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

Held at Hyatt Regency Hotel
Washington, DC
July 18–21, 1994

Session 1A – Session 2C

Sponsored by
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ABSTRACT

The 1994 Symposium on Valve and Pump Testing, jointly sponsored by the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the 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 William Russell, Ashok Thadani, Guy Arlotto, James Pelletier, Brian Sheron, and Steven Weinman for the opening presentations; to the foreign speakers and attendees; and to the symposium banquet guest speaker P. J. O'Rourke. Appreciation is expressed to the symposium steering committee: John Allen, Patricia Campbell, Gerry Eisenberg, Joe Philipps, and Edmund Sullivan with a special acknowledgment of the efforts of Gerry Eisenberg in coordinating the logistics of the symposium. Appreciation is also expressed for the efforts of the session chairs in reviewing the papers: Don Cavi, Kevin DeWall, Gerald Dolney, Chris Hansen, John Hosler, Thomas Parrent, Wes Rowley, Dr. Nabil Schauki, Thomas Scarbrough, and John Zudans. 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 Third NRC/ASME Symposium on Valve and Pump 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.

CONTENTS

VOLUME 1

Opening Addresses

Session 1A: MOV General Session
Session Chair: Kevin DeWall, Idaho National Engineering Laboratory

Session 1B: Inservice Testing General Session
Session Chair: John Zudans, FP&L Chair, Florida Power & Light Co.

VOLUME 2

Session 4B: Innovative Valve Testing
Session Chair: Thomas Parrent, Illinois Power

Opening Addresses

John Allen
Tennessee Valley Authority

Edmund Sullivan
United States Nuclear Regulatory Commission

Vigilance and Reason—The Keys to Continued Credibility[a]

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Office of Nuclear Material Safety & Safeguards
U.S. Nuclear Regulatory Commission
Vice President, American Society of Mechanical Engineers
Chairman, Board of Nuclear Codes and Standards

INTRODUCTION

I have entitled my speech "Vigilance and Reason—The Keys to Continued Credibility." A partitioning of the words gives insight into my view of where we are, where we may be going, and that we have control of our fate. "Continued" indicates that I believe, at present, the American Society of Mechanical Engineers (ASME) Codes and Standards process has credibility. "Keys" connotes that we are at a crossroads and something must be done to stay on track. And "Vigilance and Reason" suggest that we can achieve our goal of continued credibility through exercising vigilance and reason. We cannot rest on our laurels. We must take conscious, positive actions to strengthen the credibility of our products; otherwise we will backslide. It is up to us.

THE CONSENSUS PROCESS

Credibility is absolutely essential to the success of the ASME volunteer nuclear standards program. Without it, ASME would have no stature in this arena, the codes and standards would not be used, and the benefits to be gained from the process would not be accrued. We must ensure that erosion of credibility does not occur. The foundation upon which credibility is established in the voluntary standards program is the consensus process.

In simple terms, the consensus process establishes the rules of fair play that cover writing, reviewing, voting, and publishing codes and standards. The consensus process ensures that, at all levels of codes and standards development, all sides of an issue are heard in good faith, negative ballots are considered and fairly resolved, and issues are moved forward only when there is reasonable agreement. An item should be delayed, if necessary, to ensure proper application of the consensus process. More times than not, however, proper scheduling of presentations and a willingness to reason together can avoid any delays. A negative ballot that challenges the consensus process can cause a far longer delay than implementing the process appropriately in the first place. Due process is applicable within the ASME and American National Standards Institute (ANSI) voluntary standards process to give added assurance of an unbiased hearing for application of such a process to rise to such levels.

I would like to make a few points regarding the consensus process:

1. Having a process is not enough—it must be implemented in spirit as well as in fact. It starts by each of you being your own judge and jury. Upon reflection, your conscience should tell you if the planned actions are within the spirit of the consensus process and conflict of interest principles.

2. Evaluating issues with an open mind, listening to others, and being fair are difficult

a. This paper was prepared by an employee of the U.S. Nuclear Regulatory Commission. It presents information that does not represent a current staff position. The USNRC has neither approved nor disapproved its technical content.

given our bias, but we must fight within ourselves to achieve these goals.

3. The fact that you have a majority vote for a standard or even have an overwhelming vote is not enough in itself to satisfy the consensus process. Proper timing for evaluation of data, proper fora for presentation of material by those supporting the code or standard, proper fora for presentation of all other views, fair resolution of comments and negatives, and sufficient documentation, particularly for controversial issues, are all necessary for satisfying the consensus process.

4. When in doubt, particularly if the process is challenged, err on the conservative side to avoid the appearance of not adhering.

5. Get challenges regarding the process, particularly those that may smack of conflict of interest, exposed, discussed, and resolved early.

6. Get resolution within the ASME family and head off having the issues raised outside that may unnecessarily question ASME credibility.

7. Due process is applied both with the ASME and outside.

THE OPERATIONS AND MAINTENANCE COMMITTEE

Let me turn to the Operations and Maintenance effort. As a result of the Pressure Vessel Research Committee's (PVRC) independent review of ASME nuclear codes and standards, the responsibility for developing and maintaining the requirements for inservice testing (IST) of pumps, valves, and snubbers was moved from Section XI to the Operations and Maintenance (O&M) Committee. This organizational change has resulted in the development and publication of the ASME O&M Code. I believe the change, which was based largely on user input, has resulted in more credible IST rules because the expertise is now in a more focused committee.

Following long established ASME efforts, Section III and XI were initiated to ensure pressure boundary integrity of nuclear components. Now that we have moved the IST rules out of Section XI, the scope (i.e., pressure boundary) is more in line with the original mission. While pressure boundary integrity is certainly important, I believe given the current configuration, there is a potential for a bigger payoff by focusing on component operability. Turning to the subject of this meeting, there are elements that should be satisfied to ensure the operability of pumps and valves in service:

1. Integrity of the pressure boundary (valve body, pump casing) to ensure the component stays together

2. Qualification of the pumps and valves to ensure they can operate over the range of conditions required and in the environment they will experience during service life

3. Installation has been accomplished properly

4. Proof testing after installation

5. Inservice inspection and testing to ensure no degradation of operability below acceptable levels

6. Proper preventive maintenance or corrective maintenance

7. Proper training of operating staff and oversight of actions

8. Aggressive quality assurance program.

Some of these elements are within the scope of Sections III and XI, some are for qualification of mechanical equipment, some do not lend themselves to solutions by ASME codes and standards, and some are for O&M. O&M has undertaken inservice testing.

Several years ago, the Board of Nuclear Codes and Standards (BNCS) directed the O&M Committee not to expend efforts on maintenance codes and standards because there was

uncertainty of direction. Now that the U.S. Nuclear Regulatory Commission (USNRC) has issued the Maintenance Rule (effective July 10, 1996), I believe the BNCS needs to reassess its previous directive. It may be time for the O&M Committee to begin its expansion into maintenance.

Let's look at the assurance of nuclear plant safety, broadly. If there is water, the boundary stays together and the pumps start and stop when they should and the valves open and close when they should—we have come a long way to ensuring safety. Sections III and XI address the boundary well. Pumps and valves appear to be a fruitful potential for upgrading safety.

The single major incident in the history of the U.S. nuclear power industry, the March 1979 Three-Mile Island accident, involved valves at several stages. Valves that could not, and in any case did not, operate properly either because there were locked in an incorrect position, leaked excessively, would not open against the differential pressure that existed across the valve disc, were closed when they should have remained open, or were finally opened to control the accident progression. If history is an indication, valves will probably play a part in initiating, controlling, aggravating, or mitigating any accident in the future. The need for good O&M codes and standards is evident.

But, what makes a good code or standard? First, it is needed—there is an identified user, purpose, and scope; second, it is feasible and warrants the investment of resources into its development; third, there is sufficient knowledge and experience to support its technical requirements; fourth, it does not constitute an unnecessary burden; fifth, it is clear and concise; sixth, it is enforceable; and seventh, it takes into account user feedback to assess the quality of the product and to identify where modifications are required. This can only come about if all affected parties are involved and everyone has an attitude of "let us sit and reason together" and "let us implement the consensus process in good faith."

USNRC/ASME—MUTUALLY BENEFICIAL

Let us turn to the programmatic question: "What is in it for me?" as may be posed by the USNRC or the ASME, the sponsors of this symposium.

Experience shows that the best contract with lasting staying power is one that has essentially equal benefits to both sides. Is this satisfied in the arena of ASME codes and standards?

The USNRC derives benefit from several viewpoints. First, a good base of technical requirements for safety regulation should reflect perspectives from various phases of product development and use. For example, those designing a component bring knowledge different from those fabricating, and those operating different from the other two. The ASME process brings this breadth of input to a code or standard.

Second, a consensus standard is likely to be more useful and is probably more practical for application in the regulatory process. Third, the USNRC use of ASME codes and standards in its regulations is advantageous because they are more likely to be accepted by hearing boards, the public, and the industry because of broad participation and opportunity for public comments in both the consensus and rulemaking processes. Fourth, even if the USNRC cannot accept the code or standard "as is" and takes exception, it is done from a good knowledge base because the USNRC was party to the process, and the basic document is sound. Fifth, the USNRC gets the benefit of resources for which the government is not paying.

Let us now turn to the question: "What is the advantage to the ASME of the USNRC participation in its process?" The ASME gets a large commitment of highly qualified technical people in the development of its nuclear codes and standards. Second, USNRC participation ensures proper consideration to safety, which is the primary objective of ASME codes. Because the USNRC, which is responsible by law for protecting public health and safety, is involved in the

process, issues that are of public interest will be identified and codes and standards that evolve will address the right issues. A final benefit is that the probability is increased for a code or standard to be used if the USNRC participates in its development, particularly if the USNRC endorses it in its process.

I believe the mutually beneficial objective of a good contact has been met.

SENSITIVITY

Going back to the beginning, I suggest that in a voluntary process, a vigilance and reason resulted from mutual respect of the participants that can only evolve after consciously putting oneself in another person's shoes. What is driving the other person? What constraints is he or she under? In short, are we sensitive to his or her perspective?

Let's explore this sensitivity question from the USNRC's perspective and then the ASME's perspective.

The USNRC must be sensitive to a whole litany of items in its relationship with the ASME. First, it must recognized that the ASME process is a volunteer program. Second, the USNRC must understand that, because of the voluntary nature of participation, the ASME does not control the scope, content, or timing of a code or standard. The third key to the sensitivity issue is that the USNRC must recognize that feedback from USNRC staff on the ASME committees during the development process is extremely important. Fourth, if the USNRC staff participating in the development of a standard do not express concerns regarding the content or timing, and the product is then not used in the regulatory process, the USNRC must recognize that a significant problem has been created for the ASME regarding credibility with its volunteers. Fifth, if there is a positive reaction all the way up the line regarding the technical content and then the USNRC votes negative at the consensus committee level, indicating a significant problem exists, the ASME has a problem. Sixth, the USNRC must be sensitive to the fact that timely response on a document after it is published is very important to ensure the continued credibility of the program. And last, a variation on the theme of the last four points is that the ASME problem is aggravated if the USNRC itself requested the effort and then ignored it, did not give it a high priority for review within the USNRC, or did not accept it for use. Thus, there are many ASME-associated items that the USNRC (as an organization and as its individual volunteer members) must be sensitive to if we are to have a continued outstanding relationship and produce useful documents.

There is an equally long list of items to which the ASME must be sensitive regarding the USNRC's participation in this process. First, the ASME must be sensitive to the fact that codes and standards are not the highest priority item competing for the very limited resources available to the USNRC. Second, it must recognize, by its own process, that the USNRC technical people act as individual professionals using their best judgment, often on the spot, at meetings. Third, the ASME must understand that things change. Fourth, and important, the USNRC's identification of an issue for resolution, its participation on a committee, and even a positive ballot at various levels, does not and cannot commit the USNRC to use a code or standard officially in our process. Fifth, the ASME must be sensitive that the USNRC process for evaluation and potential endorsement of a standard is complex. For a code or standard applicable to a nuclear power plant, a minimum of three line offices may be involved with evaluating the technical content for potential endorsement. These are the Office of Nuclear Reactor Regulation, the Office of Research, and the Office for Analysis and Evaluation of Operating Data. In addition, the Office of the General Counsel often is required to make a finding that there is no legal objection to the standard.

Sixth, even with concurrence of the key line offices and no legal objection from the Office of the General Counsel, there are still two key advisory committees that usually get into the process before a code or standard would be endorsed for use in the USNRC process: (a) the Advisory Committee on Reactor Safeguards, whose focus is on the technical aspects, and (b) the Committee

for Review of Generic Requirements, whose focus, in addition to reviewing the technical requirements, is to ensure conformance with the regulatory process.

The last sensitivity I would like to discuss in this list of sensitivities that the ASME must reflect on is backfitting. Backfitting considerations are a key part of the USNRC process to exercise control and ensure, to the extent practical, that it will not adopt unnecessary new requirements, particularly as they relate to operating plants. Considering the process for use of newly developed standards, the backfit rule may preclude imposing the requirements on operating plants. This result should not be viewed as an indication of a shortcoming in the quality of the standard. The standard may be used in regulating future nuclear plants or may be used as acceptable alternatives to other requirements.

The current state of the nuclear power industry, the real world, leads to my last sensitivity. Utilities are being driven to cut costs more now than during earlier times.

Fuel costs are essentially fixed, and rates are established by state regulatory bodies. This leaves the operations and maintenance costs as the prime areas licensees can attempt to control. The USNRC has been sensitive to this situation and responded by inviting licensees to submit "cost-beneficial licensing actions (CBLAs)" and by forming a regulatory review group to recommend changes to eliminate regulatory requirements that do not impact the overall safety of the plants. These efforts present a challenge to all ASME codes and standards committees, but in particular to the ASME O&M Committee, to maintain the quality of its codes and standards in the face of efforts to eliminate unnecessary provisions and cut costs.

Thus, the USNRC must be sensitive to the emphasis that the ASME may place on cost, which reflects the volunteers who are primarily from industry. Similarly, the ASME must be sensitive to the USNRC's responsibility by law to ensure public health and safety; cost is not the primary focus.

SUMMARY AND CONCLUSION

In summary, we have a good process—the consensus process. We currently have a credible sponsor, ASME, that produces credible standards. We know what makes a good standard. We have a good contract between the USNRC and the ASME. It is mutually beneficial. O&M has challenging issues and has the opportunity to make a significant contribution. We cannot become complacent; we must continuously exercise vigilance and reason. It is difficult to visualize successful initiation or continuation of this relationship if we do not have mutual respect or are not sensitive to others. In short, we must sit down and reason together in good faith. I can think of no area that will be more challenging than the development and promulgation of codes and standards for pumps and valves to contribute to ensuring nuclear power plant safety. The challenge is laid at your feet. This symposium, held jointly by the USNRC and the ASME, is a key ingredient. You have your work cut out for you. I wish you well.

An Overview of Valve and Pump Testing Regulatory Issues

Status of Generic Letter 89-10, Future for Rulemaking—10 CFR 50.55a, Application of Probabilistic Risk Assessment to Testing Requirements, Inservice Testing, and Advanced Reactors[a]

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

INTRODUCTION

Since the last American Society of Mechanical Engineers/U.S. Nuclear Regulatory Commission (ASME/USNRC) symposium, the USNRC has observed improvement in the performance of pumps and valves at nuclear power plants. Nevertheless, reports of problems and component failures continue to occur more frequently than the USNRC and plant management would like. Therefore, the nuclear industry and regulators must continue their efforts to improve the performance of pumps and valves in nuclear power plants.

The third joint ASME/USNRC symposium presents an opportunity to exchange information, ideas, and suggestions to improve the performance of pumps and valves. Symposium participants include plant personnel involved with the daily operation of pumps and valves, coordinators of pump and valve programs at nuclear plants, individuals who support the ASME Operations and Maintenance Code for testing pumps and valves, and members of the USNRC staff responsible for evaluating licensee activities to ensure the proper performance of these components.

STATUS OF GENERIC LETTER 89-10

Background

Motor-operated valves (MOVs) installed in safety-related systems in nuclear power plants are required to be designed, fabricated, erected, and tested to quality standards commensurate with the importance of their safety functions to be performed. However, operating experience and research continue to reveal problems with the performance of these valves. Responding to concerns about MOV performance, the USNRC issued Generic Letter 89-10 (GL 89-10), "Safety-Related Motor-Operated Valve Testing and Surveillance," dated June 28, 1989, with several supplements. In GL 89-10, the USNRC staff requested that nuclear power plant licensees 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

a. This paper was prepared by an employee of the U.S. Nuclear Regulatory Commission. It presents information that does not represent a current staff position. The USNRC has neither approved nor disapproved its technical content.

conditions where practicable, improving evaluations of MOV failures and necessary corrective action, and looking for trends in MOV problems. The staff asked licensees to complete the GL 89-10 program within three refueling outages or 5 years from the issuance of the generic letter, whichever was later.

Recent Activities

Since the last symposium, the USNRC issued Supplement 5 to GL 89-10, requesting licensees to reexamine their MOV programs and to identify measures to account for inaccuracies in MOV diagnostic equipment. Licensees were requested to notify the staff of their diagnostic equipment and to report their plans for addressing the information on the accuracy of MOV diagnostic equipment. The staff has reviewed the responses to Supplement 5 to GL 89-10 and sent replies to individual licensees. USNRC inspections are addressing specific aspects of licensees' actions to address the inaccuracy of MOV diagnostic equipment.

On March 8, 1994, Supplement 6 to GL 89-10 was issued to clarify guidance for licensees on the completion schedule of GL 89-10 programs, to provide guidance on the MOV grouping methodology for using comparative test data, and to respond to questions raised at the public workshop held in February 1993 to discuss the generic letter. Also, the staff requested that if the licensee wishes to extend the schedule for completing the GL 89-10 program, the licensee must submit specific information on the capability of those MOVs whose test schedule will be extended. However, the staff stated that even if its GL 89-10 test schedules are extended, a licensee is expected to have the safety-related MOVs set up using the best-available MOV test data by the original completion date accepted by the USNRC.

The USNRC staff contracted Brookhaven National Laboratory to perform a core-melt frequency study of the inadvertent operation of MOVs in pressurized water reactor (PWR) nuclear power plants. Supplement 4 to GL 89-10 addressed inadvertent MOV operation in boiling water reactor (BWR) plants. Similarly, Supplement 7 to GL 89-10, which is being prepared for public comment, discusses inadvertent MOV operation in PWR plants.

Though many MOV problems have been revealed and corrected as a result of the GL 89-10 programs, the USNRC staff believes that the number of MOV problems and the operating events caused by those problems are being reduced by the actions taken in response to GL 89-10 and will continue to decrease as licensees complete their GL 89-10 programs. To assist licensees in completing their GL 89-10 programs, the Nuclear Energy Institute (NEI) has submitted for USNRC staff review a topical report describing the Electric Power Research Institute (EPRI) MOV Performance Prediction Program. EPRI tested gate, globe, and butterfly valves and analyzed the results of additional valve tests in their development of a methodology to predict the performance of MOVs. Additionally, several licensees and NEI have proposed a ranking of MOVs for establishing test schedules based on probabilistic risk assessment methods. The USNRC is receptive to such proposals as discussed in GL 89-10, Supplement 6, and will review the proposals on a case-by-case basis.

Pressure Locking and Thermal Binding of Valves

In March 1993, the USNRC issued NUREG-1275, Volume 9, *Pressure Locking and Thermal Binding of Gate Valves*, which gives the history of pressure-locking and thermal-binding events, describes the phenomena, discusses the effects of locking or binding on valve functionality, summarizes preventive measures, and assesses the safety significance of the phenomena. On February 4, 1994, the USNRC staff held a public workshop to discuss the potential for pressure locking and thermal binding in nuclear power plants. Notwithstanding the fact that both industry and the USNRC have identified this design weakness, USNRC inspections and operating experience show that this is a continuing problem. The staff is developing a proposed generic letter to request that each licensee identify safety-related power-operated gate valves that

could be susceptible to pressure locking and thermal binding and implement corrective action for those valves within a schedule discussed in the generic letter.

Inspections

Over the last 3 years, inspection efforts have been devoted to the programs developed by licensees in response to GL 89-10. The most significant concerns identified during the inspections were (a) slow progress in performance of dynamic testing, (b) the adequacy of test acceptance criteria, and (c) feedback of test results. Detailed results of the inspections will be discussed in the Regulatory Session.

Future

Many licensees are nearing completion of their GL 89-10 programs associated with the review of MOV design bases, verification of initial MOV switch settings, testing of MOVs under design-basis conditions where practicable, and improvement of evaluations of MOV failures and necessary corrective action. Licensees will need to establish processes to ensure that the long-term actions for GL 89-10 programs, such as periodic verification of MOV capability and the trending of MOV problems, are developed and maintained. It is particularly important for the industry to implement appropriate maintenance and condition monitoring to ensure degradation of MOVs is detected early and appropriate corrective action is taken to maintain the licensing basis for safety-related MOVs.

FUTURE FOR RULEMAKING—10 CFR 50.55A

Proposed Rulemaking

In the area of inservice testing of pumps and valves, the most recently proposed rulemaking for 10 CFR 50.55a will incorporate the ASME Operations and Maintenance Code (O&M Code) into the regulations. This rulemaking will complete the separation of the operation, maintenance, and testing of components from Section XI of the ASME Boiler and Pressure Vessel Code (the Code), which is structured to ensure the integrity of components, systems, and structures. The O&M Code was initially issued in 1990 to compile the standards for pump and valve testing. The scope of the O&M Code and standards now goes beyond pumps and valves to include inservice testing of snubbers, monitoring of heat exchanger heat transfer capability, and monitoring of emergency core cooling system capability.

Standard Technical Specifications and Rulemaking

The most recently proposed change to the revised Standard Technical Specifications is the deletion of the administrative section on inservice testing (except that the frequency table will remain) and licensees will be expected to develop and implement their inservice testing programs in accordance with the regulations without the specific reference in their technical specifications. The deletions are intended to narrow the scope of technical specifications to those items not covered by a regulatory requirement.

Containment Isolation Valves

The proposed rulemaking calls for the deletion of the previous modification to O&M Part 10 (now O&M Code, Section ISTC) that retained certain leak-testing requirements for containment isolation valves. After reviewing a study undertaken by a special task group under the O&M Committee, the staff agreed with the results, which indicated that no additional failures (excessive leakage) of containment isolation valves would be identified by continuing to impose inservice testing requirements, redundant or in addition to the local leak rate testing requirements in 10 CFR Part 50, Appendix J. The task group did, however, recommend that the O&M Code be clarified to state that containment isolation valves with another leaktight function (such as pressure isolation, system or train boundary, Code class boundary, etc.) are subject to inservice testing

leak rate testing requirements in addition to the Appendix J local leak rate testing.

Other Rulemaking Activities

The staff is taking comments from NEI and an ad hoc industry group on suggested revisions to the regulatory requirements that will focus the inservice inspection and inservice testing programs and Code process more on the safety aspects of inspection and testing.

INSERVICE TESTING

Guidelines

The USNRC developed draft NUREG-1482, *Guidelines for Inservice Testing at Nuclear Power Plants*, to assist the industry in eliminating unnecessary requests for relief and to approve the use of an alternative method of inservice testing (IST) if that method conforms to the latest edition of industry Code and standards approved by the USNRC. For those cases where relief is required, guidance is given on information that should be included in relief requests for prompt staff approval for a number of generic issues. Also included are a format for relief requests and information on the justifications needed for deferring tests to cold shutdown or refueling outages. A number of issues are discussed that have been identified in USNRC inspections, from licensees' questions in telephone calls or meetings, and through USNRC staff participation on the O&M committees. The final NUREG report is expected to be issued by the end of August 1994 for use by the industry. The staff plans to hold a workshop on its revised inspection procedure (IP-73756) for IST, which will reference both the guidelines in NUREG-1482 and in Generic Letter 89-04, "Guidance on Developing Acceptable Inservice Testing Programs."

Design-Basis Verification of Components

I spoke at the 1992 symposium about design-basis verification testing. Before the 1992 symposium, the USNRC had asked the O&M Committee to evaluate the need for IST to verify that the components could operate under the conditions expected for performing their design-basis functions, similar to the need described in Generic Letter 89-10 for MOVs. Since that time, the Electric Power Research Institute (EPRI), under contract to ASME, conducted a study to evaluate the need for design-basis-type testing. The results of EPRI's study are given in EPRI-TR-102240, *Evaluation of the Safety Benefits and Costs of Proposed Revisions to Inservice Testing Requirements for Pumps and Valves.*

Notwithstanding the results of the EPRI study, the O&M Committee has moved in the direction of including design-basis-type testing in the code and standards, most notably, the "comprehensive pump testing" recently incorporated into Section ISTB of the O&M Code, and Part 8 on MOV testing. Several licensees have expressed interest in requesting relief to implement the new pump testing because the overall testing time is decreased, and the periodic test with substantial flow may provide a better indication of pump performance.

ADVANCED REACTORS

To ensure the adequacy of component capability for the design-basis conditions, the staff's evaluation of evolutionary and passive advanced light-water reactor pump and valve issues includes design, qualification, preoperational, and IST requirements. The staff conducted its IST review in accordance with the requirements of the ASME Code supplemented by the Commission-approved requirements for IST of pumps and valves contained in SECY 90-016 dated January 12, 1990. To date, the staff has reviewed the IST programs for the General Electric Advanced Boiling Water Reactor and the Combustion Engineering System 80+. The staff expects the Westinghouse AP-600 IST program to be submitted and reviewed by the end of 1994.

CONCLUSION

Overall, I am pleased with the direction that the O&M committee and the USNRC are taking in the area of testing pumps and valves. I believe that ASME, through its consensus standards

process, works to the advantage of the industry and the regulators. I am also encouraged by the industry initiatives in creating and participating in users groups for various types of components, such as the Motor-Operated Valve Users' Group, the Air-Operated Valve Users' Group, and the Nuclear Industry Check Valve Group. These groups show that the industry is attempting to address the design, operation, testing, and maintenance needs for these specific types of valves. I encourage the members of these groups to interact with, or participate directly in, the O&M committees. The USNRC will continue to participate in these activities.

Session 1A
MOV General Session

Session Chair
Kevin DeWall
Idaho National Engineering Laboratory

High-Energy Flow Interruption Testing of Anchor/Darling Valve Company Gate Valves

Drew Wright
Anchor/Darling Valve Co.

ABSTRACT

The Anchor/Darling Valve Company (A/DV) has recently developed an improved flex wedge design intended for use in high-energy flow applications. The improved design was incorporated into a 6-in. 900 lb class flex wedge gate valve. Under a similar project, a double disc gate valve was also manufactured for the purpose of evaluating the high-energy flow performance of A/DV's double disc design. Both valves were subjected to rigorous testing at Wyle Laboratories in Huntsville, Alabama. This paper describes the test specimens, valve preparations, test programs, results of the test programs, and the benefits gained as a result of the program.

INTRODUCTION

The Anchor/Darling Valve Company (A/DV) initiated two test programs, one in 1991 and one in 1992, for the purpose of evaluating the performance of A/DV's double disc design and a modified flex wedge guide design under high-energy flow conditions. The test programs were intended to address utility concerns regarding Generic Letter 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance" issued by the U.S. Nuclear Regulatory Commission (USNRC).

Modifications were incorporated into a standard 6-in.-900-lb class flex wedge valve design prior to A/DV's testing. Previous tests performed by the Idaho National Engineering Laboratories (INEL) (Steele et al., 1990; DeWall and Steele, 1989) on a standard 6-in.-900-lb A/DV flex wedge design resulted in significant valve damage. The effectiveness of the modification was determined through absolute results, as well as a qualitative comparison with previous results. The double disc gate valve was a 6-in.-900-lb class valve of standard design. The performance of the double disc valve was evaluated solely on the basis of the actual test results.

It was A/DV's intent to be able to support the utilities' efforts to address their motor-operated valve (MOV) issues through the availability of the modified flex wedge design, which may be incorporated to enhance the operability of existing flex wedge gate valves in high-energy systems. Satisfactory test performance of the double disc design would support the use of the design for high-energy systems where the performance of existing valves is unsatisfactory and modifications are not practical. It was also anticipated that the test results would be useful in evaluating similar modifications and improvements in other sizes and pressure classes of A/DV flexible wedge gate valves.

In addition to the high-energy flow interruption tests, both valves were subjected to conditions intended to indicate the valves' susceptibility to thermal binding.

DESCRIPTION OF TEST SPECIMENS

Both test valves were 6-in., American National Standards Institute (ANSI) 900-lb class gate valves with American Society of Testing and Materials (ASTM) A216-WCB bodies. Both test valves were equipped with a Teledyne Engineering SMARTSTEM™ for direct measurement of stem thrust and torque. In addition, the flex wedge test valve was modified to incorporate machined steel (ASTM A108-1018) guides

welded to the body, and a disc with lengthened guideways that were hardfaced with Stellite No. 21. The modified guide design of the flex wedge valve resulted in a total axial guide slot clearance of 0.109 in. maximum, as compared with the standard design previously tested, which had a 0.317-in. maximum axial clearance. The bottom of the machined body guides was located approximately 2.5 in. below the centerline of the valve port; whereas the body guides of the standard design were located approximately 1.1 in. below the valve centerline. Figures 1 and 2 illustrate the guide and seat configuration of the flex wedge test valve. The seat flat width was designed based on the worst-case stress condition under full flow. The 5/16-in. seat flat was oriented on the downstream side of the valve throughout the test program.

The double disc gate valve was essentially a standard design. The nominal disc pack clearance was 0.033 in. This relatively tight disc pack clearance served to minimize disc tilt during opening or closing, as well as to reduce vibration with the valve in the open position. The only non-standard feature of the double disc internals was the seat flat width. As with the flex wedge test valve, the seat flat width was designed based on the worst case stress conditions under full flow. One seat was machined to have a 3/8-in. flat; the other was machined to have a 7/32-in. flat. The 3/8-in. flat seat was oriented on the downstream side of the valve throughout the test program.

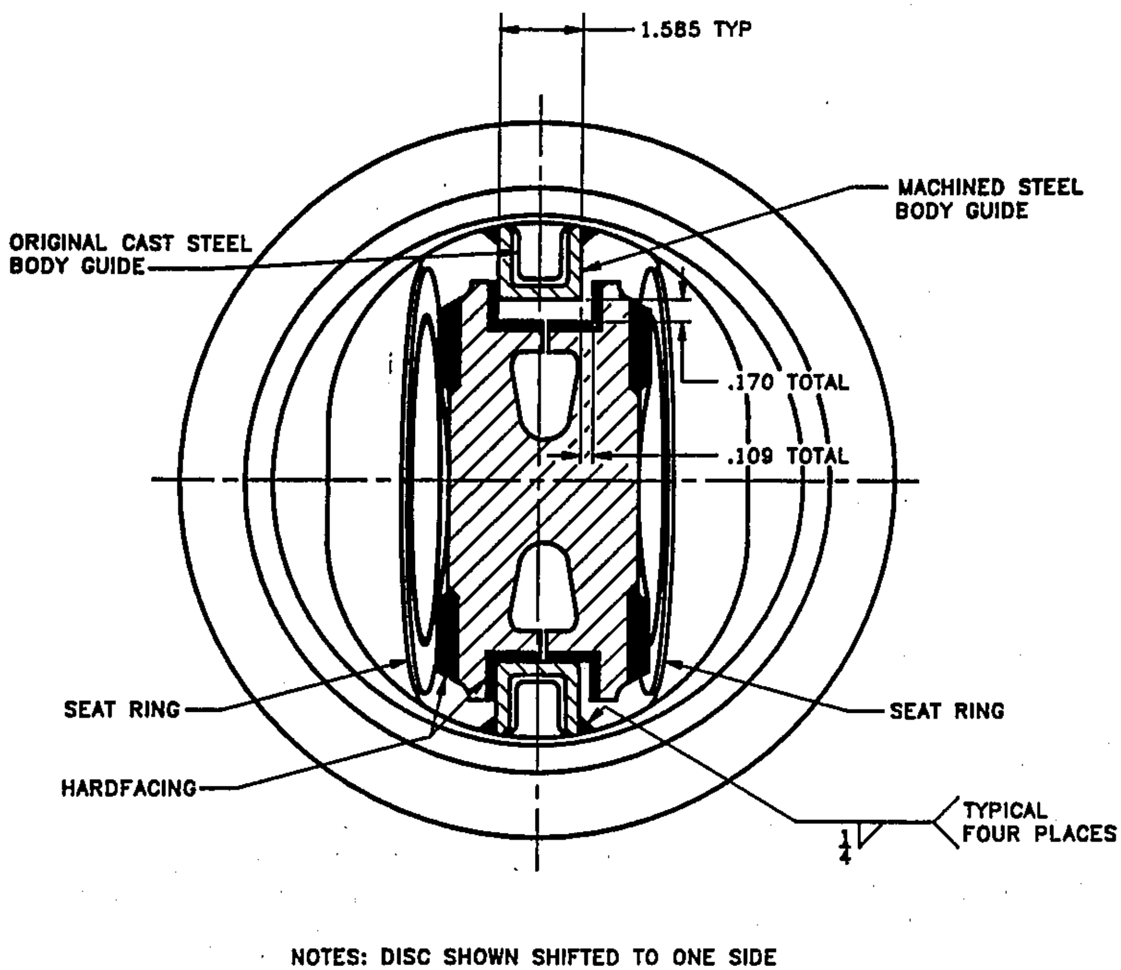

Figure 1. Top view of guide modification.

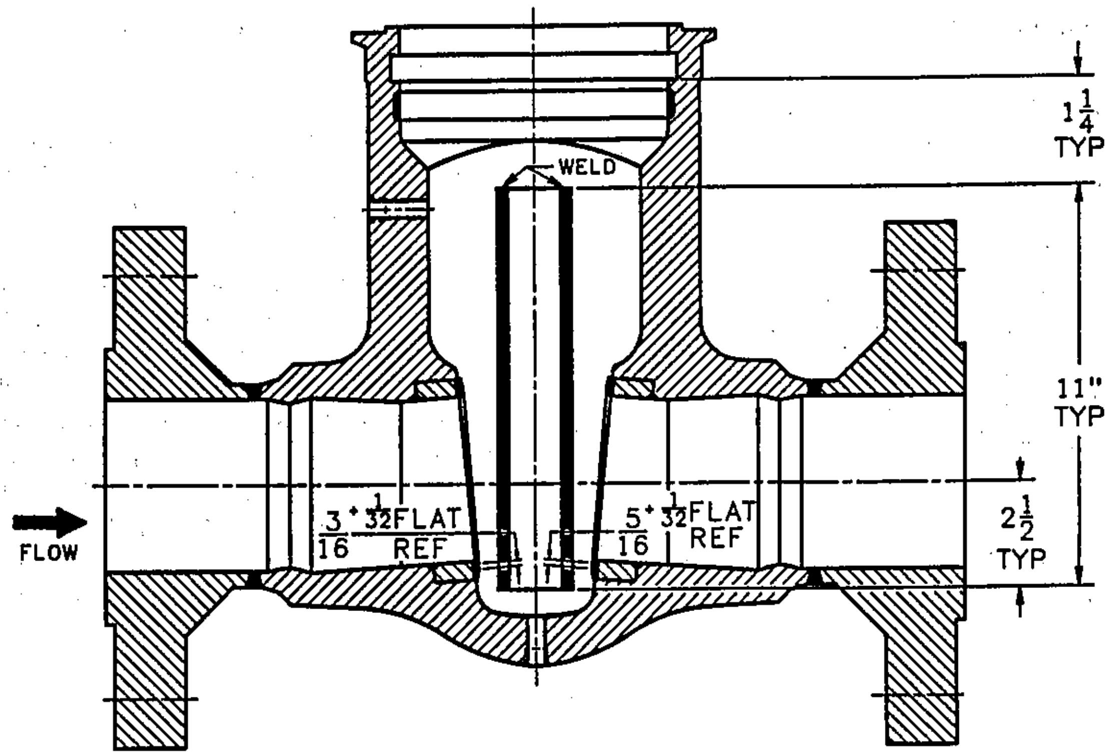

Figure 2. Side view of guide modification.

The seating surfaces of the seats and discs of both valves were hardfaced with cobalt chrome alloy. The machining chips from the final machining cut of the seat ring and disc hardfacing of each valve were analyzed and verified to contain no more than 5% iron. This criterion was imposed because of A/DV's experiences indicating that iron dilution in cobalt chrome hardfacing significantly greater than 5% can result in an increase in the coefficient of sliding friction, as well as a decrease in wear properties.

DESCRIPTION OF TEST SYSTEM AND INSTRUMENTATION

The data acquisition system employed was a 21-channel TEACK analog tape recorder. Data samples were taken at a frequency of 2,000 Hz. The data were then digitized and stored for further processing.

The valves were instrumented to measure several parameters. Table 1 lists the instrumentation employed and the associated tolerances. The SMARTSTEMs ™ were calibrated both before and after testing. The test system was an open loop arrangement that employed large nitrogen tanks to maintain system pressure during valve stroking. The system was initially pressurized with a pump before each test run. Figures 3 and 4 provide schematics of the test system and the associated instrumentation.

TEST PROCEDURE

Valve Preparation

Before the high-energy flow testing at Wyle Laboratories in Huntsville, the test valves were subjected to cycle testing, as stated in A/DV standard procedure EPS-191 R/–. This testing consisted of 50 opening strokes against 1,500 psig differential pressure, and 50 closing strokes under 1,500 psig internal pressure. Seat leakage tests were performed at 1,500 psig after every 10 cycles. Air seat tests were performed both before and after the cycle tests. All tests revealed zero seat leakage. The test valves were disassembled

Table 1. Instrumentation.

Measurement	Instrument	Range	Tolerance
Temperature, valve body surface	Thermocouple	0–1000°F	±4°F
Temperature, water supply	Thermocouple	0–1000°F	±4°F
Press, water supply	Press, transducer	0–2000 psig	±0.25%
Press, valve upstream	Press, transducer	0–2000 psig	±0.25%
Press, valve downstream	Press, transducer	0–2000 psig	±0.25%
Press, valve body neck	Press, transducer	0–2000 psig	±0.25%
Press, valve body bowl	Press, transducer	0–2000 psig	±0.25%
Press, valve differential	P transducer	0–2000 psig	±0.25%
Water flow rate	P transducer	0–2000 psig	±0.25%
Valve position	LVDT	0–10 in.	±0.25%
Voltage, motor	DVB/Transformer	0–1000 volts	±1%
Current, motor	Amp probe	0–600 amps	±2%
Torque, motor	Power meter	0–160 kW	±1%
Torque, switch	Electrical contacts	N/A	N/A
Torque/thrust data	Teledyne	0–400 ft-lb 0–30,000 lb	±3.2%

and inspected for wear or damage following the completion of the tests. No wear or damage was apparent. The valve was cycled as a precaution to ensure that the valve would operate freely without binding during any unanticipated circumstances. Motor current was monitored throughout the stroking process. Any notable increase in required operating thrust would have been revealed through higher running current. Current measurements remained stable throughout the cycling process of both valves.

Wyle High-Energy Flow Testing

The test valves were shipped to Wyle Laboratories in Huntsville, Alabama, where each valve was subjected to a minimum of five cold (i.e., ambient temperature) and five hot high-energy closure tests in accordance with Wyle procedure No. 41854 R/C. The cold closure tests were performed at nominal inlet pressures of 700 and 1,400 psig. The hot closure tests were performed at nominal inlet pressures of 700, 1,000, and 1,400 psig and at nominal temperatures of 500°F or 580°F. The test valves were instrumented as depicted by Table 1.

The general test sequence for each valve consisted of the following:

- Baseline testing, no pressure
- Initial opening test, 1,400 psig differential pressure
- Cold water high-energy closure tests (five)
- Hot water high-energy closure tests (five).

The initial cold water high-energy closure tests were performed in two stages because of capacity limitations of the system, 100–50% open and 50–0% open. Subsequent cold high-energy closure tests were performed from 50–0% open. All hot water high-energy closure tests were performed from 100% open.

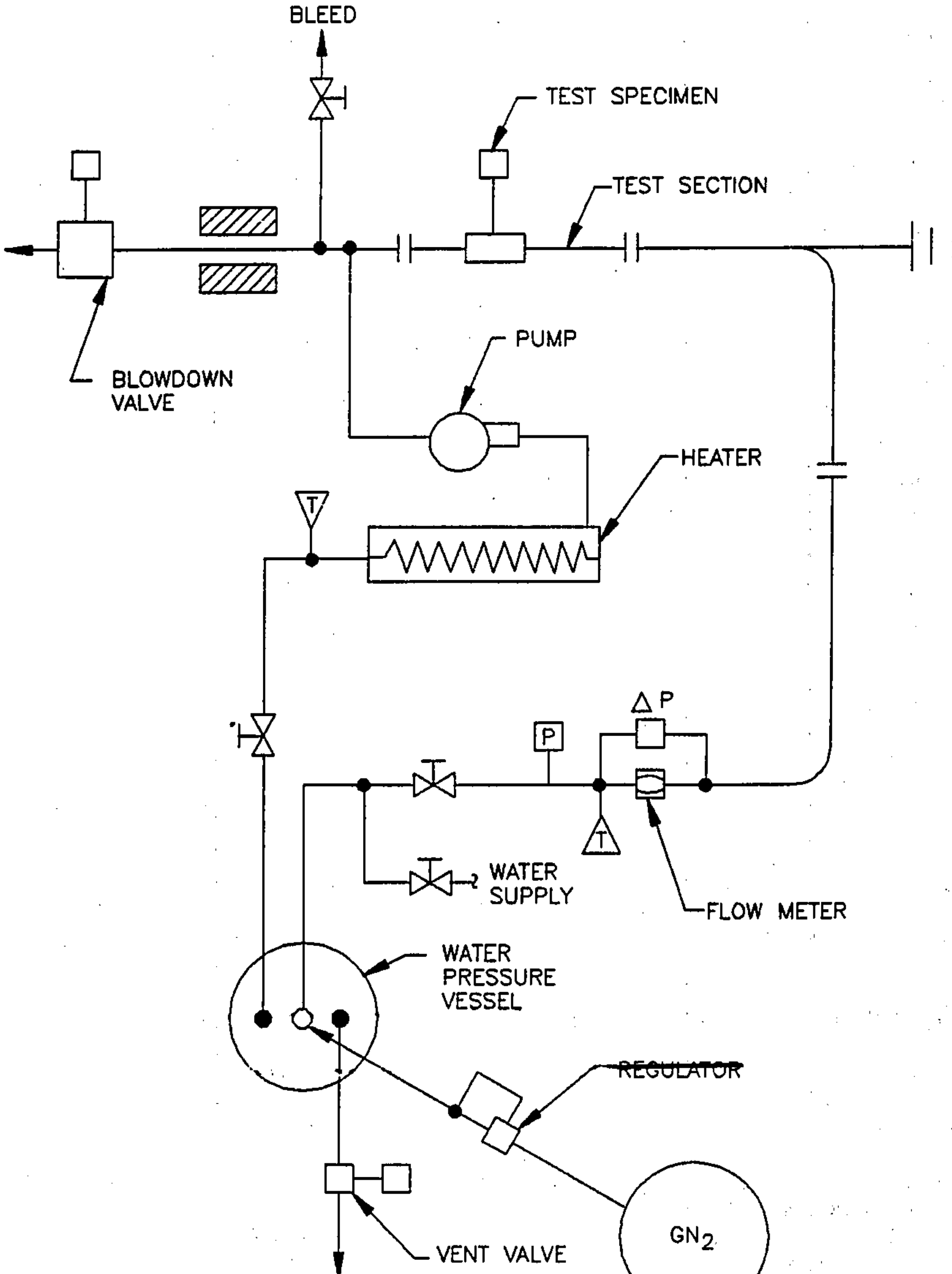

Figure 3. System schematic.

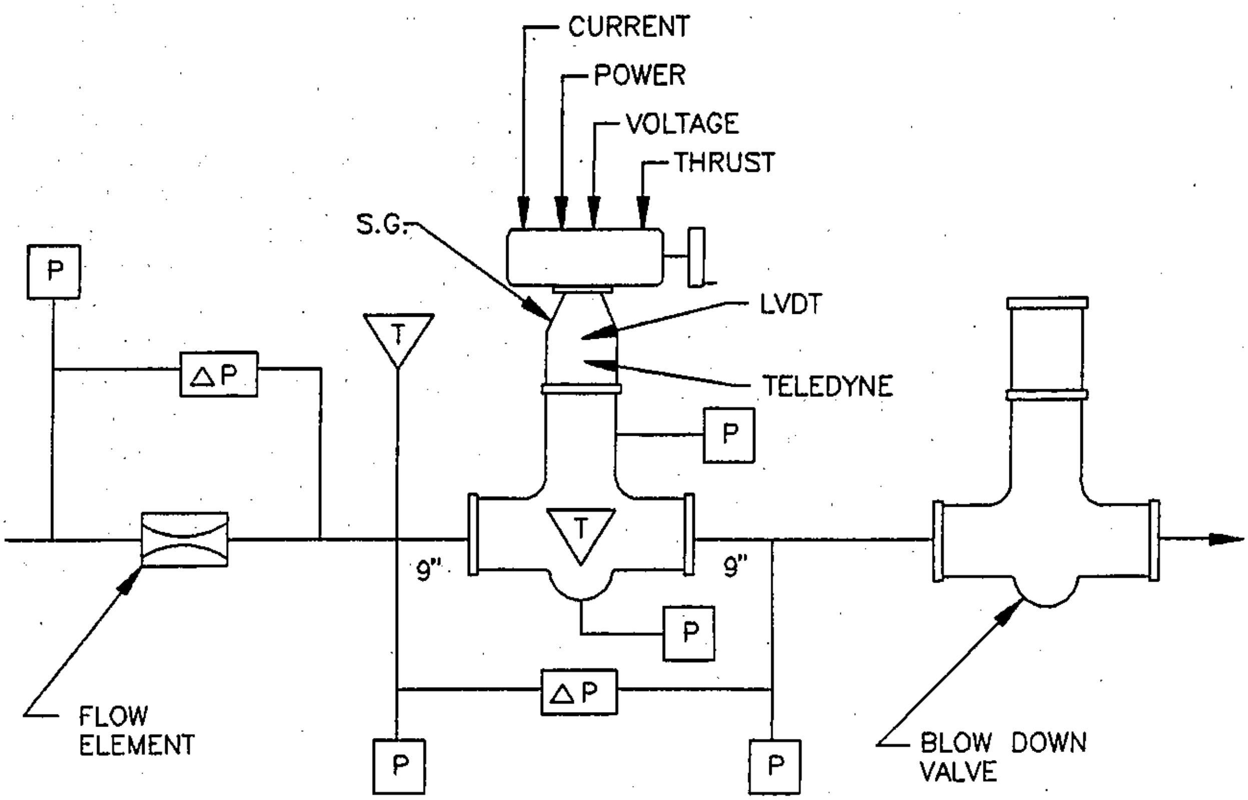

Figure 4. Test section.

After each high-energy closure test, a seat leakage test was performed and the valve was reopened. The valve was then disassembled for inspection. Seat leakage tests were performed again after valve reassembly. An exception was the next to last high-energy closure of the flex wedge valve, after which the valve was left in the line for the last high-energy closure.

During the testing of the flex wedge valve, the stem was balanced to a neutral state (i.e., no tension or compression) and the instrumentation was zeroed accordingly before each high-energy closure test. During the testing of the double disc valve, the thrust/torque instrumentation was not zeroed until after the test system had been pressurized, requiring appropriate adjustments to be made when accounting for stem end load.

Flow isolation was determined by either monitoring upstream venturi data or appropriate pressure data. The established points of isolation were verified using baseline closing and opening data.

During the hot tests, the valves were allowed to cool after closure to less than 100°F before performing seat testing and opening strokes. This procedure was intended to evaluate each valve's susceptibility to thermal binding. However, it should be noted that there was essentially no piping or system loads were imparted onto the valve assembly during these tests.

TEST RESULTS

The data for the 100–50% open and 50–0% open test runs for both valves indicated that the state of maximum stem thrust occurred during the 50–0% open portion of the stroke for both valves. The disassembly and inspection of the valves after the high-energy closure from 100–50% open revealed no damage to either of the valves. Therefore, it was necessary to test only the 50–0% open portion of the valve stroke for the remaining high-energy closure strokes performed under cold conditions.

Figures 5 through 10 present raw data of critical parameters for several strokes of both the double disc valve and flex wedge valve. Although the proper definition of "valve factor" has been the subject of much debate, A/DV defines valve factor as the net stem thrust (i.e., measured stem thrust adjusted for packing load and stem end load) at the point of isolation divided by the pressure force acting over the downstream seat. The pressure force is assumed to act over the mean seat diameter. This definition does not consider nonzero seat angles, nor does it address portions of the thrust signature beyond isolation. It is A/DV's belief that employing flow isolation as the basis for determining valve factors will provide more consistent results that enable more meaningful comparisons of valves, whether they are of seemingly identical or somewhat different designs (i.e., size or pressure class). This position is based on an assessment of some variables that may affect the valve factor beyond the point of isolation. In this regime, mechanical factors, such as engagement of double disc wedges and contact of the disc on the upstream seat of a gate valve, can contribute to the apparent valve factor. However, these effects are dependant on variables such as the stiffness of cast bodies, discs, and wedges; operating clearances; and the relative position of the disc and valve centerlines in the fully closed position. As cast parts are rarely identical and manufacturing tolerances provide for slightly different fit-up dimensions, the effect of these variables can be notable, even between two seemingly identical valves. Based on A/DV's observations, the thrust level can either decrease or increase between the point of isolation and complete wedging, subject (in the author's opinion) to the cumulative effects of the aforementioned variables.

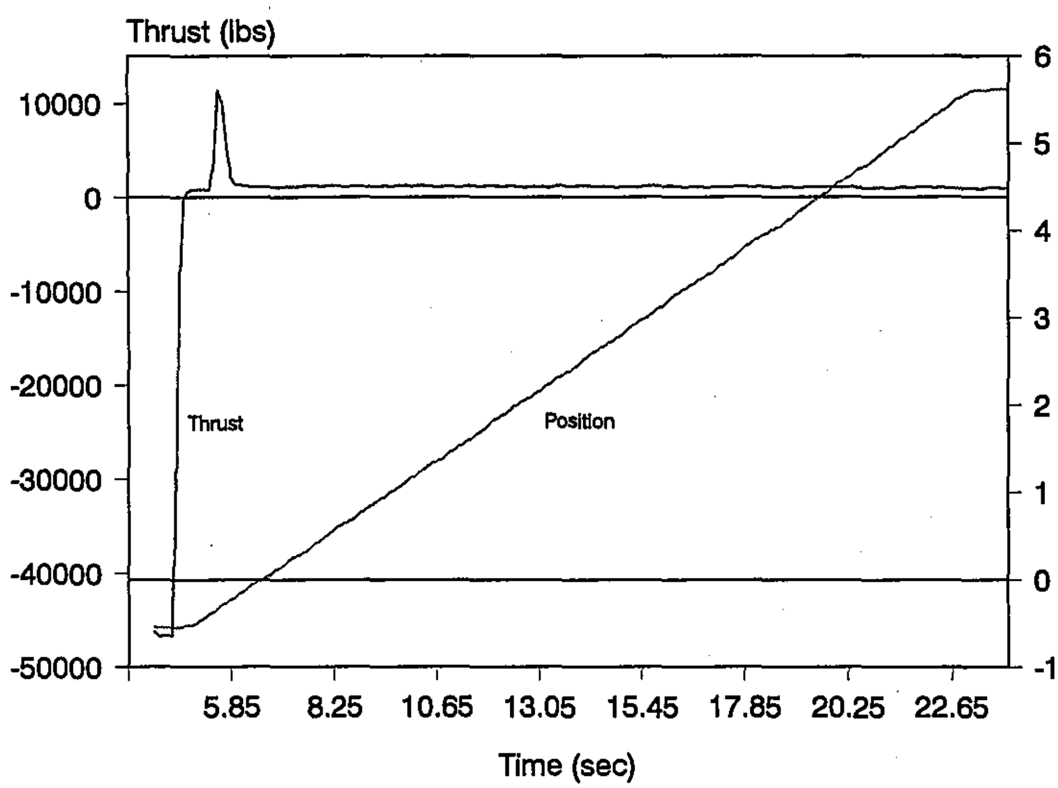

Figure 5. Wyle test: six-in.-900-lb flex wedge gate valve—opening Stroke No. 3: nominal test conditions—0 psid and 50°F.

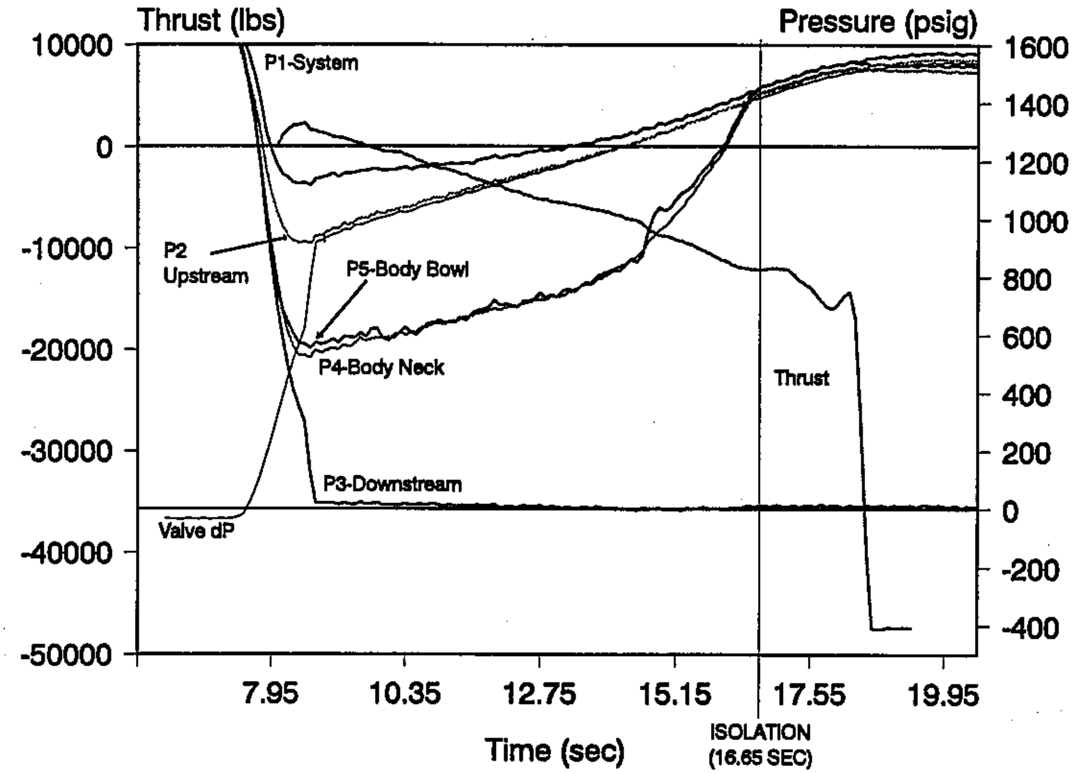

Figure 6. Wyle test: six-in.-900-lb double disc gate valve—closing Stroke No. 4: nominal conditions—1400 psid and 50°F.

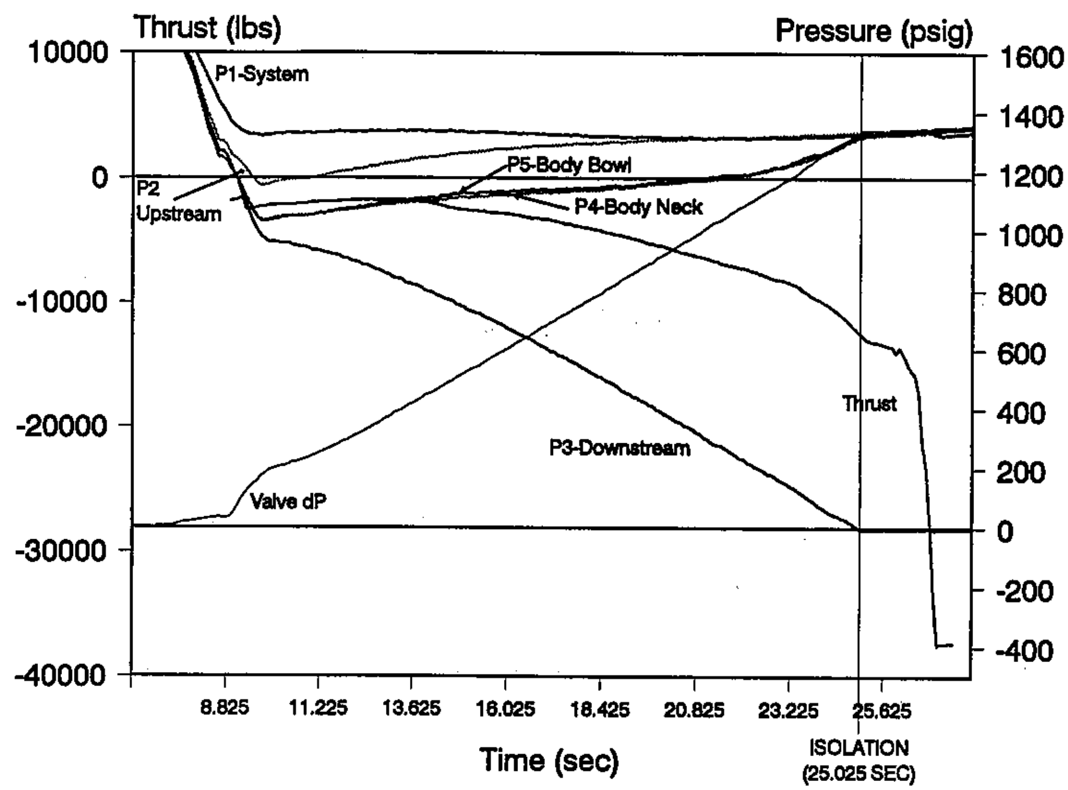

Figure 7. Wyle test: six-in.-900-lb double disc gate valve—closing Stroke No. 8: test conditions—1400 psid and 550°F.

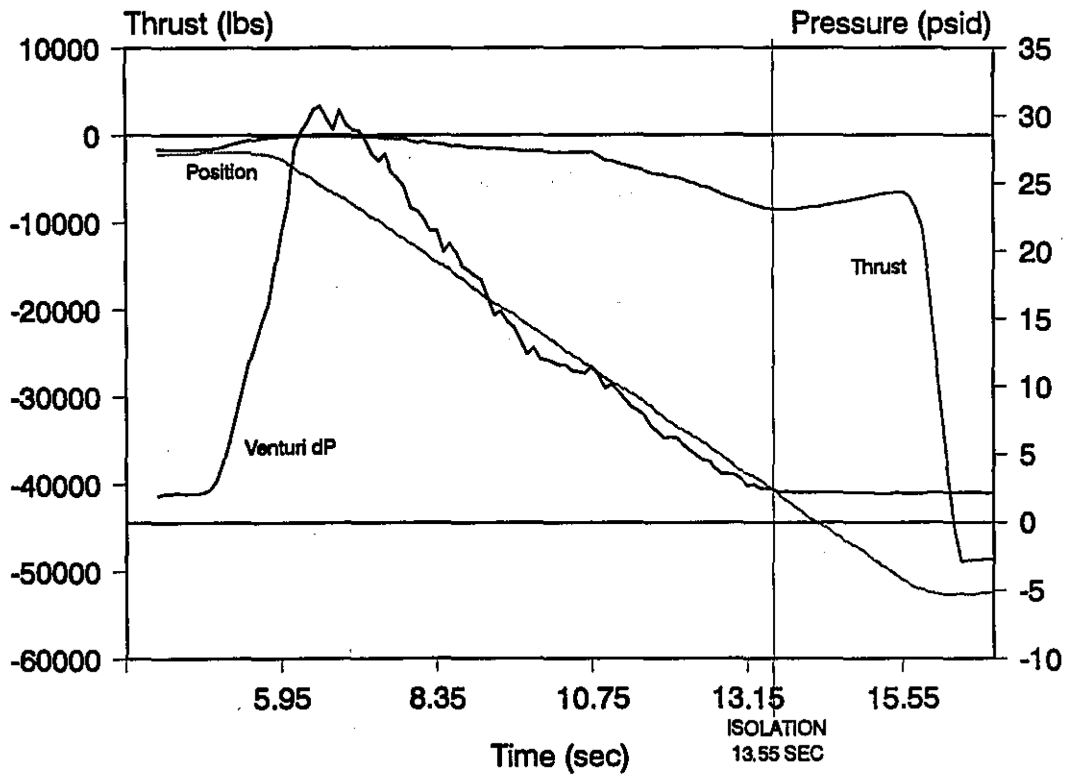

Figure 8. Wyle test: six-in.-900-lb flex wedge gate valve—closing Stroke No. 3: nominal test conditions—1200 psid and 50°F.

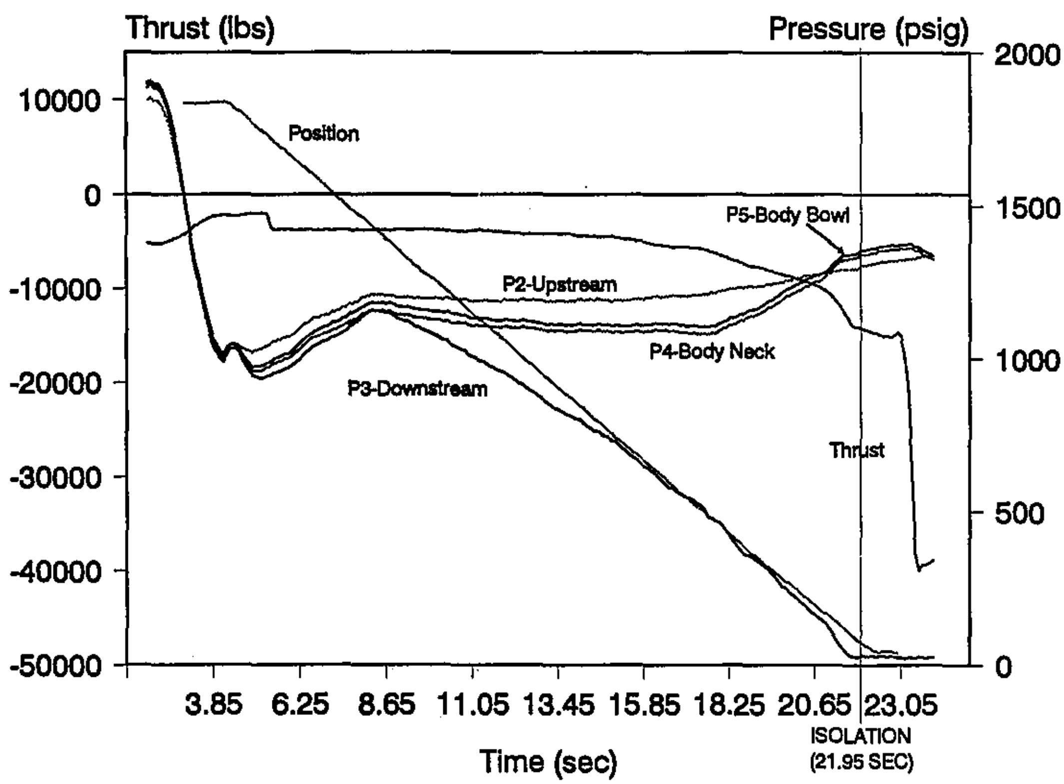

Figure 9. Wyle test: six-in.-900-lb flex wedge gate valve—closing Stroke No. 6: nominal test conditions—1400 psid and 575°F.

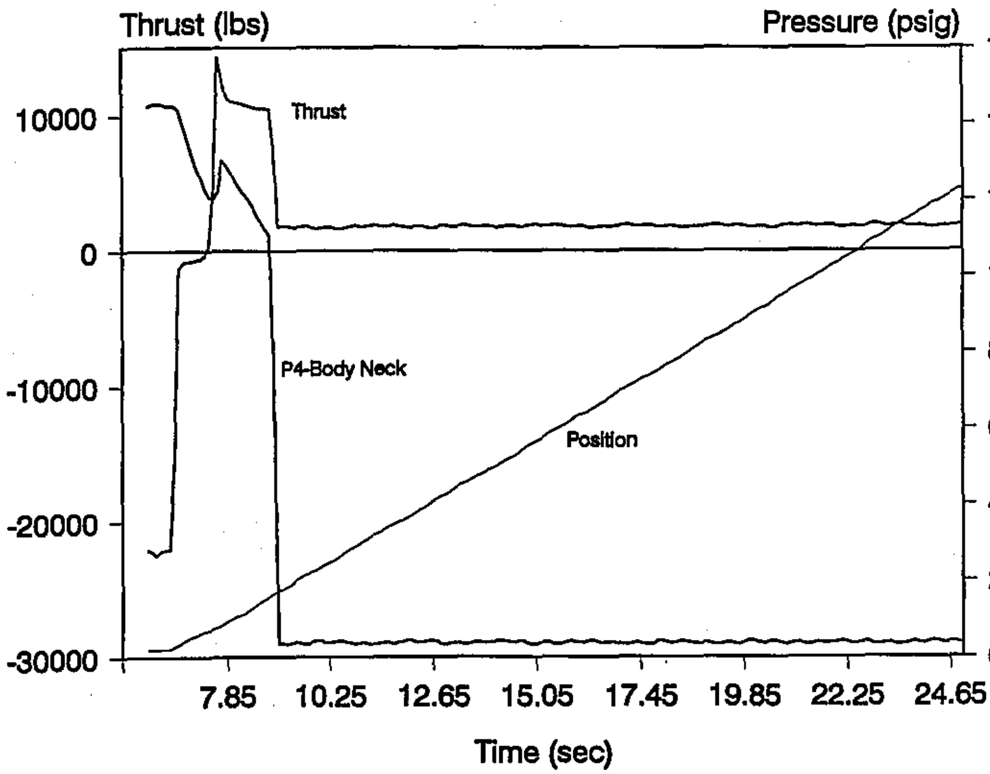

Figure 10. Wyle test: six-in.-900-lb flex wedge gate valve—opening Stroke No. 6: nominal test conditions—1400 psid (between seats) and 50°F.

Therefore, the point that appears to provide the most consistent basis for comparison is the point of flow isolation. It is A/DV's and the author's opinion that the point of isolation also addresses the concern that initiated the MOV issue: isolation of flow under high-energy flow conditions.

Double Disc Results

Referring to Figure 6, the point of isolation for closing Stroke No. 4 is identified by a convergence of the upstream and valve body pressures. The point of isolation for hot test Stroke No. 8 (Figure 7) is identified by both the convergence of the body and upstream pressures, as well as the point at which the downstream pressure goes to zero. In both cases, an inflection appears in the thrust trace at the approximate point of isolation. The slope of the thrust trace decreases at the point of isolation, as the downstream disc slides across the downstream seat, with the upstream disc being separated from the upstream seat. Beyond that region, a subsequent increase in slope indicates the onset of interaction between the upper and lower wedges.

Figure 11 presents graphical data on valve factor versus stroke for both the hot and cold series of closure tests for the double disc. Table 2 presents a summary of data for all strokes of the double disc.

The double disc valve successfully isolated flow under the referenced conditions for all tests. The valve factors tabulated in Table 2 were determined using the standard industry equation, to be discussed later in this paper. Visual inspections performed between high-energy closure strokes revealed no significant wear or damage, although slight burnishing of the disc and seat ring became apparent by the final high-energy closure stroke. The valve maintained zero leakage on all seat tests throughout the test program.

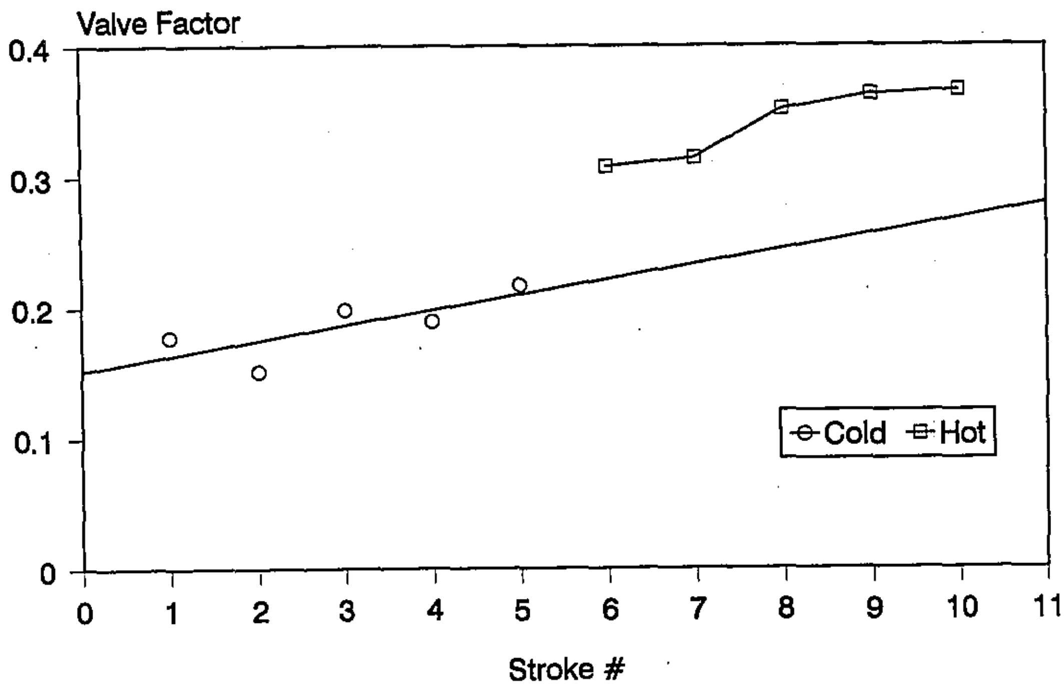

Figure 11. Six-in.-900-lb flex wedge gate valve data.

Flex Wedge Results

As shown in Figure 8, the point of isolation for closing Stroke No. 3 is identified by the point at which the differential pressure across the upstream flow meter stabilized at its minimum value. The point of isolation for hot test Stroke No. 6 (Figure 9) is identified by the point at which the downstream pressure stabilizes at its minimum value. As with the double disc, an inflection appears in the thrust trace at the approximate point of isolation. The slope of the thrust trace decreases at the point of isolation, as the downstream disc slides across the downstream seat.

Figure 12 presents graphical data on valve factor versus stroke for both the hot and cold series of closure tests for the flex wedge. Table 3 presents a summary of data for all strokes of the flex wedge.

The flex wedge valve successfully isolated flow under the referenced conditions for all tests.

The valve factors tabulated in the fourth and fifth columns of Table 3 were determined using the standard industry equation [Equation (1)]. In addition, a modified version of the standard industry equation [Equation (2)] was employed to account for the 5-degree seat angle (10-degree included angle).

The equations employed for both the double disc and flex wedge valves are presented below.

Standard Industry Equation (open or closing direction):

$$T_s = T_p \pm T_e + (vf \times A_s \times DP)$$

or

$$vf = (T_s - T_p \pm T_e) / (A_s \times DP) \ . \qquad (1)$$

Modified Industry Equation (closing direction only):

$$T_s = T_p + T_e + [(A_s \times DP) \times (\mu \times \cos\Theta + \sin\Theta) / (\cos\Theta - \mu \times \sin\Theta)] \ , \qquad (2)$$

Table 2. Six-inch-900-lb double disc data.

Close test	Calculated friction factor[a] (1,200 lb packing load)	Calculated friction factor[a] (1,200 lb packing load)	Observed packing load (subsequent opening run) (lb)	Adjusted position (in.)	Adjusted thrust at position (lb)	dp (psi)	Bowl pressure P5 (psi)	Mean seat dia. (in.)	Stem dia. (in.)	Disc area (in.^{2})	Stem area (in.^{2})	Disc force (lb)	Stem end load (lb)
Cold													
1B	0.208	0.214	990	0.34	11,769	1,580.4	1,557	5.5	1.5	23.76	1.77	37,548	2,751
2	0.256	0.296	50	0.60	10,496	1.203.7	1,125	5.5	1.5	23.76	1.77	28,598	1,988
3	0.210	0.228	627	0.67	9,982	1.322.1	1,238	5.5	1.5	23.76	1.77	31,411	2,188
4	0.248	0.272	397	0.87	12,127	1.417.7	1,455	5.5	1.5	23.76	1.77	33,682	2,571
5	0.257	0.303	75	0.99	9,273	1.020.1	1,048	5.5	1.5	23.76	1.77	24,236	1,852
6	0.229	0.251	805	0.70	6,544	751.0	711	5.5	1.5	23.76	1.77	17,842	1,256
Hot													
7	0.172	0.153	1,662	(1.27)	6,913	1,008.3	898	5.5	1.5	23.76	1.77	23,955	1,587
8	0.283	0.281	1,291	0.58	12,553	1,338.9	1,322	5.5	1.5	23.76	1.77	31,810	2,336
9	0.295	0.305	842	0.55	14,140	1,480.7	1,445	5.5	1.5	23.76	1.77	35,179	2,554
10	0.284	0.297	939	0.61	8,307	833.5	836	5.5	1.5	23.76	1.77	19,803	1,477
11	0.305	0.317	992	0.58	7,752	728.2	725	5.5	1.5	23.76	1.77	17,301	1,281

a. Friction factor = (thrust – stem end load-packing load)/disc force.

Table 3a. Six-inch-900-lb flex wedge data.

O/C[a]	A/DV test	Wyle run number	Shutoff calculated valve factor[b] (1,200 lb packing load)	Shutoff calculated valve factor[b] (observed packing load)	μ	Observed packing load (preliminary test stroke)	Position from stop (in.)	Thrust at position (lb)	dP (psi)	Bowl pressure P5 (psi)	Mean seat dia. (in.)	Stem dia. (in.)	Disc area ($in.^2$)	Stem area ($in.^2$)	Disc force (lb)	Stem end load (lb)
O	Initial	1	0.113 (unwedge)	0.10082 (unwedge)	—	1,600	0.24	4,945	1,347	0	5.6	1.5	24.63	1.77	33,177	0
—	—	—	0.127 (unseal)	0.11495 (unseal)	—	1,600	0.83	5,323	1,315	0	5.6	1.5	24.63	1.77	32,389	0
Cold																
C	1A	2B	0.170	0.17764	0.08877	1,000	0.49	7,587	1,079.0	1,056	5.6	1.5	24.63	1.77	26,576	1,866
C	2	3A	0.145	0.15122	0.06682	1,000	0.77	8,287	1,325.4	1,330	5.6	1.5	24.63	1.77	32,645	2,350
C	3	5A	0.191	0.19804	0.10866	1,000	0.75	8,537	1,155.7	1,075	5.6	1.5	49.26[a]	1.77	28,465	1,900
C	4	6A	0.174	0.18930	0.10015	800	0.75	7,623	1,068.1	1,043	5.6	1.5	24.63	1.77	26,307	1,843
C	5	7A	0.205	0.21668	0.12678	800	0.50	10,330	1,338.7	1,350	5.6	1.5	24.63	1.77	32,972	2,386
C	6	8A	0.320	0.30779	0.21452	1,600	0.11	14,291	1,361.0	1,343	5.6	1.5	24.63	1.77	33,522	2,373
O	6	8B	0.505 (unwedge)	0.52516 (unwedge)	—	600	0.3	14,258	1,223	1,223	5.6	1.5	24.63	1.77	30,123	2,161
—	—	—	0.205 (unseal)	0.21215 (unseal)	—	800	0.7	10,430	1,109	1,109	5.6	1.5	24.63	1.77	54,630	1,960
—	—	—	—	—	—	—	—	—	1,255.0	1,247	5.6	1.5	24.63	1.77	30,911	2,204
Hot																
C	7	9A	0.305	0.31424	0.22068	900	0.64	12,817	1,400.0	1,414	5.6	1.5	24.63	1.77	34,482	2,499
C	8	11A	0.340	0.35123	0.25587	800	0.58	15,410	974.0	966	5.6	1.5	24.63	1.77	23,990	1,707
C	9	12A	0.354	0.36257	0.26662	1,000	0.51	11,405	1,151	1,180	5.6	1.5	24.63	1.77	28,349	2,085
O	9	12B	0.628 (unwedge)	0.64218 (unwedge)	—	800	0.28	16,930	1,150	1,179	5.6	1.5	24.63	1.77	28,325	2,083
—	—	—	0.350	0.36387 (unseal)	—	800	0.73	9,023	986.0	973	5.6	1.5	24.63	1.77	24,285	1,719
C	10	—	0.357	0.36568	0.26956	1,000	0.43	11,600	—	—	—	—	—	—	—	—

a. O/C is opening/closing.

b. Pressure between seats; both seats loaded.

Valve factor (closing) = (thrust – stem end load-packing load)/disc force.
Valve factor (opening) = (thrust + stem end load-packing load)/disc force.

Table 3b. (continued)

O/C[a]	A/DV test	Wyle run number	Shut-off force	Wedging position	Maximum	Unwedging maximum force	Position	Unwedge ratio (shutoff)	Unwedge ratio (maximum
Cold									
O	Initial	1	—	—	—	–4,945	0.24	—	—
C	1A	2B	7,587	0.49	33,976	—	—	—	—
O	1	2C	—	—	—	–12,090	0.46	–1.594	–0.356
C	2	3A	8,287	0.77	43,098	—	—	—	—
O	2	3B	—	—	—	–9,510	0.57	–1.148	–0.221
C	3	5A	8,537	0.75	48,933	—	—	—	—
O	3	5B	—	—	—	–11,290	0.23	–1.322	–0231
C	4	6A	7,623	0.75	42,167	—	—	—	—
O	4	6B	—	—	—	–10,371	0.21	–1.360	–0246
C	5	7A	10,330	0.50	43,119	—	—	—	—
O	5	7B	—	—	—	–11,877	0	–1.150	–0275
Hot									
C	6	8A	14,291	0.11	39,932	—	—	—	—
O	6	8B	—	—	—	–16,419	0.3	–1.149	–0.411
C	7	9A	12,817	0.64	36,500	—	—	—	—
O	7	9B	—	—	—	–8,438	0.42	–0.658	–0231
C	8	11A	15,410	0.58	31,846	—	—	—	—
O	8	11B	—	—	—	–8,010	0.02	–0.520	–0252
C	9	12A	11,405	0.51	39,702	—	—	—	—
O	9	12B	—	—	—	–19,005	0.28	–1.666	–0.479
C	10	13A	11,600	0.43	37,250	—	—	—	—
O	10	13B	—	—	—	–7,678	0.32	–0.662	–0.206

a. O/C is opening/closing.

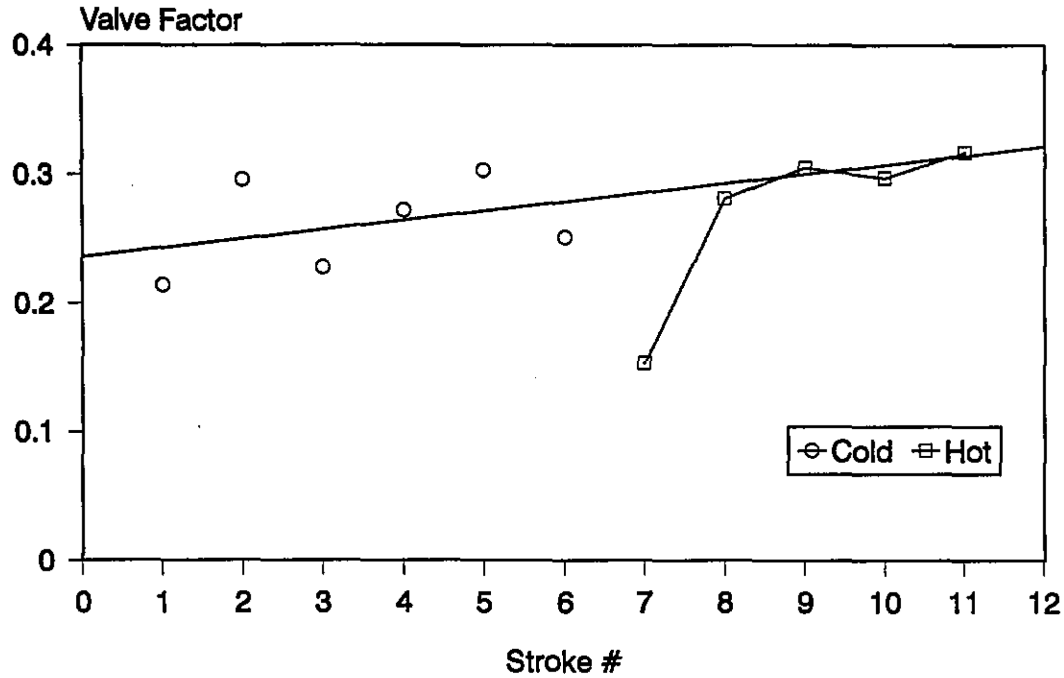

Figure 12. Six-in.-900-lb double disk gate valve data.

where

T_s = stem thrust required to *isolate* (lb)

T_p = packing friction, lb

T_e = stem end load, lb

vf = valve factor

A_s = seat area, based on mean seat diameter, in.^{2}

DP = differential pressure, psig

μ = friction factor (after accounting for seat angle)

Θ = seat angle in degrees, from a vertical plane.

Figure 13 presents graphical data on friction factor (i.e., after accounting for seat angle) versus stroke for both the hot and cold series of closure tests. The data presented below has been extracted from Table 3.

A/DV test	Valve factor	Friction factor, μ (accounting for 5-degree angle)
1A	0.178	0.089
2	0.151	0.067
3	0.198	0.109
4	0.189	0.100
5	0.217	0.127

The above data illustrate the significance of the 5-degree seat angle. The implications of this will be discussed later in the paper.

Visual inspections performed between high-energy closure strokes revealed no significant wear or damage, although slight burnishing of the disc and seat ring became apparent by the final high-energy closure stroke. The valve maintained zero leakage on all seat tests throughout the test program.

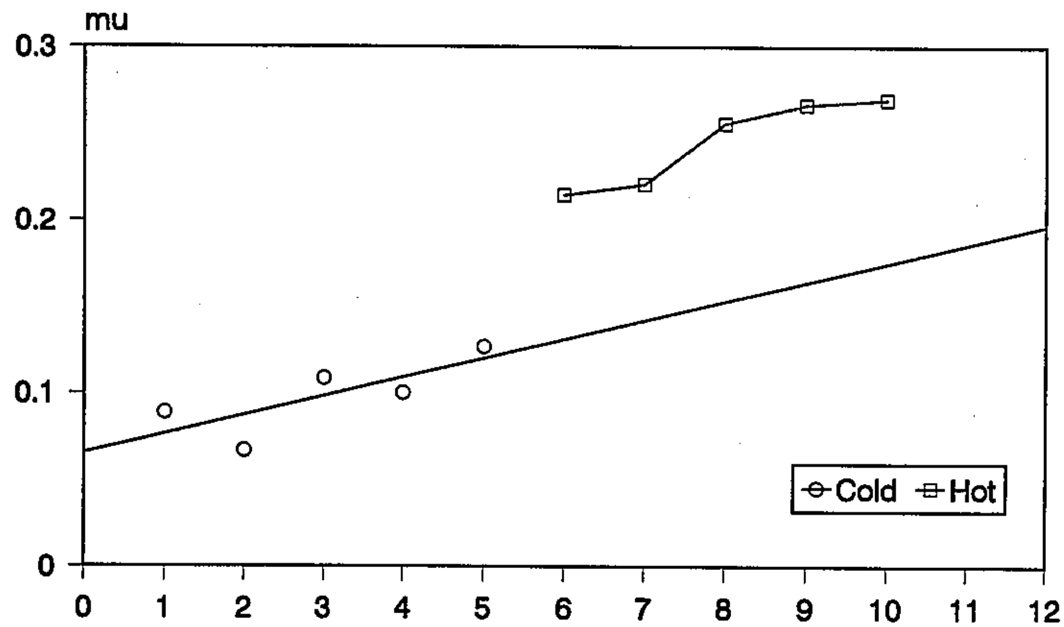

Figure 13. Six-in.-900-lb flex wedge gate valve data.

The stem thrust required to isolate, packing friction, differential pressure, and stem end load are based on experimental data gathered during each high-energy closure test. The seat area is based on the measured dimensions of the downstream valve seat. The actual packing friction is based on running loads measured during baseline testing performed without pressure before each high-energy closure test. There were no packing adjustments made between the open and closed cycles. Table 3 also presents valve factors based on packing friction established according to A/DV standard practice and the standard industry equation.

SUMMARY OF RESULTS

The results of the high-energy flow testing of the 6-in. double disc and the modified 6-in. flex wedge are as follows:

1. Both valves fully closed and isolated flow under all high-energy flow tests.

2. Neither valve sustained any damage throughout the test programs; only minor wear and superficial scuffing of the seating surfaces were observed. In contrast, previous industry testing of A/DV's standard 6-in.-900-lb design under similar conditions had resulted in notable damage to the valve. The improved performance of the flex wedge test valve was attributed to the modified guide design.

3. Seat leakage tests performed after each high-energy closure demonstrated the capability of both valves to maintain zero, or near zero, seat leakage throughout the entire test program.

4. The closing valve factors for the double disc valve, calculated at the point of flow isolation, ranged from 0.214–0.303 for the cold (i.e., ambient temperature) tests and from 0.153–0.317 for the hot tests. There is no need to adjust these values for seat angle, because the seat angle of the double disc gate valve is 0 degrees.

5. The closing valve factors for the flex wedge, calculated at isolation using the standard industry equation, were 0.18–0.22 for cold conditions (i.e., ambient temperature) and 0.31–0.37 for hot conditions. After accounting for the 5-degree seat angle, the calculated friction factor at isolation was 0.07–0.13 for cold conditions and 0.21–0.27 for hot conditions. The significant differences between the valve factors and the friction factors (i.e., after accounting for seat angle) illustrate the importance of maintaining consistency when calculating valve thrust requirements. It is important for the user to understand how a given valve factor of friction factor was derived. For example, the use of a valve factor calculated from experimental data, without consideration of seating angles, may yield overly conservative results when used in an equation that does consider seating angles. Conversely, the use of an angle-adjusted valve factor (i.e., friction factor) in the standard industry equation may cause valve thrust requirements to be underestimated.

6. Neither of the test valves exhibited indications of thermal binding for the specific conditions under which the valves were tested. However, A/DV notes that this does not necessarily ensure that either valve, and particularly the flex wedge, could not become thermally bound under actual piping system conditions and loadings.

OBSERVATIONS

A/DV has made several observations as a result of the test programs and subsequent implementations of the flex wedge guide modification. These observations are as follows:

1. Based on the flex wedge testing, the hot tests exhibited higher operating thrust requirements and higher associated valve factors than the cold tests. However, the valve factors for the double disc valve did not appear to vary significantly with temperature.

2. The calculated valve factor for the flex wedge valve trended higher throughout both the cold and hot testing and approached a plateau for the hot testing. There was a slight trend towards higher valve factors throughout the double disc tests, although it was much less apparent than with the flex wedge.

3. The valve factor did not appear to be pressure dependant (i.e., higher at lower differential pressures) for either the double disc or flex wedge valve.

4. Accounting for the 5-degree seat angle of the flex wedge valve resulted in friction factors that were less than the valve factors calculated using the standard industry equation.

5. Unwedging forces for the flex wedge valve ranged from 20.6 to 47.9% of the previously applied closing thrust.

6. For opening tests performed with differential pressure, the unwedging forces for the flex wedge valve were typically greater than the unsealing forces.

CONCLUSIONS

Based on the results and observations of the two test programs, A/DV has concluded that both the double disc and modified flex wedge designs are capable of satisfactory performance under high-energy flow conditions. The double disc design represents a viable option for new installations or retrofits in systems potentially subject to such severe conditions. The modified flex wedge design represents an option for utilities wishing to improve the performance of existing valves that might be susceptible to damage and malfunction under high-energy flow conditions. It should be understood that such modifications are limited to those valve sizes that provide the physical space necessary for access and installation of the modified parts. In general, A/DV believes that valve sizes of 6-in. and larger are candidates for the modification. As an alternative, the features of the modified flex wedge design could be

incorporated into new flex wedge valves, without restrictions on valve size.

The lack of damage to either test valve and the repeatability of the results indicate that the thrust requirements for double disc and similarly modified gate valves can be estimated with reasonable accuracy. However, because of the inaccuracies of the instrumentation involved and the desire for additional thrust margin to ensure closure and initial wedging, it may be desirable to employ slightly higher valve factors when performing actuator sizing for valves to be installed in high-energy flow systems. As a result of the test programs, A/DV now employs a valve factor of 0.4 for all new flex wedge valves, new double disc gate valves, and modified flex wedge valves that are identified as being subject to high-energy flow conditions. In addition, special design considerations regarding guide clearances and guide materials are deemed appropriate for flex wedge gate valves manufactured for use under high-energy flow conditions.

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ACKNOWLEDGMENTS

The author wishes to thank T. C. Bartlett for reduction of data and for graphic plots, F. A. Bensinger for preparation of the test plans, J. J. Chappell (A/DV) for engineering support, W. G. Knecht (A/DV) for technical support, L. J. Millsaps and T. Parks (Wyle Laboratories) for project engineering, and A. C. Wheal for the valve figures.

The Stem Factor Challenge[a]

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ABSTRACT

One of the most important challenges that still needs to be met in the effort to understand the operation of motor-operated, rising-stem valves is the ability to determine stem factor throughout the valve's load range. The stem factor represents the conversion of operator torque to stem thrust. Determining the stem factor is important because some motor-operated valves (MOVs) cannot be tested in the plant at design basis conditions. The ability of these valves to perform their design basis function (typically, to operate against specified flow and pressure loads) must be ensured by analytical methods or by extrapolating from the results of tests conducted at lower loads. Because the stem factor tends to vary in response to friction and lubrication phenomena that occur during loading and wedging, analytical methods and extrapolation methods have been difficult to develop and implement. Early investigations into variability in the stem factor tended to look only at the tip of the iceberg; they focused on what was happening at torque switch trip, which usually occurs at full wedging. In most stems, the stem factor is better (lower) in the wedging transient than before wedging, so working with torque switch trip data alone led many early researchers to false conclusions about the relationship between stem factor and load. However, research at the Idaho National Engineering Laboratory (INEL) has taken a closer look at what happens during the running portion of the closing stroke along with the wedging portion. This shift in focus is important, because functional failure of a valve typically consists of a failure to isolate flow, not a failure to achieve full wedging. Thus, the stem factor that must be determined for a valve's design basis closing requirements is the one that corresponds with the running load before wedging.

For a given stem/stem-nut combination and for a given value of torque, the only variable in the conversion of torque to thrust is the coefficient of friction at the stem/stem-nut interface. Results of tests conducted at the INEL indicate that the stem/stem-nut coefficient of friction determined at test conditions that are less severe than design basis conditions can provide consistent, useful information about the coefficient that can be expected at design basis conditions. This result provided the insight for the initial development of two straightforward methods for determining the stem factor for a valve that cannot be tested at design basis conditions. Both methods require that torque and thrust be measured directly. The first method (we call it the *threshold method*) would require that a specified minimum stem load (usually a lower load than the design basis load) be imposed on the valve

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stem during the running portion of the stroke before wedging begins. The coefficient of friction determined from such a test could be used directly to calculate the valve's design basis torque requirement. The second method (the *fold line method*) estimates a bounding coefficient of friction from the wedging load in a test with a running load below that required for the threshold method. The threshold method is the more accurate of the two, but a stem factor determined using either of these methods is likely to be more accurate and entail less unwanted conservatism than any of the default values being used or any available extrapolation method. This paper explains both methods and provides the research results that support them.

INTRODUCTION AND BACKGROUND

The Idaho National Engineering Laboratory (INEL), under the sponsorship of the U.S. Nuclear Regulatory Commission (USNRC), is performing research in support of the USNRC's efforts regarding the implementation of generic letter (GL) 89-10 "Safety-Related Motor-Operated Valve Testing and Surveillance." GL 89-10 recommends the reevaluation of the design basis requirements and the control switch settings of safety-related motor-operated valves (MOVs). In response, the nuclear industry has found it necessary to also reevaluate the methods used to assure valve operability.

Years ago, valve and motor operator manufacturers developed a number of design rules by which they calculated valve loads, sized the operators for the valves, and recommended settings for the control switches. When valves were initially installed in nuclear power plants, the design basis loads for these valves were typically defined using the manufacturers' design rules. As time passed, a number of nuclear plant transients occurred during plant operation that required the installed valves to perform their design basis function. Far too frequently, the valves failed to function as designed, because the operator motor was not powerful enough or the control switches were not set high enough to overcome the actual design basis loads. After one specific instance where auxiliary feedwater valves failed to open after being closed, the USNRC issued Bulletin 85-03, which required the utilities to reevaluate a specific number of safety-related valves in select systems, an average of about 25 valves per plant.

Later, the USNRC prioritized Generic Safety Issue (GSI) 87 "Failure of the HPCI Steam Line without Isolation." The HPCI (high pressure coolant injection) steam line is a supply line that communicates directly with the reactor primary steam system in boiling water reactor plants and runs through the containment wall. In the event of a guillotine break in the line outside of the containment, the plant's ability to maintain the containment boundary depends on the ability of at least one of two isolation valves to close against the flow load.

Two other systems include isolation valves with similar design basis requirements: the reactor water coolant cleanup (RWCU) system and the reactor core isolation cooling (RCIC) system. Thus, these valves are included in the GSI 87 concern.

FULL-SCALE VALVE TESTING

The USNRC requested the INEL to investigate how these valves were originally qualified to perform their design basis function. We found that very little testing had been performed at high flow rates. At the conclusion of our investigation, we recommended to the USNRC that a full-scale test program be performed as the only conclusive method to ensure that the valve closing requirements were correctly defined.

We conducted our first full-scale test program in 1988 at Wyle Laboratory's Huntsville facility. Two 6-in. valves representative of those in the RWCU system were tested at various flow and pressure conditions. We conducted our second full-scale test program in 1989 at the Kraftwerk Union (KWU) facilities near Frankfurt, Germany.

The KWU facilities were the only available facilities in the world that were big enough to test the 10-in. HPCI steam valves that were included in this test program. In addition to the three 10-in. valves, three 6-in. valves were tested at a number of pressure and flow conditions. Together, these two test programs produced several important findings:

- Some valves are susceptible to internal damage when subjected to high flow and pressure loads. Disc-to-guide clearances are the main factor that determines this susceptibility. This issue still needs to be resolved by valve researchers.

- Predictions based on the design rules the industry was using at the time to size motor operators were not conservative. The 0.3 disc factor in the industry's disc load equation was not high enough, and the equation itself did not adequately model valve operation. In response to this result, we developed an improved model and a correlation for conservatively estimating stem thrust requirements for valves in medium to high flow applications. That work was reported in Steele et al. (1993).

- For a given torque switch setting, the thrust at torque switch trip tends to be higher in tests with low running loads (static tests) than in tests with flow and pressure loads. In effect, the stem factor in the equation varies with the load (stem factor equals torque divided by thrust). This variability in the stem factor is the subject of this paper.

Figure 1 shows the stem force measurements for four tests of the same valve at four different flow and pressure loads. The torque switch setting was the same in all four tests. The long vertical line at the end of the low-load trace indicates the sudden increase in stem thrust at wedging; the running load before wedging was fairly low. In the design-basis-load test, the valve did not seat.

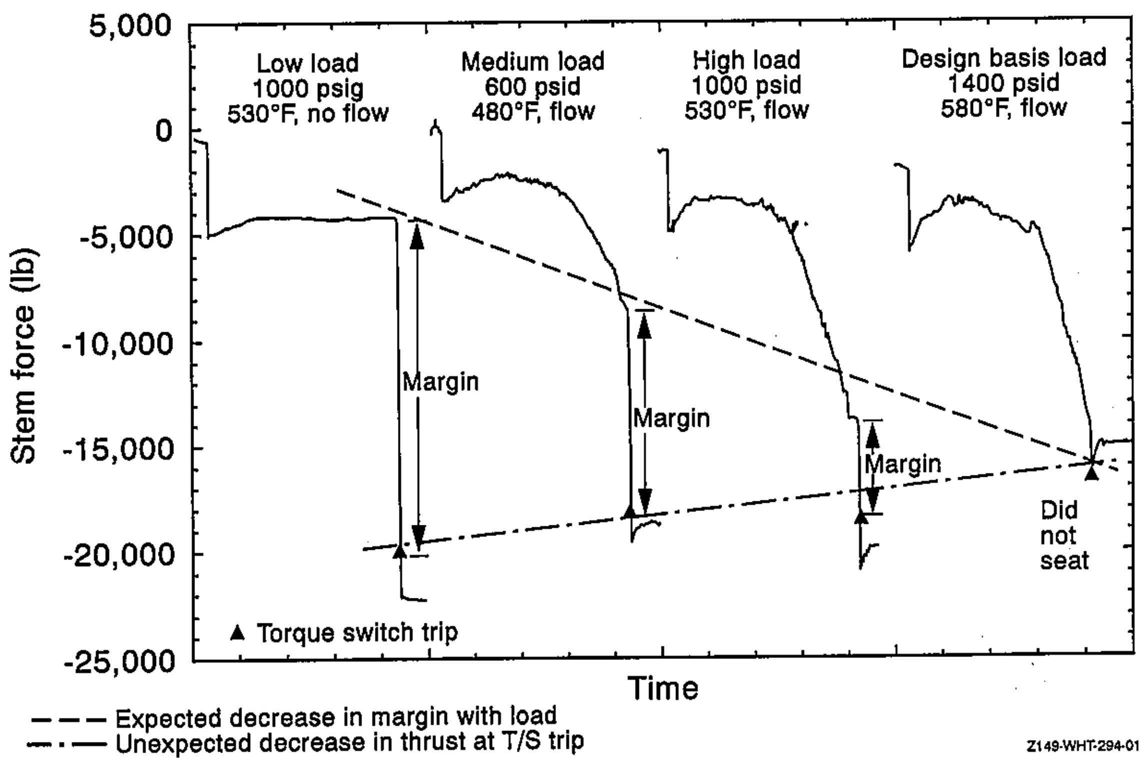


Figure 1. Four valve tests at the same torque switch setting. At the design basis flow and pressure loads, the valve failed to completely close.

Note that in the low-load test, the thrust measured at torque switch trip is considerably higher than in the design-basis-load test. This change in the thrust is due to a change in the stem factor. We call this phenomenon *load-sensitive behavior.* In some circles it is known as the *rate-of-loading effect.* We had expected that the margin at wedging would decrease with load (as indicated in Figure 1), but we had not expected the thrust to decrease at a given value of operator torque. The failure to close in the design-basis-load test demonstrates the seriousness of the problem: during an in situ test conducted by a utility at low-load conditions, the stem thrust measured at torque switch trip may be sufficient to overcome the calculated design basis load; however, the utility still has no assurance that the valve will fully close at design basis conditions. Changes in the stem factor must be accounted for.

TESTING ON THE MOVLS

In preparation for the full-scale flow tests conducted in Germany, we built an MOV load simulator (MOVLS). The purpose of the MOVLS was to test MOV instrumentation techniques and data acquisition methods. After the tests in Germany were completed, we further developed the MOVLS so we could use it to conduct additional tests to address the stem factor issue. The MOVLS, shown in Figure 2, uses motor-operators, yokes, stems, and stem nuts just as they are assembled on the valves. The flow load is simulated as the valve stem compresses a hydraulic cylinder that discharges to an accumulator. The specific valve load profile is controlled by the water level and gas charge in the accumulator, which is set up before testing. The MOVLS is instrumented to take all of the measurements (direct and indirect measurements) that are used by the commercially available valve diagnostic systems, as well as a few measurements unique to the MOVLS. Stem thrust is measured by a load cell mounted in the stem, and stem torque is measured by a calibrated torque reaction arm mounted on the stem. The simulator also has a torque cell mounted between the electric motor and the gear box to provide a direct measurement of real time motor torque. Motor speed is also measured directly.

Figure 3 shows the stem force traces for three tests conducted on the MOVLS at the same torque switch setting but at different simulated valve loads. The same load-sensitive behavior we observed in full-scale flow testing (Figure 1) is evident here.

Stem factor for a rising stem MOV is defined as the operator output torque (or stem torque) divided by the stem thrust. Figure 4 is a simplified diagram showing the important mechanical components involved in the conversion of torque to thrust. Except for very small changes (due to worm/spline friction, for example), an operator at a given torque switch setting will deliver a specific amount of torque to the stem nut. The variables that affect the conversion of torque to thrust are the stem diameter, the stem pitch and lead, and the friction at the stem/stem-nut interface. For a particular valve with a particular stem and stem nut, the only variable in the stem factor equation (assuming a constant value for the torque) is the coefficient of friction between the stem and the stem nut; the other components of the equation are constants. Thus, any change in the relationship between operator torque and stem thrust is the result of a change in the stem/stem-nut coefficient of friction.

We recently conducted a test program on the MOVLS that included eight typical valve stems with acme threads. Seven of the stems were provided by nuclear valve suppliers, and the eighth stem was built by Teledyne Engineering as part of their Smart Stem™ development program. (The Smart Stem™ is a valve stem that has been strain gaged and calibrated to measure both thrust and torque directly in the stem.) Three different sizes of Limitorque motor-operators were used in the test program. The technical details of the test hardware can be found in Table 1. The test program included two tests of each stem at each of three different torque switch settings. In each case, the first test was a static test, simulating a valve closure against a packing load and a stem

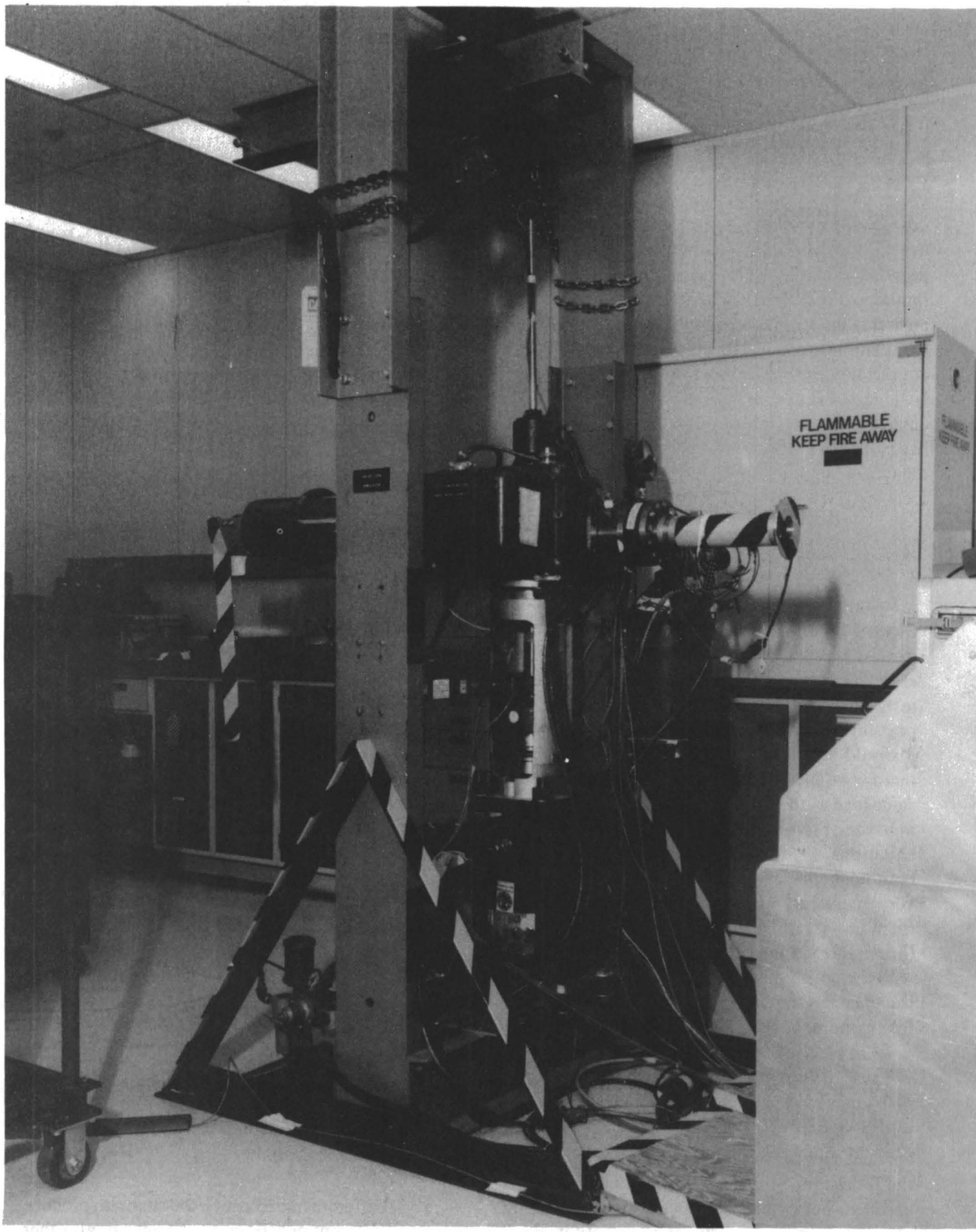


Figure 2. The INEL's load simulator for testing valve stems (the MOVLS).

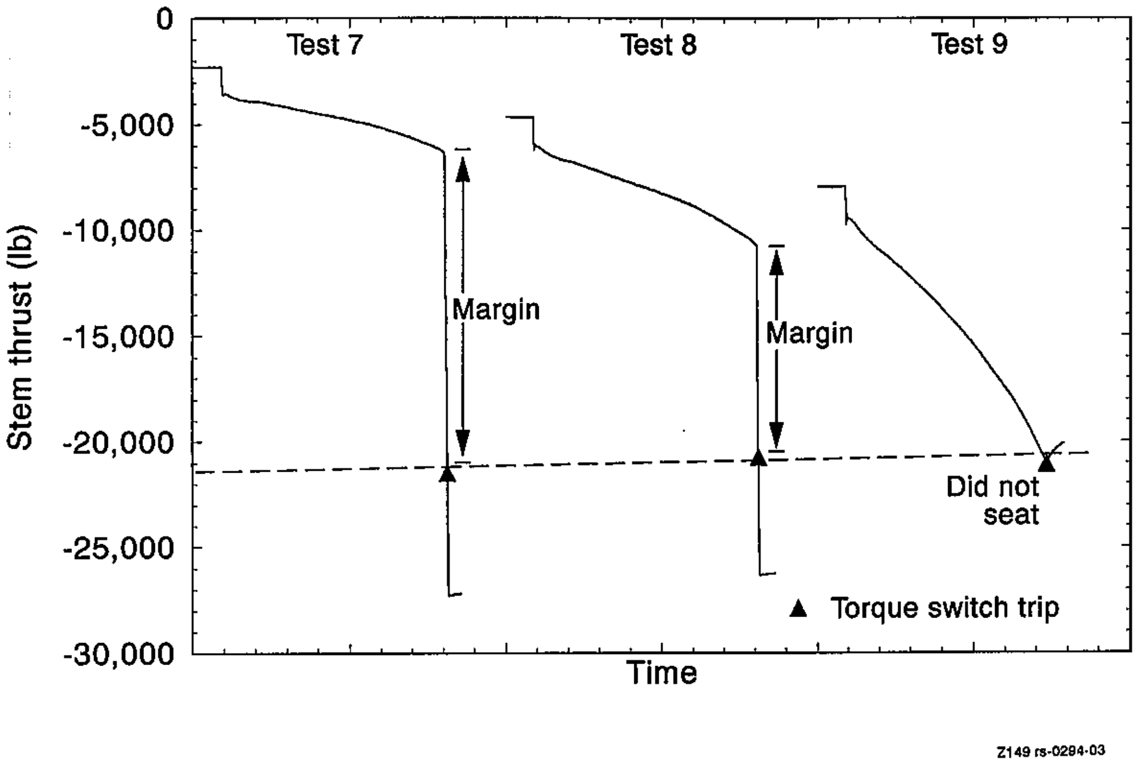


Figure 3. Load-sensitive behavior is simulated on the MOVLS.

rejection load, and the second test was a dynamic test, simulating valve closure against flow and pressure loads. Each stem was loaded in the range it would be expected to experience in the plant; that is, we did not load the 1.25-in. stems to the same load as the 2-in. stems. The test results were analyzed using the industry's power thread equations. (We have reviewed these equations both mathematically and from the evaluation of very accurately measured test results and found them to be valid.) By using the measured stem thrust and the measured stem torque, together with the power thread equations, we were able to calculate the stem/stem-nut coefficient of friction for any point of interest during the closing stroke.

The entire test sequence was conducted with two popular lubricants: SWEPCO Moly 101 and Exxon Nebula EP-1. The purpose here was not to conduct a lubricant study, but simply to find out if the lubricant influenced the results. The stems and stem-nuts were cleaned very carefully during changes from one lubricant to the other. Each stem and stem-nut was washed in three different solvent baths, the last one using previously unused solvent. The stems were also subjected to a light abrasive (Scotch Brite™) surface scrubbing between the second and third baths to ensure that the previous lubrication film was broken. Several stems were lubricated, cleaned, lubricated with the other lubricant, then recleaned, relubricated with the original lubrication, and retested. The results from repeat tests with the same lubricant were compatible with the results from the first round of tests.

Testing on the MOVLS has produced three important findings regarding the stem/stem-nut coefficient of friction:

- The coefficient of friction varies with changes in the load. This is true of both the running portion and the wedging portion of the closing stroke.

- Different lubricants on the stem can produce different coefficients of friction, all other conditions being the same.

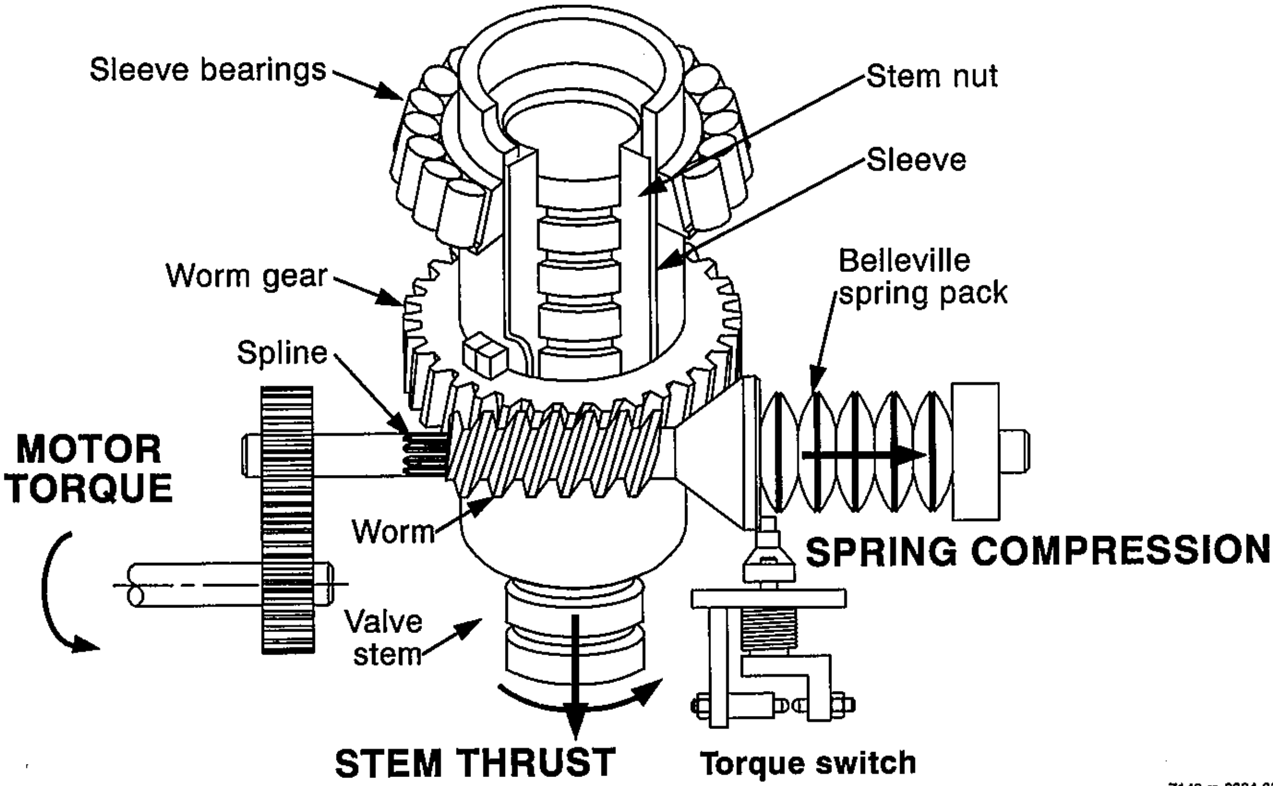


Figure 4. Simplified diagram showing the key components of a Limitorque motor-operator.

- Each individual stem/stem-nut combination is unique, with its own particular coefficient of friction profile. Some stems are more likely to exhibit load-sensitive behavior than others.

These findings underscore the difficulty valve researchers have experienced in attempting to analytically predict the coefficient of friction for any given valve; no two valves (even valves of the same size and model) can be expected to behave exactly alike, and the same valve can behave differently, depending on either the lubricant or the load. One way to address this difficulty is to simply assign a value for the coefficient of friction that is high enough to cover any worst possible case; for example, Limitorque's sizing manuals recommend a value of 0.2. The problem with this solution is that in most cases such a value is excessively conservative, and in many cases it would force the utilities to unnecessarily replace valve motor operators with larger ones. It would also subject yokes and valve internals to unnecessarily high loads, with a potential to contribute to fatigue failure. Another possible solution is to test all valves in situ at their design basis conditions so that no analytical predictions would be needed. In many cases, however, such testing is simply impossible. Some utilities have attempted to address this issue by testing a valve at static conditions (packing load only), deriving a coefficient of friction from the results at torque switch trip, and using that value in calculations to predict the operator torque needed for the design basis case. As our test results have shown, load-sensitive behavior makes this method unreliable.

Although the coefficient of friction derived from a static test tends to differ significantly from the design basis coefficient, results from testing on the MOVLS show that it is possible to get reliable, useful information from static tests and low-load tests. The following discussion proposes two new methods that use data from tests conducted at conditions less severe than design basis conditions to either predict or bound the design basis coefficient of friction. The methods are based on

Table 1. Technical data for eight stems and three operators used in the MOVLS test program.

	SMB-00	SMB-0				SMB-1		
Operator stem	S7	S1	S2	S3	S8	S4	S5	S6
Motor set ratio	22/43	37/35	37/35	25/47	37/35	21/51	32/40	27/45
Overall ratio	87.8	34.96	34.96	69.56	34.96	82.55	42.50	56.64
Running efficiency	0.50	0.55	0.55	0.50	0.55	0.50	0.50	0.50
Stall efficiency	0.50	0.55	0.55	0.50	0.55	0.50	0.50	0.50
Pull out efficiency	0.40	0.40	0.40	0.40	0.40	0.40	0.40	0.40
Application factor	0.90	0.90	0.90	0.90	0.90	0.90	0.90	0.90
Motor rpm	1800	1800	1800	1800	1800	1800	1800	1800
Motor rated torque (ft-lb)	5	25	25	25	25	60	60	60
Motor stall torque (ft-lb)	6.6	29	29	29	29	67	67	67
Operator rated torque (max) (ft-lb)	250	500	500	500	500	850	850	850
Operator torque (motor rated) (ft-lb)	220	481	481	870	481	2477	1275	1699
Operator torque (motor stall) (ft-lb)	290	558	558	1009	558	2765	1424	1897
Stem diameter (in.)	1.25	1.5	1.75	1.25	1.75	2	2.5	2.125
Stem pitch	0.333	0.25	0.25	0.25	0.25	0.333	0.333	0.250
Stem lead	0.667	0.25	0.25	0.5	0.25	1.000	0.667	0.500
Stem area (in.^{2})	0.567	0.540	0.638	0.442	0.638	0.960	1.222	0.785
Stem force (max rated (lb_f)	14000	24000	24000	24000	24000	45000	45000	45000
Stem force (rated torque) (lb_f)	15308	40642	35941	35198	35941	32956	35056	42958
Stem force (motor rated (lb_f)	13441	39073	34554	61209	34554	96018	52585	85876
Stem force (motor stall) (lb_f)	17742	45325	40082	71003	40082	107220	58719	95895

the results of our testing of eight stems. We believe that with additional validation, these two methods will represent a major breakthrough in stem factor research.

THE THRESHOLD METHOD

The appropriate coefficient of friction to use in a design basis calculation is one that corresponds with the highest stem load during running (throughout this discussion, the word *running* is used to refer to the running portion of the closing stroke, and the word *wedging* is used to refer to the wedging portion). In valves that exhibit what we call a linear response, this highest running load occurs just before wedging, at a point that corresponds with flow isolation. This is the point where the entire area of the disc is exposed to the maximum differential pressure. Some valves exhibit what we call a nonlinear response. In these valves, the highest stem force occurs before flow isolation, a phenomenon that is due to internal valve geometry and flow and pressure effects. In either case, the coefficient of friction at the highest stem force before wedging is the one that is important. Figure 5 is a scatter plot of such data from tests on the MOVLS with Moly 101 lubricant on the eight stems. The MOVLS does not simulate nonlinear behavior, so all these data are from linear responses. Running coefficients of friction just before wedging at three torque switch settings are included for all eight stems. Repeated tests for some stems are also included. The coefficients of friction were calculated using direct stem torque and stem thrust measurements.

In Figure 5, coefficient of friction is plotted against stem thrust. Thrust is not the best variable to use in a study of stem friction with stems of different sizes and thread geometries. A 10,000-lb thrust is a very different condition for a 1-in. stem versus a 2-1/2-in. stem. A more appropriate variable that includes thrust and also normalizes the effect of thrust on stems of various sizes is stem

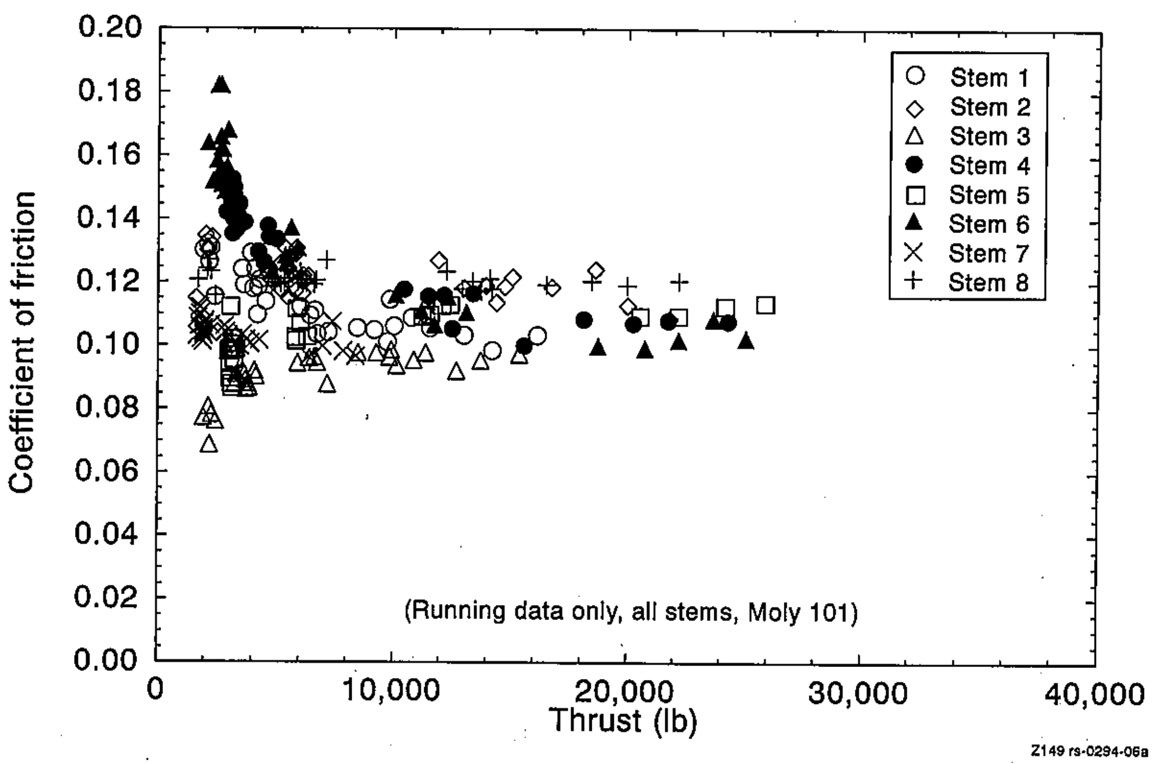


Figure 5. Coefficient of friction versus stem load; the data scatter decreases as the load increases.

thread pressure. We calculated thread pressure (in pounds per square inch) using the measured thrust and a nominal area based on one stem thread revolution. Use of this nominal thread area is consistent with standard practices in stress calculations for threaded fasteners. Coefficient of friction is plotted against thread pressure in Figure 6. Although the envelope of the data hasn't changed dramatically, the relative position of data for the various stems has shifted somewhat.

A careful review of Figure 6 shows that there is a lot of scatter in the data at low thread pressures, but the coefficient of friction traces for the individual stems flatten out above a thread pressure of about 10,000 psi. This is the key to what we call the *threshold method* of predicting the design basis coefficient of friction for a given stem. In practice, if a particular valve test can be set up to yield running data at stem pressures above the stem pressure threshold, the coefficients of friction derived from the test can be used directly in the calculation to determine the valve's design basis torque requirements. For many valves, the load that achieves this threshold stem pressure is significantly lower than the design basis load; for example, it might consist of the packing load combined with a pressure load (stem rejection load), or it might be a combination of the packing load, a pressure load, and a low flow load.

The data shown in Figure 6 are from tests with the stems lubricated with Moly 101 grease. All of the stems we tested on the MOVLS were also tested with EP-1 grease, and a similar analysis of those data has been performed (see Figure 7). From that analysis, we expect the threshold method to work as well with EP-1 as it does with Moly 101. However, the coefficients of friction obtained from testing of a given stem are likely to be different for different lubricants. With most stems, EP-1 produced slightly higher friction coefficients than Moly 101, but some stems performed better with EP-1 than with Moly 101. Figure 8 compares Moly 101 data with EP-1 data from testing of Stem 3. Note that although the coefficients of friction are different for the two

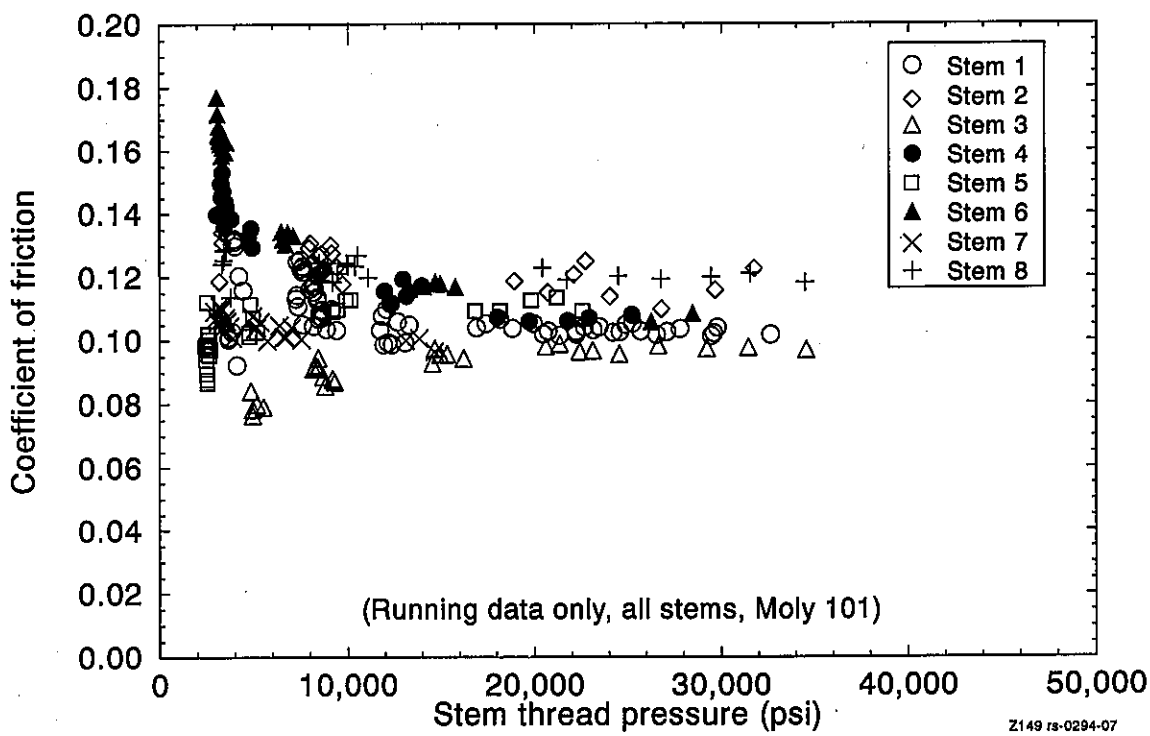


Figure 6. Friction data plotted against a normalized load, with coefficient of friction versus stem thread pressure for tests with Moly 101 lubricant.

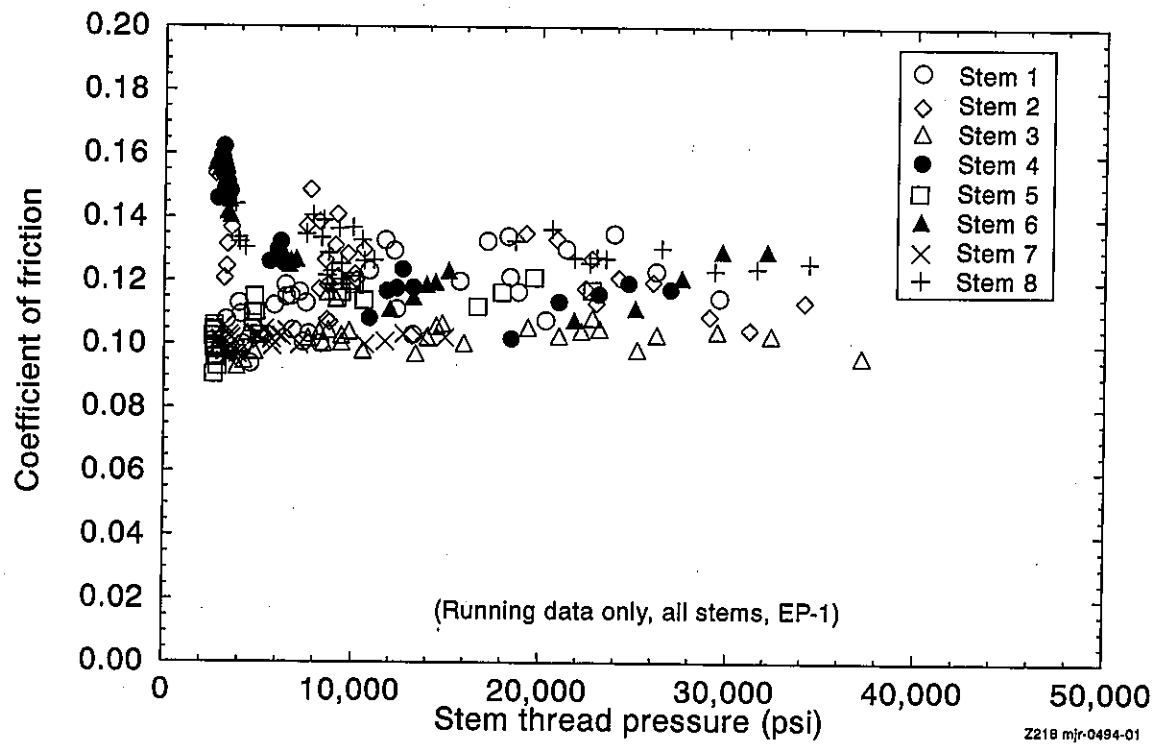


Figure 7. Coefficient of friction versus stem thread pressure for tests with EP-1 lubricant.

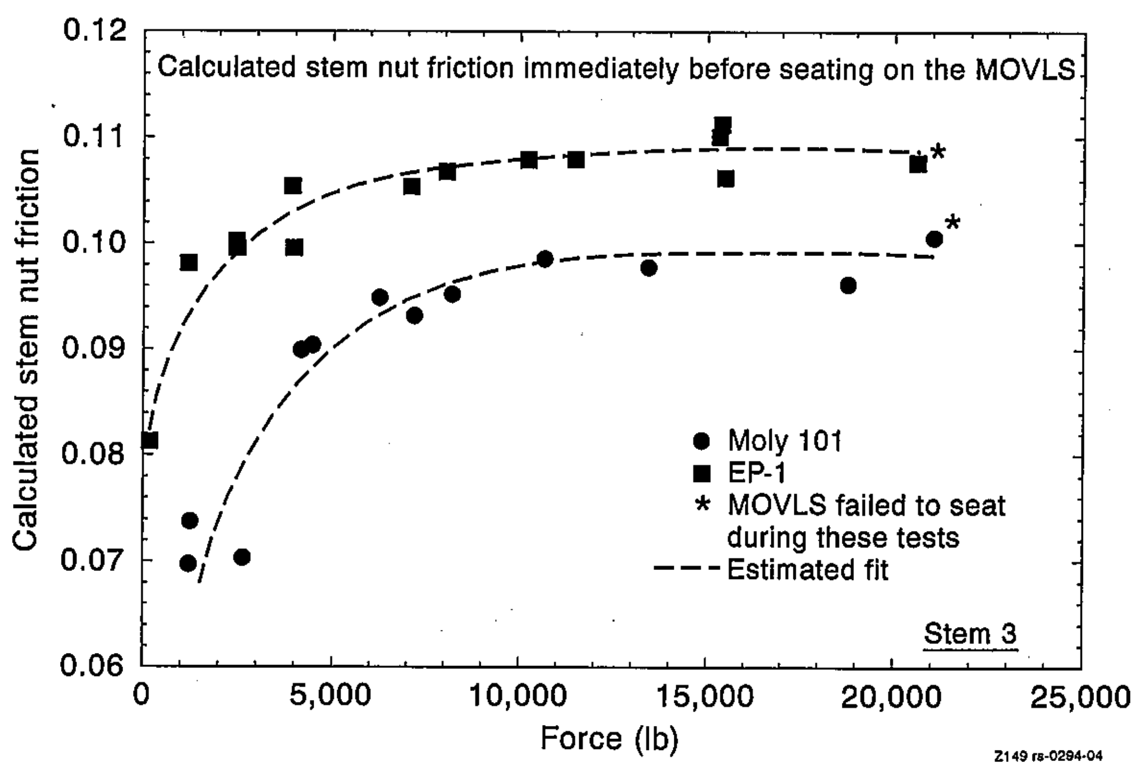


Figure 8. Coefficient of friction versus load, comparing two greases on the same stem.

greases, the results are consistent with our understanding of the threshold method; both traces reach a plateau, and the plateau occurs at about the same stem force threshold.

The work in developing the threshold method is not done yet. This method is based on empirical data from a sample of eight valve stems. Before the threshold method is put to use, it needs to be validated. To do this, it will be necessary to increase the sample size to ensure that the method is consistent for a larger population of stems. In addition, more data are needed to determine the exact threshold for stems in general, especially for smaller stems.

However, the threshold method already appears to be very promising for those valves where a partial differential pressure test can be run. It is certainly an improvement over the use of default coefficients of friction. The threshold method is consistent with the observation that the coefficient of friction for most stems is somewhere near 0.12, as compared with the best default value of 0.15 or the more conservative default value of 0.2.

In summary, the purpose of the threshold method is to provide a stem factor that can be used in calculations to determine a valve's design basis requirements. The threshold method is based on the observation that above a certain load threshold, the stem factor during the running portion of a closing stroke does not continue to change as the load increases. Use of the threshold method would entail an in situ test. The load in the test would have to be sufficient to produce a stem thread pressure (based on the area of one revolution of stem thread) of at least about 10,000 psi at or before flow isolation. For some valves, a packing load combined with a stem rejection load (pressure load) might be enough; for others, a flow load might be necessary. Continuous measurement of torque and thrust would be needed in order to calculate the stem factor. Indirect measurements of operator torque (torque based on spring pack measurements) might be sufficient, provided that measurement and calibration inaccuracies are accounted for. Measurements taken at the peak thrust at or before flow isolation would be used to determine the stem factor. That value, plus a small margin to account for possible lubrication degradation, bounds the stem factor expected at design basis conditions. The threshold method would be useful for valves that can be tested in situ against pressure and moderate flow loads but cannot easily be tested at their design basis loads. The results from testing of a particular valve would not be applicable to other valves.

THE FOLD LINE METHOD

Predicting stem factor from the results of a packing load test (static test) is very attractive because this is the best that can be done in situ for many valves. Such a prediction has been difficult to develop because the coefficient of friction behaves differently when the valve is wedging as contrasted to that while running. It is also difficult because each stem/stem-nut combination behaves uniquely. This is generally shown in Figure 6 in the overall responses of the individual stems. Figure 9 shows examples of running coefficients of friction that make up Figure 6. These are representative responses derived from the running portion of the closing stroke, just before wedging, for three stems. These three response trends represent all of the observed trends in the running coefficients of friction: increasing with load, decreasing with load, and relatively constant with load.

Consistent with what we know about load-sensitive behavior, the coefficient of friction at torque switch trip in a static test is lower than the running coefficient of friction just before wedging. This phenomenon is probably the result of lubrication performance at the stem/stem-nut interface. This difference is shown in Figure 10. Note that although the coefficients of friction are lower, the single data points for torque switch trip in the packing load tests generally follow the trend observed in the running data: increasing, decreasing, or a nearly flat response. This insight provided the first clue that there might be a link between static wedging and threshold running friction coefficients.

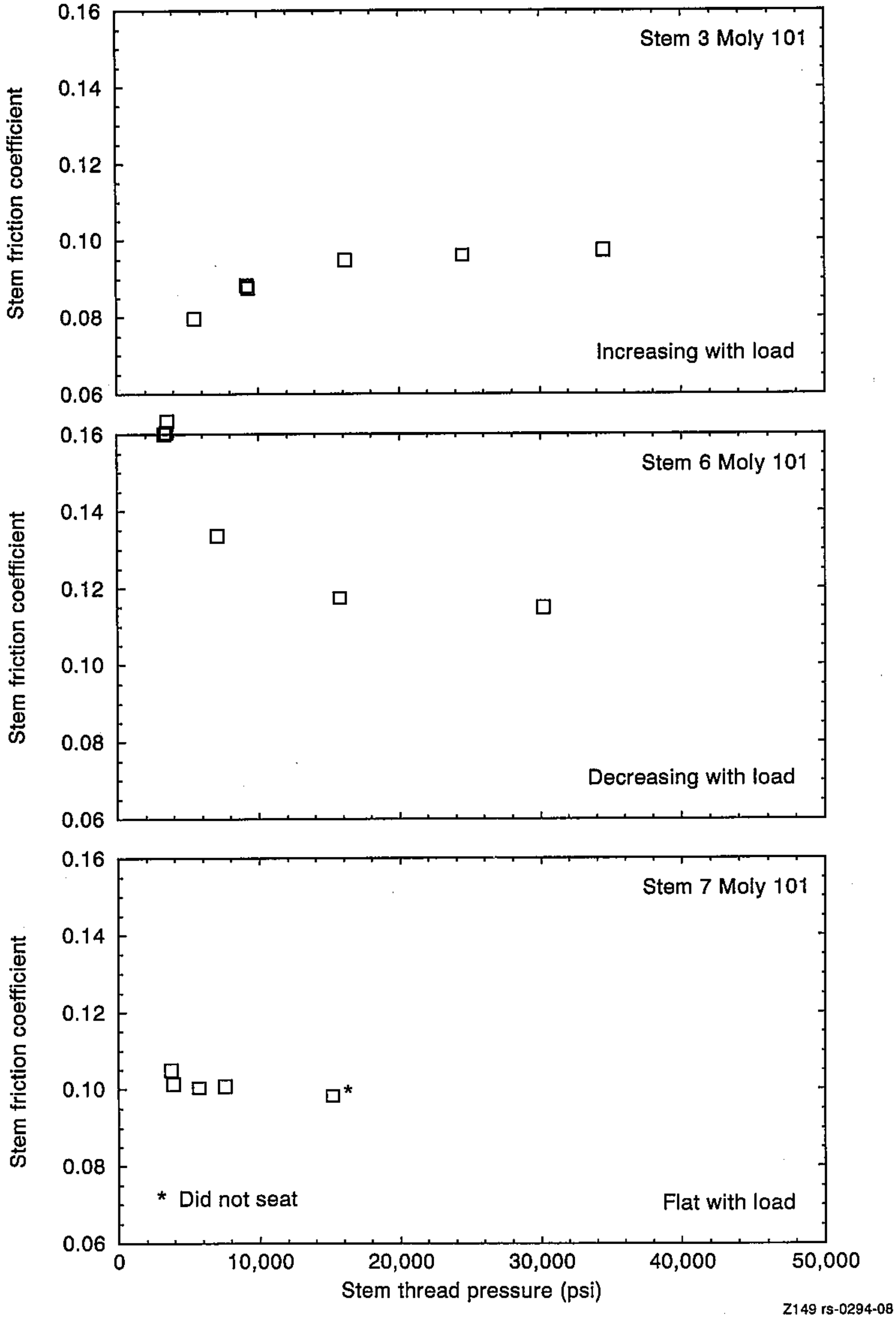


Figure 9. Running coefficients of friction (just before wedging) versus load for three representative stems.

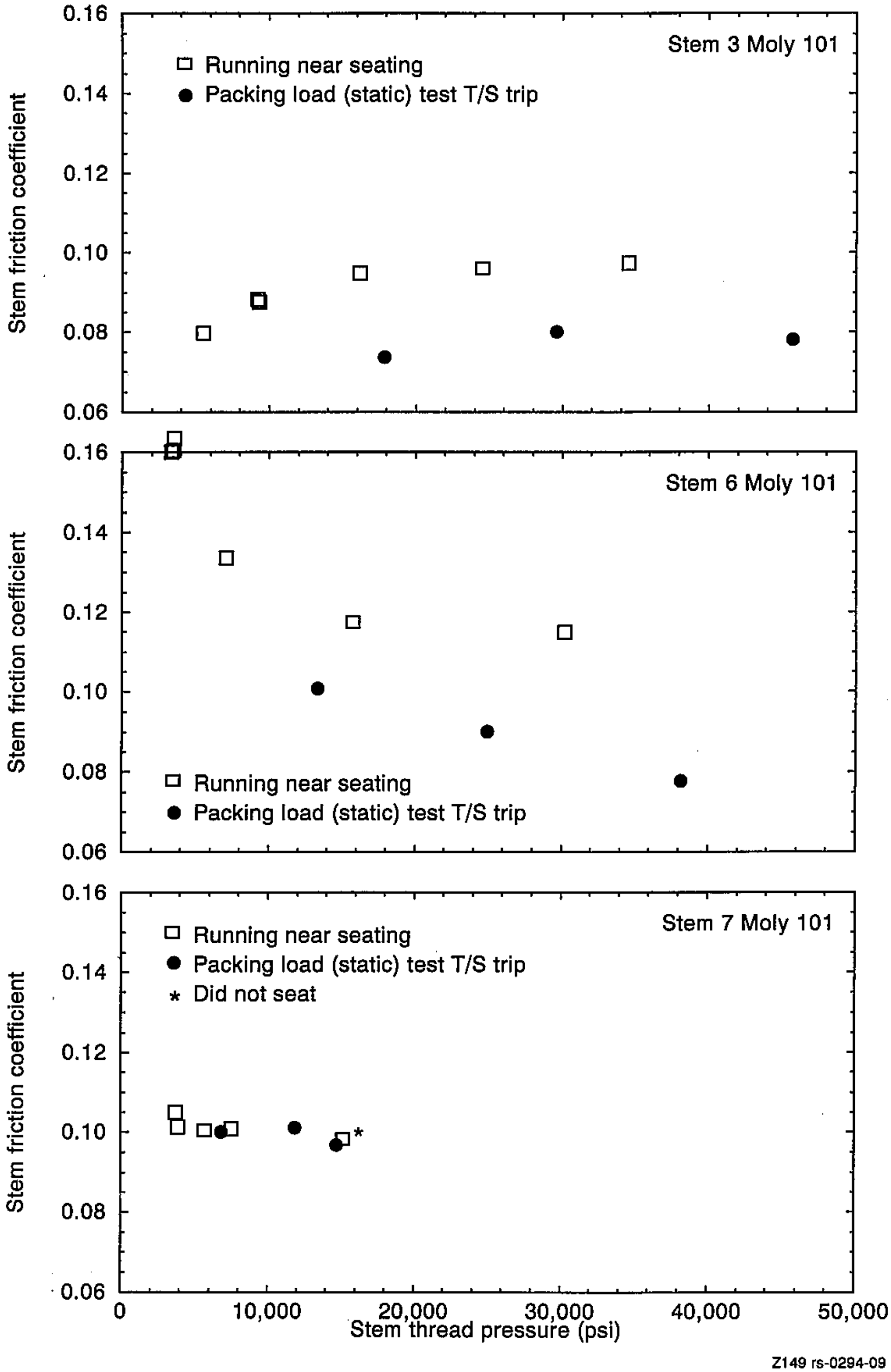


Figure 10. Running and wedging (torque switch trip) coefficients of friction versus stem thread pressure for three stems.

Although the single value for coefficient of friction taken at torque switch trip does not tell us what we need to know about the design basis coefficient, a close look at the entire wedging transient does provide some important information. The wedging transient consists of the small interval of time from initiation of wedging (at about 5,000 psi thread pressure) through torque switch trip to the final maximum thrust. Figure 11 plots stem/stem-nut coefficient of friction against stem thread pressure during the wedging transient derived from the static test of Stem 6 at the highest torque switch setting. (Wedging transients derived from static tests at the medium and low torque switch settings are discussed later in this paper.) The trace in Figure 11 represents the value calculated from the measured stem thrust and the measured stem torque during the small interval of time during which wedging occurs. Figure 11 shows how the coefficient of friction changes as the load in a static test suddenly increases during wedging.

A comparison of the Stem 6 wedging transient (shown in Figure 11) with the Stem 6 running and wedging data is presented in Figure 12. These data points represent the friction coefficients just before wedging and at torque switch trip for the dynamic and the static tests at three torque switch settings. Note that this wedging transient, represented by the trace, generally follows the trend in the wedging data from the static tests, and it roughly resembles the shape of the trend in running data. This similarity provided the second clue of a link between running data and wedging data.

To better understand the nature of the coefficient of friction transient during wedging, we plotted a family of wedging transient curves for each of the three representative stems (Stem 3, Stem 6, and Stem 7), as shown in Figure 13. For Stem 3, the top three traces (labelled Tests 3, 5, and 7) are the dynamic wedging transients; the test with the highest running load and the highest torque switch setting is represented by the trace on the right (Test 7). The bottom three traces are the static wedging transients; the longest (Test 6) is from the test with the high torque switch setting, and the shortest (Test 2) is from the test with the low setting. The six traces represent three pairs of tests (a dynamic test and a static test) for each of three torque switch settings. For Stem 6 there are only two traces from the dynamic tests (Stem 6 did not seat in the dynamic test with the highest load and the highest torque switch setting, so there is no wedging transient for that test). For Stem 7, the traces are on top of each other, making it difficult to distinguish which is which. Stem 7 has a very flat response with almost no tendency toward load-sensitive behavior.

We observed that a wedging transient provides a snapshot of the characteristic behavior of a stem. For each stem, the shapes of all the traces are similar, regardless of the absolute value represented by the trace, the load before wedging, or the torque switch setting. More important, the amount of difference among the absolute values represented by the traces corresponds roughly with the amount of change in the friction coefficient during a single wedging transient.

This last observation is shown more clearly in Figure 14. The upper plot is the same as Figure 12, showing the wedging transient and the individual data points for running and torque switch trip for Stem 6. Compare these data with the lower plot, which shows the same data for Stem 8. Note that in the upper plot, the large change in the friction coefficient during the wedging transient corresponds with large differences among the individual data points, while in the lower plot, a small change during the wedging transient corresponds with small differences. It is evident that the wedging transient tells a lot about the stem's propensity for load-sensitive behavior; the greater the change in friction during the wedging transient, the more load-sensitive behavior is seen in the comparison between the static torque-switch-trip data and the dynamic running data. This observation provided the third clue about the relationship between running data and wedging data, and it proved to be the birth of what we call the *fold line method* for bounding the design basis friction coefficient from the results of a static test.

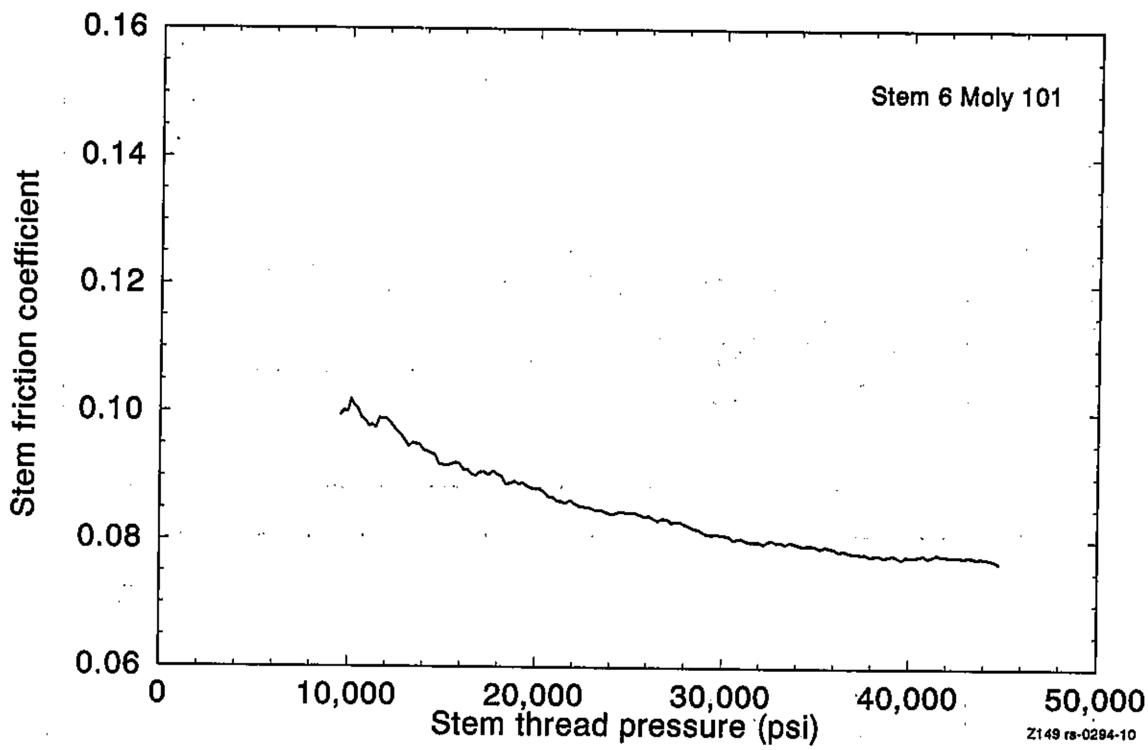


Figure 11. Wedging transient for Stem 6 (static test, high torque switch setting), with coefficient of friction plotted against thread pressure.

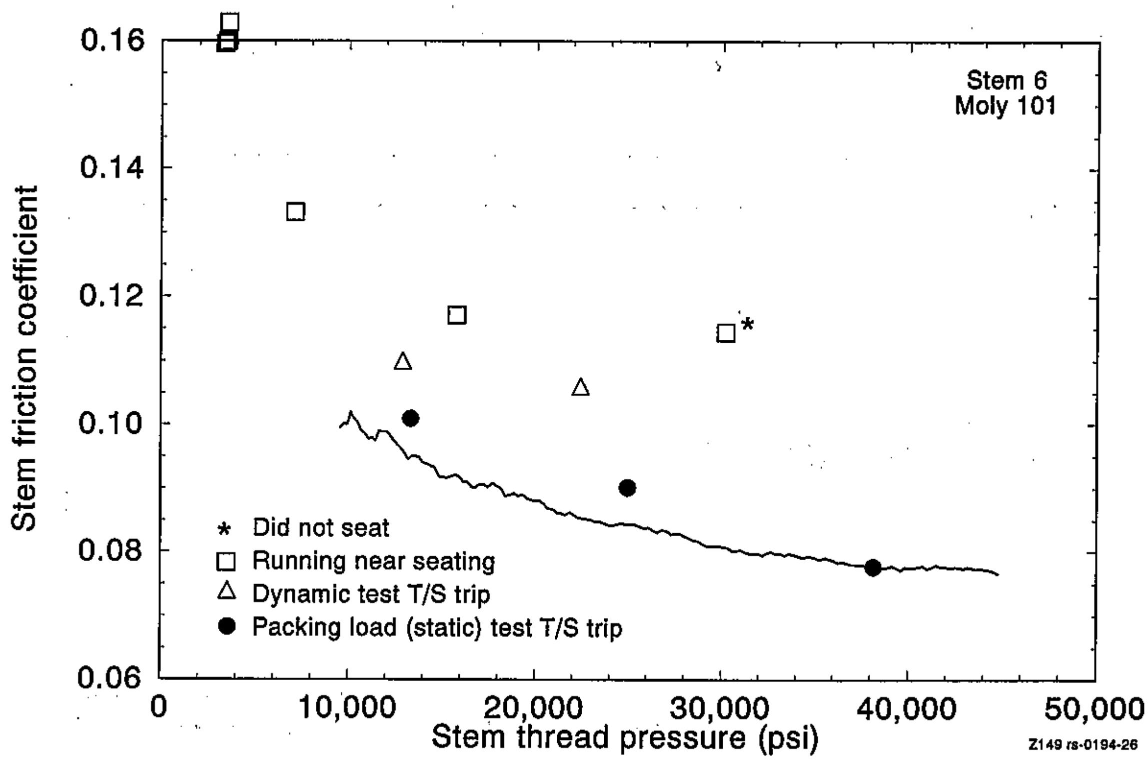


Figure 12. Coefficient of friction during the wedging transient for Stem 6, compared to individual data points for running and for torque switch trip.

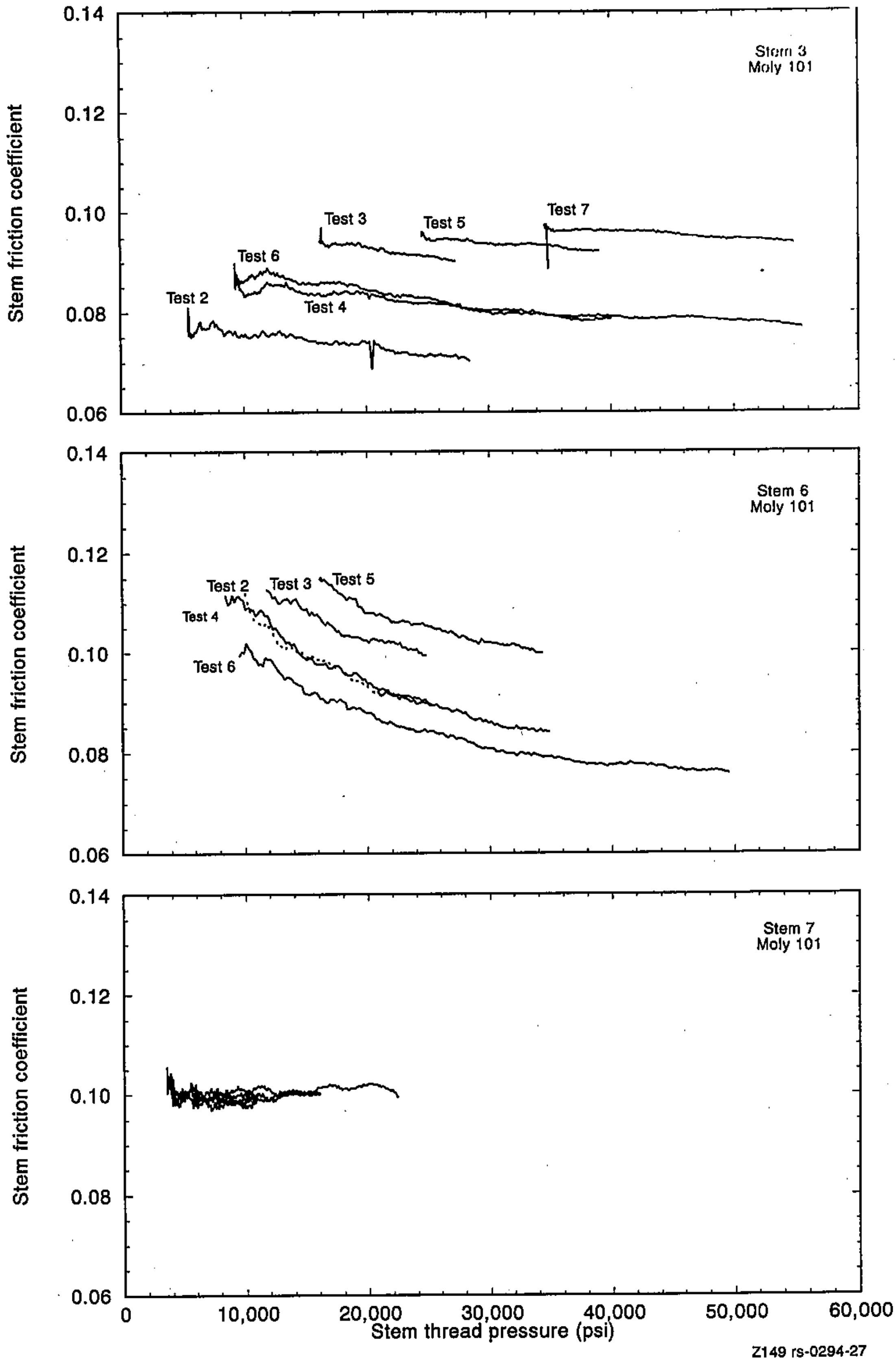


Figure 13. Wedging transients for three representative stems. Note that for any given stem, the traces are similar.

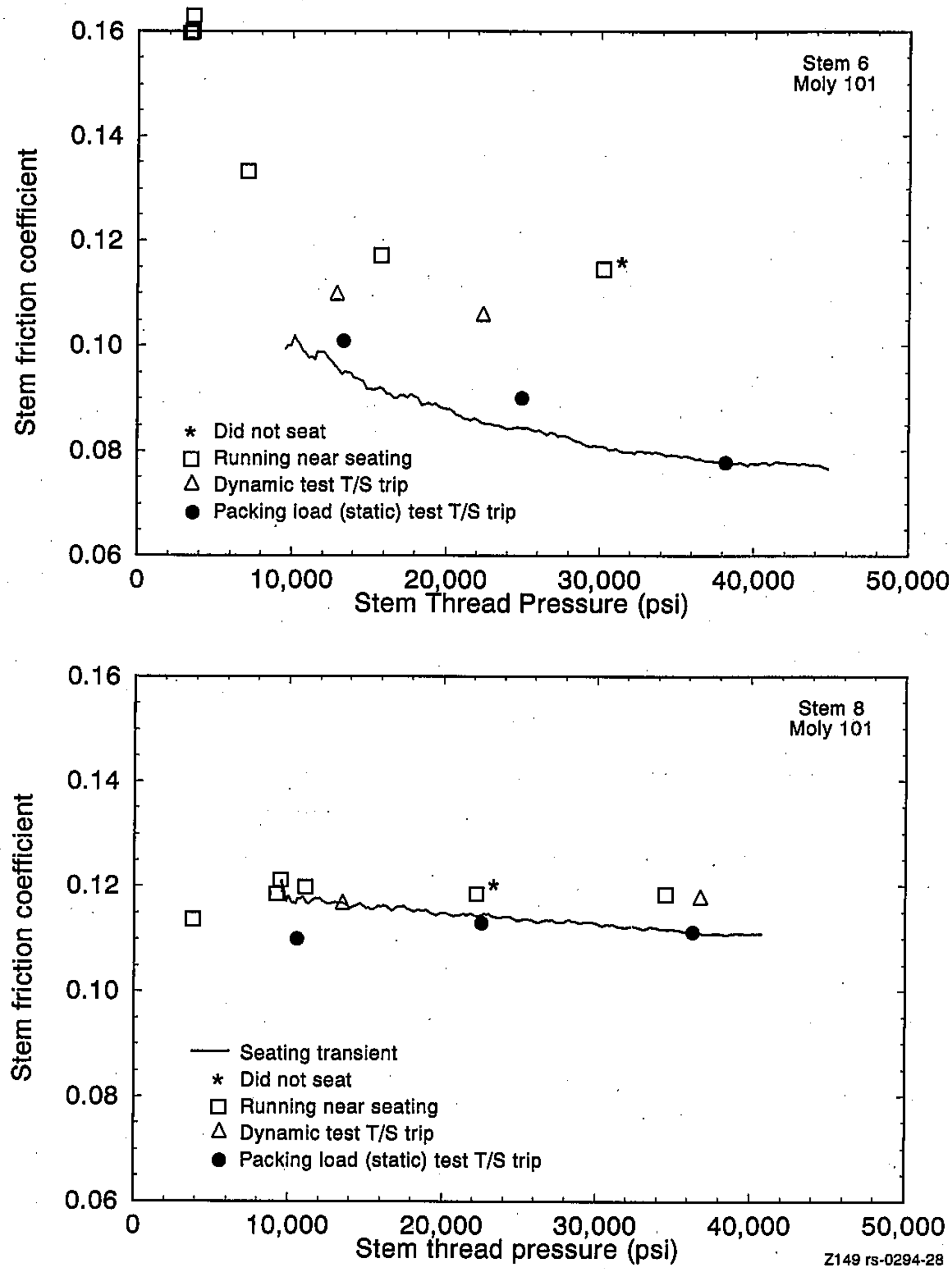


Figure 14. Stem 6 data compared with Stem 8 data.

We also observed that the friction coefficient at the beginning of the wedging transient in the static test provides a benchmark from which to extrapolate. A careful study of Figure 13 shows this to be true of Stems 3, 6, and 7, in the relationship between the data from the static test and the corresponding dynamic test. (Tests 2 and 3 are a pair of tests, Tests 4 and 5 are a pair, etc.) We found it to be true of seven of the eight stems we tested. (The exception is discussed later in this paper.) By coupling this knowledge with the expected difference in stem friction coefficient, as defined by the wedging transient, we can now bound the design basis running coefficient of friction.

The following exercise demonstrates how the *fold line method* works. The upper plot in Figure 15 is the same as Figure 11, except that we have drawn two horizontal lines to mark the change in the friction coefficient during the wedging transient. The top line represents the benchmark or *fold line* from which we intend to extrapolate, and the difference between the top line and the bottom line represents the amount of difference we expect between the static wedging coefficient and the dynamic running coefficient. Thus, by folding on the *fold line*, we can identify the location of a third line that will envelope the running data. The effect is to use the amount of change below the benchmark to bound the expected coefficient in the dynamic running data above the benchmark. This effort is demonstrated in the middle plot in Figure 15. The lower plot shows that the result bounds the running coefficients of friction near wedging and the coefficients at torque switch trip for all thread pressures above 10,000 psi. This completes the basis of the *fold line method* for bounding the design basis friction coefficient.

Figure 15 demonstrates the fold line method for Stem 6. Figures 16 through 18 provide the same information (in a slightly condensed format) for Stems 1 through 4 and Stems 7 and 8. (Stem 5 is discussed below.) In each case, the dotted lines identify the change in the coefficient of friction during the wedging transient, and the solid line identifies the bound that envelopes the running data. These results show that the fold line method consistently provides conservative results for all seven stems.

The fold line method also provides conservative results for Stem 5, but the analysis is a little different. As stated earlier, most stems experience a decrease in the coefficient of friction during the wedging transient of the static test. With Stem 5, the friction coefficient increases during the wedging transient. However, the running data for Stem 5, like that for the other stems, reaches a plateau at about 10,000 psi stem thread pressure (Figure 6). Also, the fold line method applied to Stem 5 still bounds the response. This result is shown in Figure 19. As with the other stems, the wedging transient provides a snapshot of the stem's overall behavior, and we can use the wedging transient to define the expected amount of change in the friction coefficient. As with the other stems, we place the fold line (the benchmark) at the highest value observed during the wedging transient. The only difference is that for Stem 5, this highest value occurs at the end of the wedging transient instead of at the beginning.

The data plots shown in Figures 9 through 19 are from tests with the stems lubricated with Moly 101 grease. All of the stems we tested on the MOVLS were also tested with EP-1 grease, and a similar analysis of those data has been performed. From that analysis, we have determined that the fold line method works as well with EP-1 as it does with Moly 101. Figure 20, derived from testing of Stem 3, is typical of the results of testing with EP-1 lubricant. Note that although the friction coefficients are slightly higher than those for Moly 101 (compare to Figure 17), the fold line method nevertheless bounds the responses.

We believe that with further validation and refinement, the fold line method can serve as a straightforward, useful tool for bounding the stem factor in calculations of a valve's design basis requirements. It provides the most accurate bound of all the methods we have studied to date for predicting design basis response using data from static tests. It will provide results that are considerably more accurate than the default values for the coefficient of friction (0.15 or 0.2) that are being considered for blanket application.

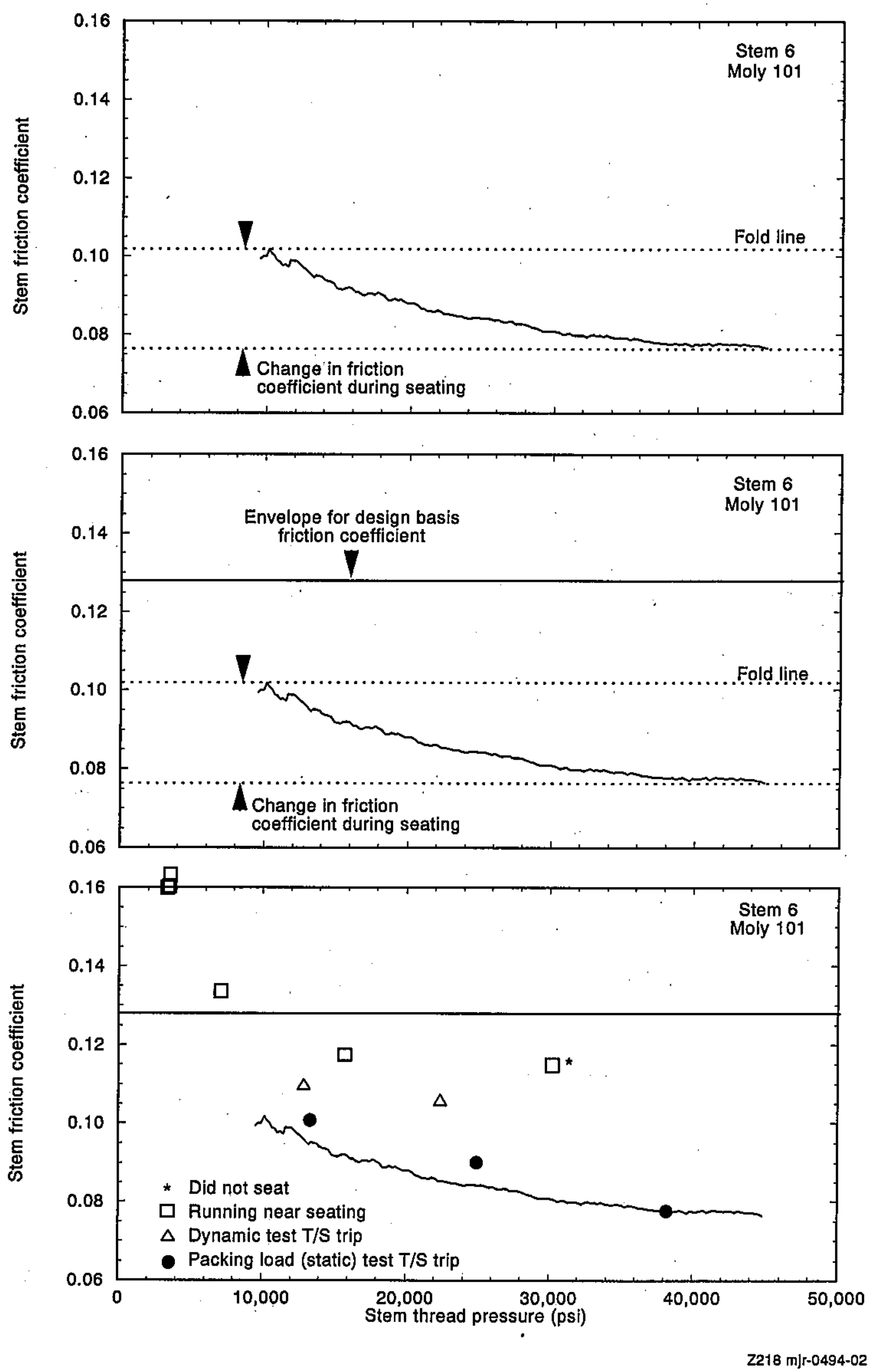
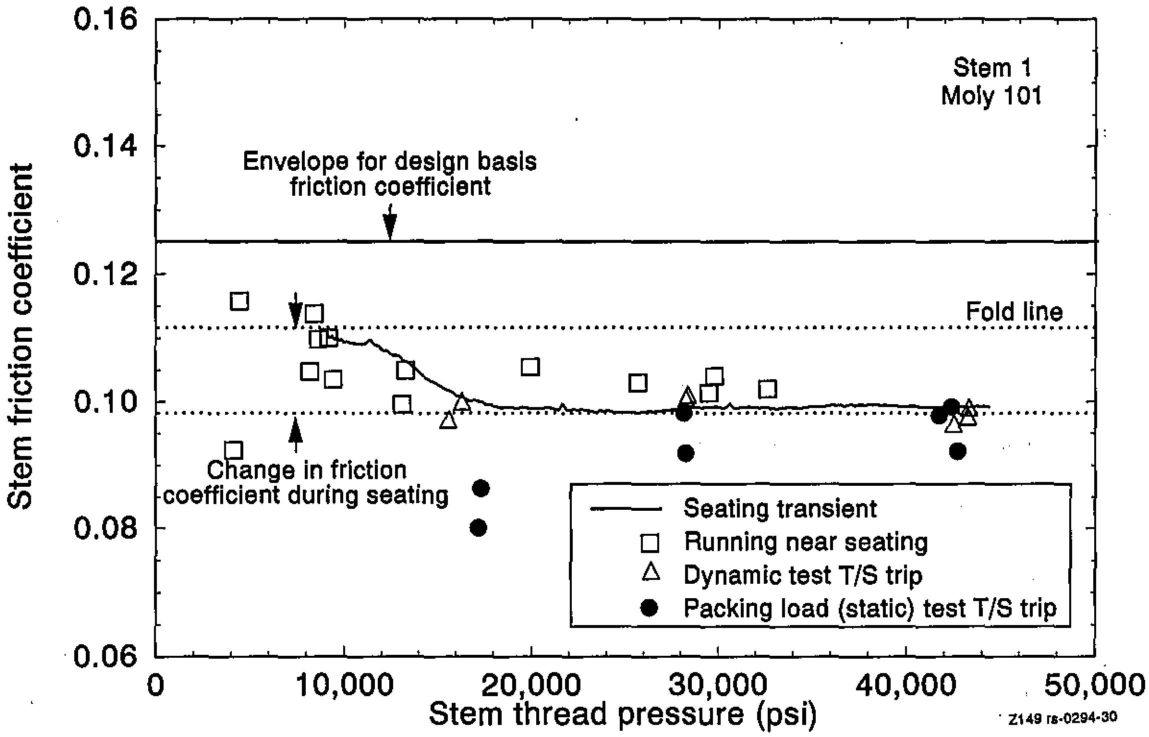


Figure 15. The fold line method demonstrated in three successive plots of Stem 6 data.



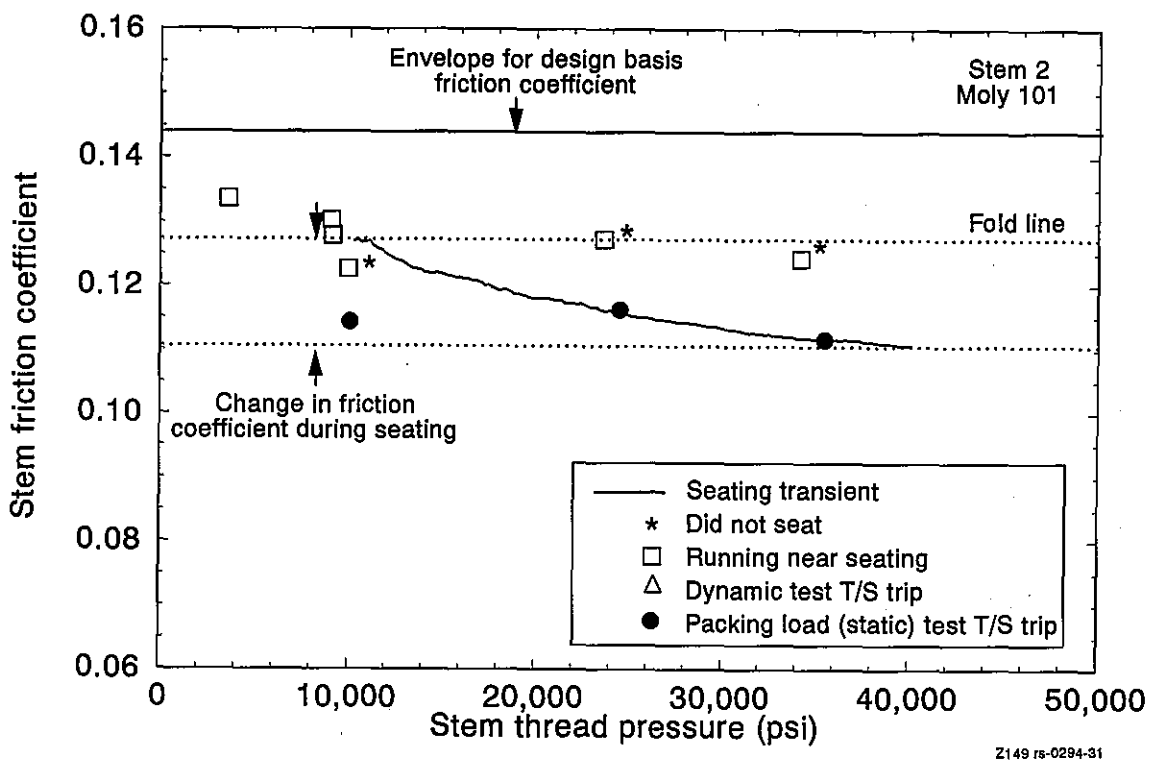


Figure 16. The fold line method bounds the performance of Stems 1 and 2.

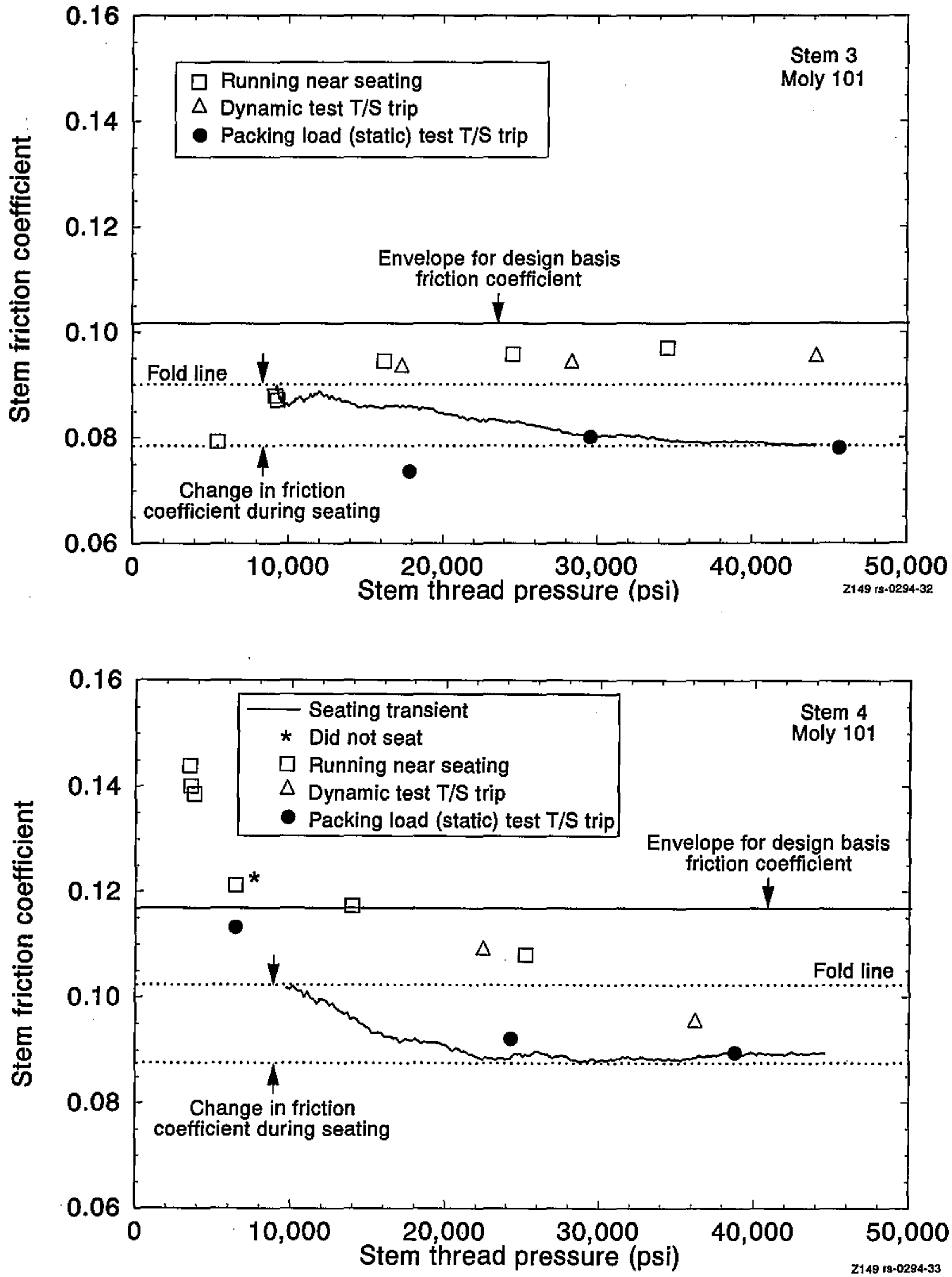
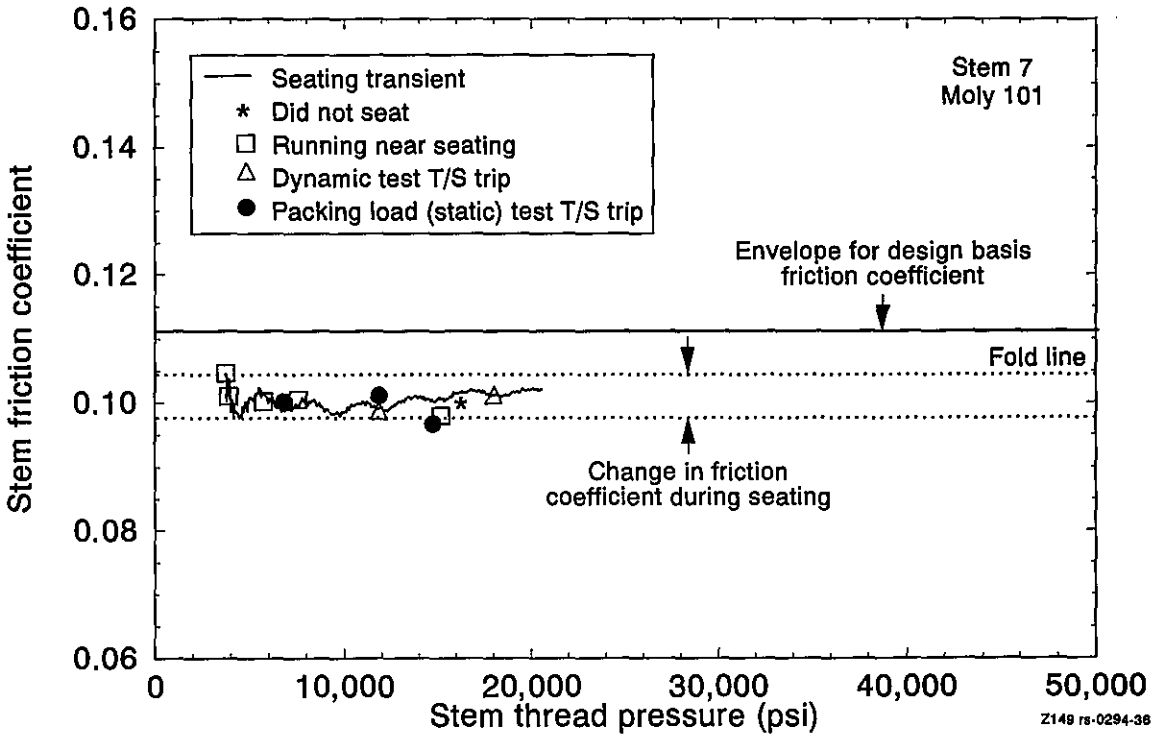


Figure 17. The fold line method bounds the performance of Stems 3 and 4.



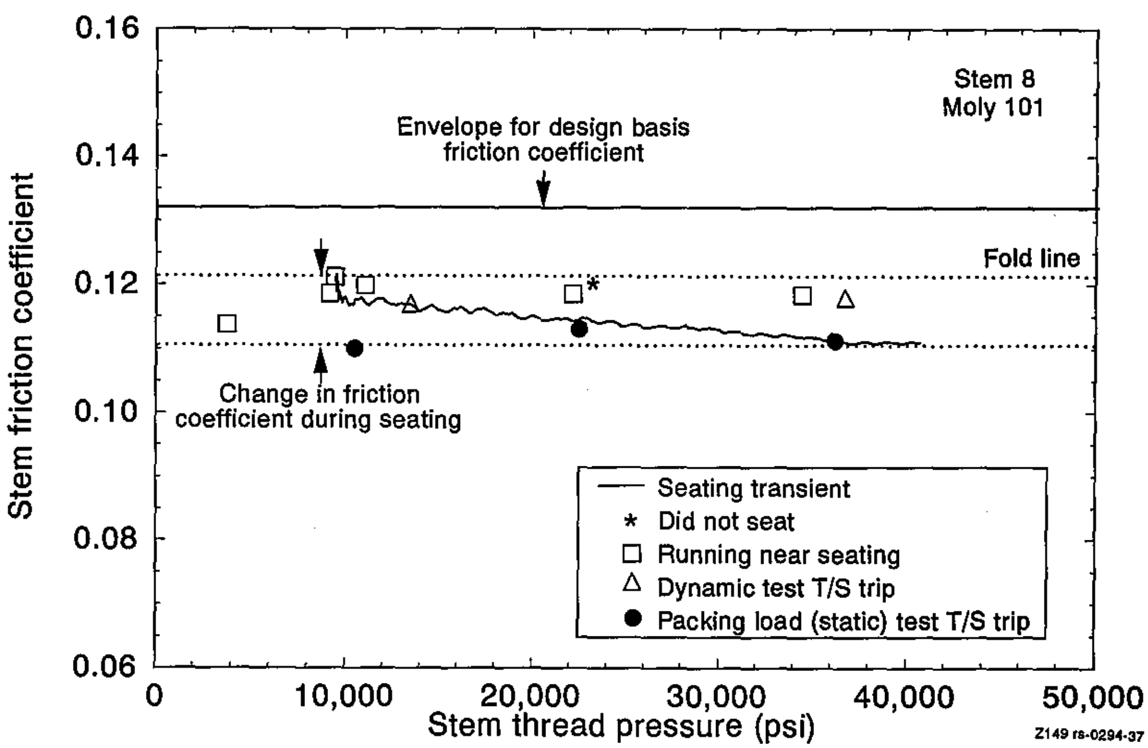


Figure 18. The fold line method bounds the performance of Stems 7 and 8.

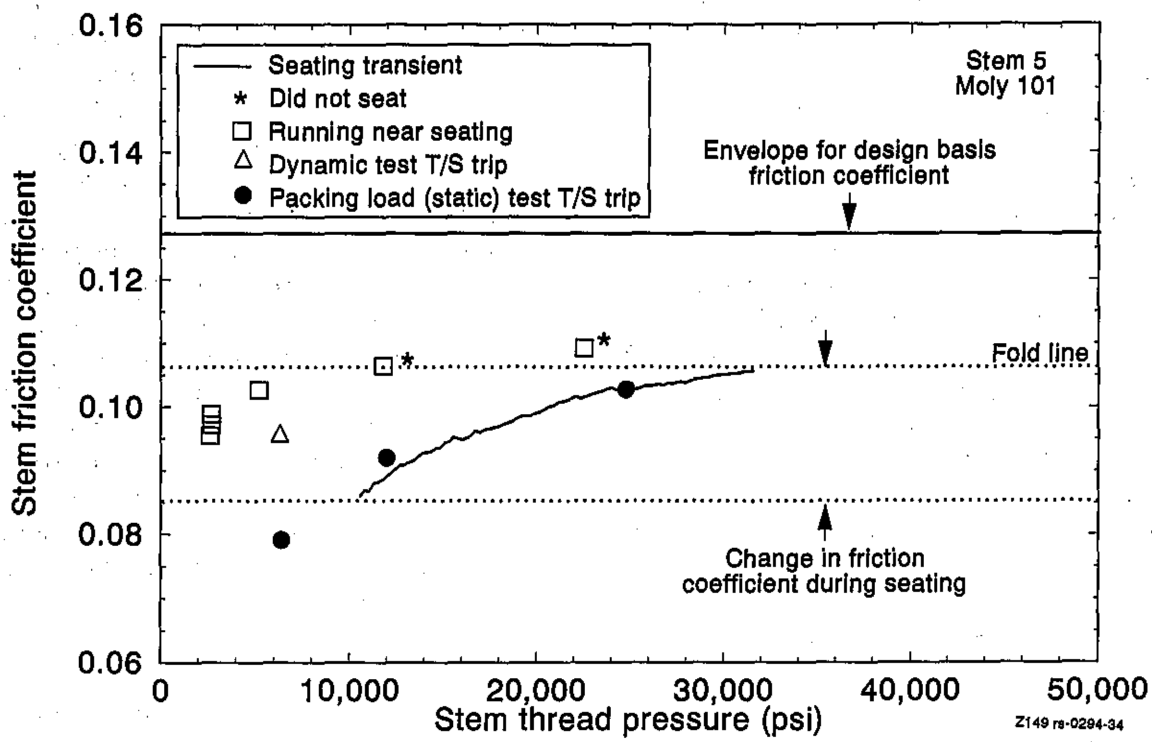


Figure 19. The behavior of Stem 5 is different from that of the other stems, but the fold line method still bounds its performance.

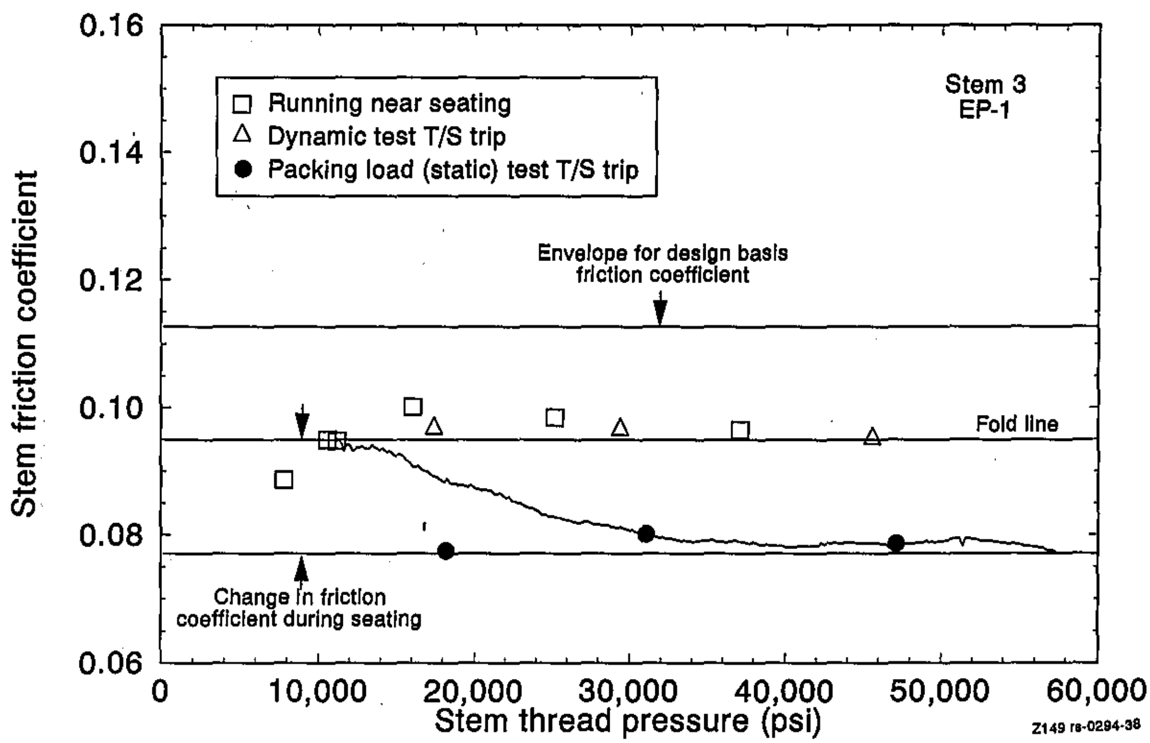


Figure 20. The fold line method bounds the performance of Stem 3 in tests using EP-1 lubricant instead of Moly 101.

With further development and validation, an analysis using the fold line method might not always require that the coefficient of friction be plotted against stem thread pressure, or that the wedging transient be plotted as we have plotted it in this presentation. Figure 21 is an ordinary plot of the friction coefficient over time for Stem 3 for two pairs of tests: a static test and a dynamic test at the low torque switch setting and at the medium torque switch setting. For Stem 3, it is easy to identify the wedging transient in the static test and to use the fold line method to draw the bounding line. Note also that in each case, this line bounds the design basis running coefficient just before wedging in the dynamic test. (The last 20% of the trace from the dynamic test represents the value that needs to be bounded.)

The data presented in Figures 15 through 19 demonstrate the fold line method using the wedging transient from static tests with high torque switch settings. These results indicate that the use of data from the static test with the high torque switch setting is conservative for bounding the friction coefficients at the low running loads. However, the use of data from the static tests with a low torque switch setting might not be conservative for bounding the results at the high running loads. In contrast to Figures 15 through 19, the fold line method as demonstrated in Figure 21 uses the wedging transient from static tests at the low and medium torque switch settings to bound the results of the corresponding dynamic tests. We performed a similar comparison of the other pairs of tests for the other stems and the other torque switch settings. In each case, the fold line method based on the wedging transient in the static test bounded the running friction coefficient in the corresponding dynamic test. This result lends confidence that the fold line method is applicable not only at high torque switch settings, but also at lower torque switch settings.

In summary, the purpose of the fold line method is to provide a stem factor that can be used in calculations to determine a valve's design basis requirements. Use of the fold line method

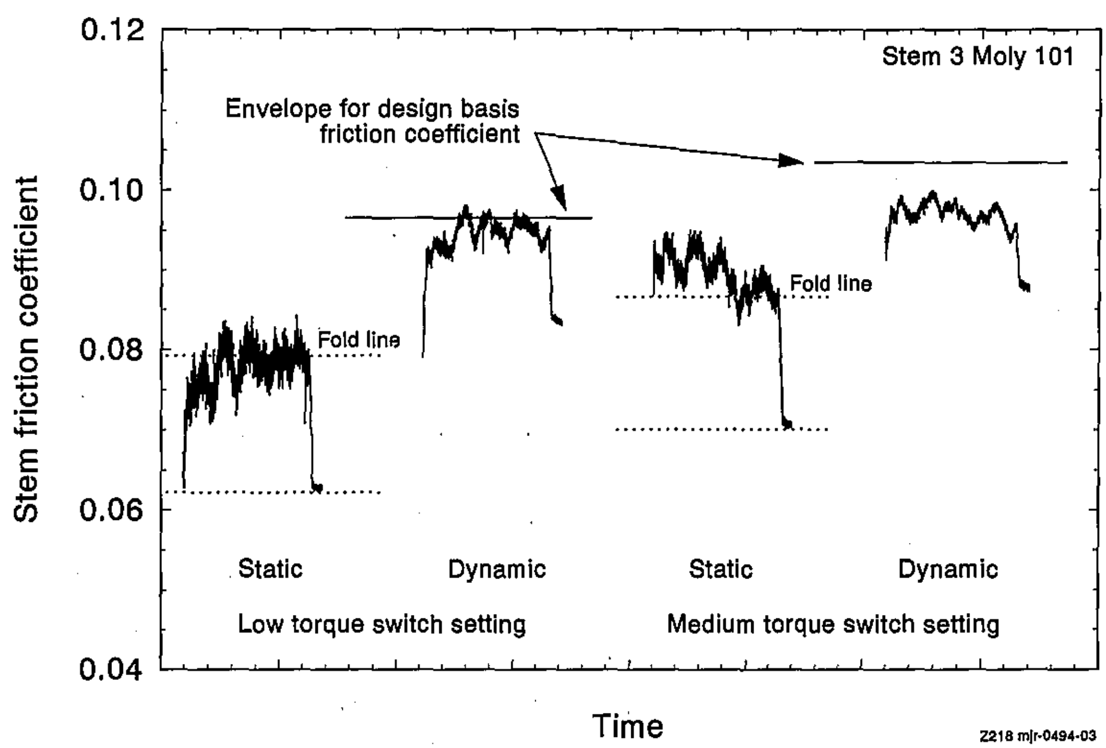


Figure 21. Real-time coefficient of friction data from two pairs of tests at two torque switch settings for Stem 3.

would entail an in situ static test (packing load only, little or no pressure load, no flow load), with the torque switch set high enough that the valve will close against its design basis load. The fold line method uses the results of the test and accounts for the difference between the low stem factors observed during wedging in a static test and the higher stem factors typically observed at or before flow isolation when a valve closes against its design basis loads. The procedure for using the fold line method would include the following considerations. Continuous measurement of torque and thrust would be needed in order to calculate the stem factor. Indirect measurements of operator torque (torque based on spring pack measurements) might be sufficient, provided that measurement and calibration inaccuracies and the effects of operator momentum are accounted for. Measurements taken during the test would be used to define the wedging transient, as shown, for example, in either Figure 21 (static test) or Figure 11. Identifying the beginning of the wedging transient is straightforward for some stems, but it is a bit tricky for others, depending on the response of the particular stem. Next, identify the highest and lowest values for the stem factor recorded during the seating transient. Add to the highest value the difference between the two. The resulting value bounds the stem factor expected at design basis conditions. The fold line method would be useful for valves that can be subjected to an in situ static test but cannot easily be tested against flow and pressure loads. The results from testing of a particular valve would not be applicable to other valves.

OBSERVATIONS

Our stem factor research to date has provided information in the form of a technical basis for the USNRC effort in evaluating industry methodologies for extrapolating stem factor from the results of tests less severe than a design basis test. The research has provided the basis for two possible methods. We present these methods here in the intent that industry may choose to develop them further and eventually implement them. The Electric Power Research Institute (EPRI) and the MOV Users Group (MUG) have both shown interest in developing the methods further.

More data for a larger population of stems is needed to more closely define the threshold value to be used in the threshold method. Additional running data for a wider range of stems would also increase confidence that the uniform flat response above the threshold is a universal characteristic.

The data used to develop the methods discussed here were recorded with ideal lubricant conditions. Stems were carefully cleaned before lubricant application, and the tests were conducted shortly afterwards. Aging, dirt degradation, and dryout degradation need to be addressed.

The data used to develop both the threshold method and the fold line method were obtained from direct measurement of torque and thrust in the stem. For valve diagnostic tools that determine operator torque indirectly, using either spring pack force or spring pack displacement, additional validation would be necessary.

The amount of change in the coefficient of friction for a given stem, as obtained during the wedging transient of a static test, appears to depend on the thread pressure achieved during running and on the duration of the wedging transient. As presented in this discussion, the fold line method uses static tests with low running loads (simulated packing drag loads) and typical torque switch settings to define the amount of change in the friction coefficient. More study is needed with very low packing loads and very low torque switch settings to determine if the fold line method is still applicable at those conditions. If there are lower limits to the packing load and the torque switch setting, additional research may be able to fine-tune the fold line method to make it applicable below those limits.

Of the two lubricants we used in the tests, Moly 101 had the lowest overall coefficient of friction, but on some stems EP-1 had the lowest friction coefficient. The reason probably has to do with surface finish and lubricant performance

relative to that surface finish. Future research might undertake (a) to identify which surface finish characteristics lower the coefficient of friction and limit load-sensitive behavior and (b) to determine which lubricants are best for a stem with a given surface finish.

CONCLUSIONS

The threshold method and the fold line method represent a major breakthrough in an area that has challenged valve researchers for several years. Both methods are presented here as being in the initial stages of development. Both methods need further work and validation before they can be used with complete confidence in the field. This work should include more testing with other lubricants and additional stems. It might also be necessary to address the effects of stem and lubricant degradation.

We have not endeavored to precisely identify the friction and lubrication phenomena that produce the effects that are evident in the data. There are several possible explanations as to why the stem/stem-nut friction reaches a plateau at a certain stem thread pressure. Similarly, there are several possible explanations for the relationship between the wedging friction in a static test and the running friction in a dynamic test. We have endeavored instead to perform the initial development of methods that are supported by test data. So far, the data show that these methods work. Both methods are based on the kinds of tests that can be performed in the plants, and both methods use simple, straightforward analyses. Both methods provide conservative results without imposing excessive conservatism.

The results presented here are based on testing of eight stems at three dynamic loads (and three corresponding torque switch settings) using two different lubricants, for a total of 48 pairs of tests (one static and one dynamic). The fold line method, based on the results of the static test, bounded the corresponding dynamic running coefficient of friction in every case. From the results of those tests, we can say that the following relationship appeared without exception: (a) the net change in the friction coefficient during the wedging transient of the static test (whether the change is an upward change or a downward change) defines the amount of difference that can be expected between the wedging friction in a static test and the running friction in a dynamic test, and (b) the highest friction recorded during the wedging transient (whether it occurs at the beginning or the end of the transient) serves as a benchmark from which a prediction can be made that will bound the running friction.

REFERENCE

Steele, R., Jr., J. C. Watkins, K. G. DeWall, and M. J. Russell, 1993, *Motor-Operated Valve Research Update*, NUREG/CR-5720.

Predictable or Not Predictable? The MOV Question

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Wyle Laboratories

ABSTRACT

Over the past 8 years, the nuclear industry has struggled to understand the dynamic phenomena experienced during motor-operated valve (MOV) operation under differing flow conditions. For some valves and designs, their operational functionality has been found to be predictable; for others, unpredictable. Although much has been accomplished over this period of time, especially on modeling valve dynamics, the unpredictability of many valves and designs still exists. A few valve manufacturers are focusing on improving design and fabrication techniques to enhance product reliability and predictability. However, this approach does not address these issues for installed and unpredictable valves. This paper presents some of the more promising techniques that Wyle Laboratories has explored with potential for transforming unpredictable valves to predictable valves and for retrofitting installed MOVs. These techniques include optimized valve tolerancing, surrogated material evaluation, and enhanced surface treatments.

INTRODUCTION

From a design and testing perspective, a valve is referred to as unpredictable when its performance cannot be predicted by industry-accepted performance prediction models. In other words, an unpredictable valve is one for which the thrust required to operate the valve is different from the requirement predicted by the valve thrust equation. This usually excessive thrust requirement can be caused by the occurrence of internal damage and out of tolerance or excessive tolerance of internal dimensions, allowing disc tipping that can cause the pressure distribution around the disc to change. From the U.S. Nuclear Regulatory Commission (USNRC) Generic Letter (GL) 89-10, predictability can also be defined in terms of operability. Therefore, in addition to the above, predictability can also be dependent upon a number of other factors including improper material selection, actuator sizing, and installation and maintenance techniques. Wyle Laboratories recently completed a large research and engineering project sponsored by the Electric Power Research Institute (EPRI). This project, which supported EPRI's joint utility motor-operated valve (MOV) Performance Prediction Program effort, focused entirely on the GL 89-10 MOV recommendations. Thirty-four MOVs, which were representative of a wide range of manufacturers, sizes, types, and pressure ratings, were tested over a variety of differential pressure and fluid conditions. The program included cold water, hot water, and steam blowdown testing on selected valve specimens.

A number of the tested valves performed predictably and consistently. However, a significant number sustained considerable internal damage during blowdown testing. These included a 6-in., 900-lb, flexible wedge gate valve and a 6-in., 1,500-lb gate valve. Both sustained considerable damage to the body guide rails and disc guides during the tests. Damage was in the form of severe galling and gouging of the sliding surfaces of the guide rails and guides. In addition, a 2-1/2-in., 900-lb gate valve experienced a thrust excursion because the as-built dimensions were out of specified tolerance. Thus, those valves whose behavior were anomalous prompted concern and further investigation.

In an earlier MOV test program conducted by the Idaho National Engineering Laboratory (INEL) in support of the resolution of Generic Safety Issue 87 (GSI 87), "Failure of HPCI Steam Line Without Isolation," several of the tested valves sustained internal damage during blowdown testing as well. Those valves were referred to as unpredictable and included 6-in. and 10-in. flexible wedge gate valves (USNRC, 1990).

In some cases, the increase in required thrust associated with unpredictability can be significant and might exceed the capability of the motor or operator. Under that condition, the thrust requirement to close an unpredictable valve during a design-basis event cannot be accurately determined without testing the valve (either individually or as a prototype) under design-basis conditions. Research has also shown that testing a valve under static or low flow conditions cannot always be used to accurately predict the behavior of the valve under design-basis conditions because extrapolation methodologies are often unrealistic predictors of design-basis behavior. For example, the valves previously referred to as sustaining damage during blowdown testing operated normally when subjected to less severe flow conditions. Thus, low flow tests should not be used to identify valves that may require significantly more thrust than that predicted by the valve thrust equation (USNRC, 1990). Nevertheless, monitoring a valve's propensity for nonlinear behavior at low flow can provide insight into the valve's behavior at higher flow.

Therefore, the results of these and other MOV research, engineering, and test programs indicate that the current predictability of some safety-related valves to operate under normal and design-basis conditions is questionable. These programs also point to the fact that the problem appears to be related to deficiencies in design tolerancing, valve material, and friction between internal parts. In a quest for potential engineering solutions, this paper describes a research program conducted by Wyle Laboratories, primarily under Virginia Electric Power Company (VP) sponsorship, to explore these deficiencies. Specifically, the research focused on modeling valve physical parameters to determine optimal design tolerancing envelopes and on investigating various surface enhancement techniques to improve common valve operability and reliability, and thus predictability.

Optimal Valve Tolerancing

Increased attention in recent years has been focused on the issue of optimal valve tolerancing. Tight tolerances can cause binding and lead to increased wear on sliding components (i.e., disc guide to body guide rail clearances). Loose tolerances can allow contact between moving parts during valve operation under high differential pressure and flow conditions. These scenarios can lead to internal valve damage and thus unpredictability.

Under this research effort, Wyle developed a computer program to model gate valve kinematics in order to study the valve tolerancing issue. The design parameters of a high quality 3-in. wedge gate valve were used to develop a mathematical model to determine areas where valve internals may contact or bind during operation. Algorithms were then constructed from which a computer program entitled MODGVALV was developed and validated. This program can be run on most personal computer systems. By entering specific valve parameters, MODGVALV calculates critical valve clearances to determine whether premature contact can occur between the seat and disc sealing surfaces. Premature contact may impact valve closure and result in deterioration of critical valve components. MODGVALV also accounts for linear thermal expansion and guide-offset impact and allows for reasonable acceptance of gate valve parameters from other manufacturers (Smith, 1991).

ENHANCED SURFACE TREATMENTS

A variety of surface treatment techniques were evaluated by Wyle to determine their application and potential for reducing the friction and wear associated with valve operation. Of these, two material modification technologies, ion

implantation and a multilayer infusion coating process called Magnaplate Hi-T Lube,[a] appeared to offer good, if not dramatic improvement potential and are therefore reviewed first. Immediately following, an overview of other enhancement technologies that were considered but eliminated from further research is presented, along with the reasons for their exclusion.

Ion Implantation

Ion implantation is a process whereby energetic ions of selected materials are accelerated and made to strike the surfaces of workpieces in a vacuum chamber (Sioshansi, 1986). The ions, typically with energies to hundreds of kiloelectron-volts (keV), penetrate hundreds of atomic layers into the surface, where they are slowed down and eventually stopped through collisions with atoms of the host material.

The collision cascade upon host atoms creates a region of extensive radiation damage within the surface layer of the target. A term borrowed from solid-state physics, radiation damage refers to alterations produced in the crystal structure of a material. In ion implantation of metal workpieces, radiation damage is extremely desirable because it, along with the foreign implanted ions, alters surface properties in a number of desirable ways.

The resultant combination can create an amorphous layer with no grain boundaries, which is believed to provide superior wear performance and low friction in tribological service. In ferrous alloys, an amorphous surface can be responsible for reducing corrosion, which often initiates at grain boundaries.

From work performed by Mr. Piran Sioshansi of Spire Corporation (1986), the ion implantation process has been applied to a variety of industrial products and tools, such as

- Bearings—To impart corrosion resistance and to improve wear resistance of precision bearings like those used in gyroscopic inertia systems, a combination of titanium and carbon has provided the best results to date. The amorphous, glassy titanium carbide precipitates formed on the surface have been shown to decrease the coefficient of friction by a factor of 2 and provide superior wear performance.

- Nuclear reactor components—Because of its excellent corrosion resistance, zirconium has been used in this highly corrosive environment. However, zirconium surfaces are susceptible to fretting and adhesive wear. High dose implantation of carbon and nitrogen increases the microhardness and significantly improves the resistance of zirconium to both types of wear by creation of zirconium carbonitride layers. Other ions implanted, such as chromium and carbon, have been chosen for evaluation of zircaloy components for nuclear reactor applications.

- Prosthetics—Many orthopedic implant devices are made from titanium-based alloys. Ion implantation has proven to be valuable for increasing the wear resistance of titanium alloys, while also improving fatigue performance and corrosion resistance.

- Stamping, cutting tools, injection molds, ceramic parts—These applications have also used ion implantation technology to improve performance factors.

As stated in Sioshansi (1986), the benefits of ion implantation can be summarized as follows:

- Surface properties can be selectively and independently added to a material's bulk properties.

- Virtually any element in the periodic table can be implanted.

- The concentration profile of implanted ions is easily controllable.

a. A registered trademark of General Magnaplate Corporation.

- In contrast to conventional coating processes that produce a discrete layer, ion implantation changes the near surface composition of the substrate to create a new alloy or to alter its surface properties.

- There are no problems related to bonding failures or delamination of coatings because there is no sharp interface in the implanted layer.

Alloys that can be created by the process are not limited by classical thermodynamic parameters of diffusion and solubility.

It is a low-temperature process that can be applied to a finished product without altering the product's dimensions.

Magnaplate Hi-T Lube

Magnaplate Hi-T Lube is a patented dry film lubricant consisting of a multilayer system that is applied to wear surfaces by means of a series of "synergistic" electrodeposited metals and alloys that are permanently bonded to the substrate metal (General Magnaplate). Hi-T Lube is a registered trademark of the General Magnaplate Corporation. It consists of metallic and nonmetallic layers that minimize dimensional changes.

Synergistic coatings are not true coatings in the conventional sense. These coatings become an integral part of the top layer of the base metal rather than just a surface cover. These coatings are referred to as synergistic because the resulting surfaces are superior in performance to both the base metal and the individual components of the coating. Each layer of the Hi-T Lube matrix has beneficial features. However, upon final diffusion, they form a metallic/oxide matrix that is significantly better than any one of the individual layers or the base metal.

The first layer applied is an extremely hard coating. Before application of this first layer, the surface of the base metal is metallurgically cleaned. During the precleaning process, special provisions are made to avoid hydrogen embrittlement of the metal because it not only contributes to the hardness of the total matrix structure but actually forms the critical interface with the base metal. The second layer applied is a semisoft, compressible metal layer. It is composed of metals that can withstand high temperatures and loads. The final surface layer is composed of a blend of highly effective lubricants. The applied layers are then diffused in a controlled atmosphere chamber (General Magnaplate).

As stated by the General Magnaplate Corporation, the benefits of Hi-T Lube can be summarized as follows:

- Can be applied to virtually all metals that are normally used in components that are subjected to high compressive forces, such as steel, stainless steel, copper, and copper alloys

- Adheres very well to the base metal irrespective of temperature and environmental changes

- Can withstand nuclear exposure to high radiation environments

- Wears well for a long life under extreme operating conditions of sliding and rolling frictions

- Is compatible with hydrocarbons and synthetic lubricants

- Allows high control of the "coat" thickness.

Testing Process

To determine the feasibility and potential of these two material surface enhancement technologies, Wyle established a test program that would subject treated and untreated samples of common valve material to a form of testing that would yield friction test data. For this program, a pin and V-block testing process was selected because of its universal acceptability, economy, and usability in measuring friction and wear simultaneously. Although the objective was to determine friction factors, the benefit of obtaining wear factors was

obvious. Twenty sets of material samples (SS 316 and stellite No. 6) were manufactured in addition to spare pins and V-blocks. Each set was fabricated to conform to American Society of Testing and Materials (ASTM) Standards D2625, D2670, and D3233, and was composed of two V-blocks and one pin. Table 1 shows the distribution of pins and V-blocks that were treated by the two processes.

Ion implantation was performed by the Spire Corporation using their "IONGUARD 2001" process. Titanium and carbon ions (Ti+ and C+) were implanted to a nominal depth of penetration of 1,000 angstroms. Hi-T Lube processing was performed by General Magnaplate of Texas to a coating of 0.001 in., plus or minus 0.0003 in.

Testing was conducted on both untreated stainless steel and stellite samples and the ion implanted and Hi-T Lube processed samples. The test matrix in Table 1 shows the various pin and V-block combinations. Each test was run for a total of 20 minutes at a linear speed of 3.8 in. per second, which is equivalent to 4,560 in. of travel. The samples were subjected to an initial load of 30 lbf, and the coefficient of friction was measured at 1, 2, 3, 4, and 5 minute intervals. The load was then increased to 50 lbf and an initial measurement was recorded at time t = 5 minutes, followed by measurements at 10, 15, and 20 minute intervals. All samples were lubricated with distilled water. A wear measurement was taken at time t = 20 minutes.

Test Results

The test results presented in Tables 2 and 3 clearly show that stellite improves wear when compared with stainless steel. However, among the surface-modified samples, the results are not as clear (see Figures 1 through 4). Without further investigation, the Hi-T Lube process "appears" to be superior to the ion implantation process. It shows apparent improvement in both friction and wear for both stainless steel and stellite "hard-faced" samples. This improvement is greater if only one of the two material components of a set is surface treated. The ion-implanted samples showed no improvement in friction, but showed good improvement in wear. The stainless steel/stainless steel reference sample results are not clear. On one hand, the wear factors are as anticipated, but on the other hand, the friction factors appear to improve (decrease).

Table 1. Pin and V-block test matrix.

Block set material	Pin material[a]			
	ST	SS	ST-II	SS-II
Ion implantation process				
ST	◆	◆	—	—
SS	—	◆	—	◆
ST-II	◆	◆	◆	◆
SS-II	◆	—	—	◆
Hi-T Lube process				
ST	—	—	—	—
SS	—	—	—	◆
ST-HT	◆	◆	◆	◆
SS-HT	◆	—	—	◆

a. SS = stainless steel; ST = stellite No. 6; II = ion implantation processed; HT = Hi-T Lube processed.

Table 2. Pin and V-block friction and wear test results (30 lbf).

Set	Pin material and treatment[b]	Block material and treatment[b]	Coefficient of friction at time (minutes)[a]					
			0	1	2	3	4	5
A	ST	ST	0.297	0.297	0.297	0.297	0.297	0.297
C	SS	SS	0.495	0.446	0.396	0.396	0.446	0.446
D	SS-II	SS	0.495	0.446	0.545	0.594	0.594	0.545
E	ST	ST-II	0.396	0.396	0.396	0.396	0.446	0.495
F	SS	ST-II	0.396	0.396	0.396	0.396	0.396	0.396
G	ST-II	ST-II	0.396	0.396	0.396	0.396	0.396	0.396
H	SS-II	ST-II	0.396	0.396	0.396	0.396	0.396	0.396
I	ST	ST-II	0.297	0.297	0.297	0.297	0.297	0.297
J	SS-II	ST-II	0.396	0.396	0.396	0.446	0.446	0.495
K	SS-HT	SS	0.099	0.099	0.099	0.099	0.099	0.099
L	ST	ST-HT	<0.099	<0.099	<0.099	<0.099	<0.099	<0.099
M	SS	ST-HT	<0.099	<0.099	<0.099	<0.099	<0.099	<0.099
N	ST-HT	ST-HT	0.198	0.198	0.198	0.248	0.297	0.297
O	SS-HT	ST-HT	0.099	0.099	0.099	0.099	0.099	0.099
P	SS	ST-HT	0.198	0.198	0.198	0.198	0.198	0.198
Q	SS-HT	ST-HT	0.147	0.198	0.198	0.198	0.198	0.198

a. The coefficient of friction at time 0 indicates the start of testing with a load of 30 lbf. Additional friction data was recorded at 1, 2, 3, 4, and 5 minutes thereafter.

b. SS = stainless steel; ST = stellite No. 6; II = ion implantation processed HT = Hi-T Lube processed.

The overall results of the sample tests were not as anticipated. Addressing the ion-implanted samples in isolation, the above results questioned existing literature regarding friction and wear factors. Prior, independently derived test data clearly indicate that titanium and carbon ion-implanted metals, including stainless steel, show friction factor reductions and wear improvement on the order of 2 to 3. Data published by the Navy and EPRI also confirm these findings. These independent tests, however, were performed on full-scale valve components, whereas this research program was conducted on material samples.

The apparent success of the Hi-T Lube process may result from the thickness of the surface coating. Whereas the ion-implanted samples were treated to a depth of approximately 1,000 angstroms, the Hi-T Lube samples were "coated" (i.e., thickness added of approximately 0.001 in.).

In other ion implantation research engineering applications, it has been shown that treatments of as small as 10 angstrom depth are adequate to improve friction and wear. Therefore, the possibility exists that the kinematics of ASTM Standard D2670 testing may have corrupted some

Table 3. Pin and V-block friction and wear test results (50 lbf).

Set	Pin material and treatment[b]	Block material and treatment[b]	Coefficient of friction at time (minutes)[a] 5	10	15	20	Wear (teeth)[c]
A	ST	ST	0.327	0.357	0.357	0.386	21
C	SS	SS	0.416	0.327	0.297	0.268	540
D	SS-II	SS	0.535	0.505	0.535	0.565	591
E	ST	ST-II	0.446	0.416	0.416	0.416	24
F	SS	ST-II	0.446	0.505	0.624	0.594	318
G	ST-II	ST-II	0.327	0.386	0.416	0.416	9
H	SS-II	ST-II	0.446	0.446	0.446	0.476	295
I	ST	SS-II	0.297	0.297	0.297	0.327	5
J	SS-II	SS-II	0.594	0.594	0.594	0.594	495
K	SS-HT	SS	0.178	0.178	0.178	0.178	1
L	ST	ST-HT	0.060	0.060	0.060	0.089	4
M	SS	ST-HT	0.238	0.327	0.297	0.386	9
N	ST-HT	ST-HT	0.268	0.297	0.297	0.297	6
O	SS-HT	ST-HT	0.268	0.416	0.446	0.505	146
P	ST	SS-HT	0.178	0.178	0.178	0.178	2
Q	SS-HT	SS-HT	0.238	0.238	0.238	0.238	4

a. The coefficient of friction at time 5 indicates the start of testing with a load of 50 lbf. Additional friction data was recorded at 10, 15, and 20 minutes thereafter. The wear measurement was taken at time t = 20 minutes, upon completion of testing.

b. SS = stainless steel; ST = stellite No. 6; II = ion implantation processed; HT = Hi-T Lube processed.

c. 14.4108 teeth = 0.001 inches of total wear.

of the sample material because it is believed that the linear velocity at (pin V-block) contact and the contact stress far exceeded normal or abnormal valve operation. Therefore, further research may be necessary to clarify the results and determine an alternate path to choosing the final surface modification technique for a full-scale valve test program. All material samples, regardless of composition or surface treatment, exhibited moderate to extreme wear. Microscopy analysis (scanning electron microscope) may be required to determine the composition of the surface layers of the samples and to see if any surface treatment remains. If the treated surfaces no longer exist, then the final reported results may require further clarification.

Notwithstanding these observations, the overall testing results were very encouraging. The Hi-T Lube processed samples showed improvement in friction and wear. The ion implantation processed samples showed improvement in wear. And, the testing confirmed the conclusion that stainless steel sliding surfaces should be "hardfaced" where possible.

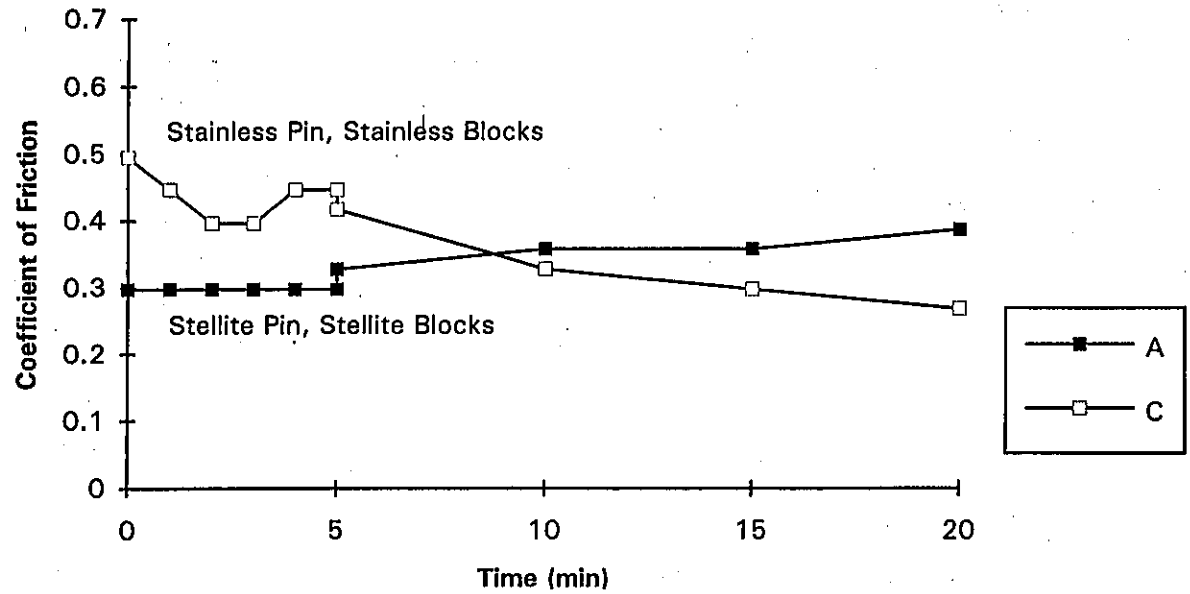

Figure 1. Friction results for untreated stainless steel and stellite samples.

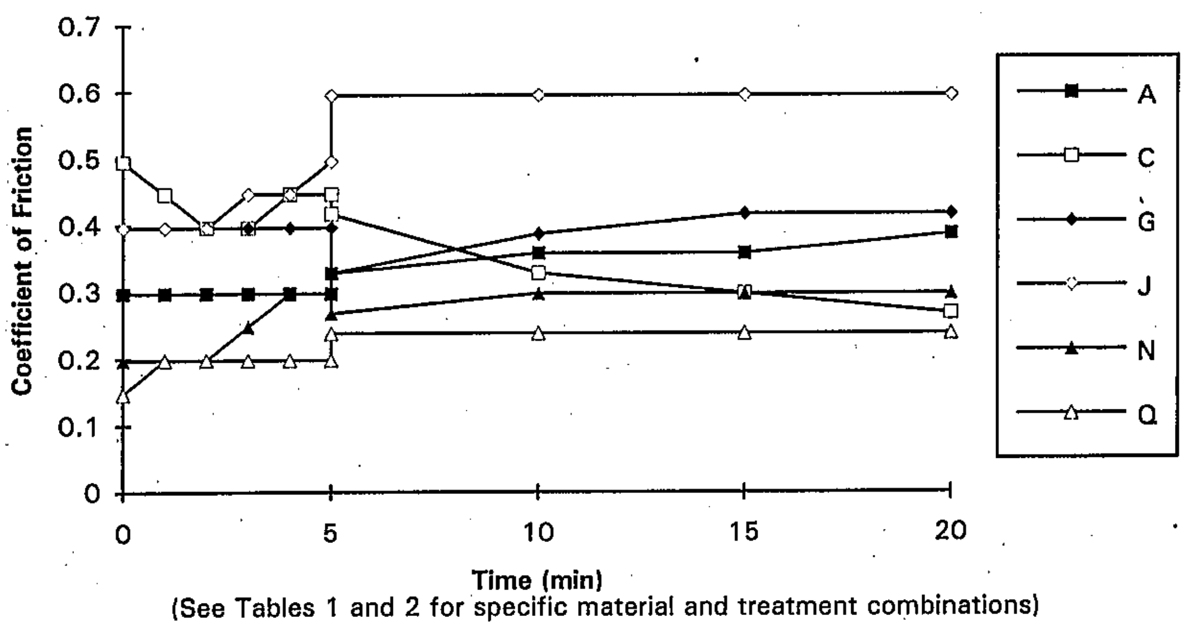

Figure 2. Friction results for samples with matching pin and V-block materials and treatments.

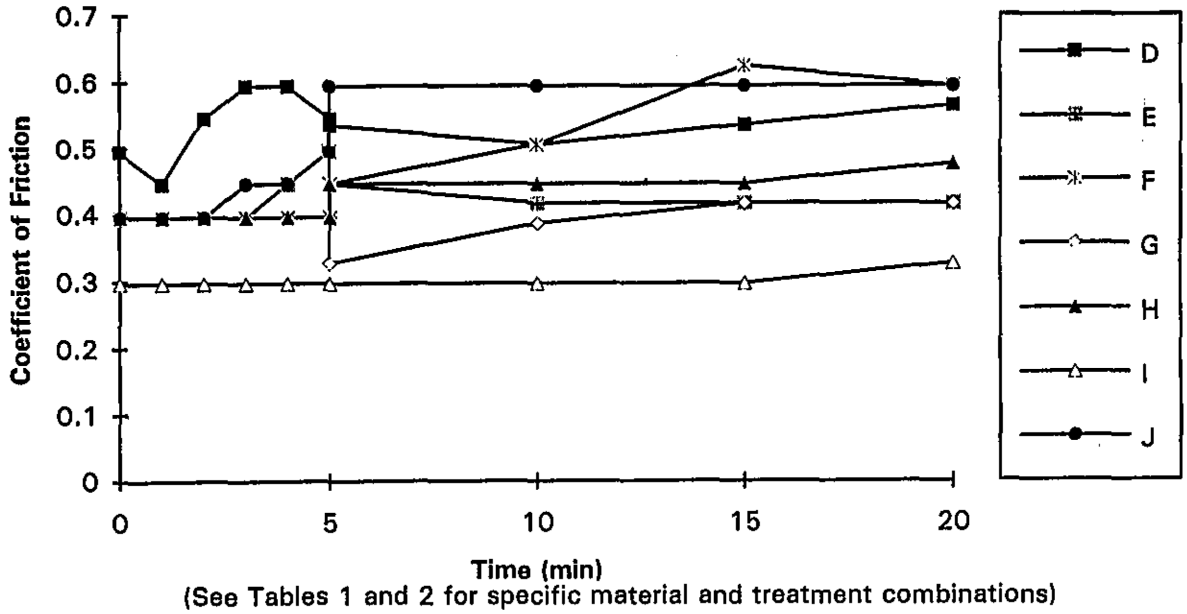

Figure 3. Friction results for ion implant processed samples.

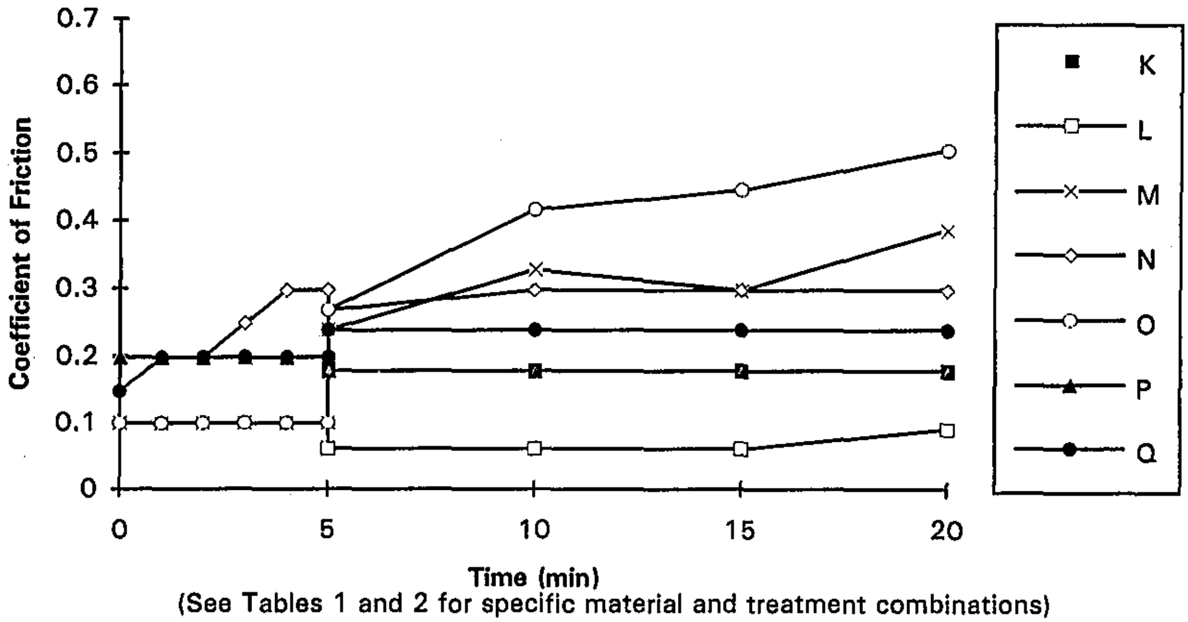

Figure 4. Friction results for Hi-T Lube processed samples.

Other Enhancement Techniques Considered

Several other enhanced surface treatment techniques were considered, but were eliminated from further research and sample testing. Plasma surfacing was discarded as a viable surface treatment technique for major internal valve components because its application would increase part dimensional limits beyond acceptable tolerances without major component modification and its adhesive lamination properties were suspect for its intended use in the program. Micro sealing may prove viable for reducing friction at the stem-stem nut interface (stainless steel on brass); however, the process does not result in true surface hardening and therefore was not considered suitable for use on wear-related internal components. And finally, vapor deposition was eliminated from further consideration because it is not suitable for long term wear in the extreme operating and postulated design-basis environments.

CONCLUSIONS AND RECOMMENDATIONS

Regarding the optimal valve tolerancing issue, MODGVALV is a simple yet effective tool that designers and manufacturers can use to statically model gate valve kinematics. Using BASIC as the programming language, MODGVALV can predict the location of contact on a gate valve's disc and downstream seat ring and thus denote potential locations of surface damage that can lead to abnormal closing forces and valve failure. The program also provides a method to study the generic effects of operational wear by selectively increasing the clearances between guiding surfaces, which, of course, will increase the likelihood of premature contact.

During its development cycle, it was noted that selective enhancements to the MODGVALV program could provide additional user benefits. Although beyond the scope of this research, enhancements, such as (a) modeling localized wear points, (b) quantifying wear factors for operational cycles, and (c) investigating elastic and plastic deformation, can provide valuable information to the designer and manufacturer alike. Whether these enhancements can be incorporated within the existing program or require a more advanced method of evaluation (i.e., finite element analysis) remains to be explored.

Regarding enhanced surface treatment techniques, the research results are very encouraging. The Hi-T Lube process showed apparent improvement in both friction and wear for both stainless steel and stellite hardfaced samples. This improvement was greater when only one of the two material components of a set was surface treated. The ion-implanted samples showed no improvement in friction, but showed good improvement in wear. The data for the untreated stainless steel and stellite hardfaced samples clearly support the conclusion that sliding surfaces in all stainless steel valves should be "hardfaced" where possible.

For unpredictable valves, enhanced surface treatments may be used to decrease friction and improve wear factors of MOV sliding surfaces. Such sliding surfaces include the disc guides and body guide rails, the stem, and the disc seats and body seat rings. Improvement of these surfaces, along with proper tolerancing, can prevent or mitigate internal damage such as gouging, galling, and excessive wear, which testing has shown can occur in some MOVs under severe operating conditions (high flow and high differential pressure). Prevention of these types of internal damage can significantly help make unpredictable valves predictable.

Further, these enhanced surface treatments and technologies may be suitable for applications involving predictable valves as well. To illustrate, the EPRI MOV Performance Prediction Program recently completed by Wyle demonstrated that even the valves that performed predictably and consistently exhibited apparent disc coefficients of friction somewhat higher than the widely accepted value of 0.3 during cold water pumped flow testing (after preconditioning) as well as during hot water (530°F) and steam blowdown testing. Because the maximum thrust required to open or close a valve is directly related to the

value of disc coefficient of friction, the higher the disc μ, the greater the stem thrust required. Application of one or more of the enhanced surface treatment techniques to the disc, guides, or valve body sealing surfaces could reduce the disc sliding coefficient of friction to 0.3 (or less), thereby avoiding more costly modifications required to counteract the effects of a high apparent disc μ. Further research and testing would be required to determine the actual improvement in friction and wear for a full-scale valve. And, other factors such as cost effectiveness, feasibility of application, life span of the proposed treatment (i.e., will the proposed treatment continue to be effective after 100, 200, 1,000 strokes?), and, of course, retrofitting techniques need to be evaluated. However, the benefits associated with even modest improvements in friction and wear factors for installed valves are readily apparent.

To maximize material sample testing results, further surface-modified (ion-implanted and Hi-T Lube) stellite material sample tests could be conducted using an alternate ASTM testing methodology (i.e., pin and disc). If undertaken, processes for cobalt reduction could also be explored. Nevertheless, the results of this research clearly show that the Hi-T Lube process should be highly regarded as a candidate for full-scale valve tests because it appears to provide a promising technology for retrofitting and refurbishing safety-related valves and valve components such as seats, seatrings, discs, disc guides, valve body guide rails, stems, and stem nuts.

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Allowable Stem Nut Wear and Diagnostic Monitoring for MOVs

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ABSTRACT

After a motor-operated valve (MOV) stem nut failure in 1991 that contributed to a forced plant shutdown, the FitzPatrick Plant staff developed criteria to check for excessive stem nut wear in MOVs. Allowable stem nut wear monitoring uses both direct dimensional measurement and diagnostic test data interpretation. The wear allowance is based on the recommended permitted backlash discussed in the Electric Power Research Institute/Nuclear Maintenance Assistance Center Technical Repair Guideline for the Limitorque SMB-000 Motor Actuator. The diagnostic analysis technique measures the time at zero load and compares this with a precalculated allowable zero force time. Excessive zero force time may be the result of other MOV problems, such as a loose stem nut lock nut or excessive free play in the drive sleeve bearing.

Stress levels for new or nominal stem nuts and stem nuts with the full wear allowance were compared. Bending and shear stresses at the thread root increase for the maximum wear condition when compared with a "new" stem nut. These stresses are directly related to the thread root thickness. For typical MOV loading and common stem threading (with two diameters of thread engagement), the thread stresses are well within acceptable limits for ASTM B584-C86300 (formerly B147-863) manganese bronze (typical stem nut material).

INTRODUCTION

Stem nut failures on motor-operated valves (MOVs) have occurred at several nuclear power plants resulting in increased concern for both utilities and regulators. Stem nut failures usually result from excessive wear from poor lubrication (lacking anti-wear characteristics), foreign material abrasion, or both. Nevertheless, significant wear that manifests itself through an on-demand or service failure will still occur with service over a considerable period of time. After an on-demand stem nut failure at the New York Power Authority's FitzPatrick plant, senior plant management requested that the MOV Engineering Group develop a method to identify excessive stem nut wear during overhaul maintenance and diagnostic testing. This paper will focus primarily on the diagnostic testing method for determining excessive wear.

DIAGNOSTIC TESTING

As part of the analysis procedure for most MOV diagnostic tests a zero load condition must be located on the traces that represent stem thrust. (Note: For this paper examples and nomenclature used will be for Liberty Technologies VOTES system. The same principles should apply to other test systems that measure MOV stem thrust.) Figure 1 shows the zero force transition that occurs going from stem compression in the close position to beginning the open valve stroke (O4 point from VOTES notation). Figure 2 shows the typical zero transition which takes place from the end of the open stroke to the beginning of the close stroke (C3 point from VOTES notation).

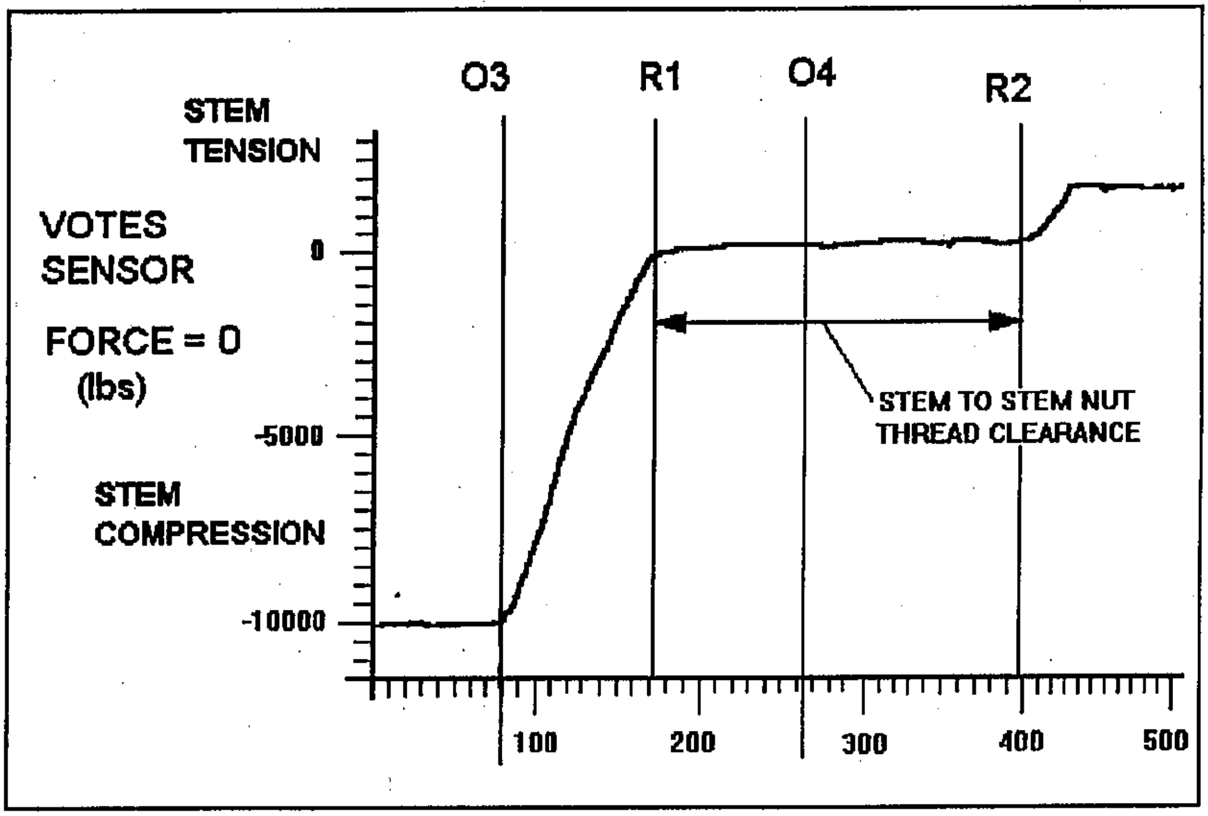

Figure 1. Opening zero force transition.

(Note that in Figures 1 and 2 the horizontal axis represents time in milliseconds and the vertical axis represents stem force in pounds with the sign convention that stem compressive or closing forces are negative and tensile or opening forces are positive.) For both of these cases the length of the zero force time (time between reference marks R1 and R2) corresponds to the time required for the stem nut to travel through the backlash clearance, δ_n, as shown on Figure 3. For a new or nominal stem nut this backlash clearance is a direct consequence of the Acme power screw thread tolerance and allowance. As the "softer" manganese bronze stem nut material wears (softer than the 400 series stainless steel materials commonly used for valve stems), the clearance, δ_w, increases as shown in Figure 4. Thus the zero force time for a "worn" stem nut may be significantly longer than for a "new" or nominal stem nut. The zero force time may be determined from the trace reference points R1 and R2, as shown in Figures 1 and 2.

The backlash clearance for a "worn" stem nut, δ_w, may be determined using Equation (1):

$$\delta_w = \text{Stem Speed} * \text{Zero Time} \qquad (1)$$

(Note: The effect of tolerance and allowance on backlash clearance is discussed in Appendix 1.)

ACCEPTABLE WEAR

We (the FitzPatrick Plant) adopted the backlash limit recommended by the Limitorque Corporation[a] from EPRI/NMAC Report NP-6229 (Myers et al., 1989, Table 16-1) as listed in

a. Personal communication from D. S. Warsing (Limitorque Corporation) concerning the technical repair guidelines for Limitorque SMB-000 actuator, April 5, 1988.

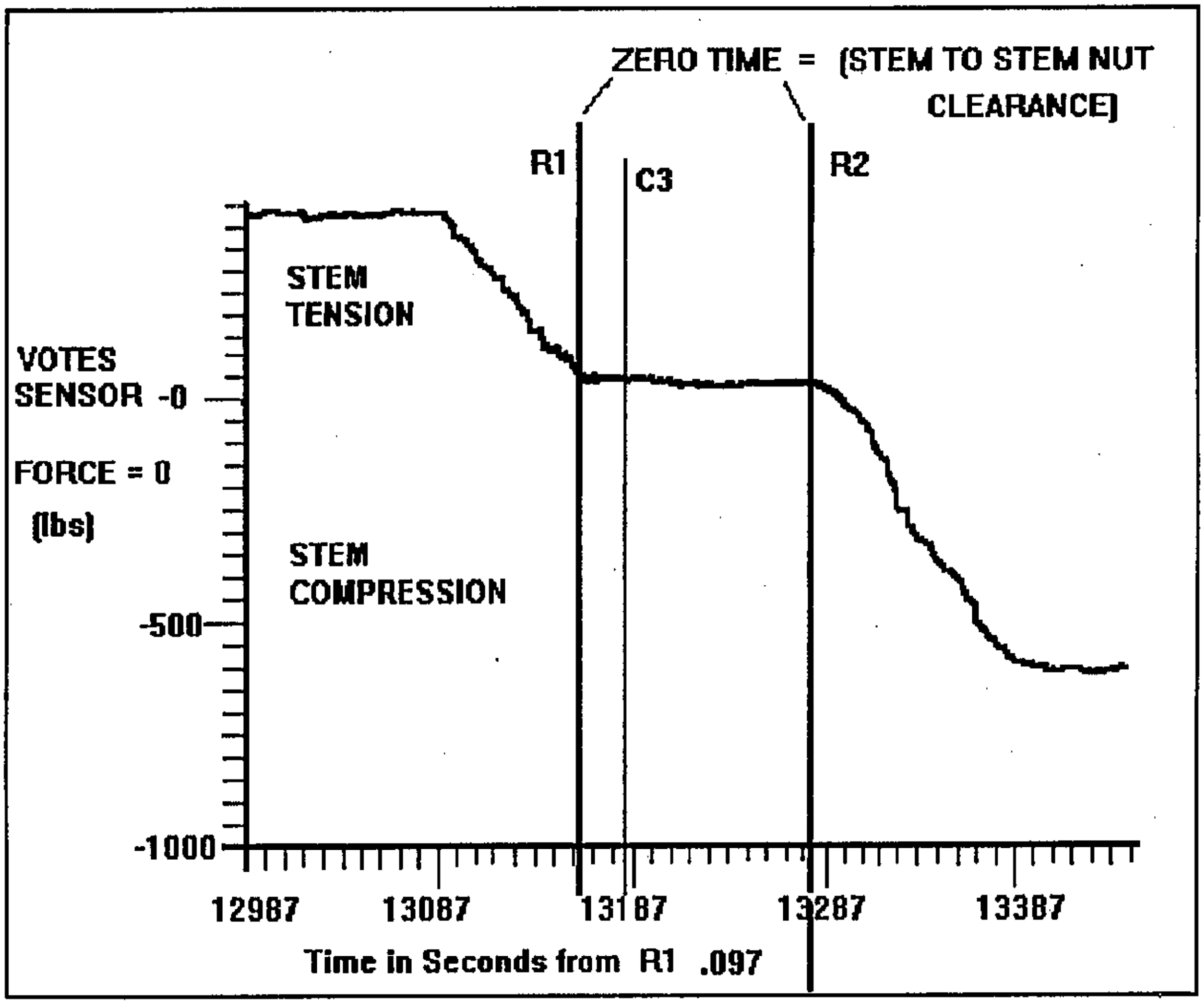

Figure 2. Closing zero force transition.

Table 1. As may be noted from Table 1, the allowed backlash is equivalent to one-eighth of the pitch length or 25% of the thread thickness at the pitch diameter. Because of the allowance and tolerance in Acme threads, you may have this limiting backlash with considerably less than 25% of the thread thickness worn. Our usual method for diagnostic testing requires calculation of a maximum zero force time based on the allowed backlash and the nominal or design motor speed, stem lead, and actuator overall gear ratio (OGR). During diagnostic test analysis we compare the measured zero force time with the calculated allowed zero force time. A measured zero force time less than the calculated limit indicates acceptable backlash, which implies an acceptable level of stem nut wear. A measured zero force time greater than the calculated allowed time indicates only a potential problem with stem nut wear. This is because the excessive "free play" or backlash (zero force time) could result from other factors, such as a loose or backed out stem nut lock nut (see the following discussion on Limitations).

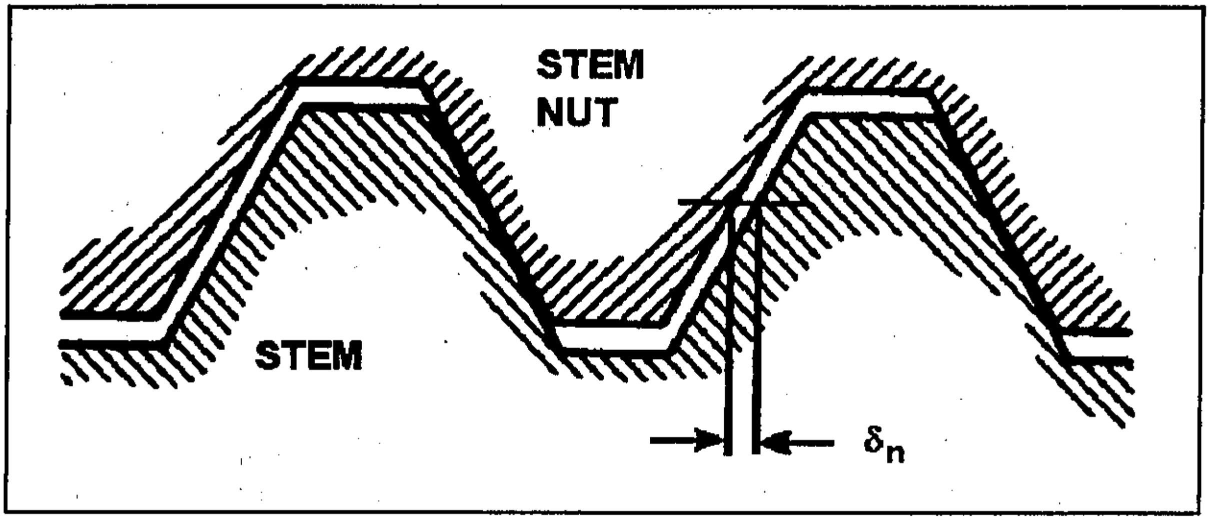

Figure 3. Backlash clearance for new stem nut (from Shigley et al., 1986).

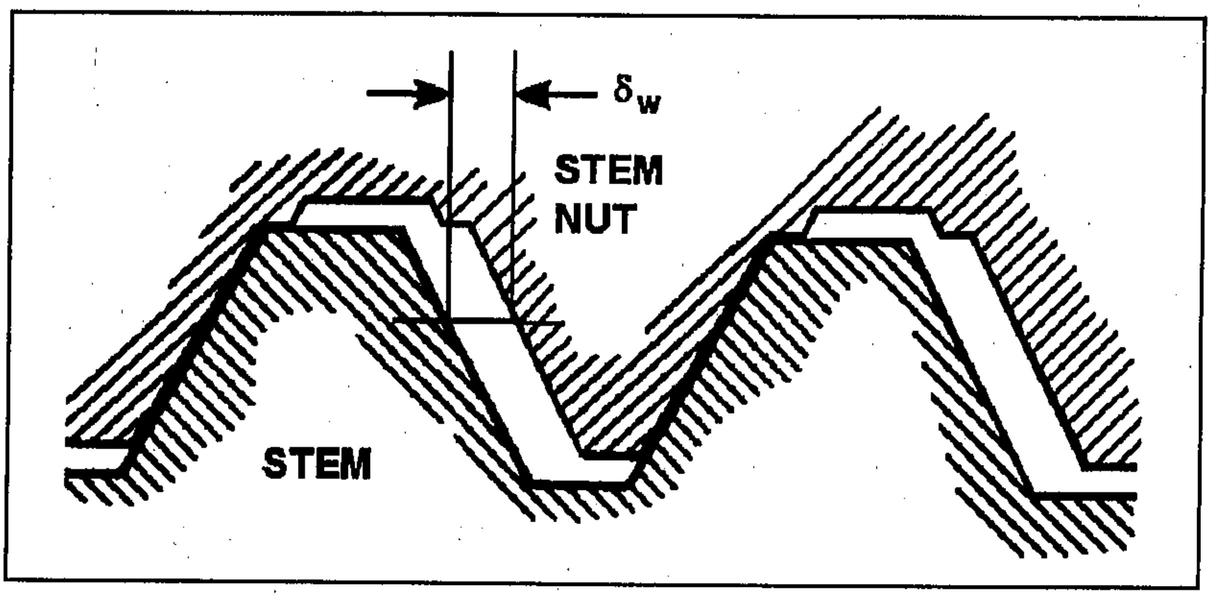

Figure 4. Backlash clearance for worn stem nut (adapted from Shigley et al., 1986).

The allowed or maximum zero force time is given by

$$\text{maximum zero force time (s)} = \frac{7.5 * OGR * p}{RPM * L} \quad (2)$$

where

OGR = actuator overall gear ratio (unit ratio)

P = stem thread pitch (in.)

RPM = nominal motor speed (rpm)

L = stem lead (in./rev).

We generally use the nominal Limitorque design motor speeds (1,700 and 3,400 rpm for three-phase ac motors and 1,900 rpm for dc motors). Because the zero stem force transition results in a relatively unloaded motor running

Table 1. Allowed backlash from stem nut wear (from Myers et al., 1989).

Threads per inch	Pitch (in.)	Allowed backlash (in.)
2	0.5000	0.0625
3	0.3333	0.0417
3.5	0.2857	0.0357
4	0.2500	0.0313
5	0.2000	0.0250
6	0.1667	0.0208
7	0.1429	0.0179
8	0.1250	0.0156
10	1.1000	0.0125

condition (just unloaded gear train drag), we could expect a slightly higher motor speed. For three-phase ac motors the speed may approach the synchronous speed (1,800 or 3,600 rpm for 60 Hz). Using the synchronous motor speed in place of the design speed will reduce the calculated allowed zero force time by about 5.6%, which is not significant for the purposes of this analysis. For dc motors, the motor speed may increase much more significantly when running under low load conditions such as during the zero force transition. For dc motors it may be advisable to calculate the zero force time based on a low torque load motor speed as determined from a generic or test motor curve. Motor current measurements from diagnostic testing for operation during the zero force transition may be helpful to estimate this low torque load and motor speed. Also the "lost motion" or hammerblow time may be used to obtain a better estimate of motor speed at low load conditions. Motor speed may be estimated determining the angular rotation for hammerblow action from the drive sleeve and worm gear geometry. The time for the hammerblow action may be determined from MOV diagnostic test data (current or power and stem thrust). With this information and the actuator overall gear ratio, one can easily calculate the motor speed at low load conditions.

LIMITATIONS

The zero force time diagnostic check has several limitations, but is still a useful check for stem nut wear or other causes of excessive drive train backlash. It should be used to indicate a potential problem that needs additional investigation with disassembly and inspection. Specific limitations and considerations are

- Unclear Beginning and End for Zero Force Area

  The zero force area may not have clearly defined beginning and ending points. This makes it difficult to mark and calculate the zero force time. Limitorque SB type operators, which allow the stem nut to "float" and load a compensating thrust spring, are particularly prone to "noisy" and unclear zero force transitions.

- Other Causes for Long Zero Force Time

  A long zero force time may result from some other source of free play in the thrust load drive train. A likely cause is a loose or "backed-out" stem nut lock nut. A loose or improperly positioned stem nut lock nut will allow the stem nut to slide up and down along the drive sleeve splines as the load direction changes. A loose stem nut lock nut will often cause a very long zero force time (on the order of seconds), much longer than the calculated limit. Free play in the drive sleeve bearing can also result in an excessive zero force time. The free play usually results from missing shims under the lower drive sleeve bearing that are used to compensate for casting differences and to properly align the worm and worm gear. Under these circumstances the whole drive sleeve may shift up and down as the load direction changes.

- Thrust Bearing Clearance in Early SMB 4 Actuators

  An early design Limitorque SMB 4 actuator (before 1971) incorporates a separate

dual-acting ball-type thrust bearing rather than the tapered roller thrust bearings employed in other Limitorque operators. The early SMB 4 design incorporates a clearance in the thrust bearing to allow the bearing to transmit thrust loads in both directions. This clearance adds to the zero force transition time. Therefore, the measured zero force time results from both the stem to stem nut clearance and the thrust bearing clearance.

- Effect of Allowance and Tolerance on Zero Force Time

The combination of pitch diameter allowance and maximum tolerance difference may result in a significant contribution to the zero force time. The new stem nut clearance, δ_n, as shown in Figure 3, results entirely from allowance and tolerance. The amount that tolerance and allowance can contribute to the total clearance and zero force time is relatively greater for small diameter stems with a low number of threads per inch (TPI) (or long pitch length). The minimum clearance results from only the allowance (minimum nut with maximum stem pitch diameters). The maximum clearance occurs with allowance and maximum tolerance combination (maximum nut with minimum stem pitch diameters). From ANSI B1.5-1977 and Machinery's Handbook (Oberg et al., 1988), the new or nominal minimum and maximum axial clearances, $\delta_{n,min}$ and $\delta_{n,max}$, for the 2G fit class generally used for MOV stems, are given by

$$\delta_{n,min} = \tan(14.5) * 0.008 * \sqrt{D} ; \quad (3)$$

$$\delta_{n,max} = \tan(14.5) * \left[0.020 * \sqrt{D} + 0.060 * \sqrt{p}\right] , \quad (4)$$

where

D = stem nominal (outside) diameter (in.)

p = pitch length (in.).

Table 2 tabulates these new or nominal clearances and a compares the allowed backlash for common stem diameter and pitch thread per inch combinations. (The details of the calculations for Table 2 are provided in Appendix 1.) The maximum clearances are the result of a limiting or maximum combination to tolerance on both the

Table 2. Allowed backlash and new stem nut clearances.[a]

Diameter (in.)	Threads per inch	Pitch (in.)	Allowed backlash (in.)	$\delta_{n,min}$	$\delta_{n,min}$ as percent backlash	$\delta_{n,max}$	$\delta_{n,max}$ as percent backlash
0.75	6	0.1667	0.0208	0.0018	8.65	0.0108	51.92
1.00	5	0.2000	0.0250	0.0021	8.40	0.0121	48.40
1.25	5	0.2000	0.0250	0.0023	9.20	0.0127	50.80
1.50	4	0.2500	0.0313	0.0025	7.99	0.0141	45.05
2.00	4	0.2500	0.0313	0.0029	9.27	0.0151	48.24
2.50	3	0.3333	0.0417	0.0033	7.91	0.0171	41.01
3.00	2	0.5000	0.0625	0.0036	5.76	0.0199	31.84
4.00	2	0.5000	0.0625	0.0041	6.56	0.0213	34.08

a. Based on Table 16-1 from Myers and White, (1989) and ANSI B1.5 and B1.8 (Oberg et al., 1988).

stem and stem nut. This should be a fairly unlikely combination. The worst result of this maximum clearance combination would be that the zero force time analysis could indicate excessive wear when actual wear is considerably less than 25% of the thread thickness at the pitch diameter or 0.125 pitch (0.125p). This situation would be discovered when the suspected stem nut was removed for inspection.

INSPECTION LIMITS

For inspection of suspected worn stem nuts, we apply a very conservative acceptable wear limit of less than 25% of the thread thickness at the pitch diameter or 0.125p. We estimate the amount of wear by measuring the stem nut (internal) thread crest width. Nominally, this width is 0.3707p for a "new" stem nut with General Purpose Acme threads and 0.4224p for Stub Acme threads. Thus, our limits for acceptable wear are stem nut thread crest measurements of 0.2457p for General Purpose and 0.2974p for Stub Acme threads.

STRESS ANALYSIS

Because we are considering a worn stem nut thread as an allowable condition, we need to calculate the effect of reduced thread width on stress levels. This is particularly important now as many MOV users are applying the Kalsi Engineering research (Kalsi et al., 1991) to extend the thrust ratings of their Limitorque actuators. Shigley (1986) provides formulae for thread stress for bearing, bending, shear, and circumferential or hoop stress in the nut. Only bending and shear stress depend on the thread width and thus would be increased by wear. For this analysis and as discussed in the examples in Appendix 2 and 3, we will only consider bending and shear stress to increase as a result of stem nut wear. From Shigley, (1986) the bending stress at the thread root may be approximated by representing the thread as a cantilever of width, $\pi N_e d_p$, and depth, w, and supporting a concentrated load, F, at a distance of $(D - d_p + 2a)/2$ from the fixed end. From the flexual formula, Equation (5) gives the bending stress as

$$\delta = \frac{3\ F(D - d_p + 2a)}{\pi d_p\ N_e\ w^2}\ , \tag{5}$$

where

F	=	force
D	=	major diameter
d_p	=	pitch diameter
2a	=	allowance plus maximum tolerance for nut major diameter
N_e	=	number of threads engaged
w	=	width of thread.

Assuming a General Purpose Acme thread such that $d_p = D - p/2$ with 2 diameters of thread engagement ($N_e = 2D/p$), Equation (5) may be rewritten as

$$\sigma = \frac{6Fp(\frac{p}{4} + a)}{\pi w^2 D(2D - p)}\ . \tag{6}$$

For the worn stem nut case we need to consider the bending stress at the reduced thread thickness. Thus, the bending stress for a worn stem nut thread, σ_w, with a thickness, w_w, is given by

$$\sigma_w = \frac{3Fp^2}{2\pi w_w^2 D(2D - P)}\ . \tag{7}$$

Also from Shigley, the shear stress for the internal nut thread may be approximated by using an area, $\pi D w N_e$, and the commonly accepted 4/3 factor associated with a parabolic shear stress distribution across the tooth width. Thus Equation (8) gives

$$\tau = \frac{4F}{3\pi D w N_e}\ . \tag{8}$$

For two diameters of thread engagement ($N_e = 2D/p$) we can substitute in Equation (8) to give

$$\tau = \frac{2Fp}{3\pi D^2 w} \quad . \tag{9}$$

Equation (9) may be used for both a new or worn stem nut (by using the new or worn thread thickness).

Equations (6), (7), and (9) are applied in Appendix 2 to calculate bending and shear stress levels for specific examples. (Note that all Appendix calculations were performed with MathSoft MATHCAD 5.0 software.) The Appendix 2 calculations show how both bending and shear stress increase for decreasing stem diameter. They also demonstrate that for some typical stem nut and Limitorque actuator configurations, the stress levels for a 25% worn (on pitch diameter thread thickness) stem nut are well below the minimum yield strength (60,000 psi) for the stem nut material (B584-C86300).

For unusual cases of relatively small stems with larger, high-thrust Limitorque actuators, specific analyses should be performed [using Equations (6), (7), and (9)]. If needed for such a special application, the stress levels may be reduced by increasing the length of thread engagement above two diameters. This simplified stress analysis assumes even distribution of stress among the threads engaged. Because of different elastic stiffness between the stem and stem nut, increased thread engagement may result in a more uneven stress distribution. This may partially counteract the peak stress reduction from increased thread engagement.

The Appendix 2 calculations considered bending and shear stresses separately. Appendix 3 combined these stresses (with the simplifying plane stress assumption) to calculate the principal stress across the width of the worn thread root. This calculation shows that the bending and shear stresses do not combine to create any principal stress greater than the maximum bending stress.

CONCLUSION

MOV thrust diagnostic testing may be used to monitor wear by measuring the zero force time. An allowed backlash equivalent to 25% of the pitch diameter thread thickness (or 0.125p) is a very reasonable and practical operational limit for stem nut wear. Because of tolerance and allowance, the apparent wear based on zero force time will generally be conservative compared with an actual wear limit of 25% of the pitch diameter thickness. Stress analysis shows significant margin for the "worn" stem nut when compared with stem nut material yield strength limit.

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Cubberly, W. H., et al. (eds.), 1979, *Metals Handbook Ninth Edition, Volume 2 Properties and Selection: Nonferrous Alloys and Pure Metals,* American Society for Metals, Metals Park, OH, pp. 383–411.

Kalsi, M. S., and Wang, J. K., 1991, *Thrust Rating Increase of Limitorque SMB-000, SMB-00, SMB-0 and SMB-1 Actuators,* Document No. 1707C, Revision 0, Kalsi Engineering, Inc., Sugar Land, TX.

Myers, R. F., and White, M. L., 1989, *Technical Repair Guidelines for the Limitorque Model SMB-000 Valve Actuator,* Report NP-6229, Nuclear Maintenance Application Center, Electric Power Research Institute, Palo Alto, CA, p. 16-14.

Oberg, E., et al., 1988, *Machinery's Handbook 23rd Edition,* Industrial Press, New York, pp. 1556–1583.

Shigley, J. E., Mischke, C. R. (eds.), 1986, *Standard Handbook of Machine Design,* McGraw Hill, New York, pp. 20.2–20.12.

Appendix 1
Backlash and Clearance Analysis

Appendix 1 **Page 1 of 2**

BACKLASH AND CLEARANCE ANALYSIS

REFERENCE: Oberg et al, (1988) and ANSI B1.5 and B1.8

The following analysis compares the maximum allowed wear backlash and the range of "new" or nominal stem nut clearance. The minimum clearance occurs with only the 2G allowance while the maximum clearance is for both allowance combined maximum tolerance for the stem and stem nut.

Consider a range of stem diameters and threads per inch. (Note that some diameters and threads per inch would be unlikely combinations)

$i := 1,2..8$ (index for diameters) $j := 1,2..5$ (index for threads per inch)

Stem diameters / Threads per inch

$D_i :=$

0.75
1
1.25
1.5
2
2.5
3
4

$N_j :=$

6
5
4
3
2

$p_j := \frac{1}{N_j}$ (thread pitch)

$\delta_{w_j} := \frac{p_j}{8}$ (allowable backlash from stem nut wear)

$\delta_{n\_min_i} := \tan(14.5 \cdot deg) \cdot 0.008 \cdot \sqrt{D_i}$ (minimum new stem nut axial clearance, from 2G allowance only)

$\delta_{n\_max_{i,j}} := \tan(14.5 \cdot deg) \cdot \left(0.02 \cdot \sqrt{D_i} + 0.06 \cdot \sqrt{p_j}\right)$ (maximum new stem nut axial clearance, from 2G allowance and maximum tolerance)

Minimum axial clearances depend only on stem diameter:

D_i	$\frac{\delta_{n\_min_i}}{10^{-3}}$
0.75	1.792
1	2.069
1.25	2.313
1.5	2.534
2	2.926
2.5	3.271
3	3.584
4	4.138

Maximum axial clearances depend both on diameter and threads per inch or pitch:

$p_1 = 0.167$ $p_2 = 0.2$ $p_3 = 0.25$ $p_4 = 0.333$ $p_5 = 0.5$

$\frac{\delta_{w_1}}{10^{-3}} = 20.833$ $\frac{\delta_{w_2}}{10^{-3}} = 25$ $\frac{\delta_{w_3}}{10^{-3}} = 31.25$ $\frac{\delta_{w_4}}{10^{-3}} = 41.667$ $\frac{\delta_{w_5}}{10^{-3}} = 62.5$

D_i	$\frac{\delta_{n\_max_{i,1}}}{10^{-3}}$	$\frac{\delta_{n\_max_{i,2}}}{10^{-3}}$	$\frac{\delta_{n\_max_{i,3}}}{10^{-3}}$	$\frac{\delta_{n\_max_{i,4}}}{10^{-3}}$	$\frac{\delta_{n\_max_{i,5}}}{10^{-3}}$
0.75	10.814	11.419	12.238	13.438	15.452
1	11.507	12.112	12.931	14.131	16.145
1.25	12.118	12.722	13.541	14.742	16.755
1.5	12.67	13.274	14.093	15.294	17.307
2	13.65	14.254	15.073	16.274	18.287
2.5	14.513	15.118	15.937	17.137	19.15
3	15.294	15.898	16.717	17.918	19.931
4	16.68	17.284	18.103	19.303	21.317

D_i	$\frac{\delta_{n\_max_{i,1}}}{\delta_{w_1}}$	$\frac{\delta_{n\_max_{i,2}}}{\delta_{w_2}}$	$\frac{\delta_{n\_max_{i,3}}}{\delta_{w_3}}$	$\frac{\delta_{n\_max_{i,4}}}{\delta_{w_4}}$	$\frac{\delta_{n\_max_{i,5}}}{\delta_{w_5}}$
0.75	0.519	0.457	0.392	0.323	0.247
1	0.552	0.484	0.414	0.339	0.258
1.25	0.582	0.509	0.433	0.354	0.268
1.5	0.608	0.531	0.451	0.367	0.277
2	0.655	0.57	0.482	0.391	0.293
2.5	0.697	0.605	0.51	0.411	0.306
3	0.734	0.636	0.535	0.43	0.319
4	0.801	0.691	0.579	0.463	0.341

CONCLUSION:

From the preceeding analysis we can see that the maximum clearance for new stem nut considering allowance and maximum tolerance may be a significant portion of the allowed backlash wear limit. The worst case combination calculated above is for a 4 inch diameter with 6 threads per inch (an unlikely combination) which has a maximum clearance of 80.1% of the allowed backlash wear. The worst consequence of this unlikely situation would be a false indication of excessive stem nut wear from diagnostic test results.

Appendix 2
Examples of Stem Nut Thread Stress Analysis

Appendix 2 **Page 1 of 5**

EXAMPLES OF STEM NUT THREAD STRESS ANALYSIS

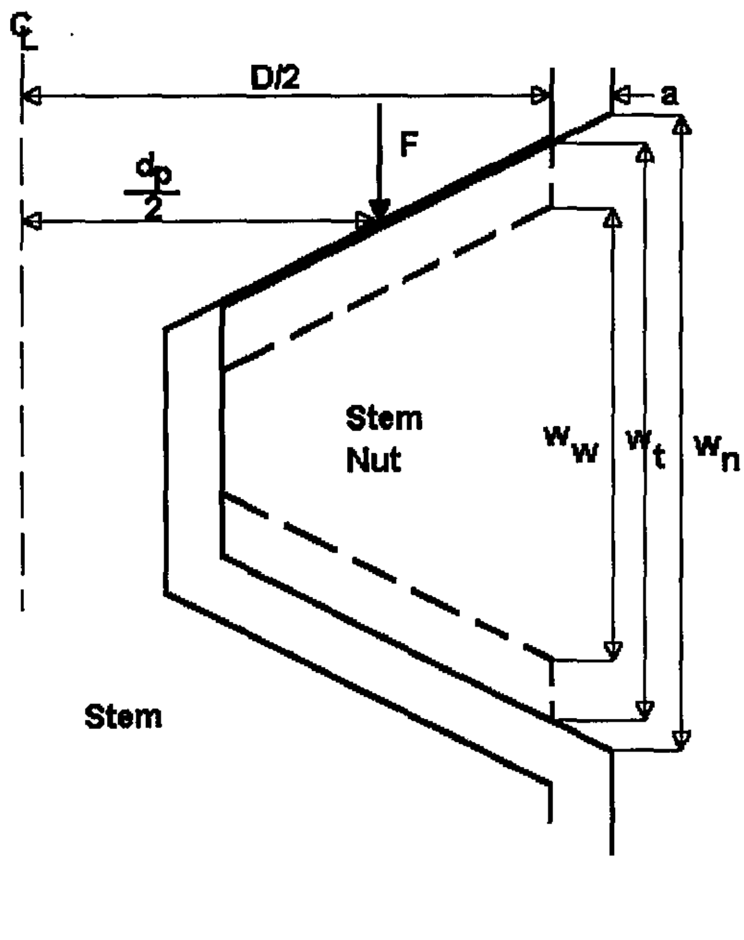
REFERENCES:

1. Shigley, et al, 1986.
2. Oberg, et al, 1988 and ANSI B1.5-1977.
3. Cubberly, et al, 1979 and ASTM B584, Rev. A, 1993.
4. Kalsi, et al, 1991.

ASSUMPTIONS:

1. Assume General Purpose Acme threads in accordance with ANSI B1.5-1977.
2. Assume thread engagement equals two diameters of the stem.
3. Assume for stress loading the thread may be modelled as a cantilever beam with a width equal to π times pitch diameter times number of thread engaged and depth equal to the root thickness with a concentrated force F at the pitch diameter.
4. For shear stress assume the commonly accepted 4/3 factor associated with the parabolic shear stress distribution across the thread root.

Figure A2-1 - Stem nut thread loading



Appendix 2 **Page 2 of 5**

Example 1 - Typical stem nut characteristics for Limitorque SMB-00

$F := 14000 \cdot lbf \cdot 162 \cdot \%$ $F = 22680 \cdot lbf$	(force for SMB-00 at Kalsi Engineering extended thrust limit)
$N := 4 \cdot \frac{1}{in}$	(threads per inch)
$p := \frac{1}{N}$ $\quad p = 0.25 \cdot in$	(thread pitch)
$D := 1.0 \cdot in, 1.125 \cdot in .. 2.0 \cdot in$	(stem diameter range from 1 to 2 inches)
$a := 0.020 \cdot in$	(radial allowance and max. tolerance on major diameter)
$w_t := 0.6293 \cdot p$	(shear width for new or nominal stem nut)
$w_n := w_t + 2 \cdot a \cdot \tan(14.5 \cdot deg)$	(width of thread root for new or nominal stem nut)
$w_w := w_t - 0.125 \cdot p$	(width of thread root for worn stem nut)

Applying the flexual formula as discussed in Shigley (Reference 1) the maximum bending stress at the thread root for a new stem nut, as a function of diameter, is given by:

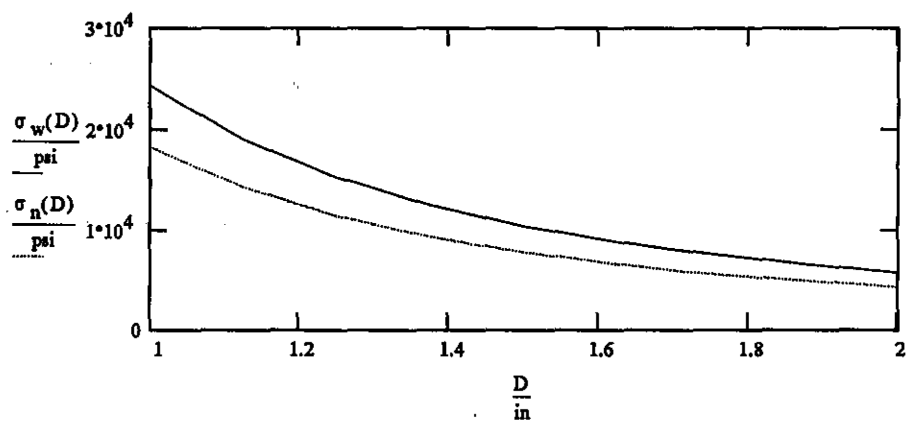
$$\sigma_n(D) := \frac{6 \cdot F \cdot p \cdot \left(\frac{p}{4} + a\right)}{\pi \cdot w_n^2 \cdot D \cdot (2 \cdot D - p)} \qquad \sigma_n(1.0 \cdot in) = 18159 \cdot psi \qquad \sigma_n(2.0 \cdot in) = 4237 \cdot psi$$

For the worn stem the maximum bending stress is given by:

$$\sigma_w(D) := \frac{3 \cdot F \cdot p^2}{2 \cdot \pi \cdot w_w^2 \cdot D \cdot (2 \cdot D - p)} \qquad \sigma_w(1.0 \cdot in) = 24331 \cdot psi \qquad \sigma_w(2.0 \cdot in) = 5677 \cdot psi$$

From ASTM B584, the minimum tensile yield stress for C86300 alloy manganese bronze is: $\sigma_y := 60000 \cdot psi$

Plotting bending stress for new and worn stem nuts vs. stem diameter:



Appendix 2 **Page 3 of 5**

For shear stress analysis consider the thread width, wt for a new stem nut and ww for a worn stem nut. As discussed in Shigley, apply the commonly accepted 4/3 factor associated with the parabolic shear stress distribution across the tooth width.

Thus the approximate shear stress for a new or nominal stem nut as a function of diameter is:

$$\tau_n(D) := \frac{2 \cdot F \cdot p}{3 \cdot \pi \cdot D^2 \cdot w_t} \qquad \tau_n(1.0 \cdot in) = 7648 \cdot psi \qquad \tau_n(2.0 \cdot in) = 1912 \cdot psi$$

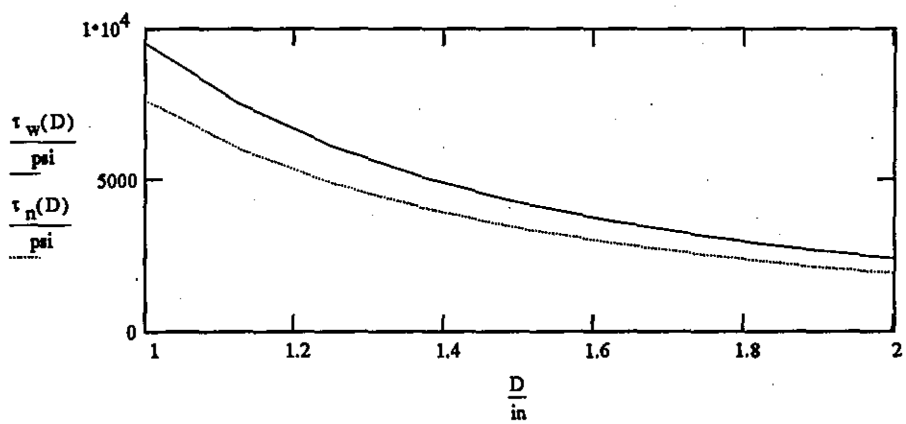
The approximate shear stress for a worn stem nut is:

$$\tau_w(D) := \frac{2 \cdot F \cdot p}{3 \cdot \pi \cdot D^2 \cdot w_w} \qquad \tau_w(1.0 \cdot in) = 9544 \cdot psi \qquad \tau_w(2.0 \cdot in) = 2386 \cdot psi$$

The shear yield stress is given by:

$$\tau_y := \frac{1}{\sqrt{3}} \cdot \sigma_y \qquad \tau_y = 34641 \cdot psi$$

Plotting shear stress for new and worn stem nuts vs. stem diameter:



This example shows considerable margin (greater than a factor of 3) to the tensile and shear yield stress limits.

Appendix 2 **Page 4 of 5**

Example 2 - Typical stem nut characteristics for Limitorque SMB-1

$F := 45000 \cdot lbf \cdot 162 \cdot \%$ $F = 72900 \cdot lbf$	**(force for SMB-1 at Kalsi Engineering extended thrust limit)**
$N := 3 \cdot \frac{1}{in}$	**(threads per inch)**
$p := \frac{1}{N} \qquad p = 0.333 \cdot in$	**(thread pitch)**
$D := 1.5 \cdot in, 1.6 \cdot in .. 3.0 \cdot in$	**(stem diameter range from 1.5 to 3 inches)**
$a := 0.020 \cdot in$	**(radial allowance and max. tolerance on major diameter)**
$w_t := 0.6293 \cdot p$	**(shear width for new or nominal stem nut)**
$w_n := w_t + 2 \cdot a \cdot \tan(14.5 \cdot deg)$	**(width of thread root for new or nominal stem nut)**
$w_w := w_t - 0.125 \cdot p$	**(width of thread root for worn stem nut)**

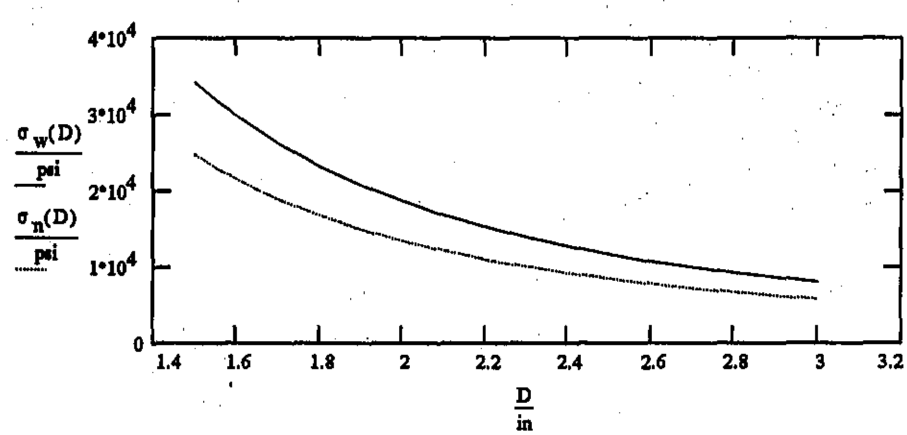
Bending stress for new stem nut:

$$\sigma_n(D) := \frac{6 \cdot F \cdot p \cdot \left(\frac{p}{4} + a\right)}{\pi \cdot w_n^{\,2} \cdot D \cdot (2 \cdot D - p)} \qquad \sigma_n(1.5 \cdot in) = 24746 \cdot psi \qquad \sigma_n(3.0 \cdot in) = 5823 \cdot psi$$

Bending stress for worn stem nut:

$$\sigma_w(D) := \frac{3 \cdot F \cdot p^2}{2 \cdot \pi \cdot w_w^{\,2} \cdot D \cdot (2 \cdot D - p)} \qquad \sigma_w(1.5 \cdot in) = 34216 \cdot psi \qquad \sigma_w(3.0 \cdot in) = 8051 \cdot psi$$

Plotting bending stress for new and worn stem nuts vs. stem diameter:



Appendix 2 **Page 5 of 5**

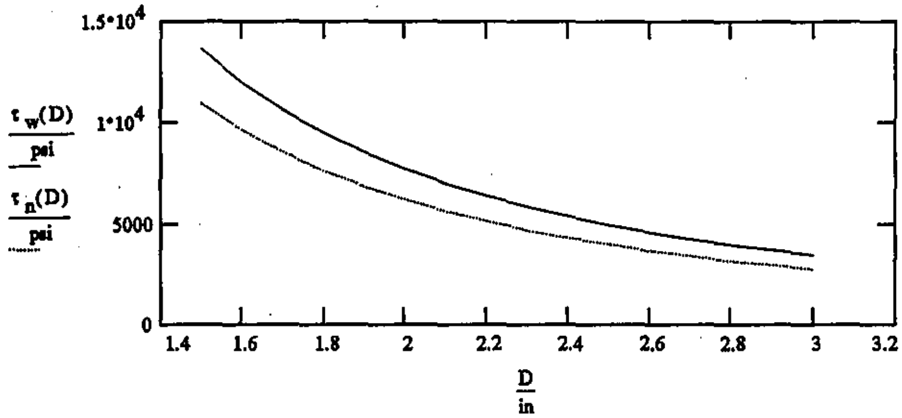
The approximate shear stress for a new or nominal stem nut:

$$\tau_n(D) := \frac{2 \cdot F \cdot p}{3 \cdot \pi \cdot D^2 \cdot w_t} \qquad \tau_n(1.5 \cdot in) = 10926 \cdot psi \qquad \tau_n(3.0 \cdot in) = 2731 \cdot psi$$

The approximate shear stress for a worn stem nut is:

$$\tau_w(D) := \frac{2 \cdot F \cdot p}{3 \cdot \pi \cdot D^2 \cdot w_w} \qquad \tau_w(1.5 \cdot in) = 13634 \cdot psi \qquad \tau_w(3.0 \cdot in) = 3408 \cdot psi$$

Plotting shear stress for new and worn stem nuts vs. stem diameter:



CONCLUSION:

Both of these examples show considerable margin to the tensile (bending) and shear yield stress limits. These examples also show that as diameter decreases the stress levels increase. Thus for most normal Limitorque application there should not be a stem nut stress problem even with extended (Kalsi) load limits and a "worn" stem nut condition. However, the diameter effects shows that for a situation where an unusally large Limitorque actuator is used with a relatively small stem diameter, then specific analysis for worn stem nut and extended thrust conditions is recommended. This analysis treated the bending and shear stresses seperately. Appendix 3 combines these stresses, with the plane stress simplification, to calculate the principal stress.

Appendix 3
Principal Stress Analysis Across Worn Thread Root

Appendix 3 **Page 1 of 4**

PRINCIPAL STRESS ANALYSIS ACROSS WORN THREAD ROOT

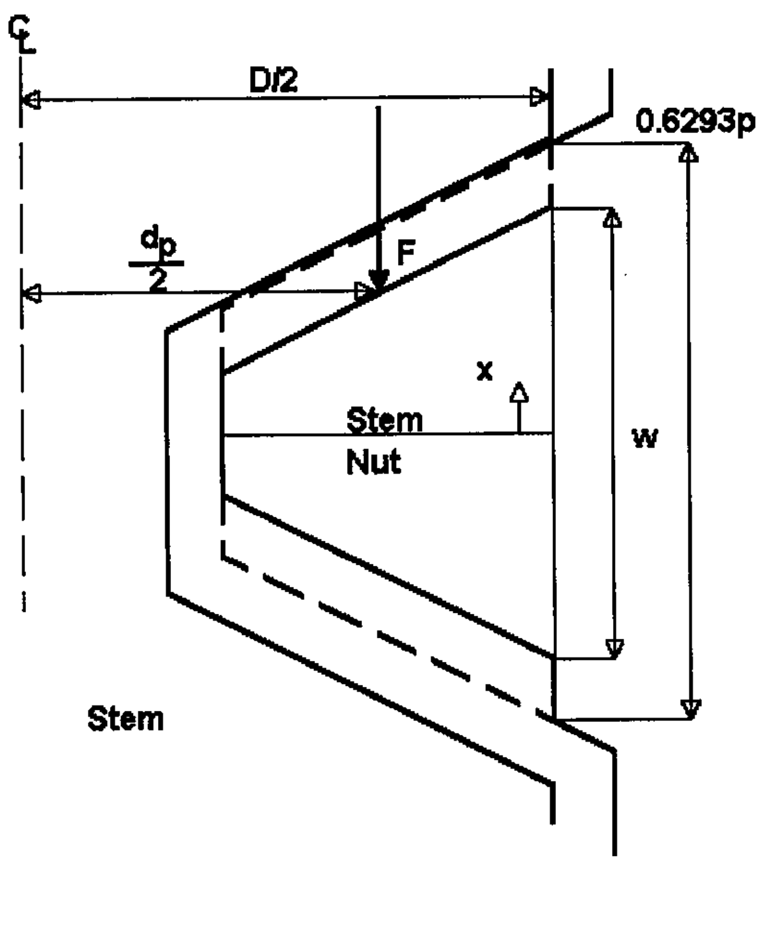
REFERENCES:

1. Shigley, et al, 1986.
2. Oberg, et al, 1988 and ANSI B1.5-1977.
3. Cubberly, et al, 1979 and ASTM B584, Rev. A, 1993.
4. Kalsi, et al, 1991.

ASSUMPTIONS:

1. Assume General Purpose Acme threads in accordance with ANSI B1.5-1977.
2. Assume thread engagement equals two diameters of the stem.
3. For bending stress assume the thread may be modelled as a cantilever beam with a width equal to π times pitch diameter times number of threads engaged and depth equal to the root thickness with a concentrated force F at the pitch diameter.
4. For shear stress assume the commonly accepted 4/3 factor associated with the parabolic shear stress distribution across the thread root.
5. Assume bending and shear combine as plane stress for simplified principal stress.
6. Assume wear of 25% of thread thickness at pitch diameter (or 0.125p).

Figure A3-1 - Stem nut thread loading



Appendix 3 **Page 2 of 4**

Example 1 - Typical stem nut characteristics for Limitorque SMB-00

$F := 14000 \cdot lbf \cdot 162 \cdot \%$ $\quad$ (force for SMB-00 at Kalsi Engineering extended thrust limit)

$F = 22680 \cdot lbf$

$N := 4 \cdot \frac{1}{in}$ $\quad$ (threads per inch)

$p := \frac{1}{N} \qquad p = 0.25 \cdot in$ $\quad$ (thread pitch)

$D := 1.0 \cdot in$ $\quad$ (stem diameter)

$w := 0.6293 \cdot p - 0.125 \cdot p$ $\quad$ (worn thread root thickness)

$M := \frac{F \cdot p}{4}$ $\quad$ (bending moment at root for force F at pitch diameter)

$I := \frac{\pi \cdot w^3 \cdot D}{6 \cdot p} \cdot \left(D - \frac{p}{2}\right)$ $\quad$ (2nd moment of inertia for cantilever thread root with 2 diameters of thread engagement)

$x := 0 \cdot in, 0.001 \cdot in .. \frac{w}{2}$ $\quad$ (varible for evaluating principal stress across the thread tooth width)

$\sigma_y := 60000 \cdot psi$ $\quad$ (from ASTM B584, the minimum tensile yield stress for C86300 alloy manganese bronze)

Bending stress as a function of x is: $\quad \sigma(x) := \frac{M \cdot x}{I}$

From Shigley maximum shear stress is given by:

$\tau_{max} := \frac{2 \cdot F \cdot p}{3 \cdot \pi \cdot w \cdot D^2} \qquad \tau_{max} = 9544 \cdot psi$ $\quad$ (assumes 4/3 factor for parabolic shear stress distribution)

For parabolic shear stress distribution with maximum at center (x=0), the shear stress as a function of x is:

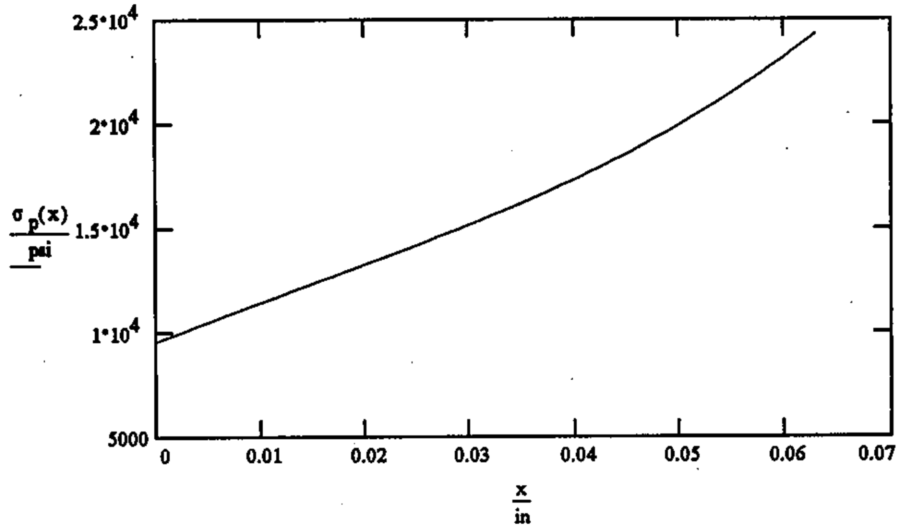
$$\tau(x) := \tau_{max} - \frac{4 \cdot \tau_{max}}{w^2} \cdot x^2$$

Combining for principal stress as a function of x:

$$\sigma_p(x) := \frac{\sigma(x)}{2} + \sqrt{\left(\frac{\sigma(x)}{2}\right)^2 + \tau(x)^2} \qquad \sigma_p(0 \cdot in) = 9544 \cdot psi \qquad \sigma_p\left(\frac{w}{2}\right) = 24331 \cdot psi$$

Appendix 3 **Page 3 of 4**

Plotting principal stress across worn thread root:



From the above graph it is clear that the shear and tensile stresses do not combine to produce at greater principal stress than the maximum bending stress at the root.

Example 2 - Typical stem nut characteristics for Limitorque SMB-1

$F := 45000 \cdot lbf \cdot 162 \cdot \%$ (force for SMB-1 at Kalsi Engineering extended thrust limit)

$F = 72900 \cdot lbf$

$N := 3 \cdot \frac{1}{in}$ (threads per inch)

$p := \frac{1}{N}$ $p = 0.333 \cdot in$ (thread pitch)

$D := 1.5 \cdot in$ (stem diameter)

$w := 0.6293 \cdot p - 0.125 \cdot p$ (worn thread root thickness)

$M := \frac{F \cdot p}{4}$ (bending moment at root for force F at pitch diameter)

$I := \frac{\pi \cdot w^3 \cdot D}{6 \cdot p} \cdot \left(D - \frac{p}{2}\right)$ (2nd moment of inertia for cantilever thread root with 2 diameters of thread engagement)

$x := 0 \cdot in, 0.001 \cdot in .. \frac{w}{2}$ (varible for evaluating principal stress across the thread tooth width)

Appendix 3 **Page 4 of 4**

Bending stress as a function of x is: $\sigma(x) := \frac{M \cdot x}{I}$

From Shigley maximum shear stress is given by:

$$\tau_{max} := \frac{2 \cdot F \cdot p}{3 \cdot \pi \cdot w \cdot D^2} \qquad \tau_{max} = 13634 \cdot psi$$

(assumes 4/3 factor for parabolic shear stress distribution)

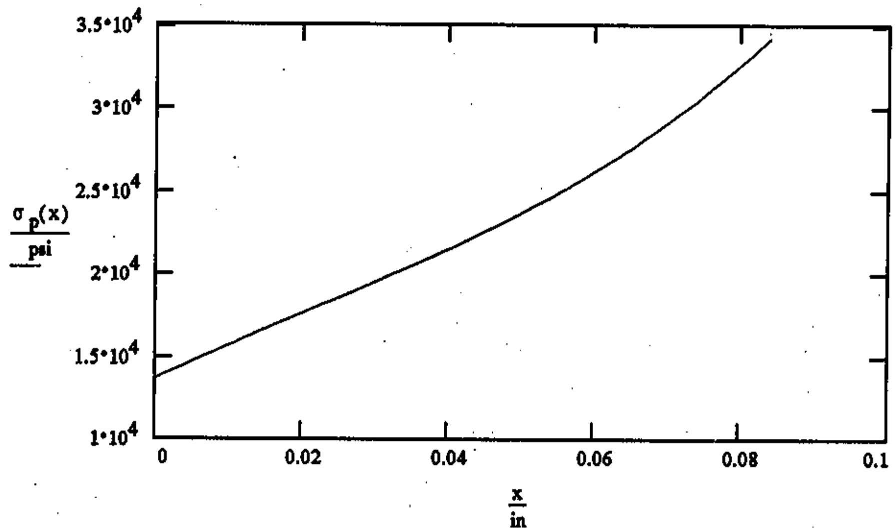
For parabolic shear stress distribution with maximum at center (x=0), the shear stress as a function of x is:

$$\tau(x) := \tau_{max} - \frac{4 \cdot \tau_{max}}{w^2} \cdot x^2$$

Combining for principal stress as a function of x:

$$\sigma_p(x) := \frac{\sigma(x)}{2} + \sqrt{\left(\frac{\sigma(x)}{2}\right)^2 + \tau(x)^2} \qquad \sigma_p(0 \cdot in) = 13634 \cdot psi \qquad \sigma_p\left(\frac{w}{2}\right) = 34216 \cdot psi$$

Plotting principal stress across worn thread root:



CONCLUSION:

Both of these examples show that for this simplified model the shear stress and bending tensile stress do not combine to create a maximum greater than the maximum tensile stress due the cantilever bending alone. For these examples the maximum tensile stresses were still well below the minimum yield strength of 60000 psi.

Results of the Motor-Operated Valve Engineering and Testing Program

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Comanche Peak Steam Electric Station

ABSTRACT

The Texas Utilities Electric Company (TU Electric) motor-operated valve (MOV) program for implementing the recommendations of Generic Letter 89-10 has typically included the following: refurbishing each actuator, verifying each actuator's as-built configuration, testing each actuator's motor on a dynamometer, testing each actuator's torque spring pack (which is used to control the torque developed), testing each fully refurbished and reassembled actuator on a torque test stand, and testing as many MOVs as practicable both without fluid flow through the valve and with the maximum test conditions reasonably achievable (static and differential pressure (DP) conditions, respectively). Test data are acquired at 1,000 samples per second for stem thrust, stem torque, stem position, actuator compensator spring pack deflection, actuator torque spring pack deflection, motor current, motor voltage, motor three-phase power, valve upstream pressure, and valve downstream pressure, wherever practicable. With this and other information, the following work has been accomplished:

- Equations used to predict stem thrust and stem torque requirements to close and open rising stem valves under both static and DP conditions have been verified for Comanche Peak gate and globe valves.

- Motor and gear train selection methods specified by the actuator manufacturer generally (but not always) underestimate motor capability; test data analysis has quantified actuator performance factors that allow credit to be taken for more motor capability to deliver torque to the worm gear.

- Actuator output torque at motor stall or at torque switch trip, if developed on a test stand that does not apply a thrust load to the actuator, is typically greater than the torque applied to a threaded valve stem (which does apply a thrust load to the actuator); this stem thrust effect has been accounted for in the methods used to determine appropriate torque switch settings, actuator gear ratios, and motor size.

- Differences between static and DP condition stem loads at close torque switch or close limit switch trip from the rate of loading effect have been justified for each group of similar MOVs; this decision permits the use of static testing to verify motor capability sufficiency and switch setting adequacy to ensure that the MOV will perform its design-basis functions under maximum design-basis conditions.

NOMENCLATURE

Disc position effect. A gate valve's disc may stop a small distance further out of the valve seat under differential pressure (DP) conditions than it does under static conditions. For limit switch controlled closure MOVs, this results in greater stem thrusts and greater stem torques being developed at and after close limit switch trip under DP conditions than under static conditions. For torque switch controlled MOVs, this difference in final disc position does not affect the torques or thrusts developed.

DP condition. Designates the condition with differential pressure across and fluid flow through the valve body.

DP test. Stroke test of the MOV under DP conditions.

Performance factor. The average value by which alternating current (ac) motor capability to deliver torque to the actuator worm gear (with power supplied at 80% of the motor nameplate voltage) exceeds the value predicted using the standard industry method.

Rate of loading effect. A given torque spring pack deflection value corresponds to greater stem thrust under static conditions than it does under DP conditions. This effect is sometimes referred to as load sensitive behavior. Thus, for a torque switch controlled MOV, the stem thrust at torque switch trip is greater under static conditions than under DP conditions. For a limit switch controlled MOV where the same thrust is developed under both static and DP conditions, the stem torque at limit switch trip is greater under DP conditions than under static conditions.

Static condition. Designates the condition without differential pressure across and fluid flow through the valve body.

Static test. Stroke test of the MOV under static conditions.

Stem thrust effect. The reduction of actuator output torque capability of the motor, the reduction of actuator output torque at torque switch trip, and (more generally) the reduction of actuator output torque for a given magnitude of spring pack force, resulting from a thrust load imparted by the valve stem to the actuator drive sleeve via the threaded stem nut.

Torque spring pack. The actuator spring pack which deflects in direct proportion to the torque output of the actuator.

BACKGROUND

In June 1989 the U.S. Nuclear Regulatory Commission (USNRC) issued Generic Letter 89-10 (GL 89-10) with a recommended date of June 1994 for completion of design-basis reviews, testing where practicable at or near design basis conditions, analysis of test data, incorporation of analysis results into the calculations and programs for appropriately setting control switches, implementation of appropriate switch settings, and establishing a program to maintain correct switch settings for the remainder of the plant operating life. This paper summarizes some of what Texas Utilities (TU) Electric Company has learned regarding motor-operated valve (MOV) performance characteristics, and the context in which TU Electric data analysis has been performed.

To verify actuator configuration and to ensure that test data were obtained for actuators in as good as new condition, all MOV actuators were refurbished before baseline testing. By means of spring pack, motor, actuator, and MOV tests, the following relationships were determined to assist in verifying the initial engineering assumptions:

• Torque switch setting	versus	Torque spring pack deflection (SPD)
• Torque spring pack force (SPF)	versus	Torque spring pack deflection (SPD)
		Actuator output torque (AOTQw) on torque test stand
		Actuator output torque (AOTQ) in situ
• Motor three-phase power and voltage	versus	Motor shaft torque and speed
		Actuator output torque (AOTQw) on torque test stand
		Actuator output torque (AOTQ) in situ
• Stem thrust	versus	Compensator spring pack deflection
		Stem torque (AOTQ) under static conditions
		Stem torque (AOTQ) under differential pressure (DP) conditions
		Differential and upstream pressure
		Stem travel under static and DP conditions

In the Summer of 1993, test data were analyzed for groups of similar actuators and similar valves to verify and adjust the initial engineering calculation assumptions. An early assessment of the impact of the test data analysis indicated that for a group of similar MOVs if the worst case values of running thrusts, valve factors, rate of loading effects, stem factors, and motor capability to deliver torque to the stem nut were all assumed to occur simultaneously for each MOV in the group, many groups of MOVs (which had demonstrated adequate capability to fulfill their design basis functions during testing) could not be shown by engineering calculation to be capable of fulfilling their design basis functions.

Thus, further analysis was performed to reduce the amount of conservatism retained in the calculation. The most significant amount of effort was expended in (a) performing a statistical analysis by which average and associated uncertainty values for each of these MOV performance parameters were determined for use in the engineering calculation, (b) revising the engineering calculation to use these results, and (c) revising the test procedures to use the revised calculation's results.

Laboratory tests have shown that the magnitude of the rate of loading effect for a given valve can be affected by adjusting the running thrusts. Therefore, it is important to note that TU Electric static and DP tests were conducted with normal packing loads, and with DP test conditions typically at or near the design-basis DP conditions. For groups of similar MOVs, TU Electric rate of loading effects may be considered statistically independent from running thrusts. This means that those MOVs that were not DP tested are most likely to have a rate of loading effect equal in magnitude to the average value determined for the MOVs tested under both static and DP conditions.

Similarly, valve factors and stem factors were determined from test data obtained under conditions at (or near) which the MOVs are expected to operate in the future. The analysis results obtained for groups of similar MOVs for running thrusts, valve factors, rate of loading effects, stem factors, and motor capabilities may therefore be considered independent of each other. A change in one parameter for one MOV is unlikely to appreciably change the average or the range of

that parameter or any other parameter determined for the group of similar MOVs.

Statistical methods have been applied to the test data for each MOV performance parameter to determine an uncertainty by which the average value of each parameter may be increased or decreased to obtain upper and lower bounds that bound the test data. Because the average and associated uncertainty of each MOV performance parameter are independent from the average and associated uncertainty of each other parameter (in the context discussed above), the various uncertainties may be combined by the square root of the sum of the squares (SRSS) technique in engineering calculations and in the test procedures. TU Electric typically determined uncertainties as the greater of (a) two sample standard deviations of the test data or (b) the difference between the average and the greatest data point. The uncertainty was then divided by the average value to obtain the uncertainty as a percentage of the average value.

It is very important to keep in mind that MOV performance parameter uncertainties calculated in this manner reflect both data scatter and measurement error. Averages tend to average out the uncertainties. By not correcting the test data before determining the averages and uncertainties, the engineering calculation reflects the true magnitudes of the uncertainties in the baseline test program's measurements.[a] The average valve factor for this group is within typical ranges, but the high uncertainty results in an unusually high upper bound value for the worst-case valve factor.

Comparison of actual test data with the results of SRSS combinations of uncertainties has confirmed the adequacy of this technique to provide a high degree of confidence that TU Electric's MOVs at Comanche Peak will operate properly under their maximum design-basis condition. The worst-case values for all parameters need not be assumed to occur simultaneously.

SUMMARY OF TEST RESULTS

Because of TU Electric MOV maintenance practices, test data collection, and analysis methods at Comanche Peak, these observations may be unique to Comanche Peak. These observations need not be assumed by others without confirmation of the effects for their MOVs, their data collection, and analysis methods. While the author has attempted to ensure that these data and discussions are correct and sufficiently detailed, it is inevitable that some aspects will not have been fully explained. The author may be contacted for further information.

The following subsections summarize the test data and analysis results. Appendix A provides a more detailed discussion of these summaries. The tables are located at the end of the text. Acronyms and terms used in the tables and in the paper are located in Appendix B.

Actuator Motor Capabilities

TU Electric has defined actuator ac motor design capability as 64% of the motor torque rating when the motor is supplied with power at 80% of the motor's nameplate voltage. Dynamometer testing in which a braking torque is gradually applied to the rotating motor shaft over 10 to 40 seconds is an easy and effective way to verify that a motor is not below this design capability. It is the author's opinion that MOV test programs should include some means to verify that the installed motors will produce the torques assumed in engineering calculations, and to justify the use of motors that may not.

In general, the actuator motors can produce their design capability. TU Electric has found a few random instances in which motors did not achieve this amount of torque before stalling. One set of nine identical motors removed from spare actuators in the warehouse all failed to produce their design capability torque.

a. High measurement uncertainty resulting from low load magnitudes contributes to the high uncertainty of the valve factor determined for valve group GT12 in Table 8c, as an example.

Actuator Performance Factors

IF (a) the actuator motor is supplied with power at 80% of the motor nameplate voltage, AND (b) the actuator output torque is gradually increased over several seconds, AND (c) the actuator drive sleeve is not subjected to a thrust load by the stem, THEN the average actuator output torque at motor stall is well above the value predicted by the standard industry methodology. Test data show that the standard industry methodology is conservative (underestimates the motor capability to produce actuator output torque) for all actuator configurations tested by TU Electric, except the Size 00 actuator with the 15-ft-lb 3,400-rpm motor, and a worm set gear ratio of 45:1. [See Tables 1 and 2, and Appendix A. (Note that the tables are located at the end of the paper before appendices.)]

Actuator Effective Moment Arms

TU Electric has determined from test data on torque test stands the average length for effective actuator moment arm, by which a torque spring pack force value can be multiplied to estimate the actuator output torque, and an uncertainty to bound the actual output torque. TU Electric has also determined the length and uncertainty using test data from in situ static and DP MOV tests, where the torque is measured by strain gages installed on the valve stem. The average effective moment arm length determined using in situ test data indicates that a thrust load on the actuator drive sleeve causes a loss of torque within the actuator that is not observed during torque stand testing. (See Table 3 and Appendix A.)

Actuator Stem Thrust Effects

TU Electric has determined (for several actuator configurations) the average reduction of actuator output torque caused by stem thrust loads of magnitudes required to properly operate Comanche Peak MOVs. Uncertainties associated with these reduction factors have also been determined. (See Tables 4 and 5, and Appendix A.)

Net Effect of Actuator Performance Factors and Stem Thrust Effects

In general, the use of the actuator manufacturer's specified pullout efficiency instead of the running efficiency is sufficient to bound the worst anticipated combined effect of the performance factor and the stem thrust effect. TU Electric test data analysis indicates that the use of the pullout efficiency is not sufficient in all cases. (See Tables 6a and 6b, and Appendix A.)

Running Thrusts

Running thrusts for Comanche Peak MOVs with stem diameters less than 1.25 in. are generally bounded by a load of 1,200 lb/in. of stem diameter. For larger stem diameters, the load per inch of stem diameter generally increases with increasing stem diameter up to 2,600 lb/in. for a 3-in. diameter stem. (See Table 7 and Appendix A.)

Valve Factors for Westinghouse and Borg-Warner Gate Valves

TU Electric data analysis has determined average closing stroke valve factors as low as 0.23 and as high as 0.56 for Westinghouse-manufactured gate valves under pumped flow conditions. The Westinghouse-specified valve factors for these MOVs are greater than the average test results. However, the data scatter indicates a slight possibility for some valves in some groups of similar valves to have valve factors greater than specified by Westinghouse. The pilot-operated relief valve (PORV) block valves tested under blowdown conditions had valve factors of 0.67, which exceeds the 0.56 value specified by Westinghouse. The valve factors determined for Westinghouse valves are based on the vendor-specified inside diameter of the valve body seat ring plus 1/16 in.

TU Electric data analysis has determined average closing stroke valve factors as low as 0.24 and

as high as 0.50 for Borg-Warner gate valves under pumped flow conditions. The data scatter indicates a possibility for some valves in some groups of similar valves to have valve factors as high as 0.63. The valve factors determined for Borg-Warner valves are based on the average of the inside and outside valve seating surface design diameters specified by the manufacturer.

TU Electric test data analysis has determined both closing and opening stroke valve factors that are demonstrated to be sufficient for predicting the stem thrusts required to close and to open Comanche Peak MOVs under their most severe design-basis conditions. (See Tables 8a, 8b, 8c, and 8d, and Appendix A.)

Repeatability of Stem Thrust at Close Limit Switch Trip

For MOVs with compensator spring packs that compress as they react to stem thrust, the close limit switch may be set to continue rotating the stem nut until the disc is fully seated under both static and DP conditions, and until any additional wedging thrust necessary to produce a seal is developed. If the limit switch is set in this manner, the disc and the stem will stop traveling at essentially the same position under both static and DP conditions. Because the same number of stem nut rotations ensures the same relative travel between the stem nut and the stem, the stem nut must compress the compensator spring pack essentially the same amount each stroke. Because the compensator spring pack preload and stiffness do not change from stroke to stroke, essentially the same stem thrust is developed in each stroke at close limit switch trip.

TU Electric analysis of 145 data points for 35 limit switch controlled closure MOVs has demonstrated that the stem thrust at close limit switch trip is generally well within ±3% for those MOVs that have close limit switches set to terminate the closing stroke after the valve disc has reached the valve seat under both static and DP conditions:

- Average variance from perfect repeatability = 0.01%
- Maximum variance from perfect repeatability = 2.69%
- Average plus three standard deviations of the variances from perfect repeatability = 2.1%.

For reasons not yet determined, one group of Borg-Warner gate valves and the Westinghouse PORV block valves produce greater thrust at close limit switch trip under DP conditions than under static conditions. The disc appears to terminate the closing stroke in a position slightly further out of the valve seat under DP conditions than under static conditions. This disc position effect has been included in the rate of loading analysis performed for limit switch controlled MOVs.

Rate of Loading Effects for Torque Switch Controlled Closure MOVs

The average rate of loading effect for torque switch controlled closure gate MOVs is a 4% greater stem thrust under static conditions than under DP conditions. A worst-case magnitude for this effect is about 30% for gate MOVs. These values are based on test results for 45 gate MOVs tested under both static and DP pumped flow conditions. For the PORV block valves under blowdown conditions, the average rate of loading effect was 30%. The postulated worst-case rate of loading effect for these MOVs under torque switch controlled closure is 58% (the greater of the two test data points is 40%). For globe valves, the average effect is 11%, with a maximum test result of 51%, for 15 test data points. (See Tables 9a, 9b, and 10, and Appendix A.)

Rate of Loading Effects for Limit Switch Controlled Closure MOVs

The rate of loading effect is primarily one of greater stem thread friction coefficients during valve strokes under DP conditions than during

valve strokes under static conditions. For limit switch controlled closure MOVs that produce the same stem thrust at close limit switch trip under both static and DP conditions, the rate of loading effect causes the stem torque at close limit switch trip to be greater by an average of 2% under DP conditions than under static conditions. The worst-case increase of stem torque under DP conditions as compared with static conditions is postulated to be 15%, based on analysis of Comanche Peak test data.

One group of Borg-Warner gate valves demonstrated higher stem thrusts at close limit switch trip under DP pumped flow conditions than under static conditions (the disc position effect). With a thrust increase under DP conditions postulated to be as much as 9%, the resulting stem torque would also be 9% greater for the same stem factor. Coupled with the rate of loading effect (greater stem factors under DP conditions), TU Electric data analysis projects a worst-case combined effect of 29% greater torque under DP conditions than under static conditions at close limit switch trip.

The PORV block valves, wired for limit switch controlled closure, demonstrated an average increase in stem thrust at close limit switch trip of 17% under DP blowdown conditions versus static conditions. The postulated maximum increase is 31%. Coupled with the rate of loading effect (greater stem factors under DP conditions than under static conditions), TU Electric data analysis determined an average increase in stem torque of 47%, a maximum test result of a 66% increase, and a projected a worst-case increase of 88%. (See Table 11 and Appendix A.)

Stem Thread Friction Coefficient Range

TU Electric has observed tremendous variation in stem thread friction coefficients for groups of nominally identical MOVs in nominally identical conditions with the same type of grease of nominally identical quality and quantity on the stem threads. Analysis of the test data suggests assuming a range of 0.05 to 0.20, with an average of 0.12, for the stem thread friction coefficient of any MOV for which there is no test data to justify otherwise. At initial unwedging of the disc from the valve seat, a range of 0.03 to 0.19, with an average of 0.10, appears to be appropriate. (See Table 12 and Appendix A.)

Margin for Stem Factor Degradation

Analysis of presently available data indicates that on the average there is no degradation (increase) of the stem friction coefficient over an outage cycle and that if any degradation does occur, it is unlikely to result in a stem factor increase greater than 10%. Thus, an uncertainty of 10% is used to account for potential degradation of the stem factor.

TU Electric presently cleans and relubricates all MOV stems each refueling outage. Over a period of N refueling cycles, the initial assumption may be that the degradation of the stem factor over these N cycles will not be more than determined by

$$\text{Stem Factor Degradation over N Refueling Cycles} = (N)^{1/2}\,(0.10) \qquad (1)$$

MOVs that have calibrated stem-mounted strain gages for measuring both thrust and torque can continue to provide test data throughout the remaining plant life. By analyzing these data, the TU Electric MOV program can refine the magnitude of the uncertainty used to account for stem factor degradation.

STATIC VERIFICATION TESTING FOR THE LIFE OF THE PLANT

All MOVs that could be tested under DP conditions were DP tested if the achievable test conditions were considered by engineering to be sufficiently close to the maximum design-basis conditions so that the test data collected could be reliably used to verify the initial assumptions made by engineering. Approximately 60% of the MOVs in Unit 1 and 90% in Unit 2 were tested under DP conditions.

TU Electric intends to use static testing as much as possible throughout the remainder of the plant operating life to verify the readiness of MOVs to perform their design-basis functions. Individual MOVs may experience increases in their valve factors from the values observed during baseline testing. However, based on TU Electric's test results and the results of industry test data, the valve factor values used by TU Electric are reasonable upper bound values.

TU Electric's test data analysis identifies groups of similar MOVs and applies the results of each group's test data analysis to all MOVs in the group. This method accommodates the potential for varying performance of each MOV over its remaining service life. Approximately one half of the MOVs in the group performed worse than the average, while the other half performed better than the average. Yet, each MOV in the group was treated as if its performance was the worst of the group.

The magnitudes used by TU Electric for the rate of loading factor, stem thrust effect factor, and the range of potential stem thread friction coefficients, contribute margin that, if not needed to compensate for the actual magnitudes of these effects, is also available to compensate for reasonably anticipated increases in valve factor values. A high degree of confidence is being provided that the MOVs important to the safe operation and shutdown of Comanche Peak are capable of performing their design-basis functions. The information and processes used by TU Electric are believed to be commensurate with the best available information and processes in the nuclear power industry.

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Texas Utilities Electric Company management is to be commended for their excellent support of the Comanche Peak motor-operated valve (MOV) program. Engineering, testing, and refurbishment personnel, most of whom have not been TU Electric employees, are to be commended for their willingness to adopt TU Electric's MOV program as their own and to make fine intellectual contributions toward its success, along with their many months of laborious implementation of the program. Without these people and their endurance of frequent changes to the engineering calculation, test procedures, and test equipment, the TU Electric MOV program would not have realized its successes. We have worked as a team, and to each individual—thank you.

Sid Chiu first proposed that TU Electric analyze the test data to determine averages and associated uncertainties, and to combine the uncertainties by the SRSS methodology. Charlie Catino and Rickey Page worked with the author for several months analyzing and reanalyzing data, while Dave Manning, Johnnie Verricchie, Jim Lee, Brian Robinson, and others assisted by compiling and checking the vast amounts of data analyzed. Sid Chiu and Scott Rosenberger reviewed the resulting analysis. Scott Rosenberger, Tom August, Tom Brown, and others worked with the author to prepare a complete revision of the engineering calculation that uses the test data analysis results. Charlie Catino and Rickey Page worked with the author to issue major revisions of in situ MOV test procedures in order to use the results of the new engineering calculation.

Those who reviewed and commented on drafts of this paper or the test data discussed herein include Kevin G. DeWall and Robert Steele, Jr., of the Idaho National Engineering Laboratory; Dr. Gerald H. Weidenhamer of the U.S. Nuclear Regulatory Commission; Norman Dingman of Nebraska Public Power District; Dr. Kalsi of Kalsi Engineering Corporation; Tim Cline of Duke Power; and Paul Damerell of MPR Associates.

Opinions expressed in this paper are those of the author and do not necessarily reflect the opinions of TU Electric, TU Electric's employees, TU Electric's contractors, or those who reviewed this paper.

Appendix A

Detailed Discussion of Test Data and Analysis Results

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Detailed Discussion of Test Data and Analysis Results

This appendix discusses in further detail most of the test results summarized in the main body of this paper. Equation (1) of this paper was presented in the main body of this paper. Thus, the first equation presented in this appendix is Equation (2).

Actuator Performance Factors

TU Electric has tested numerous actuators on torque test stands by gradually applying braking torques over several seconds until their motors stalled with power supplied at 80% of the motors' nameplate voltages. The torque test stands use splined stem adapters instead of threaded stem nuts. Thus, during torque stand testing no thrust load was applied to the actuator drive sleeve. In each of these cases, with the actuator was recently fully refurbished. It was assumed that the measured actuator output torque was essentially equal to the torque applied by the actuator worm to the actuator worm gear.

TU Electric test data indicate that when the measured motor stall torque is not less than its design capability, as defined by TU Electric, the capability of the motor to deliver torque to the worm gear (when the motor is powered by 80% of the motor nameplate voltage) during torque stand testing is generally greater than the value of AOTQ80w predicted by the common design calculation shown in as

$$\text{AOTQ80w} = (0.8)^2\,(\text{MTQ})(\text{OAR})(\text{AF})(\text{RE}) \quad (2)$$

where

0.8	=	factor for power supply at 80% of nameplate voltage
MTQ	=	motor torque rating (ft-lb)
OAR	=	overall gear ratio of the actuator from motor to stem nut (or HBC input shaft)
AF	=	application factor = 0.9
RE	=	running efficiency of actuator gear train, from actuator vendor design data.

Actuator configurations have been considered to be similar if the actuators (a) are the same size, (b) have the same motor torque rating and speed, and (c) have the same vendor-specified value for RE. For each such group of similar actuators, TU Electric has defined a performance factor (PF), which has been applied to each actuator in the similarity group. The value of PF is determined as the average of the ratios of the measured stall torque value (AOTQ80stall) divided by design value predicted by Equation (2) and shown as

$$\text{PF} = \text{average of all values of the ratio } (\text{AOTQ80stall}/\text{AOTQ80w}) \;. \quad (3)$$

The value of PF is greater than 1.00 for all groups of similar actuators tested by TU Electric. Also determined is an uncertainty (ePF) associated with the value of PF. The engineering calculation now predicts the motor's nominal and minimum capabilities at 80% of motor nameplate voltage to deliver torque to the worm gear as

Nominal (average) value

$$\text{NTQmax,w} = (0.8)^2\,(\text{MTQ})(\text{OAR})(\text{AF})(\text{RE})(\text{PF}) \quad (4)$$

and

Lower bound value

$$\text{TQmax,w} = (0.8)^2\,(\text{MTQ})(\text{OAR})(\text{AF})(\text{RE})(\text{PF})(1-\text{ePF}) \;. \quad (5)$$

Of course, if elevated room temperature decreases the motor's capability an appropriately decreased motor torque rating is used instead of the nameplate torque rating.

Only one group of similar actuators was observed to frequently produce measured actuator output stall torques (at 80% voltage) less than the value predicted by Equation (2). For this group, the value of PF is about 1.10. The least value of the ratio of measured to calculated capability is 0.91. As shown in Table 1, an uncertainty ePF of 16% is determined based on the test data.

Table 2 summarizes the results for several groups of similar actuators as determined by TU Electric engineering. The column heading PF(1 − ePF) is the ratio of (a) the minimum expected motor torque capability based on testing divided by (b) the motor torque capability predicted Equation (2). In general, there is very good agreement with Equation (2). However, for the 15-ft-lb 3,400-rpm motor on a Size 00 actuator, Equation (2) appears to overestimate the motor capability.

The Table 2 column heading PF(1 − ePF) (RE) is a modified running efficiency value based on TU Electric test data analysis. Table 2 compares this product with the design running efficiency specified by the actuator manufacturer. It may be that gear train efficiencies are not the correct explanation for the differences observed. Differences in motor characteristics may be more of a factor. This supposition is based on the data for the Size 00 and Size 0 actuators.

Actuator Effective Moment Arm Lengths

TU Electric also extracted for analysis the following torque test stand data: torque spring pack deflections and corresponding actuator output torques. The forces applied to the torque spring pack to produce the deflections were then obtained by reviewing the deflection versus force data collected during compression testing of the torque spring packs. Correspondences were thereby established between torque spring pack forces and actuator output torques produced on the torque test stand.

The torque values were divided by their corresponding torque spring pack force values to obtain the unique effective moment arm length for each set of data. The average effective moment arm length (MARMw) was then determined. Friction loads within the actuator cause the actual relationships (the effective moment arm lengths) between actuator output torques and torque spring pack forces to differ slightly from the magnitude of the worm gear pitch radius.

The products of the individual torque spring pack force (SPF) values and the average effective moment arm MARMw were next calculated. Differences between these calculated torques and the measured torques were then used to determine an uncertainty (eTQw). All measured torques are within the range provided by

$$\text{AOTQw} = (\text{SPF})(\text{MARMw}) \pm \text{eTQw} \ . \qquad (6)$$

It is assumed that torques measured on a torque test stand are appropriate for comparison with the actuator manufacturer's specified actuator gear train torque ratings (TQrt). TU Electric uses the uncertainty eTQw and the value of MARMw to convert the actuator torque rating to a corresponding maximum allowable torque spring pack force calculated by

$$\text{SPFrt} = (\text{TQrt}/\text{MARMw})\,[1 - (\text{eTQw}/\text{TQrt})] \ . \qquad (7)$$

Motor capability is similarly converted to a maximum allowable torque spring pack force (SPFmax). The result is reduced by the SRSS combination of the uncertainties, shown as

$$\text{SPFmax} = (\text{NTQmax,w}/\text{MARMw})\,\{1 - [(\text{ePF})^2 + (\text{eTQw}/\text{NTQmax,w})^2]^{1/2}\} \ . \qquad (8)$$

Effects of Stem Thrust on Actuator Output Torque

Following actuator refurbishment and torque stand testing, the actuators were reinstalled on the valves in the power plant. TU Electric

Table 1. Motor capability: actual versus initial design calculation.

OAR	AOTQ80w (ft-lb)	AOTQ80stall close (ft-lb)	AOTQ80stall open (ft-lb)	Close ratio	Open ratio
23.0:1	119.2	119.0	108.0	0.9983	0.9060
23.0:1	119.2	123.1	112.8	1.0327	0.9463
23.0:1	119.2	131.7	118.4	1.1049	0.9933
23.0:1	119.2	129.9	119.7	1.0898	1.0042
23.0:1	119.2	135.2	124.0	1.1342	1.0403
23.0:1	119.2	140.6	126.8	1.1795	1.0638
23.0:1	119.2	—	131.9	—	1.1065
23.0:1	119.2	134.5	135.0	1.1284	1.1326
23.0:1	119.2	—	138.5	—	1.1619
23.0:1	119.2	—	147.9	—	1.2408
30.0:1	155.5	172.9	155.6	1.1119	1.0006
30.0:1	155.5	—	160.6	—	1.0328
31.9:1	165.4	173.1	—	1.0466	—
31.9:1	165.4	171.6	161.4	1.0375	0.9758
31.9:1	165.4	—	184.4	—	1.1149
31.9:1	165.4	213.6	195.7	1.2914	1.1832
34.1:1	176.8	—	185.0	—	1.0464
34.1:1	176.8	—	202.0	—	1.1425
36.3:1	188.2	229.5	—	1.2194	—
36.3:1	188.2	198.0	—	1.0521	—
36.3:1	188.2	193.2	187.3	1.0266	0.9952
36.3:1	188.2	—	193.0	—	1.0255
36.3:1	188.2	200.0	199.0	1.0627	1.0574
36.3:1	188.2	233.9	216.6	1.2428	1.1509

Notes:

1. Twenty-four different actuators tested to stall at 80% voltage in close, open, or both directions. Actuator = Size 00; motor = 15 ft-lb 3,400 rpm; worm/worm gear ratio = 45:1.
2. The performance factor is the average of all close and open direction ratios: PF = 1.083.
3. One sample standard deviation of all the ratios: s = 0.087.
4. The average minus two sample standard deviations: PF – 2s = 0.909.
5. The minimum ratio of all the close and open direction data: MIN = 0.906.
6. The uncertainty associated with PF is the percentage by which PF must be reduced to obtain the lesser of (a) the value of the difference PF – 2s, or (b) the value of MIN, whichever is less: ePF = {1 – [(PF – 2s)/PF]} or [1 – (MIN/PF)] = [1 – (0.906/1.083)] = 0.16.

Table 2. Summary of PF and ePF values determined by TU Electric.

Actuator size	MTQ (ft-lb)	RPM	OAR range	W/WG ratio	RE	PF	ePF	(PF)(1 − ePF)	Verify: (PF)(1 − ePF)(RE)	Verify: > RE ? (yes/no)
000	2	1,700	23.0–63.3	50:1	0.50	1.21	0.16	1.02	0.51	Yes
000	5	1,700	33.3	50:1	0.50	1.21	0.16	1.02	0.51	Yes
00	10	1,700	23.0–63.3	45:1	0.50	1.41	0.16	1.18	0.59	Yes
00	10	3,400	31.6–34.1	45:1	0.60	1.27	0.21	1.00	0.60	Yes
00	15	3,400	23.0–36.3	45:1	0.60	1.08	0.16	0.91	0.55	No
0	25	1,700	29.6	37:1	0.55	1.25	0.08	1.15	0.63	Yes
0	40	1,700	29.6	37:1	0.55	1.12	0.06	1.05	0.58	Yes
1	60	3,400	27.2–32.1	34:1	0.60	1.10	0.08	1.01	0.61	Yes
2	80	3,400	27.8	33:1	0.60	1.10	0.09	1.00	0.60	Yes

engineering had assumed early in the TU Electric MOV program that thrust loads on the drive sleeve of the actuator negligibly affect the relationship between torque spring pack force and actuator output torque determined from torque stand testing. TU Electric analyzed test data collected during in situ static and DP condition testing to verify this assumption.

TU Electric was surprised that some test data indicated a significant loss of actuator output torque when the actuator drive sleeve was subjected to thrust loading. The actuator manufacturer indicated they did not expect this result either. This has been observed by TU Electric for actuators Sizes 00, 0, 1, and 2. The author believes if test data were available for other actuator sizes, this stem thrust effect would be noticed in that data as well.

For a small sample of MOVs, TU Electric also examined the relationships between motor three-phase power measurements and actuator output torques for both torque stand testing and in situ testing. The conclusion was reached that both the spring pack data and the power data indicated the same thing regarding actuator output torques: thrust loads on the drive sleeve tend to reduce the actuator output torque for a given value of torque spring pack deflection (such as a torque switch setting, or at the motor stall condition). The data indicated that this loss did not occur until a substantial amount of thrust was developed, but well within the normal operating load range of the actuator.

The effective moment arm lengths by which torque spring pack force values may be multiplied to obtain the torques delivered by a threaded stem nut to a threaded valve stem were then determined using data at stem thrust magnitudes appropriate for the successful operation of Comanche Peak MOVs. The method of analysis was the same as described earlier for the analysis of torque test stand data, except that the torque measurements were obtained using stems with strain gages installed in a full bridge and calibrated. Because the in situ actuator output torque is less than the torque stand actuator output torque for a given value of torque spring pack force, the length of the effective moment arm (MARM) is less than the value of MARMw. In the same manner as described earlier for torque test stand data, an uncertainty (eTQ) was determined for in situ actuator output torques calculated using MARM. All measured stem torques are within the range provided by

$$AOTQ = (SPF)(MARM) \pm eTQ \quad (9)$$

Just as actuator torque limits were converted to maximum allowable torque spring pack force values, the stem torque limits (TQSEQ) obtained from the seismic qualification documents are converted by TU Electric to equivalent maximum allowable torque spring pack forces and are shown as

$$SPFSEQ = (TQSEQ/MARM) [1 - (eTQ/TQSEQ)] \cdot \quad (10)$$

Typical of factors affected by friction coefficients, manufacturing tolerances, assembly practices, component wear, and measurement error, there is substantial scatter in the test data collected to determine the average effective worm gear moment arm length (MARMw), which relates torque spring pack force to the approximate torque (AOTQw) on the worm gear, and the average effective drive sleeve moment arm length (MARM), which relates torque spring pack force to the torque (AOTQ) applied by the stem nut to the stem. The degree of dispersion of the test data about these average effective moment arm lengths is accounted for by the uncertainties eTQw and eTQ.

Table 3 provides values for both MARM and MARMw, as determined by TU Electric for actuator Sizes 000, 00, 0, 1, and 2. The approximate uncertainty of a torque calculated as the product of torque spring pack force and one of these average effective moment arm lengths is in the column labeled Approximate eTQ.

Based on analysis of unseating test data, during unwedging of the valve disc from the valve seat the average values of the effective moment arm lengths appear to be about 10% less than the average values at other points of interest in the closing and opening strokes. Thus, TU Electric has determined different values for the average effective moment arm lengths and the associated uncertainties for use in evaluating loads during unwedging of the disc from the valve seat.

STEM THRUST EFFECT

At the February 1994 MOV Users Group meeting, the actuator manufacturer's representative stated that the manufacturer has not performed testing to verify that thrust loads do not reduce the available torque. TU Electric has introduced a stem thrust effect factor into its engineering calculation to account for the average magnitude of this effect, as well as an associated uncertainty value based on statistical analysis of the test data.

The author has postulated that when thrust loads on the drive sleeve are great enough, deflections of components within the actuator drive sleeve become great enough so that parts rub against each other, although these parts do not rub against each other at lesser thrust loads. The deflections involved may occur within one part (such as the gasket at the housing-to-housing cover joint) or within a subassembly of parts (such as a drive sleeve bearing).

The industry has been made aware that excessive torque on the actuator cover bolts may compress the actuator cover gasket too much and

Table 3. Values of MARM, MARMw, and eTQ.

Actuator type-size	MARM[a] (ft)	MARMw[a] (ft)	Approximate eTQ[a] (ft-lb)
SMB-000	—	0.11	6
SMB-00	0.14	0.17	20
SB-00	0.16	0.17	20
SMB-0	0.20	0.24	40
SB-1	0.23	0.26	60
SB-2	0.28	0.31	90

a. Values apply throughout the closing stroke and in the opening stroke after unwedging of the disc from the valve seat.

place the upper and lower drive sleeve bearings under excessive axial load, with a resultant reduction in torque delivered to the stem nut at torque switch trip (or at motor stall). The application of a thrust load to the actuator drive sleeve by the stem (through the stem nut) is conceptually very similar to the application of load by the cover bolts. The difference is that the load imparted by the stem will increase the compressive load on one bearing while decreasing the load on the other bearing.

Another possibility suggested to the author by Tim Cline of Duke Power is that movement of the actuator drive sleeve along the stem axis may result in a less efficient interface between the worm and the worm gear. Tim Cline postulated that a lesser efficiency may result from the mid-plane of the worm gear moving out of the plane in which the worm axis resides.

Much testing of actuators has been conducted using torque test stands: by the actuator manufacturer, by contractors to the actuator manufacturer, by test equipment vendors, and by utilities. If this torque stand test data have been used to determine the relationship between torque spring pack force and actuator output torque, the users of this data must recognize that the relationship may be applicable only if the actuator is used in a load condition similar to that on the torque test stand. The relationship may be valid for determining the torque at the worm gear for verification that the motor capability and the actuator torque rating are not exceeded. However, the relationship may overestimate the torque delivered to the threaded stem nut for producing stem thrust.

Using test data collected for SB-1 actuators during torque stand and in situ testing, Table 4 demonstrates the method used to determine the average magnitude of the stem thrust effect (ST) and the associated uncertainty (eST). Table 4 shows the values of TQsg and TQtts. Each measured stem torque is converted into a value comparable to torque test stand data by multiplying TQsg by the ratio (MARMw/MARM). The measured actuator output torque (TQtts) on the torque test stand is divided by the calculated torque. Because the average of the ratios is not equal to 1.00, it is clear that a better correlation between actuator output torques with and without thrust loading on the actuator drive sleeve may be obtained by multiplying TQsg by the product (AVE)(MARMw/MARM). This product is called the average ST. The eST is calculated to bound the maximum data point, which is further from the average value than two sample standard deviations. (For other actuator sizes, two sample standard deviations bound all test data.)

Because two different tests were conducted to obtain the torque stand and the in situ data for each actuator, the influence of actuator output torque repeatability (REPts) at a given value of torque spring pack deflection must be considered. The average magnitude of the ST is minimally affected by REPts because averaging the data tends to average out the effect of REPts. The spread of data about the average is affected by REPts. The values determined by TU Electric for uncertainties eST therefore include the influence of REPts. TU Electric has not performed testing to quantify actuator output torque repeatability versus torque spring pack deflection, and therefore has not reduced the magnitude of each eST value by the effect of REPts. This measure of conservatism could be removed.

Table 5 presents the average magnitude of the stem thrust effect (ST) and the associated uncertainty (eST) for several actuator types and sizes as determined by TU Electric.

Net Impact of the Performance Factor and the Stem Thrust Effect

The net effect of (a) the extra (for most actuators) capability of the motor to deliver torque to the worm gear that the performance factors justify, and (b) the reduced capability of the motor to deliver torque to the stem nut that the stem thrust effects demonstrate, are now discussed. The ratio (PF/ST) shown in Table 6a is the average combined effect. This average is then further reduced by the SRSS combination of uncertainties eST and ePF to obtain NET, the lower bound of the combined effect determined by

Table 4. SB-1 stem thrust effect test data and analysis.

SPF	TQsg	TQtts	(TQtts/TQsg)/ (MARMw/MARM)	Verify: (TQsg)(ST) (1 + eST)	Verify: > TQtts ? (yes/no)
915	199	263	1.171[a]	263	yes
1,890	425	464	0.968	562	yes
1,124	261	263	0.893[a]	345	yes
2,462	561	599	0.947	741	yes
1,789	370	414	0.992	489	yes
834	136	178	1.160[a]	180	yes
1,485	386	436	1.001	510	yes
592	159	183	1.020[a]	210	yes
1,505	388	449	1.026	513	yes
2,270	511	600	1.041	675	yes
1,219	287	321	0.991	379	yes
2,114	485	553	1.011	641	yes
2,482	577	636	0.977	763	yes
1,415	311	362	1.032	411	yes
2,482	550	636	1.025	727	yes

a. Torque spring pack deflection magnitudes were typical for settings of "1"—10% repeatability:

MARMw = 0.2572 ft, and MARM = 0.2279 ft
ST = (MARMw/MARM)(AVE) = 1.15
eST = the greater of [2s/AVE] or [(MAX/AVE) - 1] = 0.15.

Where for the column labeled "(TQtts/TQsg)/(MARMw/MARM)":
The average of the data, AVE = 1.017
The maximum data point, MAX = 1.171
One sample standard deviation, s = 0.0713
(Note: MAX > AVE + 2s = 1.159).

$$\text{NET} = (\text{PF/ST})(1 - [(\text{ePF})^2 + (\text{eST})^2]^{1/2}) \quad (11)$$

The value of NET is the coefficient by which the motor capability to deliver torque to the stem of a rising (nonrotating) stem valve as determined by Equation (2) must be multiplied to obtain a stem torque (TQmax) value which accounts for the stem thrust effect's reduction of torque delivered to the threaded stem and the performance factor's increase (for most actuator configurations) of predicted motor capability to deliver torque when there is no thrust load on the drive sleeve. The value of TQmax is determined as

$$\text{TQmax} = (0.8)^2\,(\text{MTQ})(\text{OAR})(\text{AF})\,(\text{RE})\,(\text{NET}) \quad (12a)$$

$$\text{TQmax} = (0.8)^2\,(\text{MTQ})(\text{OAR})(\text{AF})\,(\text{RE})(\text{PF/ST})\,(1 - [(\text{ePF})^2 + (\text{eST})^2]^{1/2}) \quad (12b)$$

As Table 6b shows, the net effect would be bounded if the actuator manufacturer's pullout

Table 5. Stem thrust effect magnitudes and uncertainties.

Actuator type-size[a]	ST	eST	Maximum TQtts/TQsg	Verify < ST(1 + eST) ? (yes/no)		Minimum TQtts/TQsg	Verify > ST(1 – eST) ? (yes/no)	
SMB-00	1.20	0.18	1.32	1.42	Yes	1.05	0.98	Yes
SMB-0	1.23	0.18	1.44	1.45	Yes	1.02	1.00	Yes
SB-1	1.15	0.15	1.32	1.32	Yes	1.01	0.98	Yes
SB-2	1.10	0.13	1.19	1.24	Yes	0.99	0.96	Yes

a. For each actuator type-size, the maximum test data point is less than (or equal to) the maximum predicted stem thrust effect and the least test data point is greater than the minimum predicted stem thrust effect. This provides verification that the predictions are appropriate (slightly conservative).

Table 6a. Net impact of performance factors and stem thrust effects.

Actuator type-size	MTQ (ft-lb)	RPM	Worm-to-worm gear ratio	PF	ePF	ST	eST	PF/ST[a]	NET[b,c]
SMB-00	10	1,700	45:1	1.40	0.16	1.20	0.18	1.17	0.89
SMB-00	10	3,400	45:1	1.27	0.21	1.20	0.18	1.06	0.77
SMB-00	15	3,400	45:1	1.00	0.16	1.20	0.18	0.83	0.68
SMB-0	25	1,700	37:1	1.25	0.08	1.23	0.18	1.02	0.82
SMB-0	40	1,700	37:1	1.12	0.06	1.23	0.18	0.91	0.74
SB-1	60	3,400	34:1	1.10	0.08	1.15	0.15	0.96	0.79
SB-2	80	3,400	33:1	1.10	0.09	1.10	0.13	1.00	0.84

a. Average combined effect of PF and ST.

b. Lower bound combined effect of PF and ST.

c. The value of NET includes the effects of actuator output torque repeatability, and in each case exceeds the magnitude of the actuator repeatability uncertainty specified by the actuator manufacturer:
$NET = (PF)(1 - [ePF^2 + eST^2]^{1/2})/(ST)$.

efficiency (POE) were used instead of the running efficiency (RE) for certain actuator configurations. For other configurations, a value greater than the pullout efficiency is justifiable. In other cases, even the use of the actuator manufacturer's pullout efficiency does not appear to provide a lower bound prediction of the motor's capability to deliver torque to the valve stem when the motor is powered by 80% of the nameplate voltage.

While the actuator manufacturer is continuing to use the pullout efficiency for sizing actuators for new orders, the manufacturer has stated that utilities may use the running efficiency when

Table 6b. Using a reduced running efficiency versus using the pullout efficiency.[a]

Actuator type-size	MTQ (ft-lb)	RPM	Worm-to-worm gear ratio	RE	(RE)(NET)	> POE ? (yes/no)	
SMB-00	10	1,700	45:1	0.50	0.45	0.40	Yes
SMB-00	10	3,400	45:1	0.60	0.46	0.45	Yes
SMB-00	15	3,400	45:1	0.60	0.41	0.45	No
SMB-0	25	1,700	37:1	0.55	0.45	0.45	Yes
SMB-0	40	1,700	37:1	0.55	0.41	0.45	No
SB-1	60	3,400	34:1	0.60	0.47	0.45	Yes
SB-2	80	3,400	33:1	0.60	0.50	0.45	Yes

a. Table 6b multiplies NET times the running efficiency (RE) published by the actuator manufacturer and compares the result with the pullout efficiency (POE) published by the actuator manufacturer. In most cases, the use of POE is observed to be conservative: it is less than the product (RE)(NET).

evaluating the capabilities of actuators already installed in power plants. It is the author's opinion that TU Electric test data indicates this allowance for using running efficiencies may need to be further reviewed because it may not, in all cases, provide appropriate levels of assurance that safety-related MOVs would fulfill their design-basis functions. In fact, because two configurations in Table 6b show that even the use of the pullout efficiency may not always suffice, other reviews may also be needed.

Further Investigation of the Stem Thrust Effect

It would probably be worthwhile for others to perform tests to either confirm or refine the results TU Electric has obtained. An improved technique for performing reliable tests would be to test the actuator on a test stand that can not only provide a resisting torque to cause the motor to stall but also simultaneously apply thrust loads that can be held constant or varied as the torque load increases. Testing on such a fixture may justify reducing the conservatism in the magnitudes of the stem thrust effects determined by TU Electric, especially if the effects of actuator output repeatability are removed from the analysis results.

Tests of this sort could be used in conjunction with inspections and analyses of actuator drive sleeve parts to determine what, if any, parts begin to interact adversely with each other when high thrust loads are imposed, and the magnitudes of the thrust loads that initiate this interaction. Investigation could also include identifying actuator modifications that might eliminate or reduce the magnitude of the stem thrust effect and whether such a modification would be economically feasible.

RUNNING THRUSTS

Differences between the running thrusts of nominally identical MOVs with nominally identical packing configurations and packing follower bolt torques are typically significantly greater than the differences between open and close running thrust values for these same MOVs. Thus, static condition closing and opening running thrust test data have been combined to determine an average running thrust (RT) value assumed for both directions of stem travel in the engineering calculation.

An uncertainty (eRT) associated with the average running thrust value is based on the greater of (a) the maximum test result and (b) the average value plus two standard deviations of the data. Because running thrust magnitudes are typically small, the test instrument measurement uncertainty is typically high. This may contribute to the degree of scatter of the test data. The large uncertainty is tolerable because running thrusts are typically a small part of the MOV thrust requirements.

Running thrusts result primarily from the packing friction force, which may be calculated by a packing manufacturer's equation. The thrusts are also affected by the friction loads developed from the torque load on the stem being reacted throughout the stem travel. For MOVs at TU Electric's Comanche Peak plant, there were several MOVs with measured running thrusts greater than the packing manufacturer's calculated packing friction loads.

The results of analyzing the test data are presented in Table 7. There are two sets of results for stems of 1.00-in. diameter because the difference in the packing configuration and preload, and perhaps the valves too, made a very definite difference in the actual running thrusts.

Table 7. Results of running thrust data analysis.

Dp (in.)	RT (lb)	eRT (%/100)	RT/Dp (lb/in.)	RT(1 + eRT)/Dp
0.750	620	0.50	827	1,240
1.000	310	1.00	310	620
1.000	730	0.65	730	1,205
1.125	500	0.75	444	778
1.250	720	0.75	576	1,008
1.375	1,370	0.70	996	1,694
1.875	2,100	0.70	1,120	1,904
2.000	2,100	0.60	1,050	1,680
2.500	3,200	0.30	1,280	1,664
3.000	3,650	1.10	1,217	2,555

VALVE FACTORS FOR WESTINGHOUSE AND BORG-WARNER GATE VALVES

The standard equation for the maximum stem thrust required to move a gate valve disc (gate) to the valve seat under DP conditions may be solved for the valve factor. Using test data, the actual closing stroke valve factor (VFct) may be determined as

$$VFct = [Tdpt,d - RTs,c - (Ap)(Pupct)]/[(Ao)(DPTc)] \quad , \qquad (13)$$

where

Ao = nominal seat orifice area based on a selected diameter, Do

DPTc = maximum differential pressure measured during the closing stroke of the DP test

Ap = area of the stem cross section at the packing

Pupct = maximum upstream pressure measured during the closing stroke of the DP test

RTs,c = average static test running thrust over the last 10% or so of the closing stroke

Tdpt,d = maximum thrust required at any point during the closing stroke of the DP test until the disc begins to wedge into the valve body seat

Do = For Westinghouse valves, the seat ring inside design diameter specified by the manufacturer, plus 1/16 in., for Borg-Warner valves, the average of the valve body seating surface's inside and outside design diameters specified by the manufacturer.

The standard equation for the maximum stem thrust required to move a gate out of the valve seat (after initial unwedging of the disc) under DP conditions may be solved for the valve factor. Using test data, the actual opening stroke valve factor (VFot) may be determined as

$$VFot = [TDPo - RTs,o + (Ap)(Pupot)]/[(Ao)(DPTo)] \qquad (14)$$

where Ao and Ap are as defined earlier, and

DPTo = maximum differential pressure measured during the opening stroke of the DP test

Pupot = maximum upstream pressure measured during the opening stroke of the DP test

RTs,o = average static test running thrust over the first 10% or so of the opening stroke

TDPo = maximum thrust required at any point during the opening stroke of the DP test after initial unwedging of the disc (overcoming the static friction coefficient in the valve seat).

Table 8a justifies the grouping of similar valves. Table 8b identifies the number of MOVs in each group of similar valves, the number of MOVs in each group that were DP tested in each of Comanche Peak Units 1 and 2, and the number of MOVs in each group that were tested with stem strain gage measurements. Table 8c provides a summary of closing stroke valve factor test results: the average value (VFc), the maximum value (Max VFct), and an uncertainty (eVFc) associated with the average value (which is based on the greater of (a) the average value plus two sample standard deviations, and (b) the maximum VFct value). Table 8d provides a summary of opening stroke valve factor test results: the average value (VFo), the maximum value (Max VFot), and an uncertainty (eVFo) associated with the average value which is based on the greater of (a) the average value plus two sample standard deviations, and (b) the Max VFot value. In all cases for both closing and opening, the sum of the average value plus two standard deviations was greater than the maximum valve factor derived from test data. Tables 8c and 8d compare the valve vendor's original valve factor values with the maximum test results.

For each valve in the similarity group, TU Electric now calculates by Equation (15) the maximum thrust (Tseat,d) that may be required to close the valve under DP conditions and initiate wedging of the disc into the valve body seat

$$Tseat,d = (RT)(1 + eRT) + (Ap)(Pupc) + (Ao)(DPRc)(VFc)(1 + eVFc) \qquad (15)$$

where RT, eRT, Ap, and Ao are as defined earlier, and

eVFc = uncertainty associated with the value of VFc

DPRc = closing stroke maximum design basis (required) differential pressure

Table 8a. Borg-Warner and Westinghouse gate valve grouping.

Group ID	Manufacturer[a]	Size (in.)	Do (in.)	Dp (in.)	Manufacturer's model or drawing number
GT1	W	3	2.688	1.250	Model 3GM88FNH
GT2	W	3	2.688	1.250	Model 3GM78FN
GT3	W	4	3.508	1.250	Model 4GM78FNH, and 4GM77FH
GT4	W	4	3.623	1.250	Model 4GM72FB
GT5	BW	4	4.020	1.375	Drawing 75610-1
GT6	BW	4	4.020	1.375	Drawing 75610
GT7	BW	4	4.070	1.000	Drawing 404JDB3-006
GT8	W	6	6.128	1.250	Model 6GM72FB
GT9	BW	6	6.310	1.250	Drawing 75650, 75680
GT10	W	8	6.568	1.250	Model 8GM74FE, 8GM72FB
GT11	W	10	8.818	2.500	Model 10GM78FNH, 10GM78FN
GT12	W	10	10.09	2.000	Model 10GM74FE
GT13	W	12	10.57	3.000	Model 12GM88SEH
GT14	W	14	12.07	2.000	Model 14GM74FEH
GT15	BW	16	10.57	3.000	Drawing 75790,75800, 75800-2
GT16	BW	8	8.260	1.375	Drawing 75710

a. Manufacturers: W = Westinghouse; BW = Borg-Warner.

Pupc = closing stroke maximum design basis upstream pressure accompanying DPRc

VFc = the average valve factor in the closing stroke for a group of nominally identical valves.

For each valve in the similarity group, TU Electric now calculates by Equation (16) the maximum thrust (Tod) that may be required to open the valve under DP conditions after initial unwedging of the disc (overcoming the static friction coefficient in the valve seat), shown as

$$Tod = (RT)(1 + eRT) - (Ap)(Pupo) + (Ao)(DPRo)(VFo)(1 + eVFo) \quad , \qquad (16)$$

where RT, eRT, Ap, and Ao are as defined earlier, and

eVFo = uncertainty associated with the value of VFo

DPRo = opening stroke maximum design basis (required) differential pressure

Pupo = opening stroke maximum design basis upstream pressure accompanying DPRo

VFo = the average valve factor in the opening stroke for a group of nominally identical valves after initial unwedging of the disc (overcoming the static friction coefficient of the valve seat).

Table 8b. Borg-Warner and Westinghouse gate valve test status.

Group ID	Manufacturer[a]	MOVs in the group[b]	DP tested Unit 1	DP tested Unit 2	DP tested with stem strain gages
GT1	W	4	0	2	2
GT2	W	4	2	2	4
GT3	W	14	7	7	14
GT4	W	4	0	2	2
GT5	BW	16	4	8	12
GT6	BW	4	0	2	2
GT7	BW	4	0	2	2
GT8	W	10	5	5	7
GT9	BW	10	3	5	3
GT10	W	10	2	5	6
GT11	W	14	3	7	7
GT12	W	4	2	2	4
GT13	W	8	0	4	4
GT14	W	8	0	4[c]	4
GT15	BW	12	0	2	2
GT16	BW	10	4	5	4

a. Manufacturers: W = Westinghouse; BW = Borg-Warner.

b. All valves in each group were tested under static conditions.

c. Open only.

RATE OF LOADING EFFECT MAGNITUDES FOR TORQUE SWITCH CONTROLLED CLOSURE MOVS

Actuator output torque at torque switch trip (AOTQtst) is the same within actuator repeatability (REPts), under both static and DP conditions. However, under static conditions the stem thrust at torque switch trip is sometimes noticeably greater than under DP conditions. Initially called the rate-of-loading effect (ROL), it is more correctly an effect primarily of varying stem friction coefficients from stroke to stroke. An MOV that is susceptible to ROL effects may exhibit significantly different ROL magnitudes during different strokes.

TU Electric assumes that even if only one valve in a group manifests susceptibility to the ROL effect during testing, then any other MOV in the group may manifest a ROL effect during the design-basis condition that may occur at any time in the remaining plant operating life. Also, TU Electric does not necessarily assume that the maximum possible ROL effect was demonstrated during testing. TU Electric uses statistical analysis to determine the average ROL plus two sample standard deviations, and compares this sum with the maximum ROL effect observed during testing. TU Electric assumes the greater of these two values is the maximum possible ROL effect. An uncertainty (eROL) is determined based on the greater of the two values.

Table 8c. Borg-Warner and Westinghouse gate valve factors, closing.

Group ID	DPTc range (psid)	VFc	eVFc	Verification (VFc) (1 + eVFc)	> Max VFct ? (yes/no)		Manufacturer's VFc
GT1	2010–2058	0.67	0.02	0.68	0.67	Yes	0.56[a]
GT2	1657–2846	0.29	0.63	0.47	0.41	Yes	0.56
GT3	811–2862	0.40	0.63	0.65	0.54	Yes	0.61
GT4	255–255	0.23	0.70	0.39	0.28	Yes	0.63
GT5	1528–1749	0.48	0.31	0.63	0.62	Yes	0.30[a]
GT6	2578–2878	0.50	0.06	0.53	0.51	Yes	0.30[a]
GT7	95–102	0.43	0.16	0.50	0.45	Yes	0.30[a]
GT8	131–251	0.36	0.37	0.49	0.42	Yes	0.63
GT9	290–399	0.24	0.36	0.33	0.29	Yes	0.30
GT10	126–254	0.38	0.51	0.57	0.50	Yes	0.59
GT11	51–267	0.56	0.70	0.95	0.89	Yes	0.64[a]
GT12	195–261	0.47	0.87	0.88	0.68	Yes	0.59[a]
GT13	NA[b]	—	—	—	—		—
GT14	NA[b]	—	—	—	—		—
GT15	298–304	0.39	0.11	0.43	0.40	Yes	0.30[a]
GT16	107–171	0.47	0.31	0.62	0.57	Yes	0.30[a]

a. Manufacturer's equivalent VFc value is less than maximum test result VFct.

b. There is no closing requirement within the plant design basis for this group of MOVs.

The SRSS combination of eROL with other uncertainties diminishes the impact of this effect on the minimum required stem thrust at close torque switch trip. TU Electric also determines whether the assumed valve factors and their uncertainties have enough conservatism in their values to account for the ROL effect. If so, then no additional ROL penalty is imposed. The following decision process is used to determine the values of ROL and eROL.

STEP 1: Recognize that if the magnitude of the stem factor at close torque switch trip is greater under DP conditions than under static conditions, the ratio of the DP condition stem factor (SFcst,d) divided by the static condition stem factor (SFcst,s) is greater than 1.00. Define the ratio (RSFt) of these stem factors as

$$RSFt = (SFcst,d/SFcst,s) = (Tcst,s/TQcst,s)/(Tcst,d/TQcst,d) > 1.00 \quad (17)$$

STEP 2: TU Electric has reviewed ROL effects both at the torque switch trip event and at the maximum thrust load to overcome DP conditions. They have found that the scatter of data is such that using data at the torque switch event is sufficient to approximate the effects at the maximum load to overcome DP conditions. Thus, by using Equation (18), the maximum thrust (Tdpt,d) observed during testing to overcome the test DP conditions can be converted into a corresponding minimum

Table 8d. Borg-Warner and Westinghouse gate valve factors, opening.

Group ID	DPTo range (psid)	VFo	eVFo	Verification (VFo)(1 + eVFo)	Verification > Max VFot ? (yes/no)		Manufacturer's VFo
GT1	2,081–2,128	0.53	0.13	0.60	0.55	Yes	0.56
GT2	1,815–2,851	0.32	0.63	0.52	0.43	Yes	0.56
GT3	1,651–2,740	0.37	0.44	0.53	0.48	Yes	0.54
GT4	200–200	0.44	0.49	0.66	0.52	Yes	0.55
GT5	1,521–1,637	0.41	0.33	0.55	0.54	Yes	0.77
GT6	2,579–2,628	0.39	0.09	0.43	0.40	Yes	0.53
GT7	139–158	0.36	0.78	0.64	0.45	Yes	0.42
GT8	121–211	0.39	0.40	0.55	0.51	Yes	0.56
GT9	97–215	0.38	0.62	0.62	0.48	Yes	0.75
GT10	117–198	0.35	0.46	0.51	0.45	Yes	0.52
GT11	523–1,288	0.54	0.48	0.80	0.67	Yes	0.56[a]
GT12	187–252	0.53	0.85	0.98	0.84	Yes	0.52[a]
GT13	441–459	0.53	0.33	0.70	0.65	Yes	0.56[a]
GT14	454–465	0.33	0.34	0.44	0.37	Yes	0.51
GT15	212–305	0.35	0.12	0.39	0.36	Yes	0.50
GT16	NA[b]	—	—	—	—		—

a. Manufacturer's equivalent VFo value is less than maximum test result VFot.

b. There is no opening requirement within the plant design basis for this group of MOVs.

required thrust (Tcst,s) at close torque switch trip under static conditions. This minimum thrust will ensure that the torque switch will not trip (excluding the effects of actuator output torque repeatability) before the valve disc reaching the close valve seat and initiating wedging of the disc into the seat, shown as

$$\text{Tcst,s} = (\text{RSFt})(\text{Tdpt.d}) \ . \qquad (18)$$

STEP 3: Along with other test data, use Tcst,s in place of Tdpt,d in the standard industry equation for calculating thrust requirements, and solve for the static equivalent valve factor (VFcts). Instead of solving for VFct with Equation (13), Equation (19) below is solved for VFcts as shown in Equation (20) by

$$\text{Tcst,s} = (\text{RSFt})(\text{Tdpt.d}) = (\text{VFcts})(\text{DPTc})(\text{Ao}) + (\text{Pupc})(\text{Ap}) + \text{RTs,c} \qquad (19)$$

$$\text{VFcts} = [\text{Tdpt,s} - \text{RTs,c} - (\text{Ap})(\text{Pupc})]/[(\text{Ao})(\text{DPTc})] \ , \qquad (20)$$

where

VFcts = static equivalent valve factor, which accounts for the ROL effect observed for a particular MOV during static and DP testing.

[Note: Equations (13) and (19) are applicable to gate valves; the equations for globe valves are slightly different.]

As is clear from Equation (19), the use of VFcts in the standard industry equation for predicting

thrust requirements ensures that the calculated minimum required static condition thrust at close torque switch trip is great enough so that the close torque switch will not trip at a thrust of magnitude equal to Tdpt,d under the tested DP conditions (excluding the influence of actuator output torque repeatability at torque switch trip).

STEP 4: For each group of similar MOVs, determine the average VFcts value (AVE VFcts) and the sample standard deviation. Determine the maximum potential VFcts value (MAX VFcts) as the greater of (a) the maximum test data value for VFcts, and (b) the sum of AVE VFcts plus two standard deviations.

STEP 5: Compare the value of MAX VFcts with the maximum postulated valve factor [(VFc)(1+eVFc)] for the group of similar MOVs. If the uncertainty by which VFc is increased is sufficiently great, then further increase the minimum required thrust prediction for ROL effects. The determination of ROL and eROL is

1. IF

$$(VFc)(1 + eVFc) > MAX\ VFcts \quad (21)$$

THEN

ROL = 1.000 and eROL = 0.000

2. OTHERWISE

ROL = the greater of 1.00 and the ratio (AVE VFcts/VFc) (22)

where

ROL = a factor by which the average valve factor (VFc) determined for the group of MOVs tested under DP conditions is multiplied to obtain the average static equivalent valve factor. In no case is the value of ROL less than 1.00.

This takes care of the average ROL effect for the group of valves, but the maximum potential ROL effect must still be considered. This is addressed by the uncertainty eROL, the value of which is determined as

3. IF

$$(VFc)(1 + eVFc)(ROL) > MAX\ VFcts \quad (23)$$

THEN

eROL = 0.000

4. OTHERWISE

$$eROL = [(\{MAX\ VFcts/[(VFc)(ROL)]\} - 1)^2 - (eVFc)^2]^{1/2} \quad (24)$$

Thus, the ROL effect is accounted for by adjustment of the valve factor that affects the predicted minimum required thrust. The adjusted valve factor (VFrol) is not less than MAX VFcts, shown as

$$VFrol = (VFc)(ROL)(1 + [eVFc^2 + eROL^2]^{1/2}) \quad (25)$$

Tables 9a and 9b provide the results of this analysis.

Because the author is not aware of any other organization that has addressed the ROL effect in this manner, Table 10 provides ROL results in the terms with which most are familiar (assuming the ROL factor, ROLt, for a static and DP tested MOV equals RSFt). Table 10 includes results for globe valves as well, whereas Tables 9a and 9b include only gate valve results.

Equation (26) shows the TU Electric calculation for the minimum stem thrust (Treq,s) that is required under static test conditions at close torque switch trip (excluding the effects of torque switch repeatability, margin for stem factor degradation, and test instrumentation measurement uncertainty) to ensure that sufficient thrust will be

Table 9a. Rate of loading effect for torque switch controlled closure MOVs.

Group ID	Maximum RSFt	Maximum VFcts data	Average VFcts	Average VFcts + 2s	MAX VFcts	VFc	eVFc	(VFc) (1 + eVFc)
GT1	1.396	1.029	0.941	1.189	1.189	0.67	0.02	0.68
GT6	1.176	0.600	0.547	0.698	0.698	0.50	0.06	0.53
GT7	1.111	0.493	0.492	0.495	0.495	0.43	0.16	0.50
GT9	1.028	0.305	0.262	0.383	0.383	0.26	0.47	0.38
GT11	1.024	0.897	0.523	0.916	0.916	0.56	0.70	0.95
GT12	1.150	0.765	0.724	0.841	0.841	0.47	0.87	0.88
GT15	1.082	0.439	0.405	0.501	0.501	0.39	0.11	0.43
GT16	1.174	0.567	0.508	0.594	0.594	0.47	0.31	0.62

Table 9b. Rate of loading effect for torque switch controlled closure MOVs.

				Verification		
Group ID	ROL	(VFc) (1 + eVFc) (ROL)	eROL	VFrol	> MAX VFcts ? (yes/no)	
GT1	1.40	0.96	0.26	1.19	>1.03	Yes
GT6	1.09	0.58	0.27	0.70	>0.60	Yes
GT7	1.00	0.50	0.00	0.50	>0.49	Yes
GT9	1.01	0.39	0.00	0.39	>0.31	Yes
GT11	1.00	0.95	0.00	0.95	>0.90	Yes
GT12	1.00	0.88	0.00	0.88	>0.77	Yes
GT15	1.04	0.45	0.21	0.50	>0.44	Yes
GT16	1.00	0.62	0.00	0.62	>0.57	Yes

Table 10. Rate of loading as a ratio of the DP to the static condition stem factor.

	MOV type		
	Globe	Gate Group GT1	Gate Groups GT2 to GT16
Average ROL = AVE RSFt	1.11	1.30	1.04
Sample standard deviation, s	0.15	0.14	0.10
(AVE RSFt + 2s)	1.41	1.58	1.24
(AVE RSFt + 3s)	1.56	1.72	1.34
Maximum ROL = Max RSFt	1.51	1.40	1.31
Number of data points, n	15	2	45

developed at close torque switch trip under maximum design basis DP conditions

$$Tseat,s = (RT)(1 + eRT) + (Ap)(Pupc) + (Ao)(DPRc)(VFc)(ROL)(1 + [(eVFc)^2 + (eROL)^2]^{1/2}) \quad . (26)$$

To this minimum required stem thrust, an additional load is added for selected MOVs to ensure that sufficient thrust is developed before close torque switch trip to not only reach the valve seat under maximum design basis DP conditions but also to press the disc seat surface into the body seat surface with sufficient force to effect a pressure seal.

RATE OF LOADING EFFECT MAGNITUDES FOR LIMIT SWITCH CONTROLLED CLOSURE MOVS

Limit switch controlled closure is used only with some of the actuators having compensator spring packs. In these cases, the thrust at control switch trip occurs at a specific number of turns of the limit switch rotor. The number of turns at which the close limit switch is set ensures that the disc is wedged into the valve seat, even under DP conditions, with the compensator spring pack compressed a specific amount. Because the compensator spring pack deflects the same amount under both static and DP conditions, the stem thrust at close limit switch is the same under both conditions and shown as

$$Tcst,d = Tcst,s \quad , \qquad (27)$$

where

Tcst,d = stem thrust at close limit switch trip under DP test conditions

Tcst,s = stem thrust at close limit switch trip under static test conditions.

The stem factor (SF) may be different under the two conditions, just as for torque switch controlled closure MOVs, with the DP condition having the greater friction coefficient. Consequently, the stem torque at close limit switch trip may be greater under the DP condition and shown as

$$TQcst,d > TQcst,s \quad , \qquad (28)$$

where

TQcst,d = stem torque at close limit switch trip under DP test conditions

TQcst,s = stem torque at close limit switch trip under static test conditions.

Thus, from this point of view, the ROL effect for limit switch controlled closure MOVs is the same as for torque switch controlled. For limit switch controlled closure MOVs TU Electric uses the familiar ratio of stem factors to determine from static and DP test data the ROL effect factor (ROLt) and shown as

$$ROLt = RSFt > 1.00 \qquad (29)$$

$$ROLt = (SFcst,d/SFcst,s) = (Tcst,s/TQcst,s)/(Tcst,d/TQcst,d) \quad , \qquad (30)$$

because Tcst,s = Tcst,d the above can be simplified to

$$ROLt = TQcst,d/TQcst,s > 1.00 \quad . \qquad (31)$$

Also, because Tcst,d = Tcst,s the static test minimum required stem thrust at control switch trip given by equation (26) is equal to the value calculated for the maximum design basis DP condition by Equation (15). This is accomplished for limit switch controlled closure MOVs by using Equation (26) values for ROL and eROL as

$$ROL = 1.00$$

$$eROL = 0.00 \quad .$$

The ROL effect for limit switch controlled closure MOVs must be accounted for when verifying with a static test that (a) the motor's minimum

capability is not exceeded by the torque at close limit switch trip (TQclst,s), and (b) the structural torque rating of the actuator and the qualified operating torque of the valve structure are not exceeded by the total torque (TQTOTs) developed after limit switch trip because of (a) the time it takes the electrical control system to remove power from the motor and (b) the kinetic energy of the moving parts, primarily the motor rotor, that continue to drive the disc into the valve seat until all parts stop moving. This verification could be accomplished by increasing the measured values of TQcst,s and TQTOTs by the applicable ROL factor, and verifying that the increased torques do not exceed their limits. This approach requires manipulation of the test data following the test in order to know if the test result is acceptable, which lengthens the time required to complete a test and verify that the results are acceptable. The approach TU Electric uses is to multiply the limiting torque values by the inverse of the typical ROL factor prior to performing the test, and then to compare the test results against the reduced limits.

The inverse of ROLt is the factor DTQt as shown by

$$DTQt = 1/ROLt = TQcst,s/TQcst,d < 1.00 \quad . \qquad (32)$$

TU Electric has statistically analyzed the DTQt test data for 24 MOVs to determine the average value of the ROL effect (DTQ) for limit switch controlled MOVs, and an uncertainty (eDTQ) associated with the value of DTQ. The uncertainty eDTQ is combined by the SRSS technique with other uncertainties in the test procedure, including the repeatability of thrust at close limit switch trip (REPls). The total uncertainty is used with the value of DTQ to reduce the maximum allowable torque values for comparison with the measured values of TQcst,s and TQTOTs.

Table 11 presents test data analysis results for DTQ and eDTQ. Because both static and DP tests are required to obtain test data, when determining the value of eDTQ, the effects of actuator output thrust repeatability at close limit switch trip ($\pm 3\%$) have been analytically extracted.

DISC POSITION EFFECT

Some limit switch closed MOVs exhibit another phenomenon under DP conditions that is unrelated to the change in friction coefficients. Under DP conditions the disc's final position in the valve seat may be oriented differently than under static conditions. While the author has not determined precisely what is occurring, the author has postulated from the observed effects that the stem's final position in the valve under DP conditions is slightly further out of the valve seat (a distance dL about 0.025 to 0.055 in.) than it is under static conditions. It may be that a change of dimensions of the valve internals could prevent this from occurring.

With the close limit switch setting unchanged, the stem nut's final position relative to the stem is the same. This means that the stem nut must

Table 11. Rate of loading effect for limit switch controlled closure MOVs.

Group ID	DTQ	eDTQ	DT	eDT
GT1	0.68	0.22	0.85	0.10
GT2	0.98	0.11	1.00	0.00
GT3	0.98	0.11	1.00	0.00
GT8	0.98	0.11	1.00	0.00
GT9	0.98	0.11	1.00	0.00
GT10	0.98	0.11	1.00	0.00
GT12	0.98	0.11	1.00	0.00
GT16	0.92	0.15	0.98	0.06

compress the compensator spring pack of the actuator an additional distance dL. The additional compression of the compensator spring pack under DP conditions results in a greater stem thrust at close limit switch trip under DP conditions (Tcst,d) than under static conditions (Tcst,s). In turn, the greater Tcst,d value requires a greater torque delivered to the stem nut at close limit switch trip under DP conditions (TQcst,d) than under static conditions (TQcst,s).

Thus, for some limit switch controlled closure MOVs, not only is there a ROL effect to consider when setting (or verifying the acceptability of the setting) the close limit switch under static conditions, there is a disc position effect. While caused by a completely different phenomenon than the rate of loading effect, the result is similar and is treated similarly. The ratio (DTt) of Tcst,s divided by Tcst,d is the factor by which the thrust limits must be multiplied to obtain reduced thrust limits used during a static test to verify that the thrust limits will not be exceeded during a stroke under DP conditions. The ratio (DTQt) of TQcst,s divided by TQcst,d is the factor by which the torque limits must be multiplied to obtain reduced torque limits used during a static test to verify that the torque limit will not be exceeded during a stroke under DP conditions.

Although TU Electric data analysis has not separated the individual components of the two phenomena, data from a group GT1 MOV shows the two effects are additive by

$$\text{TQcst,d} = (\text{Tcst,d}/\text{Tcst,s})(\text{SFcst,d}/\text{SFcst,s})(\text{TQcst,s})$$

$$167 \text{ ft-lb} = (1.1105)(1.1717)(128.0)$$

$$167 \text{ ft-lb} = 167 \text{ ft-lb} \ .$$

Based on analysis of DTt and DTQt test data for individual MOVs, the average magnitudes of the disc position and rate of loading combined effects are DT for the thrust limit reduction factor and DTQ for the torque limit reduction factor. Associated with DT and DTQ are uncertainties eDT and eDTQ, respectively. The values of eDT and eDTQ are determined with recognition that TU Electric is conservatively assuming REPIs equal 3% and that independent uncertainties are combined by the SRSS method. As shown in Table 11, two Comanche Peak valve groups, GT1 and GT16, showed sensitivity to the disc position effect.

Equation (33) shows how the rate of loading effect is accounted for in the engineering calculation's adjustment of the motor capability at 80% voltage in the closing stroke of a limit switch controlled closure MOV. The uncertainty shown in Equation (12b) is further combined with the uncertainty eDTQ, while the average rate of loading effect, DTQ, is directly applied to the calculated motor capability and shown as

$$\text{TQmax,c} = (0.8)^2 (\text{MTQ})(\text{OAR})(\text{AF})(\text{RE})(\text{PF/ST})(\text{DTQ})(1 - [\text{ePF}^2 + \text{eST}^2 + \text{eDTQ}^2]^{1/2}) \ . \tag{33}$$

Equation (34) shows how the rate of loading effect for limit switch controlled closure MOVs is accounted for in the engineering calculation's adjustment of the thrust load limit, which is compared in the test procedure to the total thrust developed after close limit switch trip, shown as

$$\text{TTOTmax,c} = (\text{Tlimit})(\text{DT})(1 - \text{eDT}) \ . \tag{34}$$

For torque switch controlled closure MOVs, the values of DTQ and eDTQ are such that there is no reduction of the total thrust limit, shown by

$$\text{DTQ} = 1.00 \ , \text{ and eDTQ} = 0.00 \ .$$

STEM FRICTION COEFFICIENT RANGE

Stem friction coefficients have been calculated using measured stem thrust and stem torque data at the following points of interest in the closing and opening strokes for a total of over 700 data points, which were then statistically analyzed to determine average values and standard deviations:

- Peak thrust to overcome DP condition before wedging

- Thrust at close control switch trip
- Peak thrust following close control switch trip
- Peak thrust to initiate unwedging of the disc
- Peak thrust to overcome DP condition after unwedging the disc.

There appears to be a trend of lesser stem thread friction coefficients during initial unwedging of the disc from the valve seat than at other times in the closing and opening strokes. There was no indication that closing stroke friction coefficients were in general any different from opening stroke friction coefficients at other points of interest. Some MOV groups appear to have less variation in stem friction coefficients than others. The reason for this has not been determined.

Table 12 provides the results of the analysis for all data points taken together. TU Electric uses similar stem friction coefficient analysis results to calculate six corresponding stem factors. For the average unseating stem factor, associated uncertainties are determined by which the average stem factor value must be increased and decreased to obtain the maximum and the minimum stem factors. For the average closing and opening stroke stem factor, associated uncertainties are determined by which the average stem factor value must be increased and decreased to obtain the maximum and the minimum stem factors. Any conversion between stem thrust and stem torque uses the average stem factor value and combines the stem factor uncertainty with other uncertainties by the SRSS technique.

TU Electric has assumed that any one MOV out of a group of similar MOVs can have stem friction coefficients anywhere within the range determined for that group at some point in the operating life of the plant. This risk is accounted for in the engineering calculation by combining the stem factor uncertainties with other uncertainties by the SRSS technique.

Tim Cline of Duke Power has suggested that for some MOVs the outer edge of the stem thread may rub on the stem nut thread in a way that develops radial loads at the corner of the thread and the major diameter. Radial loads would be accompanied by tangential friction loads that consume torque. Less torque would therefor be available for producing thrust. The standard ACME thread equation for relating torque to thrust does not account for this potential source of inefficiency. If the outer edge of the stem thread were more rounded, it may be that the stem thread efficiency could be increased, with the result that stem factor values would be reduced.

Equation (35) shows the calculation of the minimum stem torque required at close torque switch or limit switch trip (excluding any extra torque required for producing a pressure seal, margin for stem factor degradation, control switch repeatability effects, and test instrument measurement uncertainty) as

$$TQseat = RTQs + TQrej,c + TQdpr,c \quad (35)$$

where

$$RTQs = (SF)(RT)(1 + [(eRT)^2 + (eSFu)^2]^{1/2} \quad (36)$$

$$TQrej,c = (SF)(Ap)(Pupc)(1 + eSFu) \quad (37)$$

$$TQdpr,c = (SF)(Ao)(DPRc)(VFc)(1 + [(eVFc)^2 + (eSFu)^2]^{1/2} \quad (38)$$

SF = the stem factor based on the average stem thread friction coefficient

$$eSFu = (SFu - SF)/SF \quad (39)$$

SFu = the stem factor for the maximum postulated stem thread friction coefficient.

Table 12. Stem thread friction coefficient range.

Based on analysis of over 700 test data points from calibrated strain gages mounted on the stem	During initial unseating (unwedging of the disc)	At other points of interest: peak load to overcome DP, control switch trip, and total load in the closing and opening strokes
Maximum friction coefficient	Uun,u = 0.19	Uu = 0.20
Average friction coefficient	Uun = 0.10	U = 0.12
Minimum friction coefficient	Uun,l = 0.03	Ul = 0.05

Appendix B

Acronyms

Appendix B

Acronyms

AF — Application factor, specified by the actuator manufacturer, equal to 0.9.

Ao — Valve orifice area based on the mean diameter of the valve seat ring as specified by the valve manufacturer.

AOT — Actuator output thrust, produced at the interface of the actuator's threaded stem nut and the threaded stem.

AOTQ — Actuator output torque applied by the actuator's threaded stem nut to the valve stem.

AOTQ80stall — Measured output torque of the actuator when the motor stalls during torque stand testing with power to the motor at 80% of the motor nameplate voltage.

AOTQ80w — The standard predicted capability of the motor to deliver torque to the worm gear when the motor is powered by 80% of the motor nameplate voltage and the resisting drive sleeve torque is gradually applied over several seconds: (MTQ80)(OAR)(AF)(RE).

AOTQtst — Actuator output torque at torque switch trip.

AOTQw — Actuator output torque with no thrust load applied to the actuator drive sleeve, such as occurs during testing on a torque test stand. Also, the input torque to an HBC gearbox. Approximately the torque applied by the worm to the worm gear.

Ao — Nominal seat orifice area based on the mean seat diameter specified by the valve vendor.

Ap — Stem cross-section area at the packing.

AVE — Average of a set of values.

AVE VFcts — The average VFcts value.

CTQun,ds — A coefficient (not less than 1.00) based on static and DP test data which when multiplied by the measured static test unseating torque (TQun,s) provides a maximum predicted unseating torque under the more severe of the static and the DP conditions.

CTun,ds — A coefficient (not less than 1.00) based on static and DP test data which when multiplied by the measured static test unseating thrust (Tun,s) provides a maximum predicted unseating thrust under the more severe of the static and the DP conditions.

Cwdg	The wedging coefficient, based on analysis of test data under both static and DP conditions, by which the total closing stroke thrust can multiplied to obtain an estimate of the thrust required to unseat the valve under the more severe of the static and the DP conditions.
DPRc	Closing stroke maximum design-basis (required) differential pressure.
DPRo	Opening stroke maximum design-basis (required) differential pressure.
DPTc	Maximum differential pressure measured during the closing stroke of the DP test.
DPTo	Maximum differential pressure measured during the opening stroke of the DP test.
DT	The average value of the disc position effect (the average of the DTt values) on stem thrust magnitudes at and after close limit switch trip.
DTt	As observed from static and DP testing of a given MOV, the factor by which the thrust limits must be multiplied to obtain reduced thrust limits which may be used during evaluation of static test results to verify that the thrust limits will not be exceeded during a stroke under DP conditions: (Tcst,s/Tcst,d).
DTQ	The average value of the combination effects of the rate of loading and disc position phenomena for limit switch controlled MOVs (the average of the DTQt values).
DTQt	The net effect of the rate of loading and disc position phenomena observed from a set of static and DP tests of a limit switch controlled MOV, being the factor by which the torque limits must be multiplied to obtain decreased torque limits for use during static testing to verify the torque limits will not be exceeded at or after close limit switch trip under DP conditions, equal to the inverse of ROLt: (1/ROLt) = (TQcst,s/TQcst,d) < 1.00 .
eDSF	Uncertainty associated with potential stem factor degradation (10%).
eDT	Uncertainty associated with the value of DT.
eDTQ	Uncertainty associated with the value of DTQ.
eMARM	Uncertainty associated with the value of MARM.
eMARMw	Uncertainty associated with the value of MARMw.
ePF	Uncertainty associated with the value of PF.
eROL	Uncertainty associated with the value of ROL.

eSFu — Uncertainty associated with the value of SF relative to the value of SFu: [(SFu/SF) – 1] .

eST — Uncertainty associated with the value of ST.

eVFc — Uncertainty associated with the value of VFc.

eVFo — Uncertainty associated with the value of VFo.

GL — Generic Letter issued by the USNRC.

MARM — The effective drive sleeve moment arm length that relates SPF to the torque applied by the stem nut to the stem (AOTQ).

MARMw — The average effective worm gear moment arm length which relates SPF to the torque on the worm gear (AOTQw).

MAX VFcts — The maximum potential VFcts value, being the greater of (a) the maximum test data value for VFcts, and (b) the sum of AVE VFcts plus 2 standard deviations of the VFcts data.

MOV — Motor-operated valve.

MTQ — Motor torque rating (ft-lb), the motor nameplate start torque.

MTQ80 — Alternating current motor design capability defined by as 64% of the motor torque rating (MTQ) when the motor is supplied with power at 80% of the motor's nameplate voltage: $(0.80)^2$ (MTQ) .

MTQstall — Measured motor shaft stall torque during dynamometer testing with power supplied at 80% of the motor's nameplate voltage, and where the test was performed on a dynamometer by gradually applying a braking torque to the motor shaft over a period typically somewhere between 10 to 40 seconds.

NA — Not applicable.

NET — The coefficient by which the motor capability to deliver torque to the stem of a rising nonrotating stem valve, commonly assumed to be the value calculated for AOTQ80w, must be multiplied to obtain a value which accounts for the stem thrust effect and the performance factor: $(PF)(1 - [(ePF)^2 + (eST)^2]^{1/2})/ST$.

OAR — Overall gear ratio of the actuator from motor to stem nut (or HBC input shaft).

PF — Actuator performance factor defined by TU Electric for a group of similar actuators, equal to the average of the ratios of the measured stall torque value (AOTQ80stall) divided by design value (AOTQ80w) for each actuator in the similarity group.

POE	Pullout efficiency of the actuator, specified by the actuator manufacturer.
Pupc	Closing stroke maximum design basis upstream pressure accompanying DPRc.
Pupo	Opening stroke maximum design basis upstream pressure accompanying DPRo.
Pupct	Maximum upstream pressure measured during the closing stroke of the DP test.
Pupot	Maximum upstream pressure measured during the opening stroke of the DP test.
RE	Running efficiency of the actuator, specified by the actuator manufacturer for each combination of motor rpm, actuator size, and OAR.
REPls	Repeatability of stem thrust at close limit switch trip where the close limit switch trips only when the valve disc is being wedged into the valve seat and the actuator compensator spring pack is deflecting in response to the stem thrust load (±3% based on testing).
REPts	Actuator vendor's specified torque switch repeatability (±5, 10, or 20%).
ROL	Or, a factor equal to the average magnitude of the rate of loading effect (average of ROLt values) by which the average valve factor (VFc) of an MOV is multiplied to obtain the average static equivalent valve factor. In no case is the value of ROL less than 1.00.
ROLt	The rate of loading effect observed for a given MOV from a set of DP and static tests. Often defined as being equal to RSFt.
RSFt	The ratio of the DP test stem factor divided by the static test stem factor at close torque switch trip. It is this factor that is often considered by the indus- try to be the rate of loading: (SFcst,d/SFcst,s) = (Tcst,s/TQcst, s)/(Tcst,d/TQcst,d) > 1.00 .
RT	Running thrust.
RTs,c	Average static test running thrust over the last 10% or so of the closing stroke.
RTs,o	Average static test running thrust over the first 10% or so of the opening stroke.
s	Sample standard deviation of a set of values.
SF	Stem factor—The average stem factor at points of interest in the closing and opening strokes, except at initial unwedging of the disc (see SFun).

SFu — The maximum postulated stem factor at points of interest in the closing and the opening strokes, except at initial unwedging of the disc (see SFun,u).

SF10 — Stem factor for a 0.10 friction coefficient.

SF15 — Stem factor for a 0.15 friction coefficient.

SF20 — Stem factor for a 0.20 friction coefficient.

SPD — Spring pack deflection of the torque spring pack.

SPF — Spring pack force, the force which causes the torque spring pack to deflect.

SRSS — The square root of the sum of the squares of two or more numbers.

ST — The average value of the ratio of AOTQw divided by AOTQ, a factor to account for the stem thrust effect.

Tcst,d — Stem thrust at close limit switch trip under DP test conditions.

Tcst,s — Stem thrust at close limit switch trip under static test conditions. Or, the estimated minimum required thrust at close torque switch trip under static conditions that will ensure the thrust Tdpt,d can be produced in the closing stroke under the DP test conditions without tripping the close torque switch (excluding the effects of actuator output torque repeatability): (RSFt)(Tdpt.d).

TDPo — Maximum thrust required at any point during the opening stroke of the DP test after initial unwedging of the disc (overcoming the static friction coefficient in the valve seat).

Tdpt,d — Maximum thrust required at any point during the closing stroke of the DP test until the disc begins to wedge into the valve body seat.

TQcst,d — Stem torque at close limit switch trip under DP test conditions.

TQcst,s — Stem torque at close limit switch trip under static test conditions.

TQreq — Maximum stem torque potentially required to produce thrusts Treq,d and Treq,s.

TQsg — Stem torque value measured using calibrated stem-mounted strain gages.

TQstall — Actual actuator output torque at the motor stall condition with no thrust load on the drive sleeve.

TQTOTs — The total torque developed after control (torque or limit) switch trip because of (a) the time it takes the electrical control system to remove power from the motor, and (b) the kinetic energy of the moving parts, primarily the motor rotor, which continue to drive the stem until all parts stop moving.

TQtrp,s	Static test torque at close limit switch or torque switch trip.
TQtts	Actuator output torque measured during torque stand testing which applies a resisting torque to the actuator drive sleeve (with no thrust load on the drive sleeve).
Treq,d	Maximum thrust to reach seat under DP conditions and initiate wedging: (VFc)(Ao)(DPRc) + (Ap)(Pupc) + RT.
Treq,s	Thrust produced under static conditions by the stem torque magnitude which produced Treq,d under DP conditions.
TTOTs	The total thrust developed after control (torque or limit) switch trip because of (a) the time it takes the electrical control system to remove power from the motor, and (b) the kinetic energy of the moving parts, primarily the motor rotor, which continue to drive the stem until all parts stop moving.
Ttrp,s	Static test thrust at close limit switch or torque switch trip.
TSS	Torque switch setting.
U	The average stem thread friction coefficient at points of interest in the closing and the opening strokes, except at initial unwedging of the disc (see Uun).
Ul	The minimum postulated stem thread friction coefficient at points of interest in the closing and the opening strokes, except at initial unwedging of the disc (see Uun,l).
USNRC	U.S. Nuclear Regulatory Commission.
Uu	The maximum postulated stem thread friction coefficient at points of interest in the closing and the opening strokes, except at initial unwedging of the disc (see Uun,u).
Uun	The average stem thread friction coefficient during initial unwedging of the disc from the valve seat.
Uun,l	The minimum postulated stem thread friction coefficient during initial unwedging of the disc from the valve seat.
Uun,u	The maximum postulated stem thread friction coefficient during initial unwedging of the disc from the valve seat.
VFc	The average valve factor in the closing stroke for a group of nominally identical valves.
VFct	The actual closing stroke valve factor determined from test data.

VFcts — The static equivalent valve factor that accounts for the rate of loading effect observed for a particular MOV during static and DP testing. The use of VFcts in the standard industry equation for predicting thrust requirements ensures the calculated minimum required static condition thrust at close torque switch trip is great enough so that the close torque switch will not trip at a thrust of magnitude equal to Tdpt,d under the tested DP conditions (excluding the influence of actuator output torque repeatability at torque switch trip).

VFo — The average valve factor in the opening stroke for a group of nominally identical valves after initial unwedging of the disc (overcoming the static friction coefficient of the valve seat).

VFot — The actual opening stroke valve factor determined from test data after initial unwedging of the disc (overcoming the static friction coefficient in the valve seat).

VFrol — A valve factor not less than MAX VFcts which has been adjusted to account for the average valve factor (VFc) and its uncertainty (eVFc), and the average rate of loading effect (ROL) and its uncertainty (eROL) calculated as follows: $(VFc)(ROL)(1 + [eVFc^2 + eROL^2]^{1/2})$.

W/WG ratio — The gear ratio provided by the worm and the worm gear.

Enhancements to the Idaho National Engineering Laboratory Motor-Operated Valve Assessment Software[a]

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ABSTRACT

In January 1991, the U.S. Nuclear Regulatory Commission (USNRC) commenced Part 1 inspections to review licensee's motor-operated valve (MOV) programs that were developed to address Generic Letter 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance." In support of this effort, the Isolation Valve Assessment (IVA) software, Version 3.10, was developed by the Idaho National Engineering Laboratory (INEL) to enable rapid in-depth review of MOV sizing and torque switch setting calculations. In 1994, the USNRC commenced Part 2 inspections, which involve a more in-depth review of MOV in situ testing relative to design-basis assumptions. The purpose of this paper is to describe the latest INEL and industry research that has been incorporated into Version 4.00 of the IVA software to support the latest round of inspections. Major improvements include (a) using dynamic and static test results to determine MOV performance parameters and validate design-basis engineering assumptions, (b) determining the stem/stem-nut coefficient of friction using new research-based techniques, (c) adding the ability to evaluate globe valves, and (d) incorporating new methods to account for the effects of high ambient temperature on the output torque of alternating current (ac) motors.

INTRODUCTION

The Idaho National Engineering Laboratory (INEL) has been performing motor-operated valve (USMOV) research for the U.S. Nuclear Regulatory Commission's (USNRC) Office of Nuclear Reactor Research and inspection support for the USNRC's Office of Nuclear Reactor Regulation's efforts regarding Generic Letter 89-10 (GL 89-10), "Safety-Related Motor-Operated Valve Testing and Surveillance." One aspect of this support included development of a user-friendly personal computer (PC) based software program to perform MOV operability calculations (e.g., required stem thrust, available operator torque, and degraded voltage cable calculations) for wedge type gate valves. In July 1991, Version 3.10 of the IVA software was provided to the USNRC, and was subsequently made available for industry use. Since that time, the INEL has continued its research efforts,

a. Work supported by the U.S. Nuclear Regulatory Commission, Office of Nuclear Regulatory Research, Division of Engineering and Division of Safety Issues Resolution, USNRC Technical Monitor G. H. Weidenhamer, under DOE Contract No. DE-AC07-76ID01570.

gathered industry comments, and completed work on a revised version of the IVA software, Version 4.00.[b]

IVA SOFTWARE, VERSION 4.00

Version 4.00 of the IVA software was developed to broaden the scope of evaluation methods and to incorporate new features designed to address needs identified since the original software version was released. The following are some important features of the new releases:

- Dynamic Test Evaluation—allows determination of a valve's disc-to-seat coefficient of friction using INEL research results (hereafter referred to as the "INEL friction factor"), disc factor, hooking factor, and design-basis stem/stem-nut coefficient of friction based on the INEL's Threshold method.

- Static Test Evaluation—predicts design-basis stem/stem-nut coefficient of friction from static test results using the INEL's Fold Line method.

- Enhanced Valve Evaluation—allows evaluation of globe valves, gate valves with wedge angles other than 5 degrees (including parallel gate valves), and gate valves with hooking characteristics.

- Enhanced Motor Evaluation—determines ac motor torque loss resulting from high ambient temperature effects and adds flexibility for determining degraded voltage conditions.

In addition to these enhancements, changes were made to the graphs to help identify the MOV margin available to identify operator thrust and torque limits. These modifications were implemented while maintaining compatibility with the features and existing data file structure of the previous version of IVA.

The enhancements made to the IVA software provide a powerful tool for an auditor to validate test results or evaluate utility design-basis assumptions made as part of these very complicated MOV calculations. The following section discusses the software enhancements in the test analysis mode (Validate & Extrapolate), followed by enhancements in the thrust calculational mode (Capability Assessment).

Validate & Extrapolate Mode

The Validate & Extrapolate mode evaluates either MOV static testing (no differential pressure and no flow) or dynamic testing (differential pressure and flow). The Validate & Extrapolate mode uses a general data screen, a test data entry screen (see Figure 1), and a results screen to summarize the estimated performance parameters. The globe valve test data entry screen is similar to Figure 1.

Dynamic Test Evaluation

A primary goal for this software update was to improve the tools available to auditors, who conducted performance-based evaluations of selected valve and operator parameters, such as the INEL friction factor, the disc factor, and the stem/stem-nut coefficient of friction. Because a majority of MOV dynamic testing performed to comply with the recommendations of GL 89-10 is performed at less than design-basis conditions, the ability to validate less than design-basis testing is essential to predicting design-basis performance. The Validate & Extrapolate mode summarizes the following performance characteristics derived from the test data (depending on the type and response of the valve being evaluated):

- Applicability of the INEL correlation

- INEL friction factor

- Disc factor

- Hooking factor

b. Isolation Valve Assessment (IVA) Software, Version 4.00 Users Manual, Project Number 015488, 1994, EGG-SSRE-9777, Rev. 1 (Draft).

```
IVA Version 4.00                                       Gate Valve Test Data (Sheet 2 of 2)
File:
Variable Use:

        Test Number:<  >          Type of Test:<              >
          Test Date:              Comments:
          Test Time:
                                Dynamic Test
                               Static Test Max      Static Test Min
           Operator Torque From:  <                        >
 Spring Pack Displacement (in.):
        Spring Pack Force (lbf):
       Operator Torque (ft-lbf):
        Valve Stem Thrust (lbf):
         Stem/Stem-Nut Friction:

       Upstream Pressure (psig):
   Differential Pressure (psid):
              Packing Drag (lbf):
                Fluid Subcooling:<            >
           Stem Thrust Response:<        >

F1-Help                  F5-Valve Response              PgUp/PgDn-new data sheet
```

Figure 1. Gate valve test data.

- Design-basis stem/stem-nut coefficient of friction.

The Validate & Extrapolate mode results screen is used to summarize the results of calculations for dynamic MOV testing (see Figure 2). The top section of the results screen, Estimated Valve Response at Test Conditions, determines whether the valve tested was typical of the gate valves tested by the INEL through use of the INEL correlation. First, the software estimates the net horizontal and vertical loads acting on the disc. Then, the loads are transformed into normal and sliding loads by accounting for the seat angle of the disc. Finally, the loads from the tested valve are compared with the performance characteristics of the valves tested by the INEL. If the sliding versus normal load of the tested valve falls within the bounds of the INEL correlation, the correlation is considered valid for the valve, and design-basis estimates based on the correlation can be used. Steele et al. (1992) contains more detail regarding the derivation of the INEL correlation.

In addition to comparing valve performance with INEL test results, Version 4.00 also uses the results of dynamic testing to determine performance parameters that can be compared with engineering assumptions. The Back Calculated Valve Estimates at Test Conditions section of the results screen displays the following:

- INEL friction factor—a variable in the INEL correlation that represents the coefficient of friction between the valve disc and the valve seat

- Disc factor—a variable in the industry standard thrust equation that relates the pressure force acting on the disc to the stem force that is required to close the valve

- Hooking factor—a term that estimates the peak stem thrust at design-basis conditions for gate valves expected to experience a peak thrust that causes the force trace to have a hook shape before the plateau that indicates flow isolation. This term accounts for disc-to-guide friction, disc tippage, and the pressure distribution across the valve disc resulted from the upstream pressure, the downstream pressure, and the bonnet pressure. This factor assumes that the disc is loaded sufficiently to stabilize the contact friction and that the response is linear with

```
Validate & Extrapolate Mode
File:

ESTIMATED VALVE RESPONSE AT TEST CONDITIONS:
   Total horizontal disc load
   Total vertical disc load
   Normalized normal disc force: (    400.) or greater
   Normalized sliding disc force: (        ) to (         )

BACK CALCULATED VALVE ESTIMATES AT TEST CONDITIONS:
   INEL friction factor
   Disc factor
   Hooking factor

ESTIMATED STEM/STEM-NUT RESPONSE AT TEST CONDITIONS:
   Thread pressure: (  10000.) or greater
   Stem/stem-nut coefficient of friction
   Stem factor

ESTIMATED STEM/STEM-NUT RESPONSE AT DESIGN BASIS CONDITIONS:
   Stem/stem-nut coefficient of friction
   Stem factor

Press any key to continue
```

Figure 2. Dynamic gate valve test results.

the differential pressure. The hooking factor can be used in the Capability Assessment mode to estimate the thrust requirements for a valve that exhibits this behavior.[c]

The Estimated Stem/Stem-Nut Response at Test Conditions section of the results screen estimates the thread pressure based on the actual stem force and the stem configuration, and the actual stem/stem-nut coefficient of friction based on the thrust and torque measured during the dynamic test. Research to date indicates that if the thread pressure exceeds 10,000 psi, the observed stem/stem-nut coefficient of friction should be representative of the coefficient of friction present under design-basis conditions. The Estimated Stem/Stem-Nut Response at Design Basis Conditions section at the bottom of the results screen displays the highest coefficient of friction observed from all of the dynamic tests stored in the valve's test database.

Static Test Evaluation

When only a static test can be performed, IVA can estimate the design-basis stem/stem-nut coefficient of friction based on use of the INEL Fold Line method. The highest and lowest coefficient of friction values based on the torque and thrust measurements recorded during the seating portion of the static test (see Figure 3) are entered into the test data screen. Research has shown that plotting the change in the coefficient of friction during the seating transient is a snapshot of the overall behavior of the stem and stem-nut combination. This change is used in the INEL Fold Line method to predict the design basis running coefficient of friction. The magnitude of the change has also been shown to be proportional to the propensity of the stem to exhibit load-sensitive behavior (rate of loading). The greater the change, the higher the load-sensitive behavior will be in the stem. IVA adds the difference between the two friction values to the higher value to arrive at an estimate that bounds the design-basis stem/stem-nut coefficient of friction. This result is displayed in the bottom section of the static test results screen (see Figure 4) for the highest test results in the test database. Steele et al. (1993) provides more detail regarding the development of this method.

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c. Steele, R. Jr., K. G. DeWall, J. C. Watkins, M. J. Russell, and D. Bramwell, 1994, *1993 Motor-Operated Valve Research Update* (DRAFT), NUREG/CR-6100, EGG-2711.

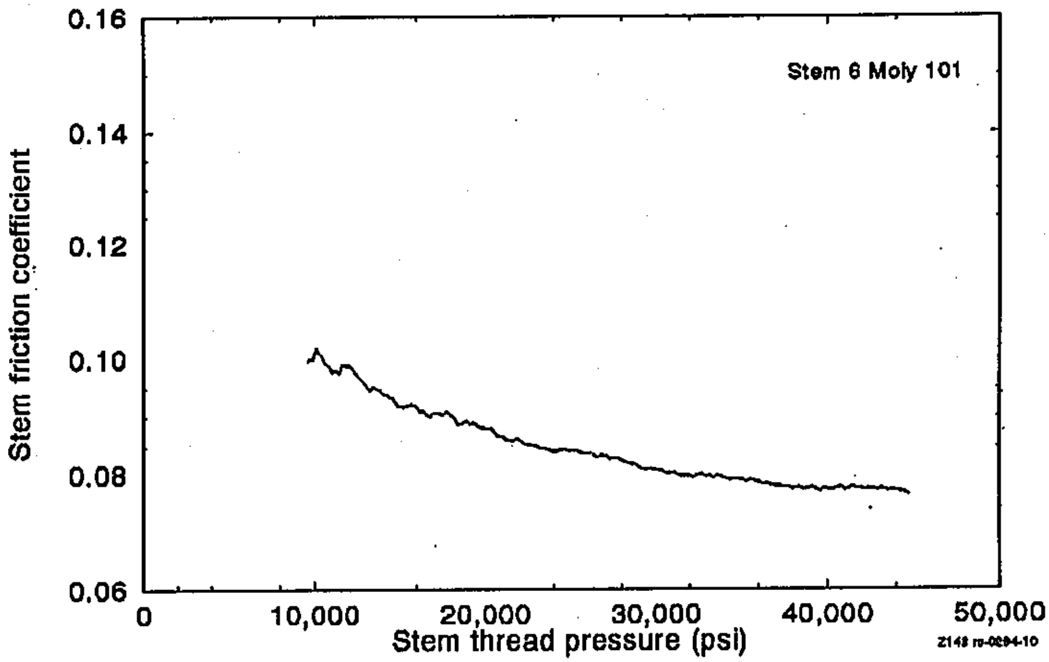

Figure 3. Coefficient of friction during the seating portion of a valve closure.

```
Validate & Extrapolate Mode
File:

Maximum   Minimum   ESTIMATED STEM/STEM-NUT RESPONSE AT TEST CONDITIONS:
                       Stem thrust
                       Stem torque

                       Thread pressure: (   5000.) or greater
                       Stem/stem-nut coefficient of friction
                       Stem factor

          ESTIMATED STEM/STEM-NUT RESPONSE AT DESIGN BASIS CONDITIONS:
             Stem/stem-nut coefficient of friction
             Stem factor

Press any key to continue
```

Figure 4. Static operator test results.

Other Validate & Extrapolate Mode Enhancements

Several graphs have been added to the Validate & Extrapolate mode that display assessments of

- A single dynamic gate valve test, or all tests in the database, relative to the results of gate valve testing by the INEL
- A stem factor (or stem/stem-nut coefficient of friction) versus stem thread pressure for all dynamic tests
- A stem factor (or stem/stem-nut coefficient of friction) versus stem thread pressure for all static tests.

The graph of sliding versus normal load (see Figure 5) shows the relationship between sliding and normal forces acting on the disc of gate valves. The outer sloping lines represent the limits of the data scatter observed during the INEL testing. The circle represents the estimated sliding and normal forces acting on the disc during the test being evaluated. The sliding and normal loads are determined through use of the INEL correlation.

Additional graphs allow the user to display the stem factor (or stem/stem-nut coefficient of friction) versus thread pressure for either the dynamic or static tests contained in the database. The stem factor versus thread pressure graph (see Figure 6) shows the estimated static stem factor versus thread pressure for each of the static tests in the database. A line connects the two common data points for each test. The horizontal line represents the maximum estimated design-basis stem factor contained in the static test database, as determined using the INEL Fold Line method.

The original version of IVA was limited to evaluating the results of a single test of a wedge gate valve. Now, Version 4.00 can store the results of up to 20 different diagnostic tests for each MOV in the database. For each database entry, the user can record the test date and time, the type of test, and descriptive comments.

In addition to thrust data, Version 4.00 allows the user to record torque-related data from diagnostic testing. Either spring displacement, spring force, or directly measured torque data can be entered. If spring displacement or spring force data from the test are entered, the software can use built-in spring calibration data, spring pack moment arm relationships, or both to estimate the corresponding operator output torque. This reduces the effort needed to determine output torque in those cases where direct torque measurements are not available. In addition, these data are also used by the dynamic gate valve test results screen (Figure 2) to estimate the stem/stem-nut coefficient of friction and the stem factor.

Capability Assessment Mode

The Capability Assessment mode estimates the design-basis thrust requirements and assesses the capability of the operator. Data entry screens are used to store general data, valve data, stem and stem-nut data, operator mechanical data, operator electrical data, and power supply data. The Capability Assessment Mode Results screen is designed to (a) compare the results of the INEL Gate Valve Stem Thrust Correlation and the standard industry gate and globe valve stem thrust equations relative to the output capacity of the operator under various voltage conditions and (b) provide an assessment of the available capability margin under design-basis conditions.

Revisions to the Capability Assessment mode provide the following functions:

- Stem Thrust Requirements—evaluation of globe valves, wedge gate valves with seat angles other than 5 degrees, and gate valves with hooking characteristics
- Operator Torque Considerations—includes operator and torque switch limits, and incorporates standard spring pack calibration data
- Electrical Considerations—estimates the effects of high ambient temperature on the output of ac motors and adds flexibility for determining degraded voltage conditions

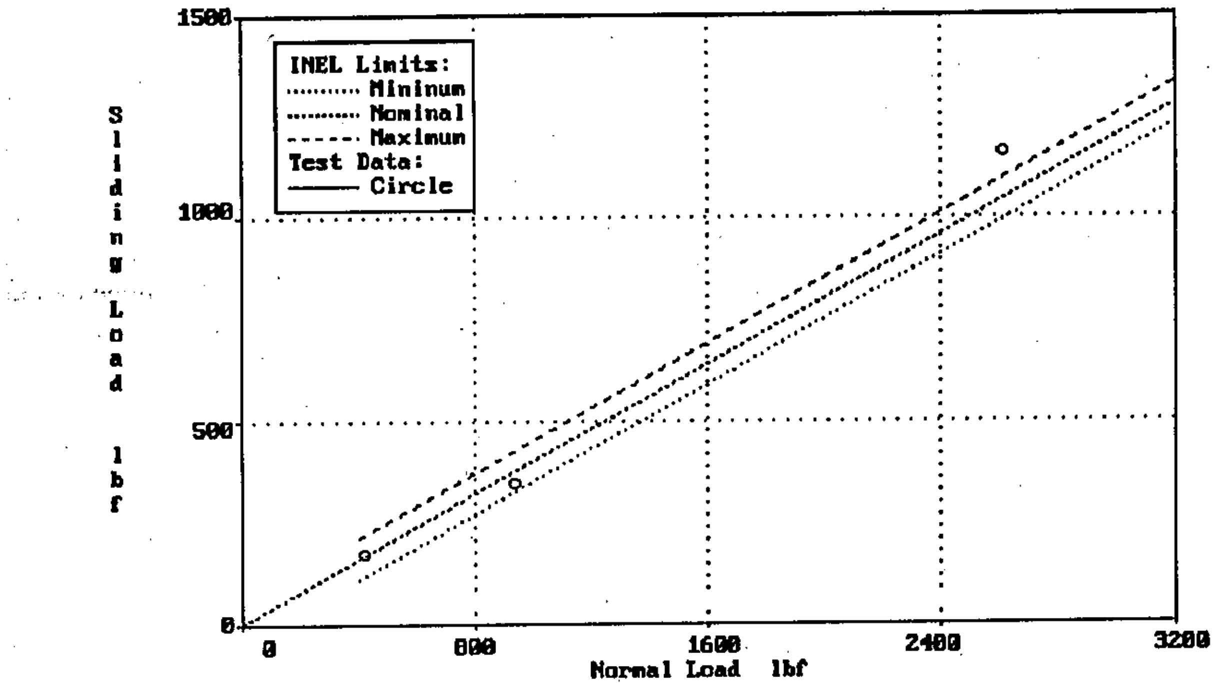

Figure 5. Sliding versus normal load.

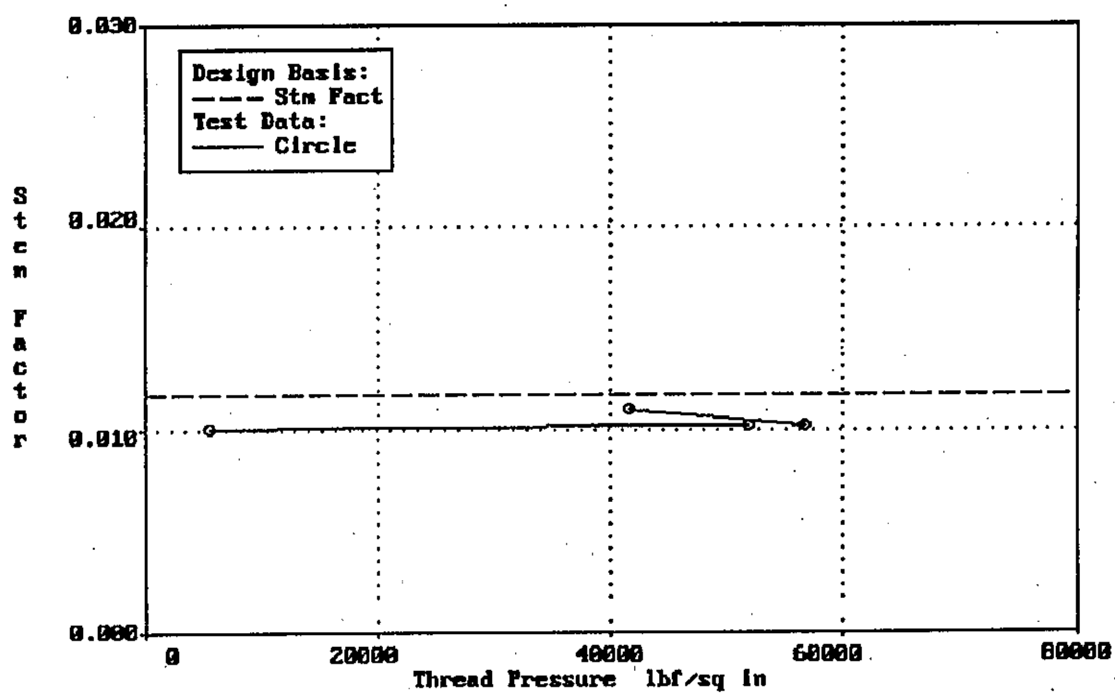

Figure 6. Stem factor versus thread pressure.

- Assessment of Available Margin—estimates the thrust margin available, based on the design-basis thrust requirements and the operator capability; graphs include operator limits and current torque switch setting.

Stem Thrust Requirements

IVA provides data entry screens (Figure 7 is an example for gate valves) for input of the system conditions and valve dimensions that are necessary for estimating MOV thrust requirements. Version 4.00 uses the INEL correlation as the primary method to determine the thrust necessary to overcome design-basis requirements. However, to make the software more adaptive to the variety of different methods used by the industry and to increase the types of valves that can be evaluated, the standard thrust equation is now available to estimate design-basis thrust requirements for both gate and globe valves. Where the original version of IVA was limited to evaluation of 5-degree wedge gate valves, the software now allows the user to enter a specific gate valve seat angle from 0 to 10 degrees. This seat angle is used in the INEL correlation to account for the forces acting on the disc that result from the angle of the seating surfaces.

For many valves, the peak force (before wedging) occurs after flow isolation when the valve disc is riding on the valve seats. This type of response is considered to be "predictable" behavior. However, as discussed in the Dynamic Test Evaluation section, some valves behave in a manner where the peak thrust occurs before the valve begins riding on the seating surface. This type of response is shown in Figure 8 and is referred to as "hooking." The software uses the entry in the Hooking Factor field multiplied by the differential pressure to determine the force necessary to overcome the hooking differential pressure effects. This term accounts for disc-to-guide friction, disc tippage, and the pressure distribution across the valve disc due to the upstream pressure, the downstream pressure, and the bonnet pressure. This factor assumes that the disc is loaded sufficiently to stabilize the contact friction and that the response is linear with the differential pressure. The hooking stem force equation is

$$F_{stem} = C_{hook}\,\Delta P + P_{up}\,A_{stem} + F_{pack}$$

where

C_{hook} = valve hooking factor

ΔP = differential pressure

P_{up} = upstream line pressure

A_{stem} = area of the valve stem

F_{pack} = packing drag.

The hooking mode should be used only if prior testing indicates that the valve is susceptible to this behavior.

In the previous version of IVA, the disc-to-seat coefficient of friction used in the INEL correlation was fixed by the fluid subcooling selection based on our testing. In Version 4.00, the user can override these coefficients by entering a value in the INEL Friction Factor field. This would be useful if the analyzed valve was known to have frictional characteristics based on other industry testing that differed from that observed during the INEL test programs (i.e., 0.4 for less than 70°F subcooled water and 0.5 for 70°F or greater subcooled water).

Operator Torque Considerations

The Operator Mechanical Data entry screen is used to record specific mechanical data for the operator, such as the type of closure, operator limiting thrust and torque data, and spring pack calibration data (see Figure 9). This entry screen has been highly modified to incorporate many of the Capability Assessment mode's new features.

First, the user can select a valve closure assessment based on rated torque or stall torque. "Rated Torque" would be used if the torque switch is used to control the operator, and "Stall Torque" would be used if the limit switch is controlling the operator. Operator limit data has also been included. The user can automatically display Limitorque's published operator torque and thrust ratings based on the identified operator size and the selected worm gear ratio. The user can also

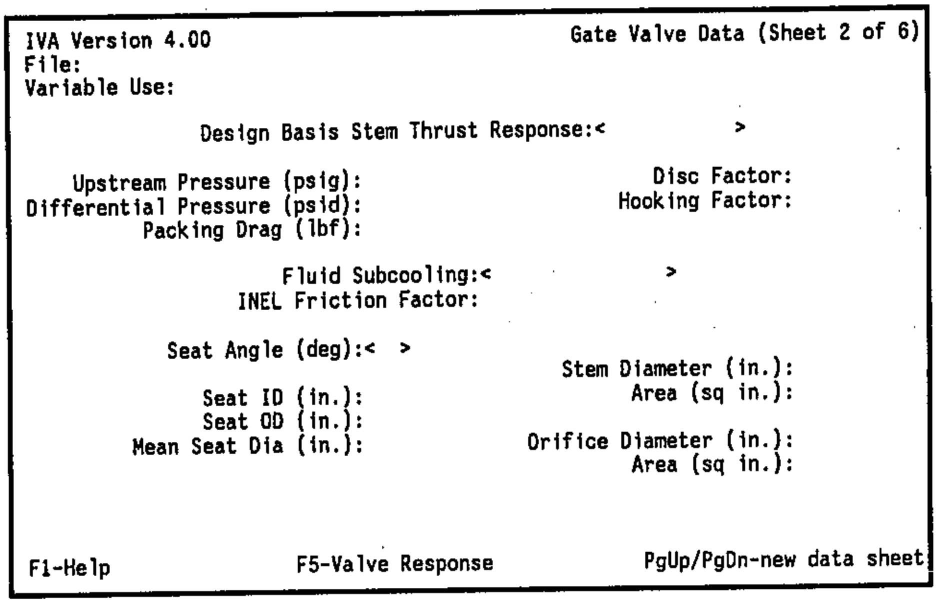

Figure 7. Gate valve data.

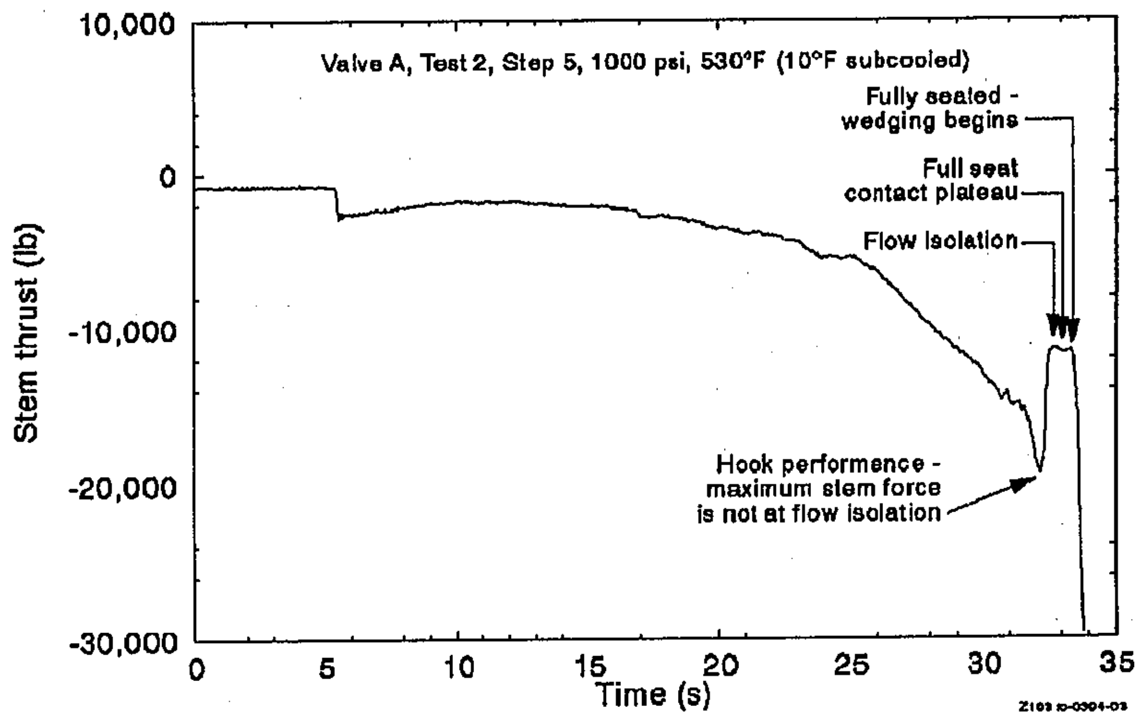

Figure 8. Valve stem thrust response showing evidence of valve hooking.

select "User Defined" to enter alternate operator ratings. Further, the graphs now include lines that correspond to the operator's torque and thrust ratings so that structural limits can be considered when analyzing operator capability margin.

Version 4.00 has added a data entry field for recording the current torque switch setting and now supports two types of spring pack calibration data. One calibration option correlates operator output torque to torque switch dial setting. The

```
IVA Version 4.00                          Operator Mechanical Data (Sheet 4 of 6)
File:
Variable Use:

                    Valve Closure On:<           >

   Overall Operator Ratio:                     Pullout Efficiency:
          Worm Gear Ratio:<    >               Running Efficiency:
Spring Pack Moment Arm (in):                     Stall Efficiency:

                  Operator Limit Data:<          >
                 Torque Limit (ft-lbf):
                    Thrust Limit (lbf):

                 Torque Switch Setpoint:

 Torque Switch Calibration Data:        Table
   <                       >            Row      Torque Switch   Operator
                                        Number      Setting        Torque
    Data Source:<          >
    Spring Pack:<         >              < >

Fi-Help                F5-Valve Response           PgUp/PgDn-new data sheet
```

Figure 9. Operator mechanical data.

other correlates spring pack force to spring pack displacement. Version 4.00 uses the latter correlation with the spring pack moment arm to estimate the operator output torque. A help screen provides estimated moment arm values, based on Limitorque operator size and the worm gear ratio, if the actual moment arm is unknown. If the user selects the "torque vs. torque switch" option, Version 4.00 will automatically enter the calibration data for the selected spring pack, as published by Limitorque (unless "user defined data" is also selected for Data Source). If the user selects "force vs. displacement," user-defined data must be entered. Twenty data sets are provided for user-defined data.

Electrical Considerations

The Operator Electric Data entry screen is used to record electrical data, such as the motor-rated torque, motor stall torque, rated voltage, motor speed, power factor, motor current, application factor, and inputs necessary to assess ac motor output degradation at elevated temperatures (see Figure 10). In May of 1993, Limitorque released results of an evaluation that determined the effect of high ambient temperature on the output torque of ac motors. This effect was determined to reduce ac motor output on a linear basis from 40 to 180°C (104 to 356°F). Version 4.00 allows the user to include this effect by entering the maximum temperature the MOV will experience, the worst-case percent current loss, and the worst-case percent torque loss for the given motor. The software uses these inputs, combined with the ac motor's rated output torque, to determine the motor output torque at elevated temperature. IVA's Help system provides easy access to tables containing the Limitorque current loss and torque loss data.

In addition to adjusting operator output for temperature effects, the software provides the following two methods for adjustment of the operator output to account for degraded voltage conditions:

- Full linear voltage
- 10% null voltage.

Availability of both methods allows the software to provide capability assessment results that are consistent with the two methods used by most industry GL 89-10 programs. If the full linear voltage method is used, the operator output is adjusted based on the ratio of the degraded voltage to the rated voltage (voltage factor). If the 10% null voltage method is selected, operator output is adjusted through use of the voltage

```
IVA Version 4.00                                Operator Electric Data (Sheet 5 of 6)
File:
Variable Use:

     Electric Motor Assessment Method:<                      >

                          Electric                                Electric
                         Motor Rated                             Motor Stall
                         Conditions                              Conditions
    Voltage (volts):                          Current (amps):
    Torque (ft-lbf):                         Torque (ft-lbf):
       Speed (RPM):                            Power Factor:
     Power Factor:

                        Application Factor:

Max. MOV Temperature (°F):                      Motor Frame:
         % Current Loss:
          % Torque Loss:

F1-Help                  F5-Valve Response               PgUp/PgDn-new data sheet
```

Figure 10. Operator electric data.

factor if the degraded voltage varies by more than 10% from the rated voltage. Otherwise, no voltage factor is used and the Application Factor is applied.

Assessment of Available Margin

The Capability Assessment Mode results screen is used to summarize the results of the design-basis calculations (see Figure 11). The left side of the screen (Capabilities section) summarizes the available motor torque, operator torque, and stem thrust at maximum, nominal, and degraded voltage conditions. The effects of voltage losses from the power source to the operator motor are also summarized. The motor torque is based on rated conditions, or stall conditions, depending on the user input. If torque switch calibration data were chosen, torque switch estimates are determined using a linear curve fit of the spring pack calibration data and estimating a maximum torque switch setting that corresponds with the available operator torque.

The right side of the screen (Requirements section) summarizes the INEL correlation (if the gate valve option has been selected) and the industry equation estimate of the required stem thrust, required operator torque, and required motor torque. If torque switch calibration data have been entered, then the minimum torque switch setting is also provided.

The Requirements section now provides the user with thrust and torque margin calculations based on the difference between the design-basis requirements and the operator's capabilities under degraded voltage conditions, or as limited by the torque switch setting. Thrust margins are provided for each thrust equation used, and a torque margin is also provided. These calculations allow the user to quickly assess whether the valve has adequate margin to allow setup of the torque switch. Further, if a negative margin exists, or if any of the Stem Thrust or Operator Torque values in either section exceed the operator thrust and/or torque limits, these values are highlighted on the screen to draw the attention of the user.

Capability Assessment Mode

File:

CAPABILITIES Closure On:			REQUIREMENTS		Current Operator Settings Torque Switch: Output Torque: Output Thrust:
Available At Max. Voltage	Available At Nom. Voltage	Available At Min. Voltage	INEL Estimate f = .000	Industry Estimate = .000	
					Stem Thrust Margin
					Operator Torque Margin
					Motor Torque
					Voltage @ MOV Source Voltage
	Reference	Maximum	Minimum	Minimum	Torque Switch Setting

Press any key to continue

Figure 11. Capability assessment mode results.

The Capability Assessment's graphs have been modified to include lines that correspond to the operator's torque and thrust ratings. In some cases, the operator's structural limits may be more restrictive than the operator's output capability under degraded voltage conditions. Therefore, the inclusion of the structural limits can provide a more complete visual representation of the available capability margin.

For the graph showing required and available thrust versus stem factor (Figure 12), the horizontal lines identify the thrust required to close the valve, as estimated using the industry equation and the INEL correlation, and the thrust limit of the operator. The vertical line identifies the estimated nominal stem factor for the operator. The four curved lines identify the thrust available from the operator under nominal and degraded voltage conditions, including the effects of voltage losses in the cables from the power source to the MOV, on the torque limit of the operator, and on the current torque switch setting. The difference between either required thrust curve and the lowest available thrust curve is identified as margin.

CONCLUSIONS

Version 4.00 of the IVA software has been improved to meet a need. We have discussed the enhancements to the evaluation of MOV testing and the determination of design-basis thrust requirements. These enhancements include the use of dynamic test data to estimate MOV performance parameters (e.g., INEL friction factor, disc factor, hooking factor, and stem/stem-nut coefficient of friction) to allow comparison of actual MOV performance to engineering assumptions. We have also discussed how the software estimates the design-basis stem/stem-nut coefficient of friction using INEL's Fold Line method. And, finally, we discussed how versatility has been added by including evaluations for different valve types, adding information regarding the effects of high ambient temperature on ac motor output, and including the capability to choose the method used to determine motor terminal degraded voltage.

REFERENCES

Steele, R. Jr., J. C. Watkins, K. G. DeWall, and M. J. Russell, 1992, *Motor-Operated Valve Research Update*, NUREG/CR-5720, EGG-2643.

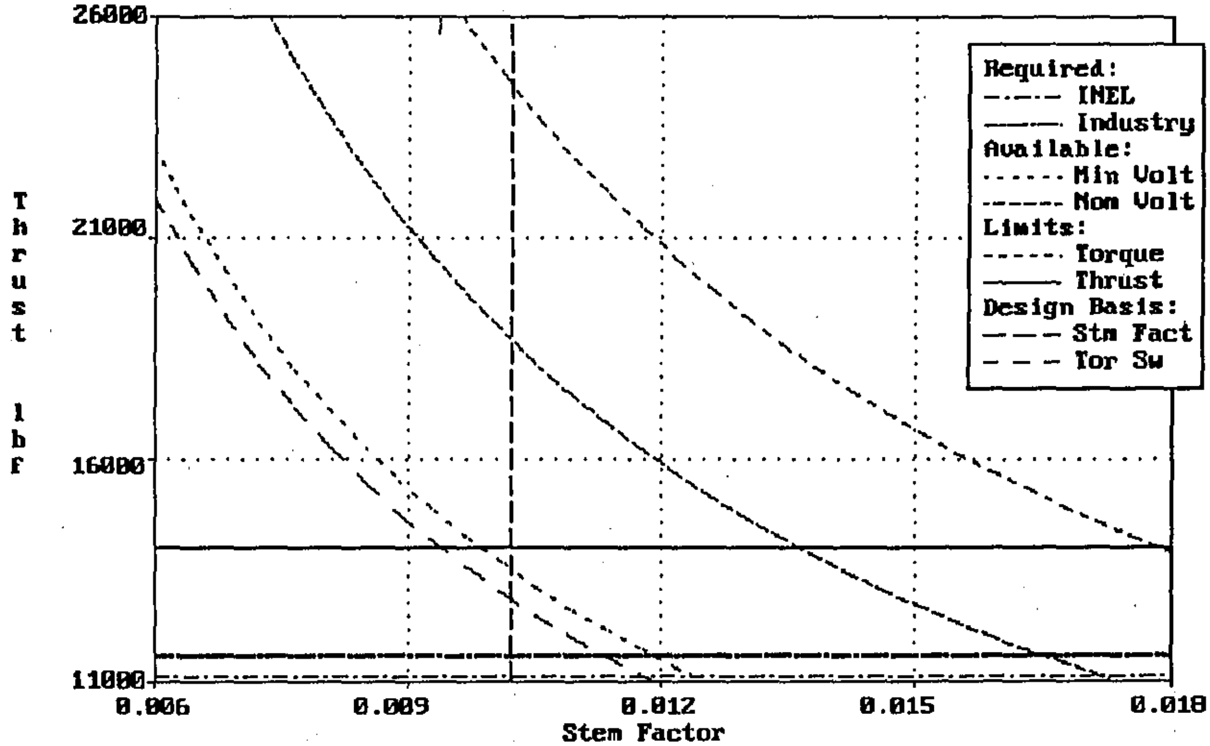

Figure 12. Required and available thrust versus stem factor.

Session 1B
Inservice Testing General Session

Session Chair
John Zudans, FP&L Chair
Florida Power & Light Co.

Ten-Year Rollover of San Onofre Inservice Testing Program for Pumps and Valves to OM-6 and OM-10

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ABSTRACT

The Pump and Valve Inservice Testing (IST) Program Sat San Onofre, Units 2 and 3, was updated for the second 120-month interval from August 1993 to April 1994. The U.S. Nuclear Regulatory Commission (USNRC) approved the OM-6 and OM-10 Codes in mid-1992. The project for the rollover to these new Codes included several elements: (a) a review of the differences between IWV/IWP and OM-6/OM-10, (b) a comprehensive audit of the IST Program scope for valves, (c) creation of the program and supporting basis documents, the Relief Requests, and implementing procedures, (d) interdivisional coordination, (e) submittal to the USNRC, and (f) training.

Subsections IWV and IWP have been used and essentially unchanged for over a decade. The new Code (Parts 1, 6, and 10 called OM-1, OM-6, and OM-10) includes several significant changes from the old Code.

Our group identified these differences and drafted revised and reorganized Inservice Testing (IST) Program documents. We also considered USNRC Generic Letter 89-04 (GL 89-04), "Guidance on Developing Acceptable Inservice Testing Programs," and NUREG-1482, *Guidelines for Inservice Testing at Nuclear Power Plants*, while revising the program.

There were six pump relief requests and 13 valve relief requests in the program for the first 10-year interval. For the revised program we needed only one pump relief request (and no valve relief requests).

Converting to the 1989 edition of the ASME Code did not require changes to the technical specifications. We revised our Updated Final Safety Analysis Report (UFSAR) to reflect the IST Program for the second 10-year interval. UFSAR changes were minor, consisting of updated references to the Code edition and 10 CFR 50.55a(f), "Inservice Testing Requirements."

A REVIEW OF THE DIFFERENCES BETWEEN THE CODE USED FOR THE FIRST 120 MONTHS AND OM-6 AND OM-10

The Pump and Valve Inservice Testing (IST) Program Sat San Onofre, Units 2 and 3 (SONGS), was updated for the second 120-month interval from August 1993 to April 1994.[a] The U.S. Nuclear Regulatory Commission (USNRC)

a. The implementation of the new 10-year revision was extended from August 1993 to April 1994. IWA-2400(c), Inspection Intervals, permits this to provide additional time for implementation.

approved the OM-6 and OM-10 Codes in mid-1992. The project for the rollover to these new Codes included several elements: (a) a review of the differences between IWV/IWP and OM-6/OM-10, (b) a comprehensive audit of the IST Program scope for valves, (c) creation of the program and supporting basis documents, the Relief Requests, and implementing procedures, (d) interdivisional coordination, (e) submittal to the USNRC, and (f) training.

Subsections IWV and IWP has been used and essentially unchanged for over a decade. The new Code (Parts 1, 6, and 10 called OM-1, OM-6, and OM-10) includes several significant changes from the old Code.

Our effort to find out the changes needed in our program was two-pronged:

- First, we systematically compared and documented the differences between the new code requirements of OM-1, OM-6, and OM-10 against our existing program. It helped us learn the new Code. Nothing can replace a good working knowledge of the Code during the development of the program documents and implementing procedures.

- Second, an engineering consulting firm was engaged to compare the new and old codes and recommend program changes. Their work resulted in an important validation of our own review results. The consultants had the advantage of looking at the new and old requirements with "new eyes." They could see things that those more familiar with the Code might overlook. They had a different perspective, however, in that they mentioned but did not emphasize differences that we later realized were very significant during implementation in the plant.

The two most significant changes in converting from IWV/IWP to OM-6/OM-10 were

- The new definition of the reference ranges for valves
- Revised hydraulic acceptance criteria and vibration measurement requirements for pumps.

REFERENCE RANGES FOR VALVES

OM-10 requires use of a "reference range" to evaluate valve stroke times. This range replaced the IWV-required evaluation of each stroke time measurement with the previous stroke time.

We made significant changes in the Operations Division implementing procedures and Control Room processes to adapt to the new valve IST Program. The Operations Procedures Group had to develop a new method of coping with the evaluation of valves that were operating outside the newly defined reference range for valve stroke time.

Before OM-10, Section XI, Subsection IWV, IWV-3413(c) stated the following:

> "If an increase in stroke time of 25% or more from the previous test for valves with stroke times greater than 10 s or 50% or more for valves with stroke times less than or equal to 10 s is observed, test frequency shall be increased to once each month until corrective action is taken, at which time the original test frequency shall be resumed. . ."

Our new procedures provided a process to identify a "reference stroke time" for each power-operated valve in the IST Program. OM-10, Paragraph 4.2.1.8, Stroke Time Acceptance Criteria, establishes requirements to compare stroke test results with pre-established reference values, similar in concept to the evaluation of results from pump testing.

Our initial approach to identification of reference stroke times was difficult. OM-10 stroke time acceptance criteria are provided in Exhibit 1. We attempted to identify the reference stroke times using a process implied by the Code of reviewing historical test results. Not all records of postmaintenance stroke testing were labeled as

such and stroke tests following design modifications were not commonly flagged in our records. We reviewed several thousand maintenance orders to find the postmaintenance tests.

Our search for reference stroke times included a review of the following:

- Trend reports showing stroke time and maintenance history for each power-operated valve.
- Maintenance orders (MOs) showing work done. Note that a MO to repaint the electrical connector box does not qualify as maintenance requiring a new reference stroke time. Consequently, we had to read each MO to separate out maintenance that had influenced only valve stroke time.
- Construction Work Orders (CWOs) showing modifications and actual work done and post-installation tests and "preservice" of new valves or actuators.
- Records from the Motor-Operated Valve (MOV) Group of gear set changes resulting from the GL 89-10 Program.
- Nonconformance Reports, which often resulted in valve maintenance and a new reference stroke time.

In the end, we used advice obtained during the public meeting on NUREG-1482. We picked the "best" reference stroke times based on a review of trending graphs showing the "usual" stroke times and compared these times with postmaintenance times for reasonableness before use.

The (±) ranges in the Code (see Exhibit 1) resulted in what we called a "reference range" for each valve stroking at greater than 2 s. Thus, a valve stroke time could be

- OK, within the reference ranges of ± 15%, for example (operable)
- Outside the reference range, but within the operability limit (operable)
- Outside the operability limit (inoperable).

From the reference values we calculated reference ranges using OM-10 rules to allow evaluation in the Control Room. Thus, we could comply with the Code requirement to immediately restroke a valve outside the reference range following the initial stroke timing.

Documenting the reference stroke times and ranges in implementing procedures was judged to be impractical. We might never be able to keep up with reference stroke time changes in a timely manner. At any time, reference values on IST valves can change depending on the frequency of maintenance. The conventional review and approval processes for issuance of changes to station procedures could take too much time.

In addition, simultaneous with the Code changes, we changed the nature of the IST Program in areas not directly governed by the Code. We converted our computer program for recording and trending test results and scheduling quarterly tests to the HOST platform. The new application, Inservice Testing System (ISTS), was developed under the San Onofre Quality Assurance Program. In addition to its other functions, the ISTS provided the means we needed to control the documentation of reference stroke times.

IMPLEMENTATION IN THE PLANT

We developed a system to make reference data available to the operators using the ISTS application. Features of this HOST-based IST application at SONGS are outlined in Exhibit 2. The system enhanced the speed with which accurate data could be provided. Updating reference stroke times is now immediate, as is determining the reference range.

Exhibit 3 provides an outline of our process for evaluating valve stroke test results. Our process for inservice testing and updating reference stroke times operates as follows:

- The operator obtains a copy of the procedure needed for the valve tests planned. Using ISTS, the operator also prints a fresh data sheet, summarizing reference and

acceptance times, for the valve(s) he or she plans to test.

- A tailboard is conducted and the tests are completed. The rules in the program, summarized in Exhibits 1 and 3, are implemented in Control Room procedures. A valve under test is immediately restroked if the stroke time falls outside the reference range. We document the evaluation using our nonconformance reporting (NCR) program. The NCR program provides for (a) identification of the problem, (b) evaluation of operability, and (c) documentation of the Code-required record of corrective action.

- After approval by the Control Room Supervisor, stroke test data is keyed into the ISTS.

- Postmaintenance and postdesign change stroke tests are flagged for evaluation by Station Engineering. From their review, Station Engineering then decides if a new reference stroke time should be established.

- New reference tests are flagged. Approval of the record on ISTS results in automatic calculation of the reference range using preestablished rules based on the valve actuator type. For a summary of these rules, see Exhibit 1.

Operators print out reference range and acceptance limit data in the Control Room as needed without the use of manual logs or hard copy records. This is one of the reasons the ISTS had to be a quality assured system. However, we maintain hard copy records as a backup in case of a computer outage.

Concerning a note on the establishment of appropriate reference stroke times, the first stroke time following "lube and tune" maintenance on, for example, an air-operated valve (AOV) is often faster than that obtained during routine quarterly testing. As a result, this initial postmaintenance stroke may not represent the best reference. Subsequent stroke times may be outside the reference range. To avoid this, we evaluate the postmaintenance time and may not use it for the new reference in all cases. A time closer to the "normal" stroke time may be a better choice.

IMPLEMENTATION OF THE NEW PUMP IST PROGRAM USING OM-6

Code Acceptance Criteria

The OM-6 pump test acceptance criteria are summarized in Exhibit 4. To implement OM-6, we revised the hydraulic and vibration measurement and acceptance rules in the program and pump test procedures. Station engineers received training in the use of these new limits.

Mechanical vibration limits in OM-6 differ from Subsection IWP. However, this did not require a change in our program. Two years earlier, we had set up OM-6 limits for vibration measurement in terms of velocity in addition to the IWP requirements for unfiltered displacement measurements.

Additional Acceptance Criteria

We made another change to the pump IST Program not directly governed by the Code. In addition to Code limits, we evaluate pump performance against the safety analysis requirements. The program established criteria for minimum pump flow and developed head.

The technical specifications provided limits on certain pumps (high and low pressure safety injection and reactor charging pumps, for example), and we had always verified pump performance against these limits. Several years ago, however, the USNRC asked us how we verified that all IST pumps retained a level of performance necessary to meet the assumptions in the safety analysis. The USNRC concern could be paraphrased as follows:

> The Code permits a 10% degradation in the hydraulic performance of a pump below the reference point. Below this limit, the pump is declared inoperable. What assurance can

the licensee offer to show this decline in performance does not exceed the degradation allowed when considering the assumed pump performance in the safety analysis? Has the licensee made a comparison of the safety analysis against the pump reference conditions used in the IST Program testing?

We developed calculations as part of our design-basis effort. These calculations established minimum safety analysis performance. Our new Pump IST implementing procedures incorporate a requirement for the engineer to compare pump test results with safety analysis limits.

ADOPTION OF OM-1

The scope of OM-1 (summarized in Exhibit 5) includes thermal reliefs in addition to safety and relief valves required for safe plant shutdown. The scope of IWV included the following:

"...certain Class 1, 2 and 3 valves (and their actuating and position indicating systems)...which 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."

However, the scope of OM-10 and OM-1 included more:

" . . . pressure-relief devices covered are those for protecting systems or portions of systems which perform a required function in shutting down a reactor to the cold shutdown condition, in maintaining the cold shutdown condition, or in mitigating the consequences of an accident."

A full awareness of this difference in scope arose at the NUREG-1462[b] public comment meeting in early February 1994. We are still reviewing the ramifications of this interpretation of OM-1. Scope could increase for relief valve testing by as much as 261 valves. The original scope under IWV included only the main steam and pressurizer safety valves and the LTOP relief valves (a total of 44 valves).

The steam system cognizant engineer implemented OM-1 for the main steam safety valves during the refueling in late 1993. We made an effort to comply with both the old PTC 25.3 and the new OM-1. Significant changes included the following:

- Seat leakage measurement was now required as part of the safety/relief IST. We use the simmering test technique for the set point test (relief valve test) and have established acceptance criteria for seat leakage based on tail pipe temperature.

  Note: Several techniques are suggested in OM-1 for seat leakage measurement other than traditional flow/volume measurement of the leakage. We developed a calculated tail pipe temperature limit based on nominal valve performance.

- Vendors are used for safety/relief valve calibration at times. Purchase order changes were necessary to provide for safety/relief valve calibration and repair by outside organizations.

- Repetitive RMOs and maintenance procedures were required to incorporate OM-1. System cognizant engineers were responsible for these for the valves within the scope of the IST Program.

- Code Class 2 and 3 safety/relief valves are tested on a 120-month schedule (versus a 60-month schedule). See Paragraph 1.3.4.1(b) in OM-1.

A COMPREHENSIVE AUDIT OF THE IST PROGRAM SCOPE FOR VALVES

The USNRC conducted an inspection of the check valve program at SONGS in April 1992.

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b. *Guidelines for Inservice Testing at Nuclear Power Plants*, Draft for comment, November 1993.

Inspectors spot-checked the scope of our testing using GL 89-04[c] as a guideline. Certain valves were not undergoing all of the inservice testing required. We had originally concluded that our program was satisfactory upon receipt of our Safety Evaluation Report (SER). Nevertheless, we had conducted an informal review or our compliance with the generic letter, and had made some program changes in 1991. Our program SER did not require this review and the resulting program changes. We had not completed a formal audit of the program scope at the time of the 1992 USNRC inspection.

As a result of the USNRC findings, the Nuclear Engineering Design Organization (NEDO) reevaluated the IST Program using the scope recommendations of GL 89-04 as a starting point. To provide direction during the scope audit, we developed and furnished a guideline to the assigned engineers. The audit objectives were to provide complete answers to two major questions:

- Of all safety-related valves in the plants, which valves should be in the IST Program?
- For IST valves, did the program impose the appropriate testing required in the Code?

NEDO reviewed the safety function of each valve. NEDO then set the Code-required testing requirements. NEDO recorded the results of this analysis in a document (Constance et al., 1992).

We added approximately 800 new valve tests to the IST Program after the audit. We concluded that all added tests resulted from our interpretation of GL 89-04. That is, none of the missing valve tests represented deficiencies in the application of the ASME Code. Accordingly, no noncompliance with the technical specifications, Code, or regulations was involved.

This massive change in valve program scope (40% increase) posed a further complication in our transition to the second 10-year interval. It imposed a large burden on the Operations Procedures Group to add these new tests to their implementing procedures when they were occupied fully with the adoption of the new code requirements into their implementing procedures.

DRAFTING THE PROGRAM AND IMPLEMENTING PROCEDURES

The Station Operations and Technical Divisions drafted the program revisions and implementing procedures simultaneously because of the tight schedule. We drafted the new program using (a) our review of the Code differences, (b) GL 89-04, and (c) NUREG-1482. We provided an early draft to the Operations Procedures Group so they could get started on their implementing procedures.

In addition to Code changes, we modified implementing procedures to enhance the speed of data entry. Test results were grouped at the end of each procedure. The original procedures sometimes required that the data entry clerks look through the entire completed test procedure page by page to locate the valve testing results.

Pump IST implementing procedures belong to the Station Technical Division (Engineering). They required only minor changes compared with the valve procedures. We had done most of the OM-6 required changes in the past 2 years because of two reasons:

c. U.S. Nuclear Regulatory Commission, "Guidance on Developing Acceptable Inservice Testing Programs," GL 89-04, April 3, 1989.

1. Vision Program[d]

2. PRR on bearing temperature in which alternate testing was to measure velocity vibration in addition to displacement for each pump.

CREATING SUPPORTING BASIS DOCUMENTS AND THE RELIEF REQUESTS

The Design Basis Documentation (DBD) Group wrote the IST topical design basis document. This controlled design document collected in one place all of the references and requirements for the IST Program. It included correspondence and other documented commitments associated with the IST Program from the time of the original plant startup.

Relief requests practically disappeared with the application of the OM Code for IST. Pump and valve relief requests for the old program are summarized in Exhibits 6 and 7. New program relief requests are summarized in Exhibit 9. These exhibits show the nature and extent of the relief requests before and after the application of the second 120-month IST Program interval.

INTERDIVISIONAL COORDINATION

Key individuals were assigned from the Station engineering staff, Operations, the Maintenance Division, the Design Basis Group, Design Engineering, Nuclear Licensing, and the Procedures groups to formulate the new program and procedures. We used the computer based Regulatory Commitment Tracking System (RCTS) to track our progress. Effective interdivisional coordination really paid off in the following areas:

- During the refueling outages High Impact Teams (HITs) were formed for the coordination of the reactor refueling testing. Each HIT consisted of representatives of the divisions involved in the outage work and associated inservice tests. ISTS came into use for the identification of testing required prior to return to service.

- Increases in scope resulted, as discussed with the comprehensive audit of the program content. Station Operations and Technical Divisions worked closely together in informal teams to write and issue the implementing procedures for the new tests added because of the audit.

- Starting in January 1994, following the refueling outages for Units 2 and 3, the SONGS IST team applied our full resources to completing the program documents and the USNRC submittals. Station Technical Division helped the Operations Equipment Control Group in setting up the data entry process. The Technical Division IST Group also helped to develop the process for using the valve stroke reference printout. The SONGS organization for IST is outlined in Exhibit 9.

- Software development required well-structured procedures and plans. We carefully analyzed our processes and set them down in writing, which was necessary to permit coding in the software. Nuclear Information Services (NIS) wrote a formal specification for the revisions to the ISTS to capture our processes and describe how they would be coded. NIS wrote a detailed file conversion specification for changing existing files into the formats needed by the revised ISTS.

d. The Vision Program is the software supporting the use of the Microlog hand-held computers used in the plant for pump IST. The Microlog guides the engineer conducting the pump IST, supplemented by the hard copy test procedure. The Microlog also provides a means of recording the test data and results for direct downloading to the Vision program when the engineer returns to the office. A directly connected vibration probe reads a full spectrum into the Microlog during the testing. The Vision program then analyzes this vibration data to provide the Code-required vibration measurements (Herrera, 1992).

Station Operations and Technical Division representatives met frequently with NIS to plan and develop the revised ISTS. We developed the process flows and assigned responsibility for various aspects of the IST activities. For example, to implement the new idea of reference ranges for valves; we developed the organization as shown in Table 1.

NRC SUBMITTAL

The two NRC submittals were based on information from NEDO and the Station Technical Division. Our Nuclear Licensing Group prepared the submittals and included elements defined by GL 89-04, draft NUREG-1482, and 10 CFR 50.55a(f).[e]

In our preliminary submittal in mid-1993 we informed the USNRC that we were extending the program cut-over date because of the two refueling outages and forwarded the schedule, the draft program documents, and our single pump relief request (although no valve relief requests were necessary). Our final submittal was sent shortly after April 1994 and included the final program procedures.

e. "Inservice Testing Requirements."

TRAINING

This section briefly describes the training given to implement the IST. There was no time block available in the tightly scheduled off-shift training program for operation for instruction on the new IST Program. Information on the new requirements were put on the required reading list, discussed in shift turnover meetings, and included in on-the-job-training.

For the Station Technical Division, the shift to OM-6 did not affect the pump testing very much because we had carried out many of the requirements long before (discussed previously). Information on the requirements was required reading, and meetings were held to discuss the basics and the practical aspects of the IST Program changes. In addition, Cognizant Engineers and supervisors reviewed procedure changes for technical accuracy associated with their components.

Table 1. Organization and responsibilities for implementing reference ranges.

Organization	Responsibility
Station Technical Division	Identification of reference tests
NIS	Development of software to provide a list of ranges for each valve so Operations would have this essential data to evaluate their test results.
Operations	Development of procedures for the Control Room staff to employ the reference range lists for valve to be tested. Operations established the format of the reference range listing as they • Established the work flow • Determined the data needed in the Control Room • Developed the procedures/process for valve testing and Results evaluation.

REFERENCES

ASME/ANSI OM-1987, "Operation and Maintenance of Nuclear Power Plants"

OM-1, "Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices"

OM-6, "Inservice Testing of Pumps in Light-Water Reactor Power Plants"

OM-10, "Inservice Testing of Valves in Light-Water Reactor Power Plants."

ASME, 1989, Boiler and Pressure Vessel Code, Section XI, "Rules for Inservice Inspection of Nuclear Power Plant Components," July 1, 1989, IWA-2400, Inspection Intervals.

Code of Federal Regulations, Title 10, Part 50, 10 CFR 50.55a(f), "Inservice Testing Requirements."

Constance, D. P., W. L. Justice, and R. S. Smith, 1992, "Inservice Testing Basis Program," *Presented at the 1992 ASME/NRC Symposium on Pump and Valve Testing, Washington, D.C., July 21–23.*

Herrera, V. M.,1992, "Computer-Based Pump Inservice Test Program at San Onofre Nuclear Generating Station," *1992 ASME/NRC Symposium on Pump and Valve Testing, Washington, D.C., July 21–23.*

ACKNOWLEDGMENTS

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Exhibits

EXHIBIT 1 (Page 1 of 2)

OM-10 STROKE TIME ACCEPTANCE CRITERIA

Code requirement	Comments
4.2.1.8 Stroke Time Acceptance Criteria. Test Results shall be compared to the initial reference values or reference values established in accordance with Paragraphs 3.4 and 3.5.	The referenced paragraphs are 3.4 "Effect of Valve or Actuator Replacement, Repair and Maintenance on Reference Values," and 3.5 "To Establish an Additional Set of Reference Values"
(a) Electric-motor-operated valves with reference stroke times greater than 10 s shall exhibit no more than ±15% change in stroke time when compared to the reference value.	Two features stand out: 1. Both an upper and lower limit are specified 2. The 15% limit used is tighter than the IWV limit (25%) formerly in use for MOVs.
(b) Other power-operated valves with reference stroke times greater than 10 s shall exhibit no more than ±25% change in stroke time when compared to the reference value. (c) Electric-motor-operated valves with reference stroke times less than or equal to 10 s shall exhibit no more than a ±25% or ±1 s change in stroke time, whichever is greater, when compared to the reference value.	The valve actuator type determines the breadth of the reference range.
(d) Other power-operated valves with reference stroke times less than or equal to 10 s shall exhibit no more than a ±50% change in stroke time when compared to the reference value.	This requirement is the same as the IWV rule, except for the existence of a lower limit.
(e) Valves that stroke in less than 2 s may be exempted from (c) and (d) above. In such cases the maximum limiting stroke time shall be 2 s.	Consistent with the USNRC Generic Letter 89-04 definition of rapid-acting valves.

EXHIBIT 1 (Page 2 of 2)

EXAMPLE OF OM-10 STROKE TIME ACCEPTANCE CRITERIA

Example

Valve: 2HV9336, Isolation Valve - SDCS to LPSI Suction
Open Stroke Limit: 80 seconds
Open Stroke Reference: 43.5 seconds (Test Dated: 1/15/94)
Actuator Type: Motor Operator (MOV) - Limits are therefore +/- 15 %

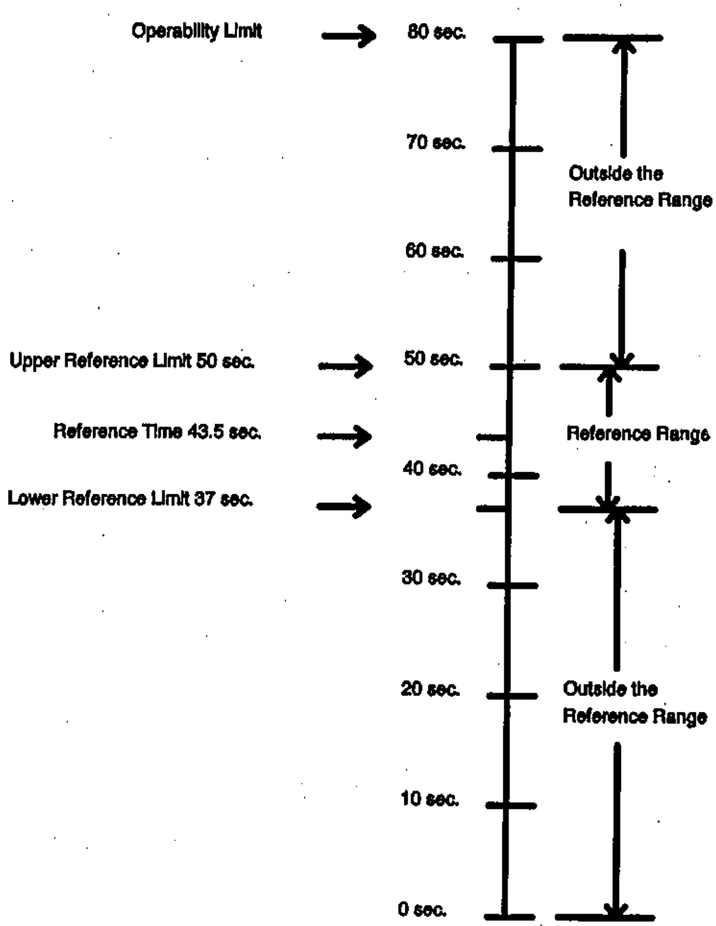


EXHIBIT 2

FEATURES OF THE HOST IST APPLICATION AT SONGS FOR THE "INSERVICE TESTING SYSTEM"

Inputs:

1. REQUIREMENTS: Valve testing requirements from the IST Program (test types, frequencies, acceptance criteria)

2. RESULTS: Valve test dates, data and pass/fail for each test.

3. TEST REASON: Identification of postmaintenance tests (PMT) versus routine quarterlies (R).

4. COMPONENT CHARACTERIZATION: Valve actuator type.

5. IDENTIFICATION OF REFERENCE TESTS: This is Engineering input.

Outputs (Displays and Reports):

1. TEST SCHEDULING: Date last tested, test due dates, drop dead (inoperable) dates, schedules.

2. RESULTS AND TRENDS: Test result summaries for each valve. This includes a stroke time history.

3. REFERENCE RANGES AND OPERABILITY LIMITS: Provided for use to the test organization to simplify prompt assessment of test stroke timing results. Also includes the reference value and the date of the reference test.

EXHIBIT 3

OM-10 PROGRAM ELEMENTS FOR EVALUATING VALVE STROKE TEST RESULTS

Code requirement	SONGS Implementation Program
4.2.1.8 Stroke Time Acceptance Criteria. Test results shall be compared to the initial reference values or reference values established in accordance with paras. 3.4 [Effect of Valve or Actuator Replacement, Repair, and Maintenance on Reference Values] and 3.5 [To Establish an Additional Set of Reference Values].	• For each group of power-operated valves to be tested we provide a printed sheet to the operator showing the reference range and operability limits of each valve. Evaluation takes place immediately following the stroke timing. • The sheet is printed from the data in the HOST-based IST application. Reference stroke time and reference range are updated each time there is a new reference test.
4.2.1.9 Corrective Action. (a) If a valve fails to exhibit the required change of obturator position or exceeds the limiting values of full-stroke time [see para. 4.2.1.4(a)], the valve shall be immediately declared inoperable.	Not new. No major program change was needed for this action.
4.2.1.9 Corrective Action. (b) Valves with measured stroke times which do not meet the acceptance criteria of para. 4.2.1.8 [quoted above] shall be immediately retested or declared inoperable. . . .	The requirement for an immediate retest is new. This not only demands immediate evaluation of the first stroke, but also procedures to provide for documentation of retest results and evaluation.
. . . .If the valve is retested and the second set of data also does not meet the acceptance criteria, the data shall be analyzed within 96 hr to verify that the new stroke time represents acceptable valve operation, or the valve shall be declared inoperable. If the second set of data meets the acceptance criteria, the cause of the initial deviation shall be analyzed and the results documented in the record of tests.	This involves Operations and Station Engineering Staff. • If it is outside the Reference Range, the valve is immediately restroked. • Station Engineering is alerted and the data are provided for evaluation. • A nonconformance report (NCR) is required whatever the outcome of the second stroke test. This provides for a properly documented engineering evaluation and record of corrective action.

EXHIBIT 4

OM-6 PUMP TEST RESULT ACCEPTANCE CRITERIA

Mechanical (Vibration) Parameters

Pump type	Pump speed	Test parameter	Acceptable range	Alert range	Required action range
Centrifugal and vertical line shaft [Note (2)]	<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 [Note (2)]	$\geq$ 600 rpm	V_v or V_d	$\leq 2.5\ V_r$	>2.5 V_r to 6 V_r or >0.325 in./s	>6 V_r or >0.70 in./s
Reciprocating	...	V_d or V_v	$\leq 2.5\ V_r$	>2.5 V_r to 6 V_r	>6 V_r

Notes:
1. Vibration parameter per Table 2 of OM-6. V_r is vibration reference value in the selected units.
2. Refer to Figure 1 of OM-6 to establish displacement limits for pumps with speeds $\geq$ 600 rpm or velocity limits for pumps with speeds <600 rpm.

Hydraulic Parameters

Test parameter	Acceptable range	Alert range		Required action range	
		Low	High	Low	High
P (positive Displacement Pumps)	0.93 to 1.10 P_r	0.90 to < 0.93 P_r	...	<0.90 P_r	>1.10 P_r
ΔP (vertical line shaft pumps)	0.95 to 1.10 ΔP_r	0.93 to <0.95 ΔP_r	...	<0.93 ΔP_r	>1.10 ΔP_r
Q (positive displacement vertical line shaft pumps)	0.95 to 1.10 Q_r	0.93 to <0.95 Q_r	...	<0.93 Q_r	>1.10 Q_r
ΔP (centrifugal pumps)	0.90 to 1.10 ΔP_r	...	...	<.90 ΔP_r	>1.10 ΔP_r
Q (centrifugal pumps)	0.90 to 1.10 Q_r	...	...	<0.90 Q_r	>1.10 Q_r

General Note: The subscript r denotes reference value.

EXHIBIT 5

SCOPE STATEMENT COMPARISON ... IWV VERSUS OM-1 AND OM-10

IWV (1977 Ed.)	Part 10 1. Introduction 1.1 Scope	Part 1 1. Introduction 1.1 Scope
IWV-1100 Scope	...	...
This Subsection provides the rules and requirements for inservice testing to verify operational readiness of certain Class 1, 2 and 3 valves (and their actuating and position indicating systems) in light-water cooled nuclear power plants, which 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.	The active and passive valves covered are those which are required to perform a specific function in shutting down a reactor to the cold shutdown condition, in maintaining the cold shutdown condition, or in mitigating the consequences of an accident. The pressure-relief devices covered are those for protecting systems or portions of systems which perform a required function in shutting down a reactor to the cold shutdown condition, in maintaining the cold shutdown condition, or in mitigating the consequences of an accident.	This Part provides general requirements for periodic performance testing and monitoring of pressure relief devices utilized in nuclear power plant systems (included in Section III ...) which are required to perform a specific function in shutting down a reactor or in mitigating the consequences of an accident. ...

EXHIBIT 6

EXAMPLES OF VALVE RELIEF REQUESTS UNDER THE OLD CODE

VRR under the old code	How it was handled under the new code
Emergency containment sump outlet valves hand stroked versus stroking with flow	Hand Stroking allowed under the new Code (Para 4.3.2.4(c)). Alternate Refuelings allowed under GL 89-04, although new Code says "Every Refueling".
Grouping valves for seat leakage testing.	Allowed under the new code (Para 4.2.2.3 - "valve or valve combination").
Certain valves cannot be stroked except during reactor refueling.	Reactor Refueling interval allowed. (OM-10, Para 4.2.1.2.)
Some valves exceed the 25 % increase allowed from the previous test even though they are in good working condition. This is usually due to a relatively fast stroke following a PM then, at the next test, the valve returns to its normal stroke time and enters the reduced interval condition. A reference stroke time was proposed as an alternate to the previous stroke time.	Reference stroke times replace the old rules in OM-10, Para. 4.2.1.8 and 4.2.1.9.

EXHIBIT 7

PUMP RELIEF REQUESTS UNDER THE OLD CODE

PRR under the old code	How it was handled under the new code
Bearing temperature measurement provides no useful information. Substitute vibration velocity measurement.	Bearing Temp. Measurement is not required under the new code. No PRR needed. NOTE: Since we were already measuring velocity vibration, adopting the new Code did not required a change in this area.
Submerged vertical Line Shaft Pump Bearing Vibration and Temperature Measurement.	Not needed in OM-10, See Para 5.1.
Instrument ranges and accuracies not in accordance with Code requirements.	Still needed under OM-6, Para 4.6.1. Renumbered, updated and reissued for second ten-year interval.
Inlet pressure measurement for DGFO and SWCS pumps is not direct.	Not needed for second ten-year interval. OM-6 adopted terminology "Determined" versus "Measured" in Paragraph 5.2(b).

EXHIBIT 8

VRRs AND PRRs STILL REQUIRED UNDER THE NEW CODE

Valves:	None.	
Pumps:	Instrument range and accuracy.	We still had some permanently installed instruments that did not meet the range and accuracy of the new Code. We showed how the combination of the range and accuracy did meet the code, however.

EXHIBIT 9

ORGANIZATION AT SONGS FOR INSERVICE TESTING

Station			Design/support (Irvine Office)	
Operations	Maintenance	Station technical	Design	Nuclear licensing
Valve testing	Relief/safety valves	Program Pump IST	Scope of testing	Submittal to USNRC
Performance	Performance			
Scheduling	Scheduling			
Implementing procedures	Implementing procedures			

Preliminary Assessment of Valve IST Effectiveness[a]

Edward Grove, Adele DiBiasio, and Joseph Carbonaro
Brookhaven National Laboratory

ABSTRACT

A preliminary review of inservice testing effectiveness for Code Class 1, 2, and 3 valves at nuclear power plants was performed. These requirements are specified by the American Society of Mechanical Engineers Boiler and Pressure Vessel Code, Section XI, and the Operations and Maintenance Standard. The Institute of Nuclear Power Operations Nuclear Plant Reliability Data System (NPRDS) database was used to provide failure reports for these components for 1988 to 1992. This time period coincides with the issuance of Generic Letter 89-04, which resulted in a more consistent application of the requirements by the licensees.

For this time period, 8,593 valve failures were identified. From the review of the NPRDS database, the primary failure causes and failure modes for motor-operated valves (MOV), air-operated valves (AOV), and check valves (CV) were identified. Solenoid-operated valves were not reviewed in this study. Plant testing programs were effective in identifying approximately 60% of the CV failures, 46% of the AOV failures, and 44% of the MOV failures.

INTRODUCTION

Brookhaven National Laboratory (BNL), under contract with the U.S. Nuclear Regulatory Commission (USNRC), is conducting a review of inservice testing (IST) effectiveness at nuclear power plants (Grove et al., 1993). The results obtained from this program will be used to identify and recommend potential changes to the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code and revisions to improve existing IST programs in identifying pump and valve degradations before failure. The results obtained from this program to date for valves will be presented in this paper. The results obtained pertaining to the IST effectiveness for pumps are presented in another paper at this Symposium.

ASME Code, Section XI, Subsection IWV; Operations and Maintenance (OM) Standard, Part 10; and the OM Code, Subsection ISTC, define the specific IST requirements to assess the operational readiness of Safety Class 1, 2, and 3 valves that perform a specific function in shutting a reactor down, maintaining it in the cold shutdown condition, or mitigating the consequences of an accident. Valves used for operating convenience, system control, or maintenance are excluded from these requirements. The codes establish test frequency (typically quarterly), parameters to be measured and evaluated, acceptance criteria, corrective actions, and records requirements.

In the event a valve cannot be exercised quarterly, testing may be deferred to cold shutdowns or refueling outages. Typically, quarterly testing may be deferred if (a) valve failure during testing could result in a loss of system function, (b) valve failure would result in degraded containment integrity, (c) testing would subject the subsystem to operating parameters that exceed design limits, (d) testing would result in personnel hazards, or (e) plant trip may result. To ensure operability of

a. Work performed under the auspices of the U.S. Nuclear Regulatory Commission.

check valves, these tests generally are performed under full-flow conditions. If this is not feasible, a partial flow test should be performed quarterly to demonstrate operability, and the full flow test deferred. The codes do not specify the system conditions for exercising power-operated valves. However, Generic Letter 89-10 (GL 89-10) (USNRC, 1989a), addresses design-basis testing of motor-operated valves (MOVs).

From 1988 to 1992, the Nuclear Plant Reliability Data System (NPRDS) maintained by the Institute of Nuclear Power Operations (INPO) was used to record all failures for Safety Class 1, 2, and 3 failures (8,593 total). As defined in NPRDS, the Safety Class of components is determined using American National Standards Institute/American Nuclear Society (ANSI/ANS) Standard 51.1 pressurized water reactors (PWRs) and 52.1 boiling water reactors (BWRs). (A legend for NPRDS codes is located at the end of this paper.) Although the scope of the new OM standards includes all safety-related valves, the current regulations require IST in accordance with only Section XI for ASME Code Class 1, 2, and 3 valves. NPRDS provides the safety class rather than the code class. Therefore, the scope of this study was limited to safety Class 1, 2, and 3 components, although this is not expected to have a significant effect on the results. Many plants include all safety-related valves in their IST program.

This 5-year time frame was chosen to coincide with the issuance of GL 89-04 (USNRC, 1989b), which provided specific, detailed instructions to nuclear utilities regarding IST. This guidance has resulted in a more consistent application of the requirements by the nuclear utilities. The USNRC has also provided additional guidance regarding compliance with IST program requirements in Draft NUREG-1482, *Guidelines For Inservice Testing At Nuclear Power Plants*.[b]

The NPRDS contains specific information (failure mode, symptom, cause, system and plant effect, and detection method) for all component failures submitted by the nuclear utilities. Failures are reported to the NPRDS when degradation of a component, part, or associated device has occurred and one function of the component has been lost or degraded, such that the performance criterion for at least one of the component's functions is not met. The performance criterion may be based upon limits specified in technical specifications, ASME Code, or system design specifications. The NPRDS database also encompasses the events reported on licensee event reports (LERs), as specified by 10 CFR 50.73.

Figure 1 shows the distribution of these failures by valve type. Table 1 identifies the systems most affected by failures of air-operated valves (AOVs,) motor-operated valves (MOVs), and check valves (CVs). This paper will review the primary failure cause, failure mode, and IST effectiveness in detecting these occurrences.

b. P. Campbell, USNRC, November 1993.

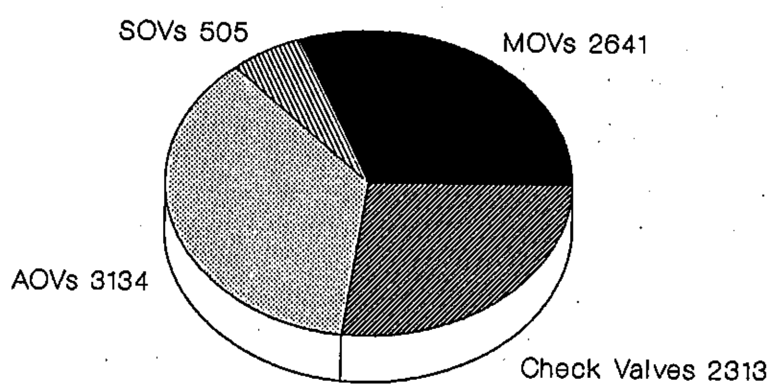


Figure 1. Valve failures by type (1988–1992).

Table 1. Systems affected by valve failures.

Valve type	Systems	Percent
Motor-operated	Residual heat removal/low pressure injection	18
	High-pressure injection	15
	Service water	10
	Main steam	8
	Chemical and volume control	8
	Feedwater	5
Air-operated	Chemical and volume control	18
	Containment isolation	17
	Main steam	15
	Feedwater	10
	Service water	6
	Residual heat removal/low-pressure injection	5
Check	Feedwater	22
	Residual heat removal/low-pressure injection	13
	Containment isolation	10
	Main steam	9
	High pressure injection	8
	Emergency diesel generators	8
	Service water	7

Because only a small fraction of the failures affected solenoid-operated valves (SOVs), they are not included in this study. Recent reports have reviewed common SOV failures (Ornstein, 1991), and the OM Committee recently approved a scope statement for the development of a new standard guide for these valves.

CHECK VALVES

The IST Codes specify that CVs should be exercised nominally every 3 months to ensure that the disc moves freely to fulfill its safety function (either open, closed, or both). Typically, a full-stroke exercise, with flow, is required to demonstrate this. If a CV cannot be tested quarterly because of a particular operational or design constraint, valves may be partial-stroke tested during cold shutdowns, and full-stroked during refueling outages. Numerous publications and information notices (INs) have been published alerting licensees to specific CV failure and program requirements [e.g., IN 82-08 (USNRC, 1982) Bulletin 83-03 (USNRC, 1983b), IN 83-54 (USNRC, 1983a), IN 88-70 (USNRC, 1988), SOER 86-3 (INPO, 1986), and GL 89-04 (USNRC, 1989b)]. Based upon the number and safety significance of check valve failures, the USNRC staff is conducting inspections to determine the effectiveness of licensees' check valve maintenance programs (Temporary Inspection 2515/110) (USNRC, 1991). Licensees generally employ a periodic disassembly and inspection program on all safety-related check valves to satisfy the SOER 86-3 concerns.

As required by the codes, adequate obturator movement may be observed through direct system indications, such as a position indicating device, or by indirect means, such as changes in system pressure, flow rate, level, temperature, leak testing, or other positive means. For valves where such observations are not possible, the USNRC has allowed valve disassembly and inspection (GL 89-04, Position 2). Although disassembly can provide useful information on the valve condition, it is a risky maintenance technique, and is not considered by the USNRC staff to be an equivalent alternative to other

testing techniques. Many nonintrusive inspection techniques have been developed for CVs, and are commercially available for use to demonstrate proper valve actuation. These methods include acoustics, eddy current, ultrasound, and radiography.

A leak test is required for CVs that perform a safety function in the closed position, and for which leakage is limited to a specific maximum amount (i.e., Code category A valves).

Figure 2 shows the main failure symptoms associated with the reported CV failures. Coolant leakage (external and internal) and demand faults were the most frequent failure symptoms. External leakage does not generally affect operability; however, internal leakage may prevent the CV from performing an isolation function, and may result in component damage and decreased plant safety. Check valve demand faults may represent both an operability and a plant safety problem. Certain standby systems (e.g., auxiliary feedwater, high-pressure injection) are isolated during normal plant operation by check valves. Upon demand (i.e., system pressure changes), these CVs must function (open or close) to permit the standby systems to operate in a timely fashion.

Figure 3 shows the failure detection method responsible for detecting each reported failure symptom. IST alone was responsible for detecting only 7% of the reported failure symptoms. However, this may be misleading, because nuclear plants may perform IST in conjunction with technical specification testing and other special testing. An overview of the reported failure narratives indicated that many of the tests classified as either a special or surveillance test were, in fact, IST. Therefore, for this study, all plant testing was combined and, for CVs, detected 61% of the failures. Operational abnormalities and routine observations were also effective in detecting these failure symptoms. A common example seen in many instances was plant personnel detecting internal CV leakage through elevated pipe temperatures, by touch, downstream of a closed check valve.

Check valve failure modes are shown in Figure 4, and detection methods for each mode in Figure 5. Internal leakage and failure to close accounted for 60% of the reported failures. Combined plant testing detected 75% of the internal seat leakages and 67% of the failure to close events. These failure modes are critical, particularly for systems that must function to mitigate the effects of an accident. Both failure modes are indicative of degraded or worn internals and seating surfaces.

This review of CV failure symptoms and modes showed that plant testing was successful in detecting the majority of failures, but that a significant fraction remained undetected and were found through operational abnormalities and routine observations. While failures will occur that are impossible to detect through testing, a review indicated that many that occurred should have been detected. Worn valve internals, corroded

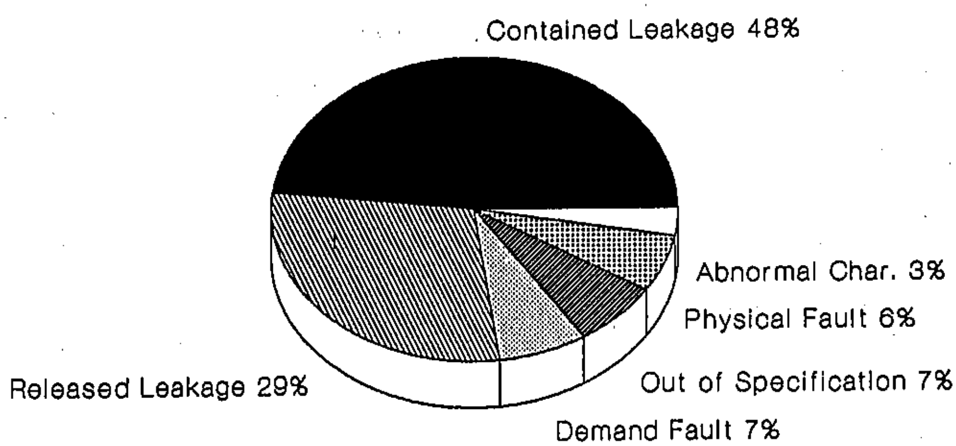
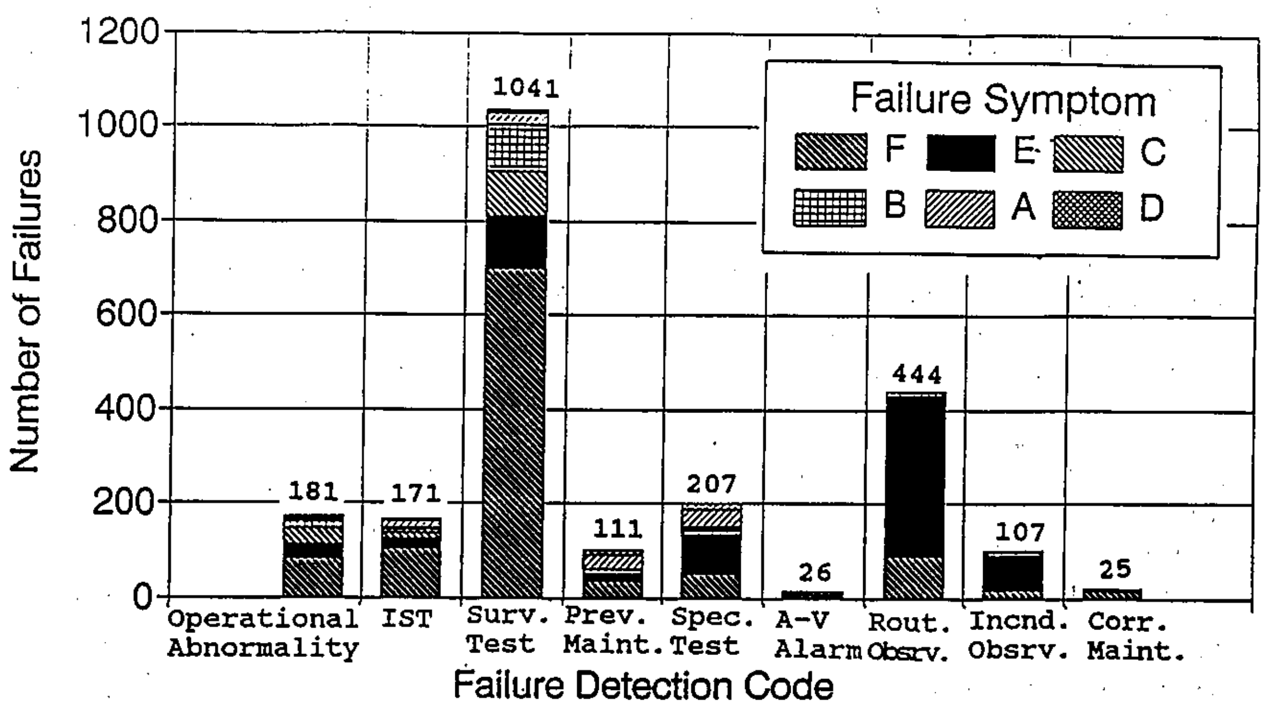


Figure 2. Check valve failure symptoms.



NPRDS Code	FAILURE SYSTEM
F	Contained Leakage
E	Released Leakage
C	Demand Fault
B	Out of Specification
A	Physical Fault
D	Abnormal Characteristic

Figure 3. Check valve failure symptoms versus detection method.

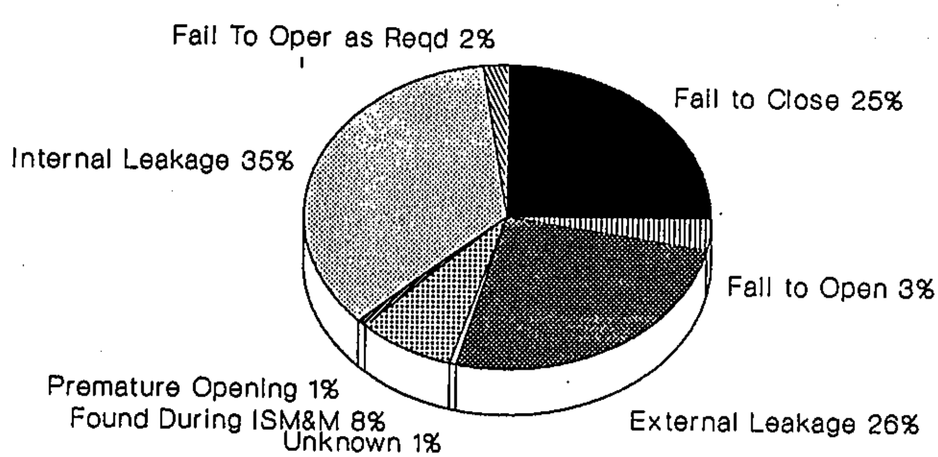
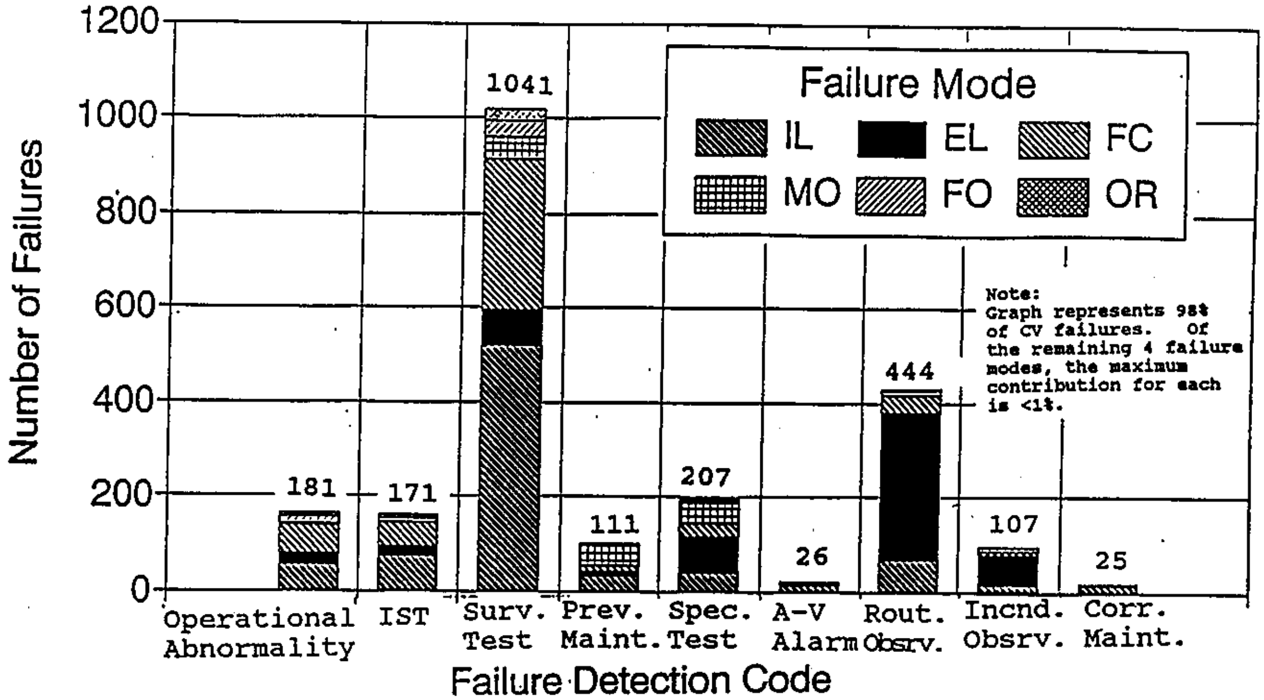


Figure 4. Check valve failure modes.



NPRDS Code	FAILURE MODE
IL	Internal Leakage
EL	External Leakage
FC	Failure to Close
MO	Found During Testing, Surveillance, Inspection, or Maintenance
FO	Failure to Open
OR	Failure to Operate as Required

Figure 5. Check valve failure modes versus detection method.

seating surfaces, and missing parts were examples of the types of degradation frequently missed by testing. A combined program consisting of flow testing, periodic disassembly on a sampling basis, and nonintrusive techniques should be able to detect many of these occurrences.

AIR-OPERATED VALVES

Of the four valve types evaluated, AOVs had the highest occurrence of reported failures. Power-operated valves are tested (i.e., fail-safe tested and stroke tested) quarterly to ensure proper functioning. This testing, similar to CVs, may be deferred to cold shutdowns or refueling outages, depending on the function and location of the valve. The primary way of monitoring power-operated valves is stroke time testing.

The primary failure symptom for the failure occurrences (Figure 6) was coolant leakage, external leakage (40%) and internal leakage (34%). Typically, external leakage does not affect valve operability, and is more a concern for maintenance and as low as reasonably achievable (ALARA). The six failure causes associated with AOV failures were wear (abnormal and normal), mechanical damage or binding, out of mechanical adjustment, aging or cyclic fatigue, dirt intrusion, and improper previous maintenance or installation. Each of these failure causes was potentially detectable through testing, including those caused

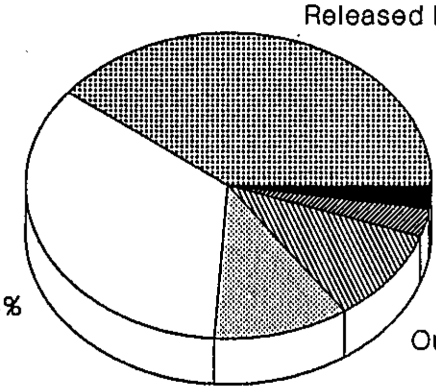


Figure 6. AOV failure symptoms.

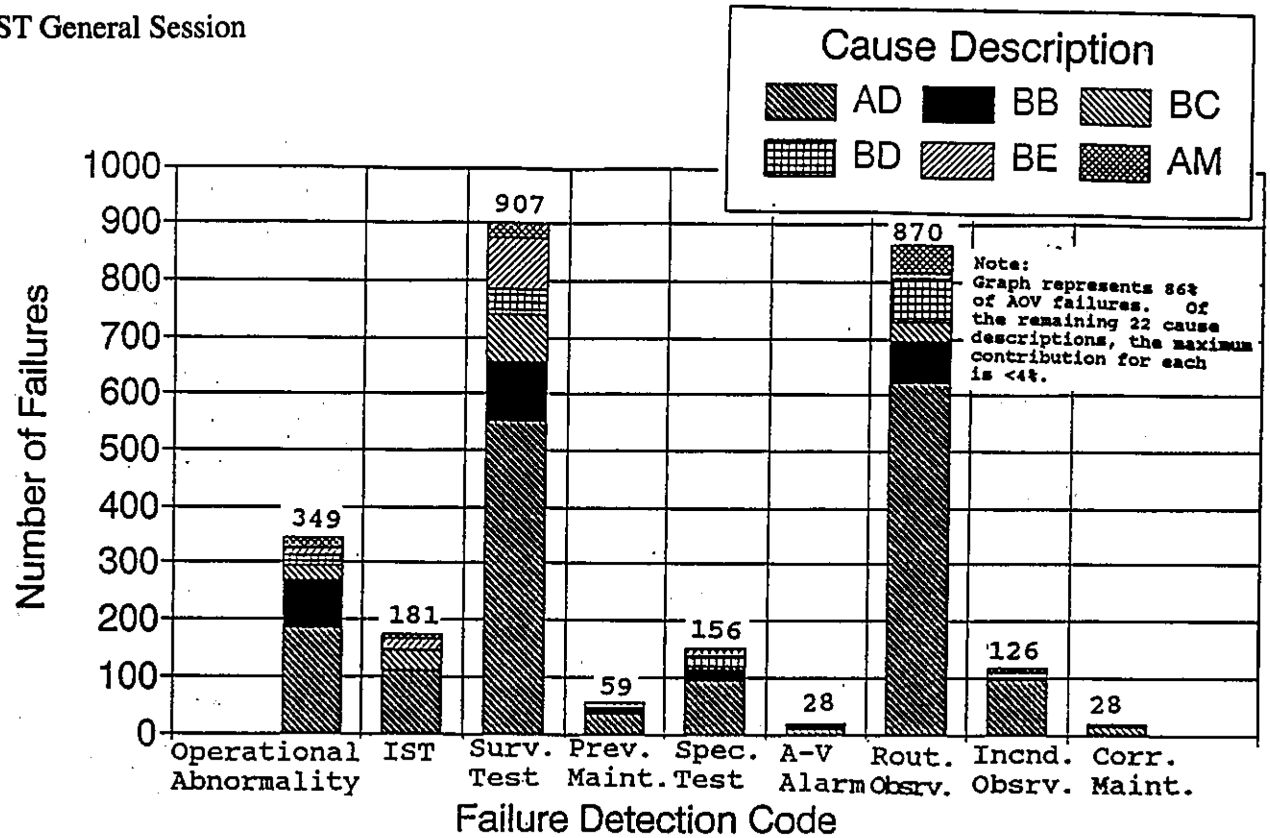
by improper maintenance. The codes specifically requires testing of repaired, replaced, or maintained components before placing them into service. Of these events, valve testing programs were successful in detecting the greatest number of failures (47%) (Figure 7), followed by routine observations.

AOV failures that resulted from aging and cyclic fatigue were of particular interest, because this type of time dependant degradation should be detected through valve testing and trending. A review of the failure narratives indicated that over 50% of these occurrences resulted from packing failures. The remaining were found to be caused by the wear of various internal components. This type of degradation, particularly for rapid-acting AOVs, may not be detectable through stroke time testing only. Examples of worn valve stems were reported, which may have been detected through stroke time testing.

Specific AOV failure modes, occurrence percentage, and percentage detected by plant testing are shown in Table 2. Coolant leakage modes were the dominant failure modes for AOVs, with plant testing successfully detecting 67% of the failures. Of those not detected by testing, routine observations accounted for 16%, and operational abnormalities accounted for 11%. Figure 8 shows the actual method of detection for each failure mode. As with the failure causes, the majority of these failure modes could potentially have been discovered during IST. Many pitted, gouged, and scored valve seats could have been detected through changes in operating parameters, leak rates, or through valve disassembly. While valve disassembly is not always recommended because it may result in valve damage, it appears that it may be useful in identifying valve seat wear and corrosion, particularly for the AOVs that operate in steam or highly corrosive mediums.

The ASME has recently introduced a new guide entitled "Preservice and Periodic Performance Testing of Pneumatically Operated Valve Assemblies in Light-Water Reactor Power Plants" (OM-19),[c] which addresses AOV dynamic testing (at system pressure or flow). The motivating force behind this guide is to detect operating failures at design conditions that may not have been detected by stroke time testing (similar to GL 89-10). In light of the success similar testing has had with MOVs, it is anticipated that this may increase the effectiveness of the current testing programs for selected risk-significant valves. As discussed, this testing and trending of additional valve parameters may detect changes indicative of valve internal degradation and wear. This ASME document provides guidance only, and does not impose mandatory requirements. Therefore, its use may be limited, given the current financial restraints on utilities.

c. ASME, 1993.

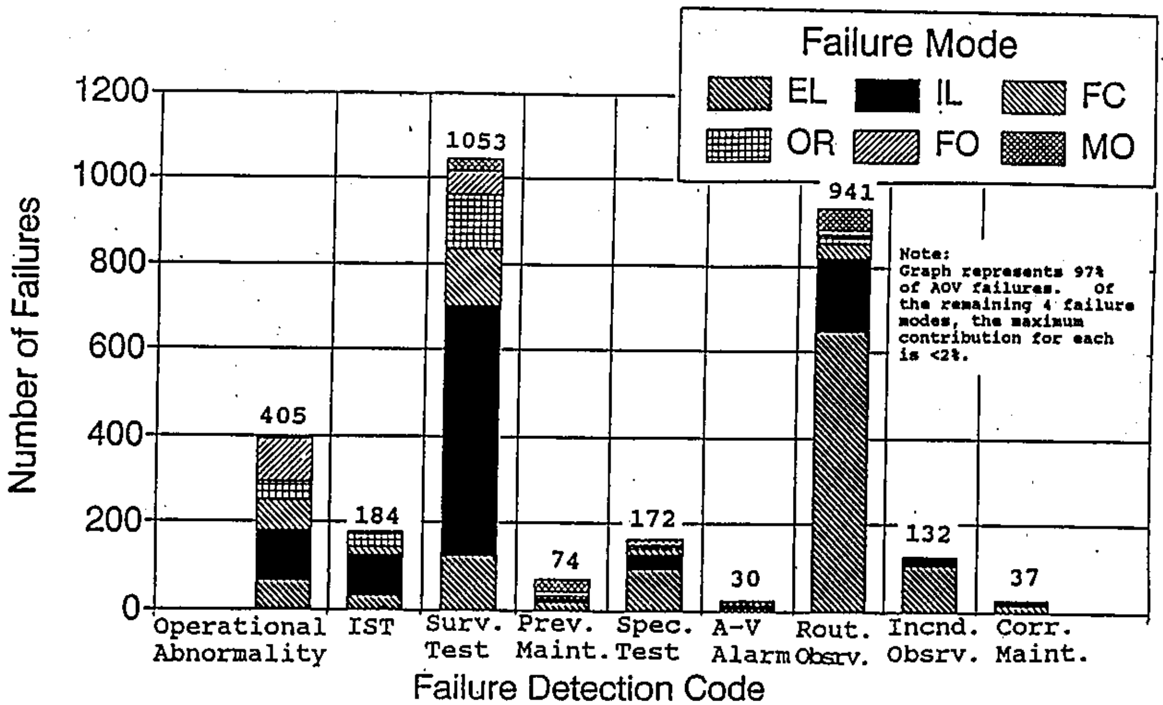


NPRDS Code	CAUSE DESCRIPTION
AD	Normal/Abnormal Wear
BB	Mechanical Damage/Binding
BC	Out-of-Adjustment
BD	Aging/Cyclic Fatigue
BE	Dirty
AM	Previous Repair/Installation Status

Figure 7. AOV failure causes versus detection method.

Table 2. Percentage of occurrence and detection of each AOV failure mode.

Failure mode	Occurrence (%)	Detected by test (%)
External leakage (EL)	35	23
Internal leakage (IL)	34	68
Failure to close (FC)	9	57
Failure to operate as required (OR)	8	67
Failure to open (FO)	6	33
Found during ISM&M (MO)	5	35
Premature opening (PO)	2	82
Failure to remain open (RO)	<1	17
Failure to operate properly (OP)	<1	53
Unknown	<1	46



NPRDS Code	FAILURE MODE
EL	External Leakage
IL	Internal Leakage
FC	Failure to Close
OR	Failure to Operate as Required
FO	Failure to Open
MO	Found During Testing, Surveillance, Inspection, or Maintenance

Figure 8. AOV failure mode versus detection method.

MOTOR-OPERATED VALVES

Following AOVs, MOVs were the most frequently reported failed (31%). A primary method of detecting MOV degradation, similar to AOVs, is stroke time testing. Recognizing that this method alone may not be sufficient for detecting MOV degradation, valve diagnostic methods [e.g., Motor-Operated Valve Analyses and Test System (MOVATS), Valve Operational Test and Evaluation System (VOTES)] were developed. These techniques monitor many of the design parameters associated with MOVs (e.g., valve stem position, torque, and thrust; spring pack displacement; control switch actuation time; motor current, voltage, and power; actuator vibration; and actuator output torque). Generic Letter 89-10 requires periodic testing of some MOVs.

Over 80% of the reported failures (Figure 9) resulted in external or internal seat leakage. External leakage is generally not detected through plant testing; however, the remaining 48% of the failures potentially are (contained leakage, demand faults, out of specification parameters, physical faults, and abnormal characteristics). The primary cause of these failures, as shown in Table 3, was mechanical degradation resulting from normal or abnormal wear, mechanical damage or binding, and aging cyclic fatigue.

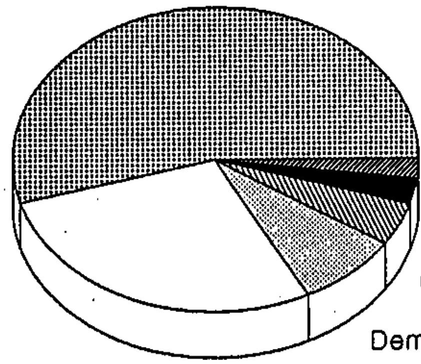


Figure 9. MOV failure symptoms.

Plant testing was effective in detecting 42% of these occurrences. Again, routine observations proved to be a very effective means of detecting MOV degradation because many of these occurrences resulted in external leakage. The frequent occurrence of failures attributed to degraded and worn internals for the AOVs and CVs were not reported for MOVs. While the exact cause for this difference was not apparent, it may be attributable to the use of the MOV monitoring and testing methods discussed previously.

Table 4 provides an assessment of the effectiveness of plant testing in detecting the various MOV failure modes. Plant testing was effective (69%) in detecting instances of internal leakage. This particular failure mode is important since many MOVs serve as containment and reactor coolant system isolation valves, and internal leakage could result in system and component degradation.

Changes in stroke time may not indicate valve degradation in all cases. Alternating current (ac) MOVs may not see significant changes in stroke time before failure because of the constant speed of the actuating device. As discussed in GL 89-10, stroke time measurement for direct current (dc) MOVs is useful in detecting certain failure modes; however, the concerns regarding adequate torque switch setting and correct actuator and motor configuration remain (Shuster, 1989). Figure 10 shows the method of detection for both ac and dc operators. Eighty-eight percent of the failures affected ac operators. Testing (IST, surveillance, and special) detected 46% of the ac failures, as compared with 57% of the dc MOV failures. Following surveillance testing, operational abnormalities detected the most failures (29%).

CONCLUSIONS AND RECOMMENDATIONS

This evaluation of operational data for check, air, and motor-operated valves demonstrated that a significant number of failures have occurred. The majority of the failures resulted in contained and released leakages. External leakage presents more of a maintenance and an ALARA problem, while internal leakage may affect valve function, particularly if the valve is used to isolate a system or component.

The effectiveness of combined plant testing programs in detecting valve failures is summarized in Table 3. The current Code requirements which potentially could have detected these failures are listed in Table 5.

The main causes for the valve failures were mechanical wear and aging. Plant testing may be capable of detecting a significant portion of these occurrences, particularly if the results are trended in a manner sensitive enough to detect the resulting operating characteristics (i.e., actuation time). The Code should consider the benefits that may be realized from trending specific parameters. For

Table 3. Effectiveness of plant testing in detecting valve failures.

Failure cause	Check valves			AOV			MOV		
	Total failures	Detected by IST	Not detected by IST	Total failures	Detected by IST	Not detected by IST	Total failures	Detected by IST	Not detected by IST
Normal/abnormal wear	1,096	642 (59%)	454 (41%)	1,721	756 (44%)	965 (56%)	1,608	667 (41%)	941 (59%)
Previous repair	83	35 (42%)	48 (58%)	136	49 (36%)	87 (64%)	104	39 (38%)	65 (62%)
Mechanical damage/binding	202	120 (59%)	82 (41%)	325	133 (41%)	192 (59%)	215	114 (53%)	101 (47%)
Out-of-mechanical adjustment	84	52 (62%)	32 (38%)	189	124 (66%)	65 (34%)	68	32 (47%)	36 (53%)
Aging/cyclic fatigue	165	76 (46%)	89 (54%)	184	68 (37%)	116 (63%)	202	82 (40%)	120 (60%)
Dirty	239	196 (82%)	43 (18%)	149	114 (77%)	35 (23%)	76	58 (76%)	18 (24%)
Totals	1,869	1,121 (60%)	748 (40%)	2,704	1,244 (46%)	1,460 (54%)	2,273	992 (44%)	1,281 (56%)

Table 4. MOV failure modes and IST detection rate.

Failure mode	Occurrence (%)	Detected by IST (%)
Failure to close (FC)	7	61
Failure to open (FO)	5	52
External leakage (EL)	52	28
Internal leakage (IL)	27	69
Failure to operate as required (OR)	2	67
Premature opening (PO)	0.15	0
Failure to remain open (RO)	0.15	75
Found during ISM&M (MO)	5	44
Failure to operate properly (OP)	0.3	75
Unable to classify (UA)	0.3	40

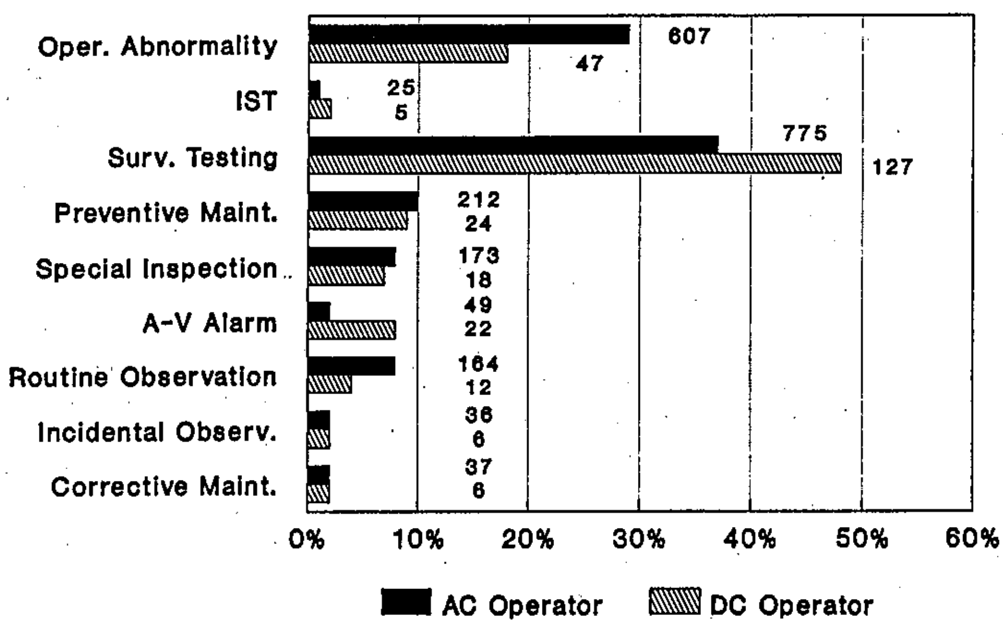


Figure 10. MOV failure detection method by valve operator type.

example, MOV stroke time monitoring for ac operators may determine operability, but not valve degradation. Evaluation of the trends could identify degradation and permit maintenance to be performed before component operating capability is affected.

Check valves may be disassembled and inspected if they cannot be functionally tested through operation. Even though there are inherent risks with such a procedure, the recurring examples of degraded internals indicate that testing is not detecting all of these conditions before failure. Certain valves, particularly those in steam and corrosive environments, may benefit from periodic sampling inspections. The USNRC has recommended for the next generation of nuclear power plants (i.e., the Advanced Light Water Reactor), that a disassembly and inspection program be developed on all safety and relief valves to detect unacceptable degradation that cannot be detected through the use of advanced nonintrusive techniques (Taylor, 1990).

Table 5. Valve failure causes detectable by current IST requirements.

	Potentially detectable by IST		
Failure cause	Leak rate testing (required for Category A valves only)	Exercising and stroke time measurement (for AOVs and MOVs)	Postmaintenance testing
Normal/abnormal wear	Y	Y	NA
Mechanical damage/binding	Y	Y	NA
Out-of-mechanical adjustment	Y	Y	NA
Aging/cyclic fatigue	N	Y	NA
Dirty	Y	Y	NA
Previous repair/installation	Y	Y	Y

While common failure causes and modes were seen for all the valve types, the frequency of occurrence was less for MOVs. This may be the result of the MOV diagnostic techniques that are used as a result of GL 89-10. Through the monitoring of several valve operating parameters, in addition to actuation time, slight variations may be detectable, which results in maintenance before valve failure.

The increased use of nonintrusive valve inspection techniques may increase the efficiency of testing. For check valves, techniques such as acoustic testing, ultrasonic inspection, internal permanent magnetic, and external ac and dc magnetic techniques have demonstrated success in detecting certain valve degradations. Variations in motor current signatures may also be indicative of valve degradation.

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NPRDS Code Legend

The NPRDS database utilizes a unique coding system. A PC-based computerized spreadsheet was used to sort the data presented in this paper using the NPRDS codes. The codes are presented here to serve as an easy reference.

Failure Symptom Code	
A	Physical Fault
B	Out of Specification
C	Demand Fault
D	Abnormal Characteristic
E	Released Leakage
F	Contained Leakage

Pump Failure Modes	
FS	Failure to Start
FR	Failure to Run
MO	Found During Testing, Surveillance, Inspection, or Maintenance

Failure Detection Code	
A	Operational Abnormality
B	Inservice Testing
C	Surveillance Testing
D	Preventive Maintenance
E	Special Inspection
F	Audiovisual Alarm
H	Routine Observation
J	Incidental Observation
K	Corrective Maintenance

Valve Failure Modes	
FC	Failure to Close
FO	Failure to Open
EL	External Leakage
IL	Internal Leakage
OR	Failure to Operate as Required
PO	Premature Opening
RO	Failure to Remain Open
OP	Failure to Operate Properly
MO	Found During Testing, Surveillance, Inspection, or Maintenance

Failure Cause Description Code	
AD	Normal/Abnormal Wear
AG	Abnormal Stress
BB	Mechanical Damage/Binding
BC	Out-of-Adjustment
BD	Aging/Cyclic Fatigue
BE	Dirty
BF	Blocked/Obstructed
BG	Corrosion
AM	Previous Repair/Installation Status

Failure Cause Category Code	
A	Engineering/Design
B	Incorrect Procedure
C	Manufacturing Defect
D	Installation Error
E	Operating Error
F	Maintenance/Testing
H	Wearout
J	Other Devices
K	Unknown

How to Determine an IST Program Component Scope

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ABSTRACT

A clear scope statement is vital for successful inservice testing (IST) or inspection (ISI) programs. This paper discusses the need to agree on the mission and objectives for a program before defining the scope of components to be tested or inspected. The paper then points out useful source documents (codes, regulatory guides, NUREGs, and standards) for specifying the components by safety class.

INTRODUCTION

Each individual has an image of what an inservice testing (IST) program should be. The mission, goals, scope of components, and method of implementation will vary by the organization reviewing the document. Such organizations may include the following individuals:

- Utility—management, licensing, operations, quality, or engineering
- U.S. Nuclear Regulatory Commission—site inspector, regional inspector, projector manager, safety evaluation report (SER) reviewer, technical evaluation report (TER) reviewer, or legal counsels.
- American Society of Mechanical Engineers (ASME)—Member of Section XI or Operations and Maintenance (OM), Parts 1, 6, and 10, committees.

This paper is directed at the individual in the trenches who must satisfy all these organizational personnel. This individual must be flexible and dedicated to his or her missions and goals to survive the daily trials. Because no individual can possess all the knowledge required to develop an IST Program that will satisfy everyone's expectations, there must be well-defined missions and goals. The missions should state the overall purpose of the program, and goals should state specific tasks necessary to achieve that purpose. Table 1 identifies the objectives covered by this paper.

Table 1. Objectives to determine IST Program scope using safety class boundaries.

Major objectives	Minor objectives
Establish IST Program mission and goals	Identify the need for ASME Class boundaries
Formalize the method to determine IST Program scope	Provide mandatory requirements for ASME Class boundaries
	Discuss the documents that require mandatory enforcement
	Compare ASME Class to quality groups
	Supply additional source documents
	Clarify bounds of a safety-class boundary
	Develop a self-check to verify scope
	State advantages and disadvantages of incorporating as a safety class system or component

Statements of mission clarify the project for everyone. Consider the following:

- To satisfy the 10 CFR 50.55a(f) requirements and technical specification requirements
- To test safety-related pumps and valves in accordance with ASME Code, Section XI, or OM Code, Parts 1, 6, and 10.

These examples would be expected as a mission from a newly assigned IST Program lead with little prior practical experience or familiarity with an IST Program and its impact on plant operations. The IST Program lead's mission should be more specific. The following mission statements are more specific, and therefore more appropriate:

- When the control room operator turns on the pump or opens a valve, the component performs its intended function!
- When the infrequent event or equipment malfunction occurs, all the automatic functions perform as required!
- When any auditing group comes in to devour the Program, it can withstand the continuous onslaught of questions and open issues!

In order to meet the simplest of the stated missions, the IST Program lead must set attainable goals, such as the following:

- Identify the code and standards for compliance
- Determine the scope of components within the IST Program
- Define the test requirements for specific pumps and valves
- Delegate responsibilities
- Implement the IST Program with a site procedure and test instructions
- Formulate a method for maintaining and upgrading the IST Program.

The effectiveness with which a lead can develop and meet the goals will determine how well the missions can be satisfied. If each of these items is perfected, the desired pump or valve will function properly for the control room operator or automatically when called upon. Finally, the IST Program will be everything the interested parties had hoped for, even though they may not agree with portions of the program.

DETERMINING THE SCOPE OF COMPONENTS WITHIN THE IST PROGRAM

One of the best methods of determining the scope of components is by developing the plant's inservice inspection (ISI) boundaries or, to be more precise, the plant's ISI pressure testing boundaries, to include all the necessary safety-class components and systems. This method will be the same for both pressurized water reactors (PWRs) and boiling water reactors (BWRs), even though they vary in design, operation, and maintenance. To understand that they can be similar, refer to Subsections IWV and IWP of the ASME Code and OM Parts 6 and 10, which do not mention or specify either PWRs or BWRs. Only OM Part 1 distinguishes between PWRs and BWRs because of the uniqueness of Class 1 safety relief valves.

Most nuclear power plants designate their ISI boundaries as ISI safety class, ASME safety class, or safety class boundaries. Power plants do not use the ASME Code terminology of ASME Class. Newly designed plants will have their design boundaries as ASME Class. However, because design rules for a power plant do not normally match the inservice and preservice requirements for class boundary separations, the use of ISI safety classes or the others is still appropriate.

Using the plant's ISI safety class boundaries is the best method to determine the component scope because Subsections IWV and IWP of the ASME Code both identify ASME Classes as the

major criterion for the scope of components to which they apply.

The following scope statements from Subsections IWP and IWV, respectively, clearly define the affected components:

> "This Subsection provides the rules and requirements for inservice testing of **Class 1, 2, and 3** centrifugal and displacement type pumps which are installed in light-water cooled nuclear power plants and which are provided with an emergency power source. The results of these tests are to be used in assessing operational readiness of the pumps during their service life."

> "This Subsection provides the rules and requirements for inservice testing to verify operational readiness of certain **Class 1, 2, and 3** valves (and their actuating and position indicating systems) in light-water cooled nuclear power plants, which 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."

Because the repeated theme of each subsection pertains to Classes 1, 2, and 3 pumps and valves, we can clearly see that identifying the ISI safety class boundaries is very important to the IST Program development. Now the difficulty in making the scope determination starts. All nuclear power plants have different ASME Safety Class 1, 2, and 3 boundaries. If the class boundaries can be established using codes and standards, our scope determination would be partially solved. Unfortunately, only new plants are built to the ASME Code, Section III, which assists in establishing a standard set of Class 1, 2, and 3 safety class boundaries. Therefore, we must look at past boundary development documents.

Determining the proper boundaries requires reviewing historical documents to identify enforcement of ASME Code, Section XI. The creation of Section XI had a unique purpose:

> "The rules of this Section constitute requirements to maintain the nuclear power plant and to return the plant to service, following plant outages, in a safe and expeditious manner. The rules require a mandatory program of examinations, testing, and inspections to evidence adequate safety. The rules also stipulate duties of the Authorized Inspector to verify the mandatory program has been completed, permitting the plant to return to service in an expeditious manner."

With the development of Section XI and the statement of its purpose, Title 10 of the Code of Federal Regulations, Part 50, mandated the ASME Code, Section XI, requirements. 10 CFR 50 categorizes the implementation by construction permits for a boiling or pressurized water-cooled nuclear power facility.

Inservice inspection, according to 10 CFR 50.55a(g), must be performed in the following:

- Nuclear power facility whose construction permit was issued prior to January 1, 1971
- Nuclear power facility whose construction permit was issued on or after January 1, 1971, but before July 1, 1974
- Nuclear Power Facility whose construction permit was issued on or after July 1, 1974.

This section also provides the start date for all nuclear power plants to implement the ASME Code, Section XI, requirements:

- 10 CFR 50.55a(g)(5)(iii) states that "For a facility whose operating license was issued prior to March 1, 1976, the provisions of paragraph (g)(4) of this section are effective after September 1, 1976, at the start of the next one-third of a 120-month inspection interval."

As written, 10 CFR 50.55a(b) approved the 1974 Edition and addenda's through summer 1975. This edition provided the standard for

boundary classification. IWA-1100 Scope (a) defines the rules and requirements for inservice inspection of Class 1, 2, and 3 pressure-retaining components and inservice testing of pumps and valves in light-water cooled nuclear power plants.

Now the real research begins. 10 CFR 50 was used to define the reactor coolant pressure boundary (ASME Class 1). 10 CFR 50.2(v) states the following:

> "Reactor coolant pressure boundary means all those pressure-containing components of boiling and pressurized water-cooled nuclear power reactors, such as pressure vessel, piping, pumps, and valves, which are:
>
> (1) Part of the reactor coolant system, or
>
> (2) Connected to the reactor coolant system, up to and including any and all of the following:
> (i) The outermost containment isolation valve in system piping which penetrates primary reactor containment,
> (ii) The second of two valves normally closed during normal reactor operation in system piping which does not penetrate primary reactor containment,
> (iii) The reactor coolant system safety and relief valves.
>
> For nuclear power reactors of the direct cycle boiling water type, the reactor coolant system extends to and includes the outermost containment isolation valve in the main steam and feedwater piping."

10 CFR 50.55a Note 2 states the following:

> "Components which are connected to the reactor coolant system and are part of the reactor coolant pressure boundary defined in 50.2(v) need not meet these requirements, provided:
>
> (a) In the event of postulated failure of the component during normal reactor operation, the reactor can be shut down and cooled down in an orderly manner, assuming make-up is provided by the reactor coolant make-up system only
> (b) The component is or can be isolated from the reactor coolant system by two valves (both closed, both open, or one closed and the other open). Each open valve must be capable of automatic actuation and, assuming the other valve is open, its closure time must be such that, in the event of postulated failure of the component during normal reactor operation, each valve remains operable and the reactor can be shut down and cooled down in an orderly manner, assuming make-up is provided by the reactor coolant make-up system only.

Using this material identifies Class 1 components uniquely, thus allowing the selection of Class 1 pumps and valves. The selection of Classes 2 and 3 pumps and valves becomes a little harder. Regulatory Guide 1.26 attempts to clarify which components are categorized as Classes 2 and 3. The regulatory guide uses Quality Groups A, B, C, and D. The following is a correlation between ASME Classes 1, 2, and 3 and the Quality Groups:

ASME Class	Quality Groups
1	A
2	B
3	C
Nonsafety	D

Quality groups are for safety-related components containing radioactive material, water, or steam. Systems not covered under quality groups are instrument and service air, diesel engine and its generators and auxiliary support systems, diesel fuel, emergency and normal ventilation, fuel handling, and radioactive waste management systems. However, these systems should be designed, fabricated, erected, and tested to quality standards commensurate with the safety function to be performed. Attachment A contains portions of Regulatory Guide 1.26, and provides the information required to select systems and components to be specified as ASME Safety Classes 1, 2, 3, and nonsafety class.

Another source document used before Regulatory Guide 1.26 was the initial publication of ASME Code, Section XI, 1970 Edition, which had colored drawings with suggested boundaries for Class 1, 2, and 3 components in BWRs and PWRs. The drawings depicted the following:

- Typical BWR
- Typical PWR—one plant
- Typical PWR—two plants
- Typical PWR—three plants.

The ASME Code, Section XI, supports the intent of Regulatory Guide 1.26 within Subsection IWA, Subarticles 1300 and 1400, "Applications" and "Owner's Responsibility." Attachment B contains portions of the ASME Code, Section XI, 1983 Edition, and provides examples of how the ASME Code attempts to identify safety class components.

Using the Regulatory Guide and supporting documentation provides a determination of ASME Safety Class 1, 2, and 3 systems. The application of these documents can be interpreted in many different ways by different individuals, which is another challenge. The next real problem in our quest to determine an IST Program scope is how much of the system has to be considered safety class? or where does the safety class boundary end? This is a much easier problem to solve! Using ASME Code, Section XI, 1980 Edition, Subsections IWB, IWC, and IWD, Subarticles 2500 "Examination Requirements," we can extract the pressure testing boundaries and define the safety class boundary end. Subarticles 2500 for IWB, IWC, and IWD are provided in Attachment B. The pressure testing boundaries are all inclusive because no exceptions are allowed for VT-2 examinations of the pressure test boundary.

Along with the plant piping and instrumentation drawings (P&IDs) to identify the appropriate safety class boundaries, the following documents may be used to support the overall classification. It is important to understand that these documents are not mandatory when establishing boundaries, but support the selection and aid in determining weaknesses in the scope. These documents include

- Draft Regulatory Guide, *Identification of Valves for Inclusion in Inservice Testing Programs* (see Attachment C)
- NUREG-0800, *Standard Review Plan*, Section 3.2.2, "System Quality Group Classification" (see Attachment D)
- American Nuclear Society Standard 52.1, "Nuclear Safety Criteria for the Design of Stationary Boiling Water Reactor Plants" (see Attachment E)
- Generic Letter 89-04, "Guidance on Developing Acceptable Inservice Testing Programs" (see Attachment F).

After determining an acceptable safety class boundary scope using these documents, the boundaries should be reviewed to ensure that all the appropriate safety-related pumps and valves are incorporated and the IST Program scope is all inclusive. This final review is accomplished by obtaining a list of all the electrical loads powered by the emergency power supplies (diesels or batteries). In almost all cases, these components are safety-related; therefore, the portion of the system or the entire system is safety-related and

should be included in the ASME Safety Class pressure testing boundary.

Where can the pitfalls of changing the classification of a portion or the entire system to ASME Safety Class occur? The upgrading of a system from nonsafety class to ASME Safety Class must meet the requirements of being seismically qualified and must be procured according to quality assurance requirements.

Older Plants may run into problems with placing or upgrading systems into their safety class boundaries. The incompatibility may arise because the system was not designed as safety, quality, or seismic. If this is the situation, the upgrade cannot be accomplished without major plant modifications. The way to invoke the IST Program without ASME Code, Section XI, is to augment these components or systems as non-Code or Quality Class D. The advantages of having a noncode classification is that the components can be tested using Code guidelines in situations where compliance is not practical or justification is not required.

Recently, draft NUREG-1482, *Guidelines for Inservice Testing at Nuclear Power Plants*, supported this position. Section 2.2, "Criteria for Selecting Pumps and Valves for the IST Program" stated

"However, the scope of the OM Standards and Code has been expanded to include all safety-related pumps and valves in the IST program. Until the scope of 10 CFR 50.55a is changed, the scope of the IST program will continue to include those components within the Code Classes. In future rulemaking, the NRC will consider expanding the scope to include all safety-related pumps and valves. However, if licensees elect to consolidate testing for pumps and valves, the IST program (designating any non-Code components as such) may be acceptable for meeting other testing requirements for safety-related pumps and valves. Relief requests for non-Code components may be implemented without NRC evaluation and approval."

CONCLUSION

If your program has a good foundation and proper ASME Safety Class boundaries, you are on the way to meeting one of your goals. The next big challenge will be to exempt pumps or valves that are in the ASME Safety Class pressure testing boundaries and then determine the applicable test requirements for those that remain.

Attachments

A—Portions of Nuclear Regulatory Guide 1.26

B—Portions of ASME Code, Section XI, 1983 Edition

C—Appendix A of the Draft Regulatory Guide for Identification of Valves for Inclusion in Inservice Testing Program

D—Section 3.22, "System Quality Group Classification," *Standard Review Plan*, NUREG-0800

E—ANSI/ANS-52.1 "Nuclear Safety Criteria for the Design of Stationary Boiling Water Reactor Plants"

F—Attachment 1 from Generic Letter 80-04, "Guidance on Developing Acceptable Inservice Testing Programs"

Attachment A—Portions of Nuclear Regulatory Guide 1.26

QUALITY GROUPS

Quality Group A—The initial portion of the system is described in 10 CFR 50.55a which requires that components of the reactor coolant pressure boundary be designed, fabricated, erected, and tested to the highest available national standards.

Quality Group B—Components that are either part of the reactor coolant pressure boundary defined in 10 CFR 50.2(v) but excluded from the requirements pursuant to 10 CFR 50.55a:

a. Systems or portions of systems[a] important to safety that are designed for (1) emergency core cooling, (2) postaccident containment heat removal, or (3) postaccident fission product removal.

b. Systems or portions of systems[a] important to safety that are designed for (1) reactor shutdown or (2) residual heat removal.

c. Those portions of the steam systems of boiling water reactors extending from the outermost containment isolation valve up to but not including the turbine stop and bypass valves[b] and connected piping up to and including the first valve that is either normally closed or capable of automatic closure during all modes of normal reactor operation. Alternatively, for boiling water reactors containing a shutoff valve (in addition to the two containment isolation valves) in the main steam line and in the main feedwater line, Group B quality standards should be applied to those portions of the steam and feedwater systems extending from the outermost containment isolation valves up to and including the shutoff valve or the first valve that is either normally closed or capable of automatic closure during all modes of normal reactor operation.

d. Those portions of the steam and feedwater systems of pressurized water reactors extending from and including the secondary side of steam generators up to and including the outermost containment isolation valves and connected piping up to and including the first valve (including a safety or relief valve) that is either normally closed or capable of automatic closure during all modes of normal reactor operation.

e. Systems or portions of systems[a] that are connected to the reactor coolant pressure boundary and are not capable of being isolated from the boundary during all modes of normal reactor operation by two valves, each of which is either normally closed or capable of automatic closure.

Quality Group C—Applies to water-, steam-, and radioactive-waste-containing, pressure vessels; heat exchangers (other than turbines and condensers); storage tanks, piping, pumps, and valves not part of the reactor coolant pressure boundary or included in Quality Group B but part of

a. Cooling water and auxiliary feedwater systems or portions of these systems[a] important to safety that are designed for (1) emergency core cooling, (2) postaccident containment heat removal, (3) postaccident containment atmosphere cleanup, or (4) residual heat removal from the reactor and from the spent

a. The system boundary includes those portions of the system required to accomplish the specified safety function and connected piping up to and including the first valve (including a safety or relief valve) that is either normally closed or capable of automatic closure when the safety function is required.

b. The turbine stop valve and the turbine bypass valve, although not included in Quality Group B, should be subjected to a quality assurance program at a level generally equivalent to Quality Group B.

fuel storage pool (including primary and secondary cooling systems). Portions of normal reactor operation and (2) that cannot be tested adequately should be classified as Group B.

b. Cooling water and seal water systems or portions of these systems[a] important to safety that are designed for functioning of components and systems important to safety, such as reactor coolant pumps, diesels, and control room.

c. Systems or portions of systems[a] that are connected to the reactor coolant pressure boundary and are capable of being isolated from that boundary during all modes of normal reactor operation by two valves, each of which is either normally closed or capable of automatic closure. [c]

d. Systems, other than radioactive waste management systems, not covered by items 2.a. through 2.c. above that contain or may contain radioactive material and whose postulated failure would result in conservatively calculated potential offsite doses (using meteorology as recommended by Regulatory Guide 1.3, "Assumptions Used for Evaluating the Potential Radiological Consequences of a Loss of Coolant Accident for Boiling Water Reactors," and Regulatory Guide 1.4, "Assumptions Used for Evaluating the Potential Radiological Consequences of a Loss of Coolant Accident for Pressurized Water Reactors") that exceed 0.5 rem to the whole body or its equivalent to any part of the body. For those systems located in Seismic Category I structures, only single component failures need be assumed.

(However, no credit for automatic isolation from other components in the system or for treatment of released material should be taken unless the isolation or treatment capability is designed to the appropriate seismic and quality group standards and can withstand loss of offsite power and single failure of an active component.)

Quality Group D—Applies to water- and steam-containing components not part of the reactor coolant pressure boundary or included in Quality Groups B or C but part of systems or portions of systems that contain or may contain radioactive material.

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c. Components in influent lines may be classified as Group D provided they are capable of being isolated from the reactor coolant pressure boundary by an additional valve which has high leaktight integrity. Regulatory Guide 1.26, Quality Group Classifications and Standards for Water-, Steam-, and Radioactive-Waste-Containing Components of Nuclear Power Plants was issued as; Rev. No. 1 dated September 1974, Rev. No. 2 dated June 1975, and Rev. No. 3 dated February 1976 with it initially being published as Safety Guide 26, Quality Group Classification and Standards dated March 1972.

Attachment B—Portions of the ASME Code, Section XI, 1983 Edition

IWA-1300 APPLICATION

IWA-1310 Components Subject to Inspection and Testing

Components identified in this Division for inspection and testing shall be included in the inservice inspection plan. These components include nuclear power plant items such as vessels, containments, piping systems, pumps, valves, core support structures, and storage tanks, including their respective supports. The selection of components for the inservice inspection plan is subject to review by the regulatory and enforcement authorities having jurisdiction at the plant site.

IWA-1320 CLASSIFICATIONS

(a) Applications of the rules of this Division shall be governed by the group classification criteria of the regulatory authority having jurisdiction at the plant site as follows

(1) The rules of IWB shall be applied to those systems whose components are classified ASME Class 1 (Quality Group A).

(2) The rules of IWC shall be applied to those systems whose components are classified ASME Class 2 (Quality Group B).

(3) The rules of IWD shall be applied to those systems whose components are classified ASME Class 3 (Quality Group C).

(b) Optional construction of a component within a system boundary to a classification higher than the minimum class established in the component Design Specification (either upgrading from Class 2 to Class 1 or from Class 3 to Class 2) shall not affect the overall system classification by which the applicable rules of this Division are determined.

(c) Where all components within the system boundary or isolable portions of the system boundary are classified to a higher class than required by the group classification criteria, the rules of (a) above may be applied to the higher classification, provided the rules of the applicable Subsection are applied in their entirety.

(d) The portion of piping that penetrates a containment vessel, which is required by Section III to be constructed to Class 1 or 2 rules for piping and which may differ from the classification of the balance of the piping system, need not affect the overall system classification that determines the applicable rules of this Division.

(e) If systems safety criteria permit a system to be nonnuclear safety class and an Owner optionally classifies and constructs that system, or a portion thereof, to Class 2 or Class 3 requirements, the application of the rules of (a) above is at the option of the Owner and is not a requirement of this Division.

IWA-1400 OWNERS RESPONSIBILITY

The responsibilities of the Owner (1) of the power system (2) shall include the following:

(a) Determination of the appropriate Code Class(es) for each component (3) of the power plant, and identification of the system boundaries for each class of

components subject to inspection and the components exempt from examination requirements:

(1) Owner refers to the organization responsible for the operation, maintenance, safety, and power generation of the nuclear power system.

(2) Power system is that part of a nuclear power plant or unit that serves the purpose of producing or controlling the output of nuclear energy from nuclear fuel.

(3) Classification criteria are specified in 10 CFR 50.

Table IWB-2500-1, Examination Category, B-P Note(1), "The pressure retaining boundary during the system leakage test shall correspond to the reactor boundary system boundary with all valves in the normal position which is required for normal reactor operation startup. The VT-2 examination shall, however extend to and include the second closed valve at the boundary extremity."

Table IWC-2500-1, Examination Category C-H, All Pressure Retaining Components Note(7), "The pressure retaining boundary includes only those portions of the system required to operate or support the safety system function up to and including the normally closed valve (including a safety or relief valve) or valve capable of automatic closure when the safety function is required."

Table IWD-2500-1 Test and Examination Category; D-A, D-B and D-C Note (1). "The system boundary extends up to and including the first normally closed valve or valve capable of automatic closure as required to perform the safety-related system function."

Attachment C—Appendix A of the Draft Regulatory Guide for Identification of Valves for Inclusion in Inservice Testing Programs

Appendix A of the Draft Regulatory Guide for Identification of Valves for Inclusion in Inservice Testing Programs stated that the valves in the following systems and components in systems important to safety should be considered for inclusion in a comprehensive inservice testing program. The list is not intended to be all inclusive. Key components in instrumentation and auxiliary systems that are required to directly support plant shutdown or safety system function should also be considered.

1. Pressurized Water Reactors

1.1 Reactor Coolant System (RCS) and any proposed flow path for establishing natural circulation

1.2 Portions of Main Steam System

1.3 High-Pressure Injection System (HPCI)

1.4 Low-Pressure Injection System (LPCI)

1.5 Accumulator Systems

1.6 Containment Spray System

1.7 Primary and Secondary System Safety and Relief Valves and Atmospheric Relief Valves

1.8 Portions of Main Feedwater System

1.9 Auxiliary Feedwater Systems

1.10 Residual Heat Removal System (Shutdown Cooling)

1.11 Component Cooling Water Systems

1.12 Service Water Systems

1.13 Containment Isolation Valves required to change position on a containment isolation signal

1.14 Chemical Volume and Control System

1.15 Emergency Diesel Engine Fuel Oil Storage and Transfer System

1.16 Ventilation Systems that perform a safety function

1.17 Instrument Air Systems that are required to support safety system functions

2.0 Boiling Water Reactors

2.1 Reactor Coolant Recirculation System (RCS)

2.2 Portions of Main Steam Supply

2.3 High-Pressure Injection System (HPCI)

2.4 Low-Pressure Injection System (LPCI)

2.5 Residual Heat Removal System (Steam Condensing, Shutdown Cooling, Suppression Pool Cooling)

2.6 Low Pressure Core Spray System

2.7 Safety, Relief, and Safety/Relief Valves of RCS and secondary systems

2.8 Reactor Core Isolation Cooling System

2.9 Containment Cooling System (Spray)

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a. The terminology for various systems such as Accumulator Systems and others may vary depending on the preference of the individual nuclear steam system supplier.

2.10 Containment isolation valves required to change position on a containment isolation signal

2.11 Standby Liquid Control System

2.12 Automatic Depressurization System

2.13 Control Rod Drive Hydraulic System

2.14 Active Valves in Service and Backup Water, Closed Cooling Water, Firewater, or Well Water Systems

2.15 Emergency Diesel Engine Fuel Oil Storage and Transfer System

2.16 Portions of Main Feedwater System

2.17 Instrument Air Systems that are required to support safety System Functions

Attachment D—Section 3.2.2, "System Quality Group Classification," from *Standard Review Plan*, NUREG-0800

NUREG-0800, *Standard Review Plan*, Section 3.2.2, "System Quality Group Classification" states that the following fluid systems important to safety for pressurized water reactor (PWR) and boiling water reactor (BWR) plants are reviewed by the Mechanical Engineering Branch (MEB) with regard to quality group classification.

FLUID SYSTEMS IMPORTANT TO SAFETY FOR PWR PLANTS

- Reactor Coolant System
- Emergency Core Cooling System
- Containment Spray System
- Chemical and Volume Control System
- Boron Thermal Regeneration System—On some plants this system may be nonsafety-related, providing it complies with the requirements of Regulatory Guide 1.26 and portions of the system that perform a safety-related function
- Boron Recycle System—On some plants this system may be nonsafety-related, providing it complies with the requirements of Regulatory Guide 1.26 and portions of the system that perform a safety-related function
- Residual Heat Removal System
- Component Cooling Water System—Portions of the system that perform a safety-related function
- Spent Fuel Pool Cooling and Cleanup System—Portions of the system that perform a safety-related function
- Sampling System—Portions of the system to outermost containment isolation valve
- Service Water System—Portions of the system that perform a safety-related function
- Compressed Air System—On some plants this system may be nonsafety-related, providing it complies with the requirements of Regulatory Guide 1.26 and portions of the system that perform a safety-related function
- Emergency Diesel Engine Fuel Oil Storage and Transfer System
- Emergency Diesel Engine Cooling Water System
- Emergency Diesel Engine Starting System
- Emergency Diesel Engine Lubrication System
- Emergency Diesel Engine Combustion Air Intake and Exhaust System
- Main Steam System—Portions of the system to outermost containment isolation valve
- Feedwater System—Portions of the system to outermost containment isolation valve
- Auxiliary Feedwater System
- Steam Generator Blowdown System—Portions of the system to outermost containment isolation valve
- Containment Cooling System
- Containment Purge System
- Ventilation Systems for Areas such as Control Room and Engineered Safety Features Rooms

- Combustible Gas Control System
- Condensate Storage System—On some plants this system may be nonsafety-related, providing it complies with the requirements of Regulatory Guide 1.26

FLUID SYSTEMS IMPORTANT TO SAFETY FOR BWR PLANTS

- Reactor Recirculation System
- Main Steam System (up to but not including the turbine)
- Feedwater System (up to outermost containment isolation valve or shutoff valve, as applicable)
- Relief Valve Discharge Piping
- Control Rod Drive Hydraulic System—Portions of the system that perform a safety-related function
- Standby Liquid Control System
- Reactor Water Cleanup System
- Fuel Pool Cooling and Cleanup System—Portions of the system that perform a safety-related function
- Sampling System—Portions of the system to outermost containment isolation valve
- Residual Heat Removal System
- High Pressure Core Spray System
- Low Pressure Core Spray System
- Reactor Core Isolation Cooling System
- RHR Service Water System
- Emergency Equipment Service Water System
- Compressed Air System—On some plants this system may be nonsafety-related, providing it complies with the requirements of Regulatory Guide 1.26 and portions of the system that perform a safety-related function
- Emergency Diesel Engine Fuel Oil Storage and Transfer System
- Emergency Diesel Engine Cooling Water System
- Emergency Diesel Engine Lubrication System
- Emergency Diesel Engine Combustion Air Intake and Exhaust System
- Standby Gas Treatment System
- Combustible Gas Control System
- Containment Cooling System
- Main Steam Isolation Valve Leakage Control System
- Condensate and Refueling Water Storage System—Portions of the system that perform a safety-related function
- Ventilation Systems for Areas such as Control Room and Engineered Safety Features Rooms.

Attachment E—American National Standards Institute/American Nuclear Society Standard 52.1 "Nuclear Safety Criteria for the Design of Stationary Boiling Water Reactor Plants"

American National Standards Institute/American Nuclear Society (ANSI/ANS) Standard 52.1, "Nuclear Safety Criteria for the Design of Stationary Boiling Water Reactor Plants" provides general criteria which shall be used by the designer:

a. Correlation of Safety Class with ASME III Class:

(1) ASME III Class 1 design rules shall apply to Safety Class 1 components.

(2) ASME III Class 2 design rules shall apply to Safety Class 2 components.

(3) ASME III Class 3 design rules shall apply to Safety Class 3 components.

Table A-1 in the standard "Equipment Classification" lists the principal equipment with its safety class, quality assurance requirement, principal construction code, seismic requirement, safety class definition reference, and special requirement reference. The principal equipment includes the following:

- Reactor Core and Internals
- Reactivity Control Systems
- Protection System
- Reactor Coolant System
- Shutdown Heat Removal Systems
- Reactor Coolant Auxiliary Systems
- Cooling Water Systems (CWS)
- Emergency Core Cooling Systems
- Primary Containment
- Secondary Containment
- Containment Auxiliary Systems
- Safety-related Area Cooling Systems
- Fuel Storage and Handling
- Electrical Power Systems
- Fire Protection Systems
- Control Complex
- Radioactive Waste Processing (Liquid, Gaseous, and Solid Waste Processing)
- Off-Gas System
- Power Conversion System.

Table A-2 in the standard "Examples of Typical Classification of Components Comprising Complex Principal Equipment" lists the components in each set of principle equipment.

Attachment F—Attachment 1 from Generic Letter 89-04, "Guidance on Developing Acceptable Inservice Testing Programs"

POTENTIAL GENERIC DEFICIENCIES RELATED TO IST PROGRAMS AND PROCEDURES

11. IST Program Scope

The 10 CFR 50.55a requires that inservice testing be performed on certain ASME Code Class 1, 2, and 3 pumps and valves. Section XI Subsections IWP-1100 and IWV-1100 defines the scope of pumps and valves to be tested in terms of plant shutdowns and accident mitigation. The plants FSAR (or equivalent) provides definitions of the necessary equipment to meet these functions. The staff has noted during past IST program reviews and inspections that licensees do not always include the necessary equipment in their IST programs. Licensees should review their IST programs to ensure adequate scope. Examples that are frequently erroneously omitted from IST programs are

a. BWR scram system valves

b. Control room chilled water system pumps and valves

c. Accumulator motor-operated isolation valves, or accumulator vent valve

d. Auxiliary pressurizer spray system valves

e. Boric acid transfer pumps

f. Valves in emergency boration flow path

g. Control valves that have a required fail-safe position

h. Valves in mini-flow lines.

It should be recognized that the above examples of pumps and valves do not meet the IWP and IWV scope statement requirements for all plants.

The intent of 10 CFR 50 Appendix A, GDC-1, and Appendix B, Criterion XI, is that all components, such as pumps and valves necessary for safe operation, are to be tested to demonstrate that they will perform satisfactorily in service. Therefore, while 10 CFR 50.55a delineates the testing requirements for ASME Code Class 1, 2, and 3 pumps and valves, the testing of pumps and valves is not to be limited to only those covered by 10 CFR 50.55a.

Performance Based Testing and Maintenance for Check Valves

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ABSTRACT

This paper proposes a methodology for optimizing check valve testing and maintenance activities. Generic Letter 89-04, "Guidance on Developing Acceptable Inservice Testing Programs," and the 1989 Edition of the American Society of Mechanical Engineers (ASME) Code specify the current requirements for testing of check valves. Recent advances in technology and the trend toward performance-based approaches to testing and maintenance represent an opportunity to reduce check valve operation and maintenance costs.

This paper describes an evaluation methodology that leads to the specification of an optimum set of inservice testing and maintenance activities for a plant check valve population. The process involves analyzing check valve performance based on preestablished performance attributes. Attributes include relative consequence of failure, historical reliability, maintenance frequency, physical degradation, and service fluid.

Each check valve is numerically scored based on the valves' performance against the stated attributes. This score is used to justify a reprioritization of check valve work activities such that safety-significant (high-risk), poor performing valves receive a higher degree of maintenance and testing rigor, while good performing check valves receive less.

Applying this methodology may allow justifications for alternative testing and maintenance to be developed and substantiated on the basis of valve performance characteristics versus across-the-board implementation of the ASME Code requirements.

The Current State of Inservice Testing Programs at U.S. Nuclear Power Plants—A Regulatory Overview[a]

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ABSTRACT

Information is provided on inservice testing (IST) of pumps and valves at U.S. nuclear power plants to provide consistency in the implementation of regulatory requirements and to enhance communications among utility licensees who may have, like NSSS vendors, similar kinds and numbers of components or comparable IST programs. Documents discussed include the ASME *Operation and Maintenance Standards* Parts 6 and 10 (covering inservice testing of pumps and valves in light water reactor power plants), the draft NUREG-1482, *Guidelines for Inservice Testing at Nuclear Power Plants* (including review comments by Nuclear Management and Resource Council), and applicable Licensee Event Reports including summaries of several reports relating to IST.

INTRODUCTION

Inservice testing of certain safety-related pumps and valves in nuclear power plants has been required by federal regulations since February 1976. Section 50.55a of Title 10 of the *Code of Federal Regulations* (10 CFR 50.55a) defines the requirements for applying the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code (the Code), Section XI, to testing of Code-class pumps and valves. In 1975, the ASME formed the Committee on Operation and Maintenance of Nuclear Power Plants (O&M Committee) to identify, develop, maintain, and review codes and standards that would replace current Section XI requirements. In 1992, the 1989 Edition of the ASME Code, Section XI, was incorporated into Paragraph (b) of 10 CFR 50.55a. The 1989 Edition incorporated by reference ASME *Operation and Maintenance Standards* Part 6, "Inservice Testing of Pumps in Light-Water Reactor Power Plants," and Part 10, "Inservice Testing of Valves in Light-Water Reactor Power Plants," to replace Section XI, Subsections IWP and IWV. In a proposed rule that should be published by the end of 1994, the USNRC will indicate its intent to incorporate the *Code for Operation and Maintenance of Nuclear Power Plants*, which codifies the Parts 6 and 10 (OM-6 and OM-10) requirements, into Paragraph (b) of 10 CFR 50.55a.

The purpose of this paper is to provide information that may be useful in comparing inservice testing (IST) programs and implementation. Recent activity in IST is discussed below. A proposed supplement to Generic Letter 89-04 and a draft NUREG-1482 were published in late 1993, and a public meeting to discuss the draft NUREG was held in February 1994. A number of IST programs have been updated to later editions of the Code, but there are still significant

a. This paper was prepared by an employee of the U.S. Nuclear Regulatory Commission. It presents information that does not represent a current staff position. The USNRC has neither approved nor disapproved its technical content.

variations in the scope of IST programs and the type of relief requests submitted for USNRC approval. Inspection results indicate that problems in implementing the IST programs continue to occur, although many of these are programmatic and do not represent safety issues. Similarly, the Licensee Event Reports that relate to IST are generally programmatic or administrative errors in the implementation of the testing requirements. Recent examples are given. While the events show that there have been no major safety concerns identified, the problems that do occur may indicate underlying concerns with (a) management attention and (b) understanding of the Code requirements and attention to detail.

Finally, reviews of IST programs and plant specific information may be useful to licensees for comparison purposes or in identifying other utilities that may be using the same edition of the Code and have similar program scope. The information is provided so that licensees can use it to communicate with other plants that may be using the same Code edition, have the same nuclear steam supply system (NSSS) vendor, or have similar numbers of components. Also, a licensee may use the information to compare the number of components in its program with other similar plants.

GUIDELINES DOCUMENT

Since the incorporation of Part 6 (OM-6) and Part 10 (OM-10), several plants have elected to update their current inservice testing program. For plants that have not elected, or are not yet required, to update to OM-6 and OM-10, NUREG-1482, *Guidelines for Inservice Testing at Nuclear Power Plants* identified certain portions of these standards that could be used by licensees. NUREG-1482 was published as a draft for public comment in November 1993. A public meeting was held February 2 and 3, 1994, to discuss issues and questions prior to the close of the public comment period. The participation and interest demonstrated at the public meeting and by the public comments were encouraging. Each of the questions from the public meeting and each of the formal comments will be addressed in an appendix and changes will be made to the draft NUREG-1482 as appropriate. A copy of the Nuclear Management Resources Council (NUMARC) comments is included as Appendix A. One comment expressed several times is that licensees should not have to reference the section of NUREG-1482 that allows implementation of later editions, or portions of later editions, of the Code because the particular action will be in compliance with the Code. However, the regulations require that when later editions or portions of later editions are used, the use is subject to USNRC approval [see 10 CFR 50.55a(f)(4)(iv)]. If a licensee is already committed to the 1989 Edition of ASME Section XI, the NUREG recommendations on the use of the 1989 Edition do not apply and no reference is required. However, if a licensee has a program that was developed to the 1986 Edition of the Code and wants to use portions of the 1989 Edition without making a submittal to the USNRC requesting approval, NUREG-1482 gives the approval and requires only that the licensee include a statement in the IST program document stating that the NUREG-1482 recommendation has been incorporated into the program. The statement has to be made in only one place of the IST Program. If the licensee does not document that the licensee is following the NUREG-1482 recommendation, the approval given in the NUREG is not applicable and the licensee could be found in violation, particularly if a portion of a later code edition is used without complying with the related requirements.

Plant Specific Information

One purpose of NUREG-1482 was to bring consistency to the implementation of the regulatory requirements for inservice testing of pumps and valves. Tables 1 through 4 list the U.S. commercial nuclear power plants by NSSS vendor. As shown on the tables, even within NSSS vendor groups, the number of components and relief requests at different plants varies. The numbers on the tables were obtained from the latest program submittal available to the staff. The staff plans to update the tables as programs are updated and revised. The Beaver Valley, Unit 1, inservice testing program was reviewed in detail and

Table 1. NSSS VENDOR—Babcock & Wilcox

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Arkansas Nuclear One, Unit 1 Entergy Operations, Inc.	12/19/84–12/18/94 2nd interval	1980 edition with addenda through winter 1981 addenda	14	3	324	19	None included in submittal
Crystal River – 3 Florida Power Corporation	03/13/87–03/12/97 2nd interval	1983 edition with addenda through summer 1983 addenda	27	3	416	23	25
Davis–Besse Toledo Edison Company	9/21/90–9/20/00 2nd interval	1986 edition	20	7	431	24	60
Oconee – 1 Duke Power Co.	07/01/92–07/01/02 3rd interval	1986 edition	25	10	350	40	30
Oconee – 2 Duke Power Co.	07/01/92–07/01/02 3rd interval	1986 edition	15	10	327	40	30
Oconee – 3 Duke Power Co.	07/01/92–07/01/02 3rd interval	1986 edition	16	10	328	40	30
Three Mile Island, Unit 1 GPU Nuclear Corp.	—	—	—	—	—	—	—

Table 2. NSSS VENDOR—Combustion Engineering.

Plant	Interval dates	Code edition	Number of pumps	Number of Pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Arkansas Nuclear One, Unit 2 Entergy Operations, Inc.	03/26/90–03/25/00 2nd interval	1986 edition	21	9	341	12	No included in IST program plan
Calvert Cliffs 1 Baltimore Gas & Electric	04/01/87–04/01/97 2nd interval	1983 edition with addenda through summer 1983 addenda	24	7	312	14	25
Calvert Cliffs 2 Baltimore Gas & Electric	04/01/87–04/01/97 2nd interval	1983 edition with addenda through summer 1983 addenda	24	7	286	13	25
Fort Calhoun Omaha Public Power District	09/26/93–09/25/03 3rd interval	1989 edition	26	3	629	7	39
Maine Yankee, Maine Yankee Atomic Power Company	12/28/92–12/27/02 3rd interval	Pumps—OM–6 valves—1986 edition and portions of OM–10	21	1	378	3	46
Millstone 2 Northeast Nuclear Energy Company	12/26/85–12/26/95 2nd interval	1980 edition with addenda through winter 1981 addenda	26	—	—	—	—
Palisades Consumers Power Co.	11/10/83–05/11/95 2nd interval (extended due to outages)	1983 edition, addenda through summer 1983 addenda	21	7	494	25	26

Table 2. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of Pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Palo Verde 1 Arizona Public Service Co.	01/28/86–01/27/96 1st interval	1980 edition with addenda through winter 1981 addenda	—	—	—	—	—
Palo Verde 2 Arizona Public Service Co.	09/30/86–09/30/96 1st interval	1980 Edition with addenda through winter 1981 addenda	—	—	—	—	—
Palo Verde 3 Arizona Public Service Co.	01/08/88–01/08/98 1st interval	1980 Edition with addenda through winter 1981 addenda	—	—	—	—	—
San Onofre 2 Southern California Edison Co. & San Diego Gas & Electric Co.	08/18/93–08/17/03 2nd interval	1989 edition	30	1	559	1	74
San Onofre 3 Southern California Edison Co. & San Diego Gas & Electric Co.	08/18/93–08/17/03 2nd interval	1989 edition	30	1	559	1	74
St. Lucie 1 Florida Power & Light Co.	02/11/88–02/10/98 2nd interval	1983 edition with addenda through summer 1983 addenda	22	13	379	38	29

Table 2. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of Pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
St. Lucie 2 Florida Power & Light Co.	08/08/93–08/08/03 2nd interval	1986 edition	22	16	395	34	31
Waterford 3 Entergy Operations, Inc.	09/24/85–09/23/95 1st interval	1980 edition with addenda through winter 1981 addenda	25	5	509	52	15

Table 3. NSSS vendor—Westinghouse Corporation.

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Beaver Valley 1 Duquesne Light Co.	10/01/86–09/30/96 2nd interval	1983 edition with addenda through summer 1983 addenda	32	10	635	43	31 (93 valves total)
Beaver Valley 2 Duquesne Light Co.	11/17/87–11/16/97 1st interval	1983 edition with addenda through summer 1983 addenda	30	8	593	32	57
Braidwood 1 Commonwealth Edison Co.	07/29/88–07/28/98 1st interval	1983 edition with addenda through summer 1983 addenda	44	5	367	20	36
Braidwood 2 Commonwealth Edison Co.	10/17/88–10/16/88 1st interval	1983 edition with addenda through summer 1983 addenda	44	5	367	20	36
Byron 1 Commonwealth Edison Co.	09/16/85–09/15/95 1st interval	1983 edition with addenda through summer 1983 addenda	23	6	328 Code 20 Non-Code	18	42
Byron 2 Commonwealth Edison Co.	08/21/87–08/20/97 1st interval	1983 edition with addenda through summer 1983 addenda	19	6	328 Code 20 Non-Code	18	42

Table 3. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Callaway Union Electric	12/19/84–12/18/94 1st interval	1980 edition with addenda through winter 1981 addenda	21	10	687	28	33
Catawba 1 Duke Power Co.	01/17/85–01/16/95 1st interval	1980 edition with addenda through winter 1981 addenda	—	—	610	—	—
Catawba 2 Duke Power Co.	05/15/86–05/14/96 1st interval	1980 edition with addenda through winter 1981 addenda	—	—	636	—	—
Comanche Peak 1 Texas Utilities Electric Co.	02/08/90–02/07/00 1st Interval	1989 edition	35	1	693	7	40
Comanche Peak 2 Texas Utilities Electric Co.	02/02/93–02/01/03 1st interval	1989 edition	28	1	693	7	40
D.C.Cook 1 Indiana/Michigan Power Co.	10/25/84–10/24/94 2nd interval	1983 edition with addenda through 1983 summer addenda	—	—	—	—	—
D.C.Cook 2 Indiana/Michigan Power Co.	12/23/87–12/22/97 2nd interval	1983 edition with addenda through 1983 summer addenda	—	—	—	—	—

Table 3. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Diablo Canyon 1 Pacific Gas & Electric Co.	05/07/85–05/06/95 1st interval	1977 edition with addenda through summer 1978 (approved per NUREG–0675, Supp.13)	20 Code 2 Non-Code	3	409	23	27
Diablo Canyon 2 Pacific Gas & Electric Co.	03/13/86–03/12/96 1st interval	1977 edition with addenda through summer 1978 (approved per NUREG–0675, Supp.13)	20 Code 2 Non-Code	3	409	23	27
Ginna Rochester Gas & Electric Corp.	01/01/90–12/31/99 3rd interval	1986 edition	25	9	440	33	34
Haddam Neck Connecticut Yankee Atomic Power Co.	12/27/84–12/26/94 2nd interval	1980 edition with addenda through winter 1980 addenda	—	—	—	—	—
H.B.Robinson 2 Carolina Power & Light Co.	02/19/92–02/19/02 3rd interval	1986 edition	26	8	500	20	39
Indian Point 2 Consolidated Edison Co.	07/01/84–06/30/94 2nd interval	1980 edition with addenda through winter 1981 addenda	34	11	717	90	0 (included in relief requests)

Table 3. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Indian Point 3 Power Authority of the State of New York	08/30/86–08/29/96 2nd interval	1983 edition with addenda through summer 1983 addenda	35	13	564	35	51
Joseph M. Farley 1 Southern Nuclear Operating Co.	12/01/87–11/30/97 2nd interval	1983 edition with addenda through summer 1983 addenda	28	16	491	41	21
Joseph M. Farley 2 Southern Nuclear Operating Co.	Updated early to coincide with Unit 1 03/31/89–11/30/97 2nd interval	1983 edition with addenda through summer 1983 addenda	22	16	434	38	23
Kewaunee Wisconsin Public Service Corp.	06/16/94–06/14/04	1989 edition	17	3	258	9	15
McGuire 1 Duke Power Co.	12/01/91–11/30/01 2nd interval	1989 edition	27	8	789	4	95
McGuire 2 Duke Power Co.	03/01/94–02/29/04 2nd interval	1989 edition	27	8	789	4	95
Millstone 3 Northeast Nuclear Energy Co.	04/24/86–04/23/96 1st interval	1983 edition with addenda through summer 1983 addenda	17	4	551	27	31

Table 3. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
North Anna 1 Virginia Electric & Power Co.	12/14/90–12/14/00 2nd interval	1986 edition	31 Code 4 Non-Code 35	8	529 Code 77 Non-Code 606	35 Code 4 Non-Code 39	27
North Anna 2 Virginia Electric & Power Co.	12/14/90–12/14/00 2nd interval	1986 edition	31 Code 4 Non-Code 35	8	514 Code 82 Non-Code 596	35 Code 4 Non-Code 39	25
Point Beach 1 Wisconsin Electric Power Co.	12/21/90–12/20/00 3rd interval	1986 edition	30	22	391	37	37
Point Beach 2 Wisconsin Electric Power Co.	12/21/90–12/20/00 3rd interval	1986 edition	14	22	287	37	37
Prairie Island 1 Northern States Power Co.	12/16/93–12/15/03 3rd interval	1989 edition	15	5	339	1	66
Prairie Island 2 Northern States Power Co.	12/21/94–12/20/04 3rd interval	1989 edition	10	4	311	0	37
Salem 1 Public Service Electric & Gas	06/30/87–06/29/97 2nd interval	1983 edition with addenda through summer 1983 addenda	36	4	548 Code 51 Non-Code 599	39	47

Table 3. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Salem 2 Public Service Electric & Gas	06/30/87–06/29/97 2nd interval (concurrent with Unit 1)	1983 edition with addenda through summer 1983 addenda	36	4	594 Code 22 Non-Code 616	34	52
Seabrook 1 North Atlantic Energy Service Corp.	08/19/90–08/18/00 1st interval	1983 edition with addenda through summer 1983 addenda	26	2	412	18	38
Sequoyah 1 Tennessee Valley Authority	07/01/81–06/30/94 1st interval (extended)	1974 edition with addenda through summer 1975	22	11	569	18	30
Sequoyah 2 Tennessee Valley Authority	06/01/82–05/31/94 1st interval (extended)	1977 edition with addenda through summer 1978 addenda	22	11	569	18	30
Shearon Harris 1 Carolina Power & Light Co.	05/02/87–05/01/97 1st interval	1983 edition with addenda through summer 1983 addenda	18	10	300 to 400	30	7
South Texas Project 1 Houston Lighting & Power Co.	08/25/88–08/24/98 1st interval	1983 edition with addenda through summer 1983 addenda	30	7	543	43	0 (included in relief requests)

Table 3. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
South Texas Project 2 Houston Lighting & Power Co.	06/19/89–06/18/99 1st interval	1983 edition with addenda through summer 1983 addenda	36	7	545	43	0 (included in relief requests)
Summer South Carolina Electric & Gas	01/01/94–12/31/03 2nd interval	1989 edition	Program not yet submitted	—	—	—	—
Surry 1 Virginia Electric & Power Co.	05/10/94–05/09/04 3rd interval	1989 edition	33 Code 4 Non-Code	7 Code 1 Non-Code	585	11 Code 3 Non-Code	34
Surry 2 Virginia Electric & Power Co.	05/10/94–05/09/04 3rd interval	1989 edition	24 Code 2 Non-Code	4 Code 1 Non-Code	437	11 code 3 Non-Code	34
Turkey Point 3 Florida Power & Light Co.	02/22/94–02/21/04 3rd interval	1989 edition	28	5	629	2	55
Turkey Point 4 Florida Power & Light Co.	04/15/94–04/14/04 3rd interval	1898 edition	25	5	629	2	55
Vogtle 1 Southern Nuclear Operating Co.	06/01/87–05/31/97 1st interval	1983 edition with addenda through summer 1983 addenda	30	3	400	27	38

Table 3. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Vogtle 2 Southern Nuclear Operating Co.		1983 edition with addenda through summer 1983 addenda	29	2	406	26	37
Wolf Creek 1 Wolf Creek Nuclear Operating Corp.	09/03/85–09/02/95 1st interval	1980 edition with addenda through winter 1981 addenda, using OM–10 (1989 edition)	23	9	884	6	51
Zion 1 Commonwealth Edison Co.	12/31/93–12/30/03 3rd interval	1989 edition	26	10	509	9	10
Zion 2 Commonwealth Edison Co.	09/14/94–09/13/04 3rd interval	1989 edition	26	10	489	9	10

Table 4. NSSS vendor—General Electric.[a]

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Big Rock Point Consumers Power Co.	03/29/93–03/28/03 4th interval	Valves: 1986 edition, pumps: OM–6	Program not yet submitted; relief requests submitted.	2	Program not yet submitted; relief requests submitted.	12	None included in IST submittal of relief requests.
Browns Ferry 1 Tennessee Valley Authority	09/01/92–08/31/02 2nd interval (concurrent for all three units)	1986 edition	64	15	507	19	10
Browns Ferry 2 Tennessee Valley Authority	09/01/92–08/31/02 2nd interval	1986 edition	64	15	507	19	10
Browns Ferry 3 Tennessee Valley Authority	09/01/92–08/31/02 2nd interval	1986 edition	64	15	507	19	10
Brunswick 1 Carolina Power & Light Co.	07/10/86–07/09/96 2nd interval	1980 edition with addenda through winter 1981 addenda	—	—	—	—	—
Brunswick 2 Carolina Power & Light Co.	07/10/86–07/09/96 2nd interval	1980 edition with addenda through winter 1981 addenda	—	—	—	—	—

Table 4. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Clinton Illinois Power Co.	11/24/87–11/23/97 1st interval	1980 edition with addenda through winter 1981 addenda	22	5	1,401	20	Not included in IST Program Plan
Cooper Nebraska Public Power District	07/01/84–06/30/94 2nd interval	1980 edition with addenda through winter 1981 addenda	33	10	1,155	44	7
Dresden 2 Commonwealth Edison Co.	03/01/92–02/28/02 3rd interval	1986 edition OM–6 for pump vibration	38	8	336	32	11
Dresden 3 Commonwealth Edison Co.	03/01/92–02/28/02 3rd interval	1986 edition OM–6 for Ppmp vibration	38	8	301	32	11
Duane Arnold Iowa Electric Light & Power Co.	02/01/85–01/31/95 2nd interval	1980 edition with addenda through winter 1981 addenda	26	17	610	53	16
Edwin I. Hatch 1 Southern Nuclear Operating Co.	01/01/86–01/01/96 2nd interval	1980 edition with addenda through winter 1981 addenda	24	4	498	39	11
Edwin I. Hatch 2 Southern Nuclear Operating Co.	01/01/86–01/01/96 2nd interval	1980 edition with addenda through winter 1981 addenda	23	4	547	39	11

Table 4. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Fermi 2 Detroit Edison Co.	01/23/88–01/22/98 1st interval	1980 edition with addenda through winter 1980 addenda	34	7	—	35	—
Grand Gulf 1 Entergy Operations, Inc.	07/01/85–06/30/95 1st interval	1977 edition with addenda through summer 1979 addenda and 1980 edition with addenda through 1980 Addenda	19	11	1,882	14	76
Hope Creek 1 Public Electric & Gas Co.	12/20/86–12/19/96 1st interval		—	—	—	—	—
James A. Fitzpatrick Power Authority of the State of New York	07/28/85–07/27/95 2nd interval	1980 edition with addenda through winter 1981 addenda	16	12	457	29	13
LaSalle County 1 Commonwealth Edison Co.	01/01/84–11/23/94 1st interval (extended due to outage)	1980 edition with addenda through winter 1980 addenda	23	8	1,297	33	13
LaSalle County 2 Commonwealth Edison Co.	10/17/84–10/16/94 1st interval	1980 edition with addenda through winter 1980 addenda	21	8	1,190	33	13

Table 4. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Limerick 1 Philadelphia Electric Co.	02/01/86–01/31/96 1st interval	1986 edition	39	3	1,716	23	24
Limerick 2 Philadelphia Electric Co.	01/08/90–01/07/00 1st interval	1986 edition	39	3	1,623	23	24
Millstone 1 Northeast Nuclear Energy Co.	03/01/91–02/28/01 3rd interval	1989 edition	17	24	288	50	Included in relief requests
Monticello Northern States Power Co.	05/31/92–05/30/02 3rd interval	1986 edition	21	4	1,093	24	14
Nine Mile Point 1 Niagara Mohawk Power Corp.	12/01/89–11/30/99 3rd interval	1983 edition with addenda through summer 1983 addenda OM–1–1987 (Safety/Relief Valves) OM–6 (Pumps)	33	6	1,104	20	11
Nine Mile Point 2 Niagara Mohawk Power Corp.	03/11/88–03/10/98 1st interval	1983 edition with addenda through summer 1983 addenda	26	1	1,680	17	22

Table 4. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Oyster Creek GPU Nuclear Corp.	12/01/89–11/30/99 3rd interval	1986 edition	26	33	966	38	Listed as Relief Requests
Peach Bottom 2 Philadelphia Electric Co.	09/19/86–09/18/96 2nd interval (1st interval was extended)	1980 edition with addenda through winter 1981 addenda. Safety/Relief Valves: OM–1 (code case N–415)	25	3	1,366	22	14
Peach Bottom 3 Philadelphia Electric Co.	12/23/85–12/22/95 2nd interval (1st interval was extended)	1980 edition with addenda through winter 1981 addenda. Safety/Relief Valves: OM–1 (code case N–415)	16	3	1,299	22	14
Perry 1 Centerior Energy Co.	11/18/87–11/17/97 1st interval	1983 edition with addenda through summer 1983 addenda	31	8	2,356 (1,593 CRDH valves)	34	18
Pilgrim 1 Boston Edison Co.	12/01/92–11/30/02 3rd interval	1986 edition	24	8	1,286	38	11
Quad Cities 1 Commonwealth Edison Co.	02/18/93–02/17/03 3rd interval	1986 edition; OM–6 for pumps	18	4	617	28	13

Table 4. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Quad Cities 2 Commonwealth Edison Co.	03/10/93–03/09/03 3rd interval	1986 edition; OM–6 for pumps	16	4	577	28	13
River Bend 1 Entergy Operations, Inc.	06/16/86–06/15/96 1st interval	1980 edition with addenda through winter 1981 addenda	29	13	696	35	31
Susquehanna 1 Pennsylvania Power & Light Co.	06/08/83–05/31/94 1st interval (extended)	1980 edition with addenda through winter 1980 addenda	27	3	450	46	13
Susquehanna 2 Pennsylvania Power & Light Co.	02/12/85–05/31/94 1st interval (shortened to be concurrent with Unit 1)	1980 edition with addenda through winter 1980 addenda	14	3	450	46	13
Vermont Yankee Yankee Nuclear Power Corp.	11/30/92–11/20/02 3rd interval	1989 edition	26	9	1,300	11	23

Table 4. (continued).

Plant	Interval dates	Code edition	Number of pumps	Number of pump relief requests	Number of valves	Number of valve relief requests	Number of cold shutdown justifications/ refueling outage justifications
Washington Nuclear 2 Washington Public Power Supply System	12/13/84–12/12/94 1st interval	1980 edition with addenda through winter 1980 addenda	18	6	1,481	21	13

NOTE: The high numbers of valves in BWRs results from including control rod drive hydraulic control unit valves.

summarized in Appendix B. The "Summary of Relief Requests" for the Duane Arnold Energy Center IST Safety Evaluation is described in Appendix C. Appendix D includes Tables D-1 through D-10, which detail valve and pump types for several plants. The examples are presented as information only and may not represent the most current programs for the specific plants.

Inspection Findings

Recent USNRC inspection findings have indicated that there continue to be questions on the implementation of certain Code requirements. Several plants that have updated to OM-1 for safety and relief valve testing have found problems in interpreting the testing requirements. Problems may continue until the most recent revision of Part 1 to the OM Code, which corrects a number of editorial and technical issues, is incorporated into the regulations and used by licensees rather than the earlier revisions of OM-1.

One plant did not have a relief request for calculation of flow rate using a change in tank level. The USNRC has approved this method when there are no flow instruments in the system, provided the method is proceduralized. The Section XI Committee had received an inquiry concerning whether a level monitor meets the requirement of the Code to measure flow using a quantity or rate meter installed in the pump test circuit. The inquiry has not yet been issued.

Several plants had not included certain manual valves that meet the scope of 10 CFR 50.55a and Section XI because they believed they were exempt from the exercising requirements because there was no accompanying actuator.

One plant had requested relief from the range limits for pump bearing vibration alert for a high-pressure coolant injection pump that had exhibited levels consistently in the alert range. The licensee had been attempting to lower the vibration levels, but rather than continue aggressively pursuing a long-term resolution, had instead opted to request relief to allow continued use of the pump with higher than normal vibration levels.

One plant requirement in a test procedure for standby liquid control pumps called for running a pump for 5 minutes before taking readings of flow, differential pressure, and vibration. However, in implementing the procedure, readings were taken immediately after the pump started so that the run time could be limited, so as not to overfill the test tank. Several plants have requested and received relief from the pump run time requirements in this situation; however, the subject licensee indicated that he believed the Code could be interpreted to cover the situation.

One plant had extended the interval in accordance with IWA-2400 because of one or more outages longer than 6 months. When USNRC inspectors asked about the dates of the interval and why one interval was greater than 10 years, the plant staff did not have good documentation for the extension of the interval. After searching for documentation, the licensee was able to explain the extension, though the documentation was somewhat sketchy. This illustrates why it benefits a licensee to submit a letter to the USNRC stating that an ISI/IST interval has been extended, explaining the basis for the extension, and listing the new interval dates. This precludes future ISI/IST engineers who were not involved in the extension from having to explain the basis. Extensions that conform with the Code do not require USNRC approval, but the USNRC should be made aware of extensions for planning purposes.

Questions have arisen about the design capability of pumps. Although testing on recirculation with flow instrumentation in the recirculation lines will meet the requirements for inservice testing, a periodic higher flow test is desirable to ensure that the design capability of the pump remains acceptable. The new comprehensive pump testing approved by the O&M Code Committee addresses this concern. Although a few systems have demonstrated problems with reductions in the pump capability, it is typical for plants to perform design flow tests for the service water system because of the effects of erosion and corrosion in the system.

Inspections indicate that documentation of certain aspects of inservice testing is not always as good as it should be to ensure that the Code requirements or the guidance of Generic Letter (GL) 89-04 are met. Frequently, when a check valve sampling program of disassembly and inspection is implemented, all items included in the guidance of Position 2 of GL 89-04 are not addressed in an easily identifiable manner. The program document may state that all the guidance is met, but without a description of how it is met. For example, the review to extend the inspection interval to every other outage is not generally documented properly, or the documentation may not cover all the required items, such as a Nuclear Power Reliability Data System (NPRDS) search on the valve model.

LICENSEE EVENT REPORTS

Several Licensee Event Reports relating to pump and valve testing have been received since the last symposium. Most of those that relate to IST are programmatic, while several discuss equipment found in an unacceptable condition. Below are summaries of several of the event reports involving IST.

Programmatic Issues or Personnel Errors

HPCI/RCIC Declared Inoperable Because of Missed Technical Specification Surveillance Requirement. The plant was at 100% power when a problem occurred making the high pressure coolant injection (HPCI) system and the reactor core isolation cooling (RCIC) system inoperable. When transferring the pump suction paths to the torus, the HPCI torus suction check valves were determined to be leaking. The cause of the leakage was determined to be inadequate acceptance criteria in the disassembly and inspection procedure for checking and aligning the valve disk assembly. A major reason that the inadequate valve condition was not detected by IST was the failure to identify and test the closed safety function of the valves. The IST program identified the check valves as having only an open safety function, and the valves had never been tested in the closed direction. Corrective actions included realignment of the valve disks, changes to the IST program, and review of generic acceptance criteria for check valves during disassembly and inspections.

Technical Specifications Not Met Because of Personnel Error. On two occasions, incorrect actions were taken following inservice testing of an auxiliary saltwater pump: (a) the pump was declared operable even though the test results exceeded the required action range of ASME Section XI, and (b) the pump was not placed on an accelerated testing frequency when test results were within the alert range of ASME Section XI. The root causes for both events were determined to be personnel error (cognitive) by plant licensed operations and nonlicensed engineering personnel. An incorrect pump curve was used to evaluate the test results and the reviewer failed to recognize that the pump was in the alert range. Corrective actions included (a) independent verification in the IST pump procedures when data were entered into the surveillance test procedures, (b) training on ASME Section XI requirements and acceptance criteria, and (c) implementation of a Section XI computerized trending program.

Valve Out of Position and Inoperable Equipment. With the plant at 100% power, apparently during an inservice test of an air-operated valve, operators determined that a nitrogen system valve was closed when it was expected (and required) to be open. The closed valve isolated motive gas to an air-operated auxiliary component cooling water (ACCW) valve such that it could not automatically perform its required function. Approximately one month earlier, the nitrogen valve was tagged out to repair a minor packing leak identified during earlier testing. According to the maintenance procedure, the valve should have been stroked several times and then left in the open position. It was left in the closed position, preventing proper operation of the ACCW valve. The nitrogen valve was opened and the ACCW valve successfully passed the IST. The USNRC commented that "this is an example of inadequate postmaintenance testing in that the

maintenance procedure should have required the ACCW valve to be stroked following work on the air (nitrogen) supply to the valve."

Missed Technical Specification Surveillance. Three safety-related components were determined to be within the scope of the IST program, but were not tested as required prior to entry into Mode 4 at the completion of the second refueling outage. The components were (a) two check valves in the instrument air piping that isolate the nonsafety compressed air system from the backup air cylinders for the primary component cooling water temperature control valves and temperature control bypass valves and (b) one of four motor-operated isolation valves for the safety injection accumulators. The check valves and the motor-operated valves were tested with satisfactory results. Several programmatic issues were to be reviewed to address the cause of the missed surveillance testing.

Isolation of Safety Injection Pump Flow Path During IST of Minimum Flow Recirculation Line Isolation Valves. It was discovered that a quarterly safety injection (SI) valve test could lead to isolation of all available flow paths for the SI pumps. Closing isolation valves in the SI and containment spray minimum flow recirculation lines places the plant in a condition where pump damage could occur if the SI pumps automatically started while reactor coolant system pressure was greater than pump shutoff head and either of the isolation valves remained closed. Operating the SI pumps at shutoff head would cause pump damage after approximately one minute. A probabilistic risk assessment (PRA) was subsequently performed. The PRA determined that the probability of this event occurring is approximately 1.0 E–6 events/year, or an increased pump damage risk of approximately 2%. Because of the increased risk of damaging the SI pumps by testing the isolation valves quarterly, the tests will now be performed on a cold shutdown frequency.

Failure to Track Repair Activities Resulted in a Technical Specification Action Requirement Time Limit Being Exceeded. While the plant was at 100% power, work commenced to adjust the packing on a normally closed containment isolation valve. A stroke time test was to be performed before the valve could be returned to service. The valve was required to be returned to operable status or the affected penetration was to be isolated within 4 hours to meet technical specification (TS) requirements. The packing adjustment was completed within 4 hours, but the stroke time test was not performed nor were the compensatory measures taken. As a result, a TS noncompliance occurred. During the test the valve performed satisfactorily. The cause was inappropriate action by the control room supervisor because he failed to use available administrative procedures to ensure that the maintenance complied with TS requirements. Corrective actions included additional training for operations and maintenance personnel.

The "A" Loop of the Emergency Service Water System Was Inoperable as a Result of an Equipment Failure and Personnel Errors. A check valve that isolates the nonsafety-related service water system from the safety-related emergency service water system failed its quarterly inservice surveillance tests, but the operators failed to declare the valve inoperable. The evening shift performed the surveillance test and turned over the test procedure to the midnight shift for review. The procedure was signed off 2 days later as "unsatisfactory." The following day, during a review of the procedure, an operator identified that the check valve had failed and actions were taken to declare the valve inoperable. The cause was (a) less than adequate training on IST procedure steps, (b) a misleading notation on the procedure made by the nonlicensed operator who performed the test, and (c) a failure to initiate corrective action or verify that corrective actions were initiated.

Radioactive Waste Disposal System Component Cooling Water Isolation Valves Outside Design Basis. Component cooling water (CCW) isolation valves were discovered to be incapable of performing their intended function because they could not satisfy the specified leakage criteria and could not be operated from

the control room. These valves isolate CCW from the radioactive waste disposal system. The valves are air-operated butterfly valves that had been added to the third 10-year interval IST program. Short-term actions were taken to maintain an isolation boundary until the valves could be replaced with leaktight valves with control room remote operation and position indication, as well as adding a containment isolation signal coupled with a CCW radiation monitor signal to close the valves. The cause was inadequate design review during the original installation of the system between 1970 and 1973.

IST Program Surveillance Deficiency. During a review of IST data sheets, IST engineers discovered that several pump discharge check valves were not being tested as indicated in the latest revision of the IST program. Because the tests had not been performed, the valves were declared inoperable. A temporary waiver of compliance was requested, but the testing was actually completed before the limiting condition of operation time expired. The revision of the IST program had necessitated a number of procedure revisions. The IST engineer had no formal process for verifying that the procedure changes were made. Later, a review of test data included a comparison of data with the acceptance criteria; however, this comparison for check valve backflow testing did not specifically verify that the test requirements were met. Several actions were taken to improve the administrative process for procedure revisions, test implementation, and data review.

Technical Specification Surveillance Not Performed Within Required 30 Days Because of Personnel Error. IST for a containment isolation valve was incorrectly scheduled when the "finish date" was incorrectly entered into the computer schedule as the "start date." When this error was discovered, the valve was stroked and verified operable. A site quality verification effectiveness review of the surveillance program was initiated to determine the cause and actions to be taken to prevent recurrence.

Violation of Technical Specifications Because of a Failure to Adequately Test the Valves Connecting the Emergency City Water Supply to the Charging Water Pump Coolers. During a cold shutdown, it was determined that the inservice test was inadequate for valves associated with the emergency city water supply to the charging water pump coolers. The test procedure was revised, and the test was performed before startup.

Check Valves Not Included in IST Program. USNRC inspectors identified four containment spray check valves installed in 1980 that had not been added to the IST program. The engineer responsible for the installation and the managers reviewing the installation were not familiar with the requirements for IST. The valves were added to the IST program and will be disassembled and inspected during the 1994 refueling outage.

Missed IST Because of Less Than Adequate Review of Work Documents. While performing a review of work documents, the IST coordinator noted that the required IST for two spent fuel pool valves had not been performed during the required time interval. The cause of the event was determined to be less than adequate review of work documents when credit was taken for postwork testing as a surveillance test. Corrective actions included performance of the IST and a reinforcement of expectations with cognizant individuals.

HPCI Pump Inoperable Because of IST Low Flow Rate. During the HPCI pump IST, the pump achieved approximately 4,890 gpm at 1,255 psig and 3,700 rpm. The acceptable range for the pump was between 5,123 and 5,559 gpm at 1,255 psig pump discharge pressure (the Code requires differential pressure) and 3,800 rpm. The pump was declared inoperable. The cause was determined to be an inadequate new procedure and a problem with the flow transmitter.

IST on Containment Isolation Valve Missed Because of Personnel Error and Management Deficiency. A stroke-time test of a valve was performed June 11, 1993. As the IST coordinator reviewed the tests on July 1, he initiated a request to increase the test frequency because the stroke time had increased. The responsible department was incorrectly identified on the request, and the test was not performed by the required date. When the condition was recognized on July 20, the test was performed. The causes were associated with processing the test results and corrective actions.

Remote Position Indication Verification Tests of SIS Accumulator Valves Not Performed in Accordance with TS 4.0.5. During a refueling outage, it was discovered that the position indication verification test for the SIS accumulator valves had not been performed within 2 years as required by the ASME Code, Section XI. The valves had been added to the IST program, but permanent test procedures had not been issued for position indication verification. The cause of the event was that changes to the IST plan were not being reflected in associated documents such as test procedures. The position indication verification was performed, the IST plan and procedures were reviewed, and the expectations of personnel were conveyed.

Inadequate IST of Service Water System Check Valves Because of Flow Instrument Calibration Error and Personnel Error in Specifying System Flow Requirement. As a result of determining that the conversion of service water flow rate measured by an annular flow monitoring device was incorrect, it was discovered that IST of the service water pump discharge check valves had not been performed in accordance with the guidance in GL 89-04. The valves are flow tested at maximum accident flow, but because of improper conversion of the annular, the flow was 8.5% less than maximum accident flow. Additionally, the test procedure did not specify the maximum accident flow rate. The procedure was revised, and the test was performed prior to startup.

TS 3.0.4 Not Met Because of Inadequate IST of SI Pump Discharge Check Valves. In two instances in 1984 and 1985, TS 3.0.4 (restrictions on changing plant operating modes) was not met when Mode 3 was initially entered without reverse flow testing the safety injection pump discharge check valves. In 1993, a review group determined that the testing program for the valves was inadequate to verify their safety function for reverse flow closure in accordance with IST requirements. Among the root causes of this event were deficiencies in the scope of the IST program plan review, miscommunication, and personnel error (cognitive). The valves were tested and added to the IST program. Other check valves were to be reviewed to determine the adequacy of the IST.

Residual Heat Removal Service Water (RHRSW) Pump Was Not Declared Inoperable as Required by the IST Program. The "A" RHRSW pump was not declared inoperable when the differential pressure was found in the required action range and the applicable TS action statement was not entered. A review indicated that the pump was able to provide the required cooling water flow and therefore the system was operable. The root cause of the event was attributed to cognitive personnel error on the part of the shift engineer who evaluated the surveillance data. The lack of an independent review was identified as a contributing cause.

Improperly Prepared Relief Request for IST Program Results in Noncompliance With Main Steam Isolation Valve (MSIV) Testing Requirement. A conflict between the implementation of the IST program and the relief request for the frequency of MSIV testing was found. The relief request (which was actually a cold shutdown justification) stated that the MSIVs would be tested *every* cold shutdown. The tests were actually scheduled such that if the MSIVs had been tested within the previous 92 days, testing was not required during a cold shutdown, as allowed by the ASME Code, Section XI. The root cause was personnel error when the changed testing frequency was

incorrectly stated in the relief request as *every* cold shutdown.

Discovery That Certain Valves Should be Subject to ASME Code Section XI Testing. During a design basis reconstitution effort, certain feedwater valves that are required to mitigate the consequences of an accident were not included in the IST program. The available data were evaluated for each of the valves to determine their operability. The valves were added to the IST program.

Inadequate Surveillance Testing of High Pressure Safety Injection (HPSI) Check Valves. At plant operating conditions of 100% power, two HPSI pump discharge check valves were declared inoperable because of inadequate IST. The IST program specified a quarterly partial-stroke test of the two valves, which was not performed. The cause was personnel error in that the surveillance procedure was changed based on verbal commitment to submit a relief request to delete the partial-stroke test.

Containment Isolation Valve Inoperable and TS Action Was Not Taken Because of Personnel Error. The stroke-time test of a containment isolation valve was not performed in the required time to meet IST requirements. The condition was not identified for 5 days and 74 minutes, at which time the valve was declared inoperable and tested. The cause was personnel error in that the shift technical advisor incorrectly cancelled the test procedure from the surveillance testing work order.

Inconclusive Check Valve Testing Methodology. A check valve testing methodology led to inconclusive results. The test relied on determining check valve closure by verifying system flow rate while opening an upstream vent valve, which was subject to blockage. With a blocked vent valve, there would be no true indication that the check valve had seated on reverse flow as required. The check valves and vent valves are located in a domestic water system that is susceptible to material deposits and blockage. Other valve testing procedures were reviewed for adequacy.

Missed IST. During a review of outage work orders, operations personnel discovered that required postmaintenance retest activities were not performed for several valves repaired during the outage. The retest activities specified on the work order did not include all the procedures required to meet the IST requirements. Each of the valves was stroked to verify full travel, and remote position indication was verified. Corrective actions included (a) establishing new recurring tasks when valve testing frequency changes are required for individual components testing and (b) verifying that specified retest activities include all components being returned to service.

Failure to Satisfy IST Requirements. During the performance of the quarterly IST of the raw water pump, test data indicated the pump was in the low alert range. The test frequency was required to be changed to once every 6 weeks until the cause of the deviation was determined and the condition corrected. Maintenance was performed approximately 3 months later. When the postmaintenance test results were reviewed, it was recognized that the increased testing had not been performed as required. The root cause was inadequate administrative controls to ensure adjustment of the test frequency until corrective maintenance is completed.

Missed Pump Surveillance Because of Personnel Error in Evaluation of Pump Vibration Data. A review of the current IST pump vibration computerized database identified 26 instances where data discrepancies existed with the reactor at 100% power. Root mean square (RMS) values were recorded, whereas the IST requires peak values to be recorded and evaluated. Four of the instances were determined to be in the alert range for vibration levels, which would have required the test to be performed more frequently. Later tests indicated results in the acceptable range. The vibration analysis system consists of field data logger instruments and a computer software package. The field data logger stores the pump vibration spectra for later downloading and analysis using the software and

also displays the overall vibration level. The overall vibration velocity levels may be displayed in either inches-per-second peak or inches-per-second RMS. The cause was determined to be the incorrect selection of the display to RMS rather than peak. Corrective actions included defeating the ability to change the units on the data logger by placing a plexiglass cover over the keypad. A contributing cause was the failure to assign a single individual the responsibility for coordination and implementation of the IST pump vibration program, which resulted in many persons performing the comparison of the IST vibration results and a higher potential for errors.

Leak Rate Testing on Four Containment Isolation Valves (CIVs) Not Performed per TS Surveillance Requirements. During the fourth refueling outage, it was determined that 18 CIVs had not been leak rate tested under either 10 CFR 50, Appendix J, or the IST program since the first refueling outage. The cause of the failure to leak test was two fold: (a) the TS exemption to Type C testing was unnecessary and unclear and (b) after the valves were removed from the IST program, the valves were to be added to the Appendix J leak rate testing program. The valves were tested in the nonaccident direction and an integrated leak rate test was performed. Future actions will include bench testing in the accident direction.

Equipment Problems

Flow Transmitter Out of Calibration Causing Pump Flow Measurement to be Below TS Requirements. During reactor core isolation cooling (RCIC) pump IST, the flow transmitter indicated a value that exceeded the acceptance criteria for pump flow. The transmitter produces a signal proportional to the RCIC pump discharge flow rate that is used to control the RCIC turbine speed to maintain the preset injection flow rate. The transmitter was recalibrated and the pump verified operable.

Degraded Service Water Pump Found During Flow Rate Testing. In refueling outage conditions, service water flow tests were conducted following installation and functional testing of new flow instrumentation. When the data were analyzed, it was determined that one of the three service water pumps was degraded. The pump was declared inoperable and scheduled for a maintenance overhaul.

Potential Radiological Release Resulting from Small Break Loss-of-Coolant Accident. Westinghouse Nuclear Safety Advisory Letter identified a scenario in which the volume control tank (VCT) outlet check valve could become a potential leak path outside of containment during a 2 to 8-inch small break loss-of-coolant accident (SBLOCA). During post-SBLOCA recirculation, the VCT outlet isolation valves would be closed to isolate the VCT from recirculation backflow; however, piping from the seal water heat exchanger (SWHX), upstream of the VCT outlet check valve, cannot be isolated by the VCT outlet isolation valves. Backleakage could pressurize the SWHX and associated piping, lifting the relief valve and allowing post-SBLOCA recirculation flow to the VCT and beyond. The amount of containment sump water discharged to the VCT would cause Part 100 limits to be exceeded if a complete core failure source term was postulated with a 10% release of iodine activity, evaporated to the atmosphere. The safety function of the VCT outlet check valve had not been recognized in the design of the plant. The VCT outlet isolation valves upstream of the check valve were considered to be the flowpath boundary valves, and the function of the outlet check valve was considered to be only for isolation from another closed piping system. Emergency operating procedures were revised to diagnose the problem based on increasing VCT level, at which time the operators would isolate the flow path.

Normal and Emergency Service Water Corrosion, Silting, and Check Valve Problems. During the 1990 refueling outage, 61 check valves in the normal and emergency service water systems were opened and visually inspected. Of the 61 valves, 37 were in the IST program and 24 were in the maintenance program (PM). Twenty IST and 10 PM valves were initially declared inoperable based on the

acceptance criteria. The IST check valves were later shown to be operable by actual flow test or calculation. Piping inspections revealed 10 to 30% blockage in 40% of the piping, and tube plugging blocked 25% of the air handling units.

Steam Generator Blowdown Sample Valve Fails IST Stroke Timing Test. A management review of a performance test indicated a blowdown sampling valve had exceeded its closing time acceptance criteria while the reactor was operating at 100% power. Untimely review of the data resulted in violating a TS action statement. The event occurred because of personnel error in not recognizing the closure time as being outside the acceptance criteria.

Failure of Pressure Isolation Valve to Seat Tightly During Plant Operation Because of Looseness of the Hinge Mechanism. During a refueling outage, it was discovered that a pressure isolation valve in the decay heat removal system could have exceeded its leakage limits during the operating cycle. The hinge pins were found to have play (freedom of movement) such that the disk did not consistently seat tightly. Repairs were made. Even if the check valve had failed to close, the first check valve in series would have prevented leakage of the primary coolant into the decay heat removal system.

Containment Spray Isolation Valve Failed to Stroke Open. An air-operated, spring to open, containment spray header isolation valve would not open following a relay surveillance conducted after IST of the valve. A high differential pressure existed across the valve after a containment spray pump run. The failure to stroke appears to have been caused by a combination of valve degradation and high differential pressure across the valve. The high differential pressure was created by a pressure surge on pump start and pockets of entrained air in the piping. After the pressure bled down, the valve stroked satisfactorily. Additional vent valves were installed in the piping, the valve was maintained in the open position, and the redundant train was assessed for similar problems. The valve had passed IST just prior to the event. The valve actuator was not sized for the differential pressure that occurred.

High Pressure Coolant Injection (HPCI) System Inoperable Because of Torque Switch Roll Pin Failure on the Injection Valve Motor Operator. A quarterly IST valve exercise was being performed to test the discharge of the HPCI valve to core spray. The valve did not stroke full open under test conditions. A signal to override the test open circuit was initiated by the operator and the valve fully opened. It traveled to the full-closed limit on a closure signal; however, shortly after reaching the full-closed position, the valve operator thermal overload protection device tripped. The root cause was identified as a failed roll pin in the motor operator torque switch assembly. The torque switch assembly was replaced and the valve operated properly.

Sticking of Main Steam Isolation Valve (MSIV) Combined with Improper Disposition of Surveillance Test. During cold shutdown troubleshooting of an inboard MSIV, the valve failed to stroke closed when signaled. It was determined that the valve had been inoperable for a period exceeding the TS action statement. An earlier partial stroke IST of the valve had indicated improper operation, but it was not recognized at the time of the test. The valve plug was bound due to improper tolerances. Repairs were effected and the surveillance procedures were changed.

Reactor Core Isolation Cooling (RCIC) System Declared Inoperable During Surveillance Testing Because of Valve Position Indication and Overload Alarm. During a routine surveillance test of the RCIC system, the RCIC turbine steam supply motor-operated valve overload alarm occurred. The cause was galling of the valve stem and bonnet chamber. The galling resulted in binding during opening of the valve and failure of the valve operator motor. The most probable cause of the galling was minimal clearance between the valve stem and bonnet chamber in combination with thermal expansion. The stem and bonnet chamber

were machined and the operator motor was replaced.

High-Pressure Coolant Injection (HPCI) System Inoperable Because of Turbine Stop Valve Failure to Close Within Required Stroke Time Limit. A plant startup was in progress and the plant was at 14% power. IST had been performed on the HPCI system. Test data indicated that the less-than-or-equal-to-2-second time interval for the HPCI turbine stop valve was exceeded. The cause of the increased stroke time was determined to be a buildup of hydraulic fluid residue in the pilot-actuated solenoid valve and ports. The solenoid valve was cleaned and the valve stroked within 2 seconds.

Appendix A

NUMARC Comments on NUREG-1482

NUCLEAR MANAGEMENT AND RESOURCES COUNCIL

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USNRC
'94 MAR 10 P3:34

Thomas E. Tipton
Vice President & Director
Operations, Management and
Support Services Division

March 10, 1994

Mr. David L. Meyer, Chief
Rules, Directives and Review Branch
U.S. Nuclear Regulatory Commission
Washington, DC 20555

SUBJECT: Proposed Supplement 1 to Generic Letter 89-04, "Guidance on Developing Acceptable Inservice Testing Programs," 58 Fed. Reg. 95738 (December 16, 1993), Opportunity for Public Comments

Dear Mr. Meyer:

These comments are submitted on behalf of the Nuclear Management and Resources Council (NUMARC)[1] in response to the subject Federal Register notice soliciting comments on the NRC draft NUREG-1482, "Guidelines for Inservice Testing Programs at Nuclear Power Plants." It is our understanding that the draft NUREG was developed to provide clarification of NRC positions regarding inservice testing (IST) described in GL 89-04 and issues described in other NRC documents (i.e., Information Notices, letters, etc.). As such, our comments are directed toward the draft NUREG and other interfacing documents and regulations effecting inservice testing and how those interfaces can be improved to assure the appropriate implementation of inservice testing programs. In addition to the general comments provided below, the review of draft NUREG-1482 has produced some specific comments related to details in the various sections of the document. Those comments are provided in the enclosure and are noted with the specific section of the draft NUREG that is affected.

[1]NUMARC is the organization of the nuclear power industry that is responsible for coordinating the combined efforts of all utilities licensed by the NRC to construct or operate nuclear power plants, and of other nuclear industry organizations, in all matters involving generic regulatory policy issues and on the regulatory aspects of generic operational and technical issues affecting the nuclear power industry. Every utility responsible for constructing or operating a commercial nuclear power plant in the United States is a member of NUMARC. In addition, NUMARC's members include major architect/engineering firms and all of the major nuclear steam supply system vendors.

As stated in the Federal Register notice, the purpose of the draft NUREG is to focus on how a utility can handle non-conforming situations and preparation of relief requests. Our review has found that the document is generally sound in these areas and provides suitable guidance. Licensees can use the document based on how they have implemented their IST program and the cost benefits they will receive from the staff guidance. Additionally, the draft NUREG contains appropriate information regarding staff determinations that they find acceptable in providing relief from the ASME code. It is encouraging to note also that the reason most often cited in providing relief is to not impose an undue burden on the licensee.

Regarding the issue of not imposing undue burden on licensees, there are several areas that warrant discussion where the draft NUREG could be improved or other regulations could be addressed in an effort to further reduce the unnecessary burden on licensees and improve the IST programs without impacting safety.

Most, if not all, of the components covered by IST programs are safety-related. This means that they will be included within the scope of the Maintenance Rule (10CFR50.65) when it becomes effective on July 10, 1996. The Maintenance Rule and its accompanying and implementing Regulatory Guide (RG 1.160) are performance based. This means that the requirements of the prescriptive IST program and the performance bases of the Maintenance Rule will create additional need for relief or exemption from regulations. The draft NUREG as presently written makes no mention of the Maintenance Rule. Generic Letter 89-04 and the proposed draft NUREG of Supplement 1 along with the ASME IST code are very prescriptive and in many cases are contradictive between technical specifications, other ASME code requirements, and the Maintenance Rule. Additionally, the NRC program to revise Appendix J to make it more performance based will serve to further compound problems related to IST of valves in the containment isolation systems. We recommend that the NRC, ASME, and the industry strive to make the regulations, implementing guidance and applicable standards compatible to avoid future problems.

While the purpose of the draft NUREG in providing guidance for relief has been achieved, we would propose that serious consideration be given to providing guidance on the transition between the various regulations involved. For example, on page 2-3 of the draft NUREG, the last paragraph before Section 2.2 states, "The technical specifications for most plants include IST requirements which are more restrictive than the regulations." An effort to reduce the conflicting requirements between the regulations, codes and standards, technical specifications, etc., will reduce the burden on licensees by removing

Mr. David L. Meyer
March 10, 1994
Page 3

the multiple layers of regulations and requirements that constantly require interpretation, relief, or exemption requests.

With the many activities underway both within the NRC and the industry to consider performance-based concepts utilizing PRA, for Maintenance, Appendix J, QA, MOVs, etc., to prioritize the effected structures, systems, and components, we recommend that the NRC also consider these concepts when developing IST guidance.

IST activities are an important element in operating our plants in a safe and efficient manner; however, we believe that this is a prime area where changes can be made to reduce the unnecessary burden on licensees without impacting safety. We would be happy to meet with you and discuss the potential activities that could lead to improvement in the industry IST programs through the application of performance-based IST.

Should there be any questions regarding this letter or our comments, please call Warren Hall, Clive Callaway or me.

Sincerely,

Thomas E. Tipton

Thomas E. Tipton

TET/WJH:plg
Enclosure

Draft NUREG-1482, "Guidelines for Inservice Testing at Nuclear Power Plants"

The following are specific comments concerning staff positions in various sections of draft NUREG–1482.

General

Guidance on the appropriate action to be followed while a relief request is being processed is found in three locations, and it is confusing and possibly conflicting.

The guidance is contained in Sections 2.5, "Relief Requests and Proposed Alternatives," Section 6.3, "Revised Standard Technical Specifications," and Section 7, "Identification of Code Noncompliance." Section 2.5 states "For those requirements, which have been determined to be clearly impractical, the licensee may implement the proposed alternative testing while the NRC is reviewing the relief request (See Section 6),: and Section 6.3 discusses using 10CFR50.59 as a mechanism for continued operation when a Code requirement is impractical, and a Code relief request is being processed, providing T/S 4.0.5 has been revised as recommended in the draft NUREG-1482. Section 7, on the other hand, discusses the need to obtain a Temporary Waiver of Compliance when a Code noncompliance exists and NRC approval of a relief request cannot be obtained within the allowable T/S action time period.

The NRC position should be clearly delineated and placed in a single, separately identified section of the draft NUREG-1482.

§ 2.2 - Criteria for Selecting Pumps and Valves for the IST Program

Draft NUREG-1482 acknowledges that the scope of the OM Standards and Code has been expanded to include all safety-related pumps and valves in the IST Program. Until 10CFR50.55a is changed, the scope of the IST Program will continue to include those components within the Code Classes. However, the NRC stated that they would consider expanding the scope of the IST Program to include all safety-related pumps and valves.

The current scope of Code Classes is adequate. Safety-related valves and pumps outside the Code Class boundaries are addressed by other measures such as the plant specific technical specifications. Maintenance Rule and the industry post-maintenance testing programs. Including all safety-related components in the IST increases the regulatory burden without a corresponding increase in overall safety.

Draft NUREG-1482 indicates the NRC is considering future rulemaking to expand the scope of 10CFR50.55(a) to include all safety-related pumps and valves, irrespective of whether the components are Code classed or not.

Mandatory imposition of ASME inservice test requirements on non-code classed components constitutes a backfit. Many of the components were not designed with the necessary provisions to perform testing (flow instruments, gages, etc.), modification would be necessary to perform tests in compliance with Code requirements.

The industry could on a plant-by-plant basis identify the components important-to-safety which are not Code classed. Secondly, history, current testing and preventative/predictive maintenance schedules could be reviewed for adequacy and revised as appropriate. Again, this approach is in line with implementing the Maintenance Rule. The intent would be to provide documented justification of these practices providing assurance of availability. The other extreme could require a similar review, generate reliefs and probably ultimately result in unnecessary plant modifications

§ 3.1.1, Deferring Valve Testing to Each Cold Shutdown or Refueling Outage

The staff has made four recommendations concerning various issues when deferring valve testing to cold shutdown or refueling conditions based upon easing the burden on the licensee and the equipment without adversely impacting safety. These recommendations are good examples of many of the recommendations contained in draft NUREG–1482.

§ 3.1.1.1, IST Cold Shutdown Testing and § 3.1.1.2 Testing at a Refueling Outage Frequency for Valves Tested During Power Ascension

The draft NUREG-1482 states that although the requirements are as stated in the ASME Operations and Maintenance (O&M) Standards Manual and the licensee is complying with the Code, "if a licensee chooses to implement this guidance, this section (of the draft NUREG-1482) must be explicitly referenced in the IST program".

The licensee should not have to state in the IST program the methods of ensuring Code compliance, if the licensee is complying with the Code.

§ 3.1.2, Entry into a Limiting Condition for Operation to Perform Testing

Draft NUREG-1482 takes the position that licensees should perform required quarterly testing, even if entry into a Limiting Condition for Operation (LCO) is required. Relief requests to defer testing must contain additional justification in addition to the entry into the LCO. This is a rather inflexible position, and may unduly restrict prudent or cost–effective scheduling.

§ 4 Supplemental Guidance on Inservice Testing of Valves.

Draft NUREG-1482 should provide guidance on whether IWV-1100 is intended to apply to small sentinel valves installed for thermal relief of service water heat exchangers in the uncommon event they are isolated from service.

Draft NUREG-1482 indicates that the 1986 Edition of ASME, Section XI, expands the scope of pressure relief under IWV-1100. This interpretation is debatable and clearly does not provide an increase in safety.

§ 4.1.2, Exercising Check Valves with Flow "NRC Recommendation"

Draft NUREG-1482 states "the NRC determined that use of non-intrusive techniques is acceptable to verify the full stroke of a check valve, although the flow rate must be sufficient to stroke the valve to the backstop."

This statement could be interpreted to mean that non-intrusive examination results are unacceptable unless an acoustic impact is obtained (from the disc against the backstop). Some check valves were sized to accommodate large flows, which may be expected with various accident conditions. In addition, many valves were sized such that pressure drop across the valve is minimal, i.e., the valves are larger than needed to pass the design flow. As a result, in many valves, the disc will not impact the backstop even though full design flow is passed.

Some check valve non-intrusive examination (NIE) equipment employs two technologies: acoustics plus either external magnetics or ultrasonics (UT). The draft NUREG seems to be based on acoustics only. This limits the usefulness of the plant's NIE system since credit for UT or magnetic data cannot be taken. For example, if full design flow is verified through a valve but no impact is detected by the acoustic sensor, then credit cannot be taken for the NIE test. This appears to be true even if UT data shows the valve disc to be 90% open.

§ 4.2.1 Increased Frequency of testing for Valves That Can Be Tested Only During Cold Shutdown

Draft NUREG-1482 states "...OM-10 requires corrective action if a limiting stroke time is exceeded and does not allow for an increased test frequency." While OM-10 does not specifically state that frequency shall be increased, neither does OM-10 prohibit an increase in test frequency as stated in Section 4.2.1.9, OM-10, Part 10. The increase in test frequency may be done as an additional corrective action or as a temporary corrective action. This section seems to be applicable to those times when testing frequency is not able to be increased due to plant conditions (i.e., plant is operating). Clarification is needed to address this issue.

§ 4.2.7 Stroke Time Measurements Using Reference Valves

Refers to Figure 4.2 being a sample relief request for the use of stroke time reference. The location of Figure 4.2 is not apparent.

§ 4.3.1 Safety and Relief Valves

Draft NUREG-1482 requires that pressure relief valves which are installed in the applicable system to protect against overpressure be tested, even though they may not typically perform a "safety-related" function.

The classification methodology for active components is taken from Regulatory Guide 1.48 and includes those pumps, valves and pressure relief devices "that must perform a mechanical motion during the course of accomplishing a system safety function." System safety functions are defined to include any function that is necessary to assure (1) the integrity of the reactor coolant pressure boundary, (2) the capability to shutdown the reactor and maintain it in a safe shutdown condition, or (3) the capability to prevent or mitigate the consequences of accidents which could result in offsite exposures comparable to the guideline exposures of 10CFR100. This active component classification methodology and the wording of the scope statements of ASME OM (Parts 1, 6 and 10) are considered to mean the same thing (i.e. pumps, valves and pressure relief devices performing nuclear safety functions).

The passive valves in the IST program are identified through individual system reviews. The passive valves include those valves which are required to perform a nuclear safety function (as defined above) by maintaining their position and for which ASME OM Part 10 specifies leakage testing or position indicator testing requirements.

ASME OM Part 1 provides requirements for pressure relief devices which are required to perform a specific function in shutting down a reactor or in mitigating the consequences of an accident. Consistent with ASME OM-1987 Interpretation No. 1-2 and the active component classification methodology described above, the IST program scope should include those pressure relief devices which themselves perform an active nuclear safety function.

§ 4.4.6 Manual Valves

Draft NUREG-1482 clarifies that exercise requirements for manual valves be "...in accord with applicable IST requirements of IWV or OM-10 if the manual valve is credited in the safety analysis for being capable of being repositioned to shut down the plant, to maintain the plant in a safe shutdown condition, or to mitigate the consequences of an accident."

However, no direction or relief action is given.

Additional guidance should be provided on relief actions/frequency justifications (e.g., stroke time not required on manual valves and valve position verification for manual valves).

§ 5 Supplemental Guidance on Inservice Testing of Pumps

OM-6 separates all positive displacement pumps hydraulically from centrifugal pumps, however, the mechanical vibration surveillance criteria therein recognizes only the difference between reciprocating pumps and centrifugal pumps. No section within the draft NUREG addresses the vibration performance characteristics of other types of positive displacement pumps (i.e., gear, screw, etc.). Must licensees cap the alert and action ranges for these types of pumps at 0.325 and 0.70 inches/seconds, respectively, or can 2.5 and 6 times the reference values be unlimited? Do utilities have the latitude for interpretation via relief requests?

OM 6, Table 6100-1 has a tighter acceptance band for "vertical line shaft pumps" (.93 vs. 90). The basis for this tighter band should be explained. No apparent increase in safety margin is obtained when such pumps are analyzed for 10% or greater degradation.

In appears that the ASME OM Code-1990, Part 6 (OM-6) eliminates the option to perform an operability analysis to determine if a pump exceeding the action limit can

still perform its safety function, reference ASME Section XI, 1983 edition, paragraph IWP-3230, "Corrective Action" (c). This option allowed the continued operation of pumps that have a large margin between minimum system flow/DP requirements, and the 10% degraded limit. That is, once these pumps hit the low action limit, a significant margin existed between that limit and the minimum operability limit required of that pump. This offers considerable economic and scheduling advantages without impacting the safety-related function of the pump. OM-6 should retain that alternative, or the NUREG should allow for such analysis. Otherwise to capitalize on such margin, code relief requests for extended allowable ranges will be required.

§ 5.4 Monitoring Pump Vibration in Accord with OM-6

The NRC discusses pump vibration monitoring and the various changes from IWP to OM-6. However, the vibration acceptance criteria in OM-6 are not clearly stated, (i.e., $2V_R$ to $6V_R$ <.325>.7), nor is it clear if these values are for full flow testing or minimum flow testing, where vibration levels typically increase due to flow noise.

§ 5.5 Pump Flow Rate and Differential Pressure Instrumentation

Instrumentation range and accuracy is discussed using both analog and digital instrumentation. It is unclear whether computer points or printouts can be used to meet the necessary instrumentation requirements and if additional requirements are associated with the use of the computer

§ 5.5.1 - Range and Accuracy of Analog Instruments

In safety evaluation reports at some utilities the NRC has stated that analog instruments which are ±2% of full-scale are in effect ±6% at 1/3 of range. Page 5-7 of the draft NUREG is implying that analog instruments are required to be ±2% at the reference values (2% at 1/3 of range). This requirement needs to be clarified; the draft NUREG seems to be in conflict with past positions

§6.3 Discussion

Revised standard technical specifications (page 6-3) discusses test frequencies and surveillance requirements for IST. Tolerances/grace periods such as ±25% of due date should be specified. Any tolerance that is applicable during "increased frequencies" should be stated.

§ 7 Identification of Code Noncompliance

It is suggested to add guidance in this section to address certain other additions of components to a plant IST Program, e.g., when the components formerly did not clearly fall within the scope of Code requirements, but the licensee has elected to add the component to the IST Program because of a modification, revised interpretation, or philosophy change. For program additions in this category, engineering analysis or other types of testing could be used in lieu of Section XI testing to justify operability of the components before addition to the IST Program; subsequently, operability would be determined through normally scheduled testing. In other words, the licensee should be able to assume operability for certain categories of components newly added to the IST without invoking the guidance in GL 91-18 for a grace period until the next scheduled IST program testing is completed.

The staff discussion on noncompliance situations discusses what the staff believes should be done when a licensee identifies components that should be in the IST program but are not. In most cases, if testing is required, the component would be in violation of TS. 4.0.5 or equivalent, and should be declared inoperable. The staff goes halfway, and states that simply failing to perform an IST test should not cause a forced shutdown, but believes that a Temporary Waiver of Compliance or other exigent relief from ASME code requirements is necessary. Exigent relief from ASME code requirements would be required, but a Temporary Waiver of Compliance should not be necessary if, consistent with the discussion in the NUREG, other tests or operational performance data reviews show that the system is operable. This would also be consistent with the comments on GL–91–18.

Appendix A

In Appendix A of draft NUREG-1482, Question 61, the NRC requests that IST programs be re-submitted each time they are revised. The reason given for this request is that ..."it is needed to prepare for IST inspectors and to assist in the review of relief requests." It has been the experience of some utilities that the inspectors and/or revievers do not use the docketed copy of the program for these purposes. Submittal preparation and review becomes a needless administrative exercised with no apparent benefit. In addition, there is substantial review costs involved as was the case for program submitted under the original Generic Letter 89-04.

Appendix B

Review of Beaver Valley Unit 1 Inservice Testing Program

Appendix B

Review of Beaver Valley Unit 1 Inservice Testing Program

Code Edition: 1983 with addenda through Summer 1983

Total Number of Pages: 187

Section I: Pump Testing Requirements

Discusses the application of IWP requirements. Includes trending of pump test results.

Section II: Pump Testing Outlines

Each pump is listed on a single page (see example for 1A charging pump 1CH-P-1A). For each pump, the measured parameter, test procedure, test frequency, and comments are provided with the following information:

Pump Name	Pump Number	Code Class	System
Drawing Number	Drawing Coordinates	Function	Pump Type
Remarks			

Table B-1. Pumps included in the inservice testing program.

Pump identification	Code class	System identification	Function	Type
1A charging pump	2	Chemical and volume control	To provide normal RCS inventory and high head safety injection	Centrifugal
1B charging pump	2	Chemical and volume control	To provide normal RCS inventory and high head safety injection	Centrifugal
1C charging pump	2	Chemical and volume control	To provide normal RCS inventory and high head safety injection	Centrifugal
2A boric acid transfer pump	3	Chemical and volume control	Chemical shim and emergency boration supply	Centrifugal
2B boric acid transfer pump	3	Chemical and volume control	Chemical shim and emergency boration supply	Centrifugal

Table B-1. (continued)

Pump identification	Code class	System identification	Function	Type
1A residual heat removal pump	2	Residual heat removal	Long-term decay heat removal	Vertical
1B residual heat removal pump	2	Residual heat removal	Long-term decay heat removal	Vertical
1A low head safety injection pump	2	Safety injection	Low pressure—high volume safety injection and long term recirculation	Vertical
1B low head safety injection pump	2	Safety injection	Low pressure—high volume safety injection and long term recirculation	Vertical
1A quench spray pump	2	Containment depressurization	To provide a flow of borated water for containment depressurization following a DBA	Centrifugal
1B quench spray pump	2	Containment depressurization	To provide a flow of borated water for containment depressurization following a DBA	Centrifugal
4A chemical injection pump	2	Containment depressurization	Chemical injection during containment depressurization	Positive displacement
4B chemical injection pump	2	Containment depressurization	Chemical injection during containment depressurization	Positive displacement
4C chemical injection pump	2	Containment depressurization	Chemical injection during containment depressurization	Positive displacement
4D chemical injection pump	2	Containment depressurization	Chemical injection during containment depressurization	Positive displacement

Table B-1. (continued)

Pump identification	Code class	System identification	Function	Type
1A inside recirculation spray pump	2	Containment depressurization	Circulate containment sump water for long term containment depressurization	Vertical
1B inside recirculation spray pump	2	Containment depressurization	Circulate containment sump water for long term containment depressurization	Vertical
2A outside recirculation spray pump	2	Containment depressurization	Circulate containment sump water for long term containment depressurization	Vertical
2B outside recirculation spray pump	2	Containment depressurization	Circulate containment sump water for long term containment depressurization	Vertical
1A component cooling water pump	3	Reactor plant component cooling water	To provide cooling water to reactor plant components	Centrifugal
1B component cooling water pump	3	Reactor plant component cooling water	To provide cooling water to reactor plant components	Centrifugal
1C component cooling water pump	3	Reactor plant component cooling water	To provide cooling water to reactor plant components	Centrifugal
Steam driven auxiliary feed pump	3	Auxiliary feedwater	Provide emergency makeup during any loss of normal feedwater	Centrifugal
3A motor driven auxiliary feed pump	3	Auxiliary feedwater	Provide emergency makeup during any loss of normal feedwater	Centrifugal
3B motor driven auxiliary feed pump	3	Auxiliary feedwater	Provide emergency makeup during any loss of normal feedwater	Centrifugal

Table B-1. (continued)

Pump identification	Code class	System identification	Function	Type
1A river water pump	3	River water	To provide a source of water during normal emergency conditions to primary plant heat exchangers and equipment	Vertical
1B river water pump	3	River water	To provide a source of water during normal emergency conditions to primary plant heat exchangers and equipment	Vertical
1C river water pump	3	River water	To provide a source of water during normal emergency conditions to primary plant heat exchangers and equipment	Vertical
1A diesel generator No. 1 fuel transfer pump	3	Station service 4 kV	Transfer fuel from the underground tank to the day tank	Positive displacement
1B diesel generator No. 1 fuel transfer pump	3	Station service 4 kV	Transfer fuel from the underground tank to the day tank	Positive displacement
1C diesel generator No. 2 fuel transfer pump	3	Station service 4 kV	Transfer fuel from the underground tank to the day tank	Positive displacement
1D diesel generator No. 2 fuel transfer pump	3	Station service 4 kV	Transfer fuel from the underground tank to the day tank	Positive displacement

Section III: Pump Testing Relief Requests

This section includes the following relief requests for the pumps in the inservice testing program:

Relief Request 1—For 28 of the 32 pumps, use pump vibration requirements of OM-6 and discontinue measuring bearing temperature.

Relief Request 2—Use static head from tanks or Ohio River elevation to calculate suction pressure for 9 of 32 pumps.

Relief Request 3—For the boric acid transfer pumps, test quarterly through recirculation lines and measure pump differential pressure; test during refueling outages, measure pump differential pressure, and calculate pump flow based on change in tank level over time. Use separate vibration reference and acceptance criteria values for the two different tests.

Relief Request 4—For the residual heat removal pumps, test quarterly when in cold shutdown conditions or refueling outages, but do not test quarterly during power operating conditions because the testing would require personnel entry into the subatmospheric containment.

Relief Request 5—For the positive displacement chemical injection pumps, measure pump discharge pressure and flow rate to evaluate pump performance.

Relief Request 6—For the inside recirculation spray pumps, perform a dry run quarterly for not more than 60 seconds and stop the pumps before they reach 100 rpm. Run on recirculation during refueling outages.

Relief Request 7—For the outside recirculation spray pumps, run dry quarterly for not more than 60 seconds and stop after visually observing an increase in motor amperage and pump shaft rotation. Run on recirculation during refueling outages.

Relief Request 8—For the auxiliary feedwater pumps, test quarterly through fixed resistance recirculation lines while measuring differential pressure. Test during shutdown conditions with flow to the steam generators, measuring differential pressure and flow. Separate vibration reference and acceptance criteria values will be used for the different test conditions.

Relief Request 9—For the diesel fuel oil transfer pumps, measure discharge pressure as an indication of pump performance. No suction pressure or differential pressure instrumentation is installed.

Relief Request 10—For the diesel fuel oil transfer pumps, use level changes of the day tank to calculate flow rate, as no flow instrumentation is installed.

Section IV: Valve Testing Requirements

Discusses the Code requirements for valves in the inservice testing program.

Section V: Valve Testing Outlines

A listing of all valves in the inservice testing program, identifying valve Code class, category, size, type, normal system arrangement (position), drawing number and coordinates, testing requirements, specific cold shutdown justification reference numbers, relief request reference numbers, test procedure numbers, and comments.

Section VI: Cold Shutdown Justifications

Detailed technical description of conditions prohibiting the required testing of valves in the inservice testing program. Includes 31 justifications that cover 93 valves.

Section VII: Valve Relief Requests

Contains detailed technical descriptions of conditions prohibiting the required testing of certain valves in the inservice testing program. Includes 43 relief requests.

Table B-2. Pump Data Sheet—Beaver Valley Power Station, IST Program for Pumps and Valves.

Pump Name: 1A Charging Pump	Pump Number: 1CH-P-1A	Code Class: 2	Dwg. OM No.: 7-1 Dwg. Coord.: C-4	System: No. 7 Chemical and Volume Control
Function: To provide normal RCS inventory and Hi Head Safety Injection	Type: Centrifugal		Remarks: See RR1	

Parameter	1OST (frequency)	Req'd	Comments
N	NA	NA	Constant speed induction motor.
Pi	7.4 (Q)	X	Installed instrumentation or temporary test gauge at pump suction (local).
	11.14 (R)	X	Installed instrumentation or temporary test gauge at pump suction (local).
ΔP	7.4 (Q)	X	ΔP is calculated using the Pump Discharge Pressure Indicator [PI–1CH–151] (local) and Pump Suction Pressure from either the installed instrument or the temporary test gauge (local).
	11.14 (R)	X	ΔP is calculated using the Pump Discharge Pressure Indicator [PI–1CH–151] or temporary test gauge (local) and Pump Suction Pressure from either the installed instrument or the temporary test gauge (local).
Q	7.4 (Q)	X	Summation of flow rates from RCP Seal Injection Flow Indicators [FI–1CH–130], [FI–1CH–127], AND [FI–1CH–124] and Charging Flow Indicator [FI–1CH–122A] or Fill Flow Indicator [FI–1CH–160] and assumed flow through mini flow line.
	11.14 (R)	X	Summation of flow rates from RCP Seal Injection Flow Indicators [FI–1CH–130], [FI–1CH–127], and [FI–1CH–124] and Charging Flow Indicator [FI–1CH–122A] or HHSI to Hot and Cold Log Hdr Flow [FI–1SI–943].
V	7.4 (Q)	RR-1	Portable monitoring equipment using velocity units.
	11.14 (R)	RR-1	Portable monitoring equipment using velocity units.
Tb	NA	RR-1	Annual pump bearing temperature measurement will not be taken since vibration is measured in velocity units.
L	7.4 (Q)	X	Lubricant Oil Filter Pressure Gauge [FI–1CH–161A1] (local). Sightglass on oil reservoir (local).
	11.14 (R)	X	Lubricant Oil Filter Pressure Gauge [FI–1CH–161A1] (local). Sightglass on oil reservoir (local).

Table B-3. Valve Table—Beaver Valley Power Station, IST Program for Pumps and Valves (valve testing outline).

System Name: Reactor Coolant | System Number: 6

Valve mark number	Valve class	Valve category	Valve size (in.)	Valve type	NSA	Drawing OM No.	Drawing Coord.	Test require-ment	CSJ or relief requests	Comments
1RC-68	2	A/C	3/4	Check	—	6-2	B-3	QS	RR2	1BVT 1.47.5-FS, RD by Leak Test (R)
								LT	RR1	1BVT 1.47.5-Leak Test (R)
1RC-72	2	A/C	3	Check	—	6-2	C-3	QS	RR3	1BVT 1.47.5-FS, RD by Leak Test (R)
								LT	RR1	1BVT 1.47.5-Leak Test (R)
IV-1RC-101	2	A	3/4	Globe	S	6-2	B-2	QST	—	1OST-47.3A(3B)-Stroke and Time Closed (Q) (RPV)
								LT	RR1	1BVT 1.47.5-Leak Test (R)
SOV-1RC-102A	1	B	1	Globe	LS	6-2	A-1	QST	CSJ1	1OST-1.10-Stroke and Time Open (CSD) 1OST-6.9-(RPV)
SOV-1RC-102B	1	B	1	Globe	LS	6-2	A-1	QST	CSJ1	1OST-1.10-Stroke and Time Open (CSD) 1OST-6.9-(RPV)
SOV-1RC-103A	1	B	1	Globe	LS	6-2	A-2	QST	CSJ1	1OST-1.10-Stroke and Time Open (CSD) 1OST-6.9-(RPV)
SOV-1RC-103B	1	B	1	Globe	LS	6-2	A-2	QST	CSJ1	1OST-1.10-Stroke and Time Open (CSD) 1OST-6.9-(RPV)
SOV-1RC-104	1	B	1	Globe	LS	6-2	A-3	QST	CSJ1	1OST-1.10-Stroke and Time Open (CSD) 1OST-6.9-(RPV)
SOV-1RC-105	1	B	1	Globe	LS	6-2	B-2	QST	CSJ1	1OST-1.10-Stroke and Time Open (CSD) 1OST-6.9-(RPV)
1RC-277	2	A/P	1/8	Needle	S	6-2	F-10	LT	RR1	1BVT 1.47.5-Leak Test (R)
1RC-278	2	A/P	1/8	Globe	S	6-2	E-10	LT	RR1	1BVT 1.47.5-Leak Test (R)

Table B-3. (continued).

Valve mark number	Valve class	Valve category	Valve size (in.)	Valve type	NSA	OM No.	Coord.	Test require-ment	CSJ or relief requests	Comments
PCV-1RC-455C	1	B	3	Plug	A	6-2	B-10	QST	CSJ2	1OST-6.8-Stroke and Time Open (CSD) (RPV)
PCV-1RC-455D	1	B	3	Plug	A	6-2	C-10	QST	CSJ2	1OST-6.8-Stroke and Time Open (CSD) (RPV)
PCV-1RC-456	1	B	3	Plug	A	6-2	C-10	QST	CSJ2	1OST-1.10-Stroke and Time Open (CSD) (RPV)

Appendix C

Duane Arnold Energy Center
SER Table 1
Summary of Relief Requests

Appendix C
Duane Arnold Energy Center SER Table 1 Summary of Relief Requests

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
PR-005	2.1.1.1	IWP-3100: Pump test procedure requirements.	River water pumps: 1P-117A, B, C, D Core spray pumps: 1P-221A, B RCIC pump: 1P226 RHR pumps: 1P229A, B, C, D	Calculate differential pressure using linear interpolation.	Relief granted (a)(3)(i)
PR-013	2.1.2.1	IWP-3200: Allowable ranges of test quantities.	River water pumps: 1P-117A, B, C, D Diesel fuel oil transfer pumps: 1P-44A, B Standby liquid control pumps: 1P-230A, BRCIC pump: 1P-226	Establish the upper Alert Range at 103% of reference and the upper Required Action Range at 105% of reference.	Relief not required.
PR-001	2.2.1.1	IWP-3100: Vibration measurement requirements.	Diesel fuel oil transfer pumps: 1P-44A, B	Disassemble, inspect, and rebuild these pumps every other refueling outage followed by pump surveillance testing prior to declaring the pump operable.	Provisional relief granted. (g)(6)(i)
PR-015	2.3.1.1	IWP-3100: Pump test procedure requirements.	HPCI pump: 1P-226	Use empirically derived pump curve as reference values over a limited range of pump operation in lieu of varying the system resistance until the independent variable equals the reference value.	Provisional relief granted (a)(3)(i)

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
PR-004	NA	IWP-3100: Suction pressure measurement requirements.	RHR service water pumps: 1P-22A, B, C, D ESW pumps: 1P-99A, B River water pumps: 1P-117A, B, C, D Diesel fuel oil pumps: 1P-44A, B Standby liquid control pumps: 1P-230A, B	Calculate suction pressure.	Preapproved GL 89-04, relief request not evaluated in TER.
PR-007	NA	IWP-4110: Instrument accuracy requirements.	Core spray pumps: 1P-211A, B High pressure coolant injection pump: 1P-216	Measure pressure and speed using instruments with loop accuracies that are less than or equal to +/- 2.26 percent.	Preapproved GL 89-04, relief request not evaluated in TER.
PR-011	NA	IWP-4120: Instrument full-scale range requirements.	All pumps.	Take vibration measurements with instrument range selection at lowest possible scale that includes the measured parameter.	Preapproved GL 89-04, relief request not evaluated in TER.
PR-012	NA	IWP-4120: Instrument full-scale range requirement.	Core spray pumps: 1P-211A, B RHR service water pumps: 1P22A, B, C, D HPCI pump: 1P-216 RCIC pump: 1P-226	Use electronic instruments with accuracies based on actual reading instead of the full-scale range.	Preapproved GL 89-04, relief request not evaluated in TER.
PR-014	NA	IWP-4310: Bearing temperature measurement requirements.	All pumps.	None.	Preapproved GL 89-04, relief request not evaluated in TER.

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
VR-005	3.1.1.1	IWV-3512: Relief valve testing requirements.	Numerous safety and relief valves.	Test safety and relief valves in accordance with ANSI/ASME OM-1-1981 in lieu of ASME/PTC 25.3-1976.	Provisional relief granted. (a)(3)(i)
VR-017	3.1.2.1	IWV-3415: Valve fail-safe test requirements.	All valves for which fail-safe testing is required.	For most valves, normal stroking to fail-safe position considered fail-safe test MSIVs, CRD valves, and Service water valves tested to fail-safe position by means other than normal stroking.	Partial relief granted. (a)(3)(i)
VR-051	3.2.1.1	IWV-3520: Check valve exercising frequency and method requirements.	Numerous HPCI and RCIC system check valves.	Disassembly and inspection every refueling outage for individually listed valves and sample disassembly and inspection for groups of identical valves in similar applications.	Provisional relief granted per GL 89-04 for open position. Interim relief granted for valves V-23-009, -010, -012 for closed position. (a)(3)(i)
VR-021	3.3.1.1	IWV-3520: Check valve exercising frequency and method requirements.	HPCI pump suction check valve from the suppression pool: V-23-001	Disassemble and inspect during refueling outages and verify reverse flow closure capability following reassembly.	Provisional relief granted per GL 89-04 for testing to open position. Relief granted for testing to closed position. (a)(3)(i)

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
VR-021	3.4.1.1	IWV-3520: Check valve exercising frequency and method requirements.	RCIC pump suction check valve from the suppression pool: V-25-001	Disassemble and inspect during refueling outages and verify reverse flow closure capability following reassembly.	Provisional relief granted per GL 89-04 for testing to open position. Relief granted for testing to closed position. (a)(3)(i)
VR-006	3.5.1.1	IWV-3411: Valve exercising requirements. IWV-3417(a): Stroke time trending requirements.	Numerous ADS, safety relief, and solenoid valves.	Remove, test, disassemble, inspect, and rebuild at least half the valves every cycle. Exercise in situ once every refueling outage during plant startup.	Relief granted. (a)(3)(i)
VR-007	3.6.1.1	IWV-3413: Stroke time measurement requirements. IWV-3417: Stroke time trending and corrective action requirements.	Diesel generator air start solenoid valves: SV-3261A, B, -3262A, B	Start diesels on AC valve train monthly and on DC valve train quarterly, both without stroke time measurement. Ensure that diesels start within Tech. Spec. time limit using DC valve train every six months.	Provisional relief granted. (g)(6)(i)
VR-012	3.7.1.1	IWV-3521: Check valve exercising frequency requirements.	Reactor recirculation pump seal water supply check valves from the control rod drive hydraulic system: V-17-083, -096	Verify the closure capability of these valves with Appendix J, Type C leak rate tests during refueling outages.	Relief granted. (g)(6)(i)

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
VR-032	3.8.1.1	IWV-3413(b): Valve stroke timing requirements. IWV-3417(a): Stroke time corrective action requirements.	Numerous containment atmosphere monitoring system containment isolation valves.	Exercise and verify valve positions quarterly.	Interim relief granted. (g)(6)(i)
VR-035	3.9.1.1	IWV-3417(a): Stroke time corrective action requirements.	ESW return valves from the control building chillers: CV-1956A, B ESW supply valves to the emergency diesel generators: CV-2080, -2081	Estimate stroke times based on valve stem movement and compare results to maximum limiting stroke time.	Provisional relief granted. (g)(6)(i)
VR-053	3.10.1.1	IWV-3520: Check valve exercising requirements.	A side control building HVAC instrument air supply check valves: V-73-006, -007	Backflow test these series valves as a unit quarterly and verify that the total backleakage through the pair does not exceed a specific maximum amount.	Provisional relief granted. (g)(6)(i)
VR-002	NA	IWV-3412, -3413, -3417(a): Valve exercising method, stroke timing, and corrective action requirements.	All solenoid and air-pilot operated control valves without individual position indication.	Verify that the main valve has stroked within its respective time limits.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-003	NA	IWV-3521: Check valve exercising frequency requirements.	RHR to recirculation system check valves: V-19-0149 and V-20-0082	Part-stroke exercise one valve with flow during cold shutdowns and manually full-stroke exercise both valves during refueling outages.	Preapproved GL 89-04, relief request not evaluated in TER.

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
VR-008	NA	IWV-3420: Valve leak rate test requirements. IWV-3521: Check valve exercising frequency requirements.	All excess flow check valves.	Test valves in accordance with DAEC Technical Specification 4.7.D.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-011	NA	IWV-3426: Valve leak rate trending requirements.	Suppression chamber vacuum breaker valves: CV-4327A, B, C, D, E, F, G, H	Verify leak tightness of these valves during containment integrity testing.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-013	NA	IWV-3411: Valve exercising frequency requirements. IWV-3413: Stroke time measurement requirements. IWV-3417: Corrective action requirements.	Numerous control rod drive hydraulic system air and solenoid operated valves.	Test valves once each cycle per Tech. Spec. 4.3.C, compare scram time data to acceptance criteria of Tech. Spec. 3.3.C, verify that backup scram valves energize to vent scram pilot air header upon receipt of a scram signal.	Preapproved GL 89-04, see anomaly No. 8.
VR-013	NA	IWV-3411: Valve exercising frequency requirements. IWV-3413: Stroke time measurement requirements. IWV-3417: Corrective action requirements.	Scram discharge volume vent and drain valves' solenoid valves: SV-1868A, B, -1869A, B	Test solenoid valves with the associated scram discharge volume vent and drain valves during the Mode Switch Placed in Shutdown Test performed each refueling outage.	Preapproved GL 89-04, see anomaly No. 9.
VR-013	NA	IWV-3521: Check valve exercising frequency requirements.	Backup scram check valve: V-17-0062	Exercise each refueling outage by verifying that the backup scram valves vent air when energized.	Preapproved GL 89-04, see anomaly No. 10.

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
VR-013	NA	IWV-3521: Check valve exercising frequency requirements.	Control rod drive check valves to the scram discharge header: V-18-1453 through -1541	Test valves each operating cycle per Tech. Spec. 4.3.C, compare scram time data to acceptance criteria of Tech. Spec. 3.3.C.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-013	NA	IWV-3521: Check valve exercising frequency requirements.	Charging water header to control rod drive check valves: V-18-0118 through -0206	Verify closure capability by depressurizing the control rod drive charging header and verifying that each hydraulic control unit accumulator remains in a charged condition during the test.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-013	NA	IWV-3521: Check valve exercising frequency requirements.	Control rod drive cooling water supply check valves: V-18-0919 through -1007	Verify closure capability by normal rod motion as required by Tech. Spec. 4.3.A.2.a.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-019	NA	IWV-3521: Check valve exercising frequency requirements.	Main steam isolation valve and ADS relief valve accumulator supply check valves: V-14-009, 014, 015, 016, 032, 100, 104, 108, 112, 116, 120, 124	Exercise to the closed position during refueling outages.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-020	NA	IWV-3521: Check valve exercising frequency requirements.	Standby liquid control injection line containment isolation valves: V-26-008, 009	Exercise open and closed each operating cycle in accordance with DAEC Technical Specifications 4.4.A.2.b and 4.7.A.2.c.	Preapproved GL 89-04, relief request not evaluated in TER.

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
VR-025	NA	IWV-3521: Check valve exercising frequency requirements.	Drywell nitrogen supply line containment isolation valve: V-43-214	Verify closure capability by during refueling outages by Appendix J, Type C, leak testing.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-031	NA	IWV-3521: Check valve exercising frequency requirements.	TIP system nitrogen purge line containment isolation valve: 1S266/CK	Verify closure capability of Appendix J, Type C, leak rate testing each operating cycle.	Preapproved GL 89-04, relief request not evaluated in TER
VR-033	NA	IWV-3521: Check valve exercising frequency requirements.	Core spray injection check valves to the reactor vessel: V-21-072, -073	Full-stroke exercise each refueling outage.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-034	NA	IWV-3417(a): Stroke time corrective action requirements.	Numerous CAD, TIP, PASS, and RHR containment isolation valves.	Assign maximum limiting stroke time of 2 seconds to these valves in accordance with GL 89-04, Position 6.	Approved per GL 89-04, Position 6. Relief request not evaluated in TER.
VR-037	NA	IWV-3427(b): Valve leakage rate trending requirements.	All containment isolation valves six inches or greater in size.	None	Approved per GL 89-04, Position 10. Relief request not evaluated in TER.
VR-040	NA	IWV-3200: Test requirements following valve replacement, repair, and maintenance.	Feedwater supply line outside containment isolation valves: MO-4441, -4442	Perform leak rate test to ensure these valves will perform their containment isolation function when they are subjected to repair or maintenance that could affect their performance.	Preapproved GL 89-04, relief request not evaluated in TER.

RELIEF REQUEST NUMBER	TER SECTION	SECTION XI REQUIREMENT & SUBJECT	EQUIPMENT IDENTIFICATION	ALTERNATE METHOD OF TESTING	ACTION BY USNRC
VR-041	NA	IWV-3200: Test requirements following valve replacement, repair, and maintenance.	Feedwater supply line inside containment isolation valves: V-14-001, -003	Perform leak rate test to ensure these valves will perform their containment isolation function when they are subjected to repair or maintenance that could affect their performance.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-050	NA	IWV-3521: Check valve exercising frequency requirements.	Suppression chamber vacuum breaker check valves: CV-4327A, B, C, D, F, G, H	Part-stroke exercise quarterly using installed air operators and mechanically full-stroke exercise to open and closed positions at least once each refueling cycle.	Preapproved GL 89-04, relief request not evaluated in TER.
VR-051	NA	IWV-3520: Check valve exercising frequency and method requirements.	RHR and core spray pump minimum flow recirculation line check valves: V-19-014, -016V-20-006, -008V-21-009, -012	Part-stroke open quarterly, verify full-stroke open capability using disassembly and inspection on a sampling basis during refueling outages.	Approved per GL 89-04, Position 2. Relief request not evaluated in TER.

Appendix D

Valve and Pump Types Associated With Inservice Testing Programs at Indicated Nuclear Plants

Appendix D

Valve and Pump Types Associated With Inservice Testing Programs at Indicated Nuclear Plants

Table D-1. Beaver Valley 1 IST Program listing of valves by system/type/code/noncode.

System	Code class	Check valves	Globe valves	Needle valves	Plug valves	Diaphragm valves	Gate valves	Three-way valves	Safety/relief valves	Ball valves	Butterfly valves	Solenoid valves	Total
Reactor Coolant	1/2	2	8	1	3	1	3	6	3	—	—	—	27
Chemical Volume and Control	2/3	17	14	—	—	1	17	—	6	—	—	—	55
Reactor Plant Vents and Drains (Aerated)	2	—	2	—	—	—	—	—	—	—	—	—	2
Reactor Plant Vents and Drains (Nonaerated)	2	—	4	—	—	—	—	—	—	—	—	—	4
Residual Heat Removal	1/2	2	—	—	—	—	6	—	1	1	—	—	10
Safety Injection	1/2	36	15	—	—	—	22	—	7	—	—	—	80
Containment Vacuum	2	—	8	—	—	—	—	—	—	—	2	—	10
Leakage Monitoring	2	—	2	—	—	—	—	—	—	—	—	—	2
Containment Depressurization (Quench Spray)	2	6	—	—	—	2	12	—	2	—	—	—	22
Reactor Plant Sample	2	—	19	—	—	—	—	—	—	—	—	—	19
Reactor Plant Component Cooling Water	2/3	6	52	—	—	—	—	—	60	—	8	—	126
Fuel Pool Cooling and Purification	2	—	—	—	—	—	—	—	—	4	—	—	4
Main Steam	2	11	6	—	—	—	9	—	15	—	—	—	41
Feedwater	2/3	21	12	—	—	—	11	—	1	—	—	—	45

Table D-1. (continued).

System	Code class	Check valves	Globe valves	Needle valves	Plug valves	Diaphragm valves	Gate valves	Three-way valves	Safety/relief valves	Ball valves	Butterfly valves	Solenoid valves	Total
Steam Generator Blowdown	2	—	3	—	—	—	6	—	—	—	—	—	9
Auxiliary Steam	2/3	1	1	—	—	—	2	—	—	—	—	—	4
River Water	2/3	24	—	—	—	—	4	—	9	4	32	—	73
Compressed Air (Instrument Air)	2/3	3	—	—	—	—	1	—	—	—	—	—	4
4KV Station Service (Diesel Air Start)	3	4	—	—	—	—	—	—	12	2	—	4	22
4KV Station Service (Diesel Fuel Oil)	3	6	—	—	—	—	2	—	4	—	—	—	12
Control Area Ventilation	3	—	—	—	—	—	5	—	5	—	—	—	10
Containment Area Ventilation	2	4	—	—	—	—	—	—	—	1	8	—	13
Post-Design Bases Accident Hydrogen Control	2	—	5	—	—	—	7	—	—	10	—	—	22
Containment	2/3	—	—	—	—	—	4	—	—	6	—	—	10
Total		363	171	1	3	4	114	6	125	28	50	4	635

Table D-2. Review of Dresden IST Valve Program revision 1 Unit 2 and common systems.

System name	CK	ERV	GA	GL	RPD	RV	SCK	SV	XFC	TRV	Code	Non-Code	Total
Reactor Recirc/Main Steam/Feedwater	22	4	9	14	—	—	—	8	67	1	119	6	125
Control Rod Drive (177 HCUs)	5	—	14	—	—	—	—	—	—	—	19	0	19
Shutdown Cooling	—	—	7	—	—	—	—	—	—	—	7	0	7
Standby Liquid Control	4	—	2	—	—	2	—	—	—	—	8	0	8
Reactor Water Cleanup	1	—	3	1	—	—	—	—	—	—	5	0	5
Isolation Condenser	1	—	4	2	—	—	—	—	4	—	11	0	11
Core Spray	4	—	10	2	—	2	6	—	2	—	26	0	26
Low Pressure Coolant Injection/CCSW	16	—	22	5	—	6	2	—	—	—	51	0	51
Radwaste	—	—	3	—	—	—	—	—	—	—	3	0	3
High Pressure Coolant Injection	13	—	10	6	2	2	3	—	2	—	38	0	38
Containment Atmosphere Monitor	2	—	8	—	—	—	—	—	—	—	10	0	10
ACAD	5	—	—	10	—	—	—	—	2	—	17	0	17
Reactor Building Component Cooling Water	—	—	3	—	—	—	—	—	—	—	3	0	3
Diesel Cooling/Service Water	6	—	—	—	—	—	—	—	—	—	6	0	6
Reactor Building Equipment and Drains	—	—	—	—	—	4	—	—	—	—	4	0	4
Control Room Ventilation	—	—	2	1	—	—	—	—	—	—	3	0	3
Totals	79	4	97	41	2	16	11	8	77	1	330	6	336

Key: CK–Check, ERV–Electromatic Relief, GA–Gate, GL–Globe, RPD–Rupture Diaphragm, RV–Relief, SCK–Stop Check, SV–Safety, XFC–Excess Flow Check, TRV–Target Rock

Table D-3. Review of Dresden IST Valve Program, revision 1, unit 3.

System name	CK	ERV	GA	GL	RPD	RV	SCK	SV	XFC	TRV	Code	Non-Code	Total
Reactor Recirc/Main Steam/Feedwater	16	4	5	14	—	—	—	8	67	1	115	0	115
Control Rod Drive (177 HCUs)	3	—	14	—	—	2	—	—	—	—	19	0	19
Shutdown Cooling	—	—	7	—	—	—	—	—	—	—	7	0	7
Standby Liquid Control	4	—	2	—	—	2	—	—	—	—	8	0	8
Reactor Water Cleanup	1	—	3	1	—	—	—	—	—	—	5	0	5
Isolation Condenser	1	—	4	2	—	—	—	—	4	—	11	0	11
Core Spray	—	—	8	—	—	—	6	—	—	—	14	0	14
Low Pressure Coolant Injection/CCSW	16	—	16	2	—	6	2	—	—	—	42	0	42
Containment Cooling Service Water	—	—	4	2	—	—	—	—	—	—	6	0	6
Radwaste	—	—	3	—	—	—	—	—	—	—	3	0	3
High Pressure Coolant Injection	15	—	9	2	2	2	2	—	2	—	34	0	34
Containment Atmosphere Monitor	2	—	8	—	—	—	—	—	—	—	10	0	10
ACAD	5	—	—	10	—	—	—	—	2	—	17	0	17
Reactor Building Component Cooling Water	—	—	3	—	—	—	—	—	—	—	3	0	3
Diesel Cooling/Service Water	—	—	3	—	—	—	—	—	—	—	3	0	3
Reactor Building Equipment and Drains	—	—	—	—	—	4	—	—	—	—	4	0	4
Totals	63	4	89	33	2	16	10	8	75	1	301	0	301

Key: CK–Check, ERV–Electromatic Relief, GA–Gate, GL–Globe, RPD–Rupture Diaphragm
RV–Relief, SCK–Stop Check, SV–Safety, XFC–Excess Flow Check, TRV–Target Rock

Table D-4. Review of Dresden IST Valve Program revision 1, units 2, 3 and common systems.

Pump name	Vertical line shaft	Centrifugal	Positive displacement	Other	Code	Non-Code	Total
Standby Liquid Control	—	—	4	—	4	0	4
Core Spray	—	4	—	—	4	0	4
Component Cooling Service Water	—	8	—	—	8	0	8
Low Pressure Coolant Injection	—	8	—	—	8	0	8
High Pressure Coolant Injection	—	2	—	—	2	0	2
Diesel Cooling	—	3	—	—	3	0	3
ECCS Keep Fill	—	2	—	—	2	0	2
Fuel Pool Cooling	—	4	—	—	0	4	4
Diesel Oil Transfer	3	—	—	—	0	3	3
Total	3	31	4	0	31	7	38

Table D-5. Review of Duane Arnold IST Pump Program Revision 12.

Pump name	Vertical line shaft	Centrifugal	Positive displacement	Other	Code	Non-Code	Total
Residual Heat Removal Service Water (RHRSW)	4	—	—	—	4	0	4
Diesel Fuel Oil (DFO)	2	—	—	—	0	2	2
Emergency Service Water (ESW)	3	—	—	—	3	0	3
Screen	—	—	—	2	0	2	2
River Water	4	—	—	—	4	0	4
Containment Spray	—	3	—	—	3	0	3
High Pressure Coolant Injection (HPCI)	—	1	—	—	1	0	1
Reactor Core Isolation Cooling (RCIC)	—	1	—	—	1	0	1
Residual Heat Removal (RHR)	—	4	—	—	4	0	4
Standby Liquid Control (SLC)	—	—	2	—	0	2	2
Total	13	9	2	2	19	7	26

Table D-6. Review of Duane Arnold IST Valve Program Revision 12.

System name	ANG	AV	BAL	BTF	CK	XFC	SH	GA	GL	PLG	RV	RPD	SV	SCK	2WY	3WY	4WY	Code	Non-Code	Total
Transverse Incore Probe	—	—	3	—	1	—	3	—	—	—	—	—	—	—	—	—	—	0	7	7
Condensate and Demin.	—	—	—	—	—	—	—	2	—	—	—	—	—	—	—	—	—	2	2	2
Reactor Building Cooling	—	—	—	—	—	—	—	2	—	—	1	—	—	—	—	—	—	0	3	3
RHR Service Water	—	—	—	—	4	—	—	8	2	—	2	—	—	—	—	4	—	14	6	20
Nuclear Boiler	—	—	—	—	30	18	—	2	11	—	12	—	2	2	8	25	8	49	69	118
Reactor Vessel Inst	—	—	—	—	—	35	—	—	4	—	—	—	—	—	—	—	—	37	2	39
Reactor Recirculation	—	—	—	—	—	35	—	2	2	—	—	—	—	—	—	2	—	38	3	41
CRD Hydraulic	—	—	—	—	8	—	—	2	6	—	—	—	—	—	—	10	—	11	15	26
Residual Heat Removal	2	—	—	—	10	—	—	29	15	—	5	—	—	—	—	—	—	57	4	61
Core Spray	—	—	—	—	6	2	—	10	2	—	4	—	—	—	—	—	—	24	0	24
High Pressure Coolant Inj	—	—	—	—	11	4	—	12	3	1	3	—	—	3	—	5	—	30	12	42
Reactor Core Isolation Cool	—	—	—	—	8	4	—	11	4	—	3	—	—	1	—	3	—	8	26	34
Standby Liquid Control	—	—	—	—	4	—	—	2	—	—	2	—	—	—	—	—	—	2	6	8
Reactor Water Cleanup	—	—	—	—	—	—	—	2	1	—	—	—	—	—	—	—	—	3	0	3
River Water Supply	—	6	—	—	4	—	—	—	—	—	—	—	—	—	—	—	—	4	6	10
Compressed Air	—	—	—	—	—	—	—	1	—	—	—	—	—	—	—	—	—	0	1	1
Diesel Generator	—	—	—	—	14	—	—	—	—	—	6	—	—	—	4	—	—	0	24	24
Radwaste Sump	—	—	—	—	—	—	—	4	—	—	—	—	—	—	—	4	—	0	8	8
Containment Atmospheric Cntr	—	—	—	11	17	—	—	20	2	—	1	1	—	1	—	19	—	4	68	72
Service Water Pumphouse	—	4	—	3	7	—	—	—	—	—	—	—	—	—	—	3	—	14	3	17
Drywell Cooling Water	—	—	—	—	—	—	—	4	4	—	—	—	—	—	—	4	—	0	12	12
Standby Filter Unit Cntr Bld	—	—	—	—	4	—	—	—	—	—	—	—	—	—	—	—	—	0	4	4

Table D-6. (continued).

System name	ANG	AV	BAL	BTF	CK	XFC	SH	GA	GL	PLG	RV	RPD	SV	SCK	2WY	3WY	4WY	Code	Non-Code	Total
Containment Atmosphere Mon	—	—	—	—	—	—	—	—	20	—	—	—	—	—	—	—	—	0	20	20
MSIV Leakage Control	—	—	—	—	—	—	—	12	—	—	—	—	—	—	—	—	—	4	8	12
Post Accident Sampling	—	—	—	—	—	—	—	—	2	—	—	—	—	—	—	—	—	0	2	2
Totals	2	10	3	14	128	98	3	125	78	1	39	1	2	7	12	79	8	299	311	610

Key: ANG–Angle, AV–Auto Vent, BAL–Ball, BTF–Butterfly, CK–Check, XFC–Excess Flow Check, SH–Explosive Shear, GA–Gate, GL–Globe, PLG–Plug, RV–Relief, RPD–Rupture Diaphram, SV–Safety, SCK–Stop Check, 2WY–2 Way, 3WY–3 Way, 4WY–4Way

Table D-7. Review of River Bend IST Pump Program Revision 6.

Pump Name	Vertical Centrifugal	Horizontal Centrifugal	Positive Displacement	Code	Non-Code	Total
Service Water—Normal	—	4	—	4	0	4
Standby Liquid Control (SLC)	—	—	2	2	0	2
High Pressure Core Spray (HPCS)	1	1	—	2	0	2
Residual Heat Removal— Low Pressure Coolant Injection (LPCI)	3	1	—	4	0	4
Low Pressure Core Spray (LPCS)	1	1	—	2	0	2
Reactor Core Isolation Cooling (RCIC)	1	1	—	2	0	2
Service Water—Standby	4	—	—	4	0	4
Diesel Generator—Fuel Oil	3	—	—	3	0	3
HAVC Chilled Water	—	4	—	4	0	4
Fuel Pool Cooling	—	2	—	2	0	2
Total	13	14	2	29	0	29

Table D-8. Review of River Bend IST Valve Program Revision 6.

System name	BF	CK	DP	GA	GL	RV	SC	XP	Code	Non-Code	Total
Control Rod Drive Hydraulic	—	4	—	2	5	—	—	—	11	0	11
Reactor Recirculation	—	4	—	—	10	—	—	—	14	0	14
Condensate and Feedwater	—	5	—	8	—	—	—	—	13	0	13
Main Steam	—	64	—	1	19	16	—	—	100	0	100
Component Cooling Water—Reactor Plant	9	13	—	3	2	—	—	—	27	0	27
Service Water—Normal	18	34	—	19	8	2	—	—	81	0	81
Air—Service and Breathing	—	1	—	3	—	—	—	—	4	0	4
Air—Instrument	—	11	—	5	—	—	—	—	16	0	16
Standby Liquid Control	—	4	—	—	2	2	—	2	10	0	10
High Pressure Core Spray	—	6	—	4	4	—	—	—	14	0	14
Residual Heat Removal—LPCI	—	20	—	24	21	8	3	—	76	0	76
Low Pressure Core Spray	—	4	—	3	1	2	—	—	10	0	10
MSIV—Leakage Control	—	32	—	2	25	2	—	—	61	0	61
Reactor Core Isolation Cooling	—	9	—	3	7	1	—	—	20	0	20
Fire Protection—Water	—	1	—	2	—	—	—	—	3	0	3
Hydrogen Mixing Purge & Recombiner	8	1	—	3	—	—	—	—	12	0	12
Service Water—Standby	6	4	—	—	—	—	—	—	10	0	10
Diesel Generator	—	13	—	4	4	8	—	—	19	10	29
HVAC	14	14	16	9	4	2	—	—	43	16	59
Containment Atmosphere and Leakage Monitoring	—	2	—	5	34	—	—	—	41	0	41
Reactor Water Cleanup and Filter	—	—	—	12	—	4	—	—	16	0	16
Fuel Pool Cooling	—	11	—	5	—	—	—	—	16	0	16

Table D-8. (continued).

System name	BF	CK	DP	GA	GL	RV	SC	XP	Code	Non-Code	Total
Drains—Floor and Equipment	—	25	—	6	14	—	—	—	45	0	45
Sampling—Reactor Plant	—	2	—	—	6	—	—	—	8	0	8
Totals	55	284	16	123	166	47	3	2	670	26	696

Key: BF–Butterfly, CK–Check, DP–Damper, GA–Gate, GL–Globe, RV–Relief, SC–Stop Check, XP–Explosive

Table D-9. Review of Waterford 3 IST Pump Program Revision 7, Change 4.

Pump name	Vertical centrifugal	Horizontal centrifugal	Positive displacement	Code	Non-Code	Total
Containment Spray	—	2	—	2	0	2
High Pressure Safety Injection	—	3	—	3	0	3
Low Pressure Safety Injection	—	2	—	2	0	2
Component Cooling Water	—	3	—	3	0	3
Auxiliary Component Cooling Water	—	2	—	2	0	2
Emergency Generator Fuel Oil Transfer	—	2	—	2	0	2
Emergency Feedwater	—	3	—	3	0	3
Charging	—	—	3	3	0	3
Boric Acid	—	2	—	2	0	2
Chilled Water	—	3	—	3	0	3
Total	0	22	3	25	0	25

Table D-10. Review of Waterford 3 IST Valve Program Revision 7, Change 4.

System name	BL	B	CK	D	GA	GL	PR	ANG	Code	Non-Code	Total
Reactor Coolant	—	—	—	—	—	6	3	2	11	0	11
Chemical and Volume Control	—	—	13	—	6	12	—	—	31	0	31
Safety Injection	—	8	44	—	25	33	2	—	112	0	112
Containment Spray	—	—	8	—	2	—	—	—	10	0	10
Emergency Feedwater	—	—	8	—	—	8	—	—	16	0	16
Feedwater	—	—	4	—	4	—	—	2	10	0	10
Main Steam	—	—	2	—	4	4	12	2	24	0	24
Emergency Diesel	—	—	6	—	—	—	—	—	6	0	6
Chilled Water	—	8	3	—	—	30	—	—	41	0	41
Component Cooling	—	44	27	—	—	—	—	—	71	0	71
Air Conditioning	2	55	6	—	—	2	—	—	65	0	65
Air Sysytems	—	—	11	—	2	8	—	—	21	0	21
Fuel Pool	—	—	—	2	2	—	—	—	4	0	4
Waste Management	—	—	—	4	—	—	—	—	4	0	4
Boron Management	—	—	—	2	—	—	—	—	2	0	2
Demineralized Water	—	2	7	—	—	2	—	—	11	0	11
Nitrogen Gas	—	—	17	—	—	9	—	—	26	0	26
Hydrogen Analyzer	—	—	2	—	—	22	—	—	24	0	24
Sampling	—	—	—	—	4	8	—	—	12	0	12
Blowdown	—	—	—	—	4	—	—	—	4	0	4
Fire Protection	—	—	2	—	—	2	—	—	4	0	4
Totals	2	117	160	8	53	146	17	6	509	0	509

Key: BL–Ball, B–Butterfly, CK–Check, D–Diaphram, GA–Gate, GL–Globe, N–Needle, PR–Relief or Safety, ANG–Angle

Solution to Valve Failures at Braidwood Induced by Service Water Cavitation

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ABSTRACT

Control valves throttle fluid from a high pressure to a lower pressure. On water systems, this throttling process may be accompanied by cavitation, which induces valve noise, vibration, and material damage. Extensive and significant cavitation erosion has been experienced the last 10 years in most service water control valve bodies, downstream flanges, and reducers at Braidwood Station. There have been 40 different and distinct cavitation-induced failures in the service water system at Braidwood Station. These failures have created significant costs and continue to be a lingering source of operational maintenance costs to the Commonwealth Edison Company, which is incurring significant financial losses. It should be noted that almost all service water control valves experience some cavitation effects.

Cavitation and cavitation damage are complex and elusive phenomena for which no single, simple analytical model exists. The purpose of this paper is to explain features of service water control valve cavitation failures and some of the solutions used by Commonwealth Edison at their six nuclear stations.

The paper discusses the following:

- Braidwood's history of erosion from cavitation
- Erosion-corrosion considerations
- The Instrument Society of America's valve sizing equations and how they relate to cavitation
- Methods to eliminate cavitation
- Corrective actions and practical approaches used by Commonwealth Edison to eliminate cavitation.

INTRODUCTION

There are nine control valves (25 problems) in the service water system at Braidwood Station, as shown in Table 1. Braidwood has two different valve models in this application, but the eccentric rotary valve plug model, shown in Figure 1, is the most common. Most of these valves are indicated in Table 1, and all of the necessary valve pressure data needed for these problems are shown in Table 2. Each heat exchanger has its own control valve, which is controlled by a two-mode temperature controller that measures the heat exchanger outlet temperature of process fluid, and this temperature is controlled by throttling the service water control valve and thus the service water out of each heat exchanger. This installation is typical of all the heat exchangers.

Table 1. History of cavitation-induced valve failures in service water at Braidwood.

Item	Valve tag	Date	Problem and solution
1	1-061	07-89	Pinhole leak in valve body and reducer; installed a new valve and reducer.
2	1-061	05-91	Pinhole leak in valve body and reducer; installed a new valve, flange, and reducer.
3	1-061	10-92	Pinhole leak in valve body and reducer; installed a new carbon steel valve body and 316 SS reducer with Belzona.
4	1-061	03-94	Pinhole leak in valve body; installed a new 316 SS valve body; inspection of 316 SS reducer with Belzona showed no cavitation damage.
5	2-061	11-88	Pinhole leak in downstream reducer; installed a new carbon steel reducer.
6	2-061	10-91	Pinhole leak in valve body; installed new valve, flange, and carbon steel reducer.
7	2-061	03-93	Installed new valve and carbon steel reducer.
8	1-106	04-91	Pinhole leak on bottom of downstream pipe; installed new valve, flange, and weld repaired reducer.
9	1-106	09-93	Pinhole leak in valve and reducer; installed new 316 SS valve and reducer.
10	2-106	07-90	Pinhole leak in downstream of valve caused by cavitation; replaced valve, flange, and carbon steel reducer.
11	2-106	09-91	Pinhole leak in downstream reducer due to cavitation; installed new carbon steel valve, flange, and reducer.
12	2-106	03-93	Pinhole leak in downstream reducer due to cavitation; installed new carbon steel valve, flange, and reducer.
13	1-113	03-89	Pinhole leak in downstream reducer; installed new carbon steel flange and reducer.
14	1-113	08-90	Pinhole leak in downstream reducer. Installed new carbon steel valve, stainless steel (SS) flange, and reducer.
15	1-113	03-94	Installed a new 316 SS body; downstream 316 SS flange and reducer were like new.
16	2-113	01-90	Valve body and downstream pipe was eroded from cavitation; replaced with carbon steel valve parts.
17	2-113	09-91	Pinhole leak in downstream reducer; installed a new carbon steel valve and 316 SS downstream flange and reducer.
18	2-113	03-93	Pinhole leak in valve body; installed a new carbon steel valve.
19	2-113	10-93	Pinhole leak in valve body; installed a new 316 SS body; 316 SS flange and reducer showed no damage.
20	0-73A	08-90	Pinhole leak in downstream reducer; installed a new carbon steel valve, flange, and reducer.
21	0-73A	03-94	Installed a new carbon steel valve and reducer.
22	0-73B	05-89	Pinhole leak in downstream flange caused by cavitation; installed new carbon steel valve, flange, and reducer.
23	0-73B	05-91	Pinhole leak in valve body and downstream reducer; installed new carbon steel valve and 316 SS flange and reducer.
24	0-73B	10-93	Pinhole leak in valve body; installed a 316 SS body; inspection showed that 316 SS flange and reducer were like new.
25	0-135C	01-94	Pinhole leak in valve body; replaced with 316 SS body.

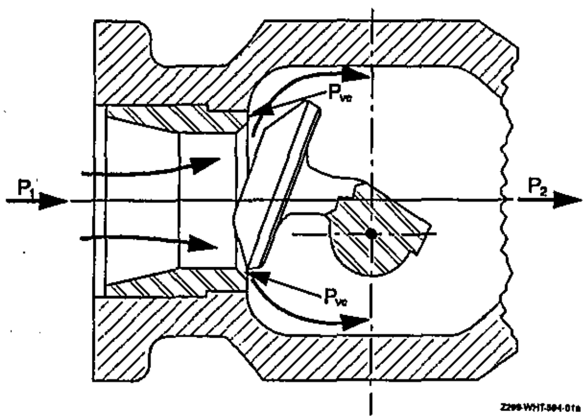


Figure 1. Masoneilan valve used in service water.

The other valves control the freon pressure in the heating, ventilation, and air conditioning systems by throttling the flow across the freon coils. These valves are operated by a pressure controller.

To control the flow as shown in Table 2 for winter flow conditions, the valves are throttled to about 5% open. At these flow conditions all the valves supercavitate. From engineering experience, during the winter months of operation the pipe downstream of all the valves is not flowing full, and therefore the downstream pressure, P2, is approximately zero pounds per square inch absolute (0 psia), as shown in Table 2. This information was used to compute the severity of cavitation, and the calculations showed that the actual valve pressure drop exceeded the allowable valve pressure drop; therefore, all valves supercavitate and are choked. Also, during the summer months of operation some of the valves are 100% open and also cavitate. Based on the preceding discussion, cavitation has induced many interesting problems, as noted by Table 1, which give the maintenance order history for eight valves. From Table 1, the two main problems found were

- Valve body erosion that resulted in pinhole leaks
- Downstream flange, reducer, and pipe erosion that also resulted in pinhole leaks.

Table 2. Valve data.

Item	Valve tag	Valve function	Valve size (in.)	CV rated	Flow (gpm)	P1 (ABS) upstream pressure	P2 (ABS) downstream pressure	Original body material
1	061	Turbine oil cooler	12	1,050	5,000	100	0–30	A216-WCP
2	106	Hydrogen cooler	12	1,050	5,000	100	0–30	A216-WCP
3	113	Stator water cooler	8	510	2,000	100	0–30	A216-WCP
4	73A&B	Service building	8	510	1,800	100	0–30	A216-WCP
5	135A-C	Auxiliary building (HVAC)	10	520	2,220	100	0–30	A216-WCP

This cavitation has induced extensive erosion in the valve bodies and the downstream reducers. Some of the valve bodies have been replaced with new ones and others have been weld repaired, as shown in Table 1 (one or the other) has occurred a number of times. Figures 2 through 14 show typical cavitation damage to the valves and the downstream reducers.

BRAIDWOOD'S HISTORY OF EROSION RESULTING FROM CAVITATION

Engineering evaluated 24 service water valves, of which 12 had experienced cavitation damage and have been repaired. These 12 valves and downstream piping were identified by exterior valve body or pipe leakage. The other 12 service water valves have not shown any external leakage, but inspection showed some interior valve trim damage. These valves have been weld repaired and their internal walls were coated with Belzona. However, after a year of operation the valves were found severely damaged from cavitation. The downstream flanges and piping were damaged by supercavitation, as shown in some of the photographs.

EROSION-CORROSION CONSIDERATIONS

The most troublesome maintenance problems encountered with valves is usually the erosion, corrosion, and pitting of valve bodies, trim, and downstream piping in contact with water under high velocities. These three different types of wear are the result of three entirely different conditions. Each type has a distinct characteristic appearance of its own, easily recognized by experienced observers.

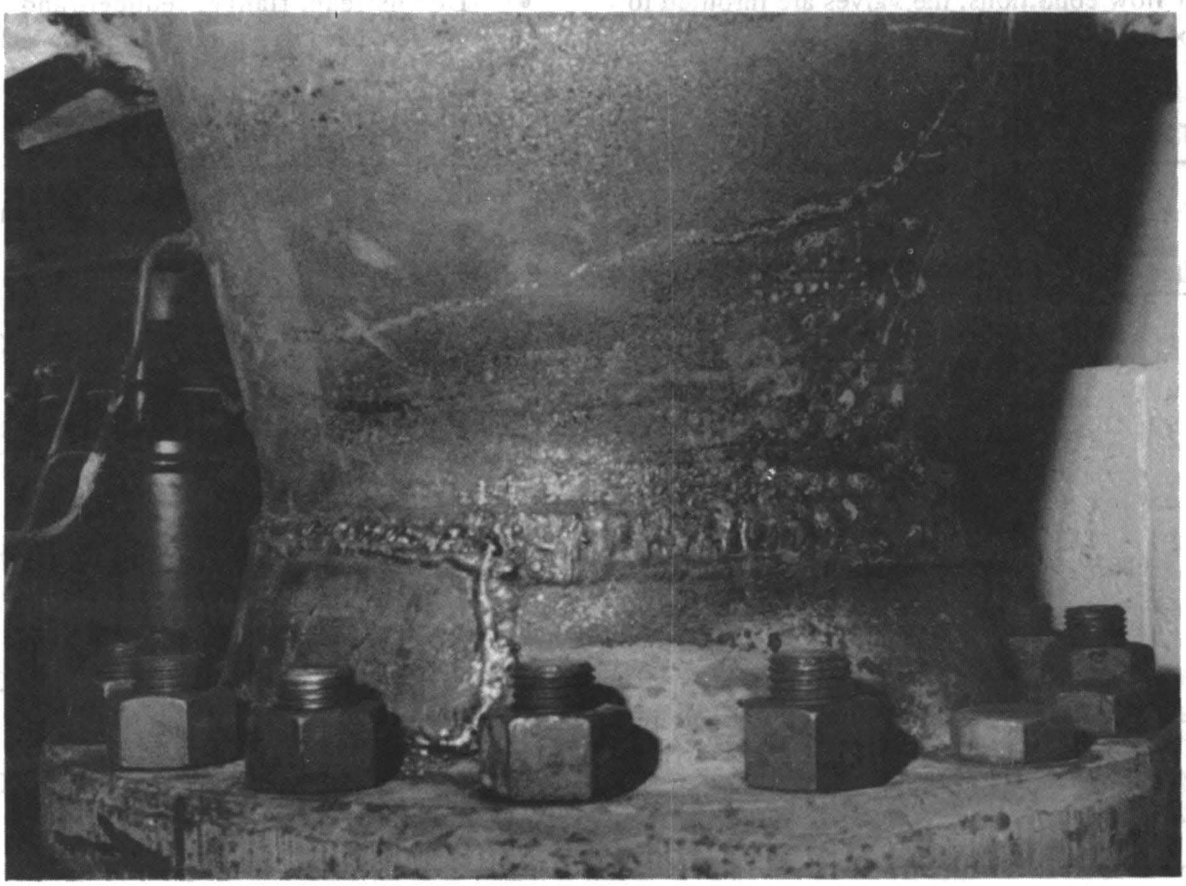


Figure 2. Note weld patch on downstream side of 1-WS-061 valve flange. Flange had a pinhole from cavitation.

Figure 3. In 2 weeks, the reducer developed a pinhole from cavitation, the pinhole is downstream of the weld patch.

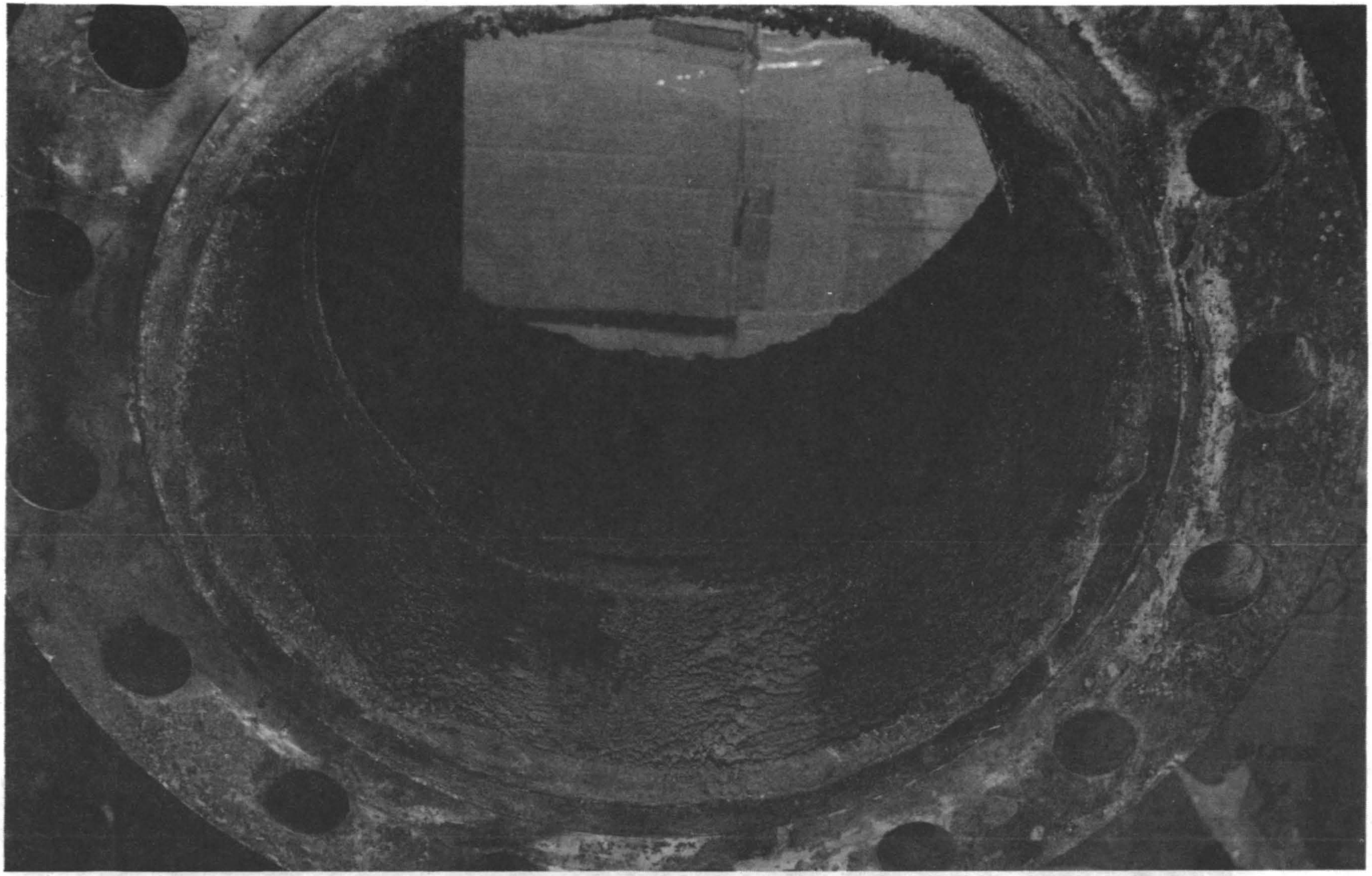


Figure 4. Cavitation damage in downstream reducer, downstream of 1WS-061 valve.

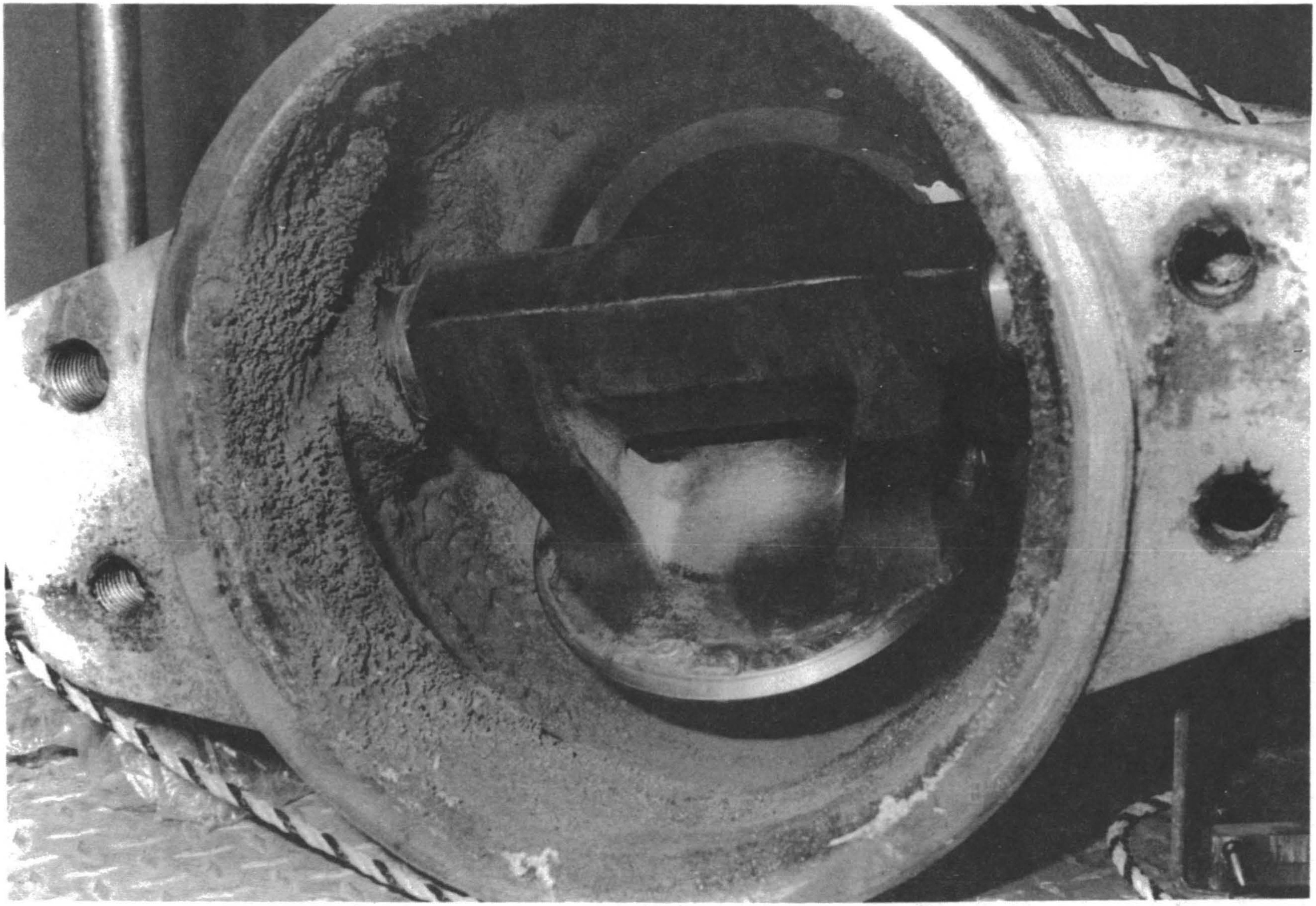


Figure 5. Erosion caused by cavitation in 1-WS-061 valve's body, flow direction is toward viewer.

Figure 6. Note the through holes on the exterior surface of the1-WS-061 valve body.

Figure 7. Cavitation damage to interior surface of 1-WS-113 valve body.

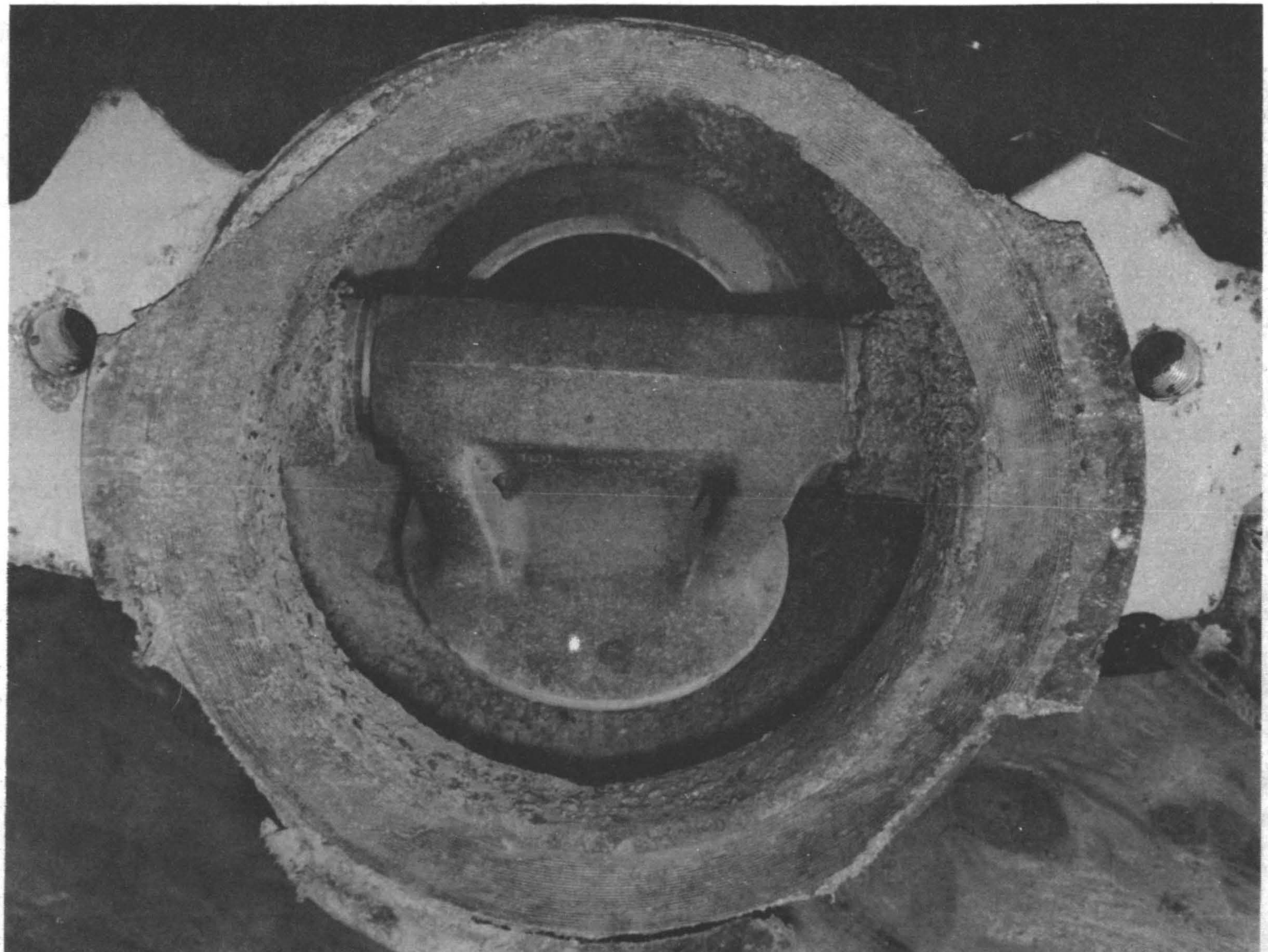


Figure 8. Cavitation damage to interior surface of WSO-73A valve body; flow is toward viewer.

Figure 9. Cavitation damage to interior surface of 2-WS-106 valve body; flow is toward viewer.

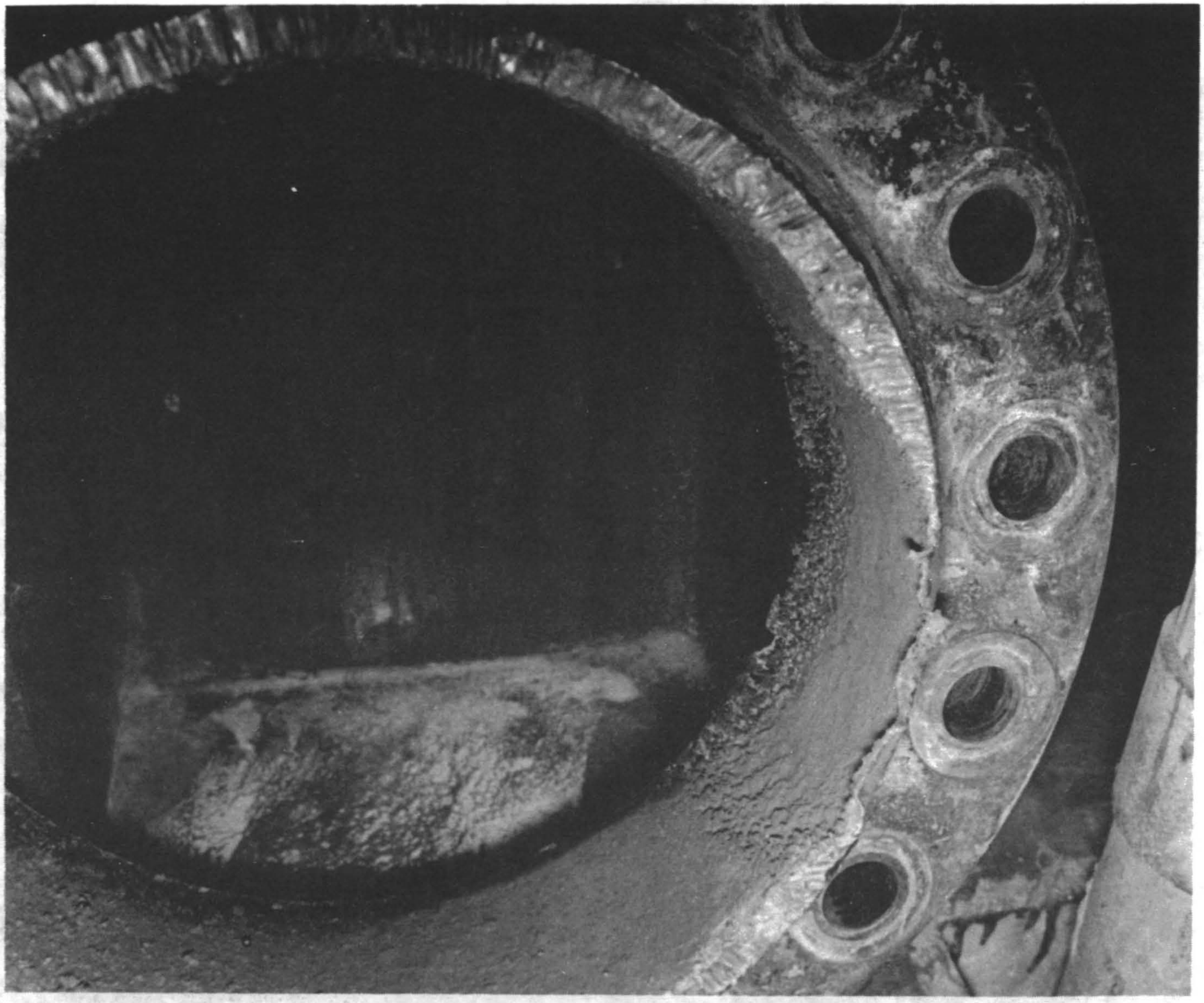


Figure 10. Cavitation damage to downstream flange of 2-WS-106 valve, flow is toward viewer.

Figure 11. Cavitation damage in downstream reducer of 2-WS-106 valve; flow is toward viewer.

Figure 12. Cavitation damage to interior surface of the 2-WS-061 valve body; flow is toward viewer.

Figure 13. Cavitation erosion in 2-WS-113 valve downstream flange and reducer.

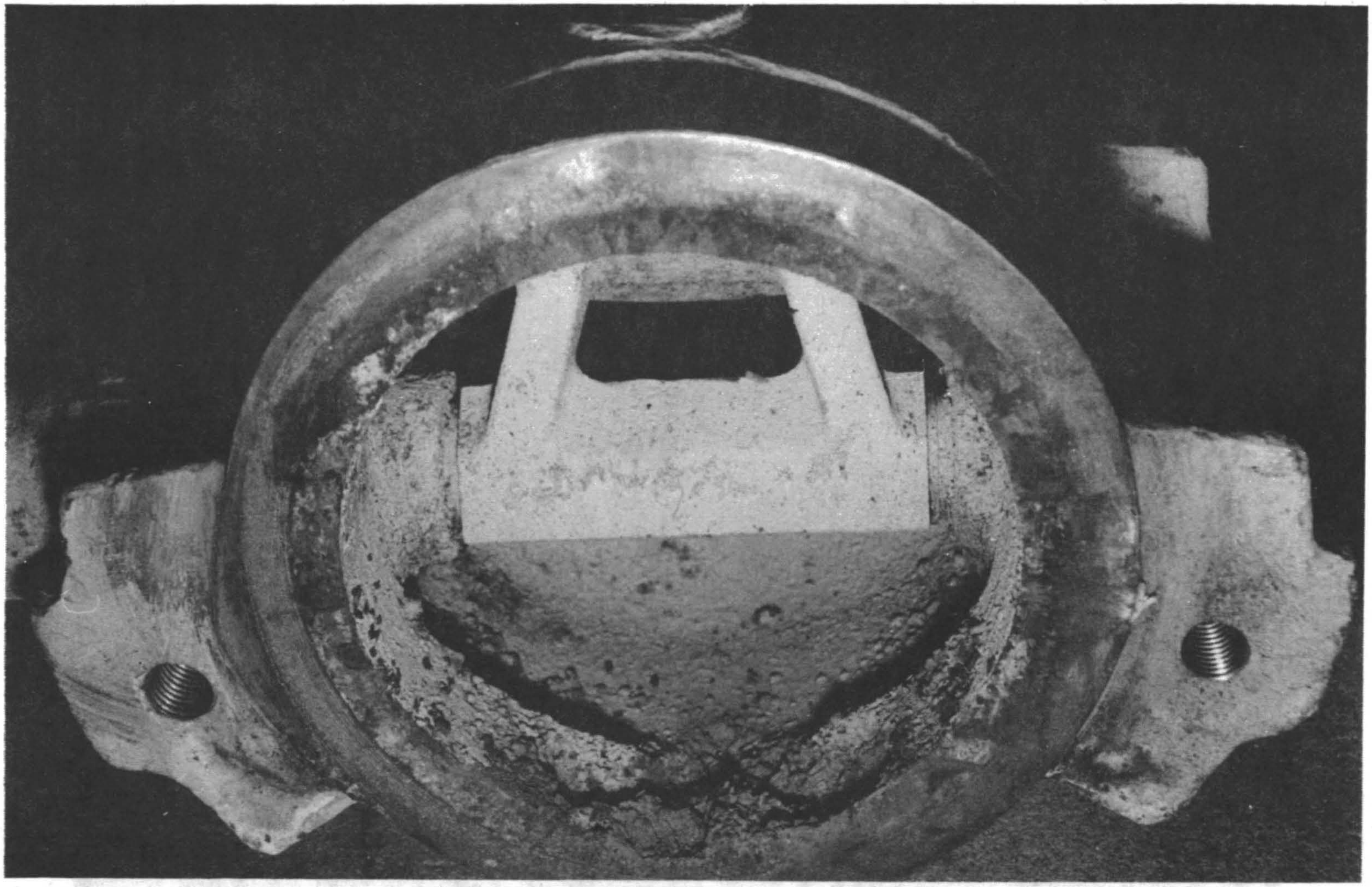


Figure 14. Cavitation erosion on interior surface of 2-WS-113 valve body, flow is toward viewer.

Erosion is caused by foreign matter, such as sand and silt in the water, and is most serious in combination with high water velocities. Eroded material usually has smoothly cut grooves that look as though they might have been gouged with a chisel.

Corrosion is caused by the action of chemicals in the water. Corrosion damage has the appearance of rusted iron or steel and looks as though the material were being removed in the form of flakes. Corrosion is most active at points of high velocity, apparently because the flow removes the protective rust coating, and a new clean surface is continuously presented to the corrosive action.

Pitting is an eroding action produced by cavitation and occurs at points of low pressure and high water velocities where there are changes in the shape of the valve passageway. Pitted material has a large number of small craters or depressions, which look as though tiny pieces of material had been removed one at a time. This gives it a distinct rough appearance, and hence the name *pitting* (see Figures 2 through 14).

Of the three types of wear, the pitting produced by cavitation is by far the most common and therefore presents the biggest maintenance problem. However, before considering possible means for preventing, reducing, or repairing pitting, it is desirable to have an understanding of the physical aspects of the cavitation phenomenon and how it attacks material to produce pitting in the valve or downstream piping.

INSTRUMENT SOCIETY OF AMERICA VALVE SIZING EQUATIONS AND HOW THEY RELATE TO CAVITATION

Cavitation is the phenomenon where vapor forms cavities in a liquid. Whenever the pressure at any point in a liquid is reduced to the vapor pressure, a cavity forms at that point, increasing as the liquid is vaporized. If the liquid is flowing, the cavity will move with the flow. When it reaches a region where the pressure is greater than the vapor pressure, it will collapse, producing shock waves that move at the speed of sound and alternate rapidly. The alternate compression and tension stresses induce mechanical fatigue in any material exposed to these waves, causing pitting.

Because cavitation occurs in many practical valve applications, it is worth considering in some detail. Cavitation tends to limit the flow and must be accounted for when sizing a valve. When cavitation is present, the basic liquid sizing equation is not a valid representation of what actually exists in the valve trim.

To simplify the discussion of flow, a control valve at any flow opening can be represented by a simple restriction in the line. As the flow passes through the physical valve seat restriction (see Figure 1 for the service water valves), there is a necking down, or contraction, of the flow stream. The minimum cross-sectional area of the flow stream occurs at a point called the vc (vena contracta), which is a short distance downstream of the physical restriction. The valve shown in Figure 1 has two vena contractas, that is, one flow path exists over the top of the disc and one flow path under the disc. Because the geometry is different for each flow path, the vena contractas are also different.

In order to understand cavitation, it is first necessary to understand the interchange between the kinetic energy and potential energy of a fluid flowing through a valve. To maintain a steady flow of liquid through the valve, the velocity must be greatest at the vena contracta where the cross-sectional area is the smallest. This increase in velocity, or kinetic energy, comes about at the expense of the pressure, or potential energy.

The pressure profile along the valve shows a sharp decrease in the pressure as the velocity increases, as shown in Figure 15. The lowest pressure will occur at the vena contracta where the velocity is greatest. Then, further downstream, as the fluid stream expands into a larger area of the valve body, the velocity decreases with a corresponding increase in the pressure. The pressure downstream of the valve, P2, never recovers completely to the pressure that existed upstream, P1.

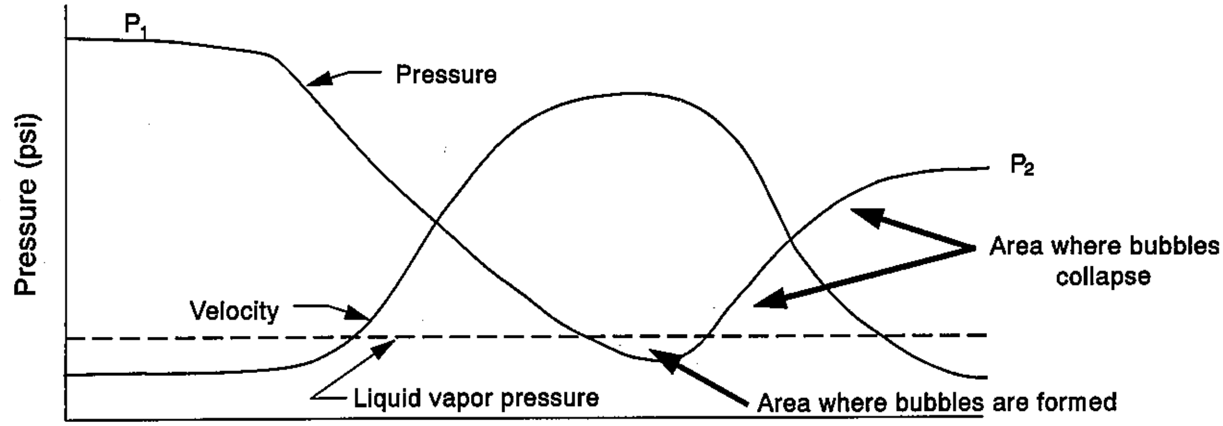


Figure 15. Pressure and velocity profile caused by flow-through.

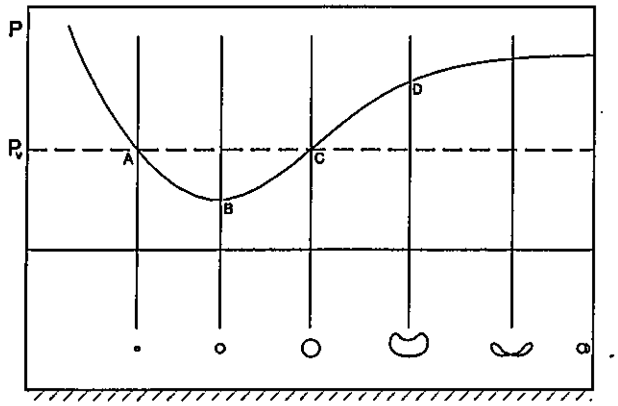
The pressure differential that exists across the valve is called the ΔP or $P_1 - P_2$ of the valve. This ΔP is a measure of the amount of energy that is dissipated in the valve. The more energy dissipated in a valve, the greater the ΔP for a given area and flow. The amount of liquid flow is determined by both the flow area and the flow velocity. If the valve's flow area is constant, any increase in flow must come from an increase in fluid velocity. An increase in velocity results in a lower pressure at the vena contracta. The pressure differential between the inlet and the vena contracta is directly related to the flow rate. The greater the flow, the larger this pressure differential.

If the flow through the valve increases, the velocity at the vena contracta must increase, and the pressure at that point will decrease accordingly. If the pressure at the vena contracta should drop below the vapor pressure for the liquid, bubbles will form in the fluid stream, shown by the area in Figure 15. The rate at which bubbles are formed will increase greatly as the pressure is lowered further below the vapor pressure.

If the downstream pressure recovery is sufficient to raise the outlet pressure, P2, above the liquid vapor pressure, the bubbles will collapse or implode as shown in Figure 15 and produce cavitation. The implosion of the vapor bubbles during cavitation releases energy, which shows up in the form of noise as well as physical damage to the valve. Millions of tiny bubbles imploding near the solid surfaces in the valve can gradually tear away the material, resulting in serious damage to the valve body or its internal parts and to downstream piping. It is usually apparent when a valve is cavitating because it will produce a noise much like gravel flowing through the valve. The area damaged by cavitation appears rough, dull, and cinderlike, as shown in Figures 2 through 14.

The vapor bubbles collapse on contact with high pressure zones, and the collapse process is so rapid that very high temperatures and pressures are created. The bubble initially decreases in size, causing the surrounding fluid to move inward, and this movement in turn increase the pressure acting on the cavity, so that it shrinks at an ever increasing rate. As a result, high pressures develop at the instant of collapse; pressures on the order of 100,000 psi have been reported. The prolonged hammering effect of imploding cavities results in brittle fracture and material wear, and also hydraulic shocks and radiated noise. This is shown in Figure 16 by the bubble collapse model. Mechanical overstressing of a metal surface is the dominate cause of damage by cavitation. High stresses are applied in impact fashion when vapor bubbles on or near a metal surface collapse and generate shock waves or liquid microjets. Corrosion and cavitation can have interactive effects that increase the rate of material loss.

High-impact blows from symmetrical bubble collapses leave spherical craters on ductile metals (see Figure 16). Ductile metals with high work-hardening characteristics can fail from fatigue if the surface hardness gained does not provide a high enough endurance limit to withstand the



Bubble collapse model

Figure 16. Bubble collapse model.

stresses from repeated blows. Hard metals and some work-hardening metals may resist immediate damage by single impacts, but they may fail from fatigue after an "incubation period." Microjet action may break out small chips or initiate microcracks on hard, brittle surfaces.

The classical pressure drop and flow relationship for single-phase, noncompressible liquid flow through valves can be written

$$Q = C_V \frac{\sqrt{P1 - P2}}{G}, \qquad (1)$$

where

Q	=	fluid flow rate in gallons per minute
C_V	=	valve sizing coefficient determined experimentally
P1 – P2	=	ΔP = Valve pressure drop
G	=	specific gravity of fluid (water at 60°F = 1.00).

If the flow is plotted against the square root of the valve pressure drop, as shown in Figure 17, a straight line relationship will be observed. The slope of the line is the value C_V. Increasing the pressure drop will ultimately result in a point where the C_V value begins to decrease. This is

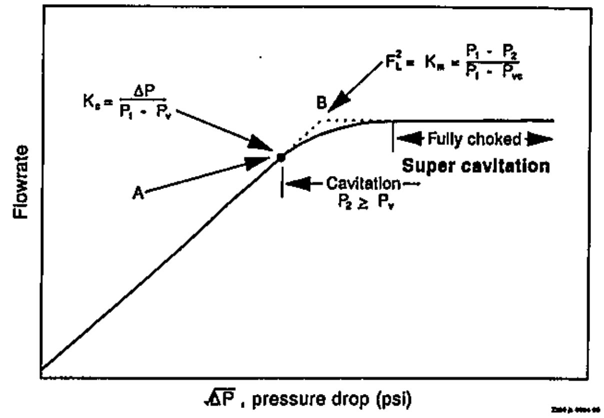


Figure 17. Pressure and flow relationship as liquid flows through a valve.

shown as Point A. The apparent decrease in flow coefficient has been shown to be an indication of cavitation occurring in the valve. This point is the point of initial cavitation and the point at which the curve deviates from a proportional relationship between flow and the square root of the pressure drop. This is the point of incipient cavitation of the liquid stream flowing through the valve. The formation of vapor bubbles causes a crowding condition at the vena contracta, which tends to restrict the amount of liquid mass that can be forced through the valve, and thus decreases the value of C_V.

A further increase in the pressure differential across the valve increases the number of water bubbles that collapse in the downstream side of the valve and produce more cavitation. Eventually, a point is reached (Point B of Figure 17) at which any further increase of pressure differential does not increase flow, and the valve chokes; this point represents severe cavitation.

Note the difference between high-recovery and low-recovery valves. High-recovery valves, such as ball, angle, Y-body, and butterfly types have a very low pressure at the vena contracta, so that vapor forms occurs there at relatively low pressure drops. Therefore, such a valve usually reaches critical flow conditions at a lower pressure drop than a low-recovery valve such as a globe valve. For example, in a globe valve handling water with a very low vapor pressure, the choked flow condition is reached at a pressure

drop greater than 80% of inlet pressure. Under the same fluid conditions, a high-recovery valve can reach choked flow with a pressure drop as low as 30% of the inlet. Hence, a high-recovery valve will cavitate much sooner and also induce cavitation in the pipe, compared with a low-recovery valve. High-recovery valves have much smaller F_L numbers [see Equation (2)] than low-recovery ones, and cause cavitation problems when used in high pressure-drop applications.

Also, it is essential to review supercavitation and its effect on valves and piping. When supercavitation occurs, the valve and pipe for several diameters downstream are completely filled with water vapor. When this happens, the valve and adjacent pipe may not be subject to any cavitation damage. However, at the location downstream where the vapor pocket collapses, the system is subjected to severe cavitation.

To show the destructive damage from supercavitation, a 20-in. butterfly valve was tested. During the test, the valve and piping were fastened to a 3-ft-thick concrete floor with 1/2-in.-diameter cables at 10-ft intervals. The cavitation caused the pipe and valve to have a displacement amplitude on the order of 1/2 in.; also, it generated a force near the valve of approximately 1,500,000 lb.

The maximum pressure drop that is effective in producing flow through the valve or pressure drop at choked flow point B on Figure 4 is given by

$$(P_1 - P_2)\ \text{allowable} = F_L^2(P_1 - F_F P_V) \quad , \qquad (2)$$

where

$P_1\text{-}P_2$ = valve pressure drop

P_1 = valve inlet pressure

P_2 = valve outlet pressure

P_V = vapor pressure of liquid at upstream conditions

F_F = liquid critical pressure ratio

F_L = valve pressure recovery factor determined experimentally.

The equation raises the question, "What is the use of the F_L pressure recovery factor?" F_L was introduced primarily to improve accuracy in flow capacity determination, not for determining cavitation damage or intensity. For example, flow at Point B in Figure 17 already causes severe cavitation.

Therefore, Equation (2) defines the maximum pressure drop that is effective in producing flow through the valve or pressure drop at choked flow.

The subtle point is that many valve manufacturers use F_L to size valves (i.e., if the actual pressure equals the allowable calculated from Equation (2), a manufacturer may recommend that the valve be installed in the piping system). Most valves in our service water system were sized this way and thus induced all of the problems described here. Also, in many cases, the actual valve pressure drop exceeded the allowable pressure drop, which induced supercavitation.

Therefore, it is clear that the pressure drop for choking cavitation equals the pressure drop at the level of cavitation predicted by the pressure recovery factor (F_L). It is recommended that because the pressure recovery factor corresponds to choking cavitation, a valve should never be operated at the conditions predicted by F_L.

The following equation is recommended for predicting cavitation conditions:

$$K_C = \frac{P_1 - P_2}{P_1 - P_V} \quad . \qquad (3)$$

The dimensionless parameter, K_C, called the cavitation index, which is calculated at Point A of Figure 17, provides an indication of the point of cavitation initiation. This point will not cause material damage to valve bodies and downstream pipe.

HOW TO ELIMINATE CAVITATION

The following four approaches *are used together or singly* to eliminate cavitation damage:

- Bypass valve and two valves in series
- Multistage orifices downstream in existing valves
- Materials
- Anticavitation valves.

The first option is to use a smaller bypass valve to provide control in the low range or winter flow condition, and let the existing valve handle the large flows or summer flow conditions. This option was investigated, but we concluded that existing valves would still cavitate during summer flow conditions. One of the disadvantages with the bypass approach is the added complexity of control and associated unreliability of operation when switching from bypass valve to the mainline valve. Also, from the analysis of using two valves in series, concluded that it added complexity and a second valve would still cavitate.

The second option is to use multistage orifices downstream of the existing valves. Downstream breakdown orifices (multistage orifices) may be inserted to artificially increase the back pressure. This is not a preferred method for two reasons. First, the effective flow range of such an installation is very limited. If the valve is throttling at a very low flow rate (in our case during the winter), fluid velocities through the orifice may be so low as to eliminate its effectiveness (that is, the bulk of the pressure drop will be seen by the valve). At higher flow rates the orifice plate will become the primary restriction and, in turn, limit or completely choke off the flow (in our case during the summer). Second, if properly sized to a particular value, the downstream orifice may prevent the valve from cavitating, but it may cavitate itself. Thus, the cavitation has not been eliminated, merely relocated. Critical components immediately downstream of the orifice will still be damaged from cavitation. Our flow requirements for the service water valves are too varied to allow the use of orifices.

The third option is to use alternate materials. An essential consideration in minimizing the damage from cavitation is the selection of materials. Unfortunately, no known material is totally immune from damage at all levels of cavitation. Selecting the best material for cavitation damage resistance for valves and trim can be roughly categorized in relative resistance groups. The grouping in Table 3 is based on references, practical experience, and value manufacturers experience.

The last option is to use anticavitation valves. This is the most effective means of reducing the risk of cavitation damage.

Many valve companies have developed special valves and valve trim to eliminate cavitation, or at least mitigate its effects. Valve designers use combinations of techniques to achieve designs that reduce or eliminate the likelihood of cavitation. These techniques include

- Reducing the pressure in multiple stages
- Directing flow away from the valve and pipe walls
- Breaking the flow into many small streams
- Forcing the flow through multiple turns or tortuous paths.

These four methods are basic combinations of the following two techniques:

- Isolating cavities—multiorifice cage valve
- Dissipating available energy—multistep, tortuous path valve.

Table 3. Relative resistance of common valve and trim metals.

Damage resistant material	Material resistance rating
Stellite (casting, hardfacing) Tool steels 440-C, 420 SS (quenched, tempered) Nickel-based hardfacing 17 Cr-7 Ni weld overlay	Excellent (best)
Inconel Aluminum bronze (casting, overlay) Nickel-aluminum bronze (casting, overlay) 410, 416 SS Austenitic SS (300 series) Monel 500 Chrome-moly steels	Good (moderate)
Manganese bronze Carbon steel Nickel Monel 400	Fair (limited)
Brass Aluminum Cast iron	Poor (low)

This section will discuss the two techniques of reducing and eliminating cavitation.

A multiorifice cage valve does not prevent cavitation, but diverts the damaging implosions away from the plug and into the cage center. The cage uses numerous pairs of small diametrically opposed flow holes through the wall of the cage (see Figure 18). Each hole admits a jet of cavitating liquid, which impacts with the jet admitted from the opposite hole at the center of the cage. Thus, the energy of residual bubble collapse is expended within the fluid and away from the metal surfaces; some cages have three-stage pressure reduction or three sets of separate orifices.

The pressure letdown through a multistep, tortuous path valve is shown graphically in Figure 19. The valve's multiple discrete passage method is shown in Figure 20, which shows one disc. However, each disc has many parallel flow paths, and many discs are stacked to form the valve trim. Each disc's flow passages are opened as the plug moves across the opening in the center of the disc stack. Each path is made tortuous by forcing the fluid to make right-angle turns. Thus, the pressure drop is achieved by the reduction of a velocity head for each right-angle turn. The velocity is controlled in two ways: first, by dividing the flow into many small streams of low mass flow rate, and second, by forcing the fluid through a series of right-angle turns to effect the pressure drop steps, as shown in Figure 20. In this design, the pressure recovery, P2, is considerably above the vapor pressure, and the valve does not cavitate.

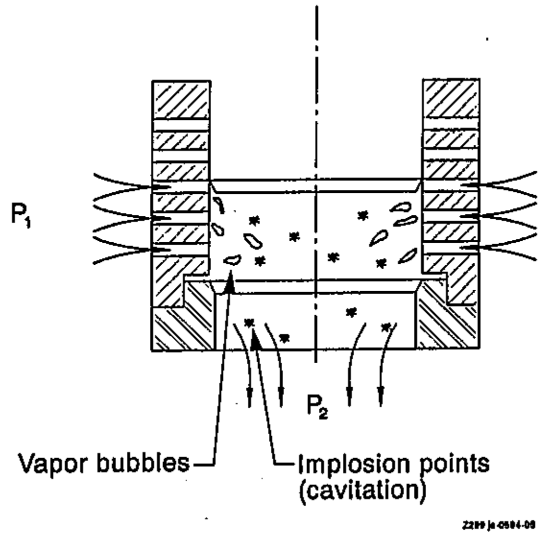


Figure 18. Multiorifice cage valve.

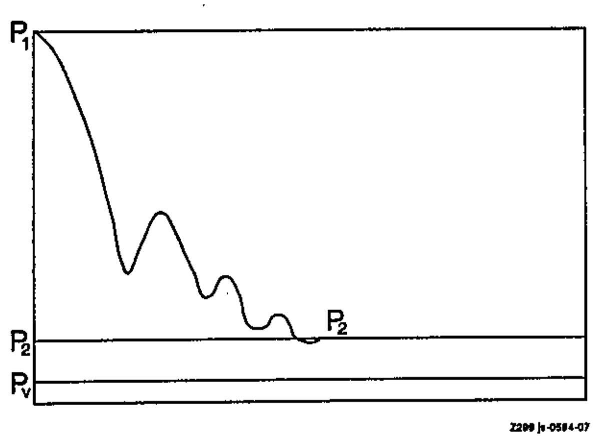


Figure 19. Pressure stages through the control valve trim.

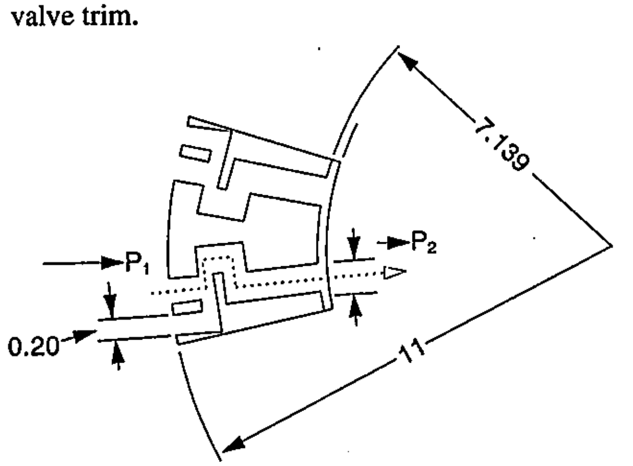


Figure 20. Recommended four-stage trim for the service water control valves—one disc.

CORRECTIVE ACTIONS AND PRACTICAL APPROACHES USED BY COMMONWEALTH EDISON TO ELIMINATE CAVITATION IN THEIR SERVICE WATER SYSTEM

Cavitation-induced damage in service water flow control valves and downstream piping has been reported for all six nuclear stations at Commonwealth Edison. All service water control valves for the six stations have the following common elements:

- All valves are carbon steel
- Fifty percent of the valves are rotary type plug, as shown in Figure 1, which are high-recovery; the other 50% are globe type, which are low-recovery
- The pressure drop across these valves is less than 100 psi; that is, upstream pressure 100 psia and downstream pressure between 15 to 30 psia
- The temperature of the service water is 100 to 120°F.

To correct the cavitation problems, the following three solutions have been tried:

- Changed valve body and pipe materials from carbon steel to 316 SS
- Installed pressure reducing orifices
- Installed anticavitation valves.

Finally, we will discuss some of our successful applications in eliminating cavitation in our service water systems.

The LaSalle service water control valves have a long history of erosion caused by cavitation. There have been 60 different cavitation-induced valve and pipe failures. These failures are typical and identical to the Braidwood failures discussed in this paper. Some of the valves that had holes in the body walls were coated with Belzona.

Belzona, which is resistant to cavitation wear, is used as another deterrent to erosion of the internal valve body walls. But after a year of operation, the valves were found damaged severely from cavitation. Also, vacuum breakers have been installed, but they did not help to reduce the cavitation. Most interestingly, the turbine lube oil cooler valves super-cavitate and have been damaged many times; they are throttled as necessary to maintain the oil temperature requirements for turbine bearings. The turbine bearings were wiped out; the root cause showed that this resulted from cold oil temperature. It is believed that the unregulated (choked flow) opening of the valve to maintain correct oil temperature was the primary contributing factor to the recent turbine bearing failure. To correct this most important problem, two new 12-in. valves with four-stage cavitation trim, as shown in Figure 20, were installed. These valves corrected the cavitation problem.

Currently at Braidwood, as shown in Table 1, some of the carbon steel valve bodies and downstream flanges and reducers have been replaced by stainless steel valve bodies and stainless steel downstream flanges and reducers. Some of these 316 SS flanges and reducers have been inspected after two cycles of operation, and they appear to have no cavitation damage whatsoever. This seems to be a successful solution for pressure drops less than 100 psi.

Byron also installed two stainless steel valves in their service water system. The old valves, which were carbon steel, were damaged severely from cavitation, and the downstream pipe was also damaged.

At Quad Cities, extensive cavitation erosion and vibration has been experienced the last 15 years in all five valve bodies on the service water system for the reactor building closed component cooling water system. To correct this problem, two new 12-in., four-stage anticavitation valves, as shown in Figure 18, were installed. This corrected the cavitation and vibration problem. Also, a three-stage pressure reducing orifice was installed in the diesel generator cooling water system. The old system had a one-stage pressure reducing orifice; this orifice super-cavitated and induced pipe vibration and pipe erosion, and it was difficult to control the flow. The new three-stage orifice eliminated all cavitation-induced vibration and flow control problems.

Dresden also installed two, 12-in., four-stage anticavitation valves in their reactor building closed cooling water service water system. Their old carbon steel globe valves experienced the same damage as the Quad Cities valves.

Extensive and significant cavitation erosion has been experienced and reported on service water control valves at Zion Station. For example, cavitation has induced extensive erosion in the valve bodies of the diesel generator system. Some of the valve bodies have been replaced with new ones; to correct the cavitation problem on the diesel generator system, pressure reducing orifices were installed upstream of the valves. Plant engineers hoped that this would reduce the cavitation, but the downstream valves were damaged by cavitation beyond repair. The upstream orifices were cavitating and inducing vapor pocket collapse in the downstream valve bodies, and one of the downstream valves stems was cut in half by cavitation and the disc had fallen off the stem.

To correct this problem, two new valves with two-stage anticavitation trim, as shown in Figure 18, were installed. Also, three 18-in. globe valves on the closed component cooling water service water system experienced severe cavitation damage. All three valve bodies were damaged by cavitation, and one valve stem was broken by cavitation-induced vibration; also, one guide busing had fallen off. This problem was corrected by installing three 18-in., two-stage anticavitation valves, as described before. Finally, to eliminate cavitation downstream of the containment ventilation cooler valves, a three-stage pressure reducing orifice was installed.

CONCLUSIONS

Control valve cavitation is one of the most difficult problems facing operating nuclear plants and a major source of operational and maintenance costs. In most cases, the conditions that produce cavitation can be predicted with current analysis methods, but providing a working solution is often difficult and always expensive. To avoid the costly plant shutdown associated with cavitation-induced failure and ensure continued plant operation, effort to monitor and inspect as well as repair damage in service water valves and piping is vitally important. Such effort is a part of our practice, as demonstrated by the examples from the six nuclear stations given in this paper.

Finally, and most importantly, all of these cavitation-induced valve failures were caused by a valve pressure drop less than 100 psi. Also, none of the anticavitation trim installed in our service water valves has trapped or clogged the small flow passages with line trash, such as weld beads, slag, scale, silt, or other large solid debris that might be in the service water line.

ACKNOWLEDGMENTS

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Session 2A

MOV Industry Research Results

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EPRI MOV Performance Prediction Program

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ABSTRACT

An overview of the EPRI Motor-Operated Valve (MOV) Performance Prediction Program is presented. The objectives of this Program are to better understand the factors affecting the performance of MOVs and to develop and validate methodologies to predict MOV performance. The Program involves valve analytical modeling, separate-effects testing to refine the models, and flow-loop and in-plant MOV testing to provide a basis for model validation. The ultimate product of the Program is an MOV Performance Prediction Methodology applicable to common gate, globe, and butterfly valves. The methodology predicts thrust and torque requirements at design-basis flow and differential pressure conditions, assesses the potential for gate valve internal damage, and provides test methods to quantify potential variations in actuator output thrust with loading condition. Key findings and their potential impact on MOV design and engineering application are summarized.

BACKGROUND

During the mid to late 1980s, motor-operated valve (MOV) failures or incidents in U.S. nuclear power plants resulted in an increased emphasis by both the U.S. nuclear industry and the U.S. Nuclear Regulatory Commission (USNRC) on improving the performance, reliability, and predictability of MOVs. In response, the Electric Power Research Institute (EPRI) initiated efforts to document existing MOV maintenance and engineering evaluation technology in the form of technical repair and engineering application guides and initiated a study to assess long-term industry needs.

MOV Application Guides

Between 1988 and 1990, EPRI's Nuclear Maintenance Application Center (NMAC) worked with utility experts to develop several technical repair guidelines for Limitorque motor actuators. In addition, NMAC developed the *Application Guide for Motor-Operated Valves in Nuclear Power Plants*[a] to document the existing state-of-the-art in conducting engineering evaluations of MOV applications. The scope of this document included rising stem gate and globe

a. W. Grant and R. Keating, EPRI Report NP-6660-D, 1990.

valves powered by Limitorque motor actuators. The Application Guide addressed definition of MOV requirements for an application, determination of required stem thrust for gate and globe valves, and determination of available thrust from Limitorque actuators. A similar guide covering butterfly valves was developed by NMAC in 1993.[b]

At the time these guides were developed, areas of uncertainty were recognized in MOV performance prediction. Examples included disc-to-seat friction coefficients for gate valves, guide friction coefficients for gate valves, flow loading and mid-stroke effects for gate valves, unwedging thrust for gate valves, sealing load for gate and globe valves, stem-to-stem nut friction coefficients, and variation in operator output thrust at torque switch trip with differential pressure (DP) loading condition. In addition, uncertainty existed in methods for predicting hydrodynamic torque loading on butterfly valve discs.

Definition Study for EPRI MOV Performance Prediction Program

During 1989, EPRI conducted a planning study to determine the extent to which the known areas of uncertainty in existing predictive methods could be addressed by a generic test program. This study also examined available data from other test programs and from in-plant tests to assess how these data could be used in such a program. To ensure industry needs were appropriately identified, the study was coordinated with the utility MOV User's Group (MUG), as well as the Nuclear Utility Management and Resources Council (NUMARC). The result of the study was a recommendation for a generic MOV research program.

As part of the study, a preliminary database of MOV applications in nuclear power plants was created. Review of the database indicated that approximately 50% of the 16,000 safety-related MOVs in U.S. nuclear units are gate valves, 20% are globe valves, and 20% are butterfly valves. The remaining 10% are other specialty-type valve designs. Valve population distributions were also defined in terms of size, pressure class, manufacturer, system, and DP. This distribution demonstrated that the valve population was very diverse, with only limited instances of identical or similar valves in service at multiple locations. This finding meant an approach to assess MOV performance using "type testing" would be impractical and cost-prohibitive.

The study recommended that an MOV test and analysis program be conducted with the objective of providing improved and validated methods for predicting MOV performance. The scope of the program was to cover common gate, globe, and butterfly valves used in safety-related applications. It was recommended that the necessary modeling to develop improved methods follow a first-principles approach that addressed known areas of uncertainty identified during development of the application guides. A combination of separate-effects testing, flow-loop testing, and enhanced in-plant testing was recommended as the most effective approach to provide the needed data. These recommendations laid the fundamental groundwork for the EPRI MOV Performance Prediction Program (PPP). Based on the results of this study, EPRI formed a utility Technical Advisory Group (TAG) composed of utility industry MOV experts to provide guidance to EPRI in the detailed formulation and execution of the Program. The Program was formally initiated in the fall of 1990 and is scheduled for completion in July 1994.

OBJECTIVES AND SCOPE

The objectives of the EPRI MOV Performance Prediction Program are to

1. Provide short-term products to utilities to allow expeditious evaluation of MOVs based on existing technology. The short-term program includes the following activities:

b. B. H. Eldiwany and M. S. Kalsi, EPRI Report NP-7501.

a. Development of an in situ test guide

b. Development of a computerized MOV general information database

c. Development of an MOV margin improvement guide

d. Review of a USNRC-sponsored gate valve test program conducted by the Idaho National Engineering Laboratory (INEL)

2. Conduct a long-term program to develop and validate improved methods for predicting MOV performance. Such methods can be used to demonstrate the design-basis capability of MOVs in cases when no unique design-basis test data are available for a valve. Key aspects of the long-term program are summarized as follows:

   a. Develop improved methods for prediction or evaluation of the following:

      (1) System flow parameters—calculation of differential pressure versus stroke position

      (2) Gate valve performance—calculation of required thrust and potential for internal damage

      (3) Globe valve performance—calculation of required thrust

      (4) Butterfly valve performance—calculation of required torque

      (5) Motor operator dynamic performance—quantification of variations in actuator output thrust with DP loading condition

   b. Perform separate-effects tests to provide information for refinement of the gate valve and operator methods:

      (1) Gate valve friction—determine friction coefficients and damage thresholds for gate valve internal components

      (2) Gate valve design effects—understand the interaction of gate valve internal components

      (3) Operator separate effects—understand observed variations in actuator output thrust with DP loading condition

      (4) Operator stem-to-stem nut lubricant performance—establish qualitative lubricant comparison data

   c. Conduct numerous MOV tests to provide data for model and method development and validation:

      (1) Flow loop testing of 34 gate, globe, and butterfly MOVs under a wide range of flow conditions

      (2) Flow loop testing of six butterfly valve disc designs to assess flow and inlet piping effects

      (3) In-plant (i.e., in situ) tests of 28 MOVs.

SHORT-TERM PROGRAM

The products developed to support near-term utility evaluation, testing of MOVs, definition of the scope, and focus of the EPRI Program are reviewed below.

In Situ Test Guide

The *In Situ Test Guide for Motor-Operated Valves* was prepared to provide guidance on the requirements for in-plant test data. The test guide considered existing industry experience by incorporating elements of the in situ testing guide developed by the MUG Thrust Calculations and Switch Settings Committee. The In Situ Test Guide includes requirements for test instrumentation accuracy and recording speed; valve inspection, measurement, and documentation; and overall documentation of the data package. For the enhanced in situ tests used in the EPRI MOV Performance Program Prediction,

additional requirements include measurement of time-history DP across the valve, as well as internal dimensions for gate valves.

MOV General Information Database

Utilities identified that an MOV database would be helpful to facilitate communication among nuclear utilities on MOV-related issues. In response to this need, EPRI prepared a personal computer (PC)-driven database covering over 5,000 nuclear safety-related MOVs. The database contains over 40 fields of information for each MOV and is updated approximately semiannually. Each participating utility has a copy of the database on diskette. In addition to facilitating communication among utilities, the database also provided a basis to assess the MOV population for selecting valves for the flow loop and in situ test programs.

USNRC/INEL Gate Valve Test Review

During 1988–1990, the INEL conducted gate valve testing for the USNRC (Steele and DeWall, 1990). The primary emphasis was to evaluate gate valve performance in boiling water reactor (BWR) blowdown isolation service in resolution of the USNRC Generic Issue 87. One key conclusion from the test program was that use of the standard industry equation in combination with disc factors historically assumed by the valve vendors may under-predict gate valve stem thrust requirements under certain flow conditions. In addition, some gate valve designs were found to be susceptible to damage during closure under blowdown flow conditions. The damage was attributed to high contact loading on the guides and the seats due to disc tipping. The tests also provided strong evidence that there were direct loads applied by the flow to the disc in not only the pipe-axis direction but also the stem-axis direction. These loads resisted the valve's opening motion and are considered to be the result of Bernoulli forces.

EPRI reviewed the USNRC/INEL tests. This review identified the apparent disc factors associated with various valve designs and conditions tested. This effort also included detailed inspections of the INEL valves and measurements of key dimensions and material properties. Finally, the review identified specific areas where the information and experience from this testing could be factored into the modeling and test activities being planned as part of the EPRI MOV Performance Prediction Program.

Margin Improvement Guide

The *Motor-Operated Valve Margin Improvement Guide* supplements the Application Guides by identifying specific actions that can be implemented to increase available actuator margin. The Guide also references the appropriate sources of additional information to support evaluation of specific situations. It incorporates detailed guidance for MOV limit switch configurations and settings, which were developed by the MUG Thrust Calculations and Switch Settings Committee.

LONG-TERM PROGRAM

The long term program includes development of improved methods for prediction of valve performance and motor-operator dynamic effects, separate-effects testing to provide a basis for method refinement, and flow-loop and in-plant MOV testing to support method validation. The following paragraphs offer a description of each long-term program activity and, where appropriate, a summary of key findings.

PREDICTIVE METHODS

System Model

To predict the performance of valves, it is desirable to accurately predict the DP across the valve disc over the full range of stroke positions. To accomplish this task, a computer-based system model was developed. The model allows simulation of a variety of piping configurations, including single and parallel line pumped flow

configurations with up to two active MOVs, as well as a single line blowdown configuration.

The system model requires plant engineering inputs to define the frictional and flow driver characteristics of the piping system in which the MOV is installed. The system model predicts the DP across the MOV at all stroke positions. The predicted DP versus disc position relationship is used as input to the gate, globe, and butterfly valve models to determine disc loading at all positions. The system model is separately validated by comparing predicted DP versus position with that measured during flow-loop and in-plant testing.

Gate Valve Model

The development of a computer-based gate valve model represents the most challenging goal of the EPRI MOV Performance Prediction Program. At the outset, it was determined that the gate valve model would need to address the fluid loading on the disc and the detailed mechanical interaction between the stem, disc, guides, and seat, including the potential for material damage at sliding interfaces.

Computational fluid dynamics analyses were performed to evaluate fluid loading. A simplified algorithm was incorporated into the model, based on the results of these analyses, to compute vertical and horizontal forces on the disc at all disc positions as a function of valve DP. A detailed mechanical model that determines disc force equilibrium over the full range of disc positions was then developed. The model accounts for disc tipping within the constraints of the guides. Results from gate valve design separate-effects testing were used to refine the model and to verify that the disc behavior is being properly calculated. Based on the results of friction separate-effects testing, a friction algorithm was added to the mechanical model to determine friction coefficient as a function of the material pair, contact mode, contact load (or stress), and fluid temperature. The resultant model can predict the stem thrust to move the disc for both opening and closing strokes using DP as a function of stroke position as input. Material damage is also predicted under modes where damage is likely to occur.

The sliding friction coefficients in the gate valve model are intended to be bounding values. The model allows the user to input a value manually for the disc-to-seat sliding friction coefficient to replace the value determined by the friction algorithm. Methods are provided to determine disc-to-seat sliding friction coefficients from valve-specific test results. These methods include use of data obtained from full DP tests, partial DP tests, hydropump DP tests, and static (zero DP) tests. The required valve internal information (dimensions and materials) for input can be obtained from the specification provided as part of model documentation by the valve manufacturers. Alternately, valve-specific internal measurements, if available, can be used to define input parameters.

The gate valve model is applicable to solid and flexible wedge gate valves with single piece discs, a conventional guiding arrangement with guide rails and slots, and a stem-to-disc connection consisting of a T-head and a T-slot. The gate valve model is validated by comparing predictions of thrust to data obtained from flow loop, in-plant, and previous INEL (Steele and DeWall, 1990) testing.

Globe Valve Model

The globe valve model predicts thrust requirements under DP loading for the full range of the valve stroke positions. The model is applicable to globe valves with T-pattern or Y-pattern bodies, rising or rising rotating stems, and balanced or unbalanced discs. Both underseat and overseat flow configurations can be accommodated. The model can compute required thrusts under incompressible, pumped flow, conditions. The model is computer based and is validated by comparison of predicted thrust with that measured during flow-loop and in situ MOV testing.

It is necessary to select the appropriate disc area (either disc seat or disc guide area) for DP application in order to accurately predict required thrust. The globe valve methodology provides

guidance in selecting the appropriate disc area based on valve internal design characteristics.

Butterfly Valve Model

The butterfly valve model determines the required torque to operate butterfly valves through their full range of stroke positions. Two types of torque calculations are performed: a seating/unseating torque that applies when the disc is near the fully closed position and a total dynamic torque that applies throughout the remainder of the stroke. Seat torque can be predicted for seats that are new or well-maintained; however, use of valve-specific test results is recommended for valves where the seats may have degraded or aged. Total dynamic torque is predicted using a bearing torque component and a hydrodynamic torque component. The key advances in this technology are the development of generic hydrodynamic torque coefficients that account for disc type, orientation, and aspect ratio, as well as multipliers to account for the influence of upstream elbows.

The butterfly valve model is computer based and is applicable to symmetric, single offset, and double offset valves with circular discs. The model can predict torque requirements with flow in either direction. The model applies to incompressible flow and compressible choked flow and is validated by comparing torque predictions to data obtained from flow-loop, in situ, and previous INEL (Steele et al., 1986) testing.

Gate Valve Empirically Based Methods

In addition to the computer-based model that addresses conventional solid and flexible wedge gate valve designs, manual calculational methods are being developed to address the following unique gate valve design configurations:

- Parallel double-disc with internal wedge
- Flexible wedge with pin-and-link stem connection
- Split wedge valve with ball and socket joint
- Split wedge valve with spacer ring joint
- Parallel expanding valve.

These methods provide guidance in applying EPRI flow-loop and in situ testing results to plant-specific MOV applications of these valve designs.

Operator Dynamic Effects Methods

EPRI and industry MOV testing revealed that a significant reduction in actuator output thrust at torque switch trip can occur when the valve is loaded slowly (i.e., under DP conditions) relative to the observed thrust output at torque switch trip under static (no DP) conditions. This phenomenon has been called the "rate-of-loading" effect or "load sensitive behavior." Because torque switches are generally set under static conditions, it is possible that insufficient thrust capability will exist when the valve is subjected to design basis DP and flow conditions.

Separate-effects testing was conducted to better understand the root cause for the "rate-of-loading" phenomenon and to develop methods for quantifying the potential effect for a given installed MOV. Based on this testing and testing conducted by Steele et al. (1992), it was concluded that this phenomenon is attributable to specific characteristics of the stem/stem-nut and lubricant combination and is not amenable to analytical treatment. Some level of unique testing is necessary to assess the effect accurately for a given MOV.

Several alternative approaches are being assessed to provide utilities with the means for accommodating potential "rate-of-loading" effects. These methods are summarized as follows:

- Impose a margin penalty if no valve specific data are available.
- Set the torque switch with a reduced loading rate (i.e., by use of the handwheel). This

method requires thrust measurement, but torque measurement is not required. While some margin penalty is required with this approach, the magnitude of the margin penalty can be minimized if torque measurements are also made.

- Set the torque switch with a DP load simulator device. If successful, this device will accurately reproduce the maximum coefficient of friction that could occur at the stem/stem-nut interface under design-basis DP loading conditions. This method requires only a thrust measurement and should require only minimal margin penalties.

- Use one of two approaches recommended by the INEL. These methods are deemed the "threshold" method and the "fold line" method. These methods require the measurement of both thrust and torque and involve testing under static or relatively low DP conditions.

- Modify the control switch logic to bypass the torque switch until flow isolation, but not necessarily leak tightness, is achieved. The torque switch would be set at a nominal setting so as not to impose excessive thrust loading during static tests. This approach would eliminate the need to add margin to accommodate potential "rate-of-loading" effects, but would still require evaluation of actuator capability to achieve the required thrust while the torque switch is bypassed.

SEPARATE EFFECTS TESTING

Friction Testing

Test fixtures were fabricated to determine sliding friction coefficients and damage threshold load levels for the range of material pairs, contact geometries and stresses, and the water/steam temperatures and pressures typically found in gate valves installed in nuclear power plants. Four predominant material combinations were tested: stellite 6 on stellite 6; stellite 6 on carbon steel; stellite 6 on stainless steel; and carbon steel on carbon steel. Tests were conducted under water and steam conditions at temperatures ranging from room temperature to 650°F.

Stellite on stellite sliding friction coefficients under room temperature water conditions were found to increase significantly, from approximately 0.2 to greater than 0.6, and eventually "plateau" at a maximum value as the number of strokes is increased. Hot water and steam friction coefficients for stellite on stellite were found to be lower than cold water "plateau" sliding friction coefficients and did not vary significantly with stroke number. Carbon steel on carbon steel friction coefficients and the potential for gouging damage were found to increase significantly as the temperature increased from 70 to 120°F.

Gate Valve Design Effects Testing

To ensure that the gate valve model would accurately predict disc orientation, contact points, and damage threshold levels, a test fixture was fabricated in which actual gate valve internal parts could be transiently loaded with hydraulic pistons to simulate DP loading conditions. A comprehensive set of parametric tests was carried out to assess the influences of variation in guide lengths, guide clearances, guide materials, and disc and body seat edge radii. The results revealed that disc and body seat edge radius or chamfer are critical parameters affecting valve performance and the potential for valve internal damage. The results of this testing were used as a basis for refinement of the assumptions made in the gate valve model.

Operator Dynamics Testing

To support development of the Operator Dynamic Effects Methods described earlier, a test fixture was fabricated to simulate the full range of MOV loading conditions. The test fixture incorporated a hydraulic cylinder that could provide a pre-programmed back loading on the end of the stem as the actuator attempted to move the stem in the closing direction. The test fixture also included a hard stop to simulate high loading

rates typical of those that occur in gate or globe valves during wedging or seating. A comprehensive set of parametric tests is being conducted in which the effects of variation in loading level, loading rate, stem/stem-nut combinations, and stem/stem-nut lubricants are evaluated. The Operator Test Fixture is being used to assess and validate a variety of approaches to account for the "rate-of-loading" effect in MOV switch set up and margin determination.

Detailed assessment of the test results indicates that the "rate-of-loading" effect is caused by a transient reduction in the stem/stem-nut coefficient of friction when the stem is loaded at a high rate (i.e., during a static closure). This effect is postulated to result from the fact that under high loading rate conditions, the load is increased to a high level in a very short time before the lubricant can be fully squeezed from the stem nut threads. During this short time (<100 ms), a mixture of hydrodynamic and boundary lubrication modes is in effect and can result in very low friction coefficients that occur briefly and change over time. If the actuator has been running for sufficient time at a sufficiently high load, most of the grease is squeezed out, resulting in predominantly boundary lubrication and somewhat higher friction coefficients typical of those expected under DP loading conditions. Only a fraction of the stem/stem-nut combinations tested exhibited a significant "rate-of-loading" effect. These findings and general conclusions regarding the cause of the phenomenon are consistent with those documented by the INEL (Steele et al., 1992).

Stem/Stem-Nut Lubricant Testing

A separate-effects test program was conducted to assess the friction and wear characteristics of various greases and solid films that are now or could be applied as stem/stem-nut lubricants in MOVs. A total of 21 lubricants were evaluated in a test fixture designed to simulate an MOV application. The effects of stroke number and loading level on friction and wear were evaluated. Although the maximum friction coefficients ranged from approximately 0.1 to 0.2 for the grease-type lubricants tested, most were bounded by a value of approximately 0.15. Friction coefficients were generally observed to decrease with increasing stroke number. The solid lubricants tested exhibited poor performance.

Data from these tests can be used by utilities as a basis for qualitatively comparing the performance of the lubricants tested. The results are not used directly as part of the predictive methodology.

FLOW-LOOP/IN SITU MOV TESTING

Full-Scale MOV Testing

Flow-loop testing was conducted in four flow loops, three of which were located in the U.S. and the fourth in Karlstein, Germany. Thirty-four MOVs were subjected to a combined total of more than 1,200 formal test strokes. The text matrix covered a wide range of flow, temperature, and DP conditions. Twenty-eight gate, four globe, and two butterfly valves were tested. Valve test candidate size, pressure class, and manufacturer were selected based on their predominance in the MOV General Information Database and on availability. Gate valves tested ranged in size from 2-1/2 to 18 in., and globe valve sizes ranged from 2-1/2 to 6 in. Two 6-in. butterfly valves were tested.

Prior to testing, all MOVs were disassembled, and comprehensive internal measurements and photos were taken. The valves and actuators were then reassembled and prepared for testing. All MOVs were tested under "baseline," 15-ft/s, cold-water flow conditions over a range of DPs up to the maximum expected in nuclear power plant applications. After cold-water testing, selected valves received parametric testing to assess the effects of variation in fluid temperature, fluid velocity, and flow media. Tests were conducted with cold-water flow velocities up to 50 ft/s, as well as hot water and steam blowdown flow conditions. Differential pressures ranged from zero to 2,650 psid. Internal valve

inspections were conducted at regular intervals to assess any valve internal damage.

High speed data were acquired to record the following parameters:

- Valve DP
- Valve inlet pressure
- Fluid temperature
- Fluid flow rate
- Direct stem force
- Direct stem torque
- Spring pack displacement
- Disc position
- Motor voltage
- Motor current
- Motor active power
- Limit and torque switch actuation.

To minimize the potential impact of the "stroke effect" on valve disc friction coefficients during planned flow-loop parametric testing, all gate valve seats were "preconditioned" by short stroking the valve into the seats under DP loading until the sliding friction coefficient reached a maximum "plateau" level. The number of strokes required to reach the "plateau" level of friction varied widely from approximately 100 to as many as 900 strokes.

Once gate valves had been "preconditioned," they were subjected to full-flow tests at cold-water flow velocities ranging from 15 to 50 ft/s. With one exception, "apparent" disc friction coefficients ranged from 0.2 to 0.9 during these tests. These "apparent" disc coefficients of friction include all valve performance phenomena and are not necessarily representative of sliding friction alone. One valve design exhibited very high thrust requirements to close under cold-water, pumped-flow conditions. The "apparent" disc coefficient of friction for this valve was approximately 1.9. The valve manufacturer's evaluation concludes that some of the test valve internal dimensions were outside manufacturing tolerances.

Although a significant range of apparent disc friction coefficients was observed, only a single gate valve sustained internal damage under cold-water, 15-ft/s flow conditions. In this case, the disc was pushed through the seats, allowing leakage above the disc on an 18-in. valve with a 3-degree seat half angle. At higher flow velocities (i.e., greater than 30 ft/s), the body guides were plastically bent in the flow direction in one valve design tested. This valve design incorporates cantilevered body guide rails.

Hot-water, pumped-flow testing resulted in lower sliding friction coefficients than for cold water, generally ranging from 0.3 to 0.5. The "stroke" effect on sliding friction was not observed to be significant under hot water conditions. Under hot water and steam blowdown conditions at 1,200 psid, "apparent" disc friction coefficients ranging from approximately 0.3 to 0.8 were observed. Although some valve designs were undamaged, others sustained significant guide and/or seat damage.

Disc-to-body-seat-friction sliding coefficients were found to decrease with increasing DP. This finding confirms the friction separate-effects testing results and supports the use of sliding friction coefficients obtained from reduced DP test results in the evaluation of thrust requirements under full DP conditions for gate valves in pumped flow systems.

Testing of globe valves under incompressible flow conditions revealed that it is necessary to select the appropriate area (either disc seat or guide area) for DP application, in order to predict required thrust accurately. Under compressible flashing flow conditions, excessive thrust loading was observed that exceeded even guide-area-based predictions. Side loading of the disc from

pressure variations within the valve body may play a role in this phenomenon.

Vendor methodologies for predicting required hydrodynamic torque for butterfly valves are generally proprietary, and as a result, little can be concluded at this time regarding the suitability of such methods based on Program test results. The EPRI butterfly valve model accurately predicts butterfly valve performance for the valves tested in the flow loops.

Subscale Butterfly Valve Parametric Testing

For a comprehensive assessment of the effects of butterfly valve disc design and upstream elbow effects, a parametric test series was conducted at small scale. Selected results were compared with large-scale data for the same disc design to confirm scaling relationships. An existing test facility was modified to allow the insertion of six different butterfly disc designs. Baseline testing was conducted on each disc design to assess flow and DP effects on the required hydrodynamic torque. In addition, selected disc designs were parametrically tested to assess upstream elbow distance and orientation, as well as flow direction, effects.

In Situ MOV Testing

To supplement the flow-loop testing of MOVs, data from 28 MOV tests conducted in nuclear plants are being obtained and formally documented. In situ test data are being obtained for 19 gate, one globe, and eight butterfly valves. Tested DPs ranged from zero to 2,880 psid. The test data obtained generally included high-speed data acquisition to measure and record the following parameters:

- Stem thrust (gate and globe valves)
- Stem torque (butterfly valves)
- Valve upstream pressure
- Valve differential pressure
- Motor current
- Spring pack displacement
- Actuator control switch actuation.

In addition, internal measurements were obtained on gate valves to support validation of the gate valve model.

The in situ data are used to demonstrate the capability of the MOV Performance Prediction Program methodologies to predict the performance of "real world" valves installed in nuclear power plants.

CONCLUSIONS

The research conducted as part of this Program has resulted in a giant step forward in the general understanding of MOV behavior and the ability to predict MOV performance accurately. As a result of this Program, fully validated methods will, for the first time, be available to confirm the adequacy of existing MOV installations and control switch settings and to support the evaluation of MOV modifications or replacements. The lessons learned from this Program should be factored into future valve and actuator development and into the design of advanced nuclear plant systems. Specific conclusions are summarized as follows:

- Cold water stellite on stellite sliding friction coefficients can be highly variable, ranging from less than 0.2 to greater than 0.6.
  - Friction coefficient variation appears to be based on the number of loaded strokes applied and the contact stress level.
  - Friction coefficients increase with stroke number to a maximum "plateau" level, then stabilize.
- Hot water stellite on stellite sliding friction coefficients are less variable, generally ranging from 0.3 to 0.5.
- Under cold-water pumped-flow conditions, gate valve "apparent" disc friction

coefficients can, in isolated cases, exceed 0.6, depending on valve-specific internal design characteristics.

- Under pumped-flow (~15 ft/s) conditions, the potential for internal damage to gate valves is extremely low.

- Under high-velocity flows (>30 ft/s up to blowdown conditions) the potential for valve internal damage increases significantly for some gate valve designs.

- Edge radii or chamfers on gate valve disc and body seats, as well as disc guide slot and body guides, can have a profound impact on the potential for valve internal damage. Sharp edges should be avoided.

- Under pumped-flow conditions, gate valve disc to seat sliding friction coefficients tend to decrease with increasing DP. This finding supports the use of friction coefficients measured under reduced DP conditions when thrust requirements at higher DPs are evaluated.

- Under incompressible flow conditions, globe valve thrust requirements can be predicted accurately if the appropriate disc area (seat versus guide) is assumed for DP application. The EPRI methodology provides guidance in selection of the appropriate area based on specific globe valve internal design features.

- Under compressible flow conditions, globe valve thrust requirements can exceed even guide area based predictions. Side loading on the plug may play a role in this phenomenon.

- For some butterfly valve designs and flow combinations, hydrodynamic torque loading can dominate total torque requirements.

- For some gate and globe valves, a significant reduction in motor-operator output thrust can occur under dynamic (DP loading) conditions relative to the output thrust attained under static (no DP) conditions.

- The EPRI MOV Performance Prediction Program provides validated methods to bound thrust/torque requirements appropriately for common gate, globe, and butterfly valves and several alternative approaches for accommodating potential "rate-of-loading" effects on actuator output thrust.

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Mr. Michael Eidson, Southern Nuclear Company (TAG Chairman)

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EPRI Flow-Loop/In Situ Test Program for Motor-Operated Valves

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ABSTRACT

The Electric Power Research Institute is undertaking a comprehensive research program to develop and validate methods for predicting the performance of common motor-operated gate, globe, and butterfly valves. To assess motor-operated valve (MOV) performance characteristics and provide a basis for methods validation, full-scale testing was conducted on 62 MOVs. Tests were performed in four flow-loop facilities and in nine nuclear units. Forty-seven gate, five globe, and 10 butterfly valves were tested under a wide range of flow and differential pressure conditions. The paper describes the test program scope, test configurations, instrumentation and data acquisition, testing approach, and data analysis methods. Key results are summarized.

BACKGROUND

During the mid to late 1980s, motor-operated valve (MOV) failures and incidents in U.S. nuclear power plants resulted in an increased emphasis by both the U.S. nuclear industry and the U.S. Nuclear Regulatory Commission (USNRC) on improving the performance, reliability, and predictability of MOVs. In response, the Electric Power Research Institute (EPRI) undertook a comprehensive research program to better understand the factors that influence MOV behavior and to develop and validate improved methods for predicting MOV performance. The program included full-scale flow-loop and in-plant (in situ) testing to assess MOV performance characteristics and to provide a basis for methods validation.

OBJECTIVES AND SCOPE

The overall objectives of the Electric Power Research Institute (EPRI) MOV Test Program are to assess valve design specific performance characteristics and to provide a data base against which improved MOV predictive methods can be validated. Specific objectives include

- Determine the influence of differential pressure (DP) and flow velocity, fluid temperature and thermodynamic state, inlet piping configuration, valve stem orientation, and flow direction on the thrust or torque requirements to open or close common gate, globe, and butterfly valves.
- Determine valve damage thresholds of DP and flow velocity.
- Determine the influence of stroke and load history on gate valve thrust requirements under cold-water conditions (preconditioning).
- Determine the influence of load history on actuator thrust output at torque switch trip during gate and globe valve closure strokes.

The scope of the test program included testing of 34 MOVs in flow loops and 28 MOVs in nuclear power plants. A total of 47 gate, five globe, and 10 butterfly valves were tested. Valve test candidate size, pressure class, and manufacturer were selected based on their predominance in the industry safety-related valve population and on availability for test.

Tests were conducted under flow conditions ranging from low pressure, pumped flow, to high pressure blowdown. To the extent possible, the range of test conditions was selected to envelop the maximum expected design-basis DP and flow conditions for each valve design in nuclear power plant applications.

FLOW LOOP TESTING

Test Matrix

Table 1 presents the matrix of tests conducted as part of the EPRI Flow-Loop Test Program. The values shown under each test condition represent the highest differential pressure to which the valve was tested under that condition. Under each test condition, a sequence of approximately 20 valve strokes was conducted. Test sequences conducted under pumped flow conditions (flow velocities ranging from 15–50 ft/s) generally included openings and closures at 33, 66, and 100% of the DP shown in the test matrix. In many cases, repeat tests were conducted at each DP level. Test sequences conducted under blowdown conditions generally included only one blowdown closure stroke. In some cases, when the valve was undamaged after blowdown closure, a blowdown opening stroke was conducted at 100% of design-basis DP, and blowdown closures and openings were conducted at reduced DP.

Description of Test Systems

Four separate test facilities were used to conduct the Flow-Loop Test Program. The following is a description of each test facility:

- The Low-Pressure, Cold-Water, Pumped-Flow Loop is located at the Wyle Laboratories facility in Huntsville, Alabama. Figures 1 and 2 present a schematic drawing and photograph of the loop, respectively. The loop is a closed system consisting of a 60,000-gallon water reservoir, five pumps, a 6-in. test section leg, a 10-in. bypass leg, and a number of control valves and related hardware. The loop can accommodate cold-water, pumped-flow testing up to 250 psid at flow velocities as high as 50 ft/s.

- The Intermediate-Pressure, Cold-Water, Pumped-Flow Simulation Flow Loop is also located at the Wyle Laboratories facility in Huntsville, Alabama. Figures 3 and 4 present a schematic drawing and photograph of the loop, respectively. The intermediate pressure loop is an open, cold-water flow system consisting of four high pressure nitrogen tanks that drive water from a 450-ft^3 vessel through a 10-in. valve test section, which then exits to the environment. The Intermediate-Pressure Loop can accommodate cold-water pumped-flow simulation conditions with DPs as high as 1800 psid.

- The High-Pressure, Cold- and Hot-Water Blowdown Flow Loop is located at the Wyle Laboratories facility in Norco, California. Figures 5 and 6 present schematic drawings of the loop in the pumped-flow simulation and hot-water blowdown configurations, respectively. The loop is an open system consisting of a large nitrogen-driven tank connected to a 300-ft^3, 3,100-psi accumulator vessel (V_2). Water is driven from V_2, through the MOV test section, and is discharged to an open collection tank. The loop can accommodate 6-inch valve testing under both cold and hot water pumped flow simulation and hot-water blowdown conditions up to 2,650 psid.

- The High-Pressure, Cold-Water, Pumped-Flow Simulation and Steam Blowdown Flow Loop is located at the Siemens/Kraftwerk Union (KWU) facility in Karlstein, Germany. Figures 7 and 8 present schematic drawings of the test loop in the cold-water pumped flow simulation and steam blowdown configurations, respectively. Figure 9 is a photograph of the facility showing an installed MOV ready for testing.

Table 1. Flow-loop test matrix—34 valves and 62 test sequences.

No.	Valve Type	Manufacturer	Size (Inch)	ANSI Class/ Material	Limitorque Actuator SMB-Type	Ambient Water 15 FPS MAX DP	Ambient Water 30-50 FPS MAX DP	450°F Water 15 FPS MAX DP	500°F Water Blowdown MAX DP	Sat. Steam 200 FPS MAX DP	Sat. Steam Blowdown MAX DP	Alternate Configuration Testing Notes
1	FWG	Anchor Darling	3	300 cs	00	740 (HI)						
2	FWG	Anchor Darling	6	150 ss	000	250 (HP)						
3	FWG	Anchor Darling	6	900 cs	0	1800 (N)	(N)1800	(N) 1200	(N) 1200			
4	FWG	Anchor Darling	10	300 ss	0	740 (HI)						A, B, D
5	FWG	Anchor Darling	10	900 cs	2-150	1800 (S)						
6	FWG	Anchor Darling	18	300 cs	2	500 (S)						
7	FWG	Borg-Warner	3	1500 cs	00	2500 (N)	(N) 2500					
8	FWG	Borg-Warner	6	150 cs	Rotork	250 (HP)						
9	FWG	Borg-Warner	6	1500 cs	1	1800 (N)			(N) 1200			
10	FWG	Borg-Warner	12	300 cs	1-25	500 (HI)						
13	FWG	New Velan	2-1/2	1500 ss	000	2500 (N)	(N) 2500		(N) 2500			
14	FWG	Crane	6	900 cs	0	1800 (N)	(N) 1800		(N) 1200			
15	FWG	Walworth/Aloyco	4	150 ss	Rotork	250 (HP)						
16	FWG	Anchor Darling	3	900 cs	00	1800 (N)						
17	FWG	Pacific	10	150 cs	000	250 (HP)						
18	FWG	Pacific	4	150 cs	Rotork	250 (HP)						
21	FWG	Rockwell	2-1/2	900 cs	000-5	1800 (N)						
23	FWG	Velan	6	150 cs	000	250 (HP)	(HP) 250					
24	FWG	Velan	6	900 cs	0	1800 (N)	(N) 1800	(N) 1200	(N) 1200	(S) 1200	(S) 1200	
25	FWG	Velan	10	300 cs	0	500 (HI)						
26	FWG	Velan	10	900 cs	2	1800 (HI)					(S) 1200	
29	FWG	Walworth	6	150 cs	Rotork	250 (HP)						
30	FWG	Walworth	6	900 cs	0	1800 (N)			(N) 1200			
31	FWG	Walworth	12	150 cs	Rotork	250 (HI)						
34	FWG	Westinghouse	3	1500 ss	00	2500/750 (N)						E
41	FDG	Anchor/Darling	6	900 cs	0	1800 (N)			(N) 1200		(S) 1200	
43	SWG	Edwards	10	900 cs	2	1800 (HI)					(S) 1200	
44	Globe	Borg - Warner	6	900 cs	2	1800 (N)	(N) 1800					
48	Globe	Rockwell/Edwards	2	1500 ss	00	2500 (N)	(N) 2500		(N) 2500			
49	Globe	Velan	2-1/2	1500 ss	00	2500 (N)						
50	Globe	Anchor Darling	10	300 cs	2	500 (HI)						C
54	BFly	Pratt 1400 Sym	6	150 cs	000 -HOBC	150 (HP)						
55	BFly	Pratt 1200 Single O/S	6	150 cs	000 -HOBC	150 (HP)						C
61	FWG	Powell	14	600 cs		500 (HI)						

HP= Wyle Huntsville Pumped Flow Loop Test Facility
HI= Wyle Huntsville Intermediate Pressure Test Facility
N= Wyle Norco High Pressure Test Facility
S= Siemens/KWU High Pressure Test Facility

Table 1. Notes.

Alternate Configuration Testing. In addition to test sequences shown on the matrix, selected valves were tested with ambient water 15 feet per second for the following conditions:

A. Test conducted with an upstream elbow parallel to stem (flow from above) at zero diameter (i.e., immediately upstream of the mating flange).
B. Test conducted with the stem in a horizontal orientation with the pipe run horizontal (with straight inlet configuration).
C. Test conducted with the flow direction reversed (from that used in the nominal test).
D. Test conducted with an elbow perpendicular to the stem at zero diameter upstream.
E. Tested to 750 psid (closures) and to 2500 psid (openings).

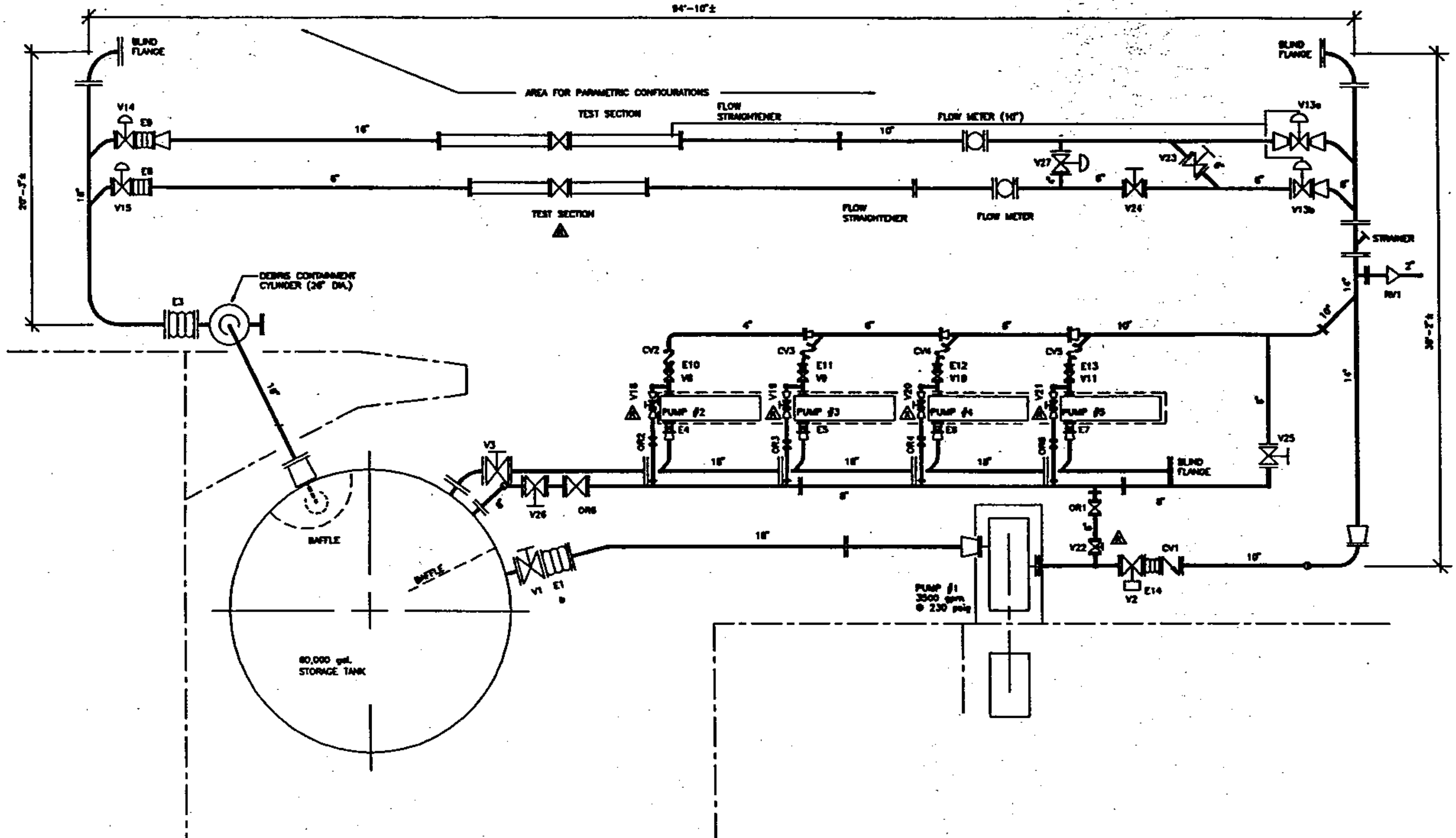


Figure 1. Cold-water pumped flow-loop schematic.

Figure 2. Cold-water pumped flow loop.

The loop is an open system consisting of a 35,000-gal accumulator vessel, high-pressure air compressors to pressurize the vessel, pumps, a test leg, and a number of control valves and related hardware. The loop can accommodate 10-inch valve tests under cold-water pumped-flow simulation conditions up to 1,800 psia and steam blowdown conditions up to 1,200 psia.

Instrumentation and Data Acquisition

Each valve was installed in an instrumented test section. The test sections were instrumented to allow measurement of the following parameters:

- Valve DP
- Valve inlet pressure
- Fluid temperature
- Fluid flow rate
- Direct stem force
- Direct stem torque
- Spring pack displacement
- Stem position
- Motor voltage
- Motor current
- Motor active power
- Limit at torque switch actuation.

Figure 10 is a schematic drawing showing the location of instrumentation on a typical test section. Figure 11 is a photograph of a typical test MOV installed in the instrumented test section. Thrust and torque measurements were made

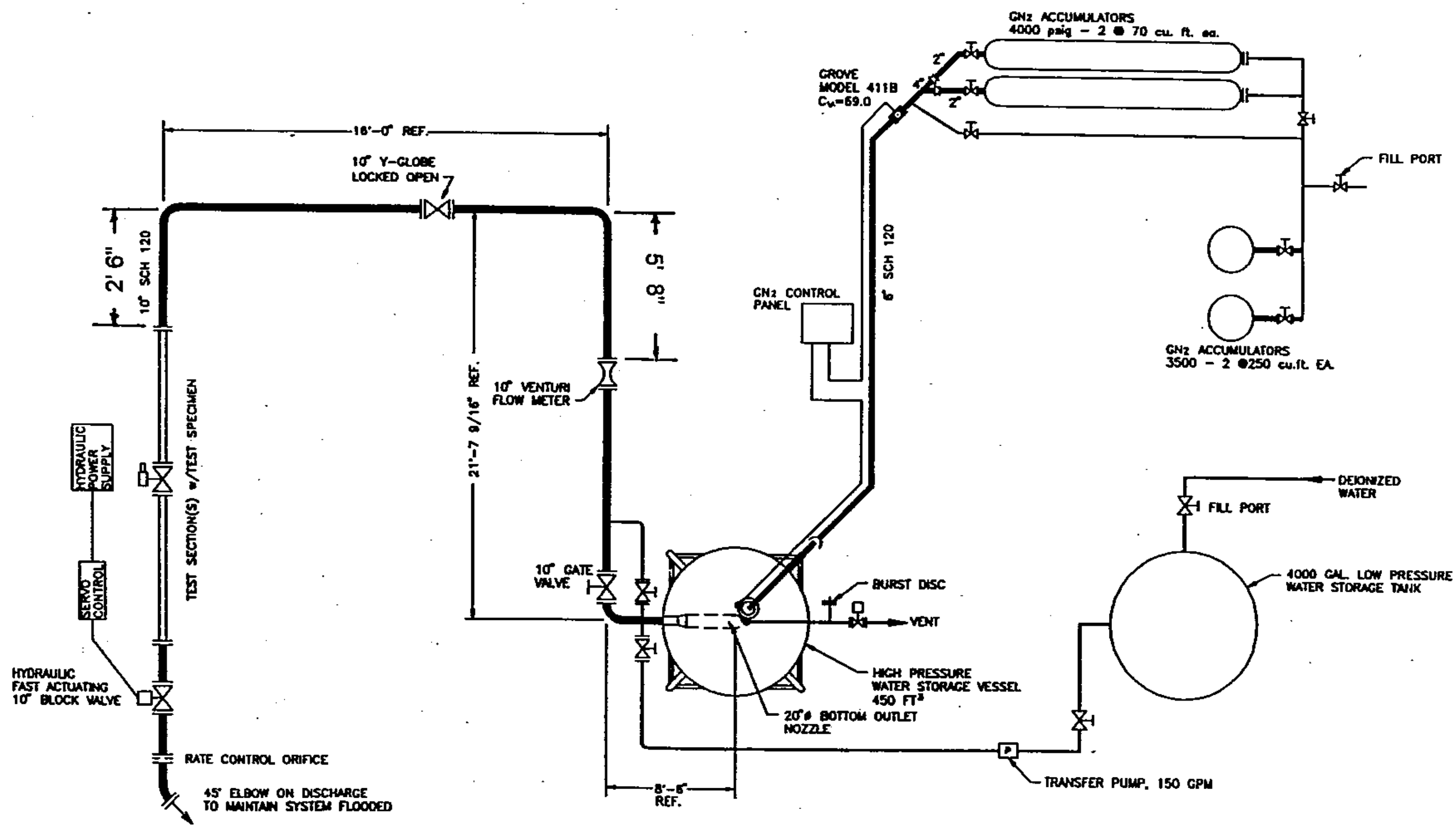


Figure 3. Intermediate-pressure, cold-water pumped flow simulation facility schematic.

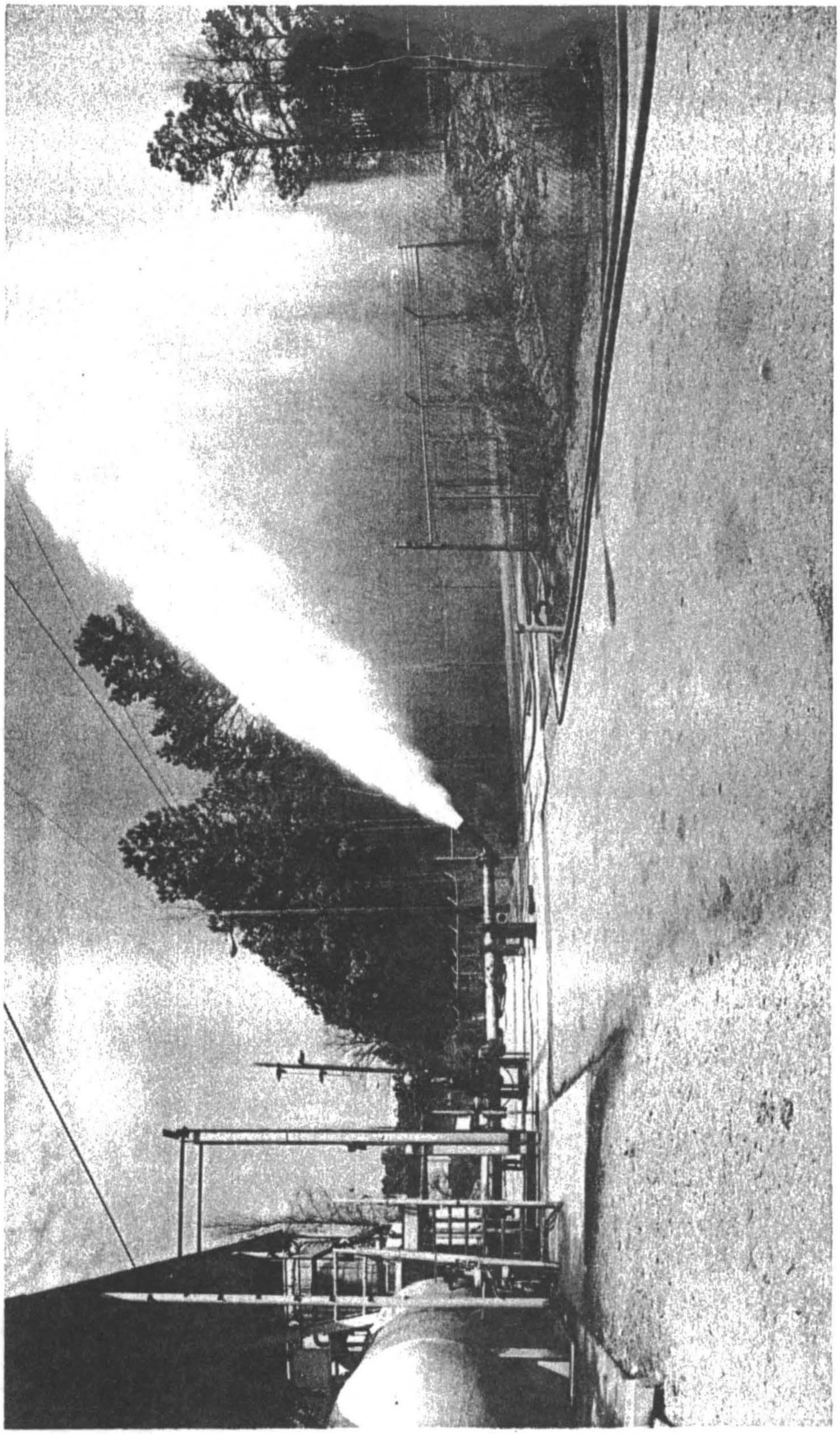


Figure 4. Partial view of the intermediate-pressure, cold-water pumped flow simulation facility.

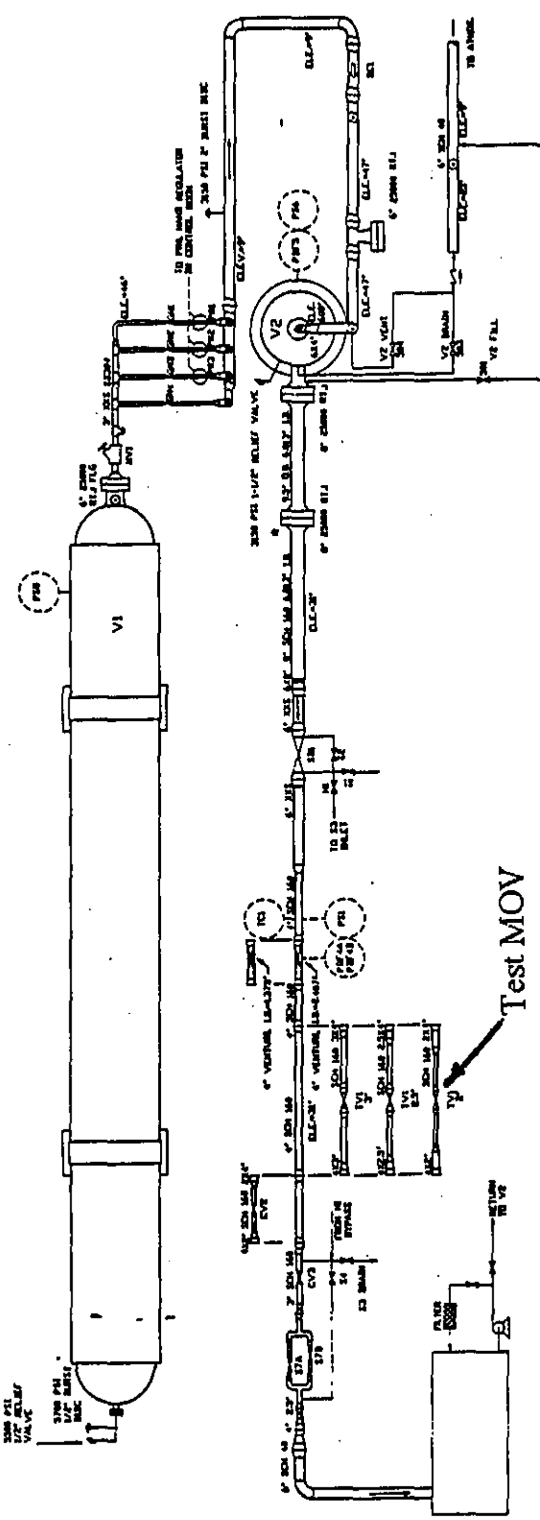


Figure 5. High-pressure, cold- and hot-water blowdown facility schematic, pumped flow simulation configuration.

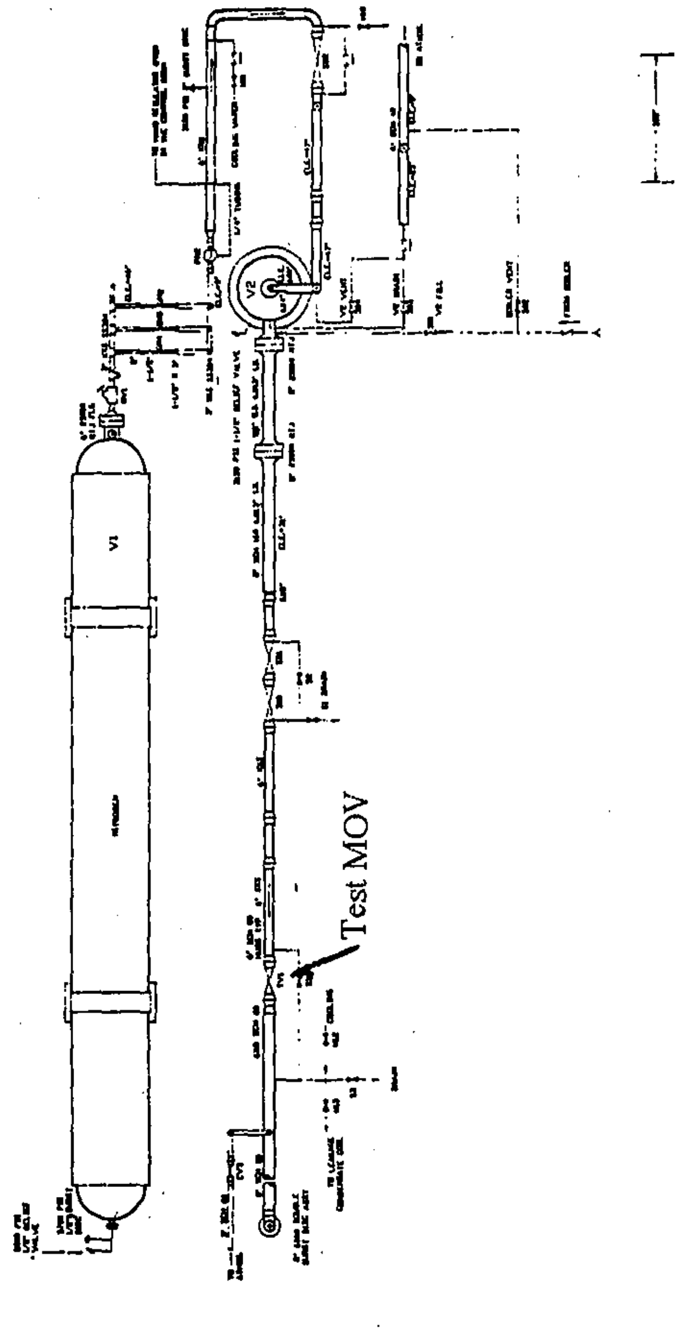


Figure 6. High-pressure, cold- and hot-water blowdown facility schematic, hot-water blowdown configuration.

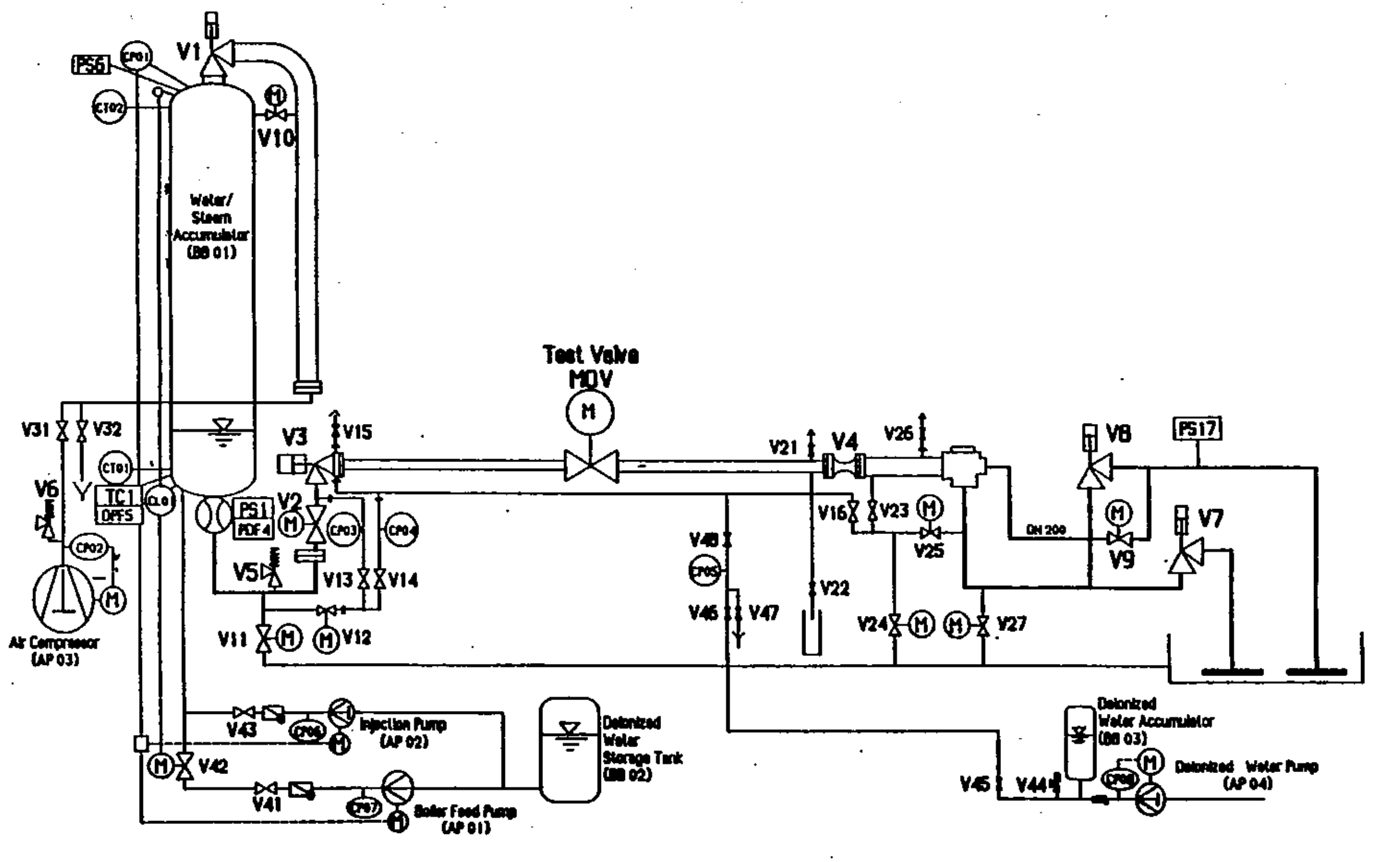


Figure 7. High-pressure water and steam blowdown facility schematic, cold-water pumped flow simulation configuration.

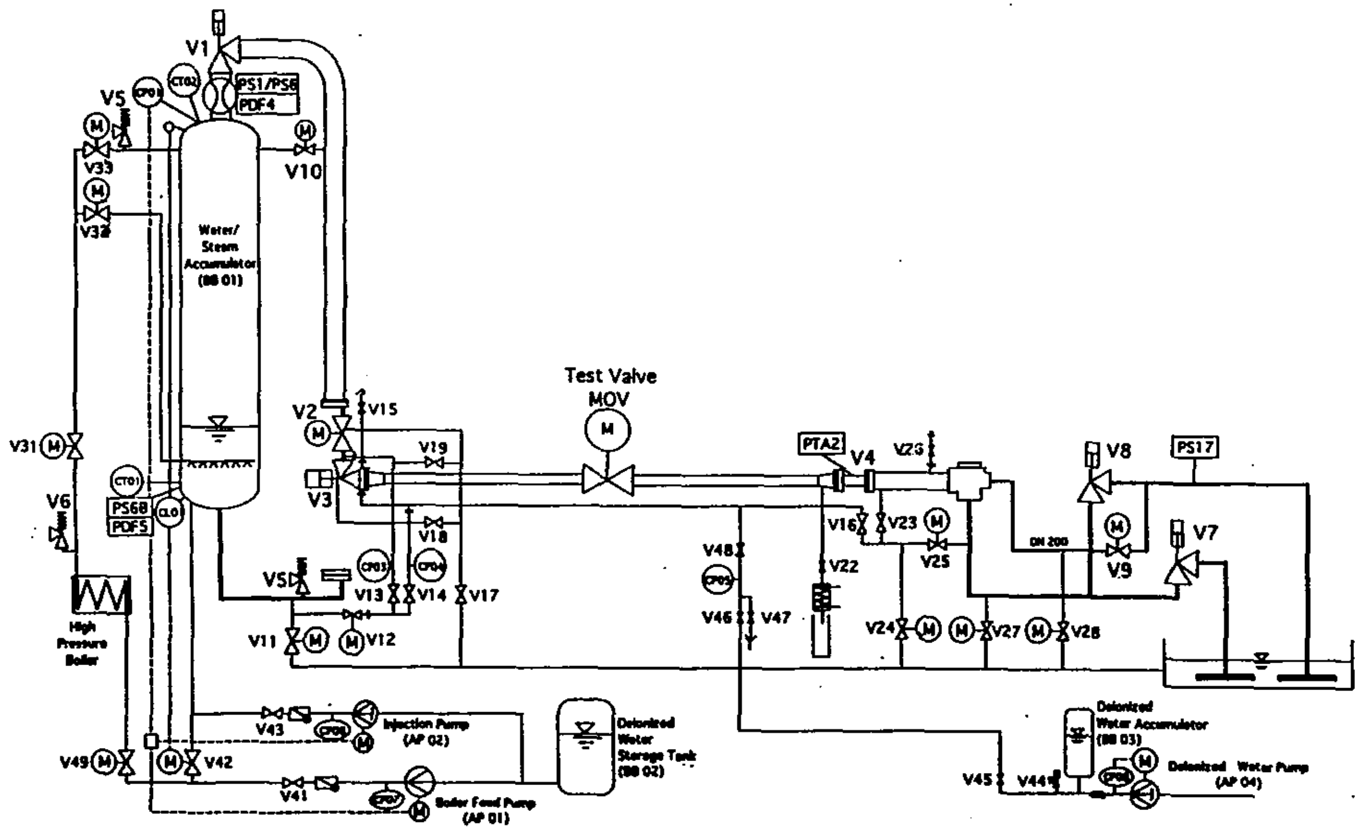


Figure 8. High-pressure water and steam blowdown facility schematic, steam blowdown configuration.

Figure 9. High-pressure water and steam blowdown facility.

using a calibrated Teledyne SMARTSTEM that included a strain gage. Use of a calibrated stem allowed measurement of thrust and torque to an accuracy of ±1% of the calibrated range.

All data were acquired at a sampling rate of 1,000 samples per second using MEGADAC data acquisitions systems. Before beginning each test sequence, all channels received electrical checks, and all signals were verified to be within expected ranges. All instrumentation was calibrated prior to installation in the loop and at the completion of testing.

MOV Preparation

Before installation in the test loops, all MOVs were disassembled, and a comprehensive set of internal measurements and photographs were taken. The valves were then reassembled and instrumented. Several partial nonwedging strokes were conducted to stabilize packing load levels, and the torque and limit switches were adjusted using a remote data acquisition system.

Testing Approach

All MOV's were tested under baseline, 15-ft/s, cold-water flow conditions over a range of DPs up to the maximum expected to occur in nuclear plant applications. To minimize the potential impact of the stroke effect on valve disc coefficients of friction during planned parametric testing, all gate valve seats were preconditioned before baseline testing by short stroking the valve into the seats under DP loading until the sliding friction coefficient reached a

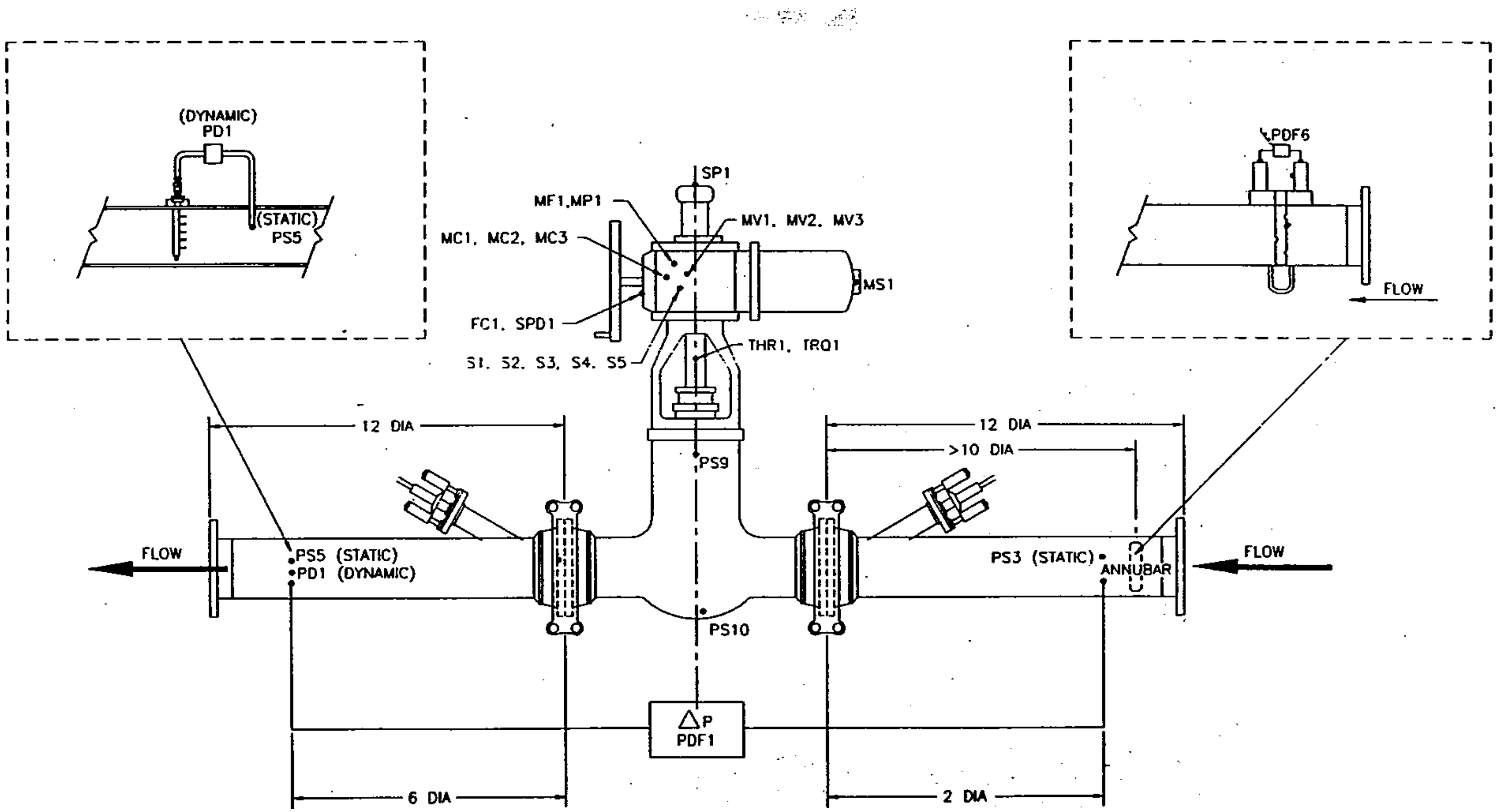


Figure 10. Typical test section instrumentation.

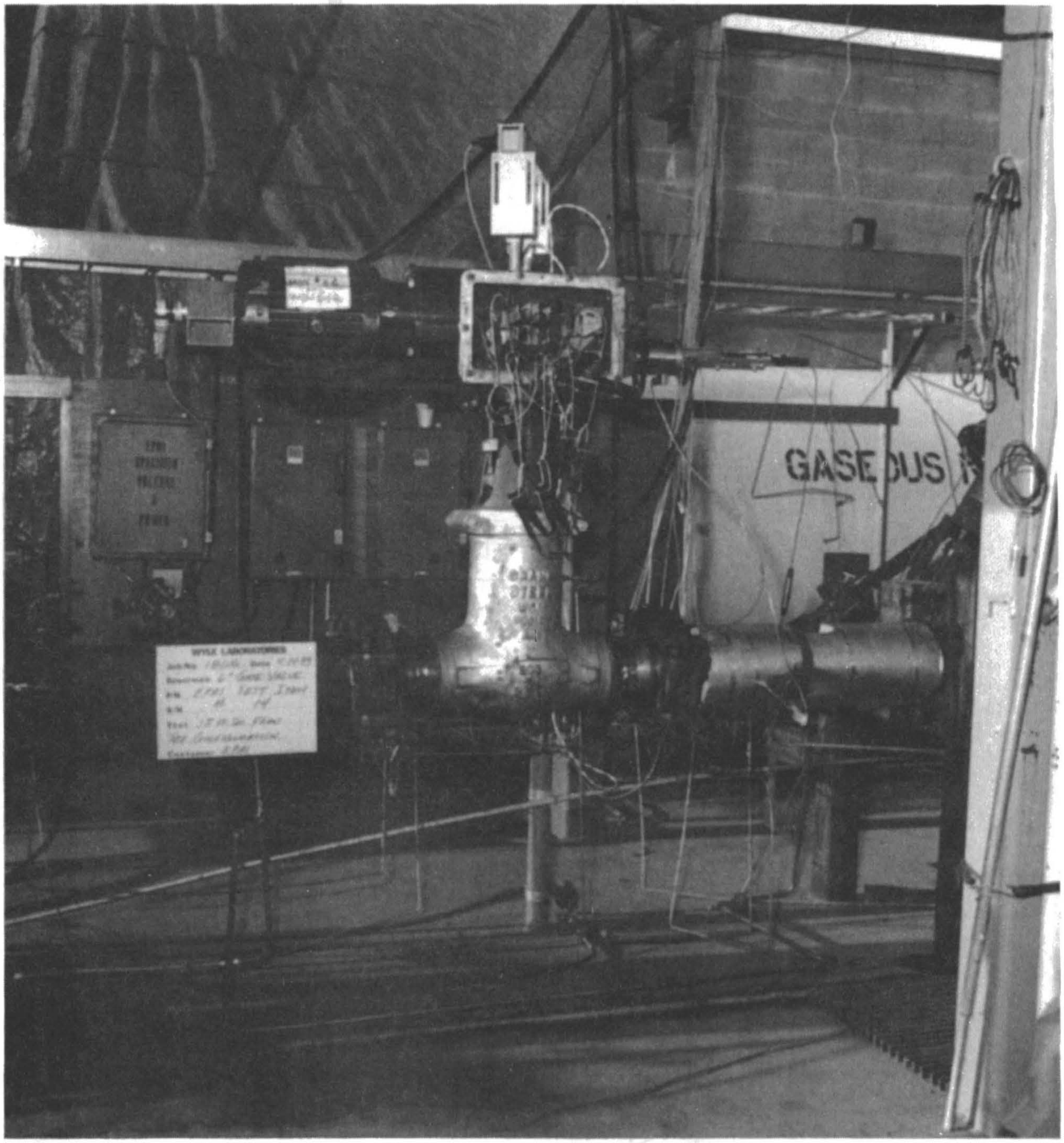


Figure 11. MOV No. 14 installed in the high-pressure cold- and hot-water blowdown facility.

maximum plateau level. The number of strokes required to reach the plateau level of friction varied from approximately 100 to as many as 900 strokes. A typical preconditioning history is depicted in Figure 12.

After completion of baseline testing, selected valves were parametrically tested to assess the effects of fluid temperature, thermodynamic state, flow velocity, inlet piping configuration, valve stem orientation, and flow direction on valve performance. Valve internal inspections were conducted at regular intervals to assess potential internal damage.

Tests were conducted under two types of flow conditions: pumped-flow simulation and blowdown. Pumped-flow simulation testing was conducted with maximum flow velocities ranging from 15 to 50 ft/s. To operate in the pumped-flow simulation mode, each test facility was configured such that the test valve DP when the valve was closed equaled the target DP, and the flow velocity when the valve was fully open equaled the target maximum flow velocity. In addition, the test valve inlet pressure was maintained nearly constant at the target maximum MOV inlet pressure throughout the stroke. By operating the test systems in this manner, a quadratic DP versus flow curve was produced, as would be the case if the flow were actually driven by a common centrifugal pump. In the Wyle Intermediate Pressure and Norco loops, as well as the Siemens/KWU loop, flow and DP were controlled by maintaining the driver tank pressure at the maximum target MOV inlet pressure and by a fixed adjustment of a flow control valve downstream of the test MOV.

In the Wyle Low-Pressure loop, flow and DP were controlled by maintaining a high bypass flow rate during the valve stroke such that test MOV operation had little impact on valve inlet pressure. The maximum test line flow was set in the low pressure loop in the same manner as described for the three remaining loops.

Blowdown testing was conducted by maintaining a nearly constant valve inlet pressure at the maximum target value and removing as much upstream and downstream flow resistance as practical.

Tables 2 and 3 present typical test sequences conducted under pumped-flow simulation and blowdown conditions, respectively.

Posttest Activities

Following the completion of all testing, each MOV was disassembled, thoroughly inspected, and then photographed. Measurements were also taken to characterize any valve damage observed.

Data Analysis Methods

Gate Valve Data Analysis. Each gate valve stroke conducted under DP loading conditions was evaluated to determine (where appropriate) the apparent disc coefficient of friction and the apparent stem coefficient of friction at various reporting points in the opening or closing stroke. The reporting points are as follows:

- Closed-to-Open Valve Stroke
    - A. At cracking
    - B. Just after cracking
    - C. Maximum after cracking
    - D. Running (zero DP)
    - E. Limit switch trip
    - F. At flow initiation
- Open-to-Closed Stroke
    - A. Running (zero DP)
    - B. Maximum prior to initial wedging
    - C. At initial wedging
    - D. At torque switch trip
    - E. Final value
    - F. At flow isolation.

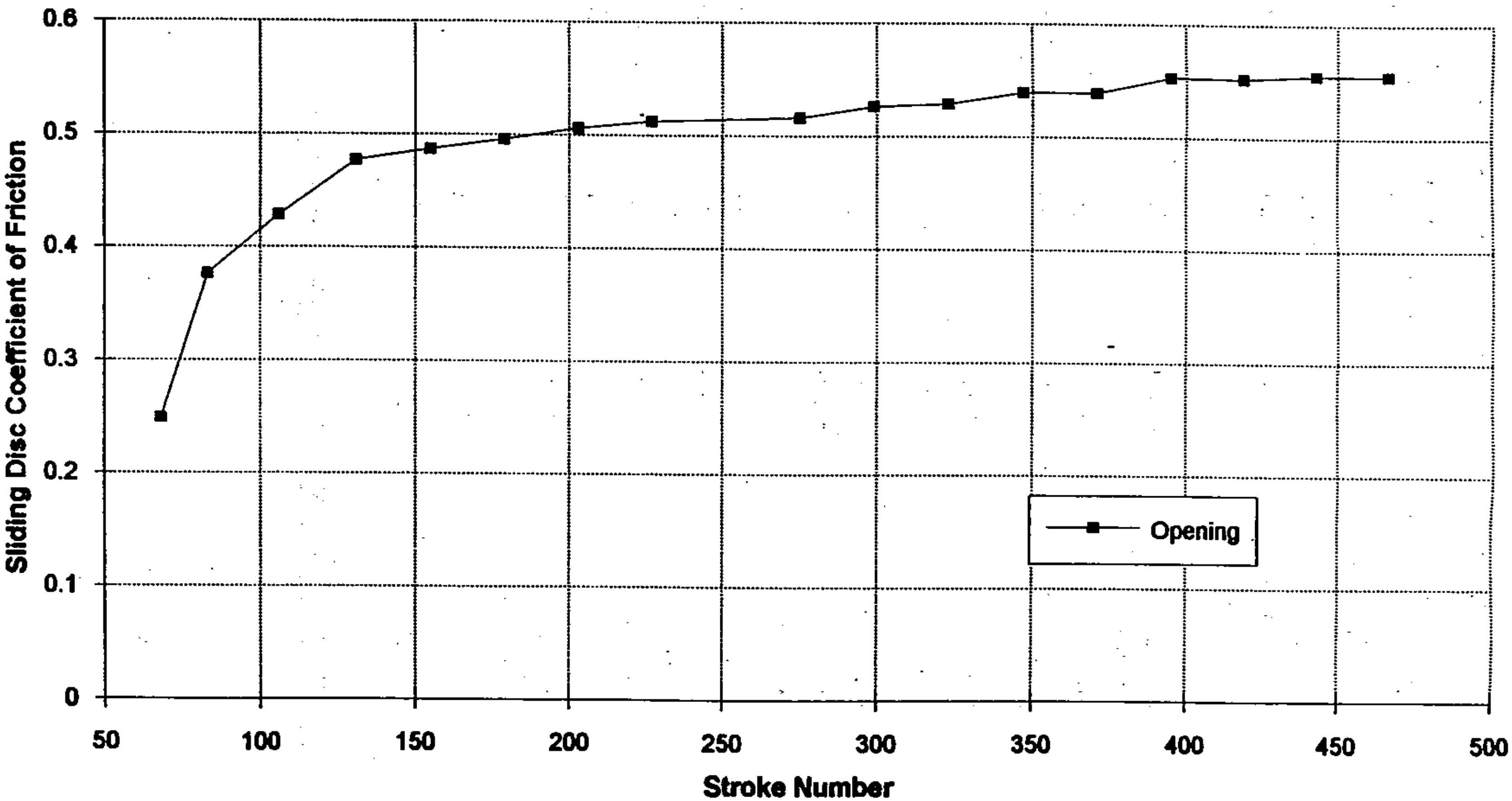


Figure 12. Typical gate valve seat preconditioning history.

Table 2. Typical pumped flow simulation test sequence.

Stroke	Description	Direction	Flow (% Nom)	Pressure (% Nom)	DP (% Nom)
1	Static	O→C	0	0	0
2[a]	Static inspection	C→O	0	0	0
3	Flow plus DP	O→C	100	33	33
4	Flow plus DP inspection	C→O	100	33	33
5	Repeat 3	O→C	100	33	33
6	Repeat 4 inspection	C→O	100	33	33
7	Flow plus DP	O→C	100	67	67
8	Flow plus DP inspection	C→O	100	67	67
9	Repeat 7	O→C	100	67	67
10	Repeat 8 inspection	C→O	100	67	67
11	Flow plus DP	O→C	100	100	100
12	Flow plus DP inspection	C→O	100	100	100
13	Repeat 11	O→C	100	100	100
14	Repeat 12 inspection	C→O	100	100	100
15	Press effect	O→C	0	100	0
16	On packing loads	C→O	0	100	0
17	Static	O→C	0	0	0
18	Hydropump	C→O	0	100	100
19	Static	O→C	0	0	0
20[a]	Static inspection	C→O	0	0	0

a. Seat leakage measurement is required prior to Strokes 2 and 20 and when significant leakage is suspected.

Table 3. Typical blowdown test sequence.

Stroke	Description	Direction	DP	P	Temp	Flow	Comment
0[a]	Cold static	O→C	0	0	Cold	0	No DAS
	Cold leak test	C	100	100	Cold	0	No DAS
0[b]	Cold static	C→O	0	0	Cold	0	No DAS
1	Cold static	O→C	0	0	Cold	0	—
2	Cold static	C→O	0	0	Cold	0	—
3	Cold static manual[b]	O→C	0	0	Cold	0	—
4	Hydro pump	C→O	100	100	Cold		—
	Condition test section to hot	O	0	100	Hot		No DAS
5	Hot pressurized	O→C	0	100	Hot	0	—
6	Hot pressurized	C→O	0	100	Hot	0	—
6[a]	Hot pressurized	O→C	0	100	hot	0	No DAS
	Hot leak test	C	100	100	Hot	0	No DAS
7	Hot mini troke equilibrate MOV DP to zero, P to 100%	C→partial O	100	100	Hot	0	—
7[a]	Hot pressurized	Partial O→O	0	100	Hot	0	No DAS
10	Blowdown closure (B/D)	O→C	100	100	Hot	B/D	—
	Hot leak test Depressurize upstream and downstream piping with test MOV closed	C	100	100	Hot	—	—
11	Hot static	C→partial O	0	0	Hot	0	—
	Video inspection; if undamaged and with EPRI approval, continue	Partial O	0	0	Cold	0	No DAS
11[a]	Cold static	Partial O→O	0	0	Cold	0	No DAS
	Reestablish hot condition	O	0	100	Hot	0	No DAS
12	Hot pressurized	O→C	0	100	Hot	0	—
13	Blowdown open cool system to ambient	C→O	100	100	Hot	B/D	—
16	Cold static	O→C	0	0	Cold	0	—
17	Cold static	C→O	0	0	Cold	0	—
18	Cold static	O→C	0	0	Cold	0	No DAS
	Cold leak test testing complete	C	100	100	Cold	0	No DAS

a. No DAS—No Data Acquisition.

b. Only the wedging/unwedging portion of the stroke needs to be manual; DAS for manual only.

Figure 13 shows the location of each reporting point and the governing equations for calculating the apparent disc coefficient of friction and the stem coefficient of friction for gate valves.

The terms $T_A - P_A A_S$ and $T_D + P_D A_S$ in the thrust equations on Figure 13 determine the packing drag at Points A or D. This formulation assumes that only stem rejection and packing drag are present when the valve is nearly wide open (i.e., there are no Bernoulli, guide friction, or other forces at this time). The advantage of this formulation is that zero shifts are automatically removed and packing drag is determined for the stroke being analyzed. The assumption is good for pumped flow tests, but can be inaccurate for gate valve blowdown tests because Bernoulli forces and guide drag can be significant at Points A or D. Thus for gate valve blowdown tests, the packing drag averaged from static strokes is used in place of the above terms. In those cases when, during valve blowdown closing strokes, the valve actuator is started before the double burst disc is actuated, a period exists in the thrust trace where only packing drag and stem rejection exist. For these cases, Point A is picked between valve actuation and burst disc initiation and the formulation of Figure 13 is used.

Determination of Gate Valve Flow Isolation. The point of flow isolation in a gate valve closing stroke is of significant interest because, at this point, hydrodynamic forces have ceased, the disc should be fully on the seat, and the valve has succeeded in essentially stopping the flow (although it may not be leak-tight). The determination of this exact point during the flow stroke is difficult because the measured parameters (such as pressure, differential pressure, and flow rate) show negligible change during the final, small increment of valve disc motion before isolation. During the closed-to-open hydropump stroke, which is conducted as part of each cold-water test sequence, the upstream pressure is much more sensitive to flow initiation because the upstream trapped volume is quite small; therefore, the pressure will drop significantly if a small leak occurs. The scheme adopted for the determination of flow isolation in test MOVs is, therefore, based upon analyses of the hydropump opening stroke and the corresponding application of these analyses to valve strokes involving fluid flow. The procedure is described briefly below for a typical MOV as follows:

1. The time corresponding to the initial movement of the valve disc during the hydropump opening stroke is determined by noting the time at the end of the uncracking spike in the thrust-versus-time trace. At this point, all lost motion in the thread clearances and T-head clearances have been taken up.

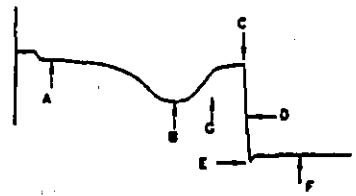
2. The time corresponding to the onset of upstream pressure decay during the hydropump opening stroke is determined from the pressure-versus-time trace.

3. The difference in stem position between these two values of time is determined from the stem position-versus-time trace. This corresponds to the disc travel from a position at unwedging to flow initiation.

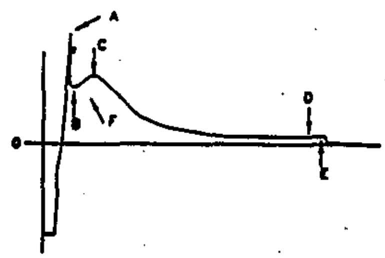
4. On any given valve closure stroke to be analyzed, the time corresponding to initial wedging is determined from the thrust trace.

5. The stem position corresponding to the above time is determined from the stem position-versus-time trace for the given closure stroke.

6. The difference in stem position determined in Step 3 is then subtracted from the stem position identified in Step 5 in order to determine the time that corresponds to the time of flow isolation.



$$T = T_A - P_A A_s + P_c A_s + \mu \Delta P_c A_o / (\cos\theta - \mu \sin\theta)$$



$$T = T_D + P_D A_s - P_B A_s + \mu \Delta P_B A_o / (\cos\theta + \mu \sin\theta)$$

Opening Stroke

T = Stem Thrust, lb.
P = Upstream pressure, psi
A_s = Stem area, in.^{2}
A_0 = Disk mean seat area, in.^{2}

ΔP = Different pressure, psi
μ = Disk coefficient of friction
θ = Half disk angle

$$\text{Stem } \mu = (24FS \cos\alpha - d \cos\alpha \tan a)/(24FS \tan a + d)$$

FS = Torque/thrust, ft.
d = Stem OD - P/2, in.
p = Pitch, in.

α = Half thread angle
a = Thread lead angle
Stem μ = Stem coefficient of friction

Figure 13. Reporting points and governing equations for gate valve analysis.

Globe Valve Data Analysis. For each globe valve stroke conducted under DP loading, the apparent valve factor and stem factor were evaluated, when appropriate, at various reporting points in the valve stroke. These reporting points are as follows and shown in Figure 14:

- Closed-to-Open Strokes
  - A. Maximum value after unseating
  - B. At flow initiation
- Open-to-Closed Strokes
  - A. Maximum before seating
  - B. At torque switch trip
  - C. Maximum after seating
  - D. Final
  - E. At flow isolation.

The globe valve apparent valve factor, f, was determined by

$$T = F_{pack} + F_p + f\, A\, \Delta P\ ,$$

where

T	=	stem thrust, lb
F_{pack}	=	packing friction load, lb
F_p	=	stem piston effect load, lb
f	=	apparent valve factor
A	=	either seat or guide area, in.^{2}
ΔP	=	differential pressure across valve, psid.

The packing friction load was determined from an average of the zero pressure, static strokes.

The apparent valve factors are evaluated using several sets of assumptions, summarized as follows:

1. f at flow isolation or initiation based on the DP at isolation or initiation and the mean seat area

2. f based on the maximum thrust, DP at the time of occurrence of the maximum thrust, and the guide area

3. f based on the maximum thrust, DP at the time of occurrence and the mean seat area

4. f based on the maximum thrust, DP at isolation or initiation, and the mean seat area.

Application of the assumptions listed under No. 4 above yielded the valve factor required to predict the maximum thrust using the method used historically by industry to predict globe valve thrust requirements. Globe valve stem factors were evaluated using the same equation as for gate valves and were not evaluated for rising rotating stems.

Butterfly Valve Data Analysis. All butterfly data are plotted and used directly to support model validation. No specific comparison to industry predictive methods was conducted.

Summary of Results

Once gate valves had been preconditioned, they were subjected to full-flow tests at cold-water flow velocities ranging from 15 to 50 ft/s. Typical time history data are presented in Figures 15 through 18 for a 6-in. flexwedge gate valve tested under cold-water, 15 ft/s conditions at a DP of 1,800 psi.

With one exception, apparent disc friction coefficients ranged from 0.2 to 0.9 during these tests. These apparent disc coefficients of friction included all valve performance phenomena and were not necessarily representative of sliding friction alone. One valve design exhibited very high–thrust requirements to close under cold-water pumped-flow conditions. The apparent disc coefficient of friction for this valve was approximately 1.9. The valve manufacturer's evaluation concluded that some of the test valve internal dimensions were outside manufacturing tolerances.

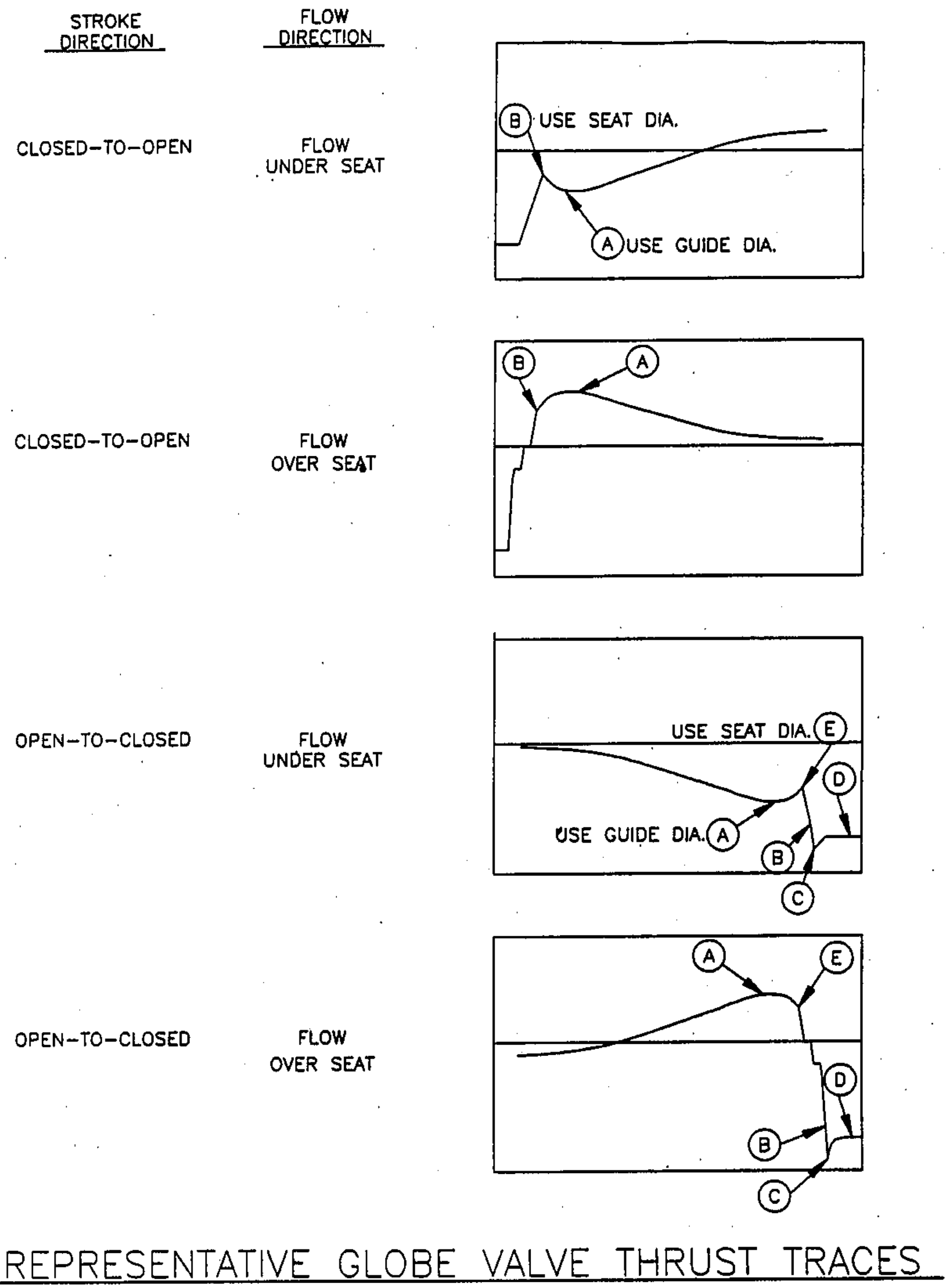


Figure 14. Reporting points for globe valve analysis.

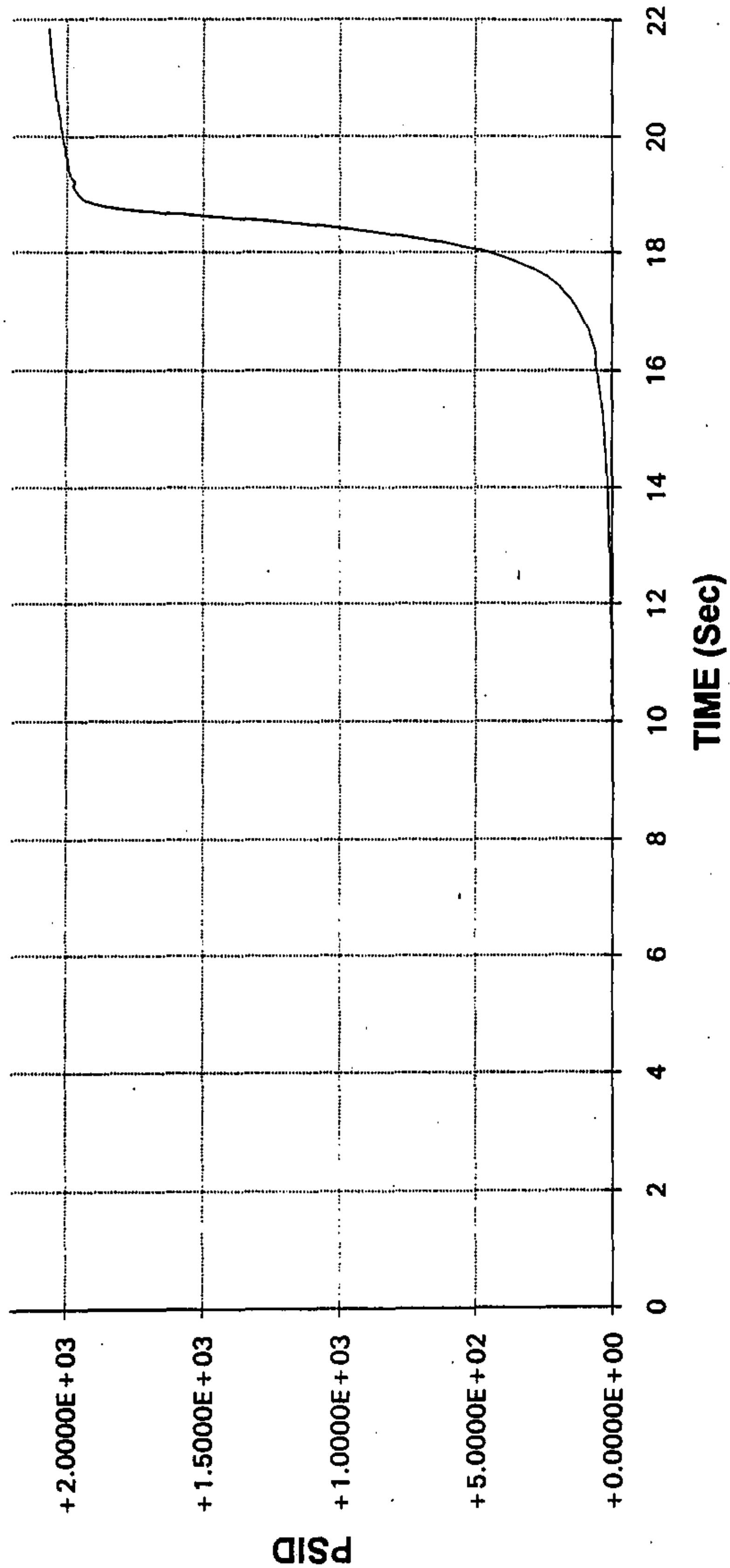


Figure 15. Typical gate valve DP time history.

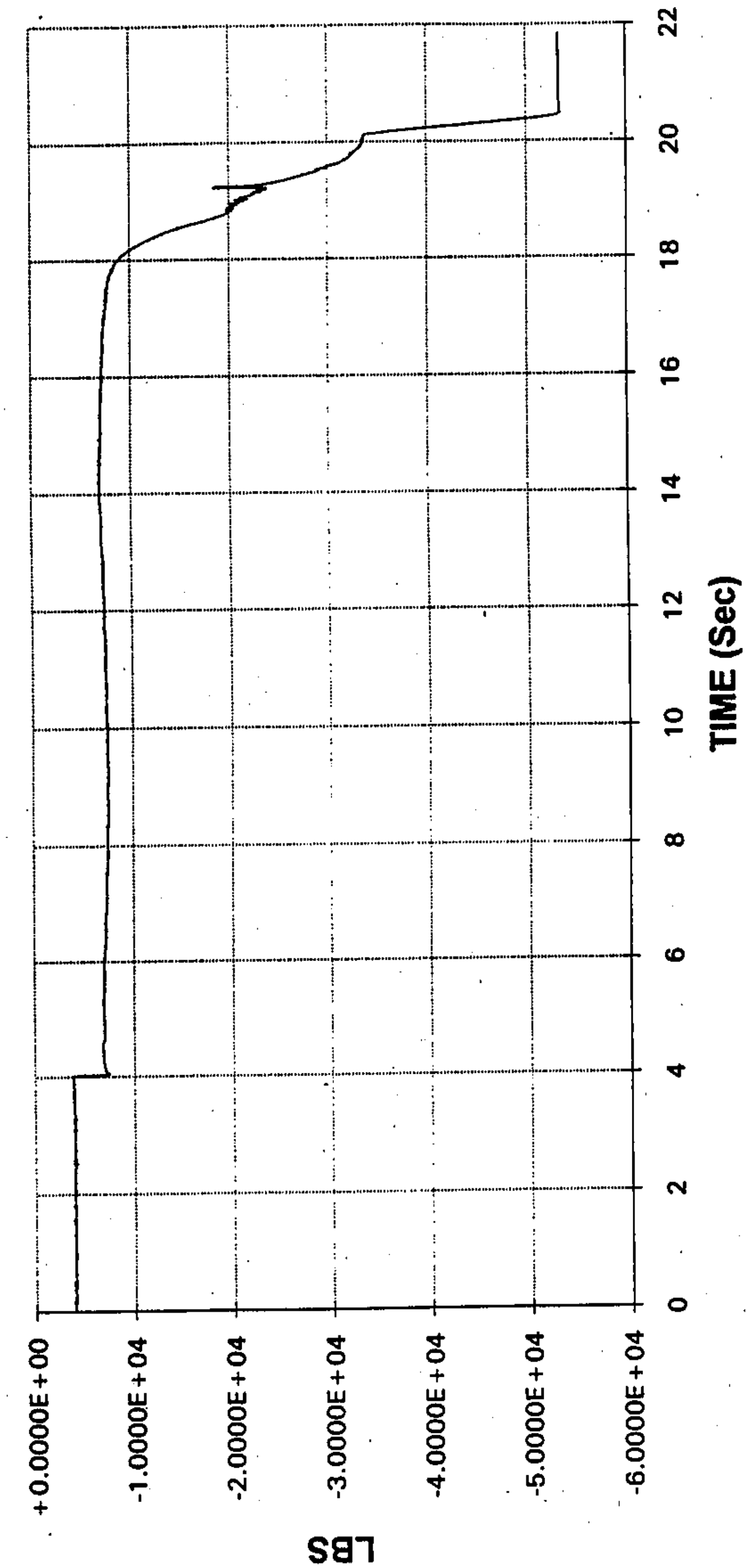


Figure 16. Typical gate valve thrust time history.

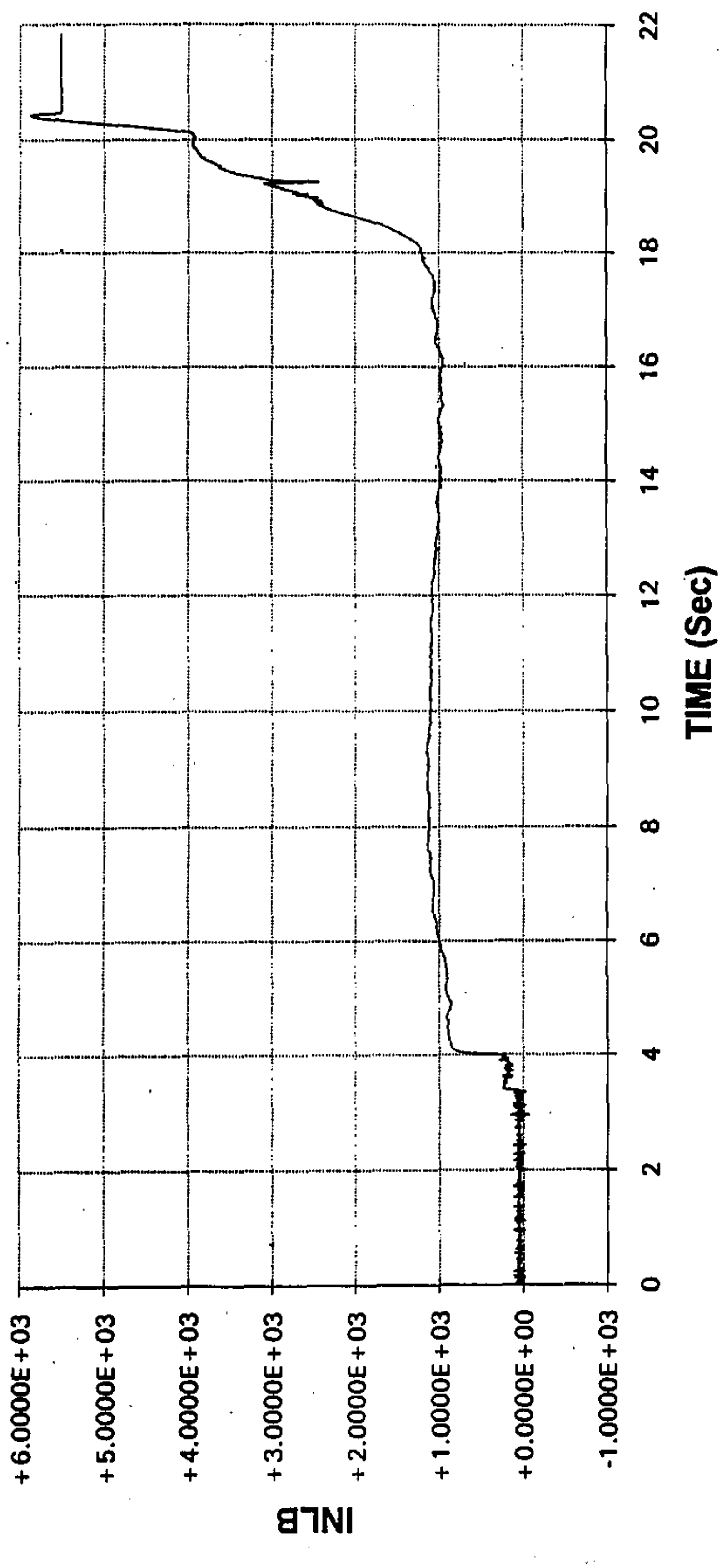


Figure 17. Typical gate valve torque time history.

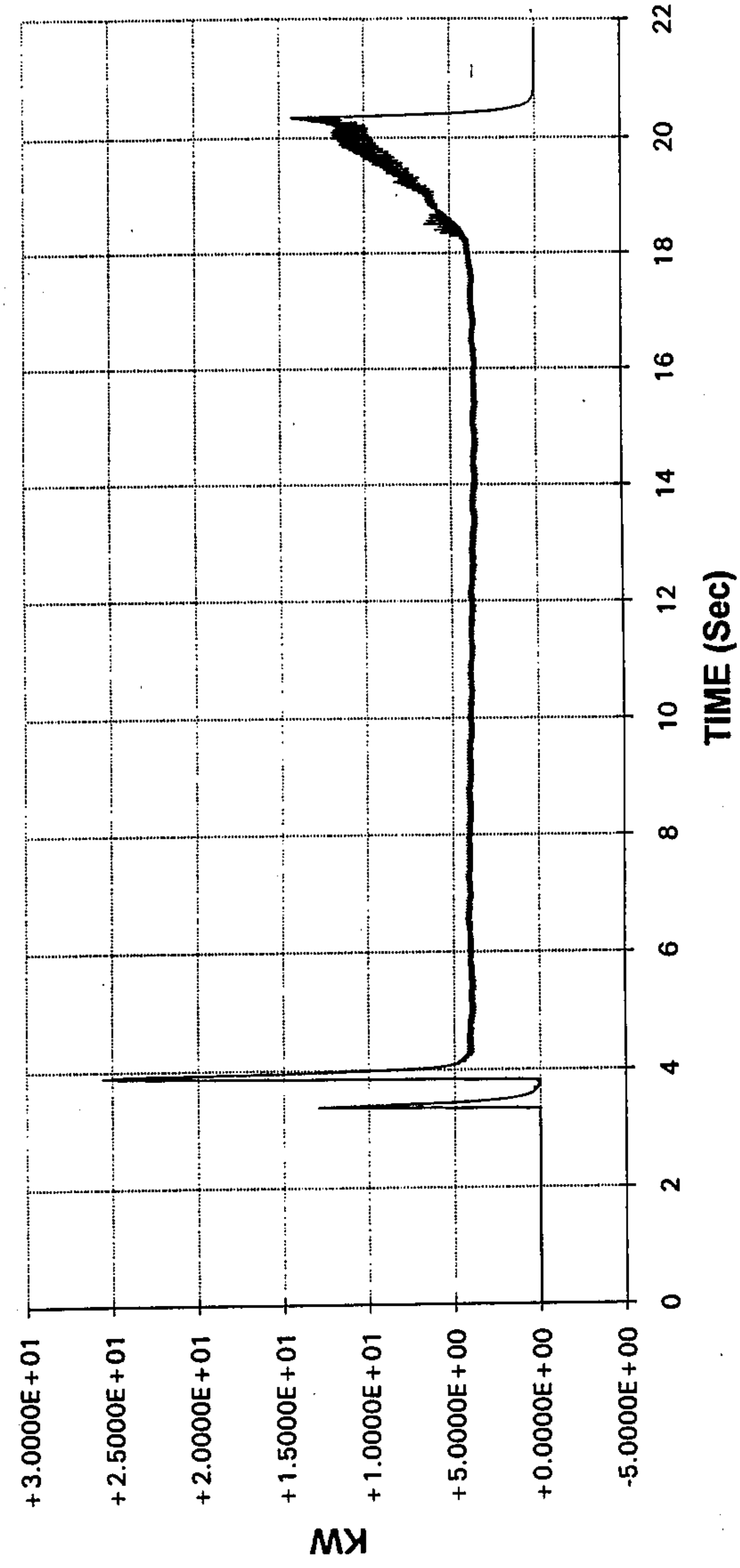


Figure 18. Typical gate valve motor active power time history.

Although a significant range of apparent disc friction coefficients was observed, only a single gate valve sustained internal damage under cold-water, 15-ft/s flow conditions. In this case, the disc was pushed through the seats allowing leakage above the disc on an 18-in. valve with a 3-degree seat half angle. At higher flow velocities (i.e., greater than 30 ft/s), the body guides were plastically bent in the flow direction in one valve design that incorporated cantilevered guide rails.

Hot-water, pumped-flow, testing resulted in lower sliding friction coefficients than for cold water, generally ranging from 0.3 to 0.5. The stroke effect on sliding friction was not observed to be significant under hot-water conditions. Under hot-water and steam blowdown conditions at 1,200 psid, apparent disc friction coefficients ranging from approximately 0.3 to 0.8 were observed. Although some valve designs were undamaged, others sustained significant guide or seat damage.

Disc-to-body-seat friction sliding coefficients were found to decrease with increasing DP. An example of this trend is shown in Figure 19. This finding confirms the results of friction separate-effects testing and supports the use of test results from sliding friction coefficients obtained from reduced DP in the evaluation of thrust requirements under full DP conditions for gate valves in pumped-flow systems.

Testing of globe valves under incompressible flow conditions revealed that it is necessary to select the appropriate area (either disc seat or guide area) for DP application in order to predict required thrust accurately. Data from a valve that is apparently guide-area based is shown in Figure 20. Note that the peak thrust occurs prior to seating when the guide area is subject to DP application.

Under compressible flashing flow conditions, excessive thrust loading was observed that exceeded even guide area based predictions. Side loading of the disc from pressure variations within the valve body may play a role in this phenomenon.

Of the 21 gate and globe valves for which analyses have been completed to date, only three exhibited reductions in output thrust under DP loading conditions exceeding 7%. The maximum rate-of-loading effect observed among this group was 12%.

Examples of DP, flow, and required torque during testing of a 6-inch butterfly valve are presented in Figures 21, 22, and 23, respectively. Vendor methodologies for predicting required hydrodynamic torque for butterfly valves are generally proprietary, and as a result, little can be concluded at this time regarding the suitability of such methods based on Program test results. The EPRI butterfly valve model accurately predicts butterfly valve performance for the valves tested in the flow loops.

IN SITU TESTING

Overview

To supplement the flow-loop testing of MOVs, EPRI procured data from safety-related MOV testing conducted at nuclear power plants. The valves were selected for procurement based on their predominance in the industry valve population and availability of test data.

Utilities provided enhanced in situ test data in accordance with the *In Situ Test Guide for Motor Operated Valves*.[a] This guide defines the requirements for enhanced in situ test data that are suitable for validating the performance prediction methodologies. The test guide specifies requirements for measurement of hydraulic system and valve performance parameters, instrumentation, internal dimensional measurements for gate valves, data documentation, and report format.

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a. M. Albers and P. Damerell, EPRI Report NP-7078, 1990.

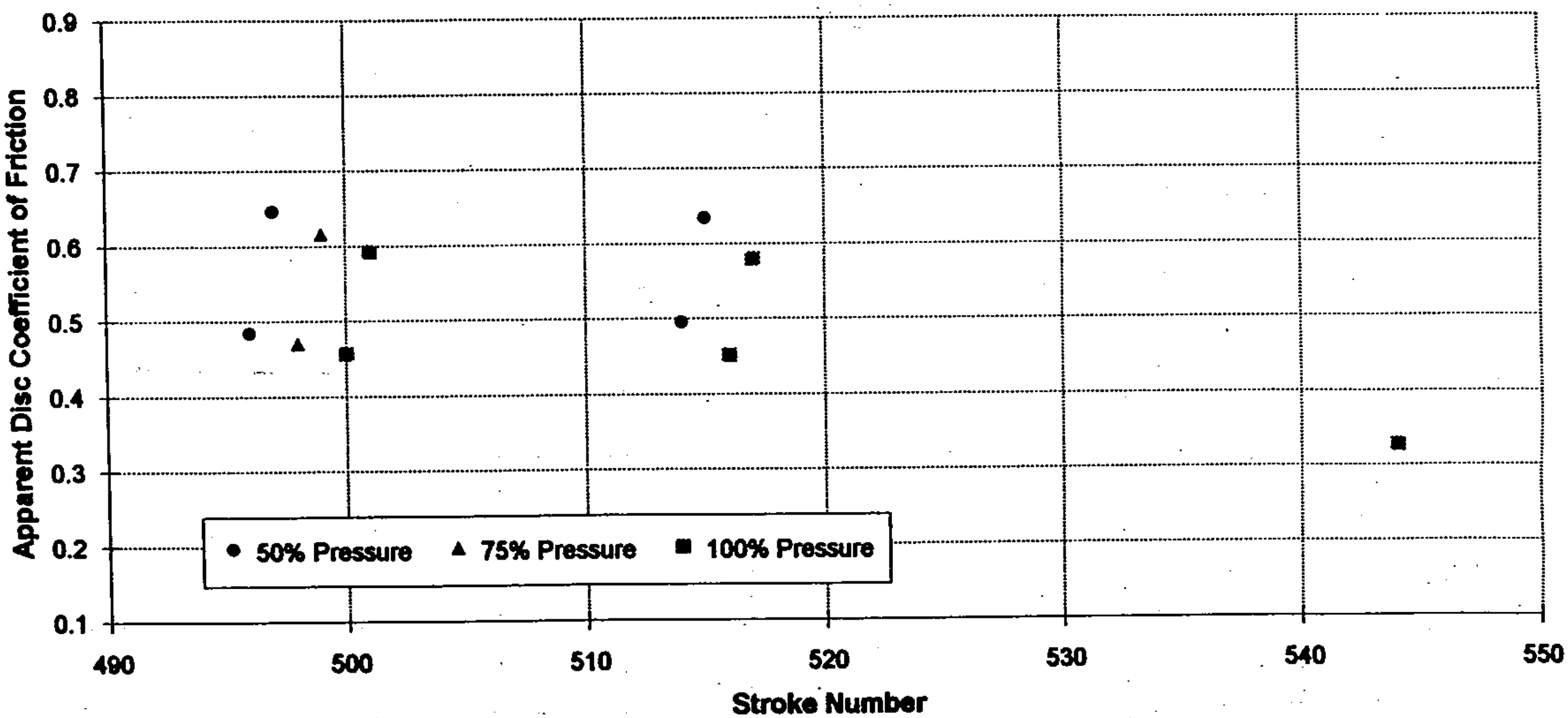

Figure 19. Effect of DP on "apparent" disc coefficient of friction—Valve 13.

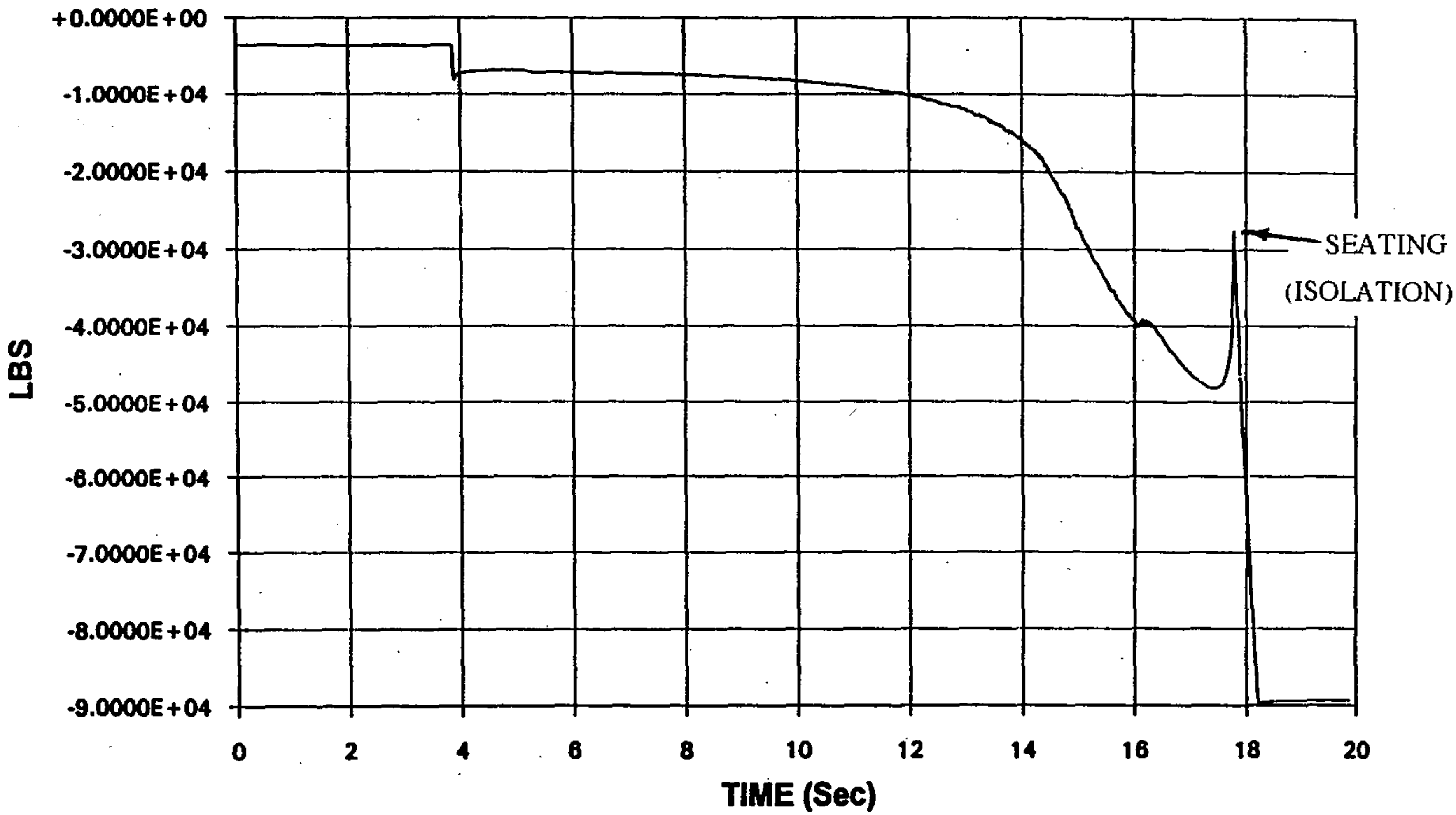

Figure 20. Typical globe valve thrust time history (guide-based valve design).

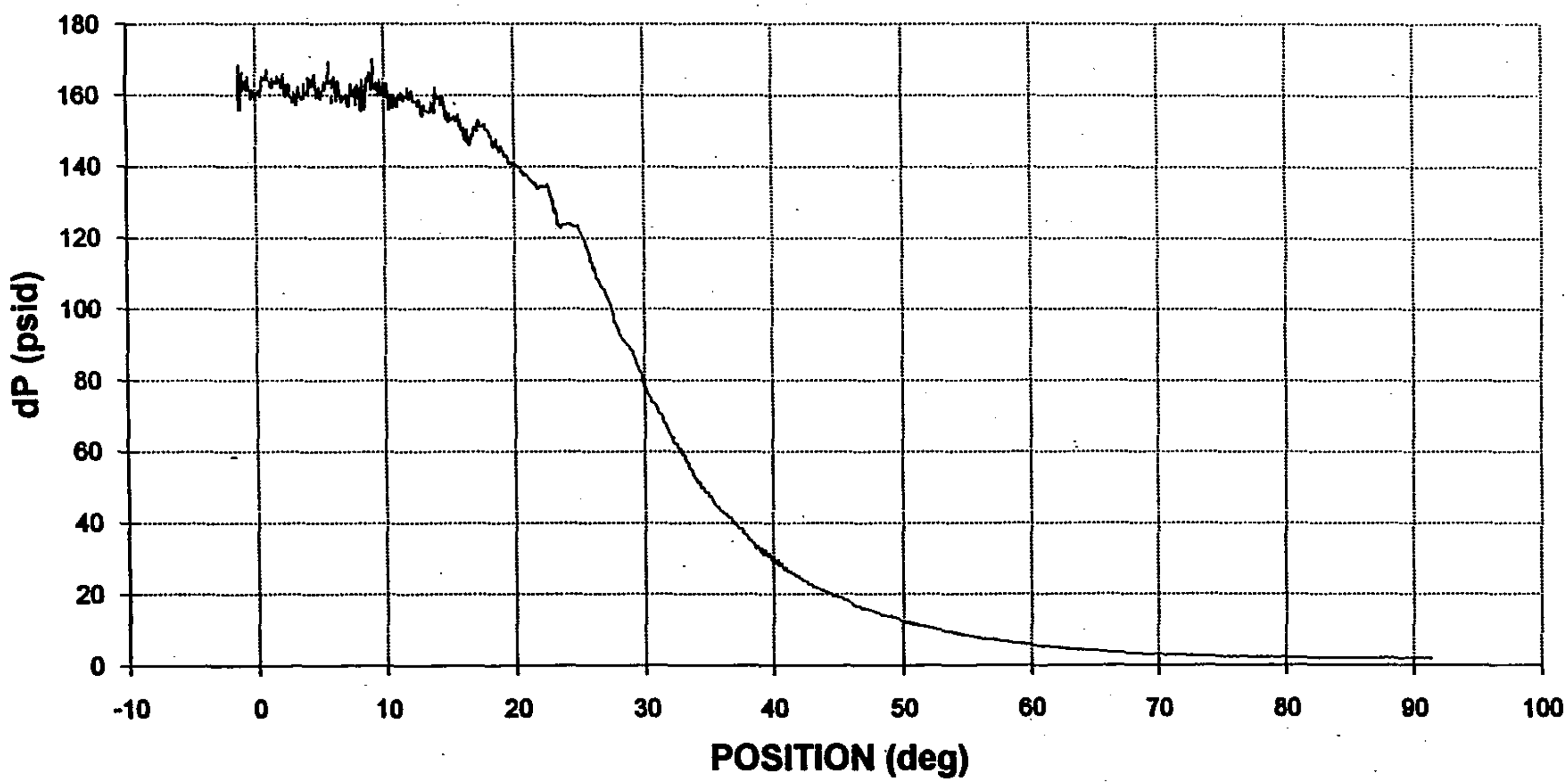

Figure 21. Typical butterfly valve DP versus disc position.

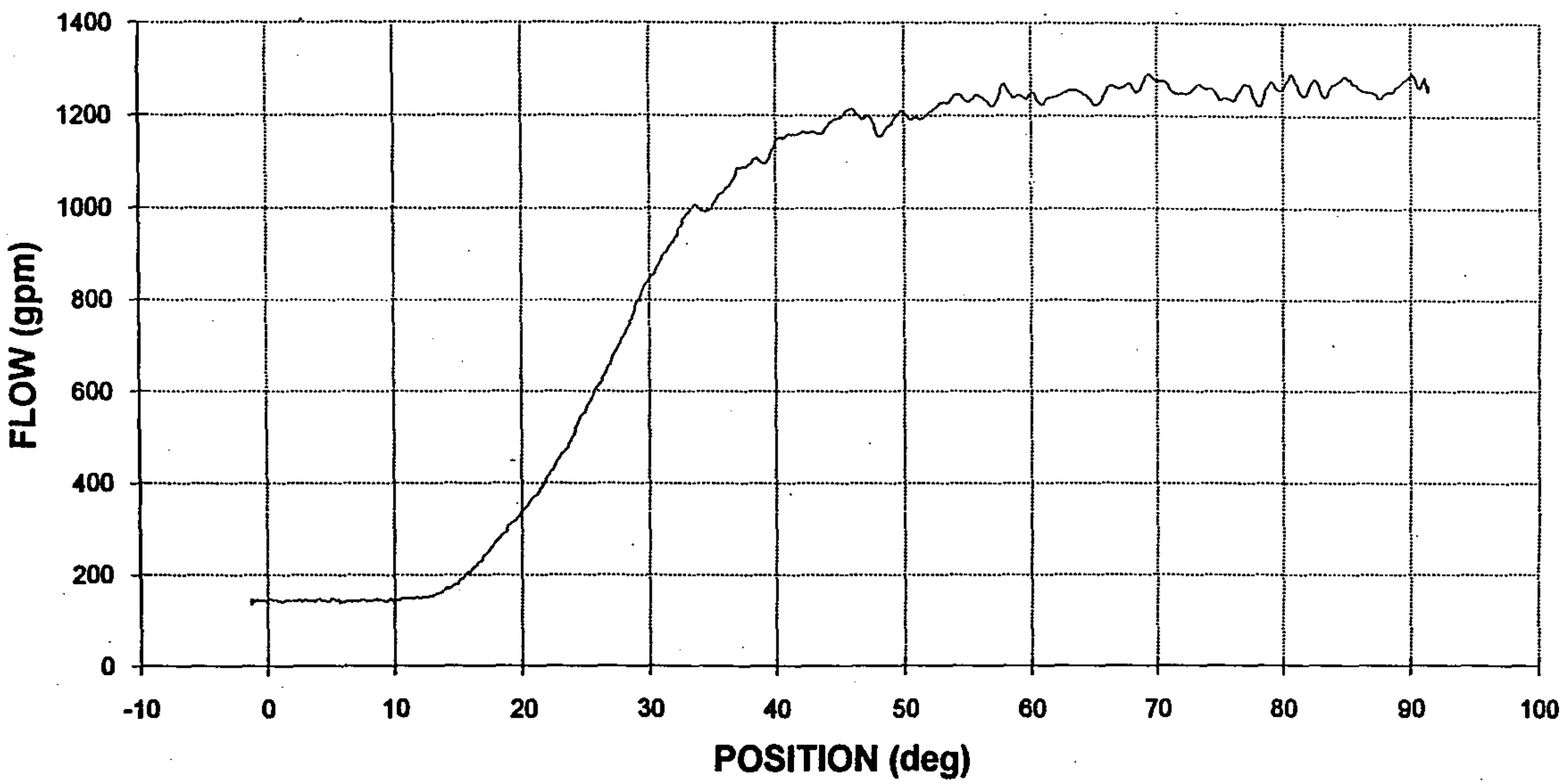

Figure 22. Typical butterfly valve flow versus disc position.

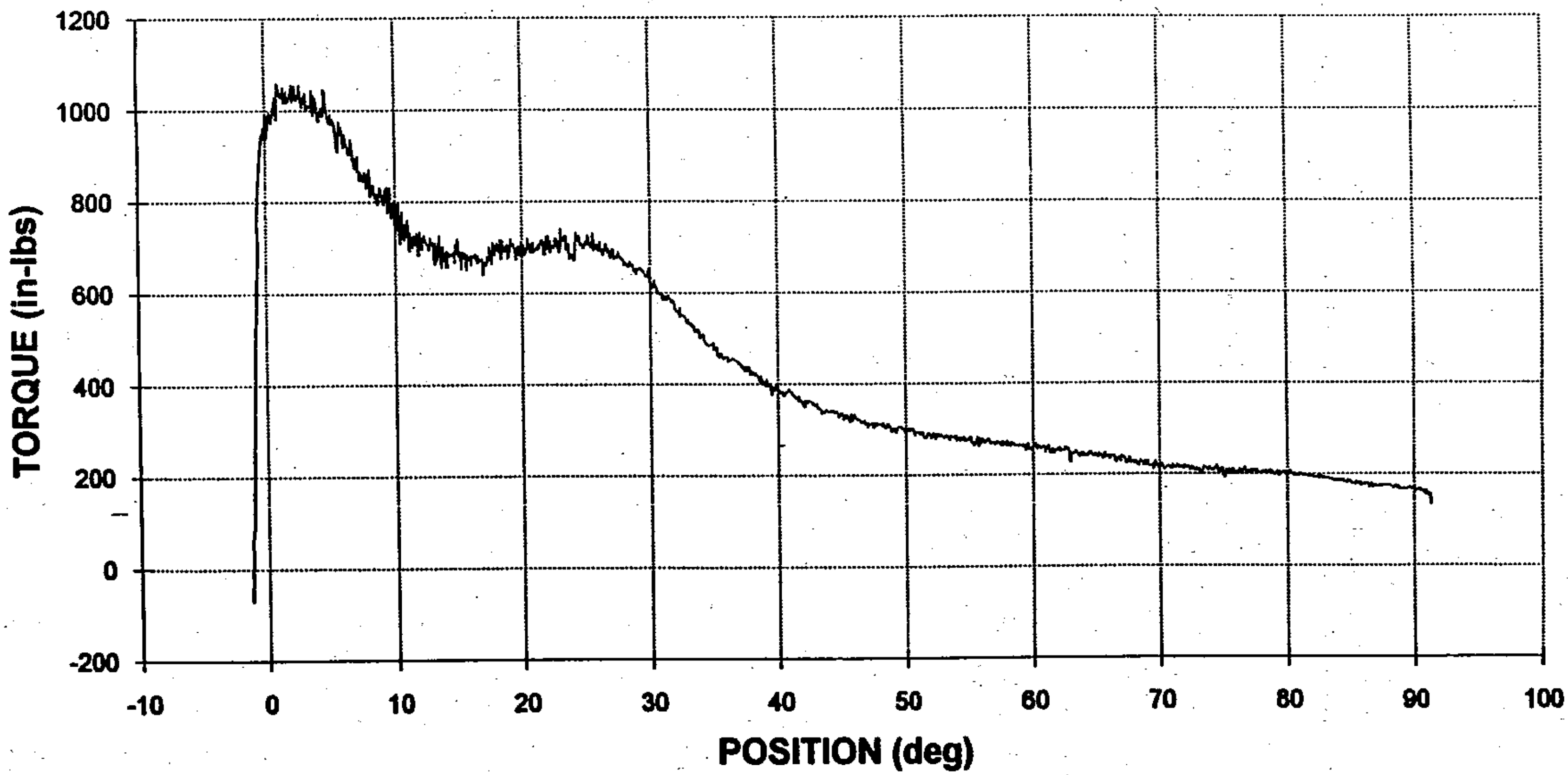

Figure 23. Typical butterfly valve torque versus disc position.

Each utility that supplied in situ data prepared a data package in which all required valve and test information was compiled. The data package included recorded data, flow system lineup, instrumentation specifications and calibration documents, applicable valve drawings, and valve internal dimensions. Each utility conducted DP and static tests using their instrumentation, data acquisition equipment, and test procedures.

The data packages were reviewed for completeness, technical accuracy, and adherence to the utilities' quality assurance program. The data were analyzed and compiled into a formal test report.

Instrumentation. The test guide establishes two categories of instrumentation: time history records and single point data. The time history record is continuous and generally includes both an opening and closing stroke for both DP and static tests. The following are required time history parameters:

- Stem thrust (gate and globe valves)
- Stem torque (butterfly valve)
- Valve upstream pressure
- Valve differential pressure
- Motor current
- Spring pack displacement
- Actuator control switch actuation.

The required single point parameters are fluid temperature, motor voltage, and maximum flow rate. Measurement of stem torque is recommended for gate and globe valves.

All in situ data were acquired using instrumentation and data acquisition systems provided by each utility. Nominally, data were acquired at a rate of 1,000 samples per second per channel.

Test Matrix. In situ test data were obtained for 28 MOVs. Nineteen gate, one globe, and eight butterfly valves were tested. Table 4 describes each valve and the test conditions under which it was tested. Testing was performed over a range of differential pressures from 100 to 2,880 psi with water-flow velocities ranging from 6 to 60 ft/s. Testing was conducted under water and steam conditions. Opening and closing strokes were generally conducted at each test condition. In addition open/close static strokes were conducted.

Data Analysis Methods. The time history data for each static and dynamic test were analyzed to verify accuracy and evaluate applicable performance parameters at important points in the valve stroke.

The analysis techniques are the same as those described in the preceding Flow-Loop Data Analysis Methods section, except that the point of flow initiation is based on valve inlet pressure decay upon valve opening during a DP test rather than on a hydropump opening stroke.

Summary of Results. Under cold-water, pumped-flow conditions, apparent gate valve disc coefficients ranged from approximately 0.2 to 0.6. Figure 24 is a histogram depicting the distribution of apparent disc friction coefficients measured during the In Situ Test Program. Two gate valves tested under steam blowdown closure conditions at a DP of 2,100 psid exhibited apparent disc coefficients of friction of approximately 0.45. The single globe valve tested was of the balanced disc design and exhibited minimal hydrodynamic torque requirements. The maximum observed reduction in operator output thrust from static to dynamic conditions was 20%. Of the 11 valves for which both thrust and torque measurements were available, only three exhibited rate-of-loading effects exceeding 7%.

CONCLUSIONS

The data obtained as part of the EPRI Flow Loop and In Situ Test Program have provided significant new insights into the performance characteristics of typical motor-operated gate, globe, and butterfly valves. These data represent the most comprehensive and accurate data base on

Table 4. In situ test matrix.

Number	System	Manufacturer	Type	Size	ANSI	Test DP	Vel., fps	Temp., F	Medium
1	CT	BW	FWG	16	300	300	13	86	WATER
2	CT	BW	FWG	16	300	300	12	86	WATER
3	AF	BW	FWG	4	900	1620	17	75	WATER
4	AFW	VELAN	FWG	4	600	900	59	535	STEAM
5	AFW	A/D	PDG	4	600	965	59	540	STEAM
6	LPCI	A/D	FWG	16	600	340	6	110	WATER
7	LPCI	A/D	FWG	16	600	350	6	105	WATER
8	SW	VELAN	FWG	18	300	205	9	64	WATER
9	CT	FISHER	GLOBE	4	300	275	40	75	WATER
10	CC	FISHER	BF-S	18	150	140	10	75	WATER
11	SW	FISHER	BF-SO	24	150	120	12	56	WATER
12	NCW	PRATT	BF-SO	10	150	110	16	80	WATER
13	SW	POSI-SEAL	BF-SO	10	150	97	22	36	WATER
14	SW	POSI-SEAL	BF-SO	42	150	75	13	36	WATER
15	CC	BW	FWG	4	1500	2880	4	75	WATER
16	RC	WESTING.	FWG	3	1525	2075	440	557	STEAM
17	RC	WESTING.	FWG	3	1525	2485	450	557	STEAM
18	SI	WESTING.	FWG	4	1525	2800	12	75	WATER
19	SI	WESTING.	FWG	10	1525	265	16	75	WATER
20	SI	WESTING.	FWG	8	316	244	20	75	WATER
21	SI	VELAN	FWG	3	1500	2650	57	82	WATER
22	SC	WKM	PDG	6	300	195	44	79	WATER
23	CC	WKM	PDG	8	150	100	19	88	WATER
24	SC	WKM	PDG	16	1500	315	0	75	WATER
25	SDSW	A/D	FWG	10	150	140	41	49	WATER
26	SP	PRATT	BF-S	24	150	23	12	120	WATER
27	SP	PRATT	BF-S	18	150	13	20	120	WATER
28	ASW	FISHER	BF-S	24	150	32	8	59	WATER

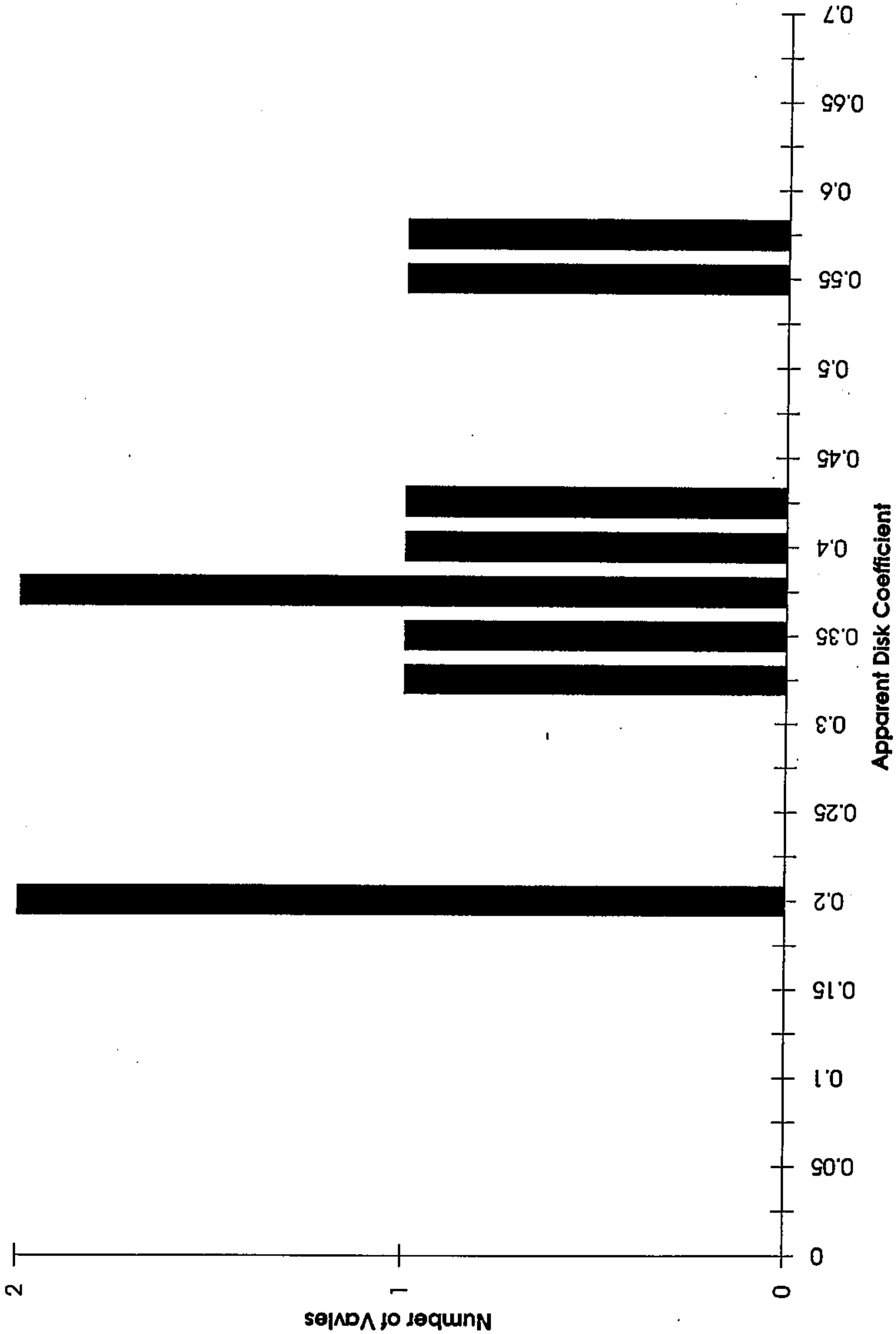

Figure 24. Histogram of "apparent" disc coefficients of friction (in situ testing, cold-water conditions).

MOV behavior currently available and provide an excellent basis for MOV predictive methods validation. Specific conclusions are summarized as follows:

- Cold-water stellite-on-stellite sliding friction coefficients can be highly variable, ranging from less than 0.2 to greater than 0.6:
  - Friction coefficient variation appears to be based on the number of loaded strokes applied and the contact stress level.
  - Friction coefficients increase with stroke number to a maximum plateau level, then stabilize.
- Hot-water stellite-on-stellite sliding friction coefficients are less variable, generally ranging from 0.3 to 0.5.
- Under cold-water pumped-flow conditions, gate valve apparent disc friction coefficients can, in isolated cases, exceed 0.6, depending on valve specific internal design characteristics.
- Under pumped-flow (~15 ft/s) conditions, the potential for internal damage to gate valves is extremely low.
- Under high-velocity flows (>30 ft/s up to blowdown conditions) the potential for valve internal damage increases significantly for some gate valve designs.
- Under pumped-flow conditions, gate valve disc to seat sliding friction coefficients tend to decrease with increasing DP. This finding supports the use of friction coefficients measured under reduced DP conditions when evaluating thrust requirements at higher DPs.
- Gate valve performance during in situ testing was generally consistent with that observed during flow loop testing.
- Under incompressible flow conditions, globe valve thrust requirements can be predicted accurately if the appropriate disc area (seat versus guide) is assumed for DP application.
- Under compressible flow conditions, globe valve thrust requirements can exceed even guide area based predictions. Side loading on the plug may play a role in this phenomenon.
- For some gate and globe valves, a significant reduction in motor-operator output thrust can occur under dynamic (DP loading) conditions relative to the output thrust attained under static (no DP) conditions.

ACKNOWLEDGMENTS

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Arizona Public Service Company; Duke Power Company; Entergy Operations, Inc.; Illinois Power Company; Northern States Power Company; Pacific Gas & Electric Company; Southern California Edison Company; Southern Nuclear Operating Company; TU Electric; Washington Public Power Supply System.

Gate Valve Performance Prediction

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ABSTRACT

The Electric Power Research Institute is carrying out a program to improve the performance prediction methods for motor-operated valves. As part of this program, an analytical method to predict the stem thrust required to stroke a gate valve has been developed and has been assessed against data from gate valve tests. The method accounts for the loads applied to the disc by fluid flow and for the detailed mechanical interaction of the stem, disc, guides, and seats. To support development of the method, two separate-effects test programs were carried out. One test program determined friction coefficients for contacts between gate valve parts by using material specimens in controlled environments. The other test program investigated the interaction of the stem, disc, guides, and seat using a special fixture with full-size gate valve parts. The method has been assessed against flow-loop and in-plant test data. These tests include valve sizes from 3 to 18 in. and cover a considerable range of flow, temperature, and differential pressure. Stem thrust predictions from the method bound measured results. In some cases, the bounding predictions are substantially higher than the stem loads required for valve operation, as a result of the bounding nature of the friction coefficients in the method.

INTRODUCTION

Gate valves are commonly used in numerous fluid systems in power plants. Although not typically used to throttle flow at partially open positions, they are often required to stroke while there is flow in the line. In such cases, the valve is loaded by the flow throughout its stroke. The load is reacted by surfaces and contact points other than those that react the differential pressure (DP) load when the valve is closed. Prediction of the stem force required to operate a gate valve with flow and DP, including proper calculation of thrust at intermediate positions, is a technical challenge that has recently received increased attention.

The Electric Power Research Institute (EPRI) is carrying out a program called the Motor-operated Valve (MOV) Performance Prediction Program to develop validated methods for predicting MOV performance. Within the EPRI program, an improved gate valve stem thrust prediction model is being developed. A previous paper (Wang et al., 1992) described the modeling approach and provided a detailed discussion of 10 important modeling aspects used in developing the model. At the time the previous paper was written, a preliminary model had been developed and example calculations had been performed. The model predicted the types of behavior seen in gate valve testing under pumped flow and blowdown flow conditions.

The gate valve model has been further developed and predictions from the model have been compared with test data, as described in this paper. The model satisfies the need for improved technical modeling in the following areas:

- The forces applied to the disc by the fluid flow are calculated explicitly as a function of disc position and DP across the valve
- The forces resulting from mechanical interaction between the stem, disc, guides, and seat rings are calculated through the full range of valve positions
- Friction coefficients and potential material damage at contact points within the gate valve are calculated as a function of materials, load, contact configuration, and temperature.

This paper describes the gate valve model developed in the EPRI MOV Performance Prediction Program. The analytical basis is summarized and the separate effects tests that support the model development are described. Stem thrust predictions from the model are compared with measurements from gate valve testing.

SCOPE OF ANALYTICAL MODEL

The analytical model addresses a wedge gate valve with a single-piece disc, which is the most common type of gate valve used in power plant applications. These valves are referred to as solid wedge gate (SWG) valves and flexible wedge gate (FWG) valves. Figure 1 shows a typical SWG valve.

The scope of the model is summarized as follows:

- Single-piece wedge disc (SWG or FWG)

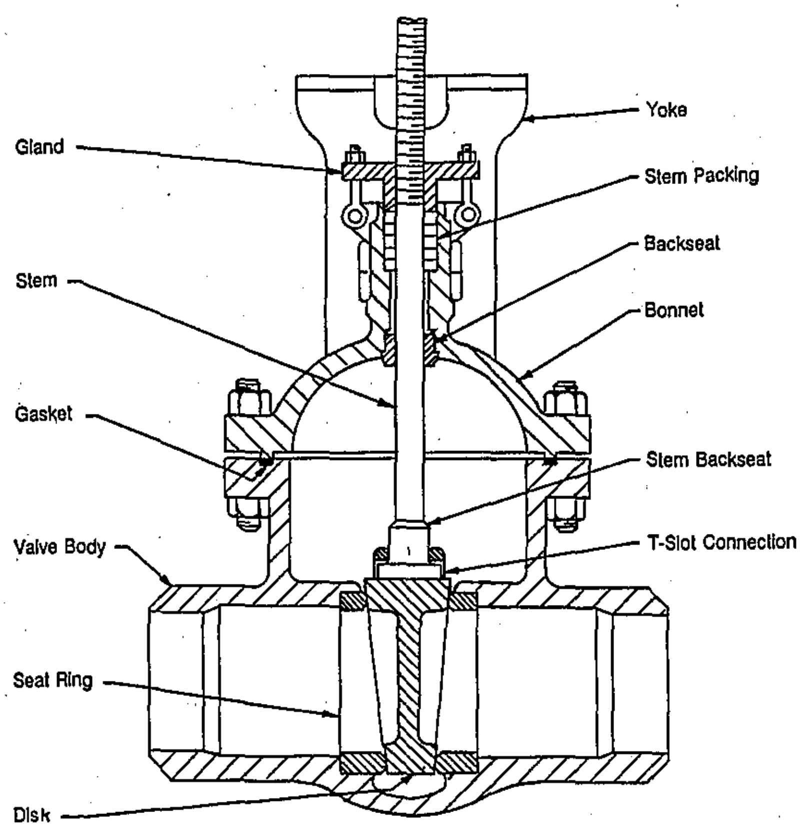

Figure 1. Typical bolted bonnet gate valve.

- Stem-to-disc connection consisting of a T-head and a T-slot
- A disc guiding system consisting of body guide rails and disc guide slots
- Flat disc and seat ring sealing faces.

Friction properties for the following material combinations are currently included in the model:

- Seating faces, stellite[a]
- Guides, either carbon steel, stainless steel, or a hardened material such as stellite.

The model can calculate the behavior of valves with other material combinations if the proper friction coefficient information is made available. Finally, the model covers valves in both incompressible and compressible flow, and calculates stem thrust for strokes in both the opening and closing direction.

The required input information for the model to support a calculation of stem thrust includes the following: key dimensions of the disc, seat rings, guides and stem, material combination at the guide interface, stroke direction, temperature, fluid medium, upstream pressure, and DP as a function of stroke position and packing friction load.

MODEL DESCRIPTION

The primary challenge in thrust prediction is the calculation of forces resulting from DP across the disc. However, the following components are not related to DP across the valve:

- Dead weight of the stem and disc
- Stem rejection created by the tendency of internal pressure to expel the stem out of the body
- Packing frictional drag
- Frictional drag load produced by reaction of stem torque outside the valve in MOVs (reaction of torque inside the valve causes an interaction with the DP load).

These loads are straightforward to evaluate and are not discussed here. Typically, the dead weight and torque reaction loads are negligible. The stem rejection and packing loads are normally not negligible, but their contribution to overall stem thrust tends to diminish as valve size increases.

Fluid Load On Gate Valve Disc

The model calculates the fluid load on the disc by first determining the DP at each stroke position. An algorithm then determines the resultant load based on the DP and the valve dimensions.

The DP across a gate valve during its stroke is dependent on the fluid system in which the valve is installed. Straight forward hydraulic models can be used to determine DP throughout the stroke if the valve resistance as a function of position is known. Data from flow testing of gate valves in the EPRI program have been evaluated to determine nondimensional flow coefficient as a function of stroke. Figure 2 shows the test results and the curve used in the model for flow coefficient versus stroke. The flow coefficient is normalized by the square of the seat ring bore diameter. Using this approach, data from six valves ranging in size from 3 to 18 in. are seen to agree closely. The disparity of observed results near the valve open position results from the very low DPs measured in this part of the stroke. The stem thrust during this part of the stroke is low; consequently, accurate prediction in this region is not important.

Figure 3 shows a gate valve in mid-stroke position with flow through the valve. The restriction of flow as it passes through the valve causes pressure differences on the surfaces of the disc.

a. Stellite is a trademark of Haynes Alloys, Inc., for wear-resistant cobalt-chromium-tungsten alloys. The most common material used in gate valves is stellite 6.

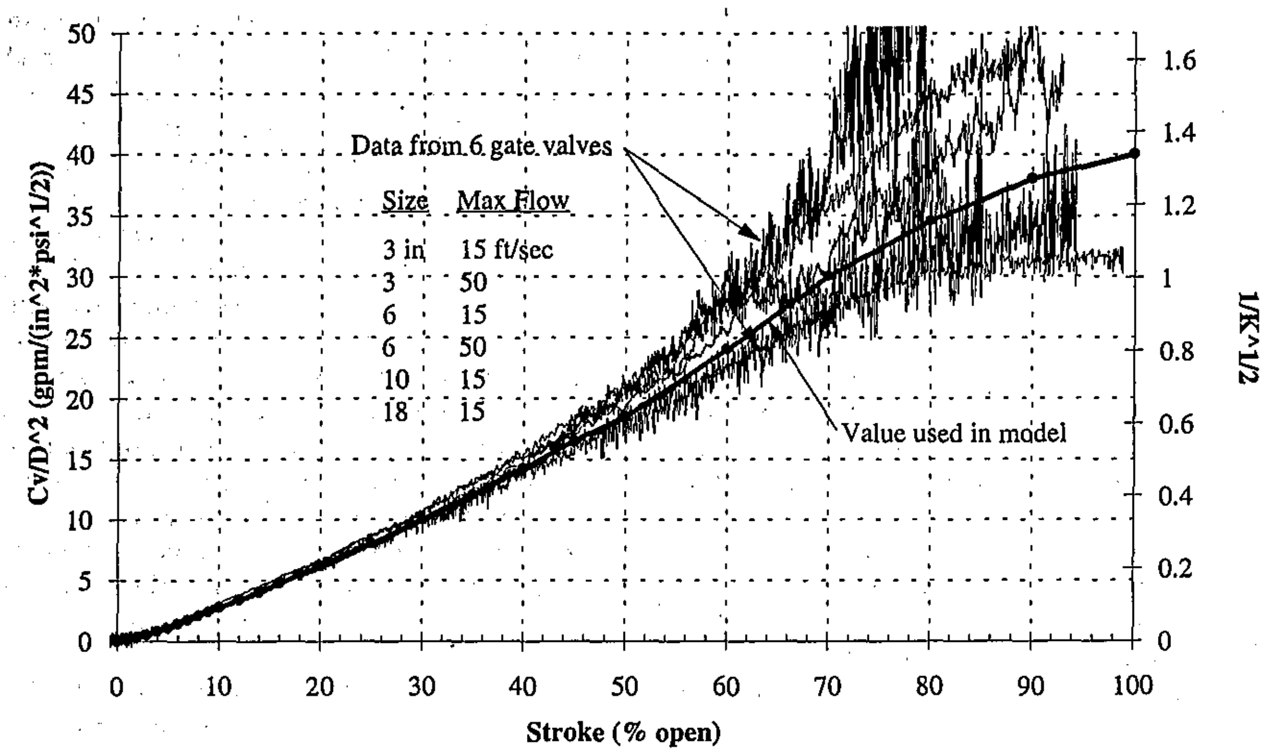

Figure 2. Gate valve flow coefficient as a function of stroke.

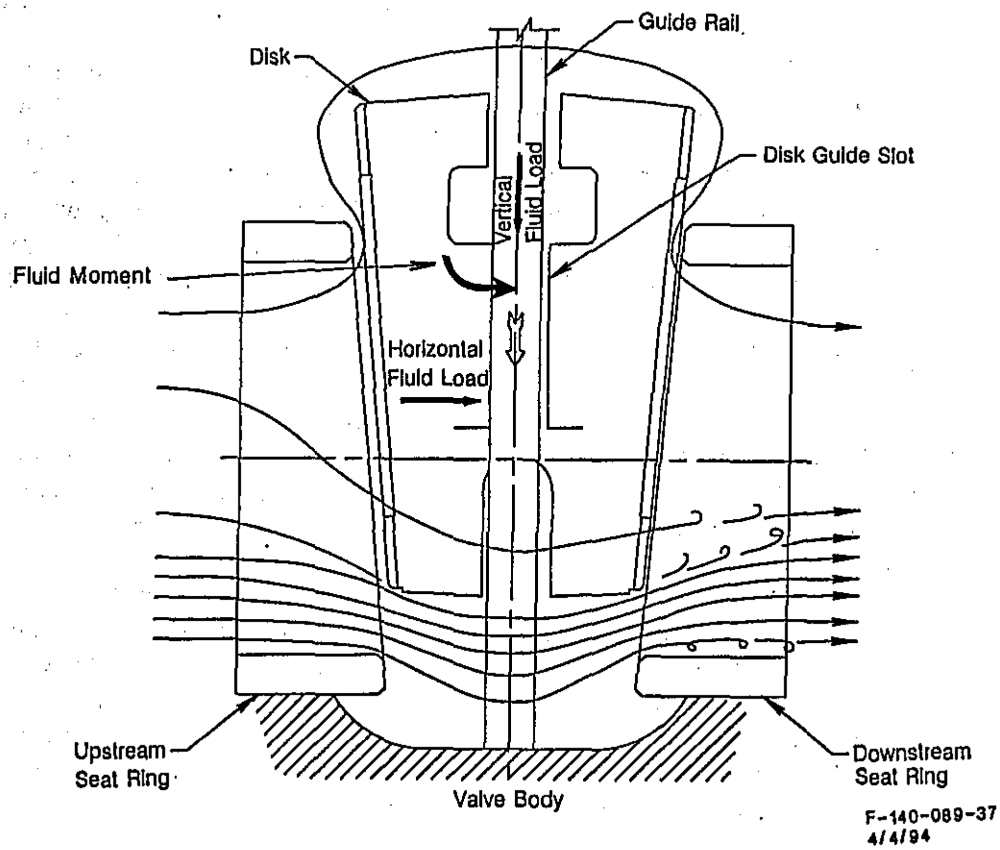

Figure 3. Fluid flow around gate valve disc.

- The pressure on the upstream face of the disc tends to be close to the upstream stagnation pressure, particularly on the portion of the disc protruding into the pipe.

- The pressure below the disc tends to be significantly reduced as a result of fluid acceleration and irreversible pressure losses.

- The pressure on the downstream face of the disc is significantly reduced from the upstream pressure resulting from wake separation at the disc trailing edge. Pressure recovery may occur downstream.

- The bonnet pressure (above the disc) is between the upstream and downstream pressures, but tends to be closer to the upstream pressure, particularly when the valve is near the closed position.

The resultant force on the valve disc has components in the directions of the pipe axis and of the stem axis. The pipe axis component acts in the same direction as the flow, and the stem axis component acts in the downward direction in Figure 3. The presence and importance of the downward component is discussed by DeWall and Steele (1989). The line-of-action of each force, which is important in determining stem thrust, is dependent on the detailed pressure distribution. The pipe axis force tends to have a line of action below the center of the disc and thus produces a counter-clockwise moment in Figure 3.

A set of computational fluid dynamics (CFD) analyses has been performed to determine the disc load as a function of DP, stroke position, disc dimensions, seat dimensions, and disc tilt angle. The methods are described by Bilanin (1992). A base valve geometry representative of typical gate valves was modeled using approximately 50,000 mesh points to represent the flow channel. Analyses were carried out on the base geometry for several stroke positions. In disc positions less than 10% open, the analysis assumed that the disc was flat against the downstream seat ring. In positions more than half open, the disc was held off of the downstream seat ring to simulate being supported by the guide rails. Between 10 and 50% open, analyses were performed both ways. In selected disc positions near fully closed, parametric analyses were performed where key disc and seat ring dimensions were varied and the disc tilt was changed. The analyses described above were conducted assuming incompressible flow; selected cases were also analyzed using compressible flow. The INS-3D computer code was used for the incompressible calculations and the PARC-3D computer code was used for the compressible flow calculations. The calculated results were processed to determine the net load on the disc in the pipe axis and stem axis directions, and the moment about the center of the disc.

The results are summarized on Figures 4 and 5. Figure 4 shows the normalized forces acting on the disc center point in both the pipe axis and stem axis directions. The forces are normalized by the product of the DP and the area based on mean seating surface diameter. The calculations for the base geometry are represented by the lines, and the parametric calculations are represented by the additional points. The pipe axis load is seen to be a maximum near the fully closed position, as expected. The stem axis load is approximately 10 to 15% of the pipe axis load when there is flow through the valve. Most of the parametric variations had a minor effect. However, the axial location of the disc (flat on the seat or supported by the guides) had some influence when the valve is about half open. Figure 5 shows the normalized moment about the disc center point. As expected, the moment is near zero at the fully open and fully closed positions and is a maximum at an intermediate disc position.

The gate valve model uses the results of Figures 4 and 5 to calculate the forces and moment on the disc at each stroke position using the value of DP. The resulting forces and moment are used in subsequent calculations that consider the interaction of the disc with the guides, stem, and seat rings.

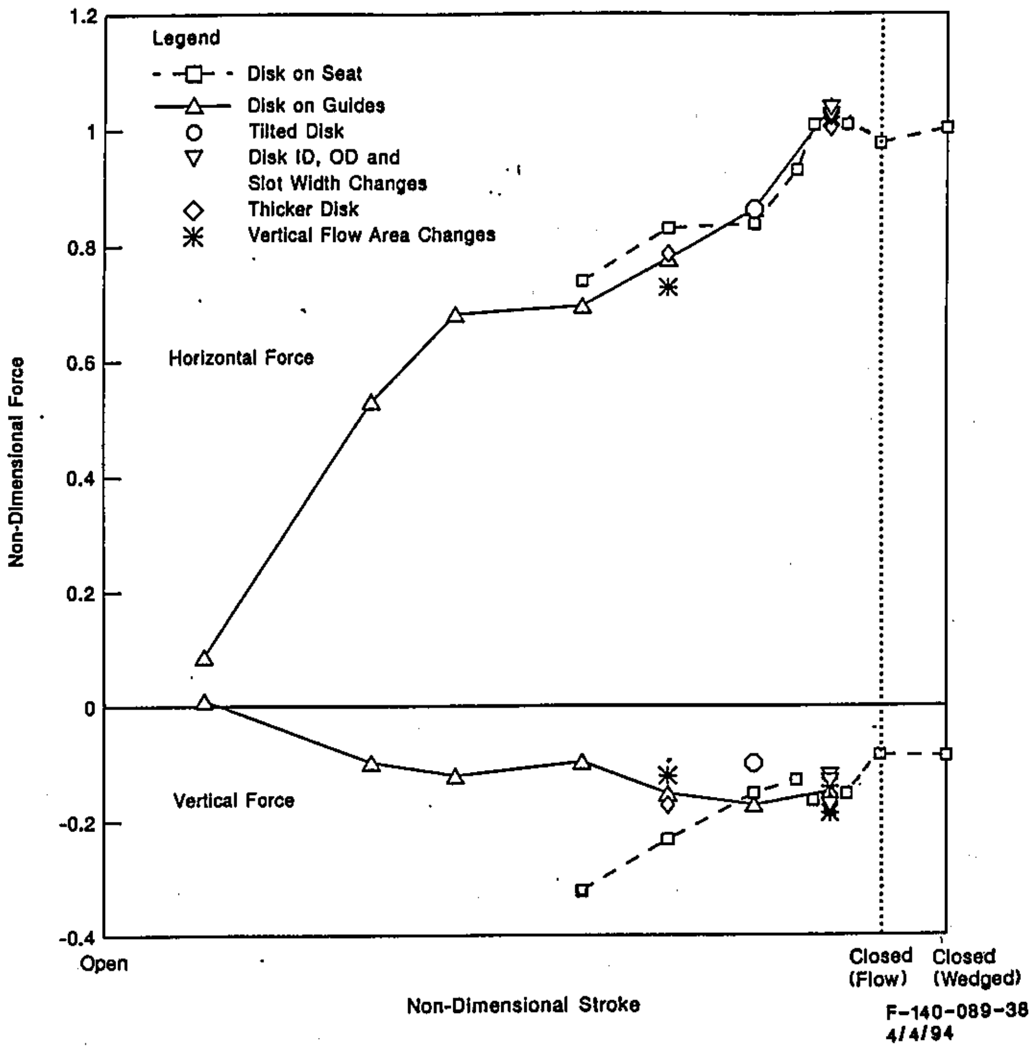

Figure 4. Horizontal and vertical fluid forces on gate valve disc.

Mechanical Interaction of Gate Valve Internals

The fluid loads applied to the disc are reacted by contact of the disc with the stem, guide rails, and seat rings. At each of these contacts there will be both normal and friction loads. The friction forces resist the relative motion (or incipient motion) of the two surfaces in contact. The relative motion is determined by the stroking direction of the disc, either closing or opening.

At any given stroke position except for the final wedged position, the disc has limited freedom to translate and to rotate (tilt) because of clearances in the guiding system and in the stem-to-disc connection. The orientation achieved by the disc is dependent on the applied fluid loads and the available load reaction points and surfaces. Figure 6 shows a disc in a position about 80% closed. In this position, if the disc tips such that the bottom end moves in the direction of flow, it can tip until it is constrained by contact with the seat (toward the bottom) and contact with the guide rail (at the top of the guide slot). Disc-to-seat contact occurs at two points where the outside of the disc contacts the inside diameter of the seat ring. If the disc does not tip, it will establish flat contact between the disc and seat.

In Figure 6, the stem is pushing down on the disc. As the disc tips, or translates along the pipe

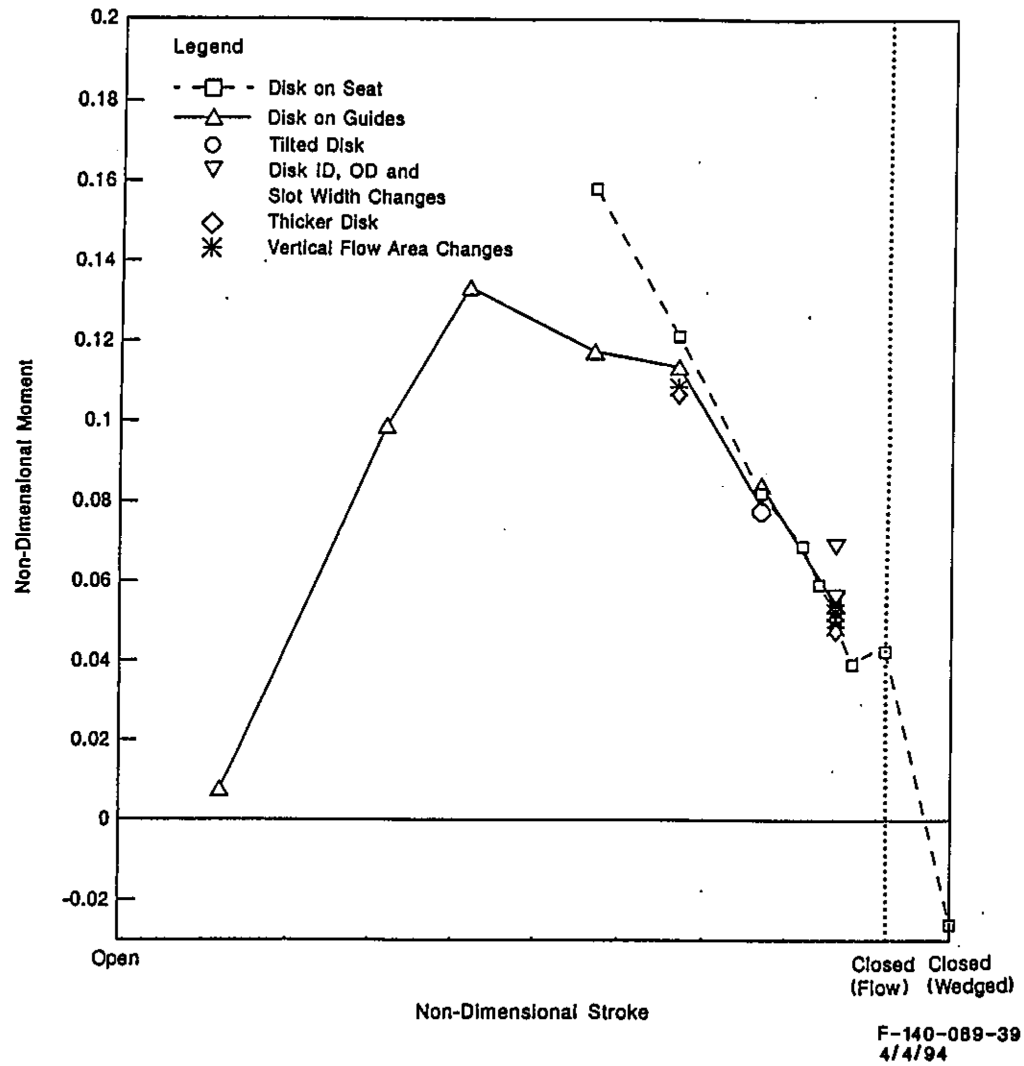

Figure 5. Fluid moment on gate valve disc.

within the clearance permitted by the guides, it tends to drag the stem with it. Because the stem has rigidity to resist this translation, a force opposing disc translation will be applied to the disc by the stem. This force is limited by the maximum friction force that can be sustained between the stem and disc; slippage will occur if additional translation takes place beyond that which develops the maximum friction force. The effect of the friction force between the stem and disc is that the disc may be constrained in a partially tipped position, rather than move fully to one of the limit configurations (tipped or untipped) based on the guide rail and seat ring constraints.

An analytical model was formulated to solve the force equilibrium for the disc as it is moved through a stroke in stepwise fashion. The model keeps track of the cumulative stem slippage in the T-slot to properly account for the effect of stem restraint. The equilibrium position of the disc within its clearances is determined at each stoke position, and the forces at each location where the disc contacts the stem, guide rails, and seat rings are determined. The calculated force component in the stem-axis direction at the stem-to-disc interface is the required stem thrust to move the disc. This force balances the summation of all of the other vertical force components acting on the disc. The model equations, although technically

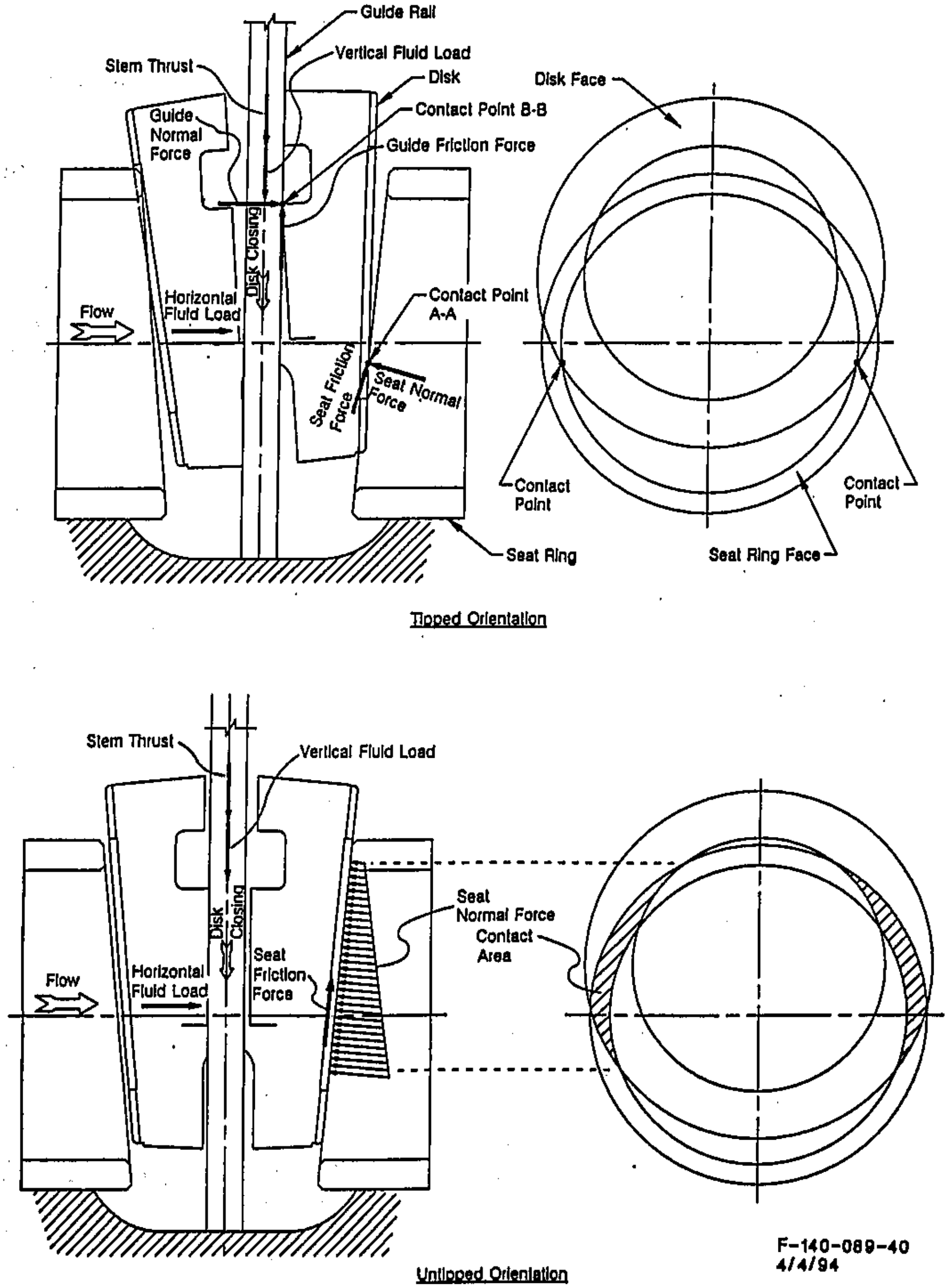

Figure 6. Gate valve forces in tipped and untipped orientation.

straightforward, are complicated and laborious to solve because of the geometry of the parts and the need to calculate contact points and contact planes for a tipped disc. For this reason, the model has been implemented as a computer routine.

Valve Design Effects Testing

To help ensure that the model correctly calculates the contact behavior of the disc with the stem, guide rails, and seat ring, separate effects tests were performed to simulate the interaction of these parts. The test fixture constructed for these tests, called the Valve Design Effects (VDE) facility, is shown in Figure 7. The key features of the facility are summarized as follows:

- The fixture body contains a gate valve disc, seat ring, guide rails, stem, and packing. Actual valve components (from 10-in. valves) or parts machined to simulate the valve components are used.
- A motor operator is used to stroke the disc up and down.
- A hydraulic loading system is used to apply simulated fluid loads to the disc during its stroke. Two cylinders are used to allow force and moment to be independently controlled. The cylinder loads are automatically controlled during testing to achieve prescribed load versus stroke profiles.
- The parts are flushed with distilled water during testing.
- Instrumentation is included for measuring cylinder forces, stem thrust, stem lateral load (at packing), stem torque, stem position, and disc tilt.

Several tests were run in the VDE facility to investigate the effects of changes in key dimensions such as guide length, guide position, and guide clearance. In addition, loading profiles were systematically varied to simulate various levels of DP and flow conditions ranging from nominal pumped flow to blowdown.

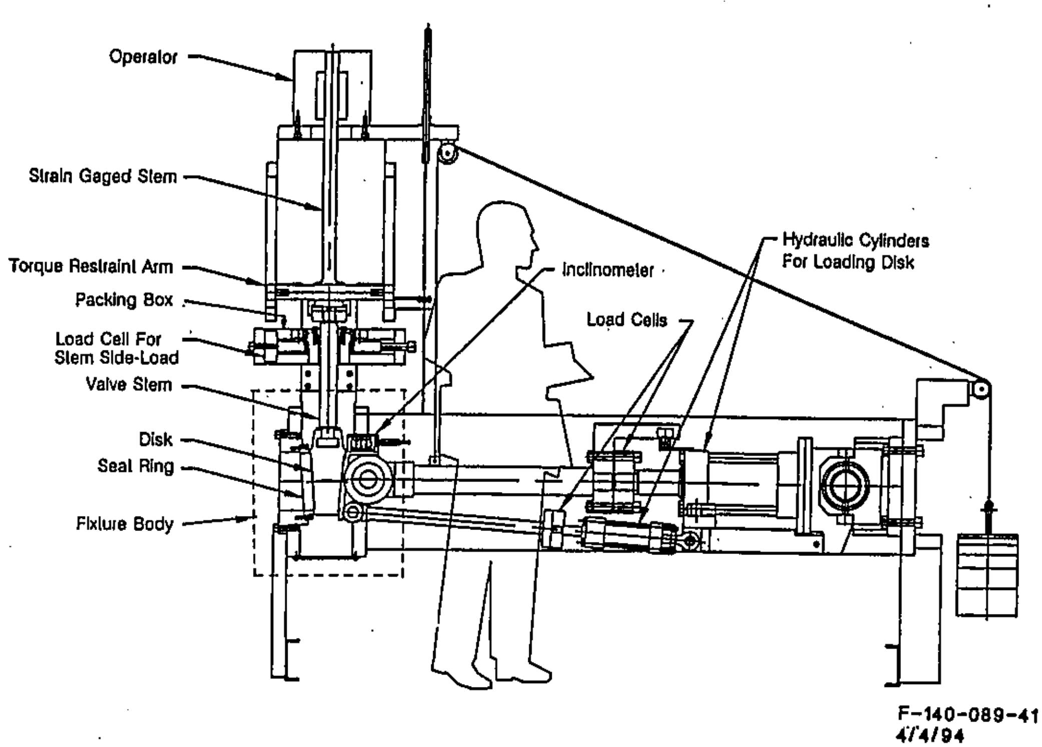

Figure 7. Schematic of valve design effects facility.

The gate valve model was used to predict the disc behavior for the tests using the measured hydraulic cylinder loads as input. The disc tip angle, the stem lateral load, and the trend of stem thrust were the key parameters evaluated against model predictions. The magnitude of stem thrust is dependent on the values of friction coefficient that occurred in the test fixture; these were determined from the test results. (See discussion of Friction Coefficients that follows.)

The comparisons of the model predictions with data confirm that the model correctly predicts the mechanical behavior of the disc as a function of stroke. Figure 8 shows a typical result and the

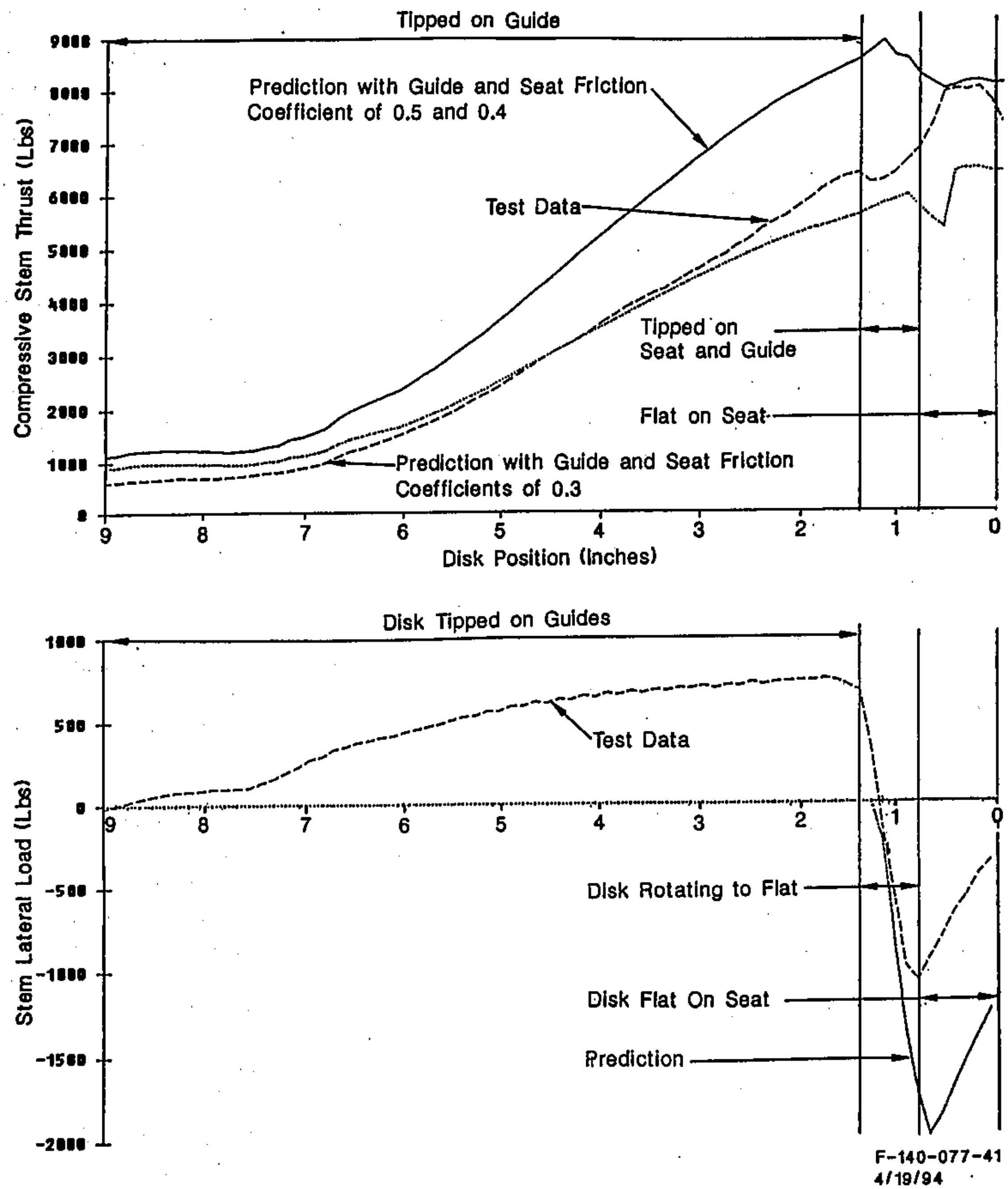

Figure 8. Measured and predicted stem thrust and stem lateral load for valve design effects test simulating 10-in. valve at 300 psi.

corresponding prediction for a simulated closure against 300 psi, pumped flow conditions. The disc is observed to progress through a range of interaction modes, including tipped on guide rails, tipped on guide rail and seat ring, and flat on seat ring. The transitions from one mode to another occur at specific positions that are seen particularly in the stem lateral load data. The model and the data show favorable agreement.

Friction Coefficients

A key parameter that determines stem thrust is the coefficient of friction at each point where the disc contacts the guides, stem, and seats. Table 1 identifies the key interfaces in a gate valve where friction forces need to be evaluated and shows typical materials at those interfaces, as well as the potential contact configurations and the typical and extreme values of contact stress (or load) that occur in nuclear service. In addition, the table identifies contact "modes" that can occur at each interface. These modes define the contact configurations that are possible at each contact point. For example, the disc-to-seat contact can occur in a "flat-on-flat" sliding mode or, if the disc is tipped, in an "edge-on-edge" contact mode. Appropriate friction coefficients were required for each of these situations. Existing literature, including McGee and McPherson (1956), Dewees (1957), Hofman and Wieling (1980), Airey (1988), Simon et al. (1989), and Wang and Kalsi (1992) did not provide sufficient data. Accordingly, tests were performed to better establish these friction coefficients. Data were obtained from two programs. The VDE test program described above provided data that primarily addressed the edge-on-edge disc contact mode. In addition, an extensive test program was conducted using small friction test specimens that provided data for all contact modes across a considerable range of load, temperature, and fluid media. These tests were performed both in a low temperature (70–200°F) water bath and in an autoclave that included high-temperature steam and water environments. Figure 9 shows the two major facilities used for the sliding specimens. A few key test results are discussed in the following paragraphs.

Self-Mated Stellite (Flat-on-Flat). The friction coefficient for stellite 6 in flat sliding is principally dependent on stroke history, temperature, and contact stress. At ambient temperature the friction coefficient increases with cumulative stroking and eventually reaches a stable level that is dependent on contact stress. The stable friction coefficient decreases with increasing contact stress, although there is considerable scatter and variability at contact stresses below 8 ksi. The maximum observed friction coefficient is between 0.6 and 0.7. As temperature increases, the friction coefficient decreases. At temperatures of 550 to 650°F, which are typical limits for nuclear service, the maximum friction coefficient is in the range of 0.4 to 0.5.

Self-Mated Stellite (Edge-on-Flat). In this configuration the friction coefficients are not as high as the values seen in flat sliding. The explanation is that regardless of load, the parts deform or wear to achieve a contact stress consistent with the material bearing stress limit, and a friction coefficient typical of high contact stress is observed. The maximum friction coefficient is in the range of 0.3 to 0.4.

Self-Mated Stellite (Edge-on-Edge). With two edges in contact, the contact stress is inherently high because of the small contact area. Deformation and wear occur to increase the contact area. If the load is not high and the edges are not sharp, this readjustment is imperceptible and the friction coefficient is similar to edge-on-flat. If the load is elevated and the edges are sharp, the readjustment results in material removal and deformation that can be seen by eye. Under these conditions, the "apparent" friction coefficient is significantly increased, with maximum values approaching 1.0. This value decreases sharply to the normal, expected levels for strokes after the first stroke, as a result of the edges being reconfigured. It was found that if a sharp-edged pair is stroked successively with gradually increasing load, the friction coefficient remained stable throughout, even up to very high loads, because the material readjustment was occurring in small, imperceptible steps.

Table 1. Key friction interfaces in gate valve.

Region	Typical material pairs	Possible contact configurations	Typical load	Maximum load
Disc-seat	Stellite-stellite	Flat-flat	Up to 15,000 psi	50,000 psi
		Edge-edge scissoring	Up to 20,000 lb	50,000 lb
Guide rail-slot	Mild steel-mild steel Mild steel-hardened Steel Mild steel-stellite Stellite-stellite	Flat-flat	Up to 5,000 psi	30,000 psi
		Edge-flat	Up to 10,000 lb/in.	100,000 lb/in.
		Edge-edge nonscissoring	Up to 5,000 lb	50,000 lb
Stem-disc	Hardened steel-mild steel	Flat-flat	Up to 40,000 lb	70,000 lb

Self-Mated Mild Carbon or Stainless Steel (Flat-on-Flat). The behavior of this combination is dominated by the potential for galling of the two surfaces. At room temperature, galling does not typically occur for normal cast or machined surfaces, and the friction coefficient is about 0.4. At temperatures above a threshold in the range of 100–200°F, galling can occur. The onset of galling often requires several strokes. When galling occurs, the apparent coefficient of friction increases to values that can approach 1.0. Typically, the value reduces after continued stroking to a stable level of about 0.6.

Self-Mated Mild Carbon or Stainless Steel (Edge-on-Flat or Edge-on-Edge). In this situation, the contact stress typically exceeds the bearing strength of the materials, and deformation is needed to supply adequate contact area. At room temperature, the materials deform readily without excessive traction force buildup. As discussed above for flat sliding, galling tends to occur as the temperature is elevated, and the friction coefficient is increased.

Stellite on Carbon or Stainless Steel (Flat-on-Flat). This combination tends to be relatively well behaved, with a friction coefficient of about 0.4, which is not strongly sensitive to contact stress or temperature.

Stellite on Carbon or Stainless Steel (Edge-on-Flat or Edge-on-Edge). This combination is well behaved, as described above with flat-on-flat, as long as the stellite edges are rounded. If the stellite edge is not sufficiently rounded or chamfered, significant damage to the mating material can occur, and the apparent friction coefficient can increase dramatically.

Discussion

The results of the tests were used in conjunction with data from existing literature to create the friction coefficient algorithm used in the model. The algorithm is intended to provide bounding friction coefficients suitable for use in designing valves or performing design-basis evaluations of valves.

The friction coefficient between the disc and the seat ring for flat-on-flat sliding is usually the most important friction coefficient for determining the maximum required thrust to open or close a gate valve. This occurs because the DP is usually at its maximum value when the valve is near

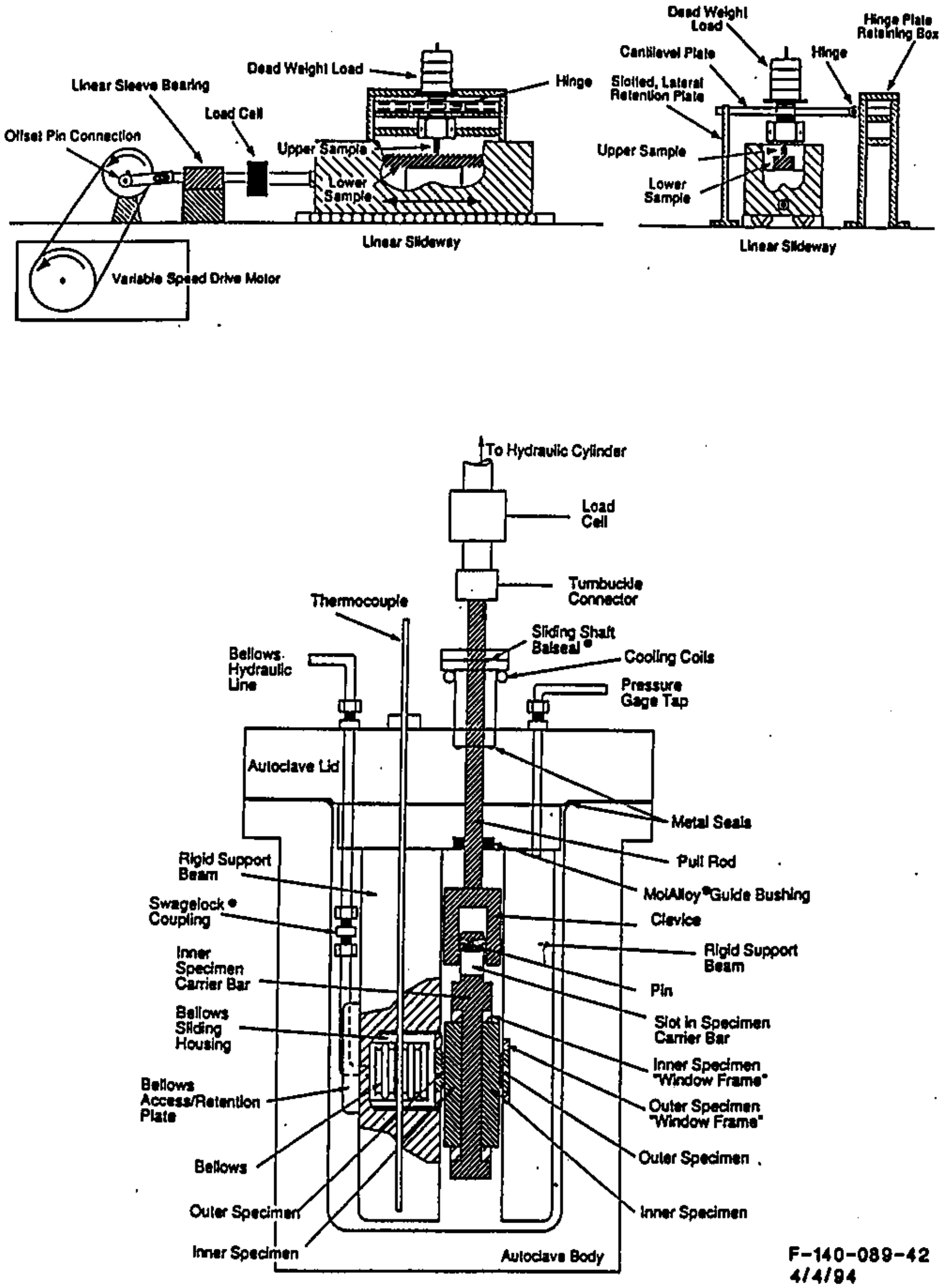

Figure 9. Ambient and autoclave facilities used for friction testing.

or at the closed position, and the disc is sliding across the downstream seat ring during this portion of the stroke. The friction tests showed that considerable variability in the friction coefficient for stellite occurs, particularly at ambient temperature and low contact stresses. The minimum value is in the range of 0.1 to 0.2 and the maximum value is in the range of 0.6 to 0.7. The reasons for the variations are not fully understood, but are thought to be related to surface metallurgical, chemical, or topographical changes.

It is desirable in many cases to be able to assess the in situ value of the friction coefficient and to base actuator setpoints on this value. Because diagnostic testing of valves has become increasingly popular, the measurements needed to determine apparent friction coefficient are available in

many cases. Evaluation of the EPRI flow-loop test data has shown that there are at least three ways in which the friction coefficient can be determined reliably. The first two ways are a hydropump test or a flow test conducted at 100% or less of full DP. The friction coefficient is determined by evaluating the measured thrust over the portion of the stroke where the disc is sliding on the seat. The fact that the friction coefficient for stellite decreases with increasing contact stress ensures that the value determined at partial DP will bound that at full DP.

A third method has been developed that, though not as accurate as the first two methods, is simple to perform. It has been found that thrust measurements made during the wedging and unwedging of a gate valve in the absence of DP provide an indication of the friction coefficient at the disc-to-seat interface. In particular, the ratio of unwedging thrust to wedging thrust (termed S) is related to the friction coefficient and the wedge angle. Flow-loop test data from several gate valves were analyzed to evaluate the relationship between S and friction coefficient. Results indicate that, except for a certain class of disc design produced by one manufacturer, wedging and unwedging thrust measurements can be used to determine the friction coefficient using this method.

Disc Unwedging Force

During initial opening of a gate valve, a high thrust is needed to unwedge the disc from the seat rings. In the absence of any changes in DP between closure and opening, the unwedging force can be determined from a free body evaluation of the forces on the disc. However, if a DP is applied (or the DP is increased) after the valve is closed, this load potentially tends to increase the disc-to-seat load and hence the unwedging load. Therefore, it is necessary to evaluate potential increased opening thrust capability to cover this scenario.

A change in DP while the valve is in the closed position affects the contact load at the downstream disc-to-seat interface. This contact load change potentially affects the required unwedging thrust. For example, if a DP is applied to a closed valve, the DP load will be reacted by a combination of increase in the downstream disc-to-seat contact force and decrease in the upstream contact force. If the decrease and increase are unequal, the unwedging load will change. By examining data from several gate valves, we have found that using a 40%/60% split of load reaction between upstream and downstream surfaces is justified for evaluations. Figure 10 shows the results of calculations performed for several gate valves that were closed with no DP and opened after DP was applied. The predictions assuming the DP load is borne 100% at the downstream seat are seen to exceed the data considerably, while calculations performed using a 40%/60% load distribution provide a more reasonable approach, and are used in the gate valve model.

COMPARISON TO GATE VALVE TEST DATA

Test Data

Flow-loop tests of 22 FWG and SWG valves were carried out as part of the EPRI MOV Performance Prediction Program. In addition, at the time this paper was prepared, detailed test data from seven FWG valves tested in nuclear power plants had been obtained.

The valves mentioned above were tested under a range of conditions including ambient water pumped flow, pressurized hot-water flow, steam flow, and high-energy blowdown with hot water, and steam. Furthermore, the flow-loop tests included parametric tests studying different levels of DP on each valve.

All of the tests had time-history measurements of upstream pressure and DP. Further, detailed measurements of the dimensions of the valve disc, stem, and body were made. These data provided the necessary inputs for gate valve stem thrust predictions using the model. Stem thrust was measured in each test and compared with

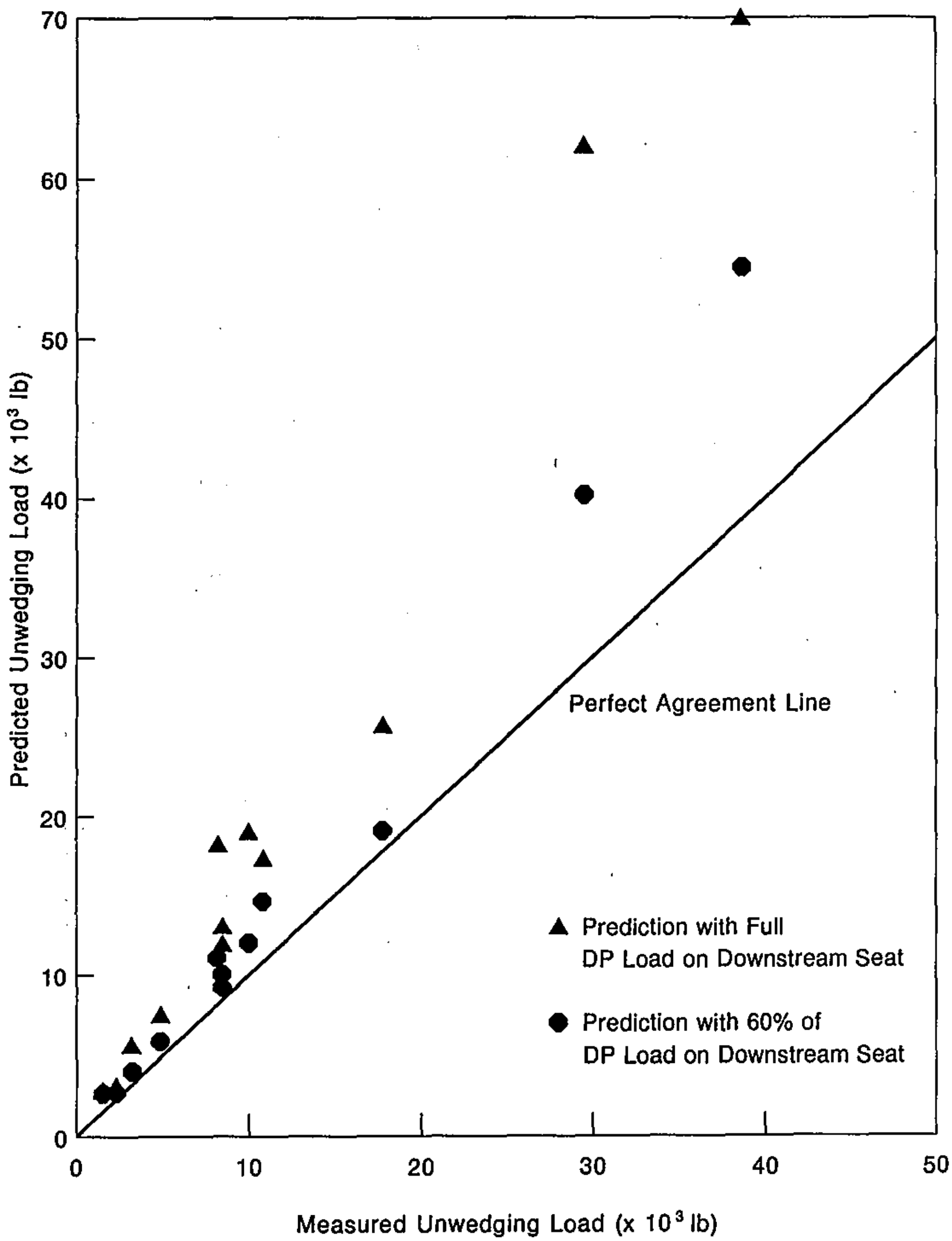

Figure 10. Comparison of predicted unwedging load to data from gate valves opened with DP after being closed with no DP.

model predictions to assess the accuracy of the model. Examples of the model-to-data comparisons are given in the following paragraphs.

Example Comparisons of Model Predictions and Data

Figure 11 shows a comparison of measured and predicted stem thrust for a 3-in. FWG valve tested with water flow at 15 ft/s and DP at 780 psi. Results for a closure stroke are plotted as a function of stroke position in percent, where 100% is fully open and 0% is flow isolation (full disc overlap of seat ring). The disc wedges at about the –15% position.

During the closure stroke, the compressive stem thrust (plotted as a negative value) remains relatively constant for the first 90% of the stroke. In the final 10% of the stroke, thrust increases as the valve approaches the closed position, because of the DP increase. From the flow isolation to the wedged position, the thrust remains relatively constant as the disc slides across the seat ring with a stable DP. At the wedging position, the thrust increases very rapidly because of the actuator characteristics. This final thrust increase is not a thrust required by the valve and is not within the scope of the gate valve model. Accordingly, predictions and measurements are meaningfully compared only before wedging occurs.

For this valve, the disc is predicted to pass through a range of behavior during the closure stroke including tipped on the guides, flat on the guides, and flat on the seat ring. Importantly, though, the thrust is dominated by the portion of the stroke where the disc is sliding flat on the seat ring near the fully closed position. This is consistent with the calculations shown in a previous paper (Wang et al., 1992), which indicated that disc tipping effects had a minimal effect under pumped flow conditions.

For this valve, the stem thrust predicted by the model bounds that measured during the test. This results principally from the fact that the friction coefficient used in the model for stellite sliding (about 0.6 for this valve under the tested conditions) bounds that which occurred in testing. As shown in the figure, if a friction coefficient of 0.38 is put into the model in place of the bounding value, improved agreement within the data is observed.

The results in Figure 11 are for a valve that had relatively loose clearances in the guides. This looseness allows the disc to slide on the seat ring for the last 10% of the stroke before flow isolation. In other valves that have tighter clearances, the disc slides on the guides until nearly the very end of the stroke, and only a short distance of seat sliding occurs. Figure 12 shows results for a 6-in. tight-clearance valve with 50 ft/s and 1800 psi DP water flow conditions. The transition between guide sliding and seat sliding occurs near the zero stroke position. Because the valve is significantly loaded when this occurs, this produces a "step" in the thrust signature, both in the data and the prediction. The step is associated with two phenomena: (a) a change in friction coefficient associated with the change in sliding surface and (b) a change in the surface angle between the guide rail and seat ring. Once again, the model is observed to bound the data. If guide and seat friction coefficients of 0.39 and 0.49 are put into the model, a more favorable agreement is observed.

Figure 13 shows predictions and data for a closure stroke of a 6-in. valve under hot-water blowdown conditions at 1,200 psi, 530°F. In this case, the valve load increases continuously during its stroke. The model predictions once again bound the data.

CONCLUSIONS

Prediction of stem thrust required to stroke a gate valve needs to consider the fluid loads applied to the disc, the mechanical interaction of the disc with other internal parts, and the friction coefficient at various interfaces where sliding occurs. A detailed model has been formulated for this purpose. The content of the model is supported by detailed computational fluid dynamics analyses to obtain fluid loading, and by separate effects testing to verify mechanical interaction of the disc and to establish friction coefficients.

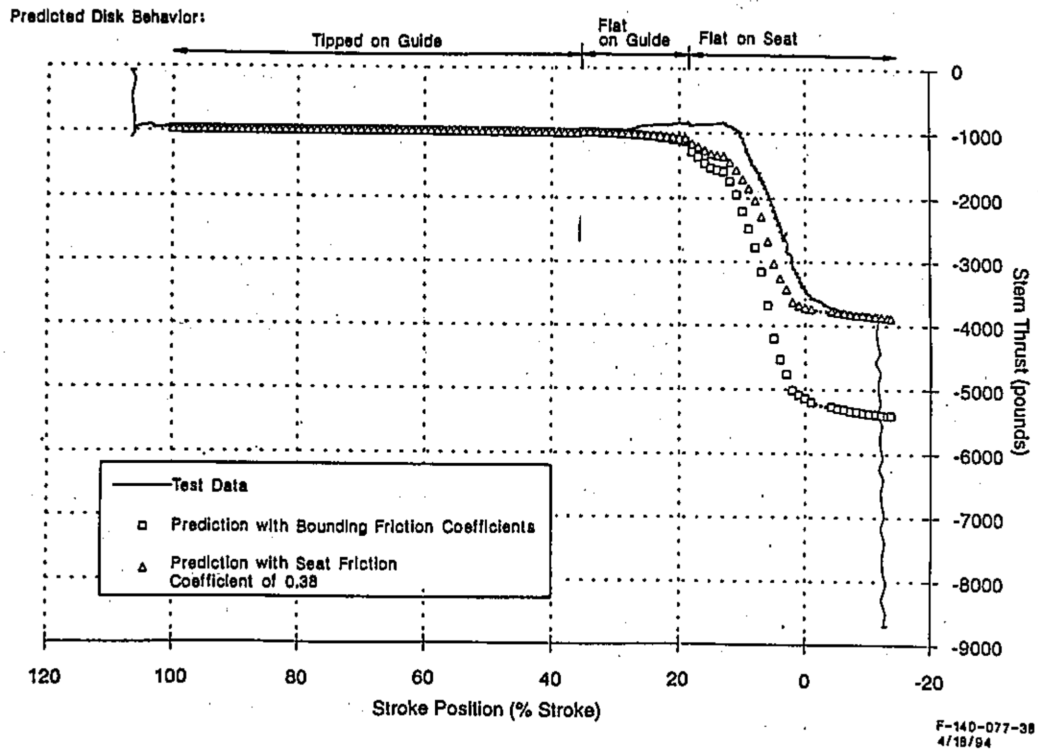

Figure 11. Measured and predicted thrust for 3-in. gate valve in water flow at 15 ft/s and 740 psi.

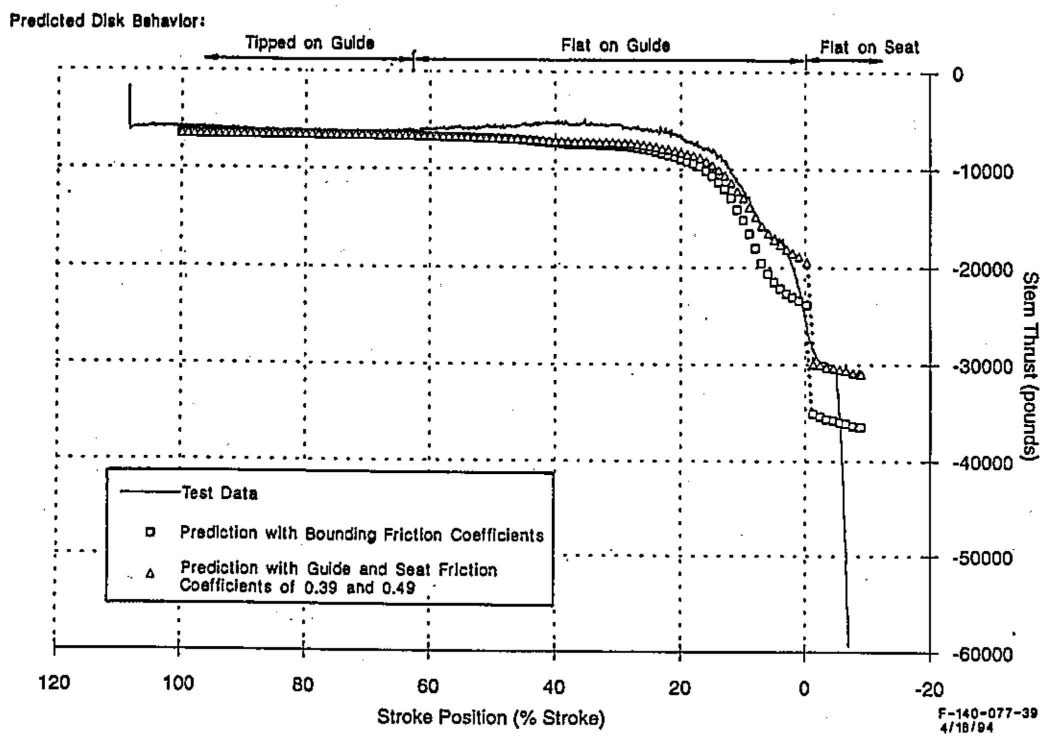

Figure 12. Measured and predicted thrust for 6-in. gate valve in water flow at 50 ft/s and 1,800 psi.

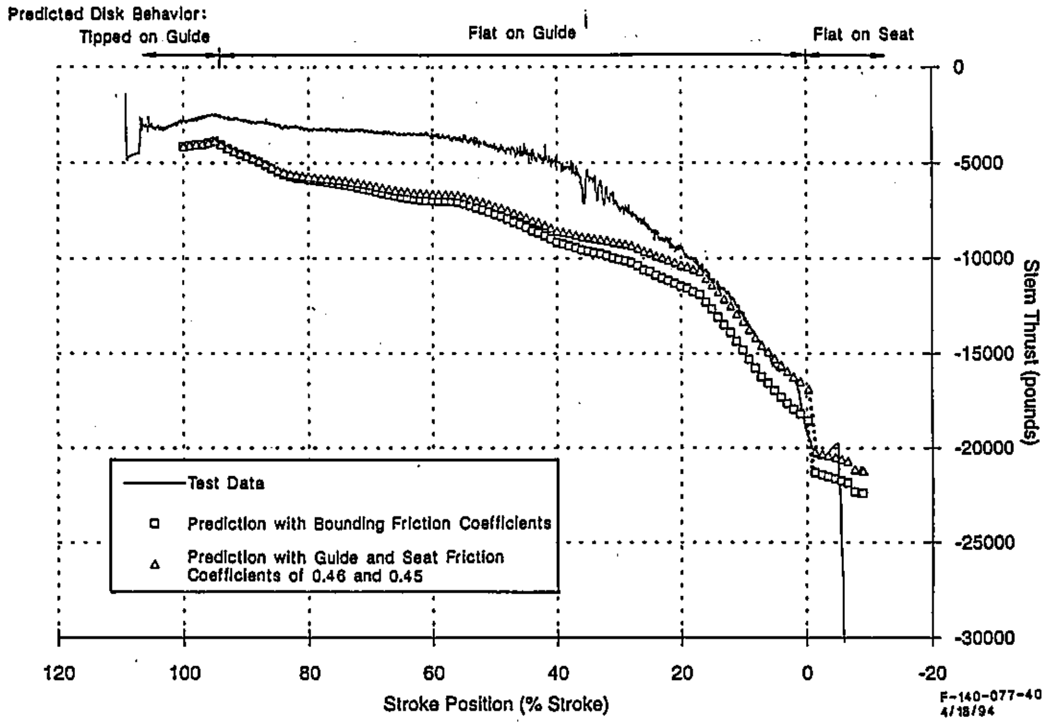

Figure 13. Measured and predicted thrust for 6-in. gate valve in water blowdown flow at 1,200 psi, 530°F.

Stem thrust predicted by the model bounds test data from gate valves, principally because of the conservative nature of the friction coefficients.

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Butterfly Valve Torque Prediction Methodology

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ABSTRACT

As part of the Motor-Operated Valve (MOV) Performance Prediction Program, the Electric Power Research Institute has sponsored the development of methodologies for predicting thrust and torque requirements of gate, globe, and butterfly MOVs. This paper presents the methodology that will be used by utilities to calculate the dynamic torque requirements for butterfly valves.

The total dynamic torque at any disc position is the sum of the hydrodynamic torque, bearing torque (which is induced by the hydrodynamic force), as well as other small torque components (such as packing torque). The hydrodynamic torque on the valve disc, caused by the fluid flow through the valve, depends on the disc angle, flow velocity, upstream flow disturbances, disc shape, and the disc aspect ratio. The butterfly valve model provides sets of nondimensional flow and torque coefficients that can be used to predict flow rate and hydrodynamic torque throughout the disc stroke and to calculate the required actuation torque and the maximum transmitted torque throughout the opening and closing stroke.

The scope of the model includes symmetric and nonsymmetric discs of different shapes and aspect ratios in compressible and incompressible fluid applications under both choked and nonchoked flow conditions.

The model features were validated against test data from a comprehensive flow-loop and in situ test program. These tests were designed to systematically address the effect of the following parameters on the required torque: valve size, disc shapes and disc aspect ratios, upstream elbow orientation and its proximity, and flow conditions. The applicability of the nondimensional coefficients to valves of different sizes was validated by performing tests on a 42-in. valve and a precisely scaled 6-in. model. The butterfly valve model torque predictions were found to bound test data from the flow-loop and in situ testing, as shown in the examples provided in this paper.

INTRODUCTION

The Electric Power Research Institute (EPRI) is developing improved and validated models for predicting the thrust required to operate gate and globe valves, and the torque required to operate butterfly valves in nuclear power plant applications. The focus of this paper is to present the key aspects of the butterfly valve model. The model will be integrated into a computer program for personal computers that will be used by the utilities to assess the thrust or torque requirements of motor-operated valves (MOVs) under user-specified design-basis conditions.

This paper summarizes the basic equations used in the butterfly valve model and the

associated assumptions and limitations. The model predictions have been compared with test data to validate the prediction methodology over a wide range of valve design features and test conditions. Typical examples of these model-to-data comparisons and conclusions regarding the scope and applicability of the butterfly valve model are presented.

Previous Research

Before the development of the EPRI butterfly valve model was started, an extensive literature search was performed to document the technical state-of-the-art and to identify areas of needed improvement. Results of the literature search are documented in the *Application Guide for Motor-Operated Butterfly Valves in Nuclear Power Plants*.[a] A thorough evaluation of data from previous analytical and experimental research, manufacturers' recommendations for actuator sizing, and the commonly used industry standard (American National Standards Institute/American Water Works Association *Standard for Rubber-Seated Butterfly Valves*, 1988) revealed that the following areas needed improvements in developing a butterfly valve torque prediction model suitable for nuclear power plant applications:

- Inclusion of the effect of disc shape and disc aspect ratio (defined as disc thickness/disc diameter) on torque

- Improved prediction of the effect of flow disturbance caused by an upstream elbow on the torque requirements, with proper accounting for the elbow configuration, its distance from the valve, and the direction of disc rotation

- Verification of the torque scaling equations used to predict performance of large valves based on tests performed on small scale models

- Validation of the model against test data obtained under rigorous quality assurance programs (e.g., satisfying 10 CFR 50 Appendix B, requirements) with documented measurement uncertainties for the key parameters.

In developing the EPRI butterfly valve model, all of these areas of needed improvement were systematically addressed by appropriate theoretical and experimental development.

TORQUE PREDICTION MODEL

The objective of the butterfly valve model is to determine stem torque from two standpoints:

- *Required actuation torque*: This is the maximum torque required to operate (open or close) the valve through its entire stroke, including total seating/unseating torque and total dynamic torque.

- *Maximum transmitted torque*: This is the maximum torque that can occur in the valve stem. The capability of the weak link in the valve (e.g., stem, disc-to-stem connection) and the actuator torque rating should both exceed the maximum transmitted torque.

The model provides predictions of both of these values.

Scope

The scope of the butterfly valve model includes the following valve design features, installation details, and operating conditions:

- *Disc Designs*: Symmetric disc and nonsymmetric disc with single offset designs are shown schematically in Figure 1. Geometrical variations in the disc shapes prevalent in nuclear power plants are shown in Figure 2. The butterfly valve model has been validated against test data for all disc shape variations in Figure 2, except Disc Shape 3.

a. B. H. Eldiwany and M. S. Kalsi, Electric Power Research Institute, Nuclear Maintenance Application Center, Charlotte, NC, 1993.

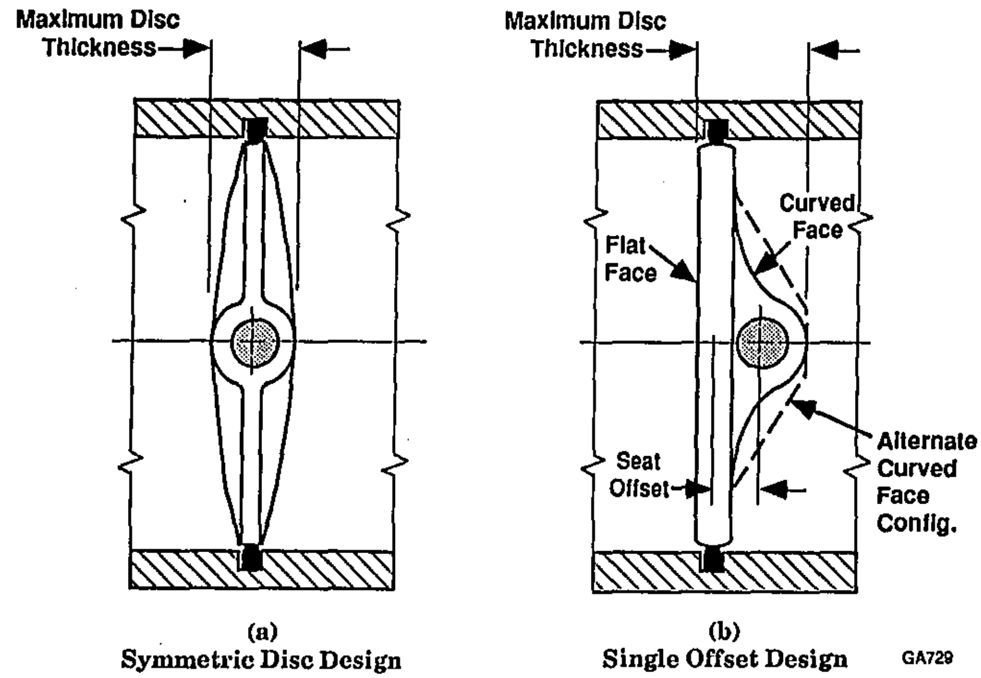


Figure 1. Symmetric and single offset disc butterfly valves.

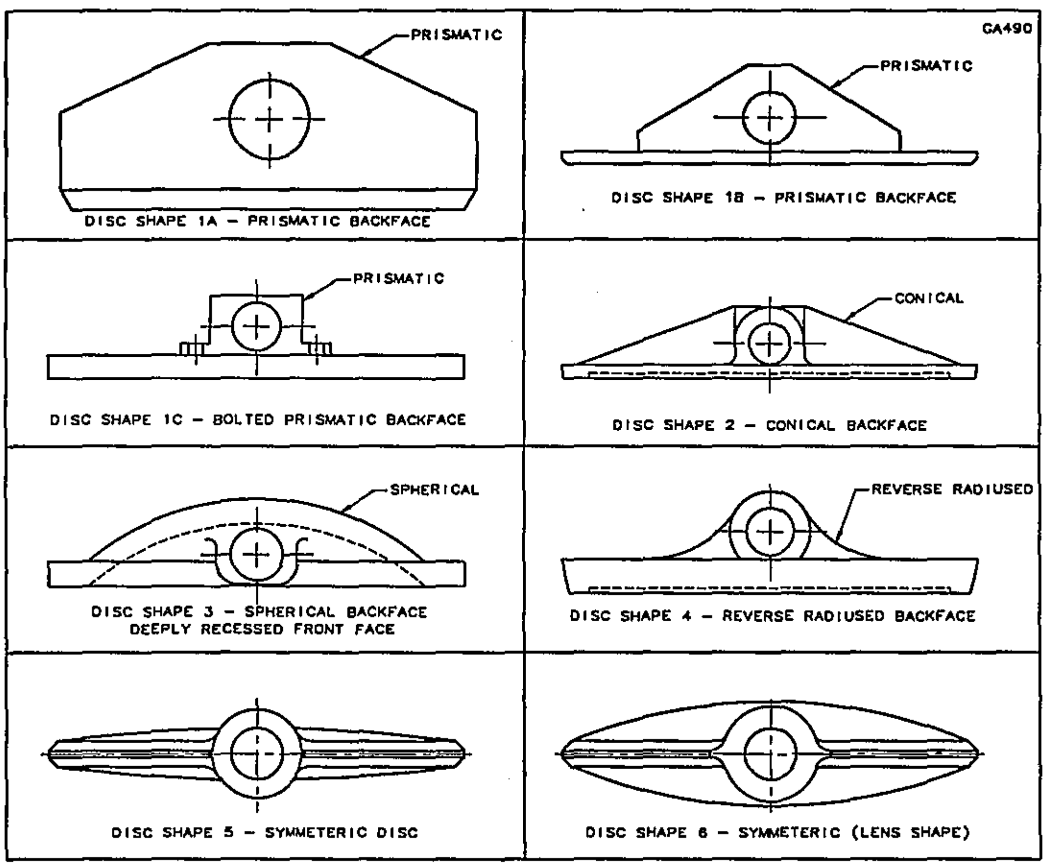


Figure 2. Most common disc shapes used in butterfly valves.

This disc shape constitutes a very small fraction of the total butterfly valve population in the nuclear power plants.

- *Operating Conditions*: Applicable to all operating conditions, including a postulated pipe break immediately downstream of the valve.

- *Seat Designs*: Interference type and pressure-energized seat designs (see *Application Guide for Motor-Operated Butterfly Valves in Nuclear Power Plants* for different types of seat designs).

- *Valve Stroke*: Both opening and closing stroke directions. The model is applicable to full or partial strokes for incompressible flow; full stroke analysis must be performed for compressible flow.

- *Flow Direction*: For nonsymmetric valves, both shaft upstream and shaft downstream flow directions (Figure 3).

- *Flow Condition*: Fully turbulent flow conditions for incompressible flow (both choked and unchoked) and compressible flow (choked).

- *Upstream Flow Disturbances*: Accounts for the effect of an upstream 90-degree elbow on the hydrodynamic torque.

- *Valve Condition and Behavior*: Assumes that the valve components have been properly maintained and are in good working order. It does not take into account anomalous or unpredictable behavior resulting from damaged or degraded seats, bearings, and packings.

Stem Torque Prediction

The model calculates two types of stem torque required to operate the valve:

- Total seating/unseating torque (T_{TS})

$$T_{TS} = T_b + T_p + T_s + T_h \,. \qquad (1)$$

Total seating/unseating torque applies only when the valve is nearly closed ($\alpha \leq$ 5 degrees).

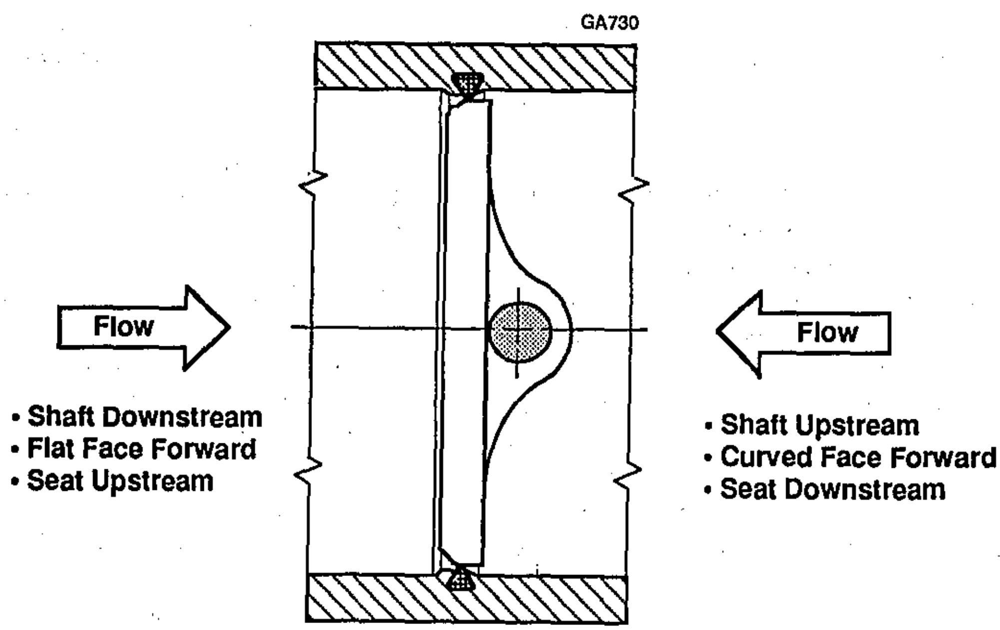


Figure 3. Valve orientation with respect to flow.

- Total dynamic torque (T_{TD})

  $$T_{TD} = T_b + T_p + T_{hub} \pm T_{hyd} \,. \quad (2)$$

  Total dynamic torque applies throughout the stroke (5 degrees < $\alpha \leq$ 90 degrees) and is a function of disc position.

The torque required to actuate the valve is the larger of the total seating/unseating torque and the total dynamic torque.

A torque applied to the stem by the actuator to rotate the disc in either the opening or closing direction is a positive (+) torque. A torque applied to the stem to restrain disc rotation in either the opening or closing direction is a negative (–) torque.

Torque Components

A brief description follows of the individual torque components in Equations (1) and (2), including the method to calculate each one.

Bearing Torque (T_b). Bearing Torque (T_b) is the torque created by the friction force between the stem and the bearings, and is calculated by

$$T_b = \mu_b \times \pi \times d_{disc}^2 \times d_s \times \frac{\Delta P_v}{96} \text{ ft-lb} \quad . \qquad (3)$$

For bronze bearings in clean systems, the model recommends a value of $\mu_b = 0.25$. The value of μ_b is, however, a user input, and lower values of μ_b (e.g., for Teflon fabric bearings used in clean systems) can be used if test data are available to justify the assumption. For bronze bearings in dirty systems and nonbronze metal bearing combinations (e.g., 17-4 PH stainless steel) against hardened austenitic or martensitic stainless steel, μ_b values can be higher than 0.25. The model recommends use of $\mu_b = 0.6$ in such cases where test data are not available.

Packing Torque (T_p). Packing Torque (T_p) is the torque created by the friction between the stem and the packing. Packing torque can vary significantly depending upon packing design, material, and the packing gland preload or torque; therefore, the user of the model is required to supply a value for packing torque. The *Application Guide* provides guidance for estimating packing torque for butterfly valves. In situ tests with no flow and no differential pressure (ΔP) (static tests) can be used to determine packing torque accurately, or to confirm that packing torque is within design values.

For symmetric disc design butterfly valves, the hub remains in contact with the elastomer liner in the body throughout the stroke to provide a seal around the stem. This position creates a frictional torque component called the hub seal torque, which remains nearly constant throughout the stroke, in a manner similar to the packing torque. The user-input value of packing torque must include the contribution from the hub seal.

Seat Torque (T_s). For interference type seats, seat torque (T_s) is calculated by

$$T_s = \frac{1}{12} \times A \times d_{disc}^2 \text{ ft-lb} \quad , \qquad (4)$$

where A is a constant that depends on fluid medium. The model recommends bounding values for the constant, A, for undamaged seats maintained in accordance with manufacturer's recommendations for incompressible flow media (wet service) and for compressible flow media (dry service) applications. The model also accepts a user-supplied value of seating torque that could be based either on in situ test data or manufacturer's data.

Hydrostatic Torque (T_h). Hydrostatic torque (T_h) is caused by the static pressure difference across the valve created when there is fluid on only one side of the disc. This torque component is calculated by

$$T_h = \frac{\pi \varrho}{64} \times \left(\frac{d_{disc}}{12}\right)^4 \times \sin \phi \text{ ft-lb} \quad . \qquad (5)$$

This torque component is significant only for large valves (typically 30 in. and larger) used in incompressible flow applications with the stem in a nonvertical orientation.

Hydrodynamic Torque (T_{hyd}). Hydrodynamic torque (T_{hyd}) is created by the hydrodynamic force imposed on the disc by the fluid flow, and is present only at disc positions other than fully closed. Hydrodynamic torque is calculated by

- Incompressible Flow

$$T_{hyd} = \frac{1}{12} \times C_t \times d_{disc}^3 \times \Delta P_v \ , \qquad (6)$$

where ΔP_v equals the smaller of actual pressure drop across valve and F_L^2 $(P_1 - 0.96\,P_v)$.

- Choked Compressible Flow

$$T_{hyd} = \frac{1}{12} \times C_t \times d_{disc}^3 \times Y^2 \times (x_T P_1) \ . \qquad (7)$$

In these equations, the nondimensional hydrodynamic torque coefficient, C_t, is a function of disc position, and depends upon disc geometry. Unlike the frictional torque components that act in a direction to oppose disc motion, hydrodynamic torque always acts in the same direction for a given disc angle. Hydrodynamic torque always tends to close the valve (self-closing), except in the following cases where hydrodynamic torque tends to open the valve:

- For offset disc designs with shaft downstream in choked compressible flow, hydrodynamic torque is self-opening throughout the stroke.

- For offset disc designs with shaft downstream in incompressible flow, the hydrodynamic torque becomes self-opening at disc positions near full open.

Dependence of Torque Coefficient (C_t) on Disc Design. The torque coefficient, which is a function of disc position (α), depends upon the disc design, disc aspect ratio, and, for nonsymmetric valves, the flow direction (shaft upstream or shaft downstream). To make torque predictions, the model requires the torque coefficient, C_t, and the related "consistent" set of flow coefficients: C_v or K_v, F_L (for incompressible flow), or x_T (for choked compressible flow) as a function of disc position for a given disc design. "Consistent" means that the torque coefficient, C_t, and the flow coefficient, C_v, are at the same disc position for a particular disc design. Using inconsistent values of C_t and C_v (for example, from different sources) can result in significant errors in the calculated hydrodynamic torque. The flow coefficients are used by a companion computer program (System Flow Model, not discussed in this paper) to calculate the value of flow rate and ΔP versus disc position, which are used by the butterfly model to calculate torque.

The butterfly valve model provides a consistent set of default torque coefficients and flow coefficients (Table 1) for the symmetric and single offset disc shapes commonly used in nuclear service. The model also accounts for the effect of disc thickness on the hydrodynamic torque and generates a consistent set of C_t and Kv values for different disc aspect ratios.

Effect of Upstream Elbow (C_{up}). The butterfly valve model includes a factor, C_{up}, to account for the effect of an upstream elbow on hydrodynamic torque. Velocity skew generated by the elbow tends to increase or decrease the hydrodynamic torque. The amount of torque created by this velocity skew depends on the proximity of the elbow to the valve and the orientation of the elbow relative to the axis of the disc stem (see Figure 4). The equations used to calculate the effect of the elbow on hydrodynamic torque are

$$T'_{hyd} = C_{up} \times T_{hyd} \ . \qquad (8)$$

C_{up} is the larger of 1.1 or the value calculated by

$$C_{up} = 1 + \left(C_{up,max} - 1\right)\left(\frac{\alpha}{90} - \frac{n}{8}\right) \qquad (9)$$

$$\frac{\alpha}{90} > \frac{n}{8} \ ;$$

$$C_{up} = 1 \text{ when } n \geq 8 \ .$$

This equation accounts for the fact that the effect of the elbow is most significant at close proximity and at high flow rates.

Table 1. Butterfly valve model coefficients for hydrodynamic torque calculations ($K_{v,min} = 0.53$).

Disk opening angle, α (degrees)	Valve resistance coefficient, K_v ($K_{v,min} = 0.53$)	Hydrodynamic torque coefficient, C_t			F_L^2	x_T
			Nonsymmetric disk			
		Symmetric disk	Shaft upstream	Shaft downstream		
0 (closed)	999999999	0.0000	0.0000	0.0000	0.7200	0.6000
1	115278757	0.0005	0.0006	0.0004	0.7200	0.6000
2	7186703	0.0009	0.0012	0.0007	0.7200	0.6000
3	1415391	0.0014	0.0018	0.0011	0.7200	0.6000
4	446391	0.0019	0.0024	0.0014	0.7200	0.6000
5	182171	0.0024	0.0030	0.0018	0.7200	0.6000
...	...	...	...	...	...	...
85	0.538	0.2600	0.3250	0.1950	0.4564	0.3179
86	0.531	0.2375	0.3074	0.1600	0.4551	0.3147
87	0.530	0.1969	0.2876	0.1200	0.4539	0.3114
88	0.530	0.1300	0.2648	0.0600	0.4528	0.3080
89	0.530	0.0700	0.2390	-0.0280	0.4520	0.3049
90 (open)	0.530	0.0000	0.2100	-0.2100	0.4515	0.3028

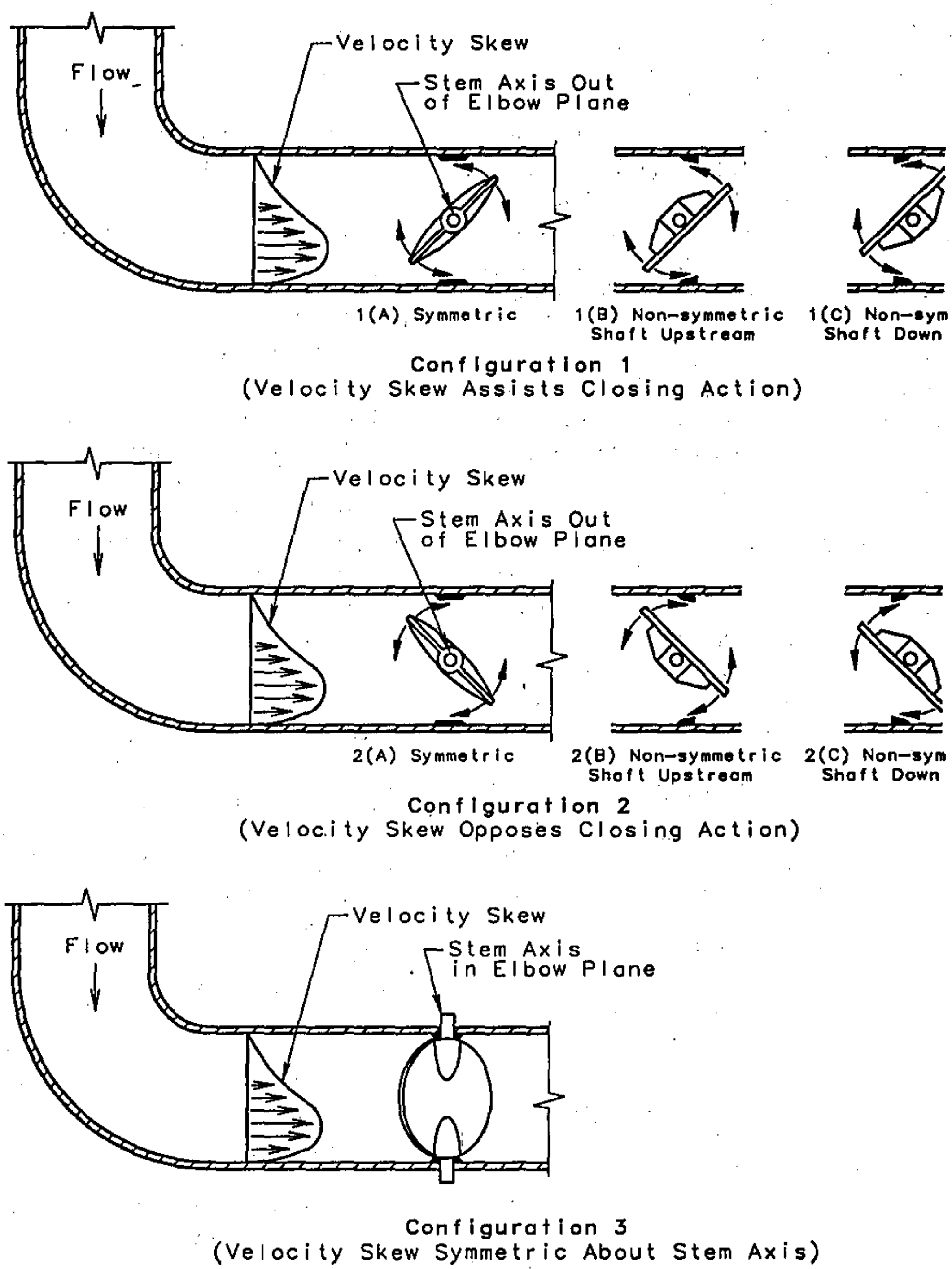


Figure 4. Upstream elbow orientation for symmetric and nonsymmetric valves.

$C_{up,max}$ depends on flow conditions (compressible or incompressible), orientation of elbow relative to axis of disc stem, and orientation of stem relative to flow direction (upstream or downstream).

MODEL VALIDATION

Test Data

The butterfly valve model was validated by comparing stem torque predicted by the model

with that obtained during flow testing of valves. Table 2 shows the matrix of test data used for model validation. The matrix includes several hundred strokes of test data on 15 valves from different flow-loop test factilities and in situ tests performed by utilities. Note that test Valves 6 through 11 in Table 2 were specifically designed to supplement the data from other sources by providing a systematic variation in disc shapes, disc aspects ratios, and the effect of upstream elbows in incompressible flow. Using this approach, the model could be validated against a range of disc shapes representative of those most prevalent in nuclear service. Figure 5 shows the disc geometries of test Valves 6 through 11.

For all tests in incompressible flow (water, both opening and closing stroke data were obtained. For compressible flow tests obtained from the USNRC/Idaho National Engineering Laboratory (INEL) containment isolation valve test program (Watkins et al., 1986), only closing stroke data were available.

The test matrix used for validation permitted the butterfly valve model to be validated for the following range of design variations and conditions:

- Symmetric and single offset disc designs
- Common disc shapes used in nuclear power plants
- Disc aspect ratios from 0.15 to 0.47
- Valve sizes from 6 to 42 in.
- Both flow directions for nonsymmetric disc valves
- Opening and closing strokes
- A range of flow rates
- A range of valve differential pressures
- Geometrically similar discs of 6 and 42 in. size
- Upstream elbows with different proximities and orientations
- Compressible and incompressible flow media
- Continuous flow and pipe rupture immediately downstream of the valve.

The validation matrix included comparisons of model predictions against test data for dynamic torque as well as for the seating/unseating torques. The following section presents a summary of the technical approach and comparison of dynamic torque results. This information demonstrates the key features of the model that overcome the shortcomings identified in the Introduction.

Technical Approach

The validation of the butterfly valve model included evaluation of the dynamic torque as well as the seating/unseating torque predictions. Comparisons of total dynamic torque predictions against the test data were made using one or more of the following four approaches, depending upon the details of the data available:

1. *Forward approach* in which the total dynamic torque from test results is compared with model predictions. The predictions are based on model torque coefficients, flow coefficients, and bearing coefficient of friction. Torque values are compared at the same value of valve flow resistance coefficient (K_V) for the model and the test valve, rather than at the same disc angle. This approach requires that the test data include ΔP and flow rate information (to determine K), as well as torque data, throughout the valve stroke.

Table 2. Validation matrix for butterfly valve performance prediction methodology.

									Test description		
										Flow	
Valve number	Valve description	Disk design[a]	Media	Data source	Flow direction (Shaft upstream or downstream)	ΔP (psi)	Max flow (gpm or lb/s)	Max V (ft/s)	Seating tests	Without elbow	With elbow
1	6-in. Henry Pratt (EPRI No. 54)	Sym	Water	Wyle	N/A	50, 100, 150	1,500	15	Yes	Yes	No
2	6-in. Henry Pratt (EPRI No. 55)	SO	Water	Wyle	Both	50, 100, 150	1,500	15	Yes	Yes	No
3	42-in. Posi-Seal	SO	Water	Duke	Both	14	46,000	11	Yes	Yes	No
4	18-in. Fisher (2-HV-4572)	Sym	Water	TU	N/A	130	8,000	11	Yes	Yes	No
5	24-in. Fisher (2-HV-4512 and 1-HV-4286)	SO	Water	TU	Upstream	85	15,000	12	Yes	No	Yes
6	6-in. model, t/d = 0.15	Sym	Water	Kalsi	N/A	90	2,700	30	No	Yes	No
7	6-in. model, t/d = 0.25	Sym	Water	Kalsi	N/A	90	2,700	30	No	Yes	Yes
8	6-in. model, t/d = 0.15	SO	Water	Kalsi	Both	90	2,700	30	No	Yes	No
9	6-in. model, t/d = .025	SO	Water	Kalsi	Both	90	2,700	30	No	Yes	Yes
10	6-in. model, t/d = 0.35	SO	Water	Kalsi	Both	90	2,700	30	No	Yes	No
11	6-in. model of 42-in. Posi-Seal	SO	Water	Kalsi	Both	90	2,700	30	No	Yes	No
12	24-in. Henry Pratt	SO	Nitrogen	INEL	Both	5 → 60	N/A	Choked	Yes	Yes	Yes
13	8-in. Henry Pratt	SO	Nitrogen	INEL	Both	5 → 60	N/A	Choked	Yes	Yes	Yes
14	8-in. Allis Chalmers	SO	Nitrogen	INEL	Both	5 → 60	N/A	Choked	Yes	Yes	Yes
15	10-in. Henry Pratt	SO	Water	APS	Downstream	120	3,815	16	Yes	No	Yes

a. Sym = symmetric disk; SO = single offset disk.

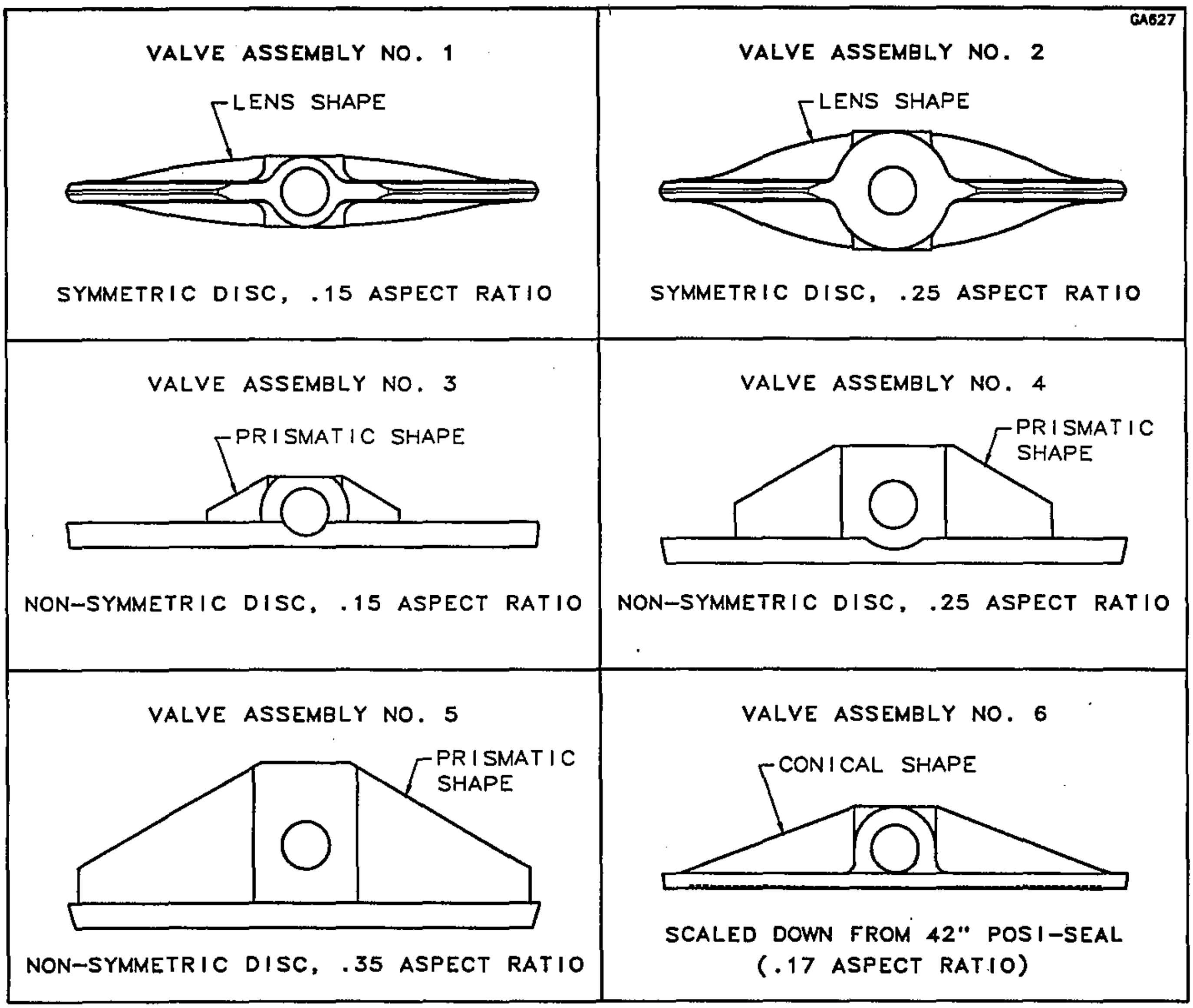


Figure 5. Disc shapes used in matrix of tests performed at Kalsi Engineering flow loop.

2. *Inverse approach* in which data from the opening and closing strokes are used to inversely calculate hydrodynamic and bearing friction torque components. These components are used to determine hydrodynamic torque coefficient and the bearing friction coefficient, which are then compared with the coefficients in the model. This approach requires data for ΔP, flow rate, and torque as a function of disc position throughout the disc stroke in both opening and closing directions. This approach was used because it provides a validation of individual components in the dynamic torque and it provides a more direct indication of the amount of conservatism in the key model features.

3. *Equivalent resistance approach* in which total dynamic torque predictions are made using an "equivalent resistance" model of the piping system determined from the maximum shut-off head in the fully closed position and maximum flow rate in the fully open position. This approach is somewhat approximate, but is used when ΔP and/or

flow rate data are not available throughout the disc stroke. The procedure is described in *The Application Guide for Motor-Operated Butterfly Valves in Nuclear Power Plants*.

4. *Normalization of upstream pressure approach* in which the test data were normalized to the nominal upstream absolute static pressure, P_1. This approach is used with compressible flow blowdown test results, for which the butterfly valve model dynamic torque predictions are calculated at the nominal value of P_1.

The packing torque component used in the predictions was based on measured data from opening and closing test strokes with no differential pressure. For symmetric disc valves the packing torque determined from the data includes the hub seal friction. The bearing friction coefficient used was selected using model guidelines.

TYPICAL EXAMPLES OF COMPARISONS

Detailed model validation included numerous comparisons with data for various test strokes, ΔPs, flow rates, and flow directions obtained from different test facilities. In all cases, the model appropriately bounded the test data. This section presents a few typical examples of comparisons and highlights of the results pertaining to the following aspects of the model:

- Total dynamic torque validation
- Hydrodynamic torque component validation
- Upstream elbow effect validation
- Scaling validation.

Total Dynamic Torque Validation

The total dynamic torque predicted by the model was validated by comparing the predicted torques with measured torques for eight valves (1 through 5 and 12 through 14). Both incompressible and compressible flow example comparisons are presented below.

Valve 1 (Incompressible Flow). Valve 1 is a 6-in., 150-lb butterfly valve manufactured by Henry Pratt. This valve has a symmetric disc of 0.31 aspect ratio. Tests were performed at three differential pressures: 50, 100, and 150 psi. Maximum flow velocity with the valve in the fully open position was 15 ft/s. Total dynamic torque predictions for this valve were performed using the forward approach for both the opening and closing directions, and the results are shown in Figures 6 and 7.

Figure 6 shows that the model bounds the test data for all three differential pressures in the opening stroke direction. The torque results are plotted against the butterfly valve model disc angle in degrees. The total dynamic torque comparisons are valid only outside the seating/unseating zone, which typically covers up to 10 degrees of disc opening. As described in the sign convention, the positive torque sign indicates that the actuator was required to supply torque to operate the valve.

Figure 7 shows closing stroke comparisons for three differential pressures. For the closing stroke, the model predicts the required total dynamic torque which must be supplied by the actuator. This is conservatively obtained by not taking credit for the self-closing hydrodynamic torque component. The model predictions for the required torque bounds the test data for all three ΔPs. A comparison of the model torque signature prediction, which does take into account the self-closing hydrodynamic torque, is also shown for the 150-psi ΔP case only. The torque signature prediction by the model closely matches the test results. It should be noted that torque signature prediction for the closing stroke is provided for

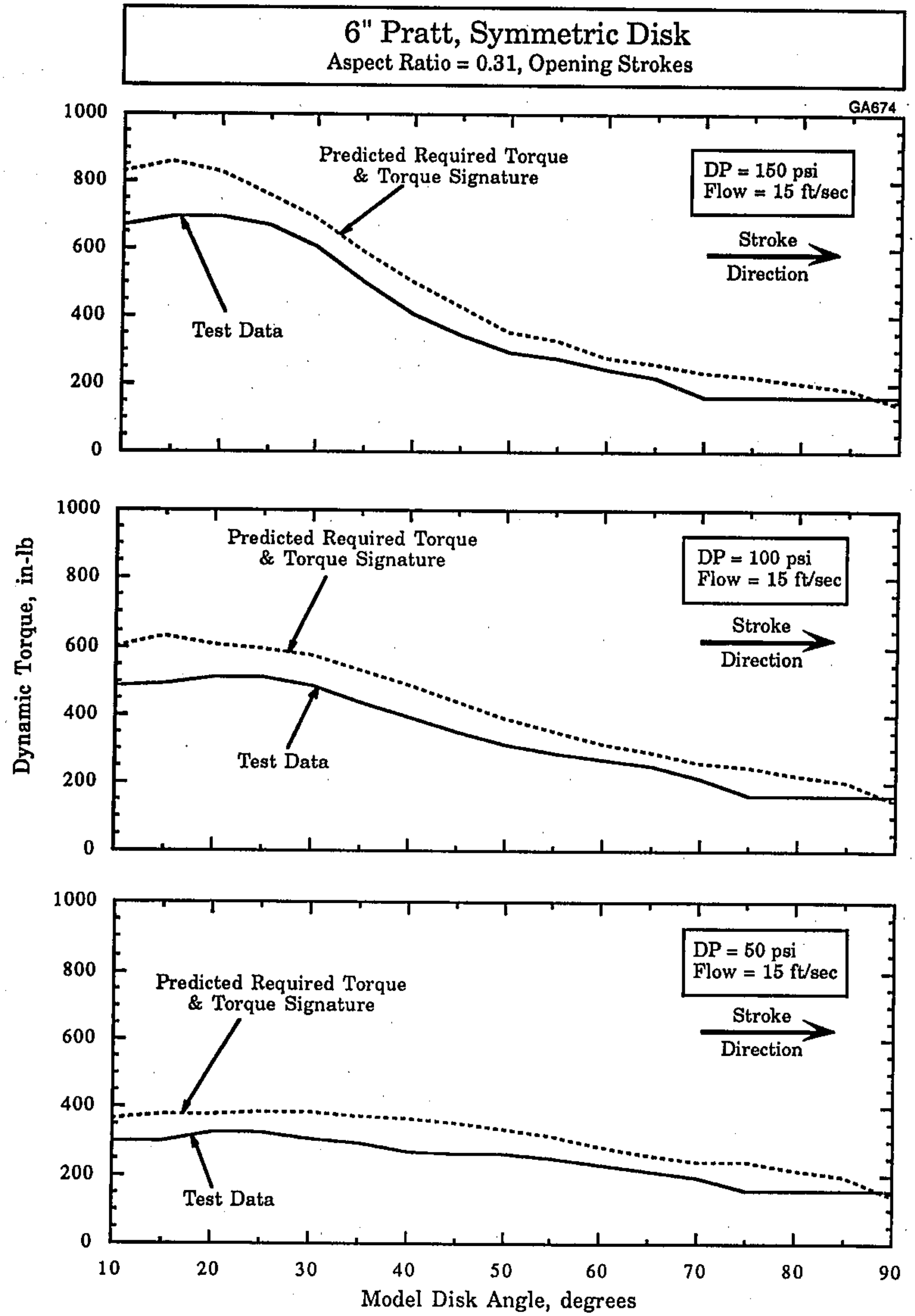


Figure 6. Model predictions and test results for opening strokes for Valve 1 (6-in. Pratt symmetric disc) in incompressible flow.

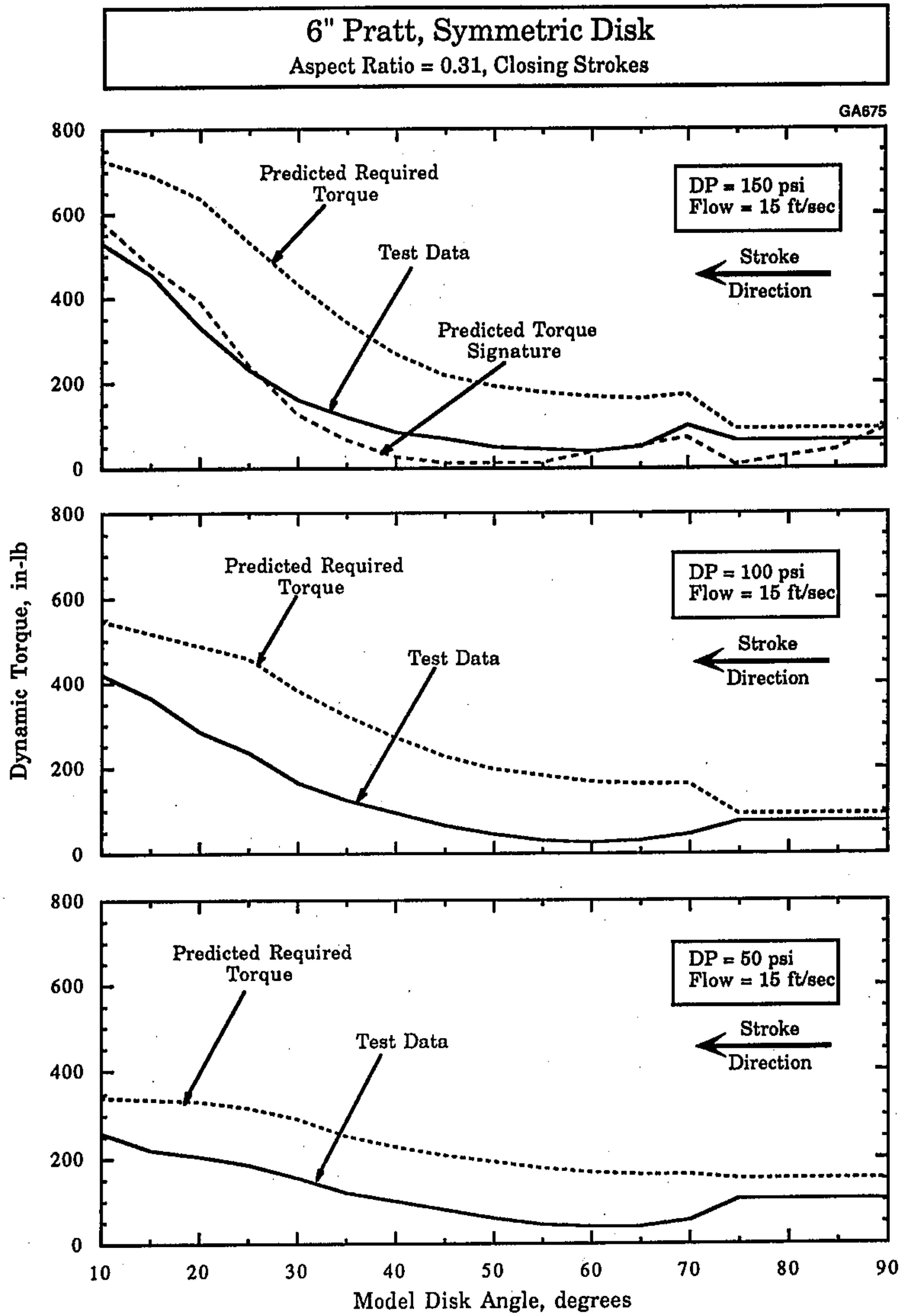


Figure 7. Model predictions and test results for closing strokes for Valve 1 (6-in. Pratt symmetric disc) in incompressible flow.

comparison and interpretation of test data only; it is not the required actuator torque.

As predicted by the model and found by testing, the required torque in the opening stroke direction is higher than the required torque in the closing stroke direction. This is the case with all valve applications that exhibit self-closing hydrodynamic torque.

Valve 2 (Incompressible Flow). Valve 2 is a 6-in., 150-lb butterfly valve manufactured by Henry Pratt. This valve has a nonsymmetric disc (single offset) of 0.47 aspect ratio. Tests were performed in both shaft-upstream and shaft-downstream orientations. Figures 8 and 9 show the results for the opening and closing strokes for the shaft-upstream flow direction only (the higher torque direction).

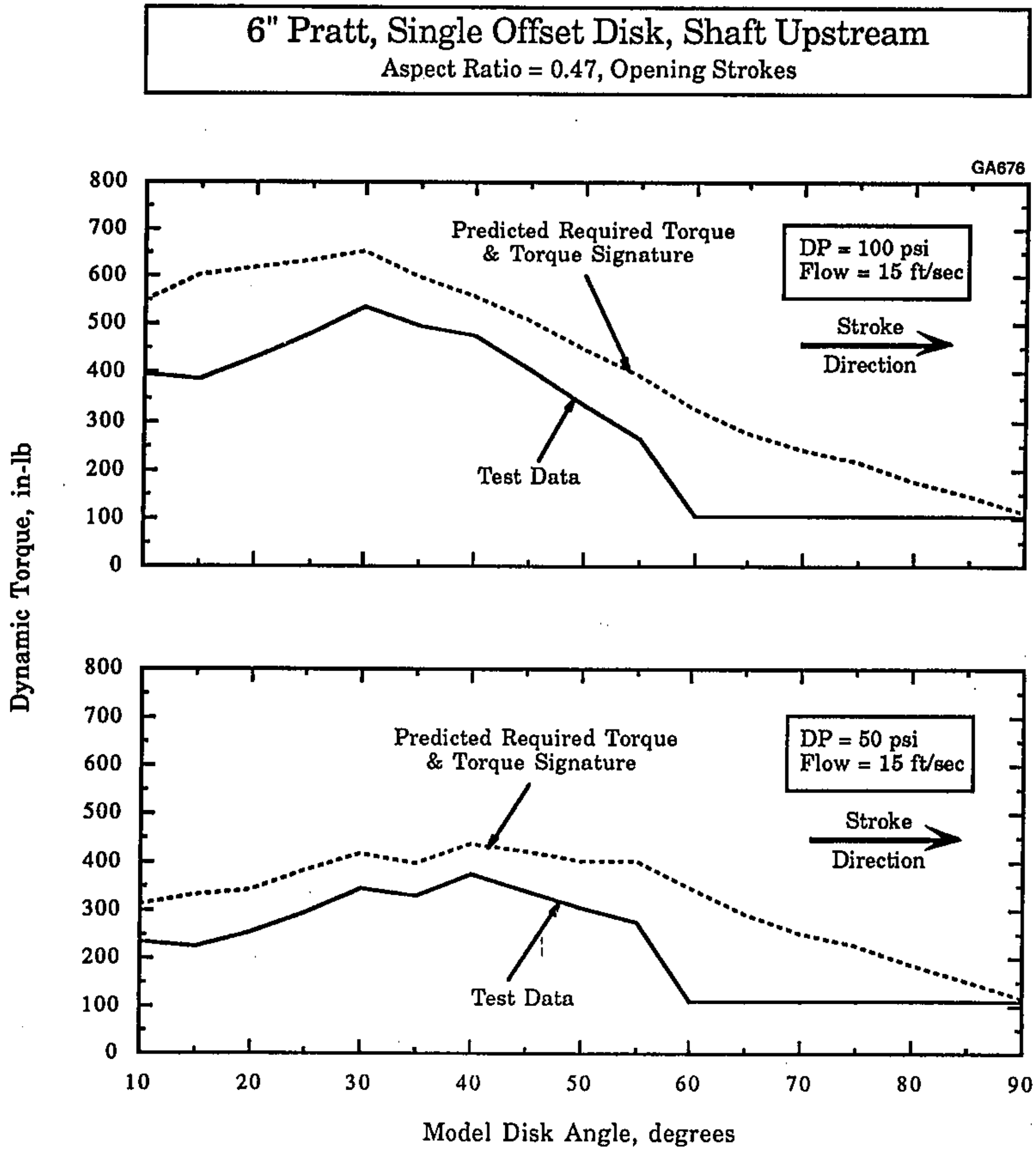


Figure 8. Model predictions and test results for shaft upstream, opening strokes, for Valve 2 (6-in. Pratt single offset disc) in incompressible flow.

The model predictions bound the test data for both the 50- and 100-psi ΔP test conditions in the opening and closing directions. Similar comparisons were found for the shaft-downstream condition.

Valve 3 (Incompressible Flow). Valve 3 is a 42-in., Class 150 butterfly valve manufactured by the POSI-SEAL Division of Fisher Controls. This valve has a nonsymmetric disc (single offset) of 0.17 aspect ratio and a conical backface shape similar to Valve Assembly 6 in Figure 5. Tests were performed at the Utah State University Water Research Laboratory, using ambient water as the flow medium, in both shaft-upstream and shaft-downstream orientations with respect to

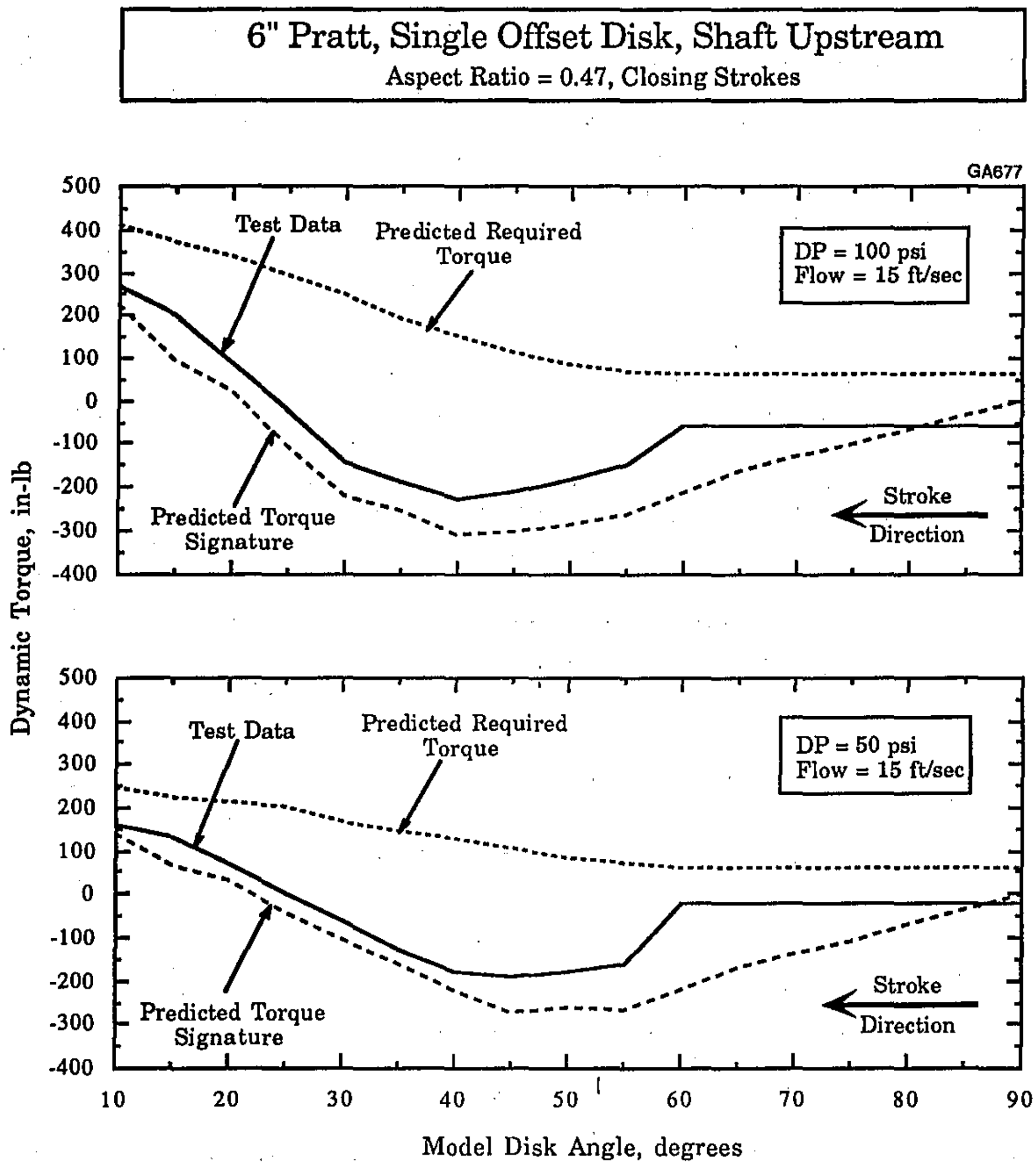


Figure 9. Model predictions and test results for shaft upstream, closing strokes, for Valve 2 (6-in. Pratt single offset disc) in incompressible flow.

flow. The maximum ΔP used in these tests was 14 psi, and maximum flow velocity with the valve in the fully open position was 12 ft/s (55,000 gpm). The torque predictions were performed using the equivalent resistance approach, and total dynamic torque comparisons for the shaft upstream and shaft downstream orientation are shown in Figures 10 and 11. The comparisons show that the model bounds the test results for both orientations with good margin.

It is noted that, for this low aspect ratio non-symmetric valve with a conical backface, the model is very conservative, especially in the shaft-upstream orientation. However, because the model is intended to cover the typical variations in disc shapes offered by other manufacturers, a reduction in the model conservatism is not justified.

Valve 12 (Compressible Flow). Validation against compressible flow data was performed using the normalization of upstream pressure approach.

Data for validating the butterfly valve model in compressible flow were obtained from NUREG/CR-4648 (Watkins et al., 1986). Supplemental information was provided by the INEL in the form of hard copy plots of torque signatures, pressures, temperatures, disc positions, and additional details pertaining to valve designs and the test matrix. The objective of the USNRC/INEL test program was to assess the ability of the valve to close under design-basis accident conditions in containment isolation applications; therefore, the digital data provided by the INEL (Watkins et al., 1986) are for closing strokes only. The supplemental information provided some data pertaining to the opening stroke; however, during the opening stroke, the upstream pressure decayed rapidly throughout the stroke. The opening stroke data were used to estimate bearing coefficients of friction for the test valves. The bearing coefficient was found to be approximately 0.15, which is bounded by the 0.25 value used in the model.

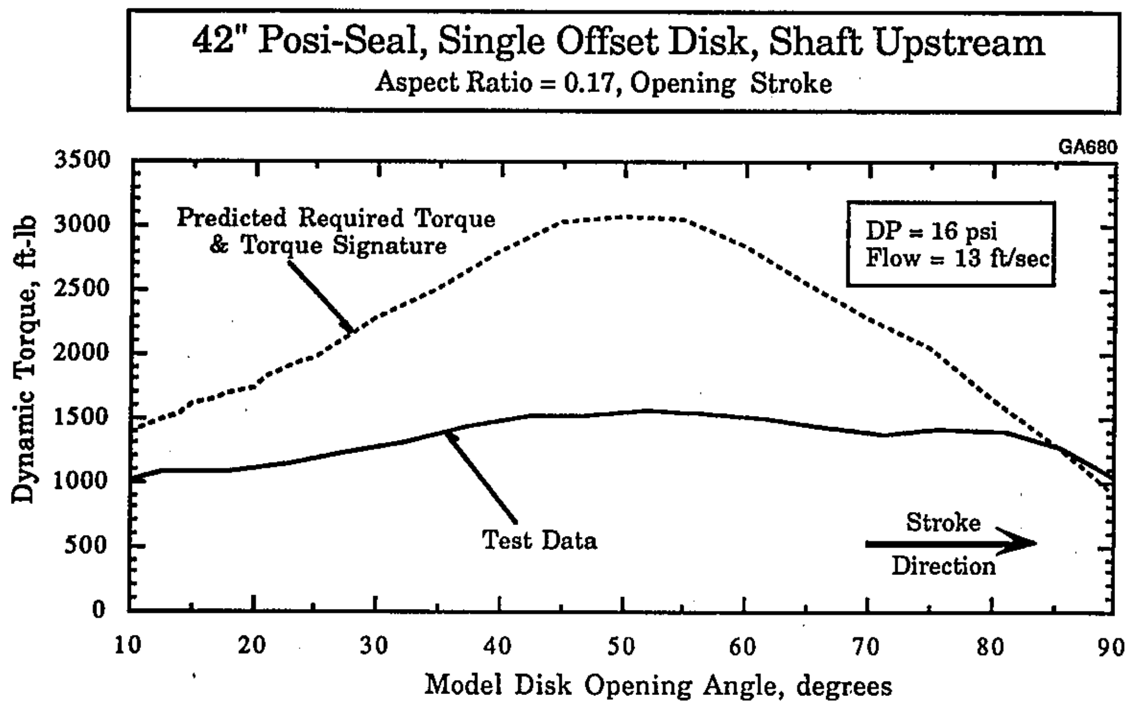


Figure 10. Model predictions and test results for shaft upstream, opening strokes, for Valve 3 (42-in. POSI-SEAL single offset disc) in incompressible flow.

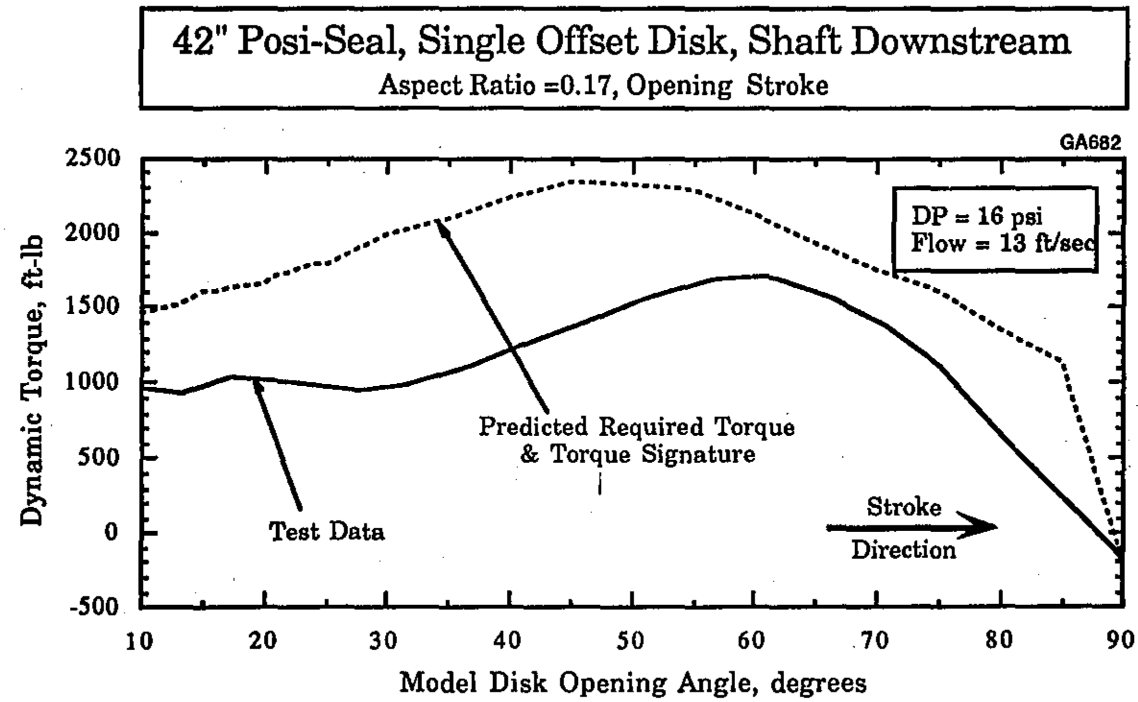


Figure 11. Model predictions and test results for shaft downstream, opening strokes, for Valve 3 (42-in. POSI-SEAL single offset disc) in incompressible flow.

The objective of the compressible flow model is to predict the peak total dynamic torque that occurs anywhere in the stroke. It assumes that the maximum upstream pressure held constant throughout the stroke. The small variations in the nominal upstream target pressure for each closing stroke in the INEL test matrix were accounted for by normalizing the data to a constant upstream pressure. Closing stroke tests were performed in both the shaft-upstream and shaft-downstream orientations. Tests were performed using nitrogen as the flow medium with maximum ΔPs maintained at nominal values ranging from 5 to 60 psi.

Note: The sign convention used for all compressible flow comparisons has been kept consistent with the sign convention adopted by the INEL, which is opposite to that of the model and that employed for the incompressible data comparisons. In compressible comparisons, a negative torque indicates that the actuator has to supply the torque to close the valve, and a positive torque indicates that the actuator is restraining the torque imposed on the shaft by the hydrodynamic forces.

Valve 12 is a 24-in. 150-lb butterfly valve manufactured by Henry Pratt. This valve has a nonsymmetric (single offset) disc of 0.26 aspect ratio and a shape similar to Valve Assembly 4 in Figure 5.

Figure 12 shows the total dynamic torque predictions by the model against the test results under 60 psi ΔP with the shaft in the upstream orientation. The total dynamic torque signature prediction envelope is defined by two values of bearing coefficient of friction: 0.25 and 0.0. The figure also shows the model prediction for the total required torque to actuate the valve. It can be seen that the maximum transmitted total dynamic torque predicted by the model bounds the test data. This maximum transmitted torque is recommended by the model to be used for evaluating the structural integrity of the MOV. The model prediction for the required actuation torque is only 700 ft-lb, which bounds the zero ft-lb found by actual testing.

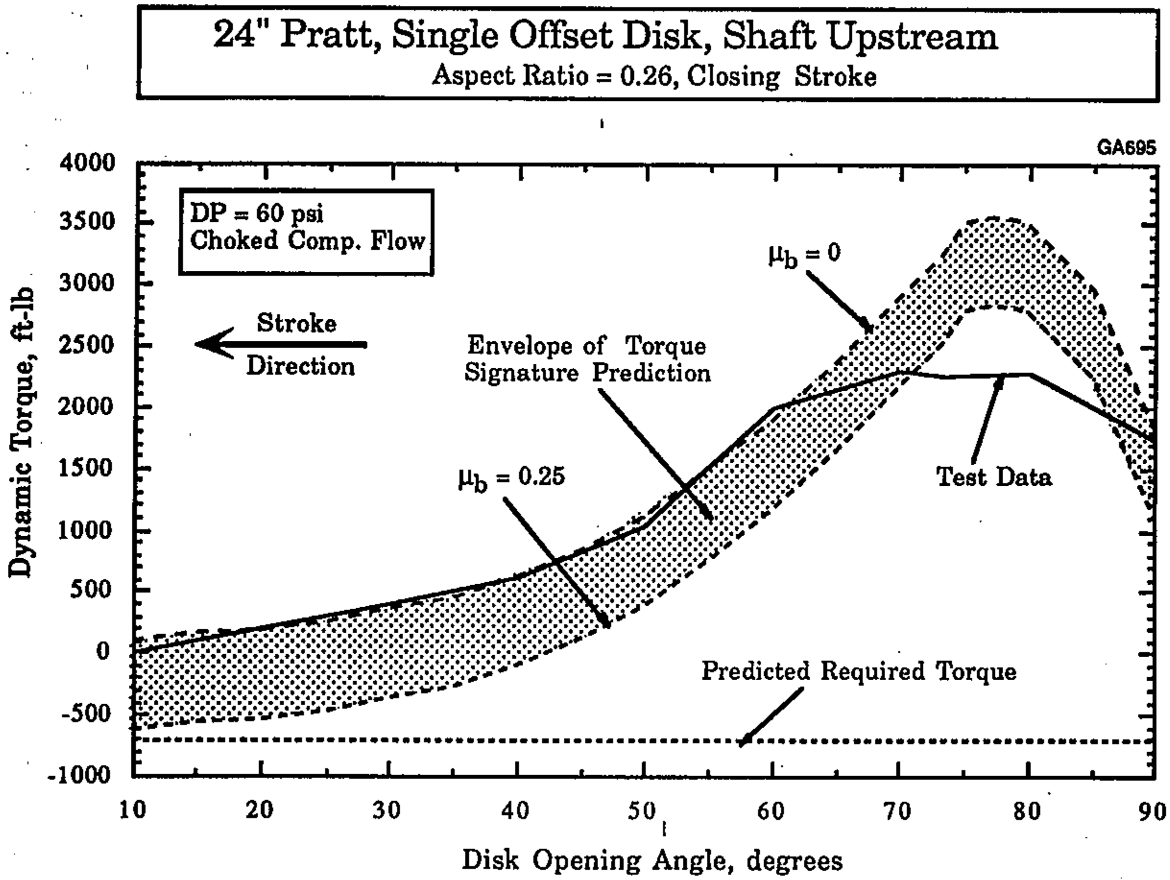


Figure 12. Model predictions and test results for shaft upstream, closing strokes, for Valve 12 (24-in. Pratt single offset disc) in compressible flow.

Figure 13 shows comparisons of the total dynamic torque predictions by the model against the test results at 60 psi in the shaft-downstream orientation. In this case the torque is negative, which means that the actuator has to supply the torque to close the valve. The peak total dynamic torque predicted by the model for 60-psi ΔP test condition bounds the test data. For this shaft orientation, this is also the maximum transmitted torque.

In the shaft-downstream orientation, Valve 12 was also tested under three additional nominal upstream pressure conditions: 15, 30, and 45 psig (30, 45, and 60 psia). The peak dynamic torque comparisons of the model predictions against the test results for all of the ΔP conditions are summarized in Figure 14. It can be seen that the model bounds the test results for all pressure conditions.

Hydrodynamic Torque Component Validation

The hydrodynamic torque coefficients used in the model were validated by comparing the model coefficients with coefficients extracted from test data for nine valves (1 through 3 and 6 through 11) using the inverse approach. Only incompressible flow data were used for this validation.

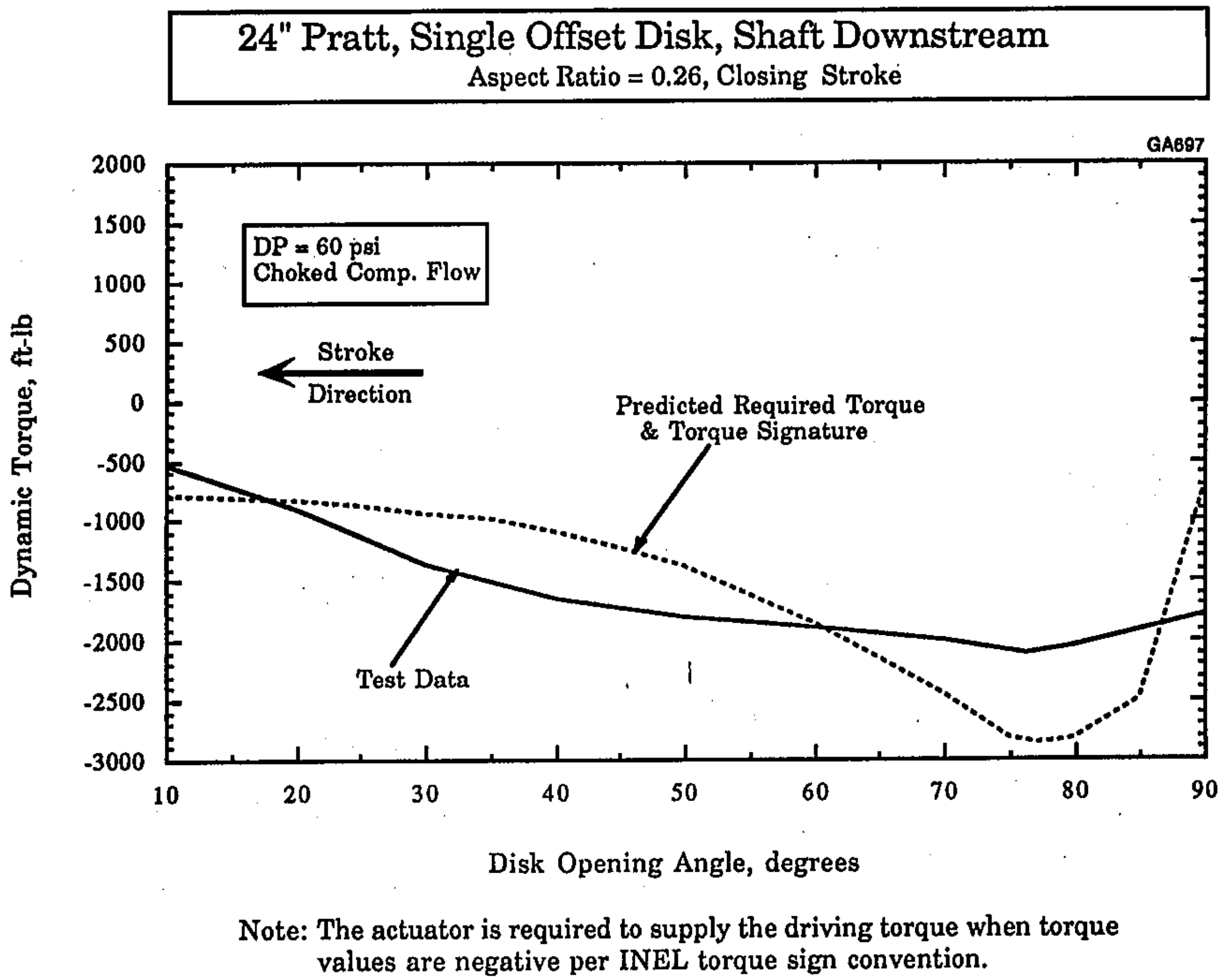
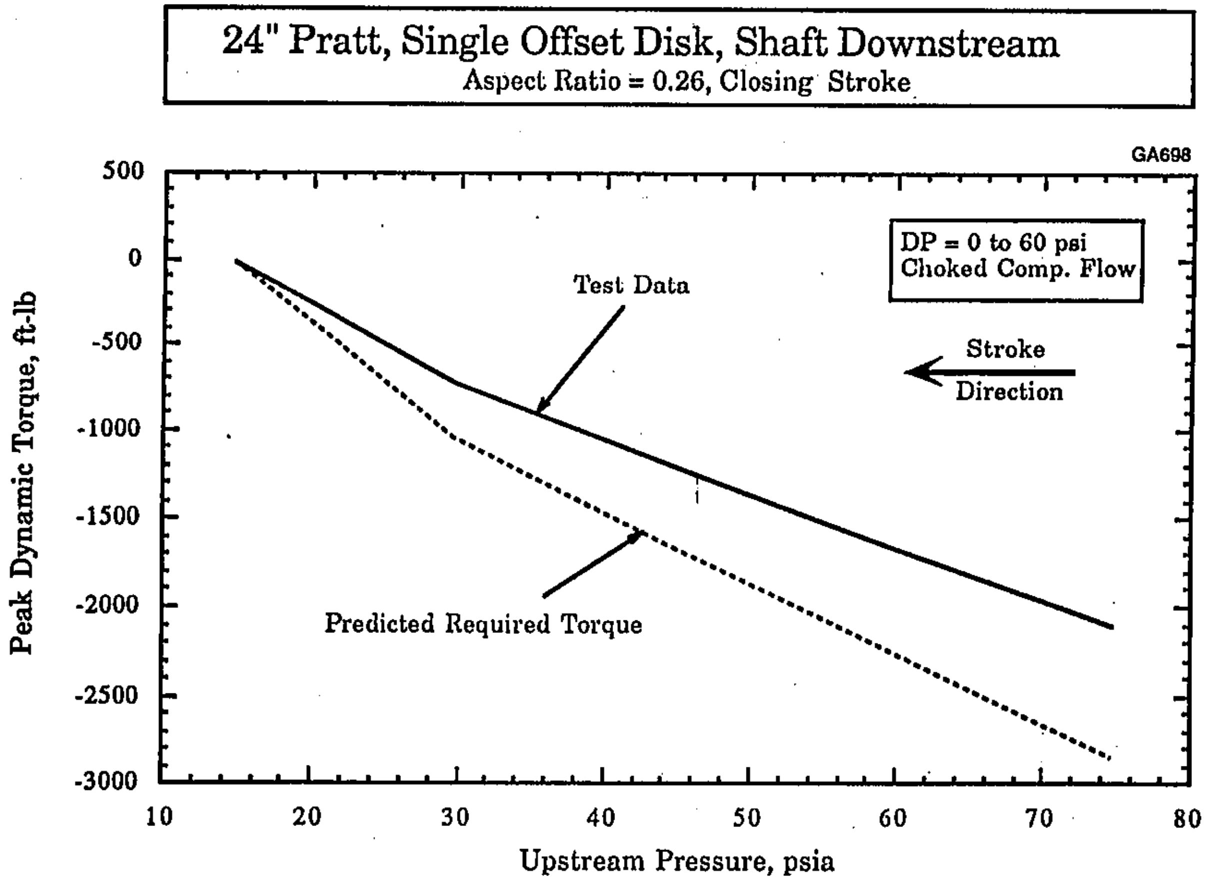


Figure 13. Model predictions and test results for shaft downstream, closing strokes, for Valve 12 (24-in. Pratt single offset disc) in compressible flow.

Figure 15 shows typical comparisons of the torque coefficients used in the model for the symmetric disc designs with aspect ratios of 0.15, 0.25, and 0.31 to those determined from test data. The comparisons show that the model bounds test data for all three aspect ratios. Similar comparisons between model predictions and test data were made for the single offset disc designs with aspect ratios of 0.15, 0.17, 0.25, 0.35, and 0.47. The model predictions for all aspect ratios tested were found to bound the test data in both the shaft-upstream and the shaft-downstream orientations.

Scaling Validation

The nondimensional hydrodynamic torque coefficients, C_t, were extracted from test data for the 42-in. POSI-SEAL valve (Valve 3) and its precisely scaled 6-in. model (Valve 11) using the inverse approach. The comparison of results for the shaft-upstream and shaft-downstream orientations for the 6-in. and the 42-in. valves is shown in Figures 16 and 17. The results show that the C_t for the 6-in. precisely scaled model is in excellent agreement with the C_t obtained for the 42-in.



Note: The actuator is required to supply the driving torque when torque values are negative per INEL torque sign convention.

Figure 14. Model predictions and test results for shaft downstream, closing strokes, for Valve 12 (24-in. Pratt single offset disc) in compressible flow under different ΔP conditions.

valve in both flow directions. From this comparison it can be concluded that C_t is independent of the valve size and that the hydrodynamic torque component is proportional to the cube of the disc diameter, as stated in Equations (6) and (7). Furthermore, the scaling validation results provide the basis for applying the butterfly valve model to all valve sizes found in nuclear power plant applications.

Upstream Elbow Effect Validation

The upstream elbow model was validated by performing comparisons against a comprehensive matrix of test data for incompressible and compressible flow.

In the incompressible flow test matrix, the valve type, elbow orientation, flow direction, and elbow distances were systematically varied to provide 27 unique upstream elbow test configurations. The matrix included symmetric and single offset disc designs of 0.25 aspect ratio tested in each of the elbow orientations shown in Figure 4 with three different spacings (0, 3, and 7 pipe diameters) between the elbow and the test valve. For each test configuration, the elbow effect was evaluated by opening and closing the valve under the following four flow conditions of maximum ΔP in the closed position and maximum flow velocity in the fully open position: 30, 60, and 90 psi with 15 ft/s, and 90 psi with 30 ft/s.

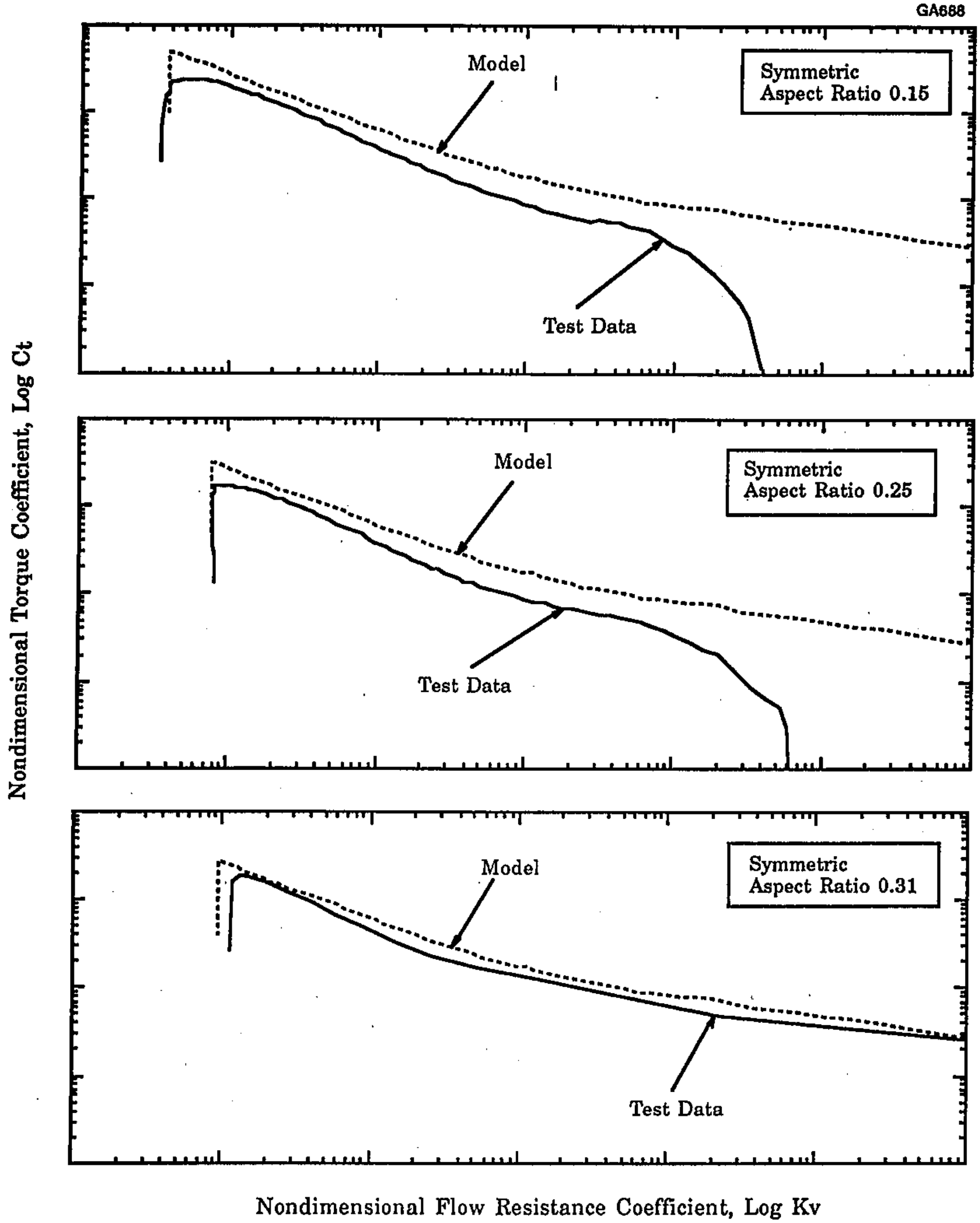


Figure 15. Comparison of model torque coefficients and flow coefficients against test data for symmetric disc valves of different disc aspect ratios.

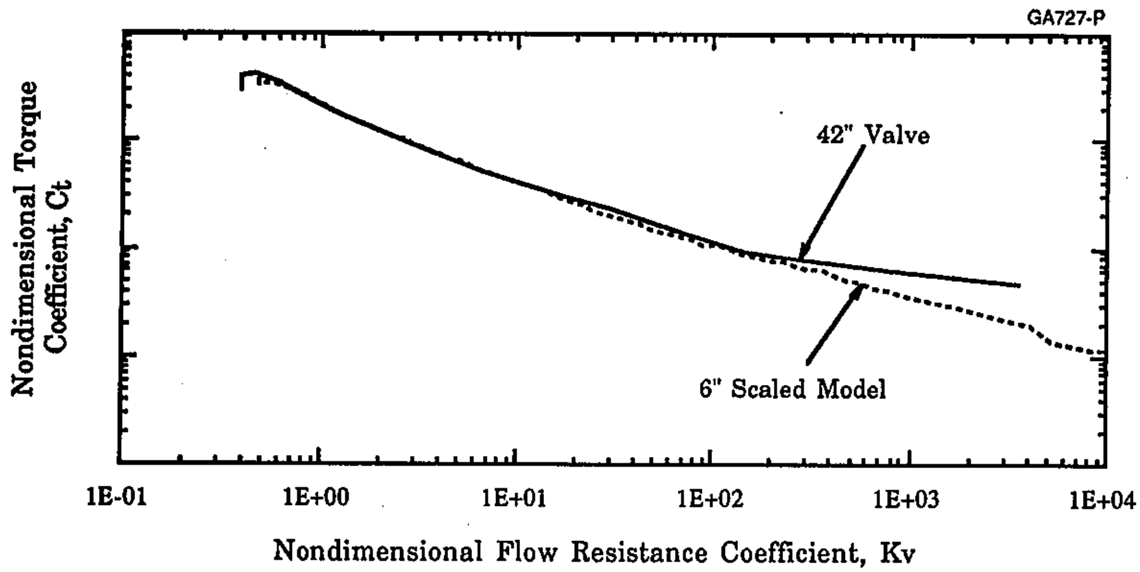


Figure 16. Comparison of test results from 6-in. scaled model test and 42-in. valve with shaft upstream orientation.

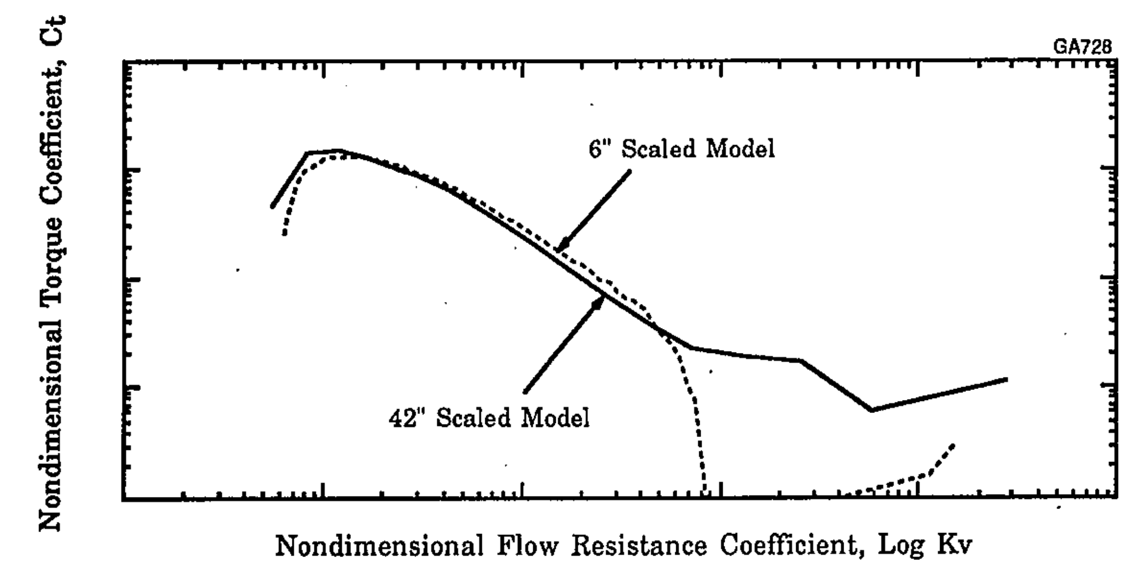


Figure 17. Comparison of test results from 6-in. scaled model test and 42-in. valve with shaft downstream orientation.

Figure 18 shows typical hydrodynamic torque results for the symmetric disc valve tested in elbow Configuration 1A identified in Figure 4 with distances of 0, 3, and 7 pipe diameters between the elbow and the valve. For comparison, the baseline test results (with 20 diameters of straight pipe upstream of the valve) are also shown in this figure. The peak torque in these tests occurred in the 55- to 60-degree range of disc position, and the increase in peak torque from the elbow effect was less than 10%. However, it is important to note that the actual location of the peak torque can vary, and the actual increase in torque caused by the presence of an elbow can be significantly higher for lower piping resistance systems that tend to shift the peak hydrodynamic torque towards the fully open disc position.

To account for this effect and extend the applicability of the elbow test results to other piping systems regardless of their piping resistance, the torque results were reduced to nondimensional upstream disturbance effect factors, C_{up}, defined by the torque ratio (T'_{hyd}/T_{hyd}) at each disc position. Typical torque ratio plots for the test data and model predictions for the symmetric disc valve in Elbow Configuration 1A are shown in Figure 19. It can be seen that the elbow has the most significant influence at the zero pipe diameter distance and full disc opening angle. The torque ratio plot also shows that the model bounds the test data for all disc openings except near the fully open and fully closed positions, where the baseline torque approaches zero magnitude and the ratios become very high (theoretically infinite). The larger values of torque ratios are meaningless from a practical torque requirement standpoint because the peak hydrodynamic torque occurs in the 20- to 75-degree range of disc opening angle.

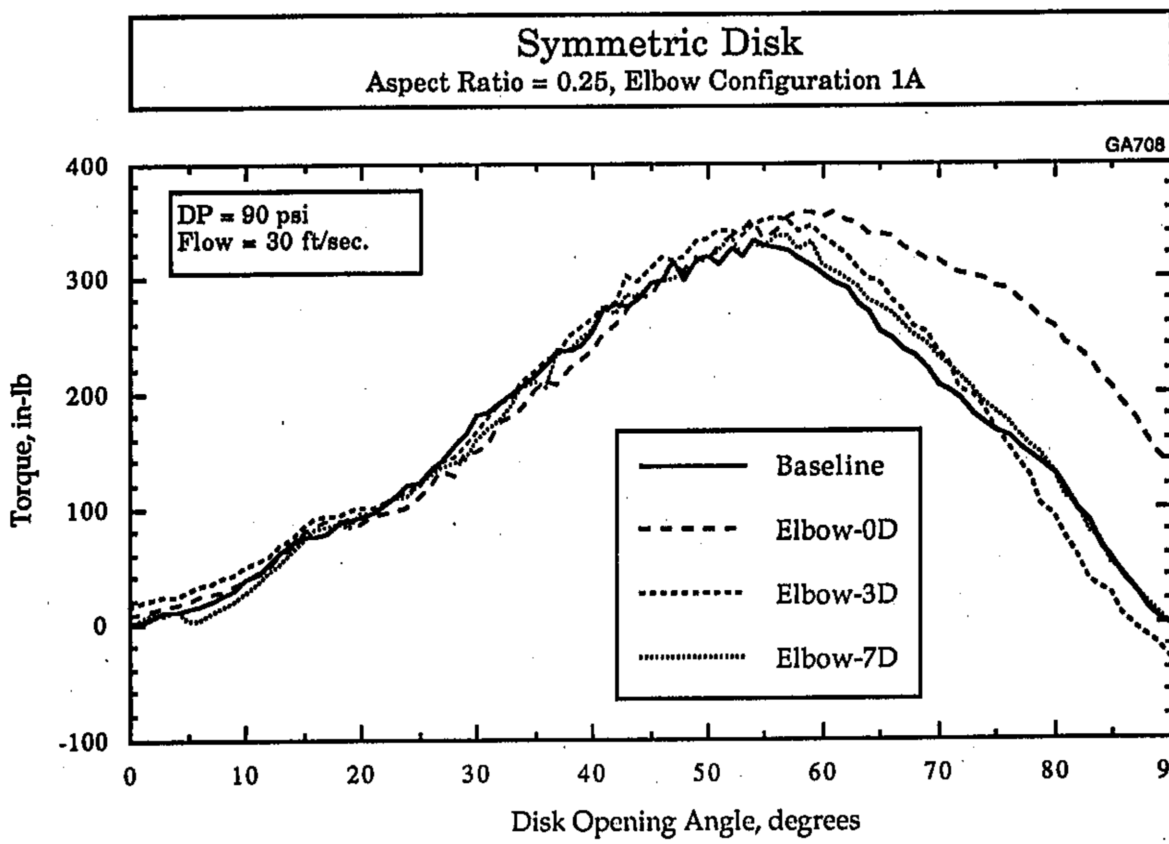


Figure 18. Hydrodynamic torque results for the symmetric disc valve tested in elbow Configuration 1A at 0, 3, and 7 pipe diameters upstream in incompressible flow.

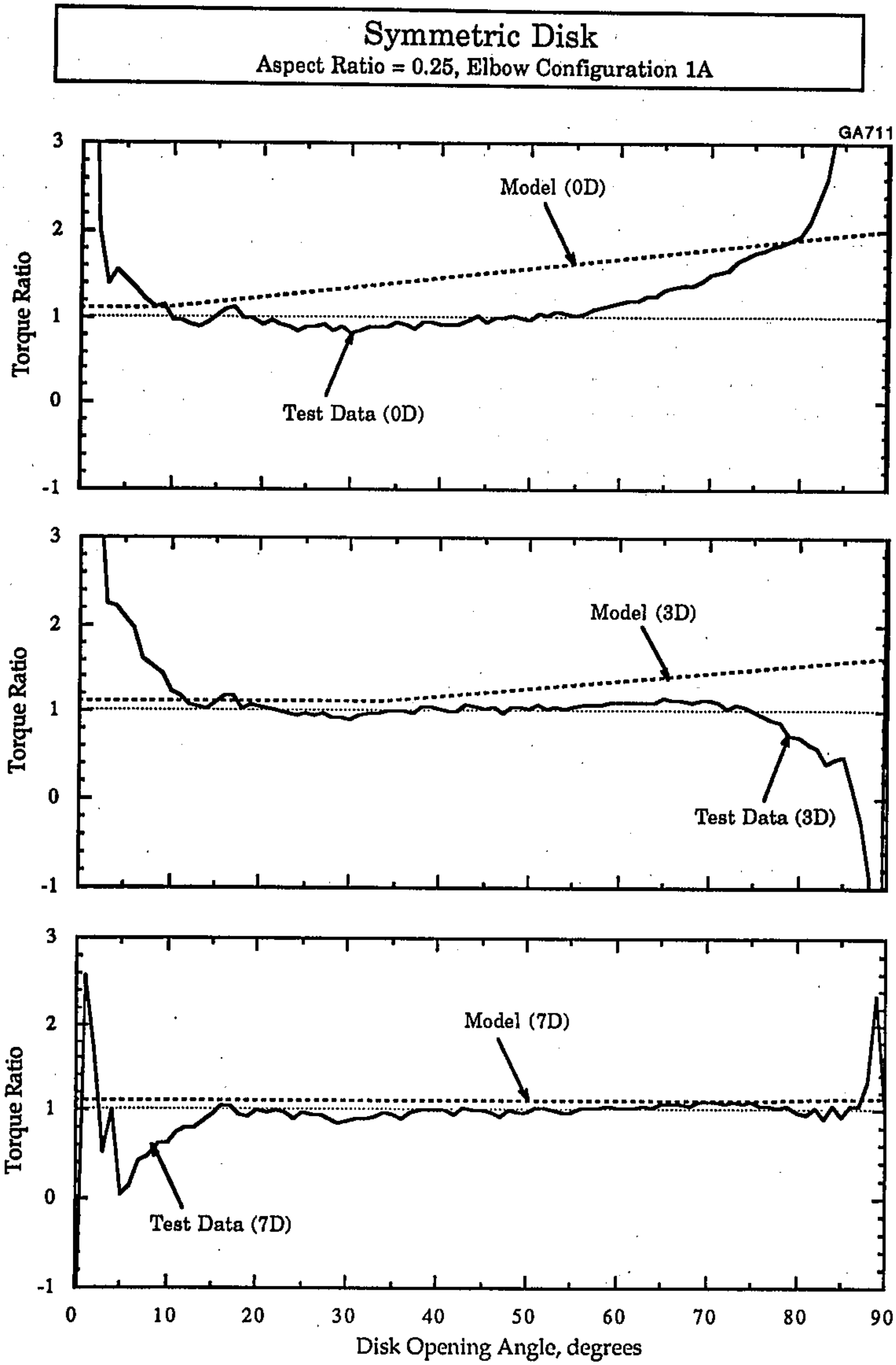


Figure 19. Torque ratio plots for Valve 7 tested in elbow Configuration 1A at 0, 3, and 7 pipe diameters upstream in incompressible flow.

Figures 20 and 21 show that the elbow effect results for the two other possible elbow Configurations (2A and 3 identified in Figure 4) for the symmetric disc valve in incompressible flow are also bounded by the model predictions. Elbow Configuration 2A creates a skew in the velocity profile that has a favorable effect (i.e., it tends to reduce the hydrodynamic torque imposed on the disc) at the zero diameter spacing; however, the model does not take credit for the reduction because this effect is very sensitive to the exact spacing between the elbow and valve and it diminishes very rapidly. Figure 21 shows the results for Elbow Configuration 3 in which the valve stem is in the plane of the elbow bend. In this configuration, the velocity skew caused by the elbow is symmetrical about the stem axis; therefore, the flow disturbance caused by the upstream elbow has the least effect on the hydrodynamic torque.

Typical comparisons from the compressible flow based on data from the USNRC/INEL tests on three different valves with Elbow Configuration 1B at a zero pipe diameter upstream of the valve are shown in Figure 22. The peak dynamic torque for compressible, choked flow occurs at disc openings of around 70 to 75 degrees. As shown in Figure 22, the compressible elbow effect model comfortably bounds the test results for these valves in the range of disc angles corresponding to the peak torque locations.

It should be noted that the $C_{up,max}$ factors provided in the model are different for different elbow configurations and incompressible or compressible flow to suitably bound the performance for different applications.

Observations Regarding Manufacturers' Data

Comparisons of actual test results against the predictions based on available manufacturers' data showed that the total dynamic torque predictions by some of the manufacturers were unconservative, whereas other manufacturers' predictions were extremely conservative. In general, it was found that some manufacturers' predictions do not take into account the variations in disc aspect ratios and disc shapes. From a survey of manufacturers' data, it was found that some manufacturers have typically performed a limited number of tests on one or two valves and extended the results to valves of other sizes and pressure ratings that have discs of different aspect ratios and shapes. The EPRI butterfly valve model overcomes this deficiency and provides the appropriate flow and torque coefficients for a given disc shape and aspect ratio.

CONCLUSIONS

The research program described in this paper has led to the development of an improved and validated butterfly valve torque prediction methodology. For dynamic torque predictions, the model provides the appropriate torque and flow coefficients based on the disc design (symmetric or single offset), disc aspect ratio, flow direction, and fluid media (compressible or incompressible). Model predictions for the total dynamic torque have been compared with test data for a number of valves ranging in size from 6 to 42 in. having symmetric and single offset disc designs and aspect ratios ranging from 0.15 to 0.47 that were tested in incompressible and compressible flow applications. The model was found to bound the test results for the required actuation torque and the maximum transmitted torque in all cases. The elbow model provides the appropriate upstream elbow effect factor, C_{up}, which depends upon elbow orientation, elbow distance, disc opening angle, fluid media (incompressible or compressible), and flow direction. The elbow model predictions were compared with test data from a large matrix of tests in incompressible and compressible flow media. The elbow model predictions were found to bound test results in all cases.

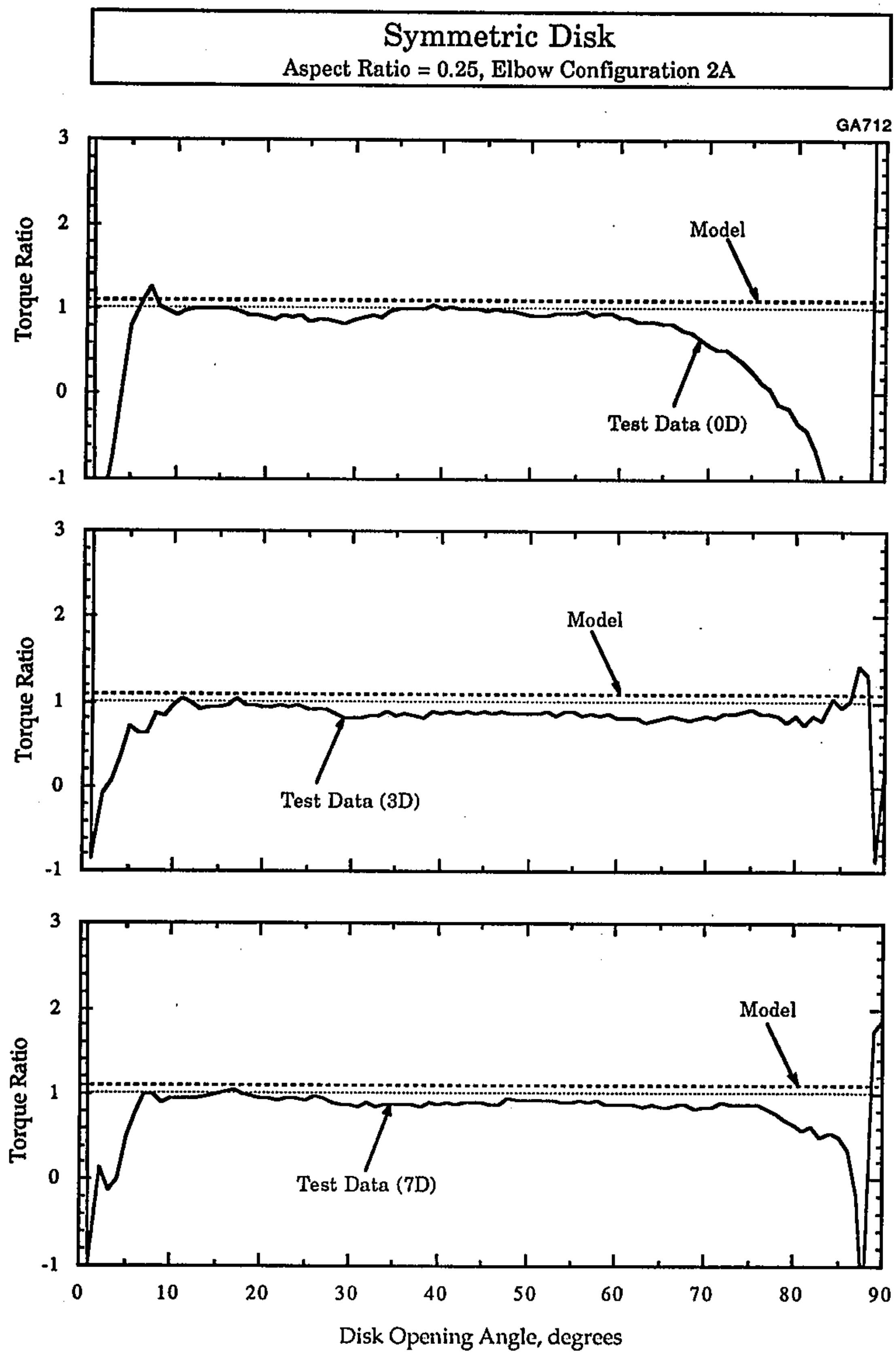


Figure 20. Torque ratio plots for Valve 7 tested in elbow Configuration 2A at 0, 3, and 7 pipe diameters upstream in incompressible flow.

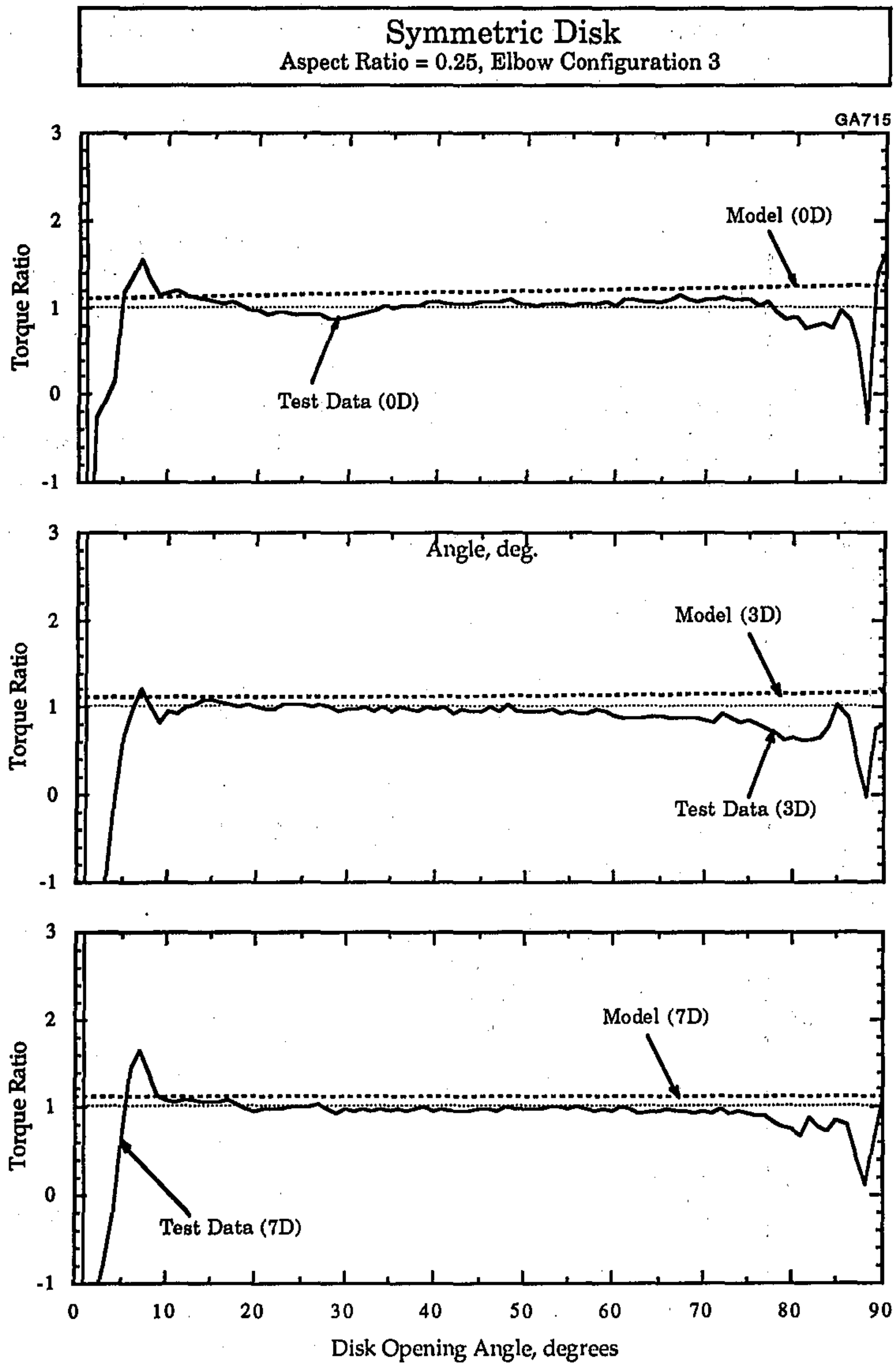


Figure 21. Torque ratio plots for Valve 7 tested in elbow Configuration 3 at 0, 3, and 7 pipe diameters upstream in incompressible flow.

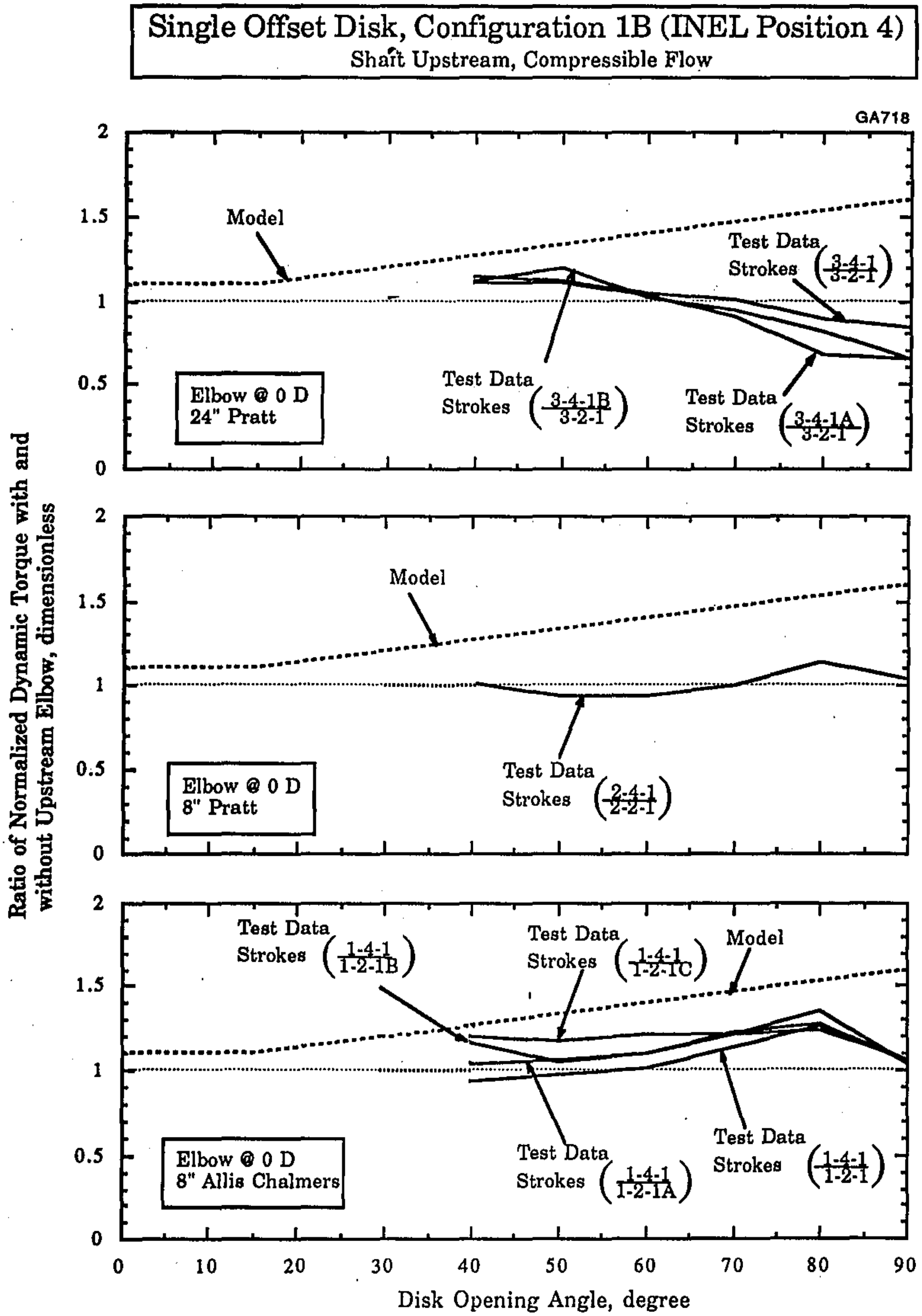


Figure 22. Torque ratio plots for Valves 12, 13, and 14 tested in elbow Configuration 1B at zero pipe diameters upstream in compressible flow.

Evaluation of test data against previously reported generalized disc model (e.g., the *Application Guide for Butterfly Valves*) shows that the earlier model did not bound some test results, especially for higher disc aspect ratios. Furthermore, the earlier elbow model did not bound the test results in some elbow configurations and was overly conservative for others. The current research eliminates these shortcomings of the earlier generalized disc model and limitations of some manufacturers' torque prediction methods and provides a validated model for butterfly valve torque prediction.

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Watkins, J. C., R. Steele, R. C. Hill, and K. G. DeWall, 1986, *A Study of Typical Nuclear Containment Purge Valves in an Accident Environment*, NUREG/CR-4648, U.S. Nuclear Regulatory Commission.

NOMENCLATURE

A	Pressure-independent seat torque coefficient, in.-lb/in.^{2}
C_t	Hydrodynamic torque coefficient, dimensionless
C_{up}	Upstream disturbance factor, dimensionless
$C_{up,max}$	Maximum value of upstream disturbance factor, dimensionless
$C_{v,max}$	Valve flow coefficient in the fully open position, gpm/$\sqrt{psi}$
d_{disc}	disc diameter, in.
d_i	Valve inlet diameter, in.
d_s	Stem diameter, in.
F_L	Valve liquid pressure recovery factor, dimensionless
$K_{v,min}$	Valve head loss coefficient in the fully open position, dimensionless
n	Length of straight pipe between an upstream elbow and valve inlet expressed in pipe diameters, dimensionless
ΔP_{choke}	Pressure differential across valve (valve/fitting assembly) at onset of choking, psi
ΔP_{max}	Maximum pressure differential at shutoff, psi
ΔP_v	Pressure differential across the valve or valve-fitting assembly, psi
R_a	Disc aspect ratio (disc thickness/disc diameter), dimensionless
T_{b_i}	Bearing torque, ft-lb
T_h	Hydrostatic torque, ft-lb
T_{hub}	Hub seal friction torque, ft-lb. This component is present only in rubber-seated symmetric disc butterfly valves
T_{hyd}	Hydrodynamic torque (without an upstream flow disturbance), ft-lb
T'_{hyd}	Hydrodynamic torque (with an upstream flow disturbance), ft-lb
T_p	Packing torque, ft-lb
T_s	Seat torque, ft-lb
T_{TD}	Total dynamic torque at disc angle, α, ft-lb
$T_{TD,max}$	Maximum total dynamic torque, ft-lb

T_{TS}	Total seating/unseating torque, ft-lb
T_{WL}	Maximum transmitted torque for weak link analysis, ft-lb
x_T	Valve pressure drop ratio factor, dimensionless
α	Disc opening angle, degree
μ_b	Bearing coefficient of friction, dimensionless
ϕ	Stem angle measured from vertical, degree
ρ	Fluid density at valve inlet conditions, lb/ft^3.

Globe Valve Performance Prediction

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ABSTRACT

The Electric Power Research Institute is carrying out a program to improve the performance prediction methods for motor-operated valves. As part of this program, a comprehensive globe valve model was developed to calculate valve stem thrust. This model includes packing load, stem rejection load, disc differential pressure (DP) load, disc weight, disc-to-body friction, and stem torque reaction friction. The model recognizes two possible diameters that may be used in calculating the area for the disc DP load—the disc seat diameter or the disc guide diameter. Guidelines have been developed to determine which diameter is appropriate for a specific valve, based on the valve configuration. The model has been assessed against data from five globe valves, ranging in size from 2 to 10 in. The model predictions compare favorably with the test data for incompressible flow. For flashing water blowdown, the data are not predicted by the model, and further work in this area is required.

BACKGROUND

Globe valves are used in numerous fluid systems in power plants. While detailed designs vary, globe valves contain a valve disc that is stroked away from and back toward a circular seat. Typically, the disc is constrained to move along the stem axis by guiding surfaces within the valve body. This configuration is referred to as a "body-guided" design.

Prediction of stem thrust for body-guided globe valves in nuclear power plants has received increased attention recently. In response to U.S. Nuclear Regulatory Commission (USNRC) Generic Letter 89-10, programs have been implemented at each plant to test and evaluate motor-operated valves. The work described in this paper was carried out to provide a method for evaluating globe valve performance. The effort includes documenting a standard stem thrust calculation approach and comparing it with data from several globe valves tested under controlled, highly instrumented conditions.

OBJECTIVE AND SCOPE

The objective of this work is to develop a model for calculating the stem thrust needed to open and close body-guided globe valves. The model was developed for body-guided globe valves with the following designs:

- T-pattern or Y-pattern body types
- Rising or rising/rotating stems
- Balanced or unbalanced discs.

The model is applicable to the complete range of stem positions for opening and closing strokes. It encompasses flow from above the seat (overseat flow) and flow from below the seat (underseat flow).

NOMENCLATURE

A_D	disc seat area, in.^{2}
A_{DG}	disc guide area, in.^{2}
$A_{\Delta P}$	disc differential pressure area, in.^{2}

A_S	stem area, in.^{2}
D_G	disc guide diameter, in.
D_S	disc seat maximum diameter, in.
ΔP_D	fluid differential pressure across disc, psi
θ	stem pitch angle, degrees
$f_{\Delta P}$	balanced disc imbalance force factor
F_{DF}	disc-to-body friction force, lb_f
F_{DP}	disc differential pressure force, lb_f
F_G	disc and stem gravity load, lb_f
F_{PF}	packing friction force, lb_f
F_R	resultant stem thrust, lb_f
F_{SR}	stem rejection load, lb_f
FS	stem factor (ratio of stem torque to thrust), in.
F_{TR}	torque reaction friction force, lb_f
G	bearing load factor, dimensionless
H	lower disc guide height, in.
μ	dynamic coefficient of friction, dimensionless
μ_T	torque reaction surface coefficient of friction, dimensionless
P_{AD}	fluid pressure above disc, psi
ϕ	stem roll angle, degrees
r_t	torque reaction moment arm, in.
T	stem torque, in./lb_f
W_D	effective weight of disc, lb_f
W_S	effective weight of stem, lb_f
X_S	absolute stem position, in.

MODEL DESCRIPTION

The stem thrust required to stroke a body-guided globe valve is calculated from the force components acting along the valve stem axis. These stem thrust components are shown on Figure 1 and described below. The components are summarized in Table 1.

Disc Differential Pressure Force (F_{DP})

This force is created by the differential pressure (DP) acting across the valve disc. For unbalanced disc valves, this load is typically the major force contributor. For balanced disc valves, the DP across the valve disc is very small. Therefore, the model neglects this force for balanced disc valves, but includes a term ($f_{\Delta p}$, discussed below) in the disc-to-body friction force to cover these small DP force imbalances that exist in balanced disc valves.

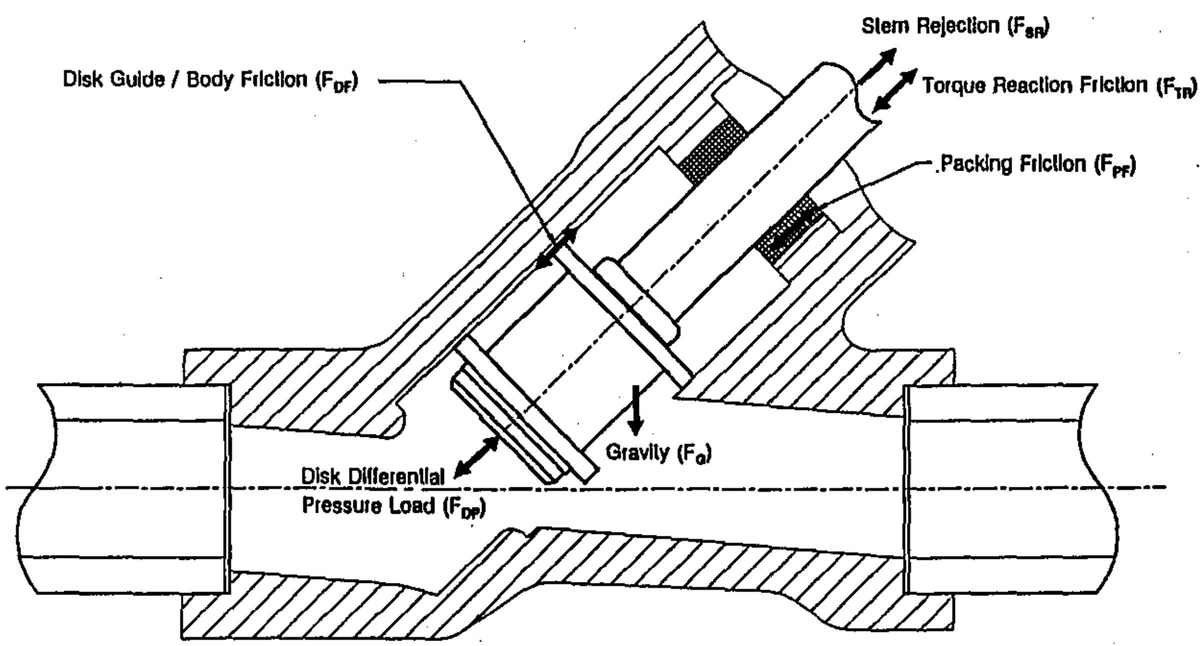


Figure 1. Globe valve stem force components.

Table 1. Globe valve stem thrust components.

Stem thrust components	Equation	Applicability	Sign	Comments
Disc differential pressure (F_{DP})	$(\Delta P_D)\ (A_{\Delta P})$	Neglected for balanced discs	Positive—overseat flow Negative—underseat flow	$A_{\Delta P} = A_D$, seat area for "seat-based" valves. $A_{\Delta P} = A_{DG}$, guide area for "guide-based" valves.
Disc guide-to-body friction (F_{DF})	$(1.1\ \mu\ G + f_{\Delta P})\ \Delta P_D\ A_{DG}$	Neglected for unbalanced discs	Positive—opening stroke Negative—closing stroke	$\mu = 0.6$ for all material combinations. $f_{\Delta P} = 0.1$ for all balanced discs.
Stem rejection (F_{SR})	$(P_{AD})\ (A_S)$	All valves	Negative	P_{AD} is downstream pressure for unbalanced discs with underseat flow or balanced discs with overseat flow. P_{AD} is upstream pressure for unbalanced discs with overseat flow or balanced discs with underseat flow.
Gravity load (F_G)	$(W_D + W_S)\left[\dfrac{\cos\theta\ \cos\phi}{\sqrt{1 - \sin^2\theta\ \sin^2\phi}}\right]$	All valves	Positive—stem pointing "up" Negative—stem pointing "down"	Gravity force can be neglected when differential pressure (psi) > 70 x valve size (in.).
Packing friction (F_{PF})	Constant (user-specified)	Neglected for rising/rotating stems	Positive—opening stroke Negative—closing stroke	User input.
Torque reaction factor (TRF)	$1 - \dfrac{\mu_T\ (FS)}{r_t}$	Applicable for rising stems	Increases thrust magnitude	TRF = 1.0 for rising/rotating stems.

The disc DP force is equal to the product of DP and the appropriate area. For unbalanced disc valves, the area is based on either disc seat diameter or disc guide diameter depending on valve design (see Figure 2). A guideline has been developed describing how to determine the appropriate area from a valve configuration drawing. Seat-based behavior occurs in valves where an open flow area is provided above the seat orifice allowing relatively unconstrained flow through the outlet port of the valve when the valve is nearly closed. However, in some valve designs, the disc guide eclipses a major portion of the outlet port near the closed end of the valve stroke. This may restrict flow through the valve exit port sufficiently to cause the full DP to act across the disc guide area. This type of valve is designated guide-based. An illustration of the two valve types is shown on Figure 3.

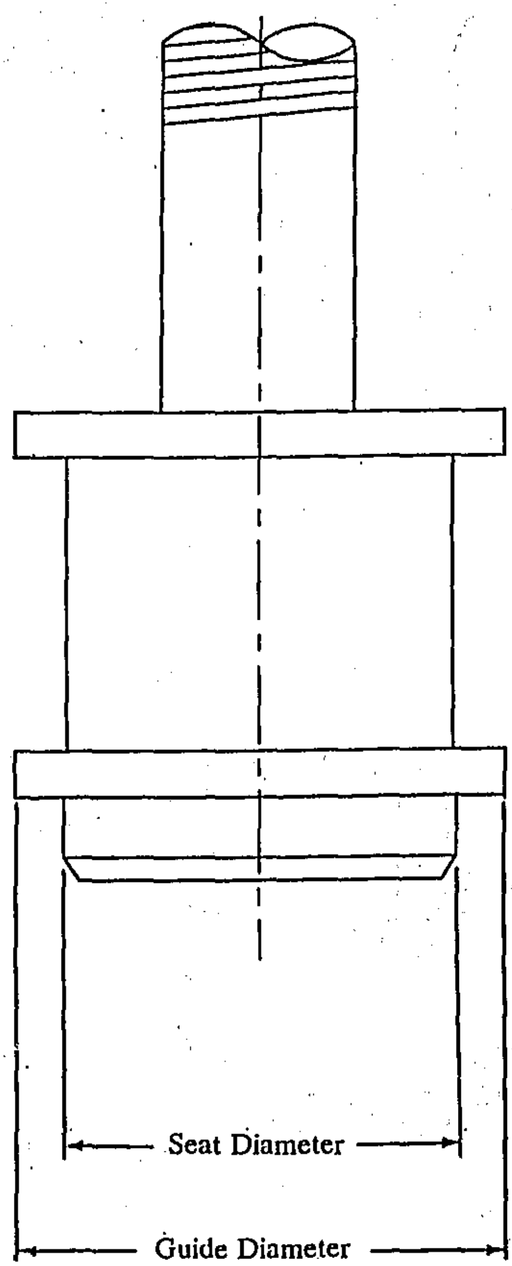
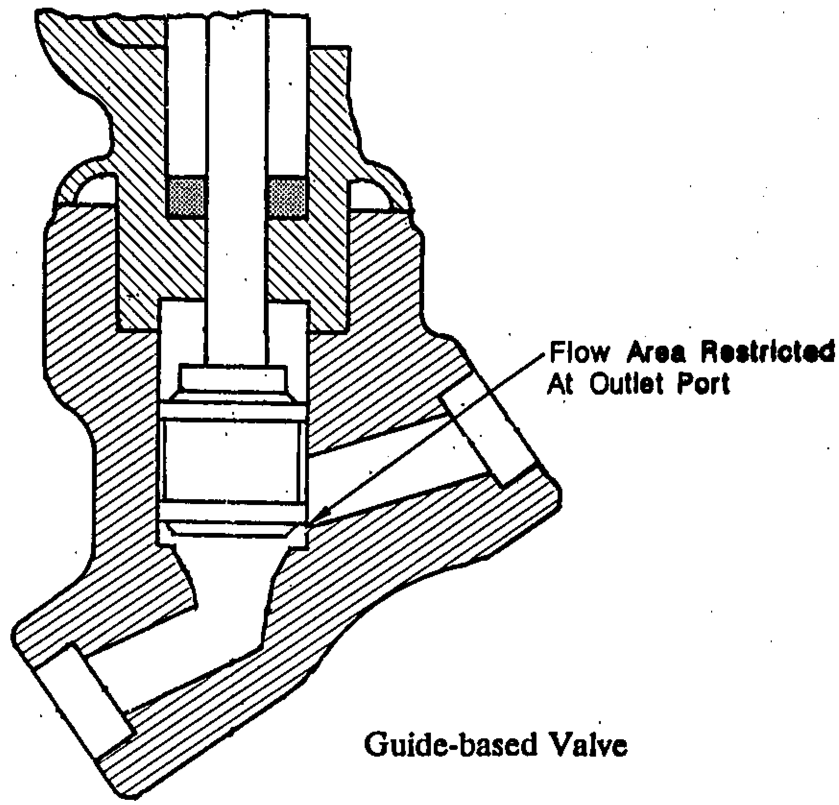
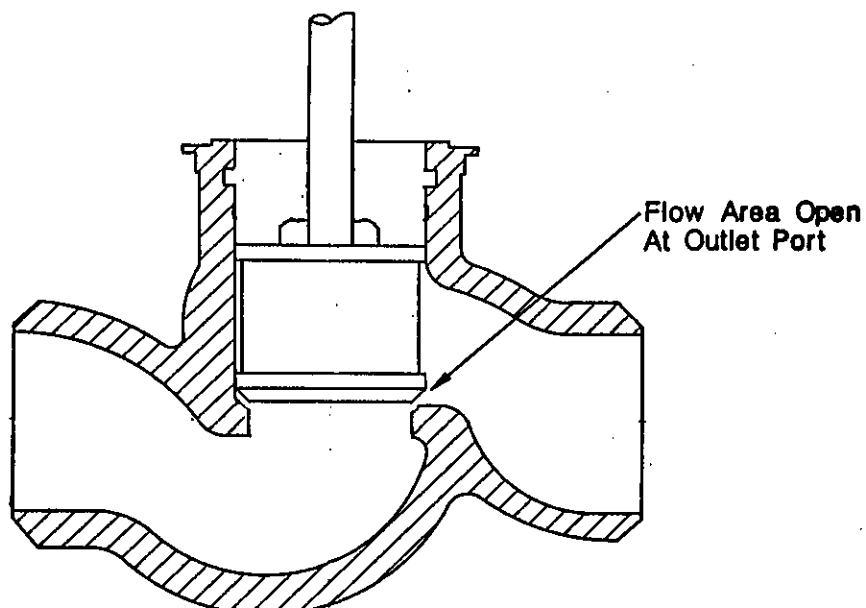


Figure 2. Globe valve disc details.



Guide-based Valve



Seat-Based Valve

Figure 3. Comparison of guide-based and seat-based designs.

Disc-to-Body Friction Force (F_{DF})

This force results from frictional resistance caused by flow-induced side loading of the disc. For unbalanced disc valves, it is neglected because its magnitude is small compared with F_{DP}. However, for balanced disc valves, this force is not negligible. Side loading on the disc is determined using a correlation derived from globe valve testing. The source data for the correlation is documented in Pool (1977). The correlation is of the form

$$G = \left[\frac{H}{D_G}\right] \cdot \text{fn}\left[\frac{X_S}{D_S}\right] \; . \qquad (1)$$

In Equation (1), G is the disc side load normalized by the product of DP and the disc guide area. The ratio H/D_G is a disc shape factor, and the correlation is considered valid for $H/D_G \leq 0.25$. The ratio X_S/D_S represents a normalized stroke position. The function $\text{fn}[X_S/D_S]$ is defined separately for underseat and overseat flow; it has a maximum value at the fully closed position ($X_S/D_S = 0$) and decreases as X_S/D_S increases. To determine a total stem load to overcome disc friction, the following equation is used:

$$F_{DF} = (1.1 \; \mu \; G + f_{\Delta P}) \; \Delta P_D \; A_{DG} \; . \qquad (2)$$

Within the parentheses, the first term accounts for sliding friction, where G is the normalized side load from Equation (1) and μ is the coefficient of friction. The value of μ can be affected by the materials in contact, contact stress, temperature, stroke history, and fluid medium. A value of 0.6 for μ is used in the model to bound the metal interfaces typical of globe valves. The factor of 1.1 covers uncertainties in the use of the disc side load correlation for a range of disc configurations. The second term in the parentheses, $f_{\Delta P}$, accounts for small pressure imbalance loads that exist in balanced disc valves. A value of 0.1 for $f_{\Delta P}$ is used in the model.

Stem Rejection Load (F_{SR})

This load results from the valve internal pressure acting to expel the valve stem from the valve body. It is defined as pressure above the disc times the area of the valve stem.

Disc and Stem Gravity Load (F_G)

This load is the component of the disc and stem weight that acts along the axis of the valve stem. It is usually a minor contributor to stem thrust. When the DP (in psi) exceeds 70 times the nominal valve size (in inches), this term is less than 1% of F_{DP}, and may be neglected.

Packing Friction Force (F_{PF})

This force accounts for the axial load generated by friction between the valve stem and the packing. The model does not provide a method for calculating this force, but depends on the individual user's experience or test data to provide an appropriate value. Typically, values in the range of 1,000 pounds per inch of stem diameter have been used.

Torque Reaction Friction Force (F_{TR})

For motor-operated valves with nonrotating stems, torque in the valve stem is reacted by a torque arm or stem key. This thrust term accounts for the friction developed as the stem slides along the torque reaction surface and is calculated

$$F_{TR} = \frac{\mu_T \; T}{r_t} = \frac{\mu_T \; FS \; F_R}{r_t} \; . \qquad (3)$$

In Equation (3), μ_T is the coefficient of friction at the sliding interface of the torque reaction surface; a value of 0.5 is used in the model. The parameter r_t is the radius from the stem centerline to the center of the torque reaction surface. FS is the stem factor that is dependent on the stem thread configuration and the stem-to-stem nut friction coefficient. The standard power screw equation with a thread friction coefficient of 0.15 is used to determine FS.

The resultant stem thrust (F_R) is the sum of the preceding six force components. The sign convention used for the model is that a positive force places the valve stem in tension, while a negative force places the stem in compression. As shown in Figure 1, some of the force components can be either positive or negative depending on the fluid flow direction (underseat or overseat) and the direction of the stroke (opening or closing). Table 1 summarizes the sign possibilities for each term.

Using the force components from Figure 1, the stem thrust is calculated

$$F_R = (F_{DP} + F_{DF} + F_{SR} + F_G + F_{PF} + F_{TR}) \ . \tag{4}$$

Because the torque reaction friction force is proportional to the resultant stem force, Equation (4) can be rewritten using a dimensionless torque reaction factor (TRF)

$$F_R = (F_{DP} + F_{DF} + F_{SR} + F_G + F_{PF}) \cdot \frac{1}{TRF} \ , \tag{5}$$

where

$$TRF = 1 - \frac{\mu_T \ FS}{r_t} \ . \tag{6}$$

The model calculates the thrust necessary to move the disc from the closed position to the fully open position for opening strokes, and from the fully open to closed position for closing strokes. During actual valve operation in the closing direction, the operator motor typically continues to increase the stem thrust after seat contact occurs. This additional thrust generates a "sealing load" that helps to ensure that the valve is leaktight. The actual sealing load developed is dependent on characteristics of the actuator and is not calculated by the model. However, an estimate of the sealing load required to ensure a leaktight seal when the valve is in the closed position (dependent on the configuration and condition of the seating faces, the fluid to be sealed, and the DP) may be obtained from *The Application Guide for Motor-Operated Valves in Nuclear Power Plants*.[a] Typically, seat contact stresses of about 4 to 8 ksi are recommended for adequate sealing.

COMPARISON OF MODEL PREDICTIONS TO TEST DATA

The model was evaluated against test data for five globe valves manufactured by different vendors. These data were obtained from testing performed in the EPRI MOV Performance Prediction Program. Table 2 lists the valves tested and provides information about the valve designs and flow conditions during testing. Key test measurements used in the model evaluation include stem thrust, upstream pressure, valve differential pressure, and stem position.

Measured values for valve dimensions, DP, pressure, and packing load were input into the model to predict stem thrust. As mentioned previously, unbalanced disc valves need to be identified as seat-based or guide-based according to the developed guidelines. Using these guidelines, two of the four unbalanced disc valves tested were classified as seat-based (Valves C and D) and two were classified as guide-based (Valves A and B). The predictions were compared with the measured test data on a plot of stem thrust versus stem position (% stroke). Predictions were made for both seat-based and guide-based behavior and then plotted along with the test data, making a total of three plots on each graph. Selected comparisons are shown on Figures 4 through 10, which are discussed individually later in this paper. On each figure, the valve designation, stroke direction, flow direction, flow rate, and DP are identified.

The stem thrust data include the additional sealing load applied by the operator after the valve seats. This behavior is shown on each of the thrust data plots as the steep-slope portion of the

a. Electric Power Research Institute, Palo Alto, CA, NP-6660-D, March 1990.

curve near 0% stroke. As previously mentioned, the globe valve model does not calculate this sealing load. Thus, it is not meaningful to compare measured and predicted thrusts in this region of the data.

Comparisons between model predictions and thrust data from valve testing are discussed in subsequent sections. The valves and test conditions listed in Table 2 were chosen so that the model could be evaluated for the following designs and conditions:

- Balanced and unbalanced disc designs
- Rising and rising/rotating stem designs
- Various valve sizes
- Various valve pressure classes
- Opening and closing strokes
- Overseat and underseat flow
- Various fluid flow rates
- Various valve differential pressures.

Table 2. Globe valve tests.

Valve designation	Size (in.)	ANSI class	Design features	Flow conditions
A	6	900	Unbalanced disc Rising stem Y-pattern body Carbon steel disc and body Stellite guide faces	Underseat flow Opening and closing strokes 15 ft/s and 50 ft/s ambient water flow 600, 1,200 and 1,800 psid ΔP
B	2	1,500	Unbalanced disc Rising/Rotating stem Y-pattern body Stainless steel body Stellite disc	Underseat flow Opening and closing strokes 15 ft/s and 50 ft/s ambient water flow 625, 1,250, 1,875 and 2,500 psid 2,600 psi, 625°F blowdown
C	2.5	1,500	Unbalanced disc Rising stem T-pattern body Carbon steel disc and body Stellite guide faces	Underseat flow Opening and closing strokes 15 ft/s ambient water flow 625, 1,250, 1,875 and 2,500 psid
D	10	300	Unbalanced disc Rising stem T-pattern body Stainless steel disc and body Stellite guide faces	Underseat and overseat flow Opening and closing strokes 15 ft/s ambient water flow 165, 330 and 500 psid
E	4	300	Balanced disc Rising stem T-pattern body Stainless steel disc and body	Underseat flow Opening and closing strokes 1,625 gpm ambient water flow 275 psid differential pressure

Seat-Based Valves in Incompressible Flow

Seat-based valves (C and D) were tested in ambient water with both overseat and underseat flow. Comparisons of measured and calculated stem thrust for Valve D are shown on Figure 4 for underseat flow and Figure 5 for overseat flow. The trend of the curves on Figure 4 is that the compressive stem thrust increases as the valve closes (from right to left on the figure), resulting principally from the DP increasing during the closure. The trend on Figure 5 is that the tensile thrust decreases as the valve is opened (from left to right on the figure). These figures show that in each flow direction the seat-based predictions bound the test data with a slight margin. If the exact seating diameter (based on valve posttest inspection) is used in the model instead of the maximum seat diameter, agreement between the seat-based prediction and test data is almost exact, as shown in Figure 6 for underseat flow and Figure 7 for overseat flow.

Guide Based Valves in Incompressible Flow

Guide-based valves (A and B) were tested in ambient water with underseat flow at two fully open flow rates. Comparisons of the model calculations and test data are shown for the low flow rate (15 ft/s) on Figure 8 and for the high flow rate (50 ft/s) on Figure 9. The trend in both figures is that increasing compressive thrust is required to close the valve (from right to left) until the valve is almost closed. Starting at about 5% open, the compressive thrust decreases significantly prior to the valve seating. The decrease in thrust at the end of the stroke results from the valve controlling area being shifted from the guide to the seat as the flow is closed off. The guide-based prediction agrees favorably with the data until this final transition region. In this region, the data approach the seat-based prediction as the controlling area changes from guide to seat. The guide-based prediction continues to increase because the DP increases slightly in the transition region. The effect of higher flow (Figure 9) is to reduce the change in DP in this region; hence, the absolute overprediction of the model is reduced at higher flow. As expected, the seat-based predictions in Figures 8 and 9 considerably underpredict the data. Therefore, it would not be appropriate to use a seat-based calculation to determine required stem thrust for these guide-based valves.

Balanced Disc Valve in Incompressible Flow

A single test was run using the balanced disc valve, Valve E. The test was run with 40 ft/s ambient water at approximately 275 psid. As expected, very low stem thrust was required throughout the valve stroke because of the balanced disc design. The model prediction bounded the test data at all stroke positions for both the opening and closing strokes of this valve.

Flashing Water Blowdown Flow

Valve B was tested with underseat flow to isolate a blowdown of hot water at 2,600 psi and 625°F. The maximum flow was 120 lbm/s, which yields a velocity of 158 ft/sec of hot water in the valve inlet piping. The DP across the valve increased from 1,000 to 2,600 psi as the valve moved from the open to the closed position. Flashing occurred at the valve throughout its stroke. A comparison of predicted and measured thrust for this stroke is shown on Figure 10. The required compressive thrust exceeded the guide-based prediction at stem positions greater than 15%. The maximum measured thrust was not bounded by the prediction. A posttest inspection of the valve revealed that scratching on the body bore and valve stem had occurred during this test stroke. It appears the disc was loaded in a manner not considered by the model. Adjustment of the model to address this flow condition will require further investigation.

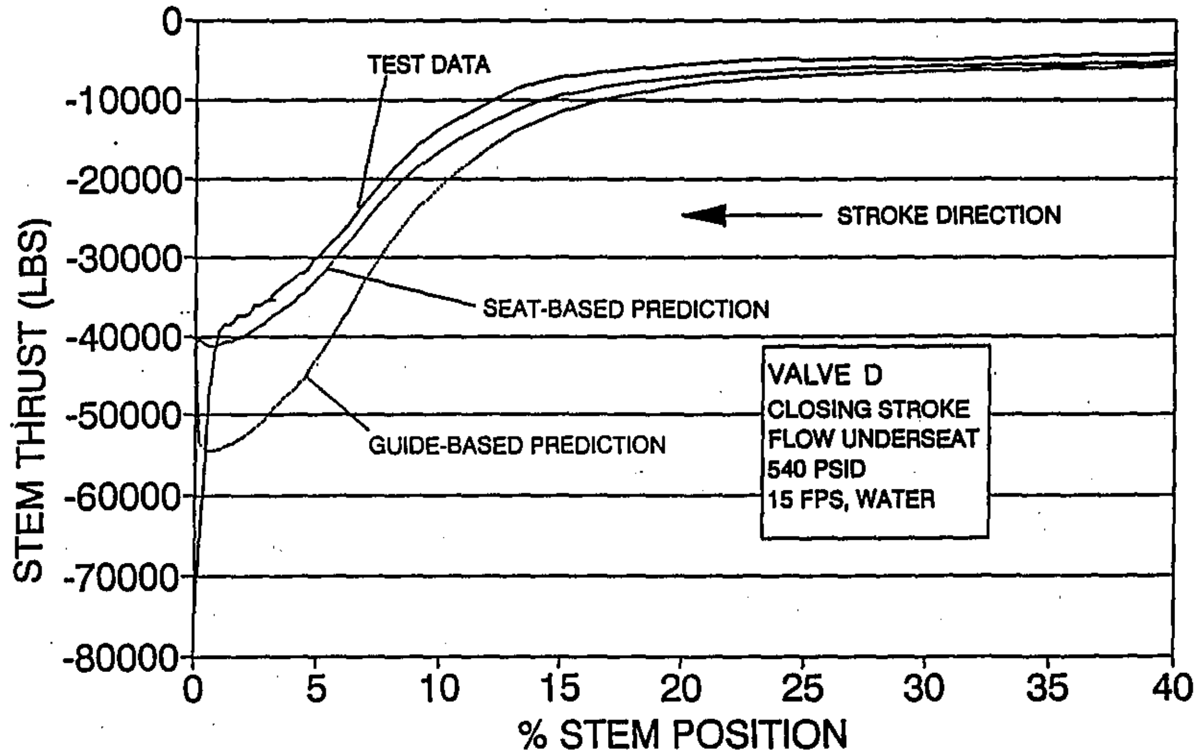


Figure 4. Measured and calculated stem thrust for a seat-based valve with underseat flow.

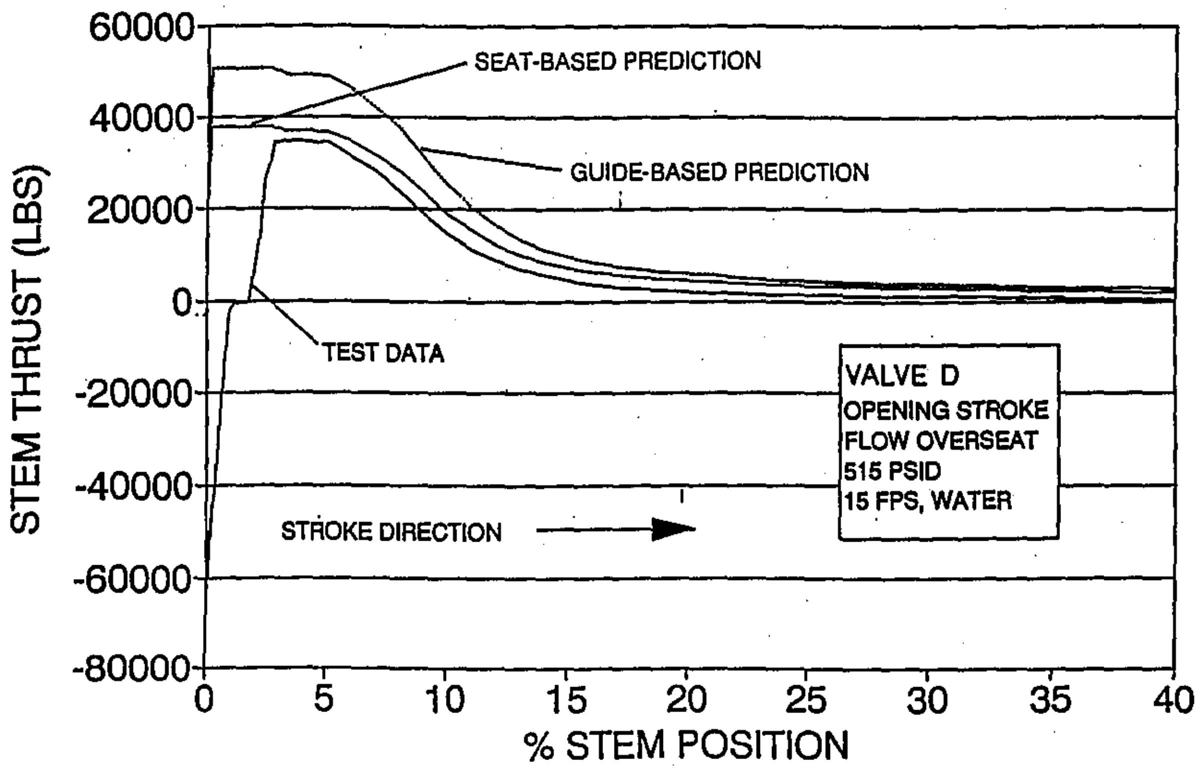


Figure 5. Measured and calculated stem thrust for a seat-based valve with overseat flow.

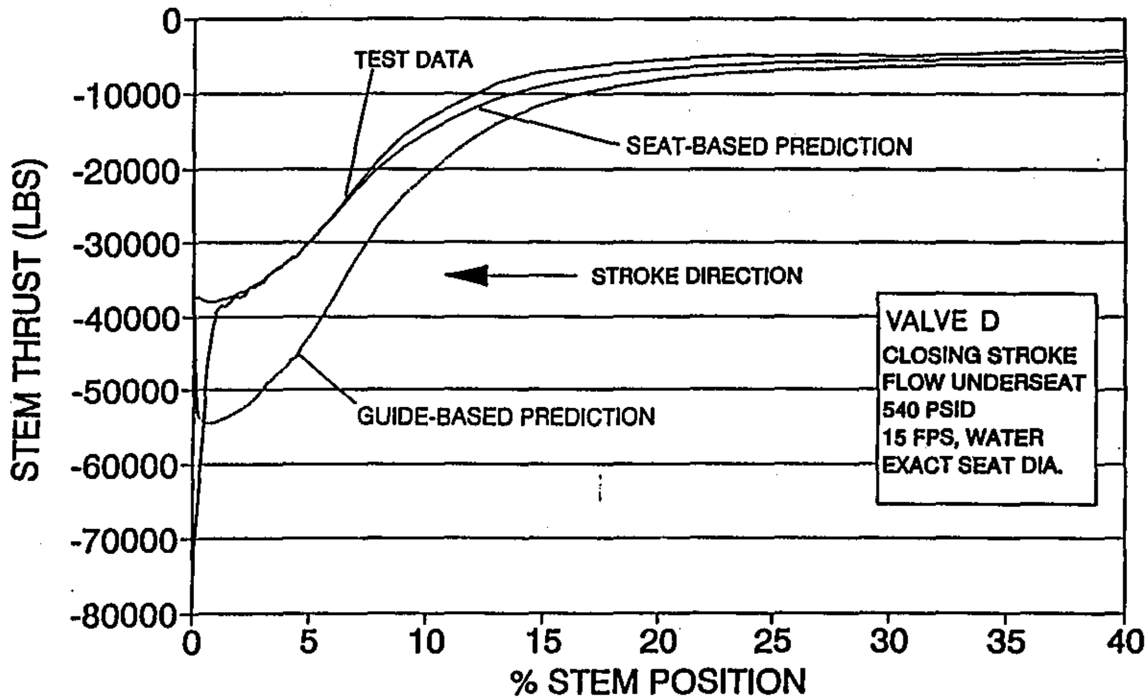


Figure 6. Measured and calculated stem thrust for a seat-based valve, with underseat flow. Exact seating diameter used for seat-based calculated thrusts.

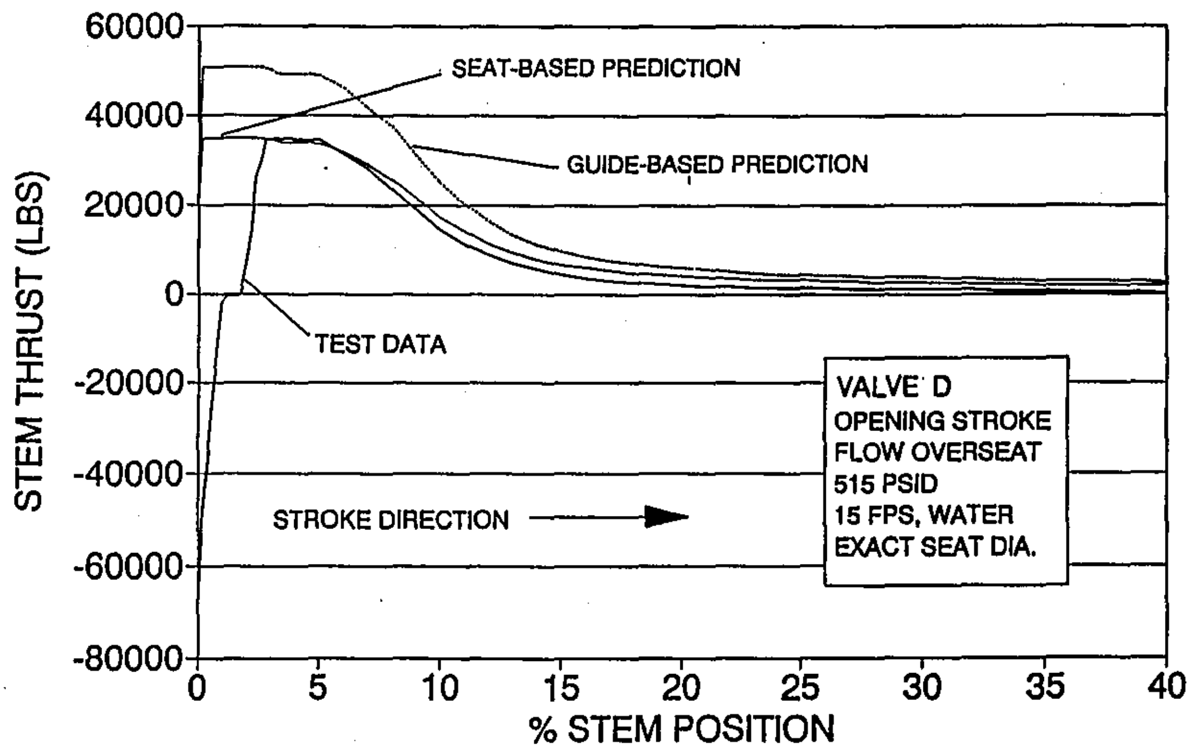


Figure 7. Measured and calculated stem thrust for a seat-based valve with overseat flow. Exact seating diameter used for seat-based calculated thrusts.

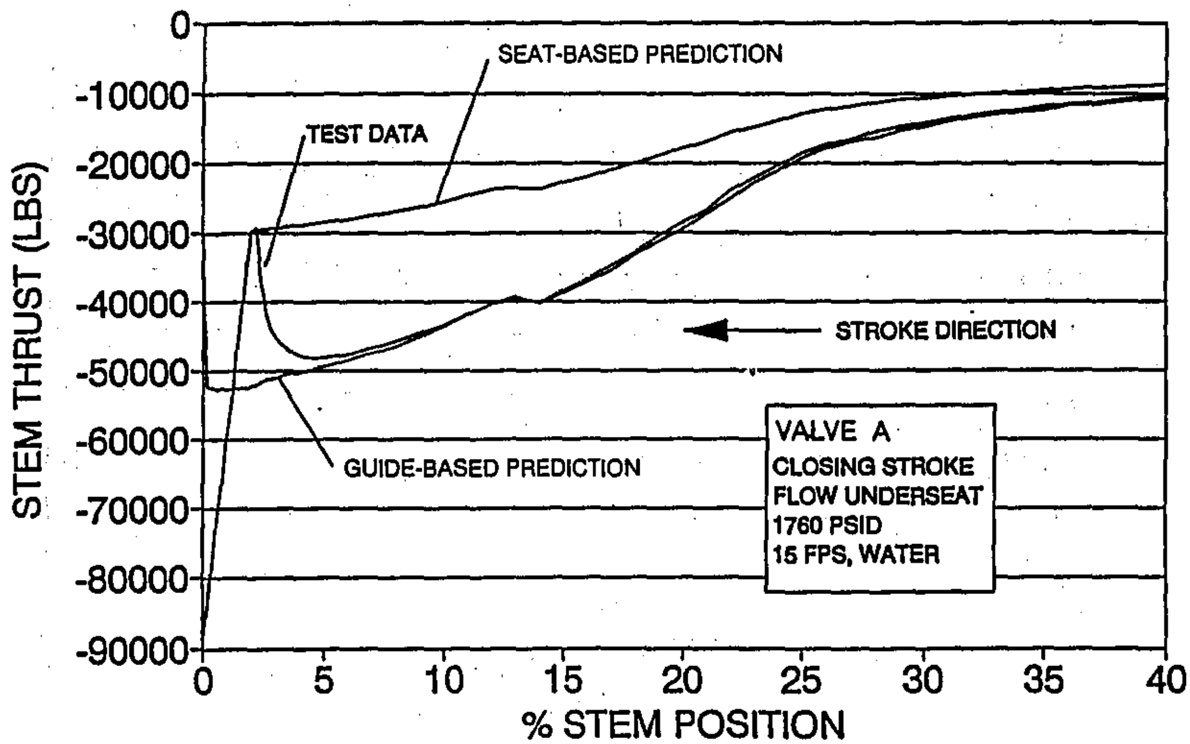


Figure 8. Measured and calculated stem thrust for a guide-based valve with 15-ft/s water flow.

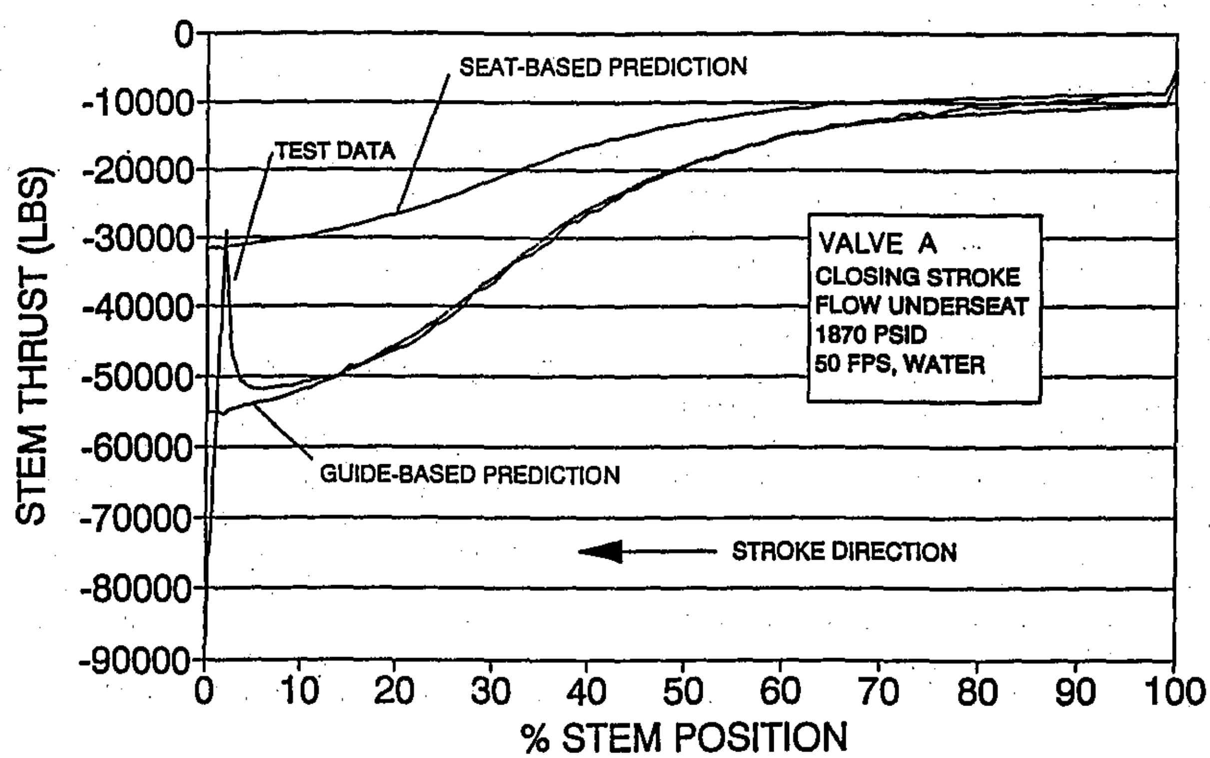


Figure 9. Measured and calculated stem thrust for a guide-based valve with 50-ft/s water flow.

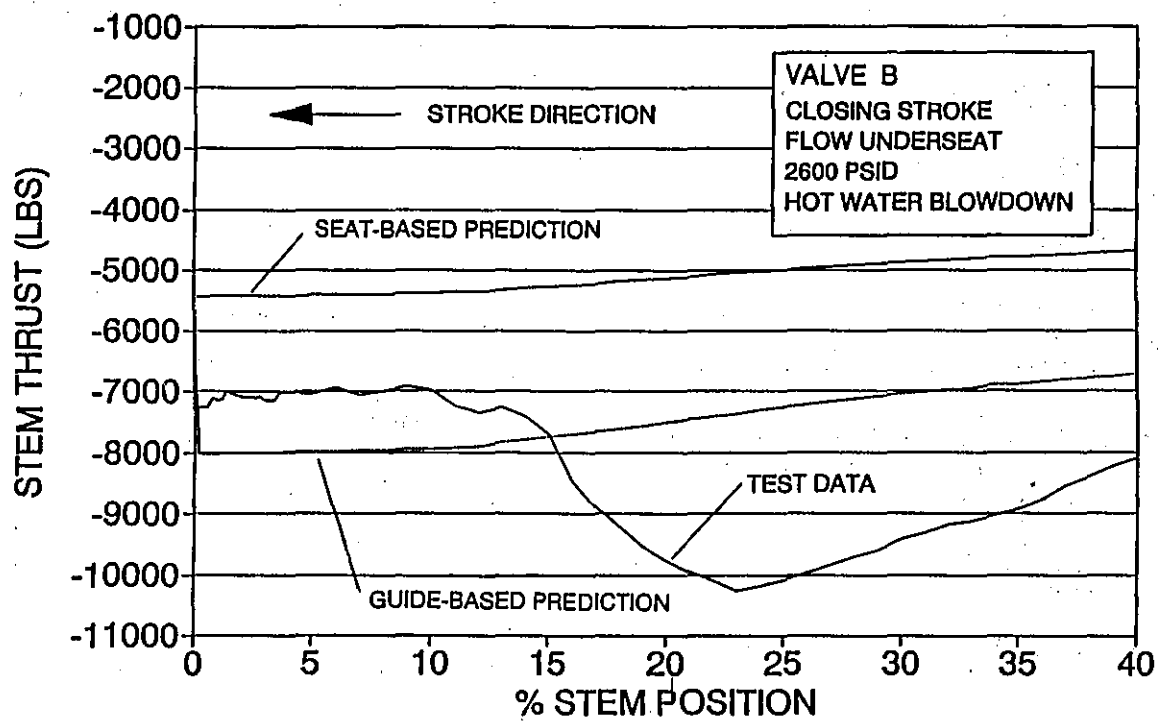


Figure 10. Measured and calculated thrusts for a guide-based valve under hot water blowdown conditions.

SUMMARY

Incompressible Flow

Table 3 provides a summary of the maximum predicted and measured thrusts for the five valves tested. For the seat-based valves, the measured results are bounded by the predictions based on disc seat area. The margin is 3 to 12%. The overprediction can be mainly attributed to the use of the maximum disc seat diameter in the model, as opposed to the actual seat contact diameter. For these seat-based valves, use of the guide area would have provided a prediction that greatly exceeds the measured thrust.

For the guide-based valves, the measured results are bounded by the predictions based on disc guide area. The margin is 7 to 23%. The overprediction can be mainly attributed to the fact that peak DP does not occur simultaneously with peak thrust. For these valves, use of the seat area would have significantly underpredicted the measured thrust.

The model predictions for the balanced disc valve bound the observed results by 20 to 25%. The conservative assumptions on disc imbalance force and guide-to-body friction used in the model account for some of this overprediction.

Flashing Water Blowdown Flow

As stated above, the model has not demonstrated the ability to accurately predict valve stem thrust under flashing water blowdown conditions. The only test stroke run under these conditions caused valve damage. Further work on flashing water blowdown is required to adequately resolve the root cause of the damage and determine the appropriate adjustments for the model.

Table 3. Summary of maximum predicted and measured stem thrusts.

Valve designation	Disc design	Valve behavior	Flow velocity (ft/s)	Nominal differential pressure (psid)	Flow direction	Stroke direction	Maximum predicted thrust (psi)	Maximum measured thrust (psi)	Percent difference
A	Unbalanced	Guide-based	15	1,800	Underseat	Closing	–52,700	–48,200	9.3
			50	—	—	—	–55,300	–51,800	6.8
B	Unbalanced	Guide-based	15	2,500	Underseat	Closing	–8,020	–6,530	22.8
			50	—	—	—	–7,960	–7,270	9.5
			B/D[a]	—	—	—	–8,030	–10,270	–21.8
C	Unbalanced	Seat-based	15	2,500	Underseat	Closing	–9,400	–8,400	11.9
			—	1,900	—	—	–7,400	–6,600	12.1
			—	650	—	—	–3,100	–3,000	3.3
D	Unbalanced	Seat-based	15	500	Underseat	Closing	–41,300	–39,200	5.4
			—	—	Overseat	Opening	37,000	34,500	7.2
E	Balanced	N/A	40	275	Underseat	Closing	–2,490	–2,077	19.9
			—	—	—	Opening	1,103	883	24.9

a. Balanced disc.

CONCLUSION

The use of a basic, first-principles model to access globe valve performance yields accurate stem thrust predictions for body-guided globe valves under incompressible flow conditions. Further examination of globe valves under flashing water blowdown will be required to be able to develop accurate stem thrust predictions for this application.

ACKNOWLEDGMENT

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REFERENCE

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Effects of Dynamic Loading of Motor-Operated Valve Actuators

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ABSTRACT

Experience has shown that valves with rising, nonrotating stems that are operated using electric-motor driven actuators can be susceptible to changes in output thrust at a constant torque switch setting as a result of changes in stem load time history. This effect is a concern because tests on these types of valves to verify thrust achieved at torque switch trip are often performed in situ under load conditions different from the required performance conditions. As part of a motor-operated valve research program being carried out by the Electric Power Research Institute, tests of typical electric motor actuators used with nuclear service valves have been performed. The test results show that changes in output thrust with load time history occur to varying degrees on different stem and stem nut combinations. When the effect exists, there is generally an increase in thrust at torque switch trip when load is developed rapidly from low initial loads, compared to when load is developed slowly. The effect is mainly a result of changes in the coefficient of friction at the stem-stem nut interface. The coefficient of friction is temporarily reduced under rapid loading conditions from low initial load, leading to increased thrust. The root cause is hypothesized to be a "squeeze-film" effect, whereby mixed-mode lubrication (hydrodynamic plus boundary) temporarily replaces boundary lubrication. This paper describes the results of tests performed to better understand the phenomenon.

BACKGROUND

The actuator thrust output during motor-operated gate and globe valve closures is typically limited by use of a torque switch. In a typical actuator, torque is applied by an electric motor through a gear train to rotate a stem nut. The stem nut, in turn, drives the valve stem in translation, thus opening or closing the valve. The gear train includes a worm and worm gear. The worm is splined to its carrier shaft and is restrained from translating along the shaft by a stack of Belleville spring washers (the spring pack). As the torque output of the actuator increases, the worm compresses the spring pack. At a prescribed compression, the torque switch opens (trips) and removes electrical power from

the motor, thus limiting the operator output torque and, consequently, the thrust.

Results of previous testing at nuclear plants (Black, 1990a and 1990b) and in laboratories (Steele et al., 1992) have shown that the stem load time history can affect the thrust output of torque switch controlled motor-operated valves (MOVs). If the stem load develops very quickly, the thrust output at torque switch trip will sometimes be higher than if the load develops more slowly. The effect has been called "rate-of-loading" (Black, 1990a and 1990b) and "load-sensitive behavior" (Steele et al., 1992) and has generally been attributed to changes in the stem-stem nut coefficient of friction. These changes in the coefficient of friction affect the efficiency with which operator output torque is converted to stem thrust.

The phenomenon is a concern because in-plant testing of MOVs to verify that sufficient thrust is developed at torque switch trip is often conducted under conditions that are different from those under which the MOV is required to function. For example, valves are often tested under "static" conditions of zero flow and zero differential pressure (DP). The valve is closed, and as the gate or globe seats, the stem load increases rapidly. However, when the valve is required to close against design basis conditions (high flow and DP), the stem load will develop more slowly. Under this condition, the torque switch may trip at a value of stem thrust that is significantly lower than in the static (zero-DP) test. Evaluation of test data from in situ tests needs to consider the potential thrust differences that may occur between static tests and valve closures under design basis conditions.

This paper describes work carried out as part of the Electric Power Research Institute (EPRI) MOV Performance Prediction Program. The objective of the work was to evaluate effects that result from dynamic loading of MOV actuators. The program included extensive laboratory testing of Limitorque motor-operators with a number of stem and lubricant combinations. Based on these tests, as well as other industry and laboratory data, the phenomenon associated with dynamic loading has been characterized.

As mentioned above, the terms "rate-of-loading" and "load-sensitive behavior" have been used. We have found that both load and its time rate of change are important parameters. The term "rate-of-loading" (ROL) is used in this paper as a matter of convenience, and is defined as a change in the relationship of stem thrust to spring pack displacement, depending on the stem load time history. Different ratios of stem thrust have been proposed to quantify operator sensitivity to ROL. In this paper, ROL sensitivity is quantified by the ratio of stem thrust at torque switch trip (TST) under static conditions to the stem thrust at TST under simulated DP conditions.

The EPRI MOV Performance Prediction Program is being carried out to develop improved methods to predict the performance of MOVs in nuclear power plants. Actuator performance and use of in-plant testing to evaluate actuator output capability were recognized as key elements at the outset of the program. Originally, it was planned that an analytical actuator model would be developed to account for and to predict the ROL effect. A preliminary model was developed, as described by Dogan and Hosler (1992). The principal feature of the model related to ROL was that the friction coefficients at the stem-stem nut interface and at the worm-spline interface were assumed to vary with the relative sliding speed at each of these interfaces. The friction coefficients were described by equations that included undetermined constants. Using this type of model, Dogan and Hosler (1992) showed that the types of ROL behavior seen in MOV tests could be predicted. Further, MOV data indicated that the changes in speeds at the sliding interfaces occurred about as predicted by the model.

The original objective of the actuator tests in the EPRI program was to determine the appropriate values for the constants in the analytical model. However, when initial test results were examined, it became apparent that the coefficient of friction required for the model could not be predicted. Specifically, although changes in friction coefficients were occurring, they could not

be reliably correlated to sliding speed. In addition, it was clear that effects other than sliding speed (such as load, load time history, even stem-stem nut fabrication details) were exerting a strong influence. Accordingly, the approach using the analytical model was abandoned. Subsequent testing and evaluation efforts were concentrated on developing a better understanding of the phenomenon and identifying test methods which are suitable for in situ evaluations of its effect on MOVs.

ACTUATOR SEPARATE EFFECTS TEST PROGRAM DESCRIPTION

Test Facility

The testing was conducted by Battelle Columbus on test stands assembled for the EPRI Program. Figure 1 shows the facility used for actuator testing. The facility was designed to provide a means to test valve actuators and stems with simulated load time histories and to accurately measure the actuator performance under these conditions. The principal elements of the facility are the support frame, the simulated valve yoke, the hydraulic loading system, and the actuator and stem.

The simulated valve yoke and top and bottom plates form the structural frame used to react the loads developed by the actuator. It was designed to be representative of a rigid valve/yoke assembly. Load time histories simulating valve operation were applied by a hydraulic cylinder attached to the lower part of the stem. The load was controlled by a closed loop feedback system, so that pre-determined load histories could be applied. The hydraulic system, however, was not able to develop the rapid load increase that is necessary to simulate a valve static test. Accordingly, the support frame included a simulated valve seat against which the stem could be loaded. The stiffness of the seat could be changed by adjusting the position of the stops and by substituting different stop beams. The stiffest configuration of this simulated valve seat is referred to as the hard seat.

Tested actuators and stems were typical of hardware used in nuclear service. Two Limitorque actuators were used that were representative of two different designs. One was an SMB-0 model with a 25-ft-lb motor and overall gear ratios ranging from 34 to 69. The other actuator was an SMB-000 model with a 5-ft-lb motor and overall gear ratios of 20 to 50. The stems had standard ACME threads and covered a range of diameter, pitch and lead (see Table 1). They were made of classes of stainless alloys (304, 410, 17-4PH) representative of those found in service. Stem nuts were fabricated from manganese/bronze stem nut "blanks" provided by Limitorque using standard industry methods.

Instrumentation and Measurement Uncertainty

The principal measurements from the test facility are actuator output thrust, actuator output torque, and spring pack displacement. Actuator output thrust and torque were measured using a torque/thrust cell (TTC) mounted between the actuator and the simulated yoke. All of the load transmitted between the actuator and the yoke is carried by internal strain-gaged structural members of the TTC. The outputs of the strain gages are combined to provide thrust and torque data that are accurate to less than 0.1 percent of full scale.

Spring pack displacement was measured with an LVDT. Spring pack force, stem position, and motor parameters (speed, temperature, voltage, power) were also routinely recorded, but not used extensively in analyzing ROL.

Torque switch trip was detected with a custom electronic circuit that sensed frequency changes in the contactor holding coil. It is important to measure torque switch trip very precisely and accurately because TST is used as a standard reference point for comparing MOV performance. The limiting factor in resolving data at TST is the sampling frequency of the data acquisition system. The sampling frequency used for the tests was 250 Hz, which allowed a resolution of

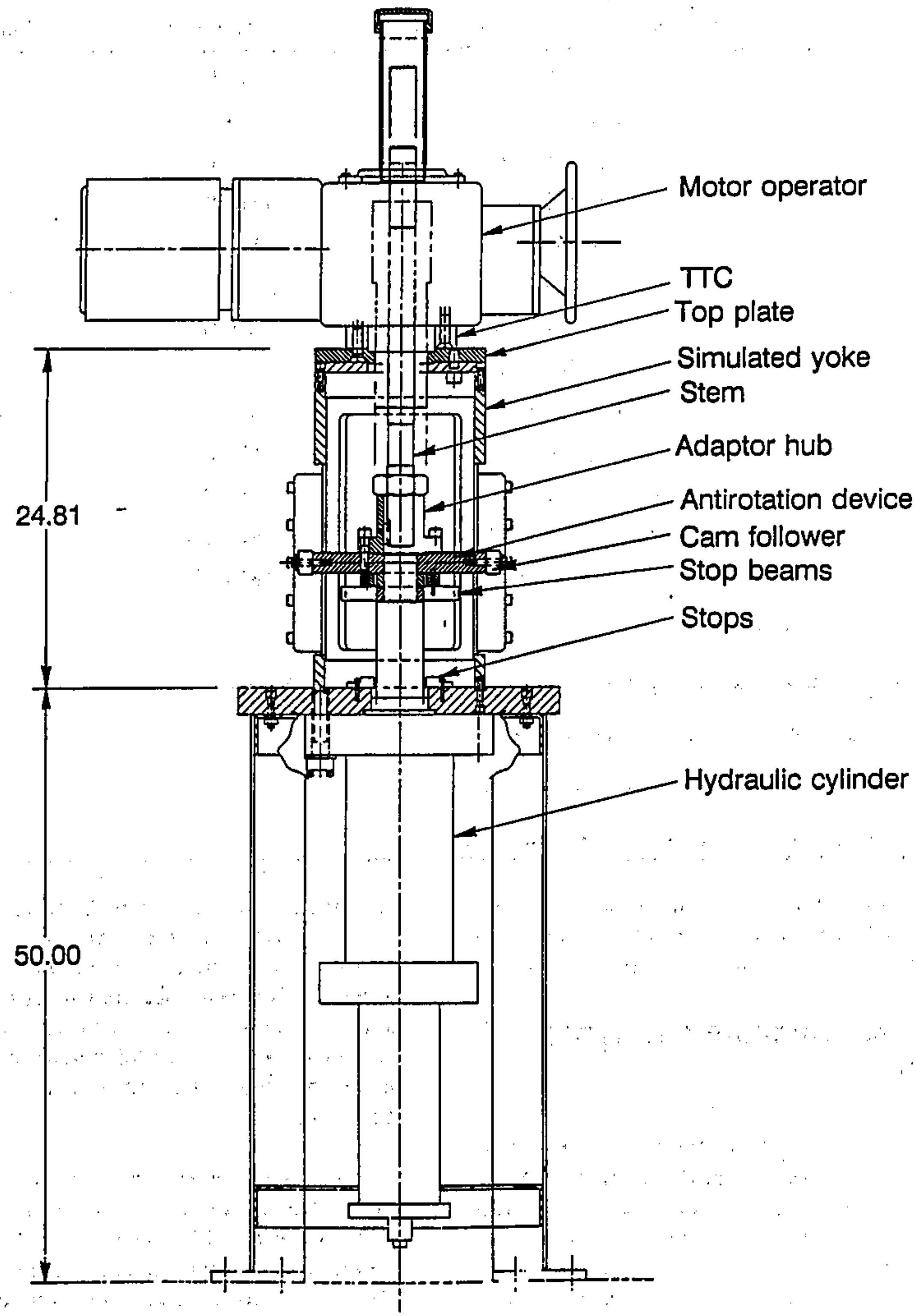

Figure 1. ROL test stand.

Table 1. Tested hardware and lubricant combinations.

Stem	Geometry (diameter × pitch × lead) (in.)	Operator	Lubricant
a	2 × 1/4 × 1/2	SMB-0	—[a]
b	1.5 × 1/4 × 1/2	SMB-0	—[a]
c	2 × 1/4 × 1/2	SMB-0	—[b]
d	2 × 1/4 × 1/4	SMB-0	—[a]
e	2 × 1/4 × 1/2	SMB-0	—[a]
f	1 × 1/6 × 1/3	SMB-0	—[a]
g	2 × 1/4 × 1/2	SMB-0	—[c]
h	2 × 1/3 × 2/3	SMB-0	—[a]
h	2 × 1/3 × 2/3	SMB-0	—[d]
i	1.75 × 1/4 × 1/4	SMB-0	—[a]
j	1.5 × 1/4 × 1/4	SMB-0	—[a]
k	1.12 × 1/5 × 1/5	SMB-000	—[a]
l	1.12 × 1/5 × 2/5	SMB-000	—[a]

a. Southwestern Petroleum Corporation, SWEPCO Moly 101, Grade 2 multi-purpose grease.

b. Never Seez, NG 165, nickel-based anti-seize.

c. FEL-PRO, C5A, copper graphite based anti-seize.

d. EXXON, Nebula EP, Grade 1, multipurpose grease.

400 lb for the thrust at TST at the maximum rate of applied load 100,000 lb/s. This relates to an uncertainty of ±1% on ROL measurements.

Typical Operation and Types of Tests

Operator performance tests were run principally in the valve closure direction. Tests were typically started by energizing the actuator with the stem positioned 2 to 4 inches from the hard seat. During the test, the hydraulic loads were controlled by a servo-hydraulic console to provide the required stem load time history. The stroke ended when the torque switch tripped to de-energize the motor. The stem was then driven in the open direction by the actuator with a low compressive load in the stem. Data were recorded in the closing direction from 1 second. before motor actuation until after the motor stopped. Typically, data were not recorded during the open stroke. In many cases, repeat closure strokes were run at the same load condition to obtain information on repeatability. Stems were not relubricated between tests.

Rising stem MOVs are used in a variety of applications, and the stem load time histories cover a considerable range. Instead of trying to simulate valve stem loading time histories in detail, a few basic load profiles were developed. The two profiles that provided the most information were a constant running load with hard seat and an increasing ramp load. Tests with the first type of load profile are referred to below as

"static" tests; tests with the second type of load profile are referred to as "ramp" tests.

In the static test profile, the stem load remains constant with time until the stem contacts the hard seat, whereupon stem load increases very rapidly. The load magnitude prior to seating simulates a running load in the valve.

The ramp load profile produces an increasing stem thrust over the stroke. Ramp rates were varied from approximately 1,000 lb/s. to higher rates that approached those achieved during hard seating 100,000 lb/s. Typically, during ramp tests, the torque switch setting, the ramp rate, and the initial distance away from the hard seat caused the torque switch to trip while the load was slowly ramping (before contacting the seat). This procedure was intended to maximize the possible differences in stem performance between ramp and static tests.

Over 700 tests were conducted. They included variations in loading profiles, actuator parameters (gear ratio, motor speed, etc.), lubricant, and, importantly, a number of different stem-stem nut geometries. The main hardware and lubricant combinations tested are summarized in Table 1.

RESULTS

ROL Variations Among Hardware

Among the stem, stem-nut, lubricant, and actuator combinations tested, a considerable range of dynamic behavior was observed. Some sets of test hardware showed essentially no dynamic effects. In other words, regardless of the stem load time history, the thrust achieved at TST did not significantly change (i.e., no ROL effect). Other sets of hardware showed a wide difference in thrust at TST (i.e., large ROL) as stem load time history was varied.

Figure 2 shows stem thrust time histories for two strokes with Stem f. One stroke has a constant running load (400 lb) with hard seat; the loading rate at hard seat is about 200,000 lb/s. The other stroke has a ramp load (1,600 lb/s.) without seating. On each curve, the point at which the torque switch tripped is marked. Two TST points are visible on the hard seat curve; they identify the thrust uncertainty band related to the determination of the exact time of torque switch trip.

For this particular set of hardware, there is essentially no difference in the thrust at torque switch trip for rapid versus gradual loading of the valve stem. Figure 3 shows the results of the same two tests plotted as torque versus thrust. The ratio of torque to thrust is known as "stem factor" (SF). For a given thread configuration, stem factor is related to thread coefficient of friction by the following equation:

$$SF = R_s * (\cos \Theta_s \tan\lambda_s + \mu_s) / (\cos \Theta_s - \mu_s \tan\lambda_s) \quad . \qquad (1)$$

In Equation (1), R_s, Θ_s, λ_s, and μ_s are the stem thread pitch mean radius, pressure angle, helix angle (or lead angle), and coefficient of friction, respectively.

Note that for this set of hardware, the stem factor and, consequently, the stem-stem nut efficiency, are not significantly affected by the type of loading. The repeatability of this result was demonstrated by additional tests.

Figure 4 is a plot of thrust as a function of time for a static and a ramp test conducted on a different set of hardware (Stem h). In this case, the thrust at torque switch trip is significantly affected by the dynamic loading. When the stem is loaded rapidly from low initial running load, the thrust at torque switch trip is considerably higher than that measured when the stem is loaded gradually during the ramp load test. Figure 5 shows results from the same tests plotted as torque versus thrust. There is a significant difference in the stem-stem nut efficiency, as reflected by the difference between the slopes of the two lines.

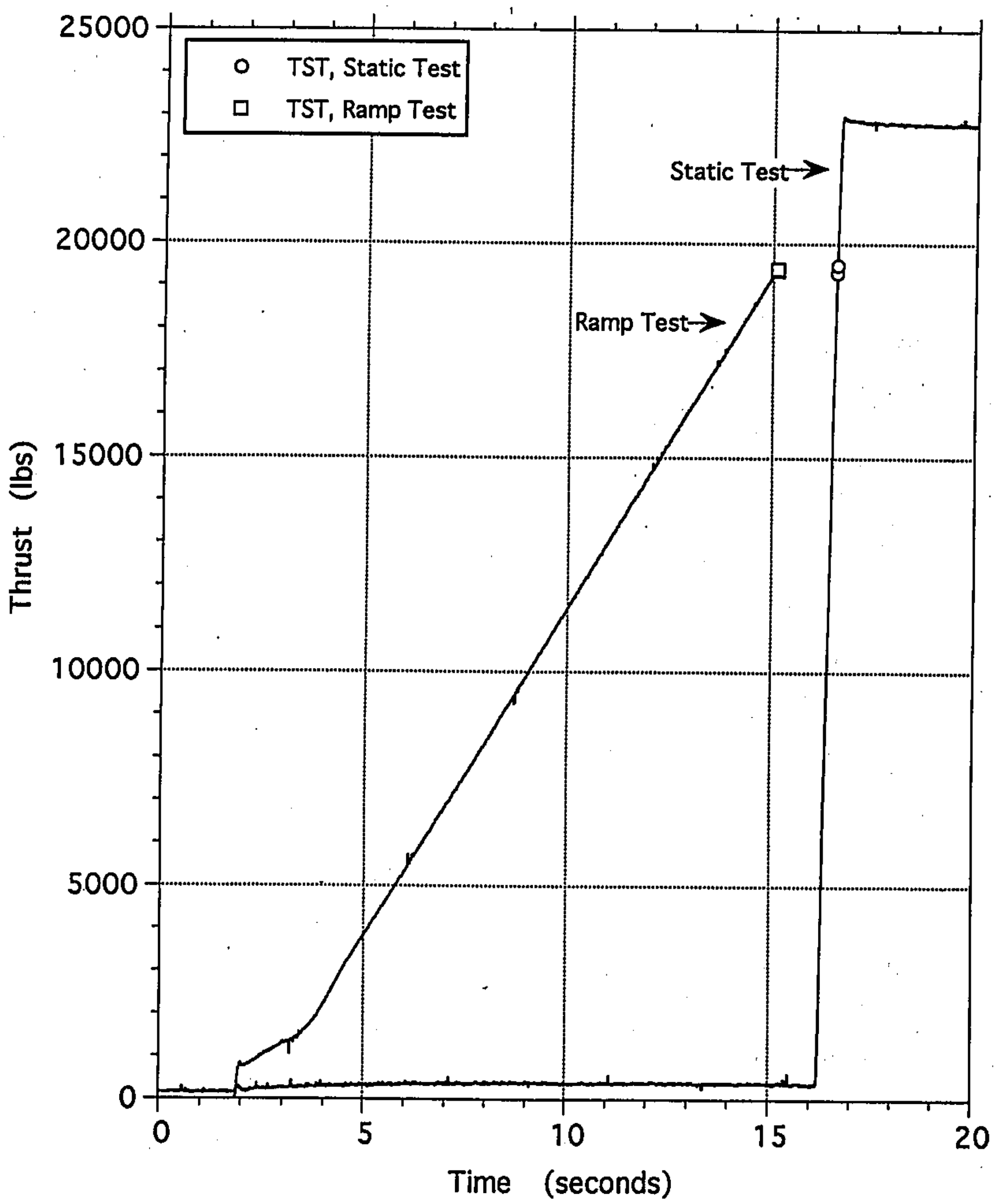

Figure 2. Load time histories for static and ramp tests—Stem f.

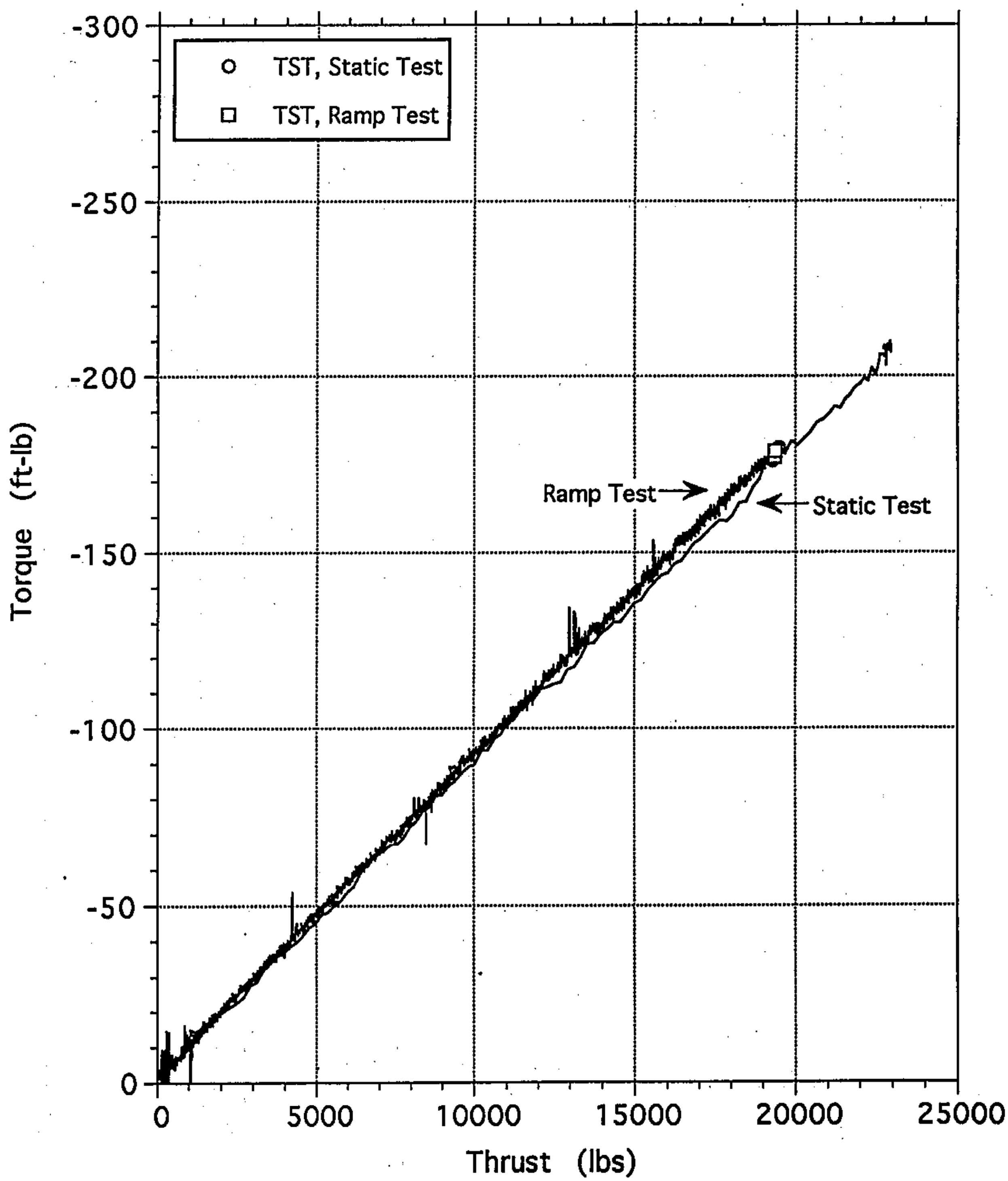

Figure 3. Torque X trust for static and ramp tests—Stem f.

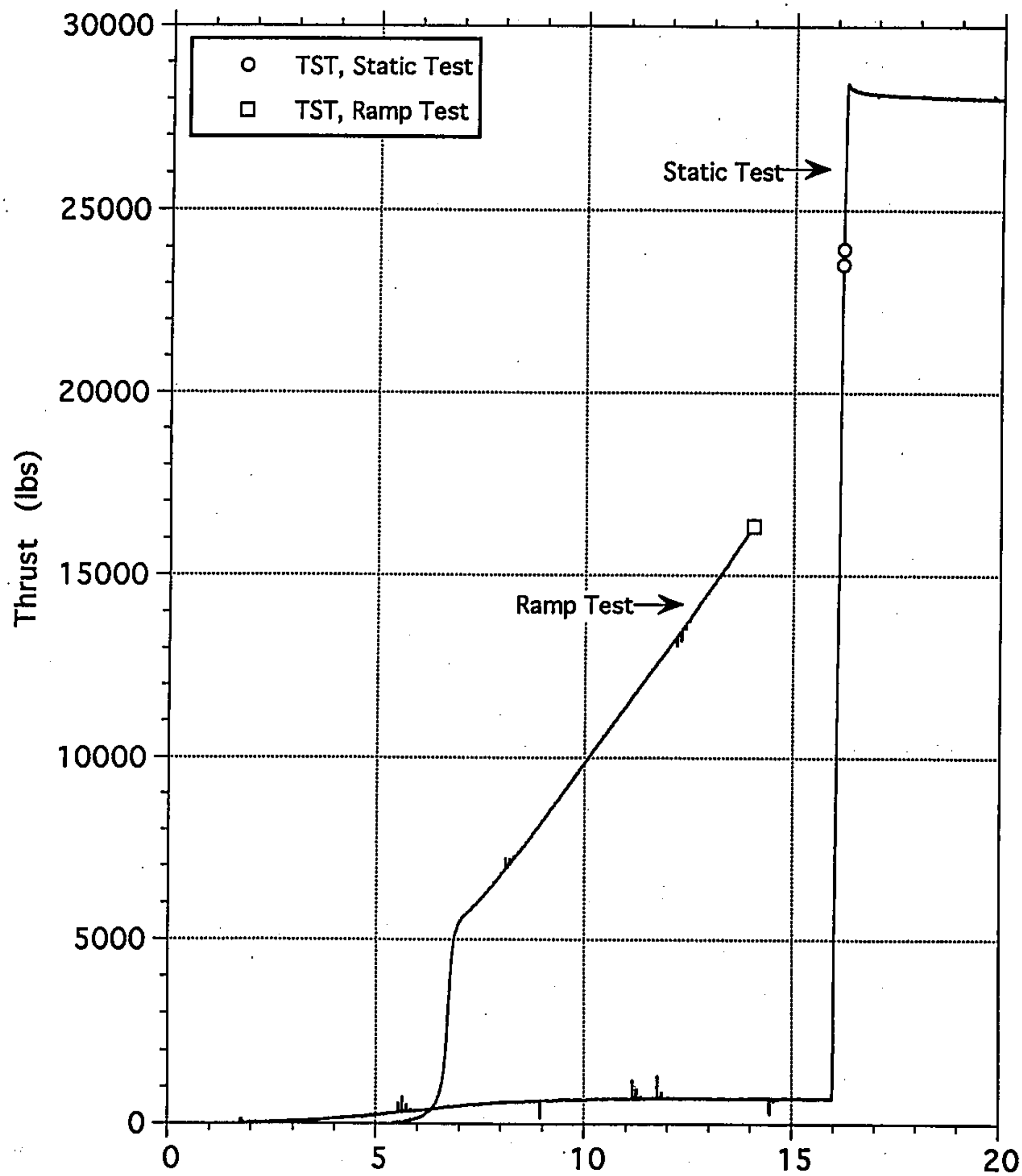

Figure 4. Load time histories for static and ramp tests—Stem h.

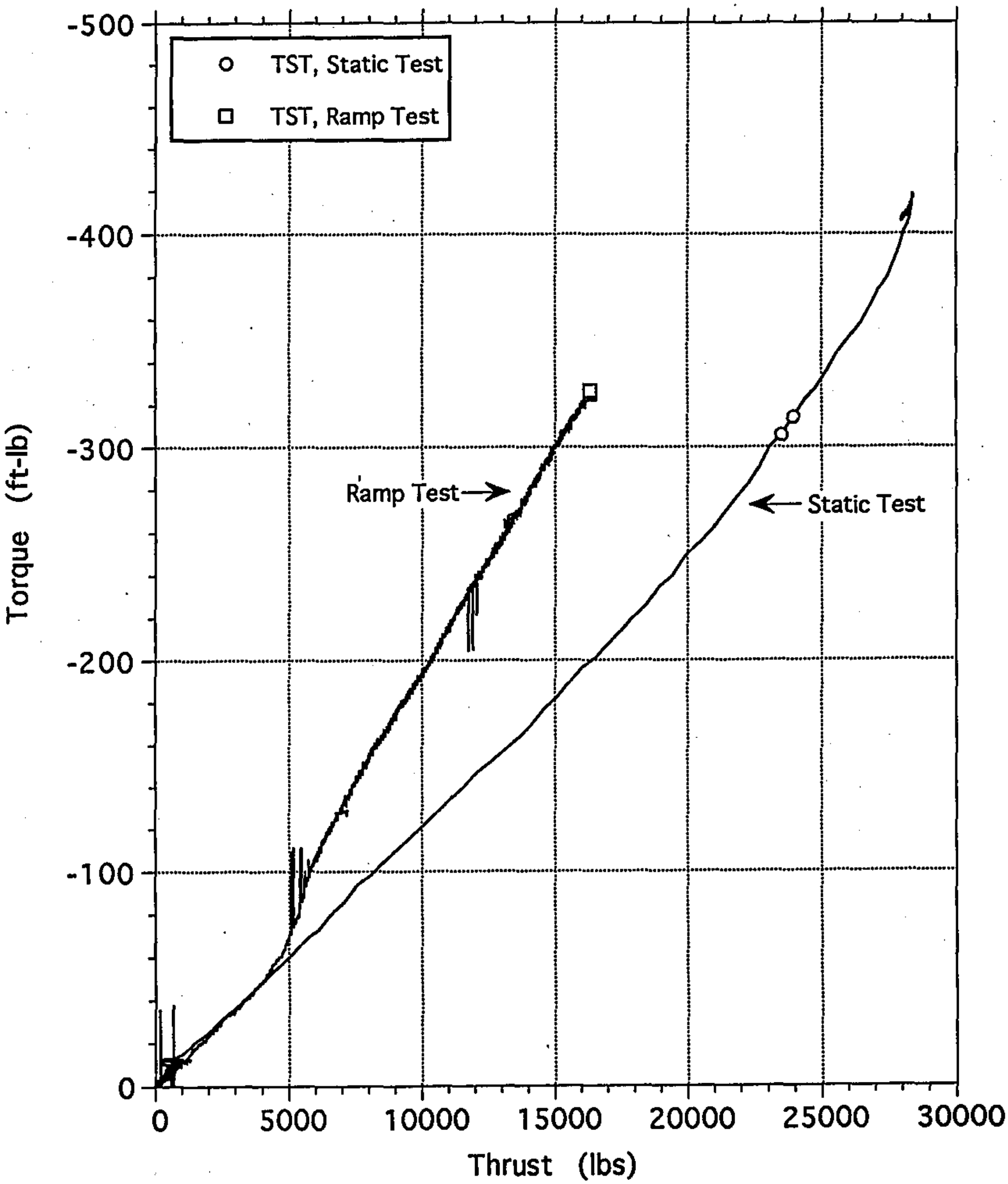

Figure 5. Torque X thrust for static and ramp tests—Stem h.

The efficiency of this stem-stem nut interface is much higher (i.e., lower friction coefficient or more thrust per unit torque) when the stem is loaded rapidly from a low initial load.

Table 2 summarizes the results obtained from all stem-stem nut combinations tested. The results reflect the wide variation of observed behavior. The results also show a good correlation between ROL effect and changes in stem factor. This correlation indicates that increases in thrust at torque switch trip during rapidly loaded tests from low initial load occur principally because of decreases in coefficient of friction at the stem-stem nut interface.

Effect of Running Load and Rate of Applied Load

Running load prior to hardseating (i.e., load sensitivity) is a factor in determining the magnitude of the ROL effect. Figure 6 shows the results of several tests conducted on Stem h, during which the load in the portion of the stroke prior to seating was increased in succeeding tests. Six strokes are shown. As the load prior to seating is increased, the thrust at torque switch trip is decreased to a level that approaching that observed during the ramp test. This implies that the behavior of the system at torque switch trip is affected by the prior history of the load and not just by the instantaneous load and rate of load conditions. Figure 7 shows that this behavior is explained by a reduced stem-stem nut efficiency (i.e., higher friction coefficient) as the system operates at higher loads. When this stem is operated at low load and then hard-seated, more thrust per unit torque is generated during the rapid load increase at hard seat. As the load prior to hard seat is increased, the thrust generated per unit torque is decreased, as can be seen by the changes in the slopes of the lines.

Table 2. Summary of results.

Stem	Lubricant	R_{TST}^{TH} [a]	$R_{TST}^{SF^{-1}}$ [b]
a	Moly 101	1.10	1.11
b	Moly 101	1.08	1.19
c	NG 165	1.02	1.01
d	Moly 101	1.01	1.03
e	Moly 101	1.02	1.04
f	Moly 101	1.00	1.00
g	C5A	1.00	.99
h	Moly 101	1.44	1.51
h	EP1	1.33	1.35
i	Moly 101	1.08	1.19
j	Moly 101	1.00	1.02
k	Moly 101	1.32	1.32
l	Moly 101	1.18	1.02

a. R_{TST}^{TH} :ratio of thrust at TST for static test over thrust at TST for ramp test.

b. $R_{TST}^{SF^{-1}}$:ratio of SF^{-1} (SF-1 = Th/Tq) at TST for static test over stem factor at TST for sample test.

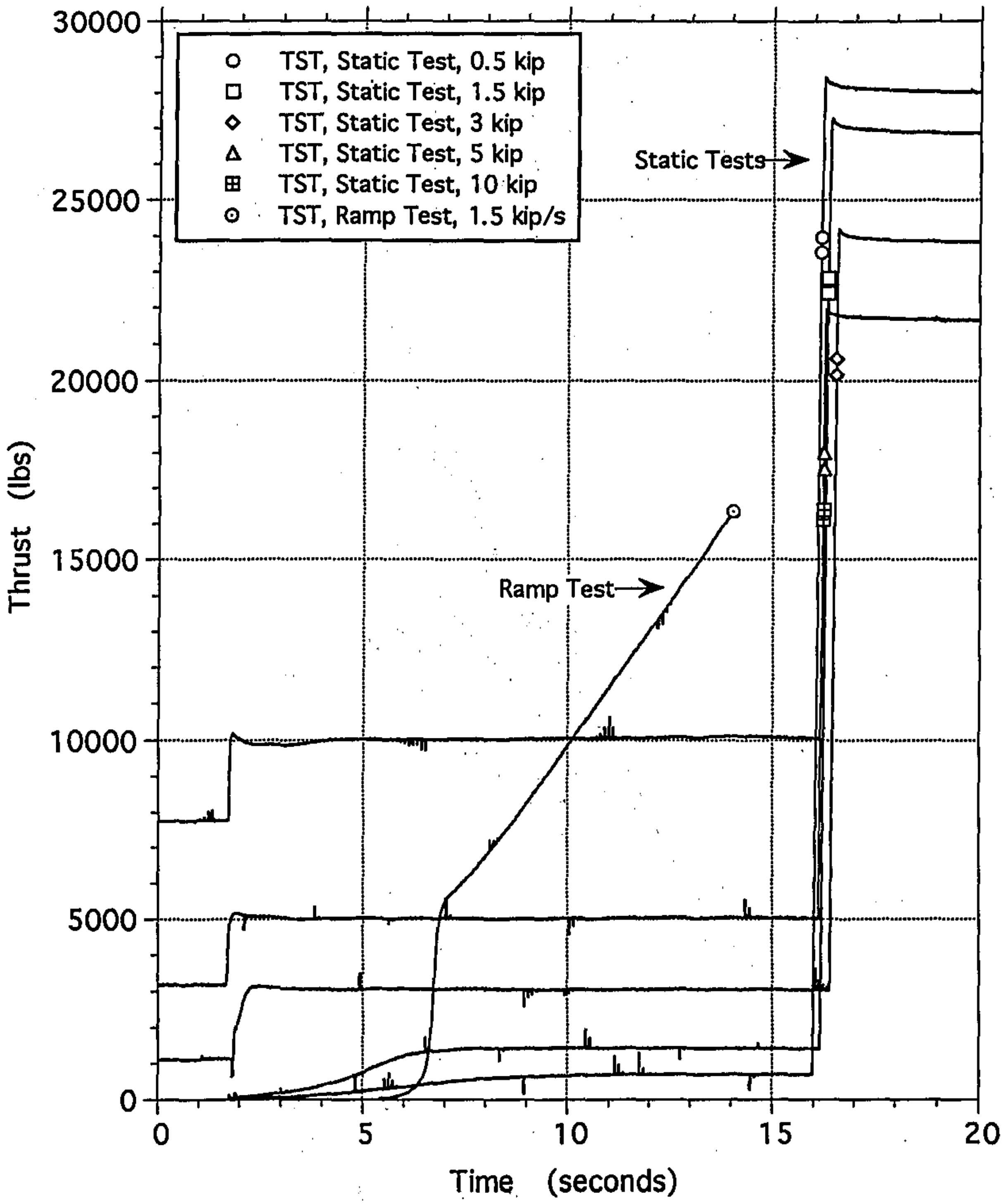

Figure 6. Load time histories showing effect of initial load—Stem h.

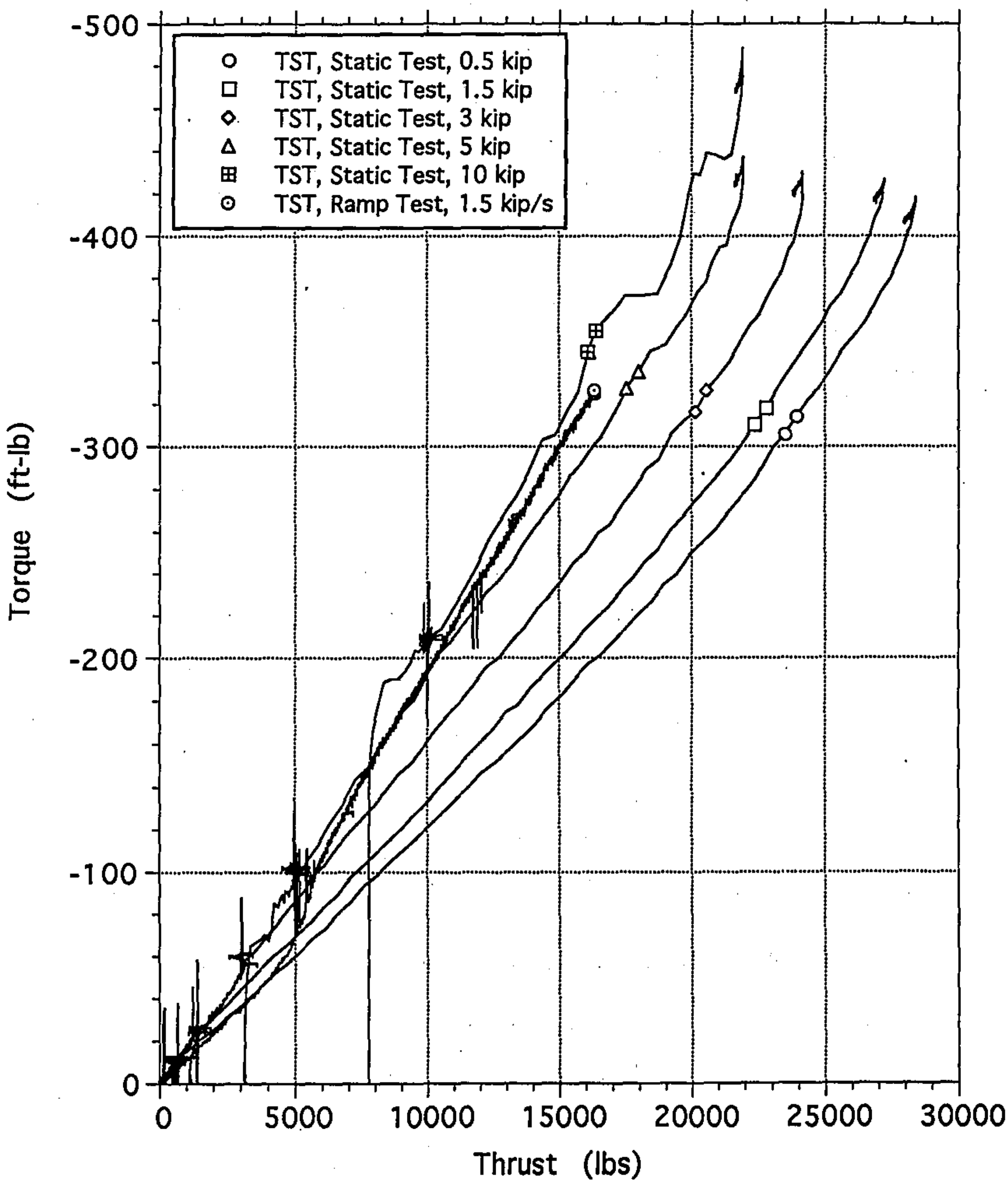

Figure 7. Torque X thrust showing effect of initial load—Stem h.

Rate of applied load is also an important factor. Figure 8 shows the results of several ramp load tests in which the ramp load rate was varied almost two orders of magnitude (from 100,000 to 1,600 lb/s). The same hardware described in Figures 6 and 7 was used for the tests shown in Figure 8. As the ramp rate is decreased from the maximum value obtained in a hard seat test, the thrust at torque switch trip decreases and stabilizes at the value achieved during the slowest ramp test. The majority of reduction in thrust occurs as a result of the first order of magnitude reduction in ramp rate (from 100,000 to 10,000 lb/s). The implication of this result is that the effect of prior load history has a limited duration. In this case, the duration appears to be a few seconds. As the ramp rate is made slower, such that the stem is being loaded over more than a few seconds, the stem-stem nut efficiency is unaffected. These load and rate-of-applied-load effects vary with different stem and stem nut combinations. For example, the effects just discussed for Stem h were not observed with Stem f, which was found to be insensitive to dynamic loads effects.

Torque Effects

Changes in torque output at torque switch trip have been observed that affect the output thrust and the corresponding ROL. For example, Figure 7 shows that, although the thrust at torque switch trip is decreasing as the load prior to hard seat is increasing, the torque shows the reverse trend, that is, torque at torque switch trip is increasing for the higher loaded tests. This torque increase partially offsets the stem factor change that is producing the ROL effect.

The observed torque increases were generally less than 10%. The torque effect results from changes in the performance of the load transmission path inside the actuator between the spring pack and the stem nut. The load path includes the worm and its supports, the worm gear and its supports, and the spring pack cartridge. The most likely source of the effect is friction either at the spline connection between the worm and its driving shaft or at the spring pack thrust washers and bore in the operator housing. As mentioned previously, the worm displaces axially on its shaft as actuator output torque increases. Friction in the worm spline connection resists worm movement. This friction adds to the resistance provided by the spring pack. If this friction force increases (for example, if the actuator is operated at high load prior to seating), there is a higher total force resisting worm motion at a given value of spring pack displacement, and the actuator output torque will be larger. Friction at the spring pack thrust washer/housing bore interface will cause a similar resistance to spring pack displacement.

CAUSE OF RATE OF LOADING EFFECT

Ratios of 1.0 to approximately 1.45 between thrust at TST for static and ramp tests were observed in the EPRI testing. This range is consistent with that reported from other laboratory tests (Steele et al., 1992) and utility in-plant tests (Black, 1990a and 1990b). The testing has confirmed the observations made by others (Steele et al., 1992) that the primary factor contributing to ROL effects is a change in the coefficient of friction at stem-stem nut interface. Specifically, Steele et al. postulated that, under high loading conditions, more lubricant would be squeezed out of the stem-stem nut interface resulting in higher friction coefficients. The EPRI testing has provided additional insights into this effect.

The friction in a grease-lubricated stem-stem nut interface is affected by the lubrication regime. Boundary lubrication is dictated by the strength of chemisorbed films (e.g., oxides formed from extreme pressure additives). The films provide lubrication by limiting metal-to-metal contact at the asperities between the sliding surfaces. For a typical stem-stem nut combination under constant or slowly changing load, boundary lubrication dominates frictional performance and the coefficient of friction is observed to range from approximately 0.1 to 0.15. In the hydrodynamic lubrication regime, the surfaces are separated by a pressurized film of lubricant and the coefficient of friction is much lower than for boundary

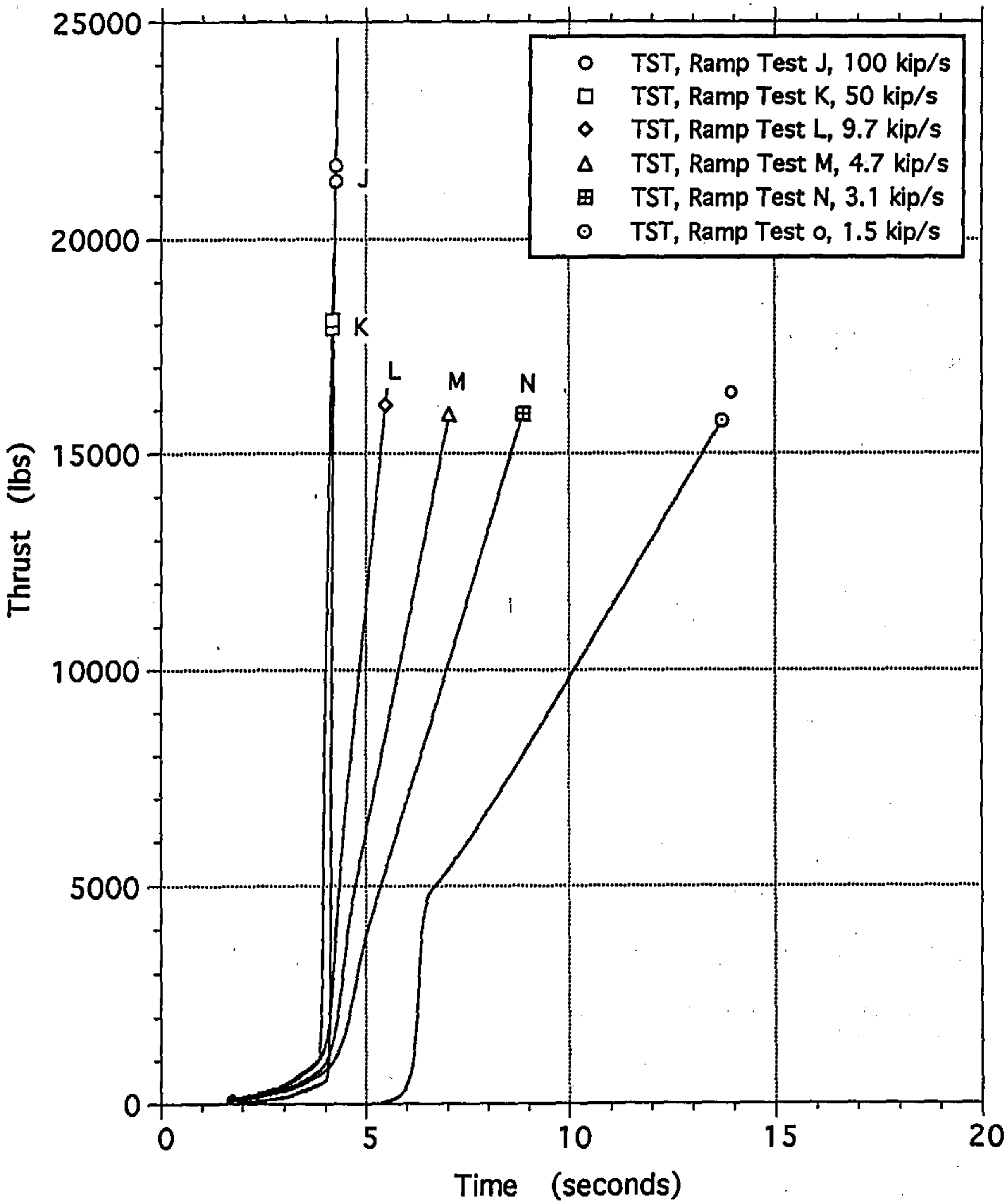

Figure 8. Load time histories showing effect of ramp rate—Stem h.

lubrication (e.g., less than 0.01). Stem thread surface velocities are too slow to obtain steady hydrodynamic lubrication. However, Hamrock (1991) notes that "it is often difficult to eliminate fluid film lubrication effects so that true boundary lubrication can occur, and there is evidence to suggest that micro-fluid lubrication formed by surface irregularities is an important effect." This effect appears to be occurring at the stem-stem nut interface during rapid load changes from low initial load.

As the load between the two thread surfaces is increased, compression of the contacting asperities occurs and the two surfaces move close together. It is possible that there is insufficient time for the lubricant to flow away from the contact areas. Temporarily, the stem threads can be partially supported on a pressurized film of lubricant until the lubricant flows away from the interface. During this time, lubrication is a mixture of boundary and hydrodynamic regimes and the effective coefficient of friction is significantly lower than for pure boundary lubrication. When the lubricant flows away from the interface, a return to boundary lubrication occurs. Depending on the properties of the stem-stem nut surfaces, the lubricant, and the load prior to seating, this transient may last up to the full seating duration (several hundreds milliseconds). By contrast, if the load increases slowly, as in a ramp test, the squeeze film effect does not occur and the coefficient of friction remains typical of boundary lubrication. Figure 9 demonstrates this effect. The friction coefficient is observed to decrease very quickly upon load application and then increase as the lubricant is squeezed out and boundary lubrication is reestablished. A fairly constant and higher friction coefficient is observed for the ramp test.

The basic squeeze film phenomenon also explains the effect of running load on ROL. When seating occurs preceded by a low running load, the amount of deformation that must occur at asperities of the thread interface is relatively large. Consequently, a large amount of lubricant must be displaced and the resulting squeeze film transient (and corresponding ROL effects) can be long. Conversely, a high running load minimizes the amount of deformation which occurs during seating and minimizes the squeeze film effect and ROL.

Further evidence of the squeeze film mechanism was obtained in another test. The grease used in the tests described in Figure 9 had its thickener removed. The remaining base oil with additives was diluted to obtain a low viscosity lubricant, but with similar chemical properties to the original grease. Results of tests with this lubricant are shown in Figure 10. The low viscosity of the lubricant increased its pumpability and drastically shortened the period of the squeeze-film transient. The thrust at torque switch trip was about the same for both a ramp load test and a hard seat test, and ROL was essentially reduced to zero.

As described above, not all stem-stem nut combinations appear to be sensitive to loading history. The specific surface properties which maximize or minimize this effect have not been identified.

QUANTIFICATION OF MOV SPECIFIC ROL EFFECTS

Several alternative approaches are being assessed to provide utilities with a means for accommodating potential "rate-of-loading" effects. These methods are summarized as follows:

- Impose a margin penalty if no valve-specific data are available.
- Set the torque switch with a reduced loading rate (i.e., by use of the handwheel). This method requires thrust measurement, but torque measurement is not required. While some margin penalty is required with this approach, the magnitude of the margin penalty can be minimized if torque measurements are also made.
- Set the torque switch with a DP load simulator device. If successful, this device will accurately reproduce the maximum coefficient of friction that could occur at the stem-stem nut interface under design-basis DP

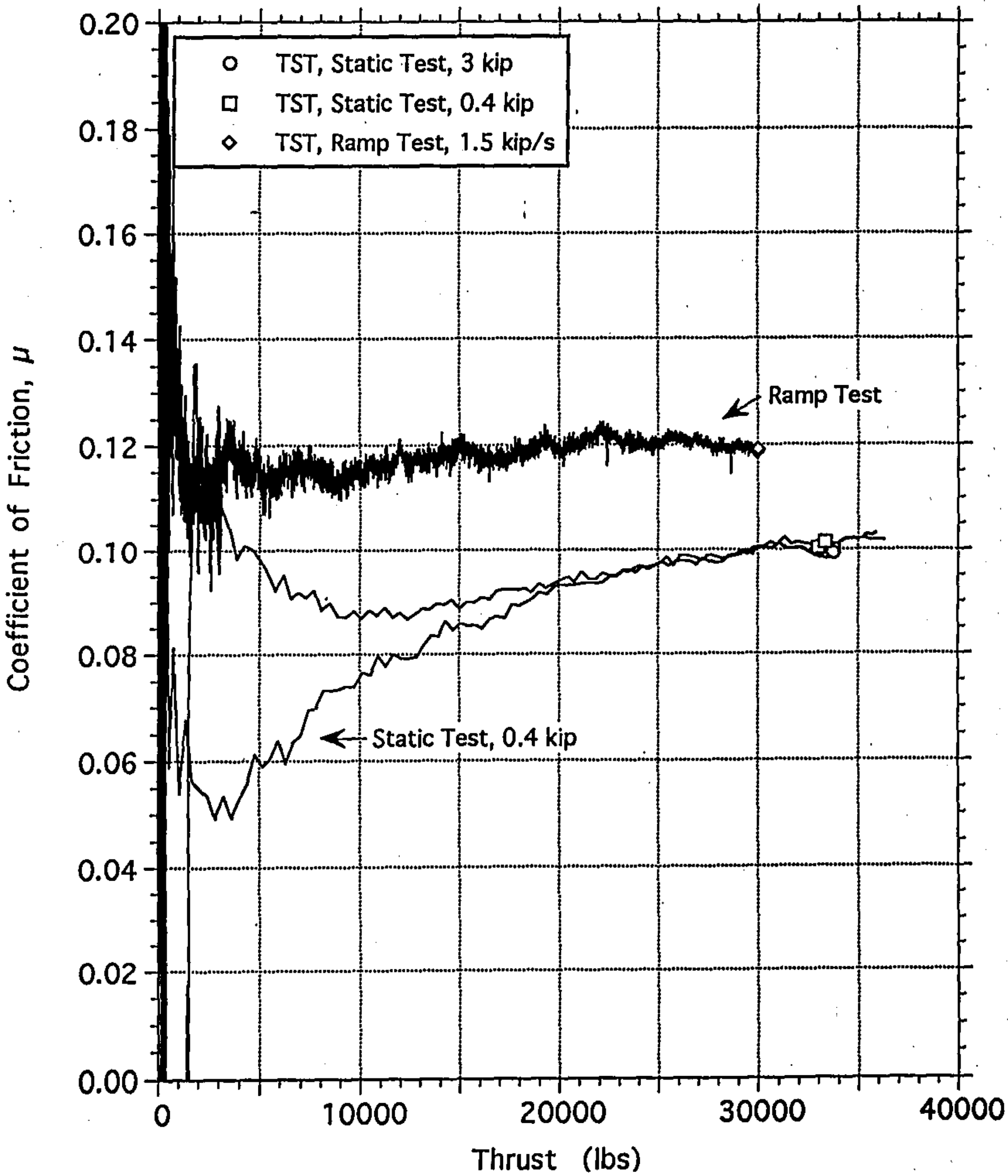

Figure 9. Variations of coefficient of friction during stem loading—Stem h.

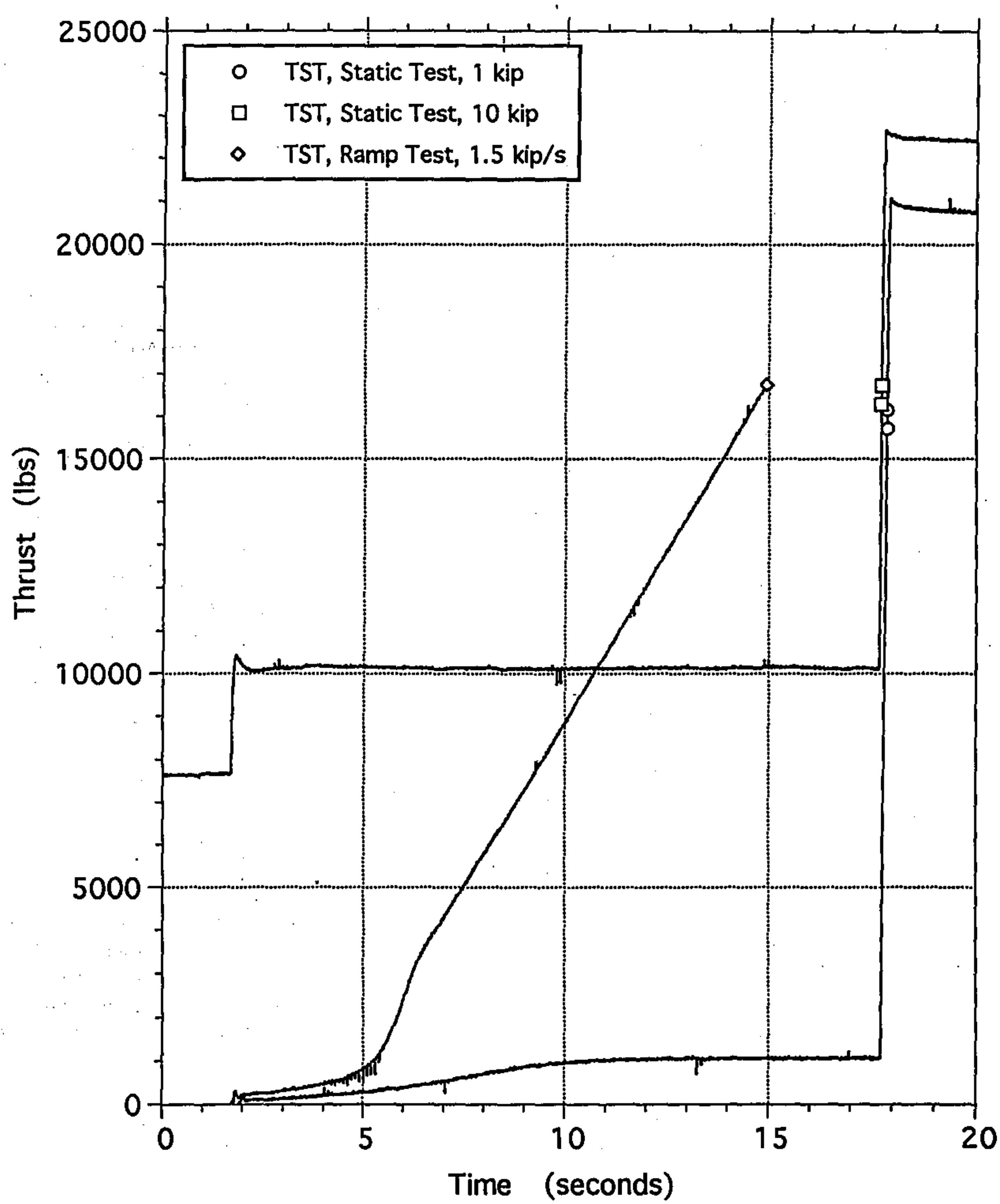

Figure 10. Load time histories showing effect of reduced oil viscosity—Stem h.

loading conditions. This method requires only a thrust measurement and minimal margin penalties.

- Use one of two approaches that have been suggested by the Idaho National Engineering Laboratory for further investigation by industry. These methods are deemed the "threshold" method and the "fold line" method. These methods require the measurement of both thrust and torque and involve testing under static or relatively low DP conditions.

- Modify the control switch logic to bypass the torque switch until flow isolation, but not necessarily leak tightness, is achieved. The torque switch would be set at a nominal setting so as not to impose excessive thrust loading during static tests. This approach would eliminate the need to add margin to accommodate potential "rate-of-loading" effects, but would still require evaluation of actuator capability to achieve the required thrust while the torque switch is bypassed.

SUMMARY

The tests carried out as part of the EPRI MOV Performance Prediction Program have provided the following principal results:

- Sensitivity to the ROL effect varies from stem to stem. The ratios between output thrust at TST for static tests and for ramp tests have ranged from 1.0 to approximately 1.45 for the 12 stems tested.

- Differences in output thrust can be directly correlated to changes in efficiency at the stem-stem nut interface. Increases in stem-stem nut efficiency occur under conditions of rapid loading from low initial load resulting in higher thrust.

- Higher torques at TST have been observed for tests performed at high initial loads. Such differences in torque at TST can partially offset the reductions in joint efficiency

- The results indicate that a squeeze-film phenomenon at the stem-stem nut interface appears to be responsible for the increases in the stem-stem nut efficiency. The physical stem characteristics that would explain the differences in sensitivity to ROL effect between stems have not been identified.

- Given that the ROL magnitude cannot be readily predicted, other means are needed to account for the effect in MOV evaluations. Testing is currently underway at Battelle to assess the effectiveness of a number of simple tests that could be conducted in situ to establish the magnitude of ROL for a specific MOV.

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Session 2B

Pump Performance and Testing

Session Chair
Don Cavi
Sargent & Lundy

Preliminary Assessment of Pump IST Effectiveness[a]

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Brookhaven National Laboratory

ABSTRACT

A preliminary review of inservice testing (IST) effectiveness for Class 1, 2, and 3 pumps at nuclear power plants was performed. IST requirements are specified by the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section XI, and the Operations and Maintenance Standard (OM Part 6). The Institute of Nuclear Power Operations Nuclear Plant Reliability Data System was used to provide failure reports for these components from 1988 to 1992. This time frame coincides with the issuance of Generic Letter 89-04, which resulted in a more consistent application of the requirements by the licensees.

For this time, 2,585 pump failures were reported. A review of these failures indicated that the majority (71.6%) resulted from external leakage. These events were excluded from the study because they typically do not affect pump operability and are not detected by the measurement of IST parameters. The remaining 733 events were reviewed to identify the primary failure causes, failure modes, and method of detection. Plant testing programs, consisting of IST, surveillance testing, and special testing, detected approximately 40% of these occurrences. Others were detected through operational abnormalities, routine and incidental observations, alarms, and while performing maintenance. This paper discusses the results of the study.

INTRODUCTION

Under the requirements specified in 10 CFR 50.55a(f), all operating nuclear power plants are required to develop and maintain a formal inservice testing (IST) program. These programs are designed to ensure the operational readiness of Code Class 1, 2, and 3 pumps and valves. Specific testing requirements and acceptance criteria are defined by Section XI of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code. 10 CFR 50.55a(b) provides the latest editions and addenda of Section XI approved for use. Licensees are required to update their IST programs every 10 years to the edition and addenda referenced in Paragraph (b). Currently, Section XI editions through 1989 are referenced in Paragraph (b).

Section XI first introduced pump and valve testing requirements in the 1971 Edition, Summer 1973 Addenda. The 1988 Addenda of Section XI omitted specific requirements and specified that the rules for pump and valve IST shall meet the requirements set forth in ASME Operations and Maintenance Standards Part 6 (OM-6), "Inservice Testing of Pumps in Light-Water Reactor Power Plants," and Part 10 (OM-10), "Inservice Testing of Valves in Light-Water Reactor Power Plants." In 1990, the ASME issued the OM Code which was written to replace the pump and valve requirements contained in Section XI. The U.S. Nuclear Regulatory Commission (USNRC) staff

a. Work performed under the auspices of the U.S. Nuclear Regulatory Commission.

is currently evaluating the OM Code for inclusion into the regulations, 10 CFR 50.55a(b).

The USNRC has also provided additional guidance to licensees regarding compliance with specific IST requirements in Generic Letter 89-04 (GL 89-04) and, very recently, in draft NUREG-1482.

Licensees prepare specific IST programs for their plants and update the program every 10 years. This program provides the specific testing requirements and test frequency for each pump and valve included in the program.

Brookhaven National Laboratory (BNL), under contract with the USNRC, is conducting a review of the IST effectiveness at nuclear power plants. The results will be used to identify and recommend potential Code changes and revisions to improve existing IST programs in identifying component degradations before failure.

IST effectiveness may be defined and assessed in several ways. For example, the following issues may be evaluated in making the assessment:

- What type of pump and valve failures are occurring, and how many are being detected by IST? Are any of the failures that are found by other means potentially detectable by IST?
- Does the IST program identify component degradation before failure?
- Are safety-significant failures and degradations being identified?
- To what extent does the program duplicate other required testing programs [i.e., Appendix J leak testing, technical specification testing, post-indicator valve (PIV) testing, motor-operated valve (MOV) Generic Letter 89-10 (GL 89-10) testing]?
- Is the program cost effective?

For this initial, limited scope review, it was decided to review the pump and valve failure occurrences to determine the specific types of failures occurring and the degree at which the IST program is identifying these failures. For the study, the Nuclear Plant Reliability Data System (NPRDS), which is a computerized information retrieval system maintained by the Institute of Nuclear Power Operations (INPO) was used. The NPRDS contains specific information (failure mode, symptom, cause and system effect, and the method of detection) on component failures submitted by the nuclear utilities. Failures are reported to NPRDS when degradation of a component, part, or associated device has occurred and one function of the component (e.g., to produce the specified flow or differential pressure) has been lost or degraded, such that the performance criterion for at least one of the component's functions is not met. The performance criterion may be based on technical specification limits, ASME Code limits, or system design limits, for example. The NPRDS data also encompasses the events reported as Licensee Event Reports (LERs), as specified by 10 CFR 50.73.

All Safety Class 1, 2, and 3 pump and valve failure events from 1988 to 1992 contained in the NPRDS were reviewed for applicability to this study. As defined in NPRDS, the safety class of components is determined using American National Standards Institute/American Nuclear Society (ANSI/ANS) 51.1 (PWRs) or 52.1 (BWRs). Although the scope of the new OM Standards includes all safety-related pumps and valves, the current regulations require IST in accordance with Section XI only for ASME Code Class 1, 2, and 3 pumps and valves. NPRDS provides only the Safety Class and not the Code Class of components. Therefore, the scope of our study was limited to Safety Class 1, 2, and 3 components. There may be some discrepancy between Safety Class and ASME Code Class 1, 2, or 3, although this is not expected to be significant. Many plants include all safety-related pumps and valves in their IST programs. The 5-year time frame was chosen to coincide with the issuance of GL 89-04, which provided specific, detailed instructions to licensees regarding

IST. GL 89-10 has resulted in a more consistent application of the Code requirements by the individual licensees. A secondary benefit resulting from this guidance has been an improved and more consistent reporting of operating pump failures to the NPRDS. This paper discusses the results of the evaluation of pump failures (Grove et al., 1993).

REVIEW OF CODE REQUIREMENTS FOR PUMPS

Section XI, Subsection IWP of the ASME Code, OM-6 and the OM Code, Subsection ISTB define the rules and requirements for the preservice and inservice testing of Class 1, 2, and 3 centrifugal and positive displacement type pumps to assess their operational readiness. The test quantities that must be measured and compared with reference values to detect degradation include pump speed, differential pressure, flow rate, and vibration amplitude (measured as either displacement or velocity). The test frequency, acceptance criteria, corrective action, and records requirements are also specified. Also included are rules and requirements on defining acceptable instrumentation, including accuracy, range, calibration, and instrument location.

As discussed in 10 CFR 50.55a(f)(6)(i), when testing in accordance with the Code is impractical, relief may be requested. Additionally, licensees may propose alternatives to the Code, as allowed by 10 CFR 50.55a(a)(3). The following are requirements of the Code from which licensees commonly ask for relief:

- Pump vibration frequency range requirements for low-speed pumps (e.g., pumps that would require the instrument range to be less than 10 Hz) [Section XI, TIWP-4520(b)]

- Pump vibration acceptance criteria for pumps with normally high vibration levels (Section XI, TIWP-3210)

- Pump inlet pressure measurement for pumps that take suction from a bay or tank (Section XI, TIWP-3100)

- Test instrument ranges (Section XI, TIWP-4120)

- Quarterly measurement of flow rate for pumps without an adequate test loop [e.g., containment spray pumps in pressurized water reactors (PWRs)] or normal plant operation does not allow the required test configuration [e.g., service water pumps Section XI, TIWP-3400(a)]

- Use of reference values for pumps with variable system resistance based on demand loads [e.g., component cooling water pumps (Section XI, TIWP-3110)].

Additionally, some licensees have requested to measure vibration velocity in root mean square (rms) rather than peak, as required by the Code, and to use specific provisions of OMa-1988 Part 6.

OPERATING DATA REVIEW

As discussed in the Introduction, the NPRDS database was used to review the pump operating failures that occurred between 1988 and 1992. Pump failure records were reviewed for specific failure modes, effects, and detection methods to provide an overview of the failures and to assess the effectiveness of the IST in detecting these failures. Failures related to the pump driver (e.g., motor or turbine) were not evaluated because IST does not directly assess these components.

For the 1988 to 1992 time period, 2,585 failures affecting Class 1, 2, and 3 pumps (Figure 1) were reported. Of these, 1,852 failures (71.6%) resulted from external leakage (i.e., packing failures). Typically, external leakage failures do not affect the pump operability, and are not detected by the required IST test parameters. Therefore, they were excluded from our study. The remaining 733 pump failure events were used for the evaluation.

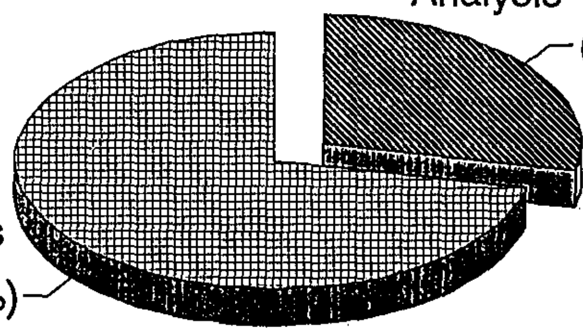

Figure 1. Class 1, 2, and 3 pump failures (1988–1992).

Failure Causes

The six most frequently occurring failure causes for the pump failures that did not result from external leakage are shown in Figure 2. These included

- Normal/abnormal wear (NPRDS Code AD)
- Previous repair (NPRDS Code AM)
- Mechanical damage/binding (NPRDS Code BB)
- Out of mechanical adjustment (NPRDS Code BC)
- Aging (NPRDS Code BD)
- Blocked or obstructed flow path (NPRDS Code BF).

These six failure causes accounted for 559 (76%) of the 773 pump failures. An additional 22 failure causes contributed to <3.7% each to the total, and were considered isolated occurrences and were not evaluated, based upon the programs scope. Pump wear [normal and abnormal (44%)], mechanical damage/binding (13%), and mechanical adjustment problems (7%) were the three most common failure causes. The system effect of these failure causes is also shown in Figure 2 and discussed later in this paper.

A review of the six most frequent failure causes revealed that each can be detected by IST, as shown on Table 1. The time-dependent failure causes (i.e., aging, wear, and mechanical degradation) should be detectable by trending specific plant operating parameters.

Failures from previous component repairs are also addressed by both Section XI and the OM Code. Both Codes specifically require pump operation and the re-establishment of reference values following maintenance, repairs, or replacement. Still, 40 (7%) pump failures were attributed to previous repairs. The need for adequate post-maintenance testing is essential to assure the proper condition of the component after maintenance. Human-related problems from maintenance (i.e., improper installation, wrong parts) can be discovered at this time. Improper maintenance, if not detected, may accelerate aging degradation or result in failure. A recent review of insights obtained from a review of USNRC Maintenance Team Inspection reports also concluded that post-maintenance testing required improvement at many plants (Fresco et al., 1993). This fact, as well as the 40 failures attributed to previous maintenance, highlights the need for thorough post-maintenance testing.

Figure 3 shows the actual method of detection for each of the failure causes. The majority of the failures were detected by surveillance tests (32%), followed by routine observation (26%), and operational abnormalities (15%). As reported by the licensees to NPRDS, only 26 (5%) failures

were detected by IST. However, a review of a sampling of the failures detected by surveillance testing indicate that many were also probably IST related. Plant technical specifications require IST in accordance with Section XI; therefore, utilities may input IST detected failures as surveillance testing. Because the scope of the study did not allow for a detailed review of each individual failure, it was decided that for purposes of this evaluation, no distinction would be made between the three testing methods (IST, surveillance, and special testing) included in the NPRDS database. This assumption is conservative because the surveillance tests (i.e., Technical Specification Tests) require system testing in addition to Section XI component testing. These three testing and inspection methods accounted for 222 (40%) of the failure causes. The remaining 60% were detected by operational abnormalities, maintenance, audio-visual alarm, routine or incidental observations, or corrective maintenance.

The number of failure occurrences detected through testing for the six most frequent failure causes is shown in Table 2. Of these six failure causes, the most critical are those which would result in the inoperability of a pump that could have affected plant operation and safety, namely mechanical damage/binding and blocked/obstructed failures. Specific examples of failures attributed to these causes included bearing failure and cracked shaft. As discussed previously, failures caused by previous repairs also should have been detected by IST. A sampling of the 25 failures attributed to this cause, not detected by testing, includes packing adjustment, missing parts, air binding, bearing installed 180 degrees out, and wrong bearings installed. Most pump failures were attributed to normal wear and aging, the majority of which were not detected by testing.

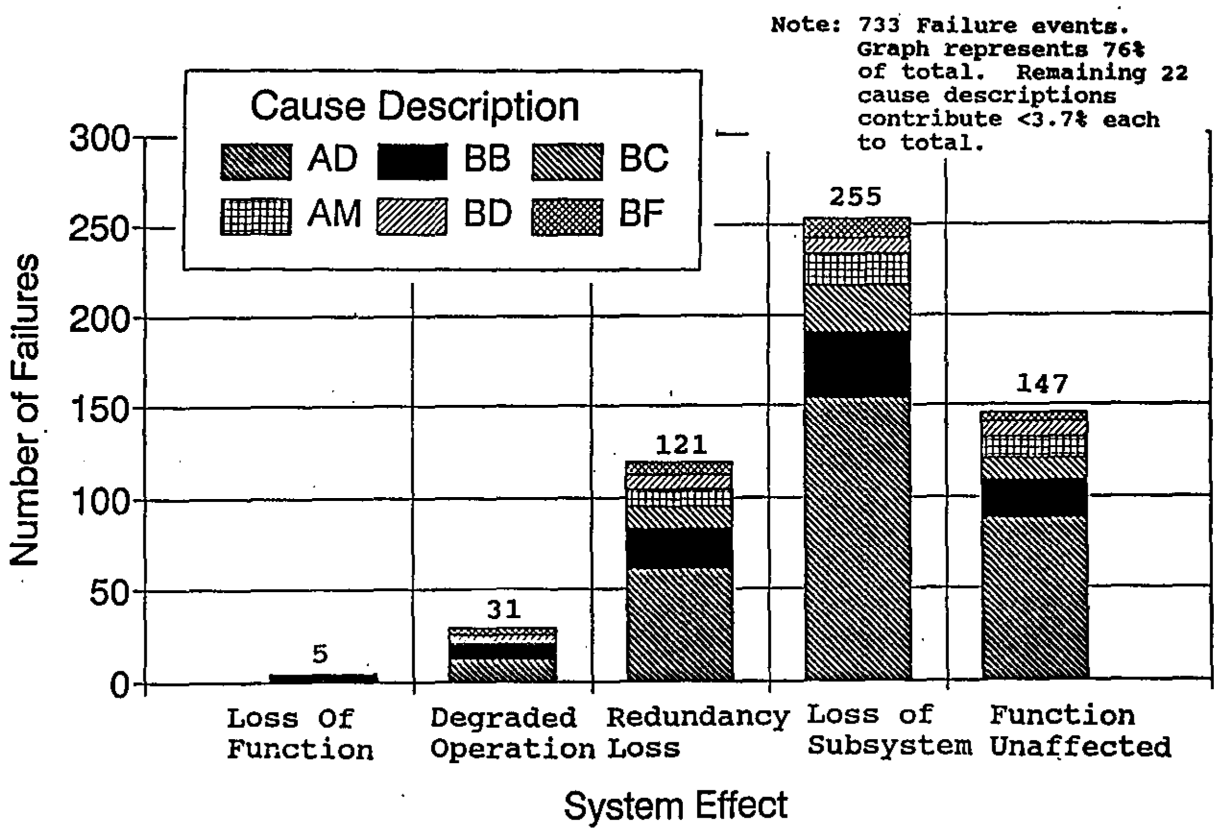

Figure 2. Pump failure cause versus system effect.

Table 1. Failure cause detectable by current IST requirements.

	Potentially detectable by IST				
Failure cause	Pump speed	Differential pressure	Flow rate	Vibration	Post-maintenance testing
Normal/abnormal wear[a]	Y	Y	Y	Y	NA
Caused by previous repair[b]	Y	Y	Y	Y	Y
Mechanical damage/binding	Y	Y	Y	Y	NA
Out of mechanical adjustment	N	N	N	Y	NA
Aging[c]	Y	Y	Y	Y	NA
Blocked or obstructed flow rate	Y	Y	Y	Y	NA
External leakage	N	N	N	N	NA
Others[d]	—	—	—	—	—

a. Depending on wear part (e.g., impeller, bearing).

b. Depending on extent of repair.

c. Depending on what part is subject to aging. (Note that aging is often used as a "catchall" failure cause.)

d. Other causes (dirt, lack of lubrication, particulate contamination, electrical, human-related) were not evaluated because of their limited frequency.

A review of the normal wear and aging failures indicate that mechanical wear of the rotating assemblies and bearing failures were the dominate failure causes. The majority of these failures were preceded by an increase in temperature and vibration. Mechanical vibration is a parameter specifically addressed by the Code. Interpretation of the data to determine the cause for increased vibration levels is often difficult. Increases in vibration could be caused by many factors (e.g., upstream flow cavitation and building structure vibration, as well as specific pump problems), and if the pump is not disassembled and inspected, the exact cause may not be readily apparent. If the levels are not large enough to exceed the Code required action or alert levels, the pump may be returned to service, regardless of the trend. Given the frequency of pump failures which experienced increased vibrations, a change to the Code may be warranted to require that an engineering evaluation be done if these trends are observed. These evaluations could be done without removing the pump from service, and if the cause is isolated, it could be fixed prior to pump failure. The OM Code Committee is considering a change that would allow the use of vibration spectrum analysis (which requires analysis of trends), in lieu of doubling the test frequency, when vibration deviations fall within the alert range. It should be noted that most plants, in addition to the Code required vibration measurements, perform spectral analysis as preventive maintenance and trend the results. As demonstrated by the two utilities recently visited by BNL to review the IST programs, maintenance vibration programs can be effective in identifying degraded conditions. The Code committees and USNRC are increasingly relying on the use of vibration to detect degradation over hydraulic monitoring, as evidenced by the changes made in

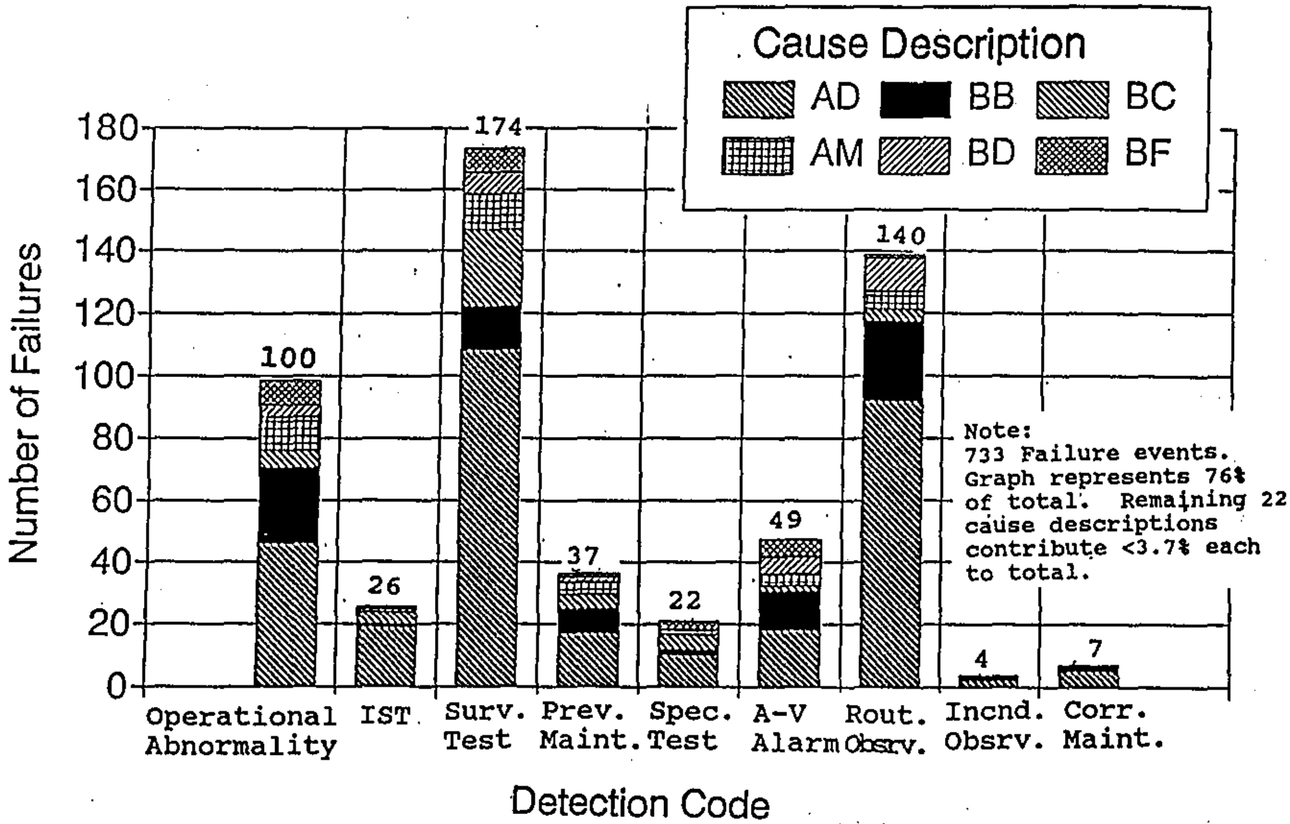

Figure 3. Pump failure cause versus detection method.

OMa-1988, Part 6 (Zudans, 1989) and Generic Letter 89-04, Position 9. Based upon this reliance and the art of interpreting vibration data, increased controls on personnel training and qualifications may be warranted.

Unlike vibration, bearing temperature monitoring is no longer required by the Code. Previously, temperature was required to be monitored yearly. The primary reason for deleting this requirement was that the ASME committees felt it was unlikely that a yearly test would coincide with the failure, which is the only time an increase in temperature would be expected. Only if the temperature is continuously monitored could impending pump bearing failure be detected. However, given the amount of failures caused by increased temperatures, it may be warranted to perform a further detailed review of these failures. Considering the potential economic and safety ramifications that could result from the failure of certain risk-significant, continuously running pumps, a continuous temperature monitoring system may be desirable.

Failures resulting from blocked or obstructed flow paths are an example of events that may not always be detectable by IST. Several failures were noted to result from foreign objects (e.g., wood) obstructing pump intakes and damaging impellers. These occurrences happen quickly, and chances are IST testing would not coincide with these occurrences. This is not to say that all of these failure causes are undetectable. Blockage from the buildup of sand over time at the service water intake would be potentially detectable through a gradual decrease in output flow and differential pressure. The trending program for specific pumps should be reviewed to ensure that changes in these parameters are investigated before failure. The OM committees are presently considering Code changes to address pump trending.

Table 2. Pump failure cause detection.

Failure cause description	Detected during testing	Not detected during testing
Normal/abnormal wear (NPRDS Code AD)	137 (43%)	181 (57%)
Caused by previous repair (NPRDS Code AM)	15 (38%)	25 (62%)
Mechanical damage/binding (NPRDS Code BB)	17 (19%)	74 (81%)
Out of mechanical adjustment (NPRDS Code BD)	34 (65%)	18 (35%)
Aging/cyclic fatigue (NPRDS Code BO)	8 (26%)	23 (74%)
Blocked/obstructed (NPRDS Code BF)	11 (41%)	16 (59%)

Failure Symptoms

In addition to ensuring that the most common failure causes were addressed by Code requirements, a review of the failure symptoms is also important. These symptoms are the first indications of an actual or imminent pump failure. The five failure symptoms for the failures reviewed included

- Physical fault—Failure because of a changed physical condition or configuration (NPRDS Code A)
- Out-of-specification—Failure characterized by pump operation outside of permissible ranges (NPRDS Code B)
- Demand fault—Failure of pump to operate upon demand (NPRDS Code C)
- Abnormal characteristic—Pump failure characterized by a response which is not normal or anticipated (NPRDS Code D)
- Contained leakage—Leakage of the pumped fluid along the system flow path (NPRDS Code F).

The majority of the pump failures exhibited out-of-specification or abnormal parameters (Taylor, 1990).

Figure 5 shows the method of detection for these failure symptoms. Of the 733 pump failures not caused by external leakage, 40% (293) were detected through the testing programs. A particular concern was the 62 pumps that failed on demand. It is essential that a pump be available to perform the designed safety function when needed, particularly those that are normally in standby. A review of these events showed that 41 failures affected operating pumps, and 21 failures affected standby pumps. Of the 21 instances that affected the standby pumps, 12 were detected through testing. Of the 9 occurrences, 7 represent events of standby components, which would not have operated upon demand if required, and were found by chance. Though these types of events were not frequently observed in the data, they highlight the importance of IST for detecting degradations that could affect the operability of standby pumps.

Failure Modes

A review of the particular failure mode for pumps, which failed from causes other than external leakage, was also performed to determine if the IST requirements were effective in identifying each. The failure modes describe how a problem or deficiency affected the function of a component when the failure was discovered. For pumps, the NPRDS database identified four specific modes:

- Failure to start upon demand (NPRDS Code FS)
- Failure to continue running (NPRDS Code FR)

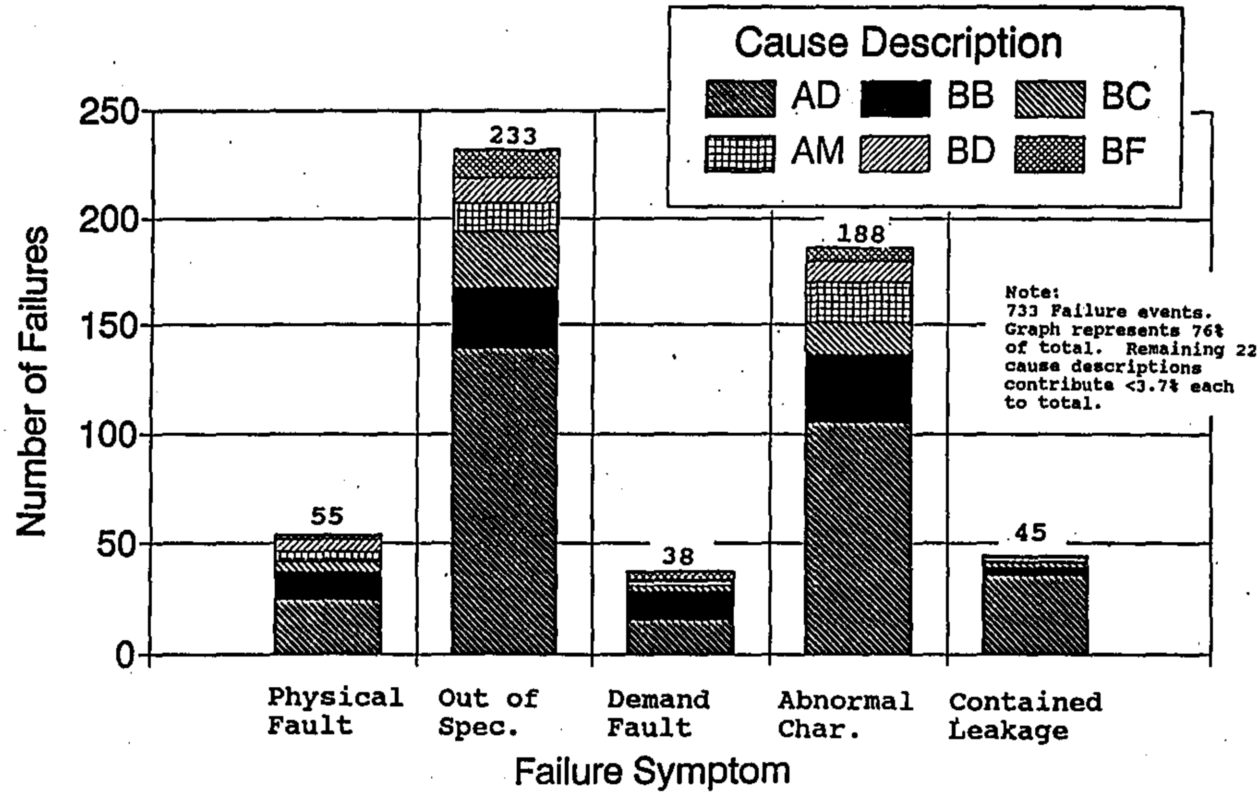

Figure 4. Pump failure symptom versus failure cause.

- Other incipient failures found during inspection, surveillance, testing, or maintenance (NPRDS Code MO)
- Failures unable to classify (NPRDS Code UA).

Figures 6 and 7 show failure modes as a function of both failure cause and symptom, respectively. Failure to start on demand was the predominant failure mode for standby pumps, while pumps that failed to run exhibited abnormal and out-of-specification pump characteristics. Pump wear was the main cause for both of these failure modes. Incipient failures were primarily from pump wear, and though they did not affect the starting or running capability, they did affect the ability of the pump to operate as designed. A sensitive trending of IST pump quantities would be required to detect these occurrences before actual pump failure.

Figure 8 shows the method of detection for these failure modes. Plant testing discovered 53% of the reported failures to start and 38% of the failures to run. Figure 9 shows the pump operating status for each failure mode. The majority of pump failures were related to operating pumps failing to continue running. Testing has not been successful at detecting these failures prior to occurrence. This highlights the importance of pump operation and performing quarterly testing, if possible. Though the plant effect is minimal, the loss of redundancy may be safety-significant for some systems. Licensees may consider rotating these alternate pumps into operation (if possible), since it appears that operation is the prime way to detect these failures and degradation. Again, it is essential that the trending program be sensitive enough to detect operating abnormalities before failures occur. The probability of detecting these failures, because of increased operating time, would be greater.

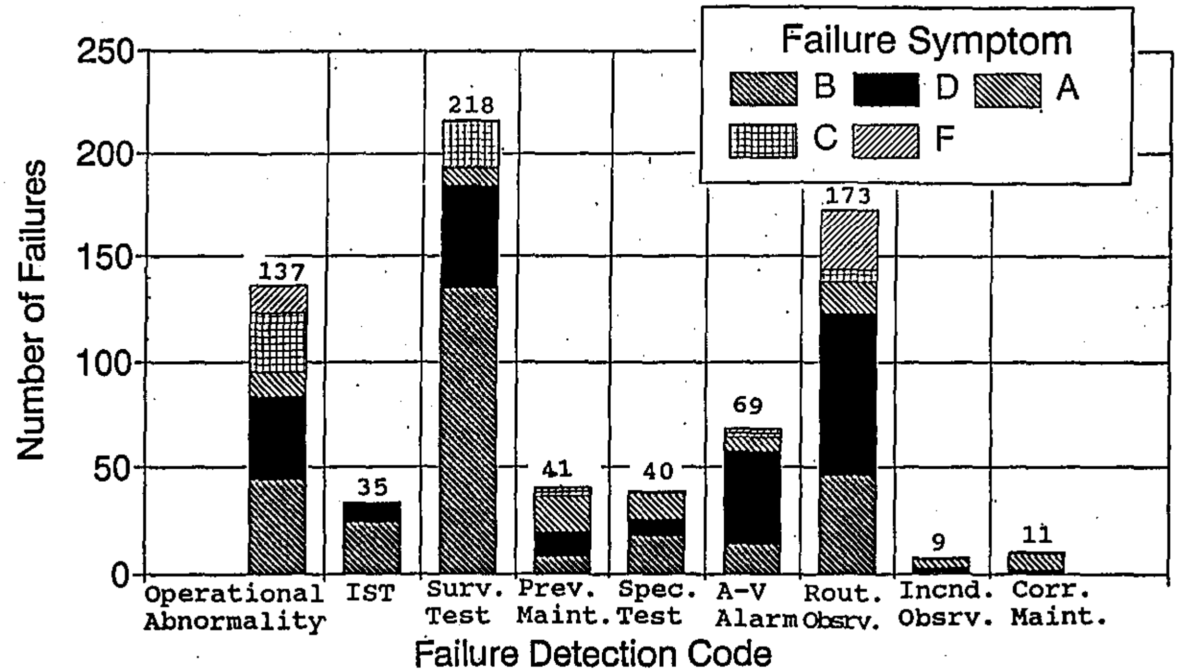

Figure 5. Pump failure symptoms versus detection method.

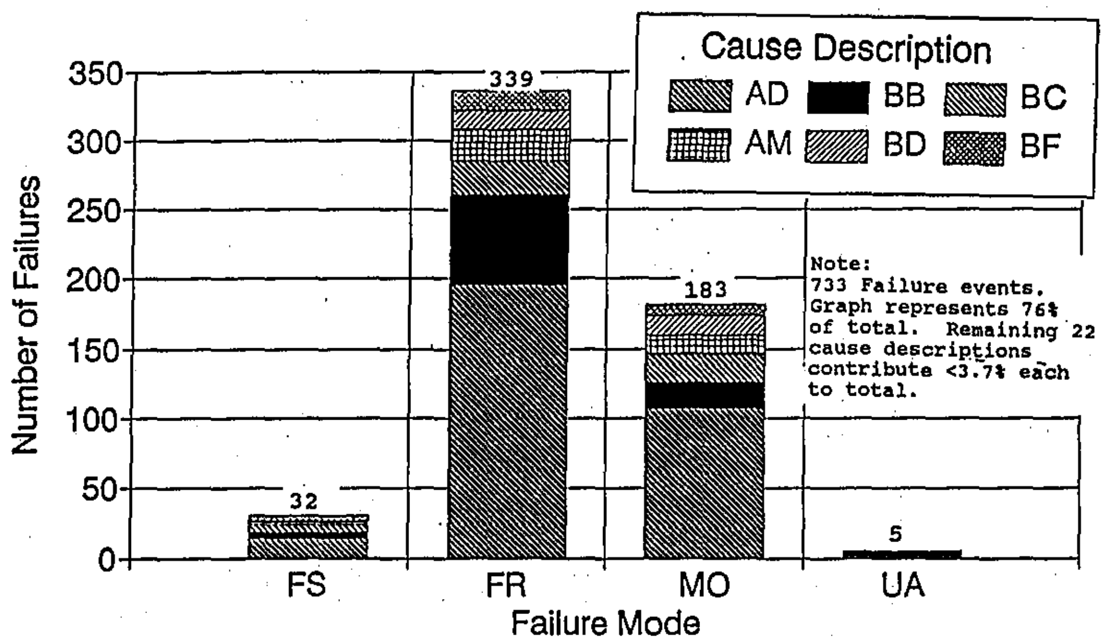

Figure 6. Pump failure mode versus failure cause.

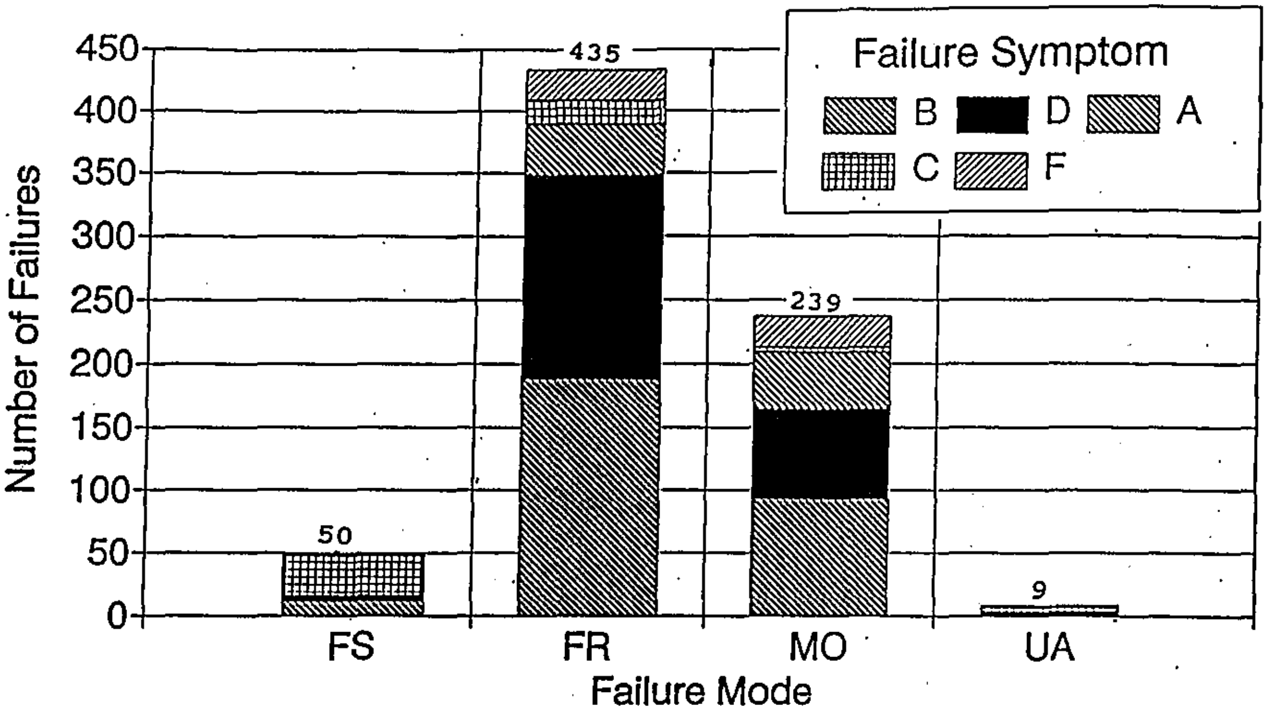

Figure 7. Pump failure mode versus failure symptom.

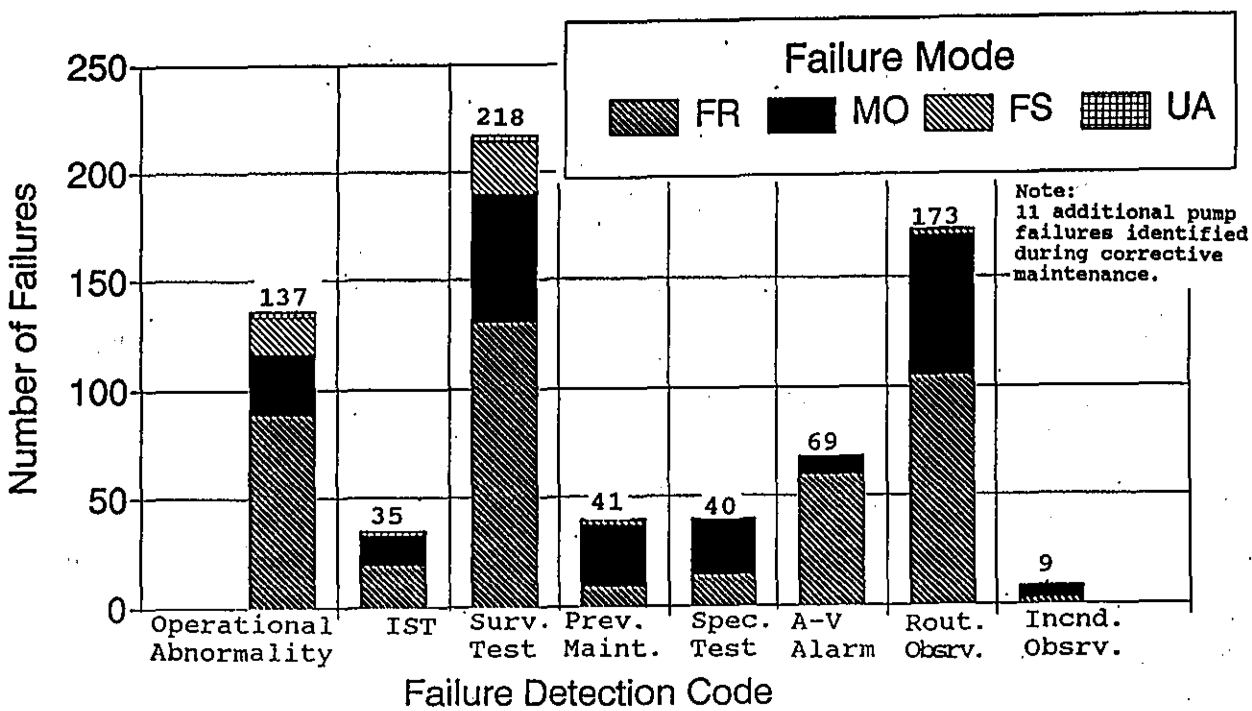

Figure 8. Pump failure mode versus detection method.

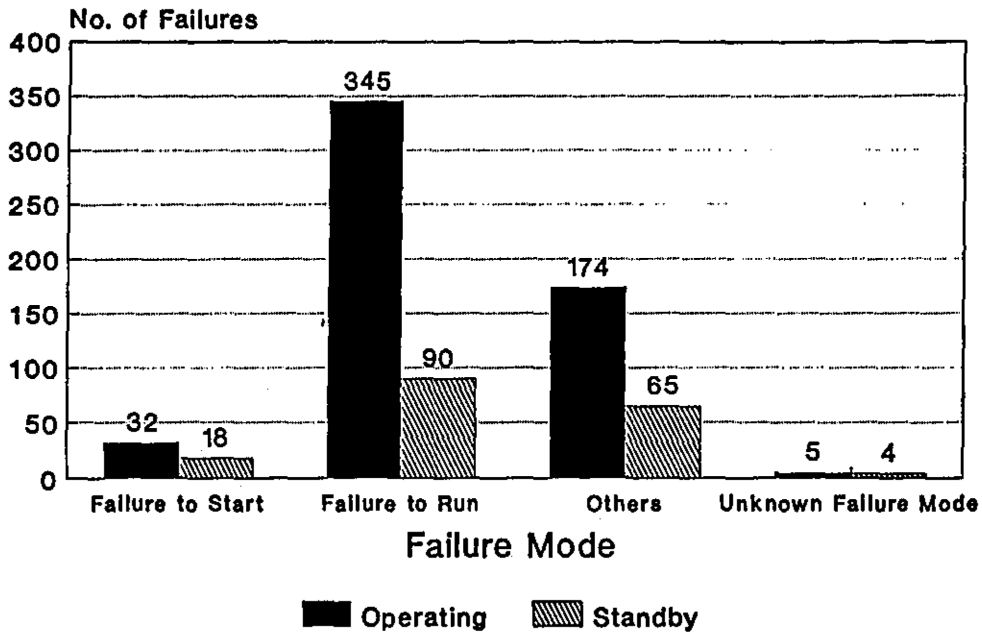

Figure 9. Plant operating status versus failure mode.

Failure Effects

The pump failure occurrences, that were not caused by external leakage affected over 40 different systems (in both BWR and PWR plants). Table 3 lists the systems most frequently affected by these failures. Service water (21%) and chemical and volume control (20%) systems were most frequently affected. The seven systems listed in Table 3 accounted for 65% of the reported failures. The remaining failures affecting other plant systems did not exceed 3% for any one system.

Table 3. Systems affected by pump failures.

System	Occurrence (%)
Service water	21
Chemical volume and control	20
Auxiliary feedwater	6
Reactor coolant	6
Control rod drive	5
Component cooling water	4
Reactor recirculation	3

A review of these failures was also made to determine if the pumps affected were in the operating or standby mode. The majority of the failures affected normally operating pumps (Figure 10). This distinction is important because it is essential that standby pumps, important to plant safety, be available to operate as designed upon demand, and that IST be used to detect and correct degrading conditions before failure during operation.

Typically, these pump failures had no significant effect on plant operations (Figure 11). Only 7% of the failures resulted in the plant being removed from service, reduced power, or a plant trip. This is not surprising, since many of the plant systems contain redundant pumps so as to minimize any disruption to power production. However, the effect on the individual systems was greater (Figure 12), as only 27% of the pump failures resulted in no system effect. Degraded system train (44%) and loss of redundancy [loss of one or more train functions (23%)] were the most frequent effects.

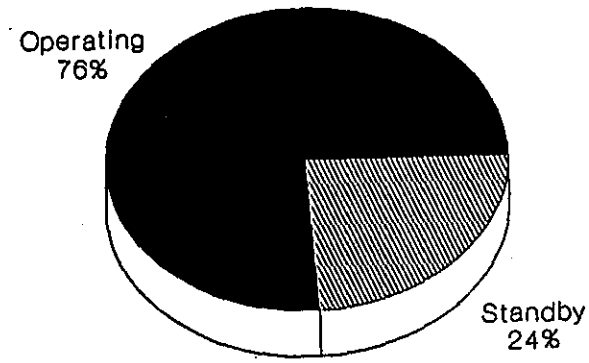

Figure 10. Pump operating status.

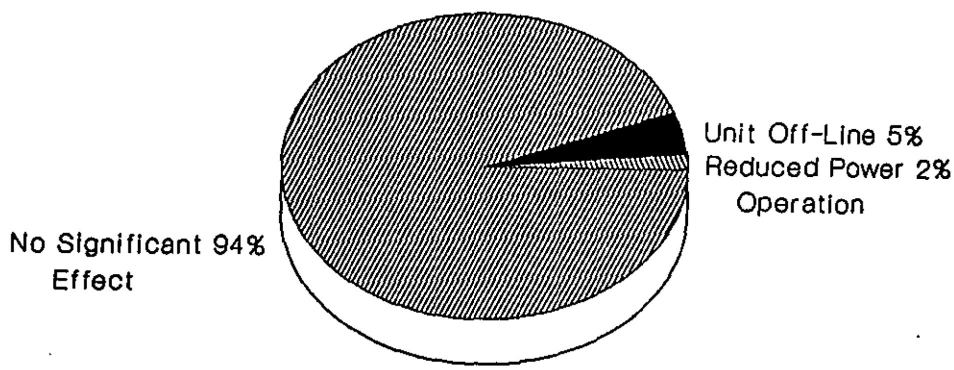

Figure 11. Plant effects from pump failures.

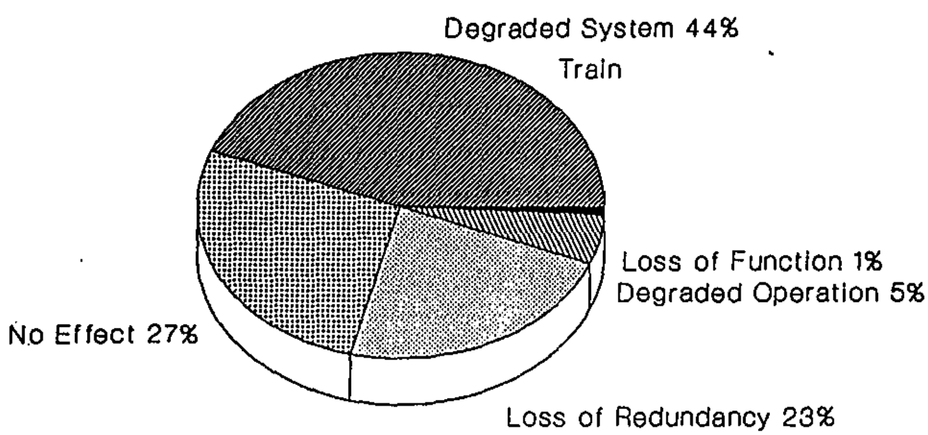

Figure 12. System effect from pump failures.

PUMP CONCLUSIONS AND RECOMMENDATIONS

This evaluation of operational data for Class 1, 2, and 3 pumps demonstrated that a significant number of failures have occurred, resulting in significant system effects (component loss, loss of redundancy, etc.), but have not resulted in significant plant effects. Numerous systems have been affected by these pump failures, with the service water and chemical and volume control systems being the most frequent. Over 75% of the failures affected operating pumps. However, the failures to standby pumps are significant, since it may have resulted in the pump being unable to perform its safety function in the event of an emergency. Table 4 shows the effectiveness of IST in detecting the six most commonly occurring failure causes. Plant testing was only effective in detecting 40% of the failures.

Pump aging, because of normal and abnormal wear of the internal components, was the most common failure cause. However, in addition to those degradations detected through testing, a significant number were discovered through routine observations. Examples of these occurrences indicate that leaks, high vibration levels, or variations in pump motor current readings alerted operators to impending failure. While not every degradation is detectable through testing, it is essential that the results from these tests be trended, and those which continue to approach Code specified Alert and Required Action Ranges be evaluated. The Code currently only requires licensee corrective action after the Required Action level is reached. It is also essential that the trending program be sensitive enough to detect such trends. Not to be lost in this evaluation is the importance of system walkdowns and observations by operating personnel. Without these observations, other failures would have occurred. An example of this are the pumps which failed because of blocked or clogged inlets. If this was occurring over time, it may have been detectable through trending flow or pressure measurements. The use of trending should be included in the Code.

Based upon the limited review of failures not detected by IST, the increased use of non-intrusive inspection techniques may help to detect degradations. Techniques such as motor current signature analysis have proven useful in correlating current variations to specific component degradations.

Some component aging caused by wear may only be detectable by periodic pump teardown and inspection. Though such a practice is required by the Code every 10 years as part of the inservice inspection program (i.e., ISI) for Class 1 components, this may not be adequate for certain risk significant pumps that have shown a tendency to this type of degradation. A more thorough analysis would be required to identify specific pumps and systems, since such a maintenance practice is not without risks, and may lead to additional failures.

Table 4. Effectiveness of testing in detecting pump failures.

Failure cause	Total number of failures	Detected by testing	Not detected by testing
Normal/abnormal wear	318	137 (43%)	181 (57%)
Caused by previous repair	40	15 (38%)	25 (62%)
Mechanical damage/binding	91	17 (19%)	74 (81%)
Out of mechanical adjustment	52	34 (65%)	18 (35%)
Aging/cycle fatigue	31	8 (26%)	23 (74%)
Blocked/obstructed	27	11 (41%)	16 (59%)
Other causes	174	272 (41%)	102 (59%)
Total	733	294 (40%)	439 (60%)

The USNRC has recommended for the next generation of nuclear power plants (i.e., the ALWRs) that a disassembly and inspection program be developed for all safety-related pumps to detect unacceptable degradation that cannot be detected through the use of advanced nonintrusive techniques. Periodic disassembly and inspection would be performed based on historical performance, analysis of trends, service life of parts (e.g., O-rings) and nonintrusive results (Taylor, 1990).

Continuous monitoring of temperature and vibration levels on some pumps may be warranted given the history of bearing failures. The Code is increasing the emphasis on vibration monitoring.

An important part of the testing program is post-maintenance testing. Failures attributed to previous maintenance were seen, which may have been preventable if thorough tests were performed prior to operation.

Testing provides one of the only ways to determine the operability of standby pumps. A significant number of standby pumps failed to continue to run after starting. If redundant pumps are available in a system that is normally operating (e.g., service water), it may be useful to rotate these into service periodically. This practice would tend to detect pump degradation, which could be addressed before the pump possibly failed on demand.

Our study highlighted the failures of Safety Class 1, 2, and 3 pumps, as reported by the licensees through the INPO NPRDS database. A significant number of pump failures were seen. Numerous occurrences were discussed highlighting failures that resulted from worn internals that were not detected by plant testing. However, a qualitative examination of these failures indicated that a significant portion may have been detectable if the trending program was sensitive enough.

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Inservice Testing of Vertical Pumps

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ABSTRACT

This paper focuses on the problems that may occur with vertical pumps while inservice tests are conducted in accordance with existing American Society of Mechanical Engineers Code, Section XI, standards. The vertical pump types discussed include single stage, multistage, free surface, and canned mixed flow pumps. Primary emphasis is placed on the hydraulic performance of the pump and the internal and external factors to the pump that impact hydraulic performance. In addition, the paper considers the mechanical design features that can affect the mechanical performance of vertical pumps. The conclusion shows how two recommended changes in the Code standards may increase the quality of the pump's operational readiness assessment during its service life.

INTRODUCTION

The inservice testing (IST) requirements of the American Society of Mechanical Engineers (ASME) Code require the pump to be tested in its regular circuit or a bypass loop. The pump is tested to determine reference hydraulic and mechanical values that may be one or more fixed sets of values, as described by Table IWP-3100-2 in Section XI of the ASME Code. Determination of operational readiness results from subsequent tests that are conducted at prescribed time intervals and compared with the reference values initially established. Each of the reference values consists of measuring pump capacity (Q), total developed head (TDH), and rotational speed (n). The measured test quantities are then compared with the reference values for the same quantity, and any deviations are compared with the limits specified in Table IWP-3100-2. The intent of this comparison is to determine if there has been any degradation of the hydraulic or mechanical performance that would impact the ability of the pump to perform its intended function.

Because the Code allows the pump to be tested on a bypass loop, the authors have assumed that the test might be conducted at a capacity that is less than the design conditions for the pump. In all likelihood, the test would be conducted at some greatly reduced flow relative to the design conditions of the pump and consist of only one test point as a result of the inability to regulate flow within the temporary bypass loop. The hydraulic performance at the reduced flow condition is used to extrapolate the performance of the pump to the full flow conditions.

Vertical centrifugal pumps have been used for safety-related services within nuclear power plants because of their versatility. Vertical pumps have the flexibility to increase flow by increasing the size of the pump, increasing the discharge pressure by adding additional stages, and meeting reduced net positive suction head requirements (NPSH) by making the pump longer. With these capabilities the vertical pump manufacturer can customize the selection to meet the hydraulic requirements of the system designer.

CLEARANCE

For this discussion, vertical pumps will be broken down into three basic categories, and several common characteristics will be discussed. The three categories are

1. Vertical free surface

2. Vertical canned mixed flow

3. Vertical canned radial flow.

Figure 1 shows the basic hydraulic characteristics of centrifugal pumps as a function of specific speed. For reference purposes specific speed (Ns) is a dimensionless number that defines the rotational speed required for geometrically similar pumps to deliver one gallon of liquid per minute at a total head of one foot.

In general, vertical free surface pumps have specific speeds of 3,000 or higher and most often are of the semi-open impeller construction. As can be seen from Figure 1, pumps of this higher specific speed range are characterized by steep head-capacity characteristics and power-capacity characteristics. These pumps are categorized as lines 4 through 8 on Figure 1.

A semi-open impeller has no shroud on the front of the vane and is shown in Figure 2. Semi-open impellers were developed because they offer improved efficiency resulting from the reduction in disk friction loss by eliminating the front shroud (Stephanoff, 1957). In order to achieve the increased efficiency, the rotating impeller vanes are operated very close to the stationary casing. The rotating impeller is lifted off the stationary casing in order to attain a prescribed clearance (A). The maintenance of the prescribed clearance is critical in maintaining the head-capacity characteristics of the pump. Tip clearance flows are an important source of turbomachinery energy loss (Engada and Rautenberg, 1989). As the clearance increases, the head-capacity characteristics deteriorate because of the leakage of the fluid from the high-pressure surface of the impeller vane to the low-pressure surface.

Tip clearance increases can be caused by wear from pumping abrasive liquids, momentary hydraulic pressure surges, increased bearing clearances, or excessive rotor or stator vibration. If the shaft and bearing combination wear, the resulting increased clearance and eccentricity will eventually result in the impeller vane tips, or impeller housing, wearing a similar amount.

Figure 3 shows the hydraulic performance of a 3,000 specific speed pump with a semi-open impeller at various clearances (Diemas, 1987). As can be clearly seen from this curve, the reduction in head for a specified-capacity value increases as the pump flow increases. Stated in other words, the increase in clearance causes the pump performance to deteriorate significantly more at the best efficiency point (BEP) flow of the pump than it does at the lower pump flows. When pump performance is evaluated at a relatively low flow, the observed reduction in pump head at low flow cannot be used as an accurate gauge of the reduction in pump head at the BEP flow of the pump.

For example, a test could be conducted on the pump whose characteristics are illustrated in Figure 3. Two all-friction system resistance curves have been arbitrarily drawn to intersect the pump head capacity curve at 200 and 600 gallons per minute. The loss in head at the various lifts is recorded in Figure 3 for the two system resistance curves. Comparison shows that at a lift of 0.100 in., 1.5 ft are lost from the 200-gpm point, while 5.5 ft are lost from the 600-gpm point. At a lift of 0.200 in., 5.5 ft are lost from the 200-gpm point, while 13 ft are lost from the 600-gpm point. Obviously if the 200-gpm data point were used as an indicator of pump hydraulic performance deterioration for the complete pump curve, it would lead to a very misrepresentative conclusion at the BEP flow of the pump.

Experimental data show that the lift-induced change in pump performance for semi-open impellers is different for the specific speed pumps as a result of curve shape, impeller vane loading, vane solidity, and special vane filing. The deterioration of pump performance from tip clearance is often found in pump literature, but practice shows these relationships to be unreliable because most of them are based upon simplified flow models where the impact of the simplifications is also unknown (Stepanoff, 1957). Consequently, no accurate generalizations can be made beyond the preceding example.

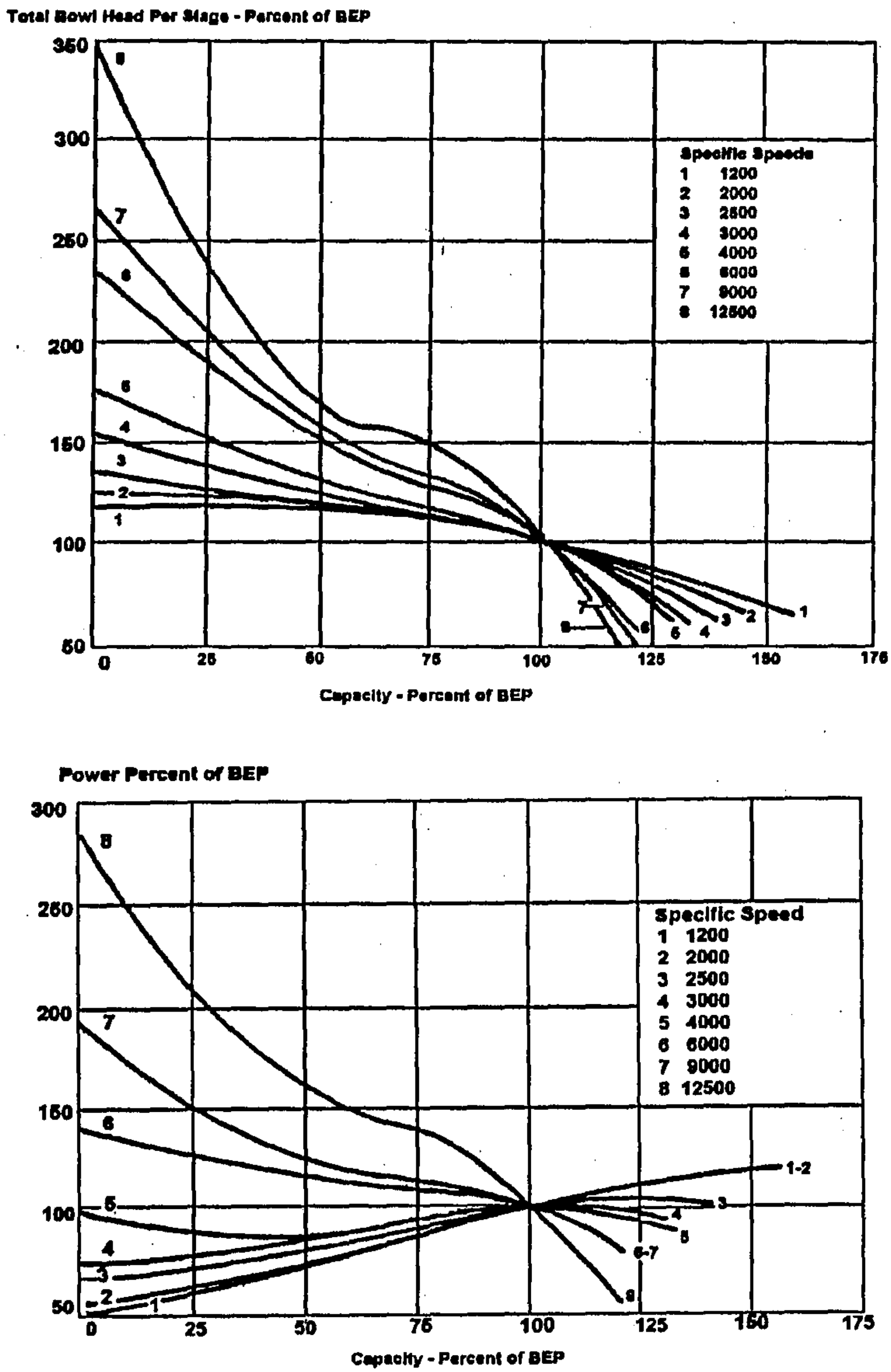


Figure 1. Head and power capacity curves at various specific speeds.

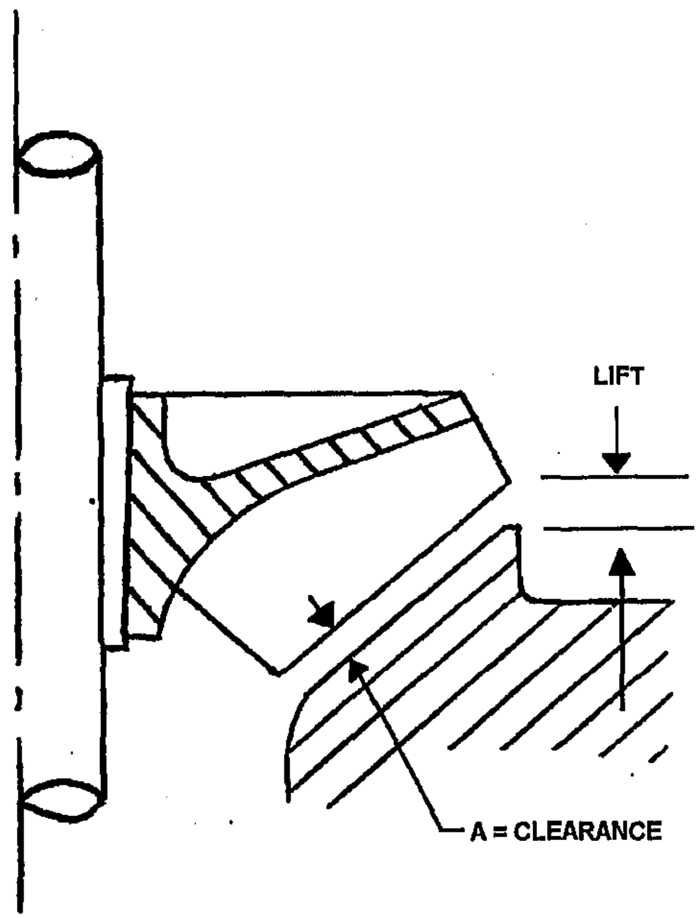


Figure 2. Mixed flow pump with semi-open impeller.

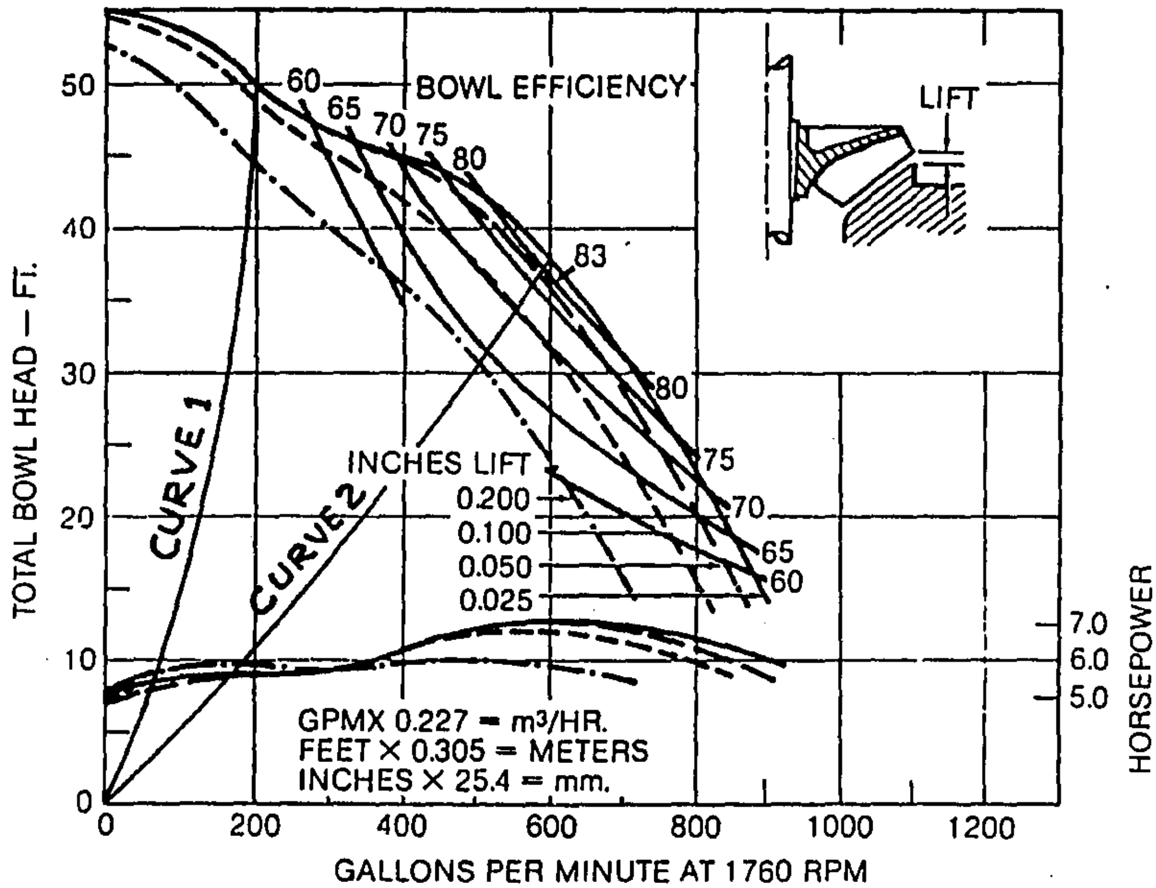
The vertical canned mixed flow and radial flow pumps are generally pumps whose specific speed is less than 3,000 and are of the closed impeller design. These pumps are categorized as lines 1 through 3 in Figure 1. Figure 4 shows the mechanical design of a typical mixed flow pump with an enclosed impeller. The performance of the enclosed impeller is affected by leakage through the clearance of the wearing rings at the front of the impeller and the casing walls. The amount of leakage is a function of the ring design, the length of the ring, and the clearance between the rings.

As with semi-open impeller designs, the ring clearances of the enclosed impeller will increase with wear. As the pump wears, the ring clearances increase and the leakage through the clearance increases, resulting in a deterioration of the pump hydraulic performance.

Figure 5 shows the performance of a 2,500 specific speed multistage canned pump. As can be seen, the change in hydraulic performance with ring clearance is a function of flow similar to the example cited for the semi-open impeller design. Consequently, the illustration used for the case of the semi-open impeller could be used for the enclosed impeller, and the same conclusion would be reached.

AXIAL THRUST

A vertical pump develops axial thrust as a result of the pressure differential across the impellers. This thrust is normally in the down direction. It is a common practice for designers of multistage vertical pumps to reduce the downward thrust in order to use drive motors with smaller thrust bearings. The most common method of reducing the impeller axial thrust is to



Lift (in)	Head-Capacity Curve 1 Head (ft)	Head-Capacity Curve 2 Head (ft)
0.025	50	37.5
0.050	50	36
0.100	48.5	32
0.200	44.5	24.5

Figure 3. Hydraulic performance of semi-open impeller at different lifts.

reduce the area of the impeller that is exposed to the developed head of the impeller. This is accomplished by adding close clearance rings to the back shroud of the impeller along with balance holes to equalize the pressure between the area inside of the back rings and the suction side of the impeller. Figure 6 illustrates a typical impeller uses back rings to reduce axial thrust.

Depending on the diameter of the back ring (Db) and the number of impellers that are balanced on a multistage pump, the pump designer can significantly reduce the axial thrust of the pump. Dicmas (1987) states that a commonly used value for thrust reduction is 50 percent balance.

The back rings reduce axial thrust because the developed head of the impeller is reduced through the circular annulus that is created by the ring. As the ring clearance is increased by wear, erosion, or corrosion, the pressure reduction across the

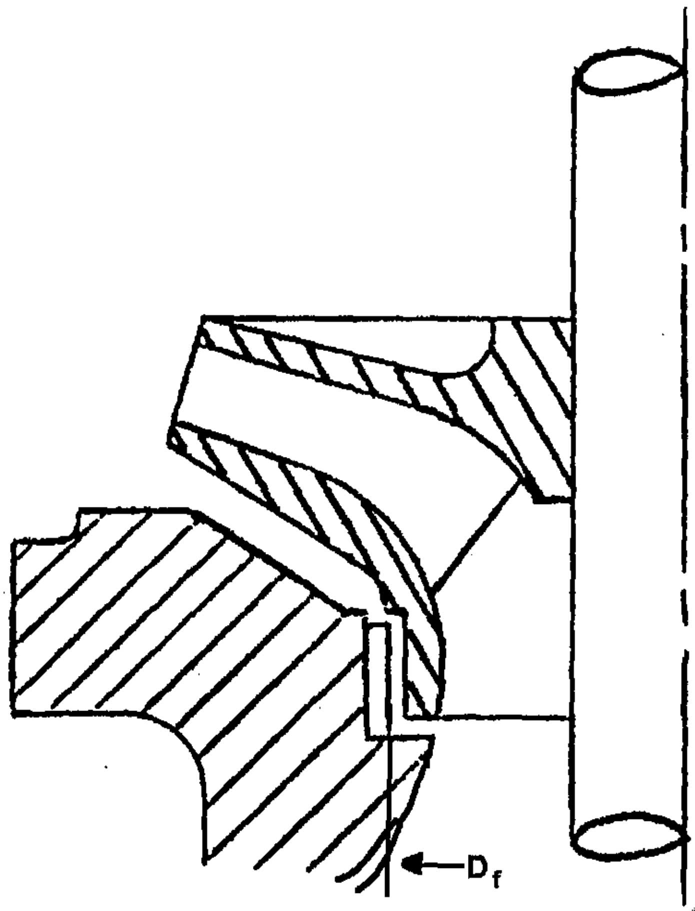


Figure 4. Radial flow impeller with front wear ring design.

ring surface decreases. This results in the area inside the back ring diameter (Db) being subjected to higher pressures, which then produce increasing axial thrust. Because the pump shaft is directly connected to the motor shaft, the load imposed on the motor thrust bearing increases.

Depending on the design of the motor's thrust bearing and the sizing criteria used, increased thrust load may result in higher bearing and bearing lubrication temperatures. All of this leads to a decrease in the expected life of the motor thrust bearing. Obviously, a failure of the thrust bearing in the motor will affect the operability of the pump.

In order to prevent a premature failure of the drive motor thrust bearing, it is necessary to measure the axial thrust of the pump because the condition of the back rings cannot be evaluated from the hydraulic performance testing of the pump. The most frequently used method to measure axial thrust in a pump manufacturing facility is to place load washers between the motor and the pump. Although this is relatively simple and inexpensive to accomplish for a short-term test, it impacts on the alignment between the pump and motor. Therefore, it is not recommended for long term installations. A more expensive method is to insert an additional component between the motor half coupling hub and the pump half coupling hub. This component is about the size of a conventional spacer used on vertical pumps with mechanical seals. The component is designed to measure the axial thrust in the rotor and transmit it electronically to direct-reading instrumentation. Because the new component is approximately 8 to 10 in. long, a spacer must be incorporated between the motor frame and the pump support.

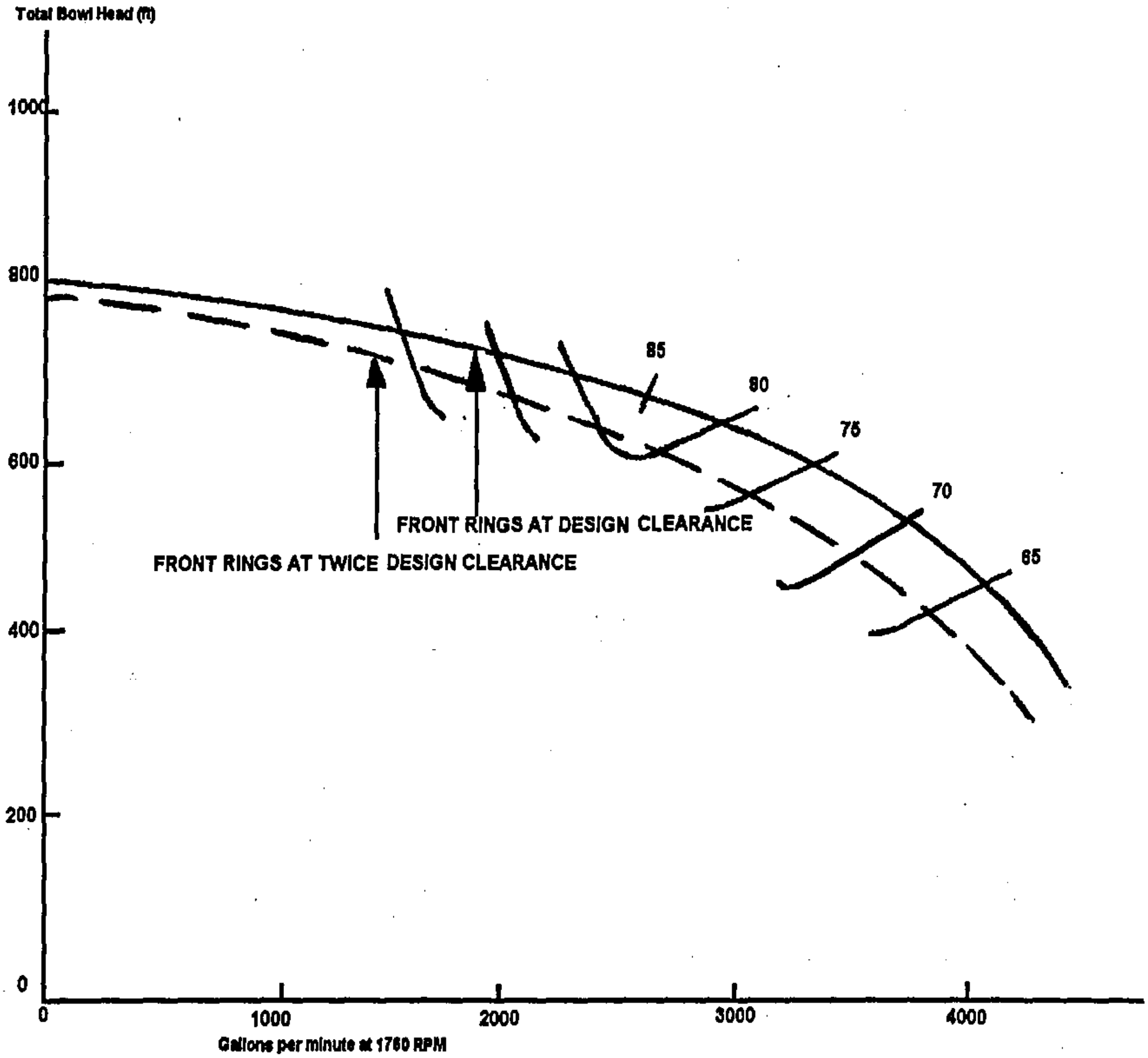


Figure 5. Hydraulic performance of mixed flow multistage pump.

The advantage of the second alternative is that the new component is commercially available and can be permanently installed in the pump. The disadvantage of this alternative is that the overall natural frequency of the motor and pump assembly is reduced because of the increase in overall height of the unit. However,.the pump axial thrust can be measured during IST, which allows the life of the motor thrust bearing to be evaluated.

CONCLUSIONS

A significant portion of this discussion relates to the affect of lift (semi-open impellers) and front ring clearance (enclosed impellers) on the shape of the pump head capacity curve. Through an example, it has been shown that the reduction in flow experienced at a low-flow condition cannot be used to predict the deterioration in the

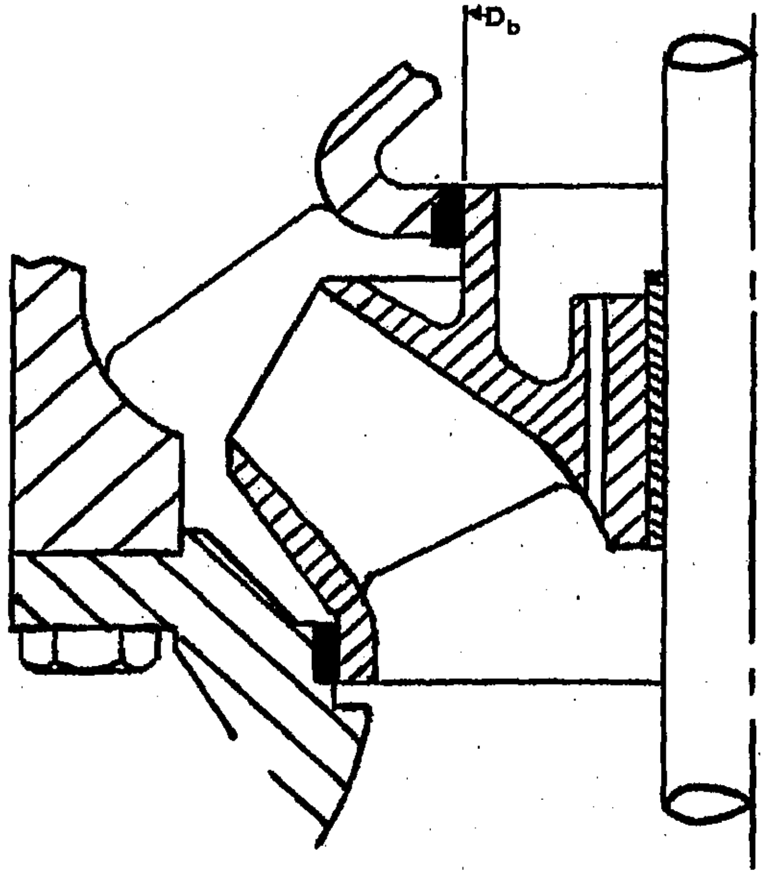


Figure 6. Typical design of closed impeller with back rings and balance holes.

hydraulic performance accurately at the best efficiency point. Consequently, IST conducted to assess the hydraulic performance of the pump must be performed using a loop that will allow the pump to operate at design flow. The authors recommend modifying the Section XI standards to require testing at the design flow of the pump. In order to accomplish this, it will be necessary to construct a bypass loop that can be throttled, as necessary, to allow the pump to operate at full design flow.

Second, in order to ensure the operational readiness of the drive motor, the axial thrust of a balanced pump design must be measured to evaluate the life of the motor thrust bearing. The authors recommend incorporating the measurement of pump axial thrust into the Section XI standards, using one of the methods discussed.

ACKNOWLEDGMENTS

The authors wish to thank the management of the Ingersoll-Dresser Pump Company for their support and acknowledge the contributions of all people involved in the preparation of the studies presented in this technical paper.

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Alternative Method of Inservice Hydraulic Testing of Difficult to Test Pumps

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ABSTRACT

The pump test codes require that system resistance be varied until the independent variable (either the pump flow rate or differential pressure) equals its reference value. Variance from this fixed reference value is not specifically allowed. However, the design of many systems makes it impractical to set the independent variable to an exact value. Over a limited range of pump operation about the fixed reference value, linear interpolation between two points of pump operation can be used to accurately determine degradation at the reference value without repeating reference test conditions. This paper presents an overview of possible alternatives for hydraulic testing of pumps and a detailed discussion of the linear interpolation method. The approximation error associated with linear interpolation is analyzed. Methods to quantify and minimize approximation error are presented.

INTRODUCTION

Section XI, Subsection IWP, of the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code and Part 6 of the ASME Code for Operation and Maintenance of Nuclear Power Plants (OM-6) require that the hydraulic performance of Code Class 1, 2, and 3 pumps be monitored to ensure their operational readiness and to identify degradation. These Codes require that the system resistance be varied until either the pump differential pressure or flow rate equals the corresponding reference value (independent variable). The remaining test quantity (dependent variable) is then measured and compared with its reference value. There are no Code provisions to allow variation in the independent variable, and power plants have encountered difficulties establishing reference test conditions for some systems.

As a general rule, inservice testing has not been a significant consideration in power plant design. In many cases, the valve used to vary system resistance was not designed for precise flow control. They are often large valves with poor throttling characteristics that are equipped with neither precise position indication nor positioning controls. In this situation, it is very difficult to precisely set the independent variable, at its reference value. Excessive valve manipulation is necessary in an attempt to repeat the reference value, which can contribute to degradation of the valve and its operator.

Several alternatives may be pursued when it is impractical to set the independent variable to an exact value. Two common approaches to this problem are either to set the independent variable as close as possible to its reference value or to perform testing using a reference pump curve. Both of these alternatives are addressed to some

extent in Draft NUREG-1482, *Guidelines for Inservice Testing at Nuclear Power Plants.*[a]

TYPICAL ALTERNATIVES

In NUREG-1482, Section 5.3, the U.S. Nuclear Regulatory Commission (USNRC) staff has taken the position that some variation in the setting of the independent variable may be allowed. However, because the reference values are fixed for both the dependent and independent variables, error will certainly be introduced into the test results. Relief is not required if the combination of the deviation from the fixed reference value and the associated instrument loop error do not exceed ±2% (the Code instrument accuracy requirement).

System design and configuration, pump characteristics, and instrument tap location may have a significant impact on flow rate measurements, but are not required to be included in the calculation of instrument loop accuracy. Pipe bends, elbows, and junctions; orifices; and valves induce turbulence that may affect achievable accuracy. Unsteady flows from pump hydraulic instabilities, recirculation cavitation, and variations in pump speed may make precise measurement difficult. If the instrument taps cannot be located according to the instrument manufacturer's recommendations, the actual measurement accuracy will be less than the rated instrument accuracy. Orifice erosion may also cause a reduction in accuracy. All these factors, when combined with the instrument loop error and the variance in the setting of the independent variable, may result in unacceptable data scatter. The pump-related problems are often easy to correct. The remedy may be as simple as testing at higher flow rates or using a more accurate tachometer (variable speed pumps). Correcting valve or piping design problems usually requires system modifications. The cost of these modifications may not be offset by a compensating increase in quality and safety.

a. U.S. Nuclear Regulatory Commission, NUREG-1482, Draft.

The degree to which pump tests are affected by flow measurement errors and variance from reference test conditions also depends on the pump hydraulic characteristics and the test conditions. If a pump has a relatively flat performance curve at low flow rates and is tested in this region with flow rate used as the independent variable (Q_R), then a substantial variation in flow rate may have little impact on the test results. This concept is illustrated in Figure 1. Conversely, when testing at higher flow rates, the same variation in the flow rates setting will have a greater impact on test results. This concept is illustrated in Figure 2.

One should not conclude that greater error may be tolerated with low flow pump testing. This example highlights a deficiency with low flow testing and illustrates that the extent of the problem with repeating reference test conditions will vary depending on the ability to throttle flow, the pump characteristics, the reference test point, and the "true" measurement accuracy. Because of the tight acceptance criteria of IWP and OM-6, it is necessary to limit the uncertainty in the test method to prevent unwarranted Alert or Required Action declarations.

Testing using a reference pump curve is a viable alternative for all situations where testing at a fixed reference value is impractical, whether system design is such that its resistance cannot be varied; reference values cannot be precisely duplicated; or flow rate is dependent on plant or climatic conditions. The usefulness of the test results obtained when using a reference pump curve will depend to a great extent on the method used to establish the curve.

It is unlikely that manufacturer's pump curves, which are developed under ideal conditions, would be acceptable for use as reference curves for inservice testing because of the differences between shop and field testing (Fehlau, 1992). Therefore, a reference pump curve would need to be generated. Although a number of different methods can be used to generate a reference curve, a third degree polynomial least squares approximation would generally be sufficient. The accuracy of the reference curve will improve as

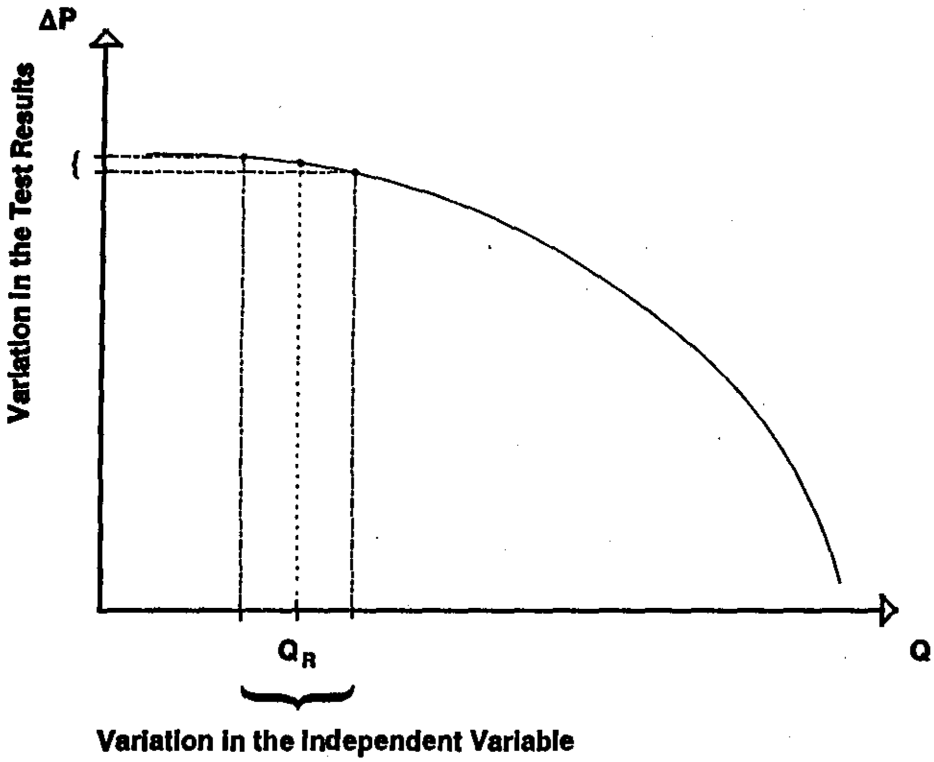

Figure 1. The effect of variation of the independent variable when testing at low flow rates.

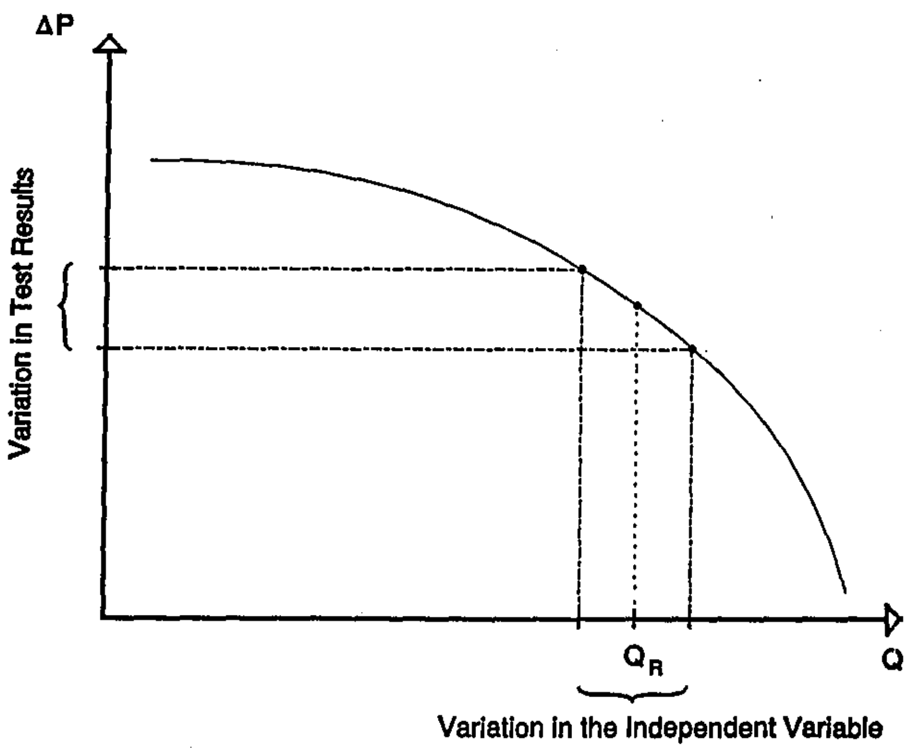

Figure 2. The effect of variation of the independent variable when testing at higher flow rates.

the number of data points increases. Other means can also be employed to improve the precision of the reference curve, such as using more accurate instruments to collect data and correcting for systematic instrument errors (Stockton, 1992). Additionally, either it must be demonstrated that vibration levels will not vary significantly over the range of the reference curve or a method of assigning reference vibration values must also be developed.

The use of reference pump curves is better suited to situations where system resistance cannot be varied, such as with "fixed" resistance test loops, or where flow rate cannot be controlled, as with many cooling water systems. NUREG-1482 provides little guidance on testing using reference pump curves; therefore, utilities must rely on their engineering judgement and expertise to develop the curve and demonstrate the acceptability of this alternative in a relief request. Although testing with a reference pump curve may be an acceptable alternative when reference values cannot be set with precision, it is a relatively complicated alternative. If feasible, a simpler alternative would be more desirable.

THE DUANE ARNOLD ALTERNATIVE

IES Utilities, Inc. (IUS) in cooperation with Vectra Technologies, Inc. (formerly NUTECH Engineers) has developed a simple alternative testing approach involving linear interpolation about the reference test point. This alternative was implemented at the Duane Arnold Energy Center following USNRC approval of their relief request.

With the linear interpolation method, the independent variable (flow rate) is set at two points. One flow rate setting (Q_H) is slightly higher than the reference value (Q_R) and the other flow rate setting (Q_L) is slightly less than the reference value. The differential pressure is measured at each test point (ΔP_H and ΔP_L). As shown in Figure 3, the straight line between these test points is used to approximate a small region of the

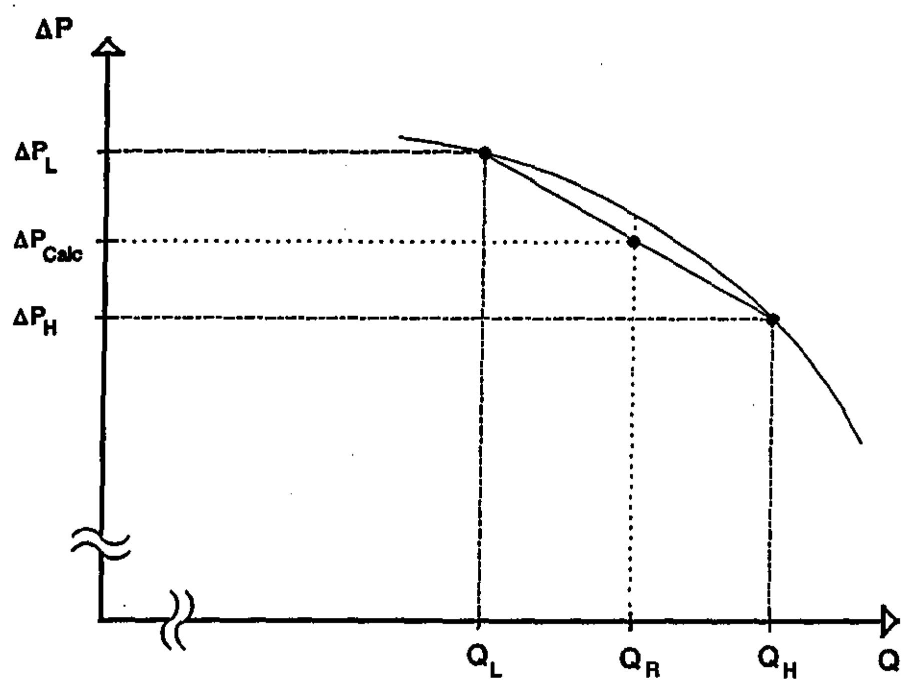

Figure 3. A small region of the pump curve is approximated with a straight line. Test differential pressure is calculated by interpolation using the linear function.

pump curve that contains the reference point. The differential pressure corresponding to the reference flow rate is calculated (ΔP_{Calc}) by linear interpolation using the values of the Q_R, Q_L, Q_H, ΔP_L, and ΔP_H. This calculated differential pressure is used, in lieu of a measured value as required by the Codes, for comparison with the reference differential pressure to monitor degradation from reference conditions.

The IUS test method has several advantages. As we shall show, the calculation to determine the test differential pressure using the linear interpolation equation is easy, and the approximation error associated with this test method can be quantified and controlled.

Linear interpolation is simpler than testing with a reference pump curve. The linear interpolation method requires measurement at only two points of pump operation, and a reference pump curve does not need to be developed and justified. It is easier to quantify the error associated with the linear interpolation method. Additionally, multiple vibration reference values are not necessary with linear interpolation because the pump test range is small and vibration levels will not vary significantly.

Regardless of measurement accuracy, the calculation of differential pressure with the linear interpolation method will always provide a more precise test result than a single measurement taken with some variation allowed between the independent variable and its reference value.

Calculations

The equation of the straight line between any two points, x_0 and x_1, which lie on a curve, f(x), may be expressed in Newton form. The linear function, p(x), shown in Figure 4 would be written

$$p(x) = a_0 + a_1\,(x - x_0) \quad . \qquad (1)$$

The coefficients, a_0 and a_1, are the Newton divided differences:

$$a_0 = f\,[x_0] = f\,(x_0) \qquad (2)$$

$$a_1 = f\,[x_0, x_1] = \frac{f\,(x_1) - f\,(x_0)}{x_1 - x_0} \quad . \qquad (3)$$

Therefore, the equation of this straight line in Newton form is

$$p(x) = f(x_0) + \frac{[f\,(x_1) - f\,(x_0)]}{(x_1 - x_0)}\,(x - x_1) \quad . \qquad (4)$$

The value of p(x) can be calculated with this equation for any value of x. When expressed in terms of our test variables, the general equation for calculation of the test differential pressure becomes

$$\Delta P_{Calc} = \Delta P_L + \frac{(\Delta P_H - \Delta P_L)}{(Q_H - Q_L)}\,(Q_R - Q_L) \qquad (5)$$

where

$Q_L = x_0$

$Q_H = x_1$

$\Delta P_L = f(x_0)$

$\Delta P_H = f(x_1)$

$Q_R = x$.

The differential pressure calculated in Equation (5) is compared with the reference differential pressure to determine whether the test results are within the Allowable Ranges of Test Quantities specified by the Codes. The difference between the calculated differential pressure and the reference differential pressure is an estimation of the amount of hydraulic degradation that has occurred, but also contains some amount of approximation error representing the pump curve with a straight line. Although that results from some additional error is introduced with this method, the error can be minimized to an insignificant level.

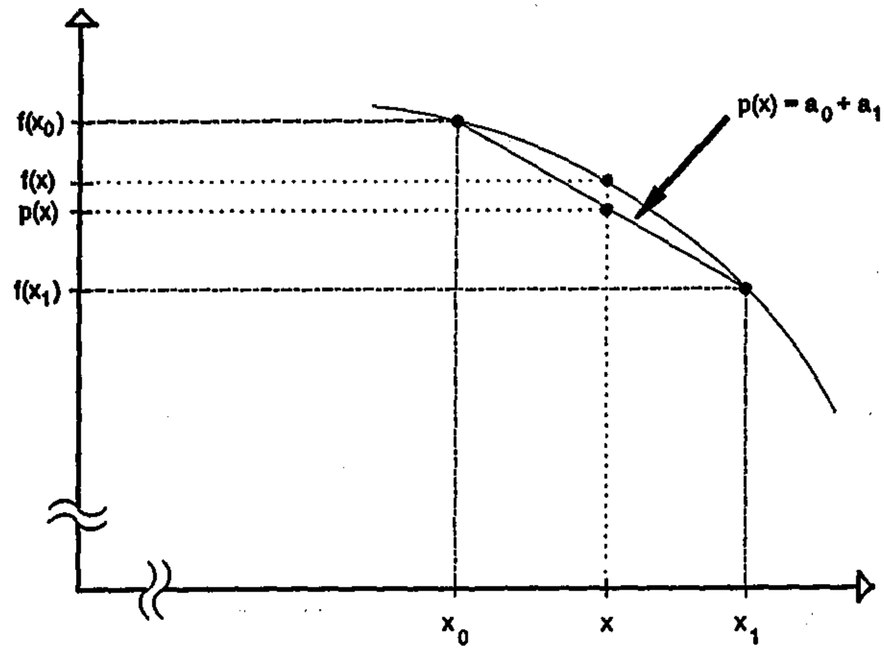

Figure 4. Variables used for derivation of the interpolating equation.

Error

The maximum error resulting from the approximation of a curve, f(x), with a linear function, p(x), is defined in de Boor (1978) as

$$\| f(x) - p(x) \| \leq \frac{1}{8} (\max \Delta x_i)^2 \| f''(x) \| \quad (6)$$

where

$$\Delta x_i = (x_{i+1} - x_i) \quad . \quad (7)$$

Since the norm of f''(x) is bounded, Equations (6) and (7) prove, and Figures 5 and 6 show graphically, that the approximation error associated with the representation of a pump curve with a straight line decreases as the distance between the test points, Δx_i, decreases. Therefore, the test points should be as close as practicable to the reference test point, and the pump test procedure should specify the maximum allowable range of the test points. The acceptable deviations of Q_H and Q_L from Q_R will depend on individual pump hydraulic characteristics. The flatter the pump performance curve, the better its approximation will be with a linear function.

If the reference differential pressure is calculated using the linear interpolation method immediately after establishing reference values (before pump performance has a chance to degrade), the approximation error can be more precisely quantified by comparison of $f(Q_r)$ and $p(Q_r)$.[b] The approximation error can also be estimated using values for $f(Q_r)$, ΔP_L, and ΔP_H obtained from a performance curve generated with field test data (if one exists).

Variance in the test results also serves as a good indicator of error. The test results for the Duane Arnold Residual Heat Removal Pump No. 1P-229A over the last 2-1/2 years are shown in Table 1. No significant maintenance was performed on the pump during this period, and the same instruments were used for each test. The loop accuracies of the test instruments used were ±0.5% of full-scale for pressure, and ±1.27% of

b. The error cannot be calculated exactly because the values of $p(Q_R)$ and $f(Q_R)$ are subject to instrument inaccuracies.

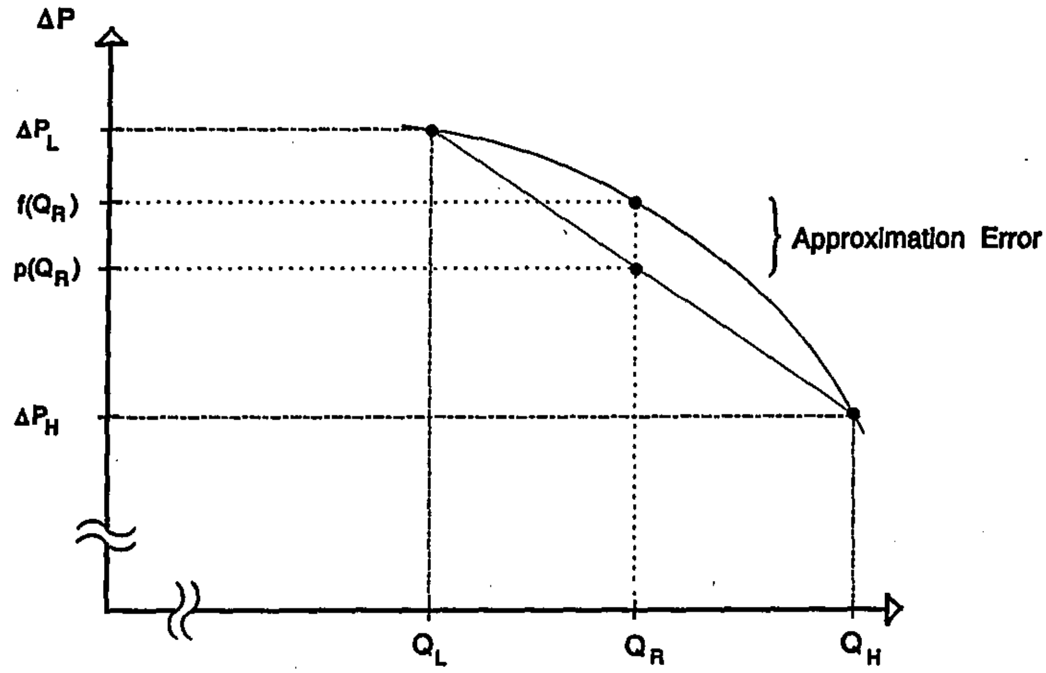

Figure 5. Approximation error from the linear interpolation method.

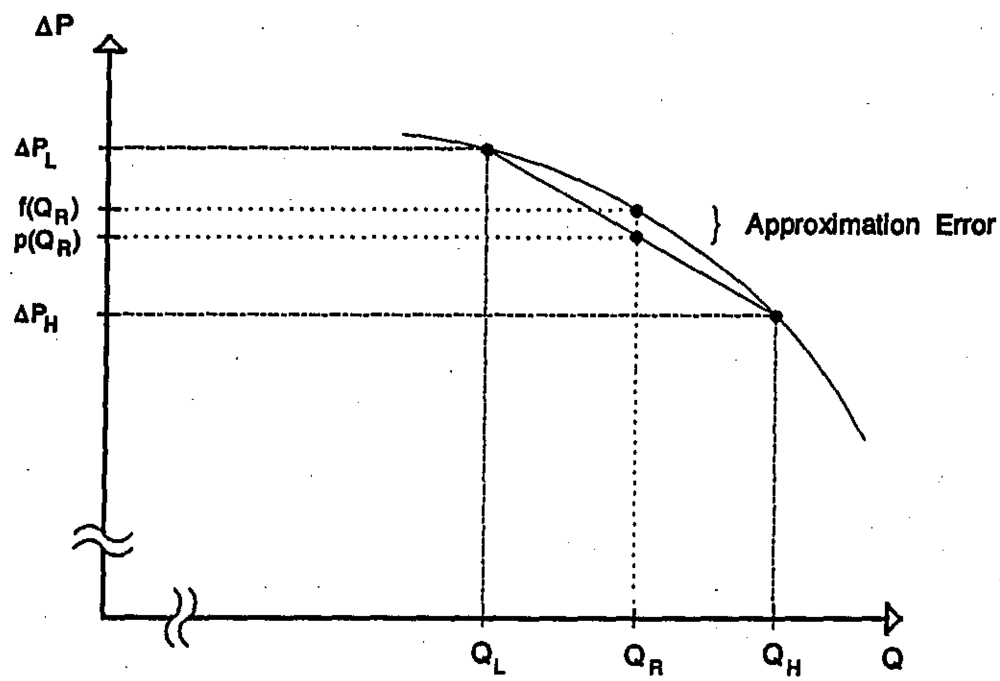

Figure 6. Approximation error decreases as the distance between the test points decreases.

Table 1. Test results for residual heat removal pump No. 1P-229A.

Test date	Test differential pressure (psid)	Reference flow rate (gpm)
05/09/91	148.64	5,200
07/31/91	150.13	5,200
10/21/91	148.84	5,200
01/24/92	148.30	5,200
03/29/92	150.06	5,200
07/02/92	148.60	5,200
10/02/92	150.49	5,200
11/21/92	151.60	5,200
11/24/92[a]	151.05	5,200
01/05/93	150.33	5,200
02/21/93	149.04	5,200
03/31/93	148.25	5,200
06/24/93	148.25	5,200
09/10/93	149.67	5,200
12/15/93	148.25	5,200
	mean = 149.43	

a. Instruments were recalibrated.

full-scale for flow rate. The reference flow rate, Q_R (the independent variable), for all tests was 5,200 gpm. By the IUS test procedure, the values of Q_L and Q_H for this pump may vary no more than ±200 gpm from Q_R (±3.85%). The maximum expected approximation error for this pump, determined using its preservice baseline curve, is estimated to be less than 1%. The total variation in these test results is 2.24% (+1.45%, −0.79%, about a mean value of 149.43 psid) which is well within the expected range of variation.

If testing is performed in a stable region of pump operation where the pump performance curve is shaped concave down, any approximation error will be in a conservative direction (i.e., in a direction indicating more pump degradation than actual). Any utility seeking to obtain relief to use this alternative should note that this in the basis of their relief request.

CONCLUSIONS

For the situation where the independent variable cannot be set with the necessary precision, linear interpolation is a superior alternative to testing either with a reference pump curve or with the Code-required method and allowing variation in the setting of the independent variable.

The linear interpolation test method is simple, the approximation error can be quantified and controlled, the calculations are easy, and any approximation error would be in the conservative direction. However, linear interpolation is a viable alternative only when the problem of compliance with the Code-required test method is the inability to precisely set the independent variable at its reference value.

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Description of Comprehensive Pump Test Change to ASME OM Code, Subsection ISTB[a]

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ABSTRACT

The American Society of Mechanical Engineers (ASME) Operations and Maintenance (OM) Main Committee and Board on Nuclear Codes and Standards (BNCS) recently approved changes to ASME OM Code-1990, Subsection ISTB, *Inservice Testing of Pumps in Light-Water Reactor Power Plants*. The changes will be included in the 1994 addenda to ISTB. The changes, designated as the comprehensive pump test, incorporate a new, improved philosophy for testing safety-related pumps in nuclear power plants. An important philosophical difference between the "old code" inservice testing (IST) requirements and these changes is that the changes concentrate on less frequent, more meaningful testing while minimizing damaging and uninformative low-flow testing. The comprehensive pump test change establishes a more involved biannual test for all pumps and significantly reduces the rigor of the quarterly test for standby pumps. The increased rigor and cost of the biannual comprehensive test are offset by the reduced cost of testing and potential damage to the standby pumps, which comprise a large portion of the safety-related pumps at most plants. This paper provides background on the pump testing requirements, discusses potential industry benefits of the change, describes the development of the comprehensive pump test, and gives examples and reasons for many of the specific changes. This paper also describes additional changes to ISTB that will be included in the 1994 addenda that are associated with, but not part of, the comprehensive pump test.

INTRODUCTION

Over the next few years, several commercial Nuclear Power Plant (NPP) operators will update their Inservice Testing (IST) programs to newer versions of the American Society of Mechanical Engineers (ASME) Operations and Maintenance (OM) Code, possibly to the 1994 addenda to the OM Code-1990. There are many changes in the 1994 addenda that were made to incorporate the comprehensive pump test and related changes. This paper was written to aid in an understanding of the changes made in the 1994 addenda, including what the specific changes are and in some cases why they were made. This paper should be consulted in conjunction with the 1994 addenda of the ASME OM Code, subsection ISTB. It provides some background on the pump testing requirements, describes the development of the comprehensive pump test, and discusses some of the benefits of the change. The paper then discusses the disposition of some of the key comments received during the approval process, identifies the Code sections affected, and gives a brief description of and reasoning for several of the more significant changes.

a. Work supported by the U.S. Department of Energy under DOE Idaho Operations Office, Contract DE-AC07-76ID01570.

BACKGROUND

ASME published the first rules for "Inservice Testing of Pumps in Nuclear Power Plants" in the 1973 edition of the ASME Boiler and Pressure Vessel Code, Section XI, Subsection IWP. Since then the ASME has issued various editions and addenda of the Code, including revisions to Subsection IWP. In 1976, the U.S. Nuclear Regulatory Commission (USNRC) published the first rule in the Code of Federal Regulations (CFR) establishing the ASME Code as requirements for pump testing at USNRC-regulated NPPs. The USNRC has since adopted several of the subsequent Code editions and addenda in revisions to the rules.

The 1988 addenda to the 1986 Edition of Section XI (ASME, 1988), Subsection IWP, states that "Pump testing shall be performed in accordance with the requirements stated in ASME/ ANSI OM (Part 6)" (ASME, 1987). The USNRC was considering adopting Part 6 in the regulations in 1988 and requested a meeting with the OM Committee. The OM Working Group on Pumps and Valves (WGPV) met with the USNRC in March 1989. The meeting was held to discuss concerns related to the newly approved pump and valve testing standards, OM-6 and OM-10, respectively. The OM-6 pump testing standard increased the upper required action limit for hydraulic test parameters from 103% (ASME, 1986) to 110%. The reasoning for the increase was that pump hydraulic performance was not expected to improve (Zudans, 1990). However, the higher limit could allow significant instrument calibration drift. The USNRC also had raised concerns about the potential for damage during low-flow testing of pumps in IST programs (USNRC, 1988). The USNRC agreed to accept OM-6 as written, if the ASME WGPV would consider improvements to the pump testing requirements.

Following the discussions, the OM-6 Task Group on NRC Issues, a task group under the WGPV, began work to develop a better, more comprehensive test for assessing pump condition. We tried to increase the overall effectiveness of pump testing with a minimal impact on plant resources. I believe we achieved these goals. The results of those efforts were called the comprehensive pump test.

As a point of information, the OM-6 pump testing standard was issued in October 1990 as OM Code-1990, Subsection ISTB (ASME, 1990). The comprehensive pump test change was written against the 1990 Subsection ISTB.

DEVELOPMENT OF THE COMPREHENSIVE PUMP TEST

The first step in developing the comprehensive pump test divided pumps into categories based on their standard operational mode. Pumps that operate the most frequently, such as service water or component cooling water pumps, are likely to degrade at higher rates than pumps that are operated only occasionally, such as standby liquid control pumps. At first, three categories of pumps were considered; however, the distinction between the categories was too vague. After much discussion, two categories were identified: Group A and Group B. Group A pumps are defined as pumps that operate continuously or routinely during normal operation, cold shutdown, or refueling operations. Group B pumps are defined as pumps in standby systems that are not operated routinely except for testing.

The next step developed testing strategies for the pump categories. Four tests were identified: preservice test, comprehensive test, Group A test, and Group B test. All pumps would initially receive the preservice test. That test would be followed quarterly by the test associated with the pump category (Group A test for Group A pump, etc.). Pumps in dry sumps are exempted from the quarterly tests. All pumps would receive the comprehensive test every 2 years.

The first test was the improved preservice or baseline test. This test would employ high-precision pressure instruments and establish accurate reference values at points of operation that would allow a precise assessment of pump performance, or operational readiness. For centrifugal pumps in variable resistance systems, the

test requires differential pressure and flow rate to be taken at five points of operation, from pump minimum to near design flow rates, to establish a baseline pump curve. The baseline pump curve can be used to establish additional reference values, if needed, and the data points are available for more detailed analysis of the pump if test data were to fall into an "Alert" or "Required Action Range." For these pumps, the reference value for flow rate must be set within 20% of the pump's design flow rate. For positive displacement pumps, the reference value flow rate must be measured at high pressure. Testing these pumps near their design pressure can allow the detection of leakage past the seals (degradation) that might not be detected at low pressures. The test requires using highly accurate, ±0.5% rather than ±2%, instruments for measurement of pressure. The higher instrument accuracy requirement helps to obtain more accurate reference values and minimize measurement uncertainties during testing. We recognized that it might require installation of temporary instruments. The preservice test is followed quarterly by a Group A or B test and at least once every 2 years by the comprehensive test.

The comprehensive test was developed to help ensure a better evaluation of pump performance characteristics at a reduced frequency. The comprehensive test must be performed only once every 2 years on all pumps. The test is performed at a single reference value at, or near (within 20%), the pump's design flow rate for centrifugal pumps. This area of the pump curve is considered to be most representative of the pump's hydraulic performance characteristics (Greenstreet, 1990; Stockton, 1992). The test is performed near design pressure for positive displacement pumps. The test specifies the same instrument requirements as the preservice test and the measured test parameters are compared with the accurate reference values determined during preservice testing. Pump flow rate, differential or discharge pressure, speed (if variable speed), and bearing vibration must be measured. The measured test parameter acceptance criteria are similar to the acceptance criteria of the preceding ASME Section XI, Subsection IWP, Table 3100-2, for hydraulic parameters and ISTB-1990, Table ISTB 5.2-2 and Figure ISTB 5.2-1, for vibration.

The Group A pump test, for frequently operated pumps, is performed quarterly and is based largely on the ISTB-1990 testing. Pump flow rate, differential or discharge pressure, speed (if variable speed), and bearing vibration must be measured. The measured test parameters and acceptance criteria are the same as in ISTB-1990. The difference between the Group A test and the ISTB-1990 test is that the test should be performed at as high a flow rate (or discharge pressure for positive displacement pumps) as practical. ISTB-1990 did not address the flow rate or pressure at which pump tests were to be performed. This should help ensure a good assessment of pump condition on a quarterly basis.

The quarterly Group B test, for standby pumps, was intended to be a quick, simple, largely qualitative test. The test would roll the pump to keep the bearings from taking a set and to lubricate and exercise moving parts. It was not intended to be used to determine hydraulic performance capabilities or to detect minor imbalances through vibration measurements. The critical performance analysis for Group B pumps is left to the infrequent comprehensive test. The Group B test can allow detection of gross mechanical or hydraulic failures or failures of electrical or control systems. The only parameters that must be measured for the test are flow rate (flow rate is the required parameter for positive displacement pumps) or differential pressure and speed (if variable speed). The test is simply a start and run test with loose acceptance criteria, 0.90 to 1.10 times the reference value of flow rate or differential pressure. The test can be conducted quickly and does not require a minimum run time or measurement of vibration.

Appendix I provides a list of the changes made to incorporate the comprehensive pump change. It identifies the changed Sections of OM Code-1990, Subsection ISTB, describes the changes, and provides reasons for changes. Minor editorial changes may not be identified. The additional changes, which were not part of the

comprehensive pump test change, are indicated as such in the appendix.

BENEFITS OF THE CHANGE

The comprehensive pump test change should have a positive impact on the safety and the cost of nuclear power production. The change demonstrates ASME's continuing commitment to the public, the USNRC, and the industry to develop and maintain high quality codes and standards. The committee took a reasonable approach with the change by improving safety without a negative impact on testing resources. The increase in safety results from an improved understanding of the pumps' condition and the decreased likelihood of test-induced damage. Many pumps receive less testing, but all pumps receive better testing. The change provides for more precise pump baselining and requires periodic testing of pumps at representative points of operation for an improved assessment of performance capabilities. All pumps will receive the preservice test followed every 2 years by the comprehensive test. The Group A pumps will be tested quarterly much the same as they are now, and the Group B pumps will receive minimal testing quarterly.

To get an idea of the potential cost benefits, I conducted an informal survey of plant operators and IST program coordinators to determine the effects of implementing this new testing methodology. From my informal survey, I found that between 30 to 60% of the pumps at a plant would qualify as standby pumps. I used information from an Electric Power Research Institute (EPRI) Plant Support Engineering presentation given to the OM Committee in 1992 to develop estimates for pump counts and test times. The EPRI study found that it took between 64 and 410 labor hours per quarter for pump tests, and there were from 16 to 41 pumps in an IST program. The average pump test takes about 8 hours [(64 + 410 hours)/(2 × 28 pumps) = 8.46 hours/pump. The average number of pumps at a plant is 28 [(16 + 41 pumps)/2 = 28 pumps]; about 45% of the pumps, or 13 (0.45 × 28 = 13), at a plant qualify as Group B pumps, and the remaining 15 are Group A.

Assumptions

I made the following assumptions for this analysis:

- The Group A test will take the same amount of time as the ISTB 1990 test, which is 8 hours.
- The Group B test for standby pumps will take only 25 to 50% of the time needed for ISTB 1990 tests: 0.25 × 8 hours = 2 hours, 0.5 × 8 hours = 4 hours.
- The preservice test will take twice as long (200%) as current tests, 2 × 8 hours = 16 hours.
- The comprehensive test will take one and a half times as long (150%) as current tests, 1.5 × 8 = 12 hours.
- The average plant is on a 2-year fuel cycle.
- IST program updates will occur once every 10 years, as required by 10 CFR 50.55 (the cost of the update is not affected by this change).
- The cost of a one labor hour is $50.

Results

Table 1 shows an analysis of costs of the ISTB 1990 testing and the 1994 addenda testing, assuming that the Group B test takes 25% of the time needed for the ISTB 1990 testing. Bear in mind that the preservice testing of all 28 pumps constitutes a heavy front-end load, which is why I stretched the analysis out over 6 years. The cost of testing 28 pumps according to the ISTB 1990 requirements for 6 years is $268,800. The cost of testing for 6 years according to the 1994 addenda is $231,200. This represents a potential cost savings of $37,600 (or 14%) over 6 years.

Table 2 assumes the Group B test takes half (50%) of the time required for the ISTB 1990 test. As shown in the table, the savings is $6,400 (or 2%) over the 6 years.

Table 1. Cost analysis assuming Group B test takes 25% as much time as the ISTB 1990 test.

Year	Quarter	ISTB 1990	Preservice test	Comprehensive test	Group A test	Group B test (0.25)
1	1	224	448	—	120	26
	2	224	—	—	120	26
	3	224	—	—	120	26
	4	224	—	—	120	26
2	1	224	—	—	120	26
	2	224	—	—	120	26
	3	224	—	—	120	26
	4	224	—	—	120	26
3	1	224	—	336	120	26
	2	224	—	—	120	26
	3	224	—	—	120	26
	4	224	—	—	120	26
4	1	224	—	—	120	26
	2	224	—	—	120	26
	3	224	—	—	120	26
	4	224	—	—	120	26
5	1	224	—	336	120	26
	2	224	—	—	120	26
	3	224	—	—	120	26
	4	224	—	—	120	26
6	1	224	—	—	120	26
	2	224	—	—	120	26
	3	224	—	—	120	26
	4	224	—	—	120	26

Number of hours spent during six years of testing 28 pumps per 1990 ISTB = 5,376	Number of hours for preservice testing of 28 pumps = 448	Number of hours for comprehensive testing of 28 pumps = 672	Number of hours spent during six years of testing: 15 Group A pumps = 2,880 13 Group B pumps = 1,248
Cost of 1990 ISTB testing at \$50.00/hour for 6 years: \$268,800	Cost of 1994 addenda testing at \$50.00/hour for 6 years: \$231,200		Potential savings if the Group B test takes 25% of the time of the ISTB 1990 or Group A test: \$37,600 percentage cost reduction: 13.99%

Table 2. Cost Analysis assuming Group B test takes 50% as much time as the ISTB 1990 test.

Year	Quarter	ISTB 1990	Preservice test	Comprehensive test	Group A test	Group B test (0.50)
1	1	224	448	—	120	52
	2	224	—	—	120	52
	3	224	—	—	120	52
	4	224	—	—	120	52
2	1	224	—	—	120	52
	2	224	—	—	120	52
	3	224	—	—	120	52
	4	224	—	—	120	52
3	1	224	—	336	120	52
	2	224	—	—	120	52
	3	224	—	—	120	52
	4	224	—	—	120	52
4	1	224	—	—	120	52
	2	224	—	—	120	52
	3	224	—	—	120	52
	4	224	—	—	120	52
5	1	224	—	336	120	52
	2	224	—	—	120	52
	3	224	—	—	120	52
	4	224	—	—	120	52
6	1	224	—	—	120	52
	2	224	—	—	120	52
	3	224	—	—	120	52
	4	224	—	—	120	52

Number of hours spent years of testing 28 pumps per 1990 ISTB = 5,376	Number of hours for preservice testing of 28 pumps = 448	Number of hours for comprehensive testing of 28 pumps = 672	Number of hours spent six during six years of testing: 15 Group A pumps = 2,880 13 Group B pumps = 1,248
Cost of 1990 ISTB testing at $50.00/hour for six years: $268,800	Cost of 1994 addenda testing at $50.00/hour for six years: $262,400		Potential savings if the Group B test takes 50% of the time of the ISTB 1990 or Group A test: $6,400 percentage cost reduction: 2.38%

It is evident from this elementary analysis that the simpler the approach taken for the Group B pump tests, the greater the potential cost savings. Another observation—boiling water reactors (BWRs), with their large compliment of full-flow test loops might find it easier to implement this methodology and achieve the high flow rates needed for the comprehensive test.

In summary, I believe that most, if not all, plants will realize a cost savings as a result of implementing this methodology. However, I think that the improved assurance of pump operational readiness (the safety benefit) is the key issue here.

DISPOSITION OF KEY COMMENTS

Many good comments received during the ballot process were not incorporated into the proposal for various reasons. Some comments were made regarding areas of the ASME Code that were not affected by the change. Those comments did not technically require a response, although responses were given. Other comments were quite involved and could employ task groups for years. For some of these, an immediate response was considered imprudent. The committee is now looking at the comments, prioritizing them, and considering additional refinements to the OM Code. The Working Group on Pumps (WGP) has prepared a matrix of comments and is currently reviewing them to ensure that important concerns are not overlooked.

The following section discusses the disposition of some key comments that were considered during the Code change process. The first had to do with a requirement to measure and evaluate pump motor current or power. The second regards pumps without full or significant test flow paths. The third and final involves the test frequency for pumps whose data during the comprehensive pump test fall into the alert range.

An early draft of the proposal included a requirement to measure the pump motor current or power with a measurement accuracy of $\pm 2\%$. Proposed acceptance criteria for required action were from 0.95 to 1.10 times the reference value of motor power. The thinking was that changes in the parameter could indicate changes in the pump's condition and that operators could use this in conjunction with the other parameters to better determine pump condition. However, many comments were received criticizing the parameter and suggesting that it be deleted. One commentor said, "...It [motor power] doesn't give pump performance information. There is no way for us to do this without modifying our electrical system to meet $\pm 2\%$.... It should be deleted." Others expressed concerns about the perceived difficulties of motor power/current measurement. There were also discussions about whether this constituted an analysis of the pump's motor and whether or not the motor's condition fell within the Code scope. We spent several meetings considering difficulties of making the measurement, appropriateness of the parameter and acceptance criteria, and scope issues. After carefully weighing these and other concerns, we decided that the remaining requirements would help to ensure an improved and accurate assessment of the pump's condition (operational readiness), and we elected to delete the parameter from the proposal.

Another comment had to do with pumps in systems that are not equipped with full or significant test flow paths. There are a few pumps in this category. However, writing code language to consider a small group of components is difficult. Also, licensee's have the option of submitting relief requests to the USNRC for requirements that are deemed impractical to implement for specific cases. No change was made to exempt these pumps from the comprehensive pump test. However, the white paper that accompanied the proposal was changed to include words similar to the following. The comprehensive test was developed with the knowledge that there may be some pumps, such as containment spray pumps, that cannot be tested at the required high flow rates because of limitations of system design. The comprehensive pump test was not intended to require installation of full-flow test loops in existing plants. As written, 10 CFR 50.55a does not require extensive plant modifications to be

performed to meet newly imposed IST requirements. However, licensees would need to request relief from the high flow rate requirements if those requirements cannot be met.

The last key comment involves an increase in the testing frequency for pumps with test data in the alert range of Table ISTB 5.2.1-1 for vibration acceptance criteria, or Table ISTB 5.2.3-1 for hydraulic acceptance criteria. Comprehensive pump test data in the alert range of these tables requires doubling the test frequency (performing the test annually). For certain pumps, this could cause a licensee to either repair the pump prior to starting up if the test was done during a refueling outage, or to shut down the plant in a year to perform the test (for some pumps). Some commentors wanted the alert ranges deleted from the comprehensive pump test. The committee discussed this issue at length. We considered several things, including ways to determine the rate of pump degradation. Ideally, the operator would have sufficient information to determine that the pump would or would not enter the required action range before the next comprehensive pump test. We believed that would be a hard call and left the alert ranges, pending further study. The ASME OM Task Group on Vibration is actively considering this issue for the vibration acceptance criteria, Table ISTB 5.2.1-1. The WGP is considering the issue in regard to hydraulic acceptance criteria.

SUMMARY AND CONCLUSIONS

This paper should aid in the understanding and implementation of the new pump testing methodology of the comprehensive pump test. It describes the major changes associated with the Code revision. It illustrates several of the issues the committee grappled with during the approval process. This paper describes the anticipated safety and economic benefits that will be gained from implementing the comprehensive pump test methodology. It showed that several of the comments received during the ballot process are under active consideration by Code groups for future enhancements to the Code. The paper also provides insights into the disposition of several key comments made on the proposal during the approval process. Appendix A identifies the affected sections, details most of the specific changes made to the Code in the 1994 addenda, and provides some of the reasoning behind the changes.

In conclusion, the comprehensive pump test change, which will be issued in the 1994 addenda to the 1990 ISTB Code, is a significant improvement to the Code. The change will have a positive impact on the safety and cost of NPP operation.

ACKNOWLEDGMENTS

Many committee members were vital to the development and final approval of these changes. Thomas Hoyle was a primary author who wrote about the test in a paper given at the Second NRC/ASME Symposium on Pump and Valve Testing (Hoyle, 1992). Thomas Ruggiero, Larry Sage, Robert Martel III, Robert Parry, Allison Pelletier, Richard Emrath, Gerald Dolney, Gary Cappuccio, Wavel Justice, Thomas Staskal, Rudolf Fehlau, Donald Zebrauskas, and Christopher Pendleton also contributed to the effort.

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Appendix A

Affected Code Sections and Descriptions/ Reasons for Changes

Appendix A—Affected Code Sections and Descriptions/Reasons for Changes

Table A-1. Affected code sections and descriptions/reasons for changes.

Code section	Description/reason for change
ISTB 1.3 Definitions	Added definitions of preservice test, group A pumps, group B pumps, reference point, and trending.[a] Revised definitions of inservice test, operational readiness, preservice test period, reference values, and instrument accuracy.
ISTB 3.1 Owner's Responsibility (b)	The section was slightly restructured. The requirement was added for the Owner to categorize pumps as group A or group B. Added a statement that a pump meeting both the group A and group B pump definitions shall be categorized as a group A pump. These changes help to ensure that the Owner correctly categorizes all pumps and helps to limit category changes.
ISTB 3.2 Bypass Loops	Revised so that bypass loops may be used for the group B test. Specific operational concerns for bypass loop testing are now explicitly stated (e.g., flow rate and time limitations). Bypass loops may also be used for group A or comprehensive tests, if the loop flow rate allows testing at the reference point. This change was made partly to recognize the high capacity bypass loops installed for most boiling water reactors.
ISTB 4 Testing Requirements	Added a statement on the purpose of the subsection and a discussion of the hierarchy of tests for preservice, comprehensive, group A, and group B tests, in that order. This allows flexibility in that a higher quality test may be substituted for a lower order test.
ISTB 4.1 Preservice Testing	This section was significantly restructured into a more logical progression as a result of the Editorial Committee's comments. It was also changed to require an initial set of reference values to be taken before implementing inservice testing. The section was divided to state test method requirements for centrifugal and positive displacement pumps separately. We added the requirement to take differential pressure and flow rate data at five points of operation, from pump minimum to at least design flow, to establish a reference pump curve. The pump curve requirement was limited to centrifugal pumps in variable resistance systems. A pump curve does not need to be established for pumps in systems where resistance cannot be varied. This change requires the pump to be tested at substantial flow and at several points of operation to better characterize its performance for comparison with subsequent test results. A single point must be designated as the reference point. Positive displacement pumps must be tested at, or near, design pressure. This testing is more likely to reveal certain types of degradation than testing done at low pressure. Vibration data must be taken at the reference point for both pump types. This should help the user navigate the code easier.

Table A-1. (continued).

Code section	Description/reason for change
ISTB 4.2 Inservice Testing	Added a paragraph with subsections identifying the paragraphs where the inservice test requirements are for group A, group B, and comprehensive tests. This should help the user navigate the code easier.
(NEW) Table ISTB 4.1-1 Inservice Test Parameters	The table is a significant revision of Table ISTB 5.2-1. It incorporates the group A and B, preservice, and comprehensive tests and shows the parameters that will be monitored during each test. Note 1 allows measurement or determination of parameter values. The table is needed to clarify test requirements due to differences between tests.
ISTB 4.3 Reference Values	This section was significantly restructured into a more logical progression as a result of the Editorial Committee's comments. A change was made to require reference values to be taken within $\pm 20\%$ of pump design flow rate for the comprehensive test and if practicable for the group A and B tests. If it is not practicable to test within $\pm 20\%$ of pump design flow rate, the group A and B reference values must be established at the highest practicable flow rate. Reference values must be taken in a region of stable flow. Added the requirement to document test results per ISTB 7. This is intended to allow an improved determination of pump condition as compared to lower flow rate testing.
ISTB 4.4 Effect of Pump Replacement, Repair, and Maintenance on Reference Values	Added the requirement to perform a comprehensive or group A test before declaring the pump operable. The Owner must determine whether reference values must be re-established. Reference value deviations must now be evaluated rather than identified.
ISTB 4.5 Establishment of Additional Set of Reference Values	Changed to require a comprehensive or group A test to be performed before establishing an additional set of reference values. This section was significantly restructured into a more logical progression. This allows data for differential pressure and flow rate from the pump curve to be used for the new reference values. Vibration values may also be determined from the curve points provided vibration was measured at points bounding the intended new reference point. Values determined in this way must be verified subsequently by test results.
(NEW) ISTB 4.6 New Reference Values[b]	This section was added to allow establishment of new reference valves where continued operation with test parameters in either the alert or required action ranges is justified. The licensee must perform an analysis and verify the pump's operational readiness at the new point and consider trends. The addition of this section resulted in number changes for the following subparagraphs (e.g., ISTB 4.6 Data Collection was changed to ISTB 4.7, etc.).
ISTB 4.7 Instrumentation	The title was changed from "Instrumentation" to "Data Collection." This is to recognize that all parameter values are not directly measured with instruments as addressed in Table ISTB 4.7.1-1.

Table A-1. (continued).

Code section	Description/reason for change
ISTB 4.7.1 General, (a) Quality	The title word was changed from Quality to Accuracy, because the section addresses instrument accuracy. Added an allowance to determine test parameter values indirectly and specified that the determined value shall meet the code-specified parameter accuracy requirements for the determined parameter. This should help ensure that high quality test data are gathered. Incorporated note (1) from Table 4.7.1-1 regarding accuracy requirements for analog and digital instruments and instrument loops.
Table ISTB 4.7.1-1, Acceptable Instrument Accuracy	Added columns of instrument accuracy requirements. Specified $\pm 0.5\%$ accuracy for pressure measurements during preservice and comprehensive tests. The increased measurement accuracy should yield better information to assess pump condition. The table note was moved and incorporated into paragraph ISTB 4.7.1.
ISTB 4.7.2(b)	Changed to delete reference to inlet "pipe" because many pumps take suction on an open bay.
ISTB 4.7.4(d)	Changed to replace the word "reference" with "measurement" for clarification.
Various	Changed 4.7.2 Pressure Measurement, 4.7.3 Rotational Speed Measurement, 4.7.4 Vibration Measurement, and 4.7.5 Flow Rate Measurement to delete the word "measurement" because some of these test parameter values might be determined.
ISTB 5.1 Frequency of Inservice Tests	Added the requirement to test pumps as specified in the new Table ISTB 5.1-1. This was done due to the various tests and test frequencies associated with this change.
(NEW) Table ISTB 5.1-1 Inservice Test Frequency	New table reflects the frequencies for performing group A and B and comprehensive tests.
ISTB 5.2 Test Procedure	The section was significantly restructured. There are now three parts: one each, for the group A and B and comprehensive tests. The new Sections, 5.2.1, 5.2.2, and 5.2.3, were structured in a similar fashion to the previous Section, 5.2.
(NEW) ISTB 5.2.1 Group A Test	Added a section with requirements for group A that describes the test. It specifically addresses testing of positive displacement and centrifugal pumps separately. The procedure specifies that speed must be set $\pm 1\%$ for variable speed drives. This better ensures that reference conditions are established. Also, instructions were added for comparison of measured vibration data with vibration data acceptance criteria. This section now specifies that the reference point pressure shall be set for positive displacement pumps.
(NEW) ISTB 5.2.2 Group B Test	Added a section with requirements for group B that describes the test. It specifically addresses testing of positive displacement and centrifugal pumps separately. The procedure specifies that speed must be set $\pm 1\%$ for variable speed drives. Pump pressure or flow rate is then compared to the reference. This is essentially a start and run test for the group B pumps.

Table A-1. (continued).

Code section	Description/reason for change
(NEW) ISTB 5.2.3 Comprehensive Test	Added a section with requirements for comprehensive that describes the test. It specifically addresses testing of positive displacement and centrifugal pumps separately. The procedure specifies that speed must be set ± 1% for variable speed drives. Pump pressure must be measured more accurately, ± 0.5%, than during group A or B tests. Instructions were added for comparison of measured vibration data with vibration data acceptance criteria. This test allows a less frequent, more detailed assessment of pump condition. This section now specifies that the reference point pressure shall be set for positive displacement pumps.
ISTB 5.3 Pumps in Regular Use	Specified that the requirement applies to group A pumps because they are, by definition, pumps in regular use.
ISTB 5.4 Pumps in Systems Out of Service	Replaced the words "of placing" with "before" for clarification.
ISTB 5.5 Pumps Lacking Required Fluid Inventory	This section was made specific to group B pumps, as any pump in a dry sump fits this category. Also specified that a group B test shall not be performed, because it could result in damage to the pump. Specified that the pumps receive a comprehensive test at least once every two years, with some exceptions.
ISTB 5.6 Duration of Tests	The section was restructured. There are now three parts: one each, for the group A, group B, and comprehensive tests. The new Sections, 5.6.1, 5.6.2, and 5.6.3, were structured in a similar fashion to the previous Section, 5.6. This takes into account some of the differences between the tests.
(NEW) ISTB 5.6.1 Group A test, Duration of Tests	Added specific group A test requirements under section ISTB 5.6 to test and measure, or determine and record, parameters identified in Table ISTB 4.1-1.
(NEW) ISTB 5.6.2 Group B test, Duration of Tests	Added specific group B test requirements under section ISTB 5.6 to reach stable conditions then measure or determine the required parameter values identified in Table ISTB 4.1-1.
(NEW) ISTB 5.6.3 Comprehensive test, Duration of Tests	Added specific comprehensive test requirements under section ISTB 5.6 to test and measure, or determine and record, the required parameter values identified in Table ISTB 4.1-1.
ISTB 6.1 Trending[c]	This paragraph requires trending of all but the fixed parameters of Table ISTB 4.1-1. It was inserted before the section on acceptance criteria, which required renumbering.
ISTB 6.2 Time allowed for Analysis of Tests[c]	This long-weekend provision (96 hours) was deleted. This is consistent with NRC Generic Letter 89-04, position 8 (NRC 1989), and good engineering practices. Once the data are recognized as being in the alert or required action range the pump should be considered inoperable.
(NEW) ISTB 6.2 Acceptance Criteria[c]	This section was subdivided into three sections to deal separately with the alert range, action range, and systematic errors.

Table A-1. (continued).

Code section	Description/reason for change
(NEW) ISTB 6.2.1 Alert Range[c]	This section calls out the tables with alert range acceptance criteria and requires the doubling of frequency as did ISTB-1990.
(NEW) ISTB 6.2.2 Action Range[c]	This section calls out the tables with action range acceptance criteria as did ISTB-1990. But it now allows an analysis of the pump and establishment of new reference values according to ISTB 4.6.
(NEW) ISTB 6.2.3 Systematic Error[c]	This section allows for the correction of systematic errors such as an improper lineup, or inaccurate instrumentation, and a rerun of the test.
(NEW) Table ISTB 5.2.2-1 Group B Test Hydraulic Acceptance Criteria	This new table provides the acceptance criteria for hydraulic performance parameters for the group B test. The acceptance criteria apply only to one parameter, and there is no alert range high or low. Discharge pressure is required to be set and flow (Q) measured for positive displacement pumps. Note also that there is no alert range.
(NEW) Table ISTB 5.2.3-1 Comprehensive Test Hydraulic Acceptance Criteria	The new table provides the acceptance criteria for hydraulic performance parameters for the comprehensive test. The acceptance criteria ranges are similar to those in Section XI, IWP, for these parameters; however, there is no alert range high. The tighter required action range "high" helps to limit instrument calibration drift.

a. The definition of trending was not part of the comprehensive pump test change.

b. This change was not part of the comprehensive pump test change.

c. This change was not part of the comprehensive pump test change but an additional change.

Session 2C
Valve Packing

Session Chair
Gerald Dolney
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Argo Packing Friction Research Update

Douglas M. VanTassell
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ABSTRACT

This paper focuses on the issue of valve packing friction and its affect on the operability of motor- and air-operated valves (MOVs and AOVs). At this time, most nuclear power plants are required to perform postmaintenance testing following a packing adjustment or replacement. In many cases, the friction generated by the packing does not impact the operability window of a valve. However, to date there has not been a concerted effort to substantiate this claim. To quantify the effects of packing friction, it has become necessary to develop a formula to predict the friction effects accurately. This formula provides a much more accurate method of predicting packing friction than previously used factors based strictly on stem diameter.

Over the past 5 years, Argo Packing Company has been developing and testing improved graphite packing systems at research facilities, such as AECL Chalk River and Wyle Laboratories. Much of this testing has centered around reducing and predicting friction that is related to packing. In addition, diagnostic testing for Generic Letter 89-10 MOVs and AOVs has created a significant data base. In July 1992 Argo asked several utilities to provide running load data that could be used to quantify packing friction repeatability and predictability.

This technical paper provides the basis to predict packing friction, which will improve calculations for thrust requirements for Generic Letter 89-10 and future AOV programs. In addition, having an accurate packing friction formula will improve packing performance when low running loads are identified that would indicate insufficient sealing force.

INTRODUCTION

Over the past 5 years Argo Packing Company has performed research in valve packing performance and friction. Improvements in packing material design have resulted in extended packing life, as well as reduced and predictable packing friction. This paper is a continuation of Argo's commitment to the nuclear industry to address various issues involving valve packing. At the 1992 summer Motor-Operated Valve Users Group meeting held in Richmond, Virginia, Argo presented a paper (VanTassel, 1992) that explained the factors that affect friction related to valve packing. Also, the issue of valve packing repeatability and predictability was discussed. Independent testing of modern graphite packing systems that use composite anti-extrusion rings demonstrate very low and predictable packing friction even when subjected to long-term cycling. If it is possible to predict the packing-related friction accurately. The following improvements in a valve program can be recognized:

- Improve predictability of packing-related friction to incorporate into operability calculations.
- Eliminate or reduce the need to perform postmaintenance testing after a packing adjustment or repack.
- Improve the performance of the packing materials by comparing tested running loads

with predicted running loads. Low running loads could indicate insufficient sealing force.

Another major improvement recognized by nuclear power plants is that the improved packing configurations with composite/graphite packing systems typically do not require repacking or packing adjustment. These packing systems offer the ultimate in leak-free and long-term performance.

PACKING FRICTION FORMULA

When predicting the amount of packing-related friction that will be present, the following factors must be considered:

- Stem diameter
- Type of packing material used
- Height of the packing set
- Amount of gland stress.

With these variables it becomes feasible to calculate the packing friction. In the past, packing-related friction was predicted by using the diameter of the valve stem and a multiplier (usually 1,000 lb times the stem diameter). This method of predicting packing friction is very inaccurate, and numerous applications to comply with Generic Letter 89-10 exhibited an actual packing friction far in excess of the value used in determining valve operability. This condition is illustrated by

- Valve stem diameter: 1.000 in.
- Calculated friction: Stem diameter (1.00 in.) $\times$ 1,000 lb = 1,000 lb.
- Actual packing load: 1,220 lb.

As this example illustrates, the actual running load exceeds the predicted value by 220 lb. The difference of these values represents a reduction of seating load that could impact valve operability. This is especially true in applications with small stem diameters, rising stems, or rotating stems. The formula for predicting packing friction is significantly improved by considering the values mentioned at the beginning of this paragraph. Using the previous example demonstrates that packing friction can be estimated more accurately by

Calculated friction: $F = 3.14 \times Gs \times Y \times f \times Ds \times Ph$,

where

Ds	=	stem diameter
f	=	friction coefficient
Gs	=	applied gland stress
Ph	=	packing height
Y	=	transfer ratio from axial to radial stress,

$F = 3.14 \times 4{,}000 \text{ psi} \times 0.85 \times 0.05 \times 1.000 \text{ in.} \times 2.500 \text{ in.}$

Calculated packing friction = 1,335 lb.

Actual packing friction = 1,220 lb.

As demonstrated by this example, the accuracy of the Argo formula is superior to estimating packing friction based solely on stem diameter. Figure 1 illustrates the frictional effect of increasing packing load and how the previous packing friction formula did not account for packing stress, height, or the type of packing used.

PACKING MATERIALS

The type of the packing used has a significant effect on friction and performance. The packing type, density, and height affects the "Y" value, or axial to radial transfer ratio. It has been proven in the laboratory and the field that increasing the packing set height does not improve sealability. In fact, having any additional packing set height

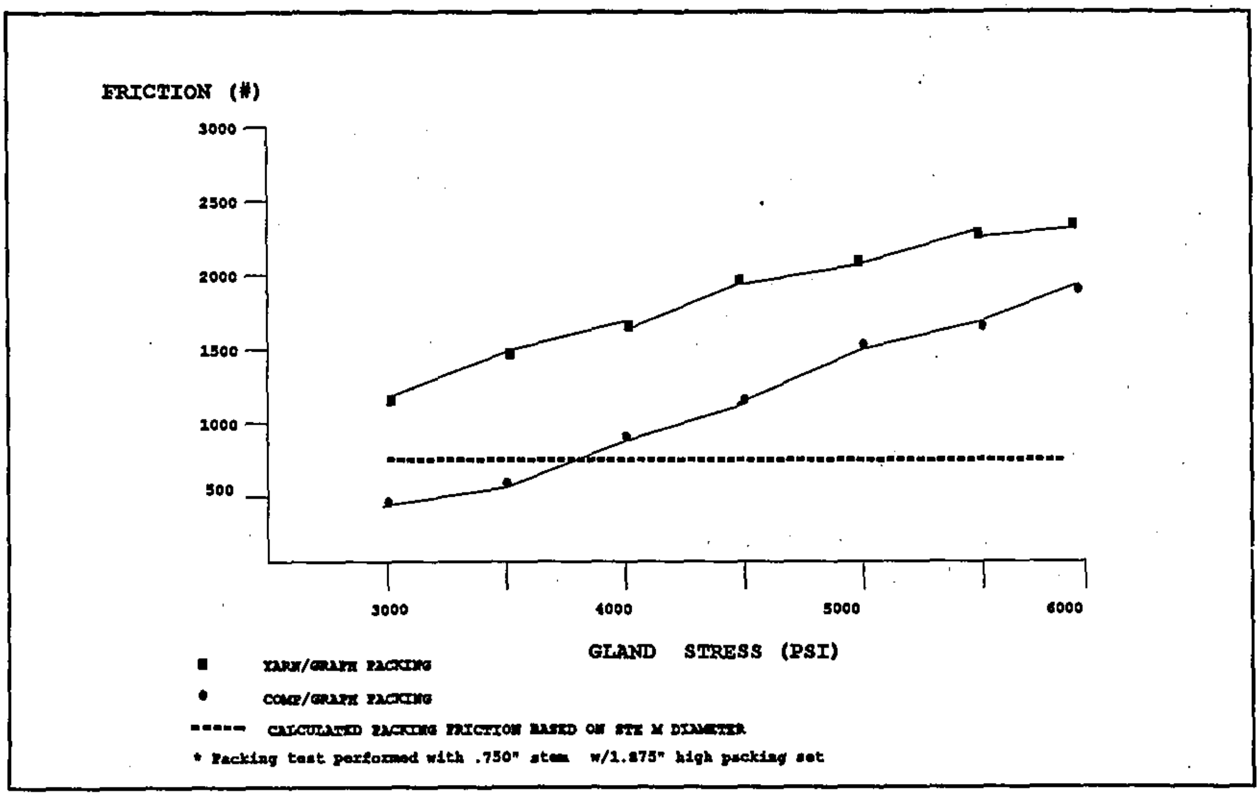

Figure 1. Effects of gland load on packing friction, comparison of yarn/graph versus comp/graph.

beyond what is absolutely necessary to accomplish sealing is actually detrimental to packing performance and increases packing related friction. The type of packing used also has a profound effect on the packing-related friction. In most U.S. and Canadian nuclear applications, combination flexible graphite packing systems are the standard. Nuclear power plants began converting to flexible graphite in the late 1970s, and many have implemented a packing "program" where procedures, training, and valve condition are emphasized (Brestel, 1992; Doyle, 1992).

There are two types of combination graphite packing systems. The difference between the two types lies in the anti-extrusion end rings that are necessary to contain the flexible graphite seal rings. Early designs of graphite packing systems used a braided graphite filament yarn as an anti-extrusion ring. Graphite filament yarn is very poor containment material because of (a) its porous construction and the tendency to fracture when subjected to stem cycling and (b) the gland stress required to achieve proper axial to radial movement of the flexible graphite. The "Y" value of braided filament yarn is very high because of its low density. Also, the coefficient of friction of braided graphite yarn is quite high. Typically a 0.2 coefficient of friction is found in nonlubricated graphite yarn. Because of the high "Y" and friction coefficient values of graphite yarn, much of the friction developed in a combination yarn and graphite packing system is located in the yarn anti-extrusion rings.

The friction developed in the yarn anti-extrusion rings affects packing performance in two ways. First, friction generated in the upper yarn anti-extrusion ring reduces the amount of axial gland stress that is generated in the flexible graphite seal rings. This significantly reduces the sealing properties of the packing system. The second effect is high friction load against the valve stem. In some valve applications this can affect valve operability. In addition, the packing stress is often lowered to reduce packing-related friction, which typically results in packing failure from insufficient gland stress.

Composite anti-extrusion rings were developed by Union Carbide and Argo Packing Company to address the deficiencies of graphite filament yarn. Composite rings are molded from graphite and carbon particles and are actually higher density than the die-formed flexible rings. Composite rings have a controlled radial expansion that is designed to contact the stem and stuffing box only enough to prevent extrusion of the flexible graphite seal rings. Combination composite and graphite packing systems have very low and predictable frictional characteristics when compared with yarn and graphite packing systems (see Figure 1). For Generic Letter 89-10, air-operated valves (AOVs), and testable check valve applications, composite and graphite packing systems are the logical choice. Typical yarn and graphite and composite and graphite packing configurations are illustrated in Figures 2 and 3.

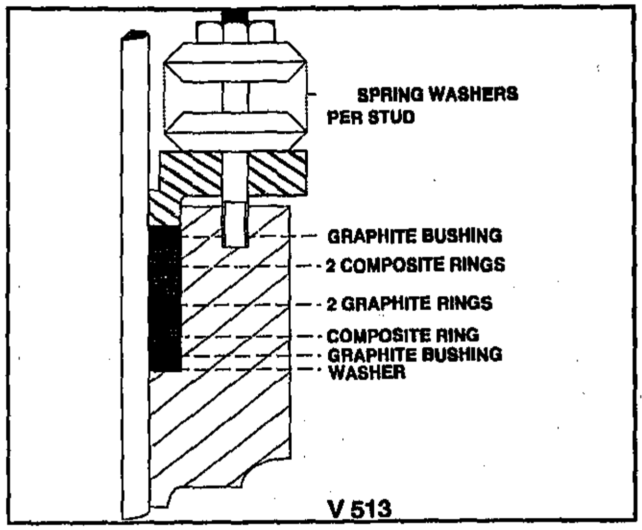

Figure 2. Composite and graphite packing system.

PACKING PERFORMANCE

The packing configuration and density can also have a significant affect on packing friction and performance. Since the mid-1980s, many nuclear power plants have standardized on a "five-ring" valve packing philosophy. The most popular packing configuration in use today is a five-ring (two yarn/three flexible graphite) combination graphite packing system. This packing system coupled with live-loaded was viewed as the cure to all packing-related problems. Unfortunately, nothing could be farther from the truth. This packing system does offer significant improvements over braided asbestos and nonasbestos fibers. However, flexible graphite should provide leak-free performance through the entire service life of most valves in a nuclear or fossil plant. This has not been the case, and numerous valves that have been packed with a combination yarn and graphite packing system do not last even one cycle before failing.

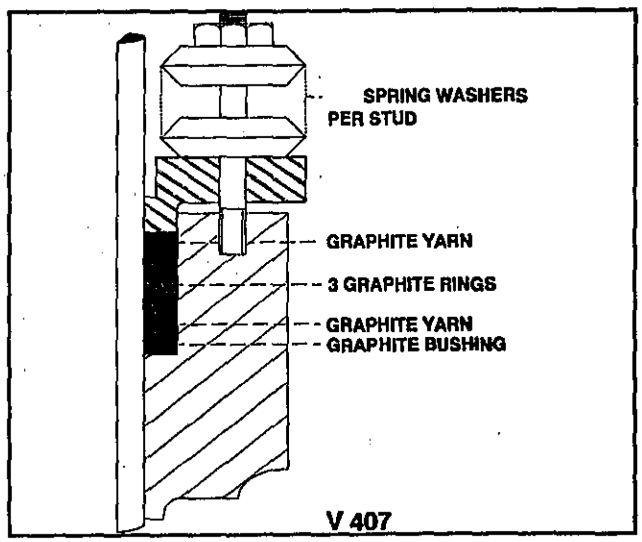

Figure 3. Yarn and graphite packing system.

The following reasons explain why a packing set will perform below expectations in regards to both leakage and friction:

- Breakdown of the graphite filament yarn anti-extrusion rings
- Improper installation of the packing materials

- Poor valve condition
- Improperly designed packing configuration
- Insufficient gland stress.

The breakdown of end rings made of graphite filament yarn reduces the volume of the packing set from both fiber loss of the yarn and extrusion of the flexible graphite. The volume loss results in a loss of gland stress, which eventually leads to packing leakage or complete failure. Converting to composite anti-extrusion rings eliminates failures from end ring degradation.

Key parts of any packing program lie in the procedure used to install the packing materials and the experience and training of the people who perform the repack. Susquehanna Station Pennsylvania Power and Light (PP&L) implemented a graphite packing program in 1985 that is considered one of the finest in the industry. To date, over 22,000 valves have been repacked with combination graphite packing systems. Packing leaks are virtually nonexistent in this two-unit boiling water reactor, and much of the success of their program can be attributed to the amount of training and support they provide to the repack crews. If a mechanic does not have the training or know the procedures on how to properly use the graphite materials, the packing program will not be successful.

The physical and dimensional condition of the valve stem and stuffing box must be taken into consideration. Valve stems that have pitting, corrosion, or a high RMS finish (above 32 RMS), will have high frictional loads and a short packing life. Bruce Nuclear Station (Ontario Hydro) and Comanche Peak [Texas Utilities (TU)] have both implemented valve refurbishment techniques to compliment the valve packing program. These techniques include processes such as flame spraying and super polishing of valve stems as well as placing sleeves in stuffing boxes that are found to be out of dimensional tolerances. By addressing valve condition, these nuclear plants and others have experienced tremendous valve packing success.

Improper design of the graphite packing configuration can result in short packing life and high frictional loads. The theory that five packing rings will work in all applications is simply not valid. In addition, not all nuclear plants have the approval to remove active leak-off lines, which requires the use of both upper and lower packing sets and lantern rings. Instead of approaching the issue of how many packing rings are necessary, it is far more important to evaluate the height of packing material necessary to seal a particular valve. For example, a 4-in. Borg Warner gate valve has a 1.00-in. stem with a 1.50-in. stuffing box inside diameter and a packing height of 1.500-in. In contrast, a 4-in. Rockwell globe valve has a stem diameter of 1.00-in. and a stuffing box inside diameter of 2.500 in. with a packing height of 3.750-in. if five rings are used. With a composite and graphite packing system loaded to 4,000 psi the predicted packing frictions would be 667 lb and 2,002 lb, respectively. If a five-ring packing set is installed in the Rockwell and loaded to the required stress, valve operability could be in jeopardy, especially considering valve that the actuator (if motor operated) would be sized for a 1-in. stem and 1,000 lb of predicted friction. It has been proven that with composite anti-extrusion rings, only two flexible graphite seal rings are necessary, and additional rings only add friction and reduce performance. In addition, the height of the composite and flexible graphite rings should be held in ratio to the stem diameter. Typical packing ring heights to valve stem diameter are as follows:

Stem diameter (in.)	Packing height (in.)
0.375 to 1.250	0.250
1.251 to 2.500	0.375
2.501 and up	0.500

Having the ability to manufacture rectangular packing rings is another advantage to using composite anti-extrusion rings because yarns are typically available only in square cross section. Using the valves previously described, a five-ring

set of 0.250 in. high rings could be used in the Rockwell valve, and packing-related friction can be reduced by 1,335 lb, potentially "saving" the valve from operator changeout or modifications.

In the past, graphite bushings have been used to take space previously occupied by excessive valve packing or lantern rings. Over the past 3 years, the function of the graphite bushings in the packing systems has expanded. By keeping clearance tolerances close and placing graphite bushings both above and below the packing set (see Figure 2), the valve stem maintains alignment. This reduces the risk of scoring of the valve stem with the gland follower or any metal parts that may contact the stem, such as back seat bushings, lantern rings, or metal spacers if present. Using the graphite bushings as guides reduces the wear on the packing system and extends packing life, especially in applications such as horizontally mounted valves, main steam isolation valves, or spring-air-actuated control valves.

If a valve application requires the use of an active leak-off, it is necessary to design a packing system that uses both an upper and lower packing set. The lower or primary packing set must have sufficient axial stress to perform. Unfortunately, in valves with an active leak-off the problem of simply too much packing material surface. The upper or secondary packing set robs much of the needed axial stress and adds additional frictional load. Because of this, a minimum number of packing rings should be used in the secondary packing set. This theory has been extensively tested by AECL Chalk River for the Candu Owners Group (COG) (Aikin, 1990; Aikin, 1992). The CANDU nuclear valves are all equipped with active leak-off ports. A seven-ring composite and graphite packing system (three upper rings and four lower) provides the lowest leak rates both in the laboratory and the field (see Figure 4). In addition, stainless steel lantern rings have been replaced by lantern rings constructed of graphite. This change provides a bearing point in the packing system, eliminates the potential for stem scoring, and simplifies future removal.

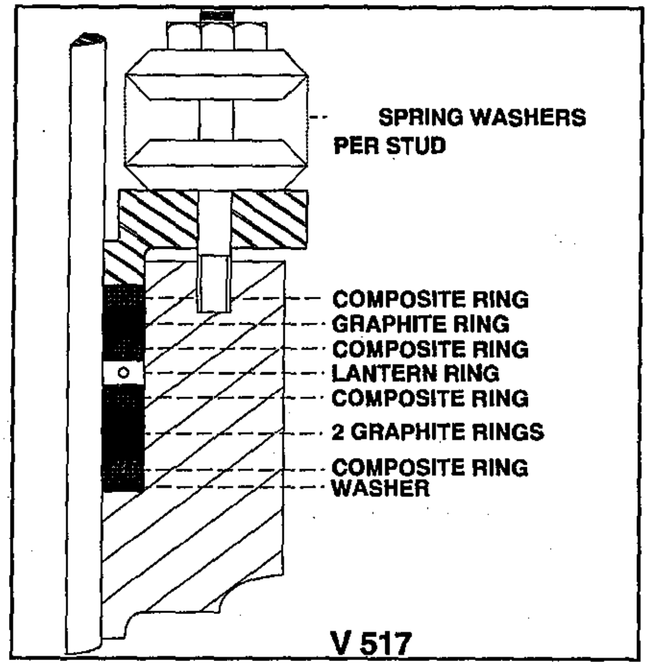

Figure 4. Composite and graphite packing system for active leak-off.

PACKING FRICTION TESTING AND EVALUATION

Background

Argo Packing Company is involved with several power utilities, valve manufacturers, valve diagnostic suppliers, and independent laboratories to identify the issues surrounding valve packing performance. Because of current regulatory mandates for motor-operated valves in nuclear applications and ongoing problems with packing friction in air-operated valves, friction research has been of the highest priority.

Over the past 4 years Argo has presented numerous technical papers conferences sponsored by motor-operated (VanTassell, 1991) and air operated users groups and the Electric Power Research Institute. The intent of these papers was primarily to inform the power industry of advancements in graphite packing and the effects of packing friction on valve operability. Information and test results revealed in these presentations were based primarily on laboratory and controlled field testing. This paper is a summary of the past 3 years experience in the field with

Argo's composite and graphite packing systems. Field data were accumulated from several nuclear power plants that have very high quality valve programs and are interested in improving their respective valve programs.

Purpose

This paper compares predicted packing friction loads with actual running loads collected during Generic Letter 89-10 diagnostic static testing, AOV diagnostic testing, and any laboratory testing. The calculation used is an industry standard calculation, but the values in regards to "Y" values and coefficient of friction are exclusive to Argo's materials. If packing friction can be predicted, it is repeatable. If packing friction can be both predictable and repeatable, issues such as preventive maintenance testing requirements following packing adjustment or replacement can be addressed. Unfortunately, the issue of valve packing friction, as with numerous other valve operability issues is not black and white. Numerous variables, such as the packing materials, installation, stem finish, and the diagnostic equipment, need to be factored before any claims or justification can be verified. The information provided in this report is intended to provide the basis to address the packing friction versus valve operability issues. This information can be melted into existing plant data and provide substantial, broad justification to current concerns. At the very least, an improved method of estimating packing-related friction and ways to reduce packing friction should be gained.

Data Collection

Data were collected for this project by a request from Argo Packing Company to all the plants that have a quality packing program where Argo materials are installed in valves that were diagnostically tested. In addition, past testing by independent laboratories or valve manufacturers was gathered to add to the database. The following plants and facilities submitted data for this evaluation:

Code	Facility/Station
AECL	Atomic Energy of Canada/Chalk River
BRUCE	Bruce Nuclear generating Station/OH
COMPEAK	Comanche Peak Station/TU
COOPER	Cooper Nuclear Station/NPPD
COPES	Copes/Vulcan Valve Co.
DARLNGT	Darlington Nuclear Station/OH
DIABLO	Diablo Canyon/PG&E
FARLEY	Farley Nuclear Station/SONOPCO
NGP	Nuclear Graphite Products
PVNGS	Palo Verde Nuclear Generating Station/APS
SUSQ	Susquehanna Steam Station/PP&L

The information provided by these facilities is illustrated in Attachment A. All valves are grouped by their stem diameter, packing height, and the gland stress. All the valves in this report are packed with a combination composite and graphite packing system. The station and valve number are listed in the table along with the following information:

DIAG SYST — Type of diagnostic system

CLOSE RT — Close running thrust

OPEN RT — Open running thrust

AVER RT — The average of the open and close running thrust

CALC RT — The calculated running thrust.

Data Evaluation

Evaluation of this data suggests that packing friction can be captured within limits. The formula used by Argo was typically greater than the average or even the maximum running thrust for each valve group evaluated. Figure 5 illustrates the summary of the data collected to date. Stem diameters from 0.500 to 4.750-in. were included

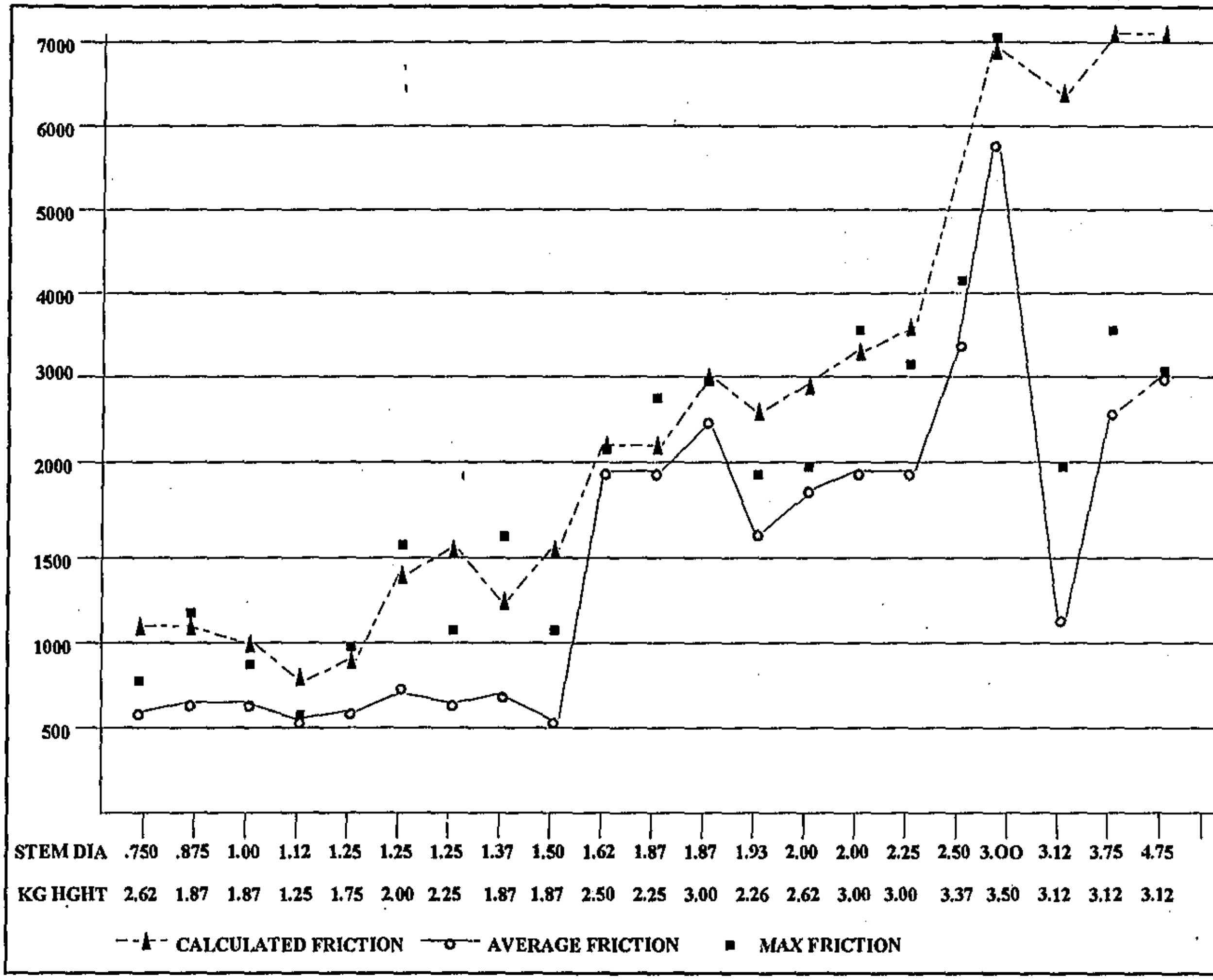

Figure 5. Summary of calculated versus actual average and maximum running loads.

in this report. The data summary grouped the valves by stem diameter, packing height, and gland stress. Yarn and graphite packing systems are being evaluated, and preliminary results indicate large data swings typical of this type of packing. The number of samples gathered for the report is shown on Attachment A, as well as the average and maximum running thrust of all samples. The variance in percent is shown between the average running thrust and calculated running thrust. In most field applications the average running thrust was 30 to 40% less than predicted. This was especially true when there were more than five samples in a valve group. It should be noted that the actual running loads for laboratory and controlled applications are typically very close to the predicted packing friction. Values tend to be very repeatable, especially when in-line load cells are used in a test fixture. A test fixture has none of the separate effects, such as gate guide friction, residual pressure effects, or anomalies in the diagnostic system, that could be present in a field application.

Figure 5 illustrates how the predicted packing friction using updated friction calculations compares with the average and maximum running load value per valve group.

Data Swings

As revealed in this report, packing friction loads can vary significantly between seemingly identical valves. However, it should be noted that in most cases the actual friction load is far **lower** than the predicted friction. This was expected

from the field data because of several factors that will be explained. The packing friction formula was built by Argo using laboratory and controlled field data. The packing system was installed in a valve or test fixture under ideal conditions. It was possible to stroke the valve several times to properly consolidate the packing system. The packing gland studs were clean and well lubricated so the torque applied to the gland nuts provided efficient transfer to the gland follower and packing system. And finally, the diagnostic system was set up and calibrated to read only the friction being generated by the packing system.

By contrast, field data are collected under far less ideal conditions. The packing system was probably installed in cramped and radiologically controlled conditions. Stroking the valve enough times to consolidate the packing may have been difficult or impossible. The gland studs may not have been in perfect condition, resulting in poor torque to stress efficiency. Even in good conditions, it has been proven that torquing on a soft joint such as packing or gaskets can vary by 40% (again usually below what was predicted). Because the majority of the field data are taken from concurrent Generic Letter 89-10 testing, calibration and set up of the diagnostics were oriented to seating loads and not to running loads. It should be noted that running loads may not be all packing friction. Friction developed in the internals of a valve, antirotation devices, and binding in AOV actuators can add significantly to running loads.

When enough valves are represented in a sample group, the high and low swings seem to even out, and the ratio between predicted packing friction and actual running loads becomes much closer.

Diagnostics

The use of valve diagnostics to determine packing frictional loads has to be evaluated very closely. Generally speaking, direct stem-mounted strain gauges or C-clamps provide the most accurate reading of running loads. The strain gauges should be calibrated in the expected range of the packing friction instead of the entire thrust window. Preliminary use of Liberty's Packing Enforcer has shown excellent results and provides a simple method of verifying running thrust and proper packing sealing force. Experience at Comanche Peak with direct-mounted stem strain gauges and MOVATs equipment has provided excellent results in reading packing frictional loads.

CONCLUSIONS

These data were collected by Argo Packing Company to expand our knowledge of the frictional effects of graphite packing systems and to assist plants with packing friction issues. This paper provides a large cross section of different valve groups that can be supplemented with existing diagnostic data. Packing friction is predictable if the following factors that affect packing friction are addressed:

- Packing Materials—Composite and graphite packing systems have very low and predictable packing friction and are used extensively in Generic Letter 89-10 applications in plants that have a packing program with Argo materials. Because of this, composite and graphite packing systems were discussed. (Note: Composite and graphite packing systems are available ONLY from Argo/NGP. Other packing suppliers' claims to providing composite anti-extrusion rings are untrue.)
- Installation Procedures and Training—Mechanics must be trained on the proper methods to install graphite packing systems according to detailed plant procedures.
- Valve Condition—The finish of the valve stem has a significant effect on packing friction. Generally, stem finish should be less than 32 RMS and free of nicks and scores.
- Database—A database that identifies the type of packing installed, configuration, and stud torque provides the necessary documentation to estimate the packing friction.

If a station implements a quality valve packing program and documents its activities, it is highly improbable that packing replacements or adjustments will be necessary. In addition, issues surrounding the need to perform preventative maintenance testing after these activities could be addressed by data on the ability to predict packing related friction.

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Brestel, R., 1992, "Control Valve Packing Systems," *Technical Monograph 38*, Fisher Controls, Marshaltown, IA.

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Attachment A

Stem diameter (in.)	Packing height (in.)	Stress (psi)	Samples	Maximum running thrust	Average running thrust	Calculated running thrust	Percent variation
0.500	1.000	6000	1	569	569	601	–6
0.750	1.875	4000	1	704	704	750	–6
0.750	2.187	3200	2	730	682	1051	–35
0.750	2.625	4000	4	760	620	1051	–42
0.875	1.500	4000	9	598	451	701	–36
0.875	1.875	4000	11	1119	701	1001	–30
1.000	1.250	4000	3	437	285	667	–58
1.000	1.875	4000	4	806	720	876	–18
1.000	1.875	3000	1	636	636	751	–15
1.000	1.875	5000	1	905	905	1250	–28
1.000	2.000	4000	1	705	705	1068	–34
1.125	1.250	4000	19	570	460	750	–39
1.125	1.562	4000	7	789	469	938	–50
1.125	1.750	5000	4	868	633	1314	–52
1.125	1.875	4000	1	687	687	1126	–39
1.125	1.875	5000	1	909	909	1406	–36
1.250	1.250	4000	4	566	507	834	–40
1.250	1.562	4000	2	558	531	1042	–49
1.250	1.750	4000	10	957	655	893	–27
1.250	1.875	3000	1	930	930	938	–1
1.250	1.875	4000	3	1011	1002	1096	–8
1.250	1.875	5000	1	1290	1290	1563	–18
1.250	1.875	6000	1	1485	1485	1877	–21
1.250	2.000	4000	22	1643	795	1335	–39
1.250	2.250	4000	20	1098	668	1501	–56
1.250	2.500	4000	2	1066	950	1668	–44
1.250	2.500	5000	3	370	309	2078	–86
1.375	1.562	4000	35	3288	1344	1146	+17
1.375	16.25	4000	1	1251	1251	1475	–15
1.375	1.875	4000	4	1663	759	1376	–45
1.375	2.184	4000	2	585	551	1582	–66
1.375	2.625	5000	8	2166	1018	2407	–58
1.500	1.875	4000	4	1179	521	1501	–66
1.500	2.250	4000	1	2781	2429	1802	+34
1.625	2.500	4000	2	2189	1959	2169	–10
1.625	2.500	5000	2	1730	1399	2709	–49
1.750	2.500	3000	1	825	825	1335	–39
1.875	1.875	4000	6	2799	1934	1878	+2
1.875	2.250	4000	9	2768	1944	2253	–14

(continued).

Stem diameter (in.)	Packing height (in.)	Stress (psi)	Samples	Maximum running thrust	Average running thrust	Calculated running thrust	Percent variation
1.875	3.000	4000	5	3022	2574	3003	–15
1.937	2.260	4000	4	1814	1169	2339	–50
2.000	1.875	4000	4	1955	1731	2002	–14
2.000	2.625	4000	3	2113	1904	2803	–33
2.000	3.000	4000	16	3602	1882	3204	–42
2.250	3.000	4000	12	3019	1880	3604	–48
2.500	3.000	4000	4	3603	3295	4005	–18
2.500	3.375	4000	9	4178	3379	4506	–25
3.000	3.500	4000	8	6146	4854	5607	–13
3.125	3.125	4000	2	2152	1243	5215	–74
3.750	3.125	4000	2	3571	2560	6307	–60
4.750	3.125	4000	1	3026	3026	7929	–62

Valve Packing Study

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ABSTRACT

The Electric Power Research Institute (EPRI) analytical model for valve packing is examined and extended to obtain important packing drag loads necessary in predicting motor-operated valve performance. First, the EPRI model is cast in a nondimensional form, resulting in further insight and analytical simplification of the packing behavior. Nondimensional tables and curves are presented to enhance the computational efficiency. Expansion of the model to include rotating stem and current packing arrangements having multiple regions of different packing material is provided. Pressure loading of packing is examined. An analytical expression for "effective Poisson's ratio" in terms of real Poisson's ratio is presented. Several examples are considered to provide a working knowledge of this methodology.

INTRODUCTION

Recent U.S. Nuclear Regulatory Commission (USNRC) concerns for operability of motor-operated valves (MOVs) resulted in the issuance of Generic Letter 89-10 (GL 89-10). Utilities, actuator and valve vendors, and nuclear steam supply system NSSS vendors, in an effort to comply with this USNRC concern, are reexamining MOV design and analysis and performing MOV in situ testing. The GL 89-10 program at Palo Verde Nuclear Generating Station (PVNGS), first initiated in 1990, consists of reevaluating set point calculations for 117 safety-related MOVs for each of the three PVNGS units. Static testing (zero pressure) using diagnostic equipment for all safety-related MOVs and dynamic testing (simulating design flow and pressure conditions) for 221 MOVs has been completed to date. In addition, PVNGS is reevaluating its nonsafety-related MOVs.

Many physical mechanisms that make up the operation of an MOV, are being examined to obtain a complete understanding of its overall behavior. Although MOVs have a long history of successful applications in other industries, the philosophy of MOVs applied to commercial nuclear industry is being revisited. The present development is only one small part of this re-evaluation. Many new questions and plausible answers concerning operability are being posed and evaluated by EPRI,[a] Toshin Dogan[b] and B. R. Black[c] have been contributing to this effort.

Valve packing has been addressed from a practical engineering point of view for many years. USNRC GL 89-10 concerns have resulted in a need to determine the packing loads that must be overcome by MOVs. In any MOV application packing drag can render an MOV inoperable. Therefore, control of this variable is a requirement.

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a. *Application Guide for Motor-Operated Valves in Nuclear Power Plants*, NP-6660-D, January 1990.

b. "Dynamics of Motor-Operated Valves," EPRI Research Project RP 3322-01.

c. "Motor-Operated Valve Testing and the Rate of Loading Phenomenon."

MATHEMATICAL MODEL

The EPRI model, explained in the *Application Guide for Motor-Operated Valves in Nuclear Power Plants* and shown in Figure 1 is used to describe the behavior of the valve packing. This model examines the axial packing pressure, $P_a(x)$, and the corresponding radial pressure, $P_r(x)$, acting on both the stem with diameter, d_i, and packing box with diameter, d_o. This radial pressure is produced by a Poisson's effect, R $P_a(x)$. The packing box produces frictional forces that resist the packing axial pressure. This resistance acting between the packing and the stuffing box is given by $\mu_o R P_a(x) \pi d_o \Delta x$ for a small differial packing element Δx, where μ_o is the coefficient of friction between the stuffing box and the packing material. A similar force acts between the packing and the valve stem, given by $\delta_{ij} \mu_i R P_a(x) \pi d_i \Delta x$. Here δ_{ij} is taken as +1 for upward motion, −1 for downward motion, and 0 for rotational motion. The use of the latter term is for rotating rising stem globe valves, as discussed in Cepkauskas and Coppock.[d] For a given Δx element these frictional forces are opposed by $dP_a(x) A_1$, where A_1 is the packing area given by,

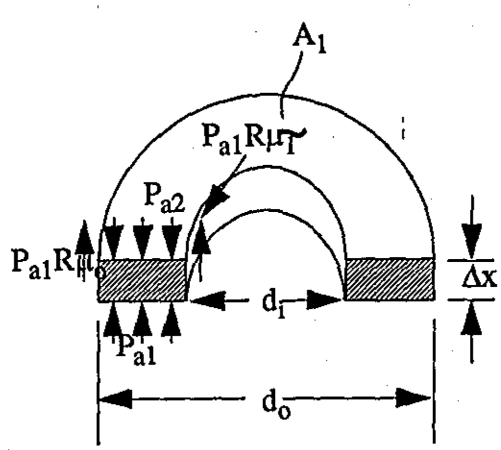

Figure 1. Forces in x-direction on differential packing element.

d. M. M. Cepkauskas and M. S. Coppock, "Packing Drag for Rotating Rising Stem Globe Valves," 1992 ANS/ASME Nuclear Energy Conference.

$A_1 = (\pi/4)(d_o^2 - d_i^2)$. The examination of these forces lead to the EPRI differential equation,

$$dP_a(x)/dx = KP(x) \ , \quad (1)$$

where

$$K = 4R(\mu_o d_o + \delta_{ij} \mu_i d_i)/(d_o^2 - d_i^2) \ .$$

NONDIMENSIONAL SOLUTION

It is convenient to cast this problem into a nondimensional form by defining:

$$\overline{P}_a = P_a(x)/S \ ,$$

where S equals the applied packing preload at the top of the packing box (psi), $\overline{X} = X/L$; where L equals the packing length, $\overline{K} = KL$; and Equation (1) becomes

$$\overline{dP}_a/\overline{P}_a = \overline{K}d\overline{x}$$

with the corresponding general solution,

$$\overline{P}_a = me^{\overline{K}\overline{x}} \ . \quad (2)$$

The boundary condition at $\overline{X} = 1.0$ and $\overline{P} = 1.0$ results in the solution,

$$\overline{P} = e^{-\overline{K}(1.0 - \overline{x})} . \quad (3)$$

Equation (3) is plotted in Figure 2. The corresponding radial nondimensional pressure, $\overline{P}_r$, is found by multiplying Equation (3) by R, the "effective Poisson's ratio." It is important to determine the seal point, the axial location, "a", (nondimensional form $\overline{a} = a/L$), where the radial pressure, P_r, equals the system pressure, P_{sys}. This can be expressed

$$\overline{x} = \overline{a} \ , \ \overline{P}_r = P_{sys}/(RS) = e^{-\overline{K}(1.0 - \overline{a})}$$

or

$$\overline{a} = 1 + (1/\overline{K}) \ln[(P_{sys}/(RS)] \ . \quad (4)$$

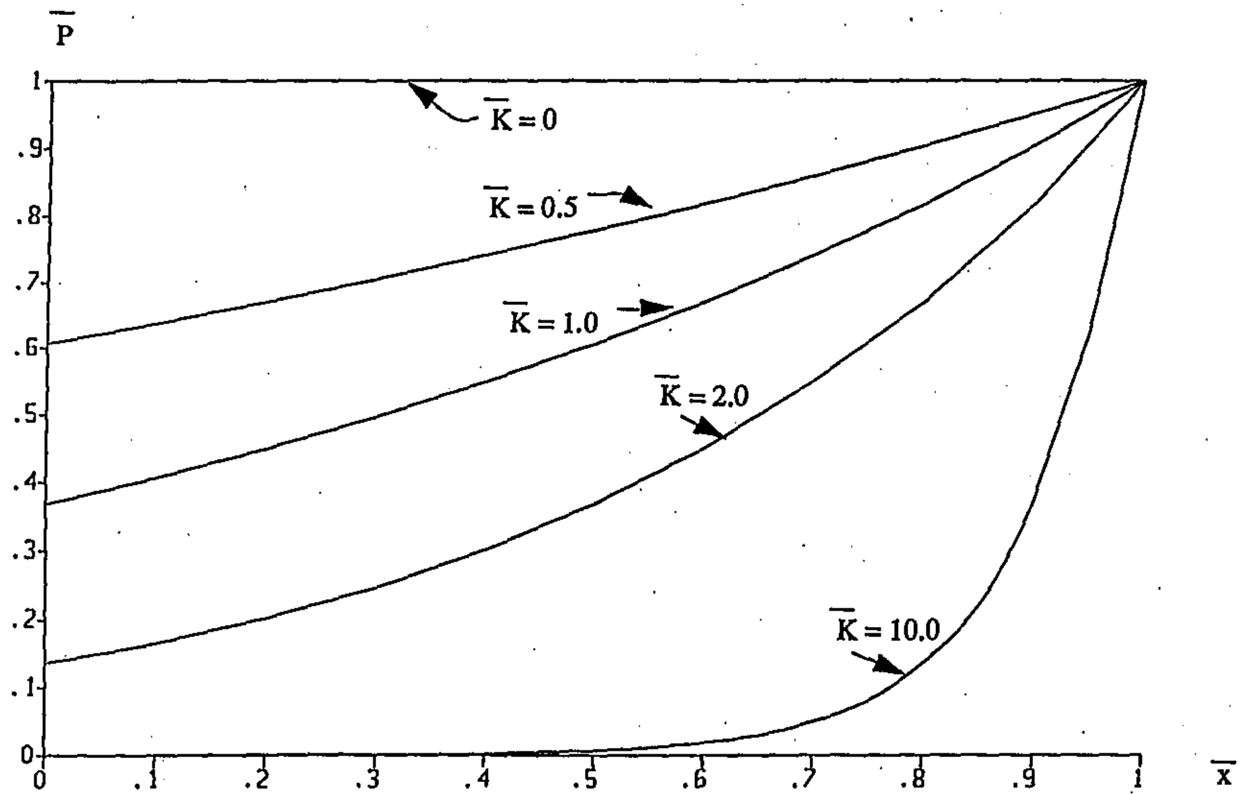

Figure 2. Axial normalized pressure distribution.

The nondimensional seal point given by Equation (4) is shown in Figure 3.

Defining the nondimensional packing drag force, $\overline{F}$, acting on the stem as $F_{packing}/(S\pi d_i L)$

results in

$$\overline{F} = \int_0^1 \mu_i R e^{-\overline{K}(1-\overline{x})} dx$$

$$\overline{F} = (\mu_i R/\overline{K})(1 - e^{-\overline{K}}) \ , \ (\overline{a} \leqslant 0) \qquad (5a)$$

$$\overline{F} = (\mu_i R/\overline{K})(1 - e^{-\overline{K}(1-\overline{a})}) \ , \ (\overline{a} > 0) \qquad (5b)$$

The normalized decay constant, $\overline{K}$, can be further simplified by recognizing that the packing length, L, is equal to the number of packing rings, N, times the height of each ring, or

$$L = N[(d_o - d_i)/2)] \ .$$

Taking the inner and outer coefficient of friction as being equal, results in

$$\overline{K} = 2R\mu N[(1 + \alpha\delta_{ij})/(1 + \alpha)] \ ,$$

where $\alpha = d_i/d_o$ (see Figure 4).

The packing drag given by Equation (5a) or (5b) can be conveniently written in the form,

$$F_{drag} = F^*_{drag}\varphi \ . \qquad (6)$$

F^*_{drag} is equal to $\mu R\pi d_i LS$, the drag force that would result if the preload, S, is not diminished along the length of packing by friction. This term may be used as an upper bound for packing drag.

The expression $\varphi = (1 - e^{-\overline{K}(1-\overline{a})})/\overline{K}$ is a correctional factor, as given in Table 1.

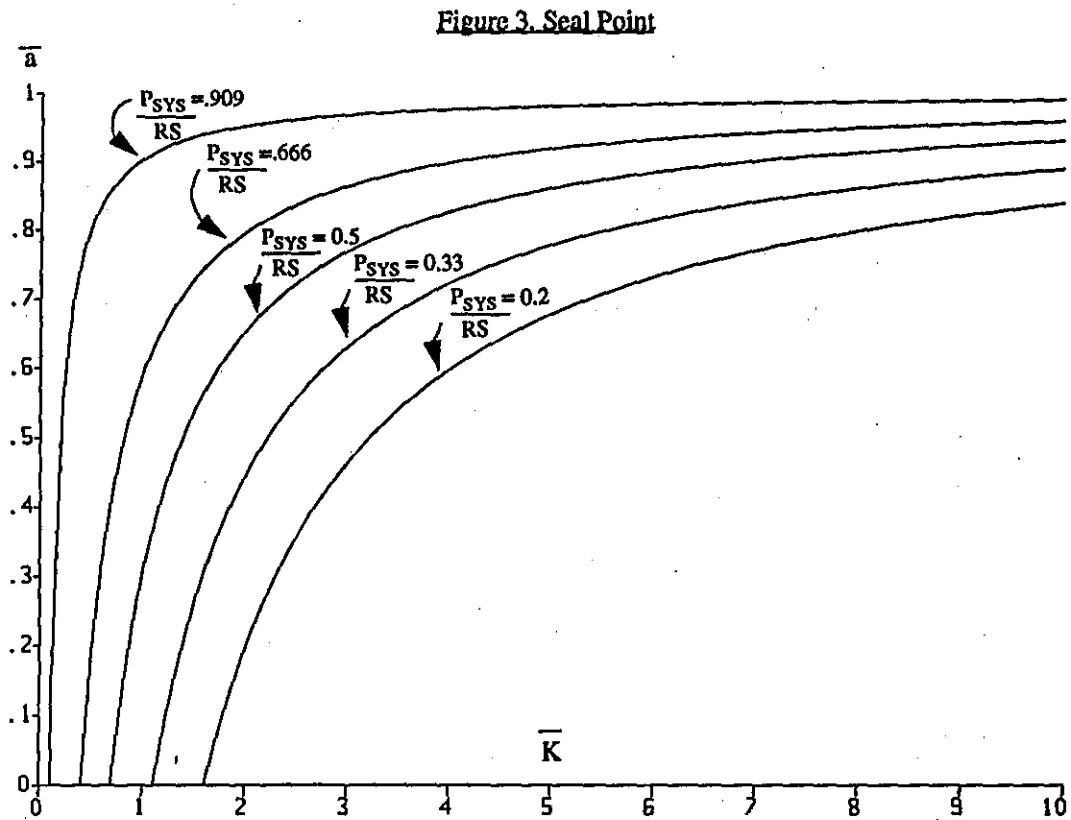

Figure 3. Seal point.

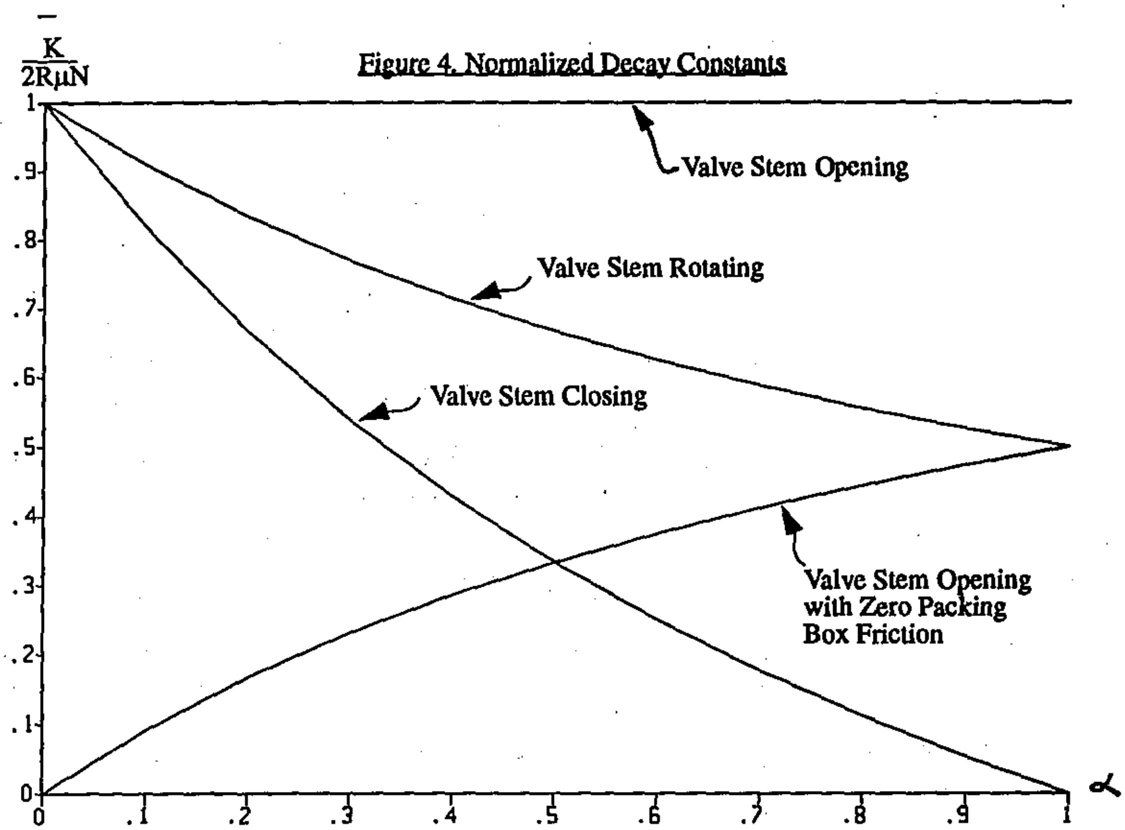

Figure 4. Normalized decay constatns.

Table 1. Valve packing correction factor.

$\bar{K}$	$\varphi(\bar{a} = 0.0)$	$\varphi(\bar{a} = 0.2)$	$\varphi(\bar{a} = 0.4)$	$\varphi(\bar{a} = 0.6)$	$\varphi(\bar{a} = 0.8)$	$\varphi(\bar{a} = 1.0)$
0.0	1.0	0.800	0.600	0.400	0.200	0.0
0.1	0.952	0.769	0.582	0.392	0.198	0.0
0.2	0.906	0.739	0.565	0.384	0.196	0.0
0.3	0.864	0.711	0.549	0.376	0.194	0.0
0.4	0.824	0.685	0.533	0.370	0.192	0.0
0.5	0.787	0.659	0.518	0.362	0.190	0.0
0.6	0.752	0.635	0.504	0.356	0.188	0.0
0.7	0.719	0.612	0.490	0.349	0.187	0.0
0.8	0.688	0.591	0.476	0.342	0.185	0.0
0.9	0.659	0.570	0.465	0.336	0.183	0.0
1.0	0.632	0.551	0.451	0.330	0.181	0.0
2.0	0.432	0.399	0.349	0.275	0.165	0.0
3.0	0.316	0.303	0.278	0.233	0.150	0.0
5.0	0.198	0.196	0.190	0.173	0.126	0.0
10.0	0.099	0.099	0.099	0.098	0.086	0.0
100.0	0.010	0.010	0.010	0.010	0.010	0.0

The EPRI formulation is easily extended to multiple packing rings with different packing materials. This is performed by requiring that axial pressure, $P_a(x)$, be continuous between packing regions. The mathematical details result in multiple solutions identical to the solution already presented with the condition that the axial pressure at the bottom of the upper packing region is used as the packing preload, S, at the top of the next region downward. The seal point is determined by starting with the bottom packing region and working upwards. The resulting packing drag has frictional contributions from all the packing above the seal point.

PRESSURE LOAD SOLUTION

A valve with pressure acting at the bottom of the packing results in an added axial pressure load acting upward. This pressure can be obtained by designing an open spacer ring at the bottom of the packing that allows pressure communications to preload the packing. For this configuration, superimposition of the mechanical preload and pressure load solutions is permissible. This results in

$$\bar{F} = \mu_i\left(\frac{\bar{R}}{\bar{K}}\right)(1 - e^{-\bar{K}}) + \frac{P_{sys}}{S}(e^{\bar{K}} - 1), \quad a \leqslant 0 \tag{7a}$$

$$\bar{F} = \mu_i\left(\frac{\bar{R}}{\bar{K}}\right)[(1 - e^{-\bar{K}(1-\bar{a})}] + \frac{P_{sys}}{S}(e^{\bar{K}} - e^{\bar{K}\bar{a}}), \quad \bar{a} > 0 \tag{7b}$$

with a corresponding seal point of:

$$\bar{a} = \frac{1}{\bar{K}}\ln\left[\frac{1}{R\left(\frac{S}{P_{sys}}\right)(e^{-\bar{K}}) + 1}\right] \tag{8}$$

The advantage of this type of packing arrangement is that with proper mechanical preload, it produces a radial stem force that is maximum at the bottom of the packing.

EFFECTIVE POISSON'S RATIO

Typically, textbooks cite extremes for Poisson's ratio as $\vartheta = 0.5$ for rubber and $\vartheta = 0.0$ for cork. For a linear isotropic homogeneous elastic material, the bulk modulus restricts the Poisson's ratio to be no greater than 0.5. The Poisson's ratio

used in the packing industry is often greater than 0.5. This Poisson's ratio is an "effective Poisson's ratio," because the packing is restrained by the packing box in the radial and tangential direction. Thus, an effective Poisson's ratio results from a Poisson's effect in two directions. Analytically, the effective Poisson's ratio can be determined in terms of the real Poisson's ratio by writing the stress-strain relations for a linear isotropic homogeneous elastic material in cylindrical coordinates, setting the radial and tangential strain to zero, and solving for the radial stress in terms of axial stress

$$\sigma_r = R\sigma_z \tag{9a}$$

where the effective Poisson's ratio, R, is given in terms of the real Poisson's ratio, ϑ, by

$$R = \left(\frac{\vartheta + \vartheta^2}{1 - \vartheta^2}\right) \tag{9b}$$

For packing with an inconel core, $\vartheta = 0.33$, Equation (9b) results in an effective Poisson's ratio of R = 0.49. For graphite, taking ϑ as 0.43 results in an effective Poisson's ratio of R = 0.75. This results in good agreement with the industry-measured effective Poisson's ratio.

NUMERICAL RESULTS

This methodology is examined using a globe valve with a 1/2-in. diameter stem, a 1.0-in. diameter stuffing box, a system pressure of 2,250 psi, a packing length of 1.75 in. (seven 0.25-in. asbestos packing rings), a preload stress of 5,000 psi with a wet coefficient of friction of 0.1 and an effective Poisson's ratio (R) of 0.5. For stem travel in the open direction, Figure 4 produces an approximate $\overline{K} = 0.7$ for a diameter ratio $\alpha = 0.5$. The axial normalized pressure distribution is easily estimated from Figure 2. Actual axial and radial pressure distributions are obtained from Figure 2. The ratio $P_{sys}/(RS) = 0.9$ with a $\overline{K} = 0.7$ results in a normalized seal point of $\overline{a} = 0.85$ from Figure 3. Table 1 provides the valve packing correction factor of 0.14, resulting in an approximate drag force of 96.0 lbf. The exact numerical value is 98.2 lbf.

Results from this example for closed and rotational stem motion are given in Table 2 as Case 1. The rotating term when multiplied by the stem radius represents a torsional component of packing drag for a rotating rising stem. Cases 1 through 5 use the entire packing length of 1.75 in. for the packing material specified. Cases 6 and 7 consists of two end spacers of 0.25 in. each, two asbestos end rings of 0.25 in. each, and three graphite center rings, for a total 0.75 in. The graphite packing, Cases 4 through 7, use a packing preload of 3,850 psi with an effective Poisson's ratio of R = 0.85.

Case 1, wet asbestos, is equal to the dry asbestos, Case 3. This is only true when the valve packing is not fully sealed. Thus, the increased frictional drag is offset by a smaller seal length. Case 2 with a wet stem and dry outer packing, substantially reduces the packing drag. This case may be more realistic than Cases 1 and 3.

Case 4, wet graphite, is substantially less than Case 1, wet asbestos. However, dry graphite, Case 5, is substantially larger than dry asbestos, Cases 1 and 3. This difference results from a larger Poisson's effect for graphite, which provides a larger seal length. The wet composite of asbestos and graphite, Case 7, is a substantial improvement over the first six cases.

In general, downward stem motion always produces the highest drag, and the upward motion, the lowest. The rotating stem is in between. As friction gets smaller, the preload increases, or both. The band decreases between upward and downward drag.

The drag force, presented in Table 2 in parentheses, represents the drag force resulting for zero pressure. This would indicate the packing drag determined during MOV static testing. Comparing these values with the pressure values (i.e., dynamic results) demonstrates a large packing drag difference between dynamic and static conditions. Case 4 is identical with and without pressure because the packing is sealed the full length.

Table 2. Numerical examples (drag force in units of pounds).

Case	μ_i	μ_o	F_{open}	F_{close}	$F_{rotation}$	$F_{upperbound}$
1—Wet asbestos	0.1	0.1	98	291	147	682
			(492)	(607)	(545)	
2—Wet/dry asbestos	0.1	0.2	59	98	74	682
			(404)	(491)	(444)	
3—Dry asbestos	0.2	0.2	98	291	147	1363
			(732)	(1089)	(888)	
4—Wet graphite	0.02	0.02	159	172	165	179
			(159)	(172)	(165)	
5—Dry graphite	0.1	0.1	236	701	353	892
			(479)	(736)	(617)	
6—Dry asbestos/ graphite	0.2/0.1	0.2/0.1	143	416	214	600
			(454)	(543)	(495)	
7—Wet asbestos/graphite	0.1/0.1	0.1	93	97	95	166
			(150)	(161)	(158)	
8—Case1, S = 5,000 psi	—	—	935	953	936	—[a]
			(492)	(607)	(545)	
9—Case1, S = 2,500 psi	—	—	528	521	464	—[a]
			(246)	(304)	(273)	
10—Case1, S = 1,250 psi	—	—	247	—[a]	97	—[a]
			(123)	(152)	(137)	
11—Case4, S = 3,850 psi	—	—	277	280	278	—[a]
			(159)	(172)	(165)	
12—Case4, S = 1,600 psi	—	—	184	180	182	—[a]
			(66)	(72)	(69)	
13—Case4, S = 800 psi	—	—	151	112	147	—[a]
			(33)	(36)	(34)	

a. No seal.

Equation (6), with the correction factor φ taken as one, was used to determine an upper bound drag force. This upper bound appears to be a good approximation for the static conditions and a conservative approximation for dynamic conditions. This upper bound is higher than the industry standard of 1,000 times the stem diameter, which for this valve would be 500 lb. The industry standard of 1,000 times the diameter does envelope the dynamic results, with the exception of Case 5 for downward stem travel. The value of 1,000 times the stem is not conservative for static

conditions. Thus, the present upper bound packing drag is recommended.

It is interesting to note that the "rule of thumb" of 1,000 times the stem diameter was probably developed for a pressurized water reactor with system pressure of 2,250 psi, 5,000 psi preload, using wet asbestos packing. For the stem geometry considered here, with these assumed parameters, Equation (7) results in a maximum packing drag of 707 x L x d_i. For L = 1.75 in., this results in 1,237 x d_i lb, close to the rule of thumb and supportive of the range found in the EPRI Application Guide. An alternate means of examining this industry standard is to equate Equation (5a), in its dimensional form, to 1,000 times the stem diameter and solve for the preload, S. This results in a required preload of 2.4 times the system pressure for a 0.5-in. valve stem and 2.05 times the system pressure for a 5-in. valve stem. The industry typically uses two times the system pressure for preload; however, the known variations in bolt friction produces variations in the preload; thus, sometimes produces the need for increased packing torque to produce proper sealing. At other times sufficient torque exists.

Cases 1 through 7 are based on Equations (1) through (6), which have packing drag based on mechanical preload only. Cases 8 through 10 use the packing parameters of Case 1, but include both a mechanical preload and a bottom pressure preload, as given by Equations (7) and (8). This produces higher drag forces and in Case 10, with a mechanical preload of 1,250 psi, the packing looses its seal for downward stem travel. Cases 11 through 13 use the packing parameters of Case 4, but also incorporate the pressure preload. For these cases, the graphite provides full-length sealing with relatively low packing drag. The pressure sealing packing needs further evaluation.

CONCLUSIONS

The EPRI formulation for packing has been expanded and the computational efficiency improved. Several numerical examples have been considered demonstrating the use of the formulation.

ACKNOWLEDGMENTS

This paper is based on the mathematical model found in The EPRI *Application Guide for Motor-Operated Valves in Nuclear Power Plants*. We extend our appreciation to E. H. Smith, Jr., of ABB-CE and W. Borrero of APS for their technical recommendations. In addition, a special note of appreciation is given to Lynne Penrice for preparing the manuscript.

Packing Force Data Correlations

Stephen M. Heiman
Liberty Technologies

ABSTRACT

One of the issues facing valve maintenance personnel today deals with an appropriate methodology for installing and setting valve packing that will minimize leak rates, yet ensure functionality of the valve under all anticipated operating conditions. Several variables can affect a valve packing's ability to seal, such as packing bolt torque, stem finish, and lubrication. Stem frictional force can be an excellent overall indicator of some of the underlying conditions that affect the sealing characteristics of the packing and the best parameter to use when adjusting the packing.

This paper addresses stem friction forces, analytically derives the equations related to these forces, presents a methodology for measuring these forces on valve stems, and attempts to correlate the data directly to the underlying variables.

INTRODUCTION

The concept behind valve packing sealing is basically a simple one (see Figure 1). The packing material is designed to fill the area between the stem outer diameter and the stuffing box inner diameter. When the material is compressed in the axial direction by the gland follower, it must expand radially into the stem and the inner wall of the stuffing box.

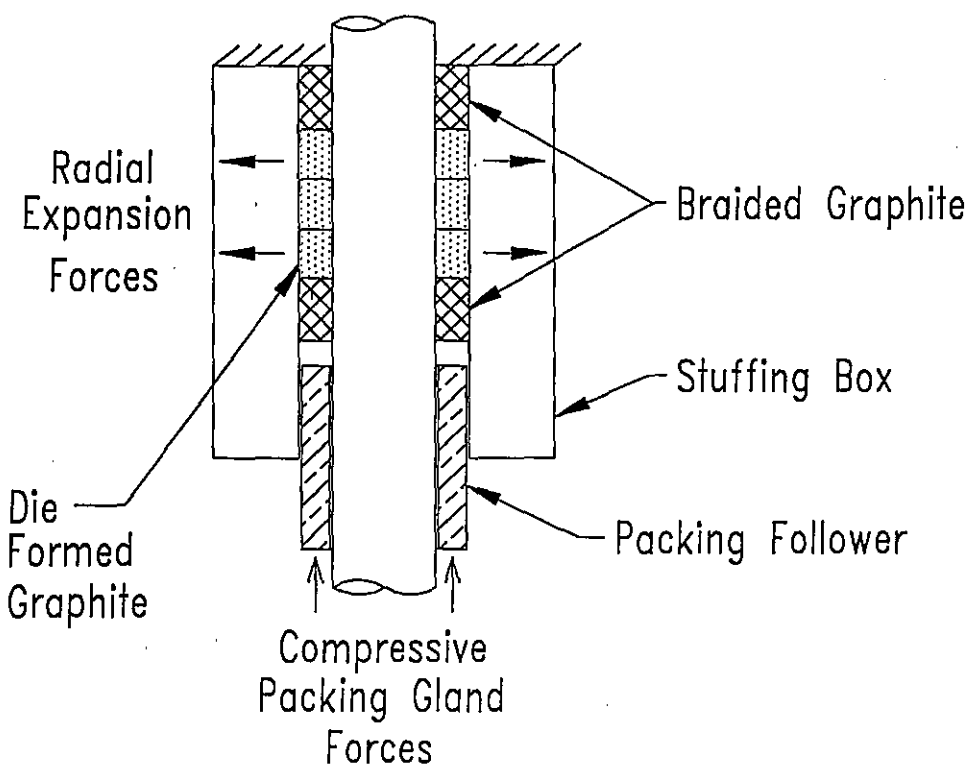

Figure 1. Valve packing seal.

Just as in any sealing methodology, this pressure must be properly set and maintained. Too low will allow the fluid to be sealed to force its way between the seal and the sealing surface, resulting in leakage. Too high will cause the packing material or sealing surface to be damaged or the pressure on the stem to be so great that the valve will be unable to operate under normal operating conditions.

In order to obtain test data to examine some of the packing variables discussed in this paper, a test stand was fabricated with special instrumentation (see Figure 2). The stand was designed to be a basic representation of a rising stem valve at the packing gland area, with a means for measuring and correlating various forces attributable to packing during a stroke.

Before examining the test data, this paper presents a review of the basic force and torque equations involved in packing bolts and packing glands.

Conversion of Bolt Torque to Force

The primary mechanism for delivering compressive force to the packing is through tightening the packing bolts that convert bolt torque to bolt force. Therefore, the analysis should begin with the standard equation for a power screw (Shigley, 1972). This equation employs a basic force summation and uses the thread geometric parameters and coefficients of friction to yield

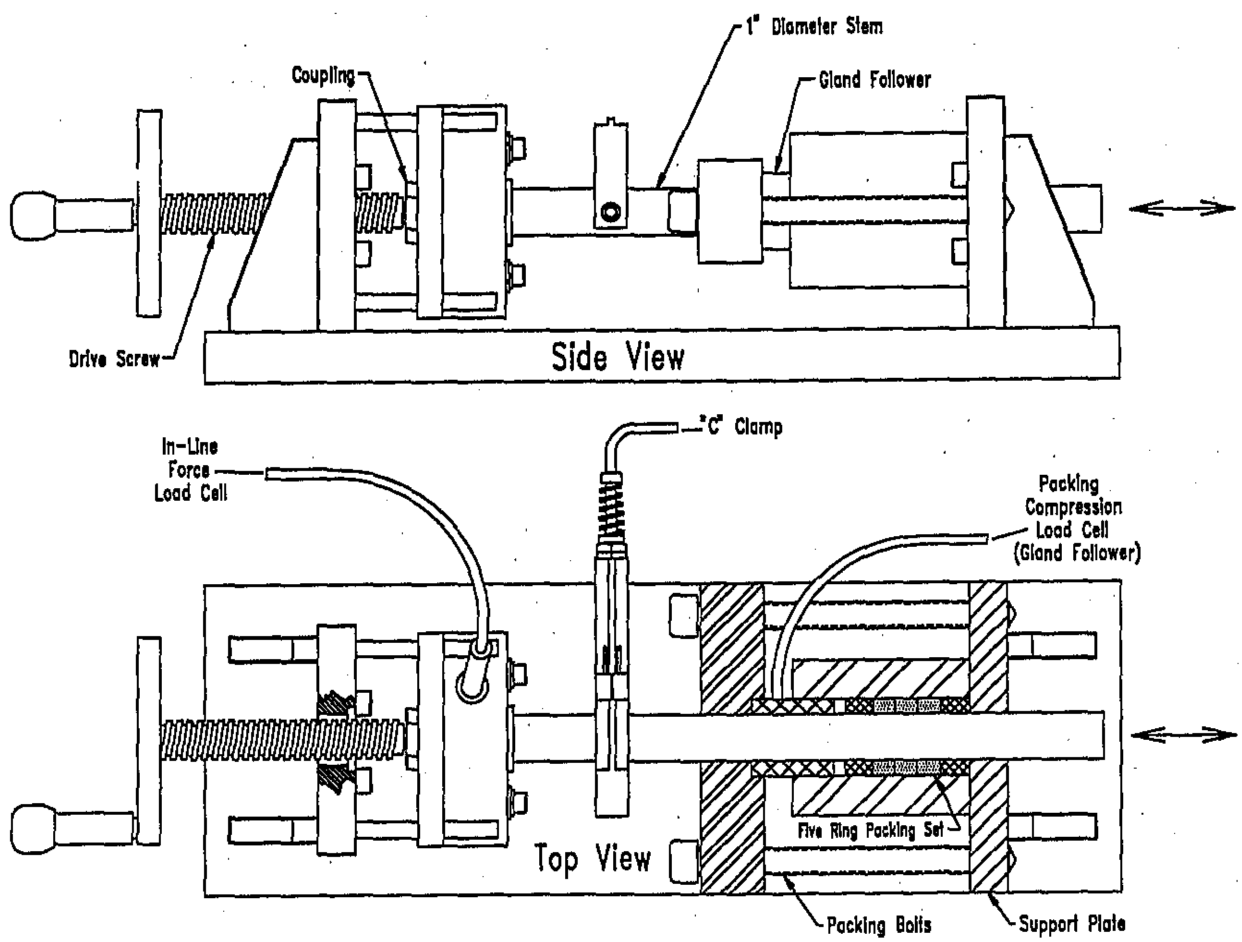

Figure 2. Standard test setup.

$$T = \frac{Fd_m}{2}\left(\frac{1 + \pi\mu_b d_m}{\pi d_m - \mu_b l}\right) + \frac{F\mu_c d_c}{2} \qquad (1)$$

where

T	=	torque applied to bolt
F	=	axial load created by bolt
d_m	=	mean bolt diameter
μ_b	=	coefficient of friction—bolt to nut
μ_c	=	coefficient of friction—collar
l	=	lead of thread
d_c	=	mean collar diameter.

The equation shows that the force to torque conversion is based on definitive, measurable quantities, such as applied torque and bolt dimensions, and also items that must be estimated, such as the coefficients of friction. In general the equation can be estimated in shortened form as

$$\frac{T}{F} = kd_m \ , \qquad (2)$$

where k is a general purpose experimental constant that includes friction and any other items that affect the relationship between bolt torque and bolt force.

As might be expected, there can be a significant amount of variation in the value of k. Experimental data show that the value of k can be expected to scatter typically by as much as ±30%. The scatter can be attributed primarily to uncertainty in the coefficient of friction, which is affected by many variables including material, surface finish, hardness, plating, and lubrication. Note that two coefficients of friction are generally independent of each other. In addition, there is also uncertainty in many of the geometric factors, such as tolerances in the thread pitch and diameter.

An additional item to consider is that k is not necessarily a constant, but may vary as a function of the applied torque. As bolt torque increases, lubricants between the thread and nut collar interfaces may migrate or break down. Also, it is possible that the high surface pressures seen on the surfaces of the threads can cause local plastic deformations that can have either a positive or negative effect on the torque-to-force conversion. In addition, k may be a function of time because the effects of corrosion, foreign material, or loss of lubricant cause an increase in the friction factors.

Another factor not often considered in the accuracy of Equation (1) is the degree to which the applied bolt torque is known. For field application of wrench torques, where potentially many different torque levels may be required, some type of adjustable torque wrench is needed. Typically, this would be either a breakaway or dial gage torque wrench. Although these devices can be calibrated to rather accurate levels, field use adds significant uncertainty. Items that can be easily controlled in a calibration lab, such as angle of the tool to the bolt, knowledge of the exact tool moment arm, and readability of the dial gage, will not be as easily controlled in or reported back from the field.

It should be noted that the effects just described involve only the uncertainty in the torque-to-force relationship immediately after the bolt torque is applied. The dynamic- and time-related effects that a packing bolt sees as the valve operates can definitely add a significant bias to the results. This bias can be controlled to some extent in a packing gland or any application by repetitive load cycles and bolt retightenings so that the packing is properly consolidated. Live loading of the packing is also used to stabilize the performance of the packing over time.

The preceding factors basically describe some of the elements involved in the conversion of bolt torque to force at the bolt. The conversion to compressive force at the packing gland follower for the packing material involves some other elements. Of primary concern is the ability to apply torque and create equivalent axial force in both

bolts simultaneously. Obviously this is not achievable in a practical manner; therefore, one of the bolts will carry a primary load, which may change to the other as the bolts are tightened in an alternating manner. In order to properly balance these unequal forces, the resultant force at the packing gland follower will not be symmetrical, but offset to one side (so that the moments will sum to zero), and the gland will rotate slightly to that side, limited by the tight clearances between the outside diameter of the packing gland follower and inside diameter of the stuffing box. In effect, the packing will be compressed on one side much more than the other. Obviously, this can have an unknown and potentially detrimental effect on the packing.

This process is somewhat self-correcting with continued and repetitive additions of torque and force. By modeling the packing as an elastic spring with uniform properties, the less compressed side should now deflect more for a similar force application and begin to level out the follower. Many small torque applications are the best solution to this problem, but there are practical limitations.

Conversion of Compressive Force to Packing Force

Figure 1 depicts the a typical packing gland. As can be seen, the gland follower applies a compressive load directly to the packing that resides in the area between the stem and the stuffing box inner diameter. In order to prevent extrusion of the packing material under load, clearances between the gland follower to stem, gland follower to stuffing box, and stem to stuffing box are kept to 0.015 to 0.030 in.

Because of these tight clearances and the compliant nature of the packing, the gland follower typically applies an even load to the top ring in the packing, which can be expressed as a stress and is derived as

$$S = \frac{4F}{(d_{sb}^2 - d_{st}^2)\pi} , \qquad (3)$$

where

F = gland follower force (bolt load times number of bolts)

S = axial stress applied to packing

d_{sb} = stuffing box inner diameter

d_{st} = stem diameter.

The packing is a compliant material under axial load that is constrained by the walls of the stuffing box and stem. As the material experiences axial strain from the compressive stress, it also experiences radial strain. As the radial clearances are taken up, the material contacts the stuffing box surfaces. Because the radial strain is now limited, the material begins to exert force on the sealing surfaces.

The radial stresses and axial stresses are related by a material property known as the "Y" value or axial-to-radial transfer ratio. Therefore

$$R = YS , \qquad (4)$$

where

R = radial stress

Y = Y value.

The radial stress can be converted to radial forces based on the surface area of the applied stress. For the valve stem, that force is calculated as

$$F_v = R\ (\pi d_v h) , \qquad (5)$$

where

dv = valve stem diameter

h = packing height.

Finally, the packing friction force for a moving valve stem is simply derived as the friction factor times the normal force calculated as

$$F_p = \mu_v F_v , \qquad (6)$$

where

μ_v = coefficient of friction between the packing material and valve stem.

Based on these derivations, an overall formula equating packing bolt force to packing friction force can be obtained as

$$F_p = \frac{4\mu_v Y F d_v h}{d_{sb}^2 - d_{st}^2} \quad . \tag{7}$$

Normally, this equation is not presented as such, because some of the intermittent variables calculated are required as input. More specifically, one of the variables used by packing vendors for performance criteria is the compressive stress that was previously derived. Presenting the equation in terms of this variable yields the following:

$$F_p = \pi S \mu_v Y d_s h \quad . \tag{8}$$

In this form, the user can input three material properties that are generally available from the packing vendors' specifications—coefficient of friction, compressive stress, and Y value—and two measurable geometric properties—stem diameter and packing height to obtain a theoretical estimate of packing force.

TEST SETUP

The test stand depicted in Figure 2 contains a mechanism for pushing and pulling a round stem through a simulated packing gland. The packing gland contains a stuffing box fastened to a support plate, a gland follower, and a separate gland follower support plate. The action of two bolts secured with the support plate creates the compressive force on the gland follower. The nuts, bolts, and collars were lubricated with a Teflon base gel. The device was designed to use a standard packing size for a stuffing box inner diameter of and 1.50 a stem diameter of 1.00. The packing installed was a standard five-ring set of die-formed graphite rings, including two anti-extrusion braided rings.

The stand uses three separate force measuring devices. The force required to move the stem can be measured by a direct in-line load cell (Lebow) connected to a Portable Strain Meter (Micro Measurements P3500). A second technique used for measuring the same force used the Liberty Technologies' Packing 'nForcer[a]— a diagnostic system that uses a special sensor temporarily clamped onto the stem. The sensor, known as a C Clamp, responds to the diametral expansion of the stem caused when the stem is under load. The diametral expansion is related directly to the stem force using the known stem geometry and stem material properties, yielding the proper stem force in pounds. Because the diagnostic system has a means for physically displaying and storing data traces, it was used as the primary device.

The compressive force on the packing is measured with a gland follower that has been converted to a sensor by applying strain gages and calibrating against a known load. For this application the easiest and most accurate method of calibrating the gland follower load cell used the in-line load cell in the fixture. It was calibrated by merely pulling the stem back into the packing area, placing a support plate across the center hole in the gland follower plate, and creating a force on the plate with an extension rod driven by the load cell, as shown in Figure 3.

Finally, torque on the gland bolts is measured using a standard breakaway torque wrench with an adjustment mechanism to vary the torque set point.

The test procedure was essentially designed to simulate a standard field packing installation.

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a. Registered trademark by Liberty Technologies, Inc.

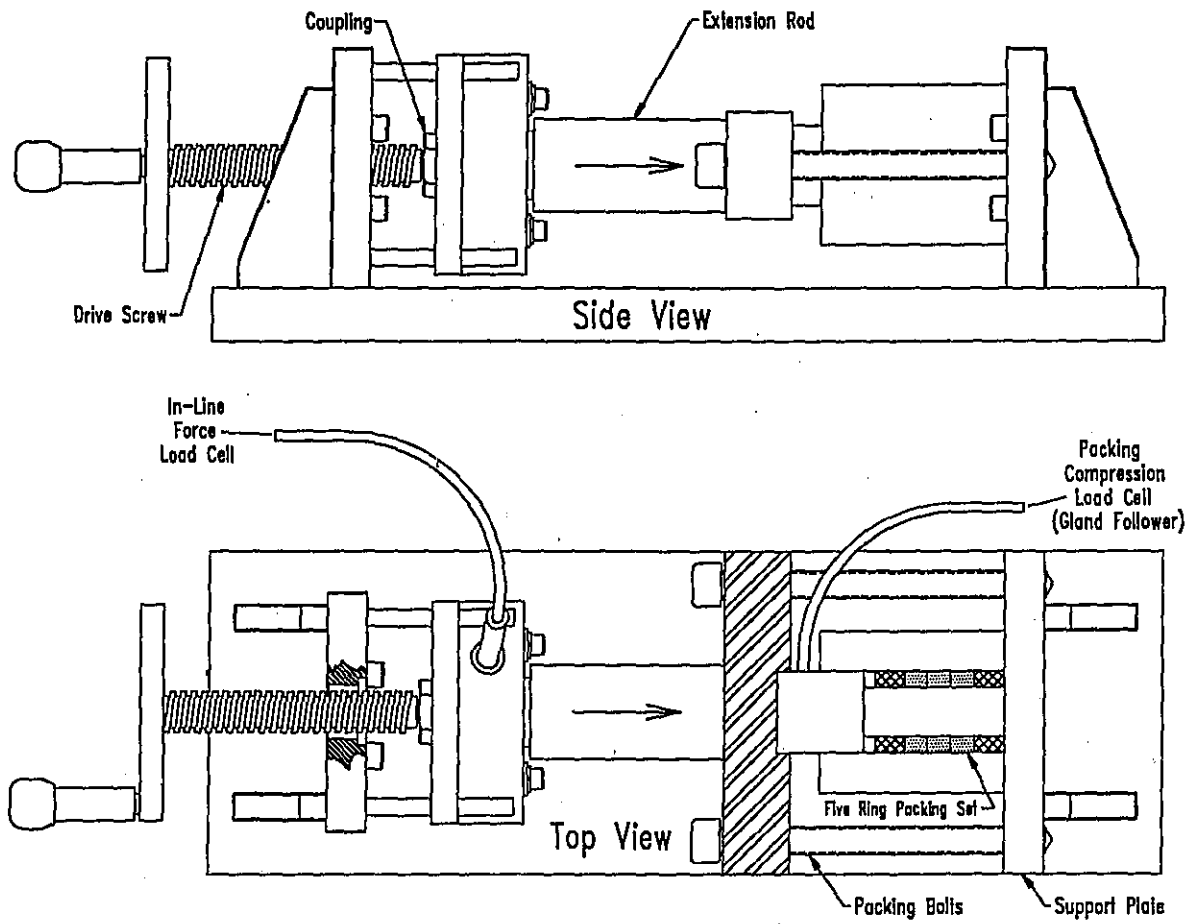

Figure 3. Calibration setup.

The test procedure consisted of the following steps:

1. With the packing installed and no load on the packing gland, note the zero level of the strain gages in the packing gland follower.

2. Apply a load to the stem to compress the packing, assisting the action of the packing gland. Apply torque alternately to each of the two bolts with the torque wrench until the breakaway occurs. Proceed slowly so as not to overrun the breakaway point, thereby applying additional torque.

3. Cycle the stem by establishing movement in both directions. Complete 10 times. Retorque the bolts, as in Step 2, to consolidate the packing after the initial, fifth, and tenth cycles.

4. Begin the data acquisition. Take data with either the in-line load cell or by using the diametral strain measuring device (C-clamp). Starting from a zero load position, advance the stem slowly in the forward direction to establish movement, stop for approximately 1–2 seconds, and pull the stem in the backward direction.

5. Move the stem to the zero load position and note the force level of the packing gland follower.

6. Review the acquired data traces to obtain the opening and closing packing friction forces. Figure 4 shows a typical acquired trace marked for packing forces.

7. Repeat the procedure for another torque setting.

```
LIBERTY TECHNOLOGIES, INC. PACKING 'NFORCER REPORT

Test Date and Time: 1994/05/02  12:52:29

Test Description:

Test 3 with 8 ft-lbs and no zero. Nut and bolt

Opening Force:        428 lbs
Closing Force:        407 lbs

Clamp Location:       Unthreaded Region

Plant:                LIBERTY
Unit:                 TEST
System:               ENGINEERING
Valve Tag:            PACKING

Stem Diameter:        1.000 Inches
Threads Per Inch:     3.0
Thread Starts:        1
Thread Type:          General Purpose Acme
User-defined Deff:    0.000 Inches
User-defined TCF:     0.000
Stem Material:        17-4 PH Annealed 16% Cr; E/v = 99.0
User-defined E/v:     0.0 x 1,000,000 psi
Manufacturer:
Packing Scheme:
Keyway Area:          0.0 Sq. Inches

Clamp ID:             A1134
Sensitivity:          0.5581 uV/V/uInch
Clamp Type:           C-Clamp
Last Calibration:     1993/11/16
Calibration Due:      1994/5/16
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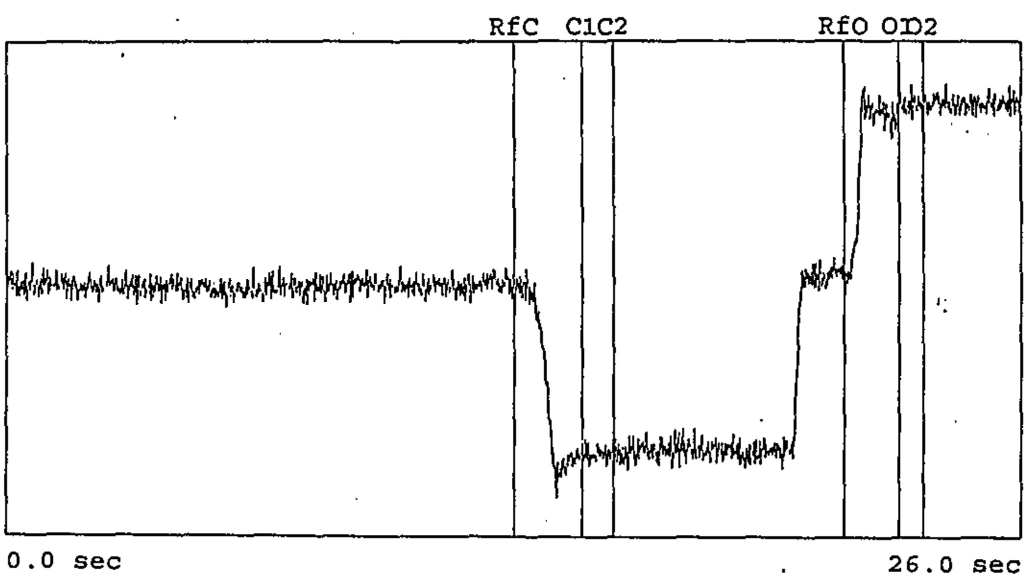


Figure 4. Typical marked data trace.

EMPIRICAL DATA

The data are presented here as tables. Data were taken at five or six different torque values for the bolts and presented in terms of the bolt torque, compressive stress on the packing, and packing friction.

Tables 1 and 2 depict the conversion of the bolt torque to axial force (as measured by the packing follower load cell). Two different bolt configurations were tested, 1/2-13 and 5/8-11. For both bolts, the data are presented in terms of the average gland load at the particular bolt torque setting. Eight data runs were taken for each of the bolts using the procedure described above. The scatter in the data is also presented in terms of the 2 sigma value of the data points divided by the mean.

If the data are analyzed using a linear regression and a best fit straight line, the average slope can be obtained. Inputting this ratio of force to torque into Equation (1), the average friction factor can be obtained (assume the same coefficient for thread and collar friction). These values are 0.22 for the 1/2-13 bolt and 0.14 for the 5/8-11 bolt. The difference in the bolt designs probably accounts for the difference, although both friction factors are within expected values. The 1/2-13 is a standard off-the-shelf nut and bolt design, while the 5/8-11 used a custom-tapped hole in the support plate for the nut, likely providing tighter clearances and increased thread contact area. In addition, the collar under the 5/8-in. bolt appeared to have a better surface finish than the 1/2-in. nut.

The coefficient of friction was further examined by replacing the 1/2-13 nut, which was made of 18-8 stainless steel with nuts of other materials. Tests were run with plain galvanized steel and zinc-plated steel. The coefficient of friction was 0.17 for the galvanized steel and 0.12 for the zinc-plated steel. For these nuts, there is an apparent high degree of material-related variability in the coefficient of friction.

The compressive stress variation for constant torque values ranges from ±1 to ±24%. These numbers can be interpreted as repeatability errors for a given bolt configuration and installation. This error is reflective of the problems discussed earlier in being able to consistently convert torque to force in a bolted joint. It should be noted that this error is independent of the error incurred in attempting to estimate the proper value of the

Table 1. Torque/force conversion for 1/2-13 bolt.

Parameter	Values					
Bolt torque (ft-lb)	4	6	8	10	12	14
Average compressive stress (psi)	615	989	1,352	1,669	2,004	2,175
Range of compressive stress (±%)	9.5	11.3	14.1	14.0	14.3	8.2

Table 2. Torque/force conversion for 5/8-11 bolt.

Parameter	Values					
Bolt torque (ft-lb)	4	6	8	10	12	14
Average compressive stress (psi)	683	1,049	1,633	2,294	2,692	2,892
Range of compressive stress (±%)	24.2	14.2	20.5	15.9	12.0	1.3

friction factors when using the theoretical equations to estimate the force-to-torque relationship. This is a bias error and can also be significant. As an example, if the coefficient of friction is assumed to be 0.15 when it is actually 0.17, the forces would be 24% (using the 5/8-in. bolt) lower than the prediction, and also subject to repeatability. The combined error, using the two 24% values and a root sum of the squares methodology, is 34%, which is close to the value discussed earlier for Equation (2).

The second phase of the evaluation involved the relationship of the compressive force on the packing to the measured friction force. The data are presented in Table 3 and should be interpreted in the following way. Using the equation developed for packing friction force as a function of material and geometric properties [Equations (8)], solve this equation for the Y value (axial to radial transfer ratio). This equation is

$$Y = \frac{F_p}{\pi S \ \mu_v \ d_s \ h} \ . \tag{9}$$

Inputting the data from the testing (packing friction–average opening/closing compressive stress), the known packing geometric parameters, and assuming a value of 0.1 for the coefficient of friction between the packing material and valve stem, the value of Y can be calculated.

The data show a significant scatter in the value of Y, ranging from ±9 to ±27% for the average values. The data tend to increase from the lower torque values and stabilize at the medium to high torque values. The average value of Y for this set of data is 1.067, which is higher than the value of 0.8 specified in the EPRI packing report (1988, pp. 4–39). In this analysis where a constant value of 0.1 has been assumed for the coefficient of friction, the Y values presented represent a relative number indicating the ability of the packing to convert compressive stress to packing friction. The actual value of Y (axial to radial stress conversion) must be lower and the coefficient of friction higher, because a value greater than one is not possible. An assumption of 0.13 for the coefficient of friction yields a calculated Y value of 0.820, closer to the EPRI value.

It was also noted that Y generally increased during the first 10 tests done with the new packing, then stabilized. This would indicate that some break-in factor is probably present, during which the load applied to the packing causes it to undergo some permanent change in its elastic properties.

The scatter in Y can most likely be attributed to the fact that as the bolt torque is increased for a subsequent test, the compressive stress does not necessarily distribute itself in the same pattern as the previous level. The distribution is generally considered to be exponential, which decays from the top rings to the bottom, and is affected by the friction between the packing and the stem and stuffing box wall. This phenomenon probably bears a strong relationship to the fact that the compressive force is developed through the action of two offset bolts, which will likely be carrying different loads and have gone through different tightening sequences.

Another item that could be examined is the bolt torque as a function of the packing friction force. This relation is shown in Tables 4 and 5.

Table 3. Calculated Y values.

	Bolt torque (ft-lb)					
Bolt torque (ft-lb)	4	6	8	10	12	14
Average Y value	0.0889	1.017	1.087	1.089	1.071	1.101
Range of Y value (± %)	22.3	27.0	22.4	20.9	13.9	9.5

Table 4. Torque/packing force relation for 1/2-13 bolt.

Parameter	Values					
Bolt torque (ft-lb)	4	6	8	10	12	14
Average packing force (lb)	166	291	428	550	647	759
Range of packing force (± %)	13.0	29.8	24.7	19.9	7.8	1.9

Table 5. Torque/packing force relation for 5/8-11 bolt.

Parameter	Values					
Bolt torque (ft-lb)	4	6	8	10	12	14
Average packing force (lb)	196	351	578	779	904	955
Range of packing force (± %)	29.7	32.7	25.3	11.3	8.4	6.9

These data present a summation of all previous factors combined into one entity, relating the input factor of bolt torque to the output factor of packing friction. The data reflect some of the nonlinearity in the value of Y for the lower torque values, and also appear to give more consistent results for the two higher torque values (12 and 14 ft-lb).

CONCLUSIONS

This study has presented test data from a fixture meant to simulate an actual valve packing setup in the field. Three elements were measured—bolt torque, compressive load on the packing, and packing friction force—and their relationship to each other was analyzed using both theoretical and empirical techniques.

The performance of valve packing must always be analyzed in terms of two main criteria, operability and leakage. Operability is basically a straightforward assessment of the maximum anticipated running loads expected during the stroke of the valve and the valve's ability to perform its intended function in spite of those loads. Using some of the equations developed and estimating the values of certain coefficients, the running load values can be inferred from the bolt torque. This procedure must be done very carefully because there is a significant potential for error not only in estimating the basic value of the various coefficients but also in the repeatability of force data for any given setup. The value can also be found using more direct means, as was done on the test fixture by using the Packing 'nForcer diagnostic system and the stem mounted C Clamp designed to measure axial force. This method eliminated the uncertainty in estimating coefficients.

The assessment of the packing's ability to seal, based on some of the measured parameters, is more difficult. This is a function of several factors, many of which are beyond the scope of the testing done in this study. One of the primary factors, however, is the gland load and resulting compressive stress examined in the testing. There are different guidelines used by packing vendors to recommend a compressive stress; in general, they are based on properties of the packing material, as well as the system pressure to be sealed.

The direct packing force measurement can be a valuable tool in assessing the gland load. As shown in the test data, there is a correlation between the packing force and gland load, although this relationship is not always predictable. In order to get the most consistent results when using the packing force measurement, use the following guidelines:

1. Follow good practice in installing and consolidating the packing, as recommended by the manufacturer. Use the direct packing force measurement to determine when consolidation has reached an acceptable level. The properties of brand new packing may change during the initial compression cycles.

2. Attempt to characterize the packing and packing gland design before installation so that the test data can be compared with theoretical values. The packing vendors who supply recommended bolt torques may also have recommended packing forces. If that information is not available, use the theoretical equations to convert the bolt torques to expected packing forces. Examine the reasons for any large discrepancies (such as poor torque-to-force conversion from corroded packing bolts).

3. Take readings at different bolt torque levels. Generally, the packing force should be a linear function of the bolt torque, so incremental torque applications can be ratioed directly to the packing force. Significant nonlinearity in this relationship could be an indicator of too little or too much force on the packing.

4. Take multiple readings at the same bolt torque level, reducing the error associated with nonrepeatability of the data.

5. Recognize that the manufacturer's recommended bolt torque specifications are likely to be conservative. The problems in trying to accurately estimate the force in a bolt based on torque are well known. Having the knowledge of the packing force may allow a somewhat more aggressive approach in using a higher gland force to potentially eliminate a future leaker.

REFERENCES

Shigley, J. E., 1972, *Mechanical Engineering Design*, McGraw Hill, pp. 295–299.

Bickford, J. H., 1990, *An Introduction to the Design and Behavior of Bolted Joints*, pp. 133–144 and 184–194.

Foster-Miller, Inc., 1988, *Valve Stem Packing Improvements*, prepared for the Electric Power Research Institute, pp. 4-35–4-40.

Maintenance Planning and Performance Software for Valve Packing Programs at Nuclear Power Stations (ValvePro Version 2.5)

Neil D. Hutcheson
Georgia Western, Incorporated

ABSTRACT

ValvePro Version 2.5 for Windows[a] was developed to help power plant maintenance personnel improve maintenance productivity and quality through a simple, attractive software program which can be installed on personal computer systems in use at many utilities today. This paper explains the functions of this software and how it can be used by a maintenance organization as a foundation for a consistent, effective valve packing program utilizing sound packing principles.

INTRODUCTION

Maintenance personnel are under constant pressure to improve maintenance quality and productivity with ever-shrinking staffs. ValvePro Version 2.5 for Windows[a] was developed to help power plant maintenance personnel improve maintenance productivity and quality through a simple, attractive software program which can be installed on the popular personal computer network systems in use at many utilities today. In implementing and managing a valve packing effort, certain data can be of tremendous help in identifying required packing sizes as well as providing trending data for various types and classes of valves.

DATA INPUT

The program input is broken into four basic input screens, all available from the main menu (Figure 1). These are Nameplate, Mechanical Dimensions, Mechanical Information, and Engineering Data. This makes for a logical sequence when entering data and for finding data when required. The main menu also includes Survey Sheet(s), Data Sheet, Inquire, Exit, and Maintenance, all which are discussed throughout this text. (Figure 1.)

Nameplate data (Figure 2) has 14 input fields of which only one, the Tag Number, is required data. The manufacturer, model number, valve type, valve size, actuator manufacturer, actuator model number, and actuator type are typical Windows pick lists which require only a point and click by the mouse to input. The valve function, valve elevation, location, and valve drawing number require keyboard inputs. An additional pick list field for the valve's status is included which allows the user to classify the valve being planned. Examples might be outage status, material type, live-load hardware, etc. (Figure 2.)

The Mechanical Dimensions screen (Figure 3) has seven required fields to be input. The fields are stem diameter, stuffing box bore, stuffing box depth, leak-off port depth, lantern ring height, stud diameter, and number of studs. Each field requires a measurement to the nearest thousandths of an inch (0.001). The actual field surveys are then rounded to the closest packing size referenced in a table loaded in the computer. The rounding limits can be adjusted as the user's plant specifications require. A graphic valve cross-section identifying the proper measurement references is shown on the screen for review.

a. Windows is a registered trademark of The Microsoft Corporation.

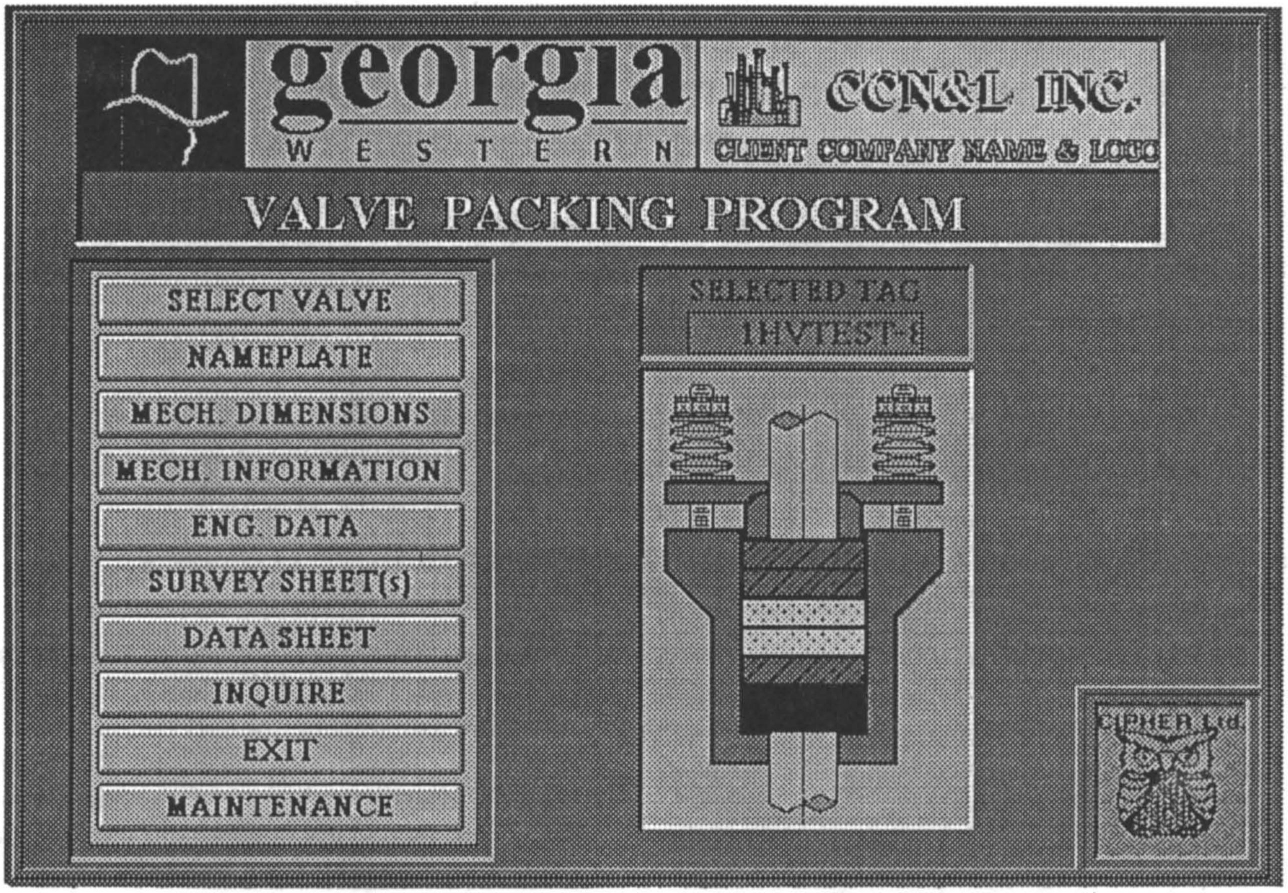


Figure 1. Main menu screen for ValvePro Version 2.5.

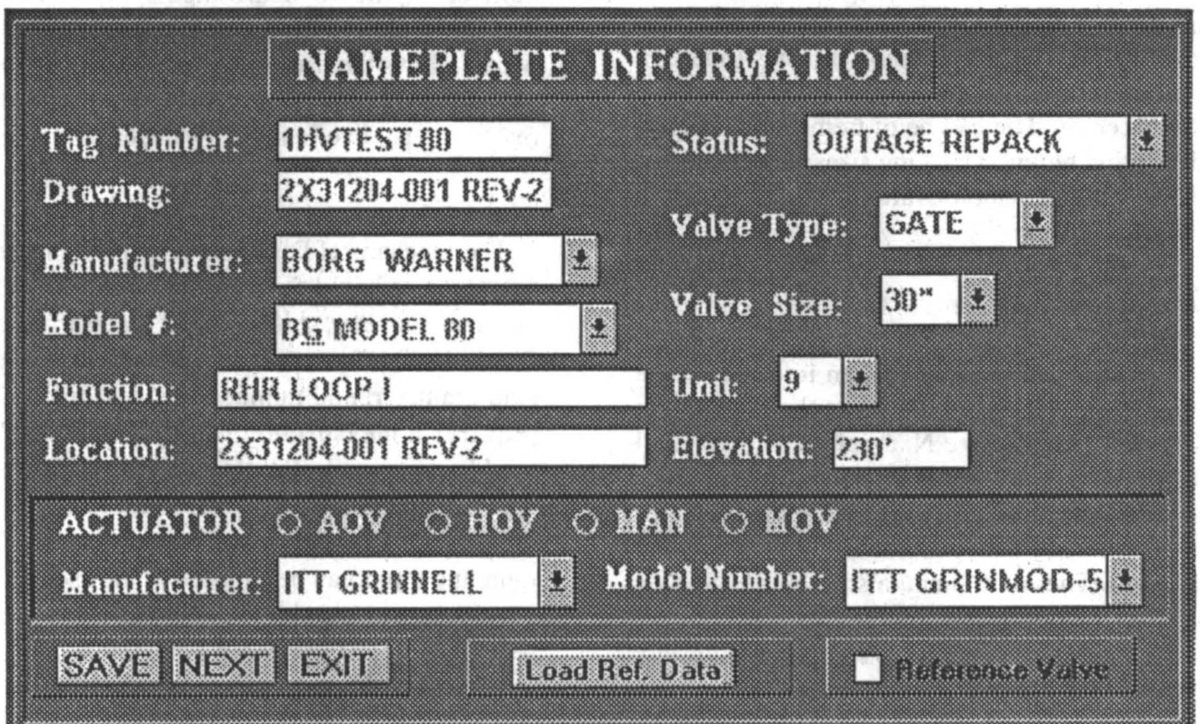


Figure 2. Nameplate data screen ("NAMEPLATE INFORMATION").

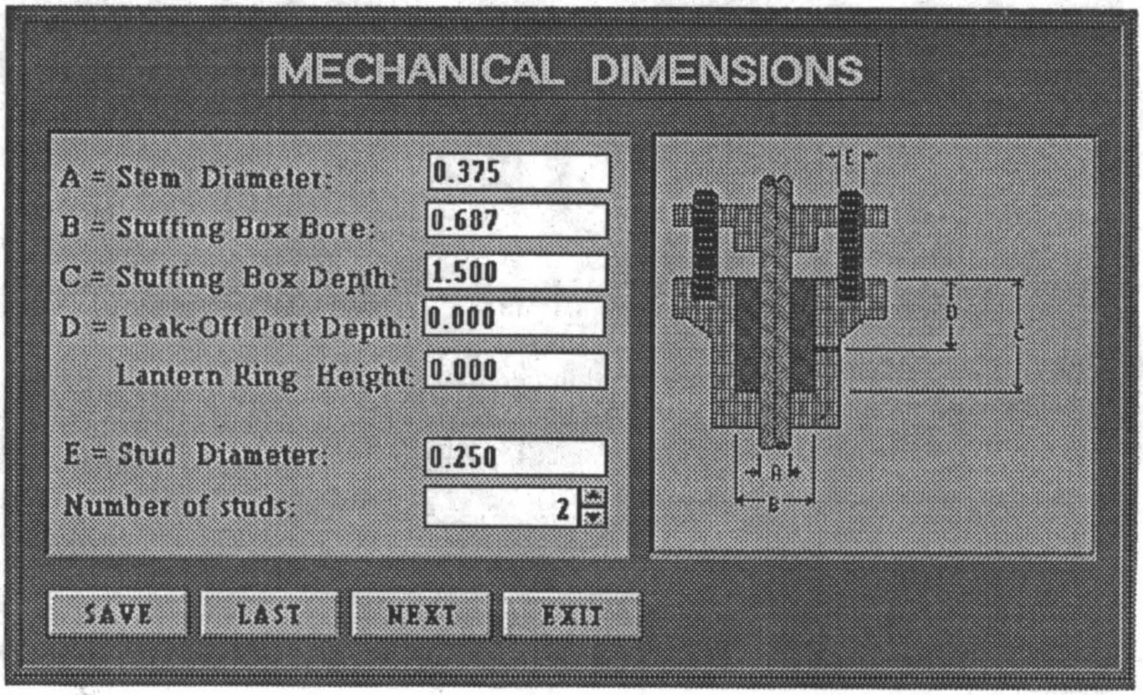


Figure 3. Mechanical dimensions screen showing seven required input fields and valve cross-section with measurement references A, B, C, D, and E.

Mechanical Information (Figure 4) requires the user to answer four questions by highlighting a button with a mouse command. The first question is Stem Orientation: Vertical or Other. Selecting Other changes the packing configuration to select both upper and lower carbon bushings if space is available. The second question is the status of the Leak-off Port: Not Applicable (N/A), Non-Active, or Active. If Non-Active is selected, the user must address the third question to indicate whether the lantern ring was Removed or Dropped (to the bottom of the stuffing box). The computer logic gives precedence to lantern ring placement if the leak-off is active and ensures the lantern ring is over the leak-off port opening. If the leak-off is active and ensures the lantern ring is over the leak-off port opening. If the leak-off port is not active and space is available, software logic places the packing below the leak-off port so that system pressure will not be present at the leak-off plug. The final question is if the bottom of the stuffing box is Flat or Beveled. Beveled stuffing boxes require compensation for the bevel, which, if left unattended, can damage the carbon bushings or packing rings. Additionally, forty lines of comments are available for the user to describe valve specifics and history. A validation field and date of entry are also shown on this screen (Figure 4).

Engineering Data (Figure 5) consists of five input fields, of which the following three fields are required and selected from a pick list; packing end ring type (braided or Composite™), live-load configuration, and desired gland pressure. System pressure, design stem friction, and actual stem friction are optional fields and require keyboard input. Other fields present on the Engineering data screen are calculated stem friction, gland nut torque, packing size, number of packing rings, and packing configuration. All of these fields are calculated automatically and are not editable.

Formulas used in calculational fields and tolerances for packing sizes are based on accepted practices used at over 40 nuclear power utilities throughout the United States and Canada.

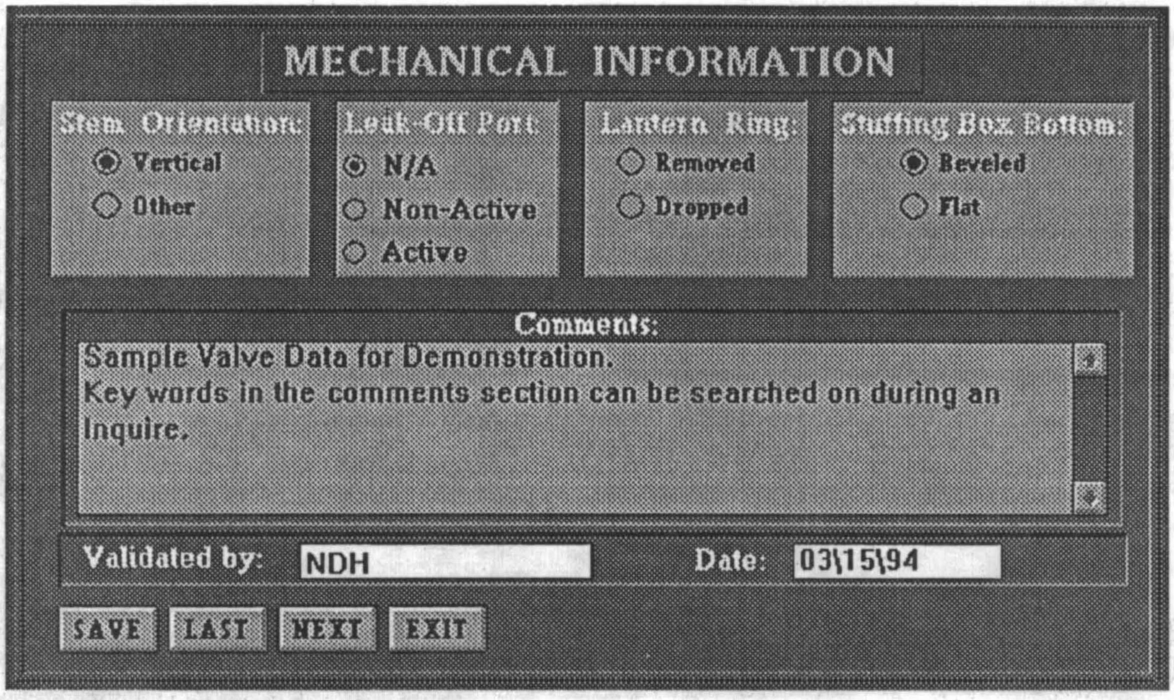


Figure 4. Mechanical information screen showing question selection options for Stem Orientation, Leak-Off Port, Lantern Ring, and Stuffing Box Bottom.

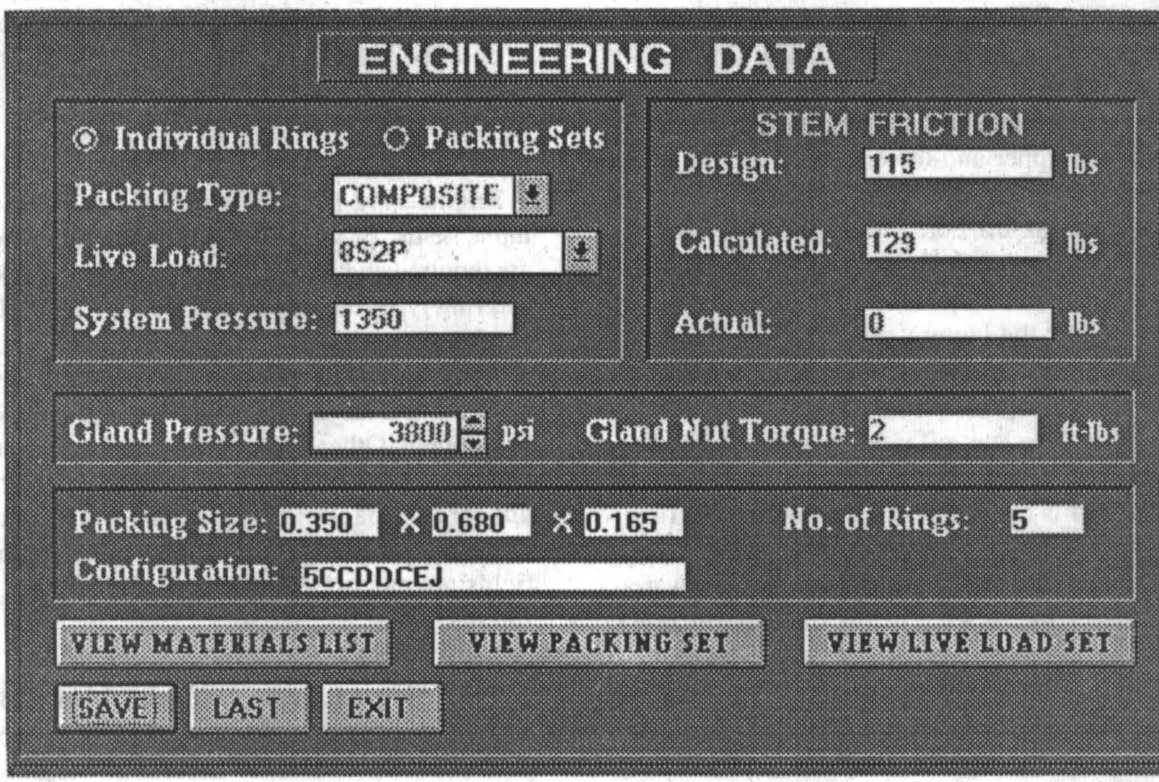


Figure 5. Engineering data screen showing input fields and sub-menu view screen options.

A submenu within the Engineering Data Screen (Figure 5) are three separate screens: View materials List, View Packing Set, and View Live Load Set. View Materials List (as selected from the Figure 5 screen and shown in Figure 6) identifies the packing size and cross-section for the dieformed graphite seal rings, the end rings selected and the quantities of each. It also identifies the size and height of the upper and lower carbon spacers (bushing), if needed; the live load spring configuration and quantities; and the size, length, material, pitch and quantity of bolting material (studs) which may be required for the live load installation. Each of the above is also identified with the vendor part number and the stock number that the utility has assigned. View Materials List is an exact duplicate of the lower portion of the Valve Packing Data Sheet (Attachment 1).

View Packing Set (Figure 7) shows the graphics printed to the middle left section of the Valve Packing Data Sheet (Attachment 1).

View Live Load Set selected from Figure 5 shows the graphics (Figure 8) of the live load springs as they are to be installed on the valve and also printed to the Data Sheet (Attachment 1) in the middle right section. The Figure 8 screen also requires the user to input the spring and stud part numbers, stock numbers, and descriptions as required.

MATERIALS	Quantity	Description	Vendor	Stock Number
DIEFORMED GRAPHITE RING	2	2.250 X 3.250 X 0.500	1411120	1411120-4111
BRAIDED GRAPHITE RING	N/A	N / A		
COMPOSITE GRAPHITE RING	3	2.250 X 3.250 X 0.500	1511120	1511120-4111
CARBON BUSHING ------- UPPER	N/A	N / A		
LOWER	1	2.250 X 3.250 X 1 3/8	91120	91120-4111
LIVE LOAD ASSEMBLY	2	Live load set 852P	51141	51141-4111
GLAND NUT STUDS	2	0.5/8in. - 11 TPI, Length: 10 in.		

Figure 6. View materials list screen.

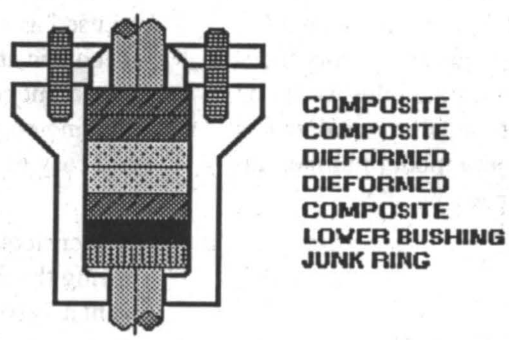


Figure 7. View packing set screen.

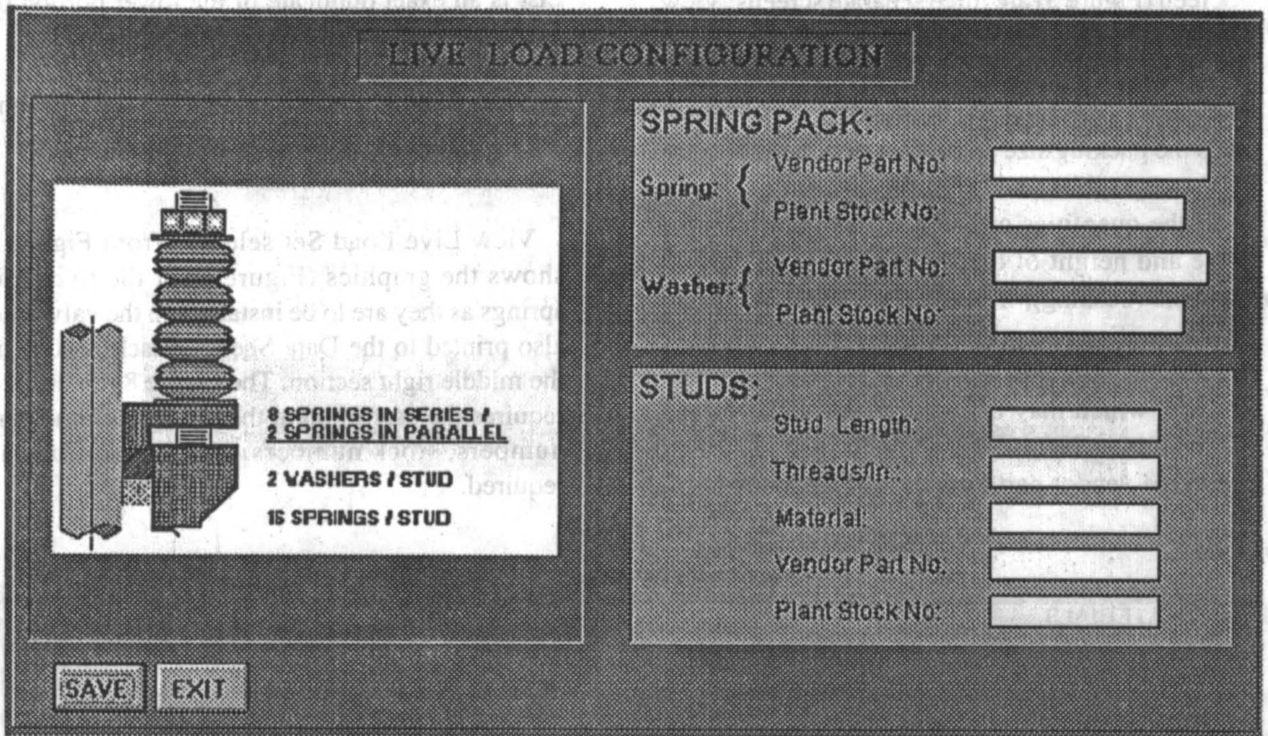


Figure 8. View live load set ("LIVE LOAD CONFIGURATION").

DATA MANIPULATION

Once the data have been input, several options are available to support the user in manipulating the data for planning material requirements, maintenance time, or department coordination. An Inquire screen (Figure 9) is available (selected from the main menu, Figure 1), which allows various levels of information search to occur. Options for inquiry are valve status, manufacturer, model, size, actuator type, stem diameter, stuffing box bore, and any key work or phrase (such as "location" or "elevation") within the comments section. The user can easily create a very specific list of valves requested by any functional group within the plant. Examples would be all MOVs in a certain location, all manual valves of a certain manufacturer and model number, all valves with the same size packing, etc.

DATA OUTPUT

Data output has five options: Valve Packing Data Sheet, Survey Data Sheet, Valve Packing Summary, Valve Packing Summary-Two, and Material Specifications. (Refer to Attachments 1, 2, 3, 4, and 5.) All five formats can be chosen from the Inquire screen (Figure 9) after a search has been completed by selecting Reports. The Data Sheet and Survey Sheet can also be printed from the main menu (Figure 1) for the valve designated by the tag number displayed.

Most important is the Valve Packing Data Sheet (Attachment 1), which is generated when the required data are input for a particular valve. The data sheet is a concise, attractive report which can be used as a packing installation reference for the mechanic and as an "As left" drawing suitable for document control. The graphics which depict the packing configuration and live load are complete and easy to view representations.

If critical information fields have been omitted during the initial data input, the software will not print a Valve Packing Data Sheet, but prompts the user to print a Survey Data Sheet (Attachment 2), which shows the user the reference points for

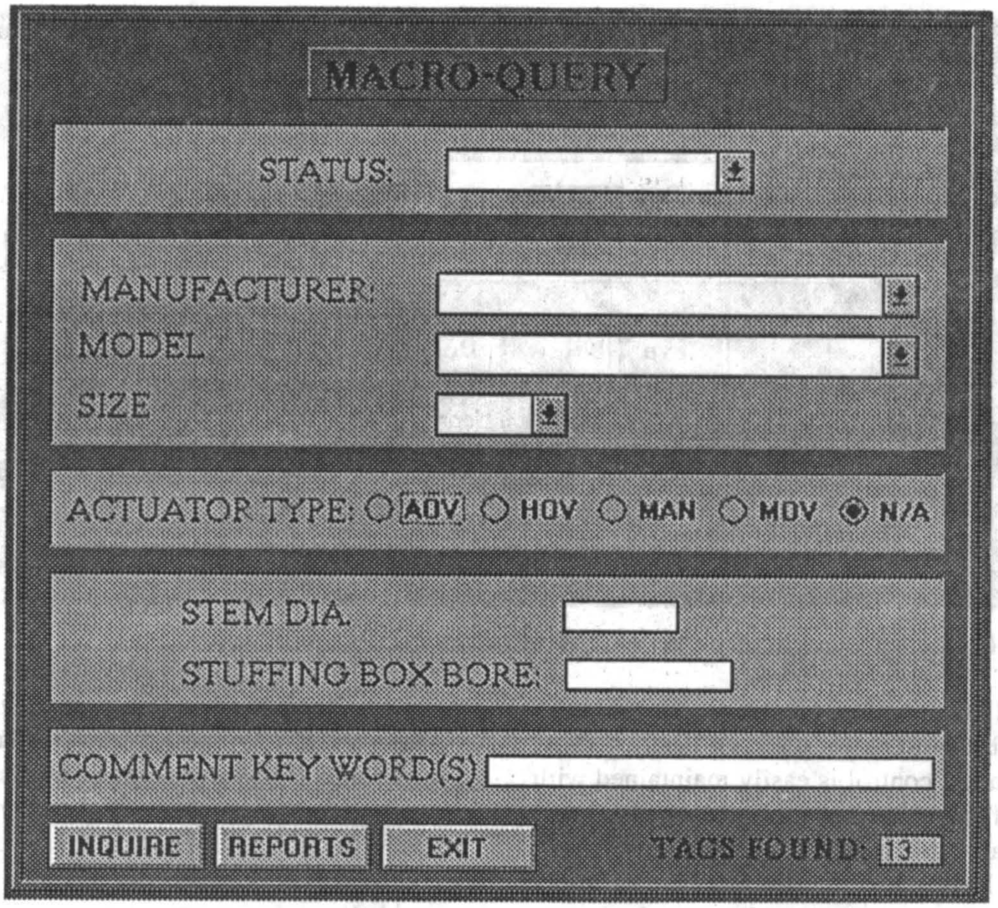


Figure 9. Inquire screen ("MACRO-QUERY") as selected from main menu (Figure 1).

surveying data in the field. The survey sheet is then brought back to the valve program manager for input of the remaining data. When printing the Data Sheets, the software asks the user for the applicable work order number so that it can also be assigned and tracked within the software.

MAINTENANCE

The Maintenance Screen (Figure 10) accessed from the main menu allows the user to add, modify, and delete existing vendor part numbers and perform the same tasks with associated stock numbers assigned by the utility.

SECURITY

A security feature is an integral part of ValvePro. A nonauthorized user may view and print existing data, but any editing or additions to the data base will not be saved. Accessing the Security feature is obtained by clicking the mouse on the pictorial of the valve which appears on the main menu (Figure 1). The initial user assigns and verifies a password for the system and enters the password at the beginning of each session. A lightly visible "Locked" or "Unlocked" is displayed at the top left corner of each input page so the user is reminded of the status of the software.

SUMMARY

By utilizing the ValvePro software, a utility can gain several benefits. The user can easily maintain control over the material requirements of a valve repacking effort, saving the utility inventory costs and expediting fees. The software promotes consistent packing configurations and torquing, which will cause less confusion within

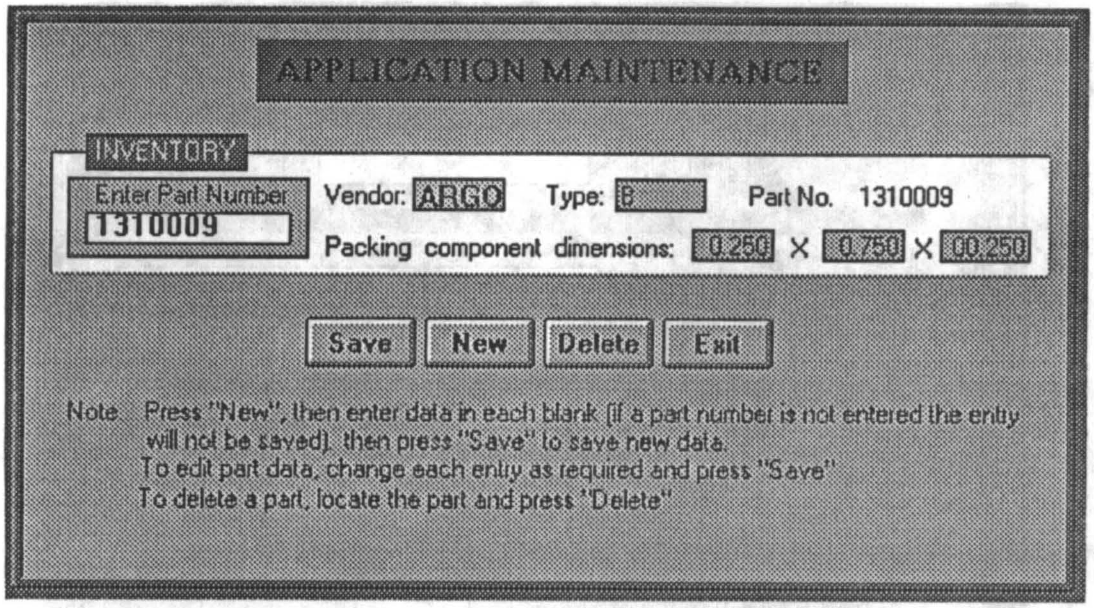


Figure 10. Maintenance screen ("APPLICATION MAINTENANCE").

the installation teams, creating better maintenance habits and better packing performance. Configuration control is easily maintained with the use of the software. The software also lends itself to actuator maintenance programs, (i.e., Generic Letter 89-10 and AOV operability issues) by estimating packing forces and ensuring available thrust requirements can be met prior to packing installation, not after. The software was written so that the user needs little experience with packing designs, thus allowing more flexibility within the maintenance staff. However, knowledge of valves and packing principles is of benefit. Minimal training is required to run this software.

With the recent popularity of network computer systems, the software is widely available within the plant or offsite to corporate locations through network ties. As mentioned, a password can be created which will allow only certain users to input and edit data, but all may view and print data and thus allow the maintenance shop to have instant access to data when preparing to initiate a work order. Ultimately, the maintenance organization will invest less time planning repacking work orders, be more consistent and concise in the instruction, and achieve better packing performance, and thus facilitate doing more work, faster and better.

ACKNOWLEDGMENTS

The author is a co-owner of Georgia Western, Incorporated, whose business was developed by initiating and supporting valve packing programs at U.S. utilities. Formulas, packing configurations and material characteristics are based on products and principles that have been widely used and accepted at over 40 nuclear sites and hundreds of fossil fuel stations in the United States and Canada. ValvePro is exclusively marketed and distributed by Georgia Western, Incorporated, which is located at 2641-A Due West Road, Kennesaw, GA 30144 (404)426-6070.

Attachments

Attachment 1

EQUIPMENT TAG NUMBER: DEMO 1

RITCHIE S.E.S.

VALVE PACKING DATA SHEET

WORK ORDER NUMBER: WO#

NAMEPLATE INFORMATION

TAG NUMBER: DEMO 1
MANUFACTURER: WESTINGHOUSE
MODEL NUMBER: 4GM88FND
SIZE: 4" TYPE: GATE

ACTUATOR

TYPE: MOV
MANUFACTURER: LIMITORQUE
MODEL NUMBER: SB 0

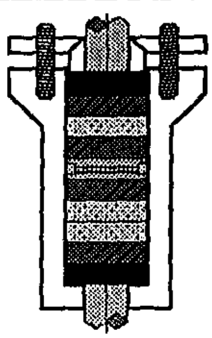
FUNCTION: RCS LETDOWN STOP VALVE
LOCATION: BELOW STEAM GEN. C
SYSTEM PRESSURE: 2485 UNIT: 1
ELEVATION: 19'

ENGINEERING INFORMATION AND DATA:

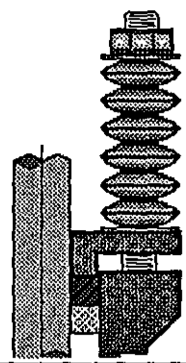
STEM DIAMETER:	1.250 in.	GLAND NUT TORQUE:	28 ft-lbs
STUFFING BOX OD:	1.750 in.	GLAND PRESSURE:	4500 psi
STUFFING BOX DEPTH:	4.250 in.	DESIGN FRICTION:	0 lbs
STUD DIAMETER:	0.625 in.	CALCULATED FRICTION:	1160 lbs
NUMBER OF STUDS:	2	ACTUAL FRICTION:	0 lbs

COMMENTS: Example of a nuclear valve application with an active leak off system and live load hardwa re.

PACKING CONFIGURATION



UPPER BUSHING
COMPOSITE
DIEFORMED
COMPOSITE
LANTERN RING
COMPOSITE
DIEFORMED
DIEFORMED
COMPOSITE
LOWER BUSHING



12 SPRINGS IN SERIES
2 WASHERS / STUD
12 SPRINGS / STUD

MATERIAL SPECIFICATIONS : ARGO

MATERIALS	QUANTITY	DESCRIPTION	VENDOR	STOCK NUMBER
PACKING SET (7 Rings)	1	1.250 X 1.750 X 0.250	2570580	NOT IN STOCK
DIEFORMED GRAPHITE RING	3	1.250 X 1.750 X 0.250	N/A	N/A
BRAIDED GRAPHITE RING	N/A	N / A	N/A	N/A
COMPOSITE GRAPHITE RING	4	1.250 X 1.750 X 0.250	N/A	N/A
CARBON BUSHING UPPER	1	1.250 X 1.750 X 15/16	90580	NOT IN STOCK
LOWER	1	1.250 X 1.750 X 3/4	90580	NOT IN STOCK
LIVE LOAD SPRING ASSEMBLY	2	LIVE LOAD SET 12S	62-M-177	
GLAND NUT STUDS	2	5/8in.- TPI, SA453 GR660	450625	
FLAT WASHERS	4			62-F-177

PERFORMED BY:________________ Date:__/__/__ FOREMAN: ________________ DATE:__/__/__

Attachment 2

DEMO 1

RITCHIE S.E.S.

SURVEY DATA SHEET

.F.

NAMEPLATE INFORMATION

TAG NUMBER: *DEMO 1*

MANUFACTURER *WESTINGHOUSE*

MODEL NUMBER *4GM88FND*

SIZE *4"* TYPE: *GATE*

FUNCTION: *RCS*

SYSTEM PRESSURE: *2485*

ACTUATOR:

[ ] AOV [ ] HOV [ ] MAN [X] MOV

MANUFACTURER: *LIMITORQUE*

MODEL NUMBER *SB 0*

UNIT: *1* ELEVATION: *19'*

LOCATION: *BELOW STEAM GEN. C*

MECHANICAL INFORMATION

A = STEM DIAMETER ______

B = STUFFING BOX BORE ______

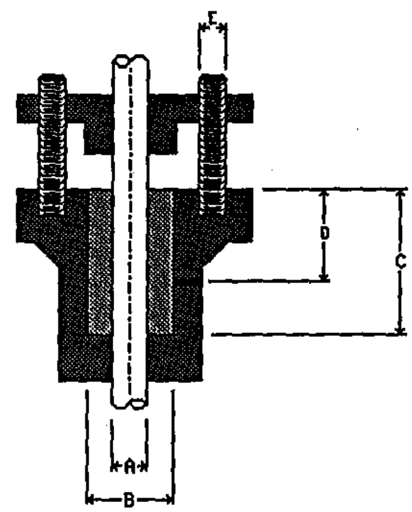
C = STUFFING BOX DEPTH ______

D = LEAK-OFF PORT DEPTH ______

LANTERN RING HEIGHT ______

E = STUD DIAMETER ______

NUMBER OF STUDS ______



MECHNICAL INFORMATION

STEM ORIENTATION [X] VERTICAL [ ] OTHER

LEAK-OFF PORT [ ] N/A [ ] NON-ACTIVE [X] ACTIVE

LANTERN RING [ ] REMOVED [ ] DROPPED

STUFFING BOX BOTTOM [ ] BEVELED [X] FLAT

Comments: *Example of a nuclear valve application with an active leak o ff system and live load hardware.*

Performed by: _____ ___/___/___ Validated by: _____ ___/___/___

Attachment 3

RITCHIE S.E.S.

VALVE PACKING SUMMARY

TAG NO:	VLV MFG:	SIZE:	MODEL:	STEM DIA:	BOX BORE:	BOX DEPTH:	STUD DIA:	TORQUE:	PACK CONF.
DEMO 1	WESTINGHOUSE	4"	4GM88FND	1.250	1.750	4.250	0.625	28	7UCDCLCDDCE
DEMO 2	FISHER	20"	EHD	2.000	2.750	4.375	0.750	99	5UBDDDBE
DEMO 3	COPES VULCAN	6"	D100-160	1.250	1.750	3.500	0.625	43	5UCCDDCE
DEMO 4	VOGT	1"	SW12144	0.500	0.875	1.062	0.375	0	5BDDDB
DEMO 5	YARWAY	2"	5515B	0.937	2.312	3.500	0.875	102	5CCDDC
DEMO 6	PACIFIC	18"	3503WE	2.250	3.250	7.500	1.000	144	5UBDDDBE
DEMO 7	ROCKWELL	14"	14411W	2.750	3.750	3.500	1.125	335	5BDDDBE

Attachment 4

RITCHIE S.E.S.

VALVE PACKING SUMMARY - TWO

TAG NO:	VLV MFG:	SIZE:	MODEL:	UNIT	LOCATION	ELEVATION	STATUS	DATE
DEMO 1	WESTINGHOUSE	4"	4GM88FND	1	BELOW STEAM GEN. C	19'	DEMONSTRATION	03/15/94
DEMO 2	FISHER	20"	EHD	1	NORTH SIDE OF BOILER	8TH FLR	DEMONSTRATION	03/15/94
DEMO 3	COPES VULCAN	6"	D100-160	2	WEST SIDE OF BOILER	6TH FLR	DEMONSTRATION	03/16/94
DEMO 4	VOGT	1"	SW12144	1	4/K MEZZANINE	4 1/2	DEMONSTRATION	03/16/94
DEMO 5	YARWAY	2"	5515B	2	OFF OF BOOSTER PUMP	2ND	DEMONSTRATION	03/16/94
DEMO 6	PACIFIC	18"	3503WE	2	AUXILLIARY BLDG	256'	DEMONSTRATION	03/16/94
DEMO 7	ROCKWELL	14"	14411W	1	ROOF OF BOILER	8TH	DEMONSTRATION	03/16/94

Attachment 5

RITCHIE S.E.S.

MATERIAL SPECIFICATIONS

03/16/94

EQUIPMENT NUMBER: DEMO 4
MANUFACTORER: VOGT
MODEL NUMBER: SW12144
SIZE: 1"

MATERIALS	QUANTITY	DESCRIPTION	VENDOR	STOCK NUMBER
PACKING SET (5 Rings)	1	0.500 X 0.875 X 0.188	2350160	NOT IN STOCK
DIEFORMED GRAPHITE RING	N/A	N / A	N/A	N/A
BRAIDED GRAPHITE RING	N/A	N / A	N/A	N/A
COMPOSITE GRAPHITE RING	N/A	N / A	N/A	N/A
CARBON BUSHING UPPER	N/A	N / A		
LOWER	N/A	N / A		
LIVE LOAD SPRING ASSEMBLY	N/A	LIVE LOAD SET NONE		
GLAND NUT STUDS	2	3/8in.		
FLAT WASHERS	N/A		N/A	N/A

EQUIPMENT NUMBER: DEMO 5
MANUFACTORER: YARWAY
MODEL NUMBER: 5515B
SIZE: 2"

MATERIALS	QUANTITY	DESCRIPTION	VENDOR	STOCK NUMBER
PACKING SET	N/A	N / A	N/A	N/A
DIEFORMED GRAPHITE RING	2	0.938 X 2.313 X 0.688	2410450	NOT IN STOCK
BRAIDED GRAPHITE RING	N/A	N / A		
COMPOSITE GRAPHITE RING	3	0.938 X 2.313 X 0.688	2510450	NOT IN STOCK
CARBON BUSHING UPPER	N/A	N / A		
LOWER	N/A	N / A		
LIVE LOAD SPRING ASSEMBLY	N/A	LIVE LOAD SET NONE		
GLAND NUT STUDS	2	7/8in.		
FLAT WASHERS	N/A		N/A	N/A

EQUIPMENT NUMBER: DEMO 6
MANUFACTORER: PACIFIC
MODEL NUMBER: 3503WE
SIZE: 18"

MATERIALS	QUANTITY	DESCRIPTION	VENDOR	STOCK NUMBER
PACKING SET (5 Rings)	1	2.250 X 3.250 X 0.500	2351120	NOT IN STOCK
DIEFORMED GRAPHITE RING	N/A	N / A	N/A	N/A
BRAIDED GRAPHITE RING	N/A	N / A	N/A	N/A
COMPOSITE GRAPHITE RING	N/A	N / A	N/A	N/A
CARBON BUSHING UPPER	1	2.250 X 3.250 X 3 1/2	91120	NOT IN STOCK
LOWER	1	2.250 X 3.250 X 1 3/8	91120	NOT IN STOCK
LIVE LOAD SPRING ASSEMBLY	2	LIVE LOAD SET 10S	100-M-177	YOUR STOCK
GLAND NUT STUDS	2	1 in.-12TPI, 193 B7	870100	NUMBERS
FLAT WASHERS	4		100-F-177	HERE

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The 1994 Symposium on Valve and Pump Testing, jointly sponsored by the Board of Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the 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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