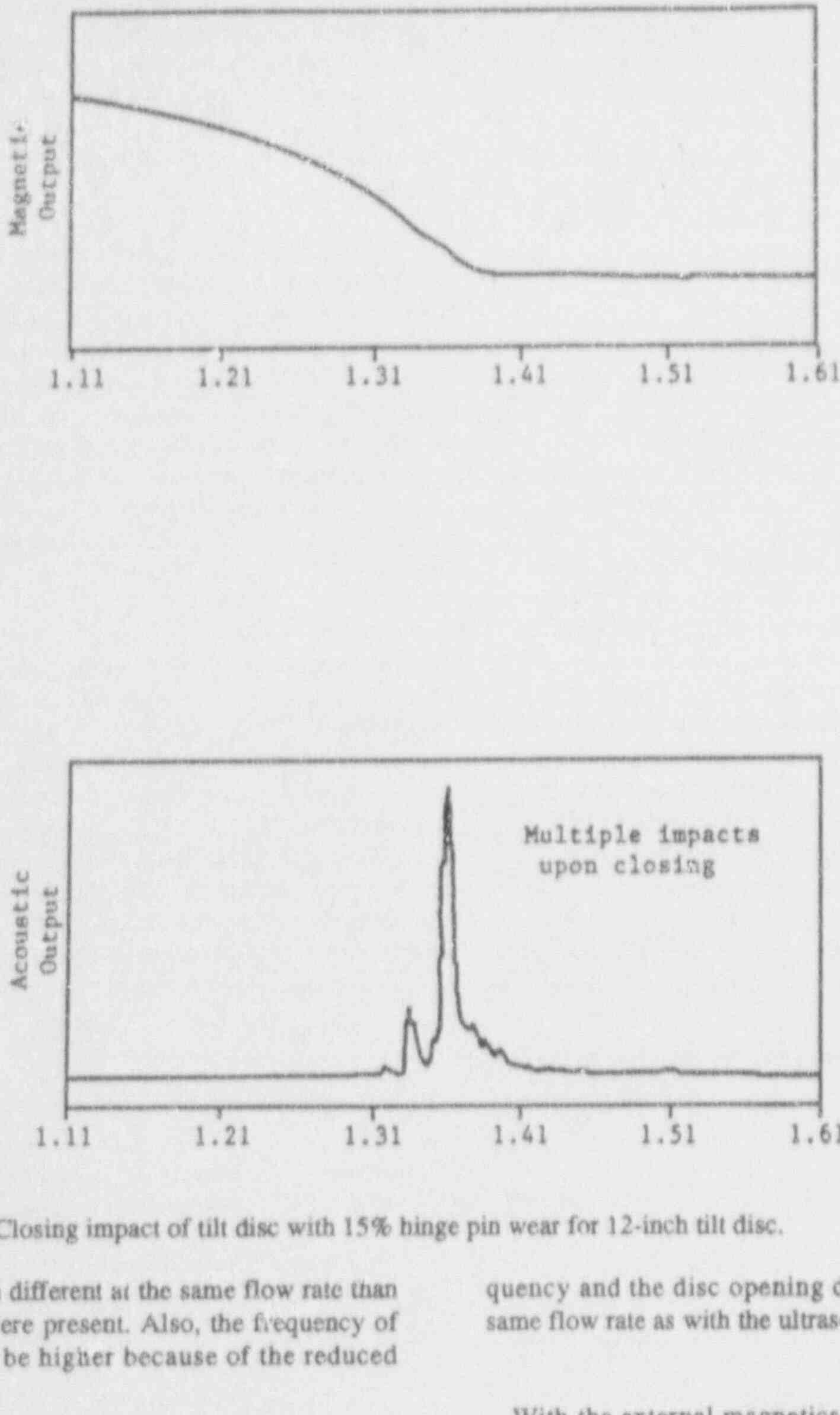

Figure 3. Closing impact of tilt disc with 15% hinge pin wear for 12-inch tilt disc.

will be much different at the same flow rate than if the disc were present. Also, the frequency of motion will be higher because of the reduced mass.

If internal magnetics are used and the magnet is on the disc and the disc laying in the bottom of the valve, there will be little change in the Hall effects probe output. If the permanent magnet is mounted on the disc arm, the flutter will be at a higher frequency and the disc opening different for the same flow rate as with the ultrasonics.

With the external magnetics, if the disc and moving parts are stuck, there will be no output from the magnetic sensor. If the disc is lying on the bottom of the valve and the hinge arm is moving, there will be some signal, but it will be considerably different than if the mass of the disc were attached to the arm.

Space does not allow a detailed discussion of the techniques used by the four different vendors. Also, much of their data processing is proprietary and cannot be disclosed in detail. As a result, the capabilities of each of the technologies are summarized in the following tables. Table 3 shows the four vendor "score cards" for predicting the various degradations.

CONCLUSION AND RECOMMENDATIONS

All three technologies provide useful information about the condition of the internals of check valves. They appear to be a viable alternative to disassembly and visual inspection of check valves in determining the condition of the valve internals. With baseline data available on a new (undegraded) valve, the technologies were generally able to distinguish between a new valve and a valve with degraded internals. They could usually identify the source of the degradation and in a few cases distinguish between degradations. They could determine if the disc was missing, stuck, or operating normally through its entire stroke. The ultrasonic and magnetic techniques were able to determine the mean disc position and identify the magnitude and frequency of disc flutter. The three technologies were able to detect seat and backstop tapping and movement of the internal parts. They could identify those valves that are operating in a stable condition with the disc firmly backseated versus those operating in a highly disturbed condition with the disc not backseated, but experiencing significant motion and corresponding wear.

One significant difference between the evaluation of the technologies in the laboratory and in the field is that for each valve tested as part of this study, the vendors were able to obtain baseline information on a valve in a new condition. Such information may not be available for plant valves. A complete operational history on the test valve will improve accuracy of the prediction for in-plant tests. However, even in the absence of baseline information, useful information can be obtained with these techniques.

As shown in Table 3, vendor score cards for detecting degradations, all three methods demonstrated the ability to determine conditions of check valves. Though no one method proved that it alone could determine the entire condition of the valve, combinations can determine most degradations, including valve position. These methods will not only enhance a predictive maintenance program, but can also enhance an inservice test program by determining valve position and the ability of the valve to perform its intended safety function.

Table 3. Vendor score cards for detecting degradations.

Degradation	Acoustics	Ultrasonics/ acoustics	Magnetic/ acoustics	Ultrasonics/ acoustics (vendor)
	Tests diagnosed correctly			
Hinge pin degradation	13 of 14	14 of 14	14 of 14	8 of 8
Stud pin degradation	2 of 2	2 of 2	2 of 2	2 of 2
Combined hinge and stud pin degradation				
Missing or stuck disc	3 of 3	3 of 3	3 of 3	1 of 1
New valve repeated (as unknown degradation)	1 of 4	2 of 4	3 of 4	2 of 2

Evaluation of Proposed Inservice Testing Procedures for Parallel-Line Safety Injection and Accumulator Discharge Check Valves

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ABSTRACT

This paper presents the results of an evaluation of proposed inservice testing procedures intended to provide full-stroke exercising of two sets of check valves in a pressurized water reactor plant. One set consists of three valves located in the three parallel, combined high- ar .ow-head, cold leg safety injection lines. The other set consists of valves installed in the safety injection accumulator discharge lines. The proposed acceptance criteria for parallel-line valves were based on total flow and pressure drop determination for the three-line system, proving that each line is functioning satisfactorily. The proposed acceptance criteria for the accumulator discharge valves were based on valve flow rate, as determined by two methods: the rate of accumulator pressure decay and the rate of pressurizer level change. The conclusions of the evaluation are that full stroking is not positively demonstrated for either set of valves, that total system flow/pressure drop is an insensitive indicator of flow through each of the parallel lines, and that temperature transients during accumulator blowdown testing should be evaluated for potential structural damage.

INTRODUCTION

At the request of the U.S. Nuclear Regulatory Commission's Office of Nuclear Reactor Regulation, Oak Ridge National Laboratory (ORNL) evaluated two relief requests pertaining to the requirements for quarterly full- or part-stroking and for cold shutdown stroke exercising of two groups of valves in a pressurized water reactor plant. The first relief request was for three combined high- and low- head, cold leg safety injection check valves that are located in three parallel injection lines, as indicated in Figure 1, and the second was for three pairs of accumulator discharge check valves, one pair for each of three accumulators, as shown in Figure 2. Alternative tests were proposed to be conducted during refueling outages.

ORNL's evaluation of each request consisted of an analysis of the proposed testing methodology to assess the validity of the methodology and to identify alternatives that might provide more positive confirmation of acceptable valve performance. This paper presents the results of the evaluations and discusses two issues of possible generic importance that may help plant operators in their inservice testing of check valves used in these applications.

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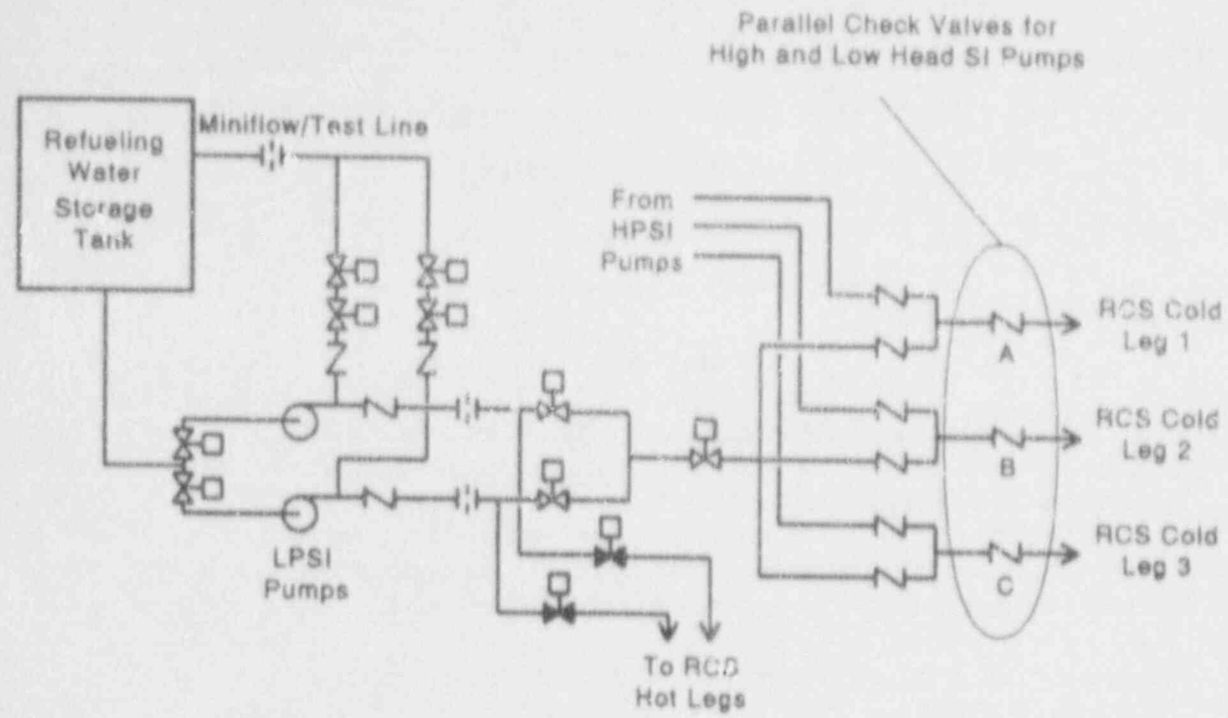

Figure 1. Simplified flow diagram for safety injection system.

COMBINED HIGH- AND LOW-HEAD SAFETY INJECTION CHECK VALVES

In this request, relief was sought from exercising the three combined high- and low-head, cold leg safety injection check valves in accordance with the requirements of Section XI of the ASME Boiler and Pressure Vessel Code. Relief from quarterly full- or part-stroke exercising at power was requested because of the lack of installed instrumentation, the relative system pressures, and the potential for a premature failure of the injection nozzles as a result of thermal shock from cold water injection. Relief from exercising during cold shutdown w[illegible] requested because such testing would requ[illegible]ll-flow injection to the reactor coolant system, where there is insufficient expansion volume to receive the additional inventory.

It was proposed that the satisfactory performance of the check valves be demonstrated by conducting a full-flow test of the low-head safety injection system during refueling outages. The test would be conducted by operating one low-head safety injection pump with suction taken from the reactor water storage tank and discharge through the three parallel injection lines containing the three valves to the reactor coolant system (RCS). With the vessel head removed, the delivered flow ultimately would be collected in the refueling cavity.

Proposed acceptance criteria would be based on establishing that measured total flow delivered by the system is equal to or greater than that determined by analysis of pump and system characteristics for the head imposed on the system at the time of the test. It was claimed that meeting the criteria would verify that acceptable flow is achieved both for the total system and in each of the three parallel branch lines. It was further claimed that the test will verify that each of the check valves is full-stroke exercised by the test, because acceptable branch line flows result in velocities greater than the minimum velocity required to produce full disk lift in the check valves [*Application Guideline for Check Valves in Nuclear Power Plants* (EPRI, 1988)]. The Licensee pointed out

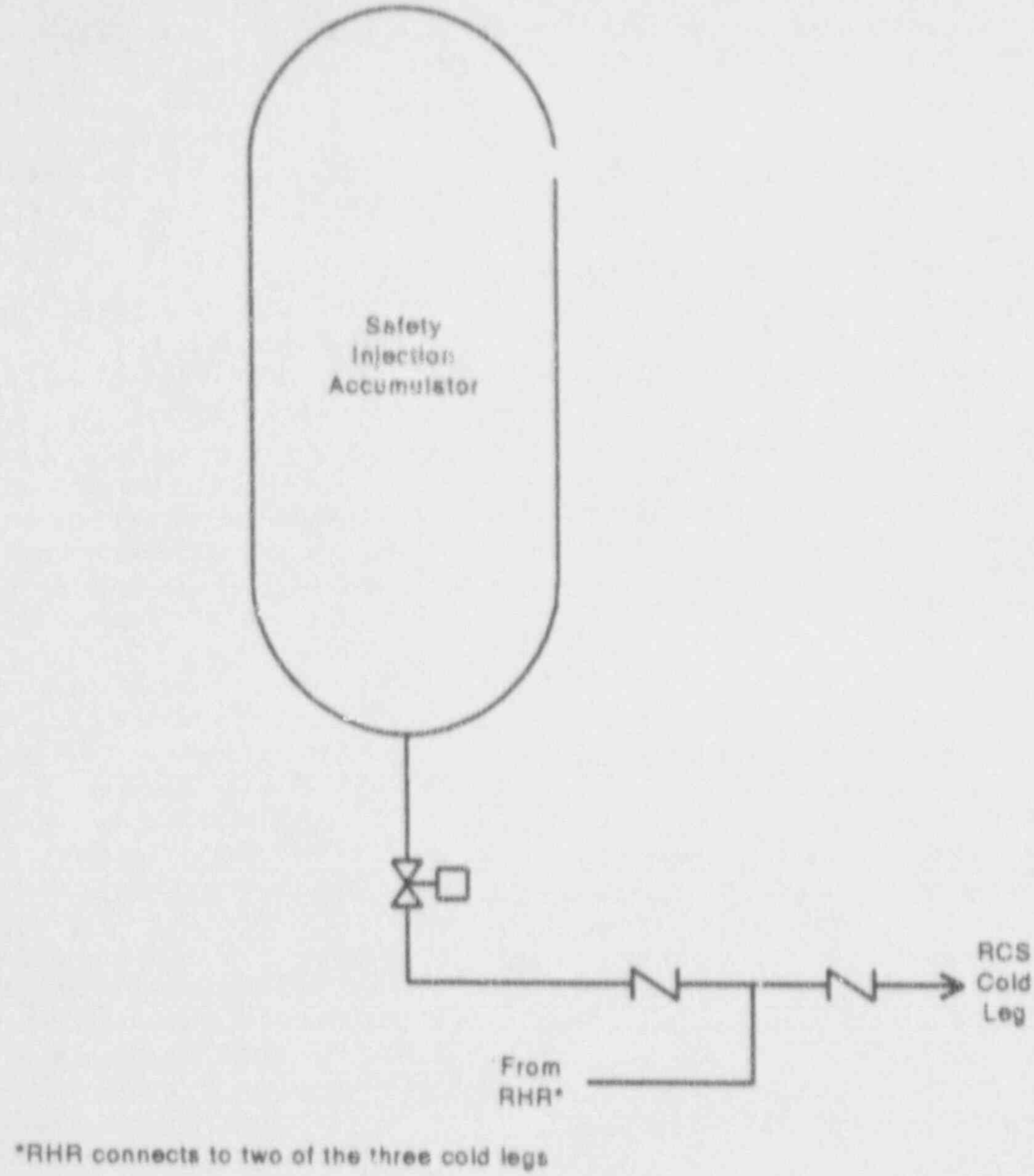

Figure 2. Cold leg accumulator (typical).

that failure to meet the acceptance criteria is an inconclusive result; the check valves may or may not be degraded, and the problem could be somewhere else in the system.

Analysis of Methodology

An independent mathematical model of the piping system and check valves was developed and used to evaluate whether total flow and overall pressure drop through the system would provide adequate definition of branch line flows with implanted deficiencies of check valve performance. The model included 10-in. Schedules40 and 6-in. Schedule 160 common lines and the three parallel 6-in. Schedule 160 injection lines with their check valves. Flow resistance coefficients were estimated for each of the line segments, based on estimates of equivalent lengths of piping for each segment, and were adjusted to reasonably match the values for flow and pressure drop from the intersection of the system resistance and pump performance curves included in the Licensee-provided material.

Three basic cases were considered. In the first two cases, the three injection lines were modeled as being two equal-length lines and one short line running from the common header to the three respective cold legs of the RCS (consistent with the physical layout). No provision for equalization of flow resistances was included. In the first case, the effects of additional flow resistance resulting from a malfunctioning check valve in the short, low-resistance line were evaluated. In the second case, the malfunctioning check valve was moved to one of the longer, higher-resistance lines. In the third case, the three injection lines were modeled as having equal flow resistances and the malfunctioning check valve was placed in one of the lines.

In each case, the total system pressure drop was held constant (at 84 psi) as the check valve resistance was changed and both total flow and flow in each of the parallel lines were allowed to change. In reality, pressure drop would not remain constant as the resistance of one element of the system is changed. Rather, the operating point would shift along the pump performance curve to coincide with the altered system resistance curve. However, since the pump performance curve is relatively flat, with small changes in head for given changes in flow, it appears that the constant pressure drop analyses should be approximately correct.

For each case, the effects of five values for check valve flow resistance, as follows, were evaluated:

Base - valve fully open, with valve disk at 70 degrees (k = 0.29)

a - valve disk at 40 degrees (k = 0.99)

b - loss of one velocity head added to fully open value (k = 1.29)

c - loss of two velocity heads added to fully open value (k = 2.29)

d - a combined line and check valve k of 5000, representing a valve stuck in the nearly closed position.

The values of k for the check valve at the two disk positions were derived from flow coefficient curves in EPRI (1988). The results of the analyses are presented in Table 1.

In Case 1, the various values for check valve flow resistance were applied in the short, low-resistance line. As shown in Figure 3, flow in the line having the added check valve resistance decreases markedly, while the flow in the other two lines increases. The changes in total system flow resulting from rather large additions (up to two velocity heads) to check valve flow resistance are quite small—less than 2%. The changes in individual line flows and in total system flow from adding additional check valve resistance in the high-resistance lines (Case 2), as shown in Figure 4, are even smaller. These changes in flow might be undetectable if the accuracy of flow instrumentation is, for example, ± 2% of full range. In both of these cases, the change in flow from essentially complete blockage of one of the three lines would make this type of failure easily detectable. It should be noted that the addition of two velocity heads to the resistance of one of the check valves produces a decrease in flow in the affected injection line of only 6 to 8%, so the minimum flow requirement for that line *may* be fulfilled (the minimum required flow was not known for this plant).

Case 3 in Table 1 presents the results of analyses where it is assumed that positive means are employed in the design for balancing the flows among the three injection lines. The changes in flow in each of the three injection lines and in total system flow are shown in Figure 5. The percentage changes in individual line and total system flow resulting from increased flow resistances for the check valve in one of the injection lines are intermediate to those resulting from the

Table 1. Modeling with different check valve loss coefficients in the individual parallel lines.

Case 1. Varying resistance in low-resistance line (unequal lines), with total ΔP constant at 84 psi

	Resistances			Flows, gpm			Pressure drops, psi					
Run	K1	K2	K3	Q1	Q2	Q3	DP1	DP2	DP3	ΣQ	ΔQ	ΔQ %
Base	11.9	11.9	8.4	1399	1399	1666	36.0	36.0	36.0	4464		
a	11.9	11.9	9.1	1411	1411	1614	36.7	36.7	36.7	4436	−28	−0.6
b	11.9	11.9	9.4	1416	1416	1593	36.9	36.9	36.9	4425	−39	−0.9
c	11.9	11.9	10.4	1430	1430	1529	37.6	37.6	37.6	4389	−75	−1.7
d	11.9	11.9	5000	1716	1716	84	54.2	54.2	54.2	3515	−949	−21.3

Case 2. Varying resistance in high-resistance line (unequal lines), with total ΔP constant at 84 psi

	Resistances			Flows, gpm			Pressure drops, psi					
Run	K1	K2	K3	Q1	Q2	Q3	DP1	DP2	DP3	ΣQ	ΔQ	ΔQ %
Base	11.9	11.9	8.4	1399	1399	1666	36.0	36.0	36.0	4464		
a	12.6	11.9	8.4	1367	1406	1674	36.4	36.4	36.4	4447	−17	−0.4
b	12.9	11.9	8.4	1354	1409	1677	36.6	36.6	36.6	4440	−24	−0.5
c	13.9	11.9	8.4	1312	1418	1688	37.0	37.0	37.0	4418	−46	−1.0
d	5000	11.9	8.4	81	1660	1975	50.7	50.7	50.7	3716	−748	−16.8

Case 3. Varying resistance in one line (equal lines), with total ΔP constant at 84 psi

	Resistances			Flows, gpm			Pressure drops, psi					
Run	K1	K2	K3	Q1	Q2	Q3	DP1	DP2	DP3	ΣQ	ΔQ	ΔQ %
Base	10.7	10.7	10.7	1488	1488	1488	36.7	36.7	36.7	4464		
a	10.7	10.7	11.4	1497	1497	1450	37.1	37.1	37.1	4444	−20	−0.4
b	10.7	10.7	11.7	1500	1500	1435	37.3	37.3	37.3	4435	−29	−0.6
c	10.7	10.7	12.7	1511	1511	1387	37.8	37.8	37.8	4409	−55	−1.2
d	10.7	10.7	5000	782	1782	82	52.5	52.5	52.5	3646	−818	−18.3

Notes:
Includes common line pressure drops
Values for common line k's: 10-in. Sch 40—10.4
(all cases) 6-in. Sch 160—0.8

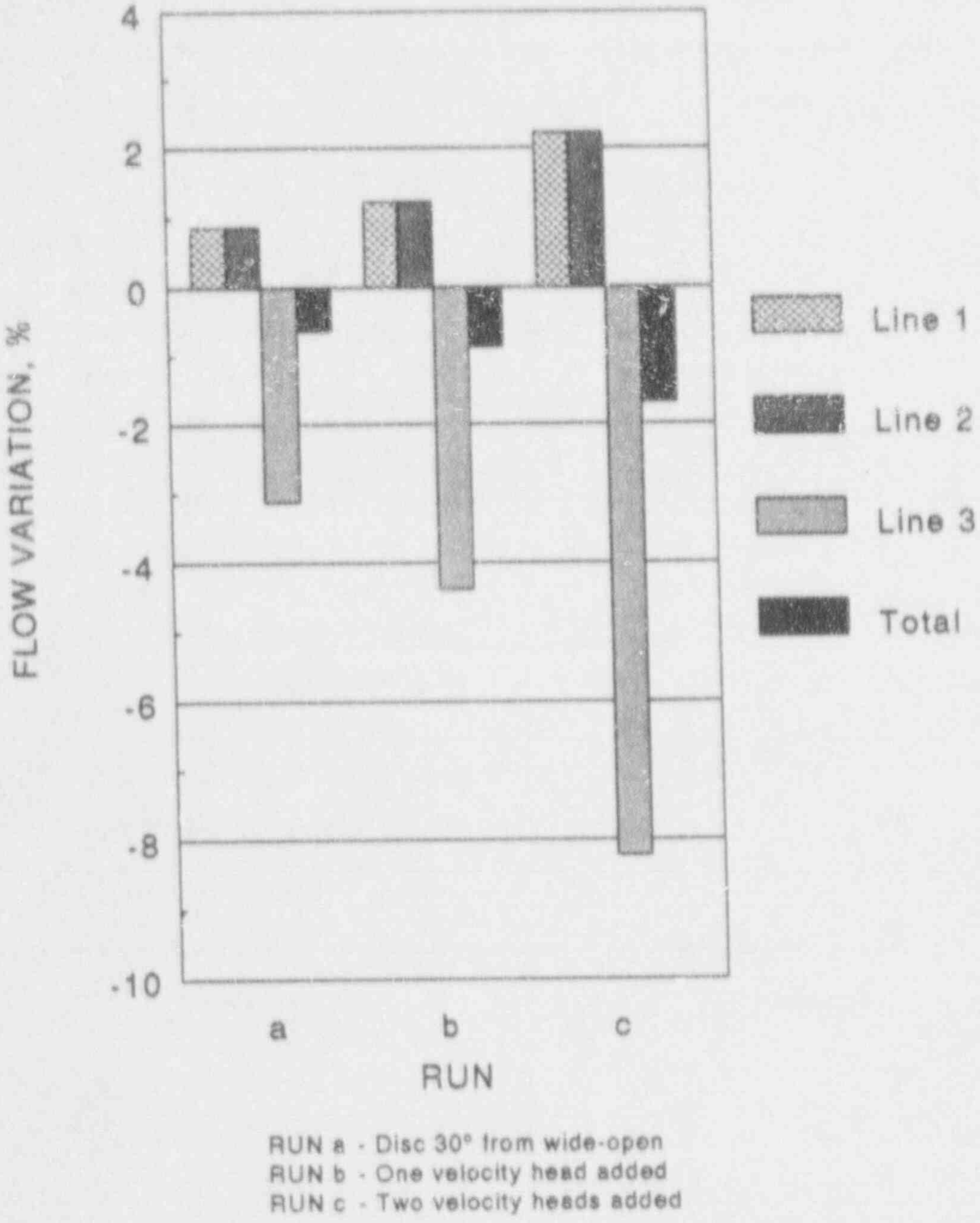

Figure 3. Variation in flow due to added resistance in low-resistance line.

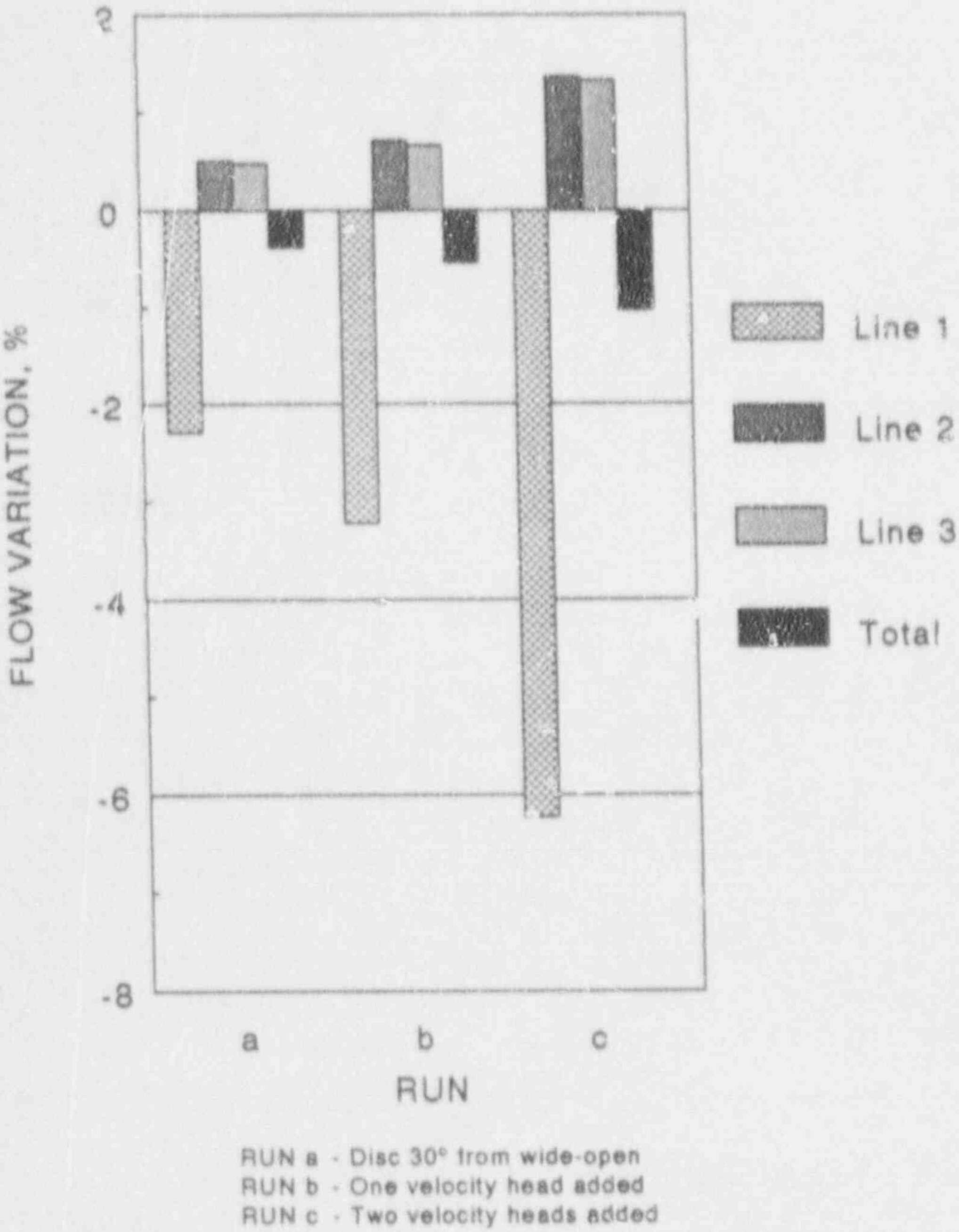

Figure 4. Variation in flow due to added resistance in one high-resistance line.

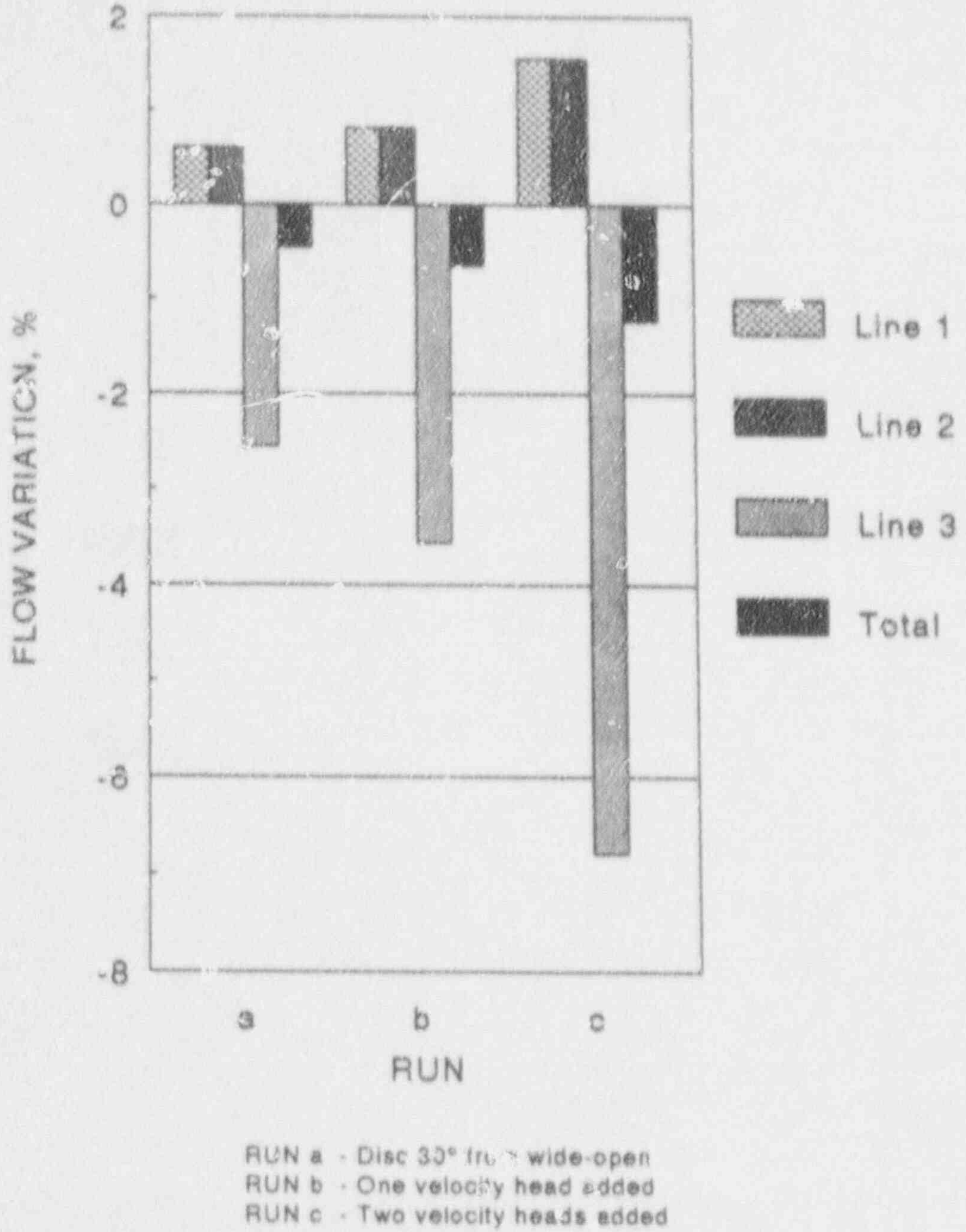

Figure 5. Variation in flow due to added resistance in one of three equal lines.

unequal injection line cases with the increased valve resistance in the low-resistance line (Case 1) and with the increased resistance added to one of the high resistance lines (Case 2). Again, the change in flow rate in the affected line from adding two velocity heads to the normal wide-open check valve resistance is rather small (less than 7%).

Conclusions and Recommendations

Establishing that the flow through a check valve is greater than the minimum flow for full disc lift as given by EPRI (1988) does not assure that a given valve will full-stroke at that flow. The primary purpose of the EPRI Guidelines minimum flow recommendations is to provide an approximate indication of the flow required to prevent disc flutter and accelerated wear of the valve internals. A degraded valve may not fully open at that flow.

Total system flow is an insensitive indicator of a malfunctioning check valve in one of the three parallel injection lines. A check valve that only opens to a 40-degree disk position, instead of to the 70-degree position corresponding to wide open, has little effect on total system flow at a fixed total system pressure drop, and the change in system flow that would result from a degraded check valve that introduces an additional pressure drop of two velocity heads is probably at the limit of detectability from total flow measurement.

It is therefore concluded that the proposed test will determine that the *system* either is or is not capable of delivering design basis total injection flow, may not determine that branch flows through the individual injection lines are acceptable, and will not positively demonstrate full-stroking the individual check valves.

It is recommended that (a) either permanent or temporary instrumentation be employed to measure flow in each of the three parallel injection lines, (b) a nonintrusive diagnostic method such as acoustic emission be employed during the proposed test to demonstrate that the valve is fully opening during the test, or (c) the valves be disassembled and inspected at some interval to prove that the valve is in good condition. Acoustic emission testing, using either vendor-supplied or utility equipment, is being effectively used for this purpose by other utilities.

Alternative (a), flow measurement, has been the historically accepted means of demonstrating check valve operability, and also provides an indication of overall flow path condition. While it is not a sensitive indicator of check valve position, it provides reasonable assurance that the valve and the associated line will pass the required flow rate. Alternative (b), nonintrusive diagnostic measurement, can provide an indication that the valve fully strokes (if flow is sufficient to accomplish this) and/or provide an indication of disk position change. Alternative (c), disassembly and examination, is the least preferred alternative. It should be noted that OM-22 is investigating the use of a sampling program using nonintrusive techniques and periodic disassembly.

ACCUMULATOR DISCHARGE CHECK VALVES

In this request, relief was sought from exercising the three safety injection accumulator discharge check valves in accordance with the requirements of Section XI. Relief from full- or part-stroke exercising at power was requested because of the high differential pressure between the reactor coolant system and the accumulators. Relief from exercising during cold shutdown was also requested because of a lack of installed instrumentation and an uncontrolled test volume change needed to achieve the flow required by the safety analysis.

It was proposed that the accumulators be blown down during refueling outages, with flow through the check valves to be determined by two independent methods. First, the decay of accumulator pressure during the blowdown would be used, via a computer model of the accumulator/pressurizer system with a wide-open check valve, to calculate the flow rate through the valves. Second, the rate of level change in the pressurizer would be used to calculate the flow rate.

Two acceptance criteria for the demonstration of full-stroking of the valves were proposed, as follows:

1. The actual measured accumulator pressure vs. time curve for each test shall lie to the left (on time axis) and below (on pressure axis) the curve of pressure vs. time produced by the computer model.

2. The actual flow rate, determined from each of the two estimation methods (accumulator pressure and pressurizer level), shall be greater than the minimum flow required for

valve full disk lift. This minimum flow value was to be obtained from EPRI (1988).

Test data and results included with the relief request showed some variation in flow from the three accumulators and for the two test series represented. These variations were attributed by the Licensee to differences in accumulator pressure at the start of the tests and to differences in venting of the pressurizer. The flows calculated from pressurizer level rate of change were well above the EPRI (1988) minimum flow required for full disk lift, but were considerably lower (16–70%) than flows indicated by the utility's accumulator pressure decay model. The curves of actual measured accumulator pressure vs. time all fell below and to the left of the model-derived curves.

Analysis of Methodology

Three aspects of the methodology and acceptance criteria were troublesome. These were (a) the consistent mismatch of the curves for measured and model-produced accumulator pressure vs. time; (b) the consistent mismatch of flow based on pressurizer level and flow based on accumulator pressure; and (c) whether a test that satisfies the acceptance criteria truly demonstrates that the check valve is full-stroked and in good condition. Further discussion of these separate aspects follows.

1. The mismatch of pressure vs. time curves, measured vs. modeled, indicates that the model does not accurately represent the actual process. Examination of the FORTRAN listing of the model revealed that the expansion of the nitrogen in the accumulator during blowdown was modeled as an isothermal process, in which

$$P_2V_2 = P_1V_1 \quad . \qquad (1)$$

Figure 6 presents the measured and calculated accumulator pressure for the test of Accumulator A described in the supporting [illegible] for the relief request. The data [illegible] peak velocity through the valve of 31.1 ft/sec, corresponding to a peak flow rate of 7800 gpm.

Actually, the expansion is undoubtedly accompanied by a decrease in gas temperature and should be represented as a polytropic expansion by the expression

$$P_2V_2^{\,n} = P_1V_1^{\,n} \quad . \qquad (2)$$

This form of equation was substituted into the Licensee's model and trial runs were made using the test data and values of n ranging from 1 (isothermal) to 1.4 (isentropic for nitrogen). The best fit of measured to modeled accumulator pressure vs. time was obtained with a value of 1.4 for n, indicating that the process closely approximates an isentropic expansion. (This result is not unexpected when the rapidity of the expansion, with little opportunity for the nitrogen to gain heat from the vessel wall and the underlying liquid, is considered.)

Figure 7 presents the results for accumulator pressure and flow obtained from the modified model for the same test. Note the greatly improved fit of calculated and measured pressures. The peak flow occurs early in the blowdown and is approximately 7350 gpm (compared with the value of 7800 gpm from the isothermal case). Figures 8 and 9 show the model results for the tests of the other two accumulators (B and C). Similar excellence of match between measured and calculated pressures is evident for these tests.

A point of interest not shown in these figures is the large temperature drop experienced by the nitrogen in an isentropic expansion of this magnitude. If an initial nitrogen temperature of 100°F is assumed, the final temperature following expansion between 135 psia and 50 psia, as in test A, would be approximately –38°F. Determination of whether such a temperature excursion imposes a significant structural stress on the accumulator is beyond the scope of the present investigation.

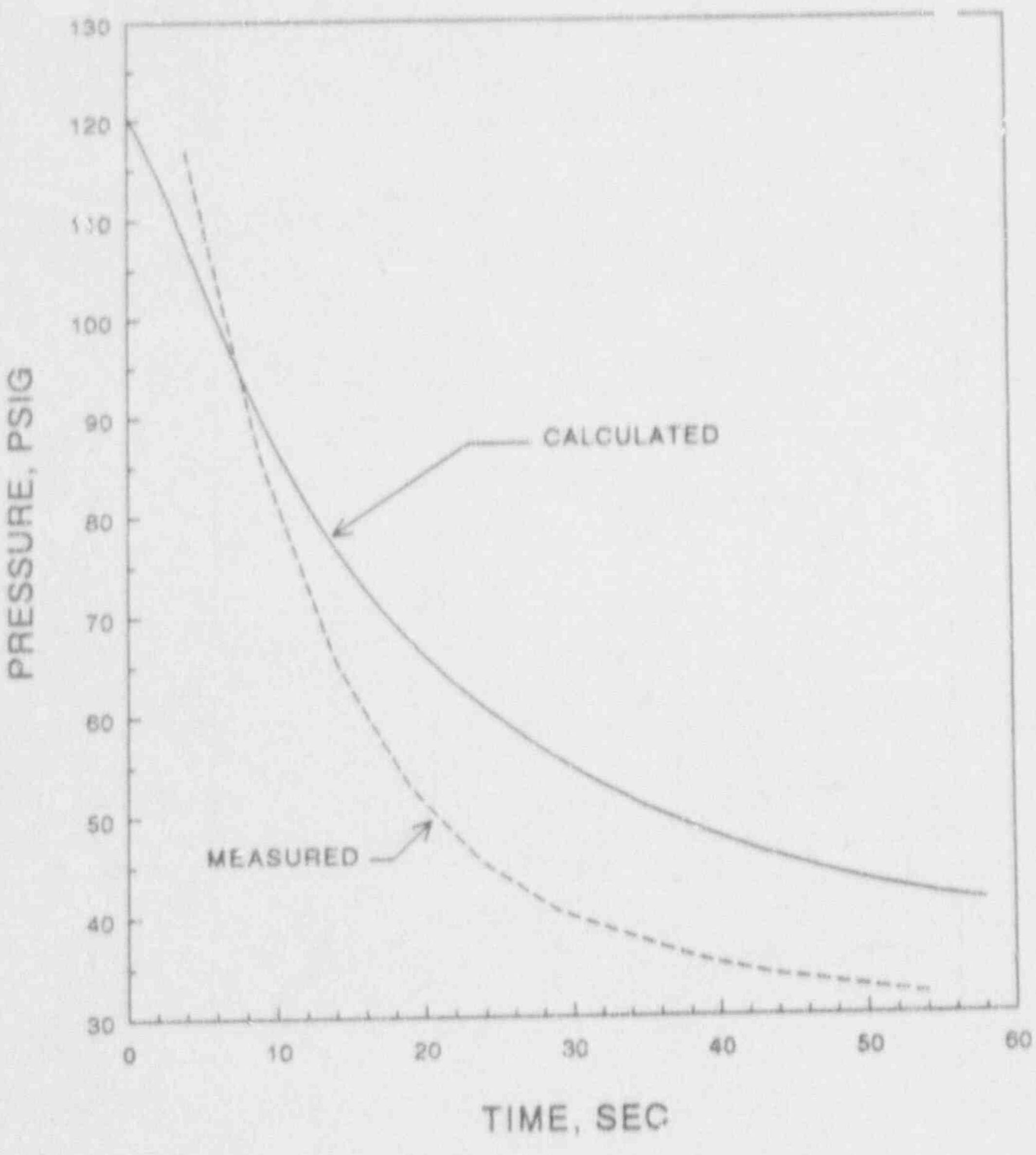

Figure 6. Accumulator pressure decay (Test A).

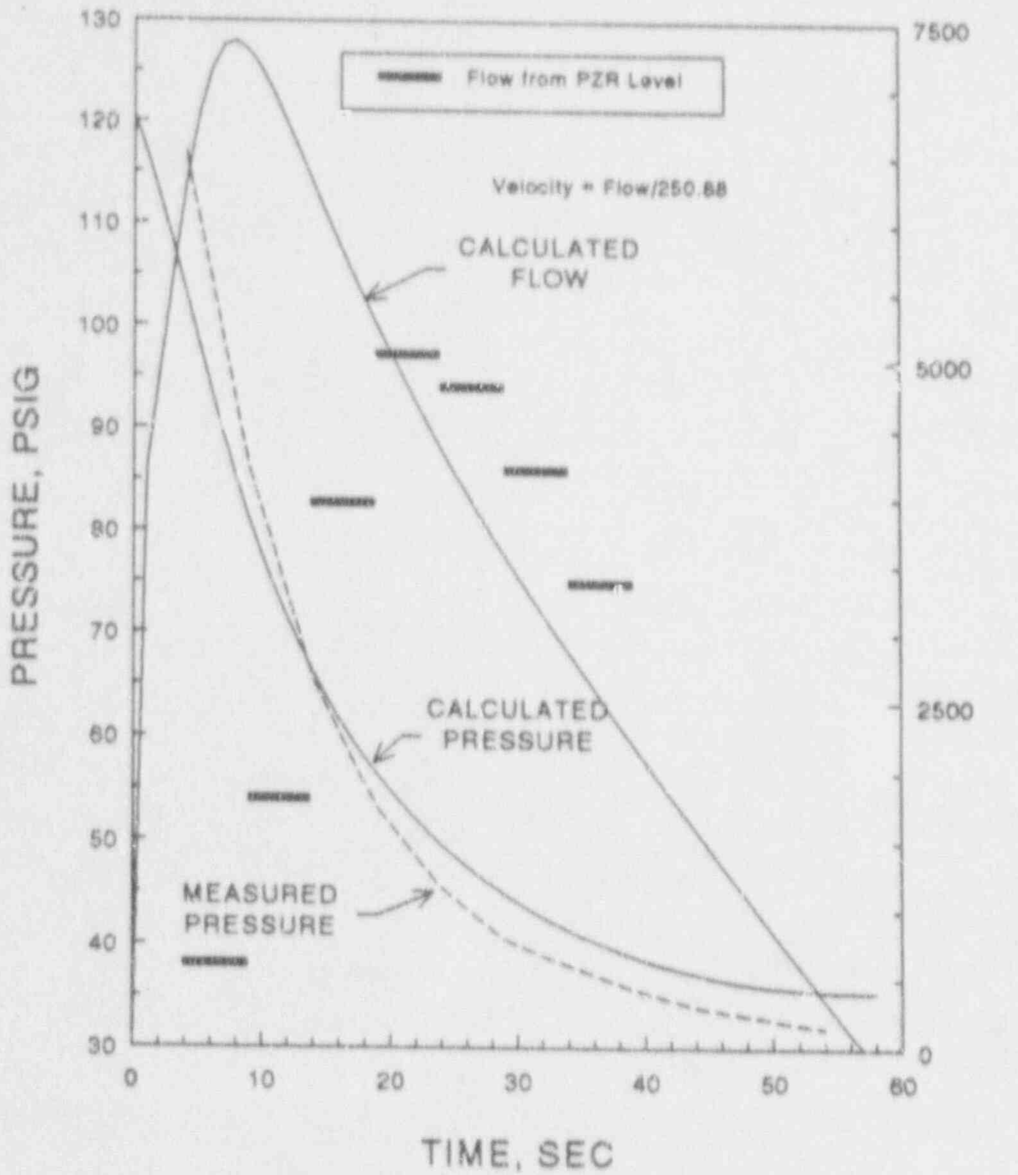

Figure 7. Flow and accumulator pressure (Test A, with n = 1.4).

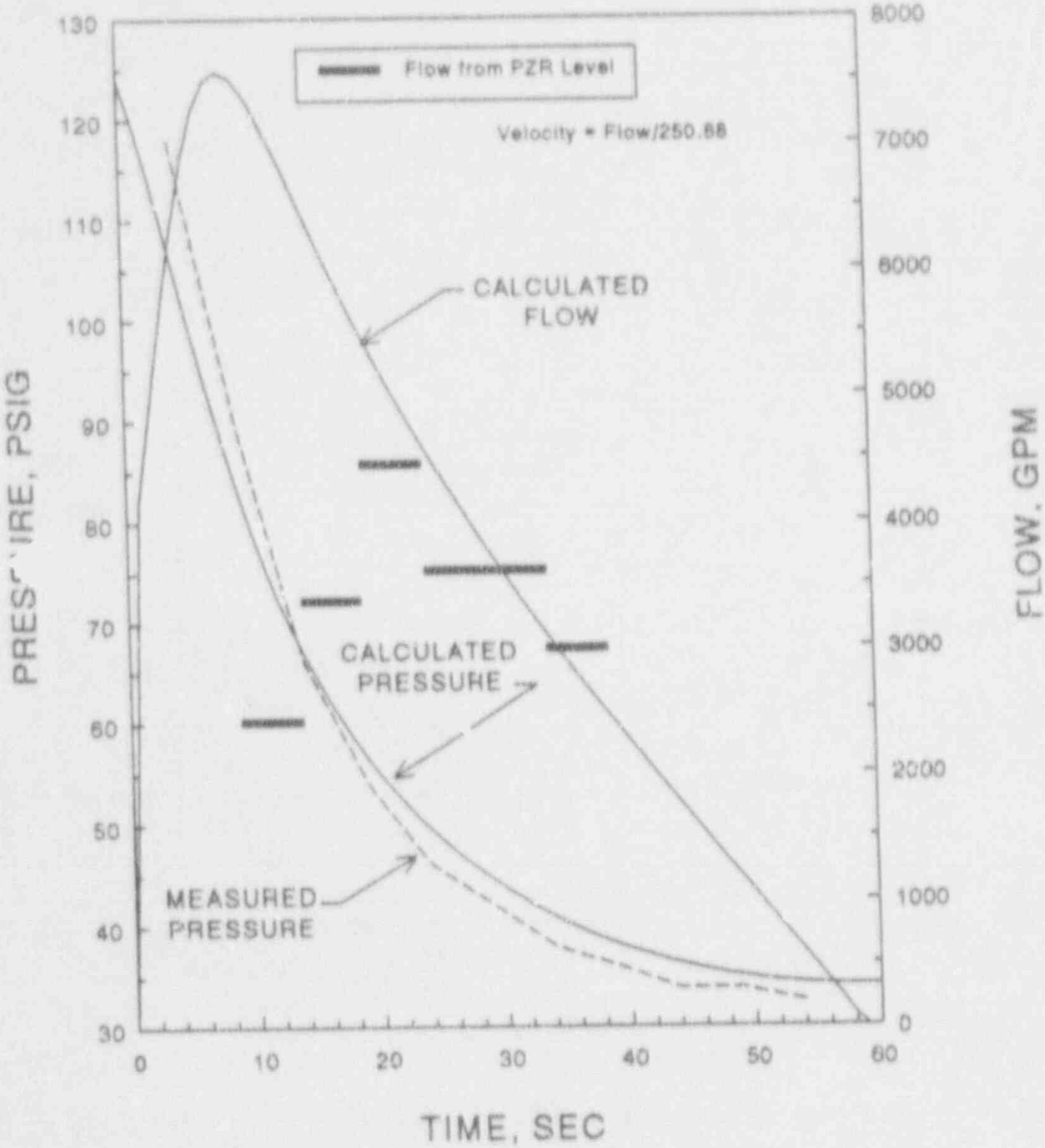

Figure 8. Flow and accumulator pressure (Test B, with n = 1.4).

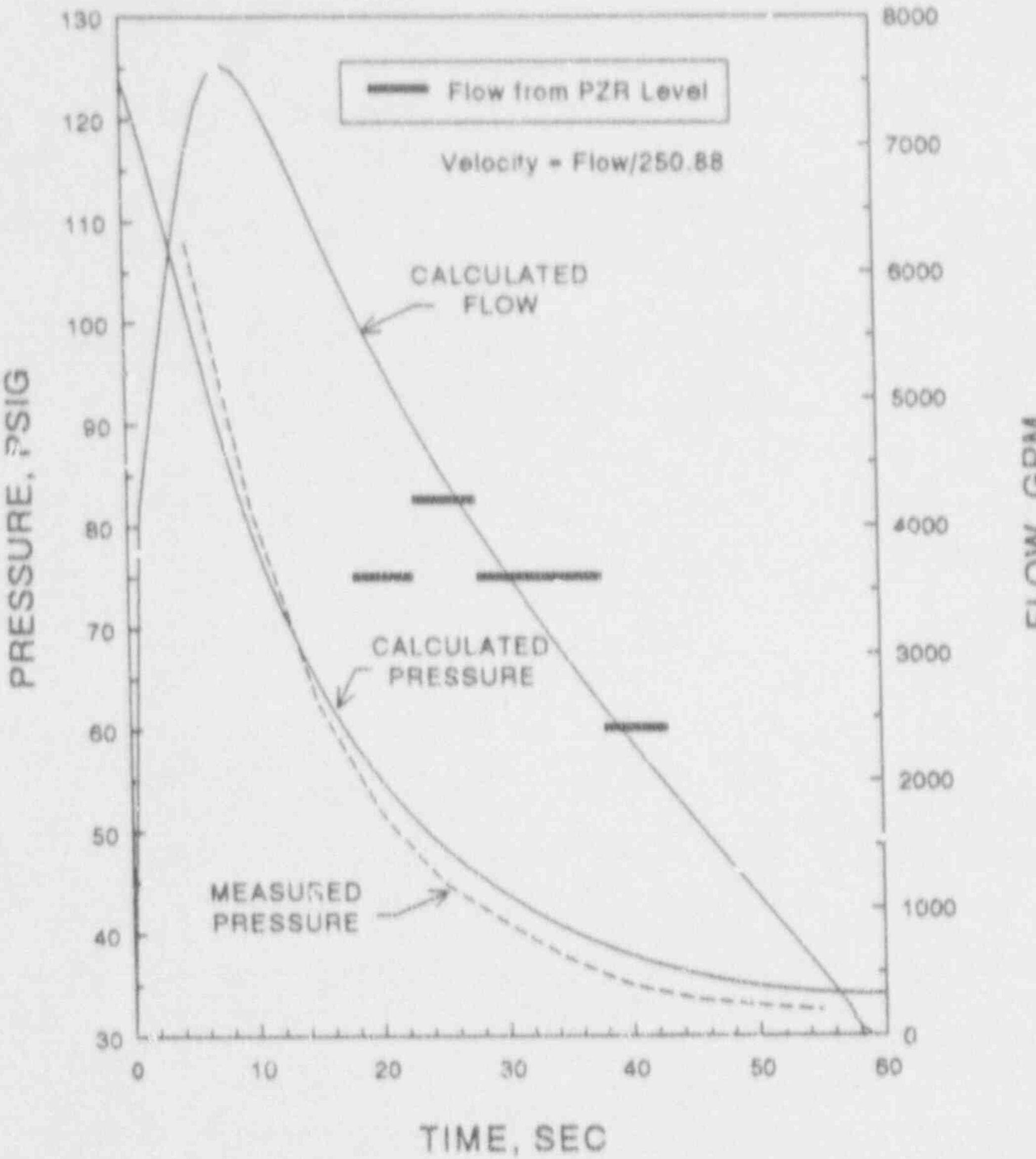

Figure 9. Flow and accumulator pressure (Test C, with n = 1.4).

2. Flow as calculated by the utility from pressurizer level change is also shown in Figures 5, 6, and 7 by the heavy bars. The pressurizer-derived flows build up much later in the blowdown and to much lower peak values than those derived from the accumulator pressure model. Later in the blowdown, the flows derived from the two methods track fairly closely. These differences are typical of what would be expected if the early portion of blowdown flow were diverted to filling an unidentified void in the RCS system. The differences may, alternatively, be the result of other unknowns, such as process computer sampling times or instrumentation response.

3. The acceptance criterion that the measured accumulator pressure vs. time curve fall to the left and below the modeled curve has dubious value in that it demonstrates, primarily, that the model is imperfect. Flow rate based on pressure decay in the accumulator should be reasonably accurate

if the model is truly representative of the process.

Flow rate based on rate of change of pressurizer level should also be accurate, if all of the flow winds up in the pressurizer and if the level instrumentation is accurate. As discussed above, however, the flow rates from the two bases are quite different, even with the modified model, so the true flow rate through the valve under test is not known with certainty.

The EPRI Guidelines (1988) value for estimated minimum velocity required to maintain a check valve in the wide-open position is provided to prevent oscillation of the disk and premature wear of the disk support hinge. Exposure of a valve to this velocity does not ensure that a possibly degraded valve will open fully. Furthermore, the Guidelines are just what the name implies, and should not be interpreted as absolute indication of a particular valve's minimum stable flow.

To provide insight into the ability to detect a partially open valve from the accumulator pressure vs. time curve, the model was further modified to permit inputting any value of k (the check valve flow resistance coefficient). From data presented in Figure D-1 of the EPRI Guidelines, extrapolating from 6 to 12-in. valves and converting C_v to k, the values of k for a wide-open (70-degree disk angle) 12-in. check valve and for one with a disk angle of 40 degrees are 0.27 and 0.86, respectively. Figure 10 presents the results obtained from runs of the modified model for these two valve positions while using test data from Test C. It is obvious that measured accumulator pressure decay is a very insensitive indicator of disk position. Further, the peak flow for the 40-degree disk angle case was only 2.5% less than the peak flow for the wide-open case— difference that probably is undetectable when flow rate is estimated from either accumulator pressure decay or pressurizer level.

To further investigate the temperature excursion of the nitrogen in the accumulator during the expansion process, a case approximating an accumulator blowdown under accident conditions was evaluated. The initial conditions of the accumulator were 150°F and 620 psia. At the time all of the water content of the accumulator had been ejected, the pressure had dropped to approximately 117 psia. Under this final condition, which was reached in approximately 45 seconds, the nitrogen in the accumulator had cooled to approximately minus 81°F.

Conclusions and Recommendations

The proposed tests and analysis appeared deficient both in the ability to determine flow accurately through the check valves during the tests and to prove that the valves have been fully stroked. The model used by the Licensee to predict accumulator pressure decay during blowdown is flawed in that the predicted pressure decay does not closely approximate the measured decay, and the acceptance criterion that the measured decay curve lie below and to the left of the predicted pressure did not provide a useful measure of valve acceptability. Even when the model was modified to provide accumulator pressure decay more closely matching the measured decay, the peak flow rate predicted by the model is considerably higher than that calculated from pressurizer level change.

If it is assumed that one or the other of the flow rates determined by these two methods is nearly correct and that it exceeds the EPRI Guidelines minimum flow to provide full-stroking of a similar valve, there is still no assurance that the particular valve being tested has full-stroked. Our analysis shows that a valve that will not stroke beyond a 40-degree disk position, where full open corresponds to a 70-degree position, will have essentially undetectable effect on either the decay pressure curve or the peak flow rate. Therefore, there is no way to detect a partially open valve from the test.

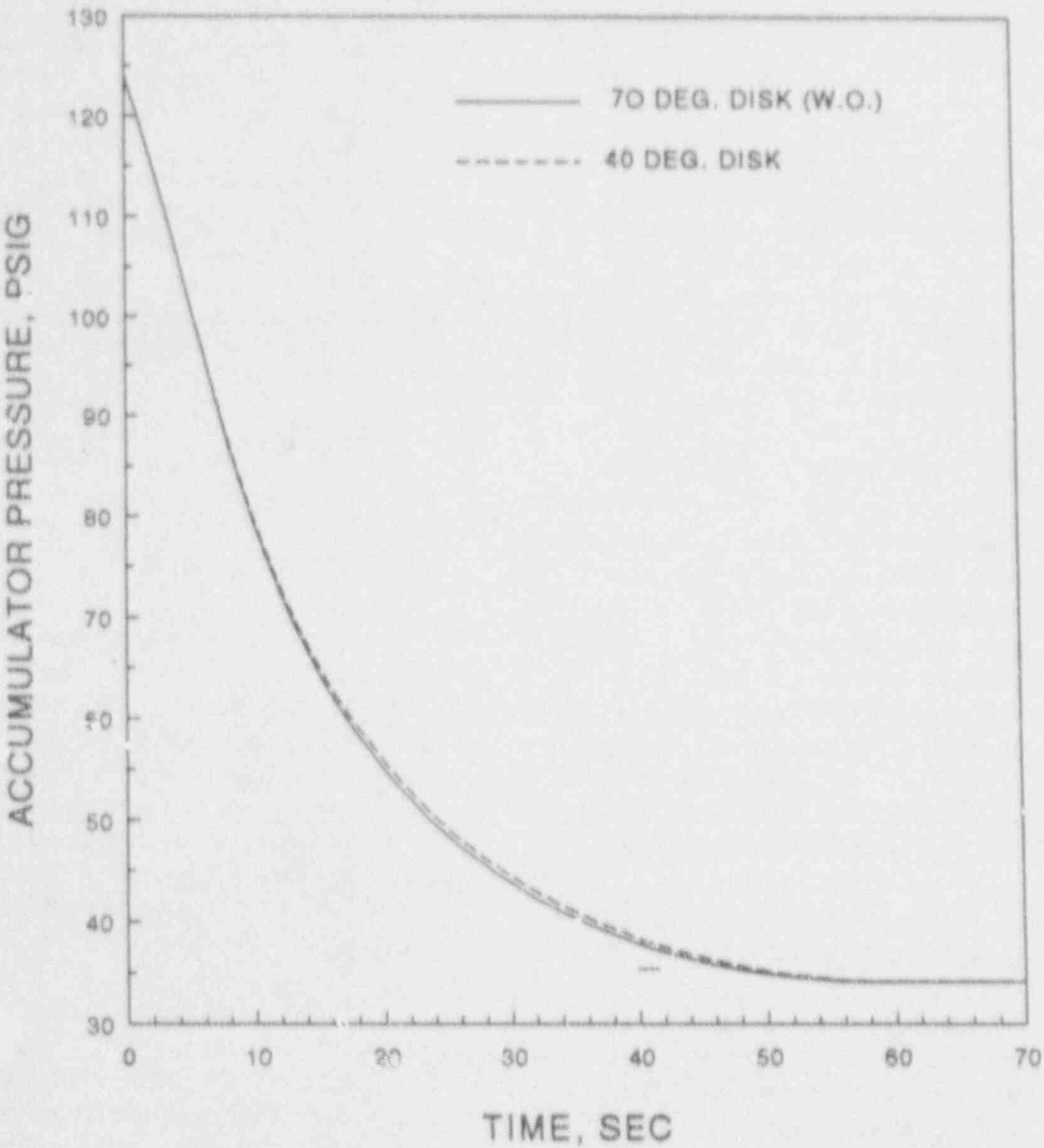

Figure 10. Effect of disk position on pressure (based on Test C).

The flow rates attainable during a test in which the accumulator initial pressure is in the range used in the reported tests do not approach the design basis flow (a total of approximately 34,000 gpm per the plant's final safety analysis report, for all operational accumulators). Thus, on this basis, the test does not fulfill the requirements for passing the maximum required accident condition flow as delineated in Position 1, Attachment 1, of Generic Letter 89-04.

It was therefore recommended that the proposed test be judged to provide proof only of partial stroking of the valves and that some alternative or additional procedure be implemented to demonstrate acceptability. These other procedures could include use of a nonintrusive testing method and/or disassembly and inspection. One nonintrusive method that has been applied to the same valve application by other Licensees is that of measuring acoustic emissions to detect the valve disk assembly striking the backstop upon full opening. A clear acoustic indication of the disk assembly striking the backstop, along with successful nonleakage performance during normal operation, is proof of full-stroking.

GENERAL CONCLUSIONS

Two issues of possible generic importance became evident from the evaluation of these relief requests. The first issue pertains to the use of flow or pressure drop in a piping run as an indication that a check valve in the run either is or is not fully open. Our analyses show that only small changes in either flow or pressure drop result from the valve disc moving from fully open to a position 30 degrees from fully open. Thus, a test using flow or pressure drop to prove full-stroke exercising of a valve is of questionable validity, and a more positive method of determination is called for to demonstrate full stroking. Conversely, the valve does not have to be fully open to pass essentially full flow at a given imposed driving pressure drop.

The second issue pertains to blowdown tests of accumulator-driven safety injection systems. The rapid expansion of the pressurizing gas during blowdown, with little opportunity for heat gain from the vessel or the underlying water results in extremely low temperatures in the gas. While the thermal coupling between the gas and the vessel wall may be rather soft, it appears that an investigation of the possibility for structural damage to the accumulator wall from such repeated thermal excursions may be warranted. [After the ORNL evaluation, it was learned that Westinghouse Electric Corporation had informed utilities having Westinghouse-designed PWRs that using accumulator pressure as the driving force for testing the operability of the check valves has the potential for causing unanalyzed thermal transients when done repetitively for inservice testing purposes. Westinghouse recommended that these transients be evaluated on a plant-specific basis. For one PWR plant, a total of 80 transients over plant lifetime was eventually authorized.]

REFERENCE

Electric Power Research Institute, 1988, *Application Guidelines for Check Valves in Nuclear Power Plants*, EPRI NP-5479, prepared by MPR Associates, Inc., and Kalsi Engineering, Inc., for the Electric Power Research Institute, Palo Alto, CA.

Session 3C
MOV Foreign Experience

Session Chair
Helmut Knoedler
Siemens-KWU

A Review of Regulatory Issues Associated with the Testing of Valves for the Sizewell B PWR in UK

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ABSTRACT

This paper describes the regulatory approach taken on valve testing for Sizewell B, the UK's first pressurized water reactor (PWR), from pre-licensing to the present day when the mechanical plant is undergoing commissioning.

In particular, the paper describes regulatory issues and concerns associated with the following valve testing programmes associated with the design of Sizewell B:

- A brief resume of endurance testing of a range of commercially available small isolation valves. This study enabled the licensee to eliminate from the design certain valve types that failed to meet the specifications set.
- A programme of testing for the pressurizer safety relief system (three tandem pairs of pilot-operated and two conventional spring loaded safety relief valves), which addressed low pressure sub-cooled operation, loop seal effects, borated water operation, valve adjustment for blow-down and accumulation, and in-situ testing.
- Testing of main steam valves (MSIV and MSSV) with regard to flow interruption capability requirements akin to NRC Generic Letter 89-10 initiatives. Also, several candidate main steam safety valves were tested with respect to control ring setting and in-service assisted lift testing. The requirement for full flow testing was identified as a result of the above work.

INTRODUCTION

In the United Kingdom (UK), the main legislation governing the safety of nuclear installations is the Health and Safety at Work etc Act 1974 and the associated relevant statutory provisions of the Nuclear Installations Act 1965. Under the Nuclear Installations Act no site may be used for the purpose of installing or operating any commercial nuclear installation unless a nuclear site license has been granted to a body corporate by the Health and Safety Executive (HSE) and is for the time being in force. HM Nuclear Installations Inspectorate (NII) is that part of HSE responsible for administering this license function.

Historically the UK civil nuclear programme has been based on the gas-cooled reactor technology of the Magnox and advanced gas reactor (AGR) stations; however, the Sizewell B Nuclear Power Station is the UK's first pressurized water reactor (PWR). It comprises the standard Westinghouse four-loop PWR, with station layout based on the SNUPPS plants, as built at Calloway and Wolf Creek in the United States.

The station reference design has been adapted in order to address regulatory requirements in UK and national grid aspects. A Public Inquiry on the Licensing of the station took place from 1983 to 1985.

Subsequent to the Inquiry, the detailed design of the plant has progressed, and discussions between the licensee, Nuclear Electric (NE) and NII on various engineering issues took place, including aspects of valve reliability, operability, and testing.

This paper describes the salient regulatory issues associated with the testing of valves for Sizewell B, from the time of the Inquiry to the present, when the mechanical plant is undergoing commissioning. Some of the more important test results are included to illustrate the description.

REGULATORY CONCERNS ON THE NEED FOR TESTING OF SAFETY-RELATED VALVES FOR SIZEWELL B

It was apparent, at an early stage in the design process, that international experience, notably in the United States, indicated numerous problems with operability of safety related valves.

In particular, HMNII was concerned that failure rate data to be used in probabilistic risk analysis (PRA) studies might not accurately reflect actual field behavior. Furthermore, the advent of PWR technology introduced new environments and demands on valves, of which there had been little experience in the UK. As an example, the use of wedge gate valves is not prevalent in the UK Civil Reactor Programme, and this proved significant in relation to the motorized valve diagnostic programme described later. Finally, full-flow testing was considered necessary for key safety related valves.

With regard to in-service testing, HMNII's position was clear from the outset: certain safety-related valves such as relief valves should be tested in-situ wherever possible and as near to full flow as possible. The concept of bench testing relief valves, which was prevalent in UK on Magnox Gas-Cooled Reactors at the time, was being re-examined and it was believed that the modern plant should be tested at conditions more akin to those that it would experience on demand under fault conditions. The pressurizer and main steam safety valves were considered of special concern. Details of testing are described in the next section.

Recognition of the difficulty of assuring repeatable set-point and valve characteristic when safety relief valves (SRVs) were subject to multi phase flow resulted in studies at the Nuclear Electric Marchwood Engineering Laboratories in UK. The effect of the loop-seal on pressurizer relief valve operation had been identified as a concern in the United States during Electric Power Research Institute (EPRI) studies and the need for further examination was accepted by the licensee of Sizewell B.

As a result of the Three Mile Island review, the licensee approached NII with a view to changing from the SNUPPS configuration of power-operated relief and block valves on the pressurizer to the use of a tandem pair of French Sebim pilot-operated valves. This valve is required to discharge both water and steam under fault conditions, and its qualification is described later.

Eventually, after consideration of considerable operational experience in France and with a pilot-operated safety relief valve system on the UK AGR at Dungeness B, it was agreed that such an approach would be acceptable.

In the case of the main steam system, the Sizewell B safety case assumes that for the main steam line break event (MSLB), the MSIV must isolate the breach in less than 5 seconds. Such an event is capable of producing the effect of a two-phase fluid front on the valve disc during closure, and it became a requirement that testing should attempt to operate the valve at the worst stage of this density front transient. In addition to the qualification of the valve for this duty, NII was also concerned that leakage through the valve should be acceptable and that the valve should not be damaged. Testing in response to Generic Letter 89-10 has illustrated the problems associated with the effect of high-flow isolation on gate valves. Finally, the valve was required to close in an event designed to mimic the passage of fluid that analysis showed could occur during reverse flow from the intact steam gate valves until isolation of the breach.

INITIAL TESTING AT MARCHWOOD TO IDENTIFY CANDIDATE VALVES FOR ISOLATION DUTY

The Marchwood Valve Test Facility was commissioned in 1987 to support valve testing and qualification for the proposed UK PWR programme.

Facilities were provided for (a) operability testing of candidate isolation valves using high- or low-pressure test loops to test valves where endurance for a simulated lifetime under PWR primary circuit conditions and (b) isolation valve flow interruption qualification tests in accordance with the requirements of ANSI B 16.41 for valves selected for *Sizewell* B procurement.

The facility is considered to have fulfilled its requirements and has now been closed. Some of the results of the test work conducted are described below with detailed qualification test work described in the next section.

Review of Isolation Valve Testing for High Pressure PWR Duty

Reference was made earlier in the paper to the need for a better understanding of the problems of safety valve reliability. The licensee, in addition to accepting the above, also needed to be confident that the final valve designs would provide good availability. Therefore, testing conducted to attempt to assess the adequacy of commercially available isolation valves and, where possible, advise the manufacturer of improvements that could be made to meet the specified performance requirements. Full details of the testing can be found in Wright (1990). Salient points are summarized in the following below.

Overview of Typical Findings of Above Testing

Gate Valves. Both parallel slide and wedge gate valves were shown to suffer from inter-gate overpressure effects; this was especially true of wedge gate valves when the fluid temperature was increased in a cold, water solid, closed valve. Solutions to the problem included drilling the upstream disc or venting. However, such a solution would not be acceptable in situations where containment isolation was involved.

During valve closure, enhanced wear on parallel slide valves was evident at the disc-to-guide contact points. In the case of wedge gate valves, high unseating torque was required, and such effects were consistent with findings in the United States during testing and PWR operation. Further detail on parallel slide isolation is included in the section on main steam isolation valve testing.

Globe Valves. Packed gland valves generally behaved well in the isolation mode with gland leakage appearing to be the most significant problem. For bellows sealed valves, in the size range 25-50 mm, there were some indications of inadequate closing torque and seat leakage on a limited number of valves. Packless metal diaphragm valves showed the worst performance of the valves tested with poor reliability of diaphragms. Therefore, packless metal diaphragm valves were therefore selected for safety duty on Sizewell B.

Solenoid Valves. The valves tested for 1000 cycles of blow-down from 172.6 bar at 343°C indicated poor seat leakage and tendencies to errors in position indication with temperature (notably the reed type). Failure to isolate against reverse pressure and disc movement independent of valve electrical state or sticking a temperature were observed. It is worth noting that reverse flow isolation was not a requirement of these valves for Sizewell B.

Conclusions on High Pressure Isolation Valve Test Results

The study (Wright, 1990), conducted at Marchwood over several years, confirmed NII's concerns that valves could be supplied for nuclear use that would show inadequate operability over station life. The results of the test work enabled better selection of safety-related valves for particular duties and also caused the rejection of certain valve types. The reduction in use of parallel slide valves (because of high leakage under flow ΔP conditions) and wedge gate valves, and the elimination of packless metal diaphragm valves were a direct result of the above test work. It should be noted that a similar study on low-pressure isolation valves was conducted, but the results are not discussed in this paper.

PRIMARY CIRCUIT SAFETY RELIEF VALVES

Pressurizer Relief System Description

The Sizewell B pressurizer relief system has three tandem pairs of pilot-operated safety relief valves (POSRV), developed for use on French PWRs, and two conventional spring-loaded safety relief valves (Figure 1).

Each tandem pair of POSRVs comprises a relief valve and a downstream isolating valve, both of which are pilot operated by hydraulic pressure in the pressurizer; or at pressures below 165 bar, (their nominal hydraulic set pressure), by means of electric solenoid actuators. The POSRV's downstream isolating valve is open under normal operating conditions and is set to close as system pressure decreases below a certain value, thus providing each relief line with enhanced reclosure reliability. The POSRVs constitute the first line of protection in the pressurizer relief system and are designed to cater for steam relief and for all water discharge requirements, including overpressure protection at cold shut-down.

The second line of protection in the system is the two conventional spring-loaded safety valves. These have a higher set pressure than the POSRVs and provide a diverse means of pressure

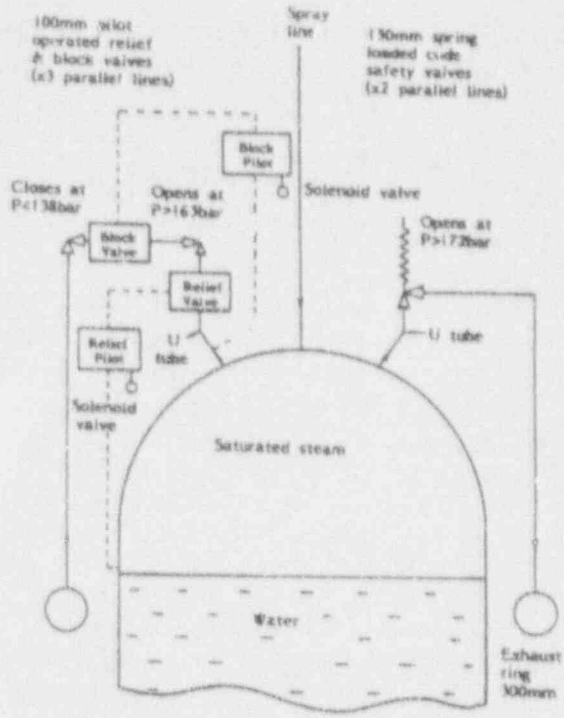

Figure 1. Sizewell B arrangement of pressurizer safety, relief, and block valves.

relief and additional relief capacity for infrequent fault sequences.

Pressurizer POSRV

Figure 2 shows the layout for the BRAVO test facility. The inlet in Figure 3 shows the typical layout of three discharge lines connected to the pressurizer. The main diagram shows the tandem of safety valves and their pilot cabinets for one of the three discharge lines.

Figure 4 shows the various components that make up the Sebim pilot-operated relief valve. The Sebim valves are actuated via sensing lines from the pressurizer, and for cold over-pressure protection, by electric actuation, of which there are diverse means. The construction of the valve is such that two valves are connected together. The first, the relief valve, is normally closed, and the downstream valve acts as an isolation valve and is normally open. The two valves differ in the balance between the sealing bellows force and a spring above the piston that acts against the main

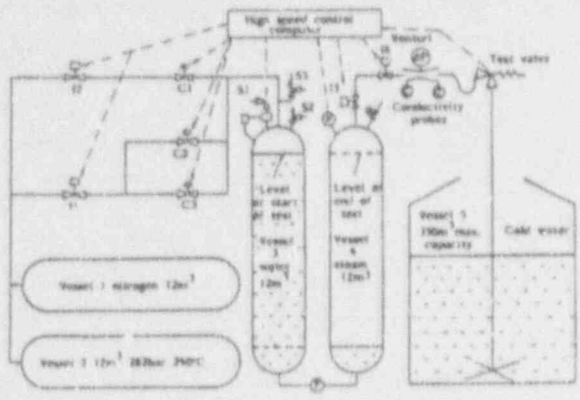

Figure 2. Bravo test facility-simplified flow sheet.

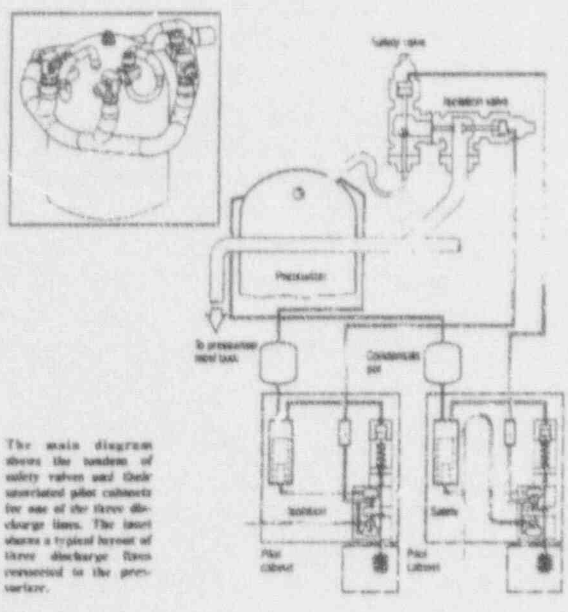

Figure 3. Pressurizer Sebim POSRV arrangement.

valve. In the case of the isolation valve, the net force tends to hold the valve open, while in the relief valve, the spring holds the disc on its seat.

The pilot assembly is connected by small bore pipework to the volume above the piston where it can apply either circuit pressure or atmospheric. A differential area effect ensures that the main valve seals effectively. The pilot valve is operated by water condensed from the steam space that

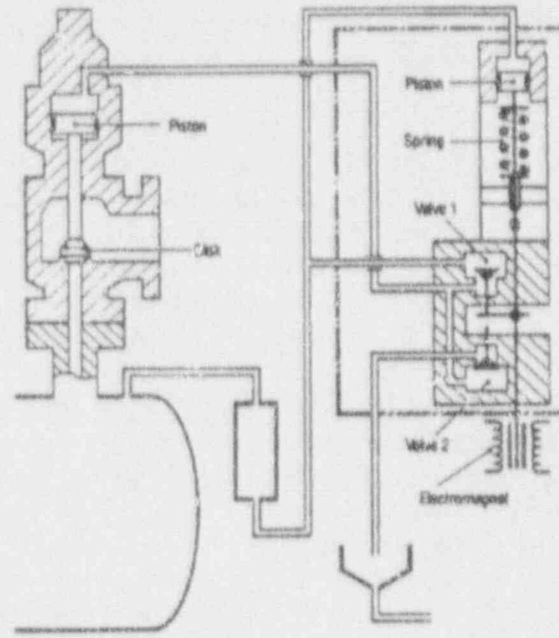

Figure 4. Components of Sebim POSRV.

passes via a filter to a pilot piston. As the circuit pressure rises to the set-point, the piston overcomes the pilot valve spring force and deflects the actuator arm, which closes valve 1 and then opens valve 2. This connects the relief valve piston to atmosphere. With the relief valve open, valve 1 is closed, thus limiting the quantity of condensate required for valve operation.

The electromagnet at the base of the pilot valve actuator arm permits an alternate means of protection should the hydraulic system be unavailable or to offer cold overpressure protection. In addition another solenoid valve is provided that again is capable of venting the main valve piston to atmosphere.

General Testing Requirements

The POSRV was selected for Sizewell B after an extensive programme of development and testing carried out by EdF. This programme addressed the duty the valve is required to perform on the pressurizer relief system of EF PWRs on which it is now installed. However, as a result of differing licensing requirements, the claims made for the valve in the Sizewell safety case differ from those made for EdF plants in certain respects. As a result, some aspects of the Sizewell B duty of the valve have not been investigated by EdF.

A programme has therefore been established in which full scale testing of a tandem pair of POSRVs has been carried out on the MEL BRAVO facility under a range of conditions representing several aspects of the anticipated Sizewell B duty of the valve not covered by the EdF test work.

Low Pressure Performance—Cold Over-Pressure Protection. In order to ensure that the reactor pressure vessel is operating with full ductility at all times, pressure in the primary circuit must be regulated to within an acceptable envelope commensurate with the temperature. A detailed survey of valve opening time with pressure has therefore been made. In addition, valve closure information is required for the safety case in order that assumptions on mitigation of a loss of coolant accident can be supported.

The tests completed to date have yielded data on the opening and closing times, lift characteristics, and discharge mass flow rates of the tandem POSRV at system pressures between atmospheric pressure and 55 bar.

The tests have also given confidence in the cyclic operability of the valve: more than 100 operating cycles have been carried out with subcooled water or steam at pressures up to 55 bar. Subsequent leakage tests and visual inspection have shown that the valve has suffered negligible wear as a consequence of this duty.

Effect of Loop Seal. Early studies at Marchwood attempted to measure the effect of loop seal water temperature on valve performance and load imposed by the resulting water hammer on the valve assembly and pipework. After examination of the loadings, it was demonstrated that the design was adequate to sustain them, and modifications were not considered necessary. This work confirmed the original EdF studies on Sebim operability and pipe loading during loop seal discharge.

Full Flow Testing. Testing was conducted at EdF on the Cumulus facility and at Marchwood. Full flow tests have been conducted over a range of water conditions from saturated steam to cold water. Having conducted over extensive testing at MEL the valve still exhibited negligible seat leakage.

Borated Water. The POSRV hydraulic pilot system is designed to operate on demineralized water to minimize the risk of the formation of boron crystals resulting from leakage through pilot valves and seals. However, to ensure that the system will remain operable in the unlikely event of the inadvertent introduction of borated water, the tests in the BRAVO facility were conducted with borated water in the pilot circuit.

Equipment Qualification. EdF had conducted their own valve performance testing prior to a joint qualification programme between themselves and NE (Fox and gerund, 1989) The mechanical aging and irradiation testing was conducted at EdF, while NE conducted the vibration aging, seismic, and harsh environment testing. This confirmed that the equipment comprising the POSRV system (i.e the tandem relief/isolation valve and pilot systems, position indicators, and electromagnet and solenoid actuators) were capable of performing under the design basis faults specified.

However, some of the more interesting test results include failure of one of the two proposed position indicators (both were included in all aspects of testing) to function correctly after harsh environment testing and minor leakage at a sensing line to pilot valve connection.

Inservice Testing. Although the submission on inservice testing has not been received from the licensee on this issue at the time of writing, it is understood that NE will follow the EdF method of Sebim valve inservice testing. The French practice is to use a mobile test bench, used at cold shut-down (RHR operation) and low pressure, to hydraulically pressurize the pilots to determine the pressure set points.

Unlike true safety relief valves where temperature effects on the valve internals are important to address during test, the same is probably not necessary on pilot operated valves where relatively cooler condensed water is used to operate the pilot valves.

However, pilot operated valves usually have high finish, small seats and in the case of the Sebim valves, NII will be interested in the means to prevent ingress of foreign particles during shutdown which might subsequently affect valve operation. In addition, maintenance practice will be examined to evaluate any other areas of concern.

Testing of Pressurizer Spring Loaded Safety Relief Valves

The Sizewell pressurizer safety valve is a conventional spring-loaded safety valve, which has accumulated a large body of relevant operational experience, through use for this duty on the majority of PWRs in the United States

Furthermore, the valve was included in a major programme of safety relief valve testing carried out under the auspices of EPRI of the United States following the Three Mile Island incident.

The general conclusion of this programme was that, providing certain rules governing the design of inlet pipework geometry were followed, the valve exhibited stable behavior during the discharge of steam. Some uncertainties remain however, and tests were carried out on the BRAVO facility on a full size pressurizer safety valve identical to the Sizewell valve. The areas addressed are discussed in the following paragraphs.

Effect of Loop Seal on Operation of Pressurizer Relief Valves. The dimensions of the loop seal on the BRAVO rig were chosen so that they represented the Westinghouses configuration on Sizewell B as accurately as possible. In particular, it was important that the effect of the water slug as it passed from the loop seal onto the valve internals was representative of conditions likely in the plant.

Figures 5, 6, and 7 indicate oscillations of the valve disc as the water slug from the loop seal interacts with the valve internals followed by hotter steam and also the performance of the valve without a loop seal. It is evident that when the valve operates by relieving a single-phase fluid, much improved performance occurs. Finally, performance of the valve with a steam valve disc shows similar performance, but with enhanced sealing capability. There is an obvious improvement in performance, which supports experience in the United States on this issue. Another aspect which NII, as the regulatory authority, would be interested in is the availability of the changed valve disc. Because there appears to be considerable experience from the chemical industry and steam side plant the change appears to provide an increase in safety.

Valve Adjustment to Meet Blowdown and Accumulation Requirements. To meet American Society of Mechanical Engineers (ASME) requirements, initial flow studies on the development valve began several years ago to show that the rig could be adequately controlled to provide correct flow conditions and that the inlet geometry did not introduce flow perturbations in the test valve. These tests were successfully concluded.

Next, experiments were conducted to determine nozzle and guide ring settings in order to

Figure 5. Valve stem lift (mm) at popping point—loop seal included.

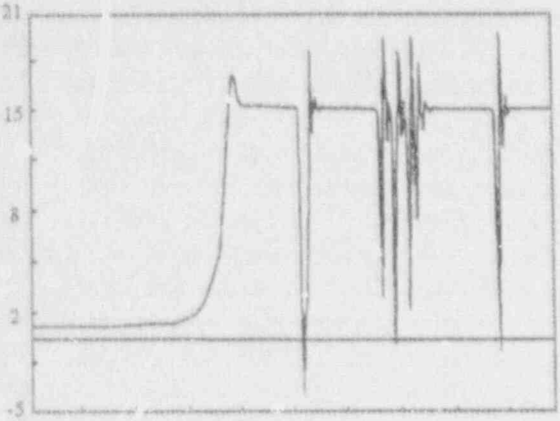

Figure 6. Valve stem lift (mm) at popping point—loop seal included.

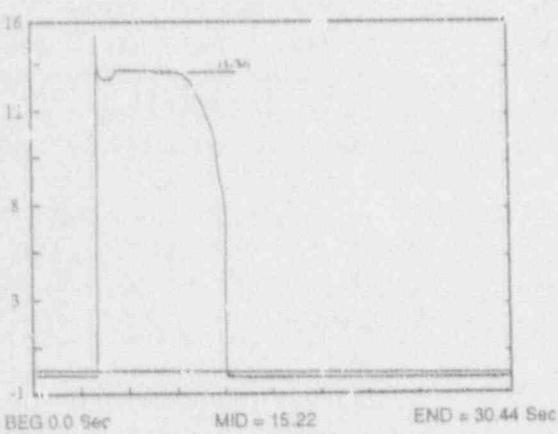

Figure 7. Valve stem iift (mm) with loop seal removed and steam internals.

produce the accumulation and blowdown targets of 3% and 5%. At this stage, assisted lift devices were used to investigate accuracy to be expected when determining setpoint.

Inservice Testing. The use of assisted lift devices for setting reactor gas safety relief valves has increased over the years on UK nuclear power plants. Hence, the results of this test work were of special interest, because accurate test data were available on a new valve, on a modern test facility, with accurate instrumentation. Typically, an accuracy of ±1% could be maintained.

The results of establishing setpoint using an assisted lift device indicate that the pressurizer SRVs for Sizewell B will be in-situ tested by such a method. Apart from reasonable accuracy and repeatability, the provision of a means of reclosing the valve by hydraulic means, should the valve stick open during test, is an important feature in assessing any case that the licensee might make on such testing.

TESTING OF MAIN STEAM AND FEED SYSTEM SAFETY VALVES

Description of Sizewell B Main Steam Valve Layout

Sizewell B has four main steam lines that converge to a common manifold supplying two turbines. Each main steam line includes five main steam safety valves, one power operated relief valve, and an isolation valve. The qualification and regulatory issue associated with testing of the main steam safety and isolation valves are discussed below.

Testing of Main Steam Safety Valves (MSSV)

Since the safety case assumes only steam relief, concerns discussed earlier on two-phase relief should not be relevant. However, at the time of licensing, NII was concerned that such large relief valves could be difficult to adjust and meet the performance requirements assumed in the safety case.

Accordingly, a number of candidate valves were tested at full system temperature and pressure (for approximately 30 tests) at the KWU facility in Germany. Such testing gave valuable information on the valves on such issues as ability to meet ASME accumulation and blowdown limits and setpoint adjustment.

Implicit in such testing is gaining a knowledge of control ring setting. Ideally, the licensee hoped, via such testing, to produce a model for the valve selected such that, once a full examination of a single MSSV had been made, the remaining 19 could be set in a similar manner but without full flow testing. It was an important finding of this work that such a model was not successful. The result of the test work was that all 20 MSSVs were required to be full-flow tested and set at the manufacturer's test establishment.

Inservice Testing of MSSVs. The UK Civil Nuclear Power programme is based on gas-cooled reactors. Conventional spring-loaded safety valves protect the primary circuit and are positioned at the boilers, such that testing is possible in situ. However, from the early Magnox days, there was a tendency to remove the valves and determine set-pressure by a bench test using a gas bottle or, as for many more modern AGRs, to perform an inter-space test, again with a gas bottle, but in situ.

Recently, the move to the use of assisted lift device has occurred. The use of hydraulic pressure to assist primary side pressure in lifting the valve disc slightly off its seat to provide a graphical indication of stem displacement is considered an improvement in safety because it removes some of the subjectivity found in the previous testing methods.

For Sizewell B PWR, a similar test method will be used. Indeed, the licensee is proposing to perform set-point measurement while at low power using an assisted lift device. Because experience such as that described above has been gained in using the technique, and testing at Marchwood have provided accurate measurements, a number of strong arguments support such a case. In addition, the test method has the ability to hydraulically re-close the valves should it not close by spring action at the end of the test.

Finally, NII is considering other factors that could provide the method with added confidence, for instance, the need for good visual data retrieval and operator procedures that can be auditable and reliable. Such efforts would help reduce the risk of dependent failure of the 20 MSSVs.

Regulatory Issues Associated with the Testing of the MSIV for Sizewell B

The Sizewell B safety case assumes that for the main steam line break event (MSLB), the MSIV must isolate the breach in less than 5 seconds. In addition to the qualification of the valve for this duty, NII was also concerned that leakage through the valve should be acceptable and that the valve should not be damaged. Testing in response to Generic Letter 89-10 has illustrated the problems associated with the effect on high-flow isolation on gate valves. Hence, the recent testing of Sizewell B's MSIVs is of special interest.

Further, at the time of licensing, NII was concerned that analysis by the licensees indicated that during steam generator blow-down, there would be a change in quality from steam to a two-phase mixture that could affect valve closure. The qualification of the valve was, therefore, required to show that the effect of such a density front would not prevent acceptable valve closure.

Finally, since MSLB in containment permits reverse flow from the main steam header back through the valve and possible blowdown of the whole of the main steam system, the licensee was also expected to consider this problem.

The outcome of the above issues was that NE elected to qualify the MSIV at the Siemens-KWU full-flow facility at Karlstein in Germany. The test objectives were to ensure that the worst-case density transient was reproduced during the test and that this occurred during valve closure.

The MSIV is a parallel slide gate valve (Figure 8) based on a design used extensively for UK power plant isolation. The valve is actuated by compressed nitrogen, and an integral pump pressurizes an oil cylinder that then compresses the gas. Solenoid valves of different manufacture provide redundant means of releasing the oil pressure and, hence, causing valve closure.

The licensee's contacts with KWU on valve isolability problems enabled him to recommend

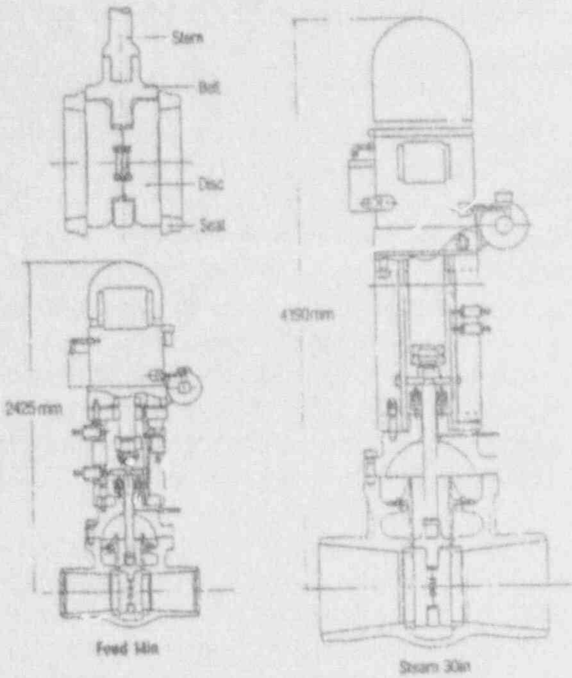

Figure 8. Diagram of MFIV and MSIV.

that the valve supplier purchased the test results from KWU-Siemens so that the findings could be incorporated in the design. In particular, the valve disc was changed to a spade type in order to reduce valve closure loads as the valve disc closed on the seat. Further details of the valve and detailed test results are given in the paper by Clayton and Lapointe (1992).

In order to model the Sizewell B flow conditions, an appropriately sized restrictor was placed at the outlet to the accumulator during the forward flow tests with both steam and two-phase mixing. The restrictor was removed for the reverse flow test.

After adjustments to ensure that the two-phase front arrived at the correct stage of valve closure, the following tests were conducted:

- Five forward flow tests; one initially with saturated steam until at the half closed position a two phase front passed through the closing valve, the remainder with saturated steam
- One reverse flow test, where the restrictor was removed, maximum accumulator pressure was attained and the valve closed against the highest flow possible.

All test results indicated successful closure of the valve in less than 3 seconds, with only a slight increase in leakage and still within specification.

Qualification of MFIV and Associated Water Hammer Analysis

ANSI B 16.41 permits the generic grouping of valves for qualification. Since the design of the MFIV is identical to MSIV except for scale, (the latter is a 30 in. compared to the MFIV at 14 in.), all but the flow interruption testing aspects of ANSI B16.41 were conducted on the MFIV. Therefore, the valve was successfully qualified.

However, the fact that the valve could isolate feed flow in less than 5 seconds for main feed line break fault suggested to NII that analysis of water hammer effects on the system might prove useful, and a study was therefore commissioned by NII. Such analysis would also provide typical flow rates throughout the main feed system, which could then be assessed against the tendency for erosion-corrosion problems.

The details of the full analysis conducted are not presented in this paper. However, a simple calculation illustrates that for isolation of all four main feed lines (which is assumed a frequent event in the safety case), for closure in 1.5 seconds, assuming the speed of sound in water of 1100 m/s, a pipe length from MFIV to pump check valve of 232 m. Reflection time =2*232/1100 or ~0.4s. The theoretical pressure rise is density*speed of sound * fluid velocity or 930*1100*7.14N/m^2 =73 bar. Such a pressure increase is likely despite valve closure in 1.5 seconds because, as Figure 9 indicates, it is only the last ~0.3 seconds during which flow isolated. The full transient analysis is in good agreement with the hand calculation and indicates a peak pressure approaching 174 bar at the MFIV.

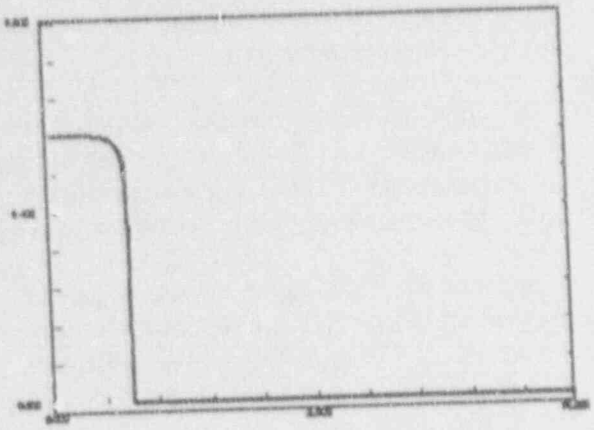

Figure 9. Assumed model of MSIV for analysis.

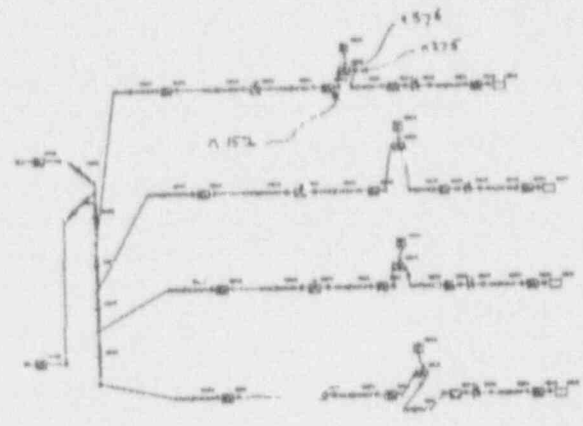

Figure 10. Computer network analysis of main feed system.

An example of the full computer analysis is shown in Figures 10 and 11. NII and the licensee are at present considering the implications of the results. It is worth noting that erosion-corrosion aspects of the system had already been addressed by the licensee via material choice, and this particular aspect is considered satisfactory.

OPERABILITY OF SAFETY RELATED MOTORIZED VALVES—NII POSITION

Contacts between ourselves and NRC have enabled us to keep up to date on this topic, which is addressed in the United States through Generic Letter 89-10. In addition, participation at the Motorized Valve Users Group by the licensee has ensured that the best international effort has been made available to support this work for Sizewell B.

In particular, it was described earlier that qualification of isolation valves to ANSI B 16.41 for each class of isolation valve, involving flow isolation valve type testing, has been conducted. During this work, a number of valves have been instrumented to give stem load and displacement, and these will provide some baseline data prior to commissioning. For the remainder, NII expects that they will be instrumented prior to flow testing to determine packing load effects, and then tested under normal and as near to design

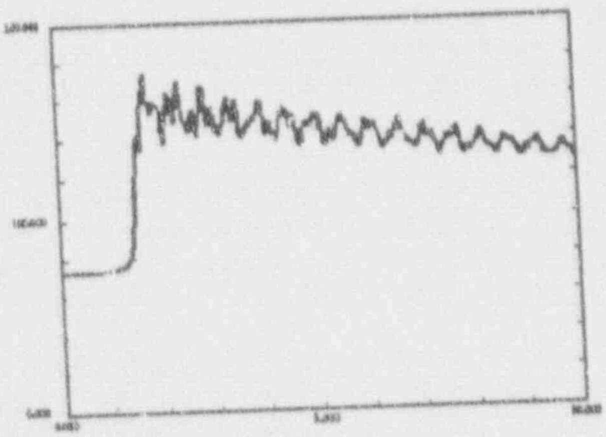

Figure 11. Pressure—time results of analysis of water hammer.

base as possible during commissioning. In the latter conditions, data will be collected that will provide the initial data base for trend analysis over the future operating years of the station. In terms of the parameters required to provide assurance of operability through life, discussions have only recently commenced between ourselves and the licensee. However, the initial view is that, as a minimum, valve stem thrust and an indicator of motor actuator output torque would seem to be required. Information relating to individual valve measurement, other than the number (160 safety related MOVs) and their position and type, is not available so far. However, problems such as inadequate available stem for transducer positioning, possible interference with what are fully qualified components, are current issues under discussion.

A more difficult question at this early stage is how to quantify the extent of the margin acceptable for each test parameter, that is, in order to inhibit premature strip-down or associated maintenance with penalties such as maintenance induced faults or operator dose, while at the same time obtaining sufficient warning of a 'cliff edge' type failure.

CONCLUSION

This paper has attempted to assemble the principal regulatory issues relating to the testing of safety related valves for the UK's first PWR—Sizewell B. From those issues, it has been shown how the licensee, Nuclear Electric, has produced a programme of qualification and testing, in association with the manufacturers, to resolve the issues described.

Probably the greatest single advantage over previous programmes has been the extent of flow testing as opposed to analysis and the international experience to draw on.

The results discussed in this paper indicate that, such are the complexities involved in the adjustment of some of the more important safety valves on a PWR, any future stations would been expected to provide similar assurance of operability by full flow testing.

ACKNOWLEDGMENT

Many of the views expressed in this paper are personal and therefore do not necessarily reflect those of HMNII. Nevertheless, I wish to acknowledge the helpful discussions with NII staff during the formulation of those views and the time made available to produce the paper.

Much of the test information has been made available by Nuclear Electric, largely as a result of our discussions relating to the qualification of safety related valves in the Sizewell B safety case.

In particular, I should like to acknowledge the help of staff from NE's PWR Project Group, Knutsford, and those at Marchwood Engineering Laboratories who performed much of the testing referred to in this paper.

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Similarity Analysis—A Basis for MOV Verification

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Because of the large number of valves in nuclear power plant systems that have to be qualified for normal operating and design base conditions, standardization is inevitable. A functionability analysis is an efficient and appropriate tool because experience from handling different types of valves from various vendors shows a similarity, especially for functional parts.

Siemens KWU developed the basis for similarity analysis according to valve design, capability, and experimental verification and successfully applied the method for valve standardization. Full flow, seismic, and endurance tests on prototypes at Siemens-KWU's test facilities demonstrated the necessity for a detailed engineering approach. This approach includes a pre-evaluation of the design and design features, considering loads and stresses as well as deflection and its influence on performance. Another important item is the material and material combination choice. The material question has to be considered extensively and the long-term mechanical and chemical behavior and threshold loads have to be verified by laboratory tests as a complement to the dynamic testing of the complete valves.

The results showed, for example, that it is not only important and necessary to specify a type of hardfacing but also to specify the procedure of welding in order to achieve the required performance and reliability. In addition, the exact environment that the material and material combination is exposed to is important. Under certain water chemistry conditions the application of a well-known and normally well-performing hardfacing, like hard chromium, can lead to component failure from undesirable corrosion.

The diversity of the design of valve internals for the same valve type makes a detailed analysis and evaluation of design features indispensable. For example, wedge gate valves from different vendors can have different guide geometries, disc shapes, and stem-to-disc connections. Clearances and tolerances in combination with specific design features also have a major impact on the valve performance.

Both internal system loads and external loads must be covered to ensure functionability and leak tightness of the seat. These loads include pressure, ΔP buildup, temperature, pipe bending, torsion, and axial forces induced by system operation and seismic and environmental conditions.

The analysis of the power supply and its possible degradation is another important parameter for verification of valve functionability. A less commonly known item is the influence of plant specific electric control systems that have to be evaluated to bound loads on components in the load path as well as to allow comparison between similar electric power operated valves. In certain cases differences in performance of identical valves with identical operators can be attributed to differences in the electric control system performance.

To fulfill the requirements of The United States Nuclear Regulatory Commission's Generic Letter 89-10 (NRC, 1989a) the above-mentioned similarity analysis was projected to motor-operated valves (MOVs). Using this process, "prototypes" of MOVs are qualified as a basic approach for "None Practically Testable MOVs." This approach is consistent with items listed in Supplement 3 of GL 89-10 (NRC,1989b), which must be addressed for performing safety and capability assessments of MOVs (Figures 1 and 2).

In addition to the above-mentioned parameters, a plant-specific history of MOVs, including long-term effects and time-dependent changes such as

ITEMS WHICH MUST BE CONSIDERED BEFORE ASSESSING THE CAPABILITY OF MOV'S FROM EXISTING TEST DATA

(FROM NRC GENERIC LETTER 89-10, SUPPLEMENT 3)

- VALVE TYPE AND MANUFACTURER
- VALVE SIZE
- DESIGN BASE DIFFERENTIAL PRESSURE
- DESIGN BASE FLOW CONDITIONS
- DISK TYPE
- INTERNAL DIMENSIONS
- INTERNAL CLEARANCES
- SURFACE MATERIALS OF DISK AND SEAT
- SURFACE MATERIALS OF GUIDES

Figure 1. Items which must be considered before assessing the capability of MOVs from existing test data.

FACTORS WHICH MUST BE ADRESSED IN PERFORMING PLANT-SPECIFIC SAFETY ASSESSMENT OF MOV'S

(FROM NRC GENERIC LETTER 89-10, SUPPLEMENT 3)

- FUNCTIONAL VALVE TEST RESULTS
- TORQUE SWITCH BYPASS SETTINGS INCLUDING POTENTIAL FOR MOTOR OVERLOAD ON A FIRST ATTEMPT TO CLOSE VALVE
- INSPECTION PROGRAMS FOR ERROSION, CORROSION AND STRESS CRACKING
- OPERATING PROCEDURES
- LEAK DETECTION CAPABILITIES
- PROBABILISTIC RISK CONSIDERATIONS

Figure 2. Factors which must be addressed in performing plant-specific safety assessment of MOVs.

aging is required. This is to ensure consideration of any material changes or maintenance work on functional parts, such as seat or stuffing box packing configuration and pre-loading, which may affect performance. Lapping or even machining of seat/disk hardfacing of a frequently leaking valve might cause unfavorable changes ranging from a loss of hardness to a position change of the disk at seating. These items are essential for allowing a retrospective identical grouping of the MOVs in operating power plants.

This solid-based engineering methodology paves the way for the "Similarity Analysis" shown in Figure 3. In addition to the above mentioned parameters Figure 3 shows the very important criterion "Disc Stability" that must be considered for performance evaluation. Disc stability is a complex process controlled by the geometries of the valve internals, pressure buildup on the disc, stem force, and friction forces during the stroke. Disc stability, clearances, and internal geometry allow an evaluation

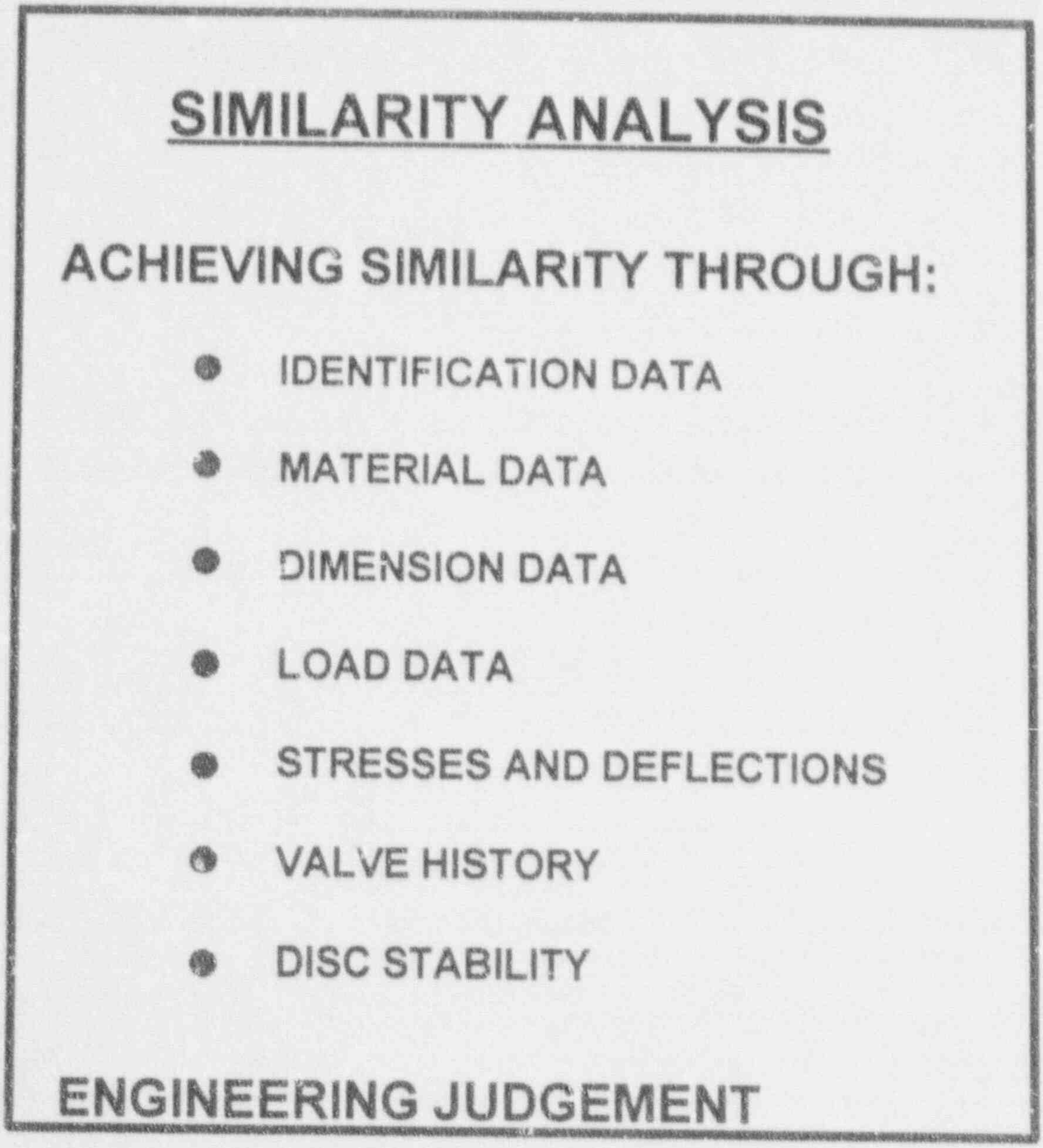

Figure 3. Similarity analysis.

of the stable disc position during stroking with the resulting disc seat interaction.

The evaluation and engineering judgment of valve performance is based on analysis of stresses in functional components, bearing stresses of gliding surfaces and deflections—all of which allow an insight to the loads and material use. The stresses and deflections are calculated with plant-specific system design data, like maximum expected ΔP, flow rate, and fluid temperature. Prototypes can then be defined for full flow ΔP testing. Figure 4 shows some basic stresses and deflections used in the methodology. With this instrument the thrust and function verification can be performed according to the flow chart steps shown in Figure 5.

This methodology will show which valves are similar and can be grouped in a family. The comparison of dimensions shown in Figure 6 allows a

STRESSES AND DEFLECTIONS: GATE VALVES

- BENDING STRESS IN UPSTREAM AND DOWNSTREAM DISKS
- DEFLECTIONS OF UPSTREAM AND DOWNSTREAM DISKS
- BEARING STRESS BETWEEN DISK AND BODY GUIDES
- BEARING STRESS BETWEEN DISK AND SEAT
- STRESS IN DISK GUIDES
- DEFLECTION OF DISK GUIDES
- STRESS IN BODY GUIDES
- DEFLECTION OF BODY GUIDES
- STRESS IN BODY GUIDE WELDS

Figure 4. Stresses and deflections: gate valves.

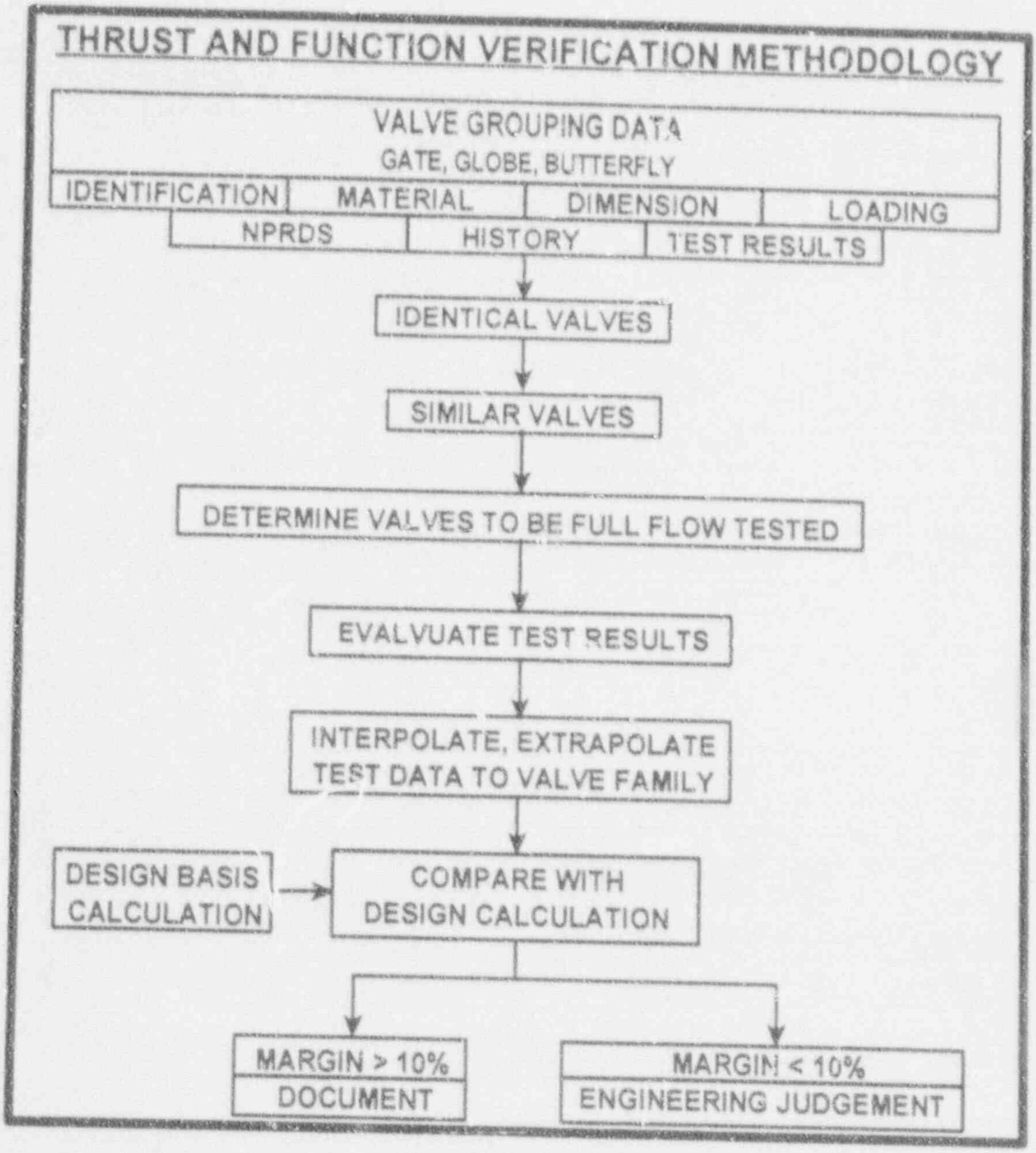

Figure 5. Thrust and function verification methodology.

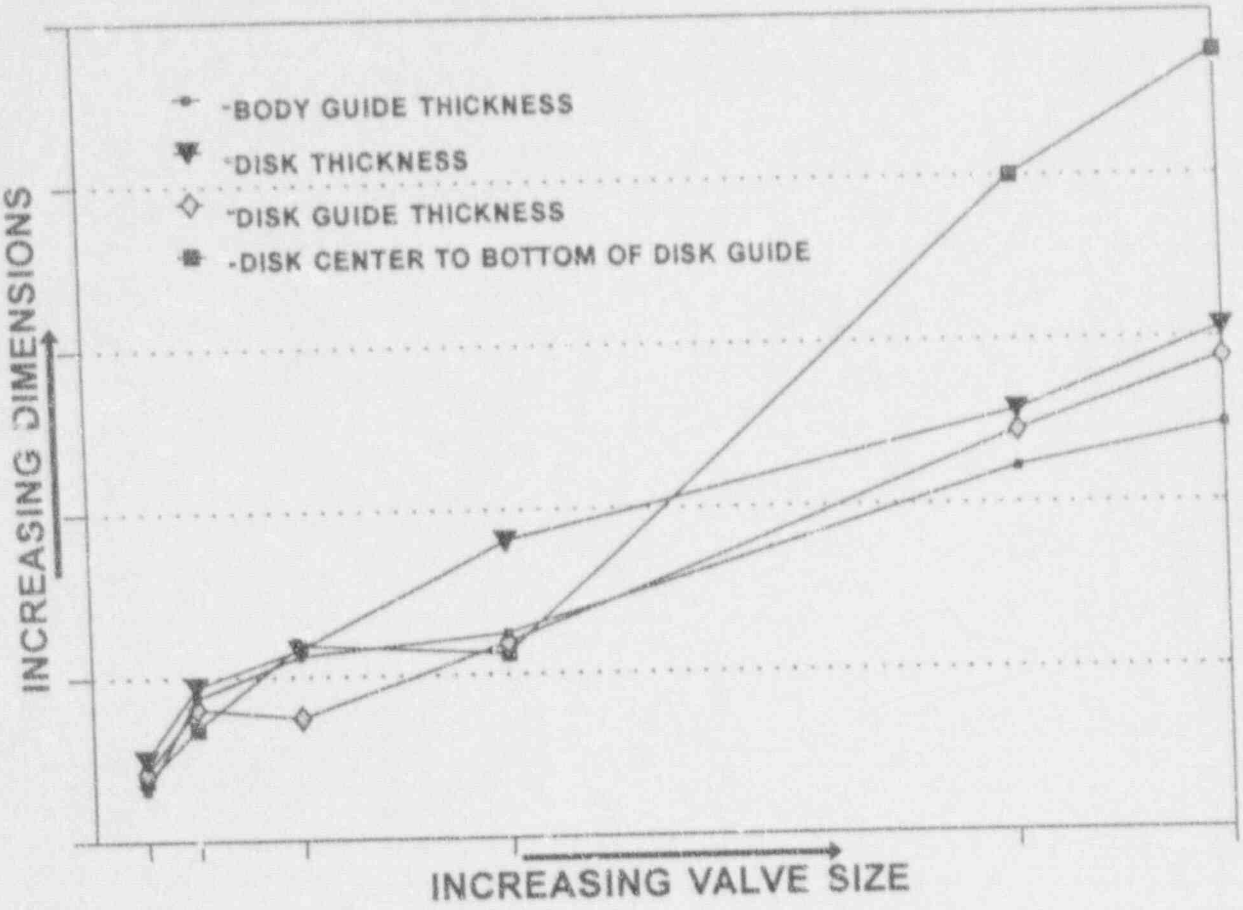

Figure 6. Gate valve dimensions—disk and guides.

preliminary insight into the functional components and provides an indication of stresses and how they change with increasing valve size for a high-pressure flex-wedge gate valve family. The results of the analysis show which family members are subjected to the highest loads and can bound the other similar valves. For example, the results shown in Figures 7 and 8 indicate that although a test in the larger dimensions is normally recommended, the test of valve number 5 will not give results that can be applied to smaller valves. In this case, a test of valve number 4 will give more and better information that will cover the other valves. In addition to the mean and local stresses induced by eccentric loads, performance and relevant deflections of functional parts is required. An evaluation of stress concentration can be achieved by reviewing calculated deflections, especially at critical stroke position.

An additional benefit of this methodology is that it gives the plant and corporate engineers documented and quality assured plant-specific data on the performance and stresses of their valves.

CONCLUSION

In-plant testing for verification of valve functionability is often not possible or practical because of dangers produced in achieving full-flow/ΔP conditions. This creates the need for justifiable, experimental and analytical methods of valve function verification to fulfill the requirements of NRC Generic Letter 89-10. Siemens NPS has developed a method based on similarity and grouping that enables verification of valve function through testing qualified "prototypes" from families of similar valves and applying the

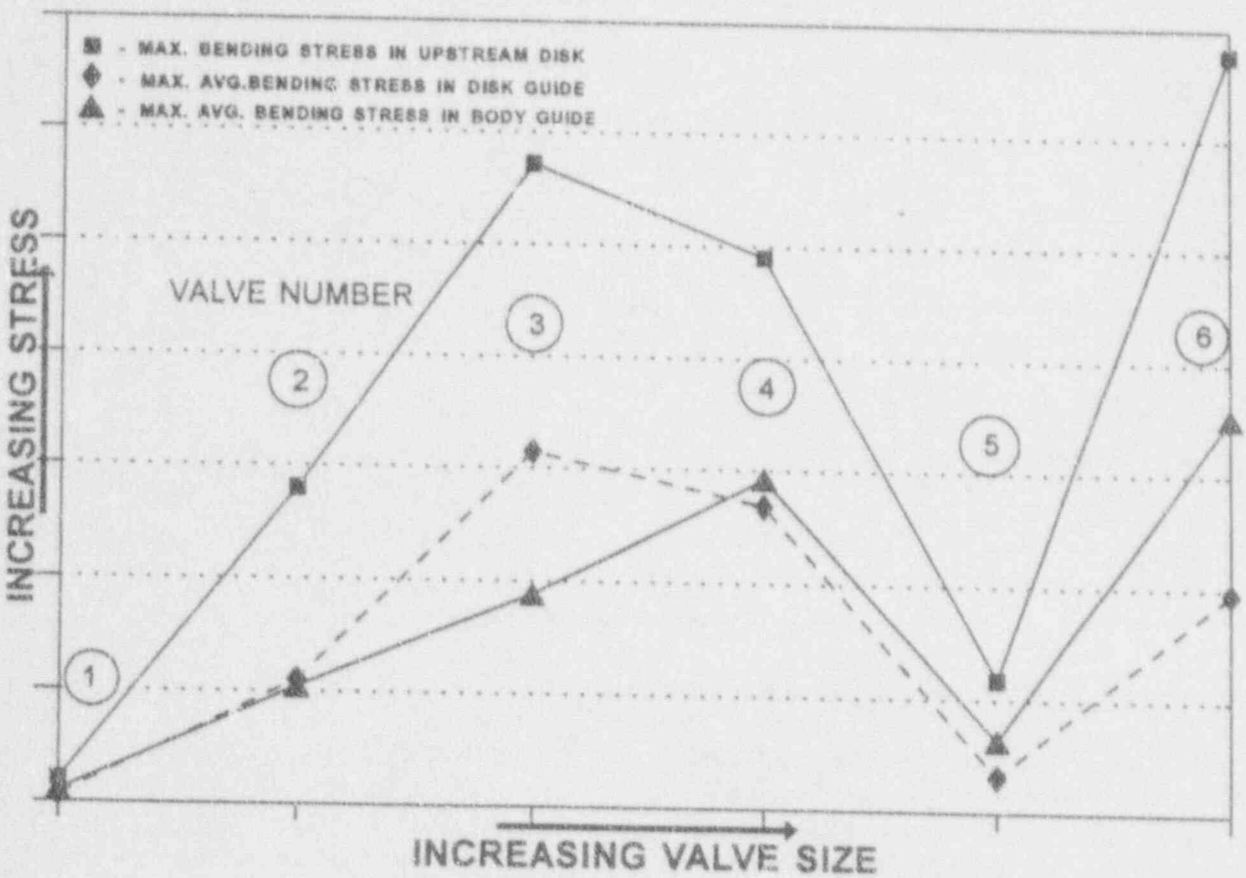

Figure 7. Gate valves: bending stress in disk and guides.

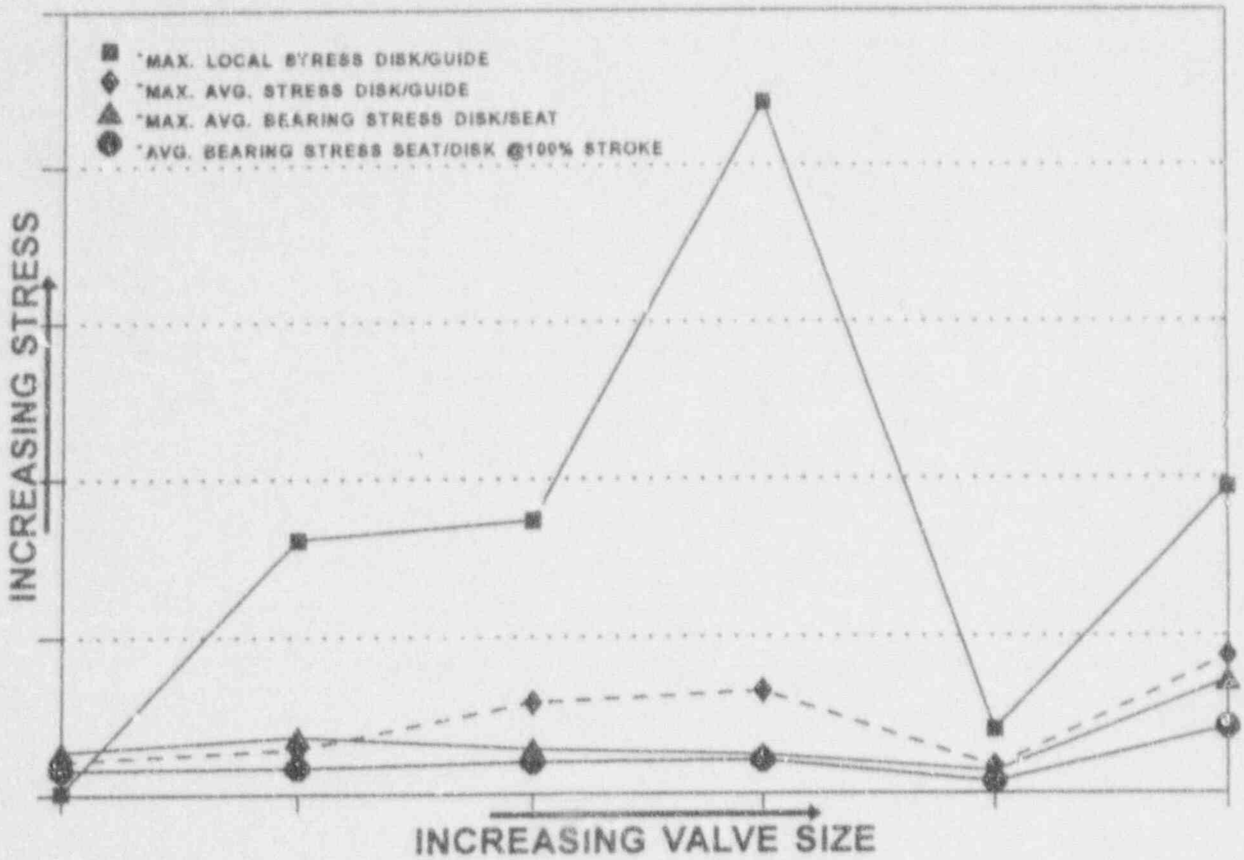

Figure 8. Gate valves: bearing stress on guides and seats.

test results to other valves in the family. The approach individually considers each valve's stresses, deflections, and disc stability using plant-specific loading data. The advantage of this approach over other methodologies is that it relies on results from test candidates from established groups rather than using results from non-similar valves.

The basis for this approach was developed by Siemens KWU, which has extensive experience in valve design and performance evaluation. The Siemens NPS method for valve function verification meets the NRC requirements for choice of prototypes.

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Nuclear Regulatory Commission, 1989b, Generic Letter 89-10, Supplement 3, "Consideration of the Results of NRC Sponsored Tests of Motor Operated Valves."

Motor-Operated Valves—French Experience

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ABSTRACT

During the startup of French 900 and 1300 MW plants, recurrent failures occurred on pressurizer, atmospheric steam dump, and safety injection isolation motor operated gate valves.

The pressurizer original wedge gate valves were not able to isolate completely the blowdown flow. After extensive testing, two improved designs with increased disc guiding length and stellite overlay on friction surfaces were found to be adequate.

On parallel-slide, double-disc, atmospheric steam dump block valves, it was necessary to increase the seating surfaces of the discs to avoid sticking in the closed position as a result of the disc-seat galling.

The failures of several remote control couplings for the safety injection valves were the consequence of over-powered operators and inadequate testing procedures.

Therefore current French research and development programs are aimed to optimize operators thrust relative to valve functions and designs.

INTRODUCTION

This paper presents French experience on three types of motor operated gate valves (MOV). Two of them specified to close under blowdown flow conditions were tested either on Electricité de France (EdF) sites or test loops at the request of French Pressure Vessel Committee (BCCN). These tests led to significant changes to the valves original design and to conclusions similar to those of NRC (EPRI).

The third case deals with a problem of a "self destruction" of a MOV due to unusual operating conditions and a too powerful actuator; not only does it underscore the need for an adequate maintenance program (NRC, GL 89-10) but also the importance of correctly matching a valve with its electric operator.

3-IN. FLEXIBLE WEDGE GATE VALVE IMPROVEMENTS

On French PWR plants, 3-in. flexible wedge motor-operated gate valves are widely used as isolation valves on nuclear auxiliary circuits. Most of these valves, which operate in liquid phase only, are required to open against high differential pressure (corresponding to the primary coolant system pressure, 15.5 MPa) but low flow. Therefore, the pressure differential decreases very rapidly when the valve starts to open, as the flow is limited by orificing or by the pumps' capacity. This is the normal operating case of isolation valves on the chemical and volume control system (CVCS) inlet and outlet lines (Figure 1). But for a few of them, such as the pressurizer relief isolation valves (PRIV) or the CVCS discharge line isolation valves, closure is required also under high differential pressure and flow conditions (blowdown condition) corresponding to a postulated break of the line. Therefore, the operability of these valves had to be proven by testing. The PRIV were the first candidates since the Three Mile Island accident underline their importance.

"Contract Programme" (900 MW) Valve Design and Qualification

Original Design. During the startup of the first 900 MW French PWR, Fessenheim 1 in 1976, random failures of operation were reported for some valves required to open or close in water under high pressure differential.

These valves were standard stainless steel valves with stellite 6 on the sealing surfaces of disc and seats. It was decided to replace them by an improved type with the following (see Figure 2 and Table 1):

- Stellite 6 deposit on the disc guides
- Reduction of manufacturing tolerances (to allow for disc interchangeability and reversibility) and disc-body guide clearances
- Adjunction of an external stem antirotation device and live loading packing
- Increased operator power: 3 kW.

Qualifications. The 900 MW design was qualified by tests in water and in steam. The trials in water were endurance cycling of EdF loop at Les Renardières; the conditions of those tests are listed on Table 2.

The testing in steam corresponded to the operating conditions of the PWRV block valves. This is less severe than a complete break of the piping, as the flow is limited by the PORV to a value of

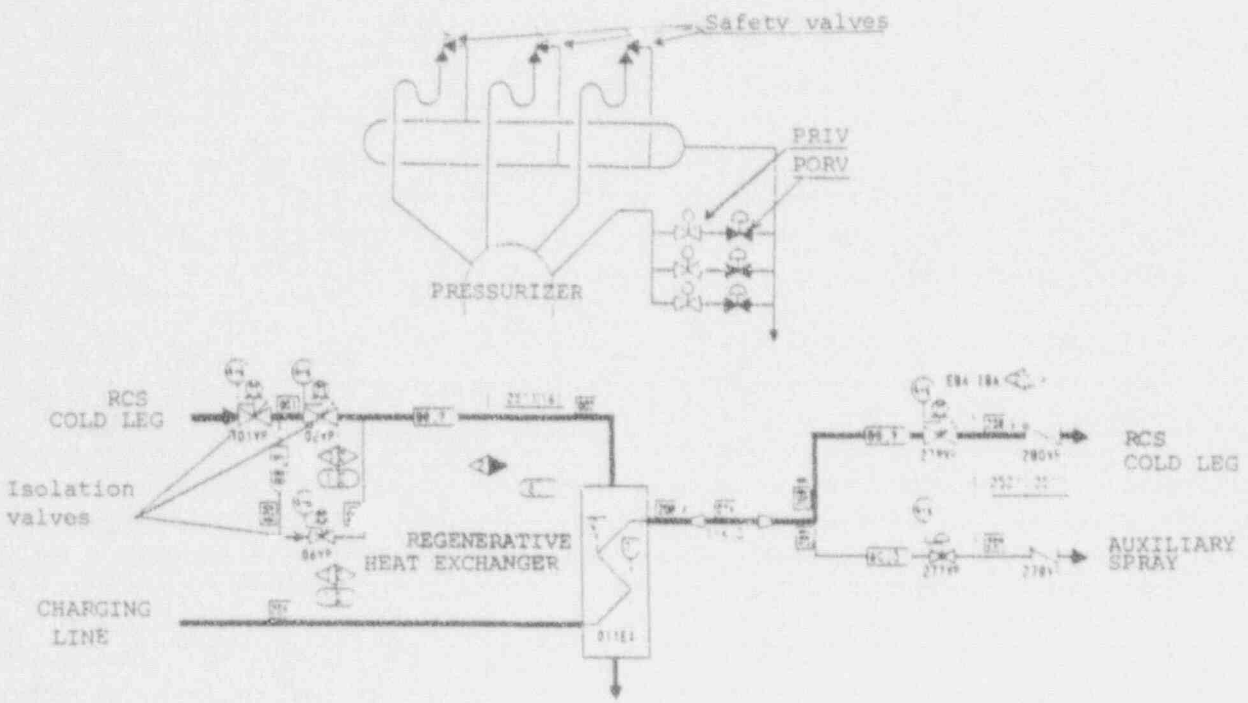

Figure 1. Pressurizer and CVCS relief lines.

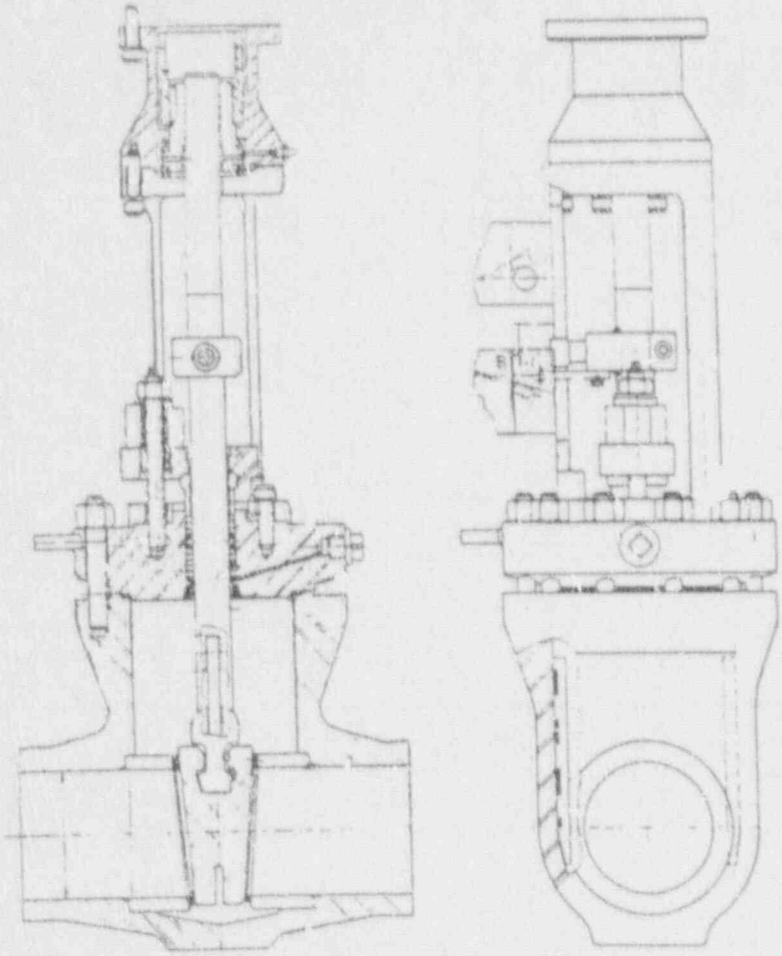

Figure 2. 3-in. flexible wedge gate valve—Alsthom Velan.

about 90,000 kg/hour and as the maximum possible pressure drop occurs only near the complete closure of the valve. This testing was done both on 900 MW sites and then on the test loops. In Dampierre 1, Tricastin 1, and Gravelines 1, one of the three PRIVs was closed five times and the others one time. Inspection of all these valves showed galling marks on the upstream side of the guides near their bottom and on the opposite disc guides. On the EdF Indira Test loop, the valve that was already subjected to endurance tests and a new one were submitted, respectively, to 30 and 20 closures under full flow, but starting from mid-stroke to limit the pressure decrease (see Table 1). Unexpectedly the "used" valve showed no marks at all, whereas the "new" one showed strong marks of galling and metal removal on the guiding bars and near the bottom of the guides.

The conclusion of the BCCN was, that although the valves closed every time, they were not reliable enough. Therefore, the BCCN required the following actions for all the plants:

- One closure, under full flow of each PRIV during hot functional testing of new plants, followed by inspection and repairs if necessary
- A systematic inspection of PRIVs after possible closure under full flow during plant operation. (This event has never occurred.)

Improved Designs

Disc with Increased Guiding Length. This new disc was intended to be retrofitted in the

Table 1. 3-in. gate valve qualification tests.

Identification	Test location	NB of cycles	NB of valves tested	Type	Medium	Temperature (°C)	Average upstream pressure (MPa)	Opening pressure differential (MPa)	Pressure differential at maximum flow (MPa)	Closure pressure differential (MPa)	Maximum flow (kg/h)
900 MW design	BP BG	500 x 3 10	1	A C	Water Water	265 280↔54	17 17	15.3 1.5		1.9 1.5	30,000 160,000
	BI BPP S1 S2	33–20 151 5–1–1 1–1–1	2 1 3 3	B1 B2 B3 B3	Steam Steam " "	355 350 350 "	17.5 16.2 16 16	0 16.2 15.6 15.6	2	15 15	95,000 70,000
900 MW +	BPP	25	1	B2	Steam	350	16.2	16.2	2	14.5	60,900
New disc	BI BP BG	25 3 x 500 10	1 2	B1 A C	" Water "	360 285 285↔60	17.5 17 17.3	0 15 1.5		15 1.9 1.5	95,000 30,000
1450 MW design	BC	7	1	B4	Water	265	15.5	15.5	8	15.5	800,000
1300 MW DESIGN	BP BG BPP BI	* 4 x 500 2 x 10 150 20	1**	A C B2 B1	Water " Steam "	285 285↔60 350 "	15.5 15.5 16 "	15 1.5 16 "	2	1.9 1.5 1.4 15	30,000 160,000 60,000 95,000

BP : Petite boucle
BG : Grande boucle
BPP : Pressuriseur Petite boucle EDF
BI : INDIRA (closed)
BC : CUMULUS
S1 : 3 plants: DA1, TN1, GR1
S2 : 18 plants : TN2 → BL4

A : Endurance
B1 : Blowdown: Downstream PORV opens at isolation valve midstroke
B2 : Blowdown: Opening while PORV closes and closing while PORV reopens
B3 : Blowdown: Start of closure with downstream PORV open
B4 : Blowdown: Pipe break just downstream the valve – 4 m of 3" sch. 160 between valve inlet and pressurized vessel
C : Thermal shocks

* : Refurbishment after 1500 cycles. Stem lubrification and inspection each 500 cycles

** : Change of internals after endurance testing

Table 2. 3-in. Flexible wedge gate valves main characteristics.

	900 MW CPY	1300 MW PQY-DPY	Improved 900 MW	1450 MW N4
Valve				
Size (body) (in.)	3	4	3	3
Pressure rating (lb)	1500	1500	1500	1500
Body material	316 SS	316 SS	316 SS	316 SS
Guide material	"	Stellite 6	"	"
Orifice diameter (mm)	67	67	67	67
Operator				
Operating time (s)	10	16	16	12
Torque switch setting (mdaN)	25	17	14	19
Speed (rpm)	73	48	48	60
Power (Kw)	3	1.5	1.5	1.5

bodies of the existing valves with only minor modification to the valve internals. Its main features included the following (Figure 3):

- Increased guiding length (× 2,4) obtained in raising the height of the disc while reducing the length of the stem
- Stellite overlay not only on the side of the disc guides but in the bottom of the groove
- Machining of chamfers of the lower edges of the disc guides to reduce the contact pressure between disc and body guides in case of disc tilting
- Better accuracy of manufacturing
- Increased stiffness of the disc for better resistance to the operator thrust.

This design was successfully qualified in 1983 after being submitted to 1500 cycles in hot pressurizer water and 20 closures under a pressure differential of 16 MPa and an initial flow at midstroke of 95,000 kg/hour (Table 2).

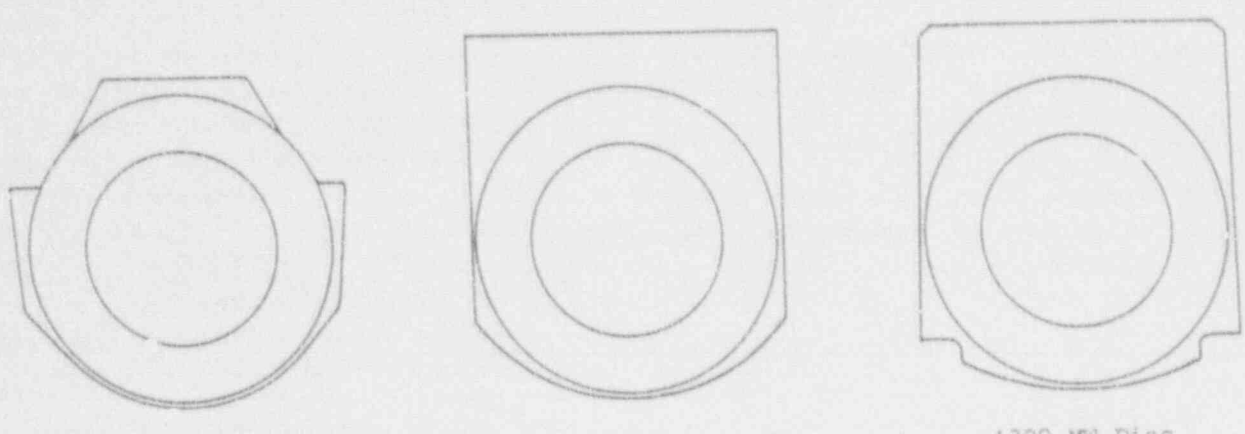

Figure 3. Comparison of 3-in. gate valve discs designed by Alsthom Velan.

Because all PORV and PRIV were eventually replaced by tandem pilot-operated globe valves, these discs were installed only on 1450 MW CVCS letdown isolation valves. At the request of BCCN these valves were also qualified in 1991 on the CUMULUS loop to a subcooled water blowdown flow corresponding to the downstream guillotine break of the letdown line. Because of the upstream piping pressure drop, the pressure differential across the valve at maximum flow was limited to 8 MPa for a tank impulse pressure of 15 MPa.

Valve with Embedded Guiding Bars. These body and disc concepts were originally intended to be used on 1300 MW units as the PRIV. For the same reason as above, the valves that were already manufactured when the decision was made to use globe valves were installed on the CVCS letdown line.

Two valves of this design were successfully submitted to an extensive qualification program including endurance testing in water (2000 cycles), steam (150 cycles) and blowdown conditions (20 closures) as shown in Table 2.

No galling marks, but only the normal wearing pattern on the rubbing parts of disc and guiding bars were observed.

This satisfactory behavior can be attributed to the following features:

- Guiding of the disc on all its length, in particular near its bottom to prevent tilting or cocking (Bake)
- Improved accuracy of guiding by using a U-shaped bar imbedded in a groove machined in a 4-in. body
- Use of stellite 6 overlay on all friction surfaces: seats and discs tracks, internal guide sides, and three apparent faces of the guide bars.

Furthermore, the disc was made rigid to increase its resistance to deformation in case of torque limit switch failure. This strengthening, however, increased the probability of thermal binding, which occurred one time during a test. The closing time was also lengthened (10 to 15 s) so that the operator power could be brought back to 1.5 kW.

Conclusion

To conclude, after numerous and extensive trials, satisfactory design were found for the required function. The main lessons that became apparent are the necessity of

- Avoiding disc tilting or cocking by adequate guiding length and clearances between disc and guides.
- Using materials on guiding surfaces that are able to support the contact stresses without damage and stellite overlay for extreme conditions.
- Keeping the stellite hardness high enough by limiting the iron dilution. Indeed an extensive study of the stellite welding process was carried out at the same time with the valve manufacturers.[a] Its aim was to obtain deposits of good quality (white sipping test) and to keep their characteristics in a tight range so as to control the friction coefficient. The program was completed by the CEA (Commissariat à l'Energie Atomique) measuring the friction coefficients of stellite 6 deposits in cold and hot conditions under representative contact stresses.[b] The values were found to be bounded by 0.4, which is consistent with some American manufacturers' published data (Chappell).

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a. AFNOR, "Rechargements dur par fusion d'alliages à base de cobalt pour matériels mécaniques de réacteurs nucléaires," NFM 64 100.

b. AFCEN, "Rechargement durs par fusion d'alliages à base de cobalt," Ptan II.

PALUEL PLANT ATMOSPHERIC STEAM DUMP ISOLATION VALVES

Background

The startup program of the first plant, of the new 1300 MW series, Paluel 1, comprised the verification of the flow capacity of the atmospheric steam dump valve. The following procedure was used:

- Close the relief valve and then close the isolation valve (without flow)
- Open the relief valve and then open the isolation valve under full ΔP
- Discharge for 5 minutes and close the isolation valve.

The first test done on Paluel 1 was not satisfactory, as the measured flow was too low, so the test was done again on Paluel 2. After the testing it was not possible to reopen the isolation valve due to limit switch tripping. Further inspection of the Paluel 1 and 2 valves' discs showed heavy marks of galling on both the valves' discs and seats.

Analysis

The isolation valves of the atmospheric steam dump valves are designed to close under full flow in the event that the downstream valve sticks open. Their characteristics are listed in Table 3.

A similar test to verify these design conditions was previously performed in July 1980 on the three 6-in. valves of the Dampierre 1 Unit (900 MW) (see Table 4).

The condition of the valves after testing was satisfactory. Only one valve showed some marks of gouging at the downstream edge of the disc.

On the 10-in. Paluel valves, it was quickly recognized that the marks were due to galling that resulted from excessive pressure between the seats and the disc.

Indeed, it has been known for a long time (for example, Armaturen technick Vog) that for this type of valves, the maximum seating pressure occurs near the closure as the ratio of the differential pressure acting on the disc to the loaded surface is greatest.

Calculation (NRC, IE) showed, however, that the maximum specific pressure from the testing remained below the damaging stress, as measured by CEA during analytical function tests. The location of the damage on the seats (inner edge)

Table 3. Atmospheric steam dump isolation valves characteristics.

Plant serie	900 MW	1300–1450 MW
Valve		
Size (in.)	6	10
Pressure rating (lb)	600	900
Body material	A42 CM	A42 CM
Orifice diameter (mm)	125	200
Operator		
Operating time (s)	30	14
Torque switch setting (mdaN)	46	[illegible]
Speed (rpm)	29	98
Power (kW)	2.2	9

Table 4. Atmospheric steam dump isolation valves qualification tests.

Serie	Test facility	NB of cycles	NB of valves	Pressure MPa	Opening ΔP (MPa)	ΔP under flow (MPa)	Closing ΔP MPa	Maximum flow (kg/h)
900 MW	Dampierre 1	1	3	6.3	6.2	—	6	180,000
1300 MW	Paluel 3	1	1	8	8	3	7	430,000
	CUMULUS	5	1	8.4	8.3	0.17[1] 0.8[2]	8.3	380,000

1. Corresponding to 0.15 relative stroke and max flow.

2. Corresponding to 0.5 relative stroke and 0.8 max flow.

— Not measured.

and on the disc (outer edge) confirmed the diagnosis of excessive contact stresses from downstream tilting of the disc. Indeed as explained in Bake, for example, this tilting results from the torque created by the fluid pressure acting below the reaction of the seats.

Solutions

A new disc design was manufactured and successfully tried, first on Paluel 3 and then on the new, large, EDF blowdown test facility, CUMULUS.

This disc (Figure 4) has a square shape to both improve the guiding and reduce the contact stress by enlarging the area of support between disc and the downstream seat.

Furthermore the edges of the disc were bevelled to prevent the disc from "ploughing" the seats during closure, and strict limits were put on the clearances between the disc and its holding collar to reduce the value of the disc tilting angle. Also, the stellite overlay welding process had to be requalified.

Finally since the valve is limit-switch closed, there was no risk of valve damage by overtorquing. Therefore, the torque limit switch was increased to 1.5 times the initial setting, which was the maximum possible value allowed by the operator.

1300 MW—SAFETY INJECTION ISOLATION VALVES

Background

During the startup functional testing of the low-pressure safety injection system for 1300 MW plants, four successive ruptures of universal joints on couplings of isolation valves were observed on three different sites: Penly 1, Nogent 2, and Saint-Alban 1 (two cases). Because this phenomenon did not occur on previous plants of the 900 MW series, and because of its potential consequences on the plant safety, an extensive study was made to understand and solve this problem.

Characteristics of the Coupling and of the Valves

Figure 5 represents a typical installation of one of the valves with its coupling. Table 5 lists the main characteristics of the quoted valves (all of

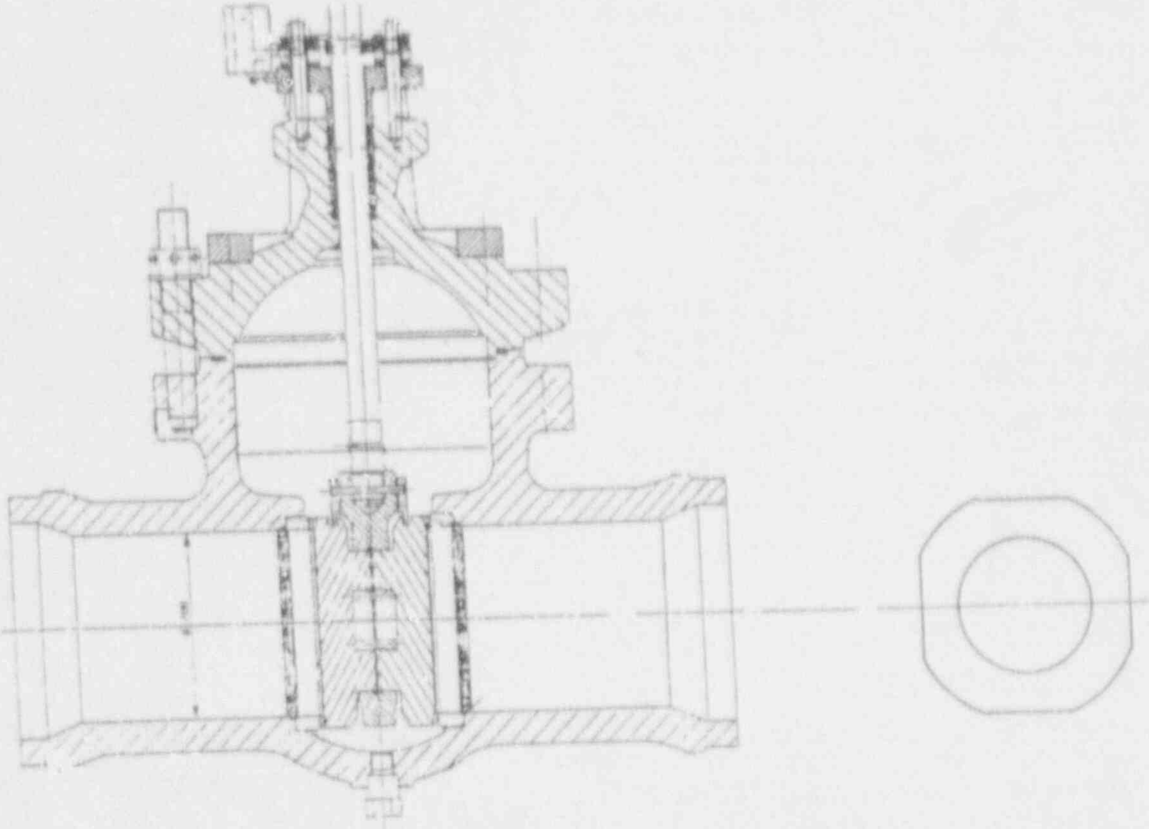

Figure 4. Atmospheric steam dump isolation valve and disc—Alsthom Velan.

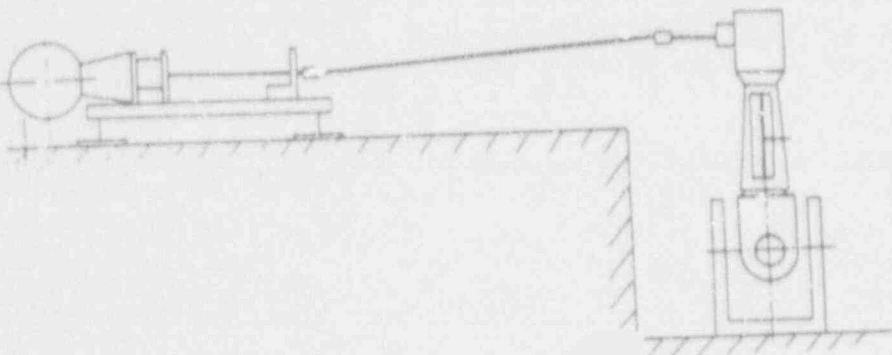

Figure 5. Safety injection valve coupling test bench.

Table 5. Low pressure 1300 MW safety injection valves characteristics.

Valve identification	Cold leg: RIS 33-34 VP Penly 1	Hot leg: RIS 29-30 VP St-Alban 1 Nogent 2
Type	Flexible wedge gate	
Size (in.)	10	8
Orifice (mm)	200	165
Pressure rating (lb)	300	300
Operator		
Operating time (s)	14	10
Speed (rpm)	48	70
Torque switch setting (mdaN)	116	80
Motor power (kW)	15	9
Maximum torque at 1.06 Un (mdaN)	480	390

them were flexible wedge gates). The important points were that they had operators of similar size from the same manufacturer (BERNARD), they were fitted with a permanent motor brake, and the motor was stopped by the torque limit switch.

Findings

Inspections.

- Onsite inspection showed, in all cases, a rupture of a universal joint cross piece arm and, one case, the rupture of one coupling jaw.

- Marks and deformation of the arm splines showed that, in three cases, the break occurred during closure. In the Penly case the break occurred during opening. Finally, inspections of the valves themselves and of their operators showed no damage except in the St-Alban cases where the brakes on the motors had failed from overheating.

Expertise. Two universal joints were surveyed in Creusot-Loire Laboratories. Figure 6 presents a view of the cracked cross bar of the first one (Penly). The metallurgic analyses showed that the rupture had been brutal resulting from excessive loading and a fragility of the hardened material resulting from its fabrication mode. On the second universal joint (Nogent 2), the rupture of the cross bar was also brutal because of excessive loading. On the contrary, the rupture of the jaw of the coupling resulted first from crack propagation corresponding to about 100 fatigue cycles followed by ductile shear. The cracks were initiated in one of the internal corners of the joint, and fatigue cracks were also present in the other corners.

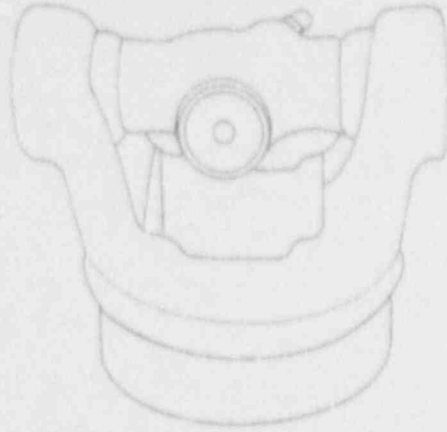

Figure 6. Cracked cross bar.

Testing. Three coupling were [illegible] breaking torque. In all cases the cross bar [illegible] at an average torque value of 530 [illegible]. Dynamic torque measurement were made on a bench (Figure 5) that reproduced the installed configuration. Maximum torque values of 500 and 580 mdaN for Nogent and Penly valves, respectively, were obtained in case of torque switch failure.

Causes of the Damage

Since the entire electrical control system of the valves operated properly and in the normal configuration, abnormal schemes during startups were investigated.

Overtorquing from manual operation was quickly discarded because the required torque was very high and would also have damaged the operator.

In the Penly case, the electrical phases had been inverted when the motor was connected. As the stem moved in the direction opposite to that expected, the open limit switches did not activate and the operator stalled when the stem stopped on backseat. This explained both the overtorquing and why breakage occurred during opening. This explanation was also applicable to one Saint-Alban case. The other case was probably the consequence of nonoperation of the torque switch because of its spurious "shunt."

In the Nogent 2 case, no explanation was found; the only difference in the normal operating condition was that the operators were operated remotely rather than from a local control box.

In a normal scheme, the demand signal is an impulse one. The motor starts to move until it is stopped by the limit or torque switch and then stays at rest. With the local control box, the demand signal is maintained as long as the operator presses the closing button. The motor is stopped, as previously mentioned, by the tripping open of the torque switch. However, if the worm and worm gear reverse, the torque switch closed back and the motor starts again. This phenomenon is called "hammering effect" and was described in detail in Information Notice 8520.

To validate this hypothesis, a test was done on the Golfech 1 plant in 1989 with a torque meter between the motor and the valve. Figure 7 shows the recording that confirms, in spite of the brake, the reversibility of the operator and the hammering effect. The progressive buildup of torque is due to switch time lag, motor inertia, and brake slipping as the brake overheats and loses its efficiency.

GENERAL CONCLUSIONS

From these previous examples, we can draw the first conclusion that correct operation of a MOV, in all circumstances, necessitates a close match of valve and operator. Indeed a very powerful actuator can ensure the operability of a damaged valve, but it can cause catastrophic failure from permanent excessive loadings. Excessive loadings may occur in normal operation from a multitude of causes, especially for valves stopped by torque, such as globe or wedge gate valves. Overloading can be the result of wrong or inaccurate torque limit switch settings, inertia effects at closure, tentative opening of pressure locked, or thermally bound valves (INPO).

At the present time, a joint research and development program between EDF, CEA, and Framatome is quantifying the opening and closing thrusts from these loadings in relationship to the design of the valve and its operator.

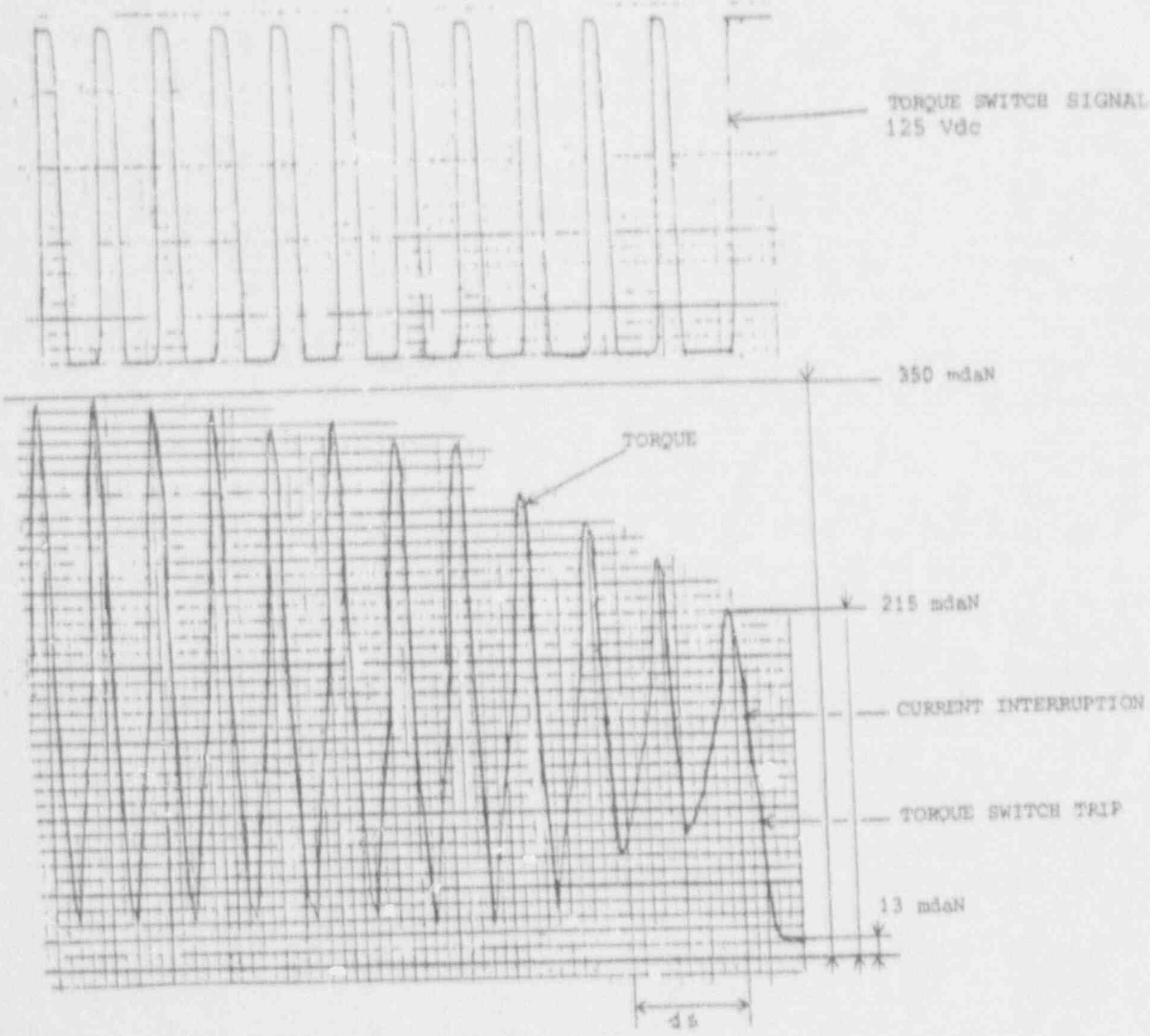

Figure 7. Golfech 1 torque recording—RIS 33 VP.

It also appears that the increase of resistance of valve components needed to support the maximum operator thrust cannot prevent damage resulting from lack of lubrication for example, it or may even induce negative effects such as thermal binding. Therefore the qualification endurance tests of valves that are systematically done by EDF on their test loops will determine limits to the range of evolution of efficiency and friction coefficient and to the required operator thrusts.

The second conclusion is that manufacturers seem now to master the design of gate valves operating under high pressure differential and flows. However, as hydrodynamic loads are not yet fully understood [for example, pressure built up between the top and the bottom of the disc or "Bernouilli" effect (EPRI)], an experimental verification under real operating conditions is needed. The experience of the 3-in. 900 MW gate valves emphasizes the fact that a successful test on one valve cannot bring an absolute guarantee, because for designs without enough operating margins, minor differences in manufacturing can lead to quite different results.

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NPSH-Requirements of Large PWR Boiler Feed Pumps at Partial-Load Conditions

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ABSTRACT

Incipient cavitation and cavitation erosion could not be calculated theoretically up until now, especially if part-load service was required. At part-load pumping conditions the flow patterns and NPSH-incipient values change dramatically. So it is often necessary to deal with the cavitation and service details of pumps with high-energy impellers or with special features related to NPSH. Some main aspects of cavitation and cavitation-caused erosion are pointed out in this paper.

As an example the development of new impeller and inlet geometry designs of feedwater pumps for pressurized water reactor, (PWR), 1300 MW power plants is pointed out. It is shown that at extreme demands on service (full operation range, especially long-time, part-load service without limits on operation time) these high-energy impeller pumps should be designed for bubble free operation.

INTRODUCTION

Cavitation, cavitation erosion, and cavitation damage have been topics of concern since pumps have been built. Cavitation can be expensive because of spare parts needed, disassembly and assembly, and maybe even plant outage.

In 1978 Makay (1978) presented the results of an examination of pump outages, where cavitation is on the fifth place behind seal failures, vibration, and bearing problems (Table 1). Ten years later Gopalakrishnan (1988) presented the results of a similar examination (Table 2). The order of succession had changed a little bit, but cavitation damage still placed fifth.

This paper gives an overview of the experiences related to cavitation and part-load service with the main feed water pumps of seven German PWR power plants. The special hydraulical requirements (i.e., big volume flow and relatively low head) led to the typical main feed water pump unit, as it is shown in Figure 1. The pumps are pumping feed water from the feed water tank via high pressure (HP)-preheaters into the steam generators. Two pump units are in service for 100% rated electrical power output of the plant, one unit is on stand by. A pump unit consists of a one-stage, double-suction booster pump, a constant-speed electrical motor, a gear box, and a one-stage, double-suction, high-speed main pump.

Before having a closer look at the experiences with these pumps, some fundamental things about cavitation shall be considered.

CAVITATION

Cavitation occurs if the value of NPSH-available becomes smaller than the required value NPSH-incipient. At these operating conditions the first bubbles appear at the blade inlet of the impeller. If the amount of bubbles raises more and more with decreasing NPSH-available, the head of the impeller is affected and a head drop can be measured.

MEASURING METHODS

Three methods are commonly used in practice, depending on different physical effects:

1. The very common measuring method is the determination of a certain head drop (mostly

Table 1. Ten major outage producing failure causes (Makay, 1978).

Number	Pump failures component, symptom or technology	Feed pump	Booster pump	Other pump types	Total number of failures
1	Seals	602	178	198	978
2	Vibration	228	85	60	373
3	Axial balancing device	337	—	—	337
4	Journal bearing	209	52	37	298
5	Cavitation	161	64	46	271
6	Impeller breakage	169	8	7	184
7	Wear ring: rapid wear	155	3	—	158
8	Unstable head curve	92	59	10	161
9	Shaft broken/damaged	77	6	51	134
10	Thrust bearing	58	11	9	78

Table 2. Failure modes (Gopalakrishnan, 1988).

	Failure mode weighting factor	Estimated cost in 1987 (million of dollars)
1. Vibration	0.0931	53.8
2. Impeller breakage/cracking	0.085	49.2
3. Shaft seal failure	0.084	48.6
4. Rapid wear of wear rings	0.081	46.8
5. Cavitation damage	0.077	44.5
6. Axial balancing device failure	0.077	44.5
7. Broken or damaged shaft	0.075	43.3
8. Journal bearing failure	0.072	41.6
9. Seizures of wear rings, etc.	0.072	41.6
10. Thrust bearing failure	0.071	41.0
11. Unstable head curve	0.068	39.3
12. Auxiliary system reliability	0.062	35.8
13. Hot misalignment	0.045	26.0
14. Gear type couplings	0.038	22.0
Total		578

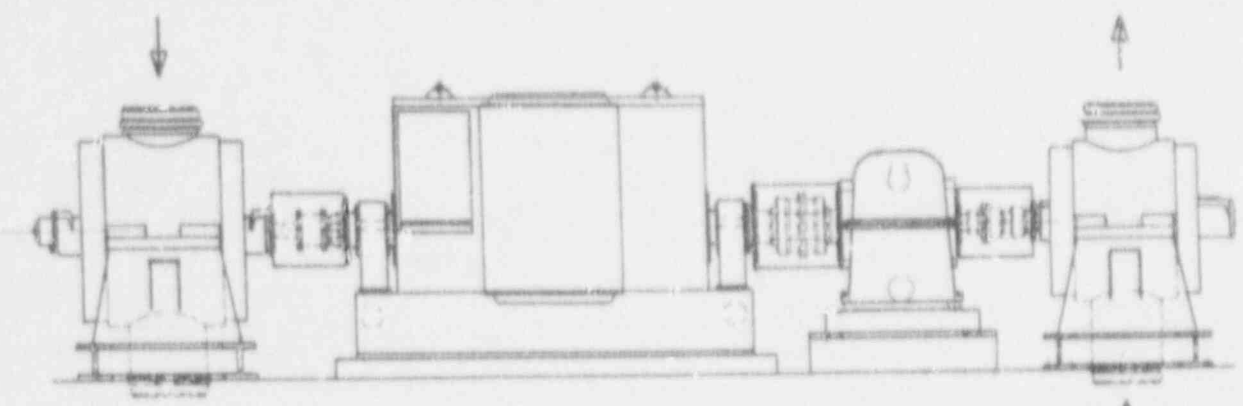


Figure 1. PWR 1300 MW main feedwater pump unit.

3% head drop) compared with the cavitation-free operation.

2. The visualization method (i.e., the direct observation with use of stroboscopic light) is another, but very expensive method. A special test rig with original size is necessary.

3. The acoustic method requires a lot of experience for interpretation of the measuring results. No special test rig is required.

CRITERIA OF ASSESSMENT

When testing one pump with the a.m. measuring methods (i.e., 3% head drop method, visualization and acoustic method), different results are obtained as shown in Figure 2. Decreasing the NPSH value in a test rig produces the first information about incipient cavitation from the acoustical method. After decreasing the NPSH value a little more, the first bubbles can be detected with the optical method. Depending on the impeller design, the NPSH value has to be decreased—possibly at a considerable amount—further until 3% head drop can be measured.

At 3% cavitation caused head drop, a considerable amount of cavitation occurs. Mostly only the NPSH value at 3% head drop is known and not the complete head drop versus NPSH characteristic. In Figure 2 the head drop characteristics of two different impellers are compared as an example. The NPSH value for 3% head drop shall be the same at both characteristics, and both impellers shall work at 5m NPSH-available, respectively. If only the value of 3% head drop is known for both impellers, no deviations in service seem to exist. But a closer look on the head drop characteristics shows that impeller #2 may have no problems because of a very small amount of bubbles, while impeller #1 is working with a considerable amount of bubbles and, thus, at least keeps the latent possibility of damage.

If an impeller is specially designed to deal with a certain amount of bubbles to produce good NPSH-3% values (a simple method is to enlarge the inlet area of impeller), it can have cavitation erosion problems instead. (refer to Figure 2, curve 1). On the other hand, this impeller, with a high suction specific speed, might have problems with recirculation flow or unsteady flow patterns at part-load conditions.

CAVITATION DAMAGE

The existence of mild cavitation seems not to be sufficient for cavitation damage, as can be learned from the practice. Many pumps operate with a certain amount of cavitation without

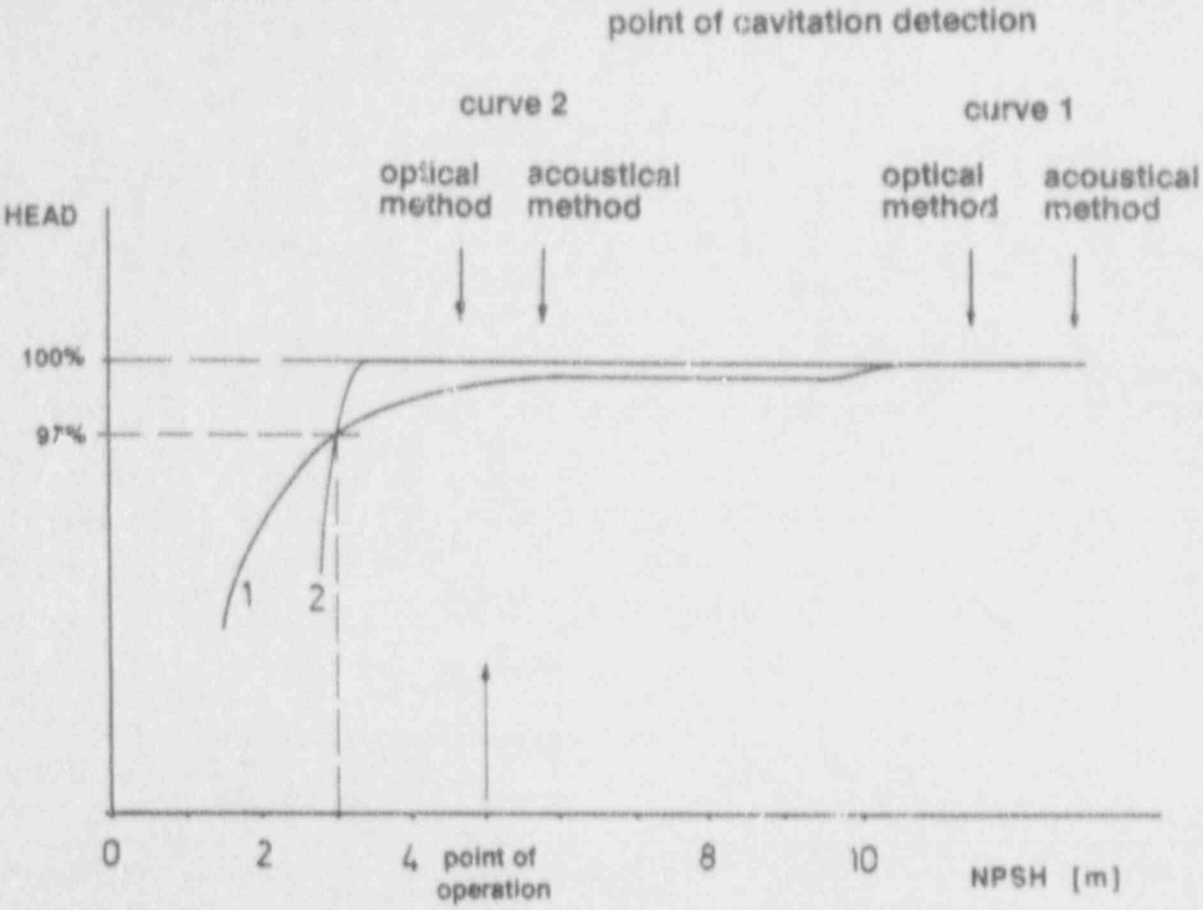


Figure 2. Head drop/NPSH—characteristics (example: 2 different characteristics with identical NPSH 3%-value.

cavitation problems. If the resistance of the material against cavitation erosion is greater than the attack of it, no cavitation erosion occurs. If the bubbles don't collapse near a wall surface, but in the middle of the impeller flow channel, no damage will occur as well.

A strong influence on cavitation erosion comes from the inlet velocities, especially from the relative velocity (blade leading edge). Cavitation erosion increases with about the 6th power of relative velocity. Other important factors are water temperature (decreasing water temperature causes increasing cavitation erosion rates), operation mode, and the material of impeller blades. Erosion factors have been published by different authors, (Doolin, 1986 and Gülich, 1986). Of course, these erosion factors can only be used for estimation of cavitation erosion if cavitation already occurs.

NPSH-REQUIRED VALUE AT PART LOAD

Reducing the flow at constant pump speed increases the incidence on the impeller blade inlets. Simplified, this is the reason for an increasing adverse pressure on the suction sides of the blades, which causes a higher NPSH-required. If the flow is decreased more and more, a reverse flow in the impeller eye commences. Because the water leaves the impeller with high circumferential velocity, a prerotation of the incoming flow starts up because of its mixing with the backflow water. This causes a decreasing incidence, and if the design of impeller and inlet geometry is sufficient, the bubbles that can be seen in the visualization test start to disappear if flow is reduced further.

A typical NPSH-versus-flow characteristic at 3% head drop, as found in many catalogues, ordinarily has a smooth shape with decreasing NPSH-required values at decreasing flow. No problems seem to exist at part load if NPSH-available is large enough at rated discharge. Choosing a stricter criterion may change things. In Figure 3, an impeller with extreme behavior is shown. It is an impeller of a submersible pump. Characteristics are printed out for total head drop, for 3% head drop, and for 1% head drop.

It can be seen that the part-load peak is the higher, the stricter the criterion. If "bubble free" were demanded, the peak would be higher again. In the literature indications can be found that the part-load peak establishes at onset of recirculation or even at somewhat larger flow rate. But here big individual differences occur between different pumps depending on hydraulic design, geometries of impeller inlet and inlet chamber, and type of diffusor (volute or vaned). Details like the inleading of internal leakage flow are important as well.

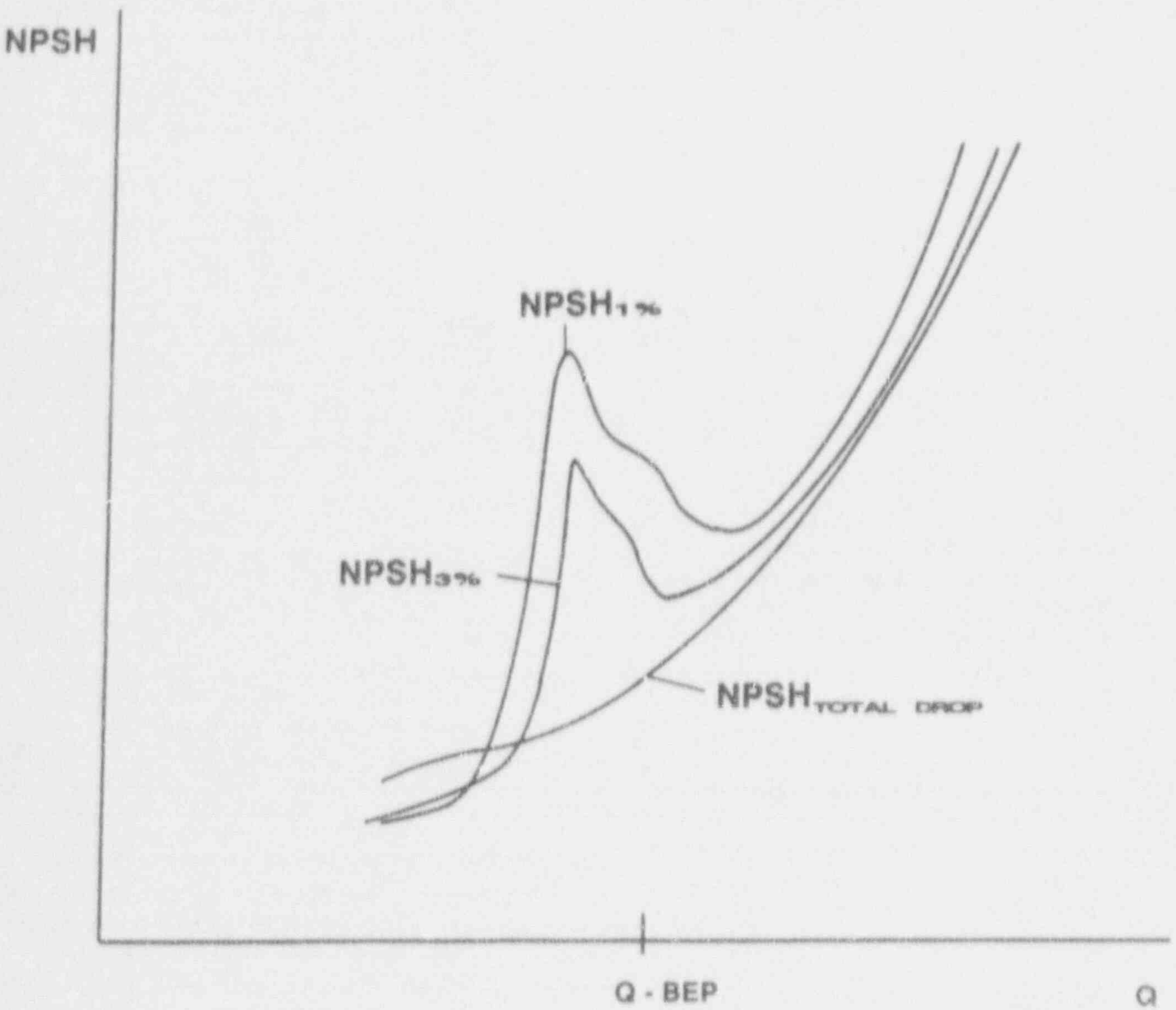


Figure 3. NPSH required—characteristics of submersible pump impeller (parameter: cavitation criterion).

At the present time no theoretical procedure is known to calculate the NPSH peak at part-load in any way. So it seems to be the best (and only) way to test the pump by visualization (or acoustic) method if the precise knowledge of the complete cavitation behavior of that pump is necessary.

OPERATING EXPERIENCES

The following section gives a short overview of the operating experiences with PWR feed pumps in reference to cavitation within the last 15–16 years. Only the main pump is of interest in view of cavitation problems because the booster pump has none. This overview focuses on experiences related to part load service with seven PWR-plants that are equipped with the pumps of one manufacturer, this overview is limited to these pumps.

FORMER MAIN PUMP DESIGN

Figure 4a shows a section drawing of a former main pump type. It has a double-suction, one-stage impeller and a double-volute type diffusor. The casing is cast. The main data are included in Table 3.

When these pumps were started up in November of 1974 for initial operation in the plant, it could be detected after a short time of service that cavitation caused erosion had occurred, although design was correct in the light of the experiences of that time. It was decided to build a bubble test rig (BTR) of original size at the workshop of the pump manufacturer and to test the impellers.

Figure 5a shows the characteristics of NPSH-incipient-versus-flow schematically as well as the head of the booster pump. The reference point is the original design of the booster pump because NPSH-available and the head of the booster pump have nearly the same value.

The curve "a" is a result of the first bubble testing. Having its minimum NPSH-incipient value at 100% load, the impellers were not bubble free even at the design point in spite of having a sufficient safety margin related to the NPSH 3% value. Cavitation occurred at the complete range of operation.

A design review was started and new impeller geometries were developed in several steps to reduce bubble length (or the amount of bubbles at the first steps). An important step was to profile the blade inlet area and to manufacture the profile shape of all blades identically. The value of NPSH at 3% head drop became insignificant.

In March of 1978 the impellers were bubble free at 100% load; at part load conditions a bubble length of 30 mm was measured (curve "b"). The NPSH-incipient curve "b" could be reduced at lower flow by modifications of the inlet casing geometry—the short diffusor type inlet geometry was developed. The head of the booster pump was raised as much as possible. By July of 1978 the result of these modifications was that the bubble length at part-load conditions was only 10–15 mm (curve "c").

This was a good result. With the so-modified pumps ("former main pump design") a long service life could be reached. After more than 65,000 hours of full-load operation, the impellers had only small areas with a little increased surface roughness. No real cavitation damage could be detected.

Sometime later one plant started a longer period of part-load service. The result at the impellers from the cavitation was considerable cavitation erosion. Finally the impellers were replaced.

ADVANCED MAIN PUMP DESIGN

In the meantime the development impellers with a shorter bubble length were designed by the manufacturer, although only the good experiences with the modified "former main pump design" were known. Experiences with part-load service over time of those pumps were not available at that time.

A new system design of power plants with more electrical power output caused development

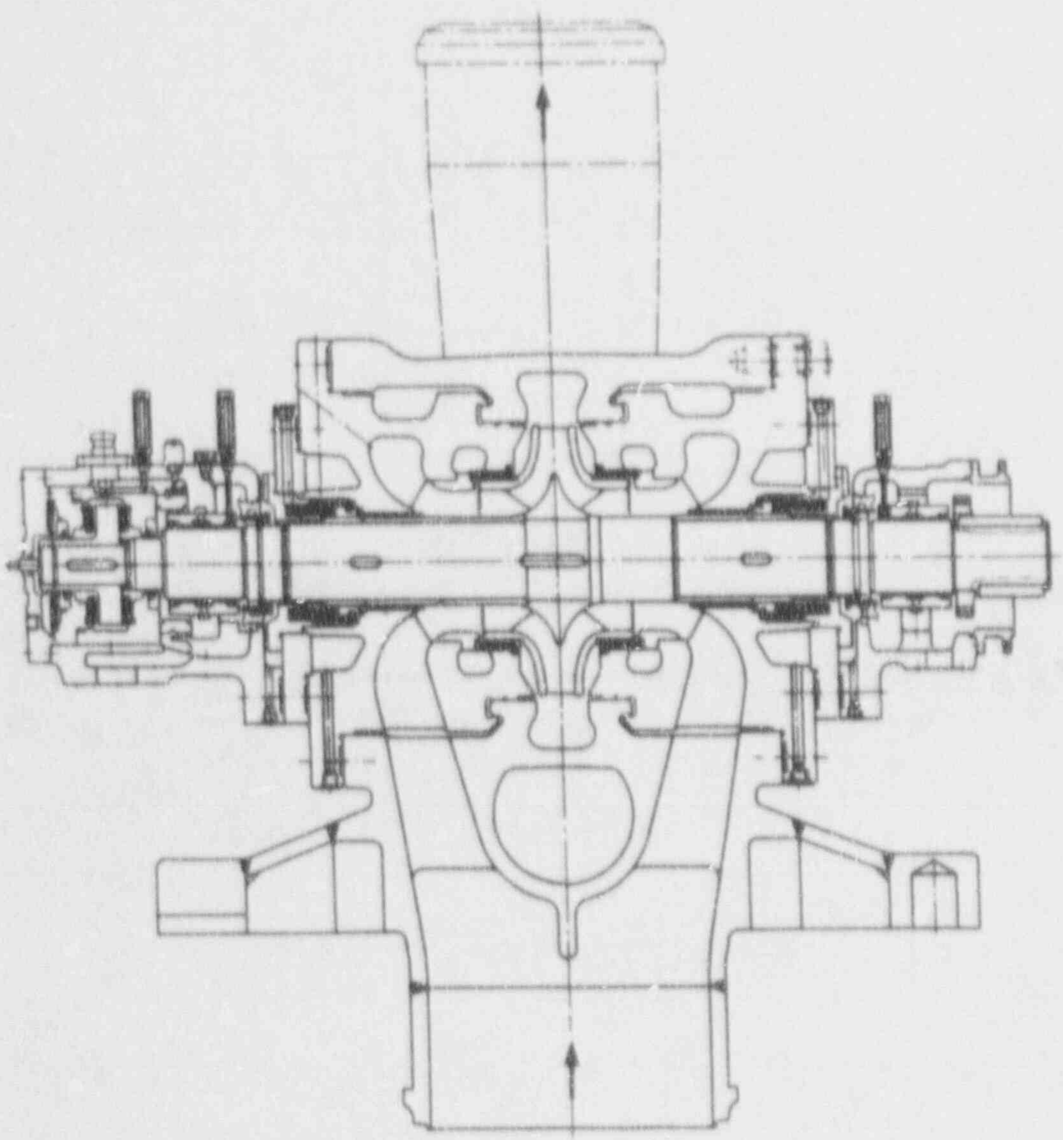


Figure 4a. Boiler feed pump, main pump, casted casing (former main pump design).

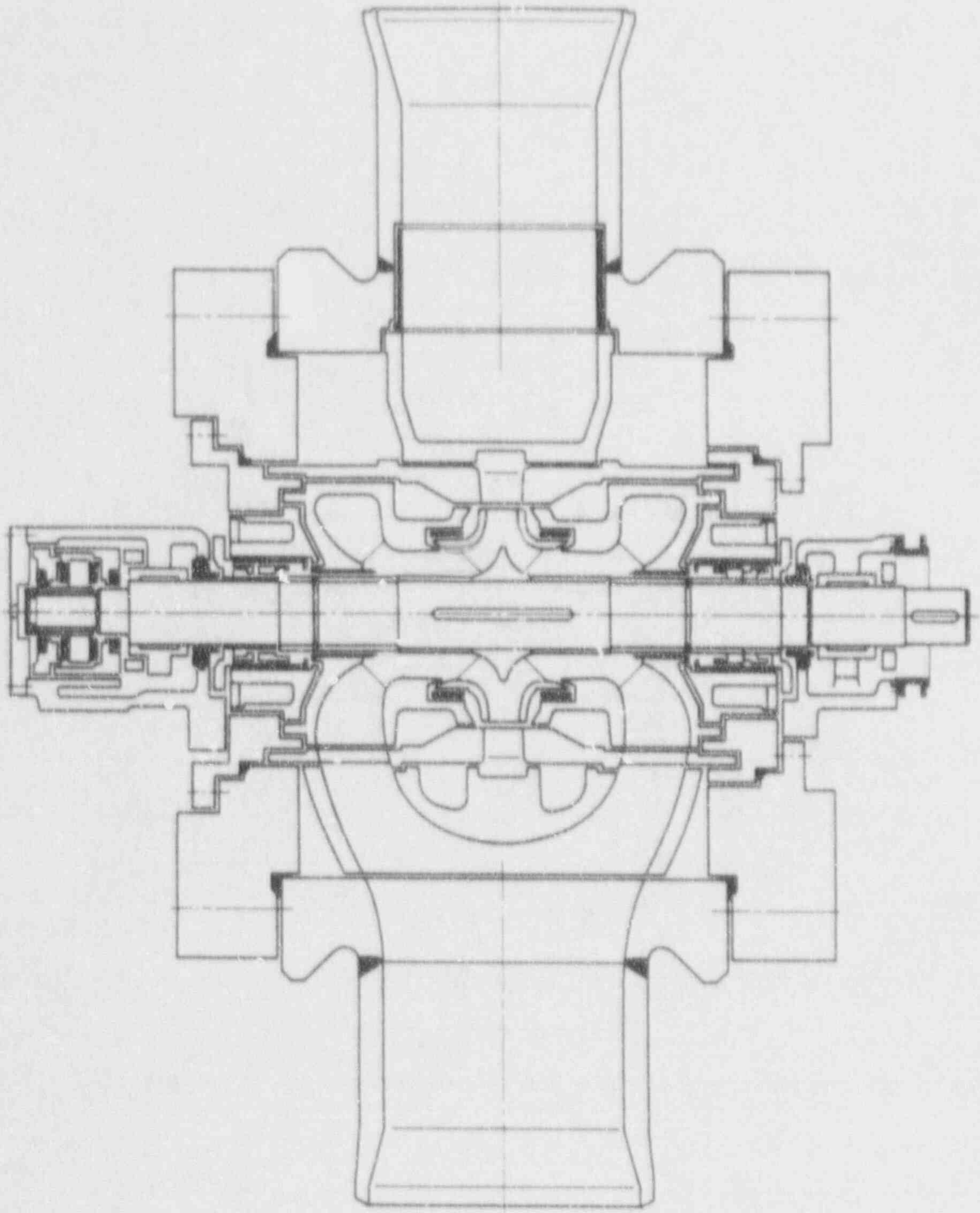


Figure 4b. Boiler feed pump, main pump, forged casing (advanced main pump design).

Table 3. Boiler feed pumps of PWR power plants—data of main pumps.

		"Former main pump design"	"Advanced main pump design"		
Plant-Date of commissioning		1975...	1985	1986	1988...
-No. in figure 6		—	4	3	1,2,5
main pump head	[m]	642	645	660	660
flow rate	[m^3/s]	0.92	1.165	1.165	1.113
speed	[rpm]	5079	5218	5296	5296
specific speed (metric units)		27	31	31	31
water temperature	[°C]	180	180	180	140
water density	[kg/m^3]	884	884	884	924
booster pump head	[m]	141	212	212	212
impeller material		1.4008	1.4313	1.4313	1.4313
hard facing		Stellited	Stellited	Stellited	Stellited
finishing		Eroded	Eroded	Eroded	Eroded
impeller inlet velocities at 100% load:					
- meridional	[m/s]	12.1	14.8	14.8	14.1
- circumferential	[m/s]	70.7	76.5	77.6	77.6
- relative	[m/s]	71.7	77.9	79.0	78.9
erosion factors related to:					
- relative velocity /5/		1	1.6	1.8	1.8
- water temperature /10/		1	1	1	1.6

of the feed water pumps. The main data for the advanced and the former feed pump design are compared in Table 3. The flow was increased by 27%; the speed was increased by 4.3%. Based on the experience with part-load service over time at the plants with "former main pump design" (feed pumps were not known at that time), the head of the booster pump was raised by 50% with a new impeller design.

As mentioned before, a higher relative velocity at impeller entrance and a lower water temperature cause an increase in cavitation erosion. At the bottom of Table 3 the erosion factors resulting from water temperature and relative velocity are calculated for comparison. (Note that these factors can only be valid if cavitation occurs!) The reference plant is the first one in Table 3 (former main pump design).

So a new pump design was necessary in spite of the increased booster pump head. In Figure 4b a section of the new pump design is shown. It is a forged-casing-type pump with new impeller design and inlet chamber (the short diffusor type inlet canals) and the pump has a vaned diffusor in addition to the double volute.

The NPSH-incipient-versus-flow characteristic of the new pump is shown schematically in Figure 5b (curve "a"). In March of 1982 these impellers produced bubbles only at a limited range of flow. Within this range the bubbles were maximum 5–10 mm long. At a service smaller than 60% rated flow or greater than 80%, these impellers were bubble free.

During the start-up commissioning of one unit the pumps were operated for a 1000-hour test at

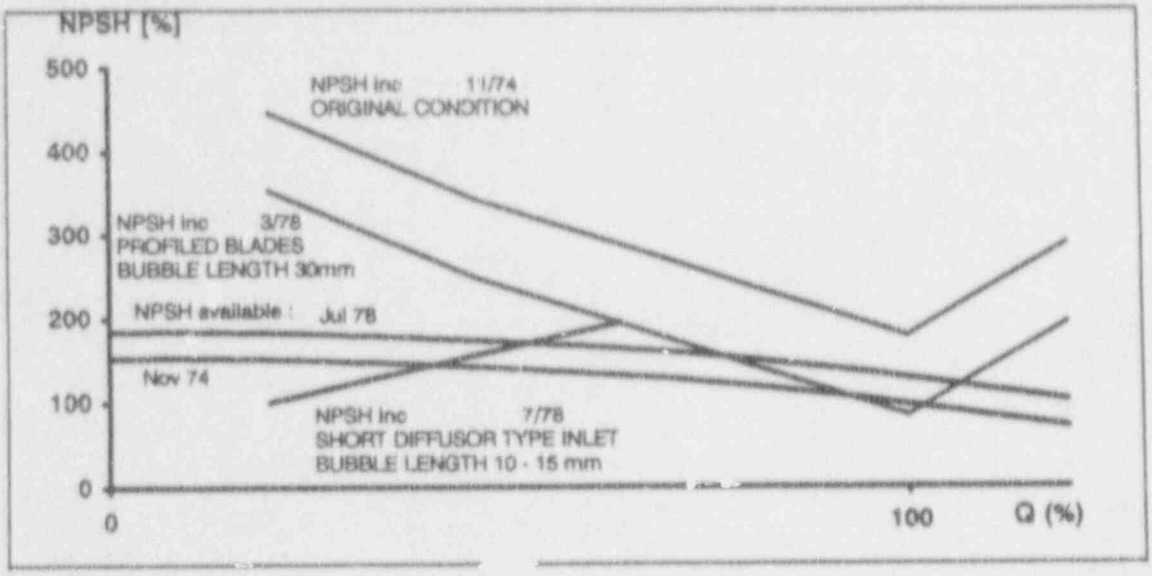


Figure 5a. Development of NPSH/flow characteristics (former main pump design).

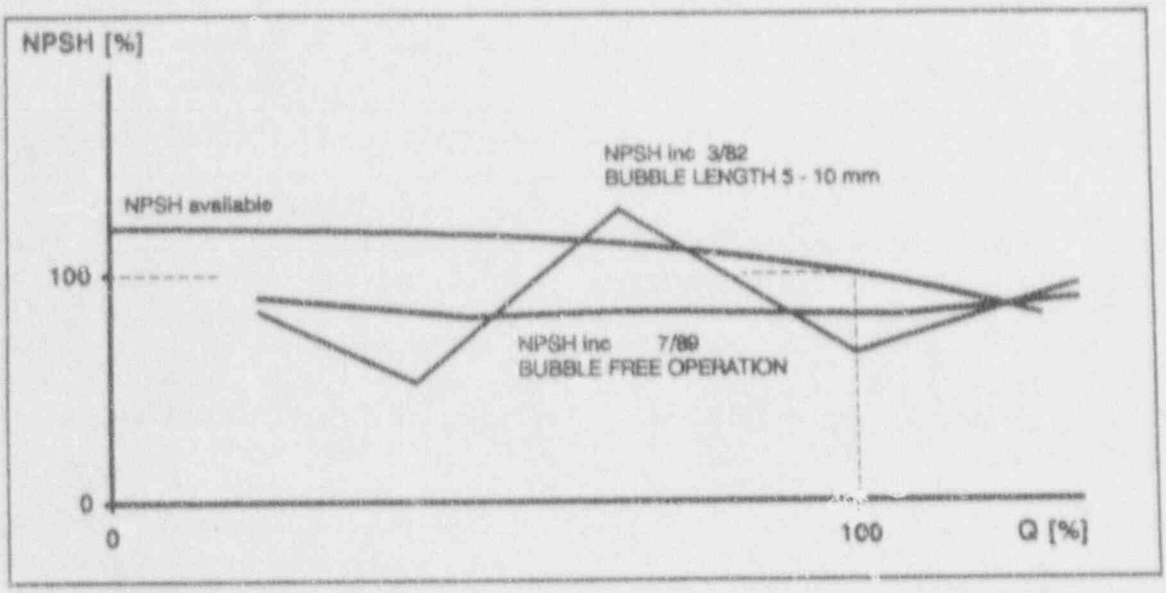


Figure 5b. Development of NPSH/flow characteristics (advanced main pump design).

minimum flow conditions. No cavitation damage was indicated at an inspection after this test.

At the beginning of 1989, this new type of pump had been in service in power plants since 1985 without cavitation problems, some of the plants were on part-load service at about 60–80% rated electrical output, about 45% of the operating time. The results are shown in Figure 6. In this diagram the average depth of cavitation erosion, the average of all blade inlet areas of an impeller, is drawn versus total time of service. The reference value is the cavitation erosion at one blade (the worst one), where the exchange of the impeller was recommended until now. In this diagram only power plants with pumps of "advanced main pump design" are represented. Because these pumps have identical geometries (only the impeller outlet geometries differ a little bit by means of slightly different outlet diameters), the main factors influencing cavitation erosion are feed water temperature, part-load operation, and impeller inlet relative velocity (refer to Table 3):

- Power plant #1 with low feed water temperature and part-load service 45% of the total operation time has the deepest cavitation erosion, as could be expected. The dotted line {1} in Figure 6 shows the erosion depth of one pump having a damaged vaned diffusor. It could be shown in model tests that the specific form of the damage creates pressure drops at the impeller suction (!) side. Thus, the flow range where cavitation occurred spread. Because this was a single failure, this pump is not included in curve 1.

- Power plant #4 has the lowest erosion rate because of its low part-load service during 10% of the operation time and a high feed water temperature.

- Power plants #2 and #5 have similar working conditions, and thus, comparable erosion rates.

- Trying to compare the power plants #3 and #4 leads to some difficulties: the plants differ slightly in relative velocity (refer to Table 3), and plant #3 has about three times more part-load service than plant #4. In this particular case, it had to be considered that cavitation could only occur in a certain range of flow rate, so cavitation erosion is limited to this range, too. Inaccuracies can occur when measuring the erosion depth from information about the operation mode. Deviations can occur between the calculated and the measured values because of inaccuracies that occur when measuring the erosion, information about the operating modes, (for example, which pump was how long in service in which kind of operation mode) and from system-induced influences. So the erosion factors should be used carefully.

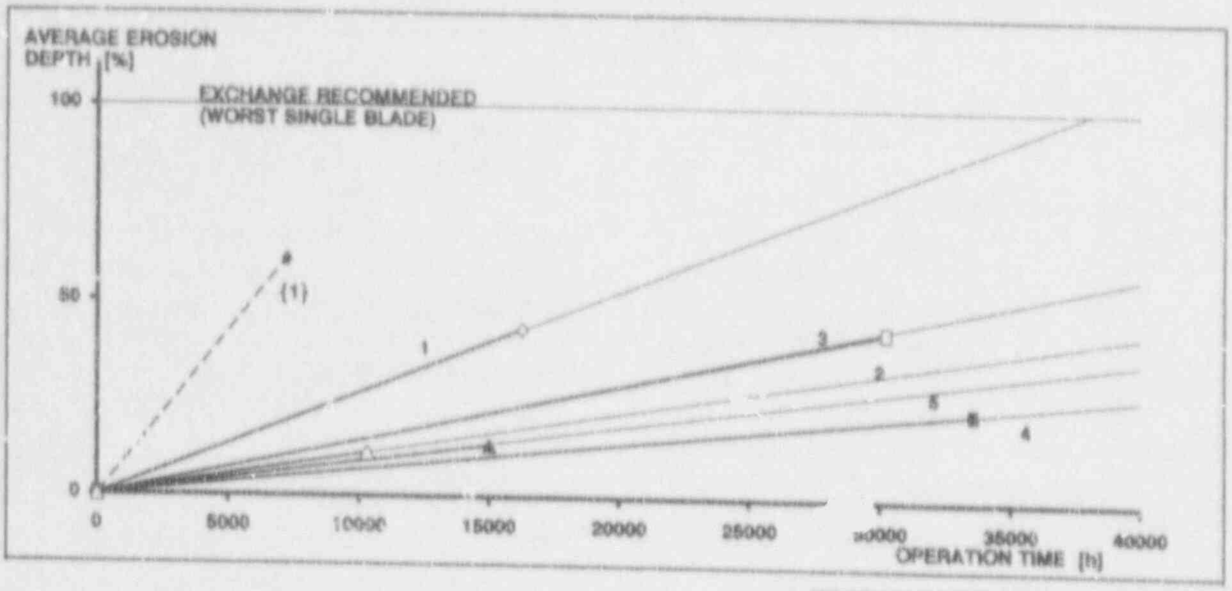
- Power plants #4 and #5 differ in feed water temperature and slightly in relative velocity. Thus, the factor between curve 4 and curve 5 should be 1.8 according to the erosion factors of Table 3. Instead, it is about 1.3 (the part-load service conditions are similar).

Coming back to our history, nobody had expected that this little bubble that only occurred between 60 and 80% rated flow would be able to cause problems. So the results of inspection at the beginning of 1989 led to the new notion that a small bubble, even if hard materials are used, would cause erosion if the operation time is thousands of hours under these operation conditions.

In July 1989 model impellers with a completely new inlet area design were tested. The blades had a special profile shape to avoid bubbles within the complete range of operation. This was the only possibility to deal with the problem. If all main parameters are given (for example, speed/inlet velocities, water temperature, NPSH-available, and materials with high erosion resistance), cavitation caused erosion can be avoided only by avoiding bubbles at the complete operation range.

So it was decided to replace the impellers of the main pumps in the three power plants with low feed water temperature (plants #1, #2, and #5). At the beginning of 1990 all original impellers with the bubble-free design were tested in the BTR (Figure 5B, curve "b"). The old impellers (advanced main pump design) were replaced during the overall maintenance inspections in 1990 even though all replaced impellers would have been able to operate.

During the overall maintenance inspections in 1991, one pump in plants #1, #2, and #5 was inspected. After about 8,000 hours of operation no indication of cavitation erosion could be detected on the blade surfaces. Because all plants have been on full-load operation, the future may show whether a part-load influence will be detectable or not.



plant No.	FW temp [°C]	part load portion [%]
1	140	46
2	140	10
3	180	30
4	180	10
5	140	10

Figure 6. Cavitation caused erosion depth (average values) recorded at different PWR-type power plants (advanced main pump design).

CONCLUSION

If a certain amount of cavitation occurs, the main parameters influencing cavitation abrasive wear are

- Impeller inlet relative velocity
- Water temperature
- Erosion resistance of impeller blade material.

The amount of cavitation (for example, the bubble length) depends on NPSH-available and operation mode. Even hard materials with excellent erosion resistance cannot withstand the impact of cavitation in high-energy impellers for a long time of service. To avoid cavitation-caused erosion, the impellers must be designed for bubble-free operation within the possible operation range. This property should be tested by the visualization method.

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Session 4

Regulatory/Operability Issues

Session Chair
Thomas G. Scarbrough
U.S. Nuclear Regulatory Commission

Regulatory/Operability Issues Inservice Testing

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ABSTRACT

Temporary Instruction (TI) 2515/114, *Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs*, was issued January 15, 1992, to provide uniform guidance for the NRC inspection of activities related to inservice testing (IST) of pumps and valves with particular emphasis on the positions in Generic Letter 89-04, *Guidance on Developing Acceptable Inservice Testing programs*, dated April 3, 1989. The TI was issued for one trial inspection in each of the five regions, with subsequent inspection to be determined based on the results from these trial inspections. Region III, Point Beach Nuclear Plant, was the first inspection performed to the TI. Region IV, Wolf Creek, followed. A summary of the results of these inspections and future activities related to the TI will be provided at the Symposium.

Temporary Instruction (TI) 2515/114, *Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs*, was issued January 15, 1992. The objective of the TI is to provide uniform guidance for the NRC inspection of activities related to inservice testing (IST) of pumps and valves with particular emphasis on the positions in Generic Letter 89-04, *Guidance on Developing Acceptable Inservice Testing Programs*, dated April 3, 1989. The TI was issued for one trial inspection in each of the five regions, with subsequent inspection to be determined based on the results from these trial inspections.

The inspection is scheduled for one week, with a minimum of two inspectors. They are considered announced inspections and a list of information the licensee is requested to have available for the inspectors is provided 2 to 3 weeks prior to the inspection. The inspection, alone, is not considered a "team" inspection, but may be conducted in conjunction with a team inspection at the discretion of Regional Management.

Sample systems are selected to review for compliance with the requirements for inservice testing in 10 CFR 50.55a, ASME Section XI, plant Technical Specifications, and Safety Analysis Reports. The implementation of alternative testing methods described in Generic Letter 89-04, Attachment 1, will be inspected. On-going inservice testing activities will be observed.

For the selected system(s), the inspectors will attempt to verify that all the applicable components are included in the IST program and that the appropriate testing is identified for each component. For testing that does not comply with ASME Section XI requirements, the inspector will verify that a relief request exists, or that the licensee has documented the applicable position of GL 89-04 in the IST program. The inspectors will review the licensee's basis for the relief requests and cold shutdown justifications to

a. This presentation was prepared (in whole, or in part) by an employee of the United States Nuclear Regulatory Commission. It presents information that does not currently represent an agreed-upon staff position. NRC has neither approved nor disapproved its technical matter.

ensure that their statements accurately describe the conditions that inhibit testing. Administrative controls for cold shutdown testing will be reviewed. Test records, program controls, implementing procedures, and post-maintenance testing will be reviewed to ensure that testing is performed as required by the IST program and corrective actions are taken in accordance with the Code requirements or approved relief requests. Reporting requirements of 10 CFR 50.72 and 50.73 will be reviewed. The licensee's process for design modifications and selected documentation will be reviewed to ensure that inservice testing requirements are incorporated as necessary.

Valve testing reviews will include a minimum of one valve in each IST category and one of each type of test included in Section XI, Subsection IWV. Check valve reviews will be performed using TI 2515/110, *Performance of Safety-Related Check Valves*, and will be a limited scope if TI 2515/110 has been used in a previous inspection.

Pump testing reviews will include pumps with no relief requests and pumps with relief requests, if applicable. Instrumentation requirements will be reviewed to ensure Code requirements are met, or relief requests exists. Corrective actions will be assessed if test results fall in the "alert" or "required action" range. The use of pump curves, expanded range limits, or any other condition that affects the test method and results will be reviewed.

Region III, Point Beach Nuclear Plant, was the first inspection performed to the TI. Region IV, Wolf Creek, followed. A summary of the results of these inspections and future activities related to the TI will be provided at the Symposium.

NRC Check Valve Action Plans and Inspection Results

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ABSTRACT

The plans to organize activities aimed at identifying and resolving the concerns about testing and performance of check valves are described in NRC's NUREG-1352, issued in June 1990. The document included a discussion of check valve problems and performance, evaluation of adequacy of current regulatory requirements, development of inspection guidance, ongoing staff research, cooperation with industry groups, participation in development of codes and standards, and evaluation of staff and industry efforts. Although industry efforts have led to some improvements in check valve performance and testing, the staff continued to receive reports and find weakness in the efforts of individual licensees to improve the performance of the check valves. The need for continued effort for improvement was reinforced by recent reports of multiple failures of check valves in main steam lines, multiple check valves rendered inoperable in service water systems because of corrosion and silt accumulation, lack of inclusion of certain essential service water check valves in an IST Program, and a report of check valve water hammer problems.

In considering the need of inspection guidance for evaluating check valve activities at nuclear power plants, the Mechanical Engineer's Branch (EMEB) and the regions developed a Temporary Instruction (TI) for evaluating the effectiveness of licensee programs regarding the performance and testing of safety-related check valves. EMEB, with assistance from the regions, proceeded with implementation of the check valve action plans as described in the May 1991 NRC Regulatory Information Conference. This paper summarizes the development and implementation of certain check valve action plan activities, including inspection results and conclusion of effectiveness of licensee programs.

INTRODUCTION

The slow progress being made in improvements in check valve performance and operational readiness as evidenced by the results of NRC-conducted inspections and reported check valve failures signaled the need for the NRC staff to take a more active role in the resolution of check valve issues. The elements of the more active role were developed and articulated in NUREG-1352 titled "Action Plan for MOVs and Check Valves" issued in June 1990 (NRC, 1990).

a. This presentation was prepared (in whole, or in part) by an employee of the United States Nuclear Regulatory Commission. It presents information that does not currently represent an agreed-upon staff position. NRC has neither approved nor disapproved its technical matter.

The implementation of the action plans was later described in a paper titled "Activities To Improve The Performance Of Check Valves" presented at the May 1991 NRC Regulatory Information Conference held in Washington, D.C. (NRC, 1991a). This paper presents background and provides a summary of the development and implementation of certain action plan activities involving the assessment of the effectiveness of licensee programs on performance of safety-related check valves.

BACKGROUND

In recent years, the importance of the need to maintain and test check valves for proper performance and operational readiness was highlighted by a plethora of problems with these valves. The importance of the need to assure proper performance of check valves and the potentially severe consequences that could result from their failure was clearly focused by an event at San Onofre Nuclear Generating Station's Unit 1 on November 21, 1985. In that event, with the plant operating at 60% power, a partial loss of plant power following a reactor trip resulted in a severe water hammer incident in the feedwater system. The water hammer occurrence caused a leak, damaged plant equipment and jeopardized the integrity of safety systems. The failure of five safety-related check valves in the feedwater system had significant involvement in the event. (NRC no date)

Recognizing the need to improve check valve maintenance, surveillance testing and performance, following the San Onofre event, an industry Owners Group Task Force formulated an initial plan for the industry's response to the check valve performance and operational readiness problems. Since then, several industry activities were initiated in an attempt to improve check valve performance and operational readiness, including an industry workshop on check valve problems, issuance of industry-recommended guidance on check valve programs, preparation of guidelines for check valve applications and plant visits that have confirmed that the recommendations have been implemented in varying degrees. More recently, the formation of a nuclear utility group on check valves (NIC) and the ASME Operation and Maintenance Working Group on Check Valves (OM-22) has evolved. NIC has completed an evaluation of the application of nonintrusive techniques for check valve testing (Utah State University Foundation) and has currently undertaken tasks, in a joint effort with Electric Power Research Institute (EPRI), to update valve application guidance and develop a check valve maintenance guide. OM-22 has developed significant check valve testing provisions for incorporation into the ASME OM Code.

REGULATORY REQUIREMENTS

NRC regulations applicable to check valves are contained in Part 50 of Title 10 of the Code of Federal Regulations (10 CFR 50). Specifically, Appendix A to 10 CFR 50, Criterion 1 states, in part, that components, structures and systems shall be designed, fabricated, erected and tested to quality standards commensurate with the importance of the safety functions to be performed and a quality assurance program shall be established and implemented to provide adequate assurance that these components will satisfactorily perform their safety functions. Further, Appendix B to 10 CFR 50 encompasses quality assurance of all planned and systematic actions necessary to provide adequate confidence that a component, system or structure will perform satisfactorily in service. This quality assurance applies to all activities affecting the safety-related functions, including inspections, testing, operating and maintaining those components, systems and structures. 10 CFR 50.55a(g) requires that Code class pumps and valves be inservice tested in accordance with the Section XI requirements of the ASME Boiler and Pressure Vessel Code. The NRC issued Generic Letter 89-04 to outline positions that described inservice valve testing deficiencies and explain certain ASME Code requirements and certain alternatives to the ASME Code that the staff considered acceptable.

CHECK VALVE INSPECTION FINDINGS

Although ensuing industry efforts have led to some improvements in check valve testing and performance, the staff continued to find weakness in the efforts of individual licensees to improve the performance of check valves. The need for continued improvement was reinforced by the results of NRC inspections and check valve failure reports. Some examples of NRC inspection findings regarding licensee check valve activities are included in NRC Information Notice No. 88-70. The Notice reported on check valve program deficiencies uncovered at McGuire Nuclear Station, Units 1 and 2, Zion Nuclear Plant, Units 1 and 2, and Ginna Nuclear Power Plant during NRC inspections conducted from September 14, 1987, through May 20, 1988. The common findings from these inspections were:

1. Not all safety-related check valves were included in the IST program
2. Check valves were not always tested in a way that verified their ability to perform their safety-related functions.

The Notice further pointed out that these deficiencies could result in undetected serious degradation of components as surfaced at the San Onofre Unit 1 event.

Some later examples of inspection findings regarding check valve issues were reported by the NRR Division of Licensee Performance and Quality Evaluation, based on Maintenance Team Inspections (MTIs) conducted in 1988 and 1989. In 1988, MTIs conducted at Diablo Canyon, Trojan, Wolf Creek, and Peach Bottom uncovered the following licensee-specific or common deficiencies in certain systems:

1. Licensee failed to establish a preventative maintenance or test program
2. Licensee was slow to implement corrective action
3. Licensee test program did not include all safety-related check valves or required testing.

In 1989, MTIs conducted at Rancho Seco, Waterford, Brunswick, Limerick and Indian Point 3 uncovered the following licensee-specific or common deficiencies in certain systems:

1. Root cause evaluations were not conducted on valve problems
2. Industry experience was insufficiently considered in problem recognition
3. Licensee was slow in starting the development of a check valve program.

NRC CHECK VALVE ACTION PLAN

Continuing staff concerns and the slow progress being made in improvements in check valve performance and operational readiness as evidenced by the results of NRC inspections and reported check valve failures signaled the need for the NRC staff to take a more active role in the resolution of check valve issues. An NRC action plan for monitoring licensee improvements in check valve performance and operational readiness was developed and published in NUREG-1352. The NUREG described NRC's ongoing and future activities to foster improvements in the performance of check valves. The major activities included in the plan were efforts to (1) identify check valve problems and testing and performance weaknesses, (2) evaluate the adequacy of regulatory requirements, (3) support the development of Codes and Standards, (4) develop inspection guidance, (5) conduct research studies, (6) evaluate industry efforts to improve check valve performance, (7) evaluate the effectiveness of licensee activities regarding testing and performance of safety-related check valves, and (8) evaluate the need for regulatory guidance.

Check valve failures contained in sample Licensee Event Reports (LERs) in 1990 and 1991, described failures attributed to hardware, procedures and personnel. The categories of cited failures included: (1) safety-related valves nc included in test programs, (2) valves not backflow tested, (3) valve disc separations, (4) valves fail to close, and (5) valve closure causes waterhammer damage.

DEVELOPMENT OF INSPECTION GUIDANCE

On the basis of NRC concerns with check valve performance, reinforced by the failure reports, and considering the need for inspection guidance for evaluating check valve activities at nuclear power plants, EMEB and the Regional Inspectors developed TI 2515/110 for evaluating the effectiveness of licensee programs regarding the performance and testing of safety-related check valves. This TI was developed to assess the effectiveness through inspections of selected activities to determine whether there are actions in place or a system of organizationally-controlled and directed processes that will assure check valve operability. The following activities are included in the inspection to provide this type of assurance: (a) management involvement and support, (b) inservice testing program, (c) preventative maintenance program, (d) failure trending program, (e) corrective action program, (f) use of industry experience information, (g) training, and (h) plant check valve failure rate history.

NRC CHECK VALVE ACTION PLAN ACTIVITY

The NRC staff-planned action for use of the TI consists of a two-phase pror

The first phase inv trial audits of three licensee plants in Regions I, III and IV conducted by EMEB with regional inspector participation. The audits included PWR and BWR plants. The purpose of the audits was to assess the need for the TI and to evaluate the TI guidance.

The second phase involved implementation of the TI if the results of phase one confirmed its need. The purpose for implementation of the TI by regional inspectors was to assess the performance of specific licensees and provide information on the need for guidance on generic check valve program activities and possible problems.

The trial audits were conducted at Byron Station Units 1 and 2, Nine Mile Point Unit 2 and Waterford 3 over the time period from July 8, 1991, to October 4, 1991 (NRC, 1991b; 1991c; 1991d). The audit results confirmed the need to implement the TI.

TRIAL AUDIT RESULTS

The check valve activities of each of the three licensees were included in two separate programs identified as an inservice testing (IST) program and a preventative maintenance or reliability program (often referred to as the check valve program).

All licensees were in the early stages of evaluating, testing and planning for the implementation of nonintrusive methods to replace certain disassembly and inspection procedures.

IST PROGRAMS

All licensees had an IST program that was comprehensive, well organized and implemented. Licensee-specific program concerns were identified during the review at each plant. It was concluded that overall the licensees' IST programs were satisfactory.

The benefits of existing IST ASME Code requirements and regulatory guidance provided by GL 89-04 were evident.

CHECK VALVE PROGRAMS

The check valve preventative maintenance and reliability programs were driven by general industry guidance. The lack of consistency and specific guidance was evident in the review of the programs.

Implementation of the check valve programs was in various stages. One plant had a documented program in place and had completed preventative maintenance procedures for about two-thirds of the valves. Another plant had a documented program in place and expected to have completed at least one inspection of each valve in the reliability program by the end of their next refueling outage. The third plant had a program in the initial stages of being addressed in procedures, but had not started implementation of their program.

The lack of sufficient guidance was evident in the similarity of one weakness found in each program. This weakness was in the criteria and review process used to select check valves for inclusion in the preventative maintenance or reliability programs. The valve selection criteria and review were limited to or dominated by valve misapplication and installation problems as opposed to further concerns such as safety significance, corrosion, cleanliness of systems, aging and higher incidences of problems in certain systems. Some programs were established with insufficient guidance with respect to priority and frequency of check valve monitoring, inspection and maintenance activities.

Two plants had a formal program document describing check valve activities and interfaces while the other plant program was being developed from several loosely-connected and significantly time-spaced documents.

CONCLUSION

The regions were issued the TI at year end 1991 with implementation instructions. The regions are currently conducting or scheduling TI inspections of specific licensee's plants. EMEB will support the regions and participate in certain inspections or provide additional expertise.

Region inspections conducted to date support the trial audit findings. EMEB will evaluate the results of all the TI inspections to identify the need for other regulatory guidance. EMEB will also determine if the need for guidance identified in the trial audits is a generic or licensee-specific issue. In assessing the need for NRC generic guidance, ongoing industry activities will be considered.

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Operating Pumps on Minimum Flow

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ABSTRACT

The Nuclear Regulatory Commission (NRC) regulations in Appendix A to 10 CFR 50 require that components important to safety be designed and tested to quality standards commensurate with the importance of the safety functions to be performed. The NRC regulations in 10 CFR 50.55a reference the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code for criteria to conduct inservice testing of pumps. The ASME Code allows the performance of pump inservice testing using mini flow bypass loops. Operating experience and studies performed for the Nuclear Plant Aging Research Program (NUREG/CR-4597, Vols. I and II) showed that a leading cause of pump problems and failures is associated with hydraulic instability phenomena induced by low-flow operation.

The NRC staff issued Information Notice (IN) 87-59 to alert all licensees to two miniflow design concerns identified by Westinghouse. The first potential problem discussed in this IN involves parallel pump operation. If the head/capacity curve of one of the parallel pumps is greater than the other, the weaker pump may be deadheaded when the pumps are operating at low-flow conditions. The other problem relates to potential pump damage as a result of hydraulic instability during low-flow operation. In NRC Bulletin 88-04, dated May 5, 1988, the staff requested all licensees to investigate and correct, as applicable, the two miniflow design concerns. The staff also developed a Temporary Instruction, TI 2515/105, dated January 29, 1990, to inspect for the adequacy of licensee response and follow-up actions to NRC Bulletin 88-04.

Oak Ridge National Laboratory has reviewed utility responses to Bulletin 88-04 under the auspices of the NRC's Nuclear Plant Aging Research Program, and participated in several NRC inspections. Examples of actions that have been taken, an assessment of the overall industry response, and resultant conclusions and recommendations are presented.

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b. This presentation was prepared (in whole, or in part) by an employee of the United States Nuclear Regulatory Commission. It presents information that does not currently represent an agreed-upon staff position. NRC has neither approved nor disapproved its technical matter.

BACKGROUND

Historically, minimum flow capacity for centrifugal pumps was based on ensuring that the temperature rise through the pump was not excessive. As a general rule of thumb, the minimum flow rate was specified so that the temperature rise through the pump would be less than 15°F (it should be noted that this rule of thumb was not universally applied, and temperature rises greater than 50°F have been used for some pump applications).

It has been recognized for many years that in higher energy density pumps at low-flow operation, destructive hydraulic forces, not temperature rise, limit safe minimum flow. Unsteady flow conditions within the pump result in substantial radial and axial forces (static as well as dynamic) on both the stationary and rotating parts. Resultant damage can be manifested in a number of ways, including impeller or diffuser breakage, thrust bearing and/or balance device failure from excessive loading, cavitation damage on suction stage impellers, increased seal leakage or failure, seal injection piping failure, shaft or coupling breakage, and rotating element seizure (Adams and Makay, 1986). In addition to the internal forces generated by unsteady flow within the pump itself, interaction between the pump and the system at low-flow conditions can result in substantial surging and vibration that can affect not only the pump, but other system components and supports.

As the effects of low-flow operation have become better understood by pump technologists, design modifications that can reduce unsteady flow conditions have been developed. Modifications to pump geometries have been demonstrated to allow operation at lower flow rates with substantially reduced impact (Makay and Barrett, 1984). Some original equipment manufacturers (OEM) and non-OEM repair shops for pumps now offer design options or retrofits that allow pumps to be operated acceptably at reduced minimum flow. However, there remain in service a large number of pumps that were not designed specifically to allow operation under low-flow conditions and for which no modifications have been made.

In May of 1988, the Nuclear Regulatory Commission (NRC) issued Bulletin 88-04, "Potential Safety-Related Pump Loss." The Bulletin addressed two general concerns:

- The potential for dead-heading one of two pumps when operated in parallel
- The adequacy of pump minimum flow protection provided by the installed minimum flow lines.

With regard to the first concern, the Bulletin specifically discussed the potential problem of parallel pump operation during miniflow operation, noting that the stronger of two pumps can dead-head the weaker pump. It was also noted that the strong/weak pump situation is not a problem at moderate to high flow conditions because of the shape of pump head-capacity curves in those regions. Relative to the second item, the Bulletin noted that pump manufacturers now advise that desired minimum flow capacity is greater than was originally specified for some pumps. The Bulletin required that all plants conduct a review of all safety-related pumps.

Oak Ridge National Laboratory (ORNL), under the auspices of the NRC's Nuclear Plant Aging Research Program, reviewed industry responses to the issues identified in the Bulletin (Casada, 1991). The principal purpose of the study was to provide a general assessment of the type and extent of actions taken in response to the Bulletin. The results of this study are summarized in this paper. The review consisted of several elements:

- Discussions with representatives of several pump manufacturers
- General review of all utility correspondence to the NRC responding to the Bulletin
- Review of the distribution of pump suppliers whose pumps are used in selected systems

- Detailed onsite review of selected plants.

The results of this study are summarized in this paper.

ORNL also participated in individual plant inspections to support NRC's Nuclear Reactor Regulation activities. The inspections were conducted to review the adequacy of the specific plants' responses to the Bulletin. Some observations made during the inspections are presented.

DISCUSSIONS WITH PUMP MANUFACTURERS

ORNL met with representatives of four of the major manufacturers of pumps used in safety-related service in U.S. plants. These four manufacturers together have furnished about 75% of the pumps used in the safety-related systems of primary concern. Three of the principal areas of discussion, and the results of the discussions are provided below.

- Which types of pumps are most susceptible to low-flow degradation?

The pumps noted to be most susceptible to low-flow degradation were high-energy and high-suction, specific-speed (high-flow, low NPSH required) pumps.

- What are the associated failure modes?

Failure/degradation modes associated with low-flow operation that were most often mentioned were seal failure, occasional shaft breakage, bearing failure, excessive wear of wear rings, and cavitation damage.

- What are some possible means for demonstrating a pump's ability to successfully operate under low-flow conditions?

The OEMs identified correlations that can provide some very general insights into the relative requirements for pump minimum flow (Gopalakrishnan, 1988 and Heald and Palgrave, 1985). However, the use of the generic correlations to address 88-04 was discouraged.

Managers agreed uniformly that there was a need to be able to measure the forces present (for example, radial thrust) in order to predict component life. It was recognized, however, that this is not practical for field-installed pumps. There was a general consensus that with the current state of understanding, a properly conducted field test could verify a particular pump's capability under the test conditions. One manufacturer noted that tolerance stack-ups and pre test service life could play major roles in the results of such testing. Another manufacturer noted that in tests conducted by their company overseas, the hydraulically induced forces associated with several pumps of the same model varied by a factor of three to four.

Several of the manufacturers emphasiz the need for a test program to address intermittent operation at low flow, in light of the fact that there is no objective data related to such operation.

The suitability of current monitoring practices was discussed. There was consistent agreement that testing pumps under miniflow conditions was of little value, from a hydraulic performance demonstration standpoint. The vendors observed that of the means that are currently practicable, spectral vibration monitoring and trending was the best indicator of potentially damaging conditions. However, they noted that monitoring capability for some pumps (specifically deep-well pumps) was limited. They also noted that the parameter that most needed to be measured was force on pump components, which could only be accurately monitored using intrusive means. Two changes to current in service testing practices were recommended:

- Periodically conduct testing at close to the pumps best efficiency point (BEP) to verify that the pump performance has not substantially degraded
- Minimize or discontinue practice of routinely testing pumps at minimum flow conditions in order to demonstrate pump operability.

ASSESSMENT OF WRITTEN PLANT RESPONSES

General Discussion

The correspondence from all plants to the NRC on Bulletin 88-04 was reviewed to provide an indication of actions taken in response to the Bulletin. The review evaluated the licensees' analyses and data for low-flow operation presented in their response and determined what actions, in terms of design changes, procedure changes, or special inspections, have been or will be made.

The level of information provided in the correspondence varied substantially. For some plants, there was a fairly detailed discussion of original and current minimum flow recommendations and existing system configuration, as well as an identification of specific design, procedural, or other changes made to address Bulletin concerns. There were also a number of responses that provided only an indication that the issues had been reviewed, with little or no system- or pump-specific information provided.

Procedural and Design Changes and Special Testing

An attempt was made to determine the extent and types of procedural and design changes made in response to the Bulletin, as well as special tests that were conducted.

The distribution of identified changes, by system, for both pressurized water reactors (PWRs) and boiling water reactors (BWRs) is provided in Figures 1 and 2. It is likely that other procedural or other administrative actions were taken that were not identified in the written responses, and thus are not reflected in these figures.

Several different types of design changes were identified in the responses as indicated in Figure 3. The majority of the design changes involved either increasing the size of the orifice in the miniflow line or otherwise modifying the minimum flow line. It is noteworthy that no modifications to pump design were identified in the written responses. After the review was completed, verbal discussions revealed one plant that is modifying auxiliary feedwater pump impeller vane and diffuser clearances to minimize low-flow operation problems.

A total of 44 special tests or inspections that either had been conducted or would be conducted to monitor pump condition were identified. About two-thirds of these tests were associated with either low pressure core spray (LPCS), low pressure coolant injection/residual heat removal (LPCI/RHR), low pressure safety injection/residual heat removal (LPSI/RHR), or containment spray (CS) systems.

A number of special analyses were also identified. Most of these involved parallel pump competition. A total of 48 such analyses were specified. Almost half of these involved the LPCI/RHR or LPSI/RHR systems. Twenty-one of the analyses were minimum acceptable flow calculations, based on either a published correlation or on other undesignated bases. Note that these analyses were performed by the utility (not by the pump manufacturer).

It was noted during the review that a substantial fraction of the design and procedural changes were made by a relatively small portion of the industry. Figure 4 indicates that 80% of the changes were made by about 30% of the plants. All noted changes were made by less than half of the plants.

Summary

The written response review made two principal observations:

- There are no generic guidelines for determining the acceptability of a pump for operation under the various modes and times required in support of both normal and emergency conditions
- The low-flow issue was not adequately addressed by all plants.

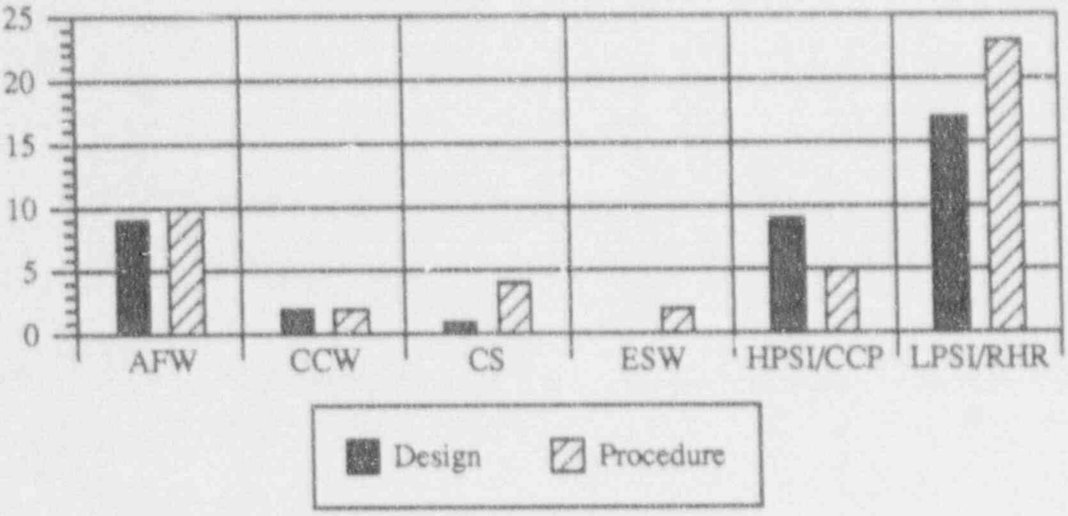

Figure 1. Number of PWR units in which design and procedure changes were made.

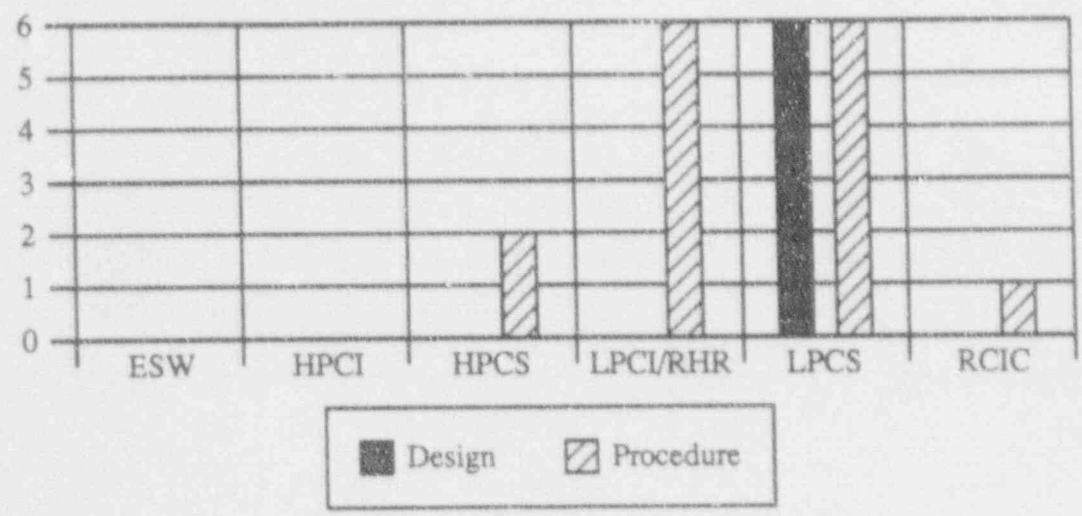

Figure 2. Number of BWR units in which design and procedure changes were made.

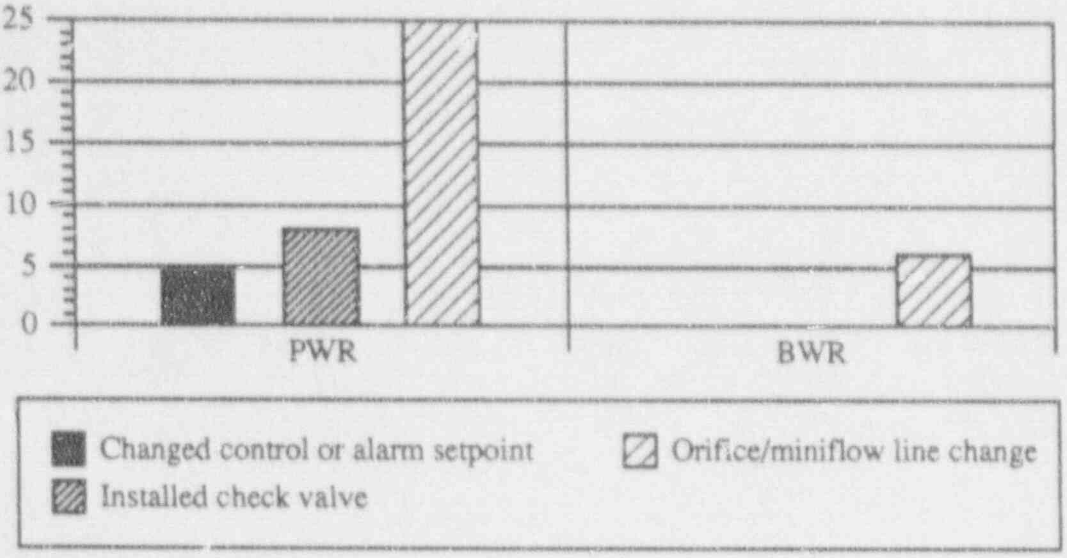

Figure 3. Distribution of design changes, by type of change.

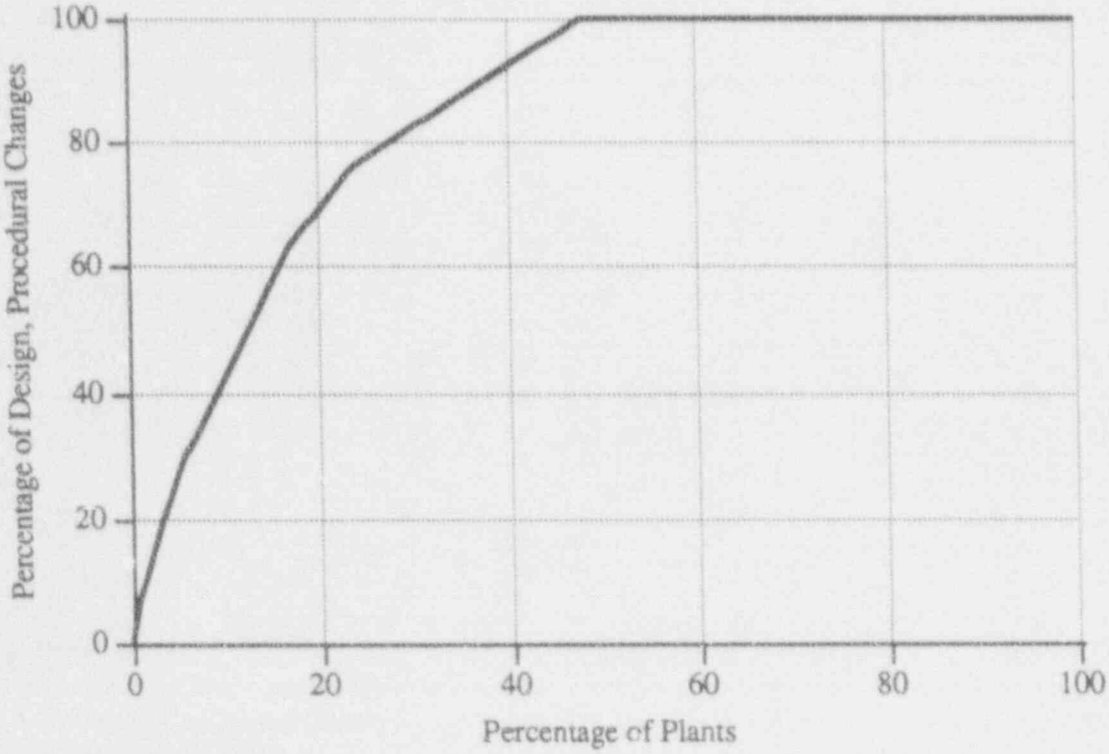

Figure 4. Distribution of procedural or design actions taken in response to NRC Bulletin 88-04.

The lack of generic guidelines essentially guaranteed that the issue would not be uniformly and adequately addressed.

INDIVIDUAL PLANT INSPECTIONS

The NRC has conducted inspections of several plants to review plant responses in more detail than was possible through the correspondence review discussed above. ORNL participated, primarily under the auspices of the Mechanical Engineering Branch, in four inspections conducted at a single-unit Combustion Engineering plant, a twin-unit General Electric plant, a single-unit Westinghouse plant, and a three-unit facility comprised of a single Westinghouse unit and two Combustion Engineering units. In addition, ORNL assisted the NRC in the review of another plant's pump test and subsequent disassembly/examination program. These inspections confirmed the observations made during the review of the written responses. Some pertinent observations made during the inspections are provided below.

Parallel Pump Competition

Personnel at one of the plants inspected had identified that parallel pump competition existed in the RHR system and were installing check valves to preclude competition in the future. The actions taken by the utility to address the pump competition were judged to be appropriate.

During another inspection, the inspectors noted the potential for parallel pump interaction to exist during a pump switch-over sequence. In this case, two RHR[c] pumps share a common minimum flow line, and normally only one pump is operated. The concern primarily exists when switching over from one pump to the other. The utility agreed that the potential exists, and identifying corrective actions.

The other two inspections also found situations in where it was not obvious that pump competition would be precluded. In one case, personnel contracted by the plant to review the Bulletin concerns had failed to note the absence of orifices in the minimum flow lines of the core spray pumps for one unit (the orifices did exist on the other). The minimum flow lines from two pumps joined into a common header, and it was not clear that the head losses before the common header connection were sufficient to prevent pump competition. Subsequent hydraulic analyses showed that competition should not occur. For the other plant, several minimum flow lines, each of which carried several hundred gallons per minute, joined in a common header. No analysis was performed to verify the absence of pump competition before the inspection.

While the latter two situations have subsequently been found to be acceptable, they had not been considered before the inspections and indicate a somewhat superficial review.

The primary generic concern of parallel pump competition (affecting several RHR systems) has been adequately identified and addressed. The only new parallel pump competition concern identified in the four inspections is specific to a single unit. It is unlikely that additional pumps are generically within any of the four scopes of supply for the nuclear steam supply system (NSSS), including the emergency core cooling system pumps, for which a parallel pump competition problem exists that has not been identified. This conclusion is based on the facts that no other parallel pump competition situations were identified and that most NSSS systems are reasonably similar in design. However, the fact that three of the four plants inspected had conditions for which potential competition existed, but had not been identified by the utilities, leaves some residual uncertainty.

c. The RHR pumps in this plant are used solely for residual heat removal, unlike the duelfunction RHR/Low-pressure injection pumps in most plant.

Minimum Flow Adequacy

Most of the inspection activities were oriented toward the adequacy of minimum flow provisions. At three of the four facilities, the majority of the OEMs had been contacted to determine the current recommended minimum flow rate. None of these three plants had contacted the service water pump OEM (Casada, 1991)[d]. However, at the fourth facility (the three-unit plant), the OEM for only one pump had been contacted.

For the 14 pump applications at the four facilities reviewed for the manufacturer's current recommended minimum flow rate, Table 1 lists the distribution of the level of conformance with supplier recommendations.

The observations made during the on site inspections relative to parallel pump competition and the potential inadequacy of minimum flow supported the conclusion reached during the review of the written responses that, generally speaking, the response to the low-flow degradation issue has been relatively superficial. There was a considerable variation in the approaches taken by individual plants involved in the inspection to address the Bulletin concerns.

The following methods were judged to be appropriate means of ensuring adequate pump reliability, as noted during both the written response review and during the inspections:

- Verification that the pump minimum flow line supports a flow rate that meets the vendor recommendations for continuous operation
- Verification that although the pump minimum flow line does not support a flow rate that meets the vendor recommendations for continuous operation, it does support a flow rate that meets a time-restricted flow rate, along with a verification that there are administrative controls are adequate to ensure that the pump is not operated for a time in excess of the vendor recommendations
- For pumps that did not meet the vendor recommendations during regular operation, a commitment by the plant to carefully monitor pump performance and to periodically disassemble and examine the pump for signs of damage.

Approaches taken to address the issue that were judged inadequate or had other weaknesses included one or more of the following:

- Reliance on static modeling of pumps
- Reliance on the absence of low-flow-attributed failures (i.e., not the absence of failures, rather the fact that none of the failures that had occurred had been attributed by plant personnel to low-flow operation)
- Reliance on non-spectral vibration data from in service testing
- Dependence upon instrumentation whose accuracy at test conditions was insufficient to form solid conclusions
- Failure to have in place administrative controls that would ensure compliance with pump manufacturer recommendations
- The assumption of orifices being present in minimum flow lines when there were none
- Failure to recognize that off-normal procedural controls created situations where the pumps would be operated outside the manufacturer recommendations (even though the plant had determined that pump operation would fully comply with the recommendations when operated under normal conditions).

While some of the above approaches are not without merit, the extent to which they were relied upon (in various combinations) did not

d. Note that, generally speaking, the service water system pumps would normally be run at a substantial fraction of design flow, and most likely minimum flow adequacy would not be a concern.

Table 1. Level of conformance to OEM recommended minimum flow.

Number of pumps	Level of conformance to OEM minimum flow recommendations
4	Fully meet current recommendations with existing configuration.
1	Meet current recommendations after completing system modification.
3	Nominally meet OEM recommendations (available flow is essentially equal to that recommended).
2	Meet current recommendations, for most plant conditions. Some off-normal (non-emergency) conditions could lead to operation outside the recommendations.
4	Do not meet current recommendations.

provide an adequate level of assurance that the pump would operate reliably.

The site inspections revealed varying levels of appreciation for the types of damage that could be manifested from operation of pumps at low flows. Some plant personnel had an excellent understanding of the damage mechanisms and appropriate means of monitoring, while others were not well aware of the potential problems.

Most plants had reviewed at least the historical failure data for the pumps at their plant; however, little had been done in the way of reviewing failure data more generically. For example, one plant used an auxiliary feedwater (AFW) pump of the same model and with similar minimum flow rates as another plant that had found substantial low-flow related damage when the AFW pumps were disassembled and examined (Gopalakrishnam, 1988).[e] The plant personnel were not aware of the experience at the other plant, even though the information was available through the Nuclear Plant Reliability Data System.

All of the plants inspected were using, or were developing, a pump spectral vibration monitoring program that goes beyond ASME Code requirements. There appeared to be a need for improved monitoring and trending of the data, however. As an example, an AFW pump at one plant showed substantial increases in vibration at vane passing frequency during the two most recent tests. However, there was no programmatic monitoring and trending practice to note the change; as a result, no actions were being taken to monitor the pump more carefully. It should be noted that the ASME Code-required testing results did not reflect the increased vibration at this frequency because the in service test program monitored only overall displacement.

GENERAL CONCLUSIONS

Based upon the review of the written responses and the individual plant inspections, pump competition concerns appear to have been adequately addressed by the utility reviews. However, additional efforts are needed to resolve low-flow-related issues. Fundamentally, there is a need for a better definition of how a pump could be qualified for low-flow operation. In order to achieve the required level of definition, additional insights into the parameters that influence a pump's ability to operate successfully at low-flow

e The damage found ultimately resulted in the issuance of a 10 CFR 21 report from the manufacturer. See discussion under "A Postscript."

are needed. Also needed is a better understanding of design or monitored parameters that could be used to make the necessary determinations.

This issue is currently being prioritized in the NRC potential generic issue process. The purpose of the prioritization is to assess the safety significance of the issue and the need for additional NRC guidance on evaluating and correcting miniflow deficiencies.

A POSTSCRIPT

In September 1991, at the time that the individual plant inspections were being completed, Ingersoll-Rand issued a 10 CFR 21 report (Young, 1991) on broken cast iron diffusers in multistage pumps used in AFW applications. The problem was initially identified at Surry Unit 2 in 1988. Following a reactor trip, low flow to one steam generator was noted. Inspections found that pump diffuser vane pieces had lodged in a venturi, thereby restricting flow.

Subsequent inspections found that cavitation damage to the AFW cast iron diffusers was evident, particularly at the leading edge of the diffuser vanes. Damage was most evident at the first stage diffuser, although damage was also seen in other stages. Damage was also found at some areas of diffuser vane to shroud junctions, which was also believed to be the result of low-flow induced cavitation erosion. Ingersoll-Rand noted that the primary cause of the breakage was cavitation damage at the leading edge of the diffuser vanes, which resulted from accumulated operation of the pump at minimum flow.

After finding indications of similar damage at other plants, Ingersoll-Rand issued the 10 CFR 21 report. In several of the instances in which damage was found, the pumps were satisfactory, according to technical specification and ASME-Code-required testing results. This could be anticipated because the pumps are tested at minimum flow conditions, where internals degradation of this nature would not be detectable.

Ingersoll-Rand recommended periodic inspection of the pumps for damage of the diffusers, and replacement of the cast iron parts with stainless steel, if necessary. They further recommended conducting periodic testing at higher flow rates, if possible.

At least one plant in our knowledge has elected to modify the AFW pumps by changing the impeller/diffuser gap clearances. This type of design change has been successfully employed on a large number of high-energy fossil plant pumps. The potential benefit offered by this change, in lieu of or in addition to the material replacement change, is that the root cause of the diffuser vane damage is being addressed more directly. This has the benefit of not only minimizing cavitation damage, but also of minimizing overall loading on the pump rotating and stationary parts, thereby minimizing other vibration-related problems that have resulted from low-flow operation.

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List of Acronyms

AFW	auxiliary feedwater
ASME	American Society of Mechanical Engineers
BEP	best efficiency point
BWR	boiling water reactor
CCP	centrifugal charging pump
CCW	component cooling water
CS	containment spray
ESW	emergency service water
HPCI	high pressure coolant injection
HPCS	high pressure core spray
HPSI	high pressure safety injection
LPCI	low pressure coolant injection
LPCS	low pressure core spray
LPSI	low pressure safety injection
NPSH	net positive suction head
NRC	U.S. Nuclear Regulatory Commission
OEM	original equipment manufacturer
ORNL	Oak Ridge National Laboratory
PWR	pressurized water reactor
RCIC	reactor core isolation cooling
RHR	residual heat removal

NRC Inspections of Licensee Activities to Improve the Performance of Motor-Operated Valves

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ABSTRACT

The NRC regulations require that components important to the safe operation of a nuclear power plant be treated in a manner that provides assurance of their proper performance. Despite these regulatory requirements, operating experience and research programs have raised concerns regarding the performance of motor-operated valves (MOVs) in nuclear power plants. In June 1990, the staff issued NUREG-1352, *Action Plans for Motor-Operated Valves and Check Valves*, which contains planned actions to organize the activities aimed at resolving the concerns about MOV performance. A significant task of the MOV action plan is the staff's review of the implementation of Generic Letter (GL) 89-10 (June 28, 1989), "Safety-Related Motor-Operated Valve Testing and Surveillance," and its supplements, by nuclear power plant licensees. The NRC staff has issued several supplements to GL 89-10 to provide additional guidance for use by licensees in responding to the generic letter.

The NRC staff has conducted initial inspections of the GL 89-10 programs at most licensee facilities. This paper outlines some of the more significant findings of those inspections. For example, licensees who have begun differential pressure and flow testing have found some MOVs to require more thrust to operate than predicted by the standard industry equation with typical valve factors assumed in the past. The NRC staff has found weaknesses in licensee procedures for conducting the differential pressure and flow tests, the acceptance criteria for the tests in evaluating the capability of the MOV to perform its safety function under design basis conditions, and feedback of the test results into the methodology used by the licensee in predicting the thrust requirements for other MOVs. Some licensees have not made adequate progress toward resolving the MOV issue for their facilities within the recommended schedule of GL 89-10.

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a. This presentation was prepared (in whole, or in part) by an employee of the United States Nuclear Regulatory Commission. It presents information that does not currently represent an agreed-upon staff position. NRC has neither approved nor disapproved its technical matter.

The NRC regulations require that components that are important to the safe operation of a nuclear power plant be treated in a manner that provides assurance of their performance. Appendices A, "General Design Criteria for Nuclear Power Plants," and B, "Quality Assurance Criteria for Nuclear Power Plants and Fuel Reprocessing Plants," to 10 CFR Part 50 provide a broad-based framework of requirements for the design, testing, operation and maintenance of components, including motor-operated valves (MOVs), that are important to the safe operation of the plant. With respect to inservice testing of MOVs, 10 CFR Part 50.55a(g) requires compliance with Section XI, "Rules for Inservice Inspection of Nuclear Power Plant Components," of the ASME Boiler and Pressure Vessel Code for MOVs within the scope of the Code. The new maintenance rule (10 CFR 50.65) also provides requirements that address the performance of certain MOVs in nuclear power plants.

Despite these regulatory requirements, operating experience and research programs have raised concerns regarding the performance of MOVs in nuclear power plants. MOV problems have included inadequate MOV design, testing and maintenance, as well as incorrect torque, torque bypass, and limit switch settings, which have led to failures of MOVs to perform their intended functions. For example, the failure of MOVs in both trains of the low pressure coolant injection subsystem in May 1991 required the shutdown of the Fitzpatrick nuclear power plant for an extended period of time. Also, in November 1991, the licensee of the Wolf Creek nuclear power plant began to identify numerous problems with safety-related MOVs that required the plant to significantly extend its planned outage.

In June 1990, the NRC staff issued NUREG-1352, "Action Plans for Motor-Operated Valves and Check Valves," which describes planned actions to organize the activities aimed at resolving the concerns about MOV (and check valve) performance. Those actions include evaluation of the current regulatory requirements and guidance applicable to MOVs, development of guidance for and coordination of NRC inspections, completion of NRC MOV research programs and implementation of the research results, and providing MOV information to the nuclear industry.

A significant task of the MOV action plan is the staff's review of the implementation of Generic Letter (GL) 89-10 (June 28, 1989), "Safety-Related Motor-Operated Valve Testing and Surveillance," and its supplements, by nuclear power plant licensees. In GL 89-10, the staff requested that licensees help ensure the capability of MOVs in safety-related systems by reviewing MOV design bases, verifying MOV switch settings initially and periodically, testing MOVs under design-basis conditions where practicable, improving evaluations of MOV failures and necessary corrective action, and trending MOV problems. The staff requested that licensees complete the GL 89-10 program within three refueling outages or five years from the issuance of the generic letter, whichever is later.

The staff issued Supplement 1 to GL 89-10 on June 13, 1990, to provide detailed information on the results of public workshops held to discuss the generic letter.

On August 3, 1990, the staff issued Supplement 2 to GL 89-10 to allow licensees additional time to review and to incorporate the information provided in Supplement 1 into their programs in response to the generic letter.

Tests performed by the Idaho National Engineering Laboratory (INEL) as part of a program by the NRC Office of Nuclear Regulatory Research reinforced concerns regarding the capability of MOVs to perform their design-basis functions. At a public meeting on April 18, 1990, the INEL researchers discussed the results of the NRC-sponsored MOV tests that revealed that more thrust was required to operate the tested valves under high-flow conditions than had been predicted using standard industry calculations. These test results were directly applicable to the safety function of MOVs used for containment isolation in the high pressure coolant injection (HPCI), reactor core isolation cooling (RCIC), and reactor water cleanup (RWCU) systems of boiling water reactor (BWR) plants. As a result of a summary review of the capability of MOVs in those systems and discussions with the BWR Owners' Group, the staff issued Supplement 3 to GL 89-10 on October 25, 1990, which requested BWR licensees to perform a plant specific safety analysis and to evaluate the capability of MOVs used for isolation in the HPCI, RCIC, and RWCU systems, and in isolation condenser lines, as applicable. Also, the staff requested all licensees to consider the results of the MOV tests in their GL 89-10 programs.

On February 12, 1992, the staff issued Supplement 4 to GL 89-10 and stated that, based on a study of core melt probability, BWR licensees need not address inadvertent MOV operation as part of their GL 89-10 programs. Nevertheless, the staff stated its belief that consideration of inadvertent MOV operation was a benefit to safety.

The staff issued Temporary Instruction (TI) 2515/109 (January 14, 1991), "Inspection Requirements for Generic Letter 89-10, Safety-Related Motor-Operated Valve Testing and Surveillance," to provide guidance for regional inspections of the programs being developed by nuclear power plant licensees in response to GL 89-10. Part 1 of TI 2515/109 involves a review of the program being established by the licensee in response to GL 89-10, while Part 2 involves a review of program implementation. The staff intends to develop a comprehensive MOV inspection procedure to encompass guidance from TI 2515/109 and information from other sources.

In January 1991, the staff initiated inspections of GL 89-10 programs and has conducted inspections of those programs at most licensee facilities. The inspections to date have focused on the review of the GL 89-10 programs (Part 1 of TI 2515/109). Some of the more significant findings with respect to GL 89-10 are summarized below.

The NRC inspections have found, for the most part, that licensees are establishing the scope of their GL 89-10 programs consistent with the recommendations of the generic letter.

With respect to the recommendations of GL 89-10 regarding design-basis reviews of MOVs, many licensees are appropriately reviewing plant documentation (such as the Final Safety Analysis Report and Technical Specifications) to determine the worst case conditions for all design-basis scenarios. However, some licensees had failed to review normal, abnormal and emergency procedures to ensure that worst case conditions are identified for various design-basis scenarios. Some licensees had focused on differential pressure and had not adequately addressed other design-basis parameters (such as flow, fluid temperature, ambient temperature, and seismic/dynamic effects). Although differential pressure is the primary design-basis parameter used to predict thrust requirements in the present industry equations, the other design-basis parameters needed to be considered to ensure that the test results demonstrate that the MOV would operate under design-basis conditions.

With respect to the recommendations of GL 89-10 regarding MOV sizing and switch settings, licensees are using various methods to determine the proper size of MOVs and their appropriate switch settings. Some licensees have increased the valve factors assumed in the industry equations (used to predict the thrust required to operate the valves) to reflect industry-wide and plant-specific experience. However, other licensees continue to use old guidance from valve vendors and manufacturers in estimating thrust requirements which may be determined to be inadequate during design-basis tests.

Among the weaknesses found by NRC inspectors at various facilities in the area of MOV sizing and switch settings are (a) lack of justification for assumptions regarding stem friction coefficients and changes in stem friction over the lubrication interval, (b) failure to consider effects that can reduce thrust delivered by the motor operator under high differential pressure and flow conditions from the thrust delivered under no-load conditions, (c) failure to consider ambient temperature effects on motor output and thermal overload sizing, (d) lack of justification for using industry databases in predicting thrust requirements, (e) failure to consider inertia in establishing maximum torque switch settings, (f) reliance on contractor studies of actuator capability without adequate justification, (g) lack of justification for reliance on motor output from generic motor curves, (h) lack of justification for removal of certain conservatisms (such as application factor) from standard industry sizing calculations, and (i) failure to consider torque switch repeatability.

Many licensees are updating their degraded voltage studies to ensure that the worst-case minimum voltage available at the motor has been determined for each MOV. Some licensees did not justify the assumptions for (a) the starting point for the degraded voltage calculations, (b) motor power factor and current used to calculate cable losses, (c) losses caused by resistance of thermal overload devices in the circuit, or (d) effects on MOV stroke time under degraded voltage conditions.

With respect to the recommendations of GL 89-10 regarding MOV testing, many licensees have committed to test MOVs within the scope of their GL 89-10 program under design-basis conditions, where practicable. Licensees who have begun differential pressure and flow testing have found some MOVs to require more thrust to operate than predicted by the standard industry equation with typical valve factors (such as 0.3 for flexible wedge gate valves) assumed in the past. The NRC staff has found weaknesses in licensee procedures for conducting the differential pressure and flow tests, the acceptance criteria for the tests in evaluating the capability of the MOV to perform its safety function under design-basis conditions, and feedback of the test results into the methodology used by the licensee in predicting the thrust requirements for other MOVs. The NRC regulations and plant Technical Specifications establish requirements for licensee actions and reporting when safety-related equipment is determined to be, or has been, unable to perform its safety functions.

For MOVs that cannot be tested under design-basis differential pressure and flow conditions, the NRC inspectors have found some licensees not to be following their commitments to the two-stage approach (discussed in Supplement 1 to GL 89-10) to test those MOVs at the maximum differential pressure and flow achievable. Where the test pressure and flow are near to the design-basis conditions, the licensee may be able to justify extrapolation of the test results so as to demonstrate the capability of the MOV to perform its safety function under design-basis conditions. For MOVs that cannot be tested near to design-basis conditions, the licensee will be able to use the results of the tests at maximum achievable conditions to help confirm valve factor assumptions in its sizing and switch setting methodology and to set the MOV using the best available data. Also, there may be Technical Specification actions and reporting requirements stemming from tests of MOVs at less than full design-basis differential pressure and flow conditions when those tests reveal that the MOVs would not be capable of performing their safety functions under design-basis conditions.

In their initial response to GL 89-10, some licensees did not commit to test MOVs where practicable, but stated that they would attempt to group MOVs to limit the extent of design-basis testing. In light of the preliminary results of design-basis tests at several plants, licensees may find it difficult to group MOVs in such a manner that a small sample of MOV tests can be used to demonstrate that all MOVs in the group are capable of performing their safety functions under design-basis conditions.

With respect to the recommendations of GL 89-10 regarding periodic verification of MOV capability, many licensees have stated that they will attempt to use tests of MOVs with diagnostic equipment under zero differential pressure and flow conditions (static conditions) to demonstrate the adequacy of torque switch settings and the continued capability of MOVs to perform their safety functions under design-basis conditions. None of those licensees, as yet, has provided justification for applying the results of tests conducted under static conditions to demonstrate design-basis capability. Many licensees are improving their methods to demonstrate continued capability of MOVs to perform their safety functions under design-basis conditions following maintenance.

With respect to the recommendations of GL 89-10 regarding MOV failures, corrective action and trending, the NRC inspectors have found some licensees to be weak in responding to MOV failures and deficiencies. Some licensees have not been thorough in their root cause analysis of MOV problems. Most licensees are attempting to improve the trending of MOV problems.

In GL 89-10, the NRC staff requested that licensees complete all design-basis reviews, analyses, verifications, tests, and inspections that were initiated in order to satisfy the generic letter recommended actions by June 28, 1994, or three refueling outages after December 28, 1989, whichever is later. During the inspections, the staff found some licensees not to have made adequate progress toward resolving the MOV issue for their facilities within the recommended schedule of GL 89-10. The staff has accepted limited extensions of the GL 89-10 schedule for particular licensees where justification has been provided.

With respect to Supplement 3 to GL 89-10, BWR licensees have completed their determinations of whether any immediate concerns exist with respect to the capability of MOVs within the scope of Supplement 3 to GL 89-10 to perform their safety functions under design-basis conditions. Many BWR licensees have determined the need to adjust or modify those MOVs to provide assurance of their capability.

In addition to the recommendations of GL 89-10, the NRC staff has inspected other licensee activities involving the performance of MOVs. For example, the staff's program of Diagnostic Evaluation Team inspections has found lack of engineering bases for torque switch settings, inadequate procedures for setting torque switches, lack of control of torque switch settings, inadequate procedures for MOV disassembly, and inadequate lubrication programs. During the inspections of the programs being developed in response to GL 89-10, the NRC staff has evaluated other MOV activities by nuclear power plant licensees. Significant findings from the inspections in those MOV areas are discussed below:

1. Some licensees have not ensured adequate management oversight and direction for the MOV program. The safety significance of the MOV program and the extensive resources necessary to develop and implement the program make it imperative that licensee management closely monitor its staff's activities.

2. Licensees have found (a) incorrect spring packs installed in MOVs, (b) incorrect MOV data on the motor or actuator nameplates and in the procurement documents from the vendor, and (c) spring packs with different performance characteristics from different manufacturers, but with the same part number.

3. Licensees are improving their documentation of current and required MOV switch settings, but some weaknesses remain.

4. Some licensees have raised torque switch settings for MOVs above the manufacturer's maximum specified value without an adequate safety analysis in accordance with the requirements of 10 CFR 50.59.

5. Some licensees rely on minor preventive maintenance or diagnostic test results in determining the need for MOV refurbishment despite the weakness of such activities in identifying degradation (such as caused by stem nut wear) in the capability of an MOV to perform its safety function under design-basis conditions. Further, some licensees are not refurbishing MOVs to establish a known MOV condition before performing an initial baseline test with diagnostic equipment.

6. Some licensees do not have maintenance procedures that (a) provide adequate torque switch balancing guidance, (b) outline specific guidance on identifying degradations or filling of the limit switch gearcase, or (c) reflect important vendor information, such as the operability effects of spring pack relaxation.

7. Some licensees appear to have strong training programs for MOV maintenance and diagnostic testing. Many licensees are significantly improving their training programs. However, the NRC inspectors have found weaknesses in the establishment of refresher training, the training of contractor personnel, and the verification of adequate knowledge of instructors provided by outside organizations.

8. Licensees are improving their processing and control of operating experience and vendor information (such as Limitorque Maintenance Updates) although some weaknesses remain.

9. During plant walkdowns, the NRC inspectors have found some licensees to have strong housekeeping programs for MOVs. At other facilities, the NRC inspectors have found (a) grease leaking from actuators, (b) dirty valve stems that do not support the licensee's assumptions for stem friction coefficients or lubrication intervals (particularly for MOVs in high temperature areas), and (c) out-of-date torque switch calibration stickers.

The NRC staff will continue to perform inspections of licensee activities involving MOVs. The NRC staff believes that the number of MOV problems and operating events caused by those problems will decrease as licensees implement their programs in response to GL 89-10.

Rulemaking Efforts on Codes and Standards

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ABSTRACT

Section 50.55a of the NRC regulations provides a mechanism for incorporating national codes and standards into the regulatory process. It incorporates by reference ASME Boiler & Pressure Vessel Code (ASME B&PV Code) Section III rules for construction and Section XI rules for inservice inspection and inservice testing. The regulation is periodically amended to update these references.

The rulemaking process, as applied to Section 50.55a amendments, is overviewed to familiarize users with associated internal activities of the NRC staff and the manner in which public comments are integrated into the process. The four ongoing rulemaking actions that would individually amend Section 50.55a are summarized. Two of the actions would directly impact requirements for inservice testing.

Benefits accrued with NRC endorsement of the ASME B&PV Code, and possible future endorsement of the ASME Operations and Maintenance Code (ASME OM Code), are identified. Emphasis is placed on the need for code writing committees to be especially sensitive to user feedback on code rules incorporated into the regulatory process to ensure that the rules are complete, technically accurate, clear, practical, and enforceable.

The "Codes and Standards" rule, which is Section 50.55a of the NRC regulations, is the primary mechanism by which NRC incorporates ASME consensus standards into the regulatory process. Specifically, Section 50.55a incorporates by reference Sections III and XI of the ASME Boiler and Pressure Vessel Code (B&PV Code). These sections of the ASME B&PV Code reference other standards, such as OM Parts 1, 6, and 10, that in turn become regulatory requirements.

Section 50.55a is amended every two to three years to include later edition and addenda of the ASME B&PV Code. Occasionally, the section is amended to modify its scope. At present, four amendments to Section 50.55a are in various stages of development. I will provide you with the status of these rulemakings, but I believe it would be useful to first provide you with an overview of the major steps in the rulemaking process.

The rulemaking process starts with the preparation of a task initiation package, which is submitted for approval to the NRC Executive Director for Operations (EDO). The package states the purpose of and need for the rulemaking, and briefly addresses various aspects of value/impact. The rulemaking is initiated if approved by the EDO.

The proposed rule and supporting draft regulatory analysis are prepared and submitted to cognizant NRC technical, legal, and administrative offices for comment. Revisions are made in accordance with staff comments. The rulemaking package is then resubmitted to the cognizant offices for approval.

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Subsequently, the rulemaking package is reviewed by the Advisory Committee on Reactor Safeguards (ACRS) and the Committee to Review Generic Requirements (CRGR). These committees advise the EDO on whether they believe the proposed rule should be issued. The CRGR has the unique responsibility to review backfit considerations and the relationship of the rule to other regulatory actions. If the committee recommendations are positive, and the EDO agrees, the proposed rule would be signed by the EDO to be published in the Federal Register for a 75-day public comment period. The Commission has delegated authority to the EDO to develop and promulgate this and certain other types of rules.

Public comments are collated, reviewed, and resolved. The proposed rule and draft regulatory analysis are modified, as necessary, to resolve public comments. The resolution of these comments is documented in the final rulemaking package. Important points on the comment resolution are specifically addressed in the supplementary information to the final rule.

The process of office review, comment, concurrence, and ACRS and CRGR review are repeated for the final rule. If the ACRS and CRGR recommend that the final rule be issued and the EDO concurs, the EDO will sign the rule for publication in the Federal Register. Although final rules for Section 50.55A are effective immediately, Section 50.55A requires that new inservice testing (IST) programs, based on referenced edition of the code, be implemented at the time of the 10-year update.

Following is a summary of the four ongoing Section 50.55a rulemaking actions. The most advanced of these amendments would update references to edition and addenda of the ASME B&PV Code through the 1989 Edition, incorporate OM -6 and -10 through a Section XI reference, separate in the regulation the requirements for inservice inspection and inservice testing, and expedite a Section XI specified reactor vessel examination. This amendment has been published for public comment, and the proposed rule has been revised based upon those comments. The final rule has been approved by the ACRS and has been submitted to the CRGR for review.

The second amendment, which was initiated a few months ago, has special interest for the Operation and Maintenance (O&M) Committee. The proposed rule would update the reference to the ASME B&PV Code to the 1992 Edition, and would incorporate by reference into the regulations, the ASME O&M Code with possible modifications. The proposed rule is scheduled to be published in the Federal Register mid calendar year 1993.

The third amendment would incorporate by reference Section XI, Subsectious IWE and IWL, which, respectively, provide rules for inservice inspection of metal and concrete containments.

The fourth amendment is in the initiation stage. The objective of this amendment, as presently defined in the task initiation package, is to (a) expand the scope of the IST program, (b) require periodic verification that each component in the program can accomplish its safety function at design basis conditions, and (c) require trending of data to identify degraded components. This potential rulemaking was the subject of a letter from James E. Richardson, Director of the Division of Engineering, Nuclear Reactor Reaction, (NRR) to Forrest Rhodes, Chairman, O&M Main Committee. The letter requested the ASME O&M Committee to consider addressing the above three issues in its codes and standards. A meeting was held in May 1992 between the NRC staff and O&M Committee members to discuss details of the letter. The response by the O&M Committee to the Richardson letter may effect the final scope of this potential rulemaking.

As you can see, the NRC staff has an extensive program to incorporate ASME codes and standards into the regulatory process. Incorporating the ASME O&M Code into the regulations is a significant action for the NRC. The determination must be made that the ASME O&M Code is acceptable as written, or with modifications, and that the infrastructure within the O&M Committee is in place for adequately maintaining the

O&M Code. This includes responding to user inquiries and developing alternative rules to the O&M Code in the form of code cases. It is reasonable to expect that the NRC would develop a regulatory guide for O&M code cases that would specify their acceptability for use.

Should the O&M Code be incorporated by reference into the NRC regulations, it would become, like the ASME B&PV Code, an integral part of the regulatory process. This would result in an expanded association between the NRC and ASME that would benefit both.

The NRC would benefit because

- The O&M Code would impart authority to the technical positions, which would be more readily accepted because of the credibility that ASME standards have on a national and international level
- NRC resources would be multiplied through the participation of industry, academia, and the public
- There would be a higher probability that better, more practical standards would be developed
- Standards not endorsed in their totality would provide a good basis for incorporation with specific modifications.

The ASME would benefit because

- The NRC staff is committed to the codes and standards program
- Participation in O&M activities by others in the nuclear industry would be encouraged
- Proper consideration of safety would be ensured
- Issues of public concern would be identified and resolved
- There would be increased probability that the standard would be used
- NRC endorsement would enhance the stature of the O&M Committee.

If the O&M Code is integrated into the regulatory process, it would be especially necessary to maintain it on a continuing basis. This maintenance should be proactive. Revisions and additions to the O&M Code should be *complete, technically accurate, clear, practical, and enforceable*. The O&M Committee must be sensitive to the need to change existing rules based upon results from IST programs and operational experience. This may seem obvious, but unless there is a mechanism within the O&M Committee to obtain, evaluate, and implement user input, it will not happen.

Many of you are aware that the ASME funded the Pressure Vessel Research Council (PVRC) to perform an independent review of the nuclear portions of the ASME B&PV Code to simplify and eliminate confusing portions of that code. The project started slowly, but eventually interest picked up and over 100 recommendations were provided to the ASME for consideration. These recommendations were reviewed by the ASME Board on Nuclear Codes and Standards and transmitted to Section III, Section XI, and the O&M Committee for evaluation and possible implementation. While this was a useful project, it should have been unnecessary. The individual main committees should have a mechanism for continuously soliciting and evaluating user feedback. I encourage the O&M committee to consider developing and implementing such a procedure.

The NRC staff looks forward to a mutually beneficial relationship with the ASME O&M committee in the interest of protecting the public health and safety, which after all *is all of us*.

Equipment Nonconformance and Degradation: Promptly Determining Operability and Establishing Corrective Action Plans

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ABSTRACT

Nine principles for dealing with degraded and nonconforming equipment are presented and some examples are discussed. The distinction between equipment operability (i.e., capability to perform the safety function) and equipment qualification (conformance to all aspects of the current licensing basis, including codes and standards, design criteria, and commitments) is discussed. The concept of finding reasonable assurance of safety for continued plant operation for equipment not covered by technical specifications is also presented. Degraded or nonconforming equipment must be evaluated for its safety impact and for operability. In all cases, degraded or nonconforming conditions must ultimately be resolved, either through prompt corrective action or through some process of showing that the changed state of the plant is acceptable for continued operation, based on 10 CFR 50.59.

INTRODUCTION

In the fall of 1991, the NRC published Generic Letter (GL) 91-18 to aid NRC inspectors in the evaluation of licensee actions to resolve degraded and nonconforming conditions of safety equipment (including pumps and valves) at nuclear power plants. Because of the thousands of components involved, licensees come upon degraded or nonconforming equipment frequently. How a licensee responds to such conditions and the decision-making process that the licensee uses are of interest to every NRC inspector. Therefore, the NRC published GL 91-18 to promulgate nine fundamental principles to keep in mind when dealing with degraded or nonconforming conditions:

1. Focus on safety.
2. Deal with operability and restoration of qualification separately.
3. Operability is the capability to perform the safety function(s).
4. Qualification is conformance to all aspects of the design basis, including codes and standards, design criteria, regulations, and final safety analysis report commitments.
5. Determining operability and plant safety is a continuous decision-making process.
6. Timeliness of operability determinations should be commensurate with the safety significance of the issue.
7. Timeliness of corrective action (i.e., 10 CFR 50, Appendix B, Criteria XVI Requirement for "prompt" corrective action) should be commensurate with the safety significance of the corrective action.

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8. Justifications for Continued Operation (JCOs) are the licensee's technical basis for operating in an otherwise prohibited manner (e.g., outside technical specification limits, regulations, or license conditions).

9. License amendments and enforcement discretion are NRC's response to a licensee's request to operate in the otherwise prohibited manner.

SOME EXAMPLES

To illustrate the application of the nine principles, it is useful to examine some examples. As we do so, we will also examine safety assessment in general; operability determinations; and how 10 CFR 50.59 safety evaluations, corrective action plans, and license amendments fit into the process.

Some examples of degraded and nonconforming equipment include equipment that doesn't meet a code or a standard, doesn't conform to the as-built condition, or doesn't respond normally or as expected. Other examples include equipment qualification; whether files are available, and whether new test data indicate that there could be a problem with equipment; or a previously undiscovered design deviation. Physical evidence, such as a fouled heat exchanger, may also indicate a degraded condition.

Information on the condition of plant equipment may come from licensee analyses, component testing in situ, bench or prototype testing of components, or from a vendor or another plant. It is the licensee's responsibility to integrate all sorts of information concerning equipment degradation and nonconformance and then take the appropriate actions.

Assuming that a degraded or nonconforming condition exists, then the first action is to focus on safety. An initial assessment is required to determine what safety impact the degraded or nonconforming condition will have, and whether continued plant operation is acceptable. In effect, this first test is whether there is any immediate threat to public health and safety. If a degraded condition arises that is a serious problem, the plant needs to be put in a safe condition regardless of any other commitments that might exist in technical specifications (TS) or any other place.

The next action is a prompt operability determination. In these determinations, it must be stressed that operability really means the capability to perform the safety function. Operability does not mean that equipment conforms to a code or standard. It does not mean that it looks like its as-built drawing. The key is the equipment's functional safety capability.

It is not unusual to have equipment that perhaps does not meet a code or standard or might have an equipment qualification problem. But there still is a legitimate judgement that can be made about its ability to perform its safety function.

Because subsequent actions depend on the outcome of the operability determination, it needs to be made promptly. We recognize that they are not being made with perhaps all the information that will ultimately be available. So when a prompt determination of equipment operability is made, it ought to be made based upon whatever analysis is available; whatever tests or partial test results are available; whatever operating experience can be brought to bear on the issue; and on engineering judgement. An initial prompt operability determination may be revised as additional information is gathered.

We now cover four examples where equipment is either covered by TS or not, and the determination is made whether the equipment is operable or not.

Example 1: Equipment in TS and Operable

The first case is one in which equipment specified in the TS is determined to be operable. This equipment, which may in fact be in some degraded state for which corrective actions may be required later, is still capable of performing the safety function(s). Therefore, by plant license and by TS, continued operation of the plant is authorized.

However, corrective action to resolve the degraded condition or lack of conformance still must be carried out, although it can be carried out while plant operation continues.

There are two ways in which corrective actions can be done. The action can be taken promptly. In this case, there is no clear time period that corresponds to "promptly." It depends on the safety significance of the situation.

Alternatively, the licensee may propose to live with the degraded or nonconforming condition without corrective action. In this case, the NRC views this as a change to the facility that must be evaluated under 10 CFR 50.59, which may or may not call for a license amendment.

Example 2: Equipment in TS and Inoperable

For equipment covered by TS and determined to be inoperable the normal course of action is to follow the technical specifications. In many instances, this would put the plant in an action statement. It could require a relatively prompt shutdown. As an alternative, the safety circumstances might dictate that the safer thing to do would be to operate the plant (perhaps in some changed s[illegible] such as at a reduced power, or perhaps with some other compensatory measures taken). In this case, an emergency license amendment or other regulatory action ought to be proposed by the licensee.

In this case also, the nonconforming or degraded state of the equipment must ultimately be resolved through corrective action or 10 CFR 50.59 evaluation.

Example 3: Equipment not in TS and Operable

For equipment that is not covered in TS there still can be safety implications because the equipment is inoperable. Therefore, a judgement about the operability of the equipment, based upon whatever information exists with respect to not conforming with codes and standards still ought to be made. If the equipment is judged to be operable (and in effect, we are using the same standards we would for equipment covered by TS) based upon operating experience and engineering judgement, then continued operation is acceptable.

In this case also, the nonconforming or degraded state of the equipment must ultimately be resolved through corrective action or 10 CFR 50.59 evaluation.

Example 4: Equipment not in TS and Inoperable

If equipment is not specified in TS and is determined to be inoperable, there is still a possibility that it is really not needed for plant operation to continue in the short-term. A judgement must be made about whether there is reasonable assurance of safety. If that can be done (and we will discuss what elements go into that determination), then continued operation of the plant would be appropriate. If not, in fact the plant ought to be placed in a safe condition. That might mean taking some compensatory measures or it might mean shutting down the plant.

As with other cases, since there is some commitment to a code, standard, or some other commitment that is not being met as a result of finding a degraded or nonconforming condition, that condition again needs to be corrected either by bringing the plant back into conformance with the original commitments through prompt corrective action, or through use of the 10 CFR 50.59 process (in effect justifying the altered state of the plant).

Reasonable Assurance of Safety

In cases where there is equipment not covered by TS and there is a need to make a judgement about the reasonable assurance of safety for continued operation, the considerations that would go into such a judgement involve the availability of redundant or backup equipment and other compensatory measures, such as operator action or stationing an operator by the equipment. Consideration should also be given to the safety function of the equipment and the events that are protected against by the equipment. The amount of conservatism in the analysis and the available margins are also factored into the judgement. Finally, the probability of needing the safety function may also be considered which is a case where judgements about the probability of remote events would play a role in the decision-making process.

SUMMARY

Degraded or nonconforming equipment must be evaluated for its safety impact and for operability. Operability is capability to perform the safety function(s) and is, in effect, a different decision than whether equipment is in a nonconforming or degraded state. In all cases, degraded or nonconforming conditions must ultimately be resolved, either through prompt corrective action or through some process of showing that the changed state of the plant is acceptable for continued operation, based upon 10 CFR 50.59.

Solenoid-Operated Valves and Related Equipment —A Status Report

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ABSTRACT

The paper presents a status report of recently performed, ongoing, and planned activities associated with solenoid-operated valves and related equipment. It includes discussions of the activities being undertaken in response to the concerns raised in NUREG-1275, Volume 6. "Operating Experience Feedback Report—Solenoid-Operated Valve Problems," and Generic Letter 91-15, "Operating Experience Feedback Report, Solenoid Operated Valve Problems at U.S. Reactors." It discusses the initiatives being taken by professional societies such as The American Society of Mechanical Engineers (ASME) and the Institute of Electrical and Electronic Engineers (IEEE); and actions being taken by individual licensees, major SOV manufacturers, and industry groups.

INTRODUCTION

In 1988, because of concerns arising from safety-significant events involving the simultaneous failures of multiple solenoid-operated valves (SOVs), the Nuclear Regulatory Commission (NRC) Office for Analysis and Evaluation of Operational Data (AEOD) began a systematic analysis of operational events involving SOVs. AEOD performed a case study which concluded that nuclear plants were susceptible to common-mode SOV failures as a result of deficiencies in SOV design, application, maintenance, surveillance testing, and manufacturing. AEOD's case study, "Solenoid-Operated Valve Problems in Light Water Reactors," was issued as a preliminary case study for peer review in June 1990. After resolution of peer review comments, it was issued as AEOD Case Study C90-01 in December 1990 and it was issued as NUREG-1275, Volume 6, in February 1991 (Ornstein). In September 1991, the NRC issued Generic Letter (GL) 91-15 (NRC, 1991) forwarding the study to licensees.

DISCUSSION

GL 91-15 described deficiencies in design and application, manufacture, maintenance, surveillance testing, and feedback of failure data on SOVs. It concluded that problems with SOVs need additional industry attention. GL 91-15 alerted licensees to the NRC's concerns for the susceptibility of SOVs to common-mode failures, the potential reductions in safety margin, and the accompanying increase in risk. The generic letter noted the vulnerability of safety-related equipment to common-mode failure or degradation of SOVs. It noted that NRC is concerned about the reliability of SOVs used in safety applications, and it stated that the NRC expected licensees to review the information presented in the case study and consider appropriate actions to avoid

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SOV problems similar to those presented in NUREG-1275, Volume 6 (Ornstein, 1991).

Individual Utility Activities

The author has had discussions with many licensees and has found that some of them have initiated aggressive programs to review SOVs that are important to safety at their plants, and to verify the subject SOVs' design, application, life, and maintenance programs. Many utilities have implemented extensive SOV root failure analyses.

Table 1 provides a brief summary of the activities presently conducted at 25 plants. In several cases, the utilities initiated their activities upon when they received NUREG-1275, Volume 6, before the generic letter was issued. Utilities that the author has not had direct contact with on the issue of response to GL 91-15 have been omitted from the table.

Professional Societies

ASME. The ASME Nuclear Operation and Maintenance (O&M) Committee has selected a task force to review the need for an ASME O&M standard on performance testing of SOVs. The task force was scheduled to provide a report to the O&M Committee in June 1992. However, (as of April 1992) the schedule appears to be slipping.

IEEE. The IEEE Power Engineer Society, Nuclear Power Engineering Committee Working Group 3.3 issued a good practices guide for SOVs in 1988 (IEEE, 1988). That guide did not recognize the susceptibility for and the impact of common-mode SOV failures upon plant safety margins. The guide was based on the assumption that the observed SOV events were isolated, random, single failures. Discussions between AEOD and the IEEE Working Group 3.3 members after the guide was published convinced the working group members that the guide should be revised to address the issue of common-mode SOV failures and degradations. IEEE Working Group 3.3 is scheduled to issue a revision to its maintenance good practices guide early in 1993.

EPRI/NMAC. In February 1991, The Electric Power Research Institute and Nuclear Maintenance Assistance Center (EPRI/NMAC) initiated a SOV Technical Advisory Group to assist a contractor formulate a SOV maintenance and application guide. The guide's intent was to inform plant operations and maintenance people about the importance and sensitivity of SOVs. The guide provides important information about principles of operation, maintenance, testing, and root cause failure analysis. The report was issued in draft in December 1991 and was scheduled to be issued in final form in April 1992.

NMAC conducted two SOV workshops (one in December 1991 and one in May 1992). The workshops included lectures on SOV operating experience, lectures on the SOV maintenance and application guide, and hands on demonstrations by several major SOV manufacturers.

INPO. NUREG-1275, Volume 6 (Ornstein, 1988) included a recommendation for systematic feedback of SOV failure experience to SOV vendors. Such a program would alert SOV manufacturers to failures of their equipment by making the failure records of their specific SOVs available to the manufacturers. The Institute of Nuclear Power Operations (INPO) has agreed to provide SOV manufacturers with failure data reports on their SOVs in response specific manufacturers' requests. No periodic updates will be provided.

Major SOV Manufacturers

Four major SOV manufacturers (Automatic Switch Company [ASCO], Automatic Valve Company [AVC], Target Rock, and Valcor) have all noted a heightened awareness of the concerns for common-mode SOV failures. They have been made more aware of 10 CFR 21 requirements for SOVs, especially in the area of user notification of newly discovered SOV deficiencies. These major SOV manufacturers have started programs to improve their SOV performance. Recently, the major SOV manufacturers have issued field service bulletins/notifications to the licensees to

Table 1. Licensee activities regarding SOVs.

Plant	Implement design verification program	Review maintenance/ surveillance test practices	Aggressive root cause failure analysis	Comments	(P) Primary contact (T) Team leader
Beaver Valley 1 & 2	Limited number of SOVs being addressed (mostly harsh environment) (EQ, SOVs)—Program being expanded	Change to the use of staggering of SOVs in turbogenerator EHC system. Expanding surveillance testing to determine operability and condition of individual valves	Has been implemented. Utilizing AOV diagnostic equipment to measure SOV operating characteristics	Three man effort addressing NUREG-1275, Volume 6, concerns	(T) Bruce Kozar (P) Jack Kweder
Brunswick 1 & 2, Robinson 2, Shearon Harris	Have implemented a corporate-wide program (7 person team) to address recommendations of NUREG-1275, Volume 6. Most of the recommendations have already been implemented at Brunswick 1 & 2. Staggered replacement of SOVs has been implemented on redundant SOVs which have had repetitive failures. Looking into diversity. Revised maintenance practices to eliminate possible introduction of contaminants into SOVs during installation or shop testing. Revising SOV changeout times to reflect actual operating experience (vs. more optimistic EQ calculations)		Participating with other utilities and ASCO on sticking of ASCO 206-832 SOVs		(T) Glenn Re[illegible]aman (P) H. Allen Walker
Crystal River 3	Yes—focusing primarily on AOVs (and their associated SOVs)	Yes—aggressive preventative maintenance program especially with regard to elastomers		Contractor report raised concern for 115 non-EQ AOVs (with their associated SOVs). Adding 115 AOVs (and associated SOVs to periodic preventative maintenance program periodic elastomer replacement program being adopted). Active participation in NMAC SOV maintenance program	(P) Larry Gastine
Davis-Besse	Yes	Active participant in NMAC SOV maintenance program and AOV users group. Developed AOV diagnostic tool which is being refined and marketed by CE/ABB			(P) John Hayes

Table 1. (continued).

Plant	Implement design verification program	Review maintenance surveillance test practices	Aggressive root cause failure analysis	Comments	(P) Primary contact (T) Team leader
Duane Arnold	Yes—initiated program several years ago as part of air system concerns. Addressing 1840 SOVs, 800 of which are safety-related. Replacing many commercial SOVs with IE qualified ones.			Active participation in NMAC SOV maintenance program	(T) Bill Simmons
Palo Verde 1, 2, & 3	Yes—committed to attacking SOV problems with a rigorous program which will address all recommendations in NUREG-1275, Volume 6.			Active participant in NMAC SOV maintenance program	(P) Bob Whiting (T) Steve Coppock
Peach Bottom 2 & 3 (lessons learned applicable to Limerick 1 & 2	Yes—have addressed over 500 "important" SOVs since September 1991. (Total number of SOVs at Peach Bottom estimated at 4000 per unit)	Yes—planning to implement staggered testing and preventative maintenance. Presently looking into diversity.	Aggressive root cause failure analyses being performed for SOVs at Peach Bottom and Limerick Station.	Initiated high frequency stroking (once or twice a week) as an interim measure prior to major SOV replacements	(P) Bill [illegible]sky (T) Allen Rausch
			Participating with other utilities and ASCO on sticking of ASCO 206-832 type SOV		
St. Lucie 1 & 2 Turkey Point 3 & 4	Yes—implemented aggressive corporate-wide program to address SOVs. Verified MOPD in detail at all four plants. Preparing an I&C "program description" to incorporate the concepts of GL 91-15, including consideration of staggering.		Perform aggressive root cause failure analysis. Have implemented detailed failure tracking and trending programs	Active participant in NMAC SOV maintenance program and AOV user group	(P) Joe Price
Vermont Yankee	Developing a plan of attack for SOVs in response to GL 91-15. Plan expected to be completed by July 1992.				(P) Mike Mettel
Waterford 3 ANO 1 & 2 Grand Gulf	Yes—corporate-wide program being evaluated with pilot program at Waterford 3. Waterford 3 report due in May 1992 for presentation to Senior Management. Report will address each of the recommendations of NUREG-1275, Volume 6, with implementation expected quickly. As of April 1992, all items recommended in NUREG-1275, Volume 6 except staggering and diversity are being recommended by the 8 person SOV task force. Task force report will have a recommendation for installing temperature sensors near many important SOVs which are located in mild (non-harsh) environments.				(P) Bob Murillof

alert licensees to the recently discovered deficiencies.

Hiller and General Electric (GE) have initiated a program to replace ASCO NP8323 SOVs on main steam isolation valve (MSIV) air packs. Thermal aging and performance tests were performed on 21 SOVs to be used on the Hiller MSIV air packs. Testing was performed on new design SOVs supplied by ASCO, AVC, and Valcor. Included were ASCO's new NS series valves, ASCO's voltage reduction model, and new model AVC and Valcor valves. The Target Rock valves were not tested because of their excessive size and weight. Of the SOVs tested, ASCO's new NS series single coil valves, NS8320, were the star performers. They operated flawlessly throughout. The AVC valves sustained several coil failures. The Valcor valves had "substantial leakage problems." ASCO's old NP series valves had shifting problems. ASCO's voltage reduction modules did not perform satisfactorily. The Hiller test report (Ralph A. Hiller Company, 1992) indicated their reluctance use the ASCO NS8320 SOVs on their MSIV air packs without additional seismic analysis. Furthermore, Hiller/GE are continuing to do engineering work to support the use of other Valcor SOVs with different seal material with the expectation that they would perform better than the Valcor SOVs that did not fare well during the aforementioned tests.

Whatever designs are finally adopted by individual BWR owners, it would be prudent to consider SOV diversity in the event that the new SOVs have design deficiencies that have not yet surfaced,[b] the use of alternate designs on inboard and outboard MSIVs would prevent common-mode failures from compromising any one main steam line's capability to isolate on demand.

CONCLUSIONS

Subsequent to AEOD's evaluation of SOV operating experience, there has been a heightened awareness of the importance for minimizing the likelihood for common-mode SOV failures and degradations. Activities are continuing on many fronts and it is hoped (and expected) that organizations other than those contacted and discussed above will take similar actions to improve SOV performance.

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Institute of Electrical and Electronics Engineers, 1988, *IEEE Maintenance Good Practices for Nuclear Power Plant Electrical Equipment; Work-in-Progress Report, Evaluation of Maintenance and Related Practices for Solenoid-Operated Valves in Nuclear Power Generating Stations*, IEEE Power Engineering Society, Nuclear Power Engineering Committee, Subcommittee 3, Working Group 3.3.

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b. For example, on November 8, 1991, GE issued a 10 CFR 21 report (GE, 1991) noting that a major error had been made in their equipment qualification calculations for ASCO dual coil NP8323 SOVs used in MSIV air packs at about half the U.S. BWRs. The report noted that GE had failed to properly consider the self-heating aspect of those valves. The use of an incorrect temperature resulted in a calculated qualified life of 5 years. GE's report noted that accounting for self-heating (self-heating addressed in Information Notice 89-66 [NRC, 1989] and an October 27, 1989 ASCO field notification report [ASCO, 1989] results in a reduction of the qualification life from 5 years to only 18 months).

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The 1992 Symposium on Pump and Valve Testing, jointly sponsored by the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the Nuclear Regulatory Commission, provides a forum for the discussion of current programs and methods for inservice testing and motor-operated valve testing at nuclear power plants.The symposium also provides an opportunity to discuss the need to improve that testing in order to help ensure the reliable performance of pumps and valves. The participation of industry representatives, regulators, and consultants results in the discussion of a broad spectrum of ideas and perspectives regarding the improvement of inservice testing of pumps and valves at nuclear power plants.

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