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

Held at Hyatt Regency Hotel
Washington, DC
July 15–18, 1996

Sponsored by
U.S. Nuclear Regulatory Commission

Board on Nuclear Codes and Standards
of the American Society of Mechanical Engineers

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ABSTRACT

The 1996 Symposium on Valve and Pump Testing, jointly sponsored by the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the U.S. Nuclear Regulatory Commission, provides a forum for 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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ACKNOWLEDGEMENTS

The Steering Committee, the ASME, and the NRC 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 James Perry, James Pelletier, Dr. David Morrison, and Ashok Thadani for their remarks in the opening session. Participation by the foreign presenters and attendees is valued for offering a broader perspective to the issues currently ongoing in the United States. The entertaining banquet speaker, Mr. Mark Shields, provided the audience with a bird's eye view of national politics. Finally, gratitude is expressed to all the attendees and their sponsoring organizations without whom the symposium would be meaningless.

DISCLAIMER AND EDITORIAL COMMENT

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

The papers have been copy edited and recast into a standard format, with certain exceptions. By consensus, English units have been used as an expression of current industry practice.

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The Need to Optimize Inservice Testing and Inspection to Enhance Safety

James A. Perry, Consultant
Vice President, American Society of Mechanical Engineers
Chairman, Board on Nuclear Codes and Standards

INTRODUCTION

Welcome to the Fourth U. S. Nuclear Regulatory Commission and American Society of Mechanical Engineers (USNRC/ASME) Symposium on Valve and Pump Testing in Nuclear Power Plants. This symposium provides a forum to exchange information on technical and regulatory issues associated with the testing of valves and pumps used in nuclear power plants. Progress made since the last symposium will be discussed along with various methods for in service testing of valves and pumps. Active participation by industry representatives, regulators and consultants will entail discussion of a broad array of ideas and points of view regarding how to improve the in service testing of valves and pumps at nuclear power plants. One of the challenges we face is the need to optimize the in service testing and inspection to enhance safety, operability and reliability. I will address this challenge from an ASME Nuclear Codes and Standards point of view.

BACKGROUND

During the early years of commercial nuclear power, ASME produced a code for the construction of nuclear vessels used in the reactor coolant pressure boundary, containment, and auxiliary systems. In response to industry growth, ASME code coverage soon broadened to include rules for construction of other nuclear components, and inservice inspection of nuclear reactor coolant systems. In the years following, the scope of ASME nuclear codes, standards, and guides has been broadened significantly to include air cleaning activities for nuclear power reactors, operation and maintenance of nuclear power plants, quality assurance programs, cranes for nuclear facilities, qualification of mechanical equipment, and concrete reactor vessels and containments.

ASME focuses on globalization of its codes, standards, and guides by encouraging and promoting their use in the international community and by actively seeking participation of international members on technical and supervisory committees and in accreditation activities. Initiatives are underway to separate the technical requirements from administrative and enforcement requirements, to include metric units, to provide for non-U.S. materials, and to provide translations into non-English languages.

The ASME Board on Nuclear Codes and Standards (BNCS) is charged with the management of all ASME activities related to codes, standards, guides and accreditation matters directly applicable to nuclear facilities and technology. The Board assesses the need for codes and standards, establishes the necessary committee structure for their development, and assures that the committees reporting to the board operate under accredited procedures and provide procedural

due process. BNCS reports to the ASME Council on Codes and Standards.

Two specific groups reporting to the Board are responsible for the ASME code requirements to maintain the nuclear power plants while in operation and to return the plant to service, following plant outages, and repair or replacement activities.

The ASME Subcommittee on Inservice Inspection is responsible for the ASME Boiler & Pressure Vessel (B&PV) Code Section XI Rules for Inservice Inspection (ISI) of Nuclear Power Plant Components. This section provides rules for the examination, inservice testing and inspection, and repair and replacement of components and systems of nuclear power plants.

The ASME Operations and Maintenance (O&M) Committee is responsible for the ASME OM Code for Operations and Maintenance of Nuclear Power Plants and for ASME Standards and Guides for Operation and Maintenance of Nuclear Power Plants. The scope of coverage of the ASME OM Code basically applies to inservice testing (IST) of pumps, valves, and dynamic restraints used in nuclear power plants.

SCOPE OF APPLICATION TO ISI/IST

Application of the rules are governed by group classification criteria of the regulatory authority having jurisdiction at the plant site as follows:

(1) The rules of IWB shall apply 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 apply to those systems whose components are classified ASME Class 3 (Quality Group C).

Title 10 CFR 50.55a(g), "Inservice Inspection Requirements," states that components which are part of the reactor coolant pressure boundary and their supports must meet the requirements applicable to components which are classified as ASME Code Class 1. Other safety-related pressure vessels, piping, pumps, and valves must meet the requirements applicable to components which are classified as ASME Code Class 2 or Class 3.

It is important to keep in mind that for Inservice Inspection, the scope of ASME B&PV Code Section XI Rules for Class 1 apply to components which are part of the reactor coolant pressure boundary and their supports. The Rules for Class 2 and 3 apply to other safety-related pressure vessels, piping, pumps, and valves. The examination categories of parts, examination and test requirements, methods, acceptance standards, and extent and frequency of examination are grouped by Class of components. The requirements for Class 1 differ from the requirements for Class 2. The requirements for Class 3 also differ from Class 1 and 2. Within each Class, however, the requirements for a given part or parts examined are the same. They are treated as equal, even though we know for example that their relative importance and performance history in operating plants may be and often are different.

Similarly in the case of the ASME OM Code, the scope of inservice testing relates to those pumps, valves, and dynamic restraints which are required to perform a specific function in shutting down a reactor to a cold shutdown condition, in maintaining the cold shutdown condition, or in mitigating the consequences of an accident. For any given type of component, such as a centrifugal and vertical line shaft pump, the testing requirements, criteria, and test frequency are basically the same. Each type of component is treated as equal, even though we know for example that their relative importance and performance history in operating plants may be and often are quite different.

OPTIMIZING ISI/IST

It is in recognizing that the components being subjected to inservice testing and inservice inspection are indeed different, that steps are being taken to analyze their differences so that they can be treated differently. The challenge then becomes one of taking advantage of the latest technology by using additional tools such as probabilistic risk assessment, condition monitoring, performance-based and other techniques in conjunction with the deterministic techniques to help us optimize the inservice testing and inspection methods, and examination and testing frequencies. The use of the latest technology will allow us to look at the components in a different light. We need to look at the components in a holistic manner. We need to take into account not just the individual examinations and tests but also to include consideration of routine operation, preventive and corrective maintenance, application of the maintenance rule, performance histories, failure rates and other aspects collectively in order to enhance safety, operability and reliability.

RESPONDING TO THE CHANGING NUCLEAR ENVIRONMENT

In response to the changing nuclear environment and looking toward the twenty-first century, I envision the ASME Nuclear Codes and Standards major emphasis will address five goals relating to:

1. Best use of volunteers, our valuable limited resource needs.

2. Streamlining codes and standards to meet the changing environment and user.

3. Encouraging frank and open discussion of issues.

4. Increasing the stature of Nuclear Codes and Standards.

5. Maintaining effective communication with regulatory bodies and other organizations.

First, making the best use of limited resources involves recruiting additional volunteers from among the nuclear community (both national and international), such as individuals attending this symposium who are not currently involved in ASME codes and standards working groups, subgroups, subcommittees or committees. Incidentally, ASME forms are available for signing up during this symposium. Making the best use of limited resources also involves focusing on the significant issues, addressing them in a timely manner and assuring that not only safety but also the cost impacts are appropriately considered. Emphasis will be placed on the use of value impact forms for major changes and new projects.

Second, streamlining codes and standards to meet the changing environment and user needs, involves recognizing that the nuclear industry is struggling to survive and to remain competitive in the United States and overseas. Shifting from design and construction of the past, to operations and maintenance of the present, and applying lessons learned to position ourselves to meet the needs of the advanced reactors projects of the future, is one of the challenges before us.

Third, encouraging frank and open discussion on issues is essential to achieve timely consensus on codes and standards, in compliance with approved procedures. By extending this thought, this is also true at this symposium. By your active participation at this symposium, a broad array of ideas and points of view will be discussed regarding how to improve the inservice testing of valves and pumps at nuclear power plants. As a result, much of the technical information presented and discussed at this symposium can be of great assistance to ASME codes and standards committees, especially the O&M Committee in keeping documents current with the latest technology.

Fourth, increasing the stature of nuclear codes and standards nationally and internationally is essential to the future success of the society. To accomplish this, we must maintain the lead as an international codes and standards organization. For example, we must respond more quickly to the urgent needs of the users and must shift emphasis, to take into account probabilistic risk assessment techniques and performance-based measures to reduce costs and optimize inservice inspection and testing.

The industry and the NRC have made considerable progress related to the use of probabilistic safety assessments (PSA). The NRC issued a policy statement in 1995 supporting the use of PSA in the regulatory environment. The ASME Center for Research and Technology Development has carried out risk-based inspection research and risk-based testing research projects that recommend risk-based processes and methods. This research is being sponsored by the NRC, the Department of Energy, the Edison Electric Institute, the ASME Council on Codes and Standards, and others. These research projects are also supported by additional NRC and industry studies involving pilot projects. In addition, the Nuclear Energy Institute (NEI) has issued a PSA Applications Guide and others such as documents pertaining to maintenance, containment testing, and diesel generator reliability. NEI task forces are currently working on risk and performance based applications involving ISI/IST and quality assurance.

Much has been done, but much more is required before we can reflect these aspects in the ASME codes and standards. We need output from the research projects and feedback from the pilot projects to be fed into the codes and standards consensus process to come up with optional alternatives to the code in the form of code cases. As more experience is added, it is anticipated that the code cases will be added to the rules.

Fifth, emphasis will be placed on maintaining effective communication with regulatory bodies and other organizations to assure common understanding and effective implementation of codes and standards. It is absolutely essential that we maintain clear and frequent communication among and between the various organizations and groups involved to assure a unified approach, consistent methodology, and desirable outcome that is mutually beneficial to all parties.

CONCLUSION

Good progress has been made, but we still have a long way to go. Let us share our ideas, experiences and points of view regarding how we can continue to improve the inservice testing of valves and pumps at nuclear power plants. Let us meet the challenge to optimize inservice testing and inspection to enhance safety, operability and reliability. I encourage you to join one of the ASME codes and standards committees if you are not already a volunteer, so your expertise can be shared with the other outstanding contributors from across the nuclear industry from utilities, suppliers, engineering and testing firms and regulators to consultants.

RECENT NRC RESEARCH ACTIVITIES ADDRESSING VALVE AND PUMP ISSUES

Dr. David L. Morrison, Director
Office of Nuclear Regulatory Research

ABSTRACT

The mission of the U.S. Nuclear Regulatory Commission (NRC) is to ensure the safe design, construction, and operation of commercial nuclear power plants and other facilities in the U.S.A. One of the main roles that the Office of Nuclear Regulatory Research (RES) plays in achieving the NRC mission is to plan, recommend, and implement research programs that address safety and technical issues deemed important by the NRC. The results of the research activities provide the bases for developing NRC positions or decisions on these issues. Also, RES performs confirmatory research for developing the basis to evaluate industry responses and positions on various regulatory requirements.

This presentation summarizes some recent RES supported research activities that have addressed safety and technical issues related to valves and pumps. These activities include the efforts on determining valve and motor-operator responses under dynamic loads and pressure locking events, evaluation of monitoring equipment, and methods for detecting and trending aging of check valves and pumps. The role that RES is expected to play in future years to fulfill the NRC mission is also discussed.

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

New Directions and Challenges for the O&M Committee

James P. Pelletier, Chairman
Main Committee of Committee on Operation and Maintenance of Nuclear Power Plants

ABSTRACT

As we move into the second half of the decade of the nineties, the imperative of finding new ways to improve efficiency while maintaining safety is taking on a new urgency. The looming deregulation of the electric industry and the expected competition in the power production business fuels this urgency. The recently completed ASME research in the area of Risk-Based Inservice Testing offers the O&M Committee an opportunity to meet this challenge. This opportunity, however, offers its own challenges. New ways of thinking about inservice testing and new technical skills will be needed to successfully incorporate this technology into Code documents. We cannot rely solely on incorporation of risk-based methods into our Code to meet this challenge. A thorough self assessment of what we do, how we do it, and the value we add will help us assure that the directions we take do, indeed, meet the challenge ahead.

Regulatory Trends Involving Pumps and Valves

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Purpose

The purpose of this presentation is to provide an overview of the various ongoing regulatory issues that have the potential to impact testing of pumps and valves.

Introduction

The long-standing inservice testing program is well established, but there are a number of activities that are creating dynamics in the program that could change the way it is developed and implemented by licensees:

(1) the move toward risk-informed and performance-based regulation,

(2) proposed changes to the governing regulations, and

(3) the Maintenance Rule.

Underlying all of the activities is the imminent deregulation of the electric utility industry of which we must be aware, but for which we cannot sacrifice safety for costs. I'd like to discuss some of these factors in light of the changing environment.

Move Toward Risk-Informed and Performance-Based Regulation

The NRC issued a final policy statement on the use of probabilistic risk assessment (PRA) methods in nuclear regulatory activities on August 16, 1995 (*see* 60 *Federal Register* 158, p. 42622 - 42629). Along with the policy statement on risk-informed regulation, NUREG/BR-0058, "Regulatory Analysis Guidelines of the U.S. Nuclear Regulatory Commission," Revision 2, was issued in November 1995, providing guidance on the implementation of the safety goals given in the NRC's policy statement issued August 21, 1986 (*see* 51 *Federal Register* 162, p. 30028 - 30033). These actions layout a framework for the implementation of risk-informed regulation as follows:

- The use of PRA technology should be increased in all regulatory matters to the extent supported by the state-of-the art in PRA methods and data and in a manner that complements the NRC's deterministic approach and supports the NRC's traditional defense-in-depth philosophy.

- PRA and associated analyses should be used in regulatory matters, where

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practical within the bounds of the state-of-the art, to reduce unnecessary conservatism associated with current regulatory requirements, regulatory guides, license commitments, and staff practices. Where appropriate, PRA should be used to support the proposal for additional regulatory requirements in accordance with 10 CFR 50.109, "Backfitting." The existing rules and regulations shall be complied with unless these rules and regulations are revised.

- PRA evaluations in support of regulatory decisions should be as realistic as practicable and appropriate supporting data should be publicly available for review.

- The Commission's safety goals for nuclear power plants and subsidiary numerical objectives are to be used with appropriate consideration of uncertainties in making regulatory judgments on the need for proposing and backfitting new generic requirements on nuclear power plant licensees.

Though performance-based regulation is not specifically addressed in the NRC's policy statement on the use of PRA technology, the past performance of components is actually an element that factors into risk methods. If a plant has one or more components that have a higher failure rate than that assumed in the PRA, it may impact the overall results in a way that would change the component from a less-risk significant component to one that is more-risk significant. The performance could also impact decisions made by the "expert panels" that review the implementation of risk-informed testing and maintenance. Regulatory Guide 1.160, "Monitoring the Effectiveness of Maintenance at Nuclear Power Plants," and the Nuclear Energy Institute (NEI) guidance for implementing the Maintenance Rule (10 CFR 50.65) use elements of both risk-informed and performance-based decisions.

The insights derived from PRA can be used in combination with deterministic system and engineering analyses to focus licensee and regulatory attention on issues commensurate with their importance to safety. We are currently working on two pilot plants for implementation of a risk-informed inservice testing program. You will hear more details on the specific programs and issues involved in the Risk-Based Testing of Pumps and Valves Session and I encourage you to become informed on the process once an acceptable plan has been completed through the pilot projects.

Assuming successful completion of the staff's interaction with the pilot plant licensees for Comanche Peak and Palo Verde and resolution of risk-informed issues with the Commission, we expect to authorize implementation of the risk-informed inservice testing programs. The authorized programs will comport with a proposed regulatory guide and Standard Review Plan that the staff is preparing concurrently with the pilot reviews. Authorization of the pilot programs and issuance of the proposed regulatory guide and Standard Review Plan for public comment are expected by late 1996. We expect that during 1997, the proposed regulatory guide and Standard Review Plan will undergo revision in response to public comments. We hope that the changes are minor. Also in 1997, we will assess implementation of the risk-informed inservice testing pilot programs.

Depending upon the changes to the proposed regulatory guide and Standard Review Plan and our experience over the period between the proposed and final publication of these documents, the pilot plant licensees may need to bring their programs into conformance with the final version of the regulatory guide and Standard Review Plan. In addition, we understand that ASME Operations and Maintenance Committee is currently developing Code cases dealing with various aspects of risk-informed inservice testing strategies. We have been following the Committee's activity with interest and believe that improved test strategies can benefit risk-informed inservice testing programs. Once ASME approves these Code cases and we find them acceptable, we may propose to revise the regulatory guide and Standard Review Plan to include the Code cases as references or as acceptable alternatives to the current Code requirements incorporated into the regulations.

The NRC staff are addressing some difficult policy issues as part of the risk-informed approach to IST. For example, we must consider whether or not a potential increase in overall risk is acceptable, including considering the relationship of a risk-informed initiative to the Commission's safety goals. We must consider the approach to implementing risk-informed IST on a trial basis at the pilot facilities and then assuring the appropriate feedback of pilot plant experience into the regulatory process.

Proposed Changes to the Governing Regulations

Currently, if a licensee wants to use a later code edition, addenda, or code case that has not yet been incorporated into the regulations, a specific plant request must be submitted to the NRC. The staff must evaluate the request, and approve or deny the request, prior to implementation by the licensees. This process is not efficient and does not encourage active participation on the committees that revise the codes. We are working toward establishing an improved method.

We are preparing a proposed change to 10 CFR 50.55a to include:

(a) retention in the rule of the concept of the 120-month update provision with imposition of certain incremental-to-safety backfits from later editions of the Code;

(b) immediate backfit of the Section XI Appendix VIII ultrasonic testing procedure and Appendix VII personnel qualification requirements through the 1995 Edition;

(c) endorsement of use of 1990 O&M Code for implementation in accordance with the 120-month update provision (considered equivalent to 1989 Section XI currently referenced in rule);

(d) endorsement of revisions in 1989 Addenda-1995 Section XI and O&M Code, and;

(e) identification of portions of the Section XI and OM Code that may be appropriate for voluntary implementation and encouragement to ASME to do likewise as code changes are made.

In the future, any safety-significant items, such as (b) above, will be subject to formal backfit procedures for immediate imposition, while incremental-to-safety backfits will be imposed only for the 120-month program update. We anticipate that the issuance for public comment of the revised rule may be late this year.

While the proposed rule change is moving forward, we will also be assessing changes that would be part of a subsequent revision to the rule. We need to incorporate performance and risk assessment techniques and streamline the process for approval of alternatives to the prescriptive requirements that now exist. By then, ASME will be issuing codes and standards that are based on risk assessment and component performance. The subsequent rulemaking would represent a substantial change, but the effort will be worthwhile. I ask that you bear with us in our attempts to improve the process.

The Maintenance Rule

The maintenance rule (10 CFR 50.65) became effective July 10, 1996. The rule is performance based and requires that safety (or risk) be taken into consideration. While it essentially umbrellas the entire scope of the inservice testing program, tests conducted pursuant to Section 50.55a may be credited for monitoring the applicable components. The rule is not intended to negate the need for the inservice testing program, though there are aspects of both programs that can be coordinated for best results.

The NRC issued implementation guidance for the rule in Regulatory Guide 1.160, relying largely on the efforts of the Nuclear Energy Institute in developing and revising guideline NUMARC 93-01. Engineers responsible for implementing the inservice testing program at nuclear plants should be familiar with the requirements of the maintenance rule. It is a less prescriptive rule than Section 50.55a and the ASME Code; it may set the stage for many elements of the risk-informed inservice testing programs that are currently being developed; and it may offer a model for less prescriptive rules in Section 50.55a.

Other Topics of Interest

Periodic Testing Program for Motor-Operated Valves

The NRC has issued a proposed generic letter to address periodic testing of motor-operated valves requesting licensees to inform the NRC of their plans to ensure that the proper valve settings are maintained throughout the life of the plant. The proposed generic letter indicates that, with certain limitations, ASME OM Code Case OMN-1 is an acceptable means of implementing such a program. The code case moves away from the requirements for quarterly exercising and stroke timing to a more informative, but less frequent, test, while at the same time recommending that the user consider the safety significance and performance history of the valves in establishing the test frequency. We believe that licensees can benefit by focusing efforts on the more safety significant valves, but continuing to monitor the remaining valves at an appropriate periodicity.

Pressure Locking and Thermal Binding of MOVs

On August 17, 1995, the NRC issued GL 95-07, "Pressure Locking and Thermal Binding of Safety-Related Power-Operated Gate Valves," requesting that licensees take actions to ensure that safety-related power-operated gate valves susceptible to pressure locking or thermal binding are capable of performing their safety functions within the current licensing bases of the facility. GL 95-07 requested that, within 180 days, licensees evaluate the operational configurations of safety-related power-operated gate valves to identify valves that are susceptible to pressure locking or thermal binding and perform further analyses as appropriate, and take

needed corrective actions (or justify longer schedules), to ensure that the susceptible valves identified are capable of performing their intended safety function(s) under all modes of plant operation, including test configuration.

The NRC has received the licensee submittals in response to GL 95-07. These submittals have provided a summary description of licensee susceptibility evaluations, further analyses, and the corrective actions, or other dispositioning for the valves identified as susceptible. The Mechanical Engineering Branch in the Office of Nuclear Reactor Regulation is currently incorporating the results of gate valve testing performed by Idaho National Engineering Laboratories (sponsored by the NRC Office of Nuclear Regulatory Research) into standard technical review guidelines. These review guidelines are being used by the staff in reviewing licensee submittals.

Use of New Technologies

One of the major purposes of the ASME Committees is to keep the codes and standards as up-to-date as possible within the constraints of the consensus process. This means that the committees should be addressing new technologies for testing and monitoring equipment such as diagnostic equipment for air-operated valves, spectral analysis for pump vibration monitoring, thermography, and infrared detection. There may be other developing technologies that will offer advancements in achieving the goals of optimizing the availability and reliability of equipment. I encourage the development of testing strategies and guidance for the use of new technologies.

Conclusions

The industry is interested in controlling costs and becoming more competitive as it moves into a deregulated environment. As regulators in this changing environment, the NRC must ensure that safety is maintained while continuing to assess the possibilities for reducing the regulatory burden on licensees where appropriate. This is not a new concept but is becoming of particular interest now because of the pending deregulation. The codes and standards groups for pump and valve testing should consider these issues during the development of new guidance or in revising current requirements. NRC participation on the industry working groups and committees should allow for adequate regulatory interaction such that we can continue to rely on the codes and standards in our regulations. However, because of the current dynamics, we may actually be leading the way for the risk-informed testing programs, being the force that merges the various competing interests into one concise methodology. We are all challenged to continue the high safety standards while being responsive to new initiatives for mechanical equipment operation and testing.

Session 1A

International Experience

Session Chair
Dr. Nabil Schauki
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Prevention of Crack Initiation in Valve Bodies Under Thermal Shock

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INTRODUCTION

On site and testing experience has shown that cracking in valves affects mainly the stellite hardfacing on seats and discs but may also be a concern for valve bodies.

STELLITE HARDFACING CRACKING

Metallurgical investigations conducted by EDF laboratories on may damaged valves have shown that most of the damage had either a chemical, manufacturing, or operating origin with a strong correlation between the origins and the type of damage. The chemical defects were either excess ferritic dilution of stellite or excess carburizing. Excess carburizing leads to a too brittle hardfacing which cracks under excessive stresses induced on the seating surfaces, via the stem, by too high operating thrusts. The same conditions can also induce cracks of the seats in the presence, in the hardfacing, of hidden defects generated during the welding process.

Reduction of the number of defects results first from controls during manufacturing, mainly in the thickness of stellite. On the other hand, maintenance must be fitted to the type of defect. In-situ lapping may lead to release of cobalt, resulting in contamination of the circuit. Furthermore, it is ineffectual in the case of a crack through the seating surface, as is often found on globe valves. The use of new technologies of valves with removable seats and cobalt-free alloys solves permanently this kind of problem.

CRACKING OF VALVE BODIES

In France, one valve of each series of all Class 1 valves is submitted to a standard qualification program. The test includes 1000 opening/closing cycles and 20 thermal shocks to check the quality of hardfacings and the bonnet/body tightness. In 1980, the French Pressure Vessel Authority (BCCN) asked to extend, for some highly loaded valves, the thermal shock program so as to obtain the same usage factor as the one computed with plant design transients.

Unexpectedly, cracks were found in internal fillets of some forged-body, austenitic valves which were designed following code body-shape rules, for fatigue usage factors lower than 1, with NB-3500 simplified calculations. With finite element analysis (NB-3200 methodology), it was possible to explain the presence of the cracks but with lower margins than previously thought. After years of arguing with BCCN about the risk induced by such cracks of small depth relative to the valve body and the validity of the fatigue calculation methodologies, increased minimum radii of internal fillets and a modified simplified calculation method were imposed and applied for all spare bodies and valves installed on new plants since CHOOZ B1. Extensive testing until apparition of cracks on

axisymmetric mockups of austenitic valve bodies and finite elements calculations were done by EDF to qualify the modified fatigue calculation methodology.

This methodology cannot be applied to shapes that vary too much from axisymmetrical shapes. Therefore, finite element analysis remains necessary in some cases. Moreover, a program comprising metallurgical investigations of small valves drawn out from operating plants has been defined to complete the qualification file of this new fatigue analysis methodology.

Latest Design of Gate Valves

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J. Stolte
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Babcock Sempell Aktiengesellschaft (Deutsche Babcock)

ABSTRACT

Babcock Sempell, one of the most important valve manufacturers in Europe, has delivered valves for the nuclear power industry since the beginning of the peaceful application of nuclear power in the 1960s. The latest innovation by Babcock Sempell is a gate valve that meets all recent technical requirements of the nuclear power technology. At the moment in the United States, Germany, Sweden, and many other countries, motor-operated gate and globe valves are judged very critically. Besides the absolute control of the so-called "trip failure," the integrity of all valve parts submitted to operational forces must be maintained. In case of failure of the limit and torque switches, all valve designs have been tested with respect to the quality of guidance of the gate. The guidances (i.e., guides) shall avoid a tilting of the gate during the closing procedure.

The gate valve newly designed by Babcock Sempell fulfills all these characteristic criteria. In addition, the valve has cobalt-free seat hardfacing, the suitability of which has been proven by friction tests as well as full-scale blowdown tests at the GAP of Siemens in Karlstein, West Germany. Babcock Sempell was to deliver more than 30 gate valves of this type for 5 Swedish nuclear power stations by autumn 1995. In the presentation, the author will report on the testing performed, qualifications, and sizing criteria which led to the new technical design.

Electric Actuator for the Sempell Gate Valve

E. C. Herbstritt
Matthias Dinse
Werner Riester GmbH & Co. KG (AUMA)

The automation of valves has a primary importance in the scope of central control and regulation of power generation processes in power plants and especially in nuclear power plants. AUMA WERNER RIESTER GmbH & Co. KG is considered a leading manufacturer of electric actuators for the automation of valves. More than 30 years experience in designing, developing, and manufacturing provide a sound basis for offering reliable products, especially for nuclear applications.

The quality assurance system of AUMA was developed according to 10 CFR 50, Appendix B and has been consistently accomplished. The program was certified by the TUV Germany (Technical Authorized Inspection Agency), according to ISO-9001, in 1994. AUMA offers two actuator type ranges for application in nuclear power plants. The range SAI is qualified according to IEEE 382-1978 and is designed for inside containment. The range SAN is qualified according to IEEE 382-1985 and KTA 3504-1988 for use in non-radioactive applications in the nuclear power plants.

Actuators SAN and SAI are in a modular design. Both type of ranges are designed by a uniform principle of design for all types. The various versions cover a torque range of 20 to 1000 Newton-meters (Nm) for SAI and 10 to 2000 Nm for SAN. Therefore, they are capable to operate almost every valve in the power industry. Specially designed motors with high starting torque guarantee reliable function. This is required for positive unseating of fully closed valves.

The gearing, contained in a robust cast iron housing, consists of a worm gear with a worm shaft and output hollow shaft. The gear case is filled with lubricant. This results in maintenance-free service for a long duration. An integrated control unit consists of the required components for control and indication of the status of the actuator. Depending upon the type of valve, the actuator must be switched off by limit or torque. For this purpose, two independent control devices are provided. AUMA actuators are provided with high value corrosion protection. This also withstands the extreme conditions as might be found in a cooling tower. Therefore, a long period of operation is guaranteed.

More than 5000 AUMA actuators for nuclear applications operate worldwide. Their reliability is proven in many nuclear power plants since 1978. In the presentation, the author will provide additional information about experiences in the automation of valves for nuclear applications.

Valve Performance Concept Move from Preventive to Condition-Oriented Maintenance

G. Zanner
Siemens Power Generation

P. Kradepohl
KWU

VALVE PERFORMANCE CONCEPT

As a turnkey supplier of nuclear and fossil power plants, Siemens must pay attention in concentrating, maintaining, and developing the expertise in many areas such as system design, components, materials, quality assurance, and qualification testing within centralized organizations. In the company segment VALVES, Siemens/KWU is staffed with experienced professionals who have serviced the power plant industry for about 25 years.

The valve engineers deal with all kinds of valve and actuator-related activities like design ratings, development, qualifications, and ongoing improvements. In this regard, the engineers are involved in nearly all actual problems and suggested solutions through continuing dialogues with utilities, authorities, and vendors of valves and actuators.

Siemens valve departments are familiar with all valve types (e.g., globe, gate, butterfly, check, control, safety, and relief valves) and vendor-specific problems. The valve engineering has reviewed or participated in various testing programs:

- INEL/NRC — Valve Testing at Wyle and Siemens
- EPRI — Separate effect testing to establish a motor-operated valve performance prediction methodology.
- Utility — In-situ testing and valve performance programs
- Vendor — Valve design and qualification
- Siemens — Valve engineering and testing

Siemens has used the knowledge gained in valve engineering and testing to develop an *EXPERT SYSTEM* to justify the change from preventive to condition-oriented maintenance. The goal is an integral valve "service concept" which includes a PC-based "trending method" and quantifies measured and calculated valves. The program tells the user if the valve performs still within accepted margins and, if not, which action (i.e., maintenance) has to be taken. The steps of the Siemens Valve Service Concept are:

- Calculation
- Baseline measurement
- Diagnostic/periodic tests
- Trending

The results indicate the readiness for function and parameter for condition oriented maintenance. A graphical example of the concept for a globe valve is shown in Figure 1.

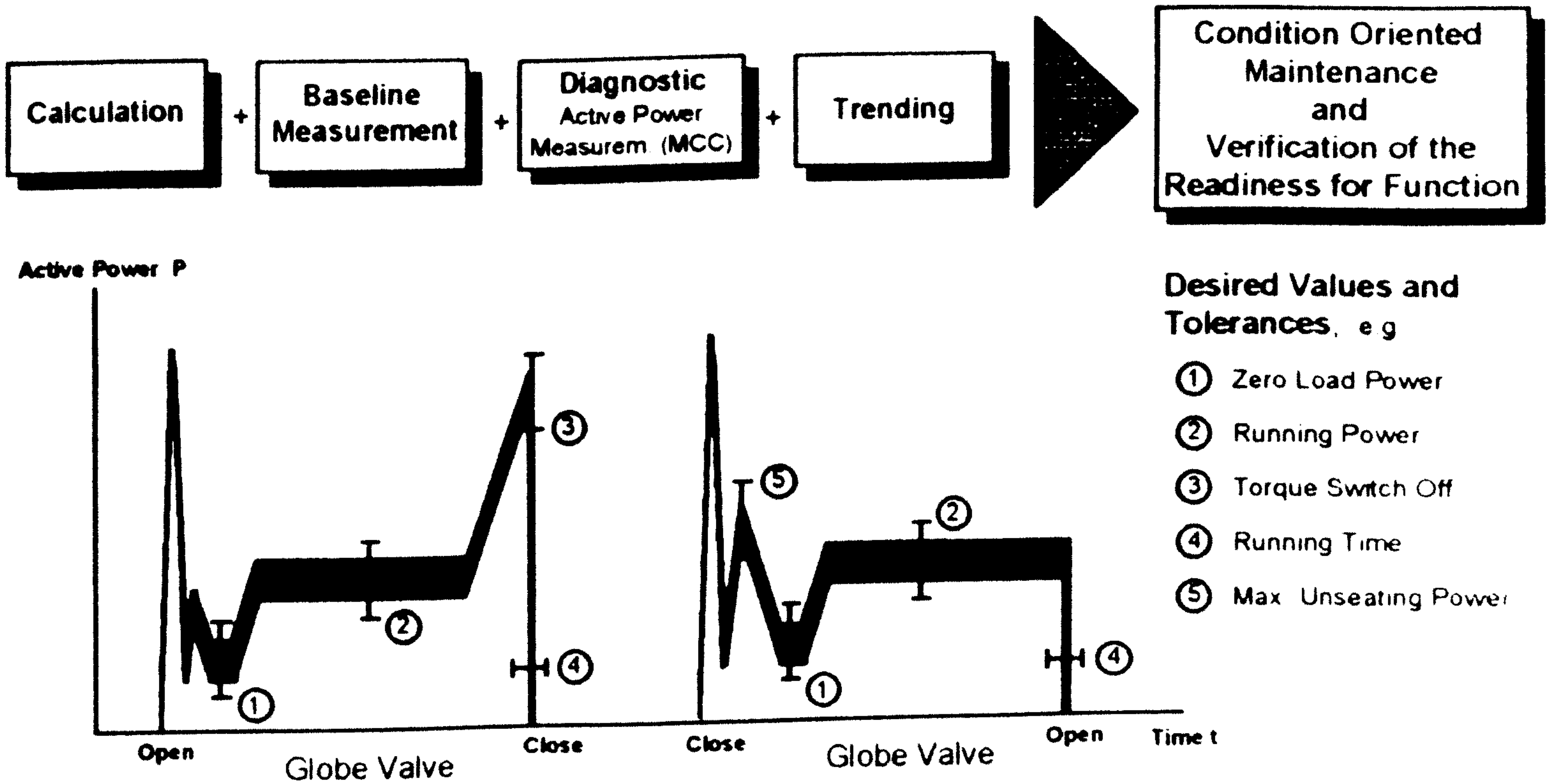
Calculation
+
Baseline Measurement
+
Diagnostic Active Power Measurem. (MCC)
+
Trending
Condition Oriented Maintenance and Verification of the Readiness for Function
Active Power P
Open
Globe Valve
Close
Close
Globe Valve
Open
Time t
Desired Values and Tolerances, e.g.
① Zero Load Power
② Running Power
③ Torque Switch Off
④ Running Time
⑤ Max. Unseating Power

Figure 1

MOTOR OPERATED VALVES PROBLEMS TESTS AND SIMULATIONS

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Direction des Etudes et Recherches - Departement M.T.C.

ABSTRACT

An analysis of the two refusals of operation of the EAS recirculation shutoff valves enabled two distinct problems to be identified on the motorized valves:

- the calculation methods for the operating torques of valves in use in the power plants are not conservative enough, which results in the misadjustement of the torque limiters installed on their motorizations,

- the second problem concerns the pressure locking phenomenon : a number of valves may entrap a pressure exceeding the in-line pressure between the disks, which may cause a jamming of the valve.

EDF has made the following approach to settle the first problem:

- determination of the friction coefficients and the efficiency of the valve and its actuator through general and specific tests and models,

- definition of a new calculation method.

In order to solve the second problem, EDF has made the following operations:

- identification of the valves whose technology enables the pressure to be entrapped : the tests and numerical simulations carried out in the Research and Development Division confirm the possibility of a « boiler » effect,

- determination of the necessary modifications : development and testing of anti-boiler effect systems.

1. Introduction

An analysis of the two refusals of operation of the EAS recirculation shutoff valves (valves EAS 13 and 14 VB in the BUGEY power plant) enabled several distinct problems to be identified on the motorized valves:

- the calculation methods for the operating torques of valves in use in the power plants are not conservative enough, which results in the misadjustment of the torque limiters of motorizations,

- a number of valves may entrap a pressure exceeding the in-line pressure between the disks, which may cause a jamming of the valve.

EDF has made the following approach to settle the first problem :

- determination of the friction coefficients and the efficiency of the valve and its actuator through general and specific tests and models,

- definition of a new calculation method.

In order to solve the second problem, EDF has made the following operations :

- identification of the valves whose technology enables the pressure to be entrapped : the tests carried out in the Research and Development Division using two different technologies try to confirm the possibility of a « boiler » effect,

- determination of the valves likely to be jammed or to lose their tightness and whose failure endanger the safety : numerical simulations give the level of pressure which could be reached in these valves,

- determination of the necessary modifications : development and testing of anti-boiler effect systems.

This paper describes the tests performed and the models developed in order to solve both problems.

2. Definition of a new calculation method

2.1. Objective of the studies

An analysis of the file of incidents which occurred in 1992 has shown that, in the French nuclear power plants, 152 incidents due to a refusal to operate were not due to the failure of a component. Some incidents could consequently be explained by the inadequacy of the actuator to the valve. This is why it was necessary to better understand the loads exerted on a motor-operated valve.

The studies currently in progress which aim to settle these problems are specific tests made on particular components or on representative sets of field-mounted equipment. In addition, analogic or numerical models have been developed for a better analysis of the phenomenons and influence parameters.

These studies have been carried out jointly by EDF, FRAMATOME and the CEA (French Atomic Energy Agency).

2.2. The tests

These tests were as follows:

- tests of specific components :

    - determination of friction coefficient for differents hardfacing alloys on a test bench (see photo 1) reproducing the geometric and ambient conditions of the valves (tests carried out jointly by CETIM Laboratories, valves manufacturers and EDF).

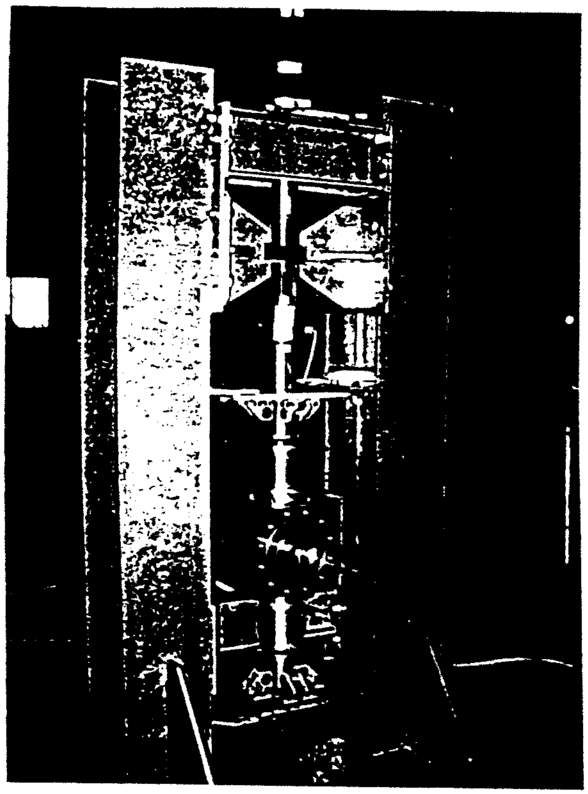


Photo 1 : Friction test bench

- determination of stuffing box friction forces on qualification test benches,

- evaluation of efficiencies in the operating stem nut using different greases (tests currently in progress with Atomic Energy of Canada Limited).

- on actuators:

  ⇐ characterisation bench tests at room temperature (see photo 2) :

  - forces developed under a degraded voltage,
  - inertia effects (on closing),
  - capacities on opening.

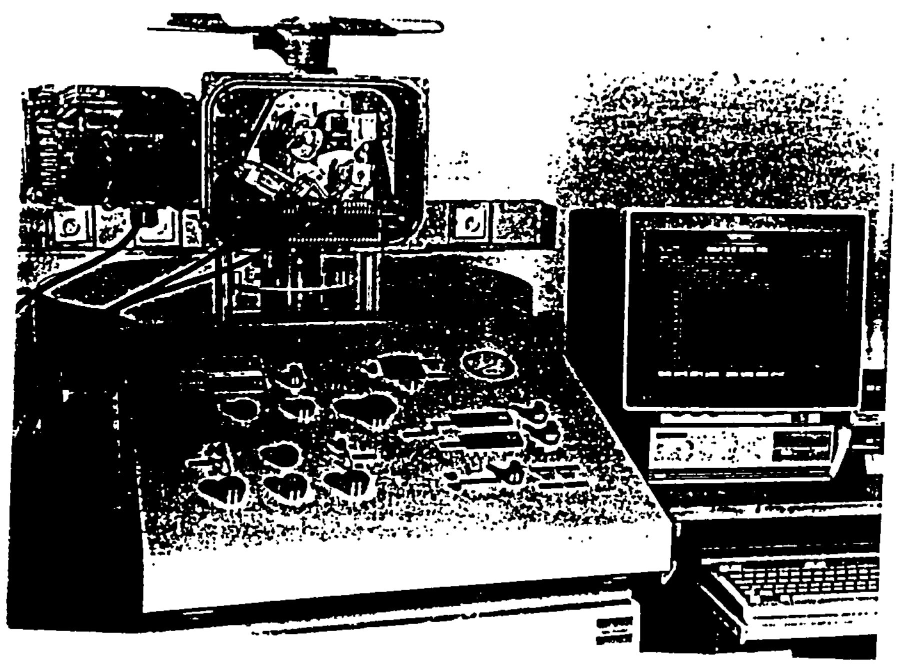


Photo 2 : Actuator test bench

  ⇐ bench tests at a high temperature (155°C): 42% drop of the torque at Un (of which 26% are due to the motor alone)

- testing of valve/actuator assembly:

  - no-flow tests with, and without, remote control,

  - loop cycling tests : a fast variation in the stellite / stellite friction coefficient was noticed during the first 300 cycles. The coefficient then became stable between 0.4 and 0.45 until after 2,000 cycles.

  - loop discharge tests under accidental-flow operating conditions:

    - SEREG free-expansion, parallel-seat gate valve, ND 250 (Turbine Bypass System) : it fully closes only after the adjusting torque has been raised (see photo 3),

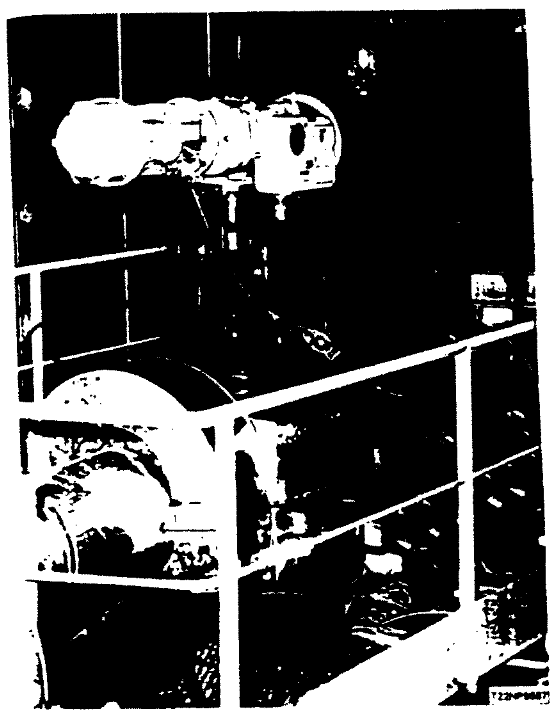


Photo 3 : SEREG gate valve ND 250 on « CUMULUS » loop

- VELAN RATEAU flexible wedge gate valve, ND 80 (Chemical and Volume Control System): no particular problem.

⇆ field tests:

- several tests have been carried out: a full-closing refusal was noticed under flow on an ALSTHOM VELAN flexible wedge gate valve, ND 80 (Safety Injection System),

- VELAN RATEAU flexible wedge gate valve, ND 250 (Residual Heat Removal System - flow rate: 1,000 m3/h). The closing force was smaller than the sizing value, and the gate exerted a « suction » effect at the end of closing (reduction on pressure beneath the gate).

- Tests in progress

Several tests are currently in progress to complete or validate a number of procedures:

⇆ under-flow loop tests on different types of motor-operated valves for validation of the models developed and definition of a simple method for checking field sizing,

⇆ « intelligent » valve: this is a valve fitted with a built-in force transducer.

2.3. Modelling

- Numerical model for simulation of a motor-operated valve:

The data processing tool used is the SIMULINK software (developed by The Math Works Inc.) which allows one to make a dynamic simulation of the systems. The valve is modelled by describing the motor-operated valve through the use of differential equations (in a simple graphic form). The equations are integrated by a variable-pitch gear type numerical integration algorithm. The model allows one to take account of inertia, flexibility and friction characteristics, plays, and mechanical reversibilities, and to simulate inertias, shocks, and so on.

The model has been developed for a valve technology (validation currently in progress), and its development is on the way for the other technologies.

- Analogic model of a motor-operated valve :

An analogic model has been developed by FRAMATOME concurrently with the numerical model described above so as to calculate the inertias involved on closing. The model is based upon the putting up of an equation of the full kinematic chain of the valve and its actuator. It takes account of the component inertia and flexibility, and has been validated by testing. It allows one to determine the inertia provided while operating the valve.

- Modelling of the heating effect upon the motor torque :

The model uses the equivalent diagram of an induction motor and allows one to evaluate the losses in motor torque during heating. A comparison with the tests shows that this is a realistic model. The reduction in torque calculated between 20 and 155°C is about 20% for a motor of a moderate size (the nominal power is around 1.5 kW). The influence of the temperature seems smaller than when the nominal power of the motor is higher.

2.4. Sizing method and consequences for the French plants

The sizing method is based upon the following principle:

« The closing torque under a maximum differential pressure must be lower than the torque delivered by the actuator supplied under a Un-15% voltage »

The tests carried out and the models developed have led to the following conclusions:

- stellite / stellite friction coefficients : new value : 0.40
- stem / stem nut friction coefficients : new value : 0.15
- remote control efficiency : new value : 0.8
- acknowledgement of the 15-to-30% margins taking account of the uncertainties about :
  - the setting of the torque limiter,
  - friction coefficients,
  - the available torque on the actuator.

Consequences for the plants :

The example for the series of P'4 type 1300-MW units gives the following results:

- Number of valves considered	:	105
- Unchanged	:	46 valves
- Modification of actuator setting	:	50 valves
- Changing the actuator or full-flow tests	:	9 valves

These modifications are currently on the way on the series in service, and they have been completed on the units under construction (full-flow check of all safety-related valves).

3. The boiler effect

3.1. Description of the phenomenon

The boiler effect phenomenon takes place when double-disc parallel seat gate valves and flexible wedge gate valves are used. The double sealing present, in turn, leads to the presence of a closed space inside the valve which imprisons some liquid; as a result of the valve being heated from the outside, the temperature of the imprisoned liquid rises; dilation of the liquid volume, limited by the valve body and bonnet, leads to a pressure rise in the liquid.

The resulting overpressure leads to blockage of the discs onto their seats, which can prevent the valve from opening or cause external leaks (in particular at the body/bonnet joint).
The tests carried out enable the boiler effect phenomenon to be better understood. The typifications obtained are as follows:

- high degree of sensitivity to ambient temperature,

- internal pressure rise of 2 bar/°C liquid temperature (instance of a double-disc parallel seat gate valve ND 400 NP 50 bar),

- no boiler effect if the initial internal pressure is low: no triggering of the phenomenon,

- no boiler effect if the valve is not tight: leaks upstream and downstream render pressure rise impossible,

- the application of heat leading to the phenomenon may originate in two causes:
  - liquid temperature rise in the pipelines upstream and downstream,
  - rise in ambient temperature (e.g. in the instance of a LOCA type accident).

3.2. Case of a LOCA type accident

The LOCA type accident is an accident inside the reactor building perimeter which follows a pipeline breakage, leading to vaporization of the primary liquid. The evolution of temperature and pressure in the reactor building is quoted in the RCCE (Régles de Conception et de Construction des matériels Electriques des centrales PWR) [Design and Manufacture Rules for Electrical equipment in PWR power plants]:

- sudden rise in ambient temperature from 50 to 156 °C and of ambient pressure from 0 to 5.5 bar,

- maintenance of these conditions for 20 minutes,

- slow drop in temperature and pressure over a 96 hour period.

Tests on valves and digital simulations were carried out on the basis of this typical profile representing a LOCA.

3.2.1. Valves tested.

Different valves were tested in order to assess the significance of the boiler effect in the event of a LOCA and of its consequences for equipment (internal and external seals in particular).

The parameters of influence studied are as follows:

- valve type,
- nominal diameter,
- nominal pressure,
- the body material.

For instance, 3 different valve types tested:

- valve 1: flexible-wedge gate valve ND250 NP250,
- valve 2: double-disc parallel seat gate valve ND100 NP20,
- valve 3: double-disc parallel seat gate valve ND350 NP50.

Valves were placed in a test vessel which enables the desired temperature and pressure profiles to be reproduced and the desired super-heated steam conditions to be obtained. The test vessel also enables the sprinkling of boric acid and soda to be reproduced.

3.2.2. Test conditions.

The initial test conditions were selected such that the following could be achieved in the most rigorous of cases:

- good internal and external seals,
- bonnet cavity space filled with water and vented,
- cavity pressurized to the service pressure plus the "piston" effect.

The instrumentation implemented enabled the temperature (environment, body, liquid) and pressure (environment, liquid) evolutions to be monitored over time.

3.2.3. Results.

The internal pressure rises were generally very quick and of high magnitude. The temperature and pressure maxima were obtained between 10 and 40 minutes after the start of the test. Figure 1 shows the evolution of the different parameters measured over time for valve 2:

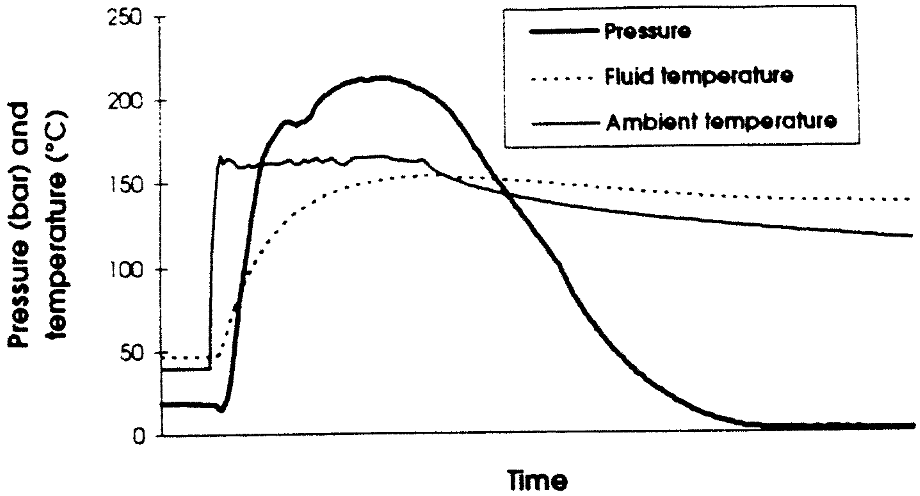


Figure 1. Evolution of temperature and pressure over time for valve 2.

The liquid temperature shadows the ambient temperature rise with a slight delay. This divergence arises as a result of the steam/body and body/liquid exchange coefficients and the thermal inertia (a) of the valve and particularly (b) of the liquid. The pressure rise is very sudden and evolves via a maximum value attained when the temperature is at maximum level.

The results obtained depend greatly on the differing characteristics of the valves tested (valve type, ND, NP, materials). It can be observed that pressure evolutions are as follows:

- the highest pressure is obtained for the valve of which the NP is highest (valve 1),
- the smallest valve (valve 2) quickly reaches maximum pressure but the latter is lower than for valve 1,
- there is no pressure rise in valve 3: an internal seal fault due to displacement of the discs at the start of the test prevented the boiler effect from being triggered. In a second test carried out with slightly different initial conditions, the boiler effect was triggered and pressure rose.

Figure 2 shows the pressure evolution as a function of the liquid temperature for valve 1:

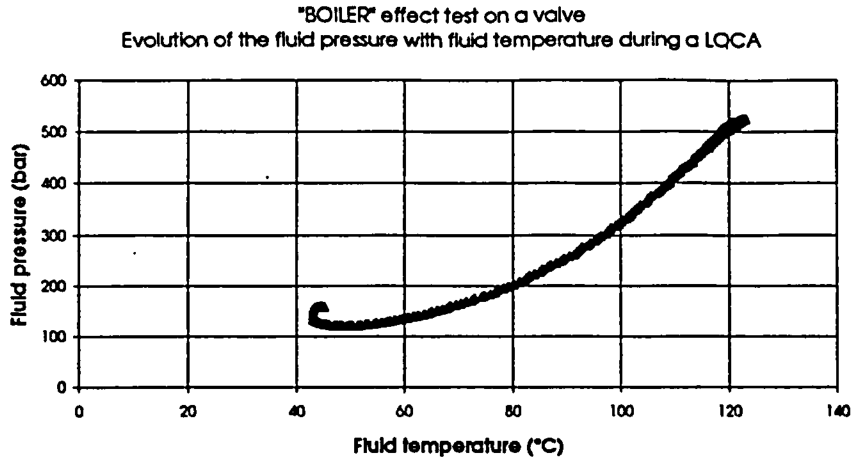


Figure 2. Pressure evolution as a function of the liquid temperature for valve 1

The graph shows the violence of the phenomenon: the pressure rise gradient reaches 9.3 bar/°C liquid temperature.

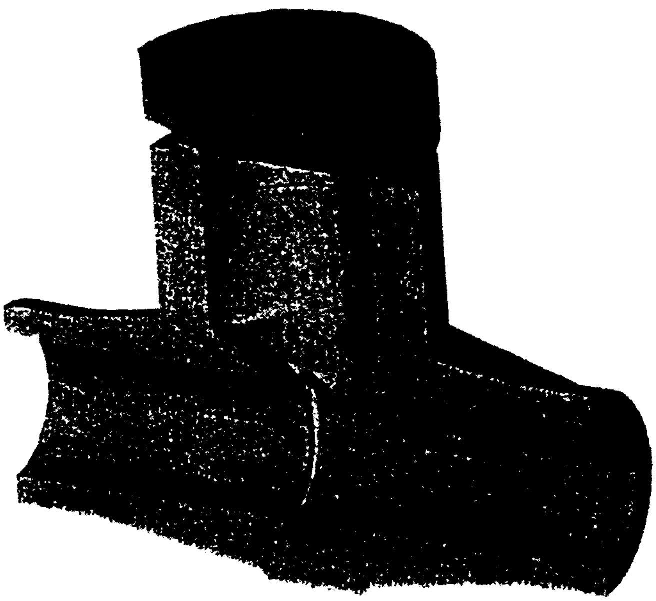
3.2.4. Conclusion.

Despite the magnitude of the pressures reached (up to ten times the nominal pressure of the valve), no external leaks were observed. Internal seals were afterwards in just as good order as before the test, except for valve 3: when subjected to pressure, the discs became deformed and a significant leak was observed after the test.

3.3. Digital approach

In order to improve our understanding of the influence of the different parameters participating in the boiler effect and in order to analyze the instances of un-tested valves, a digital approach involving modelling the valve with finite elements was used. The computations, carried out with the assistance of the ASTER code (developed by EDF), were re-encoded using the tests (see paragraphs 3.1.and 3.2.).

3.3.1. Models. These were constructed using the I-DEAS software produced by SDRC & Co.. Figure 3 shows an example of a model (valve 1) :



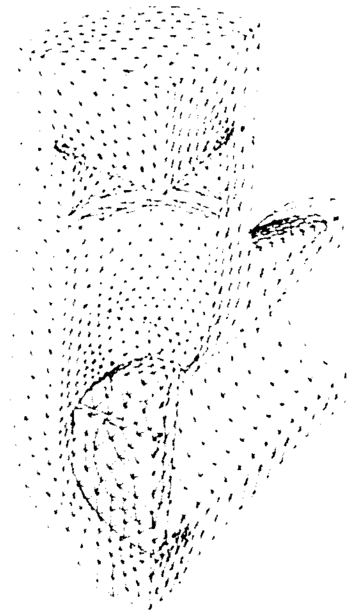


Figure 3. Views of a mesh geometry

The model integrates different components of the valve: the body, the bonnet, the disc, the body/bonnet pins. Contact components were also integrated into the model in order to simulate contact between the bonnet and body and between the body and disc.

3.3.2. Thermal and mechanical engineering computations.

These were carried out using the ASTER computation code.

Figure 4 shows the results of the thermal computation of valve 1 in the case of a LOCA type accident :

Figure 4. Temperature range in valve 1 after 3 minutes (case of a LOCA type accident).

The mechanical computations also enable the behaviour of important areas to be analyzed:

- area affecting the internal seal: by causing pressure to evolve gradually, the pressure at which the disc begins plasticization can thus be discovered.

- area affecting external sealing: the opening of the body/bonnet joint when the assembly is subjected to internal pressure can be observed.

3.3.3. Boiler effect pressure.

On the basis of the thermal computation carried out, the volume dilation for each liquid link is evaluated using the link temperature and the dilation coefficient. Taking the mechanical computation into account, we can obtain the volume variation of the valve as a function of internal pressure, thus providing the law of evolution of boiler effect pressure as a function of ambient temperature. Figure 5 shows the evolution of internal pressure and temperature obtained by computation and that obtained by test (valve 3):

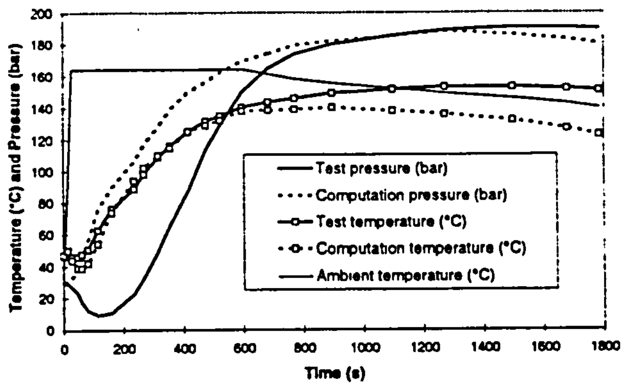


Figure 5. Evolution of internal pressure and temperature obtained by computation and that obtained by test

3.4. Technical solutions for nuclear power plants

In order to solve the problem of the boiler effect in nuclear power plants, whether operational or in the course of construction, two studies have been begun:

- on the basis of safety criteria, determination of perimeter-internal valves for which the boiler effect is unacceptable,
- design and confirmation of technical anti-boiler effect solutions.

If possible, the most simple solution is to make a small hole in the seat or in the disk in order to set up a connection between the valve body and the pipe. For valves required to be tight in one direction only, this is a good solution because the second disc/seat insures tightness. Another solution to prevent these harmful effects consists in an external bypass line between the valve body and the pipe. This bypass is equipped with a manual isolation valve to be able to cut the flow completely, if necessary. For valves required to be tight in both directions, the bypass is connected to both upstream and downstream pipes and a passive selector valve turns the overpressure to the lowest pressurised side.

4. Conclusions

The studies described in this paper show the difficulty in solving both problems:

- Because of the complexity of motor operated valves, the determination of the operating torque required is very hard to find;
- As a result of its high degree of sensitivity to specific features in the condition of the valve and its environnement, the overpressure due to boiler effect is very complicated to evaluate.

Using tests and studies carried out within EDF, these phenomena and their significance are now better known and understood. Their consequences for valves and nuclear power plants, particularly in the event of an accident, have been assessed and modifications are currently on the way.

Hydrodynamic Behavior of Check Valves

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Electricité de France

ABSTRACT

The presenter will dicuss his experiences in assessing the hydrodynamic behavior of check valves.

Optimization of Residual Heat Removal Pump Axial Thrust and Axial Bearing

Dr. Falko Schubert
Siemens KWU FTM3

INTRODUCTION

The residual heat removal (RHR) pumps of German 1300 megawatt pressurized-water reactor (PWR) power plants are of the single stage end suction type with volute casing or with diffuser and forged circular casing. Due to the service conditions the pumps have to cover the full capacity range as well as a big variation in suction static pressure. This results in a big difference in the axial thrust that has to be borne by the axial bearing. Because these pumps are designed to operate without auxiliary systems (things that do not exist can not fail), they are equipped with antifriction bearings and sump oil lubrication. To minimize the heat production within the bearing casing, a number of PWR plants have pumps with combined axial/radial bearings of the ball type. Due to the fact that the maximum axial thrust caused by static pressure and hydrodynamic forces on the impeller is too big to be borne by that type of axial bearing, the impellers were designed to produce a hydrodynamic axial force that counteracts the static axial force. Thus, the resulting axial thrust may change direction when the static pressure varies.

PROBLEM IDENTIFICATION

During an inspection in the plant, a RHR pump was found to stop rather quickly when the motor was deenergized. After dismantling the axial/radial bearing was found to be severely damaged. A root cause examination was started to evaluate the reason for this damage. It was discovered that there was no lack of oil in the bearing housing, no contamination of the oil, and no pollution that might have caused the damage. The bearing calculation was checked again and found to be correct.

A closer visual inspection on the other RHR pumps indicated axial oscillations of the pump shaft at a certain static suction pressure. Thus, the resulting axial thrust should be zero. Now the failure of the axial/radial bearing could be explained because the installed bearing type is not allowed to run without axial thrust for a longer period of time resulting in zero axial thrust. But this fact did not correspond with the axial forces calculation. Different calculational methods were then tested to find a correct calculation. But this evaluation led to widely spread results.

NEW CALCULATIONAL METHOD

At about that time, the thesis of the author was finished, providing a calculational method for axial hydrodynamic forces on centrifugal type impellers. Using this method, the calculation of resulting axial forces was repeated. In the first step, the original pump design was checked. The service conditions which lead to zero axial thrust could be verified. In the second step, the pump was modified to optimize the axial thrust regarding the different service conditions and estimated periods of time in respect of a maximization of axial/radial bearing lifetime. Additionally,

the radial/axial bearing type was changed to a different (ball type) bearing which allowed service at zero axial thrust. In the third step, one pump was modified and tested in the plant. Again, the service conditions which lead to zero axial thrust could be verified.

RESULTS

The results were submitted to the authorities. After checking the calculations and the test results, the authorities agreed to lengthen the period of time between bearing examinations so that the examinations could be done together with the routine service inspection of the pump, as was the practice before the bearing failure occurred. The remaining pumps were modified step by step. Since then, no bearing problems have occurred on the modified pumps.

VIBRATION OF SAFETY INJECTION PUMP MOTORS

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1 - OBJECT

This paper covers a fault encountered in the safety injection pump motors of the French 900 MWe unit nuclear power stations. This fault was not revealed either during the low pressure safety injection and containment spray system pump qualification tests under accident conditions or during the special tests on a test bench carried out to attempt to replicate the fault and to identify ways of remedying it. This constitutes a potential common mode of failure of the safety injection system and the containment spray system pumps. The vibration phenomena illustrate the importance of carrying out tests in the plants under conditions as close as possible to those of actual accident situations.

2 - INTRODUCTION

At the request of the Nuclear Installations Safety Directorate (French acronym DSIN), subsequent to the safety review of the 900 MWe units of early design, tests were scheduled by the operator for July 1992. These tests of the low pressure safety injection and containment spray system pumps with recirculation via the reactor building containment sump took place during a refuelling outage of Unit 2 of Fessenheim Nuclear Power Station. The purpose was to verify, on-site, the behaviour of the pumps with an inadequate net positive suction head. These tests, which placed demands on the equipment in conditions which are far closer to an accident situation than those encountered during periodic testing, revealed abnormal vibration of the pump drive motors. These vibrations are liable to constitute a potential common mode of failure of the safety injection and containment spray function in the event of an accident when use of these systems is required. The low pressure safety injection and containment spray system pumps are of identical design.

Excessive tightening of the outer ring of the lower ball bearing of the motors was thought to be the cause of the fault. Test bench trials were undertaken by Electricité de France (EDF) to confirm this hypothesis. In these tests, abnormal vibration did not occur, even with considerable tightening. However, they did show that a motor exhibiting vibration beyond the criteria for maintenance shutdown could operate for some one hundred hours without suffering damage or loss of performance. This substantial period is adequate for taking measures involving the installation of mobile equipment to avoid loss of core cooling capability. While these investigations were being carried out, the motors of all the 900 MWe unit nuclear power stations were brought to compliance with the operating conditions laid down by the manufacturers (proper sliding of the outer ring). In so far as the root cause has not been identified, the Institute for Nuclear Protection and Safety (French acronym IPSN) thinks that additional research should be performed to explain the fault observed.

3 - DESCRIPTION OF THE PROBLEM

On 29 July 1992, during a refuelling outage of Unit 2 of Fessenheim Nuclear Power Station, a test of the low pressure safety injection pump with recirculation via the reactor

building containment sump revealed vibrations reaching 230 mm and a speed of 12 mm/sec after some 6 hours of testing. These values are far higher than the shutdown criteria for this type of pump (amplitude 140 mm, speed 11.2 mm/sec). Vibration was strongest at the upper bearings of the motors. It did not result in motor damage.Testing of the low pressure safety injection system pumps with recirculation via the sump involves operation at outputs of around 830 to 850 m^3/h, which are lower than the maximum output planned for operation with recirculation via the containment sumps (1020 m^3/h).For reference, during the periodic tests on the low pressure safety injection system pumps, throughput did not exceed 240 m^3/h, with the exception of the functional tests with reactor vessel open when the pumps are required to provide high output for a period limited to around 30 minutes. During the periodic tests of the containment spray system pumps with their zero flow line, output is around 800 m^3/h.

Experience feedback from other 900 MWe capacity nuclear power stations: The operating conditions of the containment spray system and low pressure safety injection system pumps of the nuclear power stations with 900 MWe reactors of recent design are similar to those of the two power stations (Bugey and Fessenheim) of early design. There is, therefore, concern that the problem may be common to all the 900 MWe reactors.

Experience feedback from other 1300 MWe reactor nuclear power stationsIn the past, the endurance tests carried out at Saint-Alban Nuclear Power Station on the containment spray system with recirculation via the sump resulted in excessive vibration and replacement of the upper single-direction thrust bearing of the motor with a double-direction thrust bearing (double-row ball bearing) in all power stations with 1300 MWe reactors. This made it possible to solve the problem of vibration of their containment spray system pumps.

4 - SAFETY ASSESSMENT

The immediate and potential consequences of the problem found in the pumps are the following:

Immediate consequences: Apart from the appearance of high vibration, this problem had no immediate consequences. Expert appraisal of different components of the equipment failed to reveal any sign of damage. The motor-to-pump coupling, which is not of the universal joint but of the gear type, did not show the slightest sign of seizing or friction. The expert appraisal carried out on the manufacturer's premises showed that this component was undamaged. Similarly, expert appraisal of the bearings did not reveal any sign of a significant wear, and their grease did not show any signs of ageing.

Potential consequences: As the containment spray system and the low pressure safety injection system pumps are of identical design, the problem can affect either of the two engineered safety features at random. Furthermore, the defect observed may equally well affect one or both of the containment spray system and low pressure safety injection system pumps. It therefore constitutes a common mode of failure. It is extremely difficult to assess the durability of equipment subjected to vibration. The process of wear is progressive, and can accelerate to failure of the component. It is therefore probable that after a few hours, or days, there is a real risk of loss of equipment. Analysis of the potential consequences has consisted of assessing the risk of loss of the containment

spray system and/or low pressure safety injection system by comparing the conditions of appearance of excessive vibration during testing with the operating conditions planned for the containment spray system and/or low pressure safety injection pumps in the event of loss of coolant accident (LOCA).

1) Risk of loss of equipment: The low pressure safety injection system pumps are designed to provide the following outputs for the following lengths of time:

	Output m^3/h	Duration of Operation
Direct injection	740	1/2 hour
Recirculation	1020	A number of months up to 1 year

The containment spray system pumps are designed to provide the following outputs for the following lengths of time:

	Output m^3/h	Duration of Operation
Direct spraying	925	1/2 hour
Recirculation spraying	1100	A number of days to a number of months

These tables show that the onset of excessive vibration during the direct spraying or direct injection phase is unlikely. On the other hand, in the longer term, during operation in the recirculation mode, the possibility of loss of equipment ought to be taken into account.

2) Impact on the low pressure safety injection function: Several types of LOCA accidents can occur. The case of intermediate and large breaks in the reactor coolant system has been examined with particular care, as such accidents give rise to the same control and flow requirements as safety injection. The two low pressure safety injection system trains start up simultaneously, and each train is designed to be capable of providing the function alone. The two safety injection pumps discharge simultaneously into the hot and cold legs, so as to avoid crystallisation of borate in the reactor vessel. If one pump is lost, the result is partial failure of the safety injection function, but the pump remaining in service can provide the necessary output. If both pumps are lost, even if they are not lost simultaneously, loss of the safety injection function must be envisaged during its use. It is therefore necessary to resort to the H4 beyond-design-basis accident procedure (total loss of the pumping and/or heat exchange capability in the event of a loss of primary coolant accident). This procedure provides for backup of the low pressure safety injection system pumps with the engineered safety feature containment spray system, discharging through the safety injection lines, with the low pressure safety injection system pumps stopped.

3) Impact on the containment spray system function: After an accident of the large or intermediate break type, the two trains of the engineered safety feature containment spray system begin operating, on a redundant basis, for a period of 24 hours. After this period, the operator may opt to run only one train. Each train is designed to provide the output necessary for the temperature and pressure drop in the reactor building containment. Therefore there is a standby pump for the duty pump capable of providing the required output alone. It can be considered that 100 hours into the accident (with a pressure of 1.5 bar and a temperature of around 75°C inside the reactor building containment), the reactor building containment spraying phase is practically over when an engineered safety feature containment spray system pump may be called upon to provide backup safety injection (application of the H4 accident procedure).

5 - ANALYSIS OF CAUSES AND CORRECTIVE ACTIONS

5.1 - Causes:

Design of motor: The motors of the containment spray system and low pressure safety injection system pumps are of identical design (see Figure 1) in all 900 MWe units. These motors thus constitute a common risk of failure. The motor consists of a stator with a rotor supported by a shaft guided by two ball bearings. The lower bearing is of the parallel type with balls. This type of bearing is normally force-fitted on the shaft and fitted with a slight interference on its outer ring. This bearing is intended to bear radial loads only (end-play possible).The upper bearing is of the tapered type with balls. It is designed to bear downward thrust loads, specifically the weight of the rotor. The particularity of this type of bearing is that the inner ring can move relatively to the outer one in the direction opposite to that of the forces it is intended to bear (in which case guidance is lost, see Figure 2). The loading caused by vibration to which the motor is subjected consists of:

- the mass of the rotor to be borne by the upper bearing,
- the forces caused by differential expansion, resulting from heating of the motor,
- vibrations caused by the pump.

Causes: During tests at Fessenheim Nuclear Power Station, the frequency of vibration was observed to be 50 Hz, corresponding to twice the motor speed (1500 rpm). This frequency is characteristic of so-called "spinning top" operation, and Electricité de France adopted the hypothesis of binding of the outer ring of the lower ball bearing on the motor stator. Indeed, if it is considered that the shaft is anchored at this ball bearing, differential rotor/stator expansion results in unloading the upper bearing and loss of guidance (see Figure 2). The rotor then operates as a "spinning top" as the motor shaft is only restrained by the lower ball bearing.In so far as the root cause has not been identified the Institute for Nuclear Protection and Safety thinks that additional research and examination should be performed in that field.

5.2 - Bench testing: The operator held two bench test sessions (see Figure 3) to verify its hypothesis relating to binding of the outer ring of the lower ball bearing on the motor stator. The test bench includes a hydraulic brake enabling the motor to operate at its rated power. The results of these tests were submitted to the IPSN.

First series of tests: Axial movement of the outer ring of the lower ball bearing possible on the motor stator. A first campaign was carried out to prove that the motor did not suffer from vibration when the outer ring of the lower ball bearing could slide on the motor stator. Tests were carried out with clearances of 10 and 60 mm. No faults were observed during these tests.

Second series of tests: Binding of the outer ring of the lower ball bearing on the motor stator. At the request of the IPSN, the operator took steps to demonstrate, on the test bench, the hypothesis of binding of the outer ring of the lower bearing to validate the first tests. Therefore, the operator included in its test programme a configuration with maximum interference to verify the appearance of vibrations exceeding the criteria as soon as thermal equilibrium of the rotor was obtained. Interference of up to 31mm

(the maximum interference theoretically possible with the manufacturing tolerance limits) was obtained by modifying the housing diameter. No abnormal vibration was observed.

In the light of these findings, the IPSN considered that the origin of the fault had not been identified. Therefore, during the tests, Electricité de France ran a motor with excessive vibrations for some one hundred hours, corresponding to the four days necessary for fielding the mobile resources required in the H4 and U3 beyond-design-basis accident procedures (backup of the containment spray system and low pressure safety injection systems with mobile equipment). To do this, a mechanical device for artificially raising the rotor was installed to simulate loss of guidance of the motor rotor by the upper ball bearing and to obtain vibration similar to that found during the initial fault described in §2. The tests under these conditions lasted for 100 hours without any damage being detected in visual examination carried out after the test, and without the motor performance having changed.The results are interesting as they show that in the event of vibration in a motor largely exceeding the shutdown criteria, there is sufficient time to make arrangements to avoid loss of core cooling capability. The mobile equipment consists of an emergency pump connected with a flexible pipe.

5.3 - Corrective Actions:

• *Campaign for verifying and readjusting the lower bearings of the motors:* As soon as the nature of the problem was revealed, Electricité de France began checking proper sliding of the outer ring of the lower ball bearing on the basis of the functional clearance between the ring and the motor stator in all the nuclear power stations. The motors were divided into two categories depending on whether sliding was considered possible or not.

a) For the lower ball bearings with a clearance between the outer ring and the motor stator greater than 20 mm, sliding of the outer ring of the lower ball bearing on the stator was possible as the functional play was known with an accuracy of 15 mm. The ball bearings were therefore retained.

b) For the lower ball bearings with a clearance between the outer ring and the motor stator either unknown or less than 20 mm, sliding of the outer ring of the low ball bearing on the stator being uncertain, the ball bearings were therefore re-adjusted to enable sliding.

• *Supplementary and compensatory provisions made by Electricité de France:* In view of the satisfactory behaviour of the equipment in a long test with abnormally high vibration, EDF adopted the hypothesis that a motor can operate for 100 hours without damage in the presence of high vibration (second series of tests, see §4.2). EDF has provided two remedial lines of defence for partial or total failure either of the safety injection or containment spray functions:

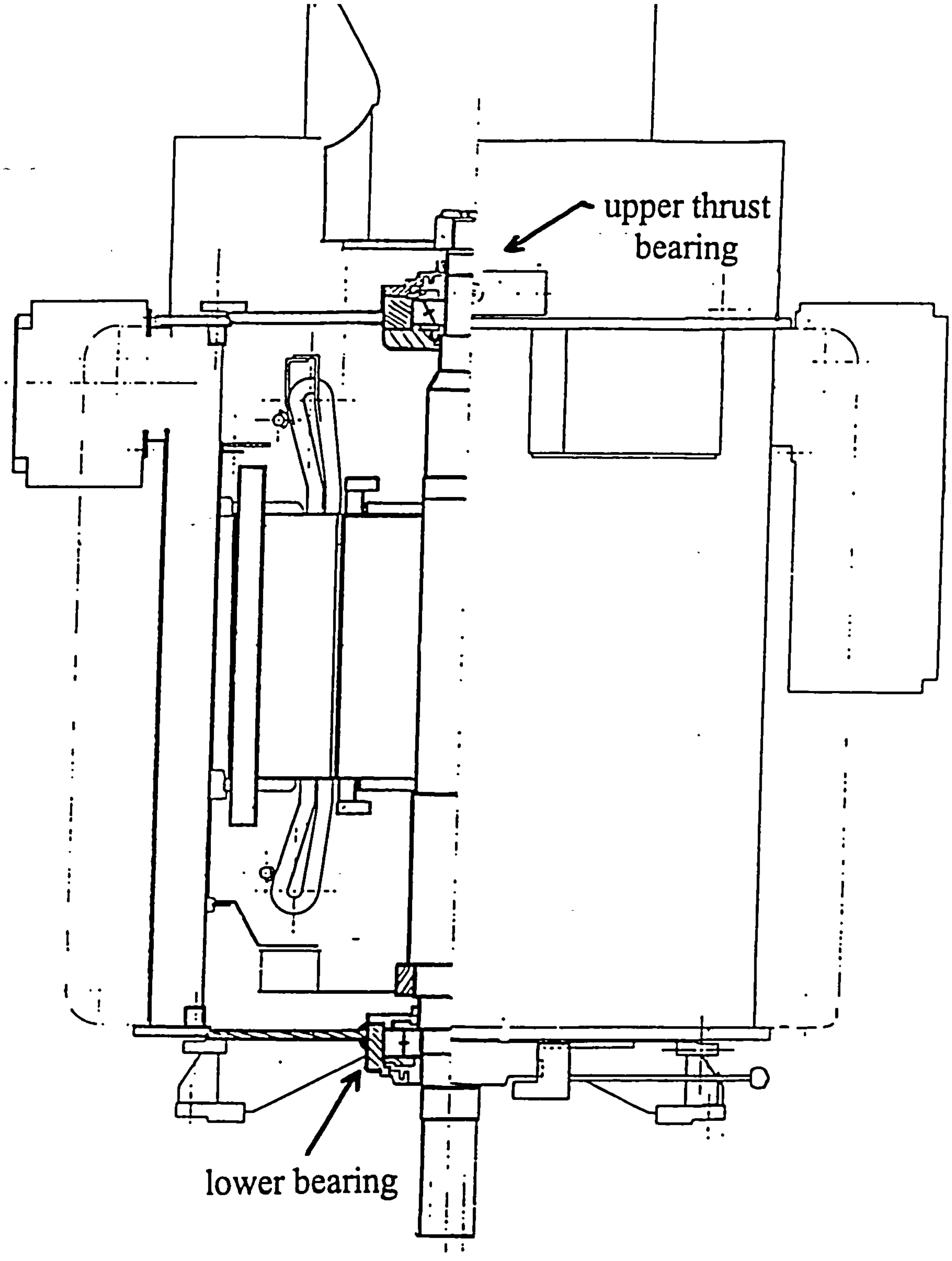
a) <u>Restoration of the function by replacement of the failed equipment.</u> Electricité de France considers that in such a situation, 60 hours would be necessary to procure and install the motor (only 35 hours if the motor is available on-site).

b) Resorting to the mobile equipment provided for in the H4 and U3 accident procedures. In the event of failure of the last safety injection or containment spray pump, partial cooling of the core is possible using the resources provided for in the H4 and U3 accident procedures.

These arrangements are provided to enable mutual backup of the containment spray system and low pressure safety injection system pumps, and make it possible to cope in the event of total loss of these pumps. Corresponding resources are permanently installed in the two 900 MWe reactor nuclear power stations of early design (Bugey and Fessenheim), whereas provision is made for mobile equipment in the other 900 MWe reactor nuclear power stations of more recent design. They necessitate the installation of flexible pipes and a backup pump driven by an internal combustion engine. When it is operating, water is injected into the reactor coolant system via the containment spray system heat exchanger and one train of the engineered safety feature low pressure safety injection system. In this situation, spraying inside the reactor containment is abandoned in favour of reactor core cooling.

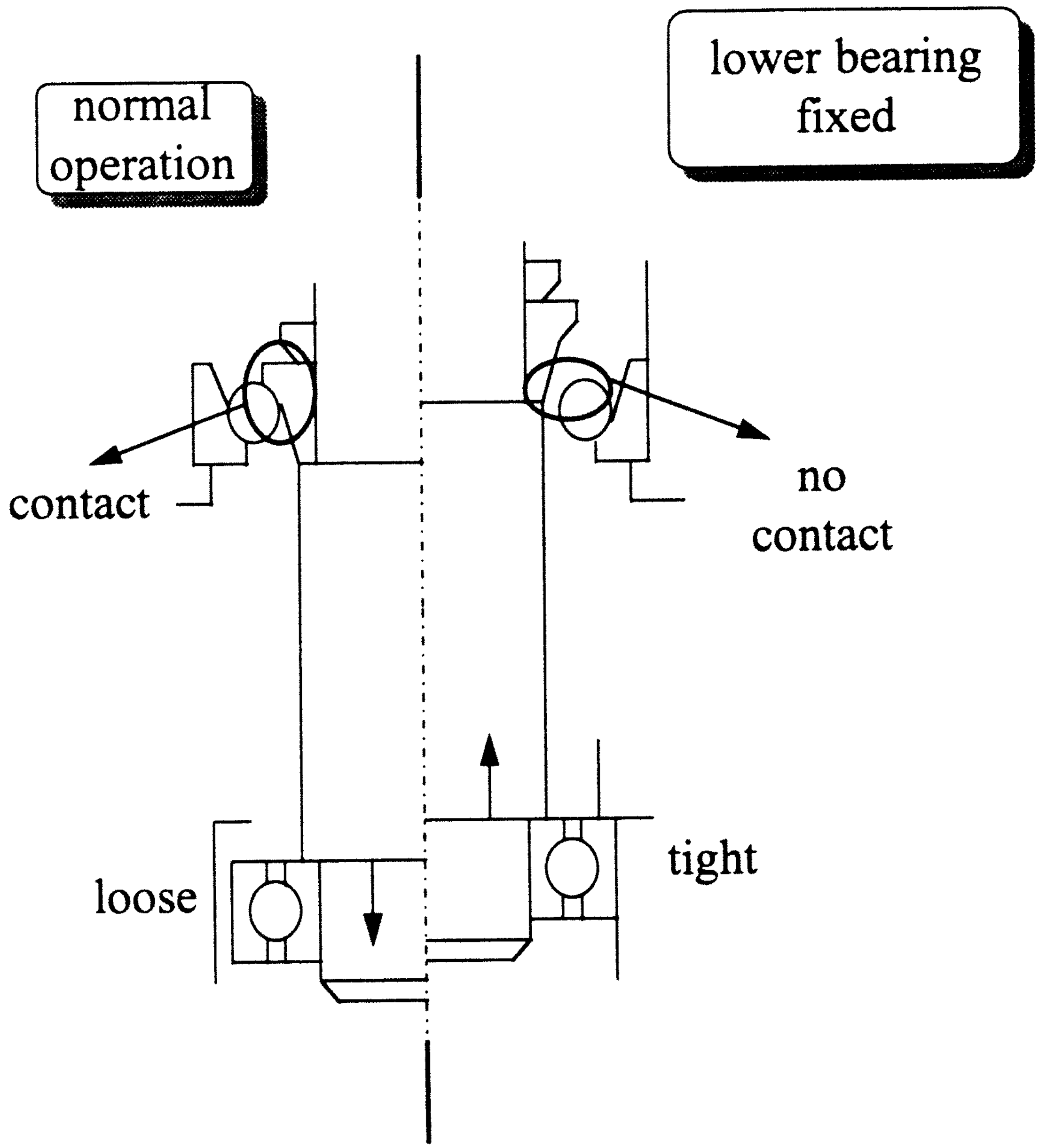
6 - CONCLUSIONS AND LESSONS LEARNED

A specific test carried out in a plant involving demands on the equipment under conditions far closer to operation during an accident than those which are applied during the periodic tests revealed a fault liable to constitute a potential common mode of failure of the pumps of both the engineered safety feature low pressure safety injection and containment spray systems. It should be borne in mind that the probabilistic safety assessments use reliability data derived from the failure rates observed during normal operation which, for this type of equipment, correspond to the periodic tests. This fault was not revealed either during the low pressure safety injection and containment spray system pump qualification tests under accident conditions or during the special tests on a test bench carried out to attempt to replicate the fault and to identify ways of remedying it. The vibration phenomena illustrate the importance of the details of assembly and supporting of equipment, and shows the need for carrying out tests in the plants under conditions as close as possible to those of actual accident situations.



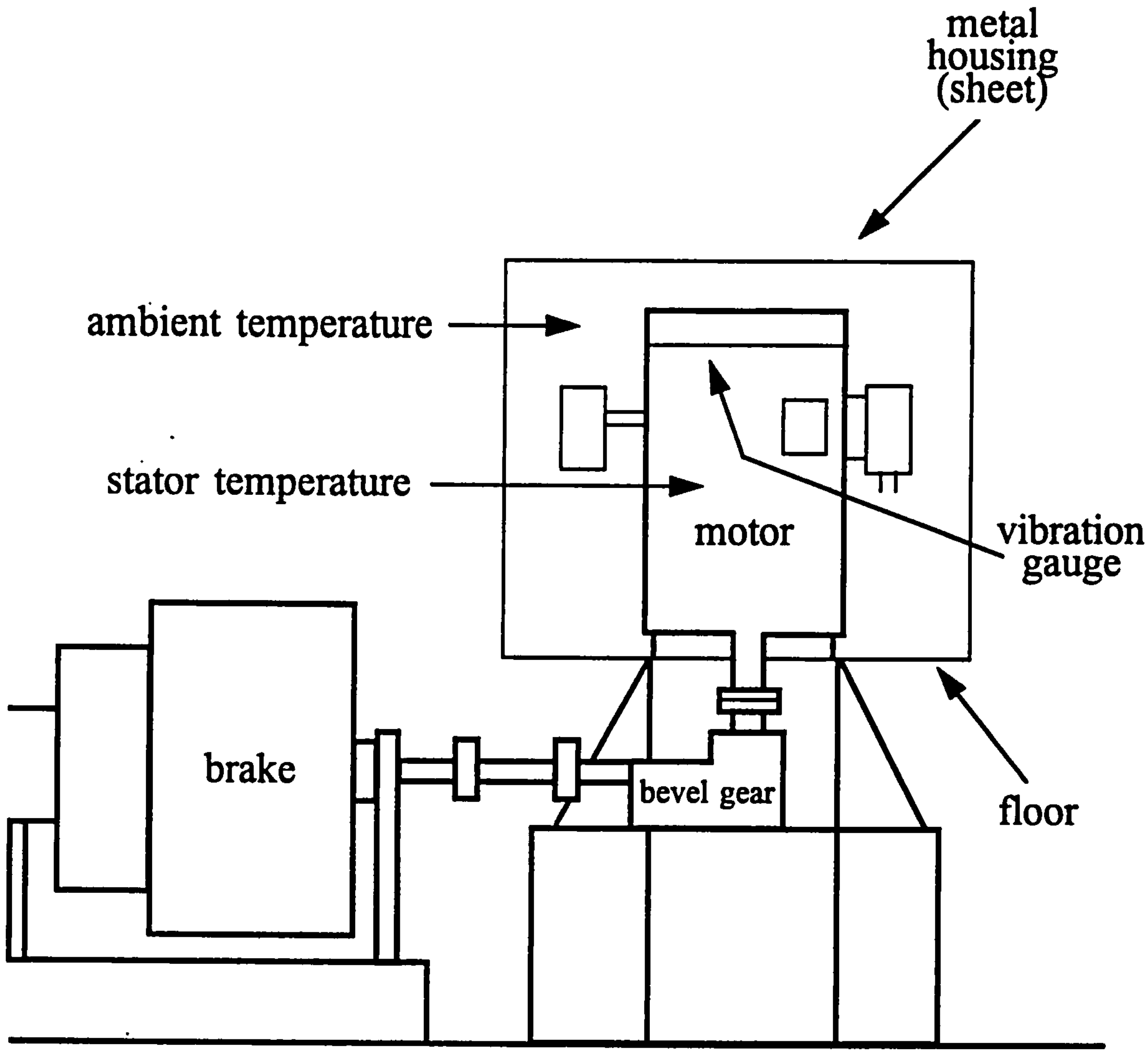
MOTOR OVERALL DRAWING

FIGURE 1



RELIEF MECANISM OF THE UPPER BALL BEARING

FIGURE 2



MOTOR TEST BENCH

FIGURE 3

FAILURES OF THE THERMAL BARRIERS OF 900 MWE REACTOR COOLANT PUMPS

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1 - SCOPE

This report describes the anomalies encountered in the thermal barriers of the reactor coolant pumps in French 900 MWe PWR power stations. In addition to this specific problem, it demonstrates how the fortuitous discovery of a fault during a sampling test enabled faults of a generic nature to be revealed in components which were not subject to periodic inspection, the failure of which could seriously affect safety. This example demonstrates the risk which can be associated with the deterioration in areas which are not examined periodically and for which there are no preceding signs which would make early detection of deterioration possible.

2 - INTRODUCTION

The reactor coolant pumps in French 900 MWe PWR power stations are equipped with a thermal barrier (see Figure 1) which is cooled by a coil to prevent hot water from penetrating in the event of injection system failure at the pump shaft seals. This coil, protected from the primary coolant system by the thermal barrier housing, is fed by the Component Coolant System (see Figure 2). During a sampling test carried out in 1990 on the No. 2 reactor coolant pump at Fessenheim 2 power station (95,000 hours of operation), to check the state of the reactor coolant pumps after ten years of operation, a crack was detected during visual examination of the outside of the thermal barrier housing. The investigations carried out revealed other cracks inside the housing and cracks on the underside of the thermal barrier flange. The first anomaly affects all the reactor coolant pumps in French 900 MWe PWR power stations. Testing of the coil isolation check valve, which was carried out following the safety analysis study, revealed a problem of jamming of this check valve. This second problem also affects all the reactor coolant pumps in French 900 MWe PWR power stations.

3 - ANOMALIES DISCOVERED

The dye penetrant examinations made on the inside of the thermal barrier housing of the No. 2 reactor coolant pump at Fessenheim 2 to check whether the crack (50 mm long) on the outside of the housing went all the way through it, revealed a circumferential crack, which extended around a 180° sector of the housing, as well as a network of cracks caused by thermal cracking affecting a 35° sector, located above it (see Figure 3). The maximum depth of the cracks found on the inside and outside of the housing was 10 mm, in other words approximately half the thickness of the thermal barrier housing, and they were located opposite each other, so the minimum thickness of the non-cracked metal was 1 mm. The tests carried out on the thermal barrier housings of the reactor coolant pumps being serviced in the operator's workshops (following various problems) did not show up cracks on the outside, but revealed some on the inside, in particular, a crack 3.5 mm deep after only 35,000 hours of operation. Dye penetrant examination of the

underside of the thermal barrier flange of this pump after removal of the housing, revealed cracks (0.3 mm deep). The discovery, in 1992 of deeper cracks (in particular one 3.5 mm deep) on the thermal barrier flange of the No. 1 reactor coolant pump at Fessenheim 1, led to the systematic dye penetrant examination of flanges when rejected thermal barrier housings are being replaced. The cracks observed take the form of networks with linear indications, located between the coil housing nozzles and around the same radius (see Figure 4). Following the safety analysis study which revealed a risk of cooling coil deterioration, the coil isolation check valves were tested, making it possible to see that numerous check valves were jammed open.

4 - STATE OF 900 MWe PWR PLANTS

Thermal barrier housing.
The external cracking phenomenon, which was discovered first, was due to the properties of the material used. The thermal barrier housings which were made of material that exhibited the same properties have been replaced. Since then, no external cracks have been discovered during testing of the housings. Considering the generic nature of the internal cracking phenomenon of the housings, a campaign was undertaken to test those in all the reactor coolant pumps of 900 MWe PWRs. The results of the tests carried out at the end of 1995, which covered almost all the reactor coolant pumps of 900 MWe PWRs (98 out of a total of 102 pumps), show that practically all the housings had internal cracks. The tests carried out on a reactor coolant pump of a 1300 MWe PWR power station did not reveal the same cracking phenomenon in the thermal barrier housing.

Thermal barrier flange.
The tests carried out on the thermal barrier flanges available in Electricité de France workshops demonstrated the generic nature of this type of fault, so it was decided to check the flange by dye penetrant examination each time a thermal barrier housing is removed. Practically all the flanges tested had cracks of varying depths (reaching 7 mm deep).

Check valves of the thermal barrier coil.
Following discovery of check valves which were jammed open, all 900 MWe PWR power stations were requested to test these check valves during refuelling outages. In addition, considering that the thermal barrier coolant circuit for the reactor coolant pumps of 1300 MWe PWR power stations is of identical design and that the check valves are also of the same design, a sampling test of these was carried out. It revealed the same fault.The expert appraisal operations carried out revealed that the check valves used (lift-type check valves) were jammed by a layer of metallic oxides which came primarily from the Component Coolant System, the piping of which is made of carbon steel. The results of the tests revealed that the percentage of check valves which were jammed was 66% for 900 MWe, and 60% for 1300 MWe PWR power stations.

5 - RISK ANALYSIS

Safety consequences - Consequences of thermal barrier housing cracking
The development of circumferential cracks in the thermal barrier housings can lead to the base of the housing becoming detached, then falling onto the wheel and rotating with it. Two similar incidents have already occurred. One was at Zaporozhe (Ukraine) on 22 March 1991, on a reactor coolant pump with a different technological design (IRS Report No.

6279.00) and the other at Fukushima Daini (Japan), on 6 October 1989, involving the rupture of a hydrostatic bearing ring of a recirculation pump in a BWR power station (IRS Report No. 959.02). In each case, significant quantities of metallic particles were spread through the primary coolant system. The possible consequences of the housing detaching are:

- jamming of the pump impeller,
- formation of loose parts,
- damage to the cooling coil.

Consequences of thermal barrier flange cracking
According to metallographical examinations carried out regarding thermal barrier flange cracking faults, the cracks are able to propagate in three directions:

- around the circumference, towards the coil nozzles, which could lead to perforation of the coil,

- towards the housing-to-flange weld, which could result in the housing or metal particles becoming detached, and the formation of loose parts could lead to wear on the coil, or

- in the thickness of the flange, which could result in its sudden disintegration.

In the first two cases, the consequences are identical to those resulting from cracking of the housing. In the case of flange disintegration, the consequence is a loss of coolant accident (LOCA) in the containment.

Consequence of jamming of the cooling coil check valve.
Perforation of the coil, whether caused by rupture of the thermal barrier housing or cracking of the flange, or by gradual wear from a loose part trapped in the thermal barrier housing cavity, results in primary fluid entering the cooling coil. Failure of the cooling coil check valve will then lead to a rise in pressure in the Component Coolant System, which is not designed to withstand primary pressure.

Risks run- Pump rotor jam.
The thermal barrier housing cracks are not always located at the same level, and, in the event of detachment of the base of the housing, the possibility of the part of it remaining attached to the thermal barrier flange and the base of the housing turning with the wheel clashing against each other, cannot be excluded. However, as the maximum inclination of the base of the housing is limited by the shape of the components present, the risk of jamming of the rotor by wedging, is remote. In addition, the experiments carried out on friction between materials, demonstrate that the risk of seizure caused by friction in the presence of water, between the thermal barrier housing and the wheel or the shaft sleeve, is low.

Formation of loose parts.
If the base of the housing becomes separated, as indicated above, the interaction of the two parts of the housing will lead to deterioration of the components and pieces of metal will become detached:

- if the base of the housing does not break up, only the small pieces (due to the gaps of around 1 mm in the labyrinths along the shaft) will go into the primary coolant system or rise towards the shaft seals, or

- if the base of the housing breaks up, larger pieces will be able to go into the primary coolant system.

The consequences of loose part formation could be as follows:

-for small sized particles:

•if the loose part is carried along by the existing flow between the thermal barrier cavity and the primary coolant system (see Figure 1), increase in primary activity due to wear of the fuel cladding (effect observed during the Zaporozhe incident) or jamming of one or several control rod clusters,

•if the loose part is carried along by the flow through the leak at seal No.1, damage of seal No.1 of the corresponding pump with the risk of a loss of coolant accident.

-for medium sized particles:

•jamming of a control rod cluster,

•local damage to internals and to the coating on the bottom of the reactor vessel.

Loss of coolant accident (LOCA) in or outside the containment.
This risk is the consequence of either:

- damage to the shaft seals in the event of formation of loose parts,
- or sudden disintegration of the flange
- or entry of primary coolant into the Component Coolant System.

Damage to the shaft seals and sudden disintegration of the flange would lead to a LOCA in the containment. The entry of primary fluid into the Component Coolant System would put it under pressure. If the leak was greater than the flow rate of the in-containment relief valve (estimated to be 20 m^3/h), the Component Coolant System would then rupture and, depending on which part of the system broke, a LOCA inside or even outside the containment would ensue, because the isolation valves of the Component Coolant System in the containment are not designed to isolate a system which is under primary pressure.

6 - ORIGIN AND DEVELOPMENT OF CRACKS

The origin of thermal barrier housing cracking.
The metallographical examinations carried out demonstrate that the external cracking discovered on some thermal barrier housings is intergranular, and that it is the result of brittle rupture of the niobium carbides where the grains join. The reason for this is the very low ferrite content (less than 0.5%) in the steel of the housing, which is Z6 CN Nb 18-11 [UNS S34700 SS]. Metallurgical expert appraisal shows that the internal circumferential cracks which are located at the blend radius (see Figure 3) and develop perpendicularly to the inner surface, occasionally slightly inclined, are the result of a fatigue phenomenon. As the thermodynamic calculations which were carried out were unable to totally explain the internal cracking phenomenon, loop and on-site tests were carried out with reactor coolant pumps equipped with a thermal barrier containing instrumentation, in order to accurately determine the existing thermal fields in the barrier and to refine the hypotheses used in the calculations.

The origin of thermal barrier flange cracking.
Current knowledge of existing thermal loads placed on the thermal barrier flange do not explain.the origin or the location of the cracks found. The thermal barriers of the pumps which were the subject of loop and on-site tests were equipped with additional instrumentation, in order to establish the thermal phenomena existing in this area more accurately and to improve the understanding of the root cause of cracks.

7 - MEASURES TAKEN

Surveillance and detection measures.
Among the different parameters envisaged for the early detection of a thermal barrier housing crack (variation of injection water flow, vibration and temperature), only variation in the temperature of the Component Coolant System water at the outlet from the coil was considered to be sufficiently representative and reliable. The principle behind monitoring this temperature consists of comparing two-by-two, the value of the Component Coolant System temperature at the outlet of each reactor coolant pump (in order to compensate for variations in the inlet temperature of the Component Coolant System water) and to generate an alarm whenever the temperature of one pump is abnormally high in relation to the other two. Surveillance measures for the early detection of a thermal barrier flange crack are currently being developed.

Action to limit consequences.
A temporary operational procedure has been set up until a final solution to the problems can be found. This procedure consists of shutting down the pump should any of the following alarms appear:

- high flow rate at the thermal barrier,
- a non-quantified primary leakage of more than 500l/h,
- high level of activity in the Component Coolant System and with a total primary leakage of more than 1,500l/h,
- very high level of water in the buffer tank of the Component Coolant System and with a non-quantified primary leakage of more than 230l/h,
- high temperature of the Component Coolant System at the thermal barrier outlet.

Development of a new type of thermal barrier housing.
In order to reduce the susceptibility of the housings to cracking, "new generation" housings have been developed (see Figure 5). The improvements in their design are:

- suppression of the 2 mm blend radius at the bottom of the thermal screen,
- adoption of a 35 mm radius between the bottom and the cylindrical part of the housing, instead of 38 mm to avoid the non-typical zone,
- manufacturing the housing by forging (Z2 CN 19-10 steel [UNS S30403 SS]) instead of casting, and
- extending the part protected by the thermal screen to the bottom of the housing and in the 35 mm radius.

These thermal barrier housings will be the subject of loop tests following the tests

currently being carried out on the original housings, in order to validate the thermal load hypotheses which have been used. Since the beginning of 1995, the utility has been preparing for the in-shop assembly of the "new generation" housings, onto the replacement hydraulic equipment which will be installed on-site from 1996 onwards.

Maintenance programme and strategy.
During refueling outages, the thermal barriers of the reactor coolant pumps of the 900 MWe PWRs were tested by ultrasonic examination using an automatic tool, to establish the state of the components. The criteria for rejection were established depending on the depths and lengths of the cracks. At the end of 1995, practically all the thermal barrier housings of the reactor coolant pumps in the 900 MWe PWR series were tested by ultrasonic examination. All the pumps tested were affected by the cracking phenomenon and twelve of them have been replaced. The housings left in service will be replaced by the "new generation" housings according to a schedule which is yet to be determined, giving priority to the worst affected in order to avoid any risk of failure. The "new generation" housings which replace the rejected components will be checked after six operational cycles. For the thermal barrier flanges, the size of the critical fault liable to lead to the failure of the flange between two tests has been determined, and a method of ultrasonic examination which enables this type of fault to be detected has been developed. The tool currently used for ultrasonic examination of the thermal barrier housings has been supplemented to enable the flanges to be tested by ultrasonic testing when the housings are being tested again. Dye penetrant examination will be carried out on the thermal barrier flanges whose housings are removed. This programme means that the state of the thermal barrier flanges will be checked three years from now. The lift-type check valves currently installed upstream of the Component Coolant System coil, will be tested during refueling outages. However, the technological design of the check valves is not adapted to the Component Coolant System coolant carried, so they will be systematically replaced with swing-type check valves during refueling outages.

8 - INFORMATION GAINED

The discovery of internal cracking of a generic nature in the thermal barrier housings, is due to two factors:

- a sampling check carried out on a reactor coolant pump during the ten-yearly inspection at Fessenheim 2,
- an external cracking of the thermal barrier housing of this pump, due to the properties of the material (used in only four pumps of the population of plants).

The discovery of cracking in the thermal barrier flanges, also of a generic nature, followed the examination of the underside of the flange (inaccessible without removing the housing) on removal of a housing. Finally, it is after the tests carried out following the safety examination (in consideration of the potential consequences) that the jamming phenomenon in the check valves upstream of the coil, again of a generic nature, was discovered. In summary, a sampling check associated with the particular nature of the material of some thermal barrier housings is the reason that several generic phenomena, which presented major safety-related risks, were discovered. In addition, the faults discovered were not envisaged at the design

stage, so no system was developed to detect them and the consequences of the resulting damage had not been taken into account. The anomalies found, the way they were discovered and the potential safety consequences they could have, demonstrate the risk which can be associated to damage in areas which are not periodically examined, and for which there are no preceding indications which would reveal deterioration phenomena.

Figure 1

THERMAL BARRIER OF THE PRIMARY COOLANT PUMP IN 900 MWe PWRs

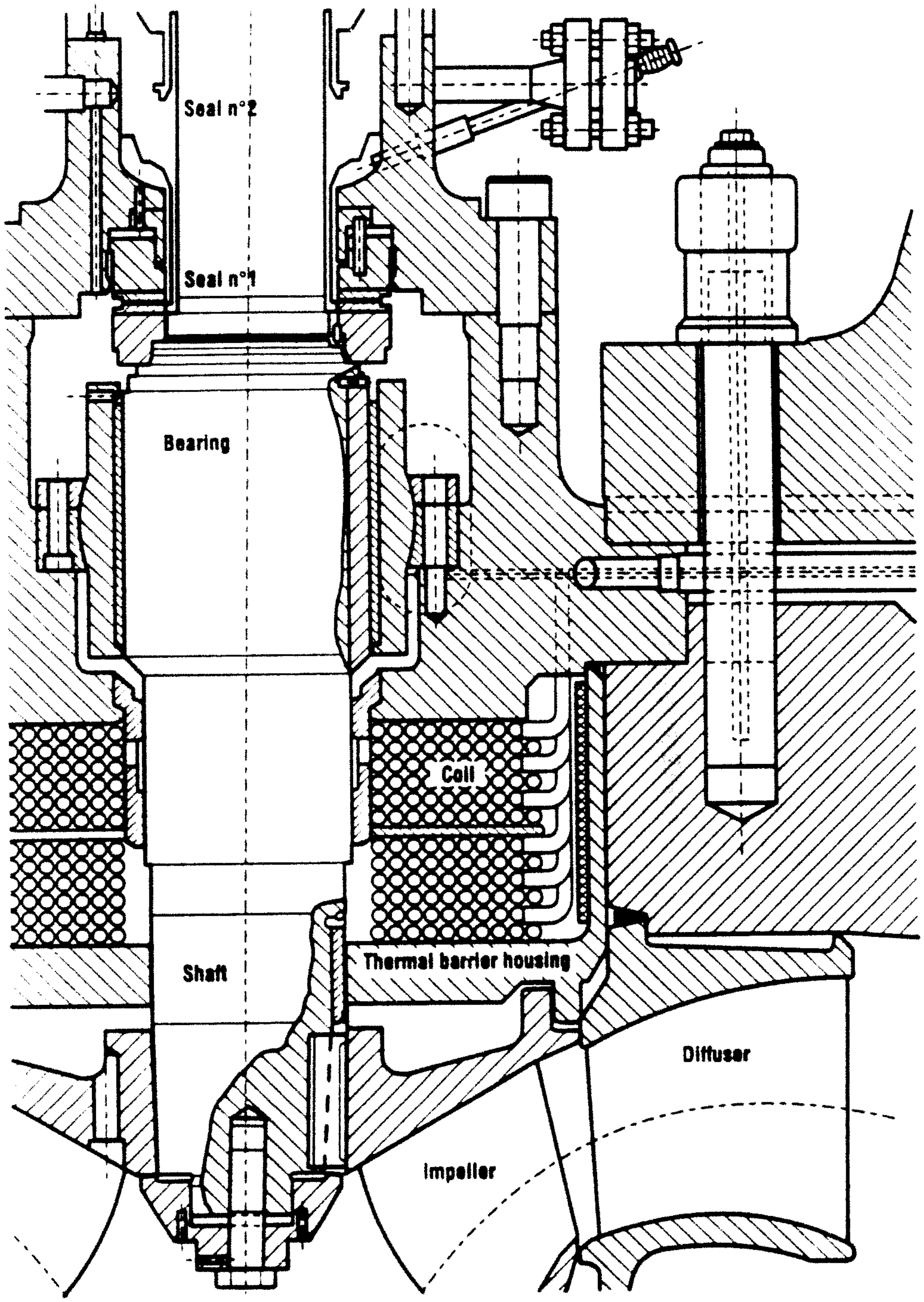


Figure 2

DIAGRAM OF COMPONENT COOLANT CIRCUIT AT THERMAL BARRIER

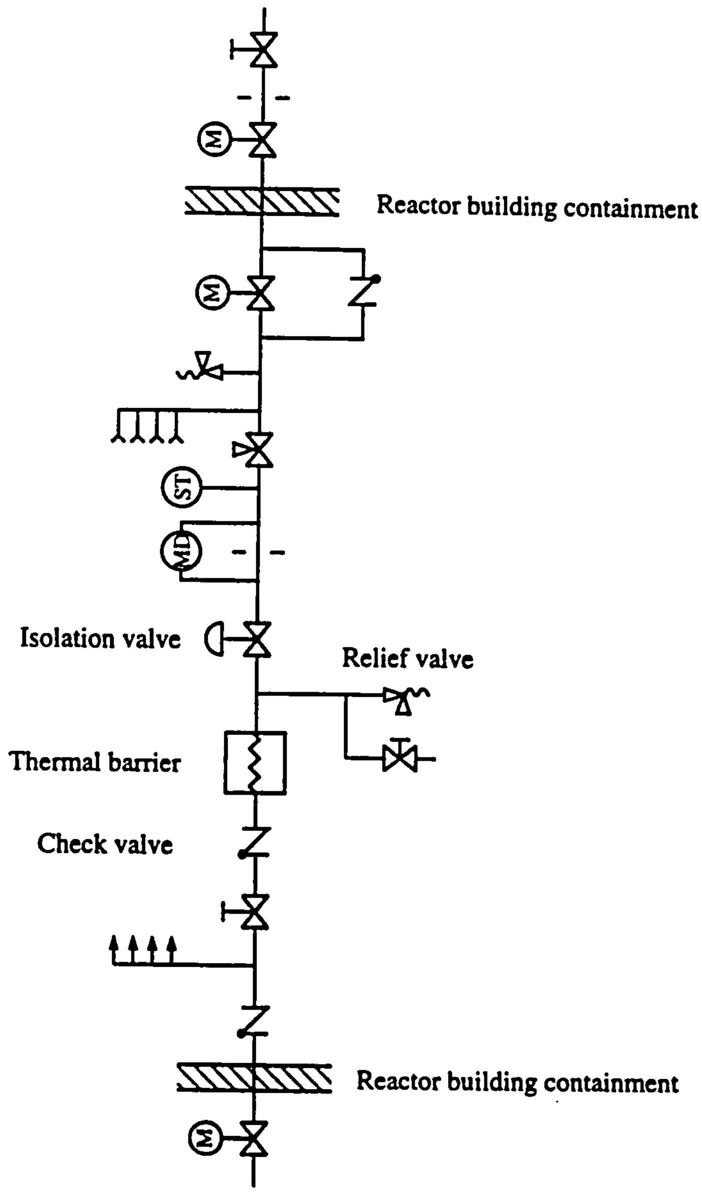
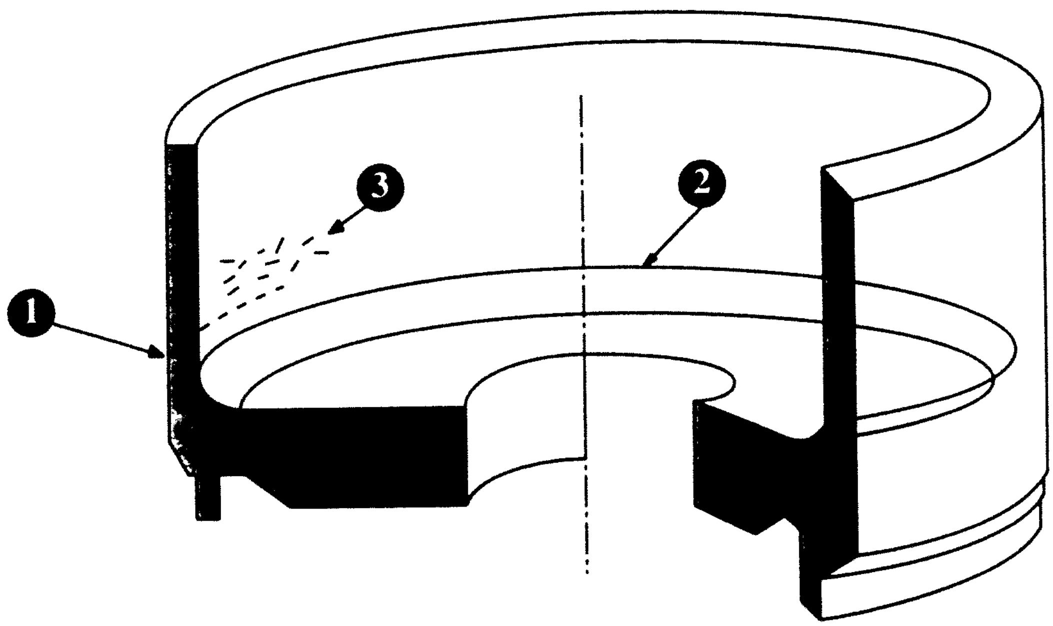


Figure 3

THERMAL BARRIER HOUSING CRACKS



❶ External crack ("Fessenheim" type)

❷ Internal circumferential crack at blend

❸ Internal thermal crazing cracks

Figure 4

THERMAL BARRIER FLANGE CRACKS

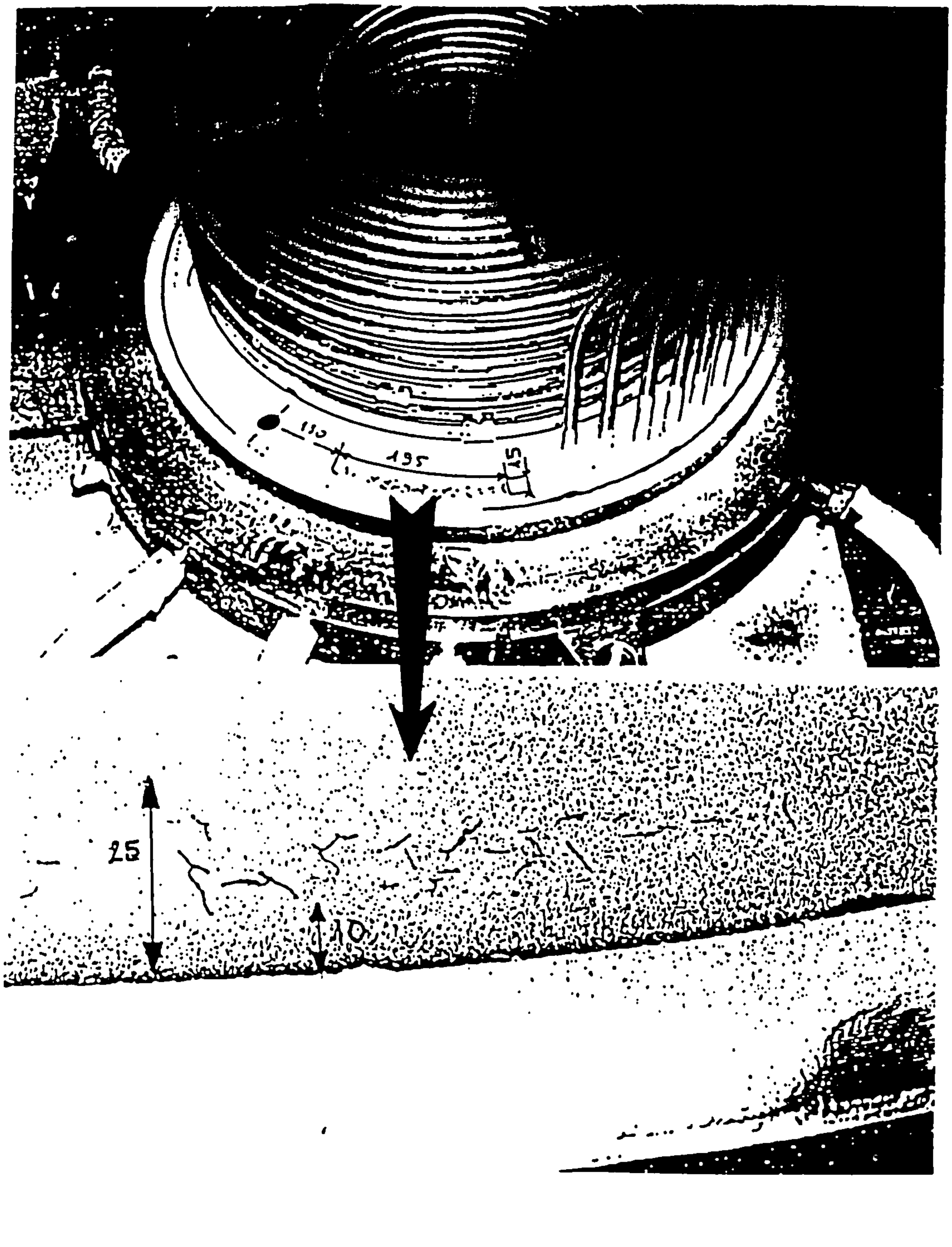
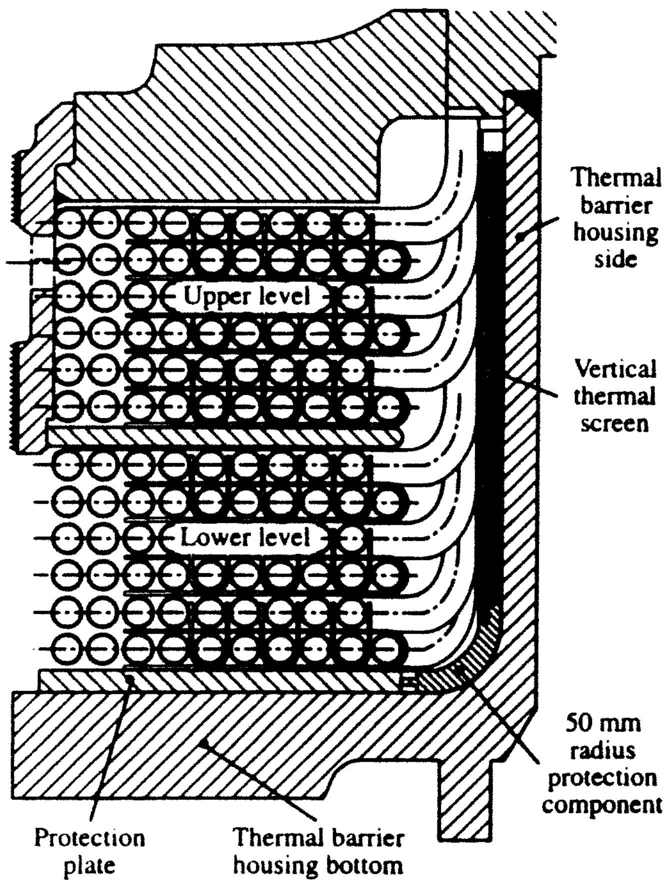


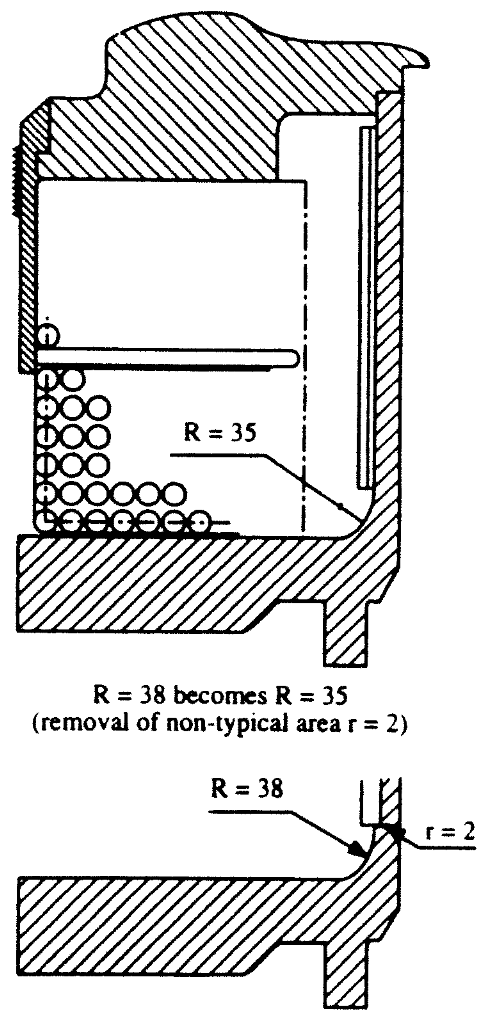
Figure 5

MODIFICATIONS TO THERMAL BARRIER HOUSING

2nd MODIFICATION



1st MODIFICATION



Session 1B

General Valve Issues 1

Session Chair
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Torsional Fatigue Model for Limitorque Type SMB/SB/SBD Actuators for Motor-Operated Valves

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ABSTRACT

Kalsi Engineering, Inc. has recently developed a computer program to predict the torsional fatigue life of Limitorque Type SMB/SB/SBD actuators for motor-operated valves under given loading levels, including those that exceed the ratings. The development effort was an outgrowth of the 'Thrust Rating Increase' test program. The fatigue model computes all pertinent stress components and their variations as a function of the loading ramp. The cumulative damage and fatigue life due to stress cycling is computed by use of a modification of Miner's rule. Model predictions were validated against actual cyclic loading test results.

INTRODUCTION

Recent industry experience during in-situ diagnostic testing of motor-operated valves (MOVs) while addressing NRC Generic Letter 89-10 issues revealed that some MOVs are frequently subjected to thrust and torque magnitudes exceeding the normal ratings provided by the manufacturers. A majority of the gate and globe valves in the U.S. nuclear power plants use Limitorque actuators of SMB, SB, and SBD models. Industry experience also revealed that Limitorque actuators do have the capability to provide satisfactory operation and fatigue life when subjected to loading levels higher than the ratings previously provided by the manufacturers to the nuclear power industry. However, no accurate quantification of overload magnitudes and corresponding number of cycles based upon a technically sound approach was available.

Under the co-sponsorship of several nuclear power utilities, a comprehensive testing and analysis program was undertaken to address this limitation. Phase I of this program resulted in a significant increase in the allowable thrust ratings of the actuators [1*] (a proprietary report, not publicly available). Under Phase II of the program, one of the key developments was an analytical model (based upon first principles) for predicting life of the torsional components. Torsional fatigue life prediction is complex and is dependent upon valve stiffness, motor speed, gear ratio, worm gear efficiency (lubrication), maximum load levels, load profile, component geometry, and materials. The model computes all pertinent stress components and their variations as a function of the loading ramp. The cumulative damage and fatigue life due to stress cycling is computed by use of a modification of Miner's rule. The modification employs the definition of a differential accumulated damage for the linearly increasing stress cycles in the loading ramps. The differential accumulated damage was then integrated over the entire actuator cycle to compute total accumulated damage, and thereby the predicted fatigue life. A computer code called LTAFLA (Limitorque Actuator Fatigue Life Analysis program) was developed [2] (a proprietary report, not publicly available). Model predictions were validated

* Numbers in brackets denote references listed at the end of the paper.

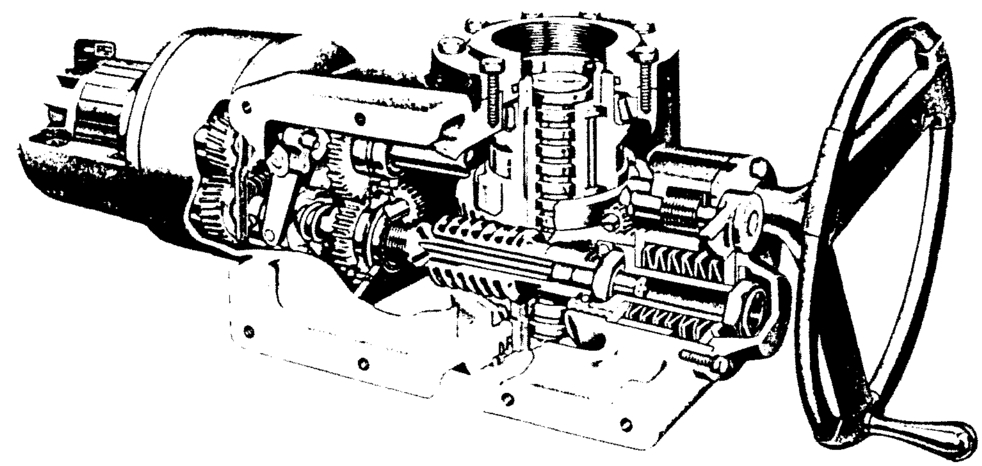

Fig. 1 Cutaway View of Limitorque Actuator

against actual cyclic loading test results obtained from five different actuators sizes under a range of loading levels [3] (a proprietary report, not publicly available).

This paper summarizes the analytical model basis and its validation against test results. The LTAFLA code has been made available to the sponsoring utilities to evaluate the life of the torsional components under their application requirements.

LIMITORQUE ACTUATOR DESCRIPTION

Mechanical Configuration. The Limitorque actuators which can be analyzed by application of the LTAFLA computer code consist of the SMB/SB/SBD class of actuators. The general configuration of the SMB/SB/SBD class of actuators is shown in Fig. 1 in a cutaway view showing the major mechanical components of the system. The vertical translational motion of the actuator valve stem is generated by a worm/worm gear set and a power screw arrangement. The worm in this design is driven by an electric motor through a relatively low ratio helical gear set and a splined worm shaft. The worm in turn drives the horizontally mounted worm gear which is directly coupled to a threaded stem nut in the same horizontal plane. The worm gear/stem nut rotation creates the linear motion of the threaded actuator valve stem and generates a large multiplication of the input torque of the motor, as well as a large speed reduction. The worm is free to slide axially on the worm shaft and thereby transmit its full axial load (which is proportional to stem torque) to a Belleville spring stack which is part of the torque switch. The torque switch interrupts the circuit and stops the electric motor at a preset stem torque, that is, at a preset compression of the spring stack.

Torque Load Characteristics. The most important factors affecting the operating life of these actuators are the load profile of the applied torque, and the gear ratios of the actuator torsional components.

Consider the typical load curve for a gate valve under static (no flow) conditions shown in Fig.2 for one closing and one opening stroke [4]. Note the wedging action displayed by the curve. In the closing stroke, the stem is increasingly compressed until it is fully seated in the wedged position (this is followed by a short dwell period of no actuation). Upon initiation of unseating, the stem compression begins to be relieved until the beginning of valve opening when it is fully relieved. The wedging and unwedging load ramps are linear and have very short durations. These steep ramps require relatively few revolutions from the

worm to perform the actuation resulting in fewer stress cycling which contribute to fatigue damage.

Actuation under dynamic (flow) conditions produces a different load curve on the actuator, see Fig.3. The load ramps now are of longer duration with only a piece-wise linear profile. The stem load undergoes a complete stress reversal from compression in loading to tension in opening. Also, the magnitude of closing torque is much larger than that of the opening torque. In comparison to static conditions, a higher number of worm revolutions is required for actuation. The number of stress cycles on the torsional components is also larger possibly resulting in more fatigue damage. The actual damage depends on load magnitude, load duration, and the required number of worm revolutions.

Thrust Overload Qualification. Thrust overload qualification [1] (a proprietary report, not publicly available) does not provide useful information in itself regarding torque qualification. This is indicated by the number of torque related failures which were experienced during increased thrust testing. Of all the actuator torsional components, it was the worm and worm shaft which demonstrated the greatest probability of failure during thrust overload qualification test program. The three consistent failure points were the worm tooth at the worm/worm gear contact, the worm shaft at the worm/worm shaft contact point, and the root of the limit switch worm. Typical torsional component failures are shown in Fig.4. Metallurgical analyses of the failed components confirmed the initial engineering observation that, based on the obvious cyclic nature of the actuator loads, the mechanism of failure was low or high cycle fatigue.

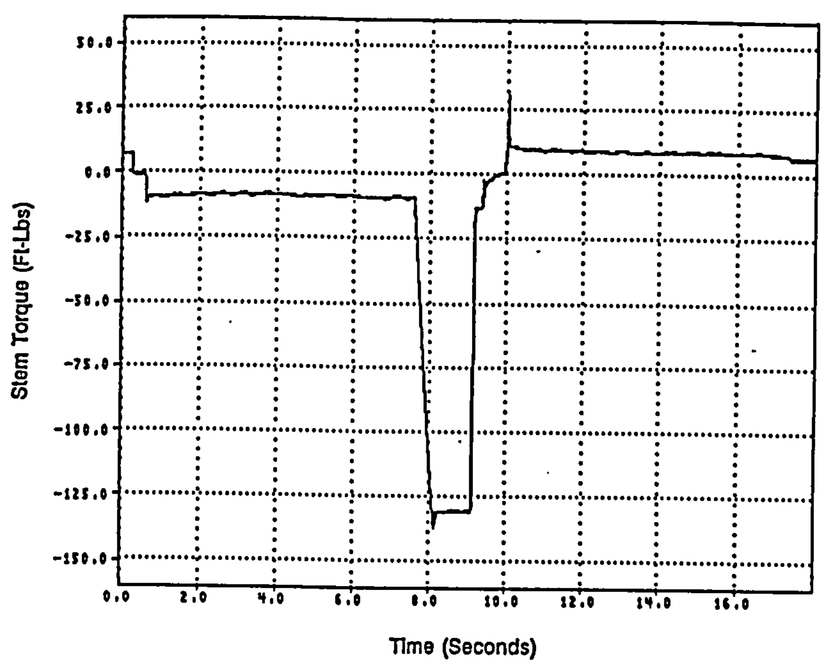

Fig.2 Typical Gate Valve Torque Curve for Static Test [4]

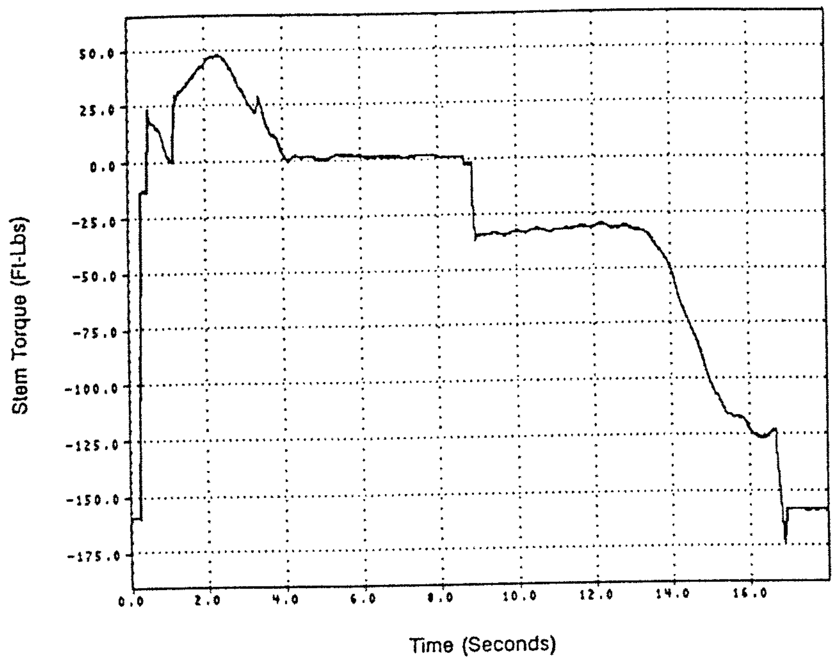

Fig.3 Typical Gate Valve Torque Curve for Dynamic Test [4]

ANALYTICAL MODEL

Mechanical Model. The mechanical model must describe the worm/shaft assembly in sufficient detail to permit the computation of the external as well as internal loads with reasonable accuracy. The type of actuators modeled here utilize two distinct worm shaft configurations. The SMB-000/00 actuators have a canti-

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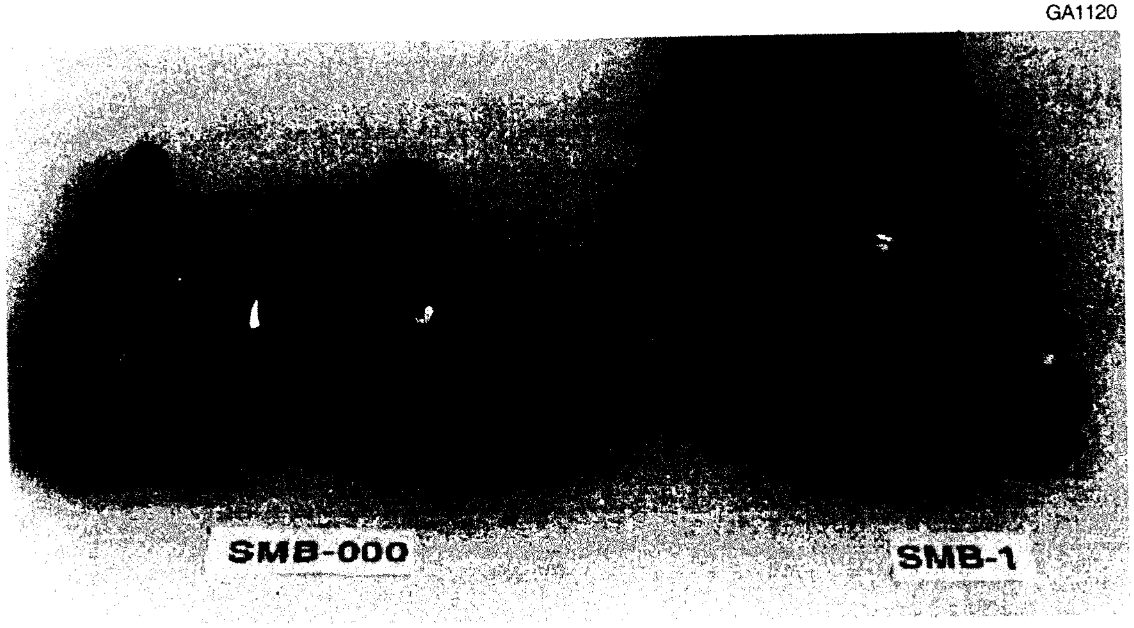

Fig.4 Typical Failures of Limitorque Torsional Components

levered shaft design at the motor pinion/drive gear, while the shafts of the SMB-0/1/2 actuators are supported by an Oilite bushing mounted in the housing. These differences in the configuration negate the use of geometrical similarity throughout the whole range of actuators. Actually, two different mechanical models are required. Two models were developed for describing the critical torsional components. Only the model for the SMB/0/1/2 actuators is shown in Fig.5. The model shows the significant forces and length dimensions for the worm/shaft assembly consisting of the worm of length L_W and shaft of length L_{SH}. The forces acting on this assembly are represented

able). These input parameters are the applied torque T_S, the stem factor, and the geometrical parameters of the valve stem power screw, the worm gear, and the helical pinion gear. Derivation of the expressions utilized the relations given by Shigley and Mitchell [5] for these mechanical elements. A functional relation for these expressions can be written as

$$F_{i,k} = F_{i,k}\left(T_s, d_m, \psi_m, \phi_m, f_m\right), \qquad (1)$$
$$i = x,y,z \quad k = w,p \quad m = s,g,p$$

where 'd' designates diameter, 'ψ' thread lead angle, 'φ' thread pressure angle, and 'f' the

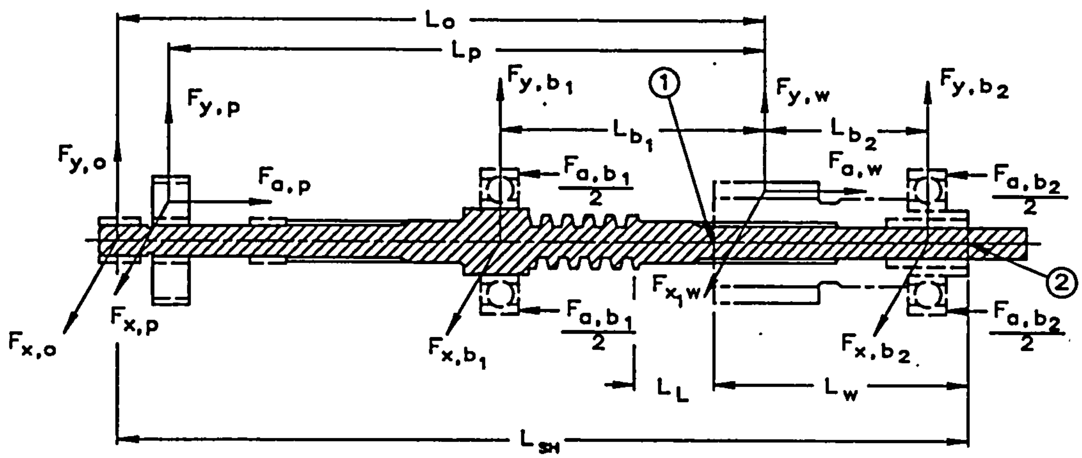

Fig. 5 Mechanical Model and Load Diagram for the Worm/Shaft Assembly

by the solid arrows. The externally applied forces act at the contact points of the worm/worm gear, designated F_W, and motor pinion/drive gear, designated F_P. The coordinate system is oriented as shown with the worm contact point on the y-axis, and the pinion contact point offset by the angle θ_P. The bearing reaction forces are designated F_{b1} and F_{b2} for the worm and shaft, respectively. The reaction force at the Oilite bushing is designated F_O.

Failure Point Loads. The loads external to the mechanical model, F_W and F_P, have been expressed directly in terms of input parameters [2] (a proprietary report, not publicly avail-

friction coefficient of the indicated mechanical element. The subscript 'k=w,p' refers to the contact points of the worm gear, and of the pinion gear, respectively. The subscript 'm=s,g,p' refers to stem power screw, worm gear, and pinion gear, respectively. For the following analysis, these external loads are considered to be known variables in the mechanical model.

The internal forces which have direct influence on the local stress field at the critical failure points are the worm/shaft assembly reaction forces at the Oilite bushing, F_O, at the internal bearing, F_{b1}, and at the external bearing, F_{b2}, and the two contact forces between the

worm and the shaft. These two contact forces are designated F_{C1} for the inner contact and F_{C2} for the outer contact. The procedure used to derive the worm/shaft reaction forces followed the standard practice of static equilibrium by summing all components of system loads, $F_{i,n}$, and of moments, $(L_n F_{i,n})$, and equating the sums to zero as expressed below

$$\sum_n F_{i,n} = 0, \quad n = o, p, b_1, b_2, w \qquad (2)$$
$$\sum_n L_n F_{i,n} = 0,$$

where the subscript 'n=o,p,b_1,b_2,w' refer to system load locations at the Oilite bushing, pinion gear, inner support bearing, outer support bearing, and worm/worm gear contact point, respectively. However, even with the external forces considered to be known, this set of equations is statically indeterminate for actuators which contain the Oilite bushing for additional shaft support. The number of unknowns is more than the number of static equilibrium equations. Additional equations were provided by considering system elasticity. This was done by letting the reaction at a support point (where the displacement is known to be zero) be the indeterminate force. The procedure was to calculate a virtual deflection at this location due to all applied loads assuming there is no support provided here. A virtual deflection was expressed in terms of all determinate forces and shaft elasticity, E_S, and moment of inertia, I_S, as

$$\delta_{i,ind} = \delta_{i,ind}(F_{i,n}, L_n, E_s, I_s), \quad n \neq ind \qquad (3)$$

In order to arrive at a zero net deflection at the indeterminate support point, this virtual deflection was equated to the deflection that would be generated by the indeterminate reaction force alone expresses as

$$\delta_{i,ind} = \delta_{i,ind}(F_{i,ind}, L_{ind}, E_s, I_s) \qquad (4)$$

Eqs. (2) to (4) form a linear system of algebraic equations which were then solved to provide analytical solutions for the reaction forces in terms of the external forces acting on the torsional system. This functional relationship can be expressed as

$$F_{i,q} = F_{i,q}(F_{i,k}, L_k), \qquad (5)$$
$$q = o, b_1, b_2 \quad k = w, p$$

where the subscript 'q=o,b_1,b_2' refers to the reaction forces at the Oilite bushing, internal support bearing, and external support bearing, respectively. Substituting the functional relationship of Eq.(1) into Eq.(5), the reaction forces were expressed in terms of the external forces, and in terms of geometric parameters as

$$F_{i,q} = F_{i,q}(T_s, d_m, \psi_m, \varphi_m, f_m, L_k) \qquad (6)$$

Similar functional relationships were derived for the worm/shaft contact forces

$$F_{i,c} = F_{i,c}(T_s, d_m, \psi_m, \varphi_m, f_m, L_k), \qquad (7)$$
$$c = c_1, c_2$$

where the subscript 'c=c_1, c_2' refers to the two worm/shaft contact points.

Induced Stresses. The induced stress fields at the critical locations are highly complex and three dimensional. Their exact solutions would require the application of numerical methods such as finite difference or finite element techniques. These type of solutions are outside the scope of the present work since they do not lend themselves to the development of a general systematic approach. Therefore, the use of classical formulas available for standard structural shapes were used where possible. Use of these formulas resulted in analytic relations which can be expressed as

$$\sigma_{ij,r} = \sigma_{ij,r}(T_s, d_m, \psi_m, \varphi_m, f_m, L_k, r_r), \qquad (8)$$
$$r = wr, sh, ls$$

where the subscript 'r=wr,sh,ls' represent the critically stressed locations at the root of the worm tooth, the contact point on the shaft, and limit switch worm root, respectively.

The torsional system exhibits definite peculiarities which make the use of classical

analysis difficult. Among these is the fact that, while bending stresses due to lateral loads are derivable from simply supported beam formulas, the presence of a clearance between worm and shaft forms localized contact points where the induced stresses can be very high. These localized stresses can only be described by the Hertz contact stress formulas. Another complicating feature is the stress field produced by the axial component of the worm contact force. This force causes a compression in the worm segment between tooth contact point and torque spring that can be treated as one-dimensional. But in the neighborhood of the contact itself, the stress field is not really one-dimensional. The transfer of torque from the splined shaft to the worm creates a similarly difficult to describe stress field. In the shaft, this stress field transitions from constant plain torsion in the free segment to distributed torsion in the splined segment. In the worm, the transition is from torsion which is distributed along the splines to circumferential shear at the base of the tooth. Despite these complicating peculiarities, the classical method of computing the stresses generated by the individual load components was applied. The pertinent individual stresses were then superimposed to generate a representative stress field. However, knowledge of the details of the stress field complexities is very important for a reasonable estimate of the stress concentration and notch factors.

Worm Tooth Stresses. In addition to load amplitudes, worm tooth stresses depend largely on the way the teeth mesh between worm and worm gear. The gear teeth rotate in the plane of containing a number of tooth contacts, while the worm teeth rotate about an axis perpendicular to this plane. It is the instantaneous cross section of the worm tooth that advances in the plane of gear rotation. Tooth contact begins near the bottom of the worm tooth, remains there until the tooth surfaces become parallel for an instant, and then almost instantaneously switches to the top of the worm tooth. Metal elasticity transforms the point contacts to surface contacts, still, the worst loading on the tooth can be represented by a tip loading. It has been shown that [5], in the general case, elasticity also distributes the worm gear load over three meshing teeth with the central worm tooth taking some fraction 'f ' of the total load, and the other two teeth sharing the remaining load equally with a fraction of '(1-f)/2'.

The worm tooth approximates the shape of a very short and deep beam with a span to depth ratio of unity. Below a ratio of three, the classical assumption of linear stress distribution in cantilever beams is no longer valid. Maximum stress becomes greater than that given by the uniform beam formula. To obtain a better approximation of maximum tooth stress, a formula given by Kelley and Pederson [6] was used in the model. The worm tooth also experiences a compressive and a shear stress. The compressive stress was computed by the use of simple compression formulas. There is uncertainty in the method of computing the shear stress at the base of the tooth. In beam theory, the shear stress distribution is parabolic across the beam with the maximum occurring at the neutral axis and being zero at the surfaces. Short and deep beams, however, approach a configuration more descriptive of a rivet or bolt with the shear stress being constant over the cross-section. To resolve this problem, a finite element analysis was made of a representative tooth geometry [2] (a proprietary report, not publicly available). The results of the analysis indicate that the corner shear stress for the worm tooth (without stress concentration) is best approximated by the constant shear formula multiplied by the factor 1.08.

Worm Body Stresses. The stresses generated in the worm body were computed by the use of simple beam formulas. However, care must be taken in the computation of bending stresses since the maximum bending stress is slightly out of phase with the worm/worm gear contact point median. However, cursory estimates showed that the maximum stress location is almost always within the loaded tooth contact domain. Therefore, maximum bending stress is computed to be later combined with the worm tooth stresses.

Worm Shaft Stresses. The shaft of these actuators has a critically stressed area at the inner worm/worm shaft contact point. Three

different stress components are generated at this point. The contact force compresses the spline, a bending moment compresses the shaft, and a torsional moment tends to shear it. The stresses generated by the compressive contact force can only be described by Hertzian formulas. The formulas outlined by Seely and Smith [7] were used in the analysis of these contact stresses. The three Hertzian normal stresses, σ_X, σ_Y, and σ_Z are compressive and are maximum at the surface with σ_Z having the largest magnitude. All three diminish with distance below the surface. Stresses σ_X and σ_Y diminish at a much faster rate than does σ_Z. The octahedral shear stress τ_G, on the other hand, is minimum at the surface and increases to some maximum value below the surface. With further increase in distance, the shear stress also decreases almost uniformly. Below the point of maximum shear stress, the normal stresses σ_X and σ_Y have almost vanished, and the octahedral shear stress τ_G is related to the compressive stress σ_Z alone. Through this relation the shear stress in the shaft was related directly to σ_Z. The solution for the compressive stress σ_Z was also quite involved. The actual solution given in terms of elliptic integrals was described by an empirical formula derived by curve fitting. The formula was dependent on an empirical factor defined as the ratio of major to minor axes of the contact surface. Worm tooth loading also induces significant bending stresses and shear stresses which were described by standard formulas of strength of materials.

Limit Switch Worm Stresses. The root of the thread of the limit switch worm has also been identified as a critically stressed location. This location is of concern only to the SMB-0/1/2 set of actuators. There are only two components of stress identifiable at this location, a bending stress, and a torsional shear stress. Both of these stresses were described by standard formulas of strength of materials.

Stress Concentration and Fatigue Notch Factors. All torsional failure points are associated with abrupt area changes or some other type of structural discontinuity. These discontinuities increase the local stress levels above the nominal stresses computed by the methods described above. The normal practice was followed to account for the increased stresses by applying stress concentration factors to the computed nominal values. Separate stress concentration factors were required for the worm tooth, the worm body, the worm shaft, and limit switch worm for each type of loading. Quantitative values for the exact configurations were difficult to obtain from the literature, very often approximations were required through the use of factors for similar configurations. The factors used in the present model were taken directly from Peterson [8].

There is an endurance limiting factor due to the size of the component being stressed as compared to the size of the original test specimen on which the fatigue data is based. An empirical equation as given by Graham [9] was used to estimate these size effects and can be written as

$$K_{size} = \left(V/V_o\right)^{0.034} \tag{9}$$

where V is an estimate of the highly stressed volume of the component, and V_O is the comparably stressed volume of the original specimen. It was estimated here that V_O= 0.0001 in^3, and

$$V = 0.014\,\pi^3\, r_f^2\, d$$

where 'r_f' is the radius of the fillet, and 'd' is the diameter of the material in question. The size factors were applied to all components under analysis even though they are generally close to unity. Another endurance limiting factor was applied to all components, a surface finish factor. This again relates the performance of a component being stressed to that of a highly polished specimen. The correction factors for surface finish were taken from Shigley and Mitchell [5].

Notch factors, defined as K_f, modify the fatigue life of notched parts as compared to the fatigue life of smooth parts. They are dependent on stress concentration factors, but are not equal to them. They are also dependent on local stress gradients and material type. An empirical expression relating notch factors to stress

concentration factors and a material constant representing stressing in the immediate neighborhood of the notch is expressed as [9]

$$K_f = 1 + (K_t - 1)\left[1 + (a_n/r_n)\right]^{-1} \quad (10)$$

where K_f is the notch factor, K_t is the theoretical stress concentration factor, a_n is a notch material constant, and r_n is the notch radius. The material constant depends on material strength and ductility and is approximated by

$$a_n = (6,463/S_u)^{1.8} \quad (11)$$

where S_U is the material ultimate strength. The procedure used for accounting for all the stress concentration effects was to compute all the theoretical stress concentration factors, apply the size and surface finish factors, and then use Eq. (11) to compute the notch factors.

Effective Stress for Cumulative Damage. The present fatigue model is based on the 'Distortion Energy Theory' of failure, also known as the 'von Mises-Hencky Theory'. According to this theory, failure is postulated to occur in a material subjected to combined stresses when the energy of distortion reaches the same value that would cause failure in tension only. The governing stress, called the von Mises stress and designated by σ' (written without critical stress location reference), represents that stress that produces distortion but no volume change, and is defined as

$$\sigma' = \left(\frac{1}{2}\right)^{1/2}\left[(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2\right]^{1/2} \quad (12)$$

where σ_1. σ_2, σ_3 are the principal stresses. Determination of the principal stresses for two critical points, the worm/worm gear and inner worm/worm shaft contact points, required finding the three roots of the cubic equation

$$\sigma^3 = -(\sigma_y + \sigma_z)\sigma^2 + (\sigma_y\sigma_z - \tau_{xy}^2 - \tau_{yz}^2 - \tau_{zx}^2)\sigma - (2\tau_{xy}\tau_{yz}\tau_{zx} - \sigma_y\tau_{zx}^2 - \sigma_z\tau_{xy}^2) = 0 \quad (13)$$

where the stress components are the stresses combined from the solutions of Eq.(8). The trigonometric form of solution was used here to obtain the roots of the above equation. Eqs. (12) and (13) were used to compute the principal and von Mises stresses at all three critical points for the static (for static failure) and dynamic (for fatigue failure) cases of the worm/shaft assembly. For the static case the computations were made using the maximum values of the stress components in the stress cycle (that is, in phase with the worm/gear contact point), without stress concentration factors.

The application of these formulas to the complicated stress situation that is encountered in the worm/shaft assembly is not straight forward. Here the principal stresses do not maintain their orientation relative to an element of rotating material. Further complications arise when the stresses have mean as well as alternating components. Shigley and Mitchell [5] suggest a method for handling such complicated stress fields, and their method is included in the present fatigue model. The method defines two stress tensors, one for the mean stresses, and the other for the alternating stresses. The elements of the mean stress tensor $\sigma_{ij,mn}$ and alternating stress tensor $\sigma_{ij,alt}$ are computed according to the usual formulas as follows (again written without the critical stress location reference),

$$\sigma_{ij,mn} = (\sigma_{ij,max} + \sigma_{ij,min})/2,$$
$$\sigma_{ij,alt} = (\sigma_{ij,max} - \sigma_{ij,min})/2 \quad (14)$$

These mean and alternating stress elements are then used in Eq.(13) to compute values of the principal mean stresses $\sigma_{1,mn}$, $\sigma_{2,mn}$, $\sigma_{3,mn}$ and of the principal alternating stresses $\sigma_{1,alt}$, $\sigma_{2,alt}$, $\sigma_{3,alt}$ for all three critical locations and all ramp loadings. Use of these in Eq.(12) then yielded mean and alternating von Mises stresses, σ'_{mn} and σ'_{alt}, for the same locations. This method thus combines any set of bending, normal, and shear stresses, all of which may have mean and alternating components. The resulting von Mises stresses were then applied to the S-N curve in the analysis as would the simple one-dimensional stresses. The method

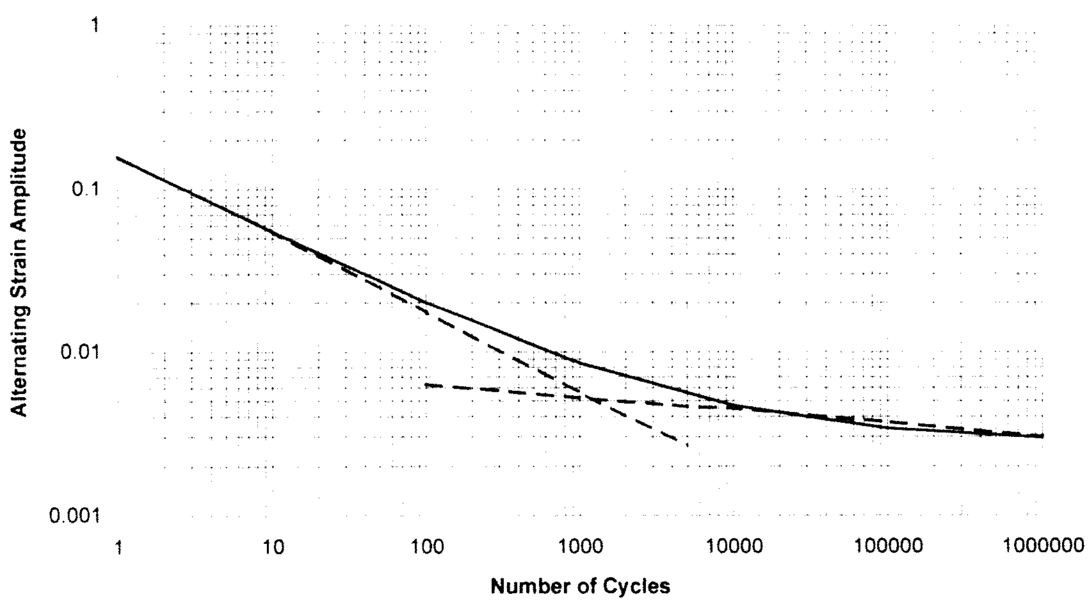

Fig. 6. Typical Material S-N Curve for 4320 Alloy Steel

follows the usual practice of applying stress concentration factors only to the alternating components.

The Material S-N Curve. The results of fatigue tests are generally plotted as stress amplitude Sa versus number of cycles N to fracture using a log-log graph. The resulting curve of data points is called a material S-N curve. For carbon and low-alloy steels, S-N curves typically exhibit a steep and straight slanting portion at low cycles changing into a straight line with a small slope at higher cycles, with some transition between the two. Such an S-N curve usually represents the median life in number of cycles for a given stress. Normally there is considerable scatter in the data measuring fatigue life.

One of the most useful and representative of all S-N curves for ductile materials such as steel is the one adopted by the Society of Automotive Engineers and is presented in the "Fatigue Design Handbook" [9]. This equation gives a good correlation for the required data, but interest here is in a form more suitable to an analytic inversion. A better equation for the present purpose is the empirical correlation given by the criteria of the "Criteria of the ASME Boiler and Pressure Vessel Code" [10] as follows

$$\sigma'_{alt} = \frac{E}{4N^{1/2}} \ln\left(\frac{1}{1-RA}\right) + S_e \qquad (15)$$

where E is the modulus of elasticity, RA is the reduction of area, and S_e is the endurance limit of the material. The above equation departs from the low cycle fatigue test data (high stress region) for the three types of material used in the actuator components. A slight modification of this equation is used here to better represent the data for all three materials used: 4320, 8620, and 41L40. The form of the modified equation is

$$\sigma'_{alt} = 0.25\ E\ C_f N^b + B$$

where now

$$C_f = \ln(1/(1-A)), \quad A = F_A RA, \quad b = -F_b RA, \quad B = F_B S_e \qquad (16)$$

and where F_A, F_b, and F_B are empirical factors to facilitate a more useful correlation with

material data. Eq.(16) is plotted for 4320 steel on a log-log graph in Fig. 6. The straight lines on the graph are the two asymptotes to the low and high cycle regimes. The plot uses F_A=1.107, F_b=1.136, and F_B=1.650. These constants are applicable to all three materials and were used throughout the analysis.

Mean Stress Effects. There are several theories in the technical literature for the accounting of mean stress effects on fatigue life [5]. One of the most commonly used is the linear criteria, also known as the Modified Goodman criteria. This criteria theorizes a linear relationship between the alternating stress amplitude and the mean stress magnitude that will result in a given constant life. Another widely used theory is the Gerber parabolic relation. Kececioglu proposed a correlation which has been verified by many experiments. An expression that covers all the nonlinear theories is written here in the form

$$(\sigma'_{alt}/S_e)^x + (\sigma'_{mn}/S_u)^y = 1 \qquad (17)$$

where the exponents can take on different values to represent the different theories. For example, if x=1 and y=1 the Modified Goodman criteria is represented. If x=2.6 and y=2.0 the Kececioglu correlation is derived. The above mean stress formula was incorporated into the material fatigue equation. The final form of the equation describing the material S-N curve including mean stress effects is

$$\sigma'_{alt} = \left[1 - \left(\frac{\sigma'_{mn}}{S_u}\right)^y\right]^{1/x} \times \left[\, 0.25\, E\, C_f N^b + B\right] \qquad (18)$$

This equation was used exclusively in the model to account for the effects of mean stress on fatigue life by including it in the computation of cumulative damage. However, it must be noted that this equation carries the mean stress correction down into the low cycle fatigue regime where its use is questionable and generates conservative results there.

Cumulative Fatigue Damage. Having established procedures for the calculation of applied loads, the induced stresses, notch factors, and mean stress effects, the concept of cumulative fatigue damage was then developed. According to Miner's rule [5], the parameter defining the damage imposed upon a material after experiencing n_i cycles of stress amplitude S_i, is given by (n_i/N_i), where N_i is the number of allowable cycles corresponding to stress amplitude S_i taken from the material S-N curve. For a repetitively applied total stress cycle consisting of several groups of individually distinct stress amplitudes of $S_1, S_2, S_3, , , S_k$, the accumulated damage imposed on the material is given by the summation $\Sigma n_i/N_i$ with the units of actual cycles/allowable cycles. According to Miner's rule, when $\Sigma n_i/N_i$ => 1 the material will fail by fatigue. This is a linear damage theory, so that after the fatigue damage for all stress amplitudes comprising a total stress cycle have been accounted for, the corresponding total fatigue life for the material experiencing repetitions of that cycle is given by Life=$1/\Sigma n_i/N_i$ in terms of allowable cycles/actual cycles.

Direct application of Miner's rule to the present problem would require a very extensive counting and summation procedure of each individual stress fluctuation of increasing amplitude within the total actuator loading ramp. However, a modification of this rule has been developed here that is easily applicable to this analysis. The modification employs the definition of a differential quantity 'dn/N' which quantifies the differential accumulated damage for the linearly increasing stress cycles in the loading ramps. The differential accumulated damage can then be integrated over the entire actuator cycle to arrive at the total accumulated damage, and thereby the component's fatigue life.

The actuator worm and shaft fluctuating stresses exhibit the linearly increasing amplitudes within each defined loading ramp as was discussed previously. The maximum stress reached in the loading ramp was defined as $\sigma'_{alt,rmp}$, and the total number of shaft revolutions in the same loading ramp as N_{rmp}. The linear relationship between number of shaft

revolutions and induced stress can be written down directly as

$$n = \left(N_{rmp} / \sigma'_{alt,rmp}\right) \sigma'_{alt}$$

$$\text{or } dn = \left(N_{rmp} / \sigma'_{alt,rmp}\right) d\,\sigma'_{alt} \qquad (19)$$

The value of N was obtained by an inversion of Eq.(18) to yield

$$N = \frac{1}{(0.25\ E\ C_f)^{1/b}} \times \left\{ \frac{\sigma'_{alt}}{\left[1-(\sigma'_{mn}/S_u)^{y}\right]^{1/x}} - B \right\}^{1/b} \qquad (20)$$

In this equation the mean stress σ'_{mn} also varies from cycle to cycle of worm rotation and must be expressed as a function of alternating stress σ'_{alt} before integration can be performed. To define this functional dependence, it was noted that the mean stress within a ramp is a linear function of time similar to the alternating stress variation, or equivalently a linear function of alternating stress. Therefore, the mean stress variation within a ramp was written as

$$\sigma'_{mn} = \left(\sigma'_{mn,rmp}/\sigma'_{alt,rmp}\right) \sigma'_{alt} \qquad (21)$$

Substituting Eq.(21) for σ'_{mn} in Eq.(20), and forming the cumulative damage integral $CDI = \int dn/N$

$$CDI = \frac{\left(N_{rmp}/\sigma'_{alt,rmp}\right)}{(0.25\ E\ C_f)^{-1/b}} \int_{S_e}^{\sigma'_{alt,rmp}} \left\{ \frac{\sigma'_{alt}}{\left[1-\left(\sigma'_{mn,rmp}/\sigma'_{alt,rmp}\ S_u\right)^{y} \sigma'^{\,y}_{alt}\right]^{1/x}} - B \right\}^{-1/b} d\sigma'_{alt} \qquad (22)$$

where the lower limit of integration is the endurance limit S_e. Use of Eq. (22) yields the cumulative damage for one of four (or as many as may be involved) loading ramps in one complete valve cycle. To obtain the total damage for a complete valve cycle, the CDI's for all ramps were summed, and the fatigue life in number valve cycles expressed as

$$N_{vc} = 1/\sum_{j=1}^{4} CDI_j \qquad (23)$$

VALIDATION AGAINST TEST RESULTS

The fatigue life model was validated against experimental data obtained in the cyclic thrust overload qualification test program. Fig. 7 shows a schematic of the test fixture that was specially designed and built by Kalsi Engineering for the test. The fixture simulated valve stiffness in the closing and opening direction by using different stacking combinations of disc springs. The stiffness of the spring stacks used had been chosen to provide approximately 0.1 inches of travel from zero to maximum test load. The approximate values of the stiffness of the spring stacks were 150 kips/in for SMB-000, 300 kips/in for SMB-00, 500 kips/in for SMB-0, 1,000 kips/in for SMB-1, and 1,000 kips/in for SMB-2. Axial load and torque applied to the stem was measured simultaneously by a strain gage load/torque cell. Typical measurements of stem torque profile imposed on the actuators is shown in Fig. 8. The profile has the well-defined characteristics of valve closing wedging and opening in both directions. This profile represents the basic actuator load cycle that was repeated continuously to a total of 4,000 cycles. The load profile was simulated exactly by the fatigue model and the predicted life was computed for each failure point [3] (a proprietary report, not publicly available).

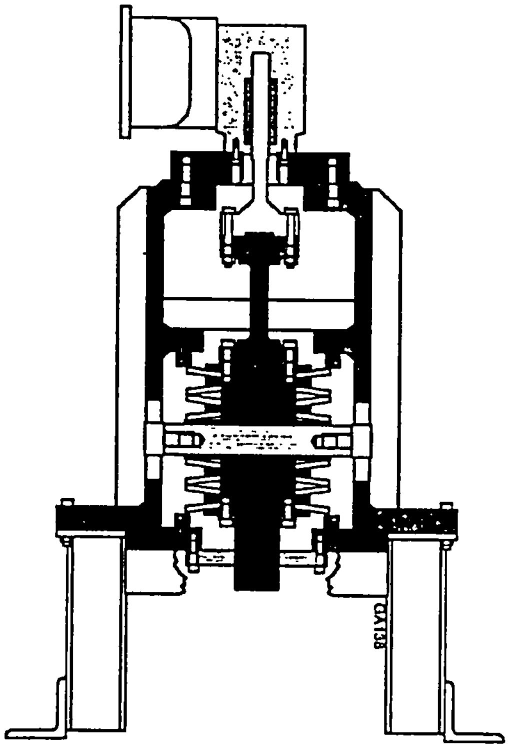

Fig. 7 Limitorque Actuator Test Fixture

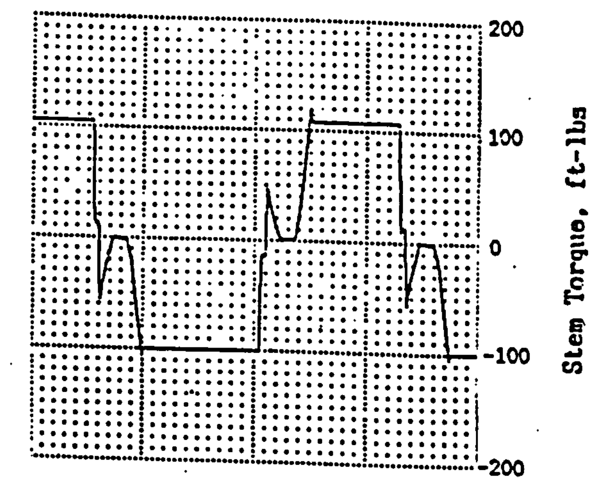

Fig. 8 Typical Stem Measured Torque Profile

A summary of predicted and experimental fatigue lives is given in Table 1 for all of the actuators tested. Tested configurations include five different actuator sizes (SMB-000 through SMB-2). The number of worms and worm shafts used for each actuator varied; for example, for SMB-000 three worms were tested, one of 8620 material and two of 4320 material. There are a total of 14 data points consisting of 8 failure points and 6 suspended test points in which no failure was predicted, and none were encountered for the duration of the test. Tabulated are the number of test valve cycles to failure, the predicted number of valve cycles to failure, and the test/predicted valve cycles to failure ratios. The test/prediction ratios show excellent agreement between test data and predicted life. The ratios range from a low of 0.61 to a high of 1.67, a high to low factor of less than three. The normal factor in fatigue test data is of the order of five or more. Therefore, the standard deviation achieved in this validation is well within acceptable limits. The averaged values of test cycles to failure (2,198) also agrees very well with the averaged predicted cycles (2,233) resulting in an average test/ prediction ratio of 0.98. This agreement between the averaged values signifies that the model is not biased toward over or under-predicting fatigue life.

Fig. 9 is a graphical presentation of the 8 failure data points on a plot of predicted versus actual component fatigue life. If the predictions and actual test data would be in perfect agreement, all of the test data points would fall on the solid line designated 'perfect correlation line'. The scatter in the actual fatigue failure data points for the various tests span a range from 61% to 167% of predicted life. This is considered a very good correlation considering the statistical nature of fatigue.

Table 1. Summary of Predicted and Experimental Fatigue Lives

Component[1]	No. of Cycles Failure in Test	Predicted No. of Cycles to Failure	Ratio Test/Prediction
SMB-000			
Worm (8620)	755	610	1.24
Worm (4320)	2,458	2,039	1.21
Worm (4320)	1,648	2,039	0.81
Worm shaft (4140)	4,870	4,818	1.01
SMB-00			
Worm (4320)	3,774	3,767	1.01
Worm shaft (4140)	none[2]	10,430	*
SMB-0			
Worm (4320)	none	none	*
Worm shaft (4140)	none	7,400	*
SMB-1			
Worm (4320)	none	none	*
Worm shaft (4140)	none (1974)[3]	1,178	> 1.67
Worm shaft (4140)	1,167	1,178	0.99
Worm shaft (4140)	714	1,178	0.61
SMB-2			
Worm (4320)	none	none	*
Worm shaft (4140)	none	none	*
Averages over all tests	2,198	2,233	0.98
Range over all tests	755 - 4,870	610 - 4,818	0.61 - 1.67

Notes:

1. See model description report [2] (a proprietary report, not publicly available) for the failure locations.
2. No failure reached in 4,000 valve cycles; hence no ratio is possible.
3. No failure point; fatigue life greater than 1,974 cycles.

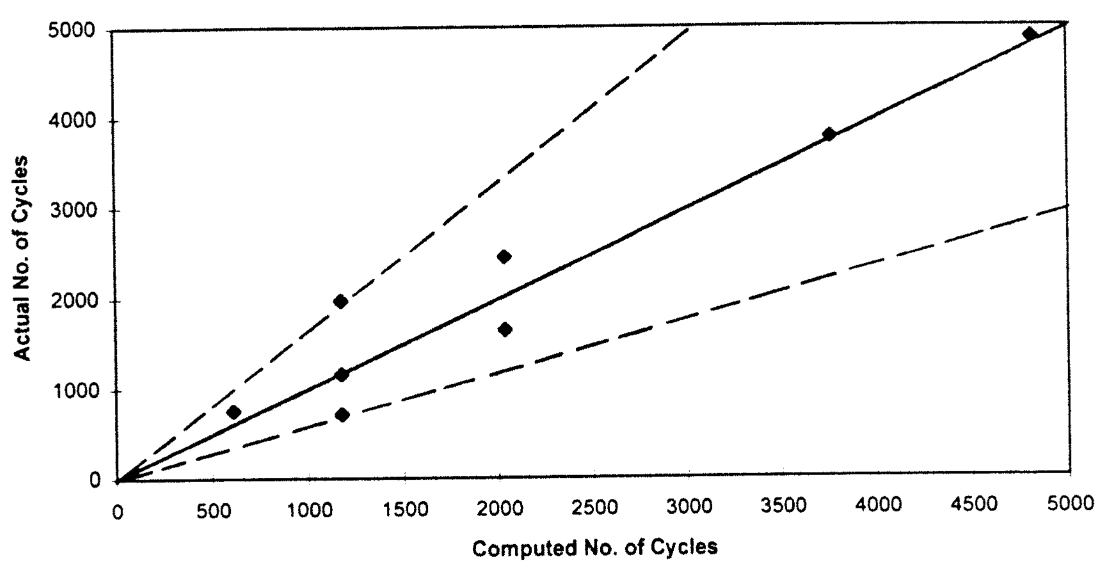

Fig. 9. Correlation of Computed with Actual No. of Cycles to Failure

APPLICATION OF MODEL

In a practical application of this model, a safety margin must be applied to cover statistical uncertainties. Section III of the ASME Boiler and Pressure Vessel Code [11] represents one such approach for determining suitable margins based on the number of replicate tests. Applied margins vary from a factor of 1.64 up to a factor of 5.24 depending on the number of test specimen available. In this program, replicate tests were performed on some of the operator components as discussed in the referenced reports [1, 2, 3]. Even though replicate tests decrease the required safety margin, similar reductions are not applicable for margins required for predictions obtained from the model. Application of the highest margin of 5.24 is always recommended.

CONCLUSIONS

A computer model LTAFLA was developed for the prediction of fatigue life of the torsional components of Limitorque type SMB/SB/SBD actuators for motor-operated valves. The model is based on first principles of engineering analysis. It was validated against test data and was found to be a good predictor model. It is recommended that for use in design, suitable margins be applied to the computed values which are based on ASME Section III, Appendix II [11] approach.

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* References 1, 2, and 3 are part of the industry-sponsored test program.

MOTOR-OPERATOR GEARBOX EFFICIENCY

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ABSTRACT

Researchers at the Idaho National Engineering Laboratory recently conducted tests investigating the operating efficiency of the power train (gearbox) in motor-operators typically used in nuclear power plants to power motor-operated valves. Actual efficiency ratios were determined from in-line measurements of electric motor torque (input to the operator gearbox) and valve stem torque (output from the gearbox) while the operators were subjected to gradually increasing loads until the electric motor stalled. The testing included parametric studies under reduced voltage and elevated temperature conditions. As part of the analysis of the results, we compared efficiency values determined from testing to the values published by the operator manufacturer and typically used by the industry in calculations for estimating motor-operator capabilities. The operators we tested under load ran at efficiencies lower than the running efficiency (typically 50%) published by the operator manufacturer.

INTRODUCTION

The Idaho National Engineering Laboratory (INEL) is conducting confirmatory research, sponsored by the U.S. Nuclear Regulatory Commission (NRC), to develop the technical basis for assessing the capability of motor-operated valves (MOVs) to meet the provisions of Generic Letter 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance." Figure 1 shows the main components of a typical motor-operated gate valve. Several valve manufacturers supply valve assemblies to the utilities, but almost all the operator assemblies are supplied by Limitorque.

There are several factors that can contribute to a gate valve's capability to fulfill its design basis functions, including (a) the magnitude of the differential pressure across the disc, (b) the friction at disc/guide and disc/seat interfaces, (c) the friction at the stem/stem-nut interface, (d) the efficiency of the power train in the operator gearbox, (e) the torque switch setting, (f) the size (torque output) of the operator motor, and (g) the power supply to the operator motor. Over the past ten years

NOTE: This paper was prepared by a contractor of the U.S. Nuclear Regulatory Commission. It may present information that does not represent a current staff position. The U.S. Nuclear Regulatory Commission has neither approved nor disapproved its technical content.

the INEL has performed full-scale and separate effects tests to evaluate each of these factors. This paper presents the results of tests conducted to investigate item (d), the efficiency of the power train in the operator gearbox. The paper is written with gate valves in mind, but the results are also applicable to other types of motor-operated valves.

The main drive train components of a motor-operator gearbox are shown in Figure 2. Motor-operator gearbox performance is measured in terms of efficiency. Gearbox efficiency is the ratio of the output torque divided by the product of the input torque times the overall gear ratio. The overall gear ratio is the total gear reduction in the gearbox--the number of motor revolutions required for one revolution of the stem nut. The input torque consists of the torque delivered by the electric motor to the input side of the gearbox, and the output torque consists of the torque delivered to the stem nut (through the worm gear) by the worm. Thus, the calculation of gearbox efficiency accounts for losses to friction at the helical gear set, the worm/spline interface, the worm/worm-gear interface, and the associated bearings. It does not include the motor electrical to mechanical efficiency or the stem-nut to stem efficiency, which are separate calculations.

In the documentation supporting their motor-operators, Limitorque publishes three efficiency factors, referred to as the pullout efficiency, stall efficiency, and running efficiency. The pullout efficiency is the lowest of the three. Limitorque assumes that this value applies when the motor is lugging at very low speed under a load or starting up against a load. The stall efficiency is higher than the others because it includes consideration of motor inertia during a sudden stall; it is typically used in evaluations of possible overload problems. Limitorque uses the running efficiency to estimate the efficiency of the gearbox at normal motor speed and normal loads.

TEST PROGRAM

As part of a larger test program investigating motor-operator performance (Steele et al. 1995), we tested five typical motor-operator configurations and evaluated the efficiency of their gearboxes. The testing was performed with the operators installed in the motor-operated valve load simulator (MOVLS), a test stand owned by the NRC and designed and built by INEL (see Figure 3). Output torque was measured by a calibrated torque arm attached to the valve stem (the reaction torque measured in the valve stem is the same as the torque applied by the stem nut), and input torque was measured by an in-line torque cell attached between the motor and the gearbox. By these means we were able to calculate operator efficiency from continuous measurements of the actual input torque and the actual output torque.

The testing included parametric studies where the motor terminal voltages were varied from 60 to 100 percent for the ac motors and 50 to 100 percent for the dc motor. Elevated temperature tests were performed by heating the motors and in one case, heating the entire gearbox.

Table 1 lists the motor-operators we tested and the published operating efficiencies (running, pull-out, and stall) for these motor-operators, along with other pertinent information. These efficiency values are from published Limitorque engineering data (Limitorque SEL-7, November 1989; Limitorque Technical Update #92-02,

October 9, 1992). The running efficiencies of the operators vary between 50 and 60%. These published values indicate that it takes about half the input motor power to overcome losses (primarily friction) in the gearbox.

Table 1 also lists the application factor for each motor-operator. The application factor is a multiplier recommended by the operator manufacturer for use in calculations of operator output and can be thought of as the service factor of the electric motor. It takes into account variations in the motor starting torque at varying voltage levels and various operator speeds and conditions. The application factor also makes allowances for any special application considerations. Limitorque Technical Update #93-03, September 1993, states that the application factor can be set to 1.0 if the motor terminal voltage is less than 90 percent (and the voltage squared calculation is used).

Table 1. Test Hardware

	SMB-00-5ac	SMB-0-25ac	SMB-1-60ac	SMB-1-40ac	SMB-1-40dc
Motor RPM	1800	1800	1800	3600	1900
Motor Gear Set	22/43	25/47 37/35	32/40	37/35	32/40
Overall Ratio	87.8	69.56 34.96	42.50	32.13	42.50
Stall Efficiency	0.50	0.50 0.55	0.50	0.60	0.50
Run Efficiency	0.50	0.50 0.55	0.50	0.60	0.50
Pullout Efficiency	0.40	0.40 0.40	0.40	0.45	0.40
Application Factor	0.90	0.90 0.90	0.90	0.90	0.90

Table 1 shows two different sets of values for the SMB-0-25 operator. The second set is listed for a second configuration (a different set of helical gears) tested by the INEL.

RESULTS

Figure 4 shows the stem torque measured during the 100 percent voltage test of the SMB-1 operator with the 60 ft-lb ac motor. The negative convention for this measurement indicates that the valve was being operated in the closing direction. Note how the stem torque gradually increases in a manner representative of a closure under design basis flow. Figure 5 shows the motor torque measured during the same test. Figure 6 shows the operator efficiency calculation made

from the data in Figures 4 and 5; the published running and pullout efficiencies are also shown for comparison. The motor operator efficiency begins at about 0.41 at low load and slowly rises to 0.51 while the operator is still under moderate load. However, the efficiency drops as the load increases, actually dropping below the pullout efficiency at stall.

Figure 7 shows the same information as Figures 4 through 6, but in a slightly different format. Here we have plotted output torque versus input torque during the reduced voltage parametric study. The slope of the data is the gearbox overall ratio times the actual operator efficiency. Figure 7 also shows the results of four calculations using the overall gear ratio times (a) the running efficiency, (b) the running efficiency and application factor, (c) the pullout efficiency, and (d) the pullout efficiency and application factor. For each of the tests, the measured efficiency is near the published running value only at lower loads.

A careful examination of this figure reveals a relationship between efficiency and the speed of the motor operator. In each of the reduced voltage tests the measured efficiency is near the running efficiency when the motor is near its normal speed, but drops toward the pullout value as the motor approaches stall. In the 60 percent voltage test, the efficiency approaches the pullout value at a motor torque of 22 ft-lb, the 70 percent test at 29 ft-lb, and so on. This indicates that the operator efficiency is related to both operator load and operator speed. For this motor operator, the pullout efficiency seems applicable for all tests up to motor rated torque.

Similar comparisons for the other ac motor operator combinations are presented in Figures 8 through 10. For the SMB-00-5, the motor torque required to spin the gear train without producing output torque (sometimes called the hotel load) is a significant percentage of the total motor torque. This distorts the data to a point where meaningful comparisons cannot be made. Thus, Figure 8 includes a 0.44 ft-lb offset, based on no-load motor torque measurements, to account for the hotel load. This hotel load is not normally considered in calculations for determining operator capabilities.

Figures 8 through 10 show that for each ac motor-operator combination, the published running efficiency does not bound operator performance at higher loads and lower speeds. The two smaller operators exhibited efficiencies lower than the pullout efficiency. The SMB-00-5 and SMB-0-25 motor operators' measured efficiencies are closer to the pullout efficiency times the application factor. Note that the application factor is not intended by the operator manufacturer to provide conservatism to the calculation of operator efficiency. We mention the application factor in our discussion only as a point of reference.

Figure 11 presents the data from testing of the SMB-1-40 dc motor operator. The shape of the curves looks slightly different, due to the speed versus torque relationship of dc motors, but the general trends are similar to those seen in the ac motor operators. Actual operator gearbox efficiency is always lower than running efficiency. As the motor speed drops under high load, efficiency drops to values representative of the pullout times application factor calculation. Since dc motor speed is linear with motor torque, the traces shown in Figure 11 for the various low voltages are more distinctly separate than in the ac motor tests, clearly showing the relationship between gear speed and operator gearbox efficiency.

Figure 12 shows the results of testing to determine if operator efficiency is affected by elevated temperature. Three tests were performed on the SMB-0-25 motor operator. The first test was a baseline test to show operator efficiency at ambient temperature. The second and third tests were performed with the operator gearbox heated to 350°F. The third test was performed immediately after the second to evaluate repeatability. The second gear ratio shown in Table 1 was used during this test. For this configuration, the measured efficiency was slightly higher than the published pullout efficiency. By comparing this figure to Figure 9, we get an indication of the variation that can occur between different gear sets used in the same operator. Figure 12 shows that the operator efficiency was not affected by elevated temperature.

CONCLUSIONS

These data show that actual efficiencies can differ from those published by the operator manufacturer. For the operators we tested, the only published value that consistently provided a conservative prediction of the actual efficiency over the entire operating range was the pull-out efficiency times the application factor.

The application factor is not intended by the operator manufacturer to provide conservatism to the calculation of operator efficiency. We mention the application factor in our discussion only as a point of reference. We do not recommend that utilities rely on the application factor to compensate for lack of conservatism in the efficiency value they use in their calculations.

Gearbox efficiency is dependent on operator speed as well as torque. Under reduced voltages the measured efficiency near motor stall drops well below the values measured at full voltage for the same motor torque. Here again, the pullout efficiency times the application factor consistently provided a conservative prediction of the actual efficiency.

We found that the operator no-load motor torque, or hotel load, can be significant for smaller motors. Hotel load consumed almost ten percent of the motor rated torque for our SMB-00-5 motor operator. We also found that the operator gearbox efficiency was not affected by elevated temperatures.

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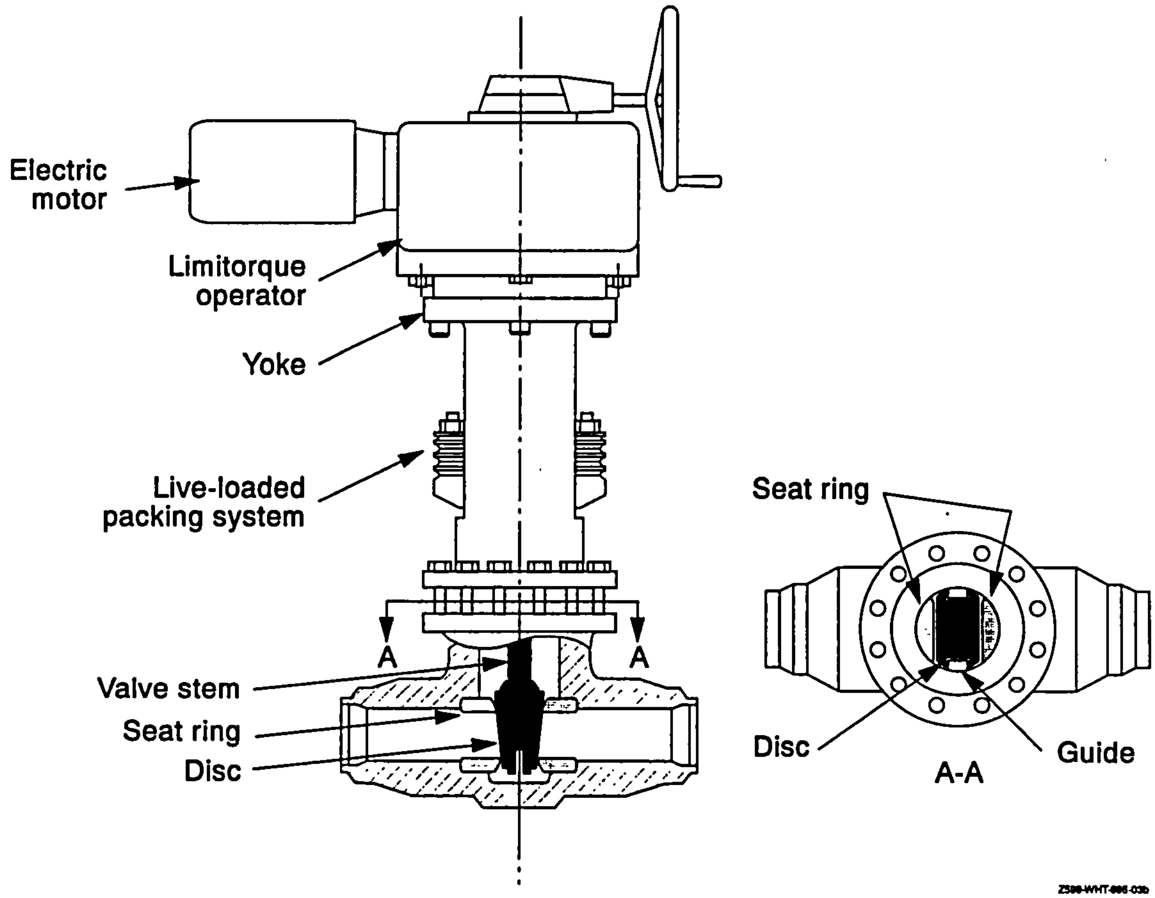
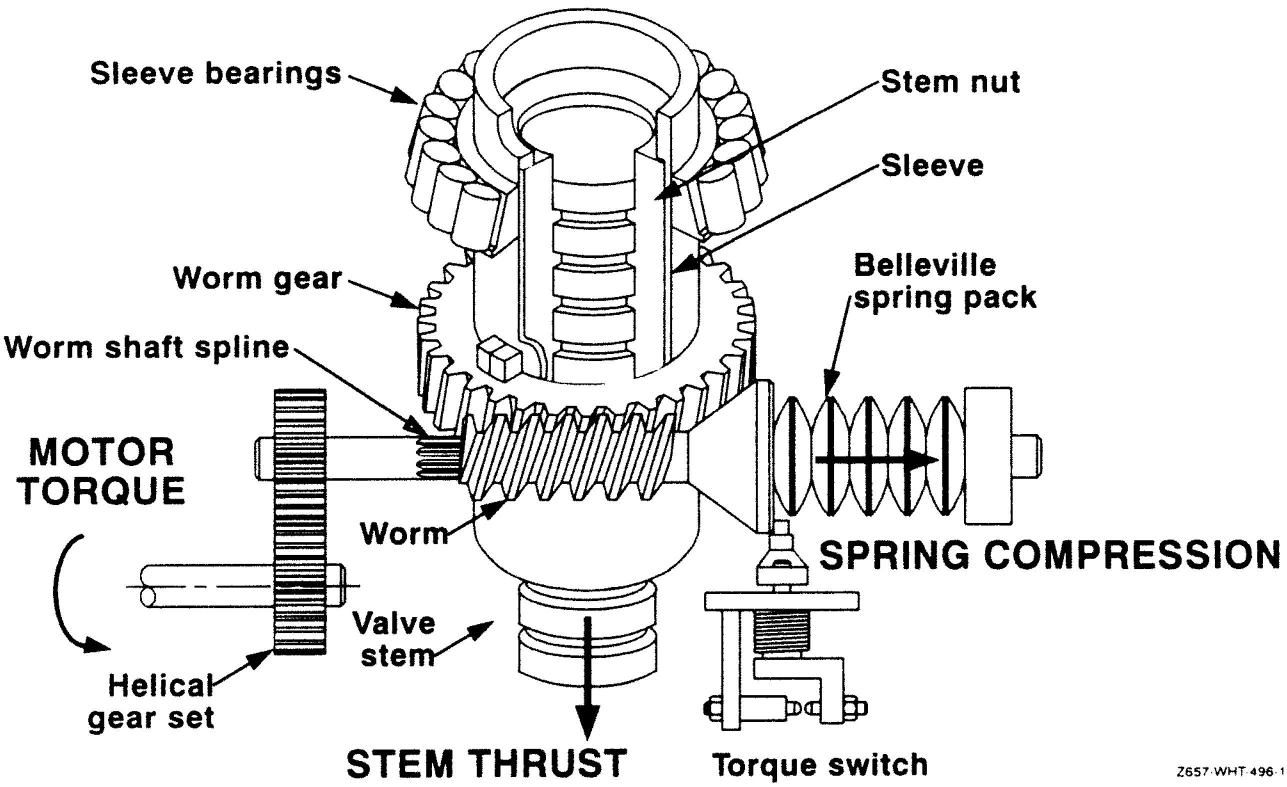
LIST OF FIGURES



MOTOR-OPERATED GATE VALVE.

Figure 1



MOTOR OPERATOR GEARBOX DIAGRAM.

Figure 2

THE MOTOR-OPERATED VALVE LOAD SIMULATOR (MOVLS).

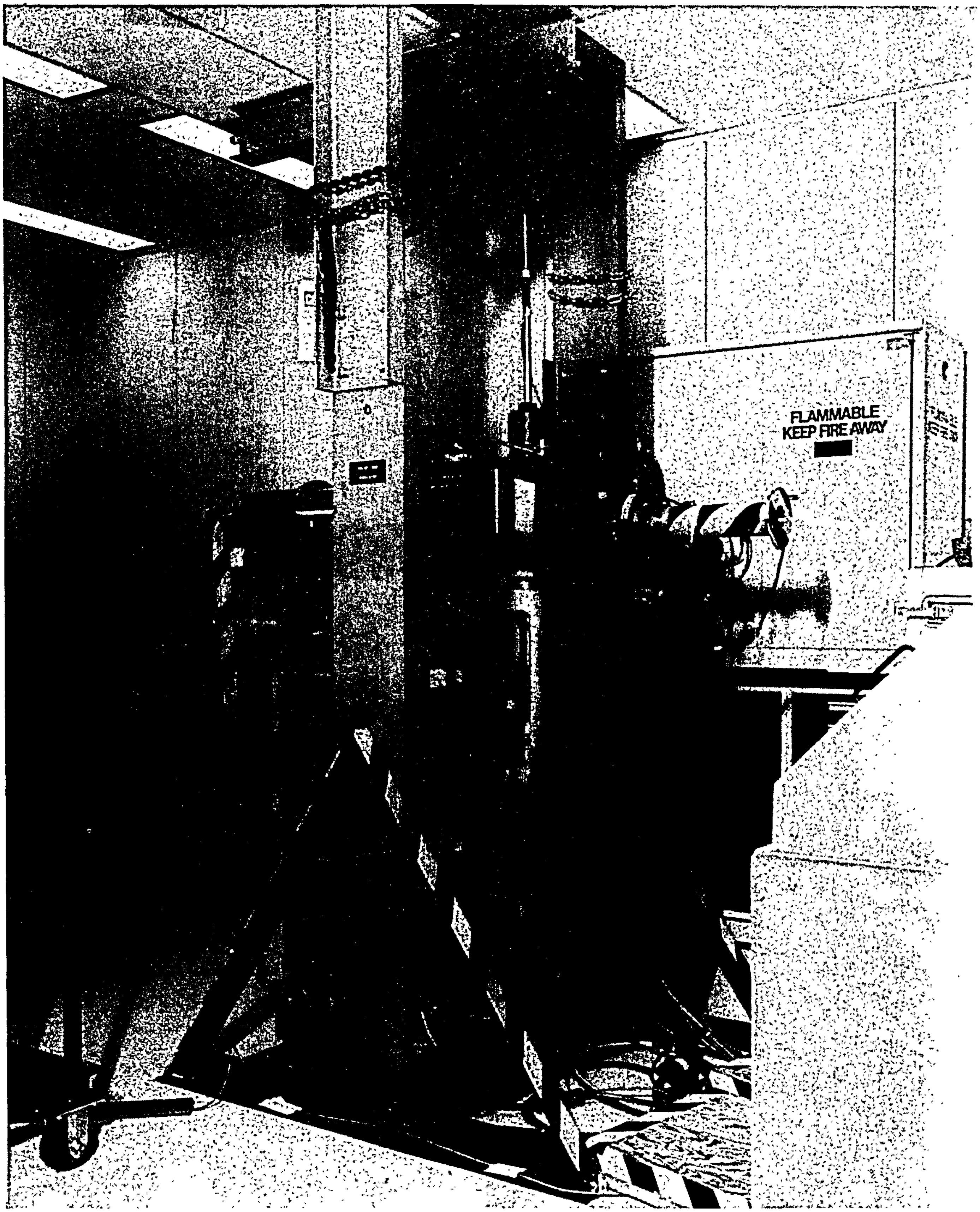
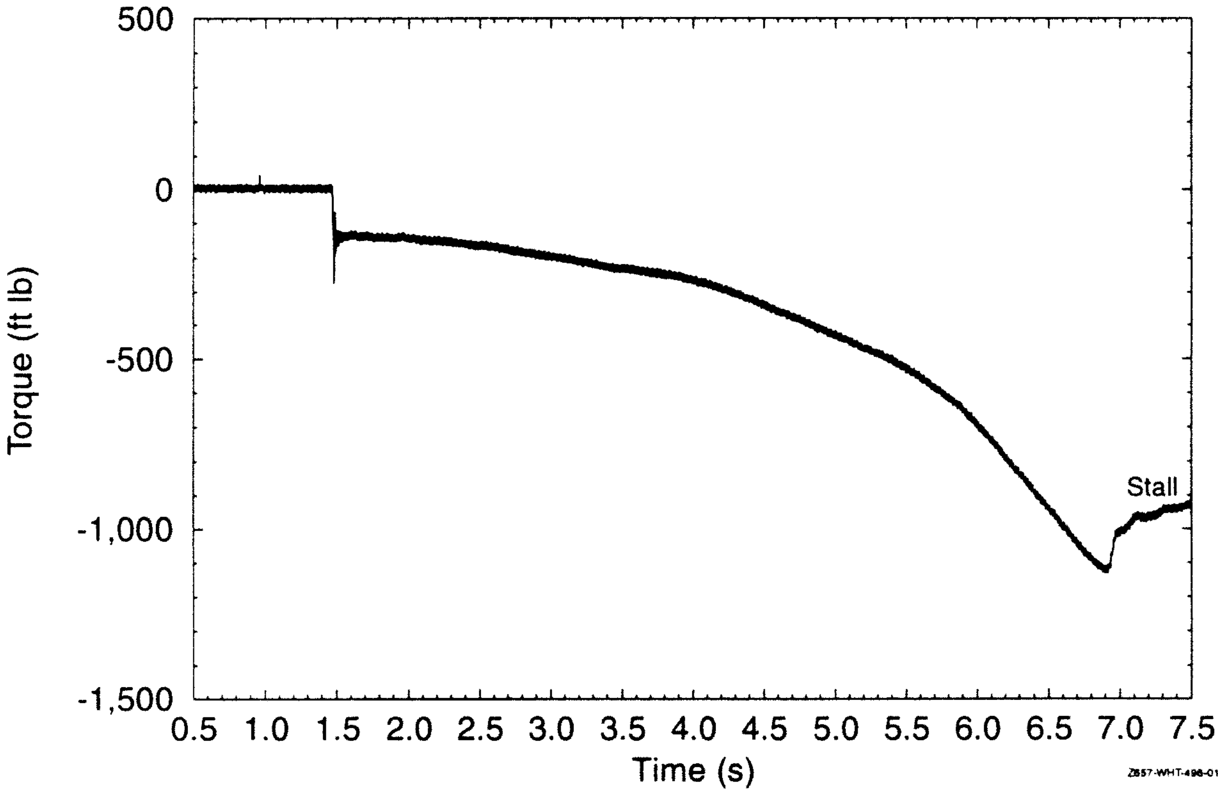
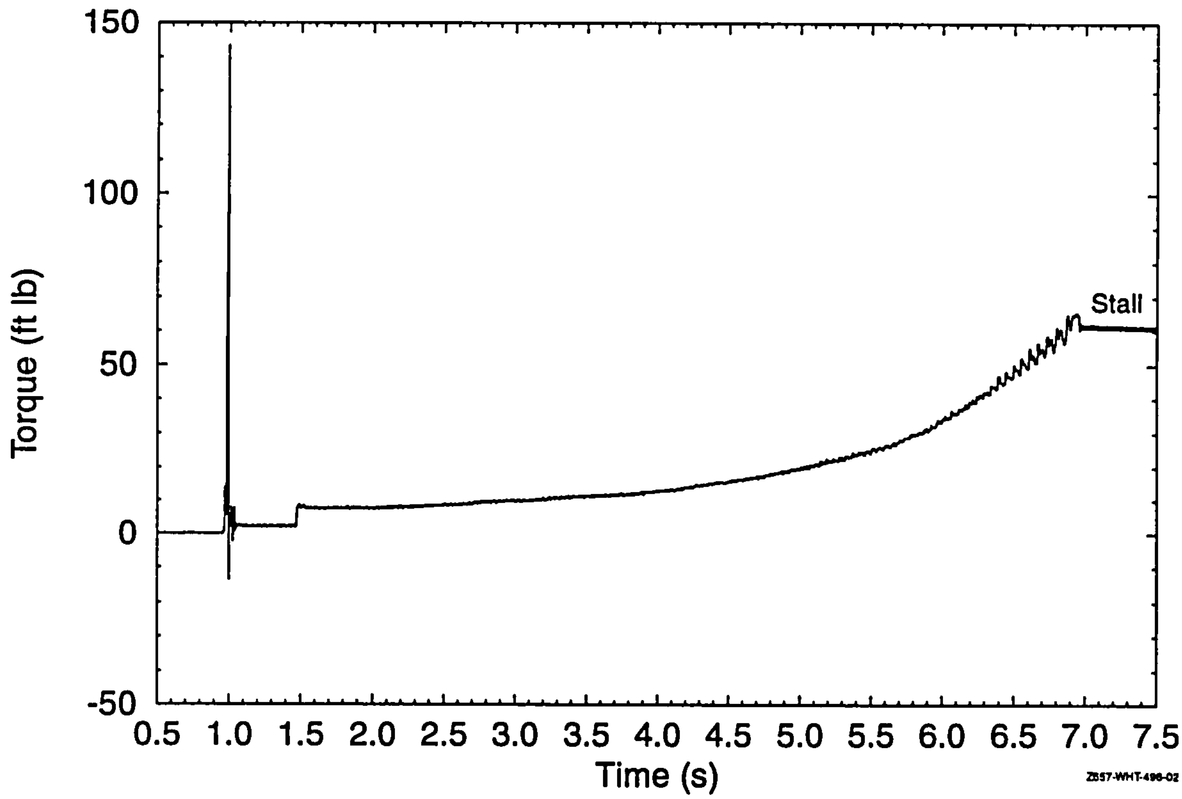


Figure 3



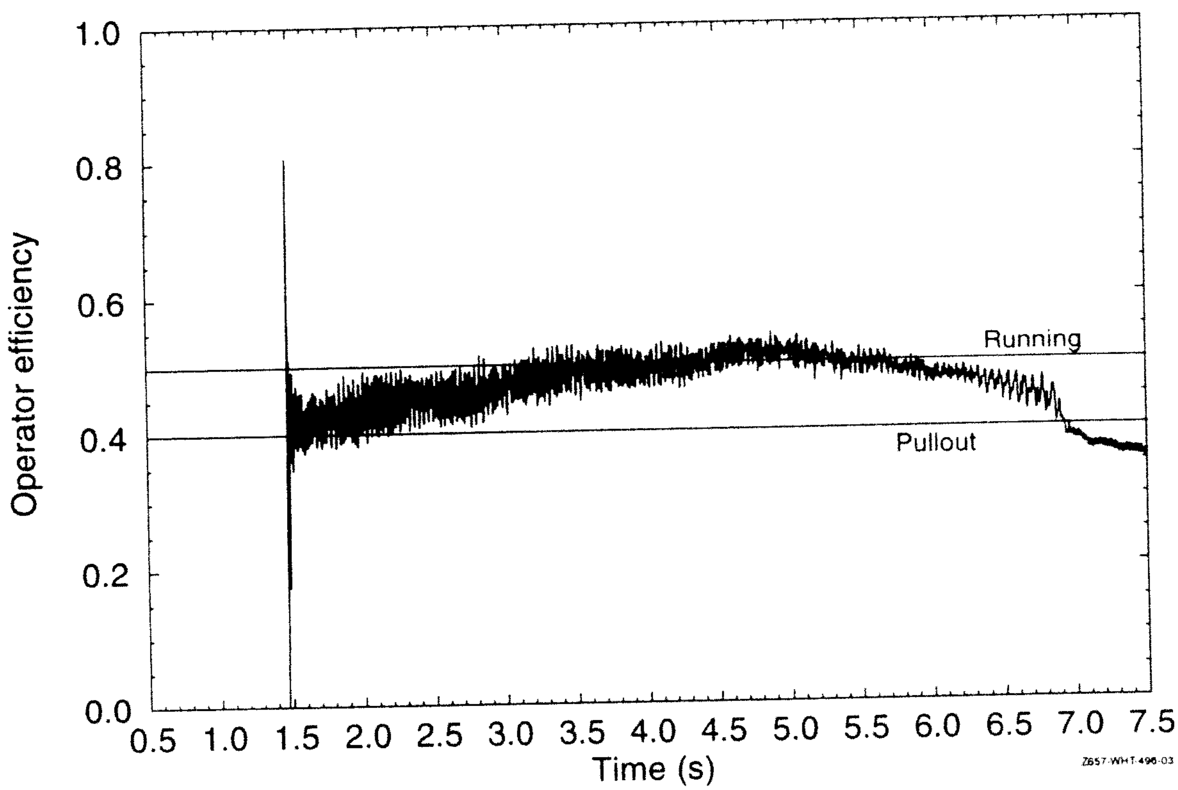
OPERATOR OUTPUT TORQUE MEASURED DURING STALL TESTING OF THE SMB-1-60 AC MOTOR OPERATOR.

Figure 4



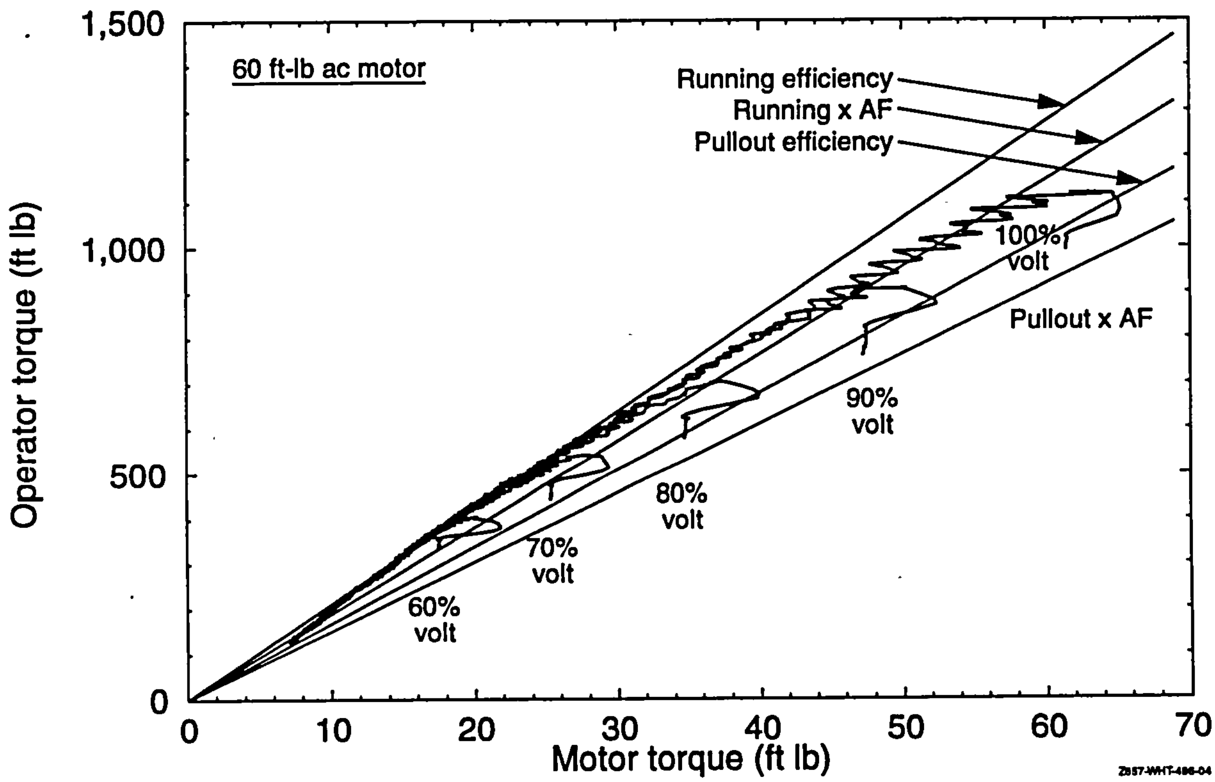
OPERATOR INPUT TORQUE MEASURED DURING STALL TESTING OF THE SMB-1-60 AC MOTOR OPERATOR.

Figure 5



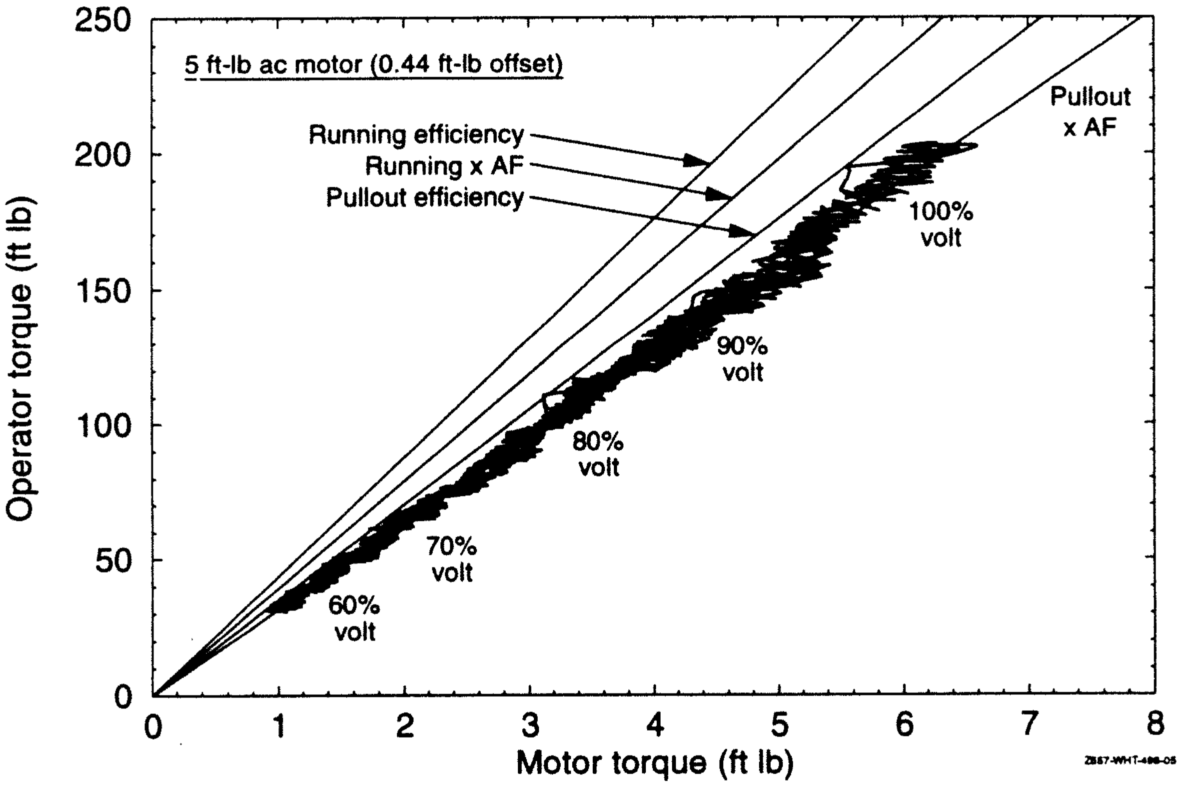
OPERATOR GEARBOX EFFICIENCY MEASURED DURING STALL TESTING OF THE SMB-1-60 AC MOTOR OPERATOR.

Figure 6



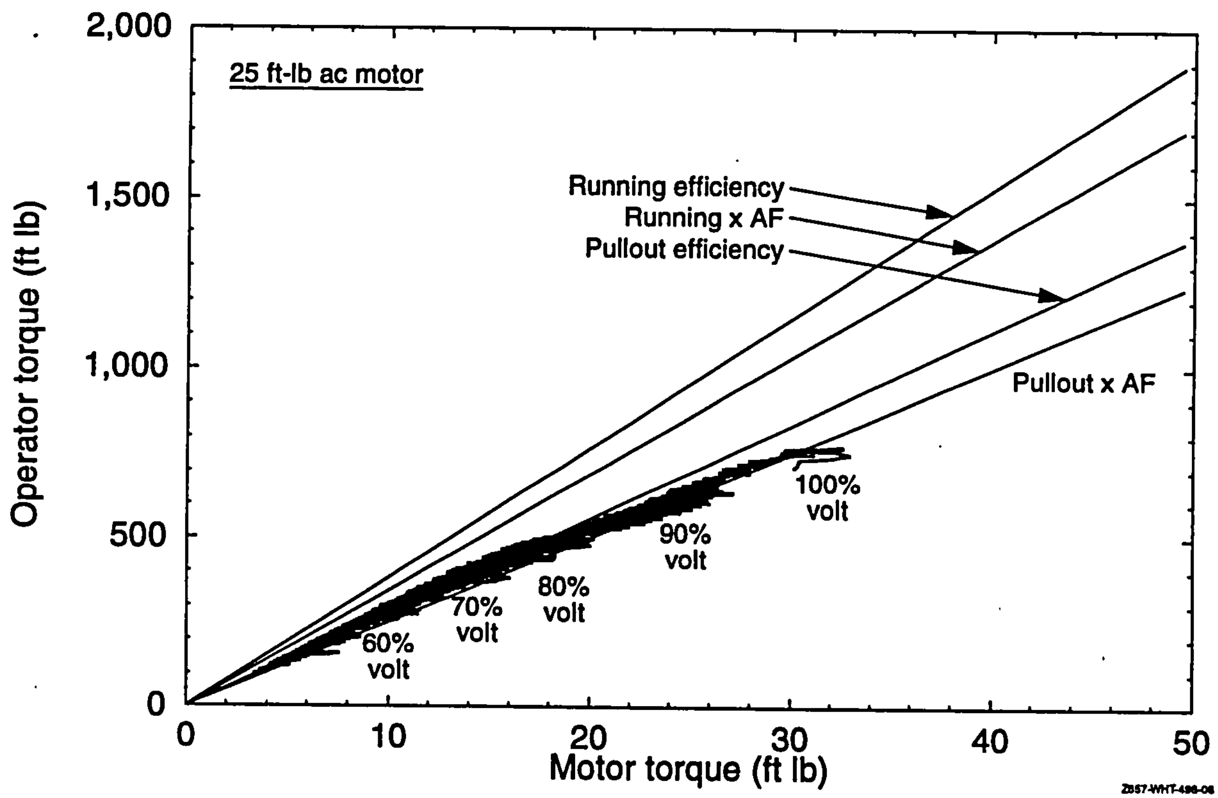
ACTUAL GEARBOX EFFICIENCIES MEASURED DURING STALL TESTING OF THE SMB-1-60 AC MOTOR OPERATOR.

Figure 7



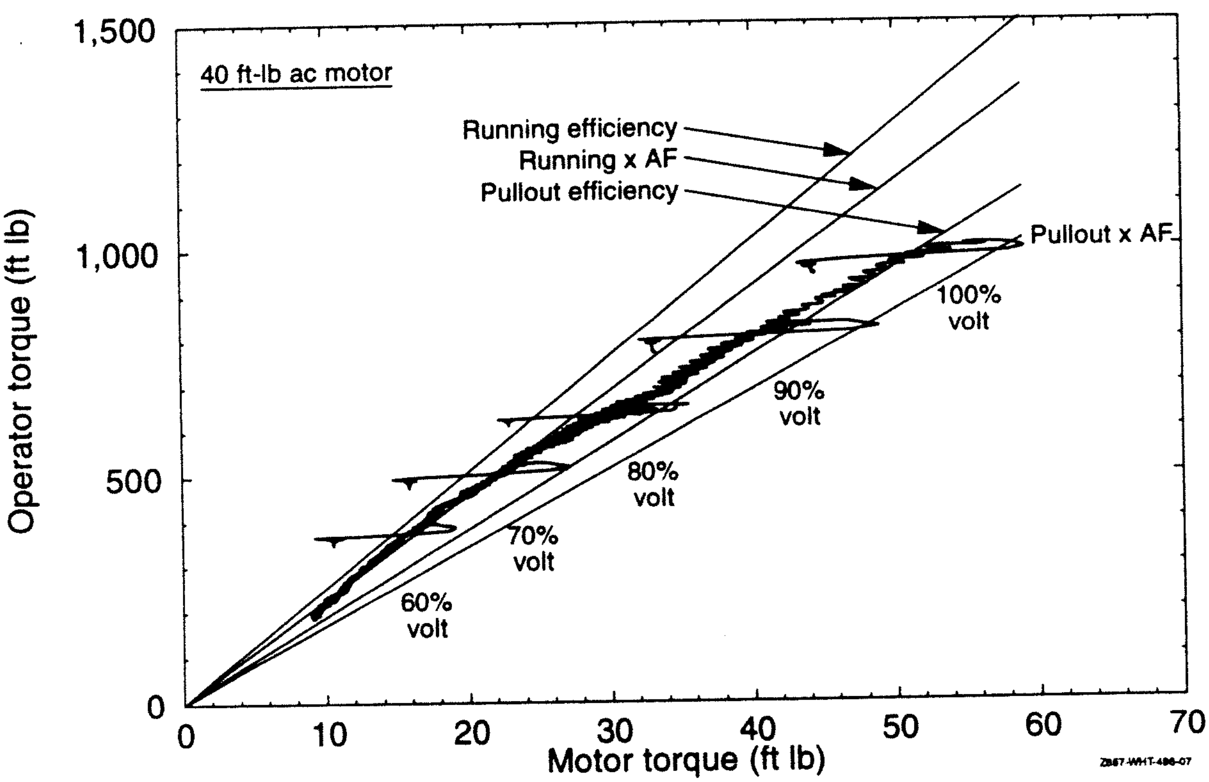
ACTUAL GEARBOX EFFICIENCIES MEASURED DURING STALL TESTING OF THE SMB-00-5 AC MOTOR OPERATOR.

Figure 8



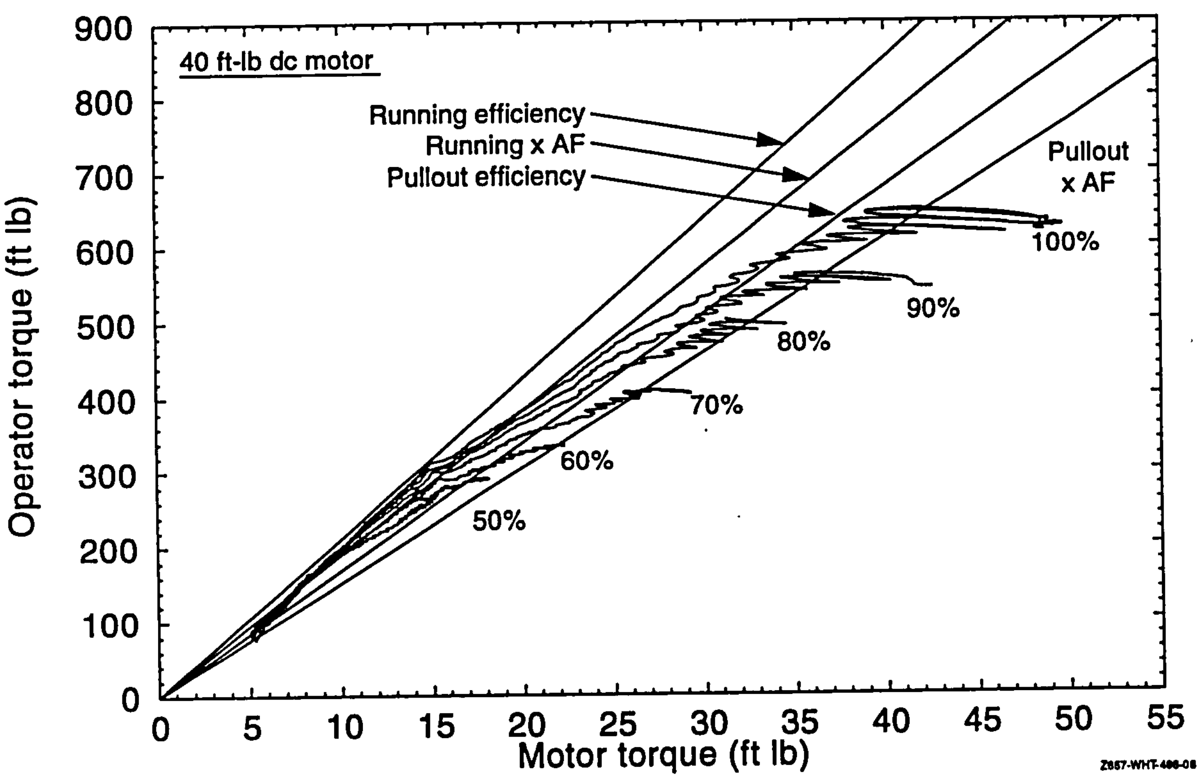
ACTUAL GEARBOX EFFICIENCIES MEASURED DURING STALL TESTING OF THE SMB-0-25 AC MOTOR OPERATOR.

Figure 9



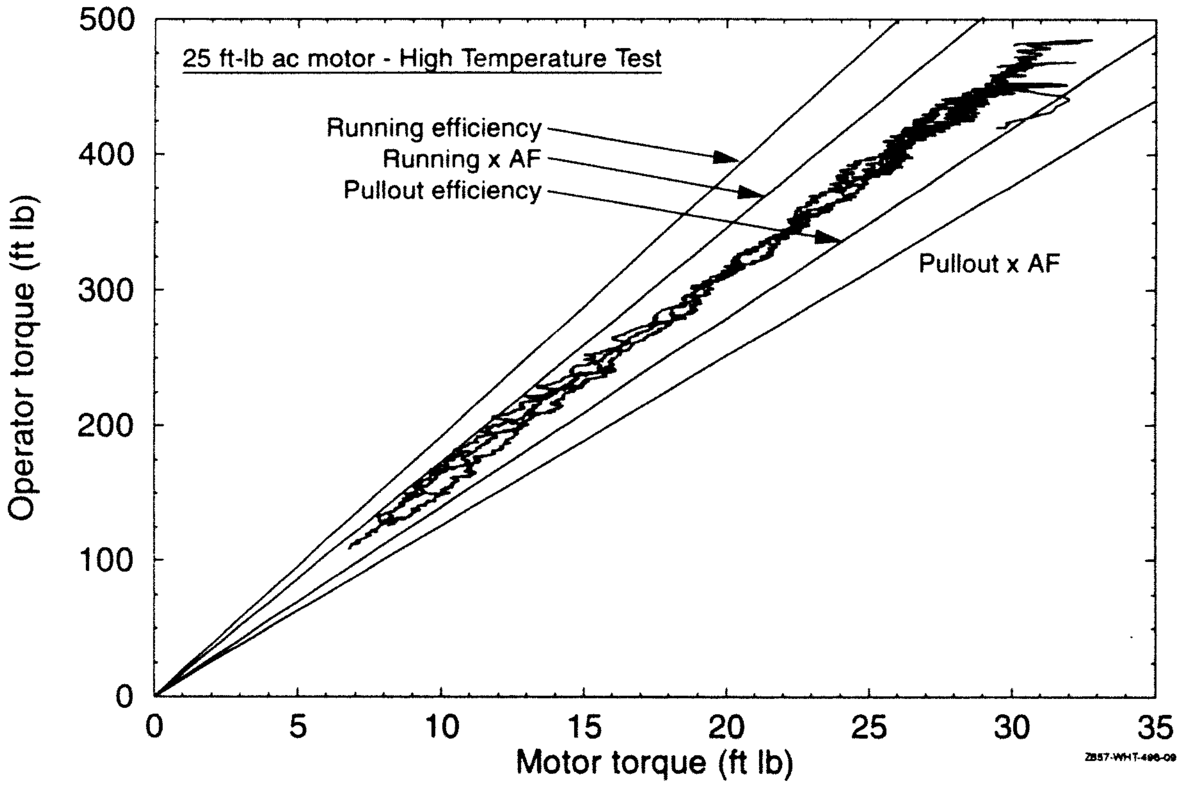
ACTUAL GEARBOX EFFICIENCIES MEASURED DURING STALL TESTING OF THE SMB-1-40 AC HIGH SPEED MOTOR OPERATOR.

Figure 10



ACTUAL GEARBOX EFFICIENCIES MEASURED DURING STALL TESTING OF THE SMB-1-40 DC MOTOR OPERATOR.

Figure 11



ACTUAL GEARBOX EFFICIENCIES MEASURED DURING ELEVATED TEMPERATURE STALL TESTING OF THE SMB-0-25 AC MOTOR OPERATOR.

Figure 12

Evaluation of Existing EPRI and INEL Test Data to Determine the Worm to Worm Gear Coefficient of Friction in Limitorque Actuators

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ABSTRACT

About the last sizing parameter for motor operated valves which has not been determined by utility or NRC sponsored testing is actuator efficiency. A by-product of EPRI testing for valve factors is the measurement of the actuator efficiencies. Motor sizing in this testing provides efficiency testing for motors running near synchronous speed. INEL testing, sponsored by the NRC, for stem factors and rate of loading provides complimentary data for motors loaded down to zero speed. This paper analyzes the data from these two test programs to determine the coefficient of friction for the worm to worm gear interface. This allowed the development of an algorithm for determining the efficiency of actuators which have not been tested. This paper compares the results of this algorithm to the test data to provide a measure of the accuracy of this method for calculating actuator efficiency.

OBJECTIVE

The nuclear power industry has tested just about every conceivable aspect of Motor Operated Valves. Most design parameters have been found or at least bounded. The output of the Limitorque actuators has not been confirmed by industry testing. After completing the testing of Limitorque AC motors, ComEd and the other participating utilities had documentation that all but a few Limitorque motors produce more torque than published curves. This provides justification for using the full breakdown torque shown on the motor curves. After discussing this, the NRC correctly pointed out that there had been no independent verification of the published Limitorque efficiencies. Indeed testing by Texas Utilities' presented at the July, 1994 NRC/ASME Pump and Valve Testing symposium, Reference 1, showed that at eighty percent of the rated voltage, the actuator output was not always bounded by the value calculated using Limitorque's methodology and pullout efficiencies.

ComEd testing showed that these motors were highly saturated and the exponent in the voltage to torque relation was higher than 2.0. This accounted for most of the difference, but there was a few actuators which were not covered by this. There was a need to independently review the efficiencies published by Limitorque.

The next question was how to do this. Testing a large sampling of different gear combinations in Limitorque actuators could prove to be an expensive undertaking.

ACKNOWLEDGEMENTS

Thankfully, Idaho National Engineering Laboratories under the direction of the Dr. Jerry Weidenhammer of the Nuclear Regulatory Commission had performed testing on the thrust and torque output of Limitorque actuators to quantify load sensitive behavior. Bob Steele and Kevin DeWall had the forethought to also measure motor torque input as part of this testing. This data was critical to finding the low speed friction coefficients for worm and worm gears.

EPRI also had the forethought to collect more the minimum information during their valve factor testing. This testing provided information on the coefficient of friction at high worm speeds. Duke power transferred the volumes of EPRI testing onto optical disks. This made transferring the data painless.

This paper would not have been possible without EPRI or INEL/NRC's testing or without Duke and INEL's hospitality in allowing me to use their data.

METHOD

The first step in verifying the actuator efficiencies is to determine what variable should be monitored. The ideal testing would allow the data from a wide variety of actuators to be reduced down to one variable. This would mean that each test would add an additional degree of freedom to our analysis. This would allow us to gain the most statistical significance from the existing tests. Without this, we would need to test several actuators of each gear ratio.

The variable must also allow us to predict the efficiencies of actuators which we did not test. If the variable could not do this, we would need to test each actuator gear combination. Without these two conditions we would have to test several repeats of each gear and actuator combination. This clearly wouldn't be practical.

Figure 1 shows a schematic of an SMB-000 or SMB-00 actuator gear train. It consists of a set of spur gears and their bearings, a worm and worm gear and the drive train bearings. The losses in the spur gears and bearings account only for a few percent efficiency loss in the drive train. These geometries are well documented and the efficiencies are published in the literature. We considered a 4% loss for these components. Any errors resulting from using the published efficiencies for these components would result in negligible errors in the overall efficiency. The worm/ worm gear interface, on the other hand, results in a loss of more than half of the motor power. This interface must be quantified.

The "Analytic Mechanics of Gears", Reference 6, as well as other machine design handbooks, provide relations between the worm geometry and coefficient of friction, and the efficiency. This formula is reproduced below:

$$f = \frac{\cos(\mathit{Normal\ Pressure}\ \angle)\sin(2\cdot\mathit{Lead}\ \angle)}{2}\left(\frac{0.96}{\mathit{efficiency}}-1\right)$$

Where:
f is the friction factor for the worm and worm gear.
cos(Normal Pressure Angle)sin(2 x Lead Angle) will be termed the "efficiency factor"
efficiency is the actuator efficiency

The geometries for both the tested and untested worms are readily available. Therefore, we can reduce efficiency data,

which would vary from one worm geometry to the next, to one variable, the coefficient of friction. Again from literature, the coefficient of friction of worm/worm gears is a function of the lubricant, the fit and finish of the worm and the worm speed. See figure 2. Typically two lubricants, EP-0 and EP-1, are used in the Limitorque actuators. The fit and finish should be the same from actuator to actuator, since they are all produce at the same factory. The worm speed can also be found from the worm geometry, the gear ratio and the recorded data. Therefore, the coefficient of friction meets the first criterium. Since ComEd motor testing used a breakdown torque of not less than 800 rpm, it would also be possible to predict the worm speed and therefore, the coefficient of friction for untested actuator and motor combinations. Therefore, the coefficient of friction meets both criteria.

The data from the all of the tests will be reduced down to the coefficient of friction between the worm and worm gear versus the worm speed. The next question is to see whether the existing test data is sufficient to allow this reduction.

DATA

The test setups of INEL and EPRI were schematically similar, as far as the actuator was is concerned. Figure 3 and Figure 4 present these setups schematically. INEL used a hydraulic cylinder to simulate the loads seen during an actual valve stroke. EPRI's data was taken during actual dynamic valve testing.

Among many other variables, INEL measured the mechanical motor torque directly using an in-line torque cell, the mechanical actuator output torque and the motor rpm. INEL tested four basic actuator setups. These configurations were described in Reference 4 and are shown on Table 1. Each configuration was tested at five voltage levels from 60% to 100% of the rated voltage. One actuator configuration was tested at elevated temperatures. These actuators were loaded to motor stall. The advantages of this testing for determining actuator efficiencies is the direct measurement of input torque, the ability to measure efficiency from full speed down to a stall, and the ability to look at the changes in efficiency, if any as the voltage is decreased.

Table 1
INEL Testing

Actuator Size	Motor Size (ft-lb)	Motor RPM	Worm/Worm Gear Ratio	Motor Pinion Gear Ratio
00	5	1800	45:1	1.951:1
0	25	1800	37:1	1.88:1
1	40	3600	34:1	0.946:1
1	60	1800	34:1	1.25:1

EPRI measured the actuator torque and motor speed directly; however, they did not incorporate an in-line torque cell so there was no direct motor torque measurement. EPRI did measure the motor electrical power and current during the testing. However they also tested the motors on a dynometer so that the motor current and power could be calibrated to the torque.

EPRI tested a total of 33 rising stem valve sat Wyle Laboratories. From these, I selected one valve from each actuator size which was tested at a significant load. This selection was not truly random. I selected the valve in each actuator size which was subjected to the highest loading. Data from EPRI valves 1, 2, 3, 6, 7, 14, 24 and 30 were used in this analysis. At least three strokes taken with significant loading were used. Valves 3, 14, and 30 had the same worm geometry. These were selected to assess the variability between actuators of essentially the same design. References 2 and 3 provide the actuator design for the EPRI valves. Table 2 summarizes the configuration of the actuators. Since the main purpose of the EPRI testing was to determine the valve factor and not the actuator efficiencies, the motors were not loaded to stall. The advantages of the EPRI testing for determining actuator efficiencies are the number of repeated tests and the ability to assess actuators on actual valves under dP conditions.

Table 2
EPRI Testing

Actuator Size	Motor Size (ft-lb)	Motor RPM	Worm/Worm Gear Ratio	Motor Pinion Gear Ratio
000	5	1800	50:1	0.957:1
00	10 *	1800	45:1	1.6:1
0	15	1800	37:1	1.323:1
1	60	3600	34:1	0.945:1
1	60	3600	34:1	0.896
1	60	1800	34:1	0.945:1
2	60	3600	33:1	1.258:1
3	60	1800	41:1	1.61:1

*The data on this motor does not consistently refer to it as a 10 ft-lb motor. The motor curve generated during the testing confirms that this is a 10 ft-lb motor.

As we will see in the data reduction section, the coefficients of friction from the EPRI and INEL data were quite close, despite the measurement method, lubricant viscosity and the purported actuator load differences.

DATA REDUCTION

Before the EPRI actuator data could be evaluated, the motor electrical data had to be converted to torque. After reviewing plots of the motor power and motor current versus actuator torque, it became apparent that the there was a slight lag in the motor power signal when compared to the current and the actuator torque. Using motor power would have resulted in a lowered power readings and higher than actual recorded actuator efficiency at control switch trip. Because of this motor current was used to correlate to motor torque. EPRI typically reported the dynometer test data for six different motor speeds. A typical set of test data along with curve fit is presented in graphical form in Figure 5.

After the current has been correlated to the motor torque, the EPRI data can be reduced in the same manner as the INEL data.

The actuator efficiency is defined by:

$$Actuator\ Torque = Motor\ Torque \cdot Overall\ Ratio \cdot \eta$$

Rearranging this to get efficiency:

$$\eta = \frac{Actuator\ Torque}{(Overall\ Ratio) \cdot (Motor\ Torque)}$$

If we graph Actuator torque versus the product of overall ratio motor torque the slope of the resulting line is the efficiency. A typical curve is shown in Figure 6. Two features of the graph are noticeable. The first is that there appears to be a small amount of motor load which is required to obtain any output torque. This has been termed "hotel" or "parasitic" load. Based on past experiences taking actuators apart, this could be due to shimming and tightening the upper housing cover. At any rate it can be measured when the actuator is assembled. The second feature is that indeed the curve is a fair approximation of a straight line. This means that it is possible to measure efficiency with these test set ups. Graphs like figure 6 were prepared for all of the closing strokes.

The next task is to calculate the coefficient of friction from this data. The efficiency factor defined above is dependent on the worm geometry. I took corresponding worms from stores and measured their geometry. I also used the input files provided with the Kalsi LTAFLA program (Reference 5) to confirm these measurements. The germane measurements and efficiency factors for the various actuator combinations is shown in Table 3.

Since Machinery's Handbook (Reference 7), the Analytical Mechanics of Gears (Reference 6) and other design handbooks show that the friction factor is a function of worm speed, we need to calculated the worm speed during each of the tests to complete the correlation.
The worm speed is:

$$worm\ speed = rpm \cdot pinion\ ratio \cdot \frac{2\pi r}{12}$$

Where:
rpm is the motor speed
r is the worm pitch radius

The data was analyzed with DADisp which facilitated series calculations of coefficient of friction and worm speed. It also allowed making an XY plot from the resulting series. Figure 7 shows a typical curve of measured coefficient of friction versus worm speed. These curves typically showed very low coefficient of friction when the motor runs near synchronous speed. As the motor slows down the coefficient of friction rises rapidly to

a semi plateau. For later reference, this point will be dubbed the "loaded point". When the motor stalls a slight rise in the coefficient of friction is seen. Because the EPRI testing did not take the motors to stall, only the coefficient of friction and worm speed at motor trip is reported.

This data reduction was completed for 12 actuator and motor combinations. There was a total of actuator 52 strokes evaluated. The repeat tests on each actuator were averaged together to reduce variance due to test measurements. These results are graphed in Figure 8. It is interesting to note that the friction factor increases rapidly at low speed when plotted against the "loaded point" of various actuators, but that it doesn't increase that much from the "loaded point" to stall for a given stroke.

There appears to be two sets of phenomena at work. The first order phenomena which shapes the overall curve as seen in Figures 2 and 9 seems to the development of a better lubrication film between the gears when they run at higher speeds. This is offset at very high speeds by an increase in the viscous loads. The second order phenomena which prevents the low rpm part of the Figure 7 from rising rapidly appears to be the transient retention of a good film as the motor rapidly slows down.

Table 3
Worm Gear Geometries

Act. Size	w/wg Ratio	Motor Ratio	Pitch	Lead	Worm OD	Worm Root Diam.	Pres. Angle	Eff. Factor	Speed Factor
000	50	0.957	.163	.163	.78	.564	20	.0721	.1838
00	45	1.6	.262	.262	1.16	.808	15	.0813	.1610
00	45	1.951	.262	.262	1.16	.808	15	.0813	.1320
0	37	1.323	.393	.393	1.555	1.211	14.5	.0869	.2737
0	37	1.88	.393	.393	1.555	1.211	14.5	.0869	.1925
1	34	.945	.499	.499	2.196	1.506	14.5	.0825	.5127
1	34	.896	.499	.499	2.196	1.506	14.5	.0825	.5408
1	34	.945	.499	.499	2.196	1.506	14.5	.0825	.5127
1	34	.946	.499	.499	2.196	1.506	14.5	.0825	.5123
1	34	1.25	.499	.499	2.196	1.506	14.5	.0825	.3877
2	33	1.258	.55	.55	2.675	1.805	20	.0730	.4662
3	41	1.61	.7188	.7188	3.06	2.031	20	.0838	.4139

Because Limitorque motors typically have a NEMA D type of curve motor and worm speed decrease rapidly from the breakdown point to stall from the breakdown point to the stall point. During this time, the worm and worm gear interface is in a transient, similar to that seen in the stem during valve seating. It is possible that ,due to the rapid deceleration of the worm, the grease film between the worm and the worm gear does not have time to degrade during this transient. This could account for the smaller change seen in the friction factor as a given actuator is stalled compared to the larger change seen between actuators running at different speeds. It is conservative to assume that the this would be the case. Therefore the "loaded point" was used to curve fit the friction coefficient. Figure 9 shows the data and the resulting curve fit for the coefficient of friction. The curve's general shape was taken from the Buckingham (Reference 6) as:

$$f = \frac{0.2}{\exp(C1 \cdot WormSpeed^{.5})} + C2 \cdot WormSpeed^{.5}$$

This curve fit is for both the EP-1 and the EP-0 data. The two points at .06 and .07, and 1000 and 1200 fpm were excluded. From the EP-1 data it is entirely possible that the coefficient of friction for EP-1 really is this low. However, my goal was to find one curve which would conservatively cover all actuators. Excluding these two EP-1 points raised the curve up to cover the EP-0 data. The result of this will be a net under prediction of the available actuator torque for a given motor and actuator combination.

C1 was found to be 0.065 and C2 was found to be 0.0023.

The standard deviation from the remaining eleven points and curve fit is 5.56% of the predicted value. Since there were two variables in the fit we are left with 9 degrees of freedom. To bound 95% of the data we need a "Student's t" multiplier of 1.83. Therefore, we need to multiply our predicted coefficient of friction by 1.10 to bound 95% of the friction coefficients.

With this curve fit for coefficient of friction, we can predict the efficiency of Limitorque actuators that we have not tested. The coefficient of friction is a function of worm speed which is in turn a function of motor rpm, actuator geometry and motor pinion gear ratio. The actuator geometry and motor pinion ratio are known or at least measurable. The motor rpm is variable throughout the stroke. From Figure 9 we can see that the friction factor is going to be highest at low rpm, therefore, we will use the lowest practical motor rpm to determine the coefficient of friction.

From the Limitorque motor curves and testing 55 Limitorque AC motors, we observed that motor breakdown occurred above 800 rpm for 4 pole motors and 1200 rpm for 2 pole motors. Table 4 is based on the 800 rpm breakdown speed of 4 pole motors. The worm speed and efficiencies are calculated with a 2:1, 1:1 and 1:2 motor pinion ratio. The efficiency is calculated from the actuator geometry and friction factor with the following equation:

$$Efficiency = \frac{.96}{\frac{frictionfactor}{efficiencyfactor} + 1}$$

OTHER CONSIDERATIONS

This analysis did not revise actuator efficiency for losses in the upper bearings of the actuator drive sleeve. Because the load on this bearing is proportional to stem thrust and not torque alone, this would introduce variation into the measured coefficient of friction. Because the standard deviation is less than 6%, the variance due to changes in stem factor for the set of actuators in this sample would have to be less than 6%. Additionally, EPRI separate effects testing (Reference 10) found that the friction coefficient is influenced primarily by worm speed and that the operator efficiency was essentially unchanged with added load. Based on these findings, it was not necessary to adjust the coefficient of friction for changes in the upper bearing drag due to stem factor changes.

This analysis did not consider the effects of motor inertia. The INEL motor torque data was measured at the shaft going into the actuator. If the motor deceleration resulted in motor inertia being converted to torque, that torque would have been measured. The EPRI data was based on motor current and any extra torque available due to rapid motor deceleration would not have been measured. This would result in a higher measured efficiency and a lower coefficient of friction for the EPRI tests. From Figure 9 you can see that the EPRI tests did not result in lower coefficients of friction. Therefore, inertial effects were not significant during the EPRI tests. Since motor capability is based on electrical rather than total available torque, inertial effects seen in actual service will provide margin above the motor capability calculated using these efficiencies.

DC motors were not considered in this analysis. However, an actuator responds to the motor torque and rpm rather than the type of potential which is applied to the motor. These curve for coefficient of friction versus motor speed should be equally applicable to DC motors. The actuator efficiency would then be calculated based on the speed at hard seat contact. For Table 4 motor speed of 800 rpm was chosen based on the AC motor curves. Since DC motors' speed varies linearly with output torque, the motor speed used to select the coefficient of friction would have to be chosen based on motor speed at the rated torque.

CONCLUSION

The efficiencies in Table 4 can be compared to the values published by Limitorque for these actuators. Table 4 doesn't include all of the actuators or worm geometries made, but it does represent a fair cross section of actuators that are in service in nuclear plants. Limitorque's pullout efficiency multiplied by the 0.9 application factor is within the range of the bounding efficiencies predicted from this testing, with one exception. The published efficiency for the SMB-1 actuator with the 66:1 worm ratio are a few percent higher than the predictions based on this testing.

Based on this analysis, two courses of actions are available when calculating the maximum available torque from a Limitorque actuator. The first is to use the published Limitorque pullout efficiency with the appropriate application factor in conjunction with the breakdown torque from the AC motor. The pullout efficiency for SMB-1 actuator with a 66:1 worm should be derated 90% of the pullout efficiency and then the appropriate application factor should be used when the breakdown torque of the motor is used. For actuators not listed in table 4 or where a more

refined actuator efficiency is required, the efficiency for a given actuator and gear ratio combination could be calculated based on the formulae given above and the bounding coefficient of friction shown in figure 9.

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Table 4
Calculated Actuator Efficiencies

Act. Size	Worm Ratio	MP Ratio	Eff Factor	WS Factor	WS fpm	COF	Eff Nom	Eff Bound
000	50	2	0.0721	0.3518	281.44	0.1058	0.3891	0.3672
000	50	1	0.0721	0.1759	140.72	0.1198	0.3607	0.3395
000	50	0.667	0.0721	0.1173	93.86	0.1352	0.3445	0.3237
00	45	2	0.0813	0.5152	412.16	0.1001	0.4302	0.4077
00	45	1	0.0813	0.2576	206.08	0.1117	0.4044	0.3823
00	45	0.511	0.0813	0.1316	105.31	0.1262	0.3760	0.3545
0	37	2	0.0869	0.724	579.2	0.0972	0.4531	0.4304
0	37	1	0.0869	0.362	289.6	0.1053	0.4340	0.4115
0	37	0.714	0.0869	0.2585	206.77	0.1116	0.4202	0.3979
1	34	2	0.0825	0.9692	775.36	0.0968	0.4418	0.4191
1	34	1	0.0825	0.4846	387.68	0.1009	0.4318	0.4093
1	34	0.8	0.0825	0.3877	310.14	0.1042	0.4243	0.4019
1	66	2	0.0466	0.8974	717.92	0.0967	0.3122	0.2925
1	66	1	0.0466	0.4487	358.96	0.1019	0.3012	0.2818
1	66	1.4	0.0466	0.62818	502.54	0.0981	0.3091	0.2895
2	33	2	0.073	1.1728	938.24	0.0978	0.4104	0.3882
2	33	1	0.073	0.5864	469.12	0.0988	0.4080	0.3859
2	33	0.795	0.073	0.4662	372.95	0.1014	0.4018	0.3797
2	60	2	0.0519	1.0472	837.76	0.0970	0.3345	0.3140
2	60	1	0.0519	0.5236	418.88	0.1000	0.3281	0.3078
2	60	1.333	0.0519	0.6980	558.37	0.0974	0.3337	0.3133
3	41	2	0.0838	1.3328	1066.24	0.0991	0.4400	0.4174
3	41	1	0.0838	0.6664	533.12	0.0977	0.4433	0.4206
3	41	1.07	0.0838	0.7130	570.44	0.0973	0.4443	0.4216
4	49	2	0.0893	1.3582	1086.56	0.0993	0.4546	0.4319
4	49	1	0.0893	0.6791	543.28	0.0976	0.4588	0.4360
4	49	1.117	0.0893	0.7586	606.84	0.0970	0.4602	0.4374

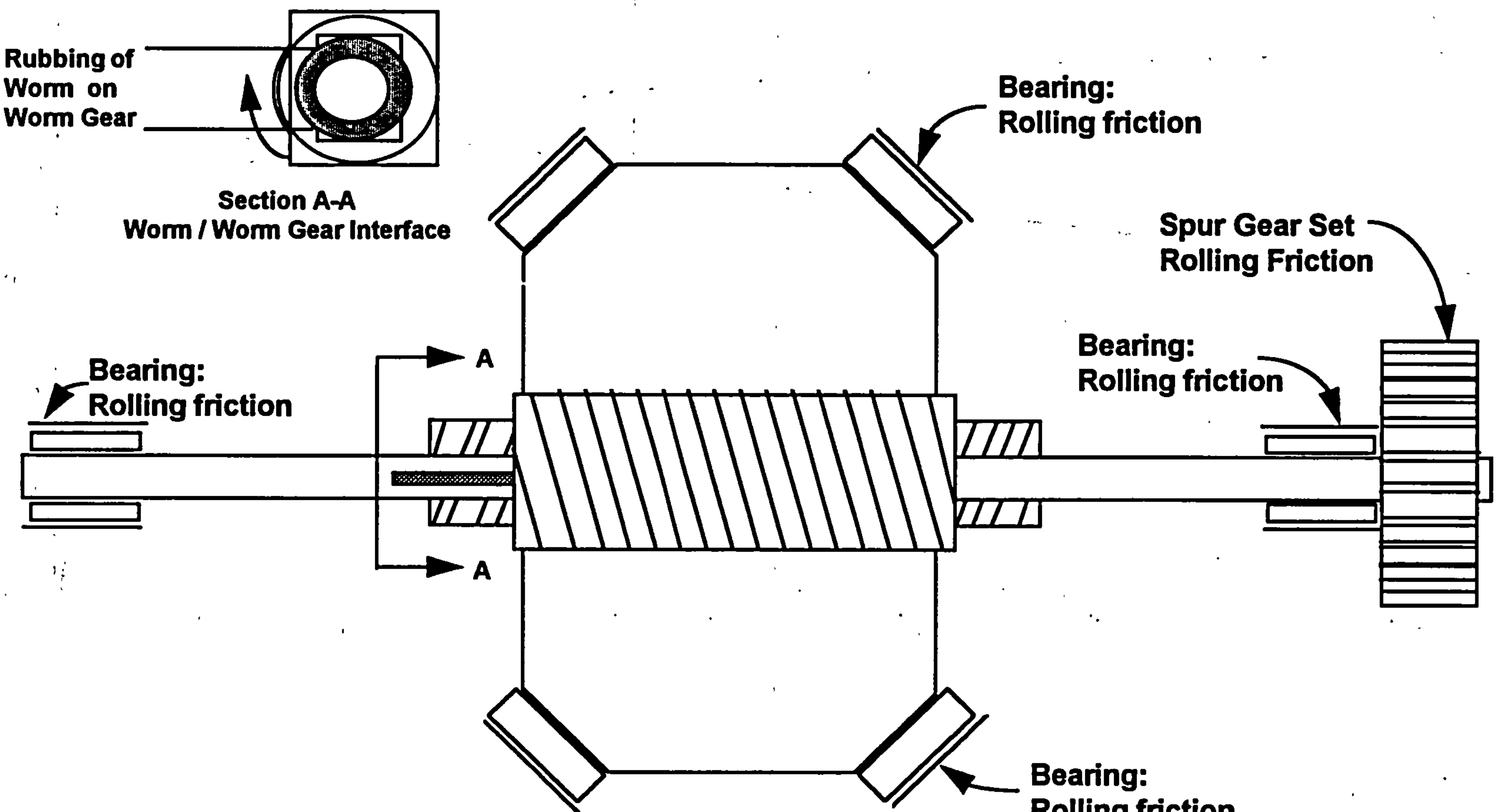

Figure 1 **SCHEMATIC OF LIMITORQUE DRIVE TRAIN**

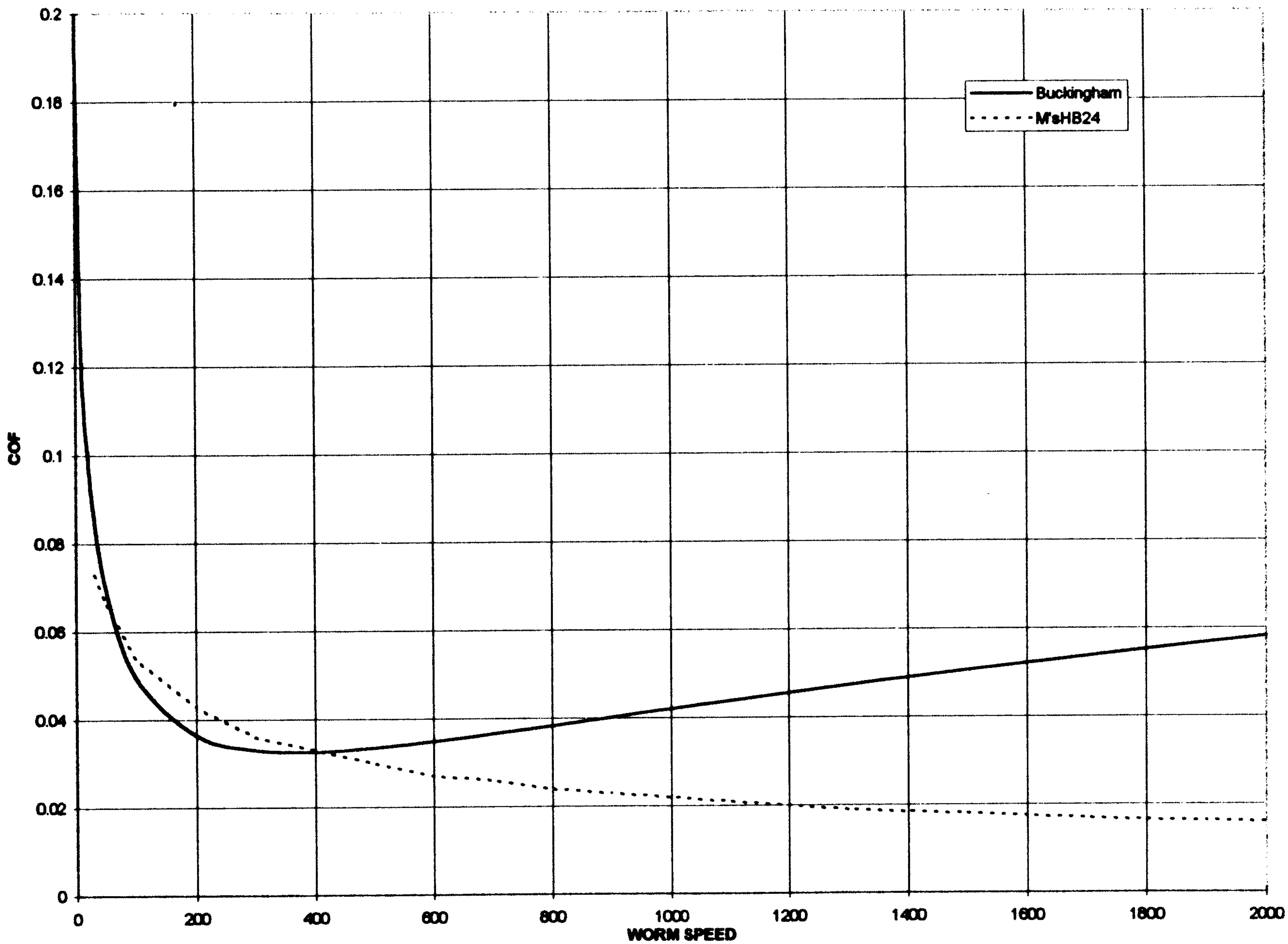

Figure 2 Coefficient of Friction From Literature

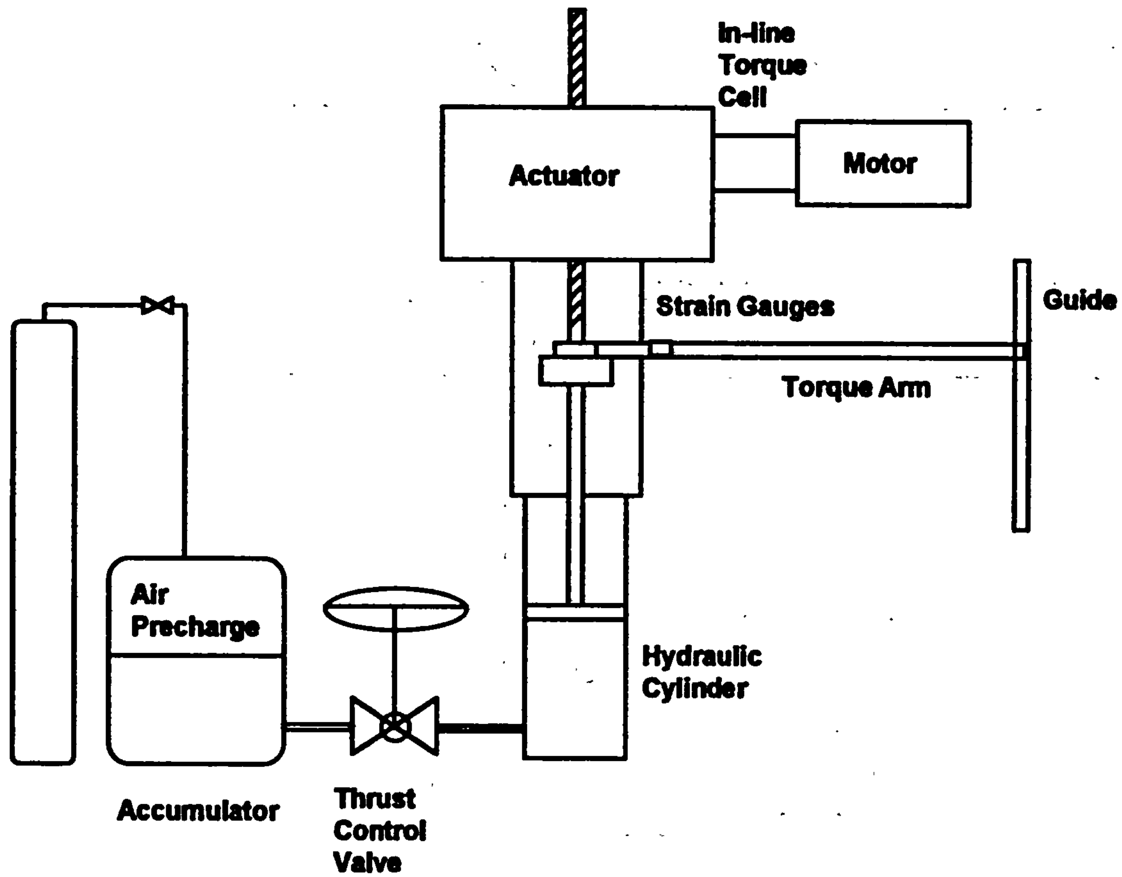

Figure 3 SCHEMATIC OF INEL MOTOR OPERATED VALVE LOAD SIMULATOR

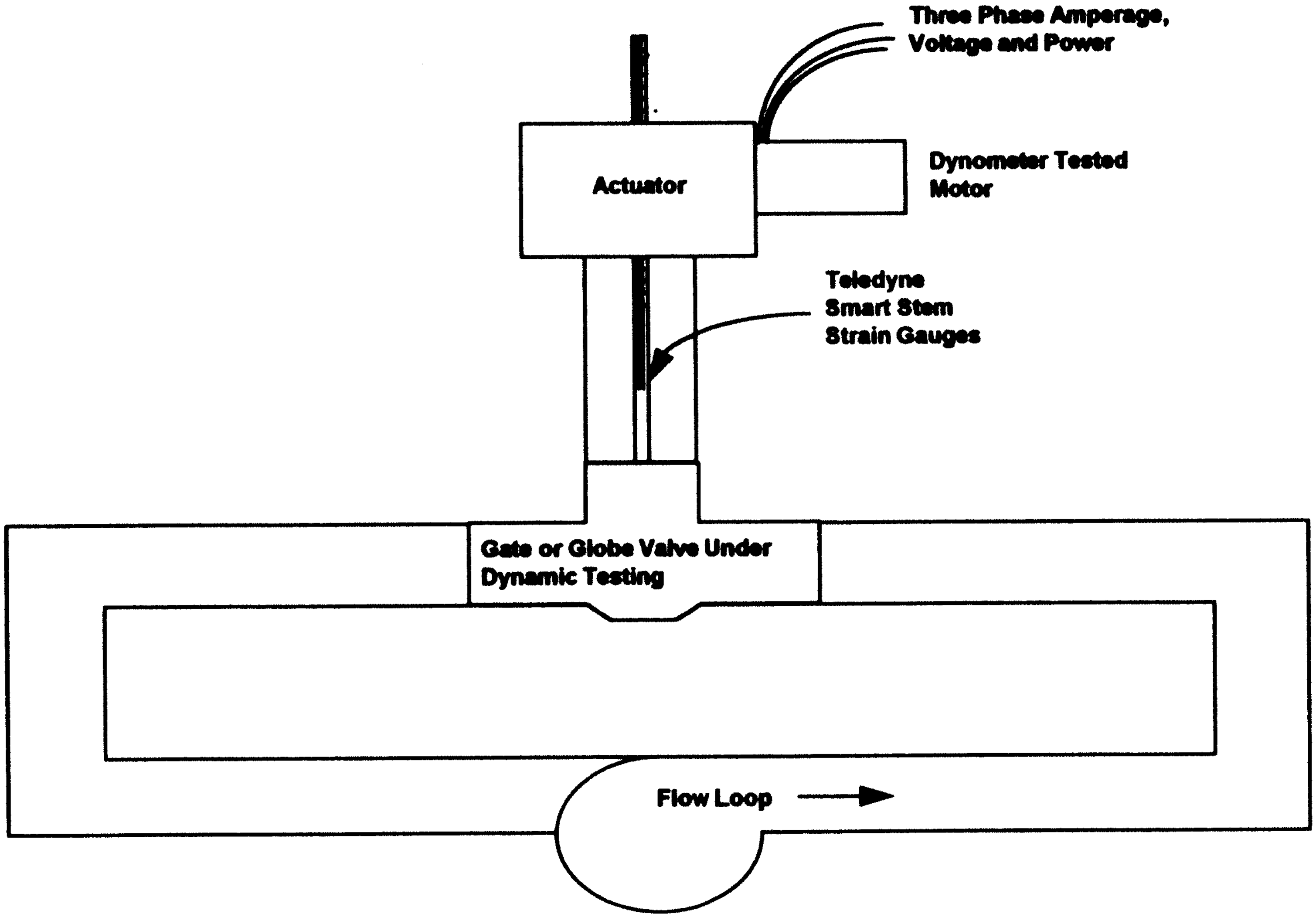

Figure 4 SCHEMATIC OF EPRI VALVE FACTOR TESTING SETUP

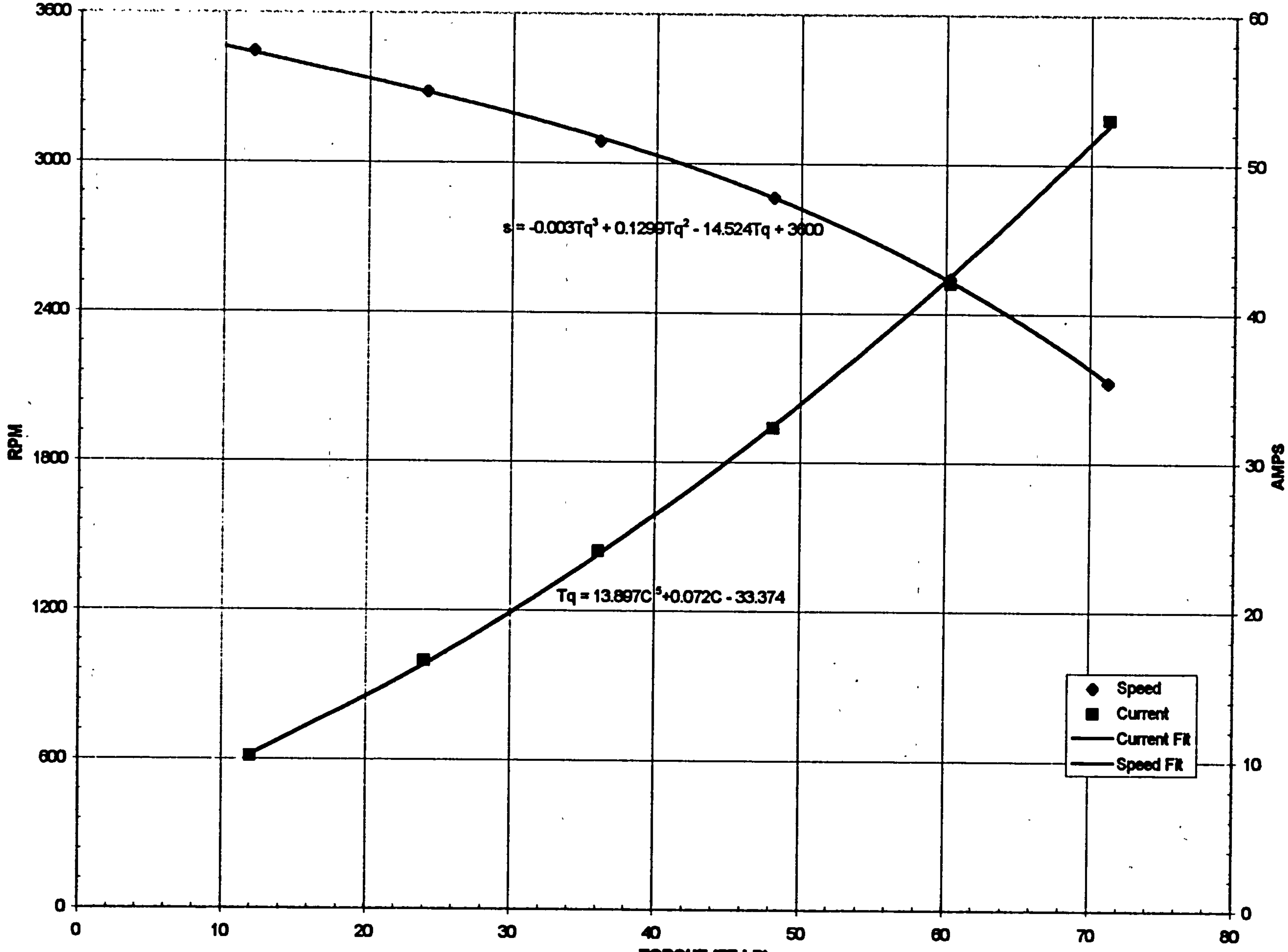

Figure 5 Motor Characteristics

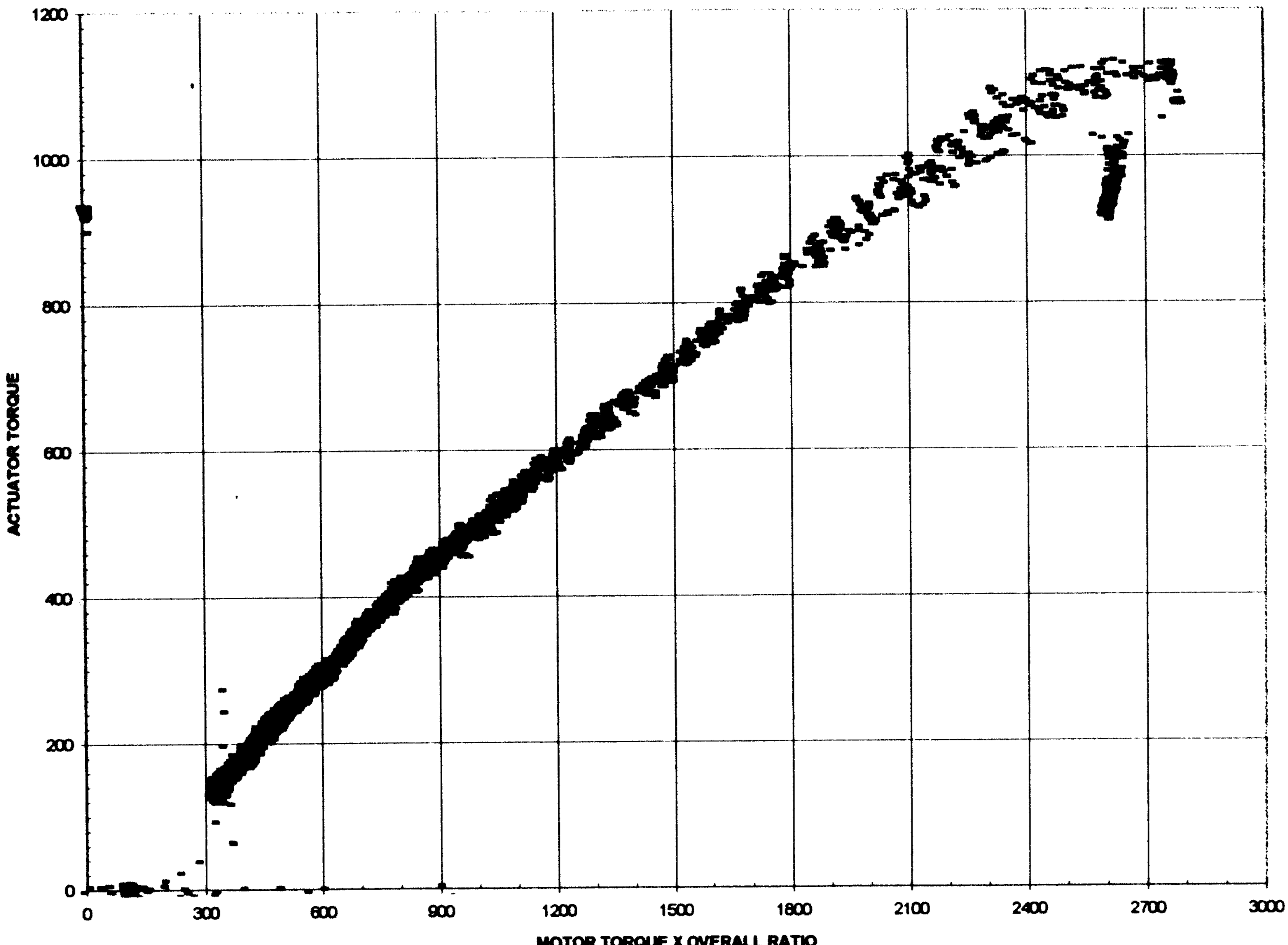

Figure 6A Raw Actuator Efficiency

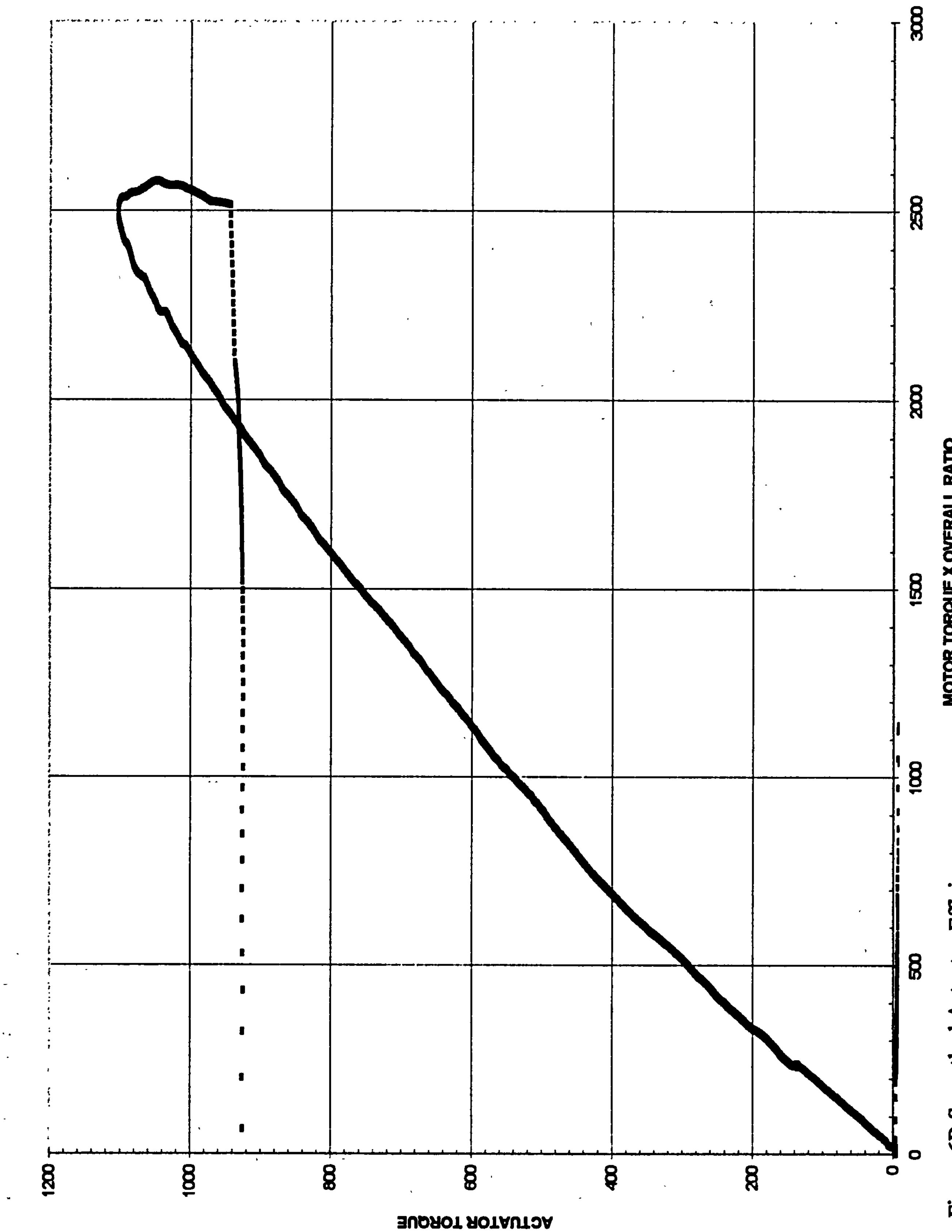

Figure 6B Smoothed Actuator Efficiency

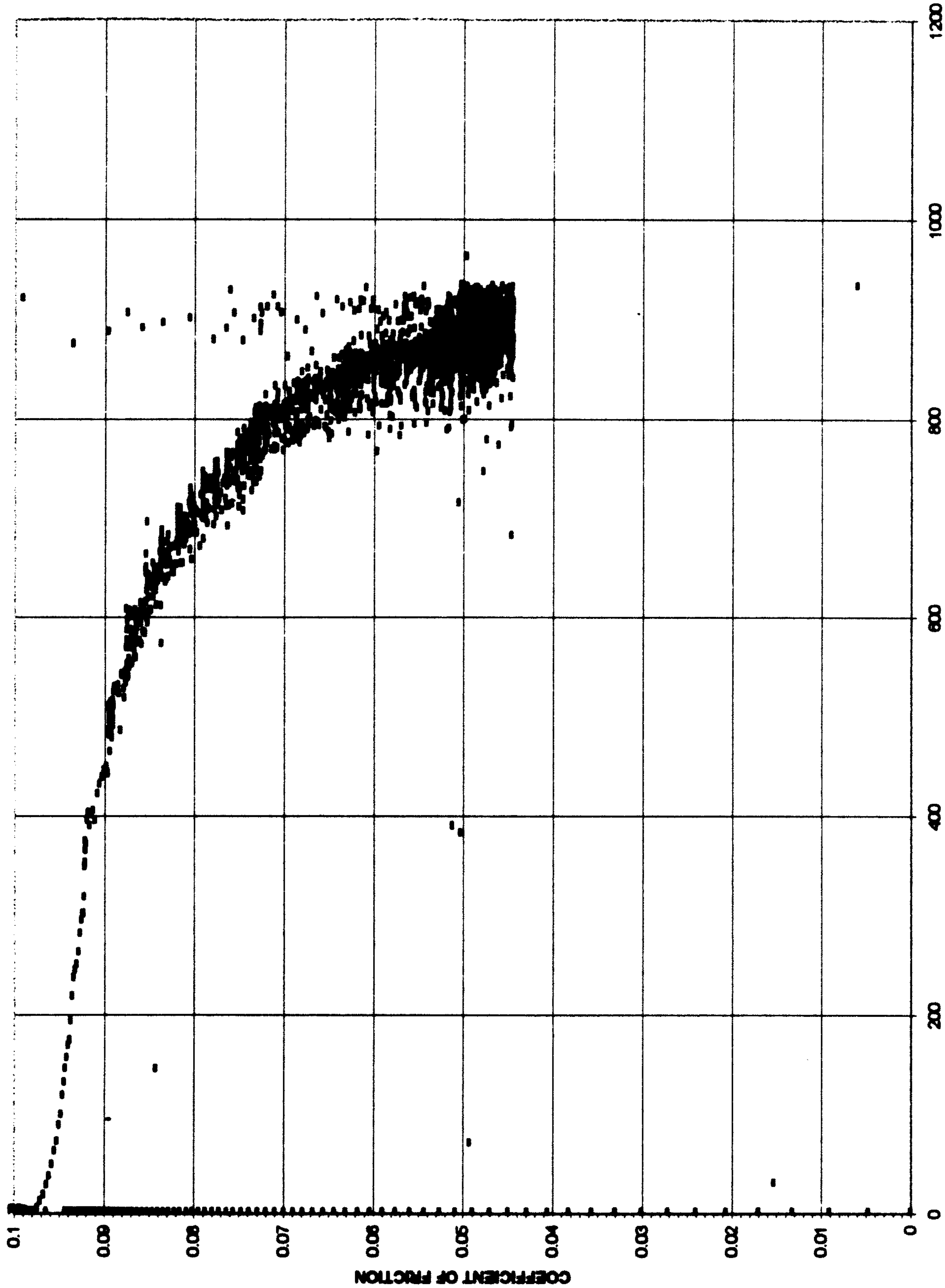

Figure 7 Raw Coefficient of Friction

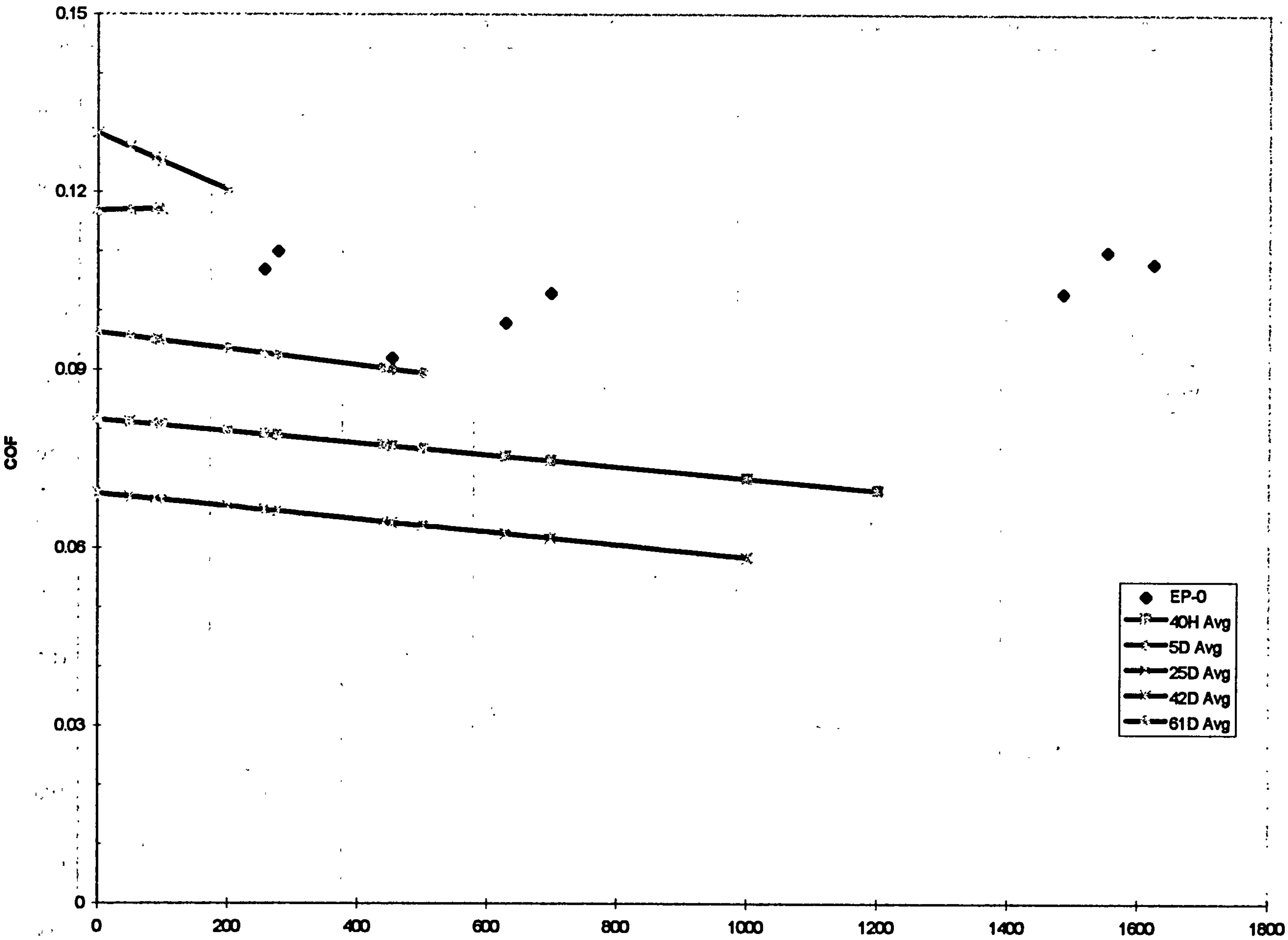

Figure 8 Coefficient of Friction Versus Worm Speed

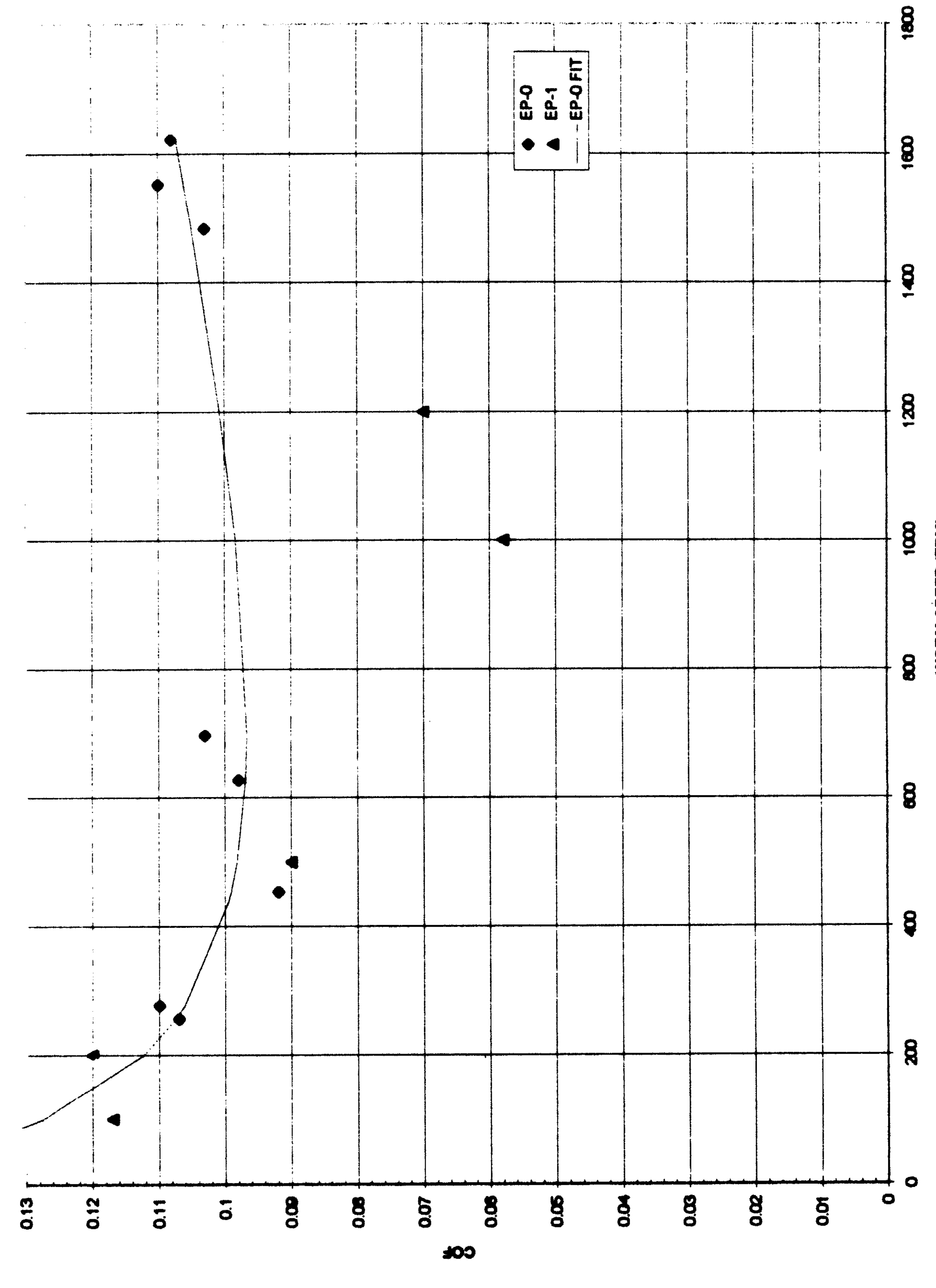

Figure 9 Worm/Worm Gear COF

An Improved Gate Valve for Critical Applications in Nuclear Power Plants

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ABSTRACT

U.S. Nuclear Regulatory Commission Generic Letters 89-10 for motor-operated valves (MOVs) and 95-07 for all power-operated valves document in detail the problems related to the performance of the safety-related valves in nuclear power plants. The problems relate to lack of reliable operation under design basis conditions including higher than anticipated stem thrust, unpredictable valve behavior, damage to the valve internals under blowdown/high flow conditions, significant degradation of performance when cycled under ΔP and flow, thermal binding, and pressure locking.

This paper describes an improved motor-operated flexible wedge gate valve design, the GE Sentinel Valve, which is the outcome of a comprehensive and systematic development effort undertaken to resolve the issues identified in the NRC Generic Letters 89-10 and 95-07. The new design provides a reliable, long-term, low maintenance cost solution to the nuclear power industry. The Sentinel Valve incorporates several innovative features and robust margins responsible for its reliable operation (predictable/ repeatable stem thrust and leak tightness) even under repeated cycling under severe blowdown conditions. One of the key features incorporated in the disc permits the disc flexibility to be varied independently of the disc thickness (pressure boundary) dictated by the ASME Section III Pressure Vessel & Piping Code stress criteria. This feature allows the desired flexibility to be incorporated in the disc, thus eliminating thermal binding problems.

In addition to the innovative features, development included a comprehensive analysis and testing program. A matrix of analyses was performed using finite element and computational fluid dynamics approaches to optimize design for stresses, flexibility, leak-tightness, fluid flow, and thermal effects. The design of the entire product line was based upon a consistent set of analyses and design rules which permit scaling to different valve sizes and pressure classes within the product line. The valve meets all of the ASME Section III Code design criteria and the N-Stamp requirements. The performance of the valve was validated by performing extensive separate effects and plant in-situ tests. Additional flow loop tests are presently being conducted to validate performance characteristics over a wider range of operating conditions applicable to both BWR and PWR plants.

This paper summarizes the key design features, analyses, and test results.

INTRODUCTION AND BACKGROUND

Problems with the reliable operation of motor-operated valves (MOVs) in U.S. nuclear power plants are extensively documented in references [1] through [16]. Tests conducted in 1989 by the U.S. Nuclear Regulatory Commission (NRC) to address Generic Issue 87 highlighted the fact that all wedge gate valves required significantly more torque and stem thrust than the recommendations established by the manufacturers, and damage to the valve internals occurred when attempting to close the valves under design basis,

high flow conditions [3, 4]. Tests revealed that the operating thrust in the opening direction also significantly exceeded manufacturers' recommendations. The U.S. NRC issued Generic Letter 89-10 for the U.S. nuclear power plants to address these issues for all safety-related valves.

Even though pressure locking and thermal binding phenomena can further increase the opening thrust substantially and cause serious valve damage [23 to 28], the NRC Generic Letter 89-10 did not emphasize the significance of these phenomena. Accordingly, even the very recently completed comprehensive EPRI research program [17, 18], which significantly advanced the state-of-the-art in predicting the performance of valves, excluded pressure locking and thermal binding phenomena from its scope. The NRC recently issued Generic Letter 95-07 to ensure that the industry does systematically identify and eliminate potential pressure locking and thermal binding problems.

Furthermore, the NRC will be issuing another generic letter in the near future with recommendations for establishing periodic verification programs. The periodic verification program is intended to ensure that the valves maintain their capability to operate under design basis conditions under age-related degradation phenomenon, including valve cycling.

Another problem that has emerged in the MOVs is inadequate margin in the weak-link structural strength. This is due to significantly higher thrust requirements (than originally recommended by the manufacturers) as well as a number of uncertainties that were not taken into account while establishing actuator control switch settings in the past. These uncertainties include control switch repeatability, stem-to-stem nut friction coefficient variations, measurement equipment inaccuracies, and rate-of-loading effect (or load sensitive behavior) [17, 20, 22]. One of the objectives of the new valve design is to provide robust margins in the weak link considering the above factors.

A comprehensive valve development program using a "clean slate" approach was undertaken by GE Nuclear Energy (GENE) in alliance with Kalsi Engineering, Inc., and Ring-O Valve, SpA, to provide the nuclear power industry with an improved gate valve that has predictable and repeatable performance, eliminates pressure locking and thermal binding concerns, maintains leak tightness, and does not exhibit degradation when subjected to a large number of cycles even under maximum ΔP and blowdown conditions.

The new valve design (GE Sentinel Valve) is the outcome of a comprehensive technical approach which included the development of a generic design methodology which is consistently applied to the entire product line, design optimization by finite element and computational fluid dynamic analyses, separate effects testing, and plant in-situ testing. Extensive flow loop testing is presently being conducted to validate performance characteristics over a wider range of operating conditions applicable to both BWR and PWR plants.

KEY DESIGN FEATURES AND TECHNICAL APPROACH

After careful evaluation of the trade-offs in alternative gate valve design approaches it was decided to base the new design on the one-piece wedge gate valve construction. We were aware of the U.K. Sizewell B test experience which showed "The high seat leakages exhibited by parallel slide gate valves at low differential pressures shown in the test programme results has led to restrictions on the use of this type of valve in SXB for isolation duties where Containment Isolation is important" [29]. The Sentinel design capitalizes on the inherent simplicity and superior leak tightness of the wedge disc principle while eliminating its shortcomings, e.g., potential for damage under blowdown conditions, susceptibility to thermal binding, inadequate margins in the structural strength of the weak link. A number of innovative, but simple, features[1] were incorporated to overcome these limitations. Furthermore, it was found that it is not possible to achieve the required margins in the structural strength of the weak links within the existing geometrical constraints of most valves. Therefore, a "clean slate" approach without being constrained by the geometrical

[1] Patents pending.

limitations of the existing valve body was pursued. Figure 1 shows the cross-section of the Sentinel Valve.

Eliminating Potential for Disc/Seat/Guide Damage

In conventional wedge gate valves, fluid force during intermediate disk travel imposes a moment on the disc that tends to cause disc tipping, which in turn is responsible for high edge loading and damage to the disc and seat faces, as well as the lower guide surfaces as shown in Figure 2. The fluid-induced moment on the disk for any given flow and ΔP condition is zero in the fully open and fully closed positions with a maxima at an intermediate disk travel position. The magnitude of the fluid-induced moment on the disk and the potential for damage increases with an increase in flow velocity. Under high energy blowdown conditions, damage to the disc and seat faces and/or the guide surfaces has been observed with many conventional wedge gate valve designs and parallel disc designs [3, 4, 15, 17, 18].

The potential for damage to the disc and seat faces is eliminated in the new valve by utilizing a flat bottom disc design. The flat bottom disc prevents disc tipping, thus ensuring an *area* contact at the seat surfaces instead of the *point* contact that typically occurs in conventional disc designs.

Disc tipping can also damage the guides due to high edge loading. Tests have shown that severe guide damage can occur even under conditions that do not cause disc tipping [4, 21]. The potential for guide damage in the new design is eliminated by utilizing full length guides, hardfacing all three guide surfaces that can potentially contact, and incorporating a new design feature that allows the guide ends to elastically flex under load. Local flexure reduces the peak stress by distributing the load over a larger area. Additionally, all of the disc, guide, and seat edges are provided with smoothly contoured chamfers and radii which further reduce the peak contact stresses that can cause rapid wear, degradation, or galling/gouging damage. The effectiveness of these design features was evaluated by a rigorous sequence of tests performed on a separate effects test fixture.

Engineered Disc Flexibility to Eliminate Thermal Binding

Eliminating pressure locking problems is relatively straightforward, and a number of acceptable approaches are available to suit the application-specific requirements [23, 26, 27]. The new design offers the option of an internal communication passage between the valve body bonnet area and the upstream or downstream side of the valve to eliminate pressure locking. This is the standard approach offered by other valve manufacturers as well.

Thermal binding problems, on the other hand, are more difficult to quantify and mitigate in conventional flexible wedge gate valves. The classical thermal binding scenario occurs when a valve that has been open and flowing hot fluid is closed and allowed to cool down. Under high temperature flowing conditions, the disc, residing in the bonnet region, becomes somewhat cooler than the valve body in the seat region. During the closing stroke, the disc does not heat up sufficiently to eliminate this temperature difference. The cooler disc is thus wedged in the warmer seat by the final wedging thrust during a closure stroke. As the valve cools to ambient conditions, the valve body and disc can potentially develop mechanical interference because of the different expansion and contraction characteristics of the valve body and disc. It is possible for the disc to be bound so tightly that reopening is either difficult or impossible until the valve is reheated.

The magnitude of increase in force at the seat faces (caused by mechanical interference due to temperature differences), as well as the increase in the unwedging/opening thrust, is directly proportional to the stiffness of the disc. The magnitude of increase in opening thrust can be substantial – in some cases exceeding the structural strength and causing failure of the weak link, e.g., stem-to-disc T-slot connection [23]. However, no quantitative stiffness criteria have been developed by the industry to eliminate thermal binding problems in flexible wedge gate valves. In fact, the stiffness of the disc in the conventional flexible wedge design is indirectly

dictated by the pressure boundary stress criteria imposed by the ASME Section III Pressure Vessel and Piping Code. Thus, conventional "flexible" wedge disc design, especially the higher pressure class valves, can be too stiff and susceptible to thermal binding problems [23, 24, 26, 27].

The disc in the new valve design has been engineered to meet a rigorous disc flexibility criteria that was developed to ensure that the worst-case thermal binding scenarios would not cause the opening thrust to be higher than the closing thrust. Extensive computational fluid dynamics and finite element analyses were performed to develop the bounding scenarios for thermal binding. The new disc geometry differs from conventional flexible wedge discs in that it has a central section in which a longitudinal slot (perpendicular to the flow axis) of appropriate dimensions can be incorporated to achieve the desirable disc flexibility. Full scale tests are being conducted to simulate postulated worst-case scenarios and to validate the analytical model predictions to eliminate thermal binding.

Robust Margins in Structural Strength

One of the problems common to wedge gate valves is that a significant thrust overshoot occurs after the actuator control switch (torque switch or limit switch) is tripped. The overshoot is the result of delay in contactor drop-out in the actuator and due to mechanical inertia. Furthermore, in determining the minimum and maximum actuator control switch settings, uncertainties in control switch repeatability, measurement system accuracy, and rate of loading effect (also called "load sensitive behavior"), must all be appropriately combined and taken into account [22]. The maximum thrust delivered to the valve can exceed the minimum required thrust by 75 percent or even more due to the above factors. It should be noted that most of the existing wedge gate valve designs were based upon valve factors of typically around 0.3, whereas the actual valve factors to ensure operation under design basis conditions may be 0.5 or higher, based on recent industry research [4, 19]. Both the higher valve factors and the uncertainties that were previously neglected in determining control switch settings result in a substantially higher thrust than originally anticipated in the design. This situation often results in the structural strength of the valve weak link being exceeded.

The new valve is designed with robust margins in the weak link, which is capable of withstanding 2.5 times the minimum required thrust under the maximum permissible ΔP and temperature combinations established for various pressure class valves within the product line. The stresses of this 2.5 times the minimum required thrust satisfy the ASME Section III criteria for normal operating conditions even when accounting for corrosion allowances. The survivable structural strength of the weak link is even higher.

Improved Leak Tightness

In the conventional flexible wedge gate valve designs, stem thrust is transmitted to the disc by the stem head directly bearing down on the two disc halves that form the sealing faces. This causes a local distortion of the disc seating faces, and the distortion increases with increasing stem thrust. This limits the leak tightness capabilities of the conventional flexible wedge gate designs. Improved leak tightness is achieved in the new valve design by eliminating the localized contact (and thereby eliminating the disc face distortion) between the stem head and the disc sealing faces. The stem thrust is delivered from the stem head to the disc center section, which is axisymmetrically connected to the two disc sealing faces (Fig. 1). This results in a uniform deflection at the disc/seat face along the entire circumference under loading conditions. The effectiveness of improved leak tightness was confirmed by several tests conducted at the manufacturing plant as well as in situ tests.

ANALYSIS

The new valve product line is based upon a generic design methodology using first principles analytical models to calculate stresses, deflections, and stiffnesses in all of the critical locations in the valve components. Computational fluid dynamics (CFD) analyses were performed to determine temperature distributions at various locations within the valve body under various scenarios that can cause thermal binding.

Parametric finite element analyses were performed to optimize the disc design. A geometric model was developed to consistently incorporate key dimensions, tolerances, and clearances in all the sizes and pressure classes of the product line.

Analysis of Thermal Characteristics for Thermal Binding Evaluation

Predicting the bounding temperature differences that can occur between the disc and the valve body under various operating scenarios that can potentially lead to thermal binding was a significant part of the analytical effort. The thermal characteristics of the new valve design were analyzed with the specific objective to evaluate and eliminate the potential for thermal binding. Two different methods were applied to model the valve and obtain solutions for the temperature field. A closed-form solution for a simplified model was used for obtaining a fundamental insight into the thermal characteristics of the valve. Figure 3 shows the turbulent mixing phenomena considered in the analytical model. Based on this insight, a finite element model was developed using the integrated ANSYS/FLOTRAN general-purpose, finite element analysis software to perform the simultaneous solution of heat transfer in the flow field and in the valve structure. The details of the fluid flow field and heat transfer field within the valve were evaluated over a range of operating conditions. Temperature distributions were computed for the structural parts of the valve with the disc in both open and closed position. The computed temperature distributions were used as an initial temperature field followed by an instantaneous valve closure and cooling down to ambient conditions to determine the increase in seat contact force and disc unwedging force. The matrix of analyses included parametric variations in flow rates, fluid properties, insulation thickness, and valve material for both open and closed disc positions. Analyses were performed using water and steam as fluid media, and flow rates were varied from 1,000 lb/hr to 500,000 lb/hr.

Figures 4, 5a, and 5b show the finite element/CFD model and the typical results of temperature distributions obtained under one of the four possible operational scenarios that were analyzed. As anticipated, the disc is cooler than the valve body region between the seat faces. It was found that the temperature difference is dependent upon the flow rate as well as fluid medium.

Disc Stiffness Analysis and Optimization

The worst-case bounding temperature differences found from the thermal characteristics analyses were used in developing the disc stiffness criteria necessary to eliminate thermal binding. In the disc center section, a slot of desired dimensions can be incorporated to achieve the required disc stiffness while fulfilling the ASME stress criteria. Figure 6 shows a typical three-dimensional finite element model mesh that was used to determine the stresses and deflections in the disc and optimize the disc stiffness. Parametric analysis approach was used by employing Pro/ENGINEER solid modeling software so that the entire product line can be efficiently and consistently designed.

SEPARATE EFFECTS TESTS

A matrix of tests was performed on the prototypical components from two different sizes of the new valve design in the "valve design effects" test fixture shown in Figure 7. This fixture was specially designed to evaluate the performance of various gate valve designs under simulated high-energy blowdown conditions and pump flow conditions in an earlier program [21]. In that program, a large number of tests confirmed that the fixture faithfully simulates actual valve performance in a flow loop, including the type and severity of damage to the seat and guide areas in conventional flexible wedge gate valves.

Variable forces and variable moments were applied to the disc by two computer-controlled servo hydraulic cylinders simulating the forces exerted on the disc by the fluid flow and differential pressure across the disc as it is stroked between open and closed positions. The test fixture was instrumented to provide measurements of forces applied to the disc by the two cylinders, stem-to-disc interface force, stem thrust, stem torque, and disc position. Tests were performed under a quality assurance program that complies with the 10CFR50 Appendix B criteria. A digital

data acquisition system was used to simultaneously acquire the data at 1,000 Hz for each variable.

Test Matrix: Tests were performed on the prototypical components of the new valve design for a 6" x 4" Class 900 stainless steel, and a 10" x 8" Class 900 carbon steel valves. The prototypical components included disc, seat, guide, and stem. Tables 1, 2, and 3 summarize the matrix of tests performed on these valves. All tests were performed using distilled water at room temperature conditions. Disassembly and detailed inspections were performed at several key intervals (identified in the tables) to determine and document any progressive degradation during the tests.

A second matrix of tests (Table 2), using new components from the 6" x 4" valve, was performed to determine the effect of applying the higher ΔP of 1800 psi in the very first stroke. Applying the highest load on new components had been found to cause more severe damage in earlier tests on conventional wedge gate valves[21]. Tests on the 6" x 4" valve also included an evaluation of the effect of extreme tolerances of the guide placement on valve performance. This tolerance affects the position at which the disc transitions from guide contact to downstream seat contact, which in turn affects the potential for disc tipping and the magnitude of the highest load on the guides and downstream seat at the point of transition.

The matrix of tests performed on the 10" x 8" valve components is shown in Table 3. Since the closing stroke under blowdown conditions has been confirmed to be the most severe test condition, tests on the 10" x 8" valve components were performed only in the closing direction.

Test Results: Typical results for a closing stroke, including the calculated valve factor, for the tests performed on the internal components of the two valves are shown in Figures 8 and 9. In all of the tests, both the 6" x 4" and the 10" x 8" valves performed very smoothly and consistently and well within the thrust predictions.

The sliding coefficient of friction between the disc to seat as well as the disc to guide surfaces remained below 0.5 in all of the tests, including repeated blowdown and high ΔP pump flow cycles on these valves. All seating and guide surfaces on both valves were found to be in excellent condition at the conclusion of the tests. Periodic and final inspections revealed only burnishing and polishing of the sliding interfaces and no damage to any leading edges or high contact stress areas. Figures 10 and 11 document the condition of the 6" x 4" valve components after being subjected to 9 blowdown closing strokes at 1134 psi, 100 close/open strokes at 1134 psi, and 5 blowdown closing strokes at 1800 psi. The valve performance was unaffected by the severity of maximum flow velocity. No adverse affects were caused by blowdown as compared to pump flow conditions.

From the second matrix of tests on the 6" x 4" valve, it was found that the valve exhibited the same smooth performance and no damage to the sliding interfaces (as observed in the first series of tests) when subjected to the highest ΔP (1800 psi) in the very first cycle. It was also found that the performance was unaffected over the extreme range of tolerances on the guide placement.

IN SITU TESTS

In situ tests were performed on four Sentinel valves installed at Boston Edison Company's Pilgrim Station using GE's BWR/3 reactor design. Two 6" Class 900 valves were installed in the reactor water clean-up (RWCU) system as inboard and outboard containment isolation valves. Two 10" Class 900 valves were installed in the high pressure coolant injection (HPCI) system as turbine steam admission and outboard containment isolation valves. All four valves were instrumented with Teledyne Smart Stem strain gages to measure thrust and torque. The total error in thrust measurement was within 3 percent. No attempt was made in these plant in situ tests to precondition the valves by repeated stroking under ΔP to determine the maximum bounding disk friction values, as in the EPRI program [18].

The following standard industry equation was used to calculate the valve factors reported below:

$$\text{Valve Factor} = \frac{F_{total} - F_{pack} \pm F_{stem}}{\left(\frac{\pi}{4} D^2 \Delta P\right)}$$

where

F_{total} = Total stem thrust

F_{pack} = Packing friction force

F_{stem} = Stem rejection force (positive for opening and negative for closing valve factor calculation)

$= \frac{\pi}{4} d_{stem}^2 \times P_{up}$

D = Mean seat diameter

d_{stem} = Stem diameter

P_{up} = Upstream pressure

ΔP = Differential pressure across the disk

Static Test Results: The in situ tests performed consisted of five static thrust strokes on each of the four valves. All four valves exhibited smooth and repeatable performance in all test strokes.

ΔP Hydrotest Results: Three opening stroke tests using hydrostatic pressure of 1125 psi were performed on the HPCI outboard containment isolation valve. The valve performed smoothly with an opening valve factor of 0.37.

Dynamic Test Results: Dynamic tests using steam were performed on both of the HPCI turbine steam admission and outboard containment isolation valves. One closing stroke test with 1,035 psi ΔP and over 100,000 lb/hr flow rate was performed on the outboard containment isolation valve, and two opening stroke tests with 1,025 psi and over 150,000 lb/hr flow rate were performed on the turbine steam admission valve. The valves operated smoothly in all dynamic tests with valve factors of 0.4 or less.

Leak Rate Test Results: The three containment isolation valves were tested for seat leakage with air at 45 psig. The three valves exhibited very low leakage rates of 0.06, 0.07, and 0.89 Std Lt/Min, which were well within the local leak rate test specification limits for these valves. Table 4 summarizes the leak rate test results.

CONCLUSION

An improved gate valve has been developed which has been demonstrated to provide predictqable performance with little or no degradation of valve internals even under repeated severe blowdown conditions. The new valve product line is the outcome of a systematic development effort which has been validated by extensive separate effects testing and in situ plant tests. Additional flow loop tests are presently being conducted to validate the product line performance over a wider range of operating conditions for both BWR and PWR plants.

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[2] Proprietary report, not publicly available.

Table 1

First Matrix of Tests on 6" x 4" Valve Components

No. of Strokes	Direction of Strokes		Maximum Simulated ΔP psi	Maximum Simulated Flow Velocity at Full Open Position
	Opening	Closing		
Qualification Testing				
25	X	X	1,134	0 (Preconditioning)
5		X	1,134	Blowdown
1	X		1,134	Blowdown*
Extended Testing to Evaluate Degradation Under Repeated Blowdown and High Velocity Pump Flow Conditions				
25	X	X	1,134	50 fps pumped flow
1		X	1,134	Blowdown*
25	X	X	1,134	50 fps pumped flow
1		X	1,134	Blowdown*
25	X	X	1,134	50 fps pumped flow
1		X	1,134	Blowdown*
25	X	X	1,134	50 fps pumped flow
1		X	1,134	Blowdown*
5		X	1,800	Blowdown*

Disassembly and detailed inspection after this stroke

Table 2

Second Matrix of Tests on 6" x 4" using New Valve Components

No. of Strokes	Direction of Strokes		Maximum Simulated ΔP psi	Maximum Simulated Flow Velocity at Full Open Position
	Opening	Closing		
28	X	X	1,800	0 (Preconditioning)
5		X	1,800	Blowdown*
5		X	1,800	Blowdown*

* Disassembly and detailed inspection after this stroke

Table 3

Matrix of Tests on 10" x 8" x 10" Valve Components

No. of Strokes	Direction of Strokes		Maximum Simulated ΔP psi	Maximum Simulated Flow Velocity at Full Open Position
	Opening	Closing		
30	X	X	1,120	0 (Preconditioning)
5		X	1,120	Blowdown*
20		X	1,120	Blowdown*
10		X	1,800	Blowdown*

* Disassembly and detailed inspection after this stroke

Table 4
In Situ Leak Rate Test Results

Valve Identification	System	Leak Rate
6" Inboard Containment Isolation Valve	RWCU	0.07 Std Lt/Min
6" Outboard Containment Isolation Valve	RWCU	0.89 Std Lt/Min
10" Outboard Containment Isolation Valve	HPCI	0.06 Std Lt/Min

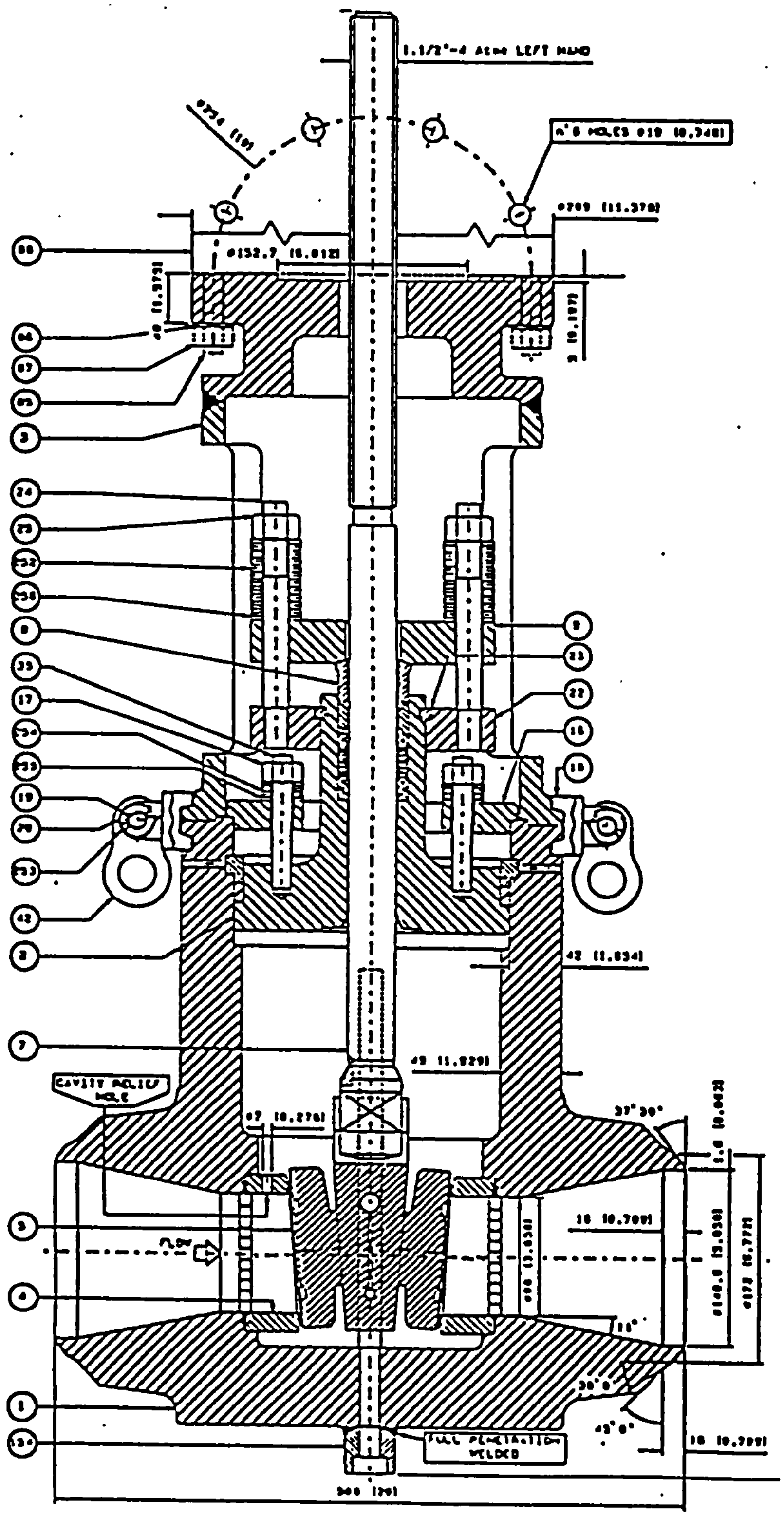

Figure 1: GE Sentinel Gate Valve Cross-Section

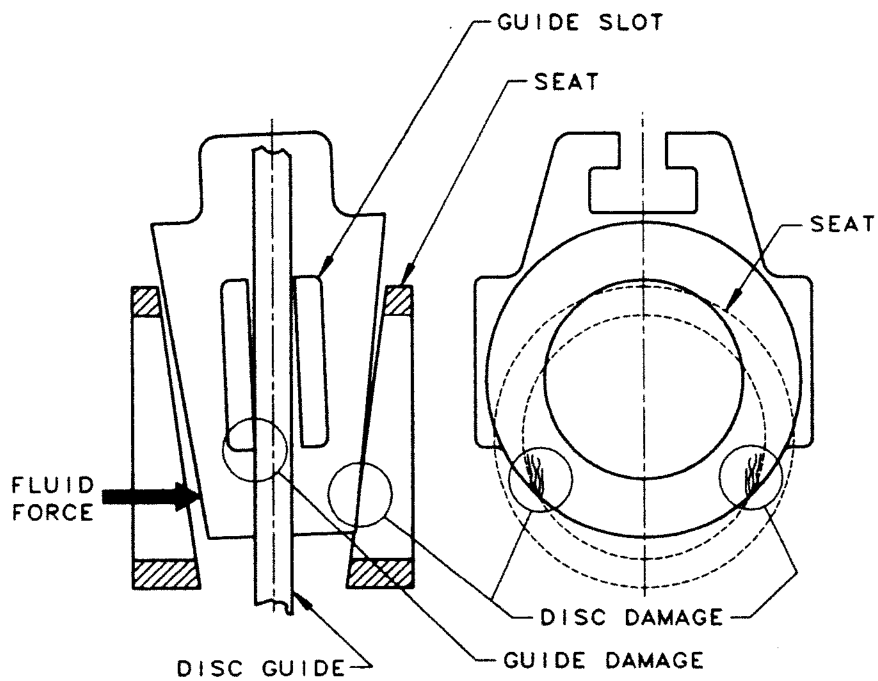

Figure 2: Typical Seat and Guide Damage Locations in Conventional Flexible Wedge Gate Valves Under High Flow Conditions

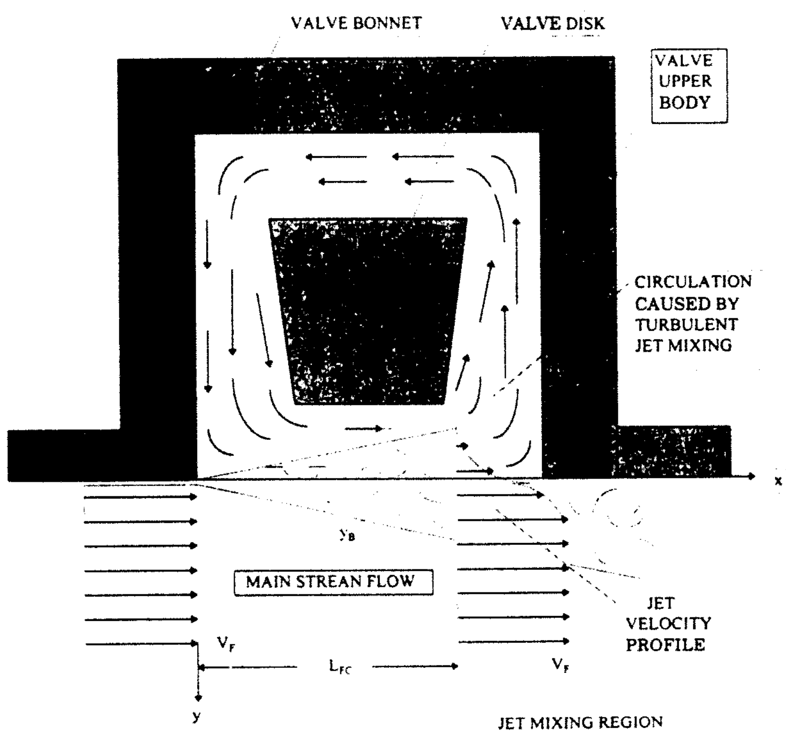

Figure 3: Turbulent Mixing Model of Two-Dimensional Jet for Analytical Predictions of Disc Temperature

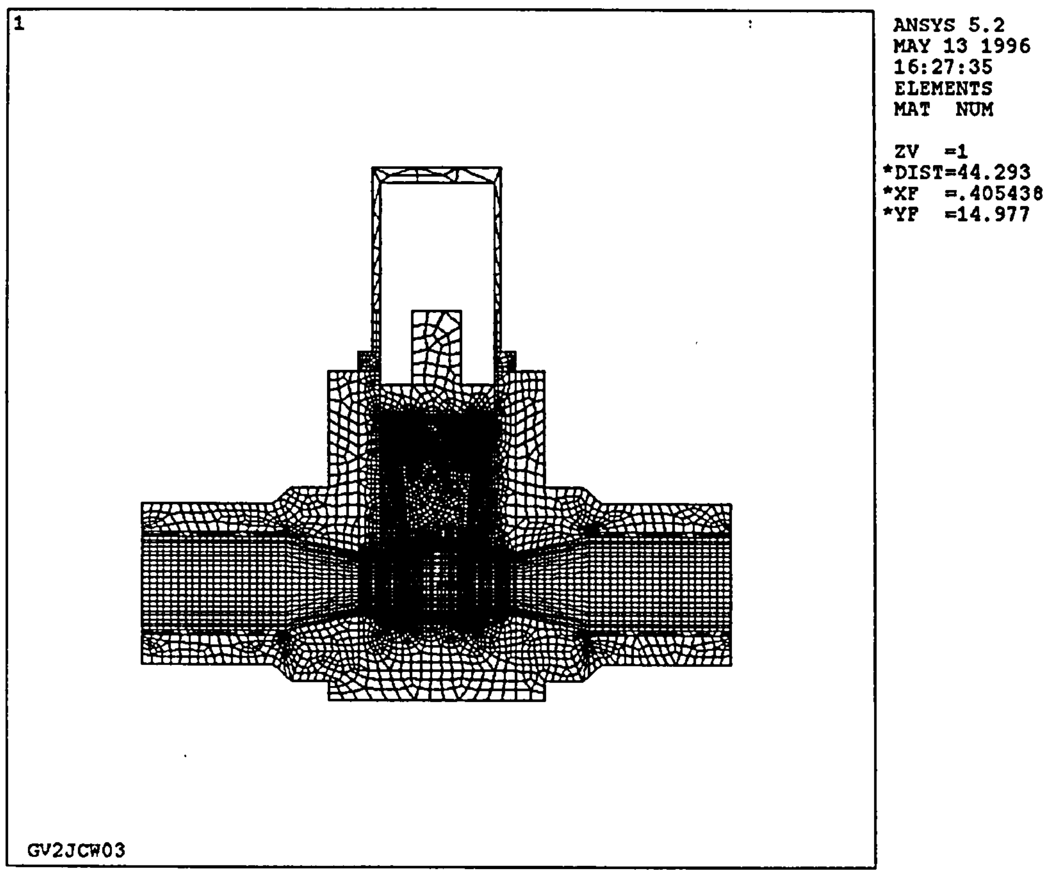

Figure 4: Finite Element and CFD Model of an Open Valve with Insulation

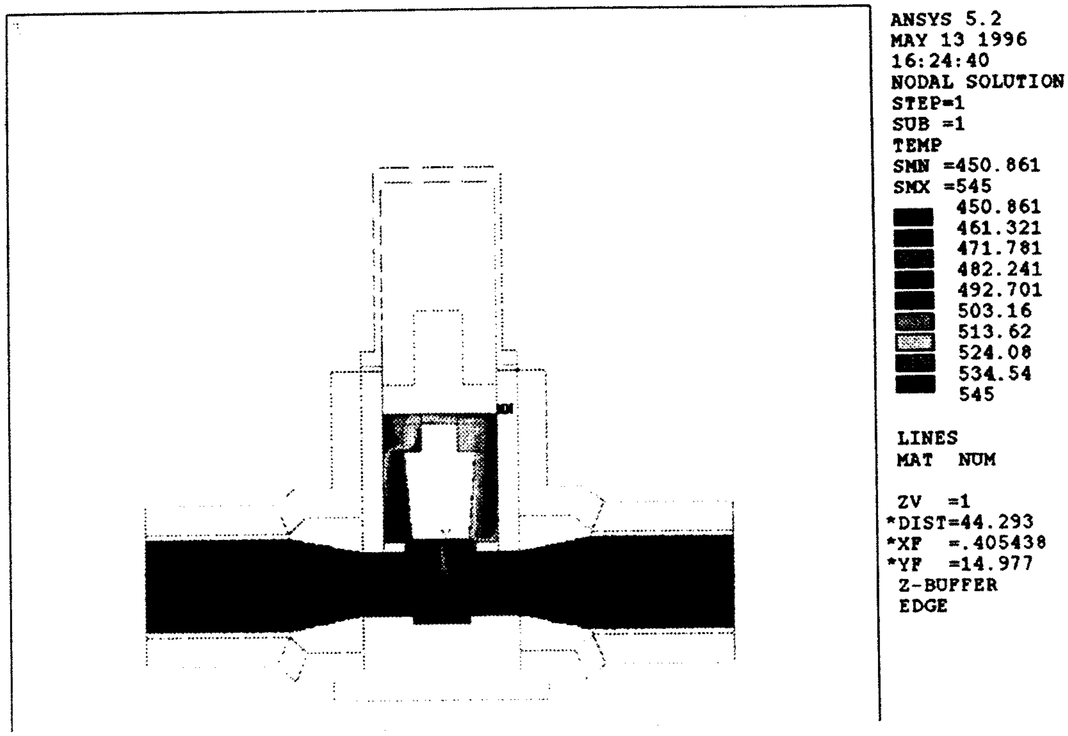

Figure 5A: CFD Analysis Results – Fluid Temperature Distribution Showing Cooler Bonnet Region

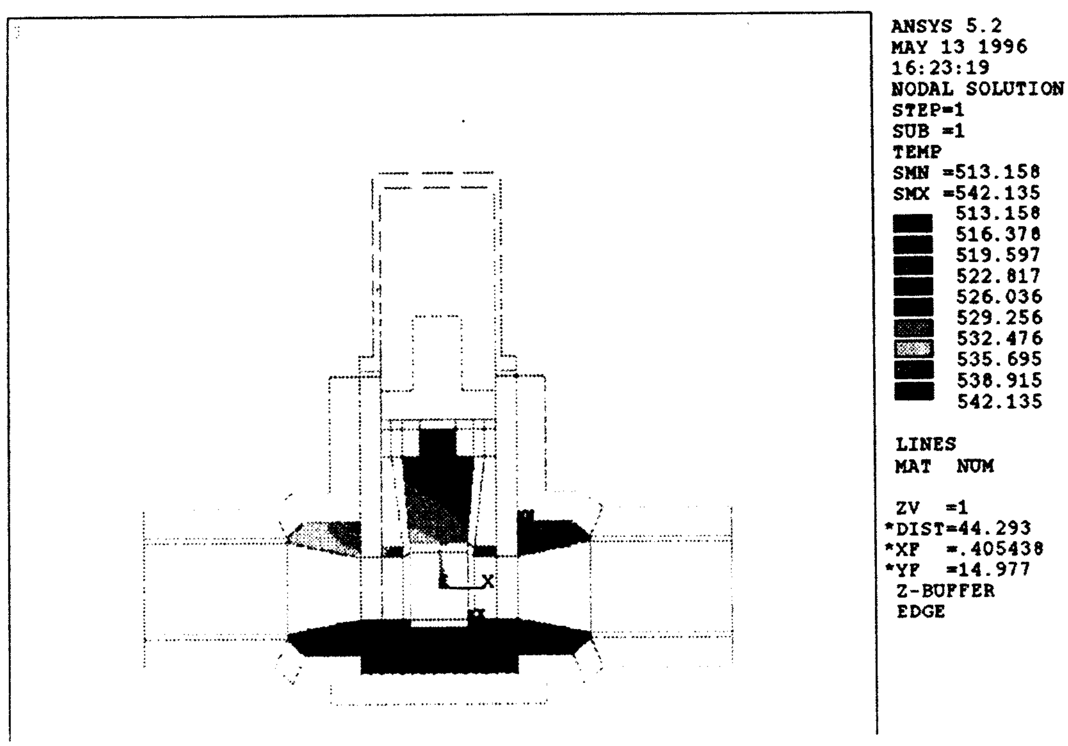

Figure 5B: CFD Analysis Results – Valve Temperature Distribution Showing Disk Cooler than Body

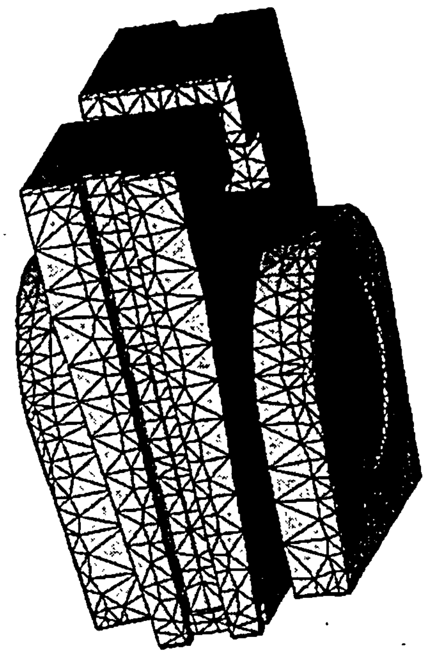

Figure 6: Finite Element Model for Optimization and Parametric Analysis of Disk

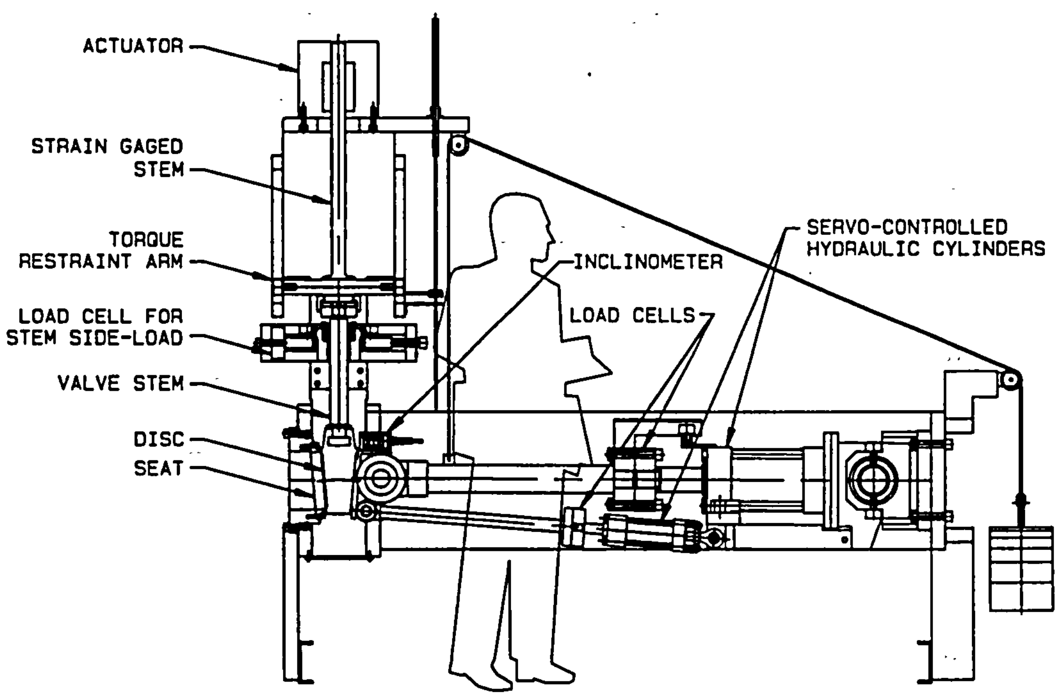

Figure 7: Gate Valve Design Effects Test Fixture

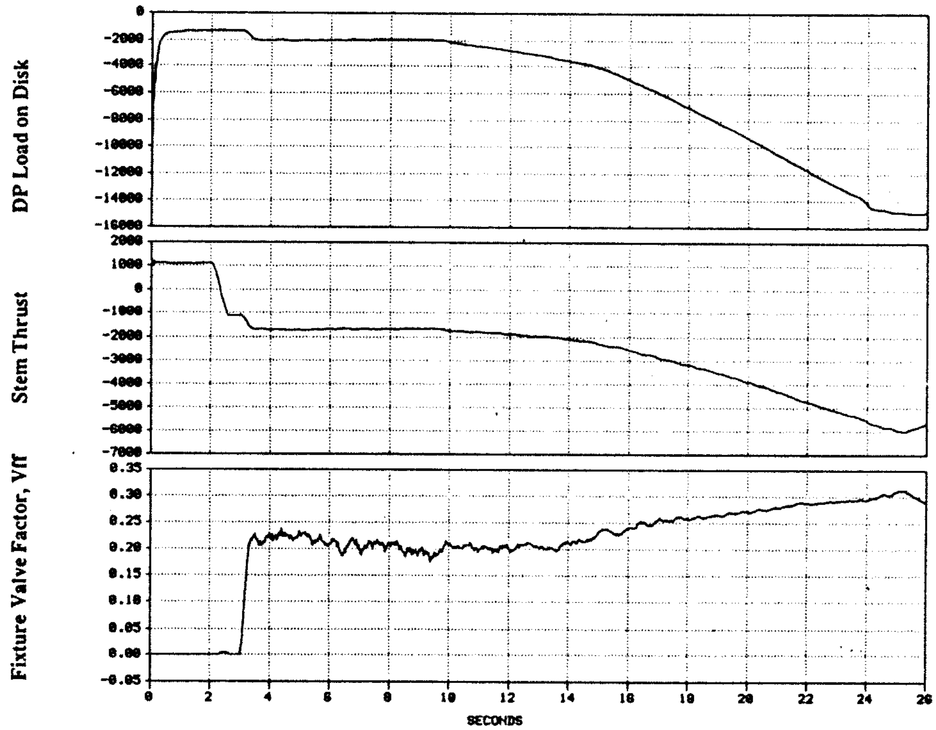

Figure 8: Typical Results from Closing Stroke of 6" x 4" x 6" Separate Effects Testing

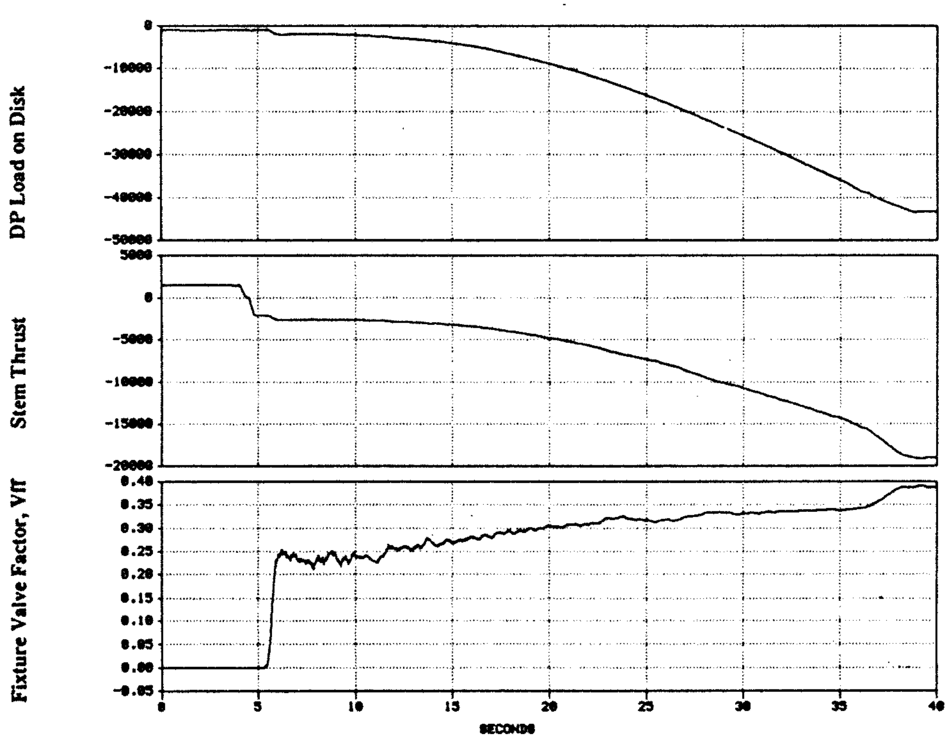

Figure 9: Typical Results from Closing Stroke of 10" x 8" x 10" Separate Effects Testing

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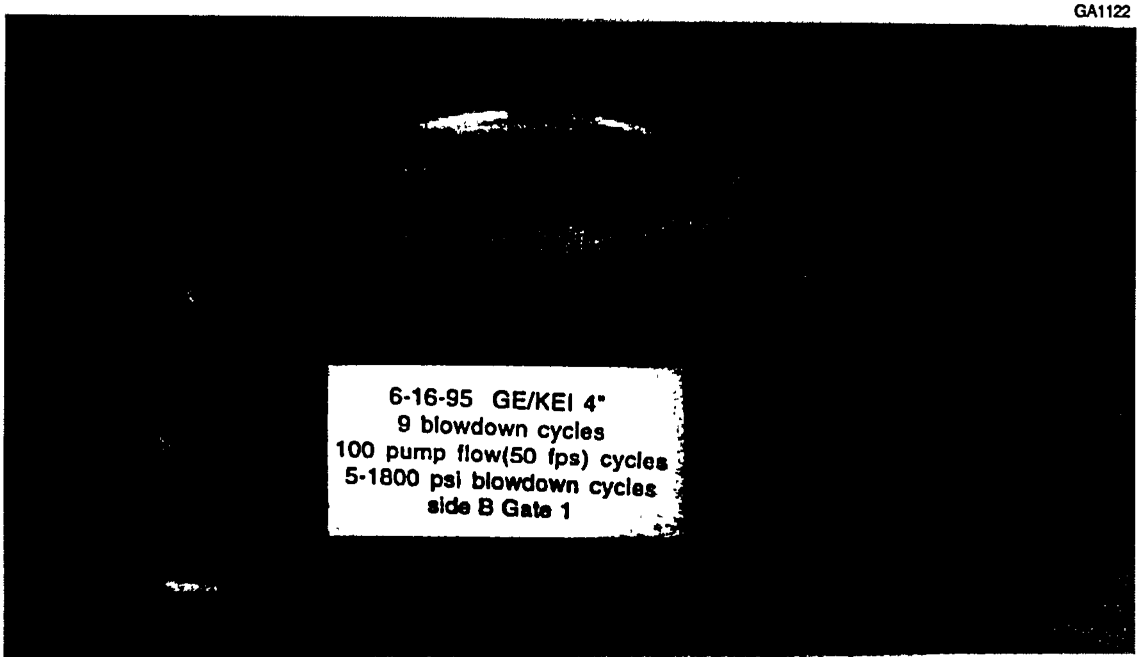

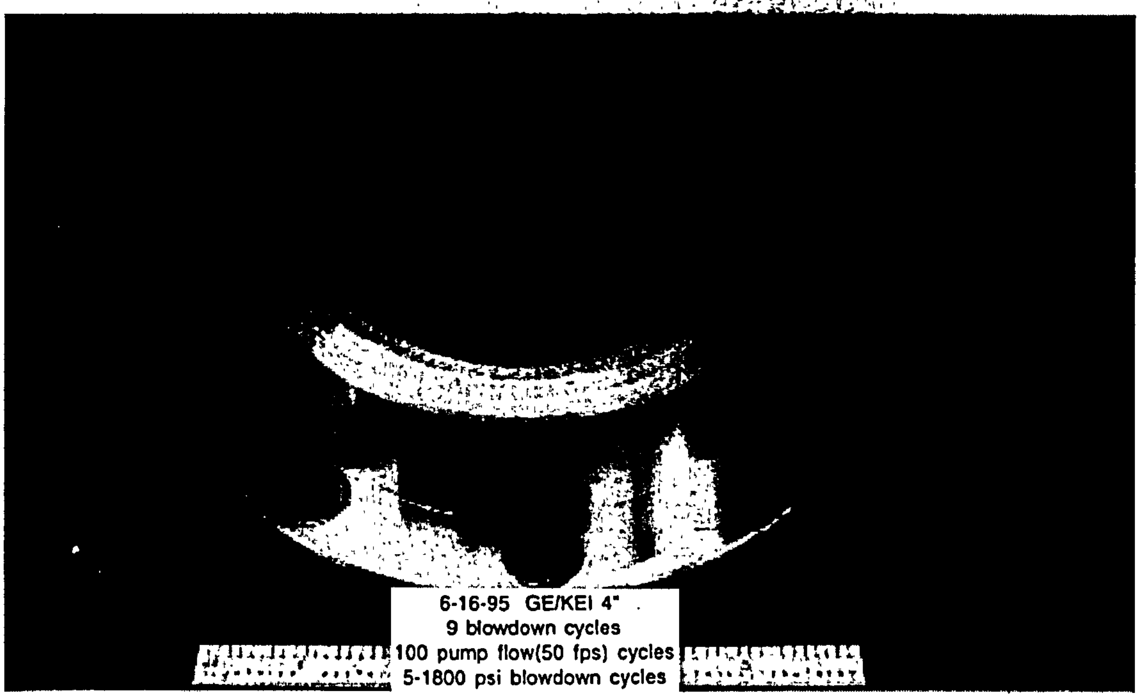

Figure 10: Excellent Condition of Disk and Seat Sliding Surfaces for 6" x 4" x 6" after 9 Blowdown Closing Strokes at 1,134 psi, 100 Close/Open 50 fps Pumped Flow Strokes at 1,134 psi, and 5 Blowdown Strokes at 1,800 psi

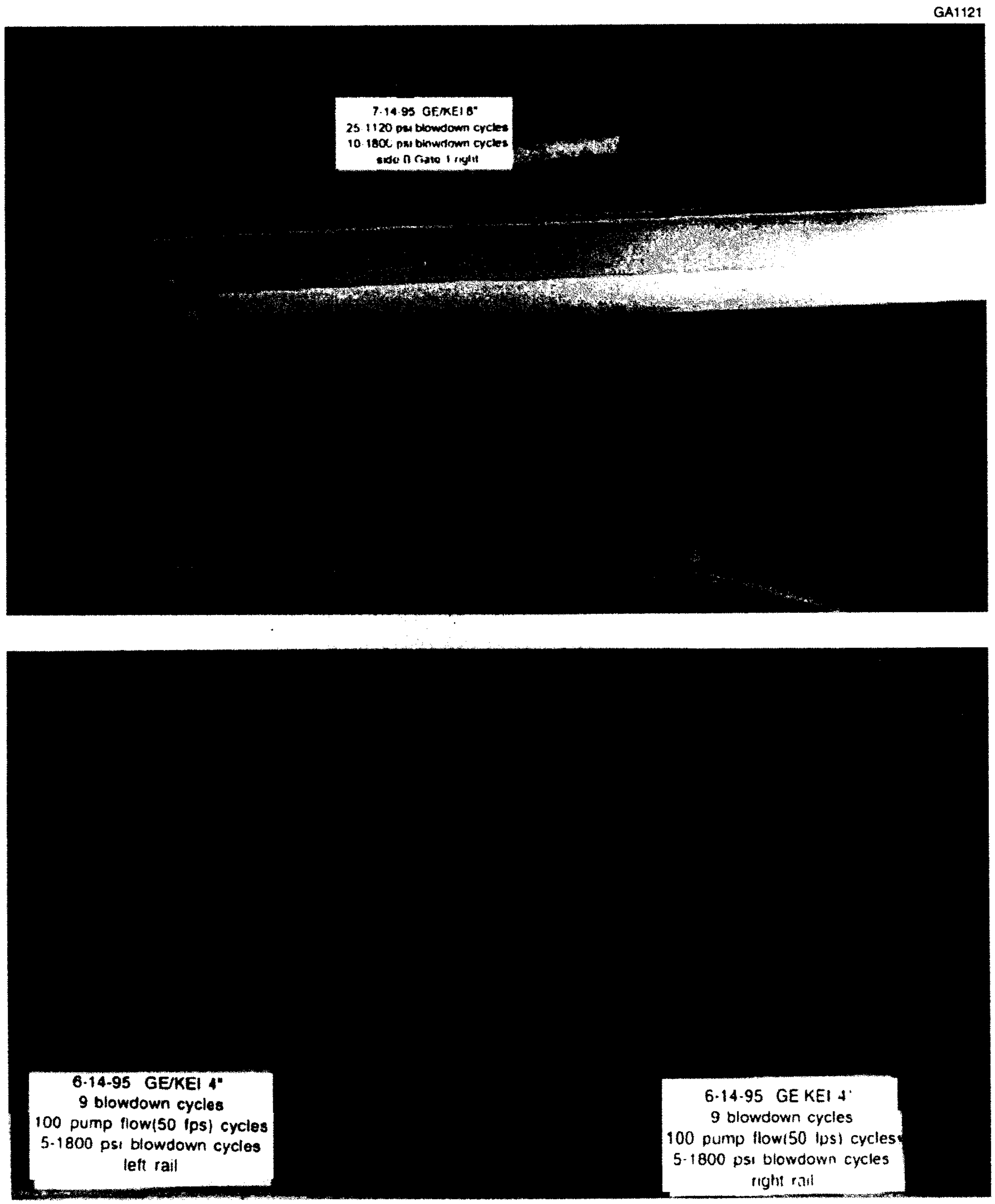

Figure 11: Excellent Condition of Disk Guide Slot and Guide Sliding Surfaces of 6" x 4" x 6" after 9 Blowdown Closing Strokes at 1,134 psi, 100 Close/Open 50 fps Pumped Flow Strokes at 1,134 psi, and 5 Blowdown Strokes at 1,800 psi

Stem Thrust Prediction Model for Westinghouse Wedge Gate Valves with Linkage Type Stem-to-Disk Connection

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ABSTRACT

The Electric Power Research Institute (EPRI) conducted a comprehensive research program with the objective of providing nuclear utilities with analytical methods to predict motor operated valve (MOV) performance under design basis conditions. This paper describes the stem thrust calculation model developed for evaluating the performance of one such valve, the Westinghouse* flexible wedge gate valve. These procedures account for the unique functional characteristics of this valve design. In addition, model results are compared to available flow loop and in situ test data as a basis for evaluating the performance of the valve model.

INTRODUCTION

The Westinghouse flexible wedge gate valve design employs a unique stem-to-disk linkage connection instead of the conventional T-head connection. This unique design causes the Westinghouse valves to behave differently than the conventional wedge gate designs. An analytical model for Westinghouse gate valves to predict the required valve stem thrust during opening and closing strokes was developed as part of the EPRI Motor-Operated Valve Performance Prediction Program (Ref. 1, a proprietary document not publicly available). The model was validated against measured stem thrust data from six valves ranging in size from 3" to 10" over a wide range of pressures, temperatures, and flow rates using water and steam as a fluid media. The methodology has been reviewed by the Nuclear Regulatory Commission (NRC) with detailed comments documented in Reference 2.

The model provides equations to determine the disk equilibrium at a given disk position using the appropriate friction coefficients and fluid flow-induced loads. Thrust calculations are performed at key disk travel positions that affect the maximum stem thrust predictions. Analysis shows that the disk equilibrium mode (flat or tipped) at a given key position can affect subsequent disk modes and required stem thrust during the remainder of the valve stroke. The unique stem-to-disk linkage imposes a lateral force on the disk, which tends to keep the disk in its existing contact mode (flat or tipped). The model also evaluates the edge contact loading on the guide and seat faces, stem buckling, and stem bending stresses caused by lateral forces exerted on the stem during the closing stroke. From this evaluation, criteria are provided to determine whether the stem predictions are valid or if the valve behavior is unpredictable. The model predictions show favorable agreement with the measured data. This paper summarizes the model description and comparison against test data. Further in-

* Westinghouse Corporation has not endorsed this paper, nor should this paper be considered an endorsement of Westinghouse Corporation products by the authors.

formation about the EPRI Motor-Operated Valve Performance Prediction Program may be obtained from references 3, 4, and 5. Stem thrust prediction models for other types of gate valves are described in reference 6 and in references 7 and 8 (proprietary documents not publicly available).

WESTINGHOUSE GATE VALVE DESCRIPTION

Design

The principal components of a Westinghouse flexible wedge gate valve are the body, stem and disk assembly, bonnet and stuffing box assembly, and yoke and torque arm. Aspects of this valve design that make it unique from other flexible wedge gate valve designs are the stem and disk assembly and the guide rails, as shown in Figure 1 (Ref. 9).

The stem and disk assembly includes the stem, double-pinned linkage, and flexible wedge. The bottom of the stem is a clevis-type stem head to which the linkage system is connected. The double-pinned linkage allows the disk to translate relative to the stem in a direction parallel to fluid flow. The upper portion of the disk contains a keystone-shaped slot which retains the bearing block of the stem-to-disk connection. The disk is a flexible, one-piece wedge with hardfaced sealing surfaces and guide slot surfaces.

Two guide rails are installed in parallel slots in the body cavity to guide the disk during the opening and closing strokes. The upper portion of the guide rails is wider than the lower portion and results in a smaller disk-to-guide clearance in the vicinity of the fully open position which keeps the disk from rattling under flow turbulence. The larger disk-to-guide clearance in the vicinity of the fully closed position allows the disk to contact the downstream seat earlier during the closing stroke than a valve with tighter clearance.

Operation

During the closing stroke of a Westinghouse gate valve, the stem lowers the disk assembly into the flow path. Fluid force pushes the disk against the guide rails initially and at some point in the stroke the disk/guide rail contact is replaced by disk/ seat contact. In the final stage of the closing stroke, the valve seal is created by a combination of differential pressure across the disk and the mechanical wedging of the disk against both the upstream and downstream seat rings.

The highest stem thrust required for the opening stroke is likely to be the unwedging force at the beginning of the stroke, especially under low differential pressure and high closing thrust conditions. After unwedging, the disk slides on the downstream seat ring until some point in mid-stroke where the guide rails pick up the disk and support the differential pressure load.

STEM THRUST PREDICTION MODEL

Scope

The valve model provides an analytical method for calculating the required stem thrust to open and close a Westinghouse gate valve with Stellite 156 hardfacing at the disk-to-seat interface, Stellite hardfacing on guide slots and 17-4 PH stainless steel material on guide rails. It is applicable to water and steam fluid flow in either direction. The model follows the sequence of a valve stroke to evaluate disk force equilibrium at selected key positions. It also examines the edge contact loading at the disk/guide and disk/seat interfaces, as well as stem axial and bending loads to determine whether a valid thrust prediction can be made.

Approach

Required stem thrust consists of five components: stem and disk assembly weight, packing friction, piston effect load, torque reaction friction, and differential pressure thrust (F_{DP}), which is usually the most significant component. The basic approach used in the model involves evaluating each term and then calculating the total required stem thrust by combining the terms. Except for the last term, F_{DP}, all of the terms are considered in

the same manner as the NMAC Application Guide (Ref. 10). Determination of the F_{DP} term for the Westinghouse flexible wedge gate valve model is different and more complex than the approach used in the NMAC Application Guide because of the unique stem and disk assembly design and because of disk tipping considerations.

Calculation of the F_{DP} component requires that the disk equilibrium for a given disk position be established using the proper friction coefficients and stem lateral force equations. An important feature of the unique stem/disk assembly design is that the disk equilibrium mode, whether flat or tipped, at any given stroke position can influence the disk equilibrium mode at subsequent stroke positions. Therefore, to make valid thrust predictions, the calculations for both closing or opening strokes need to be performed in a certain sequence. These calculations are performed only at the key disk travel positions that have been found to affect the maximum stem thrust predictions.

The key disk positions evaluated during a closing stroke are:

- early guide contact (65 percent open),
- last guide contact (before the disk begins to contact the seats),
- first seat contact, and
- flow isolation/initial wedging.

At the early guide contact position, although the F_{DP} is expected to be low, it is important to establish the disk equilibrium mode (flat or tipped) since it can affect the subsequent disk modes and thrust calculations for the closing stroke. The last guide contact can be the point of the maximum stem thrust if the pressure load and guide friction coefficients are high. The first seat contact is important in determining the disk contact mode for the remainder of the stroke in which the disk slides on the seat. The last key position, the flow isolation point, corresponds to the highest stem thrust in most cases. The thrust equations at this position were found to bound the thrust requirement for the initial wedging position; therefore, the initial wedging position does not require additional evaluation. The NMAC Application Guide may be used for sealing thrust calculations.

The key disk positions evaluated during an opening stroke are:

- unwedging,
- before flow initiation,
- maximum Bernoulli effect position (4 percent open), and
- first guide contact.

If the valve is wedged closed during the previous closing stroke, the unwedging thrust is usually the highest thrust in the opening stroke direction. It is calculated using EPRI developed equations for conventional flexible wedge gate valves. The second key disk position for the opening stroke will usually correspond to the highest stem thrust requirement if the wedging forces from the previous closure are small or negligible. After flow starts, the effect of the pressure distribution due to flow around the disk (called the "Bernoulli effect") can increase the required stem tensile thrust. To account for this increase in opening thrust, the stem thrust evaluation is done at the 4 percent open stroke position (where the Bernoulli effect has the maximum contribution) using the computational fluid dynamics results developed by EPRI for solid and flexible wedge gate valves. It should be noted that the 4-percent open position is the percentage of stroke from flow initiation to fully opened position, as defined in the EPRI methodology (Ref. 8, a proprietary document not publicly available). The last key position, i.e., the first guide contact, is evaluated since it can correspond to the maximum stem thrust requirements for conditions having high differential pressure (ΔP) and guide friction.

The model also evaluates the edge contact loading on the guide and seat faces, stem buckling, and stem bending stresses caused by lateral forces exerted on the stem during

the closing stroke by the unique disk-to-stem link design. The result of this evaluation determines whether the stem thrust predictions are valid or if the valve behavior is expected to be unpredictable.

Model Description

The valve model consists of equations to calculate the required stem thrust for closing and opening a Westinghouse flexible wedge gate valve. The equation for required stem thrust (F_R) can be written:

Closing stroke:

$$F_R = \frac{-F_W + F_{PACK} + F_P + F_{DP}}{TRF} \quad \text{(Eq. 1)}$$

Opening stroke:

$$F_R = \frac{F_W + F_{PACK} - F_P + F_{DP}}{TRF} \quad \text{(Eq. 2)}$$

where

F_R = required stem thrust load to operate valve (lb)

F_W = disk assembly and stem weight stem thrust (lb)

F_{PACK} = packing friction stem thrust (lb)

F_P = piston effect stem thrust (lb)

F_{DP} = differential pressure thrust (lb)

TRF = Torque reaction factor (dimensionless)

The individual terms in Equations 1 and 2 are described below.

Disk Assembly and Stem Weight (F_W)

This force is the component of the disk assembly and stem weight that produces an axial load in the stem. Typically, the weight of the disk assembly and stem has a negligible effect on the stem thrust needed to open or close the valve. However, the weight term can be important for large valves in low differential pressure applications.

Packing Friction Stem Thrust (F_{PACK})

Packing friction stem thrust is the load needed to slide the valve stem through the packing.

Piston Effect Stem Thrust (F_P)

Piston effect stem thrust is caused by the internal line pressure acting on the stem area. It is given by:

$$F_P = PA_{stem} = P\left(\frac{\pi}{4}\right)D_s^2 \quad \text{(Eq. 3)}$$

where

P = internal gage pressure at the stem head (psi) (i.e., bonnet pressure)

A_{stem} = stem area at packing (in^2)

D_s = diameter of stem at packing (in)

Torque Reaction Factor (TRF)

Torque reaction load is a function of required stem thrust; therefore, torque reaction depends upon the sum of the individual loads in the required stem thrust equation. The equation uses a torque reaction factor to account for the torque reaction load. For Westinghouse gate valves, torque in the stem is reacted by a torque arm. This torque reaction causes a friction load that opposes stem motion. Typically, torque reaction has a small effect on the overall stem thrust needed to open or close the valve.

Torque reaction factor is a dimensionless constant as follows:

$$TRF = 1 - \frac{\mu_t (FS)}{r_t} \quad \text{(Eq. 4)}$$

where

r_t = torque arm length (ft)

FS = stem factor (ft)

μ_t = friction coefficient for torque reaction surface

Differential Pressure Stem Thrust (F_{DP})

Differential pressure stem thrust is the required force applied to the disk by the stem in the stem-axis direction to move the disk under ΔP load. It depends upon the ΔP across the valve, the disk contact mode, and the friction coefficients for disk sliding surfaces. The calculation of F_{DP} requires detailed dimensional data to calculate internal geometric relations and reactions which are in turn used to determine disk contact mode and eventually F_{DP}. Equations needed to perform these calculations are provided in the EPRI Westinghouse Gate Valve Model Report (Ref. 1, a proprietary document not publicly available). The following paragraphs describe the iterative approach followed in these calculations for F_{DP}.

For Westinghouse gate valves, the following four disk contact modes are used for predicting stem thrust requirements:

- disk tipped on guides,
- disk flat on guides,
- disk tipped on seats, and
- disk flat on downstream seat.

The reasons for selecting these four disk modes are:

- For the majority of a valve stroke, the disk travels in either a flat or a tipped mode on the guides. The first two modes address the maximum stem thrust requirements for the portion of the stroke where the disk is riding on the guide surfaces. The guide rail design for the Westinghouse gate valve tapers from the wider section at the top to the narrower section at the bottom (Figure 1). Measurements of the valve dimensions show that the disk rides on the narrower section of the guide for a majority of the disk stroke, especially at stroke positions where the differential pressure load on the disk is significant. Therefore, only the narrow guide rail width is used in the analysis.
- For dimensional relationships present in the Westinghouse valve designs, if the disk acquires a tipped mode after leaving the guides, the disk is most likely to be restrained by contact against the upstream and downstream seats. This disk tipping mode is included in the analysis because it creates the most severe edge contact loading on the downstream seat.
- The disk-flat-on-downstream seat mode, near the fully closed position, in most cases determines the maximum stem thrust requirements since DP loading is highest at this disk position.

To predict the F_{DP}, it is necessary to follow the sequence of a stroke and perform calculations in that sequence to ensure that analysis results from a given disk position are appropriately reflected in calculations for subsequent disk positions in both opening and closing directions.

Detailed steps for calculating stem thrust in the closing and opening strokes are discussed below.

Closing stroke

Figure 2 is a flow chart showing the sequence of calculations to be performed for determining the differential pressure stem thrust for a closing stroke. (Figure 3 is for an opening stroke; discussed next.) The step-by-step calculational procedures along with key considerations are described below:

1. ***Determine ΔP versus stroke curve.***

   The ΔP versus stroke curve is needed to calculate the fluid loads on disk at each selected disk position. It can be determined using the actual test data or a detailed system flow analysis. In the absence of actual test data or detailed system flow analysis, a conservatively estimated ΔP versus stroke curve may be used with justification.

2. ***Select the first disk-on-guide position for starting the analysis.***

   The stem thrust calculations start at the 65-percent open position where the disk pressure load is expected to be low. The purpose of selecting this position is to determine the initial disk-to-guide

contact mode under relatively low ΔP loading on the disk. The disk is first assumed to be tipped-on-guide at this position. Disk moment calculations are performed to check this assumption and determine the valid contact mode (tipped or flat). In the force equilibrium calculation, if one of the reaction forces shows a tension contact force, then the assumed contact mode is not valid. Revising the contact mode assumption will provide the valid solution.

3. ***Determine stem thrust for the last guide contact.***

If Step 2 shows a flat disk contact mode, then the disk is likely to remain flat until it transitions from guide to seat contact. The reasons for the disk's remaining flat are: i) the fluid force acting on the disk is less likely to cause disk tipping as the disk moves towards the closed position, and ii) the unique design of the stem and disk assembly of the Westinghouse gate valve tends to maintain the current disk contact mode, as explained after the calculation steps. Therefore, only the last flat-on-guide contact position needs to be evaluated.

If Step 2 shows a tipped disk contact mode, at 65-percent open position, the disk may or may not change contact mode before it reaches the downstream seat contact position. Therefore, the stem thrust and disk equilibrium are calculated repeatedly in every 10-percent increment of disk stroke until the disk changes contact mode or until it contacts the downstream seat.

4. ***Determine stem thrust at the first disk-to-seat contact position.***

The disk contact mode at this position is initially assumed to be the same as that at the last guide contact position

If the disk was flat-on-guide at the last guide contact position, then the stem thrust at the first disk-to-seat contact position is calculated based on the flat contact mode. The equilibrium solution is converged if the contact forces are compressive. For the converged case, the next disk position is at flow isolation. Otherwise, the tipped mode at the first disk-on-seat contact should be analyzed. If the disk is found to be tipped-on-seat, the subsequent calculations are performed at increments of one-half of the distance between the current position and the flow isolation until the disk is within 5-percent open or the disk contact mode changes from tipped to flat. Once the disk mode is found to be flat-on-seat at any mid-travel position, it remains flat for the rest of the stroke.

5. ***Determine stem thrust at flow isolation.***

Once the fluid flow path is blocked by the disk, the pressure load is usually the highest with the maximum ΔP across the disk. The ΔP area is calculated based on the mean seat diameter. It also has stem lateral force pushing the disk against the downstream test, as shown in Figure 4. The required stem force calculated at flow isolation should bound the stem thrust from isolation to the wedged position.

The stem-to-disk link connection used in Westinghouse gate valves forces the disk to remain more stable, in both the tipped and untipped configurations, compared to the conventional gate valve designs that use a stem-to-disk connection consisting of a T-head and a T-slot. Figure 4 reveals this unique stem-to-disk behavior under a dynamic closing stroke. When the disk is flat on guides or downstream seat, the lower stem link pivot point is on the downstream side of the stem axis, while the upper stem link pivot point is on the opposite side of the stem axis. From the freebody diagrams shown in Figure 4, the disk is under a lateral force imposed by the stem link which pushes the top of disk toward the downstream seat. This lateral force tends to maintain the current flat mode. On the

other hand, if the disk is tipped on guides or seats, the lower stem link pivot point will be on the upstream side of the stem axis, resulting in a stem link lateral force that pushes the top of disk towards the upstream seat. Which reinforces the tipping mode.

For the tipped-on-guide or tipped-on-seat contact mode calculations, edge load limits were evaluated using the criteria developed for EPRI MOV performance prediction methodology for gate valves (Ref. 8, a proprietary document not publicly available). Under severe edge loading conditions, stem thrust calculations may be unreliable due to the unpredictable behavior of the highly loaded disk and seat edges.

Another source of unpredictable stem thrust behavior is stem overstressing or buckling caused by a combination of axial and lateral stem forces, as shown in Figure 4. It should be noted that the calculated stem stress may be conservative when the bounding friction coefficient values are used in stem thrust calculations.

The pressure load on disk at flow isolation is calculated using the mean seat diameter area multiplied by the disk ΔP. For some Westinghouse gate valves that use wide seat ring faces, the effective ΔP area can be increased between the flow isolation and initial wedging positions. The effect was analyzed in detail and a simple bounding method was developed to account this pressure area increase, as shown in Figure 5. The method provides a criterion for evaluating sealing ring dimensions to determine whether a pressure area increase will occur for a given valve design.

Opening stroke

Figure 3 is a flow chart showing the sequence of calculations to be performed for determining the differential pressure stem thrust for an opening stroke. The step-by-step calculational procedures, along with key considerations, for the opening stroke are described below:

1. ***Determine differential pressure (ΔP) versus stroke curve.***

   Same as the closing stroke.

2. ***Determine stem thrust before the flow initiation point.***

   Before flow initiation, ΔP load is usually a constant under the maximum ΔP. The stem lateral force on disk varies from zero after unwedging to a higher value at flow initiation. Because the stem lateral force is pulling the disk away from the downstream seat, it reduces the F_{DP}. Therefore, the bounding value for F_{DP} before flow initiation is calculated with zero stem lateral force.

   Stem lateral force in the opening stroke direction is smaller than the force encountered in the closing direction. The lateral force is smaller due to the fact that the entire disk-to-stem link assembly is on one side of the stem axis, resulting in a smaller link tipping angle and smaller stem deflection.

   For certain wide seat ring valves, the actual ΔP area may be larger than the mean seat ΔP area (as discussed in closing stroke and shown in Figure 5). This effect is also considered in the opening stroke calculations.

3. ***Determine stem thrust at the maximum Bernoulli effect position (4-percent open).***

   Before the stem thrust calculations, the transition point from flat-on-seat to flat-on-guide contact should be determined to see if it occurs before 4-percent disk open. For the valves that transition before reaching the 4-percent disk open position, the stem thrust should be calculated based on flat-on-guide equations; otherwise, the flat-on-seat equations should be used.

   The 4-percent disk open position is selected to capture the maximum Bernoulli effect on the stem thrust during an opening stroke. Computational fluid dynamic analysis show that Bernoulli

effect increases rapidly from flow initiation to 4-percent disk open. After 4-percent open, the increase is small and the ΔP drop is more than enough to compensate for it. Therefore, the 4-percent disk position is selected as one of the key positions for stem thrust calculation.

If the solution for the flat contact mode is not converged, then the transition point based on tipped disk case should be calculated and the stem thrust based on tipped-on-guide or tipped-on-seat mode at 4-percent disk open should be calculated.

4. ***Determine stem thrust for the first guide contact.***

If the transition point is less than 4-percent disk open, then the first guide contact (flat or tipped) is already calculated at the 4-percent disk open. For valves which transition at greater than 4-percent disk open, the stem thrust at the first guide contact is calculated at the transition point. A review of the test data as well as analysis results shows that the stem thrust for the guide contact modes can be higher than the stem thrust for the seat contact modes during an opening stroke.

Selection of friction coefficients

Friction coefficients for use in the model can range from 0.15 to 0.6, depending on material combination, fluid media, bearing stresses, temperatures, and contact modes (Ref. 8, a proprietary document not publicly available). Bounding friction coefficients were used to perform stem thrust calculations. To provide some potential relief from the conservative bounding friction coefficients, in situ test data may be used to determine the disk-to-seat flat-on-flat friction coefficient.

Westinghouse valve information indicates that the material pairs at the guide slot versus guide rail contact surfaces are Stellite versus 17-4 PH stainless steel. The friction coefficient for Stellite versus 316 stainless steel was used instead because this is the closest material pair for which test data are available. For determining the friction coefficients for the flat contact cases, the contact stress is calculated as an average stress on the guide or seat faces.

VALIDATION AGAINST TEST DATA

The model has been validated by comparing model predictions for stem thrust against data obtained from flow loop and in situ tests on Westinghouse gate valves. Flow loop tests were performed on a 3-inch, 1500-pound valve under ambient water flow at several levels of DP up to 732 psi in closing strokes and 2,395 psi in opening strokes. In situ tests were performed on two 3-inch, 1525-pound valves under high temperature steam blowdown flow from a pressurizer to a relief tank and on 4-inch, 1525-pound; 8-inch, 316-pound; and 10-inch, 1525-pound valves under ambient temperature pump flow conditions. A typical closed-to-open-to-closed stem thrust trace from an in situ test on a 3-inch, 1,525-pound valve is shown in Figure 6. All valves had stainless steel bodies and Stellite hard-facing on both the disk and seat sealing faces. The disk guide slots were also hard-faced with Stellite. Gate valve internal specifications were provided by Westinghouse.

For a particular valve and test condition, the stem thrust prediction procedure was followed to determine the predicted stem thrust for key points of the valve opening/closing strokes. Bounding values of friction coefficients for given contact surface conditions and material combinations were used. Test results were compared with these bounding values to determine whether the model equations are capable of accurately predicting actual valve behavior. Figures 7 and 8 contain predicted and observed stem thrusts for the closing and opening strokes based on one flow loop test valve and four in situ test valves as described above. (Data from only one of the two 3-inch in situ test valves was used because results were very similar.)

Figure 7 contains thrust values before wedging for the closing stroke (Figure 6, Point H) and after unwedging for the opening stroke (Figure 6, Point C). Figure 8 contains thrust values at isolation for the closing stroke (Figure 6, Point M) and before flow initiation for the opening stroke (Figure 6, Point F).

Results of the comparisons indicate that the model is a satisfactory predictor of the stem thrust required to open and close Westinghouse gate valves. An analysis of the detailed stem thrust traces shows that the model predictions bound all of the test data at each key disk position, and the maximum predicted stem thrust bounds the maximum measured thrust for each test stroke. The comparisons also show a significant scatter of friction coefficients among test valves.

Closing Stroke Results

For closing strokes, the predictions indicate that the disk is in the flat contact mode during mid-travel for all of the test cases. The calculated stem thrust at flow isolation bounds the maximum stem thrust data obtained from testing (Figure 8).

The average ratio of measured to predicted thrust is 0.65. Ratios vary from 0.40 to 0.99, indicating a significant scatter of friction coefficients among the test valves. However, the predictions based on the friction coefficients given in the model bound all test data.

Calculations show that the stem thrust is unpredictable under maximum friction conditions for the 4-inch, 1525-pound in situ test valve. The stem lateral force calculations indicate stem buckling/plastic deformation under excessive stem lateral force. Test results indicate that the actual friction coefficient is much less than the bounding friction value; therefore, stem damage is neither predicted nor observed under actual test conditions.

Opening Stroke Results

For opening strokes, the predictions indicate that the disk is in the flat contact mode in the mid-travel position for all test cases. Calculated stem thrust results show that the highest thrust after unwedging can occur before or after flow initiation, depending on the applicable friction coefficients for the seat and the guide surfaces. Test results show that the highest stem thrust usually occurs before flow initiation; however, for the 8-inch in situ test valve, the highest stem thrust occurred at the first guide contact. These results are in agreement with the model predictions.

The average ratio of maximum measured to maximum predicted thrust is 0.61 based on the bounding friction coefficients. The ratio of maximum measured to maximum predicted thrust ranged from 0.22 to 0.96, indicating a significant scatter of friction coefficients among the test valves. However, all of the model predictions bound the test data (Figure 8).

None of the five opening strokes revealed any significant Bernoulli effect. However, the Bernoulli effect component is still kept in the model to ensure model applicability to high flow conditions where this effect may be more pronounced.

CONCLUSIONS

Based on comparison of the model to available test data, it is concluded that the stem thrust prediction model is a satisfactory predictor of Westinghouse flexible wedge gate valve behavior. For closing strokes, the observed stem thrust to predicted stem thrust ratio ranges from 0.40 to 0.99. For opening strokes, the observed to predicted stem thrust ratio ranges from 0.22 to 0.96.

It is concluded that the stem thrust prediction model can provide bounding values for the required stem thrust. Calculations utilizing disk-to-seat friction coefficients determined from in situ valve-specific testing may provide the user some potential relief. The actual friction coefficient for a specific valve may be obtained from valve tests as described in Section 8 of Reference 11 (a proprietary document not publicly available) for the

EPRI solid and flexible wedge gate valve methodology because the friction coefficient is calculated based on stem thrust near the wedged position where the stem lateral force is small for both Westinghouse and other wedge gate valves. However, the user should consider instrument accuracy and the possibility of thrust increase with time and valve stroking. In addition to the stem thrust predictions, the model provides a methodology for evaluating the stem strength under closing stroke, and disk/guide as well as disk/seat edge contact load limitation under tipped contact modes to ensure that the model predictions are complete.

Implementation of Methodology

For each specific valve being evaluated, data needed for the evaluation includes:

- Disk assembly and stem weight
- Packing friction loads
- Stem diameter, material, and stem thread geometry
- Torque arm length
- Fluid medium, temperature
- Differential pressure, upstream pressure, and DP vs. stroke curve
- Valve internal dimensions.

Once this information is obtained, the methodology can be used to determine the stem thrust required to open/close a Westinghouse flexible wedge gate valve. The result of the method is a value for required stem thrust to actuate the valve under the desired conditions. The actuator needs to be evaluated separately to determine its capability to supply thrust to the valve.

ACKNOWLEDGMENT

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* References 1, 7, 8, and 11 are part of the EPRI MOV PPM program. For more information on these proprietary reports, the reader should contact EPRI, 3412 Hillview Ave., Palo Alto, CA 94304, (415) 855-2514.

9. *Westinghouse Instruction Book for Motor-Operated Gate Valves, Manually Operated Gate Valves, and Swing Check Valves*, Document No. 5710-99, Rev. 2, Westinghouse Corporation publication.

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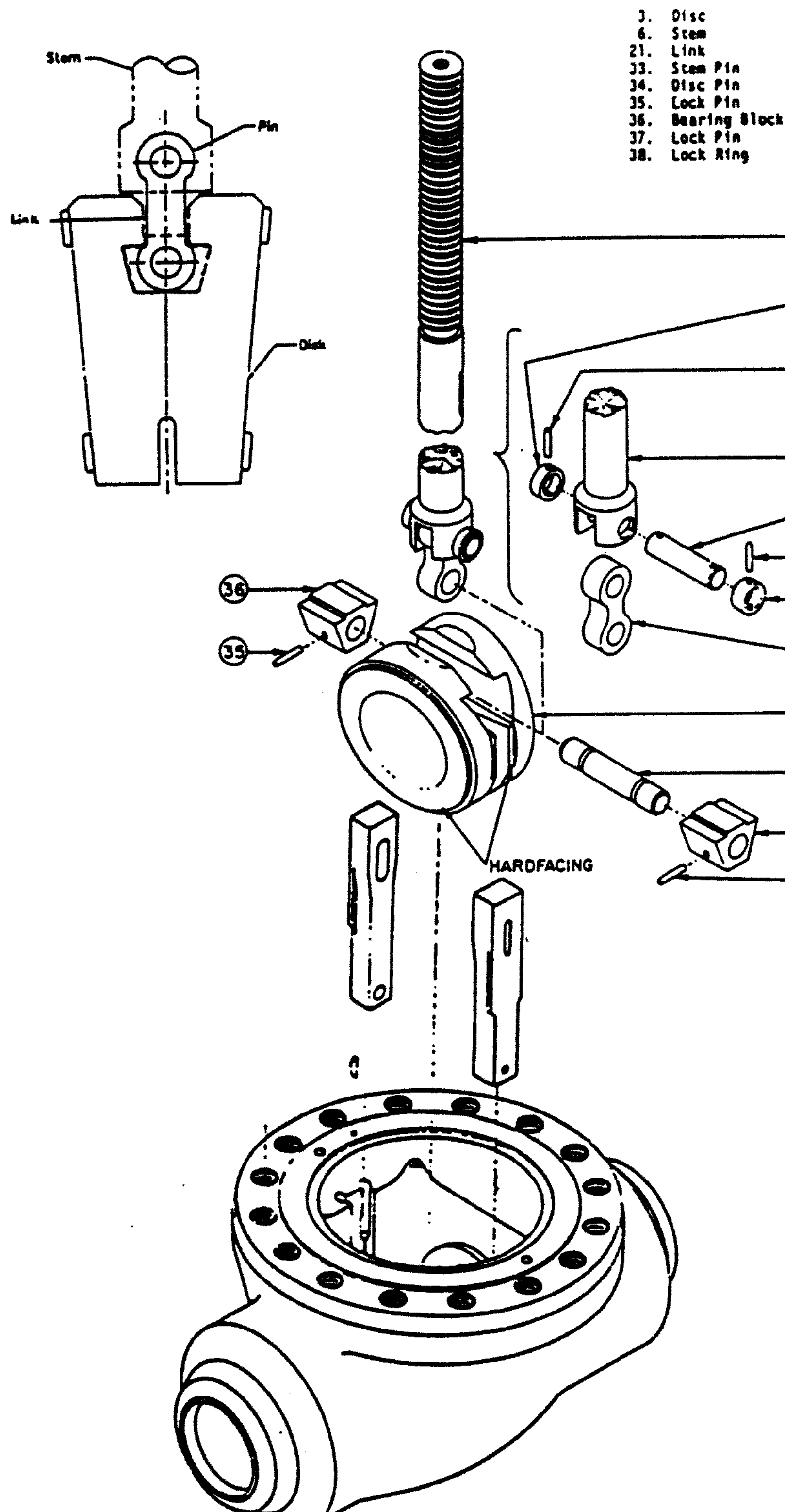


Figure 1

Key Components of a Westinghouse Flexible Wedge Gate Valve

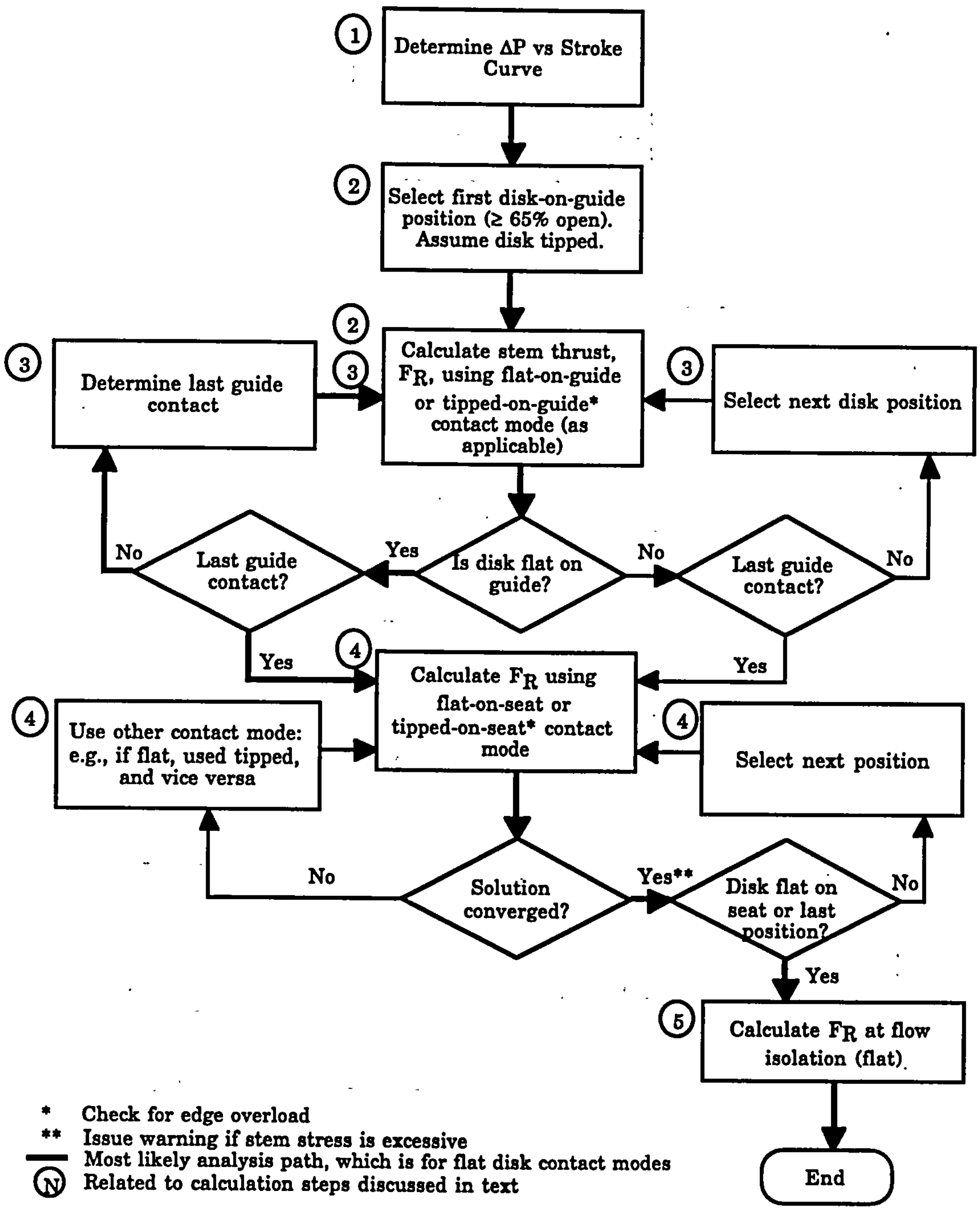


Figure 2

Flow Chart for Differential Pressure Thrust Calculations – Closing Stroke

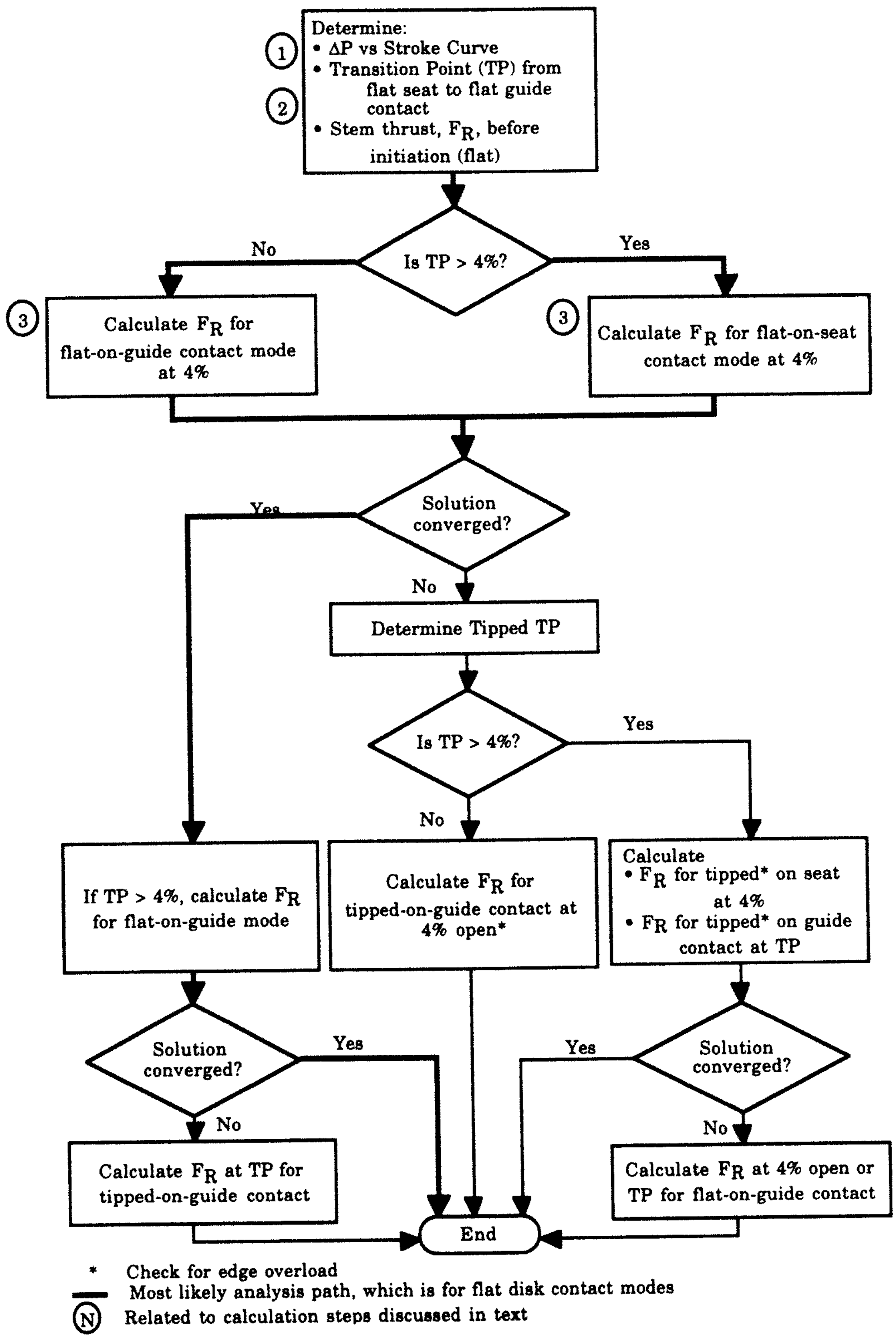


Figure 3
Flow Chart for Differential Pressure Thrust Calculations - Opening Stroke

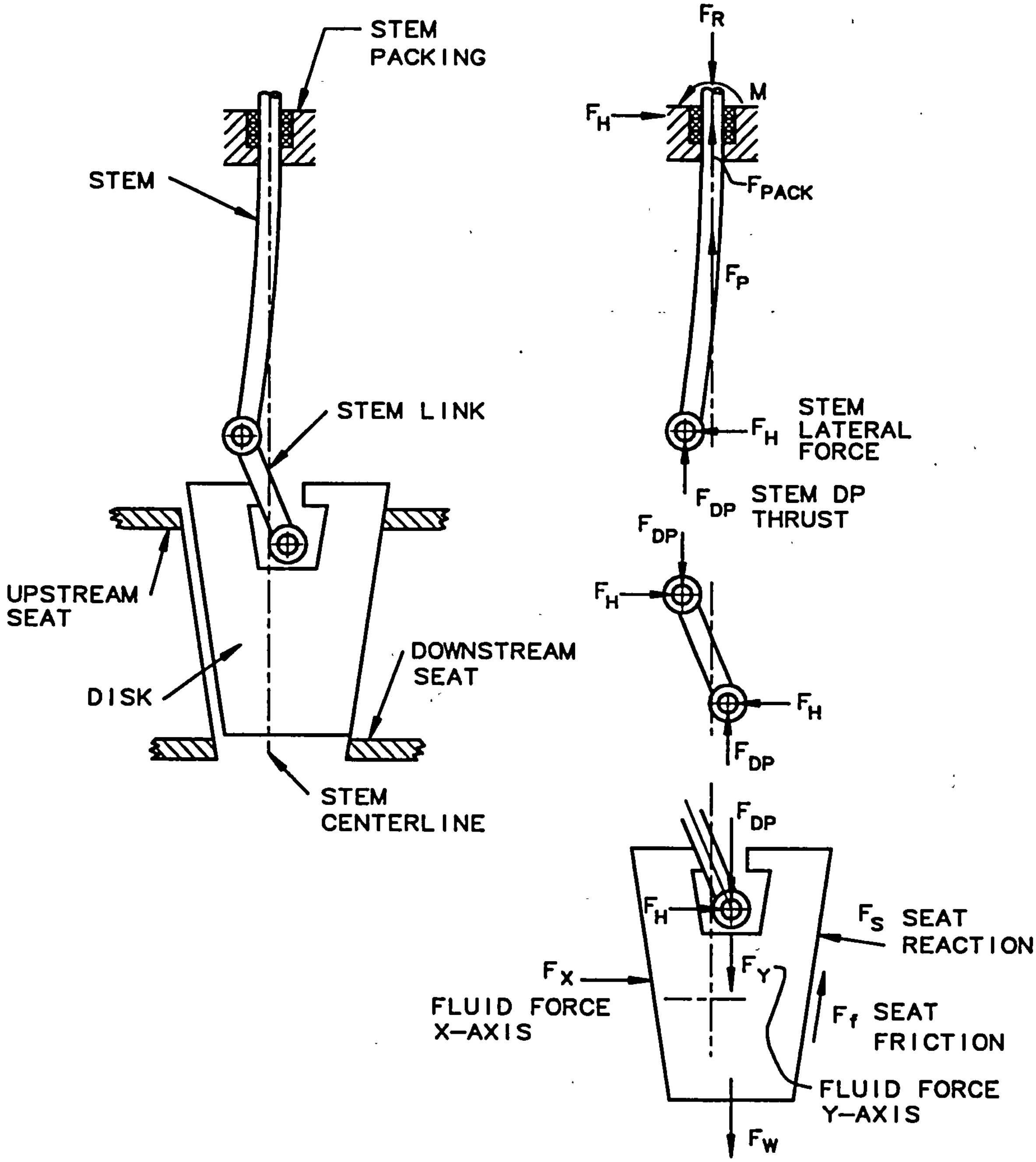
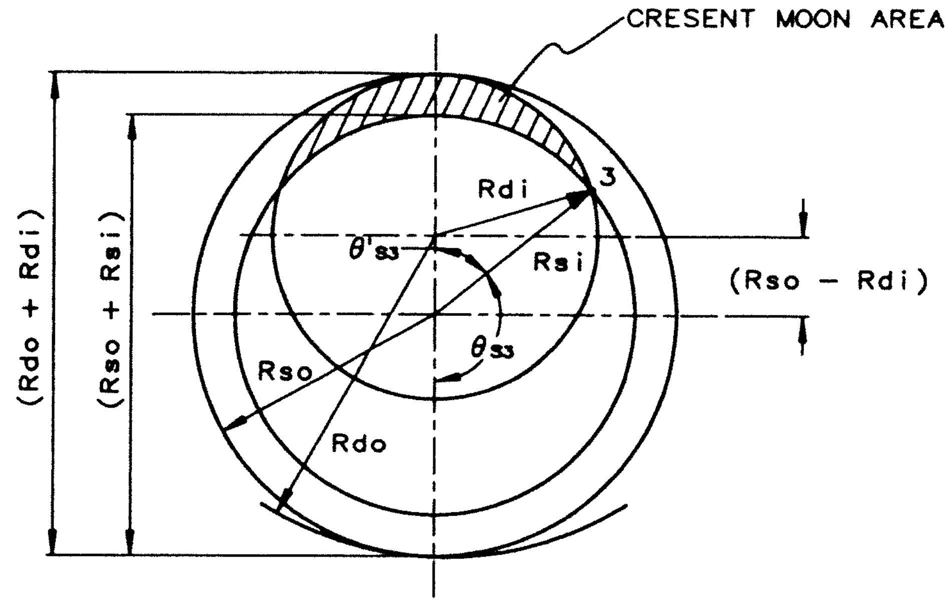


Figure 4

Stem, Link, and Disk Force Equilibrium During Dynamic Closing Stroke – Disk-on-Seat Contact



The simplified bounding pressure load calculations can be summarized as follows:

1. If $\left(R_{do} + R_{di}\right) \le \left(R_{so} + R_{si}\right)$

$$K_{area} = 1$$
$$F_P = \pi R_m^2 \Delta P K_{area}$$

where R_m = mean seat radius (in)

ΔP = differential pressure across the valve (psi)

K_{area} = constant for ΔP area adjustment

2. If $\left(R_{do} + R_{di}\right) > \left(R_{so} + R_{si}\right)$

$$K_{area} = \frac{\left(\pi - \theta'_{s3}\right)\left(R_{so} + R_{si}\right)^2 + \theta'_{s3}\left(3 R_{so}^2 + R_{si}^2\right)}{\pi\left(R_{so} + R_{si}\right)^2}$$
$$F_P = \pi R_m^2 \Delta P K_{area}$$

where $$\theta'_{s3} = \cos^{-1}\left[\frac{\left(R_{so} - R_{di}\right)^2 + R_{si}^2 - R_{di}^2}{2\left(R_{so} - R_{di}\right) R_{si}}\right] \text{ (rad)}$$

R_{so} = seat surface outside radius (in)

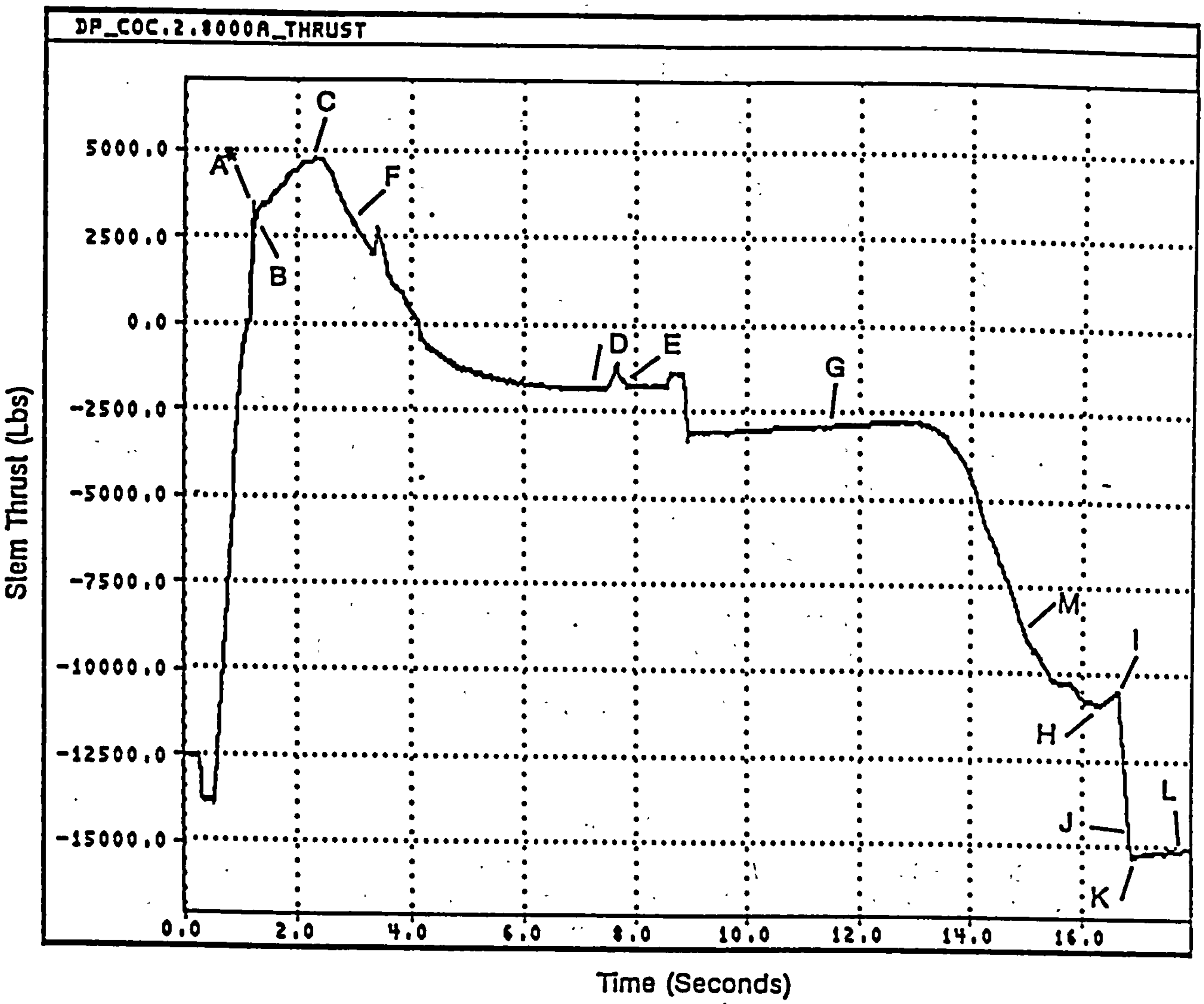
R_{si} = seat surface inside radius (in)

R_{do} = disk surface overlay outside radius (in)

R_{di} = disk surface overlay inside radius (in)

Figure 5

Simplified Bounding Area Calculation for Differential Pressure Load



* Stroke Positions:

A At cracking
B Just after cracking
C Maximum after cracking
D Running
E Limit switch trip
F At flow initiation
G Running
H Maximum prior to wedging
I At initial wedging
J Limit switch trip
K Maximum after wedging
L Final
M At flow isolation

Figure 6

Typical Closed-to-Open-to-Closed Stem Thrust for a 3-Inch, 1,525-Pound In Situ Test Valve

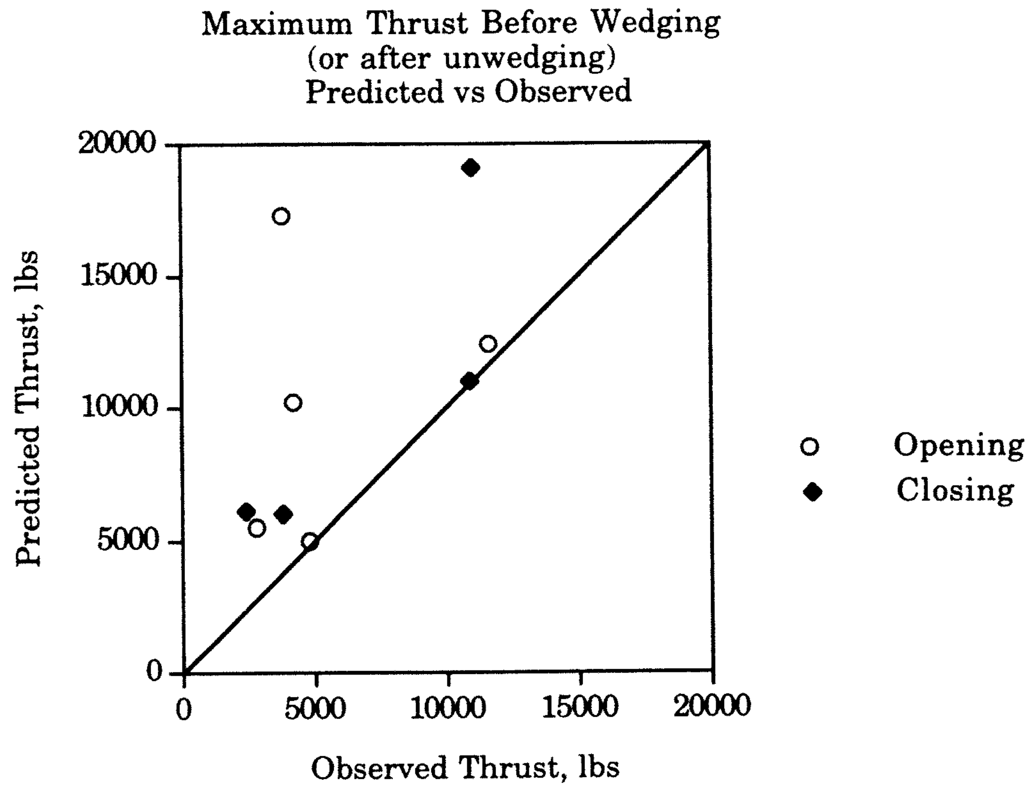


Figure 7

Predicted and Observed Stem Thrust Values Before Wedging (Closing Stroke) or After Unwedging (Opening Stroke) for Valves Ranging from 3 to 10 Inches in Size and 316 to 1,525 Pounds in Pressure Rating

Note: The large differences between prediction and test data are due to variations in actual friction coefficients, which are bounded by the friction coefficients used in the model.

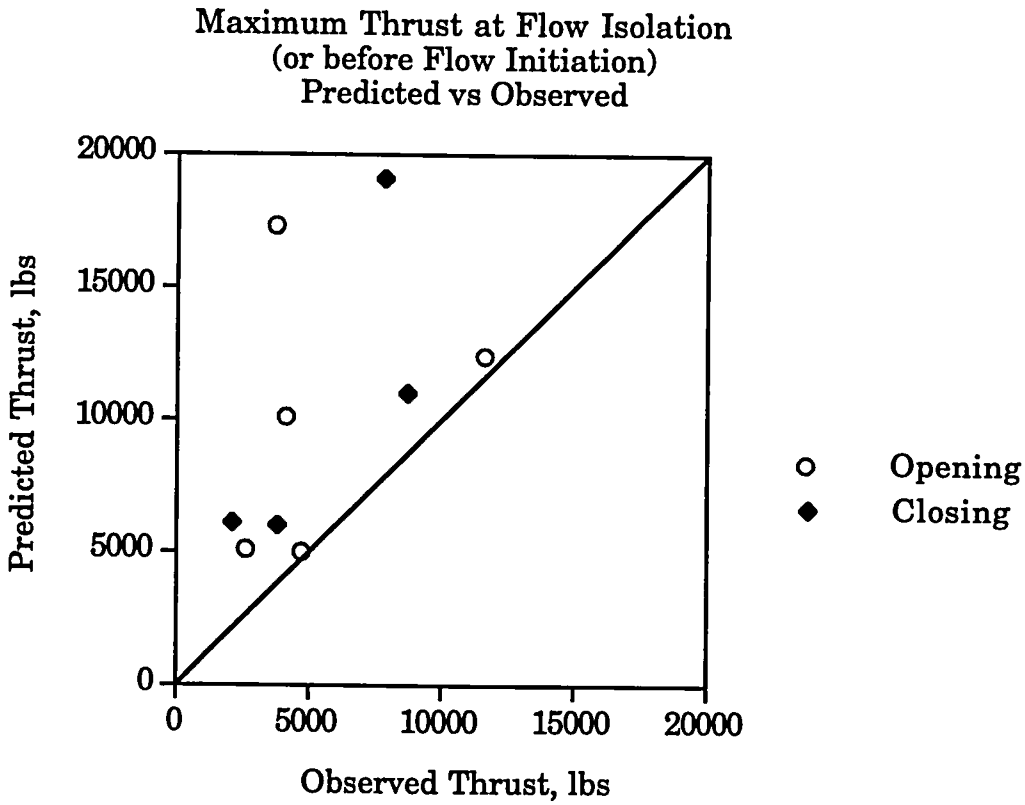


Figure 8

Predicted and Observed Stem Thrust Values at Flow Isolation (Closing Stroke) or Before Flow Initiation (Opening Stroke) for Valves Ranging from 3 to 10 Inches in Size and 316 to 1,525 Pounds in Pressure Rating

Note: The large differences between prediction and test data are due to variations in actual friction coefficients, which are bounded by the friction coefficients used in the model.

Stem Thrust Prediction Model for W-K-M Double Wedge Parallel Expanding Gate Valves

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ABSTRACT

An analytical model for determining the required valve stem thrust during opening and closing strokes of W-K-M parallel expanding gate valves was developed as part of the EPRI Motor-Operated Valve Performance Prediction Methodology (EPRI MOV PPM) Program. The model was validated against measured stem thrust data obtained from in-situ testing of three W-K-M valves. Model predictions show favorable, bounding agreement with the measured data for valves with Stellite 6 hardfacing on the disks and seat rings for water flow in the preferred flow direction (gate downstream). The maximum required thrust to open and to close the valve (excluding wedging and unwedging forces) occurs at a slightly open position and not at the fully closed position. In the nonpreferred flow direction, the model shows that premature wedging can occur during ΔP closure strokes even when the coefficients of friction at different sliding surfaces are within the typical range. This paper summarizes the model description and comparison against test data.

INTRODUCTION

The Electric Power Research Institute (EPRI) carried out a comprehensive research program with the objective of providing nuclear utilities with analytical methods to predict the force and torque requirements to operate gate, globe, and butterfly motor-operated valves (MOVs) under design basis conditions (see References 1 through 5). The EPRI program includes a combination of analyses and tests which address several gate, globe, and butterfly valve designs. The W-K-M gate valve model presented in this paper is one of several gate valve designs modeled within the MOV PPM and is documented in Reference 6.

The W-K-M gate valve is a double wedge, parallel expanding gate valve used in both water and steam applications. This design is commonly known as the W-K-M valve design and is manufactured by Cooper Cameron Corporation*. Although the methodology was developed for nuclear power plant applications, it can also be applied to non-nuclear applications (such as oil and gas, petrochemical, pipeline) where the population of W-K-M is much larger than in nuclear power plants. The W-K-M valve has two independent disks which mate against parallel seats, and has an internal wedge mechanism to solidly seat the disks against the body seat rings.

As part of the EPRI program, tests were performed on three W-K-M valves. The test results were used to validate the W-K-M valve model. This paper summarizes the analytical model and the comparison of model predictions to test data.

* Cooper Cameron Corporation has not endorsed this paper. This paper should not be considered as endorsement of Cooper Cameron Corporation/ WKM Valve Company products by the authors.

The Nuclear Regulatory Commission (NRC) has reviewed the methodology and provided comments. EPRI's response to the those comments has been prepared and submitted to NRC for review. NRC has not issued a Safety Evaluation Report (SER) to endorse this methodology at the time of submitting this paper for publication. Thrust requirements for W-K-M gate valves should be based on the original methodology report (Reference 6) and the NRC SER, not on the contents of this paper alone.

DESCRIPTION OF DOUBLE WEDGE PARALLEL EXPANDING GATE VALVE

Configuration

The internal components and nomenclature of the W-K-M parallel expanding gate valve are shown in Figure 1. Figure 2 shows four key positions during an opening stroke to illustrate the operation of the unique mechanism used to achieve wedging at either end of the stroke and to prevent wedging at mid-stroke. Another significant design feature of this valve is that the disk assembly (consisting of gate, segment, and leverlock mechanism) is of a through conduit design, and the disk assembly is twice as long as the disk used in conventional wedge gate valves. A circular hole through the lower portion of the disk assembly lines up with the flow area of the pipe when the valve is in the fully open position.

The valve seat faces are relatively wide (0.5 inch for a 6-inch valve and 1.25 inches for a 16-inch valve). The gate and segment have a figure-eight (8) Stellite 6 overlay as shown in Figure 3. The gate/segment overlays fit over the valve seats in the fully open and fully closed positions. The sealing surfaces on the valve seats are also hardfaced with Stellite 6. In the fully collapsed position, the maximum clearance between the gate/segment overlays and valve seats is less than about 0.060-inch for 16-inch and smaller valves.

The valve is equipped with two body-mounted skirts; one on the gate side of the body (called gate skirt) and one on the segment side of the body (called segment skirt). These skirts are thin plates with holes that fit over the seat outside diameter. Clearance between the skirts and the gate/segment overlays in the fully expanded position is less than 0.100-inch.

At a position slightly above full isolation, the small clearance between the skirts and gate/segment forms a passage for secondary flow. This secondary flow increases the effective differential pressure area because the skirts act essentially as extensions to the seat faces. The increase in differential pressure area increases the required thrust to stroke the valve before primary flow initiation and after primary flow isolation.

Operation

The parallel expanding gate valve design employs two wedge-type pieces (gate and segment). Whenever relative axial motion (parallel to the stem) occurs between the gate and segment, they expand in the lateral direction (normal to the flow direction) to wedge the gate assembly between the seats. The valve stem has a T-head that engages a T-slot in the gate but does not interfere with segment movement. This design requires that the two pieces travel together as an assembly without relative axial motion (or lateral expansion) during the entire stroke except at the end of travel when the assembly approaches the fully open or fully closed position. To keep the gate and segment from moving relative to each other, a special mechanism (called Leverlock*, shown in Figure 2) is employed. The Leverlock mechanism consists of a "shoe" attached to a pivoting arm mounted on the gate. The arm is equipped with a cam which engages a slot in the segment. Guide rails are fastened to skirts which fit around each seat. In mid-travel, the Leverlock shoe rides between the guide rails. This keeps the arm in a fixed orientation and prevents any relative axial motion between the gate and segment. Lack of axial motion precludes lateral expansion or mid-travel wedging of the gate and segment, which could otherwise increase the force required to open or close the valve.

At either end of the stroke, the guide rails are configured to permit the Leverlock shoe to

* Leverlock is a trademark of Cooper Cameron Corporation.

move outside the parallel restraint provided by the rails. In the opening direction, the segment stops moving when it hits the bottom of the bonnet at the end of the opening stroke. As the stem continues to pull on the gate upward, the relative axial motion between the gate and segment causes the Leverlock arm to rotate and "kick" the shoe to the left towards the gate on the downstream side of the valve. Since the arm and the shoe are now free to rotate, additional stem force causes a wedging/climbing action that forces the gate and segment to expand away from each other. This expansion also causes the sealing surfaces of the gate and segment to be firmly pressed against their respective valve seats. It should be noted that when the limit switch is used to control the opening stroke the segment may not hit the segment stop at the bottom of the bonnet, and full expansion of the gate and segment will not occur in the fully open position.

When the actuator receives the signal to close, the gate and segment assembly start to move downward with the shoe still in the kicked-out position (far left) until the shoe hits the chamfer at the top of the guide rail (see Figure 2). The chamfer forces the shoe from the extreme left position to a central position between the guide rails. Subsequently, the gate and segment assembly starts to travel through the entire stroke until the very end, when the segment stop hits a stop pad located in the upstream conduit of the valve body. After the segment stops, the stem continues to push the gate downward, and the relative motion between the other set of inclined planes again starts a wedging action. The Leverlock arm starts a camming action (similar to that mentioned above), and the shoe is kicked to the segment side. In the fully closed position, both the gate and segment expand laterally, firmly press against their respective seats, and provide the desired seating force.

Upon subsequent opening, the stem lifts the gate and the shoe is forced back to the central position. This action breaks the gate loose from the segment and the two pieces collapse and relieve the wedge force. In this case, the Leverlock arm shoe is already centered between the guide rails and no kicking take place. In valves smaller than 4 inches, a spring design is used instead of the Leverlock mechanism. The spring holds the gate and the segment in the collapsed position throughout the stroke except at the end of the stroke where the gate and segment expand when the segment is prevented from moving with the gate.

Valve Orientation

The preferred flow direction is when the gate is downstream.

Special Characteristics of W-K-M Valves

The W-K-M parallel expanding gate valve design has special characteristics that make its performance different from other gate valve designs, including:

1. At small openings, the primary flow area through the valve has a lenticular shape (like a football), whereas other gate valve designs have a flow area shaped like a crescent. For the same disk travel, the primary flow area and valve flow coefficient (C_v) for a W-K-M valve are significantly smaller than for other gate valve designs. At large openings, this difference diminishes.
2. W-K-M valves have a significant secondary flow path through the valve body cavity when the primary flow diminishes. Because of the clearances between the seats/skirts and the gate/segment overlays, the effective ΔP area caused by the secondary flow is significantly larger than the effective area at full flow isolation. Thus, the ΔP force and required gate force before flow isolation and after flow initiation are higher than those at full flow isolation.
3. The segment stop absorbs a certain percentage of the wedging force, and the remaining thrust is reacted by the seats. Theoretically, the seat contact stress and the valve sealing capability can be increased if the friction coefficient between the gate and segment is reduced, e.g., by grinding.

4. Unlike conventional wedge gate valves, the W-K-M gate valves are not susceptible to disc tilting or gouging because of the tight clearances and large contact area between the gate and downstream seat throughout the stroke.

SCOPE OF STEM THRUST PREDICTION MODEL

Fluid Conditions

The model applies to W-K-M parallel expanding gate valves used in water. For steam and two-phase flow applications, additional test data will be required.

Valve Internal Materials

The model was validated using test data from valves with low alloy steel gate/segment, Stellite 6 gate/segment overlays, and Stellite 6 body seats. The model can be applied to other material combinations and in other fluids if the coefficients of friction at the sliding surfaces can be accurately determined.

Valve Orientation

The model was validated for flow in the preferred direction only. The model shows that, for typical friction coefficients, premature wedging will occur with flow in the non-preferred direction and under reverse flow conditions. Premature wedging may prevent the valve from achieving flow isolation.

Stroke Position

The model applies to full opening and closing strokes. For closing strokes, required thrust is predicted at three key points:

1. Primary flow isolation (maximum thrust before full flow isolation),
2. Full flow isolation (also applies to onset of wedging), and
3. Full wedging with a sealing force, F_s.

For opening strokes, required thrust is predicted at three key points:

1. Cracking open from fully wedged position,
2. After cracking and prior to flow initiation, and
3. Primary flow initiation (maximum thrust after cracking).

STEM THRUST MODEL

The model is based on evaluating each force component that contributes to required stem thrust and then calculating the total required thrust by combining the individual components. The contributing terms considered include weight, packing friction, stem piston effect, torque reaction friction, and differential pressure stem thrust.

Calculation of the differential pressure thrust component (which is the key component) requires that the frictional interaction between a number of interfaces be analyzed, including:

- Gate and segment overlay-to-seat interface (μ_{s1}, μ_{s2}),
- Gate-to-segment interface along wedging surface (μ_d),
- Leverlock arm shoe to guide rails (μ_{shoe}), and
- Segment to segment-stop pad on body (μ_o).

For vertical stem orientation, the required stem thrust is given by:

<u>Closing Stroke</u>

$$F_R = \frac{F_{Gate} + F_{Pack} + F_P - W_{stem}}{TRF} \quad (1a)$$

<u>Opening Stroke</u>

$$F_R = \frac{F_{Gate} + F_{Pack} - F_P + W_{stem}}{TRF} \quad (1b)$$

Where:

F_R = Required stem thrust to operate valve (lb)

F_{Gate} = Stem thrust due to ΔP across the gate (lb). This thrust is calculated for various portions of the valve stroke and depends upon the differential pressure across the valve, the desired sealing load, and various friction coefficients for valve sliding surfaces.

F_{Pack} = Packing friction stem thrust (lb)

F_P = Piston effect stem thrust (lb)

W_{stem} = Stem weight (lb)

TRF = Torque reaction factor (dimensionless)

A thrust value to be provided by the actuator is positive (e.g., F_{Pack}). A thrust value to be resisted by the actuator is negative (e.g., F_P in an opening stroke). The individual terms in Equation 1 are described below.

Stem Weight (W_{stem})

The stem weight is relatively small but included for completeness. The weights of the gate and segment are included in F_{Gate}.

Packing Friction Stem Thrust (F_{Pack})

The packing friction stem thrust is the load needed to slide the valve stem through the packing, and can be obtained from diagnostic testing or from the packing/valve manufacturer.

Piston Effect Stem Thrust (F_P)

The piston effect stem thrust is caused by the internal line pressure acting on the stem area. The piston effect stem thrust is positive (i.e., adds to the required thrust) in the closing direction and is negative in the opening direction (i.e., helps to open the valve).

$$F_P = P_{body}\,A_{stem} = P_1\left(\frac{\pi}{4}\right)d_{stem}^2 \qquad (2)$$

Where:

P_{body} = Internal gage pressure at the stem packing (psig) (set equal to the upstream pressure (P_1))

A_{stem} = Stem area at packing (in^2)

d_{stem} = Stem diameter at packing (in)

Torque Reaction Factor (TRF)

The torque in the stem is reacted in the valve by surfaces which engage and slide. This torque reaction causes a friction load that opposes stem motion. For the W-K-M gate valve, external torque arms are not commonly used. Therefore, the model assumes that the torque is reacted through the valve seat, and the distance to the torque reaction surface is set equal to half the seat inside diameter. In general, the contribution of the torque reaction factor to the overall stem thrust needed to open or close the valve is less than 5 percent (0.95 < TRF < 1.0). The torque reaction factor is a dimensionless constant and is given by:

$$TRF = 1 - \frac{\mu_s\,(FS)}{r_t} \qquad (3)$$

Where: r_t = Distance to torque reaction surface (ft) (set equal to 1/2 seat inside diameter)

FS = Stem factor (ft), calculated from stem screw thread parameters

μ_s = Friction coefficient for torque reaction surface (set equal to disk-to-seat friction coefficient)

Stem Thrust Due to Differential Pressure Across Gate (F_{Gate})

The required stem thrust due to differential pressure across the gate is positive (i.e., adds to the required thrust) regardless of valve travel direction. This term is evaluated for different gate positions in the opening and closing directions. The required gate force is highly sensitive to the friction coefficients for the various sliding surfaces in the valve.

Closing Stroke - Gate Downstream (Preferred Flow Direction). Equations for F_{Gate} for this configuration are developed for the required gate force to reach three different points of the valve closure, as follows.

1. <u>Primary flow isolation (maximum thrust before full flow isolation)</u>: One of the features of the W-K-M expanding gate valve performance is that, even after the disk assembly blocks off the primary flow (no direct flow path visible from the upstream to the downstream side), a secondary flow through the valve body continues until full flow isolation is

achieved (see Figure 5). This secondary flow is responsible for increasing the effective area over which ΔP acts and results in a noticeable increase in stem thrust. The model accounts for the increase in stem thrust due to secondary flow effect. The maximum magnitude of the required ΔP gate force before full flow isolation is given by:

$$F_{Gate} = \mu_s F_{dp} - \left(W_g + W_{sg}\right) \quad (4)$$

Where:

$$F_{dp} = A_{eff,max} \times \Delta P \quad (5)$$

$$A_{eff,max} = 1.10 \times \frac{\pi}{4} \times d^2_{seat,O.D.} \quad (6)$$

ΔP = maximum valve pressure drop (including water inertia effect), psi

The 1.10 factor in Equation 6 is based on the fact that the effective secondary flow area (shown as a dash line in Figure 5) is about 10 percent larger than the seat outside area.

2. Full flow isolation and onset of wedging: After both primary and secondary flow isolation, the gate/segment assembly (in the collapsed position) slides over the downstream seat.

$$F_{Gate} = \mu_s F_{dp} - \left(W_g + W_{sg}\right) \quad (7)$$

Where:

$$F_{dp} = A_{eff} \times \Delta P \quad (8)$$

$$A_{eff} = \frac{\pi}{4} \times d^2_{seat,I.D.} \quad (9)$$

The seat inside diameter is used in Equation 9 because it is considered a best estimate of the sealing diameter at flow isolation. Note that the thrust is controlled by Equation 4 because F_{dp} in Equation 5 is higher than that in Equation 8.

The additional gate force to expand the gate and segment is very small, and F_{Gate} given by Equation 7 also applies at the onset of wedging.

By achieving the gate thrust given in Equation 7 (flow isolation), the valve will isolate flow at the downstream disk (differential pressure force alone will hold downstream disk against its seat). The flow isolation point represents the functionally required thrust point for many valve applications.

3. Wedging with a sealing force, F_s: Valve sealing capability can only be verified by in situ testing. Values of sealing force, F_s, can be obtained from the manufacturer or based on experience with same-size valve under similar operating conditions.

The maximum required gate force to wedge the gate and segment and to develop a seat force, F_s (after reaching isolation) is given by the following equations:

$$F_{Gate} = \mu_s \left[2F_s + \left(R_1 - R_2\right) F_{dp}\right] + F_o - \left(W_g + W_{sg}\right) \quad (10)$$

$$F_{dp} = \frac{\pi}{4} d^2_{seat,I.D.} \times \Delta P \quad (11)$$

$$F_o = \left(F_s - R_2 F_{dp}\right) \times \left(c_\alpha - \mu_s\right) + W_{sg} \quad (12)$$

$$c_\alpha = \frac{\tan\alpha + \mu_d}{1 - \mu_d \tan\alpha} \quad (13)$$

$$F_{s1} = F_s + R_1 F_{dp} \quad (14)$$

$$F_{s2} = F_s - \mu_o F_o \, ; F_s > \mu_o F_o \quad (15)$$

$$R_1 + R_2 = 1.0 \quad (16)$$

Where:

F_{dp} = Force due to differential pressure (lb)

ΔP = Maximum valve pressure drop (psi)

F_o = Force between segment and segment stop pad in valve body in the fully wedged position (lb)

μ_s = Friction coefficient between the disk and valve seat ring (dimensionless)

μ_d = Friction coefficient at the sliding surfaces between gate and segment (dimensionless)

μ_o = Friction coefficient between the segment and segment stop pad (dimensionless). This value should be set equal to μ_d.

α = Wedge angle between gate and segment (degrees)

R_1 = Fraction of ΔP force (F_{dp}) absorbed by the downstream seat, (typically $0.6 < R_1 < 1.0$)

R_2 = Fraction of ΔP force (F_{dp}) absorbed by the upstream seat

F_{s1} = Downstream seat force *including* ΔP force contribution, lb

F_{s2} = Upstream seat force *excluding* ΔP force contribution, lb

In using the above equations, the following considerations apply:

- $\alpha = 15°$ for W-K-M model D-2 valves used in nuclear service. For non-nuclear service α ranges from 12° to 18° and should be verified with manufacturer.
- If $[2F_s + (R_1 - R_2) F_{dp}] < F_{dp}$ in Equation 10, then the specified value of F_s is too small, and the entire term within the bracket should be replaced with F_{dp}. In this case, F_s is insignificant and F_{Gate} must be able to overcome $\mu_s F_{dp}$.
- The valve is not fully wedged until $F_{s2} > 0$ (i.e., $F_s > \mu_o F_o$). High μ_o reduces the upstream seat sealing capabilities.

Closing stroke - gate upstream (nonpreferred flow direction). The equations for the differential pressure stem thrust term for this configuration are developed in the model. Validation data were not obtained for the nonpreferred flow direction and the model is not applicable to this case.

The model shows that, during the closing stroke under ΔP conditions, premature wedging between gate and segment can occur after the shoe clears the guide rail but before the disk reaches the isolation point or the final segment stop position. Premature wedging is not considered a normal behavior and corresponding stem thrust predictions are not considered part of the validated methodology for W-K-M valves. The model gives the condition under which premature wedging does not occur and shows that in the absence of premature wedging, F_{Gate} predicted by equations 4, 7, and 10 for gate downstream orientation can be used for gate upstream orientation.

Closing stroke - no flow. **($F_{dp} = 0$).** Under these conditions, the gate force needed to reach the flow isolation and disk hard-seated position is zero except for the weight of gate/ segment assembly. Required gate force under no-flow conditions is only associated with developing a sealing force. To calculate F_R in this case, set $F_{dp} = 0$ in equations 4, 7, 10, and 12.

Opening stroke - gate downstream (preferred flow direction). Required gate force is calculated for three different points of valve opening stroke: (1) cracking from fully wedged position, (2) after cracking and prior to flow initiation, and (3) primary flow initiation. Required gate force for these points depends upon the differential pressure force acting across the valve assembly, the actual sealing force that exists before the opening stroke, and the change in the differential pressure across the valve since the previous closing stroke. The sealing force that must be overcome depends on:

- The stem thrust with which the valve was previously closed/wedged, including inertia overshoot,

- The amount of sealing force relieved when the compressive load on the stem is released,
- The change in differential pressure across the valve assembly between closure and opening, and
- The coefficients of friction between gate and segment wedging surfaces (μ_d) and between gate/segment overlays and valve seats (μ_{s1} and μ_{s2}).

At the beginning of an opening stroke, the stem compressive force is equal to the wedging force from the previous closing stroke. In the opening direction, both the force on the gate, F_{Gate}, and the segment stop force, F_o, drop to zero. Due to the clearance between the stem head and gate box, the stem travels a short distance before it picks up the gate/segment assembly. As the stem starts to pull on the gate, the friction forces in the upstream and downstream seats, as well as the friction between the gate and segment, reverse direction to oppose impending motion. As the segment stop force, F_o, is relieved, the segment stop friction, $\mu_o F_o$, is also relieved. Some of the stored elastic energy is relieved and some elastic relaxation takes place. In the absence of ΔP, the upstream seat force, F_{s2}, will increase to equalize the downstream seat force, F_{s1}, which will be less than its value from the previous closing stroke, F_s. Thus,

$$F_{s2} = F_{s1} = R\,F_s \qquad R \leq 1.0 \tag{17}$$

where R is a seat force relaxation ratio. Examination of test results shows that R can range from 0.5 to 0.9.

In addition, it is assumed that when a closed valve has a differential pressure applied to it, 60 percent of the differential pressure goes toward increasing the contact force at the downstream face and 40 percent is absorbed at the upstream face (seen as a reduction in the upstream disk to seat contact force).

Upon applying the cracking force, the segment may travel with the gate in the expanded position or may stay (momentarily) stationary. If the segment travels with the gate, the wedging friction locked between the gate and segment will not be broken until the Lever-lock shoe kicks the bottom of the guide rail. In this case, the required gate force at some partially open position will include the force exerted on the shoe. On the other hand, if the segment does not travel with the gate, the wedging friction between the gate and segment will be broken immediately upon cracking. In this case, the segment will collapse on the gate and the shoe will clear the guide rail without significant kicking force (if any). Analysis results show that, for typical coefficients of friction, the segment does not travel with the gate and the latter mode prevails. Test data also support this conclusion. Furthermore, the required unwedging gate force, F_{Gate}, under the prevailing mode bounds that under the other (non-prevailing) mode. The required gate force equations for both modes are given in the model. In this paper only equations for the prevailing mode are given.

At a partially open position, and before initiation of primary flow, a secondary flow develops as discussed above. The effective ΔP area with the secondary flow is the same as that given for the closing stroke.

Two different closing/opening cases are considered:

1. Valve is closed under static conditions and is opened under differential pressure conditions, and
2. Valve is opened under the same conditions in which it was closed (either with flow or without flow).

Only Case 1 is provided because the stem thrust needed to open the valve for Case 2 is bounded by Case 1. The actual seat force on the downstream seat during opening (F_{s1}) is the larger of:

$$F_{s1} = R\,F_s + R_1\,A_{eff}\,\Delta P \tag{18a}$$

$$F_{s1} = A_{eff}\,\Delta P \tag{18b}$$

Where: F_s = Seat force developed during the previous static (no differential pressure, no flow) closing stroke (lb)

ΔP = Maximum valve pressure drop during opening (psi)

A_{eff} = Seat inside area, in^2

$= \pi/4 \times d^2_{seat,I.D.}$

R and R1 are the same as given above. F_s is determined from the maximum expected stem thrust for the previous static closing stroke. F_s is calculated using Equation 19 (which is developed from equations 1 and 10 by setting F_{dp} to zero, $\mu_{s1} = \mu_{s2} = \mu_s$, $\mu_o = \mu_d$, and $W_{stem} = W_g = W_{sg} = 0$). For a given maximum expected stem thrust for the valve closing stroke, F_{RC}, the value of F_s after wedging is given by:

$$F_s = \frac{F_{RC}\ TRF - F_{Pack}}{\mu_s + \dfrac{\tan\alpha + \mu_d}{1 - \mu_d \tan\alpha}} \tag{19}$$

Although conservative values of F_{Pack} may underestimate F_s from Equation 19, the required stem thrust to open the valve (F_R in Equation 1b) will still be conservative.

Opening stroke: segment collapses on gate. This is the common mode with typical coefficients of friction. The segment will collapse on the gate when:

$$\mu_{s2} \geq \frac{(\mu_d - \tan\alpha)}{(1 + \mu_d \tan\alpha)} - \frac{W_{sg}}{R\,F_s} \tag{20}$$

Where F_s is given by Equation 19. Equation 20 gives the threshold of the coefficient of friction between the segment and the upstream seat, above which the wedging force between gate and segment is broken and the segment collapses on the gate. Equation 20 applies when the pressure drop across the segment is zero (ΔP segment = 0), which includes static conditions ($\Delta P = 0$) and dynamic conditions with body pressure equal to upstream pressure ($P_{body} = P_1$). Because the upstream seat usually does not maintain a leak-tight seal like the downstream seat, $P_{body} = P_1$ is the most common ΔP condition. Table 1 gives the threshold values of μ_{s2} for a wide range of μ_d.

The required gate force at the three key positions in the valve opening stroke are as follows:

1. Cracking from the fully wedged position: The required gate cracking force when the segment collapses on the gate is given by:

$$F_{Gate} = \mu_s F_{s1} + \left(\frac{\mu_d - \tan\alpha}{1 + \mu_d \tan\alpha}\right) \times \left(R\,F_s - R_2\,F_{dp}\right) + W_g \tag{21}$$

Where F_{s1} is the larger of those calculated by equations 18a and 18b, and F_s is calculated from Equation 19.

2. After cracking and before flow initiation: In this mode, gate movement releases the wedging force, and the segment collapses on the gate. The collapsed assembly travels together for a short distance before a secondary flow starts in the clearances at the downstream seat. The gate force required to slide the gate/segment assembly in the collapsed position before the initiation of the secondary flow is given by:

$$F_{Gate} = \mu_s F_{dp} + \left(W_g + W_{sg}\right) \tag{22}$$

$$F_{dp} = A_{eff}\,\Delta P$$

$$A_{eff} = \frac{\pi}{4} d^2_{seat,I.D.}$$

Equation 22 applies for opening strokes when μ_{s2} satisfies Equation 20. F_{Gate} given by Equation 22 is always bounded by F_{Gate} calculated by Equation 23.

3. Primary flow initiation (maximum thrust after cracking): As the secondary flow develops, A_{eff} increases until a maximum value is reached. The maximum value of A_{eff} is estimated as follows:

$$A_{eff,max} = 1.10 \times \frac{\pi}{4} \times d^2_{seat,O.D.} \tag{23}$$

The dash line in Figure 5 shows the boundary of $A_{eff,max}$.

The required gate force in this position is:

$$F_{Gate} = \mu_S \times A_{eff,max} \times \Delta P + (W_g + W_{sg}) \qquad (24)$$

ΔP is set equal to the maximum valve pressure drop because the secondary flow is too small to cause any noticeable pressure drop across the piping system. In this mode the Leverlock arm shoe does not kick the guide rail and the shoe force is negligible

Opening stroke: gate and segment travel in the expanded position. The segment collapses on the gate upon applying the cracking force with all typical coefficients of friction between sliding surfaces. When the upstream seat coefficient of friction is low (say less than 0.28) and the coefficient of friction between gate and segment is very high (say more than 0.60), the segment will fail to collapse on the gate upon applying the cracking force. The threshold value of μ_{s2} as a function of μ_d below which this mode may occur is given in Table 1. In this case the gate and segment assembly would slide on the upstream and downstream seats in the expanded position. A comparison of F_{Gate} under the two possible modes (from Reference 6) shows that the required gate cracking force for this case is smaller than that given by Equation 21. After cracking, the required opening thrust is slightly higher because of contributions from the Leverlock shoe. Detailed equations for this mode are given in the model (Reference 6).

Opening stroke - gate upstream (nonpreferred flow direction). Opening stroke equations for the nonpreferred flow direction with gate upstream are included in the model. The model has not been validated in the nonpreferred flow direction; therefore, the methodology is not considered applicable in this flow direction.

Opening stroke - no flow. For the case of an opening stroke under no-flow conditions, set $\Delta P = F_{dp} = 0$ in equations 21, 22, and 24.

MODEL VALIDATION AGAINST TEST DATA

The model was validated against EPRI in situ test data for a 6-inch 300-pound, an 8-inch 150-pound, and a 16-inch 1500-pound W-K-M parallel expanding gate valves. The 8-inch valve is a component cooling water system valve. The other two valves are shutdown cooling system valves. The valves have Stellite 6 on Stellite 6 contact between the seats and gate/segment overlays. The contact at the back side of the gate and segment is based on a base material of low alloy cast steel.

Test data include one full (closing and opening) static stroke for each valve, one full (closing and opening) dynamic stroke for the 6-inch and 8-inch valves, and three opening hydrostatic ΔP strokes for the 16-inch valve. The closing strokes for the 16-inch valve are considered static strokes because ΔP did not build up until after the valve closed. All tests were conducted using water at ambient temperature, and with flow in the preferred direction.

In the closing direction, the torque switch was used and test valves were wedged in the fully closed position. In the opening direction, the limit switch was used and test valves were not wedged in the fully open position. Test results at key points are summarized in Tables 2 and 3.

Approach

The approach used to validate model predictions using in situ test data is as follows:

1. Static test results are used to determine the packing force and the sum of the gate, segment, and stem weights for each valve.
2. Valve-specific coefficients of friction are inversely calculated from test data. The model is then used to calculate the best-estimate stem thrust predictions using valve-specific friction coefficients and the given test conditions.
3. The above best-estimate stem thrust predictions are compared to measured thrust values.

Results of Model Validation

Tables 2 and 3 show the maximum predicted thrust using best estimates of the seat friction coefficients and measured thrust at the key points of opening and closing strokes respectively. These results show that model predictions bound test data with adequate margin with one exception. The predicted thrust (using best estimate of friction coefficients) for the opening stroke of the 8-inch valve exceeded the measured thrust by less than 5 percent, which was still within acceptable range. Predictions using design values of friction coefficients (not shown) provide larger margins than those shown in Tables 2 and 3.

In addition to the test data used for model validation in Reference 6, EPRI obtained in situ cold water test results for seven 8-inch and four 3-inch W-K-M valves. Required thrust predictions using design values of friction coefficient were found to bound test results for all strokes considered.

Figures 6 and 7 show typical thrust signatures for the 6-inch and 8-inch valves for a full stroke (closing and opening). These figures show that the maximum required thrust is governed by a maximum value before full flow isolation or primary flow initiation.

CONCLUSIONS

As part of EPRI's MOV Performance Prediction Methodology, a validated model was developed to determine the required valve stem thrust to operate a parallel expanding gate valve under given operating conditions. The model calculates the required thrust at selected points of valve closing and opening strokes in the preferred flow direction. The model shows that the presence of a secondary flow can increase the required thrust above that when the valve is in the fully closed position.

Model predictions are in general agreement with measured thrust obtained during in situ testing of three different valves under different flow conditions.

In the nonpreferred flow direction (caused by installation or resulted from reverse flow conditions), the model shows that premature wedging can occur during ΔP closure strokes even when the coefficients of friction at different sliding surfaces are within the typical range for disk-to-segment wedge angle of 15 degrees used in nuclear power plant application.

ACKNOWLEDGMENT

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The EPRI PPM program includes a large number of proprietary reports and test data reports. For a detailed listing of these reports, the reader should contact EPRI at P. O. Box 10412, Palo Alto, CA 94303.

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Table 1: Threshold of Upstream Seat Coefficient of Friction, μ_{s2}, Above Which the Segment Will Unwedge and Collapse on the Gate upon Cracking at the Beginning of an Opening Stroke ($\alpha = 15°$, $W_{sg} = 0$, and Body Pressure = Upstream Pressure)

μ_d Gate-to-Segment Coefficient of Friction	μ_{s2} Threshold of Segment-to-Upstream Seat Coefficient of Friction
0.25	–0.0168
0.30	0.0297
0.40	0.1193
0.50	0.2046
0.60	0.2861
0.70	0.3638

Table 2: Predicted and Measured Stem Thrust - Opening Strokes Using *Best Estimate* Values of Seat Coefficients of Friction

Component	6-inch	8-inch	16-inch First Test	16-inch Second Test	16-inch Third Test
$F_{packing}$, lb	580.0	380.0	6,600.0	6,600.0	6,600.0
Seat I. D., inch	6.125	8.125	13.125	13.125	13.125
Seat O. D., inch	7.000	9.375	15.625	15.625	15.625
Point A; Cracking From Fully Wedged Position, Opening Stroke					
A_{eff}, Effective ΔP area, inch^2	29.465	51.849	135.297	135.297	135.297
FRC, Previous stroke wedging force, lb	8,487	7,229.0	69,000.0	69,000.0	69,000.0
F_s, Previous stroke seat force, lb	6,643	6,435	48,413	49,621	48,023
P_1, psig	211.0	114.0	101.0	211.0	291.0
ΔP, psi	189.0	85.0	98.0	208.0	288.0
F_{dp}, lb	5,568.8	4,407.1	13,259.1	28,141.8	38,965.6
Measured thrust, lb	3,398	3,751.0	25,003.0	26,319.0	30,430.0
FR, Prediction, lb	5,059.9	3,581.4	38,849.1	41,019.1	45,161.6
Ratio of measured to predicted thrust	0.672	1.047	0.644	0.642	0.674
Point C; Maximum Thrust After Flow Initiation, Opening Stroke					
$A_{eff,max.}$, Max. effective ΔP area, inch^2	42.333	75.932	210.922	210.922	210.922
P_1, psig	208.0	112.0	81.0	90.0	88.0
ΔP, psi	186.0	82.0	79.0	88.0	86.0
F_{dp}, lb	7,873.9	6,226.4	16,662.9	18,561.2	18,139.3
Measured thrust, lb	3,772.0	2,455.0	13,818.0	14,476.0	14,967.0
FR, Predicted thrust, lb	4,095.0	2,502.2	16,531.2	16,935.2	17,485.5
Ratio of measured to predicted thrust	0.921	0.981	0.836	0.855	0.856

Table 3: Predicted and Measured Stem Thrust - Closing Strokes
Using *Best Estimate* Values of Seat Coefficients of Friction

Component	6-Inch	8-Inch
Point H; Maximum Thrust Before Flow Isolation		
$A_{eff,max.}$ Max. effective ΔP area, inch^2	42.333	75.932
P_1, psig	209.0	126.0
ΔP, psi	188.0	99.0
Measured thrust, lb	3,734.0	2,728.0
F_{dp}, lb	7,958.6	7,517.3
FR, Predicted thrust, lb	4,774.1	3,356.4
Ratio of measured to predicted	0.78	0.81
Point I_1; Flow Isolation and Onset of Wedging		
A_{eff}, Effective ΔP area, inch^2	29.465	51.849
P_1, psig	211.0	115.0
ΔP, psi	189.0	86.0
Measured thrust, lb	3,200.0	1,909.0
F_{dp}, lb	5,568.8	4,459.0
FR, Predicted thrust, lb	3,619.8	2,206.0
Ratio of measured to predicted	0.88	0.87

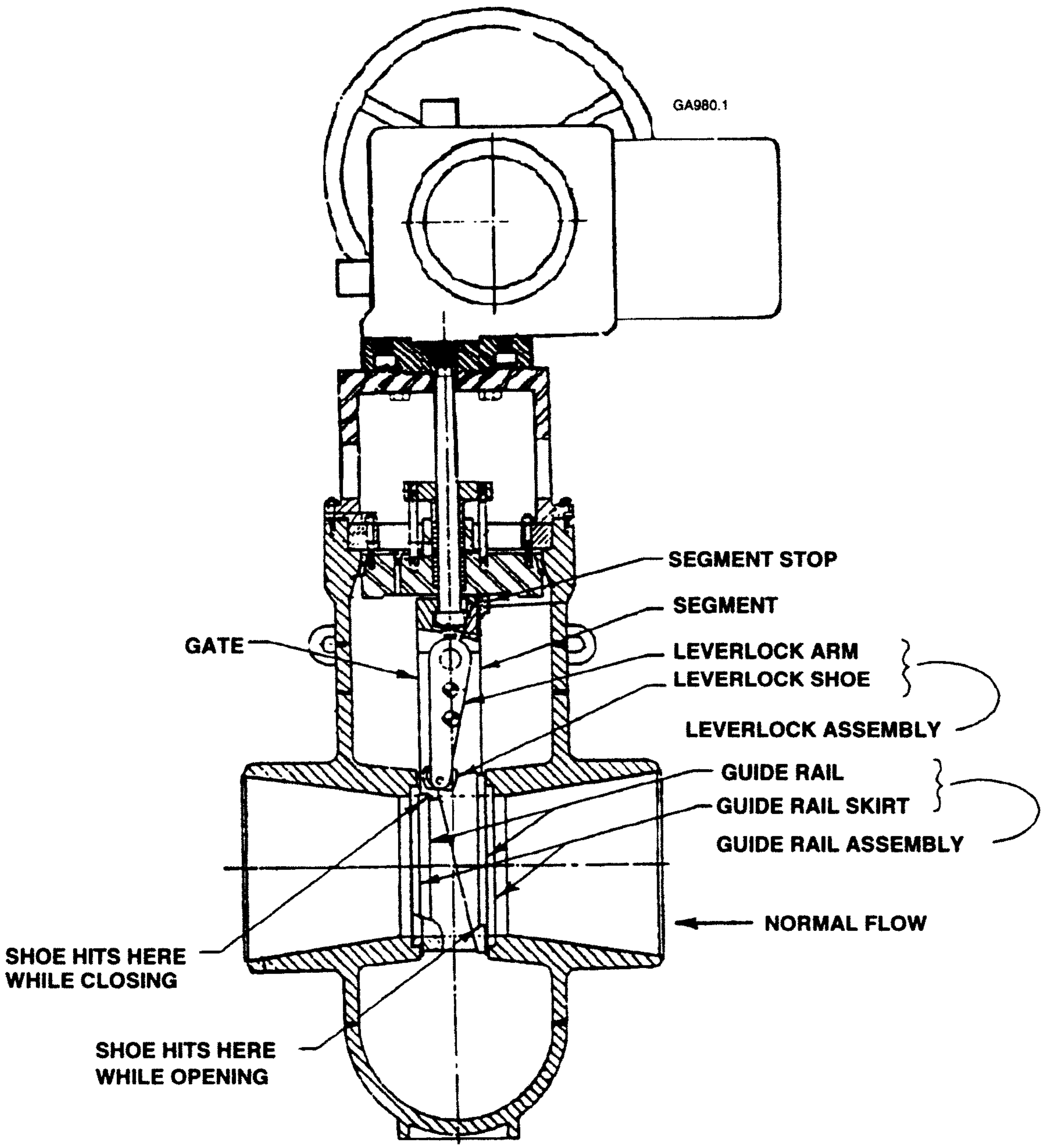
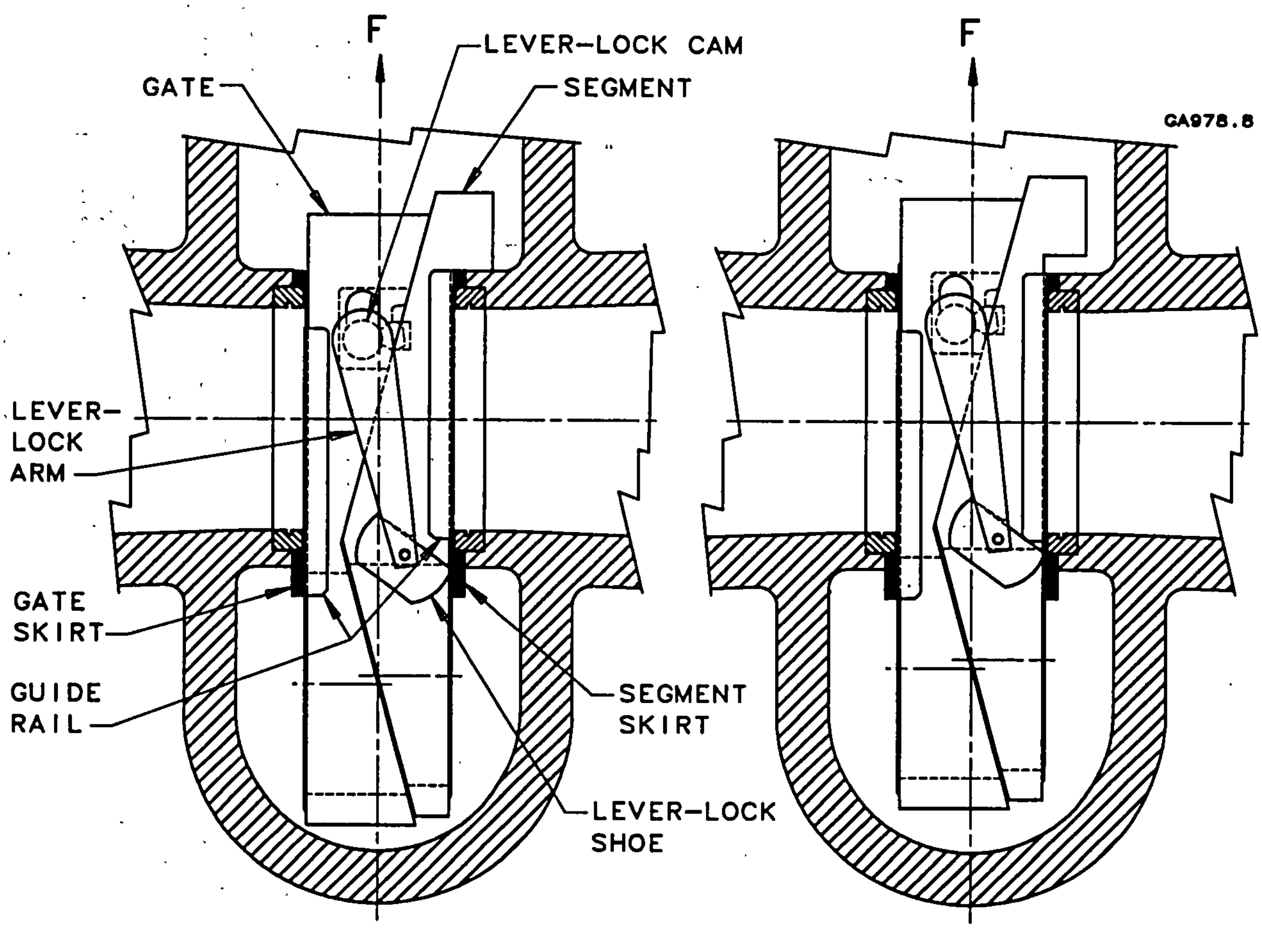


Figure 1

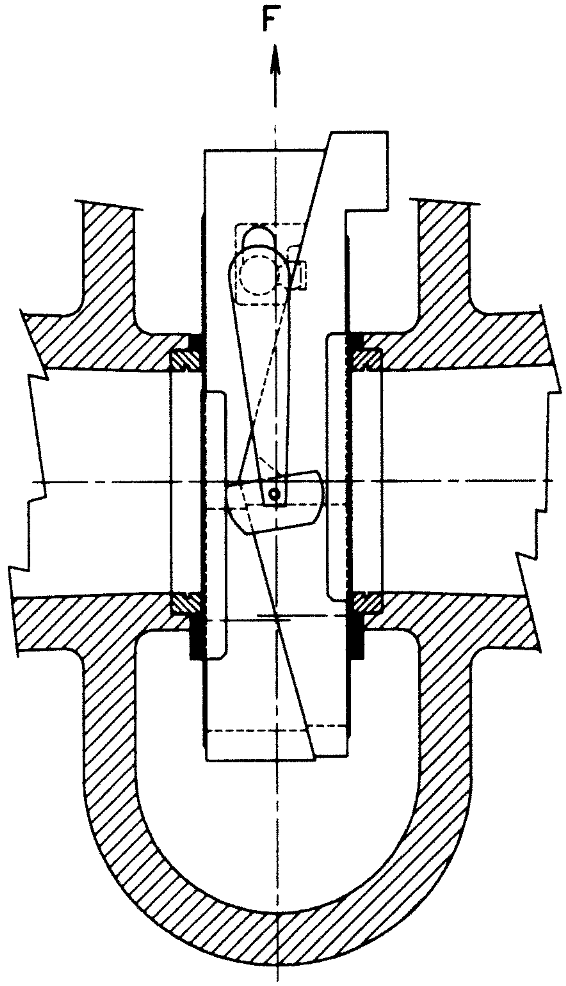
Nomenclature of Components Used in W-K-M Gate Valves



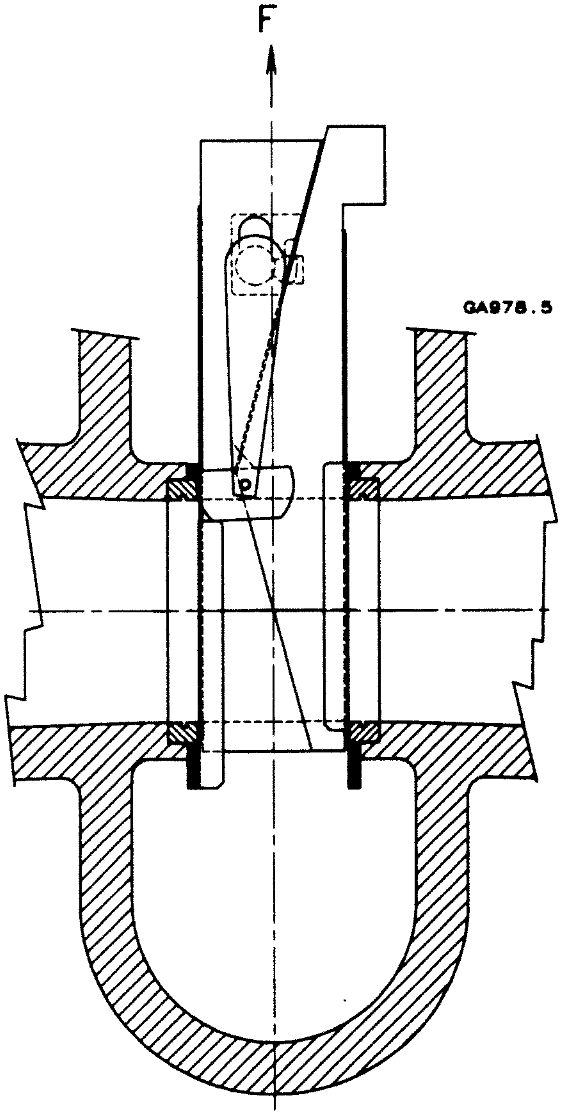
Position 1
Valve is wedged in the closed position; gate and segment are fully expanded.

Position 2
Leverlock arm mechanism ensures that gate collapses on segment.

Figure 2
Key Positions of an Opening Stroke Sequence Showing the Operation of the Leverlock Mechanism



Position 3
Gate and segment travel up in the collapsed position.



Position 4
Segment hits the open stop. Gate and segment are wedged in the open position (only for torque switch control in the open direction).

Figure 2 (continued)
Key Positions of an Opening Stroke Sequence Showing the Operation of the Leverlock Mechanism

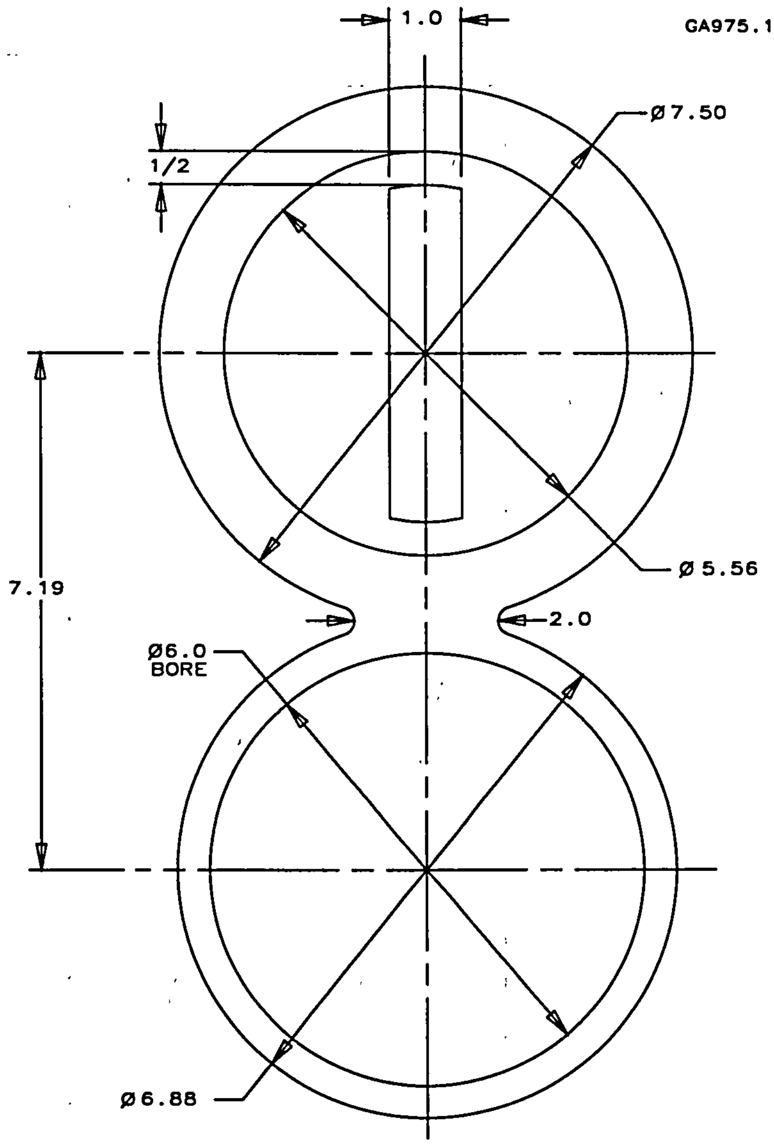


Figure 3
Gate Overlay Dimensions for 6-Inch, 300-Pound W-K-M Valve

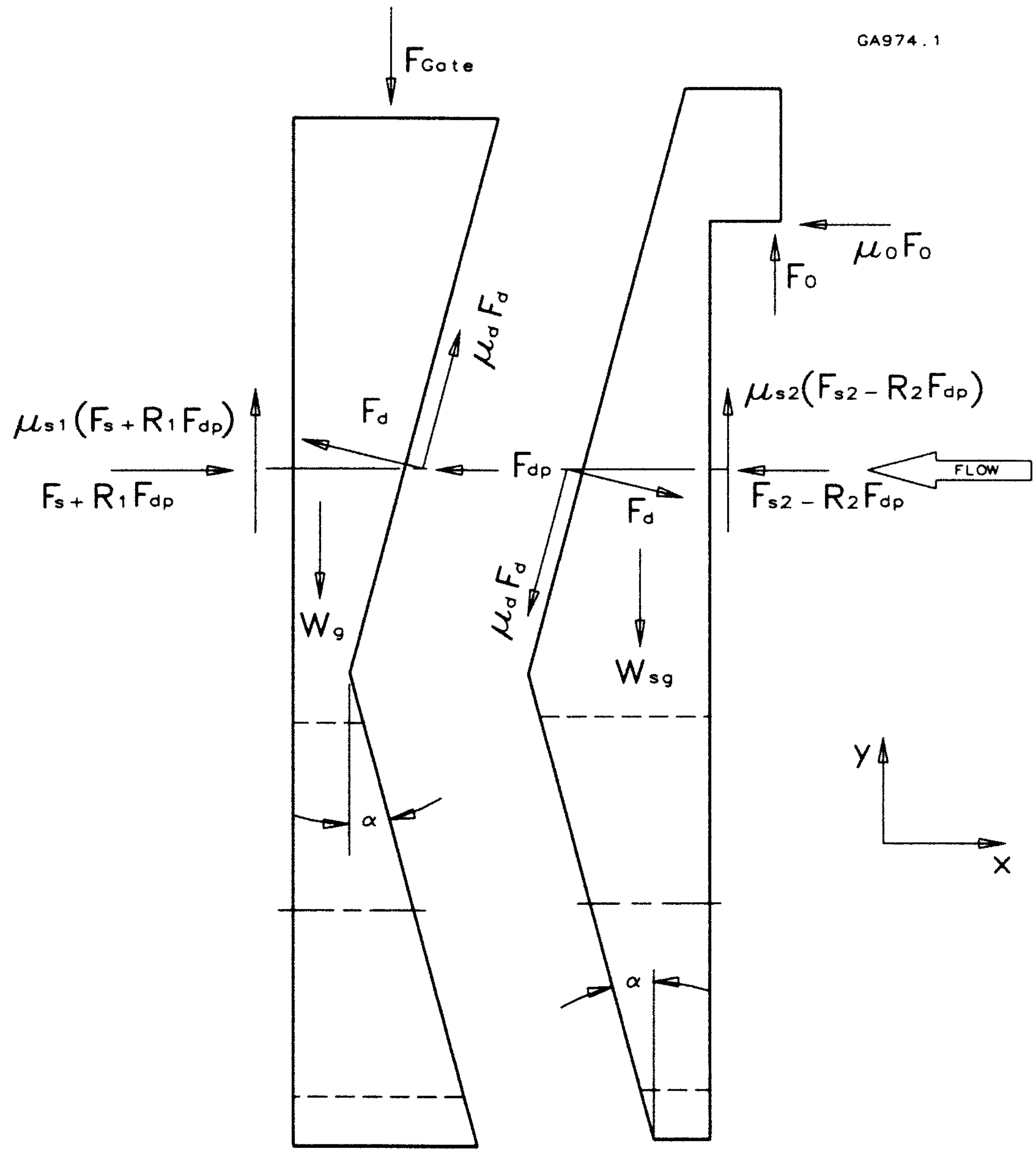


Figure 4
Free Body Diagrams for Gate and Segment During Wedging Under Maximum Pressure Drop ΔP (W-K-M-Recommended Installation with Gate Downstream)

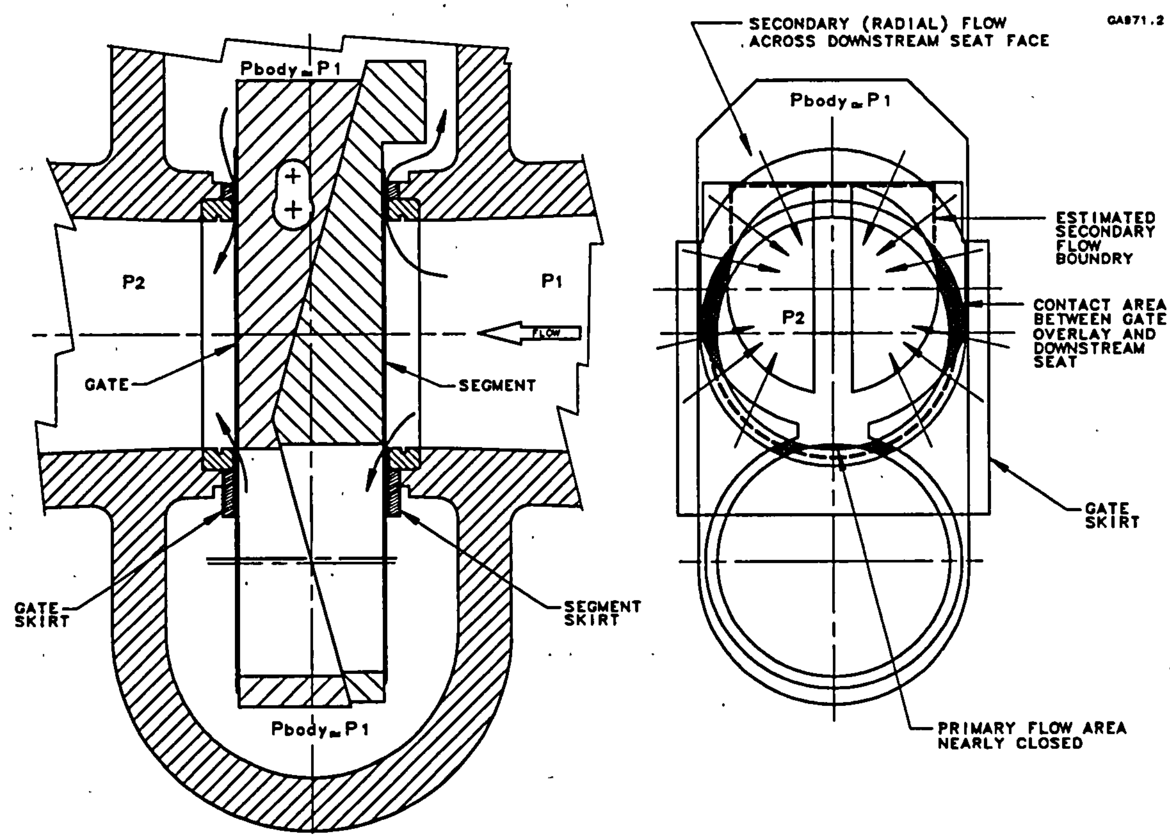
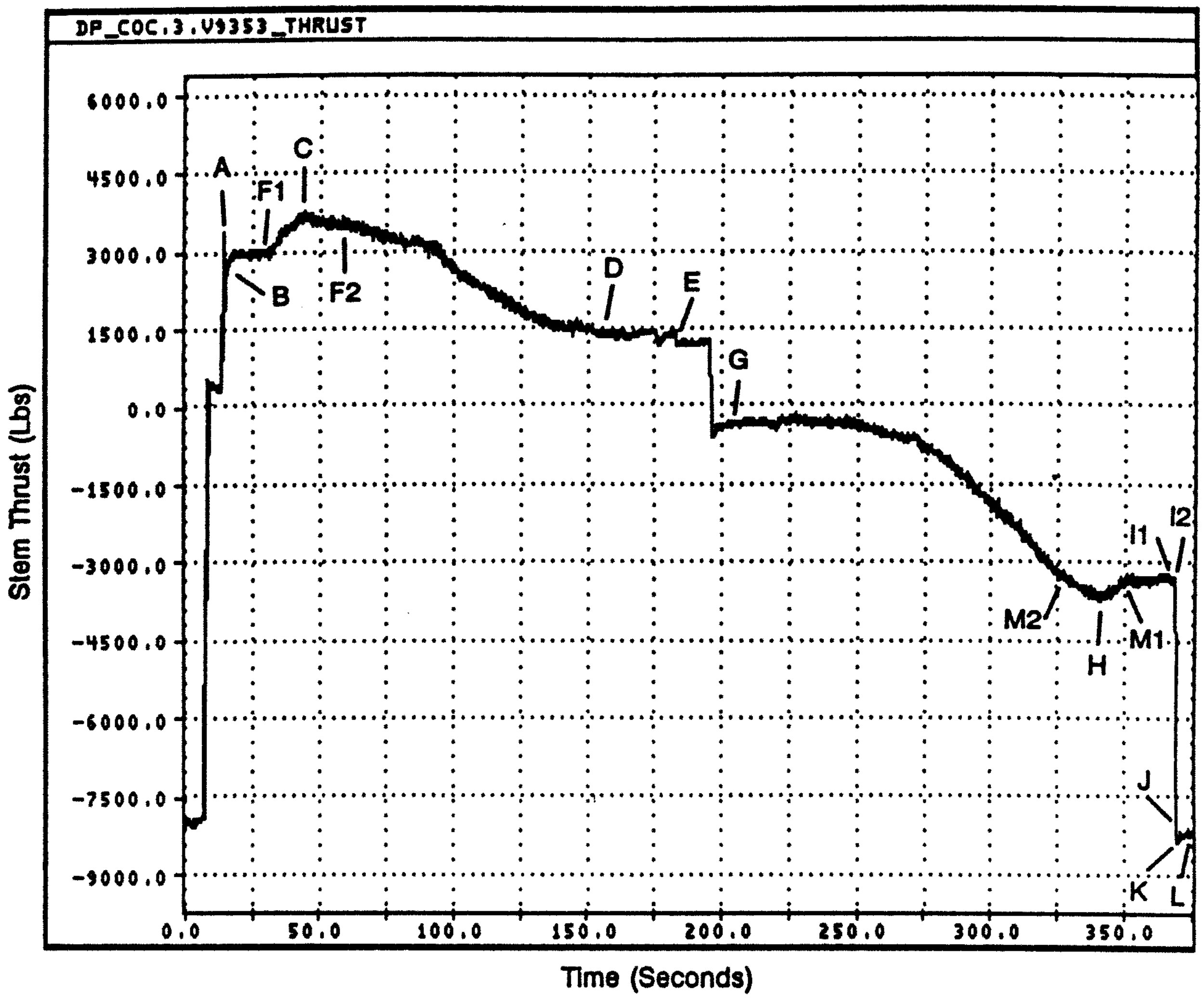


Figure 5

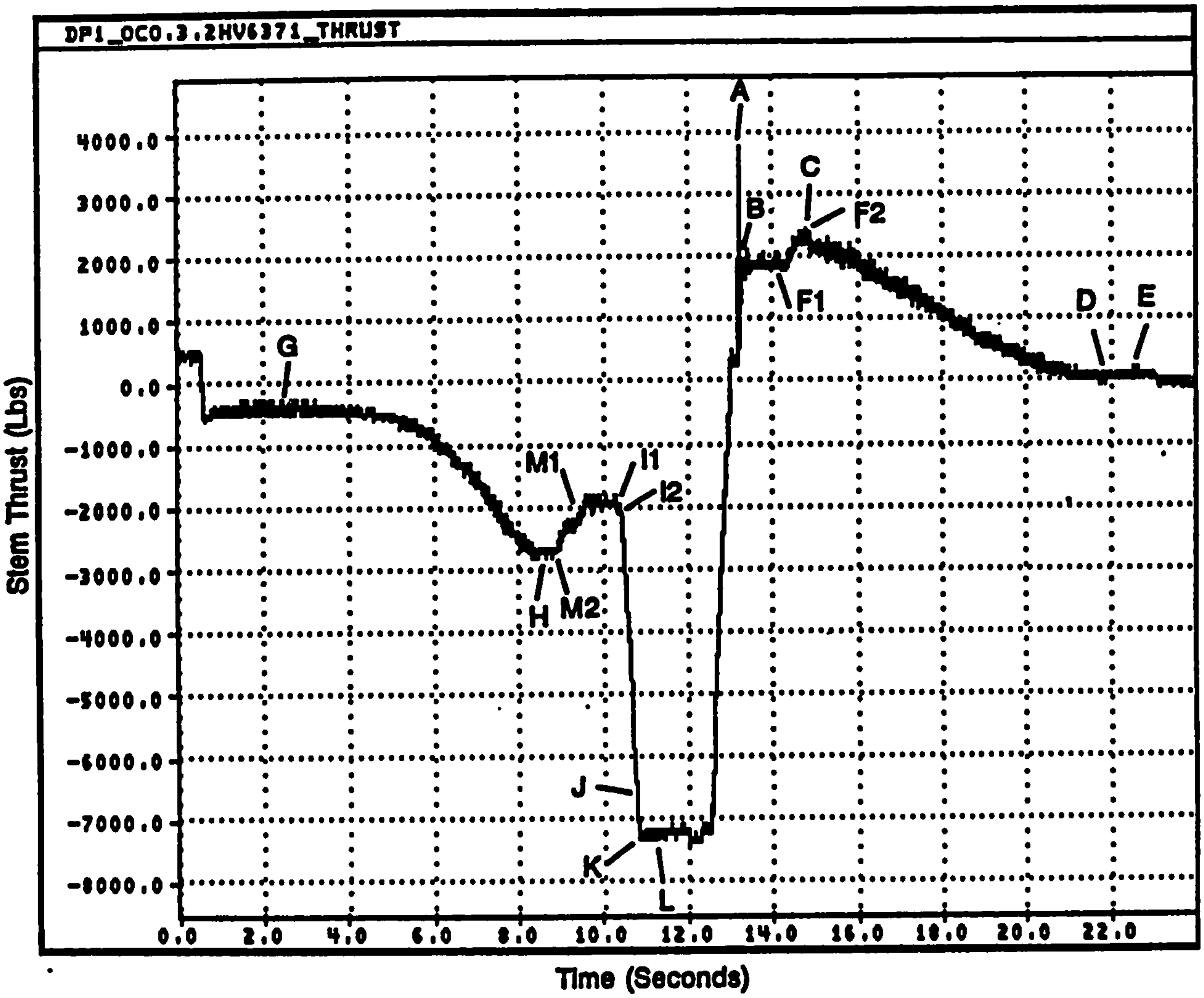
Increase in Effective ΔP Area Caused by Secondary Flow Through the W-K-M Valve Body before Flow Isolation (Dimensions used are for 6-inch, 300-pound W-K-M Model D-2)



Ambient Water; Closed-to-Open-to-Closed
Preferred Flow Direction (Gate Downstream)
ΔP = 195 psi
Flow Rate = 3,900 gpm

Figure 6

Thrust Signature from the ΔP Test for the 6-Inch, 300-Pound W-K-M Gate Valve



Note 1: The decrease in measured stem thrust below 0.0 lbs after point E indicates a source of measurement error which apparently results from transient response of the stem thrust strain gages to rapid load changes.

Ambient Water; Open-to-Closed-to-Open
Preferred Flow Direction (Gate Downstream)
ΔP = 100 psi; Flow Rate = 2,975 gpm

Figure 7

Thrust Signature from the ΔP Test for the 8-Inch, 150-Pound W-K-M Gate Valve

Session 2A

General Issues on Testing Pumps

Session Chair
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On-line PWR RHR Pump Performance Testing Following Motor and Impeller Replacement

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ABSTRACT

On-line maintenance and replacement of safety-related pumps requires the performance of an inservice test to determine and confirm the operational readiness of the pumps. In 1995, major maintenance was performed on two Pressurized Water Reactor (PWR) Residual Heat Removal (RHR) Pumps. A refurbished spare motor was overhauled with a new mechanical seal, new motor bearings and equipped with pump's "B" impeller. The spare was installed into the "B" train. The motor had never been run in the system before. A pump performance test was developed to verify it's operational readiness and determine the insitu pump performance curve. Since the unit was operating, emphasis was placed on conducting a highly accurate pump performance test that would ensure that it satisfied the NSSS vendors accident analysis minimum acceptance curve. The design of the RHR System allowed testing of one train while the other was aligned for normal operation. A test flow path was established from the Refueling Water Storage Tank (RWST) through the pump (under test) and back to the RWST. This allowed staff to conduct a full flow range pump performance test. Engineering was requested to develop a pump performance test and measurement system, that would be capable of minimizing pressure and flowrate measurement errors upon Total Dynamic Head (TDH) test results. Each train was analyzed and an expression developed that included an error vector term for the TDH (ft), pressure (psig), and flow rate (gpm) using the variance error vector methodology. This method allowed the engineers to select a test instrumentation system that would yield accurate readings and minimal measurement errors, for data taken in the measurement of TDH (P,Q) versus Pump Flow Rate (Q). Test results for the "B" Train showed performance well in excess of the minimum required. The motor that was originally in the "B" train was similarly overhauled and equipped with "A" pump's original impeller, re-installed in the "A" train, and tested. Analysis of the "A" train results indicate that the RHR pump's performance was also well in excess of the vendors requirements. The methodology, data collection, and analysis processing can be applied to any type of centrifugal pump (vertical or horizontal installation). The error analysis of measured data points can be accomplished quickly and high quality pump performance curves generated. Test personnel can quickly ascertain that the shape and magnitude of the pump curves are satisfactory. The method also addresses pump entrance / exit effects and differential elevation of the inlet / outlet nozzles.

INTRODUCTION

On-line maintenance and testing of safety-related pumps requires the performance of an Inservice Test (IST) to determine and confirm operational readiness of the pumps. Since the unit is operating, emphasis was placed upon conducting a highly accurate in-situ full range pump performance test, that would ensure that it satisfied the Design Bases Accident (DBA) analysis minimum acceptance curve. Since IST tests measure a single point, a special pump test was developed to meet plant operating conditions and test requirements. RHR System piping designs allowed a flow path to be established from the RWST Tank (water source), to the pump under test, and back to the RWST through a common discharge header. This allowed on-line testing of each train and satisfied the technical specifications of maintaining one operational RHR train. A dedicated operator was stationed near the common discharge header's return valve to the RWST, throughout the entire test. If a Safety Injection Signal (SIS) occurred, sufficient time existed for operations to close the valve.

The system's existing piping layout and installed monitoring instrumentation did not lend itself well to conducting a full range pump performance test. The location of existing pressure monitoring instrumentation was too far away from the pump and would introduce unacceptable errors in accurately determining the pump's Total Dynamic Head (TDH). To overcome this obstacle and achieve an accurate picture of the pump's performance, highly accurate pressure test gages were installed near the pump and a mathematical model was developed for each pump; using standard equations, correcting for elevation and differences in inlet and outlet piping diameters, and applying the variance error vector method to determine the pressure and flow rate errors upon measured data points TDH (P,Q) and Q. The resultant equations were then used to quantify the amount of error for various combinations of pump suction and discharge pressure gages and installed flow transmitter. This technique allowed the engineers to select a measurement system that would yield very small errors in TDH.

The IST procedures were revised to serve as a "special platform" to include the full range pump performance TDH versus Q equations, requirements for test gages, error analysis, data gathering and analysis methods. The performance tests were successfully conducted on-line and the results were compared to the Nuclear Steam Supply System (NSSS) vendor's minimum acceptance curves. This comparison demonstrated that the pump's hydraulic performance was well in excess of that required by the accident analysis.

The techniques presented in the following discussion can be applied to any type of centrifugal pump (vertical or horizontally installed) to produce high quality tests and results. The equations can be modified to meet unit-specific piping designs and programmed into lap-top personal computers (PCs). Test engineers or technicians can readily use this tool, in a near real-time environment to determine if testing is yielding satisfactory performance curves and meets the acceptance criteria.

DEVELOPMENT OF PUMP MODEL

The development of a simplified pump model was necessary to address limitations in the existing RHR pump's pressure monitoring gages. The instrumentation satisfies IST requirements for measuring a single point.

However, they are insufficient for full range performance tests. The suction and discharge pressure transmitters were located too far from the pump inlet and outlet nozzles, to provide meaningful measurements of pressure and therefore TDH. This is not a new problem for the industry. Stockton et.al (Ref. 1) have pointed out that "...as a general rule, inservice testing has not been a significant consideration in power plant design..." This is also true for our example. In addition, the installed pressure transmitters range and accuracy were insufficient to ensure an accurate measurement of pump performance and provide verification of the vendors minimum acceptance criteria.

A mathematical model of a centrifugal pump was developed for the Total Dynamic Head or TDH across the pump as a function of flow rate Q. The suction and discharge conditions and physical piping / entrance conditions of interest are denoted as 1and 2 respectively. The "TDH equation" will include corrections for inlet / outlet and entrance / exit piping effects, suction / discharge piping centerline elevation differences, and measurement uncertainty of the suction and discharge pressure test gages and installed plant flow transmitters. The measurement uncertainty will be treated using the Variance Error Vector Method (Ref. 2). The pressure gages were installed at pipe taps very close to the pump's suction and discharge nozzles. This minimized the need to make any further corrections to the pump itself. Testing and analysis was further simplified by installing the pressure gages at the pipe centerlines, thereby eliminating the need for making elevation corrections to indicated readings. Karassik et. al. (Ref. 3) gives the following expression for pump Total Dynamic Head:

$$H_d - H_s = \{ (V^2{}_d)/2g + p_d/\gamma_d + Z_d \} - \{ (V^2{}_s)/2g + p_s/\gamma_s + Z_s \} \qquad \text{Eq.-1}$$

Where H_d, H_s, $(V^2{}_d)/2g$, p_d/γ_d, Z_d, $(V^2{}_s)/2g$, p_s/γ_s, and Z_s have their usual meanings.

It can be readily shown that by re-arranging and re-defining terms { Refer to Crane Technical Paper 410 for details (Ref. 4) }, converting units and subscripts ($s = 1$, $d = 2$, $Z = h$, etc..) that a general expression for the pump's Total Dynamic Head or TDH can be written as:

$$TDH\,(Q,P) = A\,(P_2 - P_1) + B\,(Q)^2 + (h_2 - h_1) \qquad \text{Eq.-2}$$

Where:

- $TDH\,(Q,P)$ = Total Dynamic Head, as a function of pressures and flow rate, ft
- $A = 144\ \text{inch}^2/1\ \text{ft}^2/\rho\ (\text{lb/ft}^3)$
- P_2 = Pump discharge pressure, psig
- P_1 = Pump suction pressure, psig
- ρ = Density of water, (lb/ft^3)
- $B = 2.577\text{E-}3(Q)^2 \times [\,(1/d_2)^4 - (1/d_1)^4\,]$, ft
- h_2 = Discharge piping centerline elevation, ft
- h_1 = Suction piping centerline elevation, ft
- Q = Flow rate, gpm
- d_2 = Pump outlet piping inside diameter, inches
- d_1 = Pump inlet piping inside diameter, inches

Parameters P_2, P_1, and Q are test measurements and have instrument measurement uncertainties associated with them; $\pm\,\epsilon_{P2}$, $\pm\,\epsilon_{P1}$, and $\pm\,\epsilon_Q$.

These indicated test measurement errors are defined as follows:

$\pm\,\epsilon_{P2}$ = Discharge Pressure Test Gage Range (psig) x % F.S. (Full Scale) Calibration Accuracy

$\pm\,\epsilon_{P1}$ = Suction Pressure Test Gage Range (psig) x % F.S. (Full Scale) Calibration Accuracy

$\pm\,\epsilon_Q$ = Flow Transmitter Range (gpm) x % F.S. (Full Scale) Calibration Accuracy

The Total Dynamic Head or TDH will also have an uncertainty (calculated) associated with it: $\pm\ \epsilon_{TDH}$.

To evaluate the impact of instrument measurement uncertainties upon TDH, the Variance Error Vector or $\surd$ Var must be calculated. It can be shown (Ref. 2), that the Variance Error Vector for TDH can be written as:

$$\pm\ \epsilon_{TDH} = \surd\ (\text{Var TDH }(P,Q)) \qquad \textbf{Eq.-3}$$

Where Var can be expressed as:

$$\text{Var} = (\partial TDH/\partial P_2)^2 \times (\epsilon_{P2})^2 + (\partial TDH/\partial P_1)^2 \times (\epsilon P_1)^2 + (\partial TDH/\partial Q)^2 \times (\epsilon_Q)^2 \qquad \textbf{Eq.-4}$$

Equation 4 assumes that there are no errors in water density $(\partial TDH/\partial \rho) = 0$ or centerline pipe elevation measurements $(\partial TDH/\partial h_{1\text{ or }2}) = 0$.

Var can now be determined by taking partial derivatives $(\partial$ TDH (P,Q) / $\partial \rho$, ∂Q,..etc..) of Equation 2 and inputting the results into Equation 4, to calculate $\pm\ \epsilon_{TDH}$:

$$(\partial TDH/\partial P_2) = +A \qquad (\partial TDH/\partial \rho) = 0$$
$$(\partial TDH/\partial P_1) = -A \qquad (\partial TDH/\partial h_1) = 0$$
$$(\partial TDH/\partial Q) = 2BQ \qquad (\partial TDH/\partial h_2) = 0$$

Substituting these values into Equation 4 yields the following expression for Var:

$$\text{Var} = (A)^2 \times (\epsilon_{P2})^2 + (-A)^2 \times (\epsilon_{P1})^2 + (2BQ)^2 \times (\epsilon_Q)^2 \qquad \textbf{Eq. -5}$$

An expression for $\pm\ \epsilon_{TDH}$ can now be determined by substituting the results of Equation 5 into Equation 3:

$$\pm\ \epsilon_{TDH} = \{ (A)^2 \times (\epsilon_{P2})^2 + (A)^2 \times (\epsilon_{P1})^2 + (2BQ)^2 \times (\epsilon_Q)^2 \}^{1/2} \qquad \textbf{Eq.-6}$$

Equation 6 can be factored and terms combined to yield the following expression for $\pm\ \epsilon_{TDH}$:

$$\pm\ \epsilon_{TDH} = \pm\ (2ABQ) \times \{\ [\ (\epsilon_{P2})^2 + (\epsilon_{P1})^2\ ] / 4B^2Q^2 + [\ (\epsilon_Q)^2 / A^2\ ]\ \}^{1/2} \qquad \textbf{Eq.-7}$$

Equation 7 represents the TDH error due to the suction and discharge pressure test gages and flow transmitter measurement uncertainties.

Combining Equations 2 and 7, letting $\Delta P = (P_2 - P_1)$ and $\Delta h = (h_2 - h_1)$, a general expression for TDH (P,Q) $\pm\ \epsilon_{TDH}$

(Eq.-8) can now be written:

$$TDH(P,Q)\ \pm \epsilon_{TDH} = A(\Delta P) + BQ^2 + (\Delta h) \pm (2ABQ)\{[(\epsilon_{P2})^2 + (\epsilon_{P1})^2]/4B^2Q^2 + [(\epsilon_Q)^2/A^2]\}^{1/2} \qquad \textbf{Eq.-8}$$

Where:
A is now in units of (inch^2-ft^3 / ft^2-lb)
P is in units of (psig)
B is now in units of (ft- gpm^{-2})
Q is in units of (gpm)
ϵ_{P2} is in units of (lb-inch^{-2})
h is in units of (ft)
ϵ_{P1} is in units of (lb-inch^{-2})
ϵ_Q is in units of (gpm)
TDH (P,Q) is in units of (ft)
ϵ_{TDH} is in units of (ft)
$\Delta P = (P_2 - P_1)$
$\Delta h = (h_2 - h_1)$

APPLICATION OF MODEL TO ACTUAL PLANT CONDITIONS

To derive a RHR pump's TDH expression, unit-specific parameters,conditions, assumptions, and design parameters applicable to each RHR pump are inputted into Equation 8 and the expression is calculated.

RHR Pump A:

The following test instruments, conditions, assumptions, and design parameters were used in Equation 8 to model the A RHR pump:

- Suction pressure gage range 0-60 psig, $\pm$ 0.2 % F.S. accuracy
- Discharge pressure gage range 0-500 psig, $\pm$ 0.2 % F.S. accuracy
- Flow Transmitter range 0-6000 gpm, $\pm$ 2.0 % F.S. accuracy

ρ = 62.224, (lb/ft^3) @ 79.59 °F (Assumption)

h_2 = 9.31, ft $\quad\quad$ h_1 = 6.50, ft

d_2 = 10.020, inches $\quad\quad$ d_1 =13.124 , inches

A = 144 inch^2/1 ft^2/ρ (lb/ft^3) = 2.314

B = 2.577E-3(Q)^2 x [(1/d_2)^4 - (1/d_1)^4] = 1.704E-7 x Q^2, ft

ϵ_{P2} = 1.0 (lb-inch^{-2}) $\quad\quad$ ϵ_{P1} = 0.12 (lb-inch^{-2})

ϵ_Q = 120 (gpm)

$\Delta P = (P_2 - P_1)$ $\quad\quad$ $\Delta h = (h_2 - h_1)$

$$TDH(P,Q) \pm\epsilon_{TDH} = 2.314(\Delta P) + 1.704E\text{-}7Q^2 + 2.81 \pm (7.886E\text{-}7Q)\{[8.692E12]/Q^2 + [2689.1]\}^{1/2} \quad \textbf{Eq.-9}$$

RHR Pump B:

The following test instruments, conditions, assumptions, and design parameters were used in Equation 8 to model the B RHR pump:

- Suction pressure gage range 0-60 psig, $\pm$ 0.2 % F.S. accuracy
- Discharge pressure gage range 0-500 psig, $\pm$ 0.2 % F.S. accuracy
- Flow Transmitter range 0-6000 gpm, $\pm$ 2.0 % F.S. accuracy

ρ = 62.224, (lb/ft^3) @ 79.59 °F (Assumption)

h_2 = 9.31, ft h_1 = 6.50, ft

d_2 = 7.981, inches d_1 = 13.124, inches

A = 144 inch^2/1 ft^2/ρ (lb/ft^3) = 2.314

B = 2.577E-3$(Q)^2$ x [$(1/d_2)^4$ - $(1/d_1)^4$] = 5.534E-7 x Q^2, ft

ϵ_{P2} = 1.0 (lb-inch^{-2}) ϵ_{P1} = 0.12 (lb-inch^{-2})

ϵ_Q = 120 (gpm) $\Delta P = (P_2 - P_1)$ $\Delta h = (h_2 - h_1)$

$$TDH(P,Q) \pm \epsilon_{TDH} = 2.314(\Delta P) + 5.534E\text{-}7Q^2 + 2.81 \pm (2.561E\text{-}6Q)\{[8.824E12]/Q^2 + [2689.1]\}^{1/2} \quad \textbf{Eq.-10}$$

PERFORMANCE OF POST-MAINTENANCE IST AND RESULTS

Tests were successfully performed on the RHR pump's following on-line maintenance and installation of a refurbished electric motor, a new impeller, and seal packages. Existing IST procedures were modified to incorporate the full range pump performance equations and required test instrumentation. High precision pressure gages (± 0.2 % F.S. accuracy) were installed at vent and drain taps, which are located very close to the pump's inlet and outlet nozzles. The gages were mounted at the centerline of the piping to eliminate the need to make elevation corrections to recorded readings. Installed plant pressure transmitters and local pressure indicators were not used. As previously discussed, the existing IST procedures and associated monitoring instruments satisfy Inservice testing requirements. However, the taps for these devices are too far away from the inlet and outlet of the pump, their range and accuracy insufficient to ensure accurate testing was achieved, require both elevation and axial pressure-flow loss calculations to be of any use, and would introduce additional errors in the determination of TDH. For example, the installed discharge pressure gage's tap location was located a significant distance away from the pump discharge nozzle, and due to the pressure piping losses, reads 20 psig lower than a test gage mounted near the discharge nozzle. Due to this inherent design configuration, it was decided to use local high precision test gages. This approach yields more accurate testing and a clearer indication of the pump's TDH behavior. The pump's installed flow transmitter was used to measure flow (± 120 gpm). The inlet and outlet water temperatures were also recorded. All readings were recorded manually. No special training, instrumentation, or calibrations were required to make use of this method or measurement system.

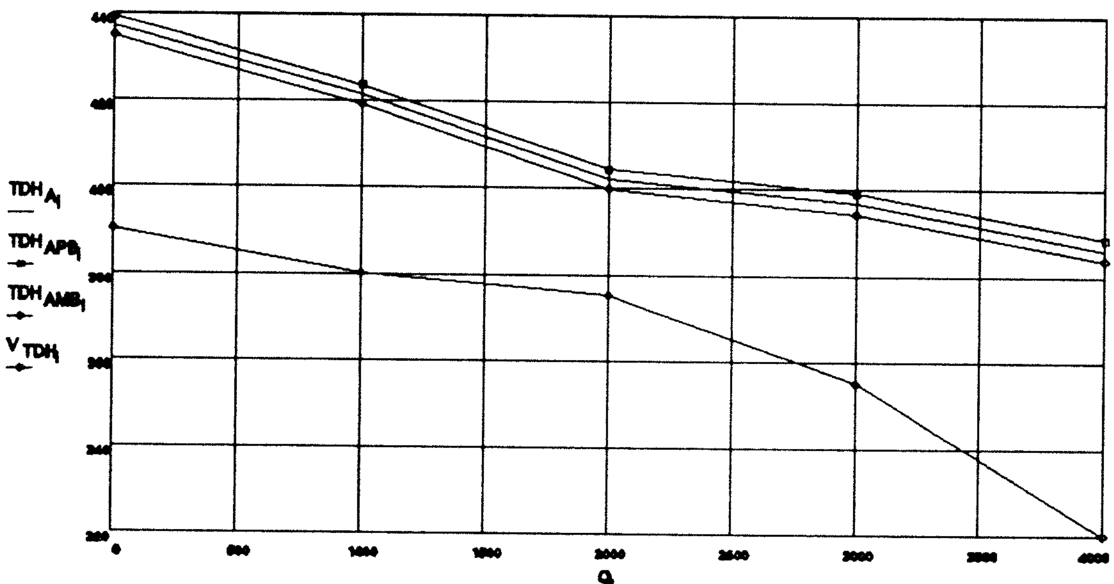
TEST RESULTS

The test measurements for the RHR pumps are shown in Table 1. Values for the measurement uncertainty in TDH and flowrate Q are also listed. Figure 2 is a graph of RHR Pump A's TDH (P,Q) versus Q. Also plotted is a graph representing the minimum acceptance curve. This graphical comparison indicates that the pump's performance exceeds that which is required. Similarly, Figure 3 indicates that RHR Pump B's performance exceeds the vendors minimum requirements.

TABLE 1 PWR RHR PUMP TEST MEASUREMENTS AND CALCULATIONS

P_2 (psig)	P_1 (psig)	Q (gpm)	TDH (Q,P) (ft)	$\pm\ \epsilon_{TDH}$ (ft)	$\pm\ \epsilon_Q$ (gpm)
RHR PUMP A					
223	35.5	650	436.757	2.326	120
213	32.5	2000	421.169	2.327	120
203	31.25	3125	401.904	2.329	120
200	30.75	3540	396.59	2.33	120
195	30.4	3680	386.002	2.33	120
RHR PUMP B					
P_2 (psig)	P_1 (psig)	Q (gpm)	TDH (Q,P) (ft)	$\pm\ \epsilon_{TDH}$ (ft)	$\pm\ \epsilon_Q$ (gpm)
227	35.5	650	446.175	7.608	120
213	32.3	2087	423.36	7.613	120
202	31.1	3015	403.303	7.618	120
196	30.5	3436	392.31	7.621	120
192	30.1	3717	385.092	7.624	120

FIGURE 2 RHR A PUMP PERFORMANCE CURVE



Equation 9

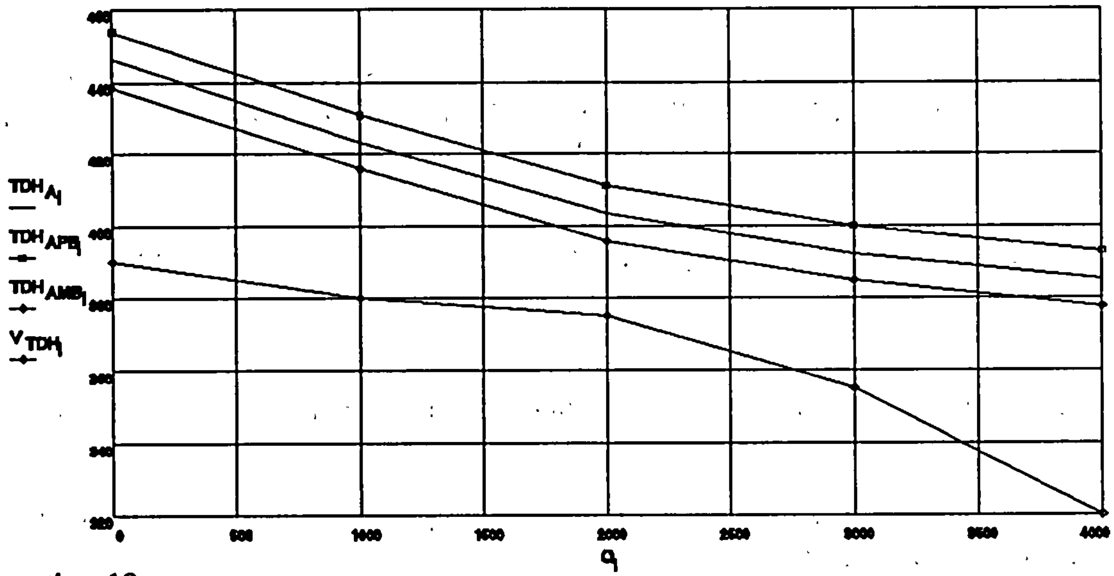
THD_{Ai} = RHR Pump A TDH (ft) $\quad$ Q_i = Flow Rate (gpm)

THD_{APBi} = RHR Pump A TDH + ϵ_{TDH} (ft)

THD_{AMBi} = RHR Pump A TDH - ϵ_{TDH} (ft)

V_{TDHi} = Vendor Minimum Acceptance TDH (ft)

FIGURE 3 RHR B PUMP PERFORMANCE CURVE



Equation 10

THD_{Ai} = RHR Pump B TDH (ft) Q_i = Flow Rate (gpm)

THD_{APBi} = RHR Pump B TDH + ϵ_{TDH} (ft)

THD_{AMBi} = RHR Pump B TDH - ϵ_{TDH} (ft)

V_{TDHi} = Vendor Minimum Acceptance TDH (ft)

SUMMARY AND CONCLUSIONS

On-line pump performance testing of safety-related pumps following major maintenance or refurbishment can be successfully completed with minimal measurement errors. Recent on-line testing of two PWR RHR pumps have demonstrated this approach is viable and produces high-quality test results. The methods presented in this paper can be applied to any centrifugal pump test configuration, to verify operability and acceptance criteria. The method uses readily available test instrumentation, first-principle data collection and analysis techniques. The variance error vector method can be used to evaluate pump measurement system errors and select the proper combination of pressure and flow measuring instruments (full scale range and accuracy).

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Examination of Pump Failure Data in the Nuclear Power Industry

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Abstract

There are several elements that are critical to any program which is used to optimize the availability and reliability of process equipment. Perhaps the most important elements are routine monitoring and predictive maintenance elements. In order to optimize equipment monitoring and predictive maintenance, it is necessary to fundamentally and thoroughly understand the principal failure modes for the equipment and the effectiveness of alternative monitoring methods.

While these observations are general in nature, they are certainly true for the "heart" of fluid systems - pumps. In recent years, particularly within the last decade, the capabilities and ease of use of previously existing pump diagnostic technologies, such as vibration monitoring and oil analysis, have improved dramatically. Newer technologies, such as thermal imaging, have been found effective at detecting certain undesirable or degraded conditions, such as misalignment and overheated bearings or packing.

The ASME Code and NRC regulatory requirements have been, like essentially all similar code and regulatory bodies, conservative in their adoption or endorsement of newer technologies. The requirements prescribed by the Code and endorsed by the NRC have, in their essence, changed only minimally over more than a dozen years.

In light of the contrasting availability of newer and improved technologies against the background of both relatively stationary requirements and suggestions for new approaches (such as risk-based alternatives), it is a particularly propitious point in time to assess the effectiveness of historically implemented pump monitoring methods in the nuclear industry.

As a follow-on to studies of check valve failure experience in the nuclear industry that have proven useful in identifying the effectiveness of alternative monitoring methods, a study of nuclear industry pump failure data has been conducted. The results of this study, conducted for the NRC by Oak Ridge National Laboratory, are presented. The historical effectiveness of both regulatory required and voluntarily implemented pump monitoring programs are shown. The distribution of pump failures by application, affected area, and level of significance are indicated. Apparent strengths and weaknesses of alternative monitoring methods are discussed.

Background, Scope, and Methodology

Although the design and normal operating functions of fluid systems used at current generation reactors are diverse, almost all normally operating and standby fluid systems share the common feature of depending upon pumps to provide motive power for the process fluid. Malfunctions of other components, such as valves, instrumentation, and controls can often be minimized or overcome by human intervention. In the case of pumps and their drivers, however, many failures cannot be dealt with by manual interaction.

Recognizing the importance of reliable pump operation in these systems, ORNL undertook a study for the NRC of pump failure data available from the Nuclear Plant Reliability Data System (NPRDS), a component failure database maintained by the Institute of Nuclear Power Operations. The pump study used, as closely as possible, the same methodology that had previously been applied in analyzing check valve failures [1, 2].

The pump failure data review [3] studied and characterized failures occurring in the years 1990-1993 of centrifugal pumps that are used in safety-related service in several critical systems at BWR and PWR plants. The systems included are identified in Table 1.

Table 1. Systems included in the pump failure study

PWR plants	BWR plants
• Auxiliary feedwater (AFW)	• Component cooling water (CCW)
• Component cooling water (CCW)	• High pressure coolant injection (HPCI)
• Containment spray (Cont. spray)	• Emergency service water (ESW)
• Charging/high pressure safety injection (CVCS/HPSI)	• Low pressure core spray (LPCS)
• Emergency service water (ESW)	• Reactor core isolation cooling (RCIC)
• Low pressure safety injection/residual heat removal (RHR)	• Low pressure coolant injection/residual heat removal (RHR)

In addition to studying the failures of the pumps themselves, the failures of motor drives used for these pump applications were studied. Non-motor drives (principally turbines used on AFW, RCIC, and HPCI pumps) were not included in the study because of detailed examinations of turbine drive experience in previous studies [4, 5, 6]. These studies indicated that turbine drives used in standby applications have been (relative to motors) unreliable. A study of AFW system operating experience [4] indicated that pump drives were responsible for almost 40% of system degradation, and that turbine drives were responsible for about 75% of all drive problems. It should be noted that the turbines themselves have been found to be rugged, durable components; it is control features, such as speed control and governor valves and trip/throttle valves that have been problematic.

A total of 7210 pump-years of experience was accumulated by the studied pumps during the 1990-1993 period (2405 at BWRs and 4805 at PWRs). There were 797 failures of the studied pumps and 143 failures of the associated motors reported to the NPRDS database during the period.

The failures were characterized by reading the failure narratives and categorizing the failures according to several important features that are briefly described in Table 2. Not all these features are discussed in this paper, but are included (and more fully described) in the Ref. 3 report.

Table 2. Failure characterization features

Feature	Coded in NPRDS?	Categories
Extent of degradation	No	Failures were classified into 6 levels of degradation, ranging from minor problems that could exist essentially indefinitely without adversely affecting reliability to complete inability to operate.
Detection process	No	Eleven categories were established including processes such as discovery by operators on routine rounds, discovered by failing to meet acceptance criteria during surveillance/in-service testing, and through remote alarm or annunciation. The categories were subdivided into three general detection method groups: • Discovered during regulatory required testing, • Discovered through plant programmatic monitoring that was not explicitly required by regulation, and • Discovered through non-programmatic means (such as demand failures).
Specific indicator or symptom	No	Eight pump and five motor categories were established for this feature. Examples are: • Hot bearing • Inadequate hydraulic performance • Failure to run or start • Excessive noise or vibration
Affected area	No	Eight pump and nine motor categories were established for the affected area feature. Example categories are: • Bearing • Shaft, coupling, or keys • Internals
Reactor type	Yes	Boiling water reactor (BWR) or pressurized water reactor (PWR)
Manufacturer	Yes	All motor and pump manufacturers are coded in NPRDS. Experience of principal suppliers of studied motors and pumps was evaluated.
Age group	Yes	Age at failure was calculated from inservice date and failure discovery date fields in the NPRDS database. Five year failure age groups were established.
System of service	Yes	NPRDS codes by systems according to individual nuclear steam supply system suppliers. The systems listed in Table 1 are generic names for the NPRDS coded systems.

Population distributions by some of these categories, such as age group and manufacturer, can significantly influence the numbers of failures occurring. In order to provide a more accurate representation of failure experience, some of the results presented are normalized to population. A term used herein to depict the normalized failure rate is "Relative failure rate." This value is calculated by dividing the failures within a particular category (such as manufacturer) per unit time by the overall failure rate (for all manufacturers). Thus, a relative failure rate of 1.4 for a particular manufacturer suggests that the pumps made by that manufacturer have a failure rate that is 40% greater than the overall average.

Results

Extent of Degradation

As noted in Table 2, all failure records were assigned to one of six levels of degradation. These ranged from conditions that could exist essentially indefinitely with minimal or no effect on the pump's ability to meet its functional requirements to conditions where the pump would not operate at all. For simplicity, the failures are segregated into two general categories – all failures and significant failures only. Those failures that are classed as significant are those for which the pump either would not operate at all, would not operate to the level required to perform its safety function, or was operating at a degraded level with near-term continued operation in jeopardy.

The reported failure rates and the distribution of all pump and motor failures by extent of degradation for BWR and PWR plants are shown in Figure 1. It should be emphasized that although the rates are shown in absolute terms, the author strongly discourages blindly using these failure rate values for other purposes without considering other factors such as reporting practices and recovery time. The extent to which misleading results could be developed from the data at this level is indicated by the differences in the *All failures* and *Significant failures* categories. A large fraction of all reported pump failures were associated with relatively minor (from the standpoint of affecting the pump's ability to meet its required functions) seal or packing leakage that did, however, require removal of the pump from service to correct. For such circumstances, other than the time during which the pump was being repaired, the pump would have been capable of fulfilling its safety-related functions. To simply use the *All failures* failure rate in performing additional calculations would be misleading. For example, if the effectiveness of different monitoring methods were assessed based on the *All failures* distribution, simple visual examination might be found to be the most important detection tool available (since it would obviously be an inexpensive but effective means of detecting minor seal or packing leakage). While the benefits of routine walkdowns and visual/audible observations are great, their relative importance to the detection of significant pump problems (particularly in the early stages of degradation) could certainly be overstated.

There were clearly more pump than motor failures (more than five times as many, considering all failures). However, there were only three times as many pump failures compared to motor failures for those failures classed as significant. This is an important finding, since the existing ASME Code test requirements for pumps [7] do not explicitly address motor monitoring. ASME Working Group OM-6 (charged with responsibility for pump test requirements) is beginning to consider the merits of specifically incorporating motor testing into the Code. While there are other factors to be considered, such as the practicality of periodic motor testing and whether periodic inservice testing or monitoring would be effective at detecting motor degradation prior to failure, the failure data clearly indicate that motors are an important factor in overall pump drive train reliability.

Another finding of particular interest is that although motor failure rates for BWR and PWR units are similar, the pump failure rate for PWRs is approximately double that of BWR units. This was found to be true for all failures and significant failures only.

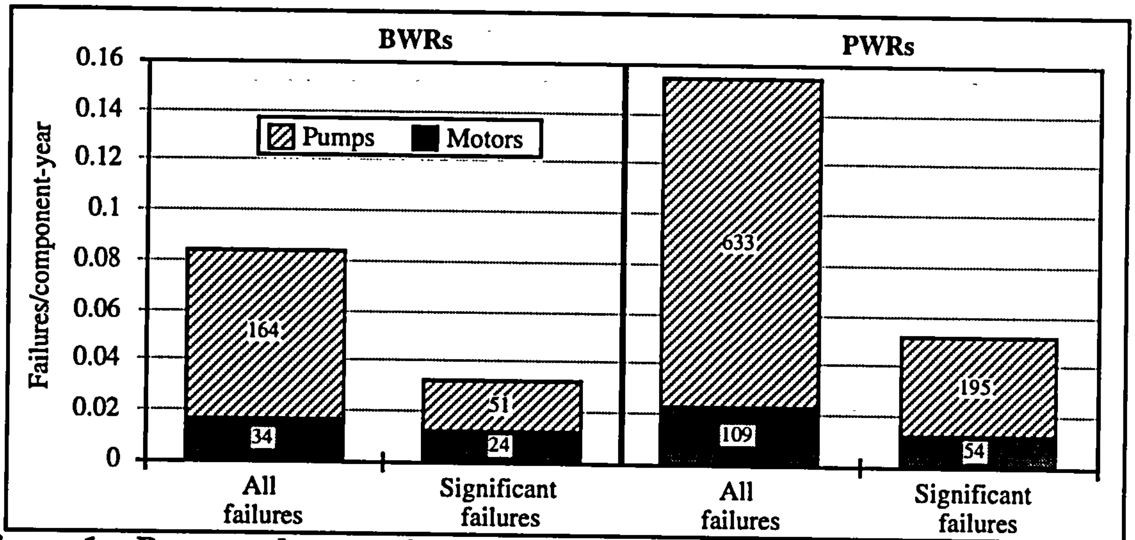

Figure 1. Pump and motor failures by extent of degradation and reactor type.

System

The system in which pumps are used was found to be an important factor in regards to both failure rate and mode. Figure 2 shows the absolute failure rates for pumps and motors by system for BWR and PWR plants. Some features of particular note are:

- ESW pumps and motors at both plant types have substantially higher overall failure rates. The failure rates for significant failures in ESW are almost three times those of the next closest system (CCW) at BWRs, almost twice that of the next highest system at PWRs (AFW).
- The failure rate for BWR system pumps is strongly related to the system usage. For example, normally operating systems (such as ESW* and CCW) have higher failure rates than do systems that are occasionally used (such as RHR and RCIC), which in turn have higher failure rates than systems whose primary usage is for testing (HPCI and LPCS).
- In general, the same system usage effect appears in the PWR data. The primary exception is AFW, which has both an overall and significant failure rate that is comparable to those of CCW and CVCS/HPSI. Note that the CVCS/HPSI data represents a mixture—at some plants, the HPSI pumps are used solely for testing or emergency response, while at others, they serve the dual function of charging. Thus, the actual usage of CVCS/HPSI pumps can be at either extreme.

Age Group

The relative failure rate as a function of component age for all pumps (motors not included) at BWR and PWR plants is shown is Figure 3. For the BWR plants, there is a clear age-related trend in that the failure rate drops significantly after a period of infant mortality. The same trend appears to occur, to a lesser extent, for the PWR plants in the transition from <5 to ≥5 and <10 year group, but then reverses in subsequent age groups. For the significant failures only, the PWR plant failure rate trend as a function of age group resembles the classical "bath tub" shape.

* ESW pump usage varies among plants (BWR and PWR), ranging from being used only for testing or emergency conditions to normally in service. Although a careful review of usage patterns by plant was not performed, a qualitative survey indicated that most problems occurred with normally operating pumps.

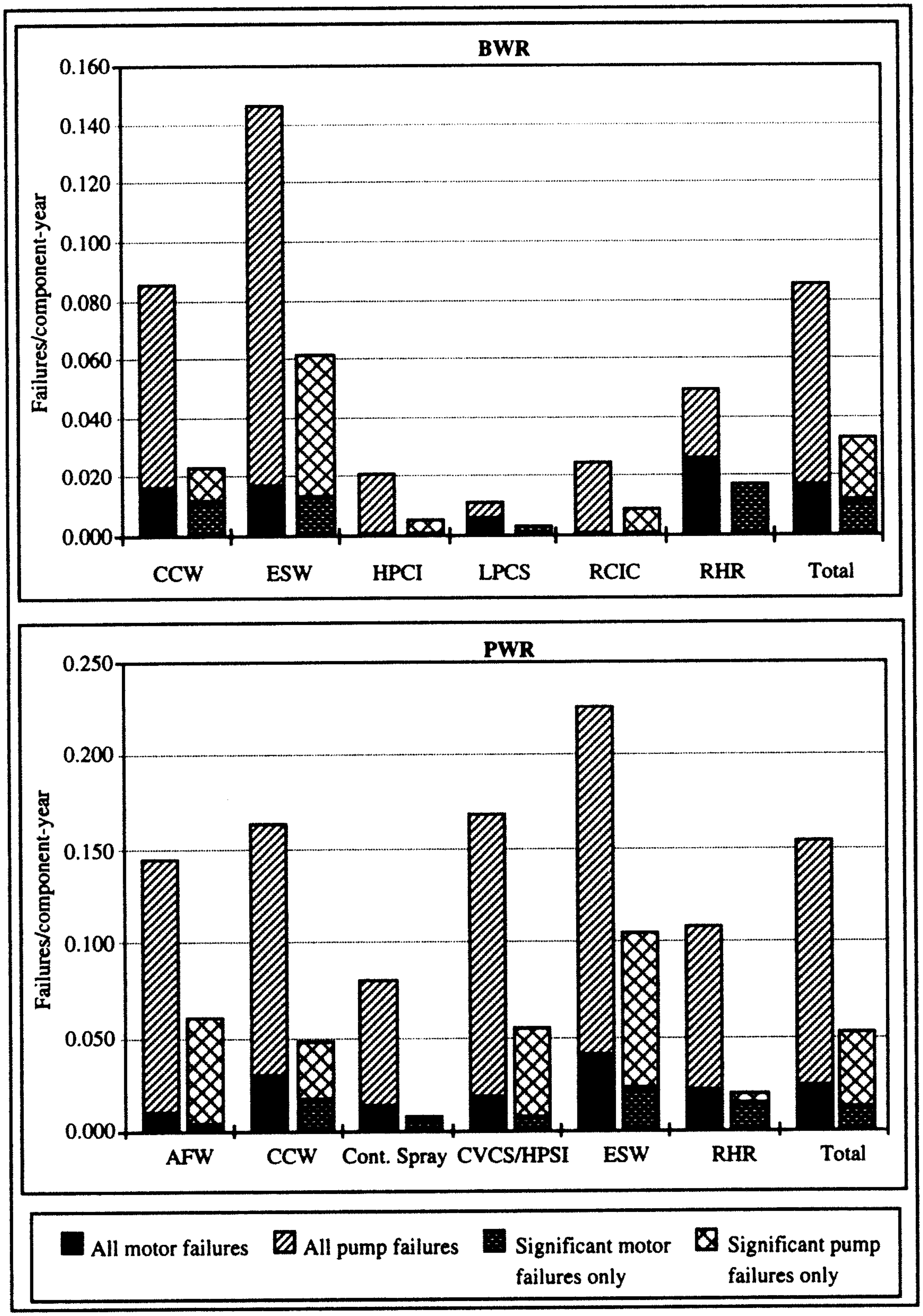

Figure 2. Absolute failure rates for pumps and motors by system and reactor type. Note that the y-axis scaling is different for the two graphs.

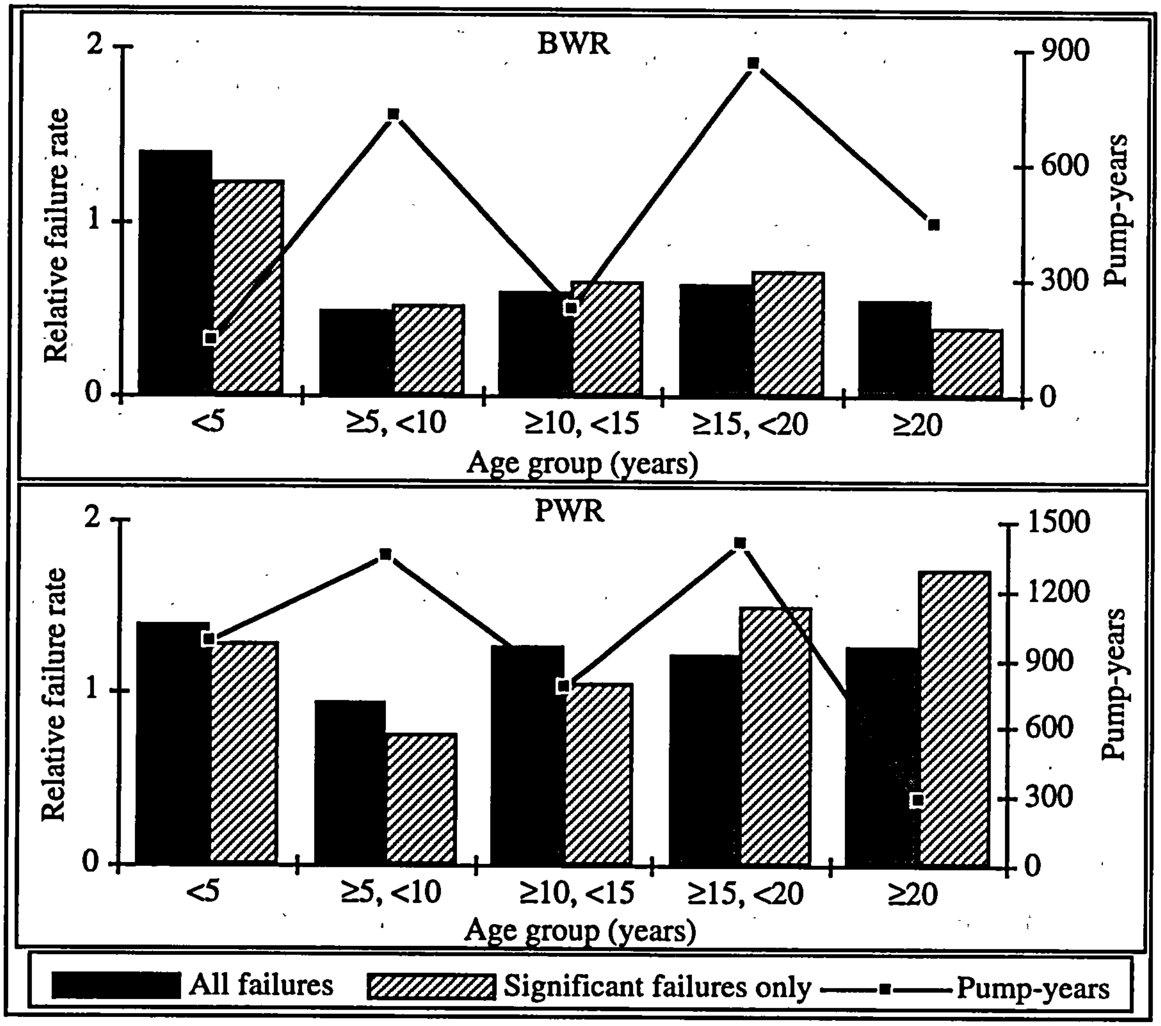

Figure 3. Relative failure rates for pumps by age group and reactor type.

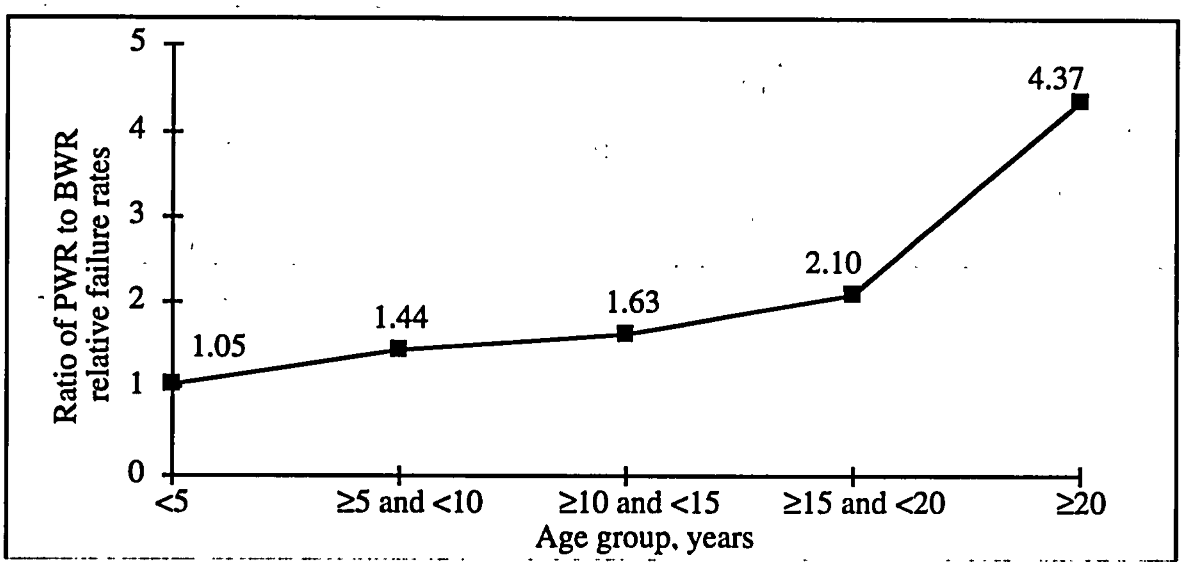

Figure 4. Ratio of PWR to BWR relative failure rates as a function of plant age for significant failures.

The fact that BWR plants have had better experience than PWR plants is clearly illustrated in Figure 4, where the ratio of PWR to BWR failure rates as a function of age group for significant failures is shown. During the early years of operation, the performance of pumps at BWR and PWR units is similar. But over time, the failure rate relationship changes dramatically. One possible explanation for this pattern is the difference in test conditions at BWR and PWR units, for at least some systems. At BWRs, most systems can be tested at or near design flow conditions, which are, in turn, usually close to pump best efficiency point (BEP) flowrates. In contradistinction, many PWR pumps are tested at minimum flow conditions. There are two important factors associated with this disparity that could contribute to increased failure rates vs. age for the affected PWR plant pumps. First, testing at minimum flow conditions provides little or no useful information about pump hydraulic performance. Second, at minimum flow conditions, pumps normally experience adverse hydraulic loading that may accelerate aging of a variety of parts, including seals, bearings, impeller vanes, diffuser vanes or volute tongues, and wear surfaces.

Affected area

All failures were characterized by the affected area. Table 3 presents the numbers of failures by extent of degradation and reactor type and the ratio of normalized failure rates at BWR and PWR plants. Almost half of all reported failures involved seal or packing leakage; in over 80% of the seal/packing failures, the leakage was the only noted problem. It is notable that PWR pumps had higher failure rates in all areas than BWR pumps (excluding the "Unknown" category).

Tables 4 and 5 tabulate the numbers of significant failures by affected area and system for PWR and BWR plants. Of 91 significant failures involving internals (at BWR and PWR plants combined), 71 were in ESW pumps; 18 of 24 shaft/coupling/key failures occurred in ESW pumps. However, only 25 of 92 significant failures in which a bearing was affected involved ESW pumps.

Table 3. Number of pump failures by affected area and reactor type and ratio of PWR to BWR failure rates.

Affected area	All failures BWR	All failures PWR	Significant failures only BWR	Significant failures only PWR	Ratio of the PWR to BWR failure rate (sign. failures only)
Alignment/balance	2	5	1	3	1.50
Bearing	23	133	18	74	2.06
Shaft, coupling, keys	8	19	6	18	1.50
Internals	26	77	22	69	1.57
Oil leak	13	11	0	0	N/A
Other	14	133	4	14	1.75
Seal/packing	87	280	5	40	4.00
Unknown	2	2	2	2	0.50
Total	164	633	51	195	1.91

Table 4. **Significant PWR pump failures by affected area and system.**

Affected area	AFW	CCW	Cont. Spray	CVCS/ HPSI	ESW	RHR	Total
Alignment/balance	2	0	0	1	0	0	3
Bearing	14	22	2	22	12	2	74
Shaft, coupling, keys	0	2	0	4	12	0	18
Internals	6	5	0	9	49	0	69
Oil leak	0	0	0	0	0	0	0
Other	7	0	0	6	1	0	14
Seal/packing	15	7	1	7	8	2	40
Unknown	0	0	0	0	2	0	2
Total	42	26	2	44	77	4	195

Table 5. **Significant BWR pump failures by affected area and system.**

Affected area	CCW	ESW	HPCI	LPCS	RCIC	RHR	Total
Alignment/balance	0	1	0	0	0	0	1
Bearing	3	13	1	0	1	0	18
Shaft, coupling, keys	0	6	0	0	0	0	6
Internals	0	22	0	0	0	0	22
Oil leak	0	0	0	0	0	0	0
Other	0	4	0	0	0	0	4
Seal/packing	0	5	0	0	0	0	5
Unknown	0	2	0	0	0	0	2
Total	3	46	1	0	1	0	51

Method of detection

Three general methods of detection were used – regulatory/code, plant programmatic, and nonprogrammatic. Failures included under the regulatory/code category were those that were detected during regulatory/code required testing by means prescribed by the ASME Code. The failures detected by plant programmatic means are those that were detected by plant programs that are routinely implemented, but not mandated by regulation. Failures detected by the third category, nonprogrammatic, are those that were detected by neither of the other two methods. It should be noted that some of the plant programmatically detected failures were found during the process of preparing for or conducting regulatory required testing, but were noted not because of the regulation or code criteria, but because of good practices and observation patterns of utility employees.

Figures 5-7 show the numbers of failures by detection means for all failures (Fig. 5), all failures except those involving only seal or packing leakage (Fig. 6), and significant failures only (Fig. 7). Although many of the failures detected by plant programmatic means involved seal or packing leakage only where visual observation was the specific means of detection, Figures 6 and 7 indicate that plant programmatic means were also the leading means for detecting other failures.

Figure 8 illustrates the distribution of significant failures by detection means and affected area for three critical areas. The number of failures for each category is shown at the top of each chart column. Over five times as many bearing problems at PWR plants were detected by plant programmatic means as by regulatory/code testing. Further review of the bearing failures detected by plant programmatic means indicated that the principal ways of detecting the problems were routine oil monitoring (either sampling or simple visual observation) and operators noticing hot or noisy/excessively vibrating bearings. Of the 10 failures involving bearings at PWRs in which

regulatory/code monitoring was employed, 5 were detected by elevated temperature (for which monitoring is not required in the more recent versions of the Code), 1 by failure to start, and only 4 by vibration.

There were surprisingly few reports of bearing problems detected by non-Code based vibration monitoring programs that are more focused on specific spectral vibration bands (there are several alternative systems, but fundamentally all rely upon the fact that important bearing-related energy occurs in discrete spectral regimes). One possible reason for this is that bearing degradation detected under such programs would normally be trended with time and the bearing replaced before the level of degradation reached the threshold for NPRDS reporting (i.e., the degradation could be considered incipient). In contrast, if a bearing had deteriorated to the point, for example, that it significantly affected overall vibration amplitude (all that is required to be monitored by the Code), the degradation would normally be fairly severe, and therefore reportable.

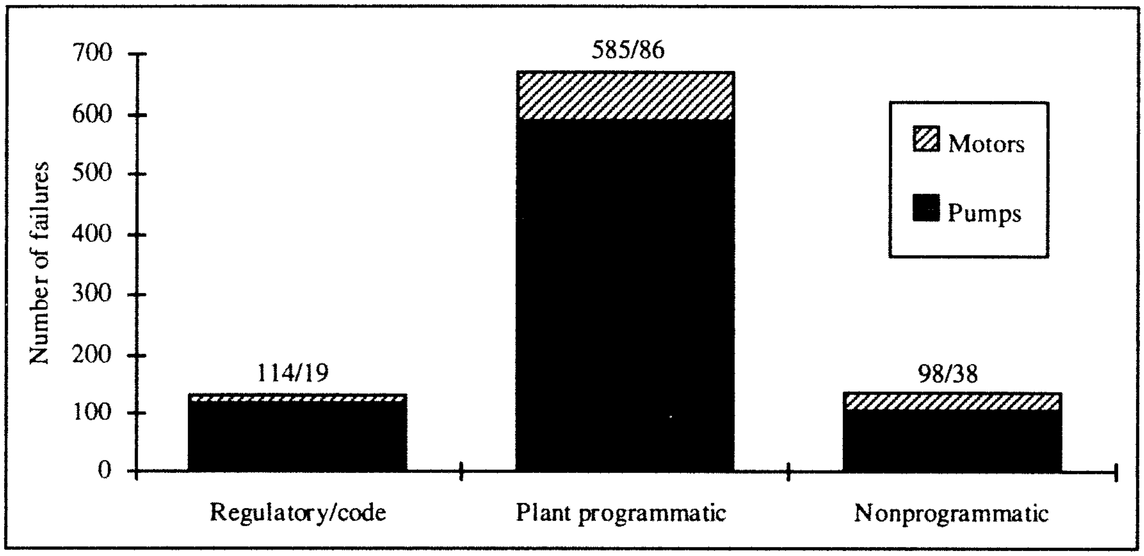

Figure 5. Distribution of failures by general detection method (all failures).

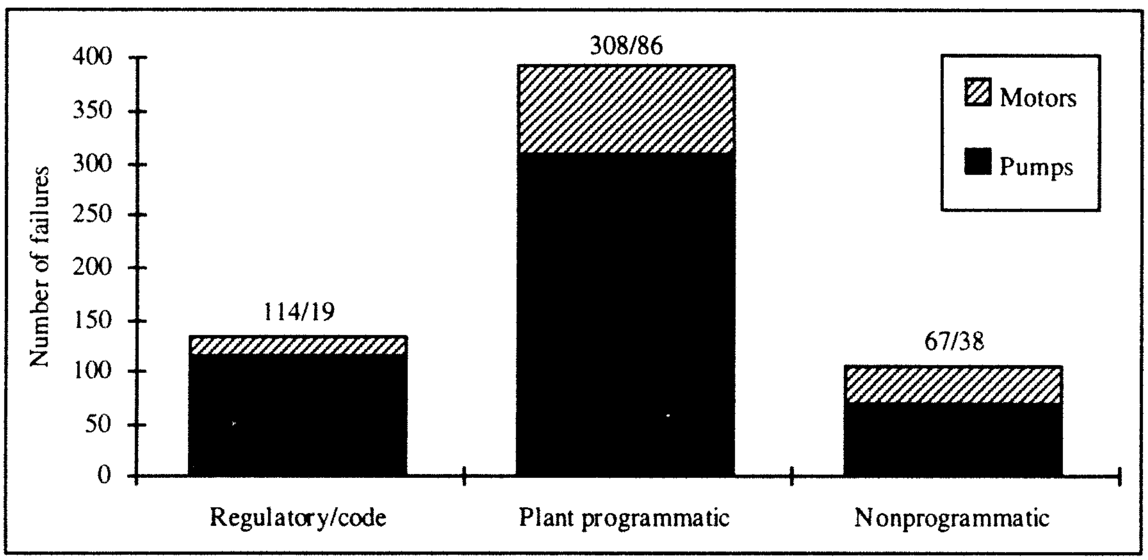

Figure 6. Distribution of failures by general method of detection (seal failures excluded).

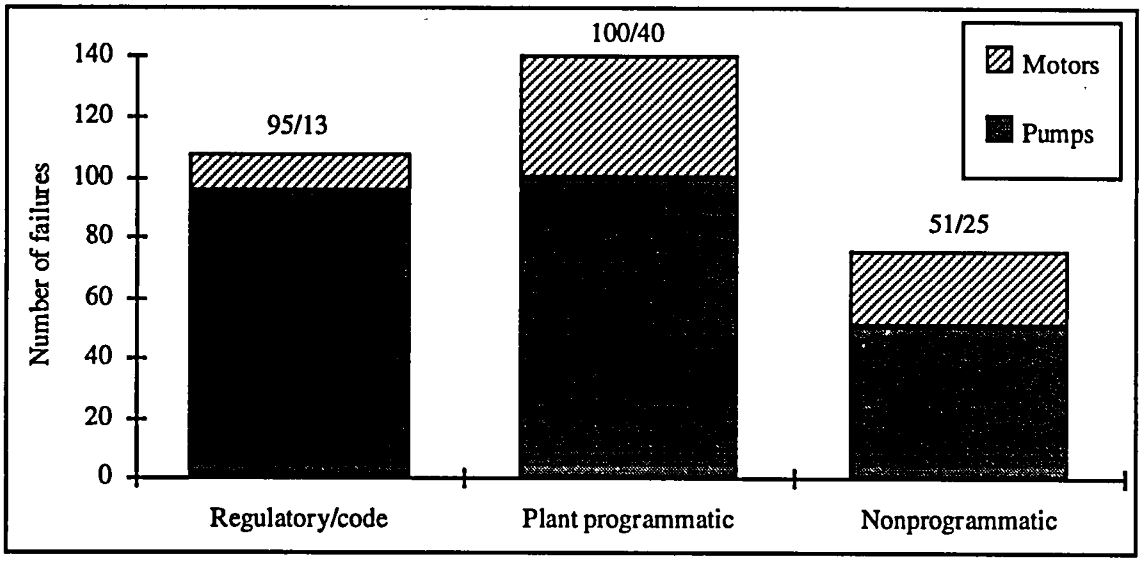

Figure 7. Distribution of failures by general method of detection (significant failures only).

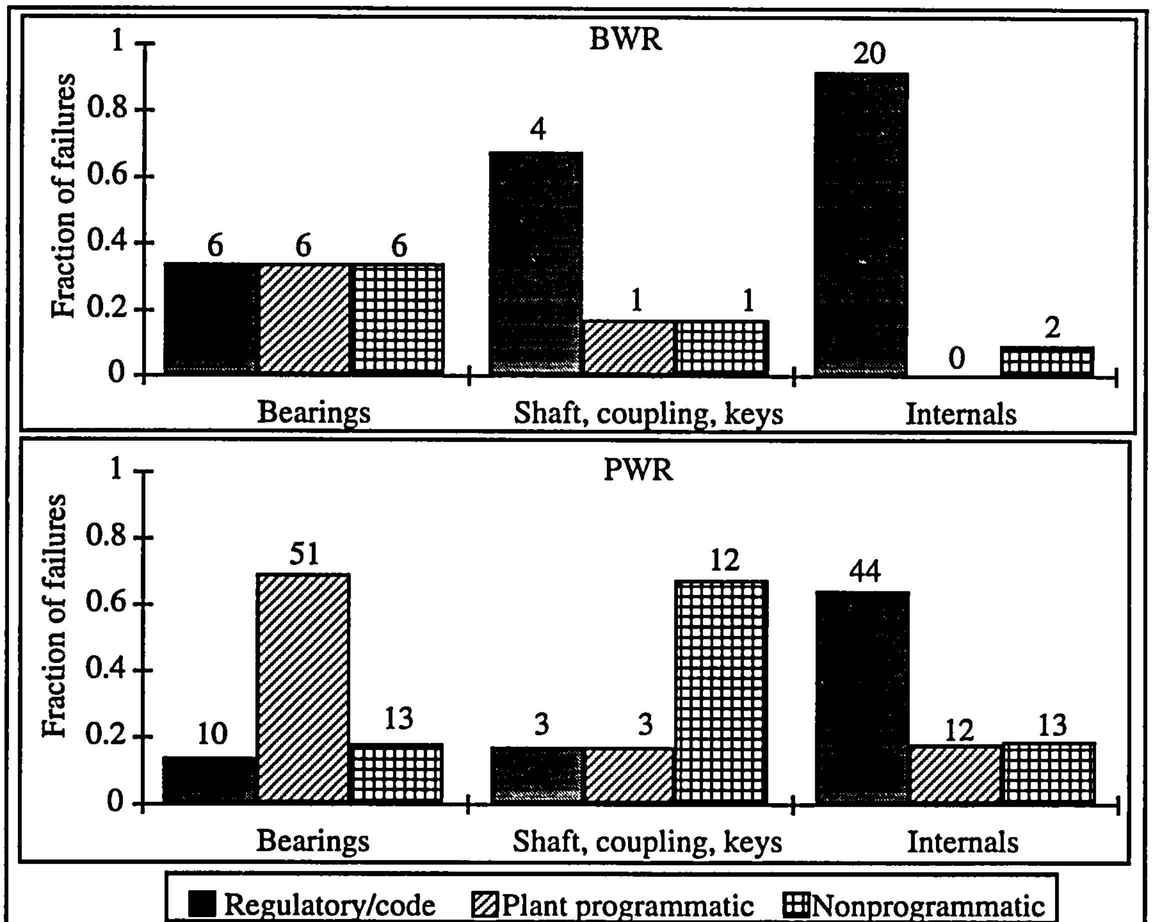

Figure 8. Distribution of significant pump failures by affected area and method of detection.

Tables 6 and 7 show the distribution of significant pump failures by system and detection process. Systems that are not included in these tables had no significant level failures.

Table 6. Significant PWR pump failures – by system and detection process

General detection process	AFW	CCW	Cont. Spray	CVCS/ HPSI	ESW	RHR	Total
Regulatory/code	11	2	1	4	45	0	63
Plant programmatic	27	21	1	25	12	3	89
Nonprogrammatic	4	3	0	15	20	1	43
Total	42	26	2	44	77	4	195

Table 7. Significant BWR pump failures – by system and detection process

General detection process	CCW	ESW	HPCI	RCIC	Total
Regulatory/code	0	30	1	1	32
Plant programmatic	2	9	0	0	11
Nonprogrammatic	1	7	0	0	8
Total	3	46	1	1	51

From the information shown in these tables and further review of the data, several particularly useful insights into the effectiveness of monitoring practices were gained:

- As was shown previously, ESW pumps had the highest failure rates of any of the systems studied. ESW pumps were an even more important contributor for those failures detected by regulatory/code mandated means. There were 95 significant level pump failures detected by regulatory/code mandated testing; 75 were in the ESW system.
- Of the 95 regulatory/code detected failures, 32 occurred at BWR plants. Of the 32 BWR failures, 30 were in the ESW system; 20 of the 30 ESW failures involved internals degradation.
- Of the 63 pump failures detected by regulatory/code required monitoring at PWR plants, 45 occurred in the ESW system. Of these 45 ESW pump failures at PWR plants, 41 involved internals degradation.
- Summarizing the last two bullet items, almost two-thirds (61 of 95) of the failures detected by regulatory/code required monitoring involved internals degradation of ESW pumps.
- Further review of these failures showed that 39 of the 95 regulatory/code detected pump failures were in ESW pumps at only five units. Thus, over 40% of all significant pump failures detected by regulatory/code required testing were found in one system at five units. The primary factor in the higher failure rates at these units was the quality of water being pumped (either high silt levels or foreign material presence). It should also be noted that variation in reporting practices could also influence this result.
- The distribution of failures detected by plant programmatic means at PWR units was decidedly more evenly distributed, with approximately equal fractions coming from four systems: AFW, CCW, ESW, and CVCS/HPSI.
- A total of 51 significant pump failures occurred at BWR units. Of these 51 failures, 46 (90%) involved ESW pumps. In contrast, 77 of the 195 significant PWR pump failures (40%) involved ESW pumps.

- Of the pump failures detected by regulatory/code testing, 83% (5 out of every 6) were classified as significant (i.e., level 4 or 5). This is not unexpected, since most failures thus reported fall into the required action range. Those failures that were not classified as significant either involved some increased vibration level (which was not deemed excessive by conventional criteria) or were nuisance type reports (for example, a pump delivered 24 gpm instead of the required 25 gpm under recirculation conditions).

- Only 17% of the significant bearing failures were detected by regulatory/code testing; almost 2/3 of the significant bearing failures were detected by plant programmatic means.

Summary

Extent of degradation

About 1 out of every 4 pump failures and about 1 out of every 3 motor failures were classified as significant. Almost half of the pump failures reported to NPRDS involved mechanical seal or packing leakage. Of 367 failure records involving seal or packing area problems, 308 only involved relatively minor leakage.

System

Failure rates were found to be highly dependent upon system of service. Both pumps and motors in the ESW system exhibited high failure rates compared to other systems.

Component age group

For BWRs, the failure rate (all failures and significant failures only) drops after a period of infant mortality and remains relatively constant afterward. There is also an initial drop in pump failure rate at PWRs, but it rises again after about ten years of age. For significant PWR pump failures only, the trend resembles the classic bath tub shape. The oldest age group for PWR pumps exhibits the highest overall failure rate.

Plant type

Pumps in BWRs have experienced lower failure rates than their PWR counterparts. Plotted as a function of age, it is clear that during the early years of operation, pump performance is similar, regardless of plant type. Over time, however, the superior performance of pumps in BWR applications becomes apparent; in the ≥ 20 years age group the relative failure rate of PWR pumps is over four times that of BWR pumps.

Affected area/method of detection

Voluntarily implemented plant programmatic controls were responsible for the detection of 73% of all pump failures and 41% of the significant pump failures. Regulatory/code required testing was responsible for only 14% of all failures and 39% of significant failures. The data also revealed that of the 95 significant failures detected by regulatory/code required testing, 75 failures involved ESW pumps. Of these 75 ESW pump failures, 61 involved internals wear; 56 of the 61 failures were indicated by failure to meet required flow or head. In summary, almost two-thirds of all pump failures detected by regulatory/code methods involved internals degradation of ESW pumps.

About 60% of all significant ESW pump failures were detected by regulatory/code required monitoring. In contrast, only 15% of the significant failures of pumps used in PWR plants in systems other than ESW were detected by regulatory/code required monitoring. More than four

times as many significant pump failures in non-ESW systems at PWR plants were detected by nonmandated plant programs as by regulatory/code required methods.

Observations and Conclusions

For pumps and their motors, voluntarily implemented plant programs have been more successful at finding degraded operation than have regulatory/code mandated methods. There appear to be several reasons why this has been the case:

(1) The ASME Code has historically allowed pumps to be tested at any reference point, including minimum flow. Hydraulic and vibration data collected at minimum flow conditions may be of minimal value. Also, operation at these conditions may contribute to accelerated pump wear and degradation.

(2) One of the leading causes of both pump and motor degradation has been bearing wear. For anti-friction bearings in particular, the types of monitoring done per Code requirements is not especially effective, even when the pump is operated near its best efficiency point. Overall vibration amplitude is normally dominated by running speed and harmonic components associated with conditions such as mechanical unbalance and misalignment. While bearing damage can be seen in spectral vibration data (or equivalently, in various treatments of the vibration signal which enhance the ability to observe bearing fault frequency-related energy), its contribution to the overall vibration amplitude, particularly in the displacement and velocity domains, is often negligible. A study conducted by a major pump manufacturer in which measurements of vibration velocity, vibration acceleration, shock pulse energy, and bearing outer race deflection* were made for various bearing and rotor conditions concluded that "The velocity system is the least effective bearing condition monitoring system. It was only effective in identifying bearing deterioration during the contaminated oil test. The bearings were defective to the point that total pump failure was imminent." [8]. The acceleration domain vibration data were more sensitive to bearing degradation, as were the shock pulse and bearing outer race deflection systems. All three measurements inherently (either because of the fundamental transducer response or accompanying filters) emphasize higher frequency data relative to velocity domain vibration.

(3) Voluntarily implemented plant programs tend to focus on effective activities. Tasks that provide no value are usually discontinued. Regulatorily mandated tasks which provide minimal return on the resource investment cannot be dismissed. It is important to recognize that external forces, such as insurance requirements appear to be moving in a direction that may effectively supersede certain regulatory or Code requirements. As an example, at least one major insurer requires that full spectral vibration analysis be performed for pumps operating with an overall vibration amplitude that essentially corresponds to the lower end of the alert range specified in the ASME Code.

(4) Human observations are a valuable diagnostic tool. For example, more bearing failures were detected by operator observation than by implementation of regulatory/code mandated testing. At a time when competitive forces and management philosophies of the day often result in reductions in available support staff, even skilled vibration monitoring technicians are sometimes placed in a mode of having to continuously defend the value of predictive maintenance programs. The vital contribution of operators and technicians who are able to maintain intimate familiarity with plant equipment diagnostic patterns – be they vibration spectra or sight and sound – to maintaining high equipment availability is often difficult to quantify, but is equally difficult to overstate.

* All data were considered in the overall amplitude sense (i.e., nonspectrally).

The observations that bearing degradation was both a leading source of pump and motor failures and was primarily detected through nonprogrammatic means suggests that alternative bearing health monitoring techniques, for example spectral analysis, high frequency demodulation, or shock pulse methods, would be likely to significantly improve programmatic detection experience.

The facts that such a large proportion (50%) of significant failures originated within one system (ESW), and that a large fraction of these ESW failures occurred at a few plants would appear to suggest that considerable dividends might accrue if available resources were allowed to focus on these "bad actor" applications. However, since at least some of the failures were attributable to circumstances that may be difficult to address (such as silt-laden water), it would presumptuous to project the extent of potential improvement in overall experience.

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Integrated Predictive Maintenance Program Vibration and Lube Oil Analysis Part 1 - History and the Vibration Program

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ABSTRACT

This paper is the first of two papers which describe the Predictive Maintenance Program for rotating machines at the Palo Verde Nuclear Generating Station. The organization has recently been restructured and significant benefits have been realized by the interaction, or "synergy" between the Vibration Program and the Lube Oil Analysis Program. This paper starts with the oldest part of the program - the Vibration Program and discusses the evolution of the program to its current state. The "Vibration" view of the combined program is then presented.

History

The Vibration Program began in 1982 with analog monitors and moved to computerized data collectors as soon as they were available. Initially the data was collected by Electricians and analyzed by dedicated Vibration Technicians. The Vibration Engineer reported to a different department, that provided additional support. This limited the effectiveness of the program because the data collectors could not detect instrument problems or significant vibration problems at the time the readings were taken. Significant vibration problems required a second trip to the field by a Vibration Technician to verify the readings and take additional diagnostic data. In 1988, dedicated vibration data collection personnel were added to the program, and within a year the effectiveness of the vibration program was significantly improved.

For years the Lube Oil Analysis Program sent samples off site for analysis with limited effectiveness. Improvements in the program were apparent in 1990 when a full time engineer was assigned, but even with a full time engineer, the use of an off-site lab resulted in a program with less technical depth and a reduced ability to perform specialized follow-up testing. Also there were often large time gaps between sampling and the receipt of analysis results. As a result, program effectiveness was still somewhat limited until a comprehensive Onsite Lube Lab was installed in Oct. 93

During the period of 1992-1994, a number of management changes affected both programs. The programs remained strong through this period because each program had dedicated technology engineers, but management attention was lacking due to the constant changes. This resulted in a fairly static vibration program, and considerable delay and difficulty in setting up the onsite Lube Lab.

During this static period (92-94) the programs were somewhat isolated from each other because they were physically isolated, and because of the concentration of each engineer on

maintaining and improving their individual programs rather than taking a larger view.

In late 1994, as a result of a company wide Re-Engineering effort, the Predictive Maintenance Program was reorganized. This revitalized the program by moving it from Engineering to Maintenance, locating the three technologies (Vibration Analysis, Lube Oil Analysis, and Thermography) together inside the protected area (except for the Lube Lab) and improving management support and attention. Another factor was the creation of a dynamic Maintenance Engineering Section with responsibilities for specific systems. This Section became the major customer of the Predictive Maintenance Section and improved the response to conditions detected by the Predictive Maintenance Program. Locating the Predictive Maintenance Program in the same organization as its major customer streamlined communication and improved the understanding of management expectations for the Predictive Maintenance Program.

The assignment of one leader for all three technologies, including both the engineers and the technicians, was another improvement. The physical closeness of the Engineers greatly increased the internal communication and support of each discipline by the other. The closeness of the Technicians increased their awareness of the strengths of the other programs and improved their interest in cross training. It also made scheduling cross training easier.

The move to locate the three technologies together had other unexpected benefits. Improvements in internal communications were expected, but we found a very significant "synergistic" effect, especially between the Vibration Program and the Lube Oil Program on roller element bearing problems.

Another key factor in this re-vitalization was the strong management support the program received after re-engineering.

The revitalized organization, called the Maintenance Support Engineering Section, has developed the following Mission Statement:

> **"The mission of Maintenance Support Engineering is to provide high quality condition monitoring services and surveillance testing to improve safety, maximize electrical production, reduce costs, and improve equipment reliability and availability."**

Organization

Figure 1 gives the organization chart for the Predictive Maintenance Section. The Section Leader reports to a Department Leader who reports to the Director of Maintenance. The Maintenance Engineers report in a parallel maintenance organization.

Note that our organization includes the Local Leak Rate Technicians. Their work load is normally moderate with peaks during outages, which is the opposite of the Predictive Maintenance technologies. These technicians are cross training into the Predictive Maintenance Program.

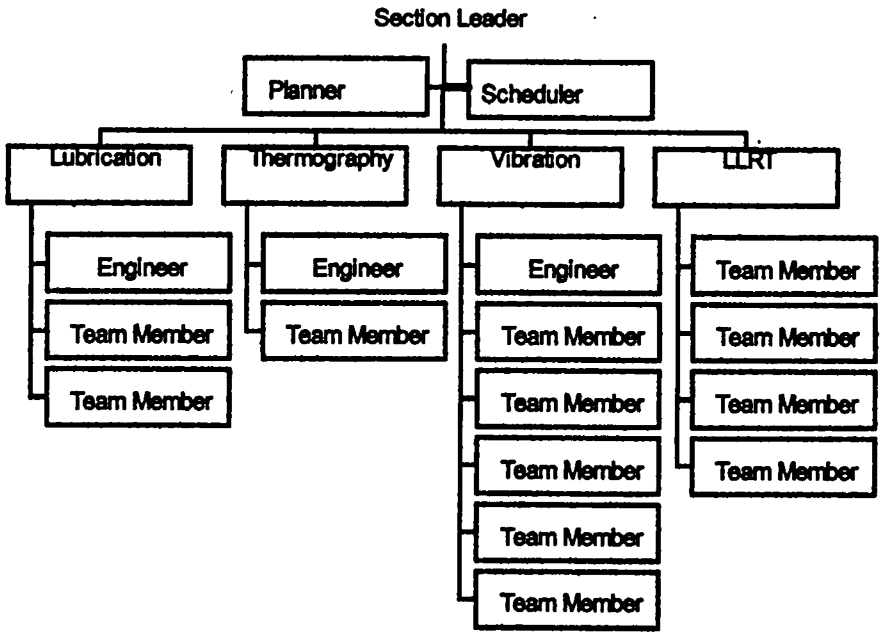
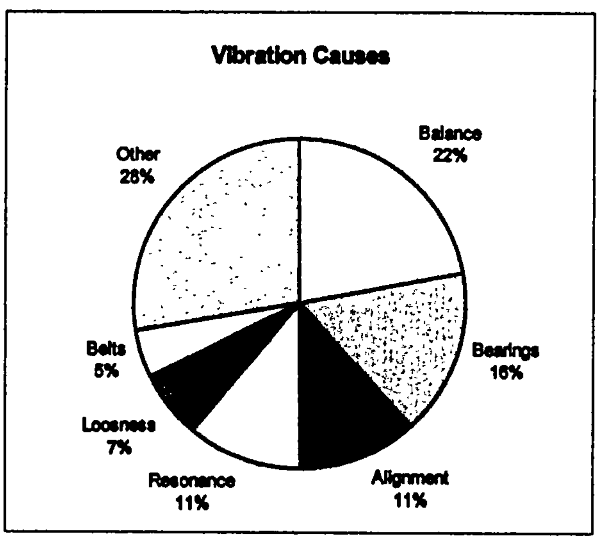


Figure 1

The Vibration Program

The Vibration Program takes readings on an average of 400 machines per month, with typically 14 points per machine. 750 machines are in the program, across a three unit station.

Figure 2

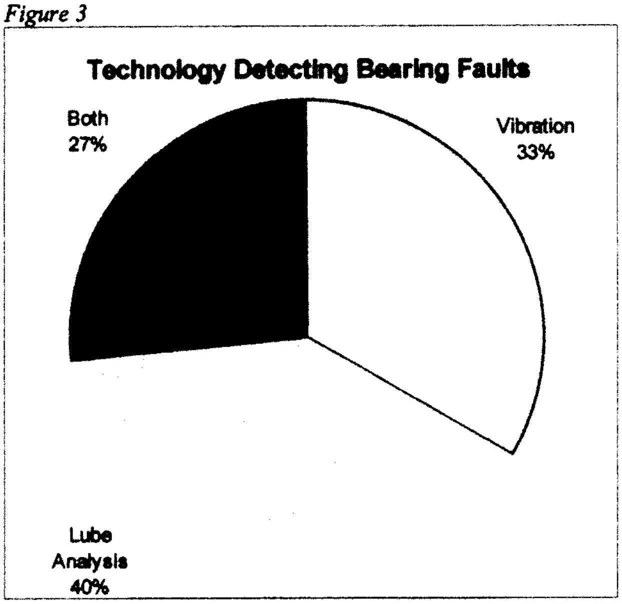


We recently analyzed our open condition report database to determine the primary causes of the station's vibration problems. As the chart in Figure 2 shows, balance, bearings and alignment are the three largest causes contributing to about 50% of all vibration problems.

A number of machines show symptoms indicating several possible causes. Also, the alignment and balance problems will eventually result in bearing problems.

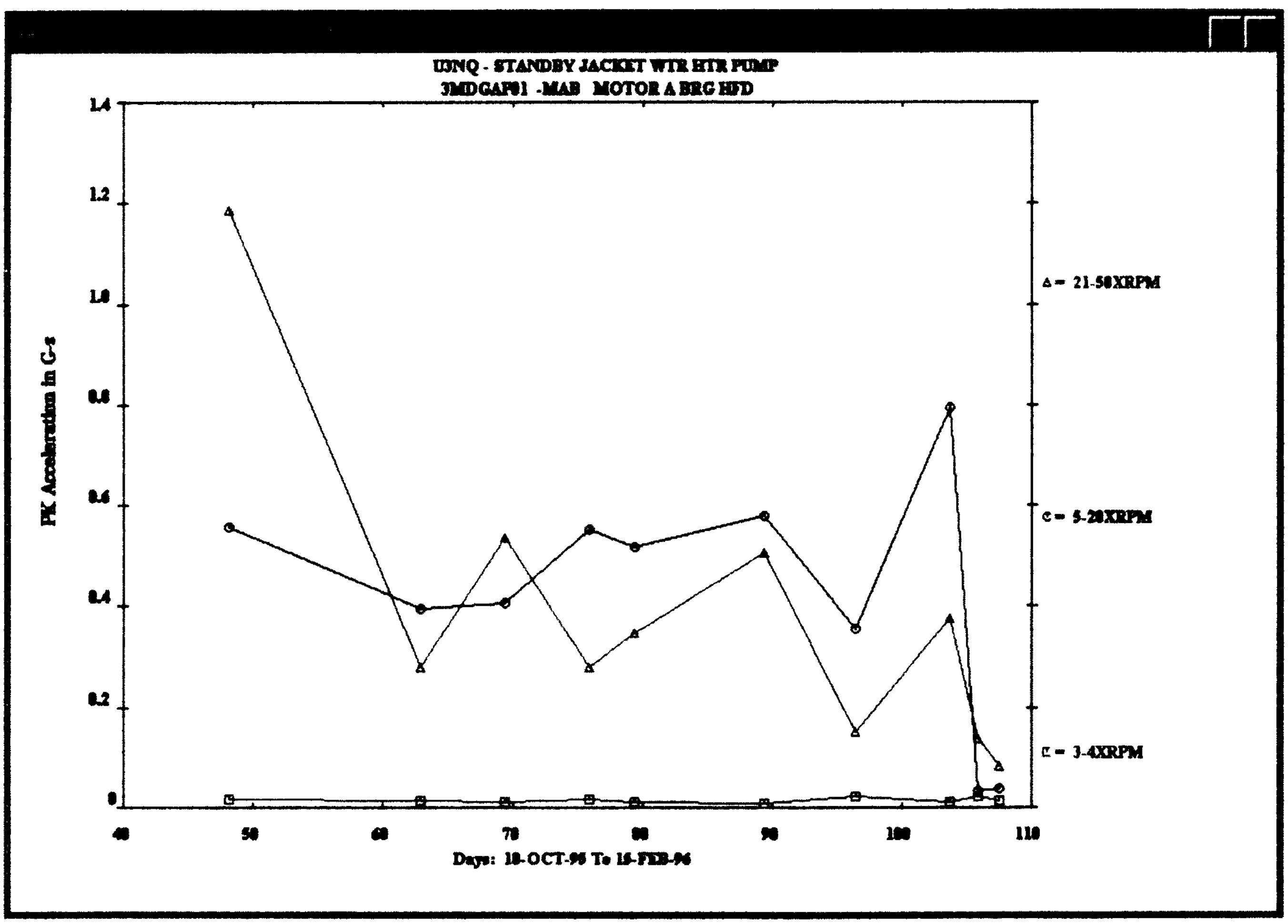
The Lubrication Program has been very helpful in improving the Vibration Program's evaluation of the 16% of the problems which are due to bearings, and in helping us determine the most likely cause of vibration. Figure 3 shows the relationship of the two programs in identifying 15 significant bearing problems in 1995. Prior to 1995 the programs were almost completly independent with minimal data correlating between technologies.

Figure 3



One recent example is on a Condensate Pump, which is a vertical deep draft pump. An analysis of the spectrum shows 1xRPM, a beating 0.63xRPM with multiples, and an erratic 0.5x with multiples. A "rumble" noise like looseness or a rub could also be heard. These symptoms indicate looseness, or a rub, or bearing damage, but where? Because the pump is buried in the ground and suspended from the motor thrust bearing, and vibration readings can only be taken on the motor, determining the location of the fault is difficult. However the upper motor bearing is oil lubricated, and the oil program has been very successful in detecting bearing damage from this bearing. So when the Lube program reported that there were no indications in the oil sample that would support a problem in the motor, the Vibration Program was able to determine that the pump is the probable source of vibration.

Figure 4

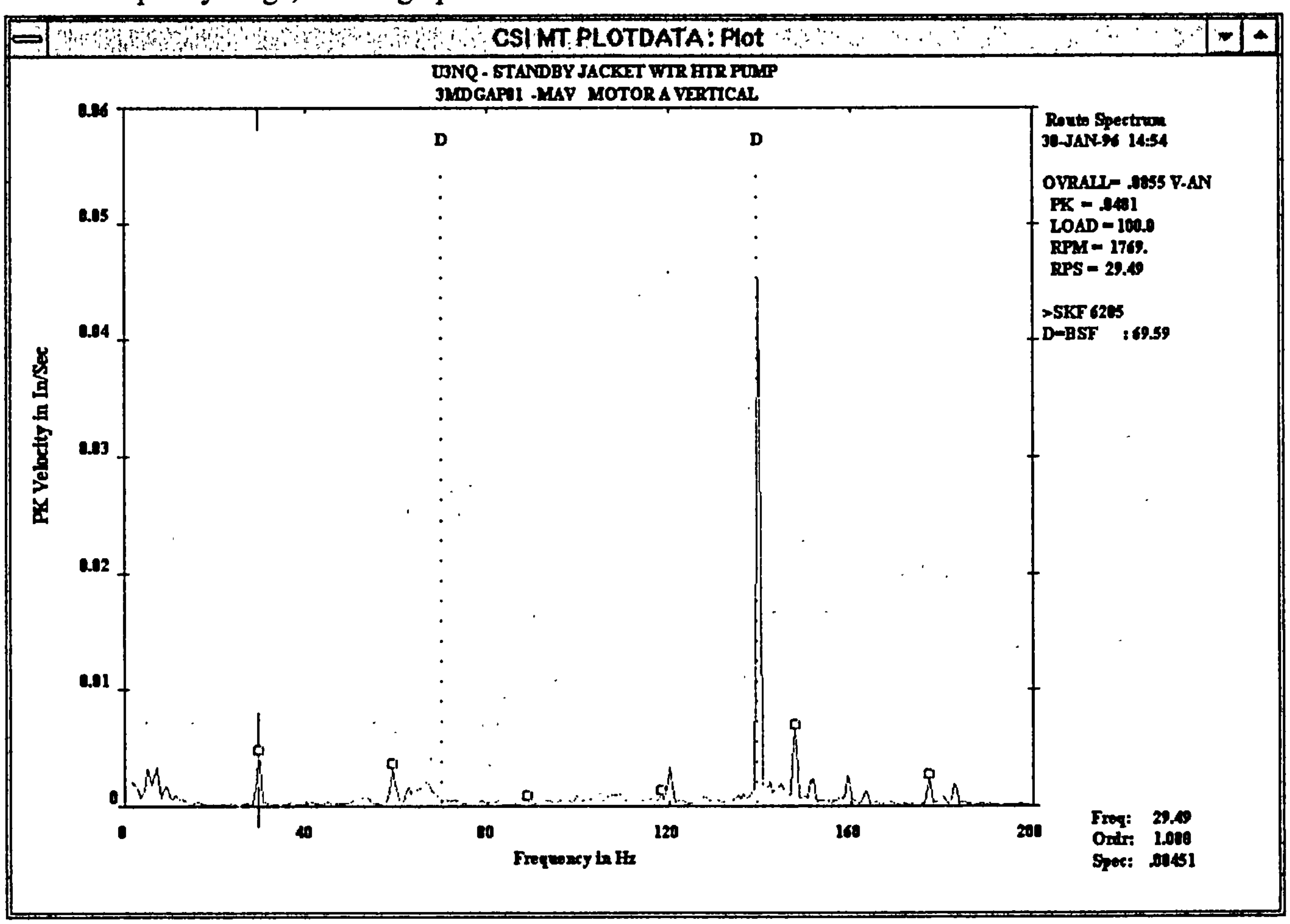


Vibration and Bearing Failures

At Palo Verde we use several techniques to determine bearing condition. Our first line of defense is high frequency bearing noise. In addition to the normal velocity vibration readings at each bearing, for rolling element bearings we take a special "bearing" reading. These acceleration readings cover two frequency ranges, the first is 5kHz -20Khz, which should provide the earliest warning (10% of life), and the second range is 2-50 orders of RPM, which should provide the second warning(5% of life). Figure 4 shows an example of this type of data. The graph shows the trend of the vibration in three frequency bands from a bearing on a motor. This bearing never did show a high amplitudes in the 5kHz - 20 kHz frequency range, but the graph does show that the frequency range of the signal lowers as the problem progresses. Note that the 21-50xRPM parameter is trending down while the 5-20xRPM parameter is trending up. The last two points of the graph were after the bearing was replaced.

We have had mixed success with these acceleration readings. They have been useful in determining when a bearing needed grease, and occasionally have indicated early bearing failures. The last, and most reliable, warning is provided by the normal velocity spectrum when the fault has moved into the low frequency range with less than 1 or 2% of bearing life left. Figure 5 shows the spectrum for the same bearing just before it was replaced. Note the 2 times Ball Spin Frequency.

Figure 5

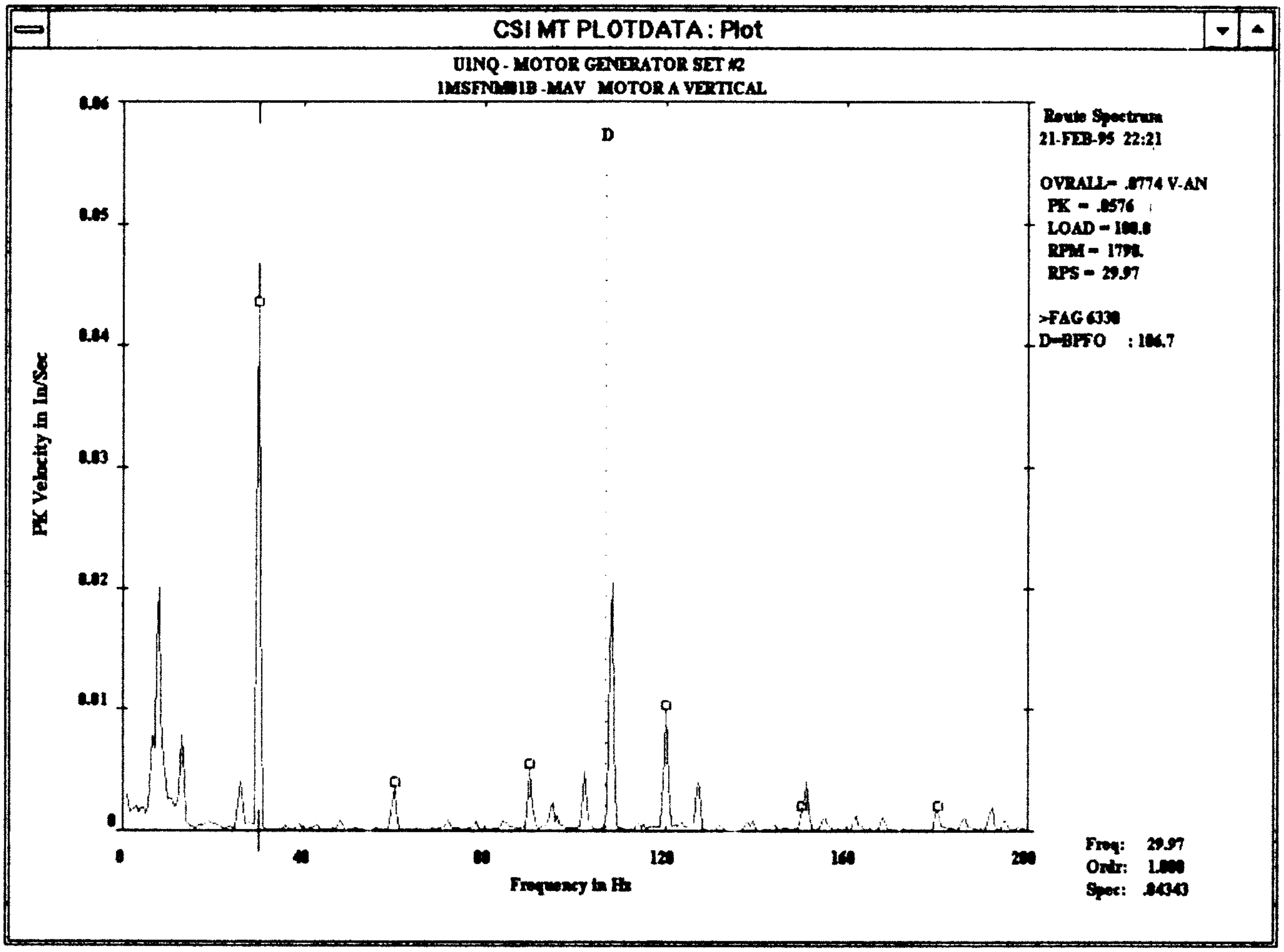


The lube oil program seems to be more reliable than the vibration program at the earlier stages of bearing failure. The most reliable indication is when both the lube oil program metal or wear debris analysis, and the presence of low frequency bearing fault frequencies confirm a bearing problem.

The Vibration Program is usually the first technology used when determining the condition of grease lubricated bearings, because greased bearings are not routinely sampled for Lube Analysis. However, on several occasions, grease samples were taken from bearings which the Vibration Program had indicated were in a failure mode. In the cases where a good sample could be obtained, the Lube Analysis confirmed the bearing problem.

One example is on a Motor-Generator Set. The vibration, shown in Figure 6, indicated a bearing problem on the motor end bearing.

Figure 6



The level was low, and did not show any upward trend. This indicated that the failure was in its very early stage, or was perhaps normal for this machine. A grease sample was taken, and confirmed bearing metal in the grease. Based on these two data points, the Predictive Maintenance Group determined that although the bearing was not in danger of imminent failure, it would probably not run longer than 9-12 months and possibly less. The plant was about to go into a refueling outage, and there would not be another convenient time to replace the bearing for about 20 months.

Based on this information the plant decided to change the bearing during the next outage.

In addition to the grease sample on the ball bearing indicated by the Vibration Program, a grease sample was taken from the other bearing, which was a roller bearing. This bearing had even more wear metals than the bearing with the vibration indication. Based on this result the scope of work was increased to replace both bearings. Both bearings were visually inspected (Figures 7 and 8) and showed the expected low level of wear, with only very early small spalls. This is a very good example of the "synergistic" effect that the two programs have on each other.

Figure 7 - Motor end bearing showing spalls on ball (largest is 130 micron).

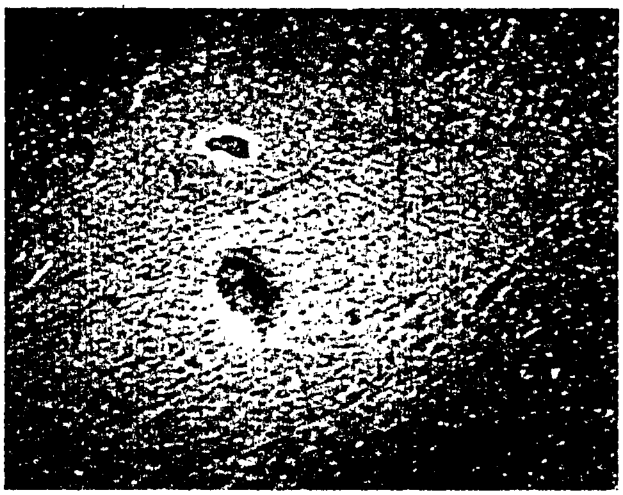
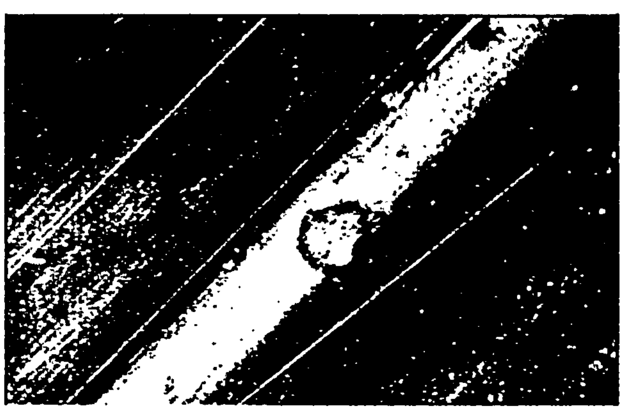


Figure 8 - 110 micron spall on roller in yellow streak from brass cage.



Conclusion

Our experience shows that a strong, up-to-date Vibration Program can be improved by closely integrating it with a strong, up-to-date lubrication program. The Lube Analysis Program also benefits in a similar way. Each of the Palo Verde programs benefited from interacting with the other, but an additional unexpected benefit was found, as the combined programs became more than the sum of the parts and our ability to detect and analyze roller element bearing problems was improved substantially.

Not all bearing problems can be detected in the early stages by a vibration program alone. A stand alone Lube Oil Program will also miss some bearing problems. But a combined program provides a significantly increased detection rate. In addition, when both technologies indicate a problem, there is an increased assurance that the indication is not a false alarm.

Part 2 -- Current Program Integrating Strategies and Lubrication Technology

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ABSTRACT

This paper is the second of two that describe the Predictive Maintenance Program for rotating machinery at the Palo Verde Nuclear Generating Station. The Predictive Maintenance program has been enhanced through organizational changes and improved interdisciplinary usage of technology. This paper will discuss current program strategies that have improved the interaction between the Vibration and Lube Oil programs. The "Lube Oil" view of the combined program along with case studies will then be presented.

Background

Predictive Maintenance programs can be used to perform maintenance on an "as needed" rather than "time directed" basis and to minimize the severity of a failure when an adverse condition is diagnosed. The Predictive Maintenance program is structured for machinery which is needed for both economic production and/or safety considerations. An integrated approach of various Predictive Maintenance technologies has improved the early detection capabilities of the program. One area of particular success is the early detection of infant bearing failure mechanisms.

Success Indicators

The most obvious success indicator is measured by the extent of the repair or how much secondary damage resulted from the original fault. Its important to not overlook the impact that the timing of the repair has on the facility. Finally the 'Root Cause of Failure Program' is used to minimize future occurrences of similar failures.

Typically a program's effectiveness is measured in dollars which are based upon the number of "saves" identified by the program. "Saves" can be counted when machinery is still operational when it is taken out of service. An ideal "save" occurs when the repair is minor in nature and secondary damage to the machine is avoided.

A recommendation to remove a machine from service can have a significant impact to the station. This impact can be particularly severe for machinery with safety or economic significance. For this type of machinery, a scheduled outage window would generally be preferable for the repair. If this is not possible, then having prior warning of a developing condition is extremely beneficial when planning the repair.

A program can not be fully successful until it has a 'Root Cause of Failure' process which identifies the source of the failure. Once the cause has been identified, corrections and modifications can be made. A successful Root Cause Program will

improve the reliability of the machinery which will reduce the number of future potential Predictive Maintenance program saves. Often equipment service life can be extended by making minor adjustments or repairs that will improve equipment reliability. Examples of this may be an oil change/flush or to tighten a loose belt.

Relative Success

The benefit of dollar savings or the avoidance of repair costs can be tabulated when reviewing a Predictive Maintenance Program. A subtle, but likely stronger benefit, is the usage of a monitored condition indicator to forecast when a failure will occur. Early knowledge of the condition can allow the necessary repairs to be made in a convenient work window which allows planning and scheduling to function in a routine manner. Secondary damage to the equipment can also be minimized by a timely repair.

Figure 1 is an as found bearing removed from a gearbox identified as having a degrading condition by the Predictive Maintenance Program. The gearbox was removed from service while the machine was still able to perform its function and to many could be classified as a dramatic save. As can be seen, the bearing has extensive damage and was estimated to have had only 'hours' of remaining service life. This bearing may not have survived an additional start. This save was universally considered successful, mainly due to the dramatic visible damage on the bearing surfaces.

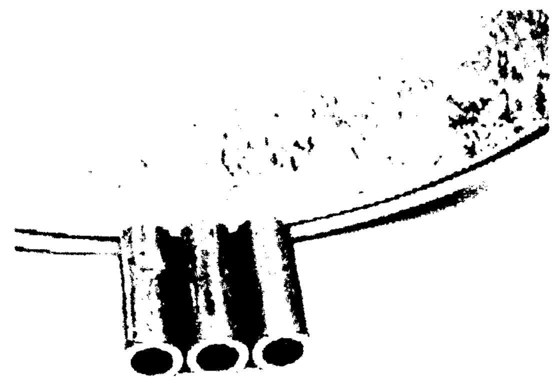

Figure 1 -- Bearing from gearbox

Later, a bearing serving the same purpose was removed from a second gearbox due to indications of a degrading condition. This bearing is shown in Figure 2 and its condition is not as apparent. (see next page) The degrading condition of the second bearing was identified by the Predictive Maintenance Program. The gearbox was removed from service to make the repair while the machine was still able to perform its function. The vertical lines on the bearing are lines of corrosion. Under microscopic examination, these lines have begun to form spalls and pits which are readily apparent.

The vibration industry has extensive documentation which correlates the amplitude of the bearing fault and its trend to expected remaining service life. Using the level of vibration produced by the second bearing just prior to its replacement, an estimated 3+ months of usable service life remained. One could argue that this bearing was removed early than necessary.

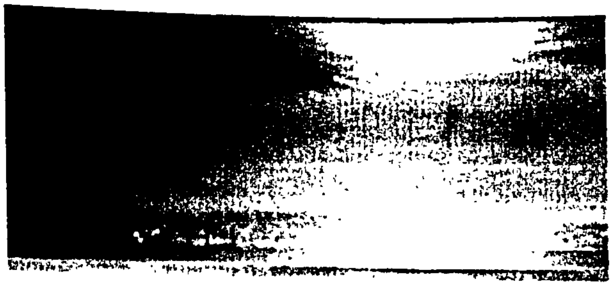

Figure 2 -- Bearing from gearbox

Considerations other than remaining expected service life need to be taken into account when determining the optimum time to replace a bearing. These two bearings can be used to discuss the merits of the three success indicators previously discussed. This paper will take the position that the bearing with less damage produced the better economic return.

The bearing fault in the first bearing was identified and the gearbox was immediately shut down. The repair was made on an unscheduled emergent basis that resulted in an increased cost of the repair. The wear debris produced by this bearing remained in the oil of the gearbox. The debris caused abrasive wear of the gearing requiring their replacement. The extent of the bearing damage obscured any chance of performing an effective root cause analysis.

The second bearing was removed with 3+ months of service life remaining. The recommendation was made with some lead time which allowed the opportunity to plan and schedule the repair in a more routine manner. This bearing had not produced a high level of debris. The oil cleanliness was more typical of oil from a normal gearbox than from one needing repair. The gear wear patterns were normal. The early removal of the bearing allowed a Root Cause of Failure to be performed. The corrosive condition mentioned was readily identified as the cause of the bearing wear.

The normal expected life cycle for the gearbox is 15 years. In light of the additional material damage caused by the increased gear wear, the emergent nature of the repair and the removal of Root Cause of Failure evidence, the second bearing provided obviously better return than the first.

Lubrication and Vibration Program Integration

The lubrication and vibration technologies had functioned as independent, largely unrelated programs until they were placed into the Maintenance organization. This was partially due to the relative technical strengths of the programs. The vibration program was a mature process with state-of-the-art technology routinely implemented in house by highly trained technicians. The lubrication testing program relied on the off-site testing of its samples and had minimal overall in-house capability. A major improvement was made to the lubrication program with the installation of a comprehensive in-house lube test laboratory.

As the lubrication program improved, two strong programs were then available to monitor and correlate information on rotating machinery. Information from the lubrication analysis program indicated that the vibration program needed strengthening in the area of bearing fault diagnostics. A project was implemented to identify which bearings were installed in monitored machinery and to set up the analysis data base with the respective bearing fault frequencies. An immediate improvement was seen in the vibration program in its early bearing failure detection capability. An improvement in general Root Cause of

Failure analysis occurred with evidence more readily found on removed bearings.

Enhanced Diagnostics Integrated

Wear debris from a lubricated surface initially forms in microscopic sizes. As a condition progresses, the relative amount and size of debris or wear particles increase. This debris collects in the oil and becomes strong evidence of a developing failure condition. Using advanced testing capabilities for the detection of wear debris, it was discovered that wear debris in oil indicative of a developing problem would be found in rotating equipment prior to its detection using vibration techniques. Since the debris which is generated at the on-set of an early stage failure is microscopic, the defects remaining in the surfaces are unlikely to cause an abnormal vibration indication. This is particularly true for bearings.

The wear debris in Figure 3 is typical of debris which originated from a rolling element bearing. The largest particle is noted as 130 micron and can be considered to be a severe wear indicator of a developing bearing problem. Note how the particles vary in size and how they all share similar characteristics. Many of these small particles are less than 5 micron in size. These small particles of wear debris are the first clue of a developing bearing failure.

The vibration analysis industry has developed very good methods that can be used to estimate the remaining life of a bearing. Blending the early detection strength of the lube wear particle analysis techniques with vibration data allows a developing machine fault to be tracked from a much earlier date to its time of failure.

Effective use of this information allows advanced planning and scheduling.

Figure 3 - Wear debris from bearing

Effective communication between the programs maximizes a monitoring strategy. This may include either increasing or decreasing either program's data gathering frequencies.

Bearing failure fault frequency detection using vibration technology can vary in effectiveness from machine to machine. Supplementing the vibration data with a lubrication wear testing program can improve the capabilities of vibration monitoring technology. For some machinery, especially those which have a standby function, this capability can be used to improve both the reliability and availability. The availability is enhanced by using scheduled work windows which do not impact the system or machine. These windows would typically be during an outage. The reliability is improved by fixing the developing problem before failure.

A normal service cycle for a machine may be many years, however, if a component within the machine is subjected to a higher than expected load due to a subtle error in

either the manufacture of the part, or its installation, then its service life can be greatly shortened. The reliability of the machine is a question of great interest for the later case. The actual failure rate may also increase due to abnormal loading or a defect condition existing in a part.

Stand-by machinery, such as safety equipment, poses the problem of very low run hours which may not be sufficient to expose the problem until a failure is near. With a higher than normal degradation rate, the relative available service time of the machine may be low when it is discovered through means such as walk downs, noise or even conventional vibration analysis. The use of the enhanced capabilities of an early bearing fault detection program which uses a combined vibration and lubrication analysis strategy will identify this type of fault at an earlier failure stage and allow better monitoring or repair of the condition. Such information could possibly be used to make a repair weeks to months earlier than by presently used methods. These weeks/months may have a direct correlation to improved reliability of the machine for that time period. When a premature wear condition exists, a significant save can result even though the actual damage to a bearing is moderate or low, because a higher than normal wear rate could lead to a quicker than expected failure.

The lubrication test techniques which are of greatest benefit for early fault detection include visual microscope methods. These methods require the preparation of a small portion of the lube sample in a manner which distributes the wear debris for microscopic viewing and removes the oil so that it does not block the view. Other detection methods include spectrographic analysis which measures the level of metals debris in the sample in units of parts per million (ppm). These techniques can target debris in various size ranges. (Typically 0-5 micron or 5+ micron)

Since most abnormal wear debris is larger that 5 micron, and standard spectrographic analysis is limited to particulate less than 5 micron, instrumentation used for analysis can have a significant impact on the effectiveness of the testing. The limitations of instrumentation must be understood to be able to evaluate the accuracy of the data. These limitations can at times be significant. PVNGS uses a spectrographic technology which is new to the lube industry and allows the monitoring of the 5+ micron particles in a convenient manner. With five distinct wear debris test techniques routinely used in the PVNGS lube lab in lieu of the typical 1-2 used by many commercial labs, the on-site testing capability for oils began to pay dividends in early bearing failure detection.

Figure 4 - Severe abnormal wear particle

An example of abnormal wear debris is shown in Figure 4. The large particle is an example of a severe cutting wear particle.

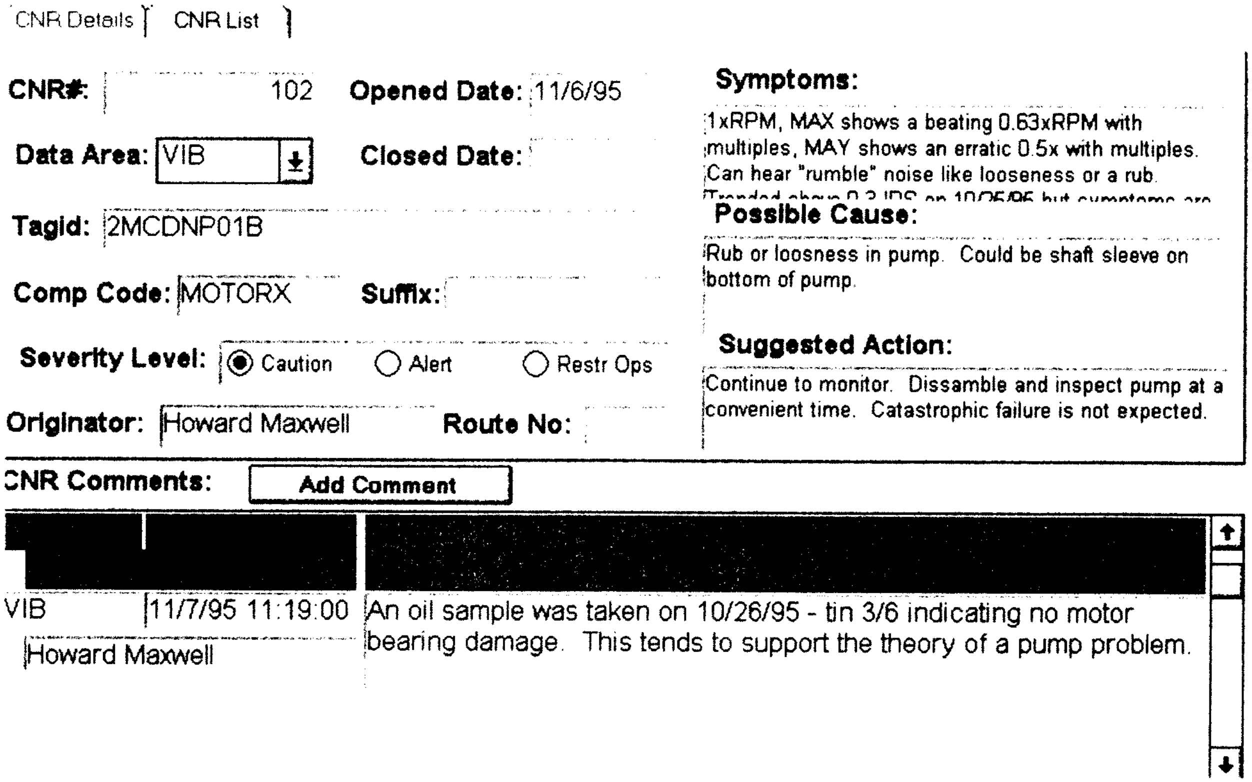

Figure 5 - Example of CNR

Program Integration into Maintenance

Technical accuracy is needed to provide a foundation for acceptance of a Predictive Maintenance Program. The integration of a Predictive Maintenance Program into the maintenance of the plant ultimately requires an effective communication process. E-mail, a widely accessible database and training were identified as the keys to successfully integrate the program within the Maintenance organization. Training seminars were organized and the capabilities and future direction of the Predictive Maintenance Program was discussed with identified program customers.

A computer software program was developed which allowed direct e-mail notification of higher severity level conditions detected using standardized definitions. The program identifies monitored machinery conditions as follows:

Caution - A condition that may indicate a trend towards an unacceptable condition

Alert - A condition exists for which it is believed that if no corrective actions are taken, the condition will progress to failure.

Restricted Operation - Failure to take immediate or short term corrective action will result in a failure.

These reports are identified to Predictive Maintenance program customers as 'Condition Notification Reports' (CNR). Putting the CNR information into a format in which it is readily available to station personnel has had benefits in scheduling and increased the integration of the program into the Maintenance Department structure. An example of a lower level condition is included in Figure 5. It should be noted that although this condition is being identified and monitored by the vibration program, the

lubrication program was used to eliminate a possible failure source.

Case Studies

Bearing failure can be monitored from early stage deterioration to final bearing failure. The gearbox bearings shown in Figures 1 & 2 are good examples of conditions which were monitored well into the end of the service availability of the bearings. Both of these bearings were identified by the lubrication and vibration predictive maintenance technologies. The vibration technology took the lead in determining when to take the machines out of service. In the case of the badly worn bearing, the gearbox was taken out of service with little opportunity available for scheduling and planning. The second bearing was not as badly worn. This allowed a longer window to schedule and perform the work.

An example of a successful repair based upon a monitored early stage failure condition of a standby machine occurred during a recent outage. The oil sample received at the beginning of the outage contained a moderate level of wear debris. The debris was further examined to characterize its type and extent. This determination is often based upon particle size with larger particles indicative of a more advanced wear condition. The sample was viewed microscopically with debris characteristically similar to Figure 3 visible. It was concluded that the sample contained strong evidence of a bearing which was in early failure.

The vibration program was consulted to obtain correlation data on the bearing condition. No indication was available of a bearing fault which indicated that the fault was an early failure. Based upon strong lubrication test data, the availability of a convenient work window and the ease of replacement of the bearing, a recommendation was made to replace the bearing.

Microscopically heavy general wear was found on the bearing ball and on its races. Inspection indicated that the bearing was on the verge of producing a fault large enough to be detected by the vibration program. This detection would likely have occurred after the outage had ended. Figure 6 shows the extent of damage on the balls. It should be noted that at 100x magnification that the ball surface for a normal ball bearing would be smooth and bright. The discoloration is due to a roughening of the surface caused by loss of material and heat.

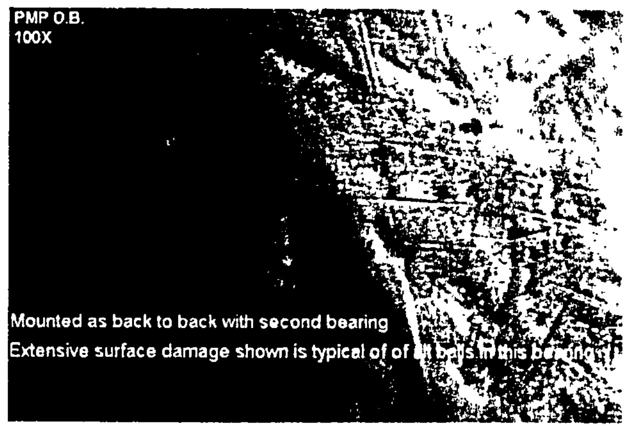

Figure 6 - Damaged bearing

As would be expected, the bearing race was also damaged. This damage was visible with the naked eye and appeared as skid marks. With 100x magnification the damage is readily apparent as can be seen on Figure 7 on the next page.

Figure 7 - Bearing inner race

Conclusion

Independent lubrication and vibration Predictive Maintenance programs can provide a valuable benefit to plant maintenance, however, when integrated into a common strategy that capitalizes on strengths, each technology can extend the diagnostic capabilities of the other. Lubrication sample wear debris testing is highly useful in determining the on-set of a wear condition. Vibration data has extensive industry correlation in terms of remaining useful service life of a bearing. Predictive technologies can be successfully integrated to obtain a better overall product. Potentially, the greatest benefit may be for stand by machinery which has a subtle material or installation fault. The benefit would be in the early detection of the problem which would allow its correction in either a more convenient time frame such as an outage or through a reliability increase from an earlier repair. When integrated into a Maintenance structure, Predictive Maintenance can be a powerful tool with benefits in both improving reliability and availability of rotating equipment and in reduced costs.

Using the Motor to Monitor Pump Conditions

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Abstract

When the load of a mechanical device being driven by a motor changes, whether in response to changes in the overall process or changes in the performance of the driven device, the motor inherently responds. For induction motors, the current amplitude and phase angle change as the shaft load changes. By examining the details of these changes in amplitude and phase, load fluctuations of the driven device can be observed.

The usefulness of the motor as a transducer to improve the understanding of devices with high torque fluctuations, such as positive displacement compressors and motor-operated valves, has been recognized and demonstrated for a number of years. On such devices as these, the spectrum of the motor current amplitude, phase, or power normally has certain characteristic peaks associated with various load components, such as the piston stroke or gear tooth meshing frequencies. Comparison and trending of the amplitudes of these peaks has been shown to provide some indication of their mechanical condition.

For most centrifugal pumps, the load fluctuations are normally low in torque amplitude, and as a result, the motor experiences a correspondingly lower level of load fluctuation. However, both laboratory and field test data have demonstrated that the motor does provide insight into some important pump performance conditions, such as hydraulic stability and pump-to-motor alignment.

Comparisons of other dynamic signals, such as vibration and pressure pulsation, to motor data for centrifugal pumps are provided. The effects of inadequate suction head, misalignment, mechanical and hydraulic unbalance on these signals are presented.

Background

Motor-driven pumps are, without question, one of the cornerstones of industrial societies. This is clearly demonstrated by simply considering two essential commodities at the residential level - water and electricity. The requirement for pumps in distributing water is obvious. The dominant sources of electrical power, coal-fired and nuclear power plants, require pumps in numerous applications for both the production of electricity and the protection of other equipment. The vast majority of the pumps used in both applications are motor-driven.

Given the importance of motor-driven pumps, there has always been an interest in improving their reliability. Historically, pump designers and users have recognized that pump designs that optimized pump efficiency were not always optimal from a reliability standpoint, and proper pump selection practices and pump design modifications have certainly resulted in improved reliability [1, 2, 3, 4, 5, 6].

Over the last two decades (and the last decade in particular), improvements in equipment condition monitoring capabilities have also greatly contributed to reduced failure rates in a variety of ways. For example, vibration monitoring equipment allows the detection of a broad variety of equipment maladies, many at the incipient level of degradation. With early indication of wear, users are able to plan equipment outages to address problems before they escalate to the level that demand failure occurs. Other important monitoring techniques, such as lubricant analysis, thermography, motor meggering and inductive imbalance measurements can be used to help the user understand and trend motor and pump conditions.

In recent years, the potential value of using on-line motor data to help understand the condition of both the motor and the driven equipment has begun to be exploited [7, 8] The inherent ability of induction motors to be used as transducers can be readily perceived by simply observing that certain motor parameters such as current, power, and power factor vary with load. Typical performance curves for these parameters, normalized to their rated load values are shown in Figure 1. It can be seen that the motor input power is relatively linear with motor load, current is essentially linear above 40 or 50% of load. The power factor is clearly non-linear, but its sensitivity at lightly loaded conditions is obvious.

Pump motor current is commonly transduced by permanently installed current transformers and displayed locally at the motor breaker cubicle and/or remotely at main control panels. Operators often use current as not only an indicator that the driven device is running, but in a diagnostic sense, to help understand the general condition. For example, operator observation of large fluctuations in indicated pump current has in some cases been cited as the first indicator of vortex formation in residual heat removal pump suction lines during reduced inventory operations.

In addition to the information conveyed from such obvious load fluctuations, there are often more subtle features* that can be extracted from motor signals that help characterize load conditions. The term "motor current signature analysis" was first coined during studies of motor-operated valve aging performed at Oak Ridge National Laboratory for the Nuclear Regulatory Commission [7]. This term has since been broadly applied to a variety of methods of analyzing motor current signals, ranging from simple overall current spectral analysis to multi-level demodulation.

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* At the time of the studies, a heavy reliance upon analog signal conditioning was necessary due to inherent limitations of affordable recording systems. As the dynamic range of recording and digitizing devices has improved in recent years, it has been possible to perform much of the signal conditioning in the digital domain, although analog preconditioning still plays a critical role in certain applications.

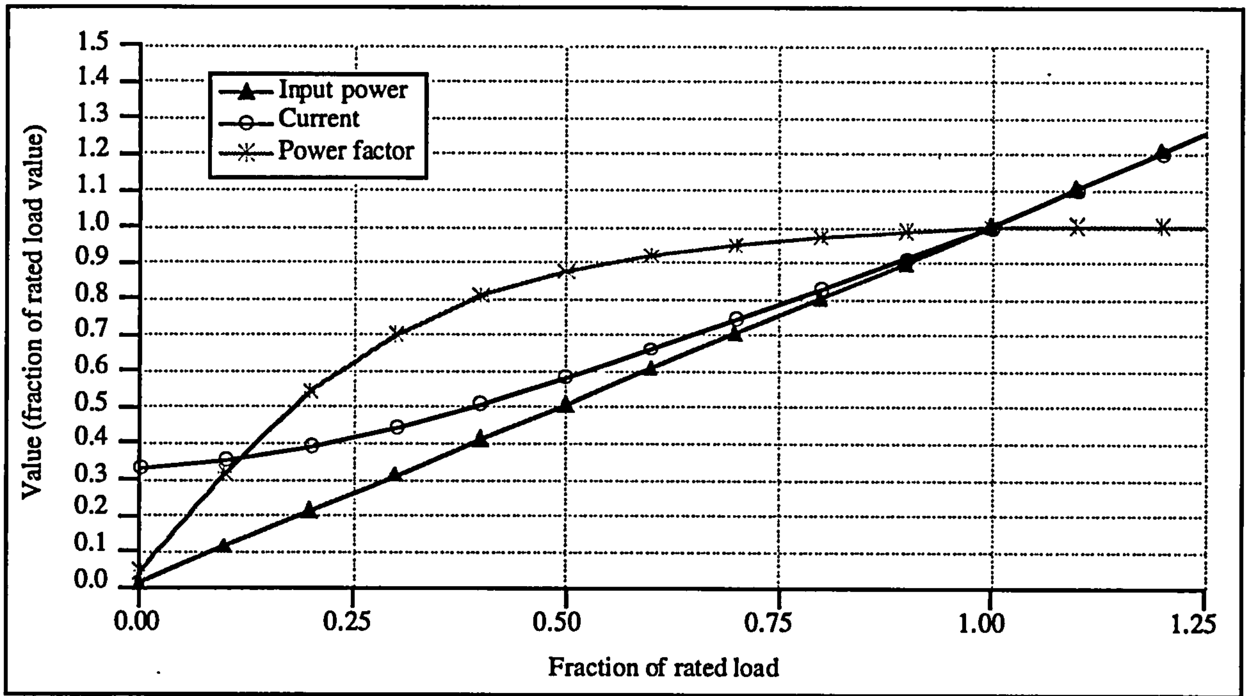

Figure 1. Performance characteristics for a typical induction motor.

For a steady load with no fluctuating component, the current spectrum is normally composed of a dominant line frequency component (i.e., 60 Hz in the U.S.) accompanied by some level of harmonics. When the motor load is fluctuating, a modulation of the amplitude and phase of the current occurs, normally resulting in spectral sidebands. Figure 2 show the spectra of a simulated steady and 15 Hz fluctuating load. For the steady load (left hand plot), all spectral energy is located at line frequency (60 Hz). The fluctuating 15 Hz load manifests itself (right hand plot) as 15 Hz sidebands of the line frequency.

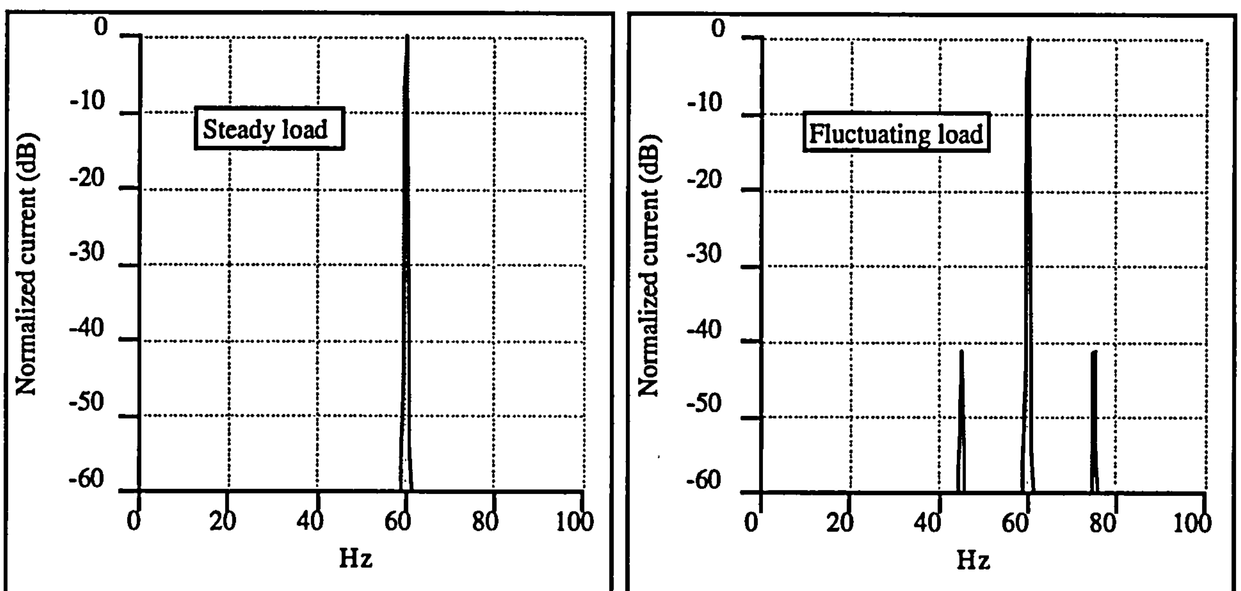

Figure 2. Current spectra for steady and fluctuating load.

Short segments of the time domain waveforms of the signals represented by the Figure 2 spectra are shown in Figure 3.

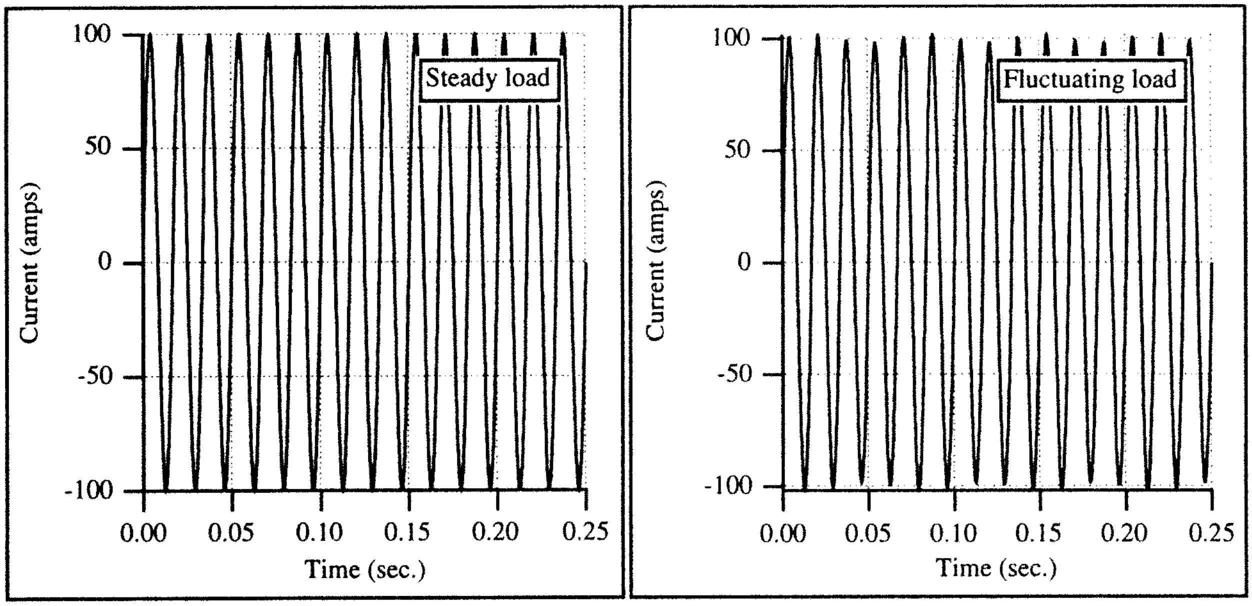

Figure 3. Time waveforms for the current of steady and fluctuating loads.

The load fluctuation can be seen by careful examination of the fluctuating waveform. In the quarter-second of data shown, 3.75 cycles of the fluctuating load occur. It should be noted that the load fluctuation for this artificially generated data is relatively high - two percent of the overall amplitude.

By demodulating the overall current amplitude, the nature of the fluctuating load becomes much clearer. The fluctuating load, expressed as a fraction of the average load, and the demodulated currents for both the steady and fluctuating load cases are shown in Figure 4.

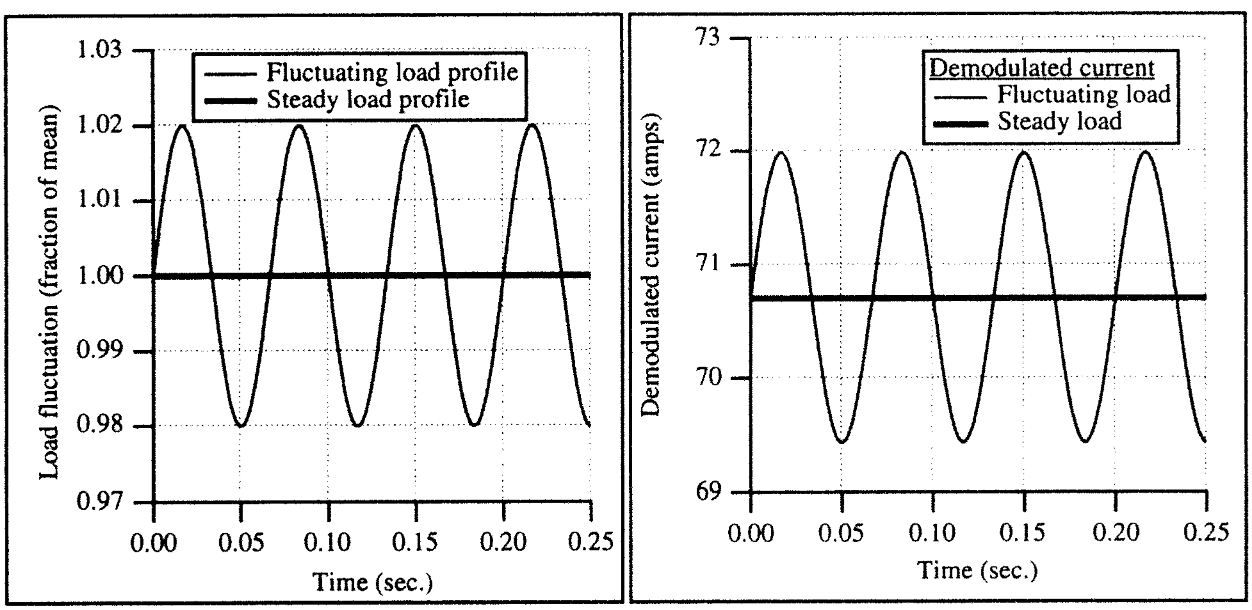

Figure 4. Simulated shaft load fluctuation and demodulated current data for steady and fluctuating loads.

Figure 4 clearly illustrates the value of demodulation in recovering the nature of the fluctuating load source. Another feature of note is that the spectra for the demodulated data indicate the load at the load frequency - not as sidebands of the carrier frequency.

The demodulation means used to develop the data shown in Figure 4 was by a digital rms (amplitude demodulation) calculation. Various alternative demodulation methods are available, including analog and digital means that use amplitude or frequency demodulation methods. While the results of these approaches may vary somewhat, depending in part on the nature of the load modulation, the general goal is the same - to extract a signal that is representative of the source of energy that caused the current modulation.

The determination of instantaneous power can be viewed as a special method of demodulation. The measurement of power has the advantage of being very nearly linear across a broad load span (note the essentially linear nature of motor input power vs. load in Figure 1). In addition, instantaneous power measurement inherently addresses both amplitude and frequency or phase modulation effects. The obvious drawback to power measurement is that it requires securing voltage waveform signals that are amplitude proportional to, and in phase with the voltage for the phase on which current is being measured. At higher voltage levels, power measurements become much more complicated, and are generally not feasible unless permanently installed potential transformers are available. Even at the 480 volt bus level, there is often significant reluctance to hot hookups necessary for using portable instrumentation. As a result, the acquisition of motor data is often limited to current signals acquired from clamp-on transformers or hall-effect pickups.

This background discussion is intended as an introduction to a few of the features of signal acquisition and processing. Subtle features that can significantly influence results abound in motor signal analysis* that are beyond the scope of this paper. With reasonable thought and care, however, some very useful information about load and motor conditions can be extracted from motor-derived signals. The balance of this paper will provide some specific examples of the use of motor data in understanding pump and drive train conditions. Where appropriate (and available), comparison with other diagnostic sensor information will be provided.

Misalignment

Alignment is always an important consideration in ensuring rotating equipment reliability. The availability in recent years of laser-based alignment tools has significantly improved the ease of achieving good alignment between drive train components. Even with such improved set-up tools, it is always helpful to have data available at operating conditions that validate alignment suitability.

In support of the Department of Energy's Motor Challenge Program, ORNL has had the opportunity to collect both laboratory and industrial test data on the effects of machinery alignment on drive train efficiency. In conjunction with these tests, vibration and motor power spectral data were evaluated for various alignment conditions.

In laboratory testing, the effects of various levels of combined parallel and angular misalignment was evaluated. For this testing, a 10-hp, 4-pole motor was used to drive a generator. A torque cell, capable of measuring both static and dynamic torque fluctuations was interposed between the motor and generator, and both shaft connections were made using a flexible coupling with an elastomeric insert.

* It should be noted that this kind of statement is almost universal in application when dealing with instrumentation of any sort. Flow, pressure, vibration and most other parameters to be measured in engineering applications require that the user understand limitations of the transducers, proper installation arrangements, necessary signal conditioning practices, etc. to acquire truly representative data.

All misalignment conditions were created by adding or removing shims from under the motor feet. Vibration at all tested conditions was very low amplitude. Motor vibration and motor input power ripple data for two of the alignment conditions are shown in Figures 5 and 6. The vibration data shown were collected at the vertical inboard location. Vibration data at other locations showed similar trends. Of particular note is the fact that while the vibration amplitude roughly doubled between these cases, the running speed power ripple increased by a factor of about 30.

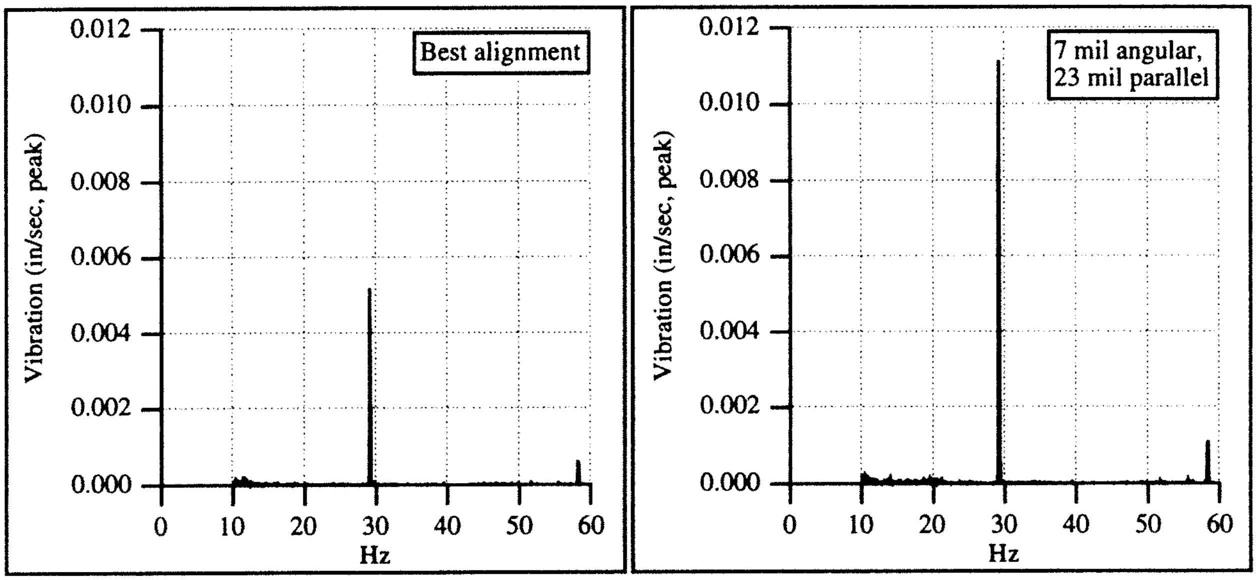

Figure 5. Vibration spectra for two alignment conditions.

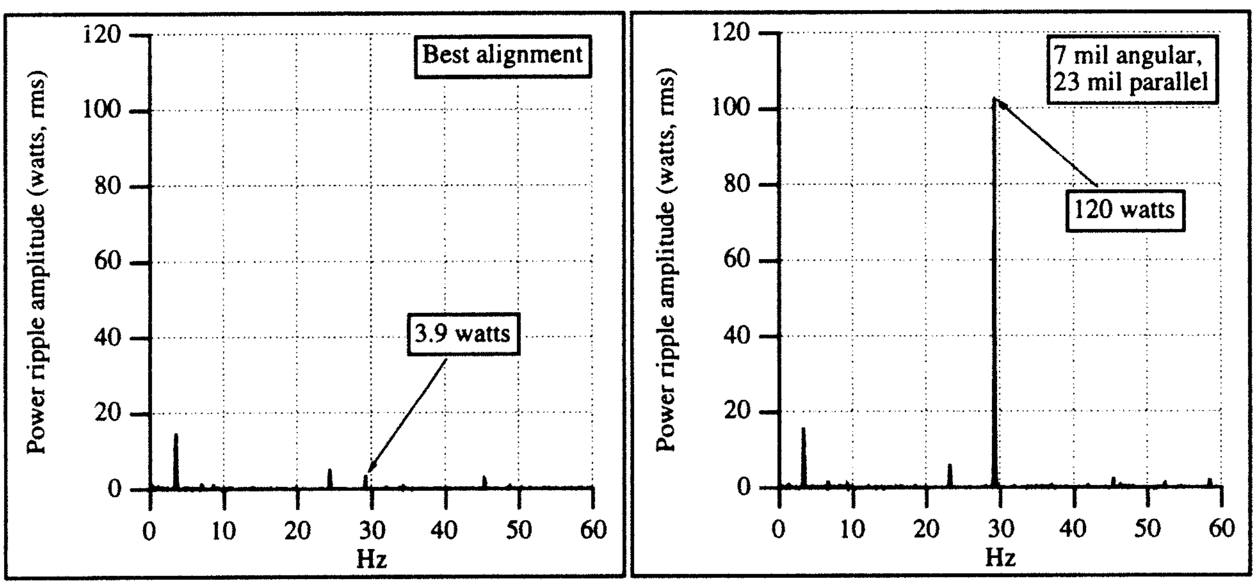

Figure 6. Motor power spectra for two alignment conditions.

Figures 7 and 8 present summary vibration and power ripple vs. measured torque ripple for all tested alignment conditions. All of the alignment conditions except the worst case (represented by the upper-right most points in Figs. 7 and 8) were within the coupling manufacturer's allowable. For this worst case, the coupling was in obvious distress (coupling sleeve became twisted during operation).

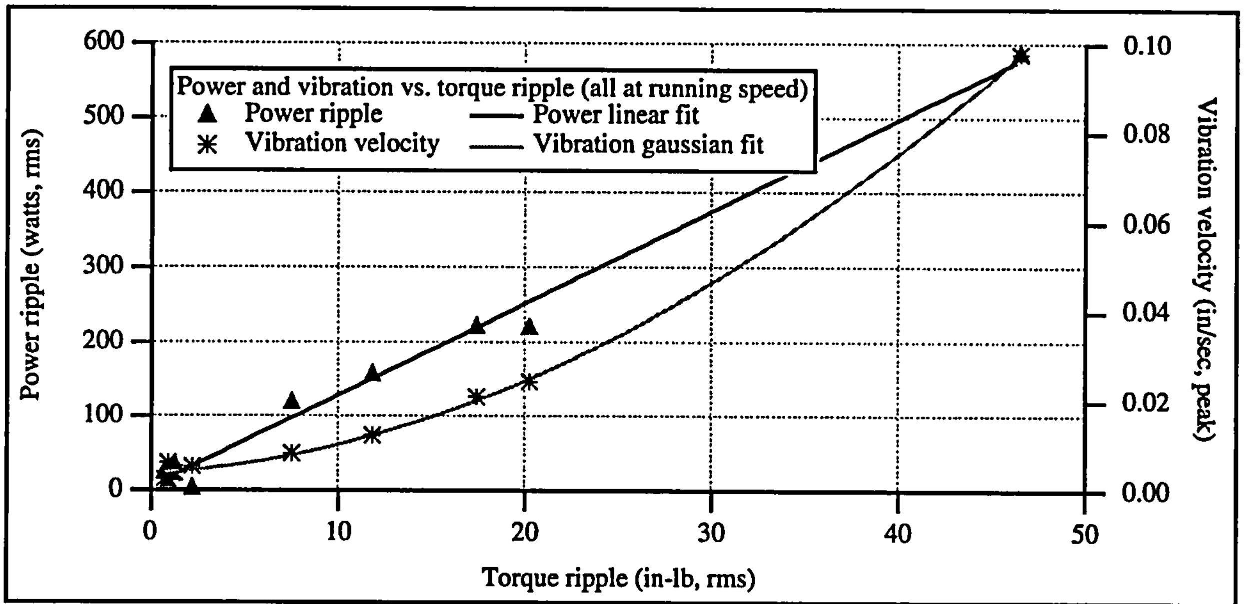

Figure 7. Motor power ripple and vibration velocity vs. torque ripple (all values at the running speed spectral peak).

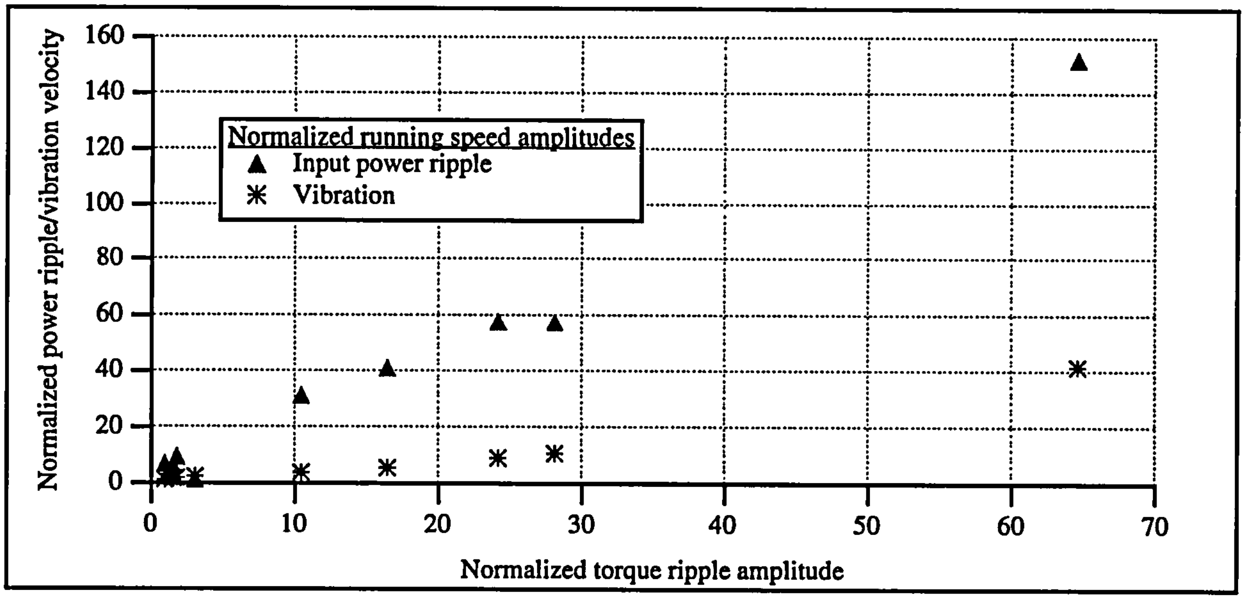

Figure 8. Normalized running speed power ripple and vibration velocity normalized to the minimum test condition values).

It should be noted that the power ripple amplitudes cannot be interpreted as true power dissipation, since this coupling, like all flexible couplings, has some means of resiliency, and tends to act like a spring. However, the coupling temperature did rise with increased misalignment (as indicated by a thermographic camera and infrared sensor)*.

A General Motors plant located in Saginaw, Michigan has an ongoing program, with the assistance of Consumers Power Company, to reduce energy consumption at their facility. One of the

* An interesting feature noted with the thermographic camera was that the coupling key acted like a fan and helped cool the coupling ends.

activities of this program is precision alignment of motor-driven machinery, with the anticipation that excellent alignment of critical plant process equipment will not only increase the lifetimes of the equipment, but result in some reduction in energy consumption as well. The alignment work is being implemented as a Motor Challenge Showcase† demonstration project. ORNL was able to acquire motor and vibration data on three 250-hp pumps before and after laser alignment. The pumps are double suction, 6000 gpm, 125 ft. head, and use steelflex couplings between the motor and pump. Two of the three pumps have a backup diesel drive. One of the combination drive pumps is shown in Figure 9.

Figure 9. Double suction, dual drive hotwell pump

Table 1 provides the pre- and post-alignment offsets for the pump that was most out of alignment in the "as found" condition.

Table 1. Hotwell pump #6 pre- and post-alignment offsets

Condition	Plane	Angular offset (mils)	Parallel offset (mils)
Pre-alignment	Vertical	-33.7	-10.5
Pre-alignment	Horizontal	-0.7	-0.5
Post-alignment	Vertical	0.1	-2.3
Post-alignment	Horizontal	0.3	2.5

Vibration data was recorded at both the motor and pump inboard bearings in the horizontal, vertical, and axial orientations. Waveforms for all three phases of current and voltage (phase to neutral) were recorded, allowing subsequent determination of total power.

Table 2 shows pre- and post-alignment vibration data for the first three running speed harmonics and the overall (rms-based) vibration velocity amplitudes for pump 6. The changes in amplitude were mixed in direction, but almost negligible at all points.

† A Department of Energy Program, a goal of which is the reduction of excess energy consumed by electric motors.

Table 2. Pump 6 vibration amplitudes before and after alignment

Condition	Running speed harmonic	Motor axial	Motor vertical	Motor horizontal	Pump axial	Pump vertical	Pump horizontal
Pre-alignment	1	0.213	0.107	0.090	0.125	0.066	0.066
Post-alignment	1	0.193	0.094	0.065	0.122	0.058	0.058
Pre-alignment	2	0.003	0.002	0.006	0.024	0.013	0.013
Post-alignment	2	0.011	0.002	0.017	0.012	0.007	0.007
Pre-alignment	3	0.009	0.013	0.001	0.006	0.003	0.003
Post-alignment	3	0.016	0.017	0.003	0.004	0.004	0.004
Pre-alignment	RMS	0.218	0.103	0.112	0.151	0.104	0.086
Post-alignment	RMS	0.208	0.080	0.107	0.153	0.110	0.081

Example spectra are shown in Figure 10. Other than a slightly greater noise level in the post-alignment spectrum (the pump was operating at a slightly different flow rate), no major differences are apparent.

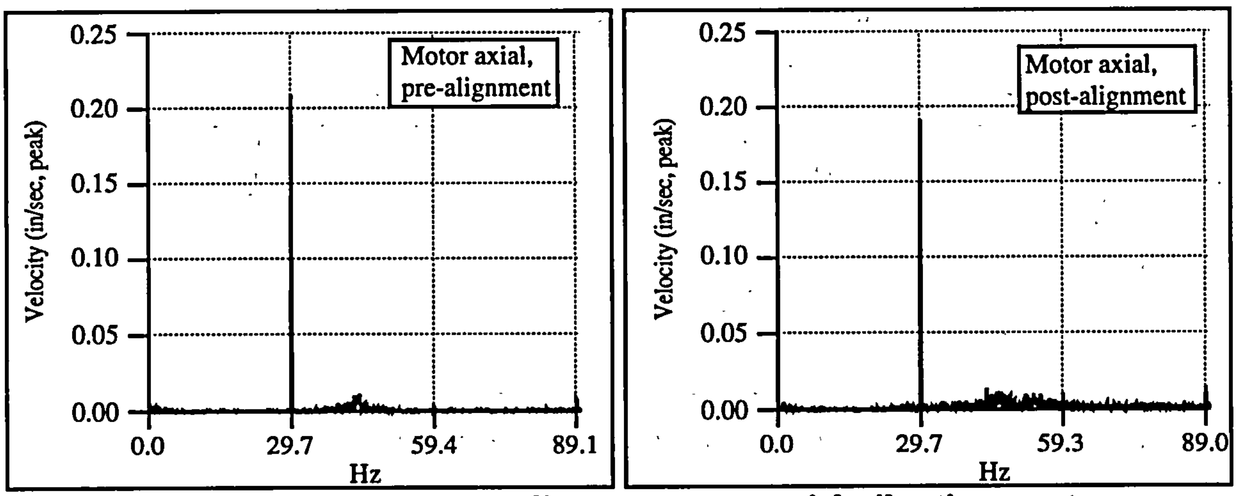

Figure 10. Pump 6 pre- and post-alignment motor axial vibration spectra.

The spectra of the first derivative of the input motor power is shown in Figure 11. The change in running speed-related energy associated with the change in alignment is clearly evident in the power data. Both the first and third harmonics energy rates dropped significantly.

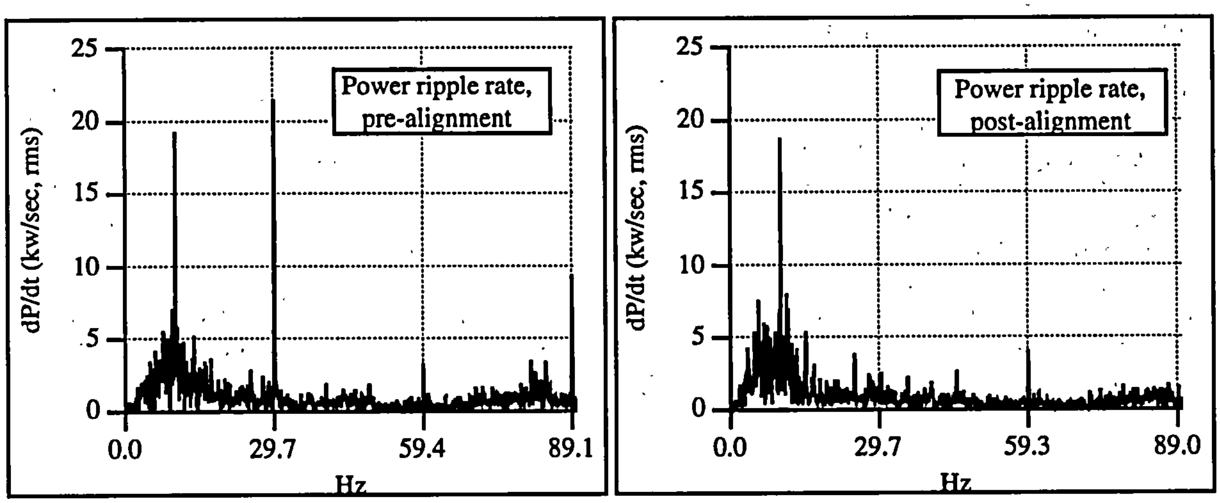

Figure 11. Pump 6 pre- and post-alignment differentiated power spectra.

It should be noted that the change in flow rate between the pre-aligned and post-aligned case could have influenced the results somewhat (see the discussion that follows on flow-related instability), but it is unlikely that the effect would have been as significant as experienced.

Hydraulic stability

Hydraulically related energy (as opposed to mechanical sources, such as unbalance) of pumps has been identified as the dominant loading factor [9, 10]. It is of interest, then, to compare the response of different transducers to off-design flow. To provide some perspective on the usefulness of motor data in assessing flow stability, three pumps were tested at flow rates from shutoff to equal to or greater than pump best efficiency point (BEP).

Spectral power data for three pumps at near the best efficiency point flow and at minimum flow conditions are shown in Figures 12-17. Pump design information for the three pumps is provided in Table 1. The spectral power data is presented as normalized power - that is, the spectral rms power divided by the average running power. By so normalizing, the power data for different pump sizes and styles can be more readily compared.

Table 3. General Pump Design Parameters

Parameter	Pump A	Pump B	Pump C
General style	Horizontal, single suction	Horizontal, double suction	Horizontal, single suction
Flowrate (gpm)	2000	1000	200
Nominal speed (rpm)	1780	1765	3500
Head at BEP (ft)	137	170	100
Specific speed	1990	1185	1570
Suction specific speed	12750	5760	10700
Motor power rating (hp)	75	50	7.5
Number of impeller vanes	6	8	5

There are several features of note in the power spectra for the three pumps:

- In general, as the flowrate was reduced from near BEP to minimum flow conditions, the level of low frequency noise in the motor power spectrum increased. This pattern has been observed in essentially all pumps studied.

- There are considerable variations in response to reduced flow. In particular, note that the spectral noise level for the two higher suction specific speed pumps (pumps A and C) grew much more dramatically than did that for pump B.

- The amplitude of the running speed peak for pump B was significantly greater at both flow conditions than for the other pumps. In the limited data collected on double suction pumps, it has generally been found that they are more likely to have higher running speed load fluctuations than single suction pumps. It is hypothesized that this may be due to the greater difficulty in manufacturing double suction pumps that are flow-wise symmetrical. Any such asymmetries which result in fluctuating head development at rotating speed (or harmonics) would also result in torsional load fluctuations. It will be recalled from previous discussion that misalignment can also influence running speed (and harmonic) energy. However, since any misalignment that existed would impose a relative constant load, regardless of flow rate, the flow-related energy can be segregated by observing running speed-related energy at various flow rates.

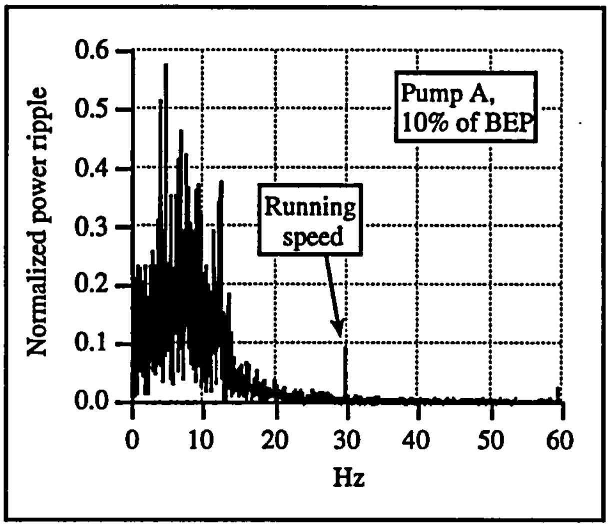

Figure 12. Pump A low flow power spectra.

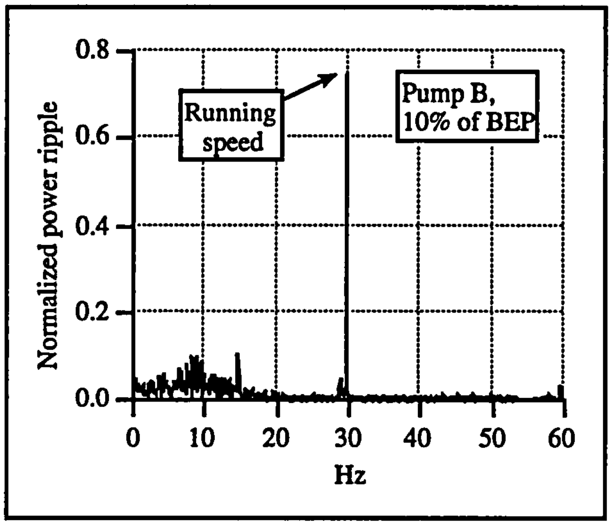

Figure 13. Pump B low flow power spectra.

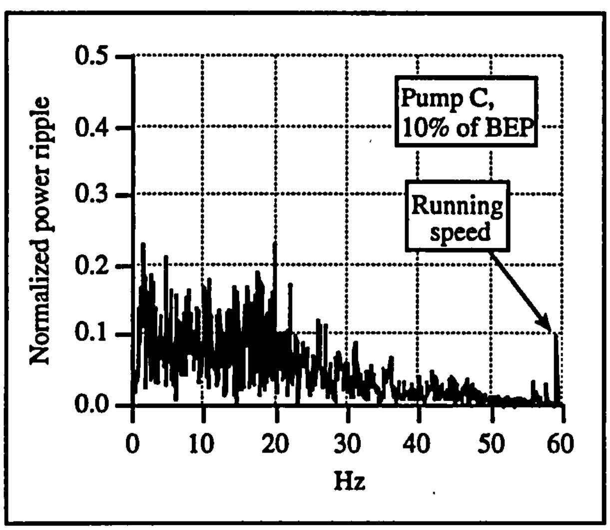

Figure 14. Pump C low flow power spectra.

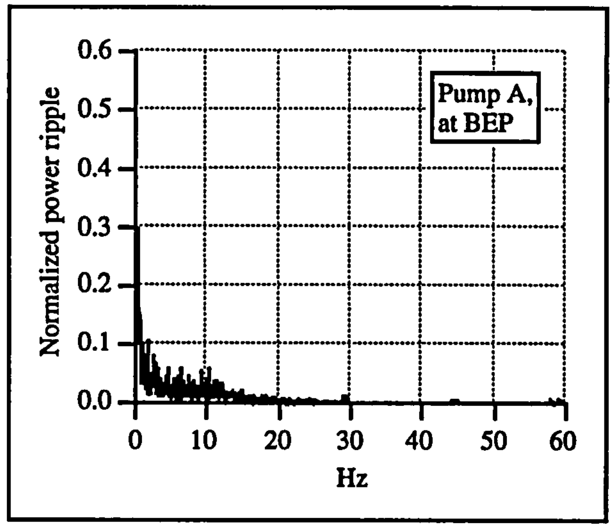

Figure 15. Pump A stable flow power spectra.

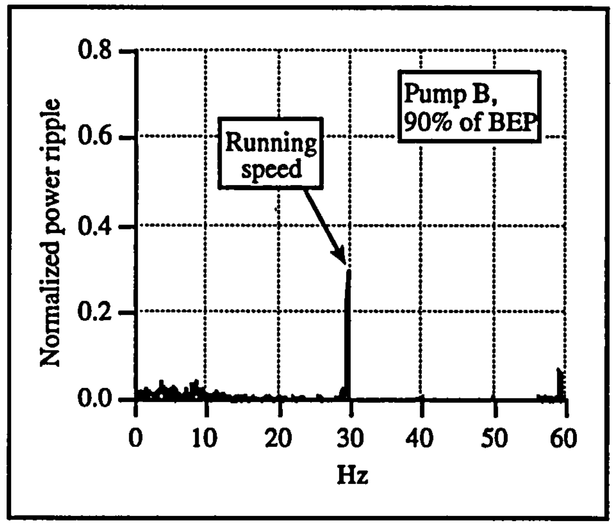

Figure 16. Pump B stable flow power spectra.

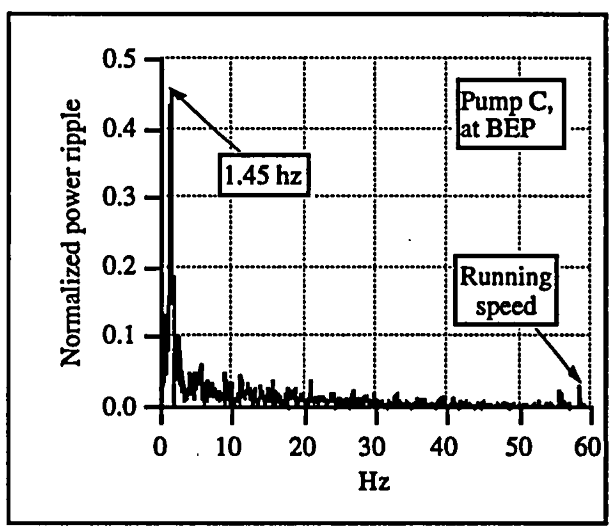

Figure 17. Pump C stable flow power spectra.

- The relative amplitude of the running speed peak for all three pumps increased as flow was reduced. This is due, in part, to the fact that the data are shown in a normalized fashion (normalized to average power), and the power at the lower flow rates for all three pumps is less than at near BEP. Even accounting for this factor, however, the running speed energy level is still higher at low flow condition.

- For both pump A and C, very low frequency load fluctuations were observed at the BEP conditions. For pump A, the frequency was less than 1 hz and poorly defined; for pump C, the spectral peak is relatively well defined, and is annotated in Figure 17 (1.45 hz). In the case of pump A, the apparent cause was system surge effects. For pump C, the peak was observed to occur from less than BEP to the maximum flowrate supported by the system in which the pump was installed (almost two times BEP). The frequency of the peak was found to be proportional to the flowrate. This observation led to the comparison of pressure pulsation and motor data provided in a subsequent section of this paper (*Pressure Pulsation*).

For comparison, vibration data collected at the pump inboard bearing on all three pumps for the same flow conditions are provided in Figures 18-23. Vibration data collected with accelerometers has a much wider flat frequency response range; the spectral data shown include the range from DC to above vane-pass frequency. All three pumps exhibited low overall vibration amplitude. Of the three pairs of vibration spectra, the only particularly notable feature is that the pump A vane-pass energy actually dropped as flow was reduced (contrary to what is often observed), and there were some minor changes in the amplitudes of some of the other running speed harmonics for pumps A and B.

Comparison of Motor Power and Pressure Pulsation Data

Further testing of Pump C was performed to compare pressure pulsation and motor power data. Water power developed by a pump is proportional to the head times the flow rate. The power input requirement to the pump is the water power divided by pump efficiency. Motor output power is equal to motor input power times motor efficiency. Thus, it is not unreasonable to expect that pressure pulsations, particularly low-frequency pulsations, would be readily detectable from pump motor input power.

Figure 24 provides time waveforms of low-pass filtered pump discharge pressure pulsation and motor input power at four flow rates, ranging from 50% to 200% of pump BEP. Even a simple review of the waveform relationships in time shows that the two parameters (power and pressure pulsation) are generally more closely related at the higher flow rate.

Figure 25 helps clarify the nature and strength of the relationships between power and pressure. To develop the data shown in this figure, the pressure and power were ac-coupled and amplitude normalized before performing the correlation. For the three higher flow rates, all the correlated energy at t=0 is positive, meaning positive coincident relationship. The nature of the relationship at 100 gpm is less clear at first glance - there are both positive and negative components.

Figure 26 provides a closer examination of both the 100 gpm correlation and a correlation performed for operation at very low flow (10 gpm). It can be seen that the positive correlation for the 100 gpm flow rate occurs at t=0 (again a positive coincident relationship), but that there is a higher negative correlation at about 0.1 second, suggesting an even stronger inverse relationship. The lack of clear correlation at 10 gpm demonstrates the extent of instability and lack of coherence at low flow conditions. From further examination of the correlations in Figure 25, it is apparent that there is a fluctuating component to the correlated data. Figure 26 shows spectra of the correlations for the three higher flow rate cases. Note that the frequency is roughly proportional to the flow rate, which suggests possible vortex shedding.

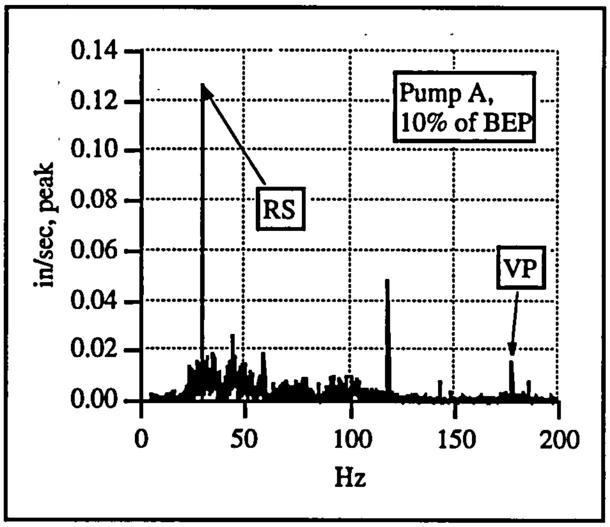

Figure 18. Pump A low flow vibration spectra.

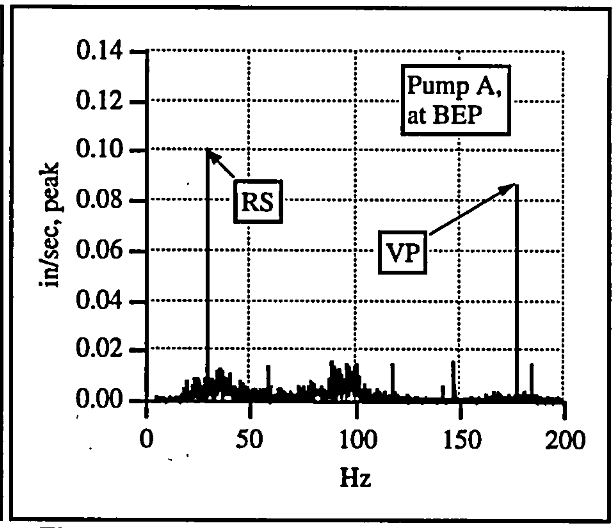

Figure 21. Pump A stable flow vibration spectra.

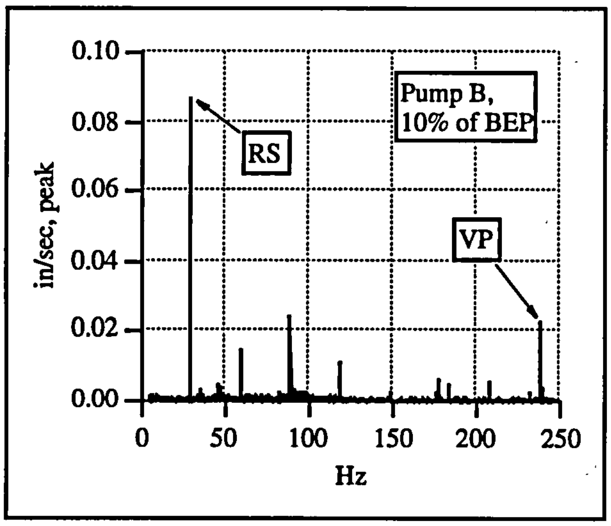

Figure 19. Pump B low flow vibration spectra.

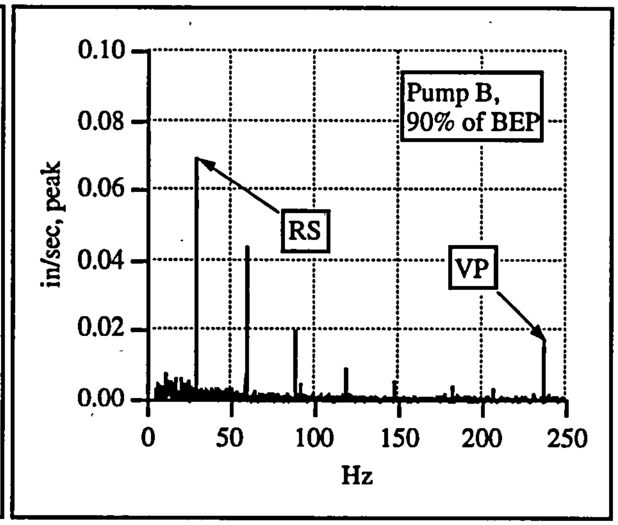

Figure 22. Pump B stable flow vibration spectra.

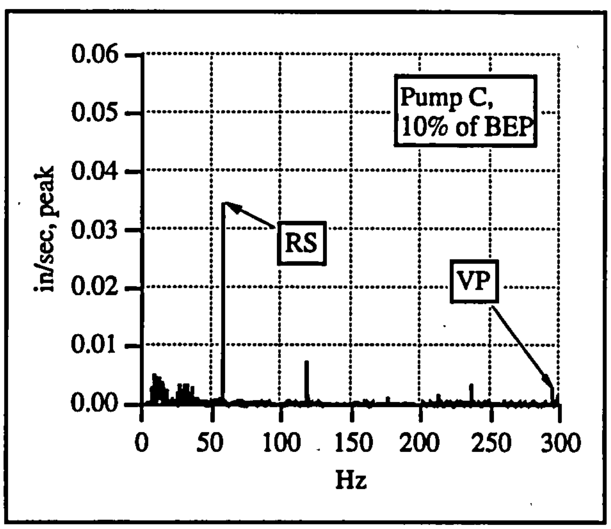

Figure 20. Pump C low flow vibration spectra.

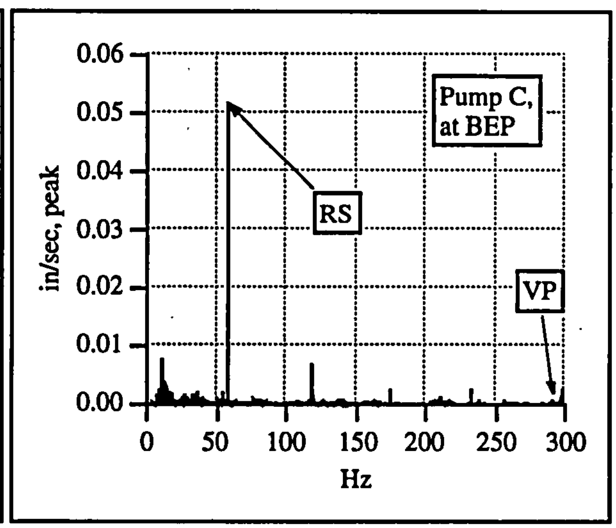

Figure 23. Pump C stable flow vibration spectra.

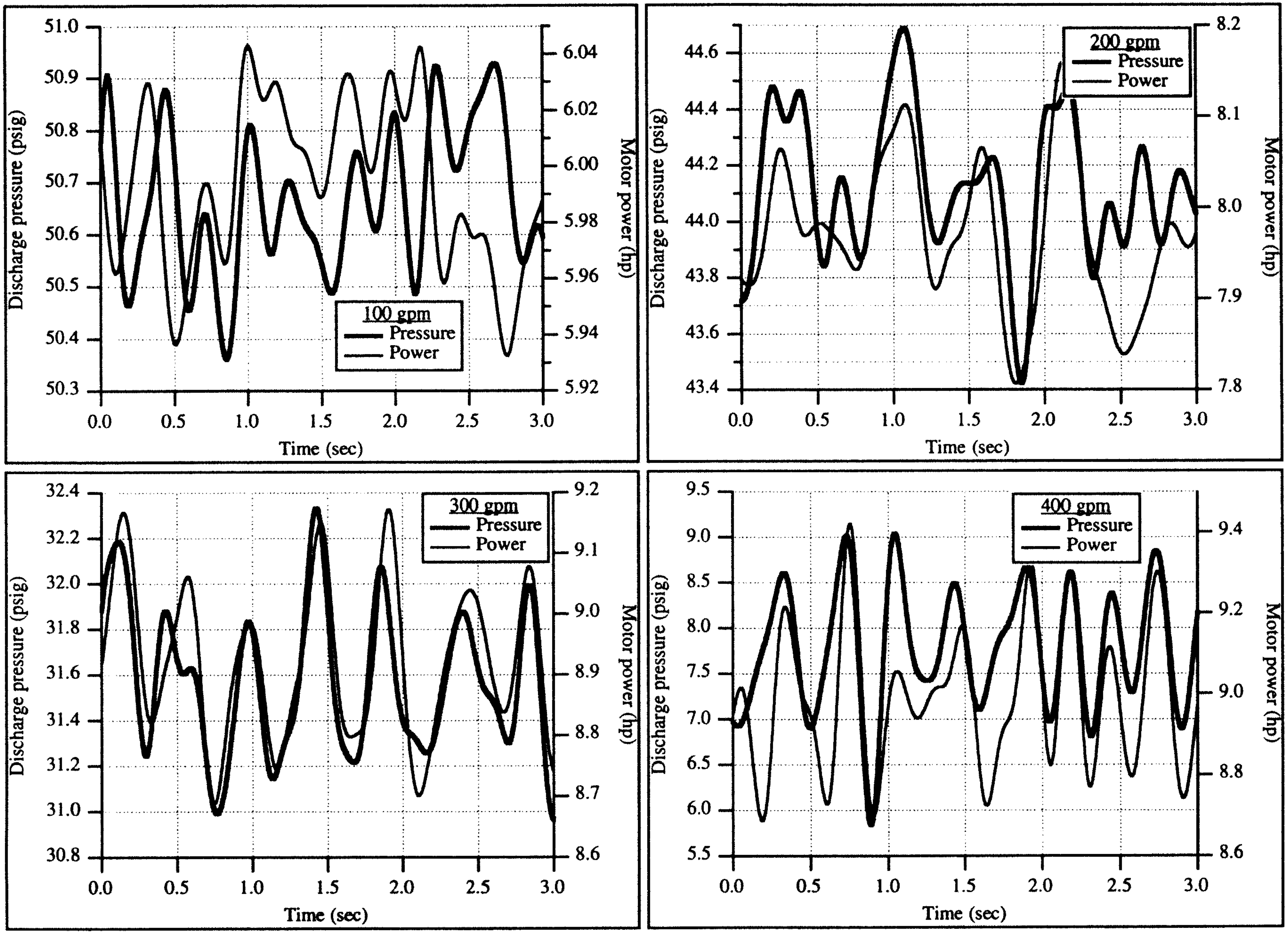

Figure 24. Low-pass filtered pressure pulsation and motor power time waveforms at four flow rates (BEP is 200 gpm).

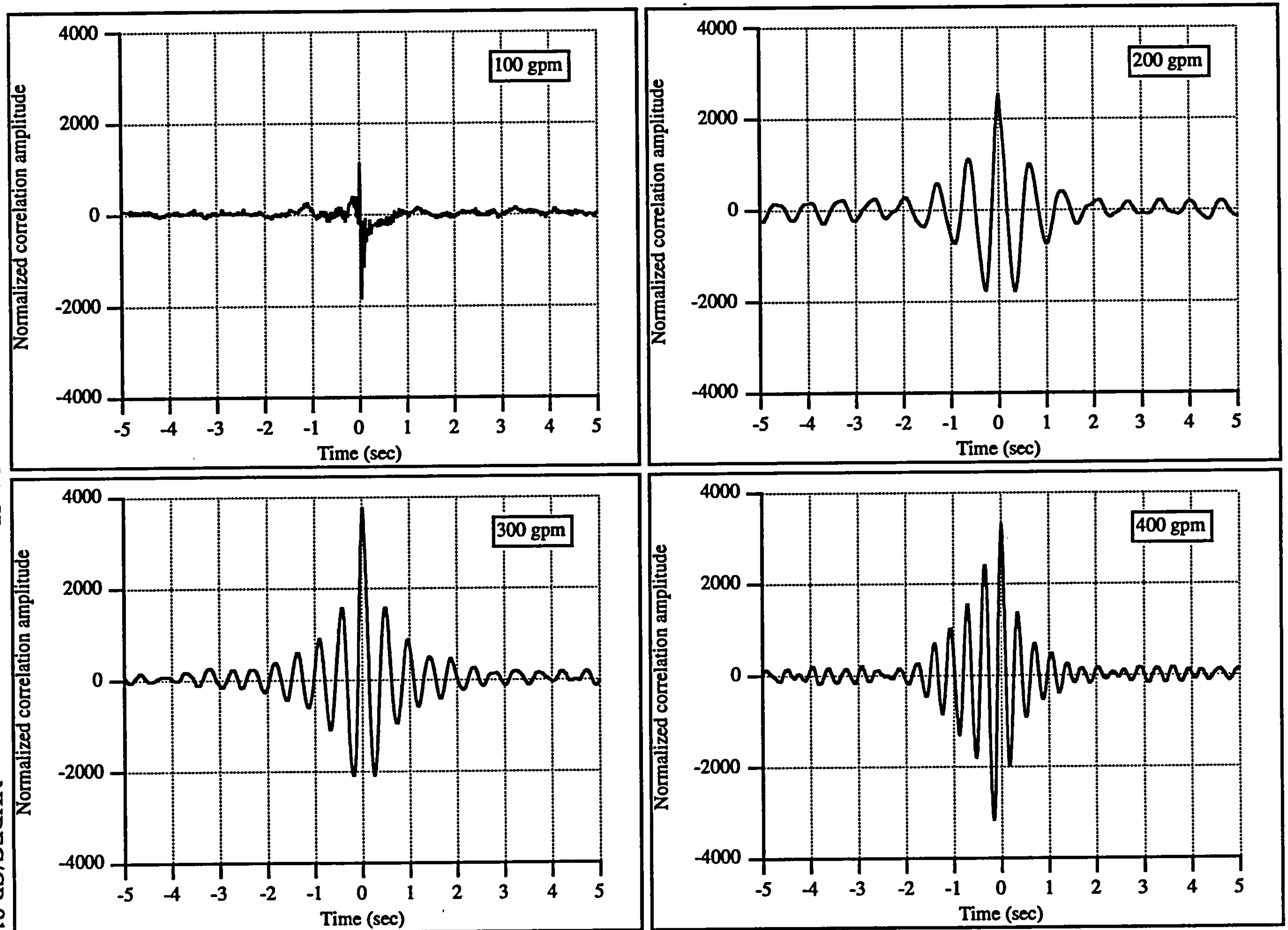

Figure 25. Normalized cross-correlations of pressure and power at four flow rates.

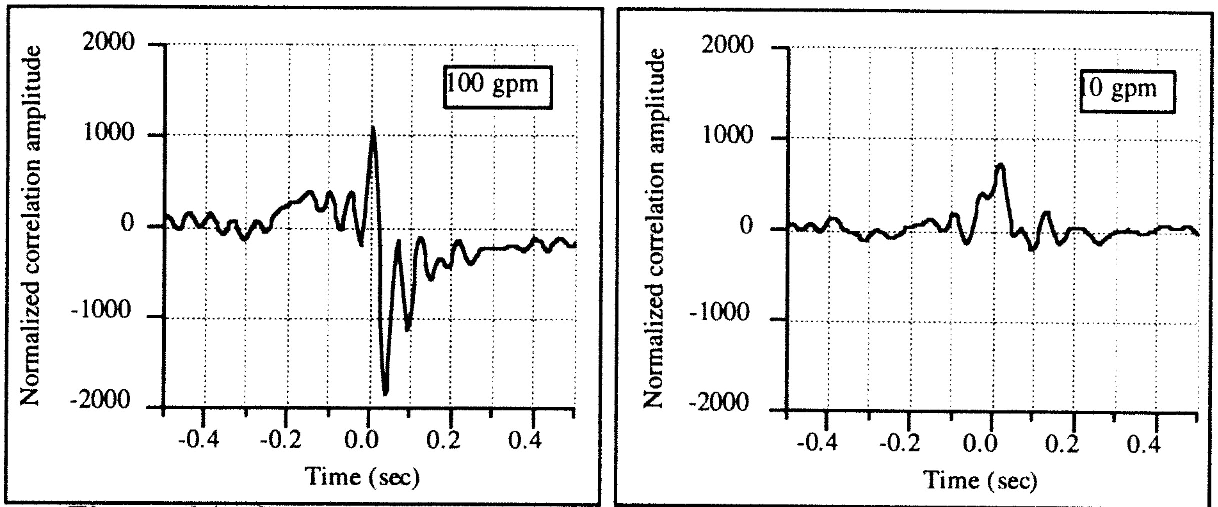

Figure 26. Correlations of pressure and power at reduced capacities (50% and 5% of BEP, respectively).

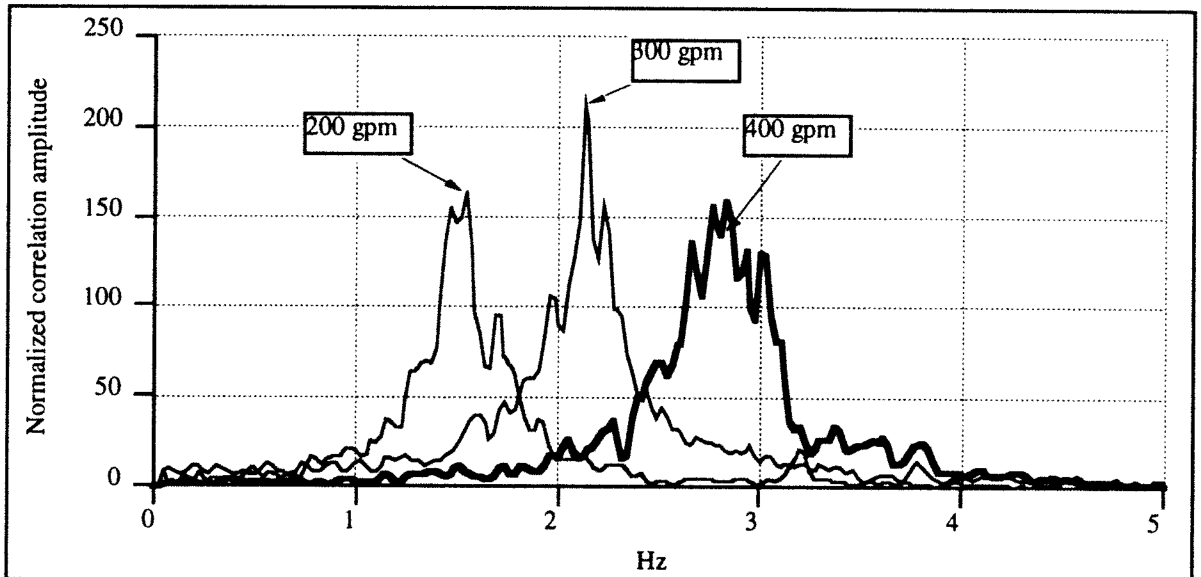

Figure 27. Spectra of pressure and power correlations.

Effect of Degraded Suction Conditions

At the Y-12 defense plant in Oak Ridge, an on-line motor current monitoring system has been installed to trend motor data on pumps, fans, and chiller compressors. The system is typically set up to collect overall current amplitudes hourly, and spectral data several times a day. One example of the value of the system was demonstrated when it was successful in detecting suction strainer clogging of a chilled water pump. Although the pump is relatively small - 10 hp - it provides chilled water in support of the maintenance of precise conditions in national calibration facilities.

A trend plot of motor current for two identical pumps operating in parallel over the course of 35 days (shortly after a chiller facility restart) is shown in Figure 28. In Figure 29, the motor power spectrum for the J102 pump at the end of the trend period is shown. Based on previous data, it was concluded that it was likely that the suction strainer for J102 was partially blocked. Suction pressure measurements were made to confirm the suspicion, and the suction was opened to inspect

the strainer. A picture of the strainer is shown in Figure 30. Power spectral data for the same pump after cleaning the suction strainer is shown in Figure 31.

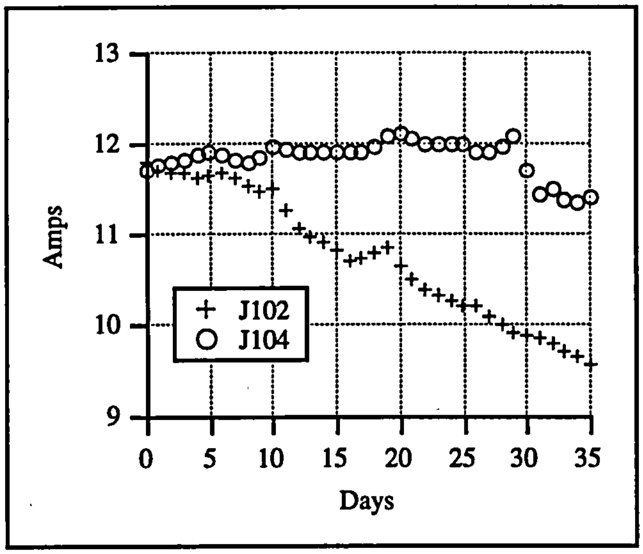

Figure 28. Parallel chiller pump current trends.

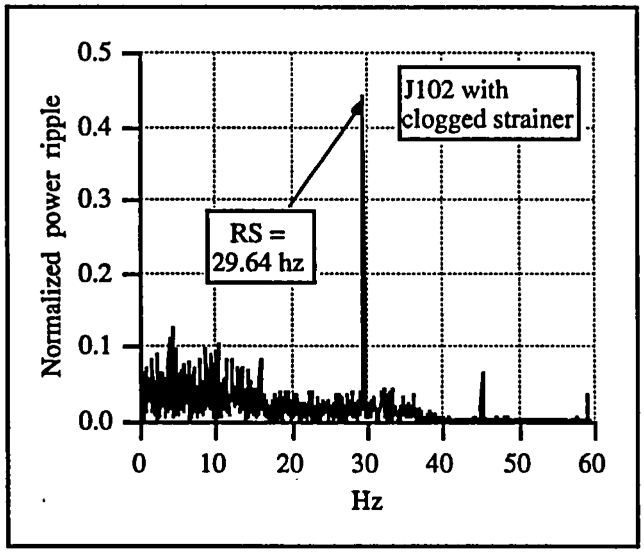

Figure 29. J102 power spectrum before strainer cleaning.

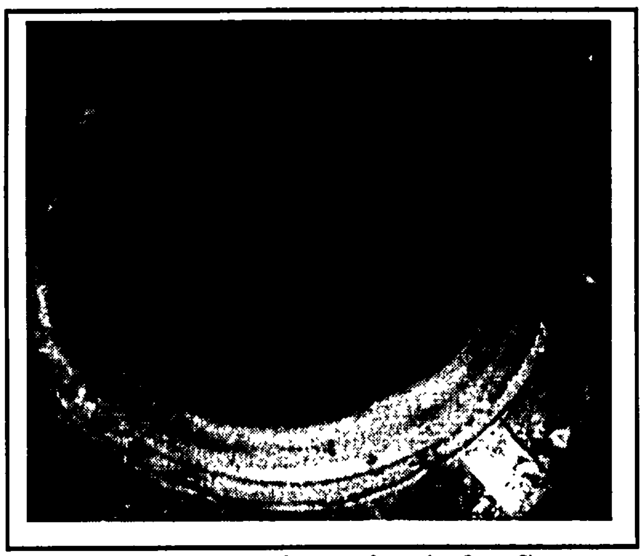

Figure 30. J102 suction strainer (as found).

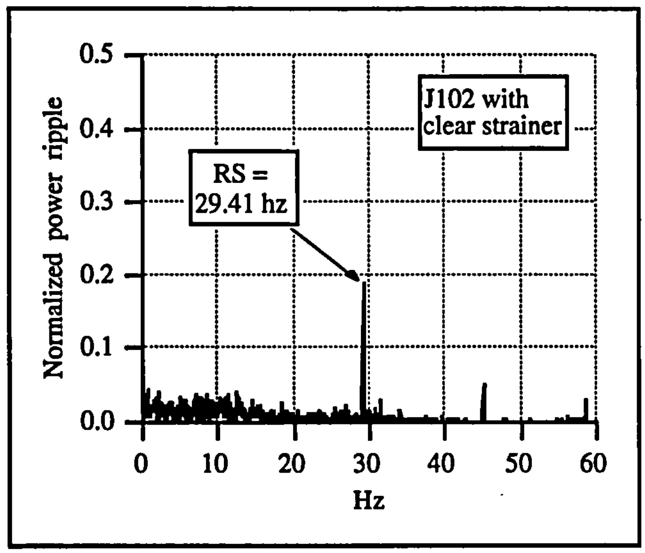

Figure 31. J102 power spectrum after strainer cleaning.

Monitoring Motor Condition

The use of motor data to help understand motor conditions would appear to be a more straightforward use of current, power, or phase data. It is beyond the scope of this paper to address the detection of stator and rotor degradation using on-line data, but it is worth noting that there are some significant uncertainties, in the author's opinion, regarding assessments of certain conditions (such as rotor bar failures or other anomalous rotor conditions) based on single measurements and comparison with some absolute criterion. Trending of data is certainly helpful in identifying degradation patterns. Advanced motor models hold promise for improving on-line degradation detection, but work remains to be done.

Limitations of the use of Motor Data in Assessing Drive Train Conditions

Although motor data has been helpful in the diagnosis of certain conditions, it has been generally unsuccessful in detecting some other conditions. Mechanical unbalance and bearing wear, except in cases where either are severe, are two degradation means that are much more readily detected by vibration (or human senses such as sound and touch). While both conditions have been seen through the use of spectral data, the level of influence is extremely small, and has not proven to be universally reliable, particularly in light of the fact that other means are so effective.

As a general rule, sources of energy that are purely radial or axial in nature have minimal effect on motor signal amplitudes. There are obviously exceptions, such as cases where the motor rotor axial or radial motion is such that it significantly affects the stator-to-rotor magnetic field relationship (and thus results in changes in produced torque).

Another important consideration when attempting to use motor data for trending is the recognition that bus voltage can significantly influence some parameters. Figure 32 shows the effect of bus voltage variations on measured rms current for a premium efficiency 100 hp, 480 V motor, with the motor operated at rated load conditions. The first data set, at rated voltage, was simply a repeat of testing at the same control conditions as for the base case, with the exception that slight variations in the three phase bus voltage distribution had occurred (in the external power supply). While there was only a minor effect for this case, there were obviously significant variations in measured average current (for the undervoltage and overvoltage cases), and among the individual phases with unbalanced voltage.

This effect is not merely an academic observation. At power plants, there can be significant variations in bus voltage, particularly between normal operation and outage conditions, since power supplies for these two conditions normally come from different transformers (unit auxiliary transformers during normal operation and startup transformers during outages).

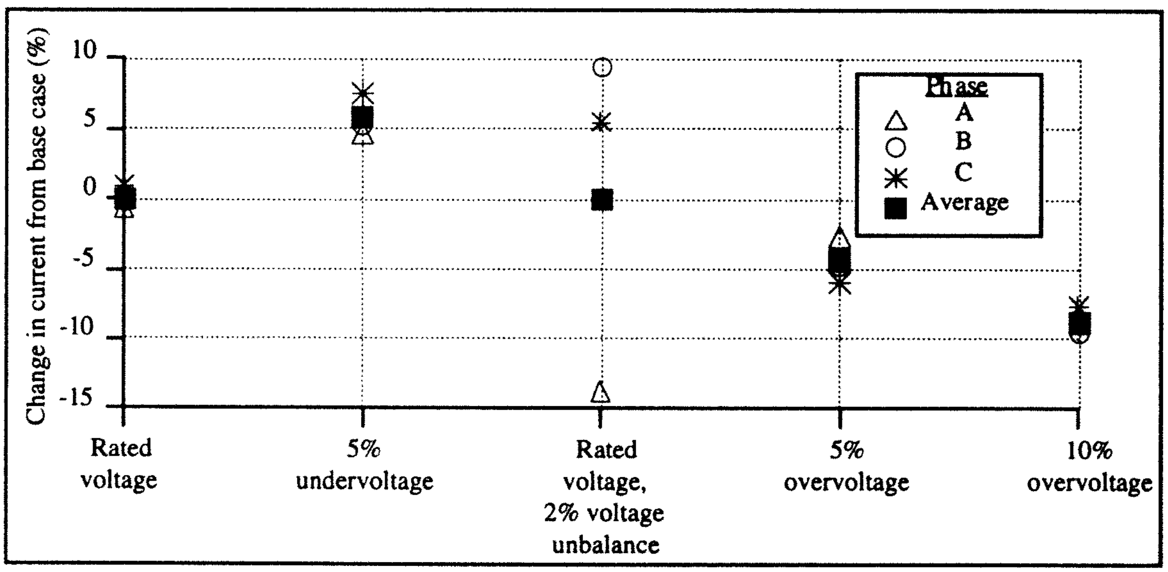

Figure 32. Effect of variation in voltage supply on current amplitude

In contrast to current amplitude, motor power is not typically as dramatically affected. The change in total motor input power is shown in Figure 33. It should be noted that the magnitude of the effect of voltage amplitude and unbalance on both current amplitude and overall power (and hence, efficiency) can vary, depending upon motor type and the operating load.

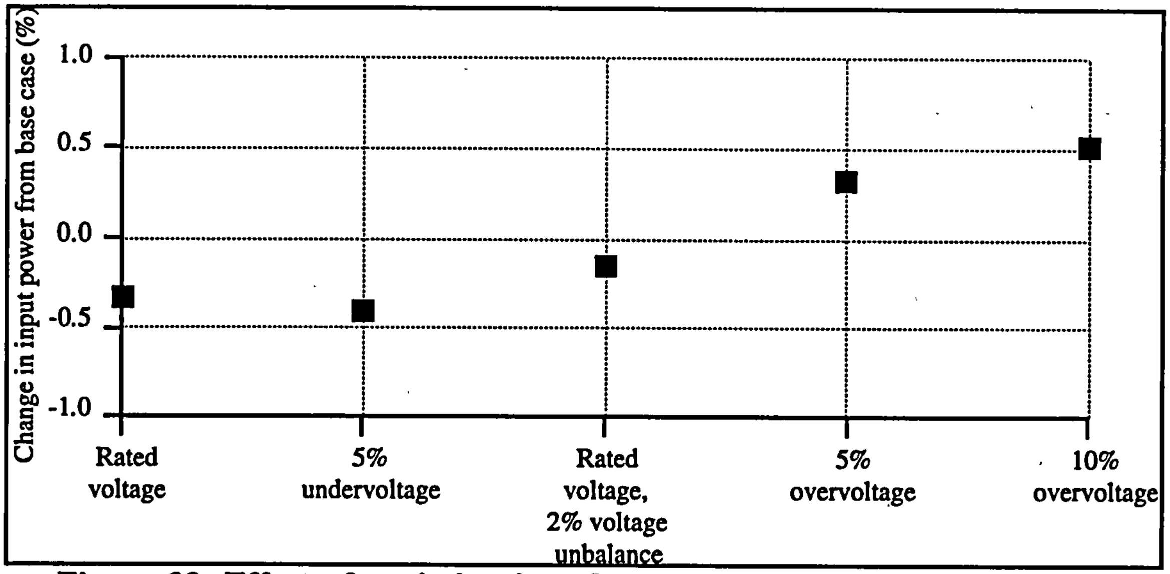

Figure 33. Effect of variation in voltage supply on total motor power

The lower sensitivity of power to bus voltage conditions is clearly a strong point in favor of power vs. simple current monitoring. However, that does not preclude the useful application of motor current - it simply means that additional care must be exercised in the analysis process. Given the inherent additional safety considerations often associated with power monitoring, a case can certainly be made for monitoring only current, and analytically accounting for voltage effects..

Summary

Motor data has been found to be capable of providing insights into driven equipment conditions that are difficult to detect with other common, field usable technologies. In the case of pumps, it can provide insights into hydraulic stability, suction conditions, alignment, and the nature of load fluctuations. Improved insights into other drive train conditions, such as gears, is also provided (although that subject has not been discussed herein).

Motor measured data is a relatively young technology. Rules of thumb developed for older diagnostic technologies such as vibration analysis need to evolve for motor-measured data in order to help its maturity. Even without hard standards (which it might be noted are difficult to apply universally, even for fields such as vibration), motor data can be extremely useful in providing an alternative view to equipment conditions.

As the move toward condition-based and risk-based maintenance accelerates, not only within the nuclear utility field, but in the broad spectrum of industries that are attempting to minimize unplanned outages and unnecessary maintenance, it will be important that diagnostic engineers and technicians carefully select the *combination* of tools that will be most effective in providing insights into critical equipment. It is envisioned that predictive programs of the next century will include combinations of periodic and on-line monitoring techniques that provide independent means of assessing the conditions that are most likely to cause equipment degradation. As has been shown in this paper, the use of the motor as a transducer to help understand pump conditions is not only possible – it often provides a perspective that cannot be gained from other technologies.

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REACTOR COOLANT PUMP TESTING USING MOTOR CURRENT SIGNATURE ANALYSIS

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Framatome Technologies, Inc.
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Florida Power Corp.

This paper describes reactor coolant pump motor testing carried out at Florida Power Corporation's Crystal River plant using Framatome Technologies' new EMPATH (Electric Motor Performance Analysis and Trending Hardware) system. EMPATH ™ uses an improved form of Motor Current Signature Analysis (MCSA), technology, originally developed at Oak Ridge National Laboratories, for detecting deterioration in the rotors of AC induction motors.

Motor Current Signature Analysis (MCSA) is a monitoring tool for motor driven equipment that provides a non-intrusive means for detecting the presence of mechanical and electrical abnormalities in the motor and the driven equipment. The base technology was developed at the Oak Ridge National Laboratory as a means for determining the affects of aging and service wear specifically on motor-operated valves used in nuclear power plant safety systems, but it is applicable to a broad range of electric machinery.

MCSA is based on the recognition that an electric motor (ac or dc) driving a mechanical load acts as an efficient and permanently available transducer by sensing mechanical load variations, large and small, long-term and rapid, and converting them into variations in the induced current generated in the motor windings. The motor current variations, resulting from changes in load caused by gears, pulleys, friction, bearings, and other conditions that may change over the life of the motor, are carried by the electrical cables powering the motor and are extracted at any convenient location along the motor lead. These variations modulate the 60 Hz carrier frequency and appear as sidebands in the spectral plot.

EMPATH ™ utilizes a unique patented signal conditioning circuit to demodulate the signal from the 60 Hz carrier and present an unambiguous spectral display. All of the known MCSA technologies are employed in EMPATH ™, including amplitude demodulation and phase demodulation, plus the option to calculate motor power and phase angle. Motor current signatures, obtained in both time and frequency domains, provide equipment condition indicators that are then trended over time to provide early indications of degradation.

The system is portable, totally non-intrusive, and very quick and easy to use. It is a complete Windows ™ based package, using standard database structure, with totally open architecture.

At the Crystal River plant, the purpose of the MCSA testing was to determine the health of the rotor, in particular to detect any deterioration in the rotor bars, as well as alert the operators to any abnormalities in

the phase current, low power factor performance, etc. The EMPATH™ technology allows an estimation of the number of damaged, cracked, or broken rotor bars, plus a measure of the static and dynamic eccentricity between the rotor and stator. The frequency spectra also provides an accurate measure of the motor slip (pole pass frequency) and running speed plus the three phase currents, which show any possible phase imbalance.

The paper presents the time-based and frequency-based current signatures for the Crystal River reactor coolant pump motors and explains the diagnostic findings regarding the condition of the rotors and stators, the currents, and the running speeds.

The data is for several runs on each pump; typically, a run at low frequency, standard resolution was made, followed by either a high frequency run or another low frequency run, but at a higher resolution, to ensure capture of all pertinent information.

Each pump has a summary sheet, noted as 'airgap & bars' at the top, that serves to collect all the results in a raw form. The ' EMPATH Results Summary' presents indications of the health of each motor, based on the results of the associated time and frequency plots. The key is in the 'Summary of Diagnostic Results' section, which shows the number of slip harmonics, their level, per-cent of the total spectra, and rotor bar health index. The index comes from empirical relations developed on the basis of numerous cases of rotor degradation. If the actual number of bars is known, this figure represents the equivalent number of bad bars, based on high resistance in the rotor. When the number of bars is not known, this figure represents a percentage of bad bars, assuming there are 100 bars, total. Thus, for Pump 1A, there appears to be the equivalent of 2-3 bad bars, or roughly 2% of the total of bars degraded. Pump 1B appears to be the least degraded on an overall basis, and really doesn't need to have its monitoring schedule altered.

Pumps 1C & D do not show as much degradation as 1A, but should be monitored more closely, at least until their normal operating characteristics can be cataloged over a sufficient period of time. Pumps 1C & D also show slight indications of static eccentricity, as the spectral amplitudes are above the background noise level. These results do not presently indicate any major problem, but this should be monitored for any changes that may occur over short time intervals. These eccentricities represent the variation in the air-gap between the rotor and stator, and should generally show no indications, as for Pumps 1A & B.

The line 'Normally expected levels --->' shows what results are expected for motors in 'new' condition, or those with virtually no detectable rotor degradation. The column headings are explained in the 'Legend', in the middle of the summary page.

The bottom of the page presents a summary of the raw current data, for each pump, for each of the three phases. This information was taken from the actual current traces that were acquired, and shown on the accompanying plots. For each time waveform, the RMS, maximum, and minimum values were found. These were then averaged and the percent difference found among all three phases. The largest difference of 3.5% was found for Pump 1D, meaning that the swing in peak current varied by this amount among the phases.

This amount of current variation could be the result of a 1-2% variation in phase voltage, which could lead to slight decrease in starting and running torque, synchronous speed, full-load speed, and starting current. It could also result in a 5-7% decrease in the full-load power factor, which would lead to inefficient operation. A full-load temperature rise of up to 7-8% could also result, which would ultimately shorten the life of the windings. Because no voltage measurements were taken at the same time, no estimates can be made of power imbalance or power factor, other than that discussed above.

As far as the authors are aware, this is the first time that MCSA has been utilized to diagnose the condition of a reactor coolant pump, while it continued to run. The results of the motor current testing have clearly shown the operating condition of these motors. In this particular situation, there does not appear to be a need for any immediate remedial action. However, an attemp should be made to correct the slight current imbalance, as this could cause uneven wear and degradation of the stator windings as operation continues. All the pumps, except B, do show some indication of high resistance in the rotor which should be checked at some convenient time during an outage. Repeated starting will continue to cause increases in the rotor resistance and should be avoided, as much as possible.

It is recommended that repeat measurements be made in approximately 3-4 months, so that an operating history can be acquired for comparison with future results. It would also be most useful to be able to acquire similar information on rc pumps at other nuclear power plants for extension of the operating database, as well as having a larger population of samples for future comparisons to be made.

Results of EMPATH current signature analysis of four reactor coolant pump motors at FPC CR-3

Data acquired 11/7/95

<--- Operating Characteristics --->				<--- Summary of Diagnostic Results --->									
		Ave run		EMPATH Slip				Power line dB diff		rotor bar	slot harmonics, freq/level		
Pump No.	RS, rpm	cur, amps	% FL	Se, fund	Se, harm	level	slip %	upper sb	lower sb	health	psh	2nd sh	se/de indication
1A	1188	488	71.2	1	2	3.49E-04	66	48.7	47.74	2.3	no	no	no/no
1B	1188	485	70.8	1	0	2.79E-04	32.9	51.38	50.08	0.9	no	no	no/no
1C	1188	481	70.2	0.94	2	4.61E-04	64.7	53.2	53.03	1.28	-69.5	-79.6	yes/no
1D	1188	488	71.2	0.94	2	3.50E-04	59.6	50.52	49.74	1.83	-66.36	-78.39	yes/no
Normally expected levels:--->				yes	none	varies	25-30	>60	>60	<<1	no	no	no/no

Note: all pumps, except 1A, showed a peak around 12 Hz, with harmonics out to the fifth, with the second generally the strongest. This could be the result of a lube oil pump, flow induced vibration, or possibly turbulence

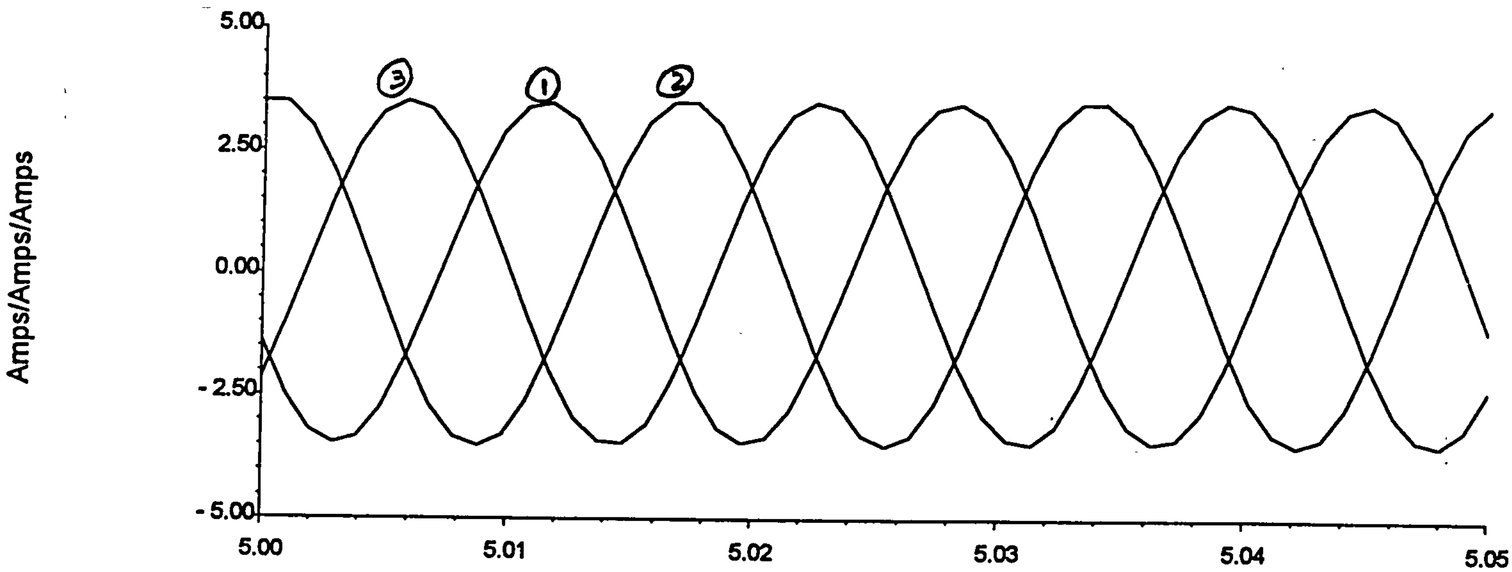
Legend:
- RS=running speed, rpm
- FL=full load amps
- Se, fund=location of EMPATH slip fundamental, Hz; (EMPATH slip is the same as pole passing frequency)
- Se, harm=number of EMPATH slip harmonics
- level=sum of spectral amplitudes of EMPATH slip fundamental and harmonics
- slip %="level" divided by rms level of RMS DEMOD spectra
- upper sb=dB level of upper slip sideband of power line peak
- lower sb=dB level of lower slip sideband of power line peak
- rotor bar health=estimate of either the per-cent or number of broken or cracked rotor bars
- psh=principal slot harmonic level, if present; frequency shown on 'airgap & bars' summary sheet
- 2nd sh=second slot harmonic level, if present; frequency shown on 'airgap & bars' summary sheet
- se/de=static and dynamic eccentricity of rotor, with respect to stator
- se/de indication=presence of eccentricity detected, as shown on 'airgap & bars' summary sheet

Raw current summary of results

	Pump 1A			Pump 1B			Pump 1C			Pump 1D			
	RMS	Max	Min	RMS	Max	Min	RMS	Max	Min	RMS	Max	Min	Overall RMS Ave
Cur 1	2.49	3.44	-3.48	2.47	3.58	-3.48	2.44	3.42	-3.43	2.48	3.42	-3.5	
Cur 2	2.54	3.53	-3.53	2.52	3.51	-3.53	2.51	3.53	-3.48	2.53	3.54	-3.54	
Cur 3	2.5	3.51	-3.49	2.48	3.48	-3.49	2.45	3.42	-3.46	2.48	3.51	-3.47	
Ave	**2.51**	**3.49**	**-3.5**	**2.49**	**3.52**	**-3.5**	**2.47**	**3.46**	**-3.46**	**2.50**	**3.49**	**-3.50**	2.49
% diff	**2.0%**	**2.6%**	**-1.4%**	**2.0%**	**2.9%**	**-1.4%**	**2.9%**	**3.2%**	**-1.4%**	**2.0%**	**3.5%**	**-2.0%**	2.2%

Note: raw current values to be multiplied by CT ratio (200) to obtain actual running current levels

Reactor Coolant Pump Motor 1A, Run 2, printed on 12/8/95 at 11:59:57 AM
Reactor Coolant Pump Motor 1A



Current 1= 2.47
Current 2= 2.56
Current 3= 2.49

Air-gap eccentricity computations for induction motors

at CR-3, RC Pump Motors PUMP 1A

fag = { (nrt*R +/- nd)*[(1-s%)/p] +/- nws }*f1

where:
- f1 = supply line frequency, Hz
- nrt = any integer
- R = number of rotor slots or bars
- nd = any integer
  - = 0 for static eccentricity, psh
  - = 1,2,..., for dynamic eccentricity, de
- p = number of pole pairs
- s% = per unit slip = EMPATH slip/(2 x f1)
- nws = odd integer

- s% = (synch - RS)/synch = Se/2f1, %
- synch = f1/p, Hz
- Se = EMPATH slip = (synch - RS) x2p, Hz
- slip = synch - RS, Hz
- bar current freq = s% x f1, Hz
- pole passage = slip x 2p = 2 x (s% x f1) = EMPATH slip, (Se), Hz

bold italic denotes user input required

				level, dB or ampl			min	max
	RS	19.8	***Hz***	4.99E-05			min	max
	f1	59.97	***Hz***	0.09		***RMS current level, amps***	2.36	2.52
	R	104	***slots or bars***			***CT ratio***	200	
	p	3	***pole pairs***			Motor current level, amps	472	504
meas pole passage = Se		1	***Hz, EMPATH slip***	2.38E-04		average running current, amps	488	
rotor bar degradation when ~ 50-60 dB				48.7	47.74			
	S	126	***no. of stator slots***			***Full load current, amps***	685	
no. Se harmonics		2	***harm level***	1.11E-04		% FL	71.2%	
	synch	19.99	Hz					
	s%	0.95%	per unit slip, %			***Demod spectra level***		
calc pole passage=Se		1.14	Hz			***RMS***	5.29E-04	
Se slip sidebands about RS, Hz		20.8	18.8			***AM***	1.24E-04	
	slip	0.19	Hz			***PM***	4.05E-04	
bar current frequency		0.57	Hz	***Upper SB***	***Lower SB***	AM + PM	5.29E-04	
Se slip sidebands, (1+/- 2s%)*f1, Hz		61.11	58.83	-48.61	-47.65			
	R*f1	6236.88	Hz			***slip level/RMS level***		
bar-pass frequency	RS*R	2059.2	Hz			***slip %***	66.0%	
2x bar pass frequency	2BPF	4118.4	Hz					
static eccentricity=	RS*R+/-f1, Hz	2119.17	1999.23					
principal slot harmonic, psh								
bar-pass sidebands	RS*R+/-2f1	2179.14	1939.26					
air-gap eccentricity	mid fag	2059.2	Hz					
	R*f1-mid fag	4177.68	Hz					
	mid fag/RS	104.00	no. slots/bars					
slot pass frequency	RS*S	2494.8	Hz					
(slot pass - bar pass) x no. poles		2613.6	Hz, na if neg					

nrt	nd	nws	fag(+)	*level, dB*	fag(-)	*level, dB*	fag(+)/RS	fag(-)/RS	del fag	del fag/RS	
1	0	1	2119.17	none	1999.23	none	107.03	100.97	119.94	6.06	psh
1	1	1	2138.97	found	1979.43	found	108.03	99.97	159.54	8.06	de
1	-1	1	2099.37	above	2019.03	above	106.03	101.97	80.34	4.06	de
1	0	3	2239.11	background	1879.29	background	113.09	94.91	359.82	18.17	psh
1	1	3	2258.91	levels	1859.49	levels	114.09	93.91	399.42	20.17	de
1	-1	3	2219.31		1899.09		112.09	95.91	320.22	16.17	de
1	0	5	2359.05		1759.35		119.14	88.86	599.7	30.29	psh
1	1	5	2378.85		1739.55		120.14	87.86	639.3	32.29	de
1	-1	5	2339.25		1779.15		118.14	89.86	560.1	28.29	de

fsh1 = f1*(R/p*(1-s%)+/-n), Hz
fshv = f1*(R/p*(1-s%)+/-2*(n-1)), Hz
fsh2 = fsh1 +/- 2*s%*f1, Hz

principal slot harmonics of flux/current, fsh1 (n=1), psh

fsh1= 2119.17 n= 1

vibration slot harmonics, fshv (bar pass frequency)

fshv= 2059.2

flux/current slot harmonic sidebands w/asymmetric rotor, fsh2, (psh+/-Se)

fsh2=	2120.31	1.14	Se sidebands around
	2118.03	1.14	fsh1 slot harmonics

Other notable frequency peaks

raw freq	*level, dB*	*demod freq*	*level*
3.85E+02	-85.7	none	
		found	
		above	
		background	
		levels	

Calculation of broken rotor bars, after Hargis, et al, for				at CR-3, RC Pump Motors			
				PUMP 1A			
Rs = sin(alpha)/[2*p(2PI - alpha)]							
dB version; n=2*N/(10^(dB diff/20)+2*p)							
					20log(x) = level, dB		
p = no. pole pairs		3		line level, dB	lower sb, dB	upper sb, dB	dB diff
N = no. rotor bars/slots		104		0.09	-47.65	-48.61	47.74
no. slip harmonics		2	x=	1.01	0.004145	0.003711	
ave run current		488	Rs=		0.004102	0.003673	
FLA		685	Rs(max)=	0.004102			
load %		71.2%	Rs(max;				
				0.011516	a=	0.069096	
Conditions and constraints					b=	0.434142	
alpha <= PI/2, (1.570796327)					alpha	0.417315	
					sin(alpha)	0.405307	
						0.434142	
						n=	2.302482
n = no. broken bars						n(int)=	3
dB version	n=	0.832729					

EMPATH Summary
Reactor Coolant Pump Motor 1A, Run 1
Printed at 10:56:40 AM on 11/15/95

CLIENT INFORMATION

Client: Florida Power Corp.
Date/Time of test: 11/7/95 11:39:49 AM
Location of plant: location of plant
Equipment name: Reactor Coolant Pump Motor 1A

Motor ID: motor id
Plant Name: Crystal River-3
Contact name/phone: Jim Bellamy x 3310 /(904)7956486

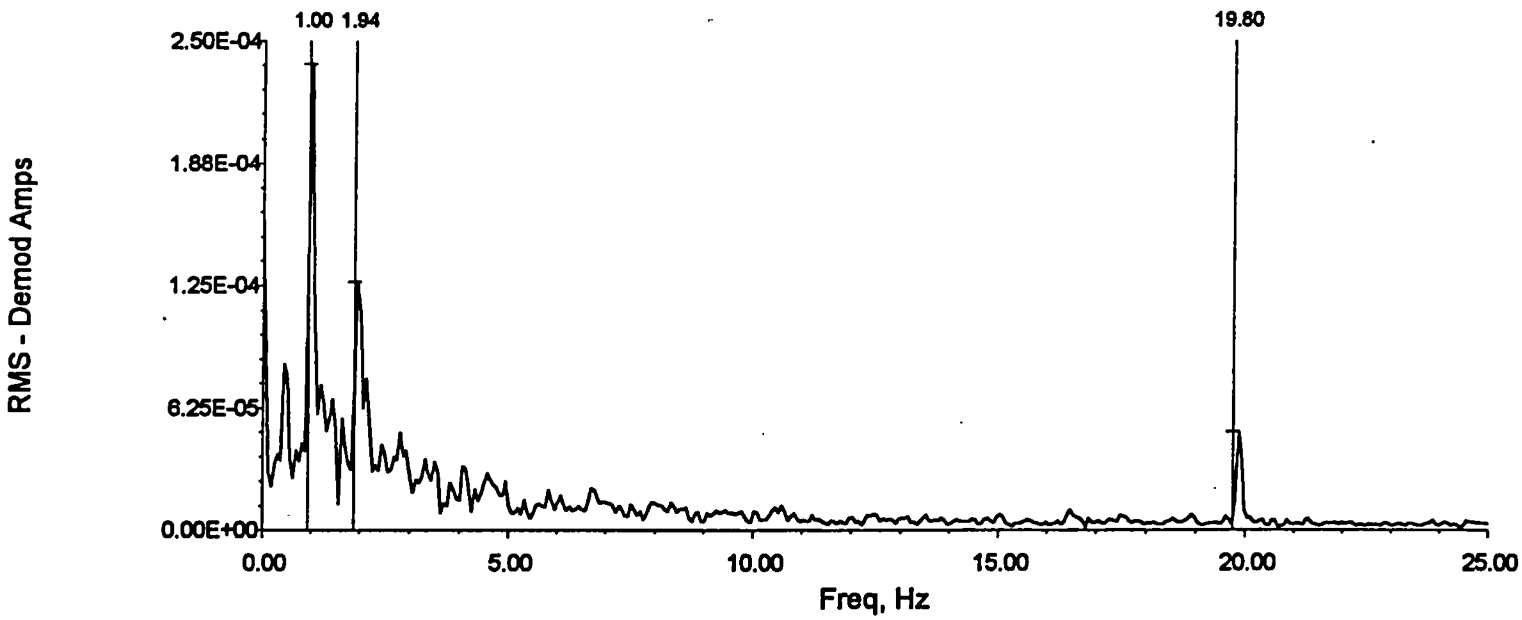
NAMEPLATE INFORMATION

Manufacturer: General Electric Co.	Model No: 295X141		
Voltage (AC/DC): ac 6600	Phases: 3	Serial number: 8367242	Torque(ft lbs): 51390
Poles/RPM: 6/1185	Full Load Current: 685		
Frame size: 6389	No Load Current:	Locked Rotor Current: 3960	Service Factor:
Amb Temp:	Ins Type: Type K	Duty: cont	No. Stator Slots:
Bearing Type-Rolling Element/Sleeve:		Opp End Brg No.:	
Dr End Brg No.:		Opp End Brg No.:	

EMPATH SYSTEM SET-UP

Analysis Freq: 400	Sample Rate: 1024	AAF Freq: 333	
Frame Size: 4096	Acq Time: 4	Freq Res: .25	No. Aves: 1
Line of Resolution: 1600	CT Ratio: 1000		
Raw Signal Gain:	Current1: 100	Current2: 100	Current3: 100
	Voltage1: 4	Voltage2: 4	Voltage3: 4
DEMOD Signal Gain:	RMS: 100	AM: 100	PM: 100
Range Settings:			
Channel 1: +-10V	Channel 5: +-10V	Channel 9: +-10V	Channel 13: +-10V
Channel 2: +-10V	Channel 6: +-10V	Channel 10: +-10V	Channel 14: +-10V
Channel 3: +-10V	Channel 7: +-10V	Channel 11: +-10V	Channel 15: +-10V
Channel 4: +-10V	Channel 8: +-10V	Channel 12: +-10V	Channel 16: +-10V

Reactor Coolant Pump Motor 1A, Run 2, printed on 11/21/95 at 2:31:14 PM
Reactor Coolant Pump Motor 1A

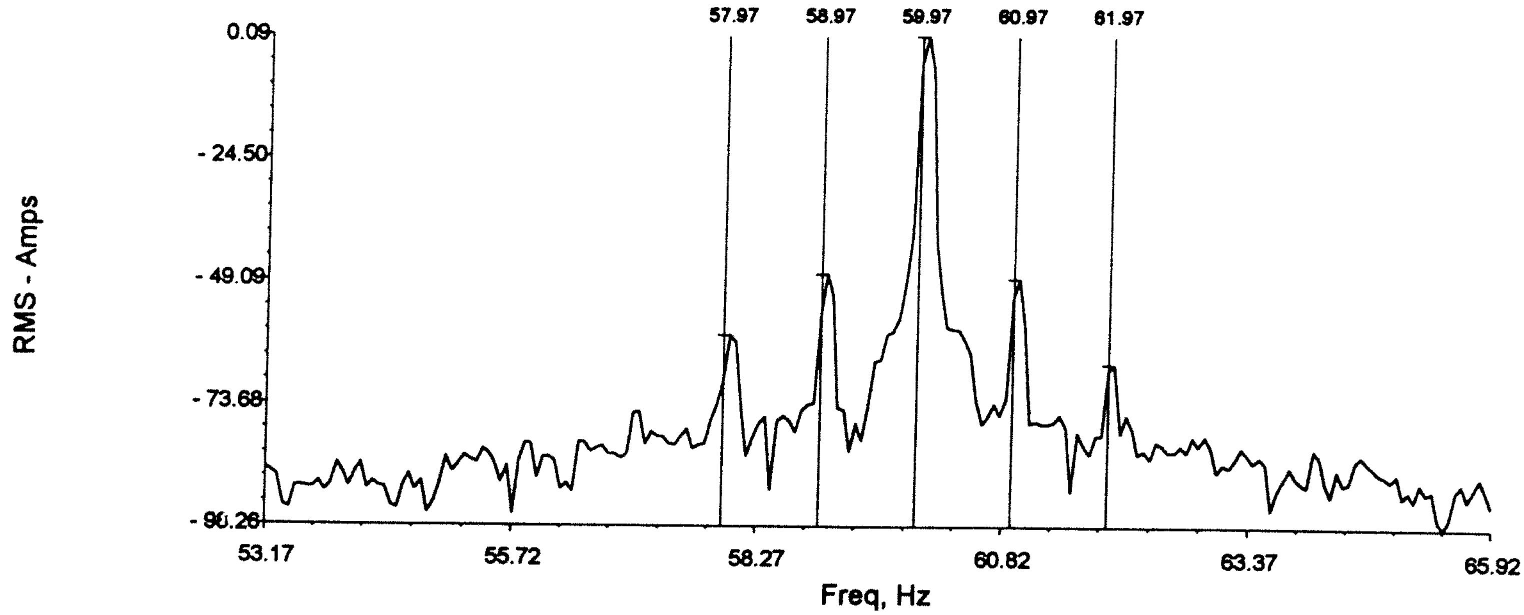


RMS Demod= 5.29E-04

Standard Cursors

X	Y1
1.00	2.38E-04
1.94	1.27E-04
19.80	4.99E-05

Reactor Coolant Pump Motor 1A, Run 2, printed on 11/21/95 at 2:16:04 PM
Reactor Coolant Pump Motor 1A

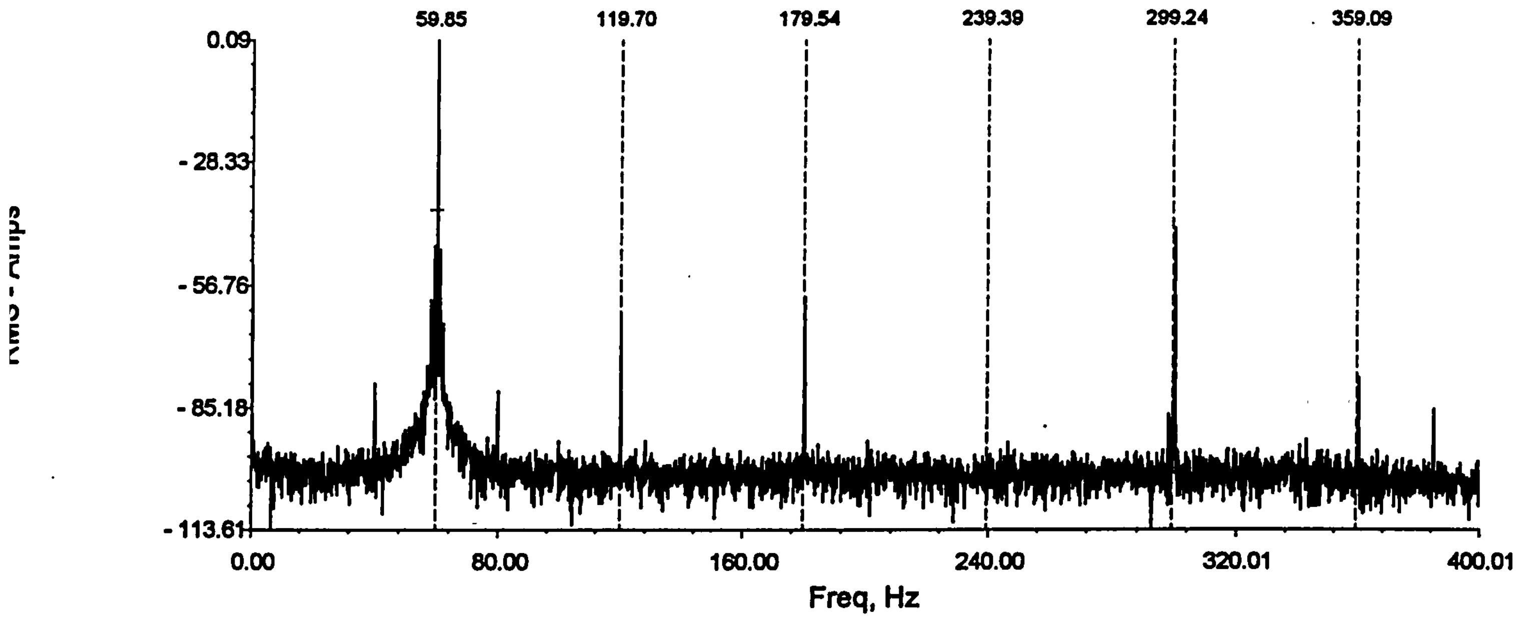


Current 1= 1.24

Standard Cursors

X	Y1
59.97	0.09
60.97	- 48.61
61.97	- 65.62
58.97	- 47.65
57.97	- 59.90

Reactor Coolant Pump Motor 1A, Run 2, printed on 11/21/95 at 2:20:10 PM
Reactor Coolant Pump Motor 1A

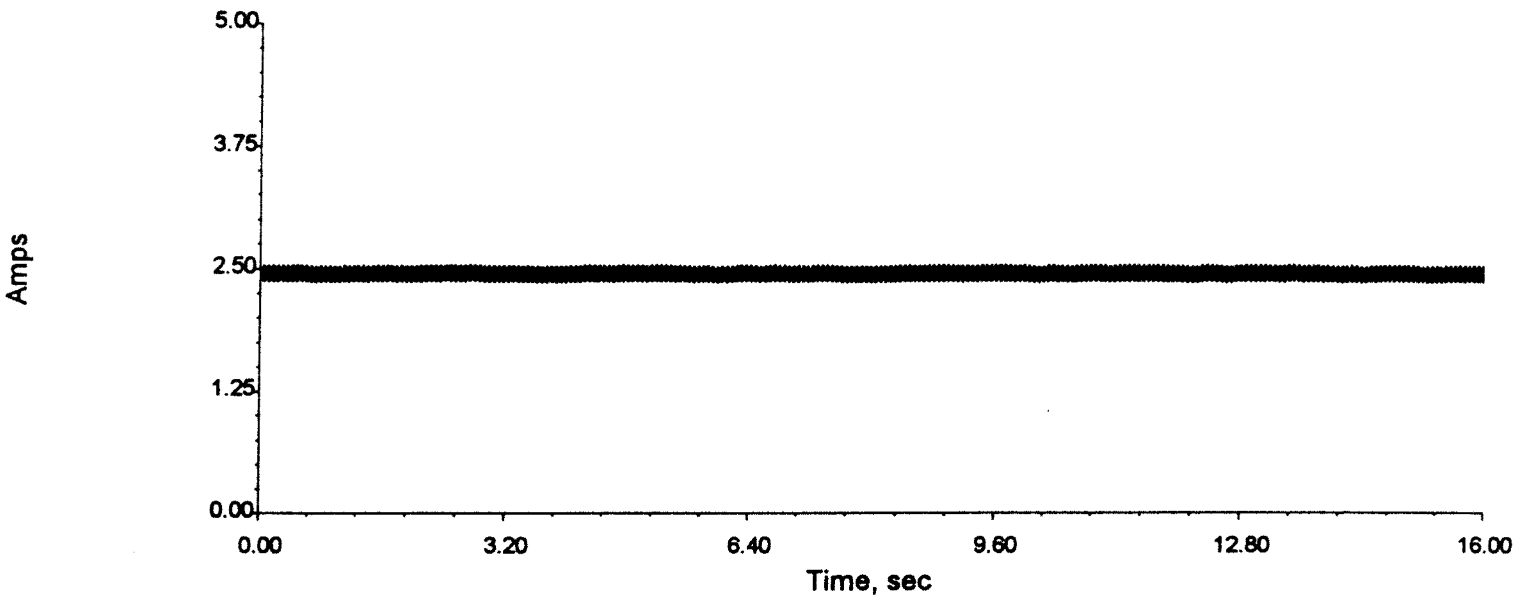


Current 1= 1.24

Harmonic Cursors

X	Y1
59.85	- 39.34
119.70	- 100.05
179.54	- 99.21
239.39	- 102.95
299.24	- 100.40
359.09	- 103.93

Reactor Coolant Pump Motor 1A, Run 2, printed on 11/21/95 at 2:28:11 PM
Reactor Coolant Pump Motor 1A



RMS Envelope= 2.45

WHAT WE LEARN FROM SURVEILLANCE TESTING OF STANDBY TURBINE DRIVEN AND MOTOR DRIVEN PUMPS

Bob Christie
Performance Technology

ABSTRACT

This paper describes a comparison of the performance information collected by the author and the respective system engineers from five standby turbine driven pumps at four commercial nuclear electric generating units in the United States and from two standby motor driven pumps at two of these generating units. Information was collected from surveillance testing and from Non-Test actuations. Most of the performance information (97%) came from surveillance testing.

"Conditional Probabilities" of the pumps ability to respond to a random demand were calculated for each of the seven standby pumps and compared to the historical record of the Non-Test actuations. It appears that the Conditional Probabilities are comparable to the rate of success for Non-Test actuations.

The Conditional Probabilities of the standby motor driven pumps (approximately 99%) are better than the Conditional Probabilities of the standby turbine driven pumps (82%-96% range). Recommendations were made to improve the Conditional Probabilities of the standby turbine driven pumps.

1 INTRODUCTION

Most nuclear electric generating units in the United States use turbine driven pumps and motor driven pumps in a safety related function as a power source to supply cooling water to selected equipment (reactor vessels or steam generators) in accident and transient conditions. These turbine driven pumps and motor driven pumps are normally "standby equipment" in that they are not running during power operation but are "function upon demand". These pumps are generally found to be quite important pieces of equipment in the nuclear unit's Probabilistic Risk Assessment. In some cases (loss of all ac power), the turbine driven pumps may be the only power source to provide water to cool the reactor core.

These turbine driven pumps and motor driven pumps are configured in essentially single train configurations with very little redundancy except for an alternate source of cooling water. All the pumps are provided with recirculation piping that enables the pumps to be checked during surveillance testing without injection into the steam generators or reactor vessel. Only in the case of an emergency signal are the injection valves opened and cooling water supplied to the steam generators or reactor vessel.

Mathematical modeling has been done to

determine the "Conditional Probability" of the turbine driven pump or motor driven pump responding to a emergency demand. Most of the performance information available about these standby pumps comes from test (surveillance) data. However, there are occasions that Non-Test data is collected. In some cases the motor driven pumps were started to control water inventory in the steam generators during reactor startup and shutdown. These startup and shutdown runs were treated as surveillance runs and not as Non-Test actuations. The Non-Test actuations for the motor driven pumps were the times when the reactor was at full power and a low water level signal in the steam generators was received following a reactor trip.

2 HISTORICAL RECORD

The total period of time covered by this draft paper is from January 1, 1988 to November 30, 1995. Not all of the units were tracked over the entire period. The time the turbine driven pumps and the motor driven pumps were in either planned or forced maintenance was tracked and compared to the time the pumps might have been demanded in an emergency function. All applicable starts of the turbine driven pumps and motor driven pumps were tracked and the ratio of successful starts to total starts was calculated for various time periods. All run hours and run failures of the turbine driven pumps and motor driven pumps were also tracked and a "failure rate to run" was calculated by dividing the run failures by the run hours and updating this information using Bayesian techniques. Bayesian techniques are a mathematical way of incorporating data from a small data population into a larger data population. The Bayesian techniques are extremely useful when there are not many demands at any one particular unit. This failure rate to run was then used in a constant failure rate model to calculate the Probability of Run for a specified mission time.

3 RESULTS

As can be seen in Table 1 and Table 2, most of the performance information comes from the surveillance testing.

In spite of the fact that most of the information (approximately 97%) comes from the surveillance testing, the Non-Test data is consistent with the overall data if one looks at the overall "Conditional Probability." The overall Conditional Probabilities from the mathematical models for the pumps are shown in Figure 1 through Figure 7.

The overall Conditional Probabilities range from 82% to 96% for the turbine driven pumps. There were 17 successful demands out of 19 attempted demands for the standby turbine driven pumps during the Non-Test actuations (17/19 = 89%). The Conditional Probabilities are over 99% for the motor driven pumps. There were 9 successful demands out of 9 attempted demands for the standby motor driven pumps during the Non-Test actuations (9/9 = 100%).

There are a number of points to be noted:

3.1 The turbine driven pumps spend more time in maintenance and have more failures, especially failures to start.

3.2. During the 19 Non-Test actuations of the turbine driven pumps, the turbine driven pumps were "Standby Available" in all cases; that is, they were not in maintenance at the time of the demand. During the 9 Non-Test

actuations of the motor driven pumps, the motor driven pumps were "Standby Available" in all cases.

3.3. The turbine driven pumps are not used for long periods of time in the Non-Test actuations. The longest run was approximately 2.5 hours. Most Non-Test actuation runs are seconds or minutes. The motor driven pumps are used for longer times in the Non-Test actuations. The longest run was 30 hours and the average run was 12 hours.

3.4. It appears that the problems during the Non-Test actuations of the turbine driven pumps are the same as the problems during the surveillances. Most of the failures during the surveillance demands are overspeed trips of the turbine during the start. Both failures during the Non-Test actuations were overspeed trips during the start.

4 DISCUSSION

As can be seen in Table 1, Table 2, and in Figure 1 through Figure 7, the motor driven pumps have a much better performance record than the turbine driven pumps. The Conditional Probabilities of the motor driven pumps are over 99% and have been consistently in this range for the time period. The Conditional Probabilities of the turbine driven pumps are in the 82%-96% range.

The lower values for the Conditional Probabilities of the turbine driven pumps can be attributed to the start cycle of the turbines. The turbines were designed for continuous running and are generally used as continuous running pumps in other industries. The start time for these continuous running applications is generally in minutes. In the nuclear commercial electric generating industry, the use of the turbines in a standby mode with a short start time (30-60 seconds) is not conducive to successful starts. The motor driven pumps are better able to handle being in a standby mode and having a short time for starting.

It should be noted that there were approximately 34 surveillance tests for every Non-Test actuation of the turbine driven pumps. There were approximately 47 surveillance tests for every Non-Test actuation of the motor driven pumps. It is the author's belief that these values are very high and less frequent testing would be appropriate. There are some indications that frequent testing with the short start cycle is detrimental to the equipment. Acceptable performance can be achieved with a more cost effective testing scheme. Also, as the nuclear units have fewer reactor trips, the number of Non-Test actuations will decrease. For example, between May 1, 1995 and November 30, 1995, there were a total of 44 starts of the five turbine driven pumps and these starts were all surveillance test starts. There were no Non-Test actuations from May 1, 1995 to November 30, 1995. If we never have a Non-Test actuation, there is no need for frequent testing.

5 SUMMARY

In spite of the smaller amount of performance information from Non-Test actuations, the performance information gathered during Non-Test actuations appears to be consistent with the performance information gathered during surveillance testing for the standby turbine driven and motor driven pumps covered in this paper.

The Conditional Probabilities for the motor driven pumps (over 99%) are better than the Conditional Probabilities for the turbine driven pumps (82%-96% range).
The most beneficial change to enhance the Conditional Probabilities of the turbine driven pumps would be to change the way the turbines are started. The automatic start should take minutes, not 60 seconds or less as presently configured. Consideration should even be given to making the start of the turbine driven pumps a manual action over perhaps 10 minutes of time. The present start cycle results in the most probable failure mode being the overspeed trip of the turbine. This is one of the most damaging failure modes because in order to restore a turbine driven pump to service following an overspeed trip of the turbine, an operator must be sent to the turbine room to manually reset the overspeed trip.

There was a very high ratio of Surveillance test to Non-Test actuations for these standby pumps. As the performance of the nuclear units gets better, there will be fewer Non-Test actuations of the standby pumps. It appears that less frequent testing would be appropriate. It is not necessary nor is it cost effective to perform a lot of tests on standby equipment that is rarely demanded. The testing frequency should be based mainly on the number of Non-Test actuations and the success probability during the Non-Test actuations. The testing should not be based on calendar time.

It is recognized that changing the testing frequency or extending the time to automatically start the turbines from seconds to minutes or to make the start of the turbine driven pumps a manual action would represent a major change in the philosophy of the design and operation of these standby pumps. The justification for the change comes from the operating experience at commercial nuclear electric generating units in the United States and from the Probabilistic Risk Assessments performed for these units. Changes in the Technical Specifications are necessary. Approval from the Nuclear Regulatory Commission must be granted.

6 REFERENCES

1. Paper - "Turbine Driven Pump Surveillances: What Do We Learn?", International Conference on Probabilistic Safety Assessment and Management, PSAM-III, Crete, Greece; June 1996; by Bob Christie

2. Paper - "Turbine Driven Pump Conditional Probability Comparison," American Nuclear Society International Topical Meeting, Seattle, Washington, USA; September 1995; by Bob Christie, Tim Morse, Dan Nilius, and Bill Stuart.

3. Paper - "Improving the Conditional Probability of the Emergency Feedwater System at Arkansas Nuclear One," PSAM-II, San Diego, California, USA; March 1994; by Bob Christie, Dwight Johnson, Jay Miller, and Dan Nilius.

TABLE 1

The total values for starts and runs for the period are:

	Turbine 1	Turbine 2	Turbine 3	Turbine 4	Turbine 5	Motor 1	Motor 2
Total Starts	119	410	62	42	33	89	344
Successful Starts	117	400	59	41	31	89	344
Failures to Start	2	10	3	1	2	0	0
Run Hours	111	138	45	30	25	93	1090
Run Failures	1	1	0	1	0	0	0

The surveillance test values for starts and runs for the period are:

	Turbine 1	Turbine 2	Turbine 3	Turbine 4	Turbine 5	Motor 1	Motor 2
Surveillance Starts	116	405	60	36	30	86	338
Surveillance Successful Starts	114	396	58	35	28	86	338
Surveillance Failures to Start	2	9	2	1	2	0	0
Surveillance Run Hours	110	136	44	27	24	92	980
Surveillance Run Failures	1	1	0	1	0	0	0

The values for Non-Test actuations are:

	Turbine 1	Turbine 2	Turbine 3	Turbine 4	Turbine 5	Motor 1	Motor 2
Non-Test Starts	3	5	2	6	3	3	6
Non-Test Successful Starts	3	4	1	6	3	3	6
Non-Test Failures to Start	0	1	1	0	0	0	0
Non-Test Run Hours	less than 1 hour	less than 2 hours	less than 1 hour	less than 3 hours	less than 1 hour	1 hr	110 hrs
Non-Test Run Failures	0	0	0	0	0	0	0

TABLE 2

Turbine Driven Pumps	Total	Surveillances	Non-Test
Starts	666	647	19
Successful Starts	648	631	17
Failures to Start	18	16	2
% Successful Starts	97.3	97.5	89.5
Run Hours	349	341	8
Run Failures	3	3	0

Motor Driven Pumps	Total	Surveillances	Non-Test
Starts	433	424	9
Successful Starts	433	424	9
Failures to Start	0	0	0
% Successful Starts	100	100	100
Run Hours	1183	1072	111
Run Failures	0	0	0

Absence of Line Marker Indicates an Outage Month

SYSTEM RELIABILITY PROGRAM - Figure 1

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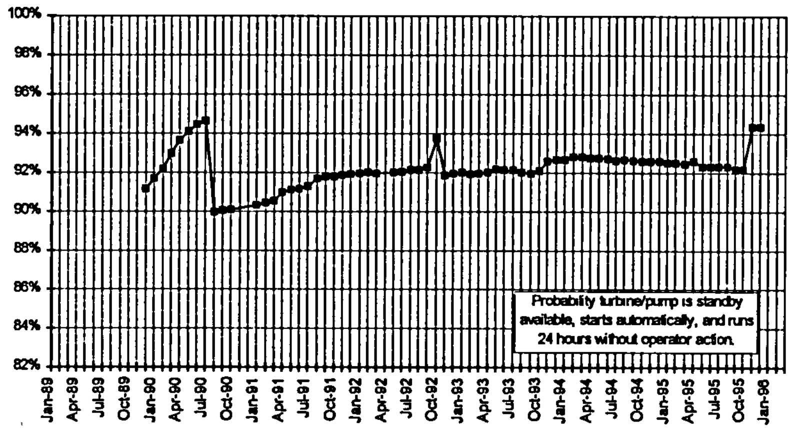
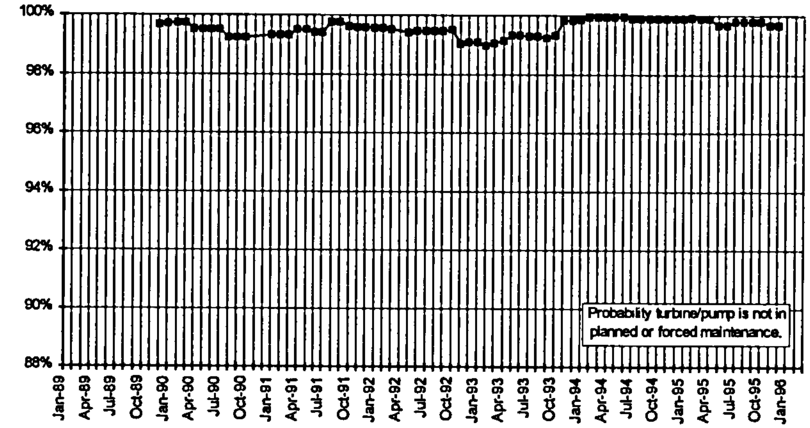
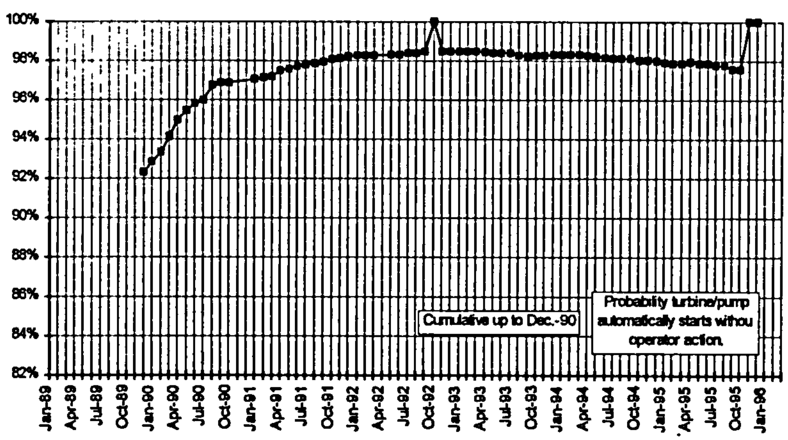
TURBINE #1
CONDITIONAL PROBABILITY
MODES 1, 2, 3



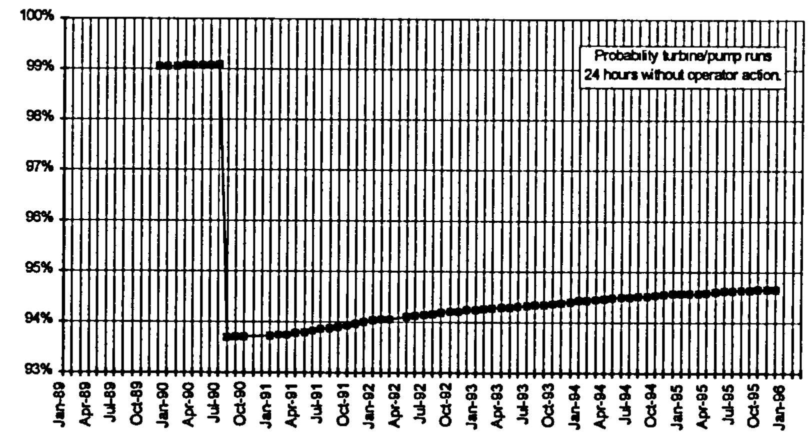
TURBINE #1
STANDBY AVAILABILITY
12 MONTH SLIDING - MODES 1, 2, 3



TURBINE #1
START PROBABILITY
36 MONTH SLIDING - MODES 1, 2, 3



TURBINE #1
RUN PROBABILITY
CUMULATIVE - MODES 1, 2, 3



Absence of Line Marker Indicates an Outage Month

SYSTEM RELIABILITY PROGRAM - Figure 2

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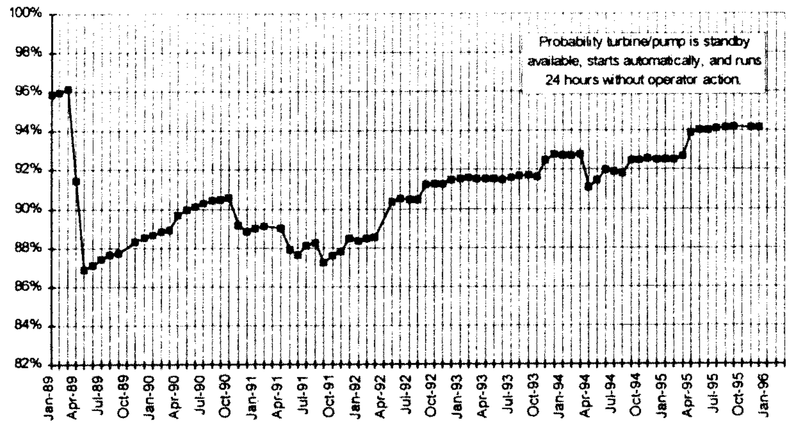
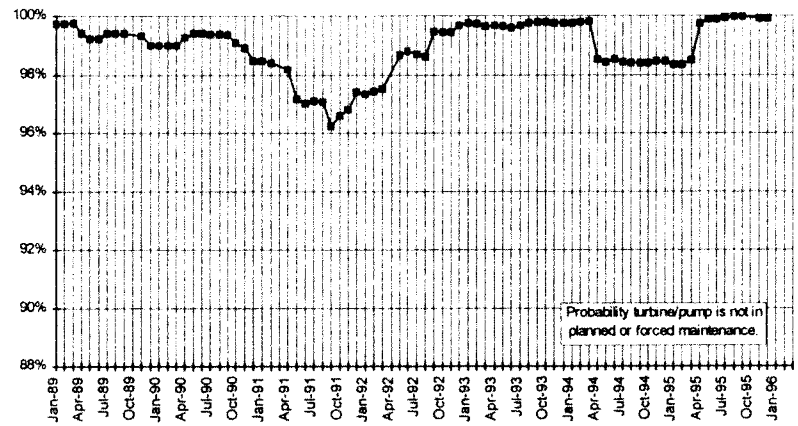
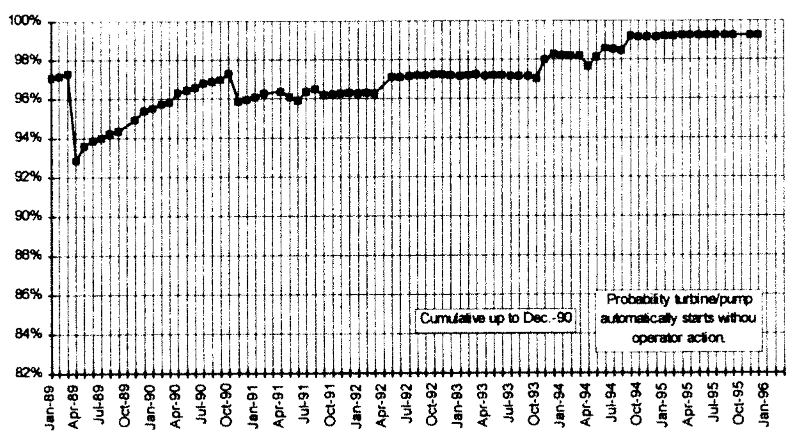
TURBINE #2
CONDITIONAL PROBABILITY
MODES 1, 2, 3



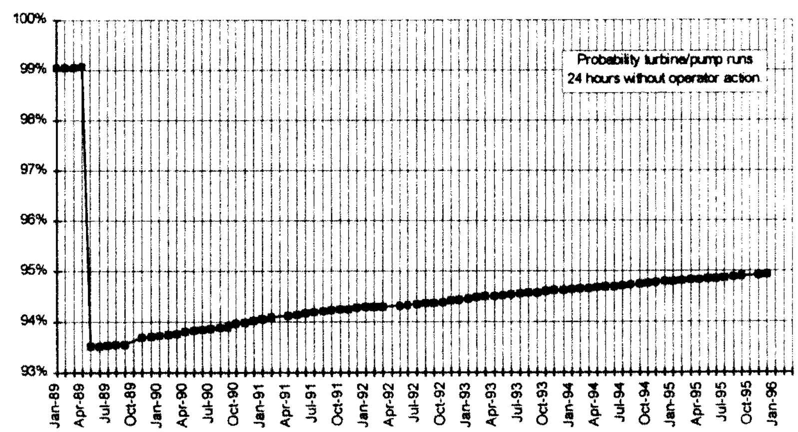
TURBINE #2
STANDBY AVAILABILITY
12 MONTH SLIDING - MODES 1, 2, 3



TURBINE #2
START PROBABILITY
36 MONTH SLIDING - MODES 1, 2, 3



TURBINE #2
RUN PROBABILITY
CUMULATIVE - MODES 1, 2, 3

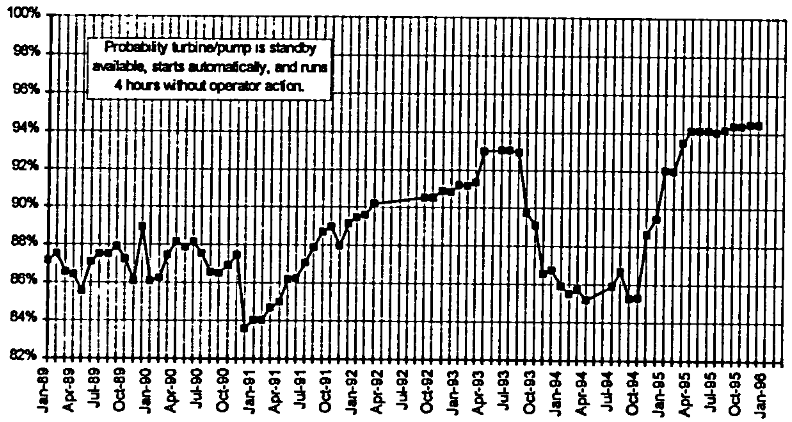


Absence of Line Marker Indicates an Outage Month

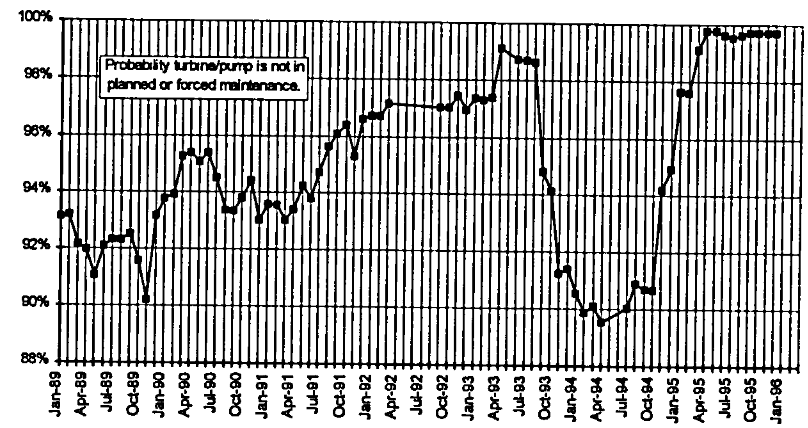
SYSTEM RELIABILITY PROGRAM - Figure 3

Performance Technology
3/26/96

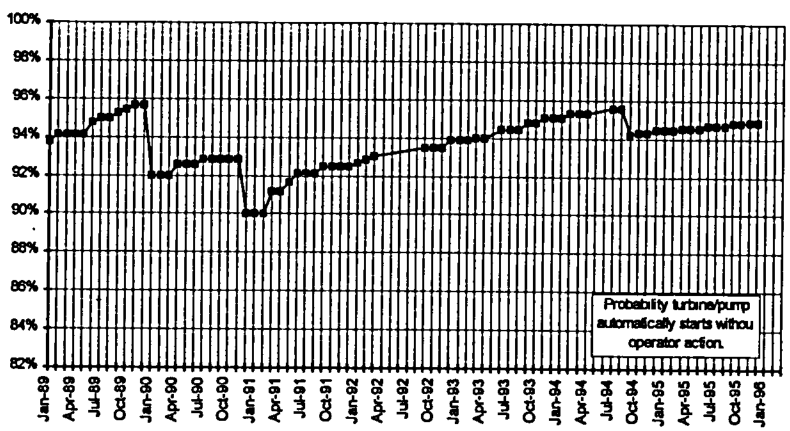
TURBINE #3
CONDITIONAL PROBABILITY
MODES 1, 2



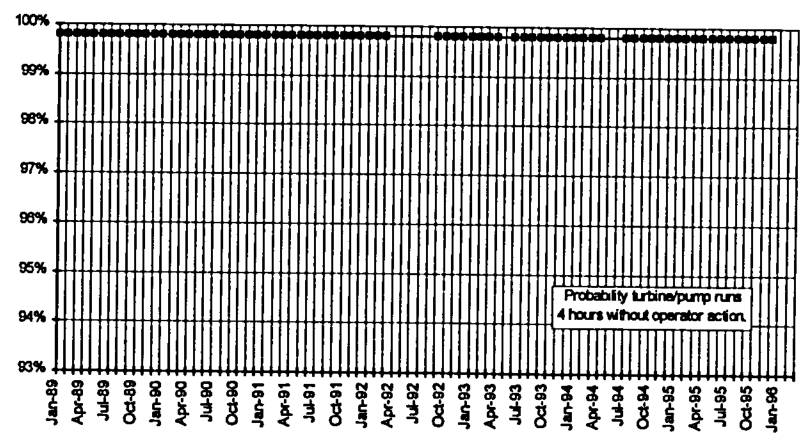
TURBINE #3
STANDBY AVAILABILITY
12 REACTOR MONTHS SLIDING - MODES 1, 2



TURBINE #3
START PROBABILITY
CUMULATIVE - MODES 1, 2



TURBINE #3
RUN PROBABILITY
MODES 1, 2

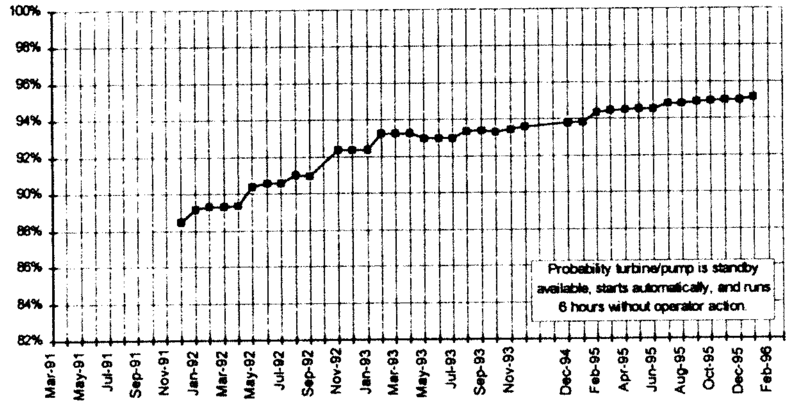


Absence of Line Marker Indicates an Outage Month

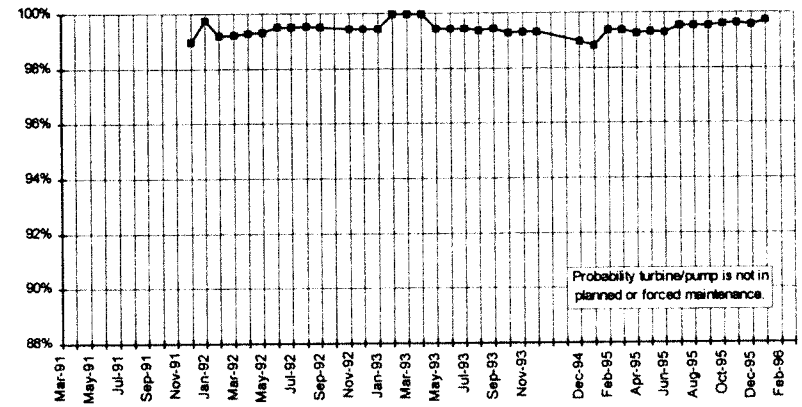
SYSTEM RELIABILITY PROGRAM - Figure 4

Performance Technology 3/26/96

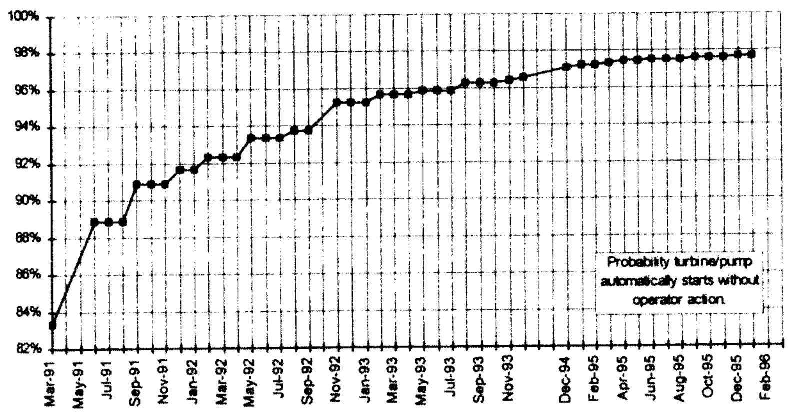
TURBINE #4
CONDITIONAL PROBABILITY
MODES 1, 2, 3



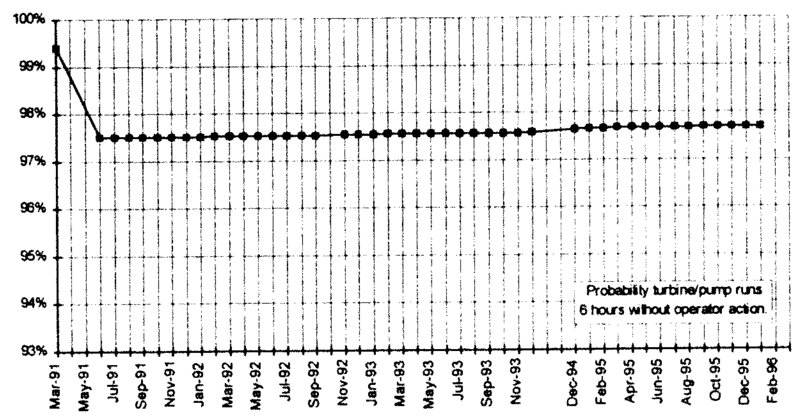
TURBINE #4
STANDBY AVAILABILITY
12 MONTHS SLIDING - MODES 1, 2, 3



TURBINE #4
START PROBABILITY
CUMULATIVE - MODES 1, 2, 3



TURBINE #4
RUN PROBABILITY
CUMULATIVE - MODES 1, 2, 3

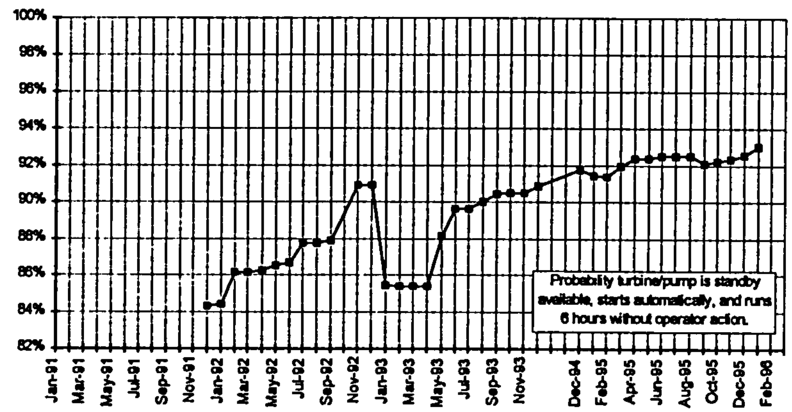


Absence of Line Marker Indicates an Outage Month

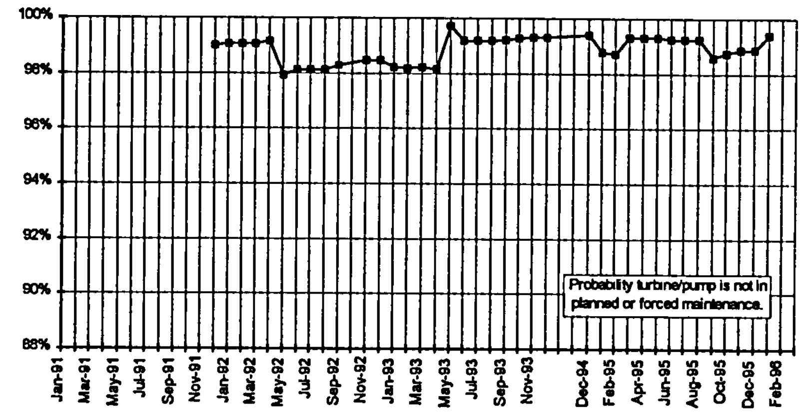
SYSTEM RELIABILITY PROGRAM - Figure 5

Performance Technology 3/26/96

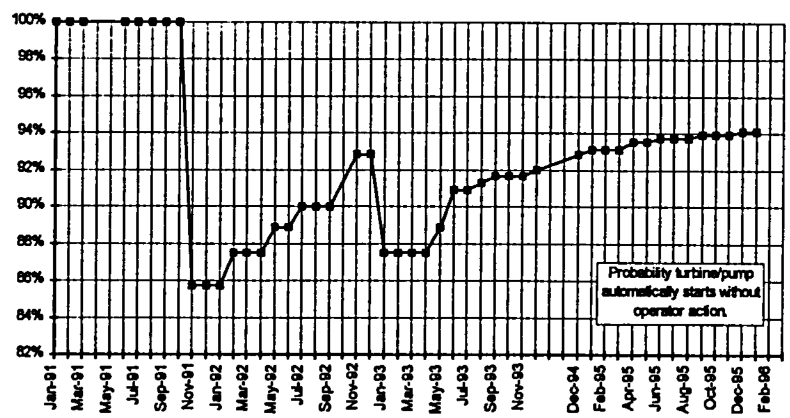
TURBINE #5
CONDITIONAL PROBABILITY
MODES 1, 2, 3



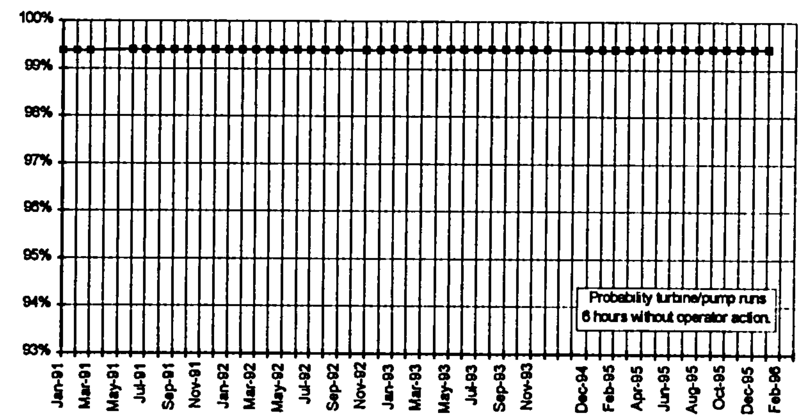
TURBINE #5
STANDBY AVAILABILITY
12 MONTHS SLIDING - MODES 1, 2, 3



TURBINE #5
START PROBABILITY
CUMULATIVE - MODES 1, 2, 3



TURBINE #5
RUN PROBABILITY
MODES 1, 2, 3

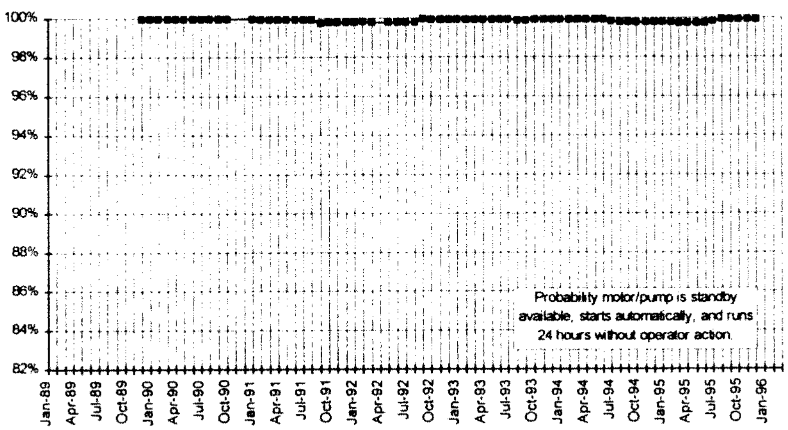


Absence of Line
Marker Indicates an
Outage Month

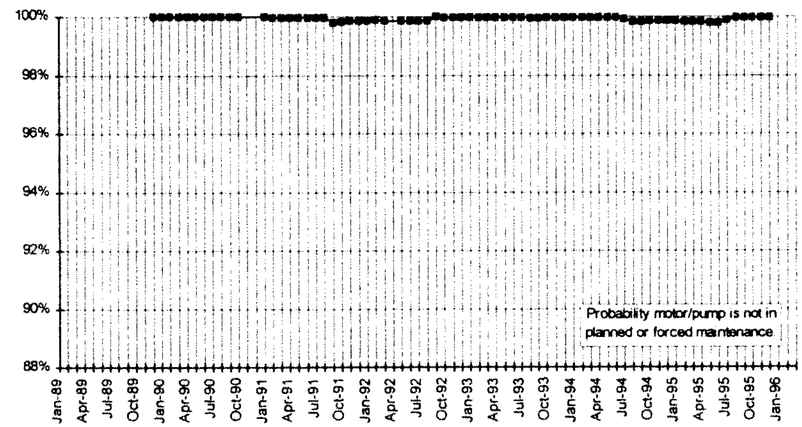
SYSTEM RELIABILITY PROGRAM - Figure 6

Performance
Technology
3/26/96

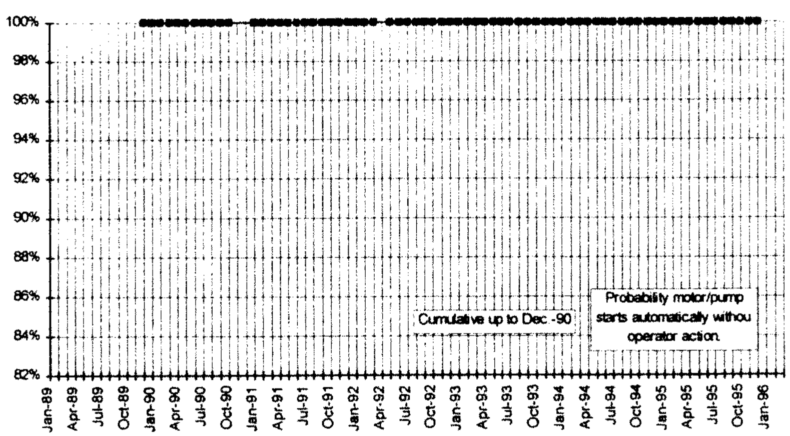
MOTOR #1
CONDITIONAL PROBABILITY
MODES 1, 2, 3



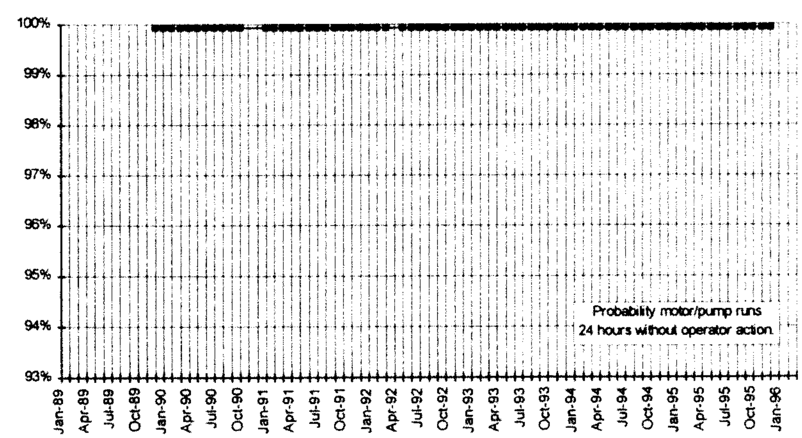
MOTOR #1
STANDBY AVAILABILITY
12 MONTH SLIDING - MODES 1, 2, 3



MOTOR #1
START PROBABILITY
36 MONTH SLIDING - MODES 1, 2, 3



MOTOR #1
RUN PROBABILITY
CUMULATIVE - MODES 1, 2, 3

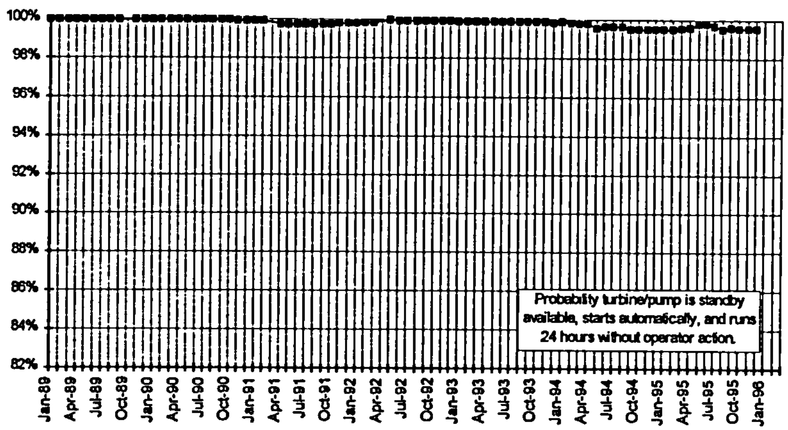


Absence of Line Marker Indicates an Outage Month

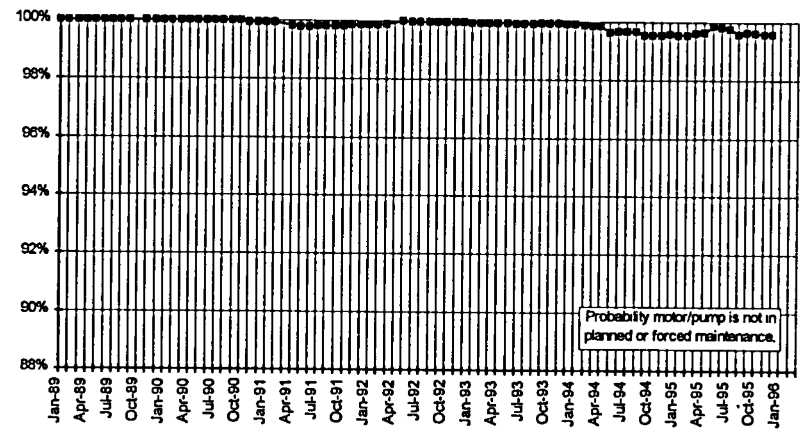
SYSTEM RELIABILITY PROGRAM - Figure 7

Performance Technology 3/26/96

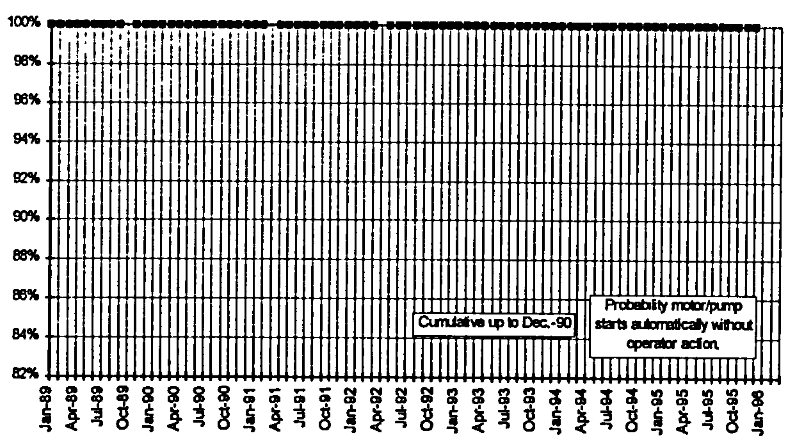
MOTOR #2
CONDITIONAL PROBABILITY
MODES 1, 2, 3



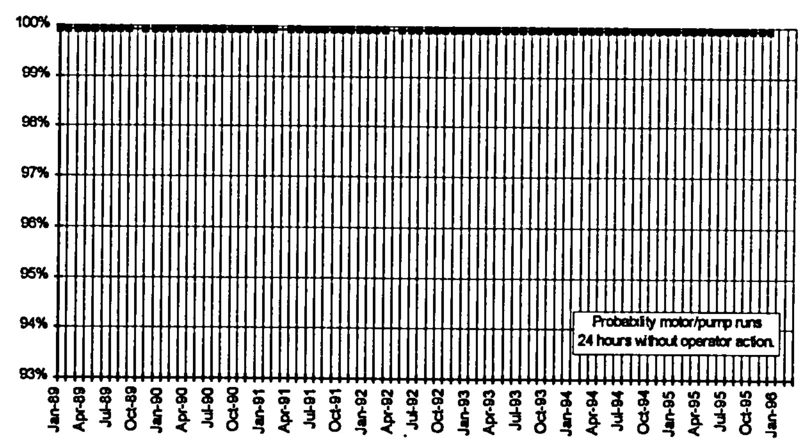
MOTOR #2
STANDBY AVAILABILITY
12 MONTH SLIDING - MODES 1, 2, 3



MOTOR #2
START PROBABILITY
36 MONTH SLIDING - MODES 1, 2, 3



MOTOR #2
RUN PROBABILITY
CUMULATIVE - MODES 1, 2, 3



Session 2B

General Valve Issues 2

Session Chair
Christopher Hansen
Senior Engineer
Yankee Atomic Electric Co.

QUENCH TANK IN-LEAKAGE DIAGNOSIS AT ST. LUCIE

J.E. Price
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Framatome Technologies

ABSTRACT

In February 1995, leakage into the quench tank of the St. Lucie Nuclear Station Unit 1 was becoming an operational concern. This internal leak resulted in measurable increases in both the temperature and level of the quench tank water, and was so severe that, if the trend continued, plant shut down would be necessary. Preliminary diagnosis based on in-plant instrumentation indicated that any one of 11 valves might be leaking into the quench tank. This paper describes the joint effort by two teams of engineers--one from Florida Power & Light, the other from Framatome Technologies--to identify the sources of the leak, using the latest technology developed for valve diagnosis.

1. INTRODUCTION

In February 1995, leakage into the quench tank of the St. Lucie Nuclear Station Unit 1 was becoming an operational concern. This internal leak resulted in measurable increases in both the temperature and level of the quench tank water, and was so severe that, if the trend continued, plant shut down would be necessary. Preliminary diagnosis based on in-plant instrumentation indicated that any one of the following 11 valves might be leaking into the quench tank:

TAG #	VALVE DESCRIPTION
SV1200	Code Safety Relief Valve
SV1201	Code Safety Relief Valve
SV1202	Code Safety Relief Valve
PV1402	Pressurizer Power Operated Relief Valve (PORV, Pilot valve)
RV1402	Pressurizer PORV (Main valve)
PV1404	Pressurizer PORV (Pilot valve)
RV1404	Pressurizer PORV (Main valve)
RV3469	Shutdown Cooling Relief Valve
RV3482	Shutdown Cooling Relief Valve
RV2199	Reactor Coolant Pump Bleed Off Valve
SOV1445	Reactor Head Gas Vent Isolation Valve

Figure 1 is a schematic of the quench tank and associated valves. The three code safety valves and the two PORVs with their pilot valves were all located on top of the pressurizer inside the pressurizer cubicle.

On February 9, 1995, engineers from Framatome Technologies' Valve Services were deployed to the St. Lucie Nuclear

Station to test the above valves in an attempt to pin-point the leak source, using the Framatome Technologies UltraCheck non-intrusive check valve diagnostic system. Although designed primarily for non-intrusively diagnosing check valves, this system contains three sensitive acoustic sensors for detecting noises, including internally generated noises inside valves caused by fluid leakage. For this and the subsequent test series, the only objective was to detect seat leakages in the aforementioned valves. Therefore, only the three acoustic sensors were used. These sensors were mounted on the high-pressure (upstream) side, the low-pressure (downstream) side, and near the seat of each valve, and the valves was tested one at a time. By comparing the amplitudes (in the time domain) and the signatures (in the frequency domain) of noises recorded by these three acoustic sensors, Framatome Technologies engineers were able to determine whether there was any through-seat leakage during the test.

UltraCheck is a self-contained, personal computer (PC)-based system with all the signal conditioning electronics built into the PC. During the entire test, the equipment was set at the four-channel mode, with a combined sampling rate of 96,000 per sec (24,000 per channel per sec.) The data was acquired directly into the RAM and then onto the hard disk of the computer. The data was retrieved later for further analysis, using the same computer. It should be mentioned that the acoustic sensors used were extremely sensitive and were able to detect tiny, high-velocity steam leaks that might not have contributed much to the overall quench tank in-leakage. In this case, the valve was classified as not leaking.

2. PRE-SHUTDOWN TEST AT FULL POWER

The first test series was performed on February 11 and 12, while the reactor was at steady state, 100% power. The two shut down cooling relief valves, RV3469 and RV3482 and the reactor head gas vent isolation valve SOV1445 were not accessible due to the high neutron field. The results showed that of the remaining eight valves tested, the two code safety relief valves, SV1200 and SV1201, were distinctly leaking. The third code safety relief valve, SV1202, also had indication of through-seat leakage. However, it was believed that the leak rate was much smaller and did not contribute significantly to the measured quench tank water level or temperature increases. The same was true for the pressurizer PORV PV1404. No leaks were detected in the other pressurizer PORV PV1402 or the two PORV pilot valves. Repeated "tapping" sounds were noticed in the RCP bleed off valve RV2199 but no seat leakage was noticed. The following paragraphs discussed the reasoning leading to the diagnostic conclusions in more detail.

SV1200, SV1201, SV1202 Code Safety Relief Valves

The acoustic traces from each of the three sensors mounted upstream (high pressure side), near the seat and the downstream (low pressure side) on these three valves are shown in Figures 2 to 4. Figures 2(a) and 3(a) show that in both SV1200 and SV1201, the root mean square (rms) amplitudes of the noise levels at the seats and on the downstream sides of the valves were significantly higher than those on the upstream sides of the valves. This is especially evident in SV1201 and indicated that through-seat leakage existed in both valves, and was worse in SV1201. Figures 2(b) and 3(b) are the corresponding

power spectral density (PSD) plots for the above time traces. The existence of significant high-frequency activities on the downstream sides and at the seats of the valves again indicated that through-seat leakage existed in these two valves.

Figure 4(a) shows the time traces from SV1202. Again, the noise level at the seat was significantly higher than on the upstream side of the valve. However, the noise level on the downstream side was about the same as that on the upstream side. Figure 4(b) is the corresponding power spectral density (PSD) plot for SV1202. The absence of significant high- frequency activity on the low pressure side, together with the small rms noise level, indicated that, although there was some through-seat leakage in this valve, it was much smaller than that in SV1200 and SV1201 and probably did not contribute significantly to the leakage into the quench tank.

RV2199 Reactor Head Gas Vent Valve
All three sensors on this valve picked up distinct tapping-like noises. Figure 5(a) shows the time domain signature of the noise picked up by the sensor mounted close to the seat. The other two signatures looked very similar. In Figure 5(b), one of these tapping wave forms is zoomed-in to reveal its detail. Its smeared front showed that this noise was not likely to have come from any component tapping inside the valve. It was suspected that this noise originated either from water burping through the seat, or from water flashing inside the quench tank located only a few feet below. Unfortunately, the three sensors were mounted too close to one and other to enable one to determine the origin of the noise by time-of-arrival technique. This valve would be re-tested with the low-pressure side acoustic sensor mounted further away from the valve. Meanwhile, although the tapping noise masked any intelligent conclusion on seat leakage from the time traces, PSD plots and in particular, waterfall plots, one of which is shown in Figure 5(c), indicated no high-frequency activities. This test showed that this valve did not have any through-seat leakage that could have contributed significantly to the quench tank total in-leakage.

3. POST SHUTDOWN TEST AT OPERATING TEMPERATURE AND PRESSURE

The second test series was performed on February 27, 1995. The plant was shut down, but holding normal operating pressure and temperature. The three valves that the Quench-Tank In-Leakage Team could not get to earlier (RV3482, SOV1445 and RV3469) because of the high neutron field were tested. RV2199 was replaced and re-tested with the downstream acoustic sensor mounted two feet from the valve and only three or four feet from the quench tank. The objective was to locate the origin of the tapping noise observed in the earlier test. After reviewing the acoustic data from S/D cooling relief valve RV3482, we decided to test MOV3480, which was the S/D cooling isolation valve upstream of RV3482. In summary, the following valves were tested in the second test series:

Tag No.	Valve Function	Leaking?
RV3482	Shutdown Cooling Relief Valve	No
MOV3480	S/D Cooling Relief Isolation	Yes
RV3469	Shutdown Cooling Relief Valve	Yes
SOV1445	Reactor Head Gas Vent Isolation	No
RV2199	RCP Seat Bleed off Line Relief	No

The acoustic signatures from valves RV3482, SOV1445 showed approximately equal noise amplitudes for all three accelerometers in the time domain, and no high-frequency activities in the PSD plots. As explained in Section 1 above, these signatures indicated a non-leaking valve (in the sense that their leak rates, if any, were negligible compared with the measurable leak rate into the quench tank). The signatures from MOV3480, which was the isolation valve upstream of the shutdown cooling relief valve RV3482, were studied in detail. Following the same reasoning as explained in Section 2, this valve was leaking slightly. However, since RV3482 was not leaking, this valve could not have contributed to the quench tank in-leakage. The signatures from the remaining two valves are discussed below.

RV3469 Shutdown Cooling Relief Valve
As shown in Figure 6(a), there seemed to be an intermittent flow through this valve consistent with the valve plug being lifted off the seat momentarily and then re-seating. The noise level was highest at the seat and apparently originated at the seat. This assessment was confirmed by zooming in around t=5.0 seconds and using the cursor on the computer monitor to determine the times of leakage flow initiation at the three sensor locations.

Figure 6(b) is the PSD corresponding to the quiet portion of the time history (2.5 to 6.0 seconds) in Figure 6(a). Apart from the broad peak around 4,500 Hz, which was observed in the upstream sensor in several other valves tested and which was not related to seat leakage, there were no high-frequency activities in any of the sensor locations. This confirmed that from 2.5 to 6.0 seconds, the valve was properly seated and there was no through-seat leakage. Figure 6(c) is the PSD corresponding to the noisy part of the time history (12.5 to 16.5 second) in figure 6(a). High-frequency energy content can be seen at the sensors mounted both near the seat and downstream (quench tank side) of the valve, showing definite seat leakage.

RV2199 RCP Seal Bleed Off Relief Valve
The acoustic signature is shown in Figure 7. This looked very similar to that acquired during the first test series. However, the larger separation between the seat and downstream sensors enables one to conclude, based on the time-of-arrival technique, that the "tapping" noise originated from the downstream side of the valve and was thus not due to seat tapping. This can be barely observed in Figure 7(b), which is an extreme blow-up of one of the tapping spikes in Figure (7a). The significantly larger rms amplitude recorded by the downstream sensor, which was much closer to the quench tank, showed that the noise originated inside the quench tank and was probably due to water flashing inside the quench tank. In fact, the acoustic wave forms from RV2199 were very similar to those from SOV1455 (not shown). Both resembled far-field impact wave forms in the time domain. However, the corresponding PSD plots showed no high- frequency activities, showing that there were no leaks in these valves.

4. ACTIONS RESULTED FROM THE FIRST TWO TEST SERIES

The results from the first two test series indicated that the code safety relief valves, SV1200 and SV1201, leaked significantly and might have contributed the majority of the increase in measured water level and temperature in the quench tank. The other code safety relief valve SV1202 also showed indications of through-seat leakage, although by far not as much as the other two. In

addition, the shutdown cooling relief valve RV3469 appeared to have through-seat leakage due to the plug's lifting off the seat. The shutdown motor-operated isolation valve (MOV) upstream of RV3469 was not tested. However, since this MOV was closed during the test, this valve must also be leaking. The motor operated isolation valve (MOV3480) upstream of the shutdown cooling relief valve RV3482 was also diagnosed as leaking. However, since RV3482 was not leaking, this could not have contributed to the quench tank in-leakage.

As a result of these findings, all three code safety relief valves SV1200, SV1201 and SV1202 were replaced with re-built units. The shutdown cooling relief valves RV3469 and the RCP bleed off line relief valve R2199 were also replaced.

5. POST REPAIR TEST

On March 5-7, a third series of tests were performed when the plant was re-started. As plant pressure was increased the acoustic data was monitored real time on the computer monitor and then recorded. Leakage from all three code safety relief valves was detected. This was cross-checked and confirmed by in-plant monitoring equipment. Plant pressure was dropped to allow the safety relief valves to reseat. Plant pressure was then increased slowly to normal operating pressure. All 11 valves listed in the Introduction were tested again. No leakage was detected in any, except in the shutdown cooling relief valve RV3469. This valve was in a high vibration area and the vibration still caused the valve seat to "relax" occasionally, thus resulting in through-seat leakage. However, the leak rate was much smaller than before. This is discussed in more details in the following paragraphs.

Code Safety Relief Valves SV1200, SV1201 and SV1202

Figure 8(a) shows the time histories of the signals from the sensors on code safety relief valve SV1200 during the first "power up." The significantly higher rms amplitudes at the seat and the downstream sensor locations compared with those at the upstream side indicated that this valve was leaking. Figure 8(b) shows the corresponding PSD plots. The high-energy contents in the high-frequency range at the seat and the downstream side of the valve confirmed there was through-seat leakage. Figure 8(c) shows waterfall plots of the signals from the upstream and downstream sensors on SV1200.

Figure 9(a) shows the time histories from the sensors on the same code safety relief valve after the second power up and the valve was re-seated. The rms amplitudes at all three sensor locations were about the same, indicating no significant through-seat leakage. However, Figure 9(b), which shows the corresponding PSD plots of the time traces, indicated high-frequency activities at the seat, but at much lower energy levels than those in Figure 8(b). These results indicated that some seat leakage still existed in SV1200. However, the leakage was much less than what was observed before the valve was replaced with a rebuilt unit and would not cause any significant increases to the water level or temperature in the quench tank.

Acoustic signatures from the other two code safety relief valves, SV1201 and SV1202, generally showed the same trend.

RV2199 RCP Bleed Off Valve

The tapping noise disappeared completely. This confirmed the conclusion made after the second test series, that this "tapping" noise was actually flashing noise caused by the hot

water leaking into the quench tank. This noise naturally stopped when the leak was stopped.

RV3469 Shutdown Cooling Relief Valve
The acoustic signatures from sensors on this valve generally resembled those acquired before the valve was rebuilt, but at a much lower energy level. Intermittent through-seat leakage could still be observed, though not nearly as bad as before. This valve was in a high-vibration area. Far-field impact noises were recorded in some tests on this valve. The vibration source was not identified during this test. However, the vibration seemed to affect this valve, causing the valve seat to "relax" occasionally and resulting in low-level, intermittent through-seat leakage. It is not expected this would contribute measurably to the quench tank water level and temperature.

6. CLOSURE

The Unit was brought back to full power. The quench tank appeared to be quiet. Water level and temperature were holding steady. Thus, it appeared that the leak sources were properly identified and the leaking valves fixed. The three replaced code safety relief valves were subjected to a bench-top hydro test under pressure. All were found to be leaking, with SV1202 leaking less than the other two.

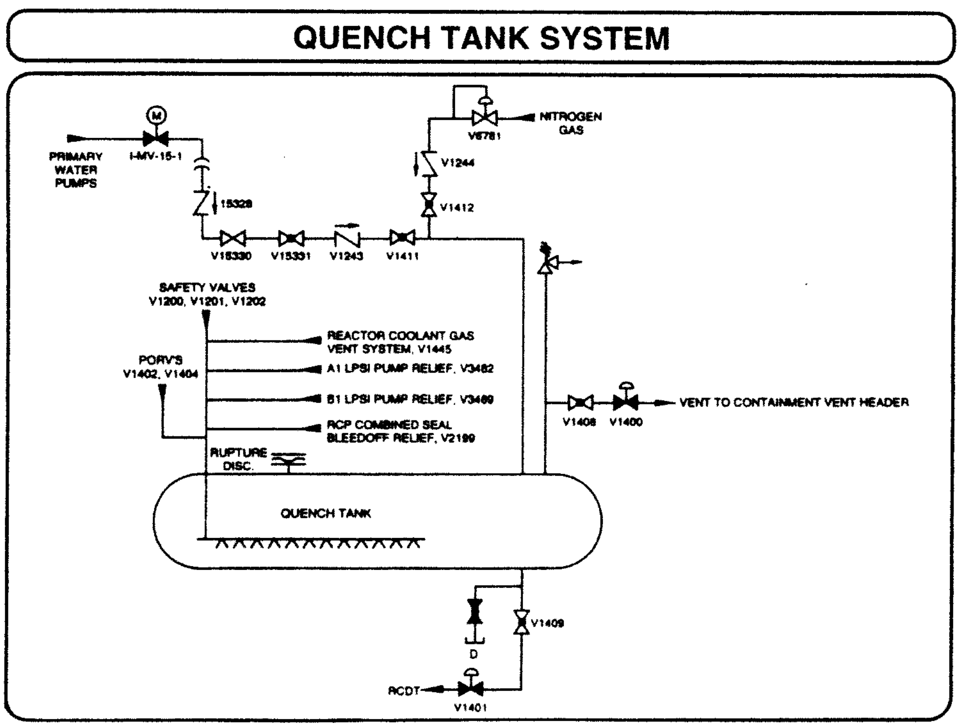


Figure 1(a): The St. Lucie Nuclear Station Quench Tank System

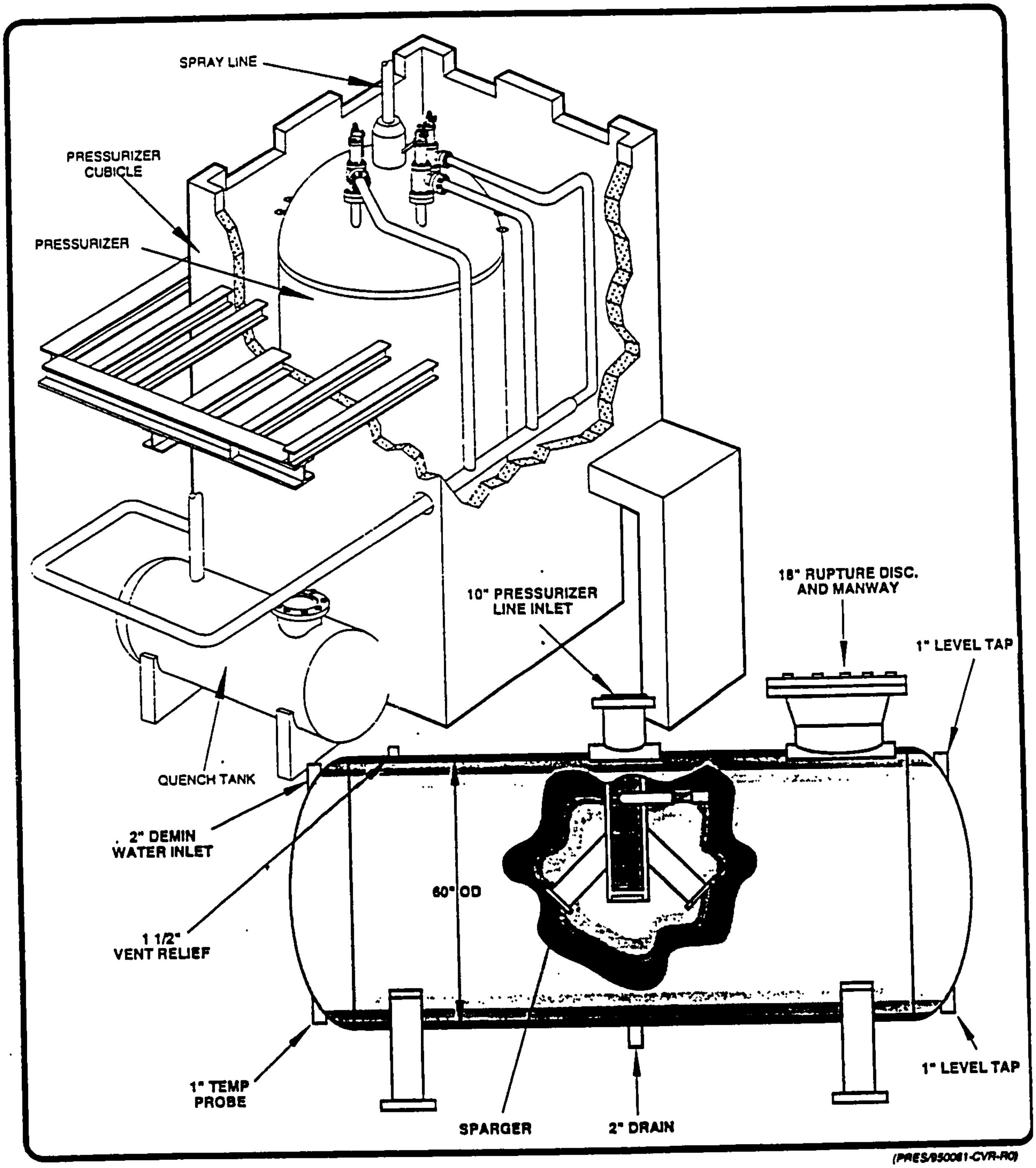
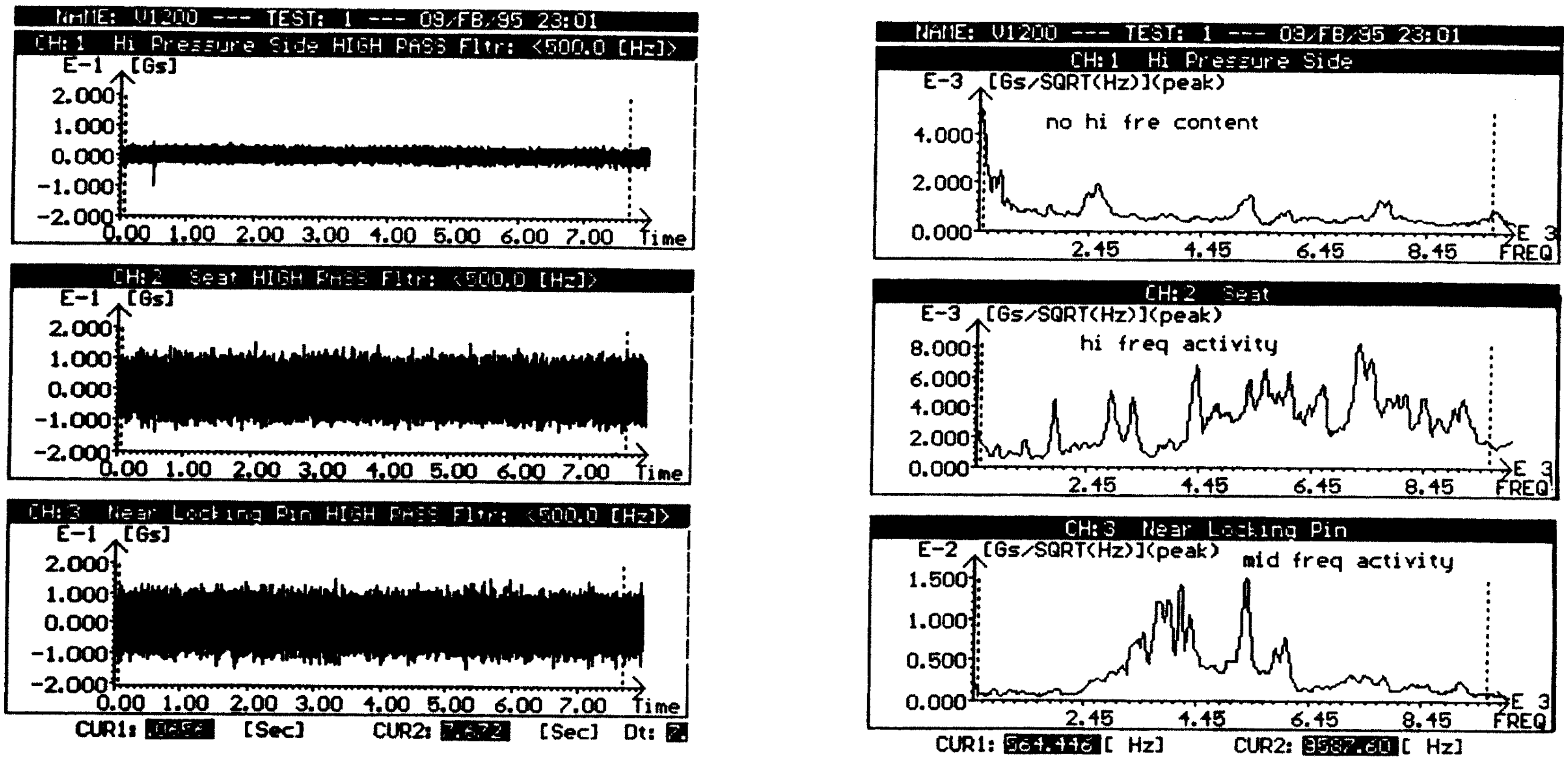
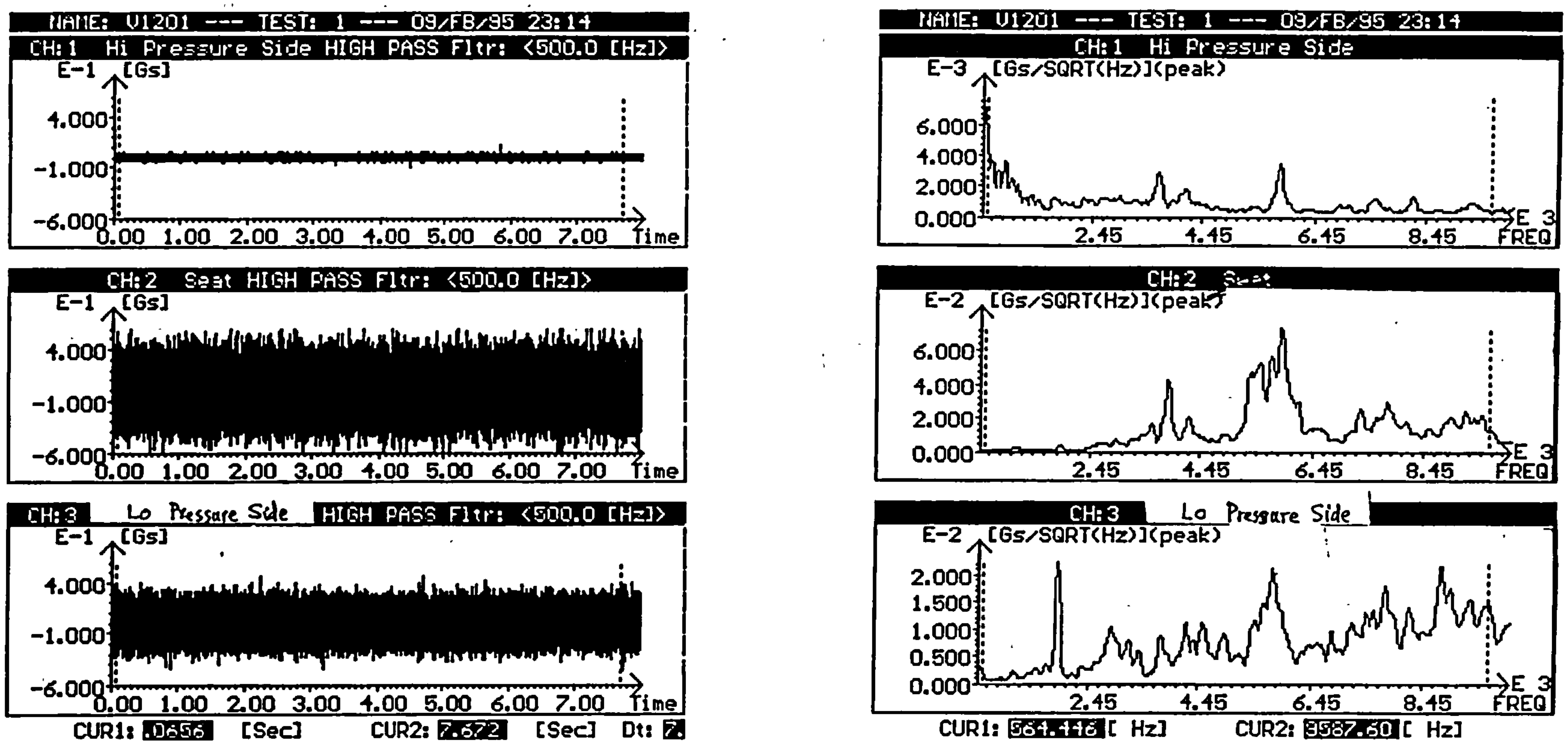


Figure 1(b): Schematic of the Quench Tank and the Pressurizer Cubicle



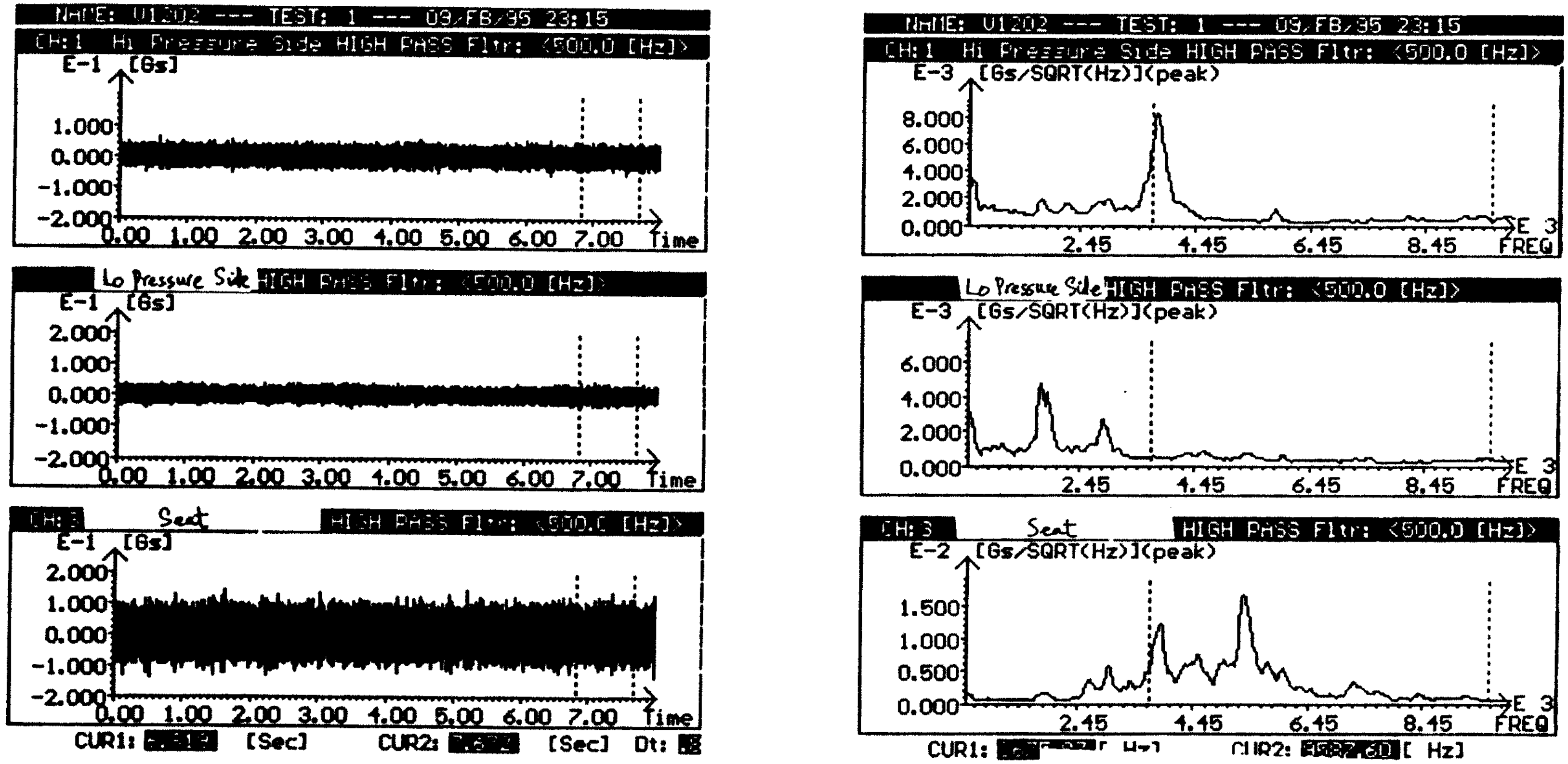
(a) (b)

Figure 2: (a) Time histories of acoustic signals from code safety relief valve SV1200 and (b), their corresponding power spectral density (PSD) plots.



(a) (b)

Figure 3: (a) Time histories of acoustic signals from code safety relief valve SV1201 and (b), their corresponding power spectral density (PSD) plots.



(a) (b)

Figure 4: (a) Time histories of acoustic signals from code safety relief valve SV1202 and (b), their corresponding power spectral density (PSD) plots.

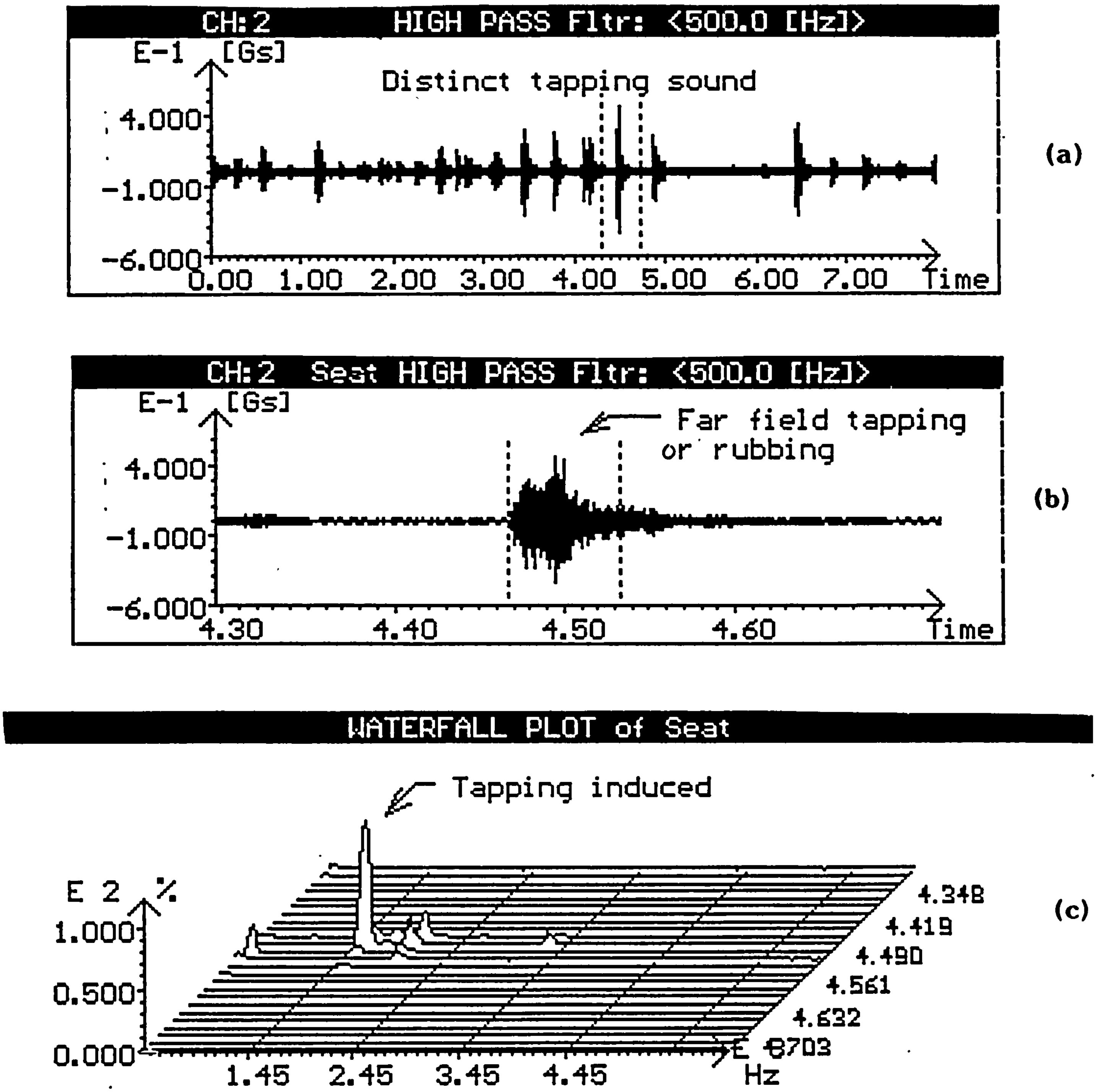
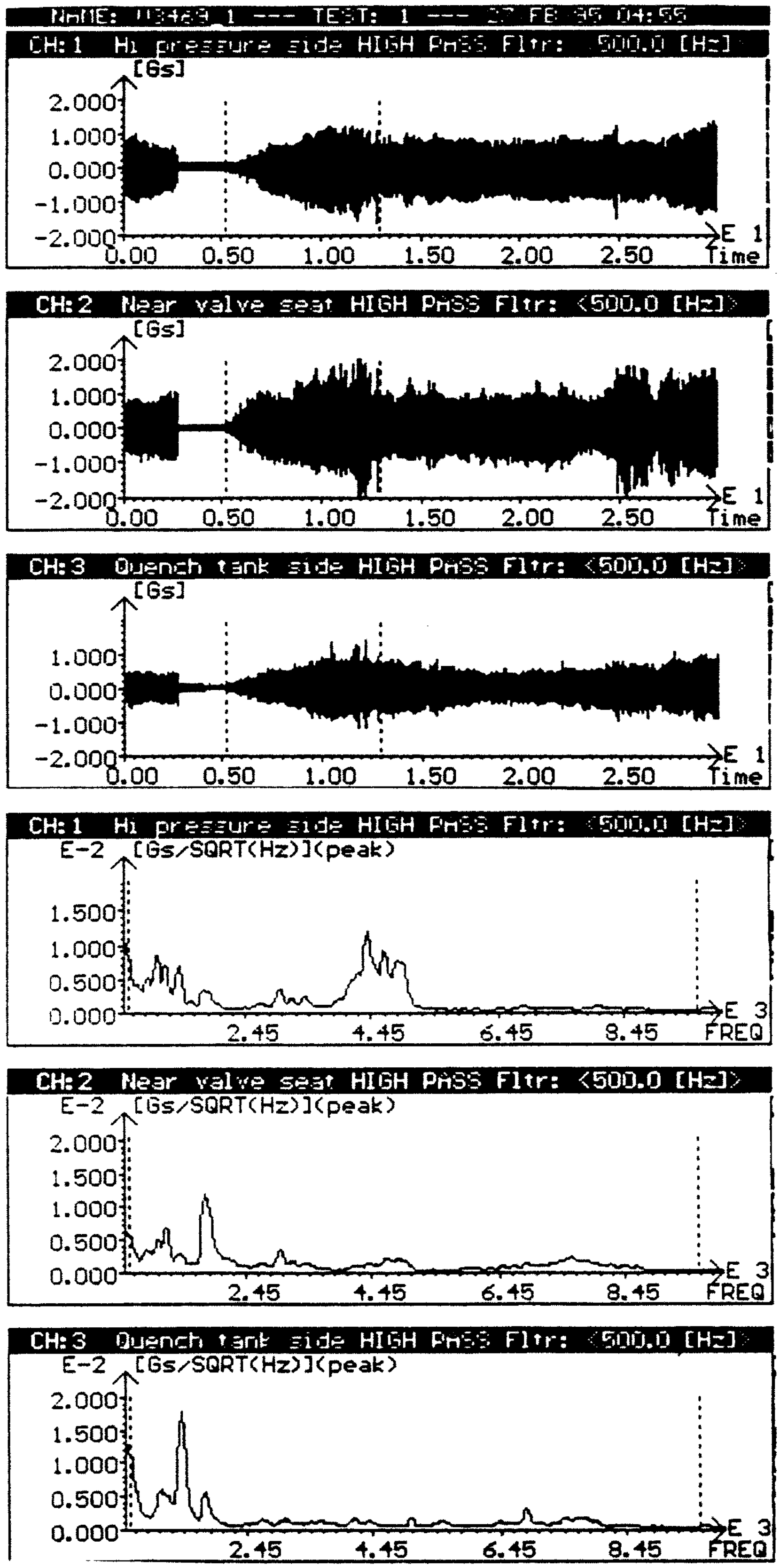


Figure 5: (a) Time history of acoustic signal acquired from the sensor mounted near the seat of the RCP bleed off valve RV2199 showing continuous "tapping" noise. (b) Zoom-in of one of the noise spike reveals slow rise wave front, an indication of far field and/or non-impact noise. (c) Corresponding waterfall plot showing one isolated spike.



(a)

(b)

Figure 6: (a)Time history of acoustic signals from sensors mounted on shutdown cooling relief valve RV3469 showing intermittent leak. (b)Power spectral density (PSD) plots corresponding to the no leak portion of the time history showing little high frequency energy content.

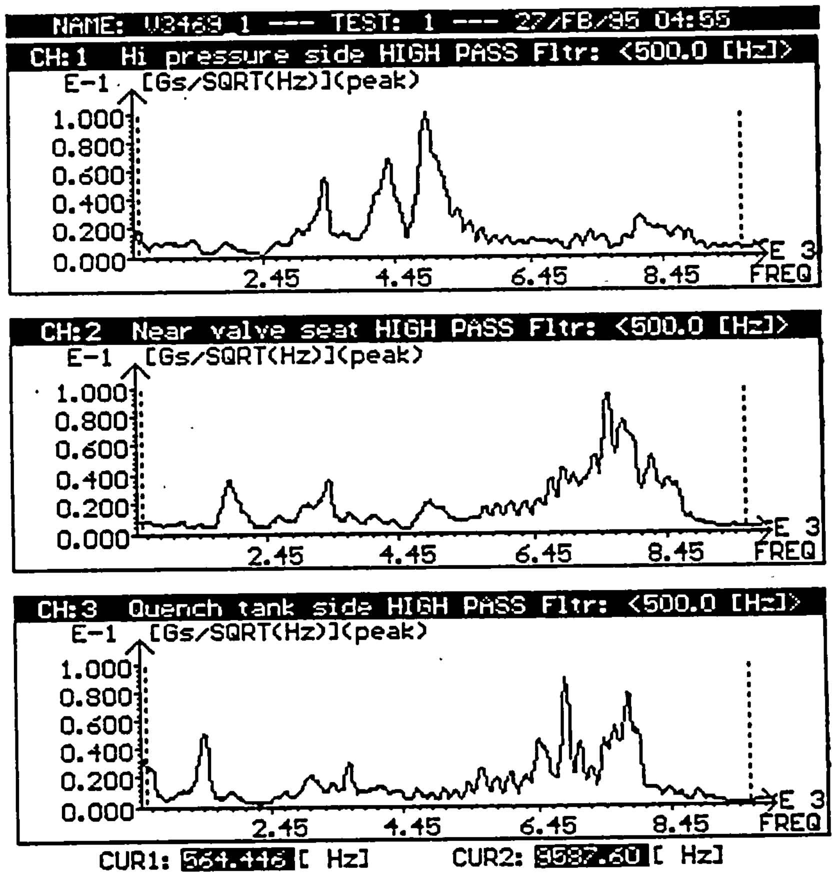
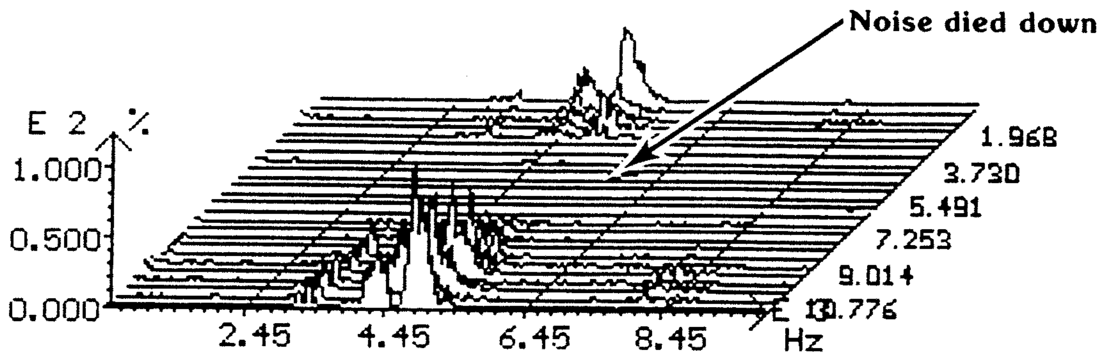
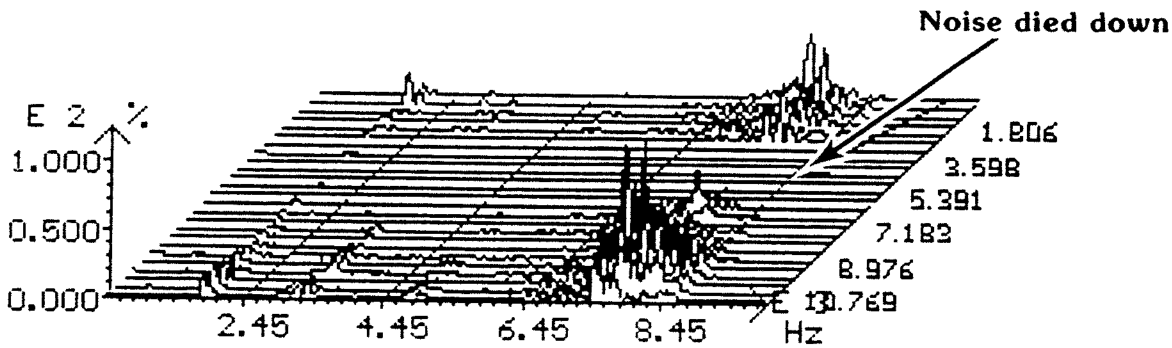


Figure 6(c): Power spectral density (PSD) plots corresponding to the high noise portion of the time history showing high frequency energy content imparted by the leakage flow.

High Pressure Side



Near Seat



Quench Tank Side

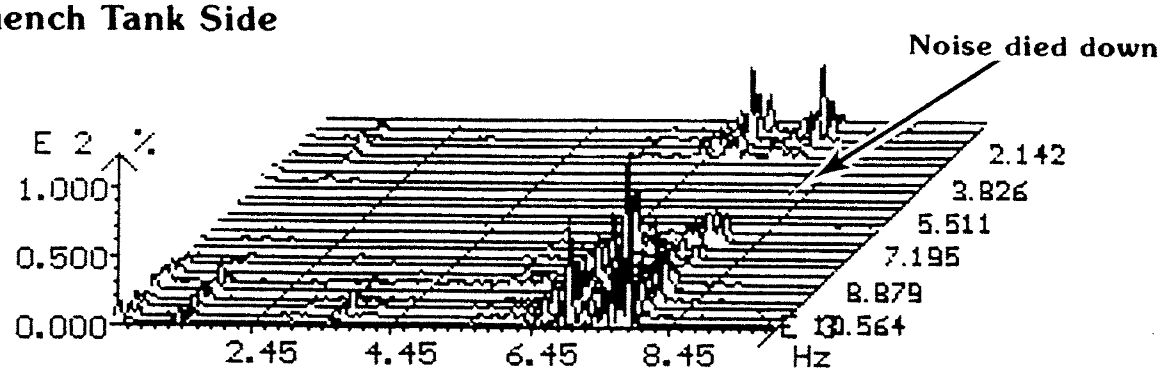
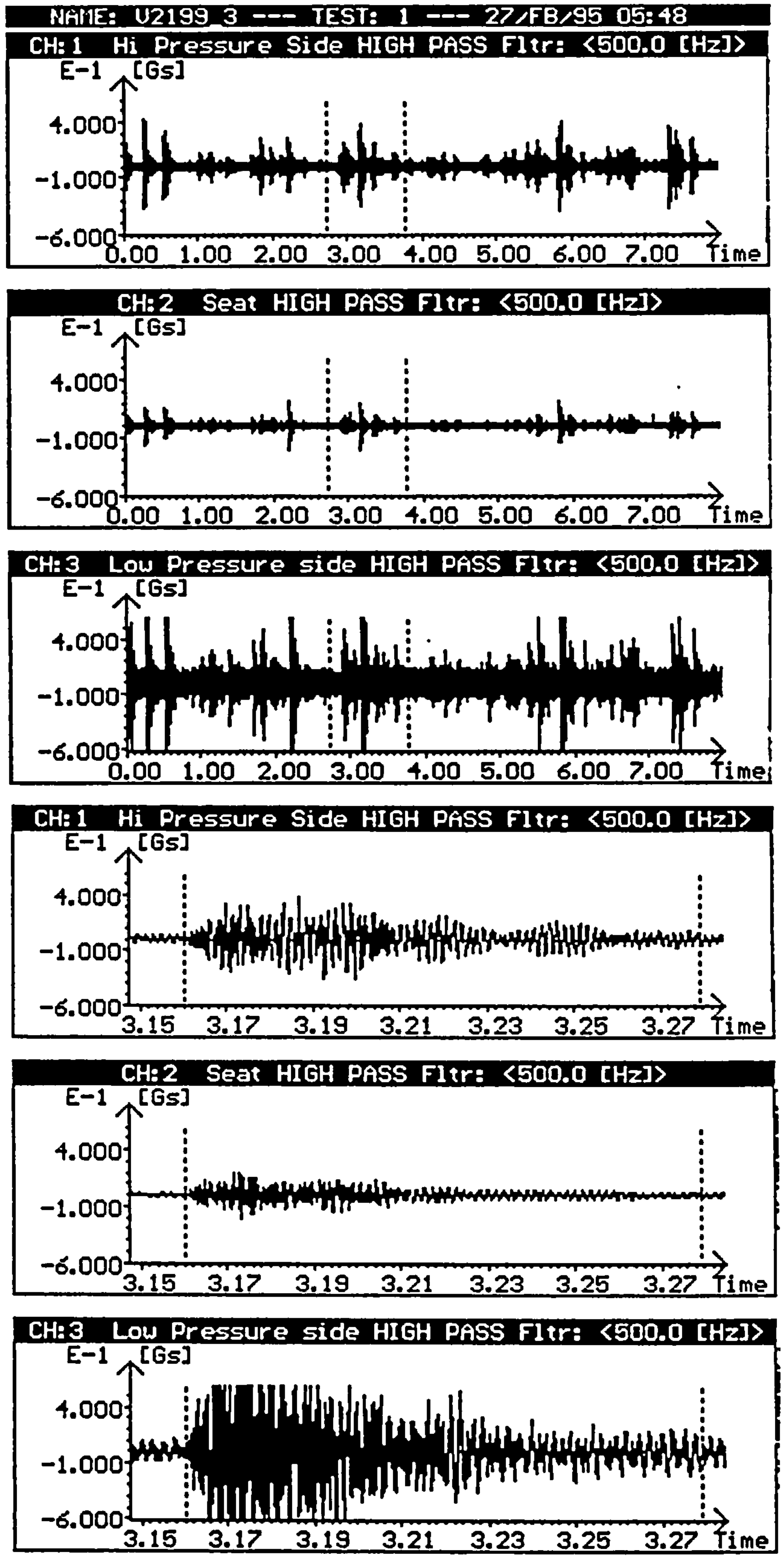


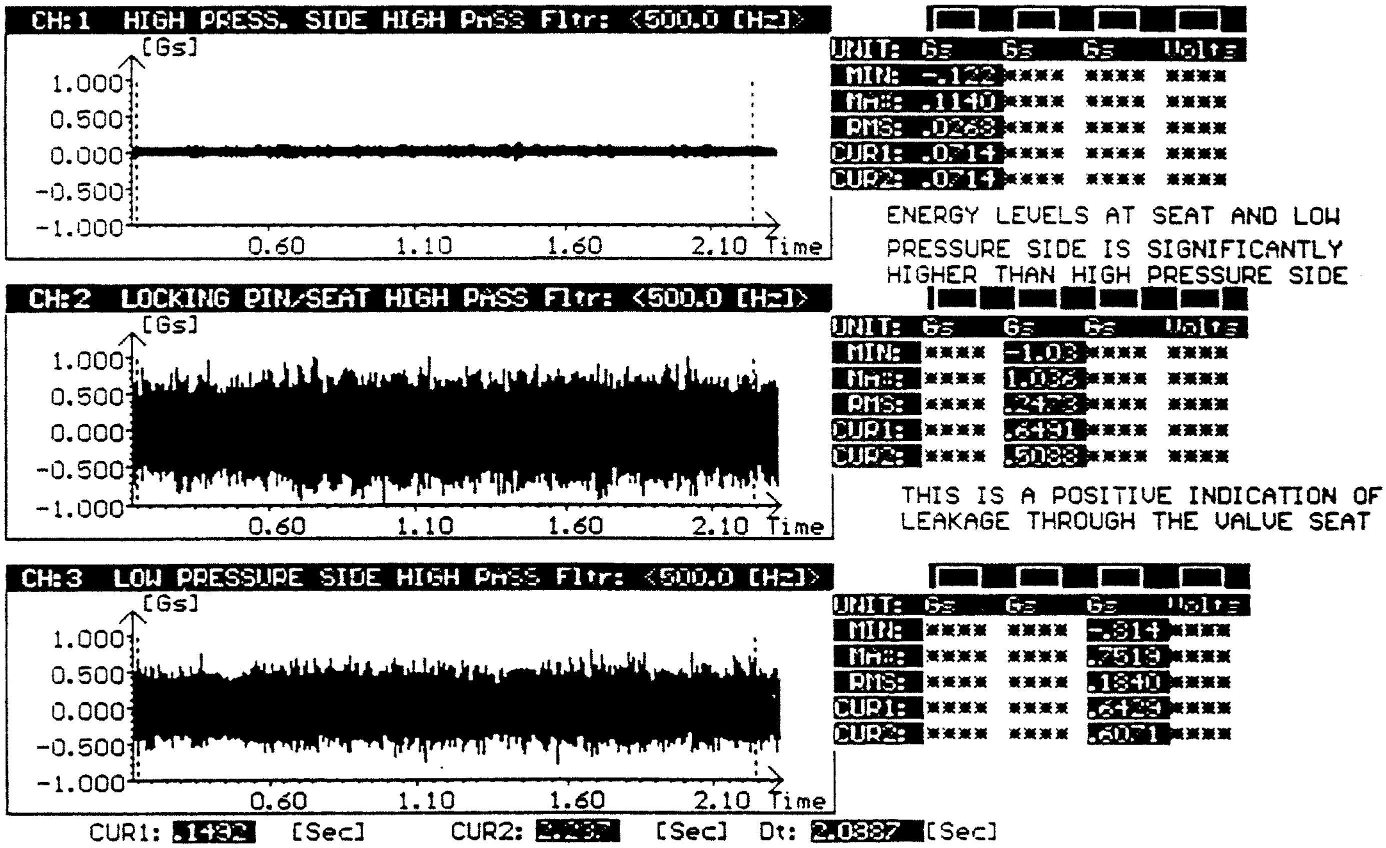
Figure 6(d): Waterfall plots showing noise died down and then gradually reappeared as the leak resumed.



(a)

(b)

Figure 7: (a) Time histories of acoustic signals from sensors mounted on the re-built RCP bleed off valve RV2199 still show numerous "tapping" like noise spike. (b) Extreme zoom-in reveals not only far field wave form but also this wave arrived at the downstream side sensor first.



(a)

Figure 8: (a) Time history of acoustic signals from sensors mounted on the re-built code safety relief valve SV1200 during the first power up test shows much higher amplitudes at the seat and downstream side--an indication of seat leakage.

(b)

(c)

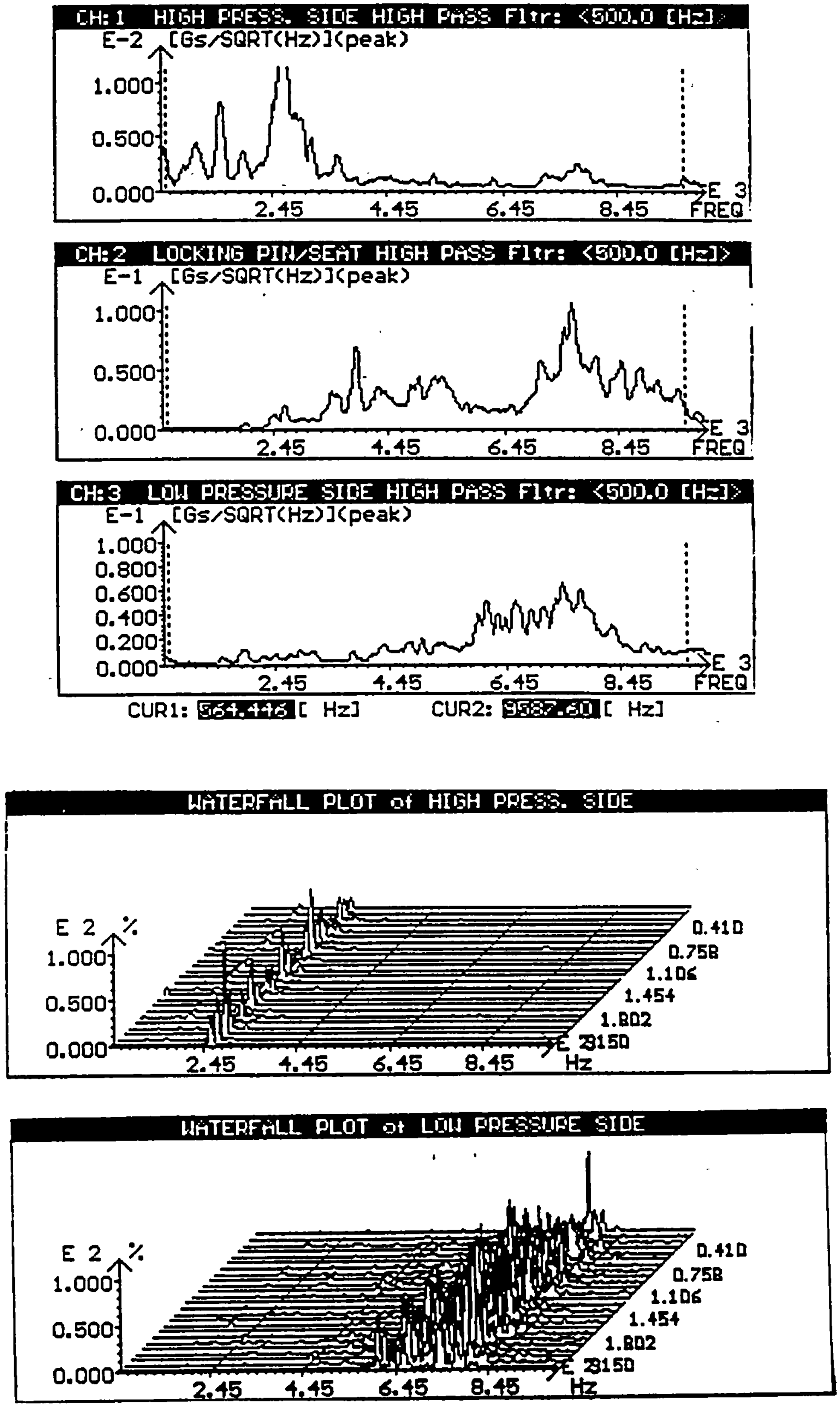
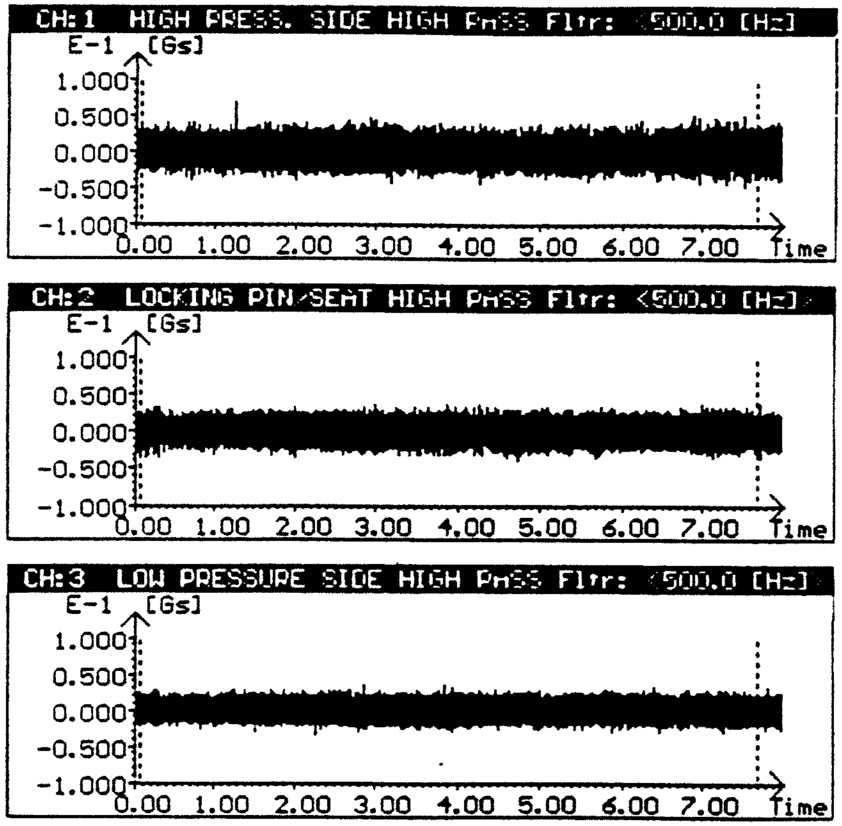
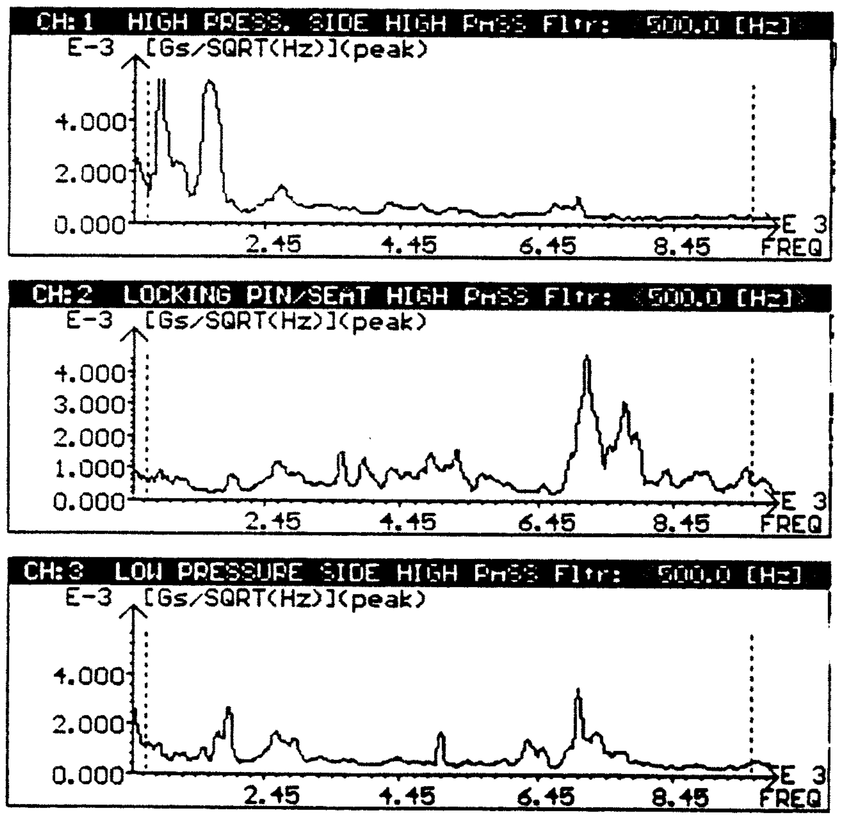


Figure 8: (b) Corresponding power spectral density plots show high frequency energy content at the seat and the downstream sensors, another indication of through seat leakage. (c) Corresponding waterfall plots.



(a)



(b)

Figure 9: (a) Time history of acoustic signals from sensors mounted on the re-built code safety relief valve SV1200 during the second power up test shows about equal amplitudes at the seat and downstream side--an indication of no major seat leakage. (b) Corresponding PSD plots show some high frequency activities at the seat, indications that some small leak existed.

ASME XI Stroke Time Testing of Solenoid Valves at Connecticut Yankee Station

Charles W. Martin, Engineering Programs Group
Connecticut Yankee Atomic Power Company

ABSTRACT

Connecticut Yankee Atomic Power Company has developed the capability of measuring the stroke times of AC and DC solenoid valves. This allows the station to measure the stroke time of any solenoid valve in the plant, even those valves which do not have valve stem position indicators. Connecticut Yankee has adapted the ITI MOVATS Checkmate 3 system, using a signal input from a Bruel and Kjaer(B&K) Model 4382 acoustic accelerometer and the Schaumberg Campbell Associates(SCA) Model SCA-1148 dual sensor, which is a combined accelerometer and gaussmeter.

INTRODUCTION

ASME XI requires that power operated valves with an active safety function be exercised and that the valve stroke time be measured. Many small solenoid valves, such as ASCO two way solenoid valves, do not have valve stem position indicators. The indicating lights for some solenoid valves, if they are installed, only provide indication that the solenoid is energized and do not indicate the actual valve stem position. Until recently, technology adaptable to measuring solenoid operated valve stroke times did not exist. The introduction of test equipment for other types of valves has made solenoid stroke time measurement possible.

Connecticut Yankee Station has developed the capability to measure solenoid valve stroke times. The following test equipment is used to measure SOV stroke times:

1. ITI MOVATS Checkmate 3 data acquisition computer.
2. One B&K Model 4382 Accelerometer.
3. SCA Model SCA-1148 dual sensor.

TEST EQUIPMENT DESCRIPTION

A description of the test equipment used by Connecticut Yankee for solenoid valve stroke testing is provided:

1. The ITI MOVATS Checkmate 3 system was originally developed for the non-intrusive testing of check valves. The Checkmate system has four data acquisition channels available. Any three channels can be used at any time. There are two acoustic channels, one UT channel, and one auxilliary channel.
2. The B & K Model 4382 accelerometer is connected to one of the acoustic channels. This accelerometer records acoustic impacts and is used to detect valve opening and closing impacts.

3. The SCA dual sensor is connected to the other accelerometer channel and the Aux channel. The SCA dual sensor is a combination gaussmeter accelerometer instrument. The gaussmeter detects the presence of a magnetic field, and indicates when the solenoid is energized. The accelerometer provides a second indication of valve opening and closing impacts.

4. The Checkmate 3 system is set up for raw data acquisition by the acoustic and auxiliary channels. The length of the data acquisition window can be varied between 8 and 179 seconds. A data acquisition window of between 15 to 30 seconds has generally been used.

ASCO AC SOLENOID VALVE TESTING

There are three ASCO AC solenoid valves in the Connecticut Yankee IST Program that are exercised and stroke time tested using the Checkmate data acquisition system. The three valves are the suction vent isolation valves for the charging pumps, CH-SOV-242 & 242B, and the Containment Air Monitor Supply Header Trip Valve, VS-SOV-12-1. All three valves are ASCO two-way solenoid valves. These valves are tested on a Cold Shutdown frequency. These valves are tested on a cold shutdown frequency because it is not practical to stroke these valves at power. A cold shutdown testing frequency is allowed by ASME XI, Subsection IWV, Paragraph IWV-3412, and is documented with an evaluation in the CYAPCO Inservice Testing Program Manual. The charging pump suction constant vent solenoid valves can not be stroked at power because that requires closing one of the charging pump suction MOV's. This cannot be done if a charging pump is required to operate. Both the opening and closing strokes are recorded on the same raw data trace. The opening stroke time is measured starting from the time that the Gaussmeter detects that the solenoid coil is energized until the time that an opening impact is detected by the acoustic accelerometers. The closing stroke time is measured from the time the Gaussmeter detects the decay of the magnetic field until a closing impact is detected. The measurement of the closing stroke time is somewhat arbitrary, depending on the time instant the analyst determines represents coil deenergization. Solenoid valves are considered to be rapid-acting valves and must open/close within two seconds. RFO 18 AC solenoid valve stroke times are listed on Table 1. The test data show that ASME XI stroke time acceptance criteria of 2 seconds were met. Figure 1 shows the opening stroke event for CH-SOV-242, and Figure 2 shows the closing stroke event for that valve. These graphs clearly indicate the events monitored during the test.

Table 1 Connecticut Yankee RFO 18 AC Solenoid Valve Stroke Times

Valve ID	Open Stroke Time(msec)	Closed Stroke Time(msec)
CH-SOV-242	13	5
CH-SOV-242B	13	11
VS-SOV-12-1	6	17

DC SOLENOID VALVE TESTING

The Checkmate system is also capable of testing DC solenoid valves. A similar instrument setup is used for testing DC solenoid valves, with a accelerometer and the dual sensor providing input signals to the Checkmate data acquisition system. The four DC Solenoid valves tested in the CYAPCO Inservice Testing Program are the remote

actuation solenoid valves for the dual purpose main steam safety relief valves. When DC Solenoid valves are tested, the gaussmeter is only capable of detecting one solenoid coil event, either the coil energization or deenergization. After one valve stroke event is recorded on the Checkmate data acquistion system, the polarity of the gaussmeter leads must be reversed, and the valve exercise/stroke timing test repeated to record the other solenoid coil event. This is a phenomenon that was observed the first time that Connecticut Yankee tested these solenoid valves.

Figure 3 shows a DC solenoid valve opening valve stroke test for MS-SOV-1615-1 and Figure 4 shows a DC solenoid valve closing stroke test for the same valve. RFO 18 DC solenoid valve stroke times are listed in Table 2.

Table 2 Connecticut Yankee RFO 18 DC Solenoid Valve Stroke Times

Valve ID	Open Stroke Time(msec)	Closed Stroke Time(msec)
MS-SOV-1615-1	58	375
MS-SOV-1615-2	48	374
MS-SOV-1615-3	54	299
MS-SOV-1615-4	41	375

INSERVICE TESTING PROGRAM ISSUES

Generic Letter 89-04 provided guidance on measurement of stroke times of rapid acting valves. Connecticut Yankee surveillance tests measure the stroke time of these valves and verify that the stroke time was less than 2 seconds. Connecticut Yankee does not trend the stroke times measured by non-intrusive solenoid valve testing. Hard copies of the diagnostic graphs that have been marked with the valve events and measured stroke times are attached to the Automated Work Orders for the surveillance tests, and are stored in plant nuclear records. The Checkmate system non-intrusive test data is stored electronically in Connecticut Yankee Engineering Programs files. The IST Program Engineer expects a stroke time for certain solenoid valve designs, for some valve manufacturers and valve sizes, based on observed test data. If non-intrusive solenoid valve testing indicated a significant variation from the expected stroke time for that style valve, additional evaluation would be performed to determine the cause for the increased stroke time.

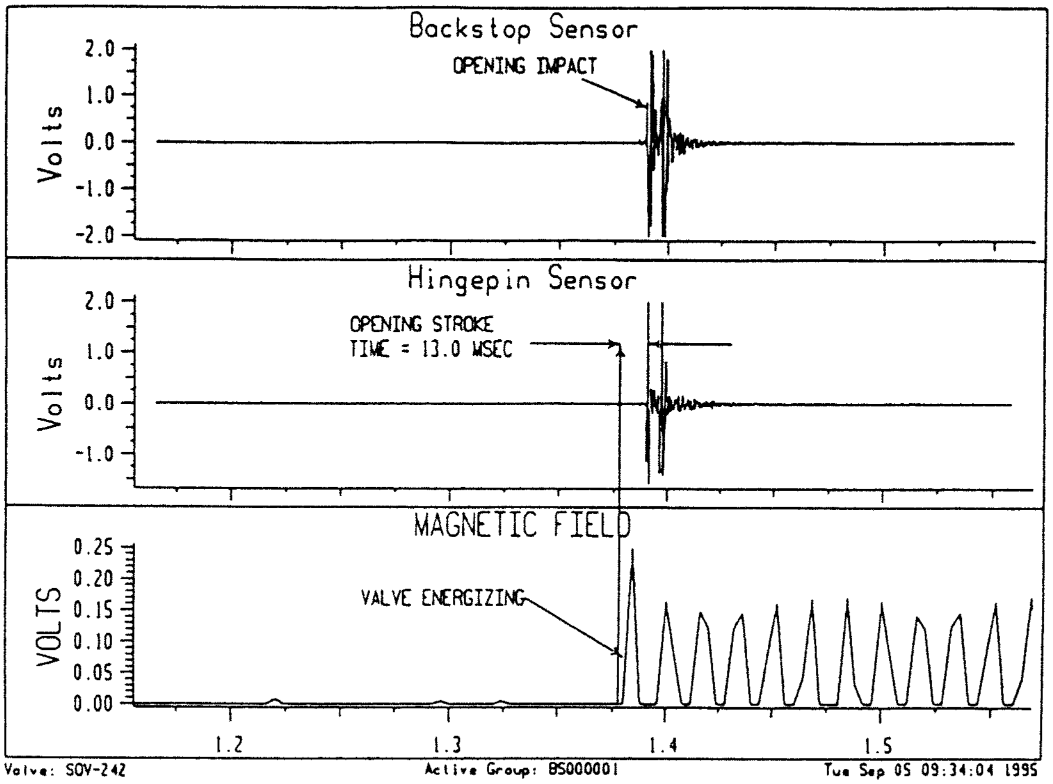


Figure 1 Opening Stroke Test Diagnostic Trace for CH-SOV-242 Recorded 1/31/95

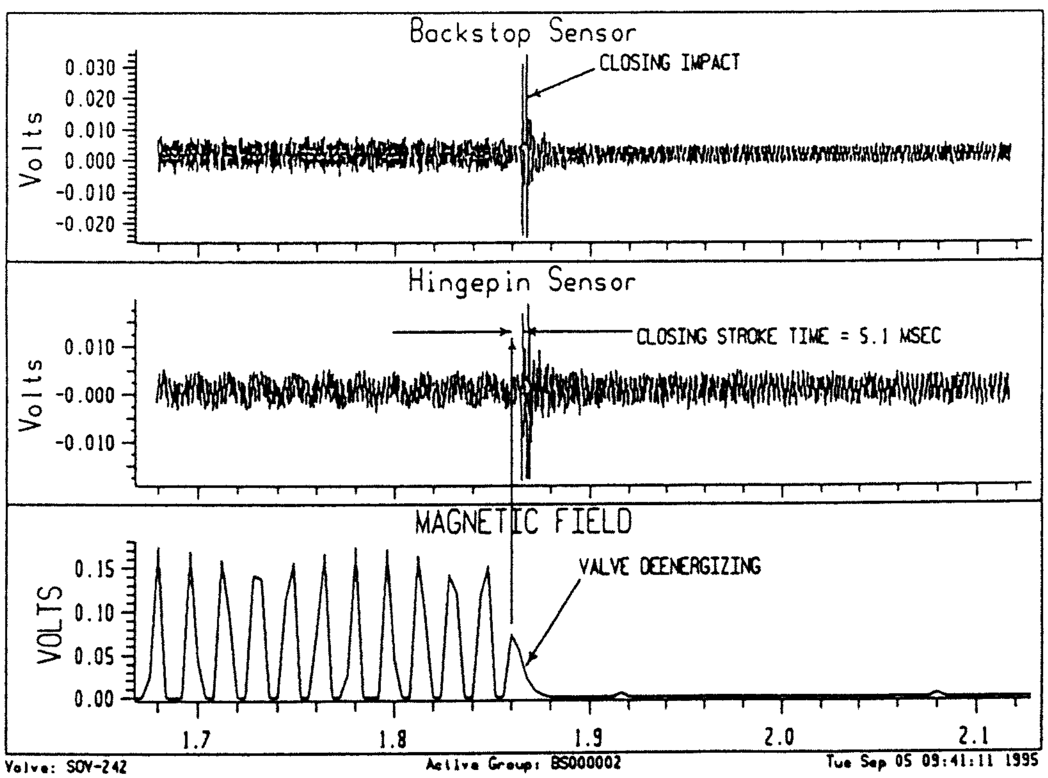


Figure 2 Closing Valve Stroke Diagnostic Test for CH-SOV-242 Recorded 1/31/95

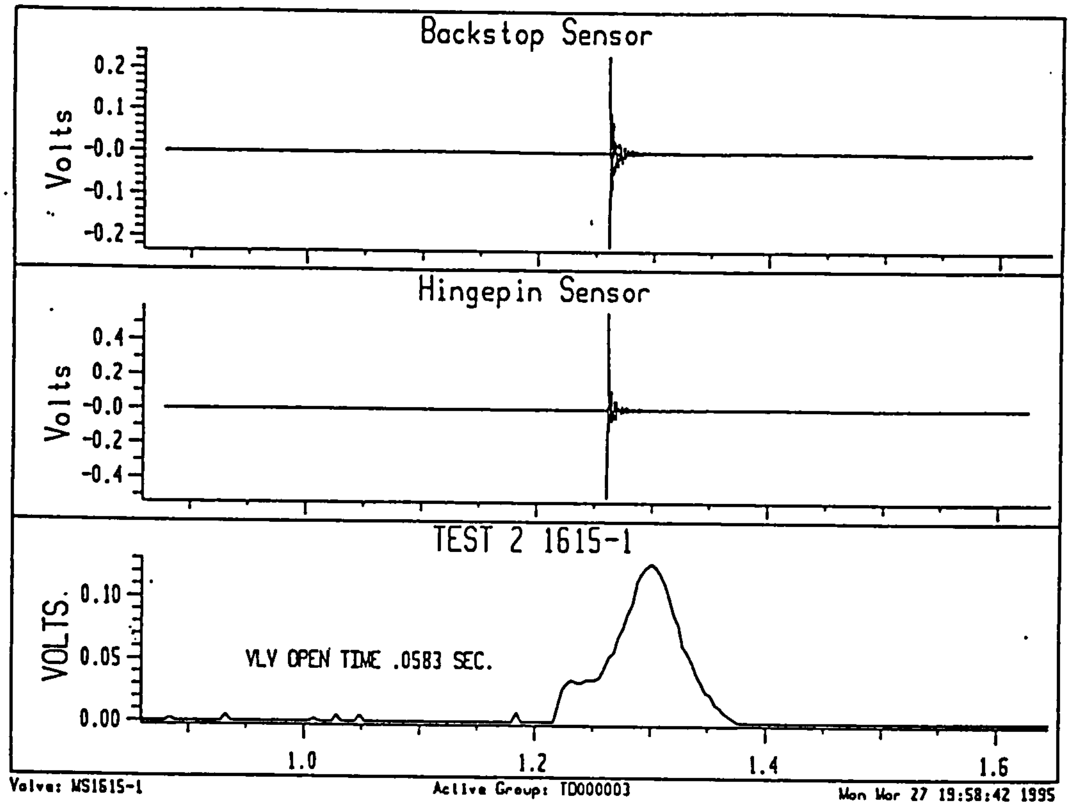


Figure 3 Opening Stroke Test Diagnostic Trace for MS-SOV-1615-1 Recorded 3/24/95.

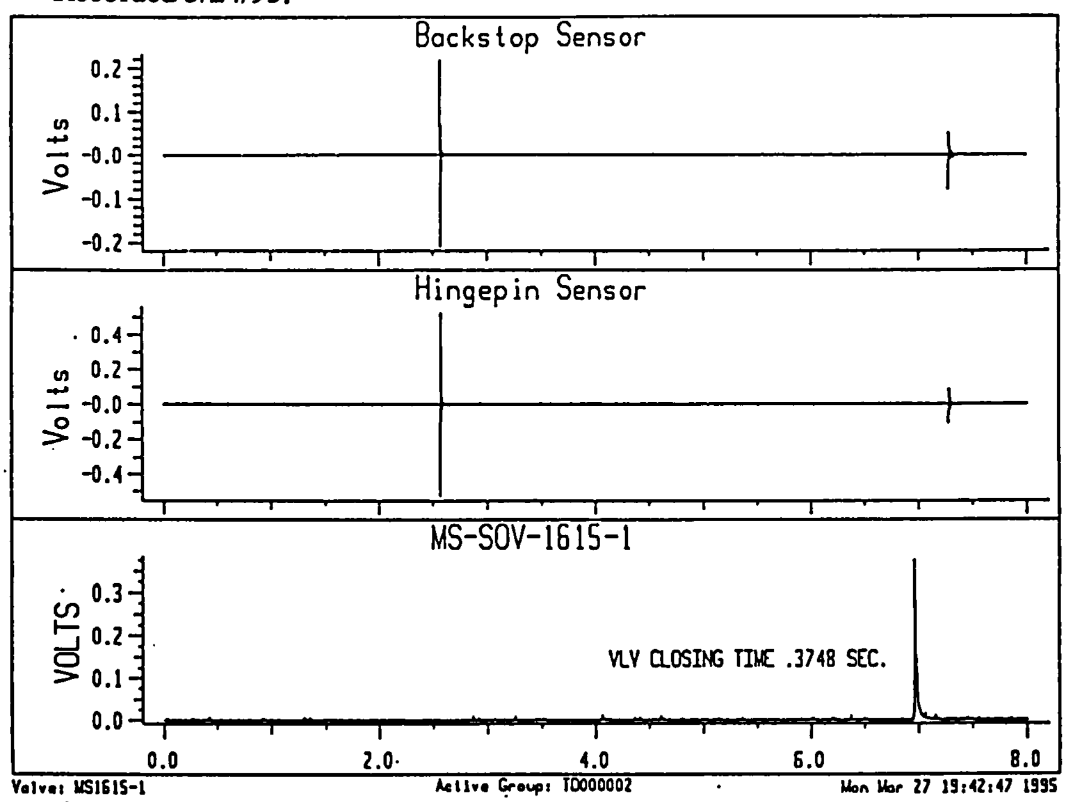


Figure 4 Closing Stroke Test Diagnostic Trace for MS-SOV-1615-1 Recorded 3/24/95.

The Use of Check Valve Performance Data to Support New Concepts (Probabilistic Risk Assessment, Condition Monitoring) for Check Valve Program

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ABSTRACT

The concept of developing an integrated check valve database based on the Nuclear Power Reliability Data System (NPRDS) data was presented at the last Symposium. The Nuclear Industry Check Valve Group (NIC), working in cooperation with the Oak Ridge National Laboratory (ORNL), has completed an operational database of check valve performance from 1984 to the present. NIC has committed to the nuclear industry to periodically update the data and maintain this information accessible.

As the new concepts of probabilistic risk analysis and *condition monitoring* are integrated into the American Society of Mechanical Engineers (ASME) Code, a critical element will be performance data. From check valve performance data, feasible failure modes and rates can be established. When a failure rate or frequency of failures can be established based on a significant enough population (sampling), a more solid foundation for focusing resources and determining appropriate frequencies and testing can be determined.

The presentation will give the updated status of the NIC Check Valve Performance Database covering (1) methodology used to combine the original ORNL data; (2) process/controls established for continuing update and refinement of the data; (3) discussion of how this data is being utilized by (a) OM-22 for *condition monitoring*, and (b) risk-based inservice testing work of Westinghouse Owners' Group; and (4) results/trends of data evaluations.

At the 1994 Symposium, ORNL provided an update as of 1991 to their original work of 1984 - 1990 which they had performed to characterize check valve degradations and failures in the nuclear industry. These characterizations will be updated to 1995 and additional reviews provided to give insight into the current condition and trends of check valve performance.

NIC Check Valve Database Graphics

The charts and graphs on the following pages represent a compilation of data obtained through NPRDS from 69 nuclear stations (111 units) from 1984 through 1995. There is currently a total of 2705 failure records in the database. Of this total, 197 records have been re-coded as "non-failures" based on failure narratives reporting external leakage, dirty internals, minor wear, etc. It is the remaining 2508 records upon which the graphs and charts are based.

The following definitions are provided to more clearly express the meaning of the terms used in the attendant graphs.

General Detection Methods

Programmatic: Failure observed during the conduct of a surveillance test, inservice inspection or test, leak rate test, post-modifiction test, bench test, or periodic preventative maintenance (test, scheduled inspection, etc.) on the valve or a related piece of equipment (such as a diesel generator).

Routine (or incidental) observation: Failure observed by off-normal plant instrumentation readings (such as level, pressure, etc.) during the course of normal operation. These include such observations as elevated piping temperature by feeling of piping. Includes normal operator rounds and system walk down.

Abnormal equipment operation: Failure observed by off-normal operation of plant equipment, such as reverse flow of a pump, frequent cycling of a compressor, or lifting or a relief valve.

Special inspection: A degraded condition discovered during an inspection performed due to failure of a similar valve at either the plant in question or some other plant (such as an inspection performed as a result of an NRC Notice on some particular manufactuter's valve) or as part of an inspection process that was not routine in nature.

Miscellaneous or unclear: A failure that did not fit into any of the above categories. Includes failures found as a result of correcting other valve problems (such as finding a disc/seat clearance problem when replacing a leaking gasket) or when performing maintenance on another component. Also includes those failures for which the general detection means were not identified.

Failure Modes

Improper seating: Includes all failures in which the valve failed to properly seat (excluding stuck open and restricted motion cases), resulting in internal leakage (in most cases, in excess of some specific limit). Includes failures described as "valve leaking by its seat", dailure of a seat leakage test, or where small amounts of foreign material or dirt prevents valve from fully seating.

Detached or Broken (disc or other part): The disc or some other internal part was loose (detached from the internal assembly) or cracked or broken.

Free or loose (not detached) or impact/friction damaged part: Some portion of the assembly, generally in the hinge pin or disc stud area, was found to be loose or otherwise not in

proper assembly condition (with no attendant problems, such as stuck open, etc.).

Restricted motion or reduced flow: A condition in which free motion of the valve was restricted. Includes obturator sticking, binding, or unable to move freely. Partial obturator movement was either stated or implied in narrative.

Stuck Closed: Valve failed to open upon demand (when forward pressure was applied).

Stuck Open: Valve failed to full close upon demand. Includes cases in which the disc was clearly stuck open or cocked, or when the disc was cocked in the seat due to wear of the disc stud.

Unknown or miscellaneous: Includes failures not applicable to any other category or not explicitly described in the narrative.

Corrective Actions

Disassemble and clean/tighten/adjust: No machining required.

Clean/tighten/adjust/lubricate: External only; no *disassembly* required.

Replace valve: Replace valve.

Clean and refurbish internals: Includes machining, lapping, or welding of plug, seat, or disc (or other internal part(s)), but where no internal part replacement was necessary (except bonnet or flange gaskets).

Clean and rebuild internals or replace some internal parts: Including seat seals and o-rings; including machining or refurbishing of internals.

Other: Unidentified in narrative or not in the above categories.

Extent of Degradation

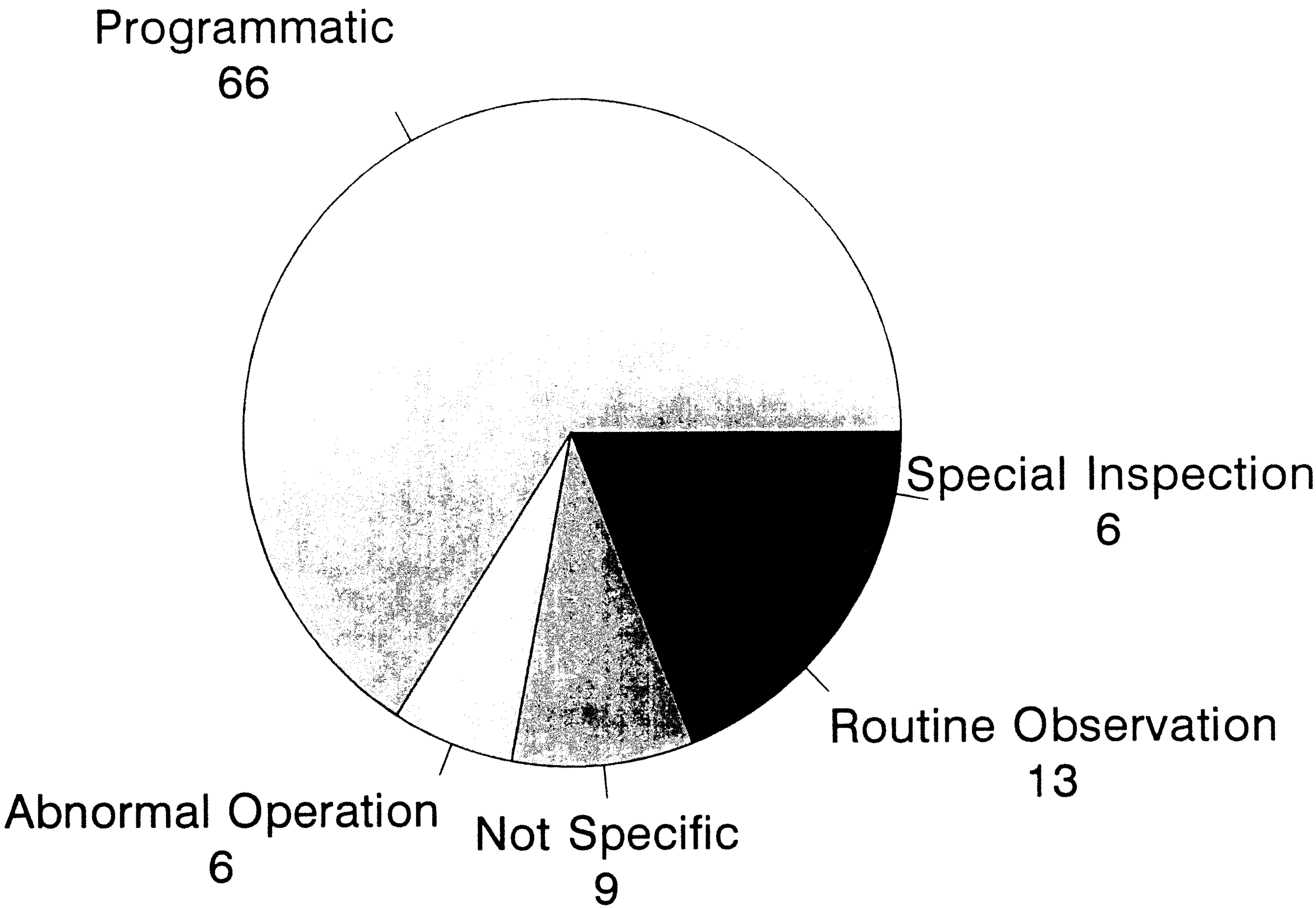
Moderate: Includes failures to seat properly (excluding stuck open and restricted motion failures), and generally includes any major quantifiable local leakage rate test (LLRT) leakage where seat lapping or machining or parts replacement was necessary or where moderate internal seat leakage, loose internal assembly (without attendant problems, such as stuck open), or a miscellaneous failure in which the level of degradation was not evident from the narrative.

Significant: Includes broken or detached internals, restricted motion, stuck open and stuck closed cases, and cases where relief valves failed to meet set pressure. Also includes gross internal leakage (LLRT "off scale" or "would not pressurize").

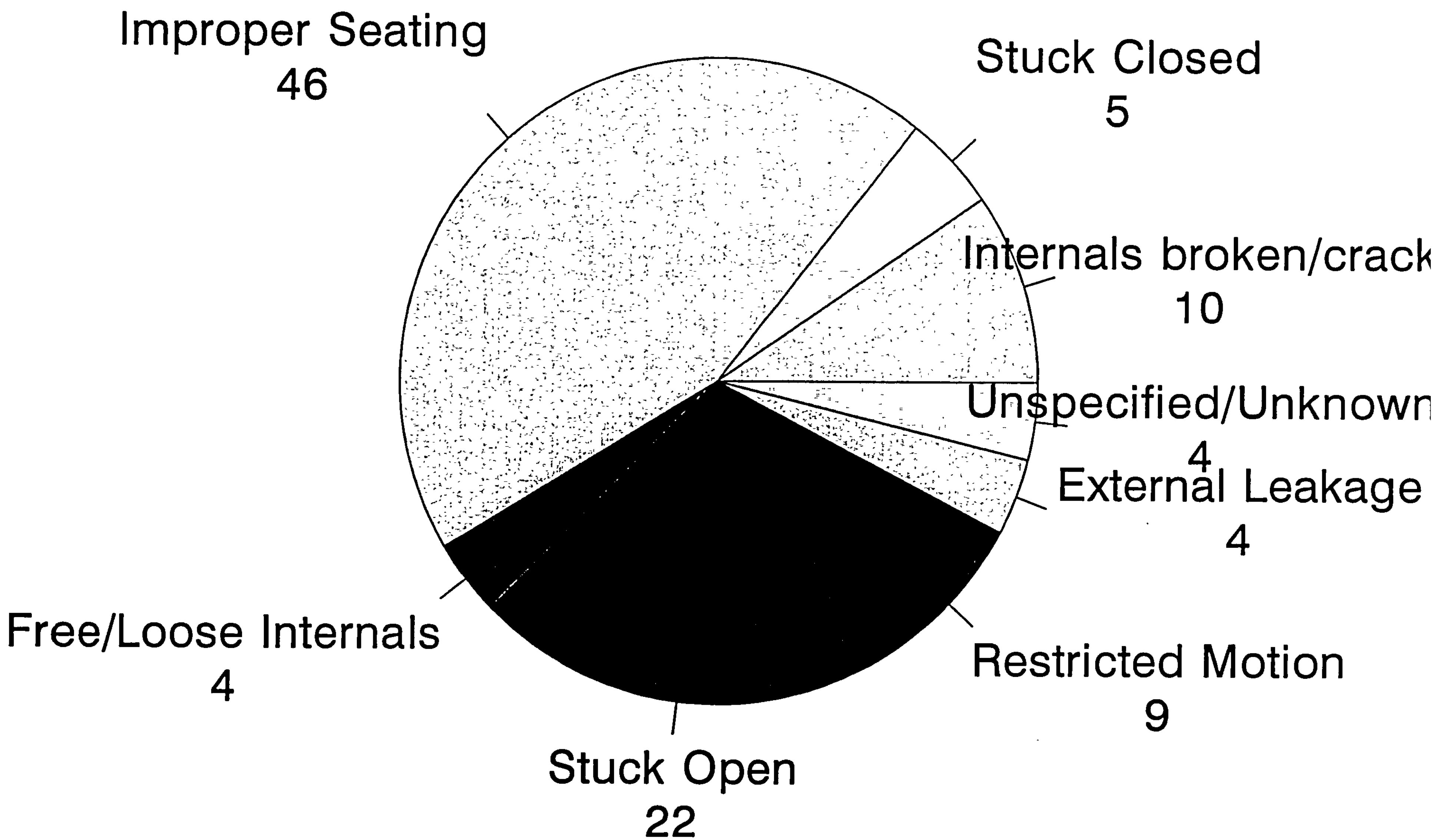
Not a failure: Generally, any LLRT failure to meet specified limits but where internal leakage was quantifiable (not "off scale" or "would not pressurize") and no repairs or replacements were required, or where narrative indicates valve continued to function even though some wear was found.

NOTE: The graphs are generally given in percentages.

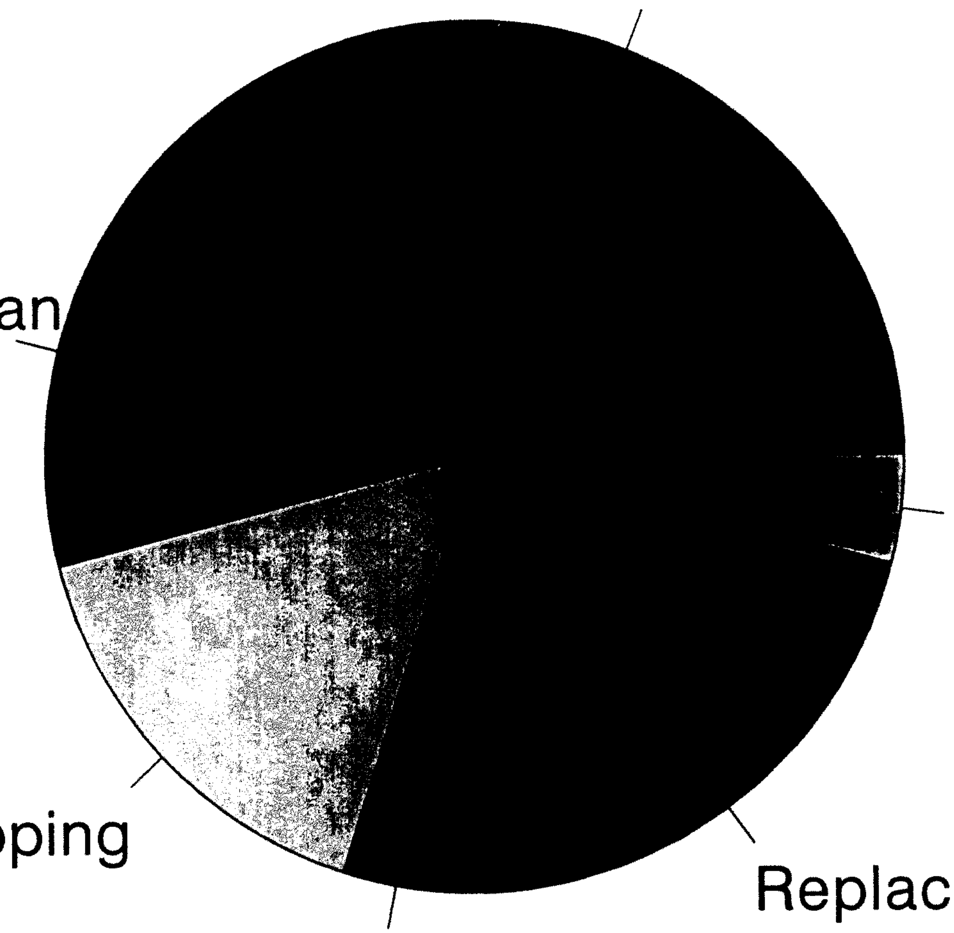
General Detection Methods



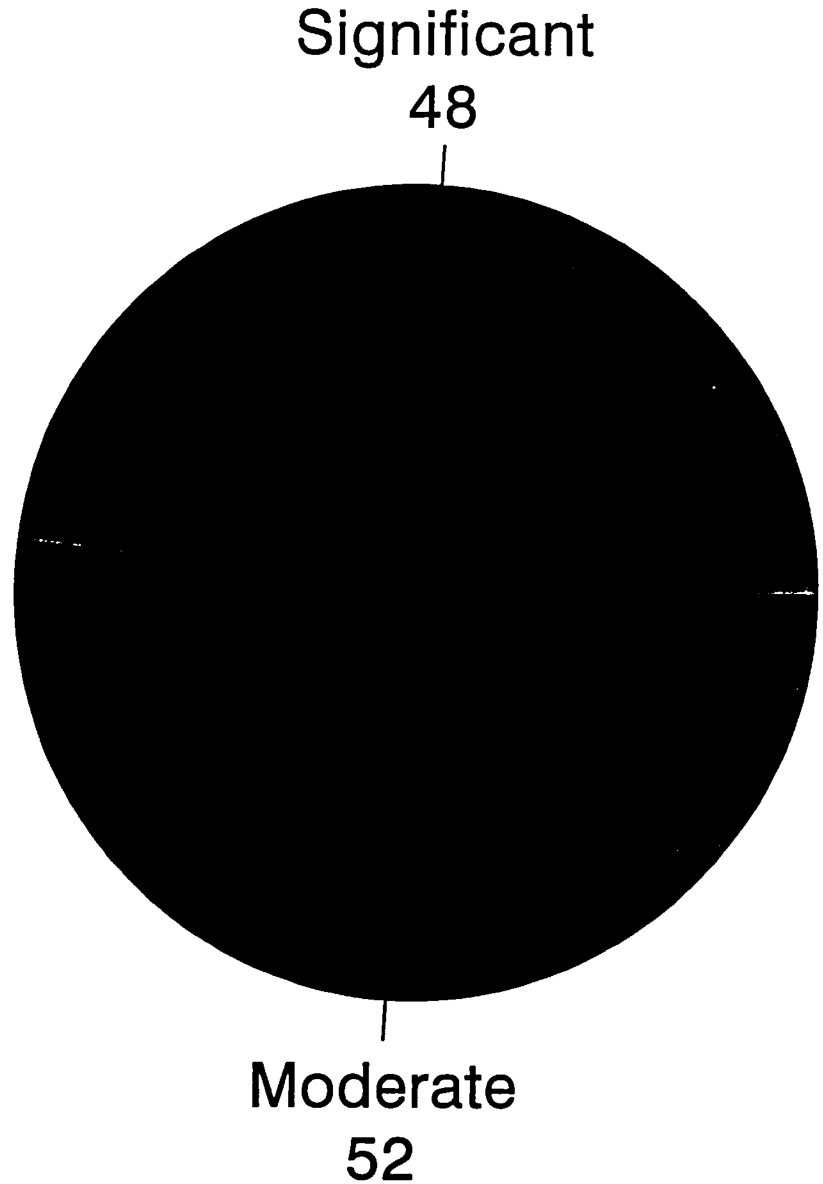
Failure Modes



Corrective Actions

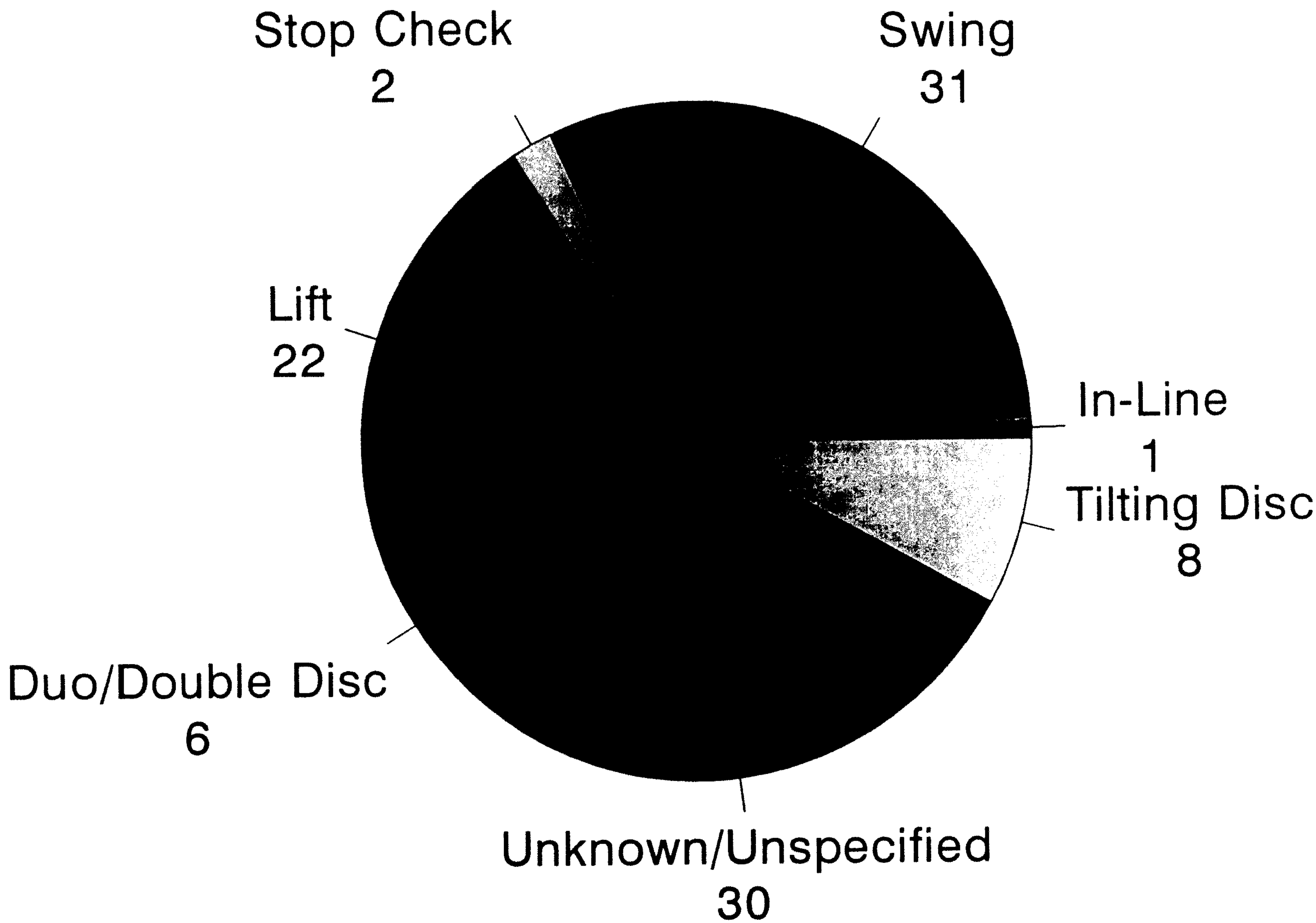


Extent of Degradation

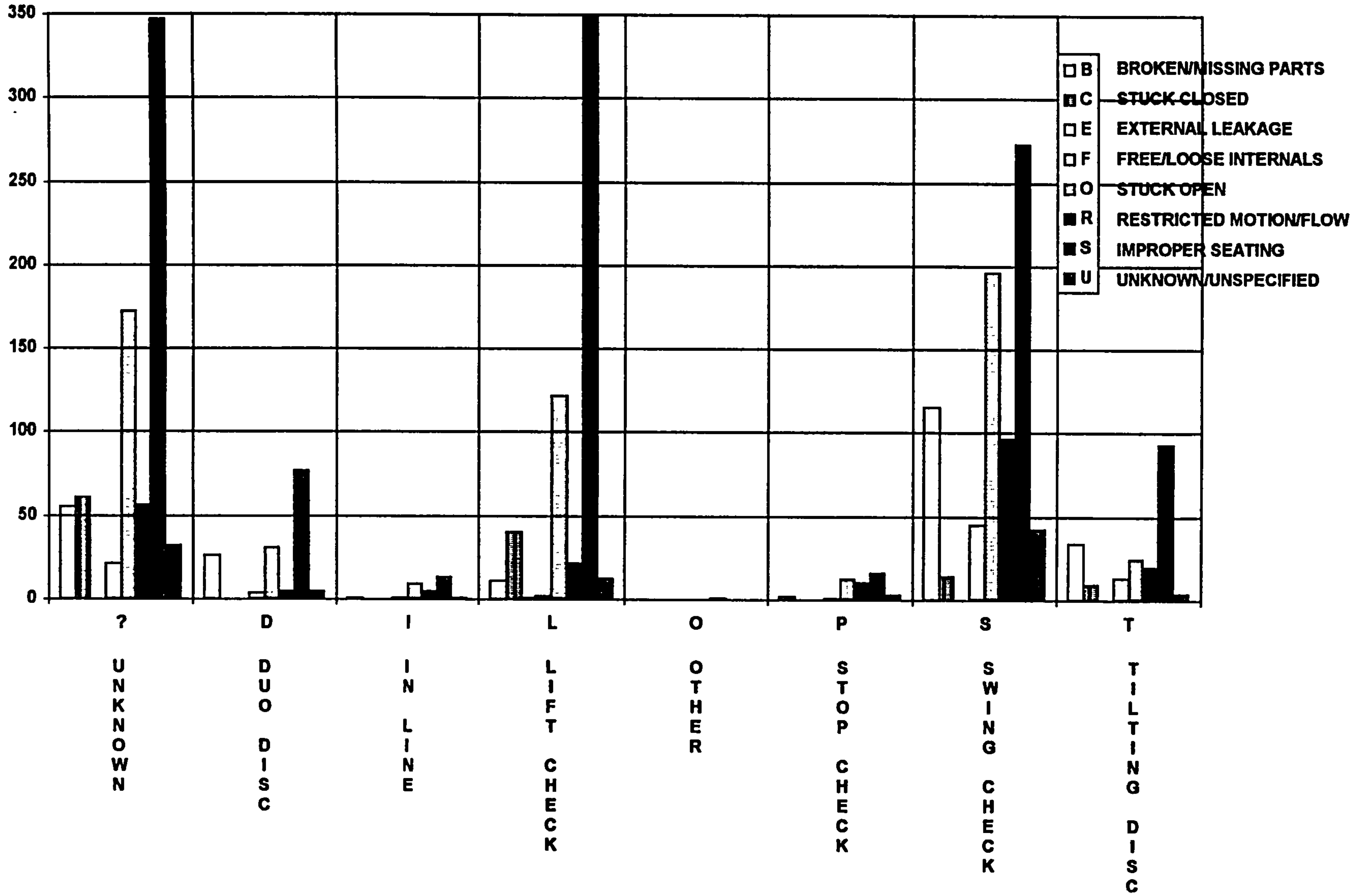


Percentage of Significant and Moderate

Failures by Valve Type



FAILURE MODES BY VALVE TYPE



DEVELOPMENT OF AN EFFECTIVE VALVE PACKING PROGRAM

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ABSTRACT

Current data now shows that graphite valve packing installed within the guidance of a controlled program produces not only reliable stem sealing but predictable running loads. By utilizing recent technological developments in valve performance monitoring for both MOV's and AOV's, valve packing performance can be enhanced while reducing maintenance costs. Once known, values are established for acceptable valve packing loads, the measurement of actual valve running loads via the current MOV/AOV diagnostic techniques can provide indication of future valve stem sealing problems, improper valve packing installation or identify the opportunity for valve packing program improvements.

At times the full benefit of these advances in material and predictive technology remain underutilized due to simple past misconceptions associated with valve packing.

This paper will explore the basis for these misconceptions, provide general insight into the current understanding of valve packing and demonstrate how with this new understanding and current valve diagnostic equipment the key aspects required to develop an effective, quality valve packing program fit together. The cost and operational benefits provided by this approach can be significant impact by the:

elimination of periodic valve repacking
reduction of maintenance costs
benefits of leak-free valve operation
justification for reduced Post Maintenance Test Requirements
reduced radiation exposure
improved plant appearance

BACKGROUND

Over the past nine years, a detailed, programmatic approach to valve packing has been implemented at PP&L Resource's, Susquehanna nuclear station. From the 12,000 valves repacked at Susquehanna, many lessons have been learned the hard way. Our valve packing supplier, Argo Packing, provided not only technical support but also added new insights from their experiences

gained at other utilities. Due to our "total valve" maintenance approach (all valve work is centralized in Maintenance Engineering), information from both motor and air operated valve efforts is coordinated with valve packing data to provide new insights into understanding and predicting valve packing performance.

As PP&L Resources has shared our experiences with valve packing in training classes and utility meetings, we consistently find two things. One, the same misconceptions of valve packing which impacted our program, though subtle, keep most valve packing programs from being as cost effective as they could be. Second, few utilities take advantage of the vast amount of motor operated and air operated valve diagnostic data available to feedback as a predictive tool into their valve packing effort.

Addressing and correcting the impact of these misconceptions and integrating valve diagnostic data into a programmatic valve packing program has demonstrated reduction in maintenance and material savings are achievable. Arguments against the traditional need for periodic valve repacking will be presented. A methodology for establishing a predictable packing load and utilizing motor and air operated valve diagnostic systems to monitor actual packing loads to insure long term performance are discussed.

HISTORICAL MISCONCEPTION ABOUT VALVE PACKING

Asbestos valve packing because of its long term use for decades in the power industry prior to the late 1980's provided the basis for our historical experiences with valve packing performance. Because of both the specific properties of asbestos valve packing and past misconceptions on the mechanism of valve stem sealing, several inaccurate assumptions and practices have long been accepted, which no longer apply to the modern expanded graphite packing materials and can frequently interfere with efforts to improve valve packing performance today.

VALVES REQUIRE PERIODIC REPACKING

Asbestos valve packing includes both an inconel wire for strength and various binders and fillers to improve performance. In time, the nature of the various additives allow the packing to "dry out" when continually exposed to high temperatures. Eventually because of this deterioration the sealing capability of the asbestos packing degrades. The remaining material has typically been described as "concrete" when valve were unpacked. Once the packing material has lost it's flexibility and elasticity, it clearly can no longer provide a dynamic stem sealing to a valve as the stem is raised and lowered through the valve packing. This "drying out" over time has lead the industry in general to assume that valve packing has a finite life and will require valve repacking at some periodic frequency.

VALVE SEALING MECHANISM "by pressure breakdown "

In part because of the inherent strength of asbestos packing, it was assumed that the sealing mechanism between the valve stem and the valve packing was produced by a series of pressure breakdowns, similar to the labyrinth seal design. With each ring of valve packing, it was thought that the internal pressure was reduced, until the point that the valve packing could hold back the pressure and totally seal off the leakage path. This assumption led to the design of valves with deep stuffing boxes to accommodate large numbers of packing rings (10-12) and more packing rings for higher pressure systems. No one thought you could have too much packing!

VALVE PACKING WILL LEAK

If you accept as fact that valve packing must be periodically replaced, it follows that if you do not repack a valve before it's packing "wears out" it will leak. That valve packing will leak is such an accepted fact, that in the 1970's leakoff systems were added into plant designs in an attempt to indicate when valve packing starts to leak. One nuclear steam supplier even designed a triple stuffing box to have two totally redundant packing sets in a valve.

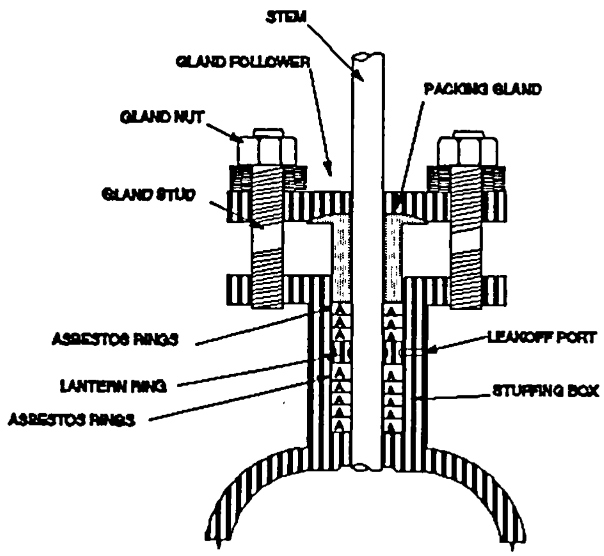

Figure 1. TYPICAL VALVE STEM LEAK-OFF SYSTEM

Valve leak off systems doubled the amount of valve packing in the valve and included a leak off point halfway through the packing to monitor for leakage of the lower packing. Since the assumption is that the packing has a specific life and the installation of additional packing will improve pressure breakdown and therefore sealing, the leakoff system was thought to have no adverse impact on packing performance

Valve suppliers today continue to offer leakoff systems.

PACKING LOADS ARE 1000 LB. PER INCH OF STEM

Historically valve packing loads have been assumed to be 1000 lb. per inch of stem diameter. Though for many valves this has proven to be a fair estimate, there exists no empirical or analytical basis for this statement. Even today this assumption is still used in many valve sizing equations.

Assumptions used for air operated valves are even less analytical. To minimize the effect of valve packing loads on air operated valves, many manufacturers in the past provided instructions to adjust valve packing "just to the point that it does not leak". With this direction extremely low packing loads could be achieved. Lower packing loads on an air operated valves significantly improves it's control and minimizes the valve operator size.

With no definitive basis for actual valve packing loads, no guidance was generally provided to the mechanic on the proper torque value to tighten packing gland nuts to. It was assumed the mechanic's **basic skills provided him with sufficient information to properly tighten valve packing.**

RESULTS OF RESEARCH & MATERIAL IMPROVEMENTS

Valve packing research in 1980's identified that the real challenge to valve stem sealing came in handling the diametrical stem size changes as it is stroked through the stuffing box. The change of stem size is caused by the temperature change seen by a valve stem between when it is in contact with the hot process fluid and then is withdrawn and subjected to the external environment of the valve (ambient air temperature). Once drawn out from the valve,

the stem cools and as it cools the thermal expansion of the stem material allows it to decrease. When the valve stem is then driven through the stuffing box, the stem diameter is slightly smaller than the diameter of the hot stem which had been pulled through the stuffing box when the valve was opened. The valve packing MUST remain flexible if it is to handle this situation. As the asbestos packing "dried out" it lost the ability to handle this situation.

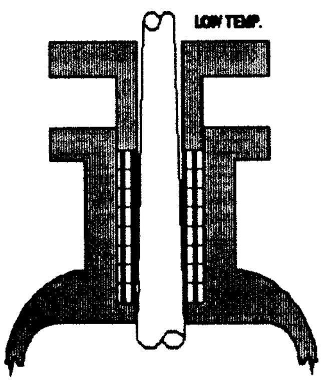

Figure 2. EFFECT OF INTERNAL TEMPERATURE ON VALVE STEM DIAMETER

The key to achieving stem sealing is to insure adequate radial load is applied to a packing ring. To obtain adequate radial loads in valve packing material requires (1) adequate axial load be applied to packing gland nuts, and (2) packing material to convert axial load to radial.

To accomplish this conversion the packing system must remain flexible and be able to store adequate energy to respond to the change in stem diameter discussed above.

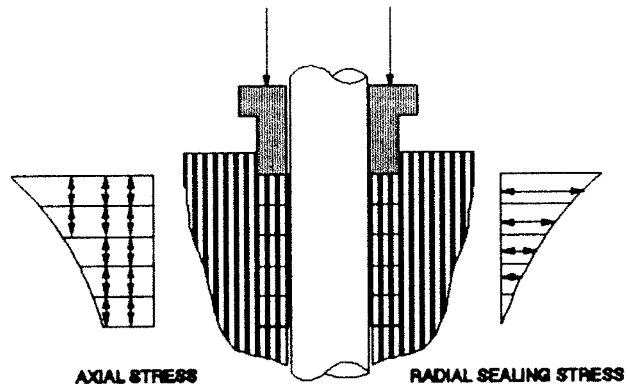

Figure 3. TRANSFER OF GLAND LOAD TO RADIAL SEALING LOAD

Understanding of this concept turns valve packing design 180 degrees around. The goal now is to insure radial load to the valve packing to seal against the valve stem not pressure breakdown. The packing material must remain capable of transferring axial stress to radial stress to the stem and stuffing box. As the asbestos packing "dried out" it lost this ability.

A more subtle impact which was found is that each additional ring of packing makes the transfer of adequate axial loading to the next lower packing ring more difficult. Typically the lower rings of packing will see little if any load. Once placed in service stroking of the valve stem through the valve packing and internal system pressure, will distribute whatever load exists in the packing rings, uniformly between all rings. Frequently after this redistribution, the final packing load achieved is barely adequate to allow the valve packing to seal the stem and valve leakage occurs as the plant is starting up or soon there after. Typically in the past and even at plants today, packing leaks on plant startup are expected and considered normal. When the mechanic tightens the gland nuts on these leaking valves it should not be surprising that the leaks stop, since he then provides adequate axial load

to the packing for sealing. But if this same load could have been provided to the valve packing during the original valve packing work, this leak would have never occurred.

INTRODUCTION OF EXPANDED GRAPHITE VALVE PACKING MATERIAL

The general elimination of asbestos as a viable material and the introduction of expanded graphite provided an improved material for valve packing in the 1980's. Exfoliated expanded natural graphite proved to be an excellent sealing material for high temperature applications. It required none of the fillers, binders or wire which had been included with asbestos. Without these added materials which would eventually burn off or dry out, the graphite's characteristic properties remain unchanged with time and service. The basis for "periodic repacking" is **eliminated.**

Expanded graphite packing rings are die formed from ribbon in a press to the desired dimensions and density. It is critical to note that the forming pressure of the ring must be exceeded before the graphite ring can convert axial load from the gland nuts to radial "sealing load" on the stem.

Because of graphite's ability to flow (the reason it provides such excellent sealing) a mechanism must be provided to prevent the graphite from escaping from the valve stuffing box area. This confinement is provided by a containment packing ring (typically a braided graphite yarn or composite) The containment ring provides no effective high pressure sealing of the valve stem.

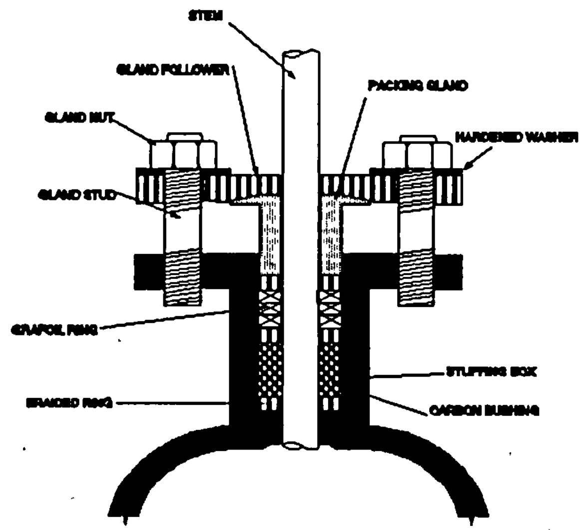

Figure 4. STANDARD 5 RING GRAPHITE PACKING SET

Valve packing seals via adequate radial load being applied to the packing, not by pressure breakdown thorough numerous packing rings. Extensive testing has shown that only one died formed ring is required to provide adequate sealing, but to insure backup protection, the standard graphite packing set typically includes additional die formed rings and has a containment (wiper) ring on the top and bottom.

From the research on graphite packing the technical basis to begin to correct misconception exists.

INITIAL PACKING PROGRAMS IN THE LATE 80's

The conversion from asbestos packing to graphite packing in general was treated as an improvement based on the material change and this new understanding that sealing was achieved via radial

loading of the packing ring. The misconception of sealing via pressure breakdown was discarded. The elimination of excess packing rings via the use of a carbon spacer was accepted.

The challenge to early packing programs was to insure radial loading was achieved, since it was not a parameter which could be easily confirmed by a mechanic out in the plant. To achieve adequate radial load several factors must be addressed, adequate load must be applied to the packing gland nut, the load applied must be higher than the force used to form the packing ring, and the load must be distributed uniformly to all packing rings. These techniques were clearly not considered with asbestos packing, and a continuing reluctance to them can still be found today among some mechanics. To accomplish the proper loading of valve packing the following techniques were initially used.

> Specifying torque values for packing gland nuts provides the mechanic with a key guideline to insuring that at least adequate axial load was applied to packing gland.
>
> A concept of packing consolidation was instituted to insure that the axial load applied by the mechanic was distributed to all packing rings. Packing consolidation is achieved by the repetitive valve stroking and torquing.
>
> An alternative method utilized by some packing vendors is to measure packing compression to insure adequate consolidation is achieved

Early efforts concentrated as noted above on the massive conversion from asbestos to graphite and on simple techniques to provide some confidence that adequate radial loads were achieved. It is crucial to point out that not all packing misconceptions were corrected. The need for periodic repacking, the attitude that valve packing will leak and prediction of 1000#/ stem inch of valve packing running load remained and continued to impact decision making and attitudes toward valve packing

WHAT HAVE WE LEARNED IN RECENT YEARS

The material improvement to graphite from asbestos and the understanding of the need to achieve radial load of the packing ring for sealing laid the foundation for further packing improvements. During the early conversion from asbestos to graphite packing, some utilities elected to institute programmatic maintenance techniques along with the material change to graphite packing. The programmatic aspects included such areas as:

- attention to detail
- root cause evaluation
- specialized training
- technical field support
- identification of PM requirements

From this programmatic approach, additional insights were developed or reinforced on valve packing performance. The impact of valve condition on performance became clearer when only certain valves or valve types continued to leak. As reasons for leaks are searched for by root cause evaluations and vendor technical support attention focuses on other factors which effect packing performance more insight is gained. The factors which are discovered which impact performance include:

- valve clearance
- stem condition/finish
- condition of stuffing box
- stem alignment/maintaining centered
- condition of stud and nuts
- packing gland clearance

When any of these factors were grossly deteriorated it was obvious that packing performance would be impacted, but it was found that more minor defects had an impact.

The last few misconceptions of valve packing can now be addressed.

Valves do not require periodic repacking. With the conversion to graphite which does not have additional binders and fillers, there is no lose of flexibility and no drying out associated with asbestos. The critical parameter for graphite packing which must be maintained is "adequate radial load" on the packing ring to the stem. By the use of periodic packing retorque and live loading of packing, radial load can be maintained without repacking.

Though the material superiority of graphite and need to maintain radial load was known, it has been difficult to change this past misconception. Most laboratory data exploring packing life is based on cycle life (i.e. number of valve strokes) not years in service. With the past experience with asbestos and a lack of attention to this point the misconception has continued.

Actual field experience over more than 8 years at Susquehanna and other utilities now show that with periodic restoration of radial load to packing, valve repacking is not necessary. At many sites it is difficult if not impossible to reach this conclusion, since after valves were initially repacked, the valves were ignored. It was then assumed when they developed leaks it was to be expected and that the valves required a repack.

By evaluating data obtained from periodic packing retorques (i.e. remaining take-up, as found torque) any valve which could require a repack due to constant stem cycling or poor stem condition will be identified prior to a packing leak occurring.

Valve packing should never leak. As obvious as this statement may sound it is at times the hardest misconception to correct. It requires that personnel believe in it. It means that when a valve does develop a leak, it is not simply repacked again, but investigated to determine the cause of the leak.. It means that when investigating valve packing leaks the cause is not believed to be the packing "just wore out".

To make this statement that packing does not leak, assumes that adequate radial load on the packing is maintained. Too frequently when AOV or MOV stroking problems are encountered the immediate solution is to loosen up the valve packing. Though never easy the solution to typical valve stroking problems must be a good compromise of all valve parameters, an increase in torque switch setting, adjustment of the bench set of an AOV, reduced packing loads with more frequent retorquing should all be considered.

WHAT ABOUT VALVE PACKING LOADS?

Initially as valve packing programs were developing, there existed no way to accurately measure valve packing loads in the field, so except for laboratory data, little attention was paid to running loads. When actual valve stroking became a problem estimates of packing loads could be determined by performing handwheel torque readings on motor operated valves or by calculations of air pressure and diaphragm size on air operated valves. Even when data was obtained it was difficult to utilize it when no reasonable calculation of expected running loads were available except for the historical 1000 lb./in.

In the early 1990's motor operated valves (MOV) issues and Generic Letter 89-10 drew attention to all aspects of the sizing of forces associated with a motor operated valve. The historical concept of 1000 lb. per inch of stem was used for most

initial MOV sizing calculations. As soon as diagnostic testing of MOV's began to measure actual packing loads the over simplicity of the 1000 lb./in formula appeared. The most obvious factor missing for 1000 lb. formula was consideration of how tight the packing nuts were tightened. Other factors which effect packing loads are not considered in this formula, such as:

number of packing rings
dimensions of packing rings
type of packing
valve position
stem finish
packing is live loaded
valve type
type of valve motion
(rising, rising/rotating)

The conflict between the requirement to maintain MOV operability and to maintain adequate radial load on packing to prevent leakage caused increased attention as never before to be placed on a better calculation of packing loads. Based on the successful valve packing performance at the time, to simply reduce packing loads arbitrarily to comply with 89-10 needs was not the correct solution

At this time, the foundation was laid for developing a better formula. The more programmatic valve packing programs were already operating under a standard uniform controlled process which maintain configuration of installed packing, detailed specific torque values for packing gland nuts, and utilized trained personnel for consistent installation. Laboratory data and initial field data in support of 89-10 appeared on initial review to show a consistency and relationship to basic valve packing parameters which would support some type of calculational model.

Working closely with our packing supplier we provided him not only available field data but performed various mockup tests at varying packing loads and packing configurations to provide additional data. Based on the data that other utilities and we were able to supply, a packing load prediction model for his material was developed. Refer to Attachment A for their packing load prediction formula.

Attention was concentrated on developing a formula to predict the **maximum** packing load for which a valve operator would be sized. These formulas are inherently conservative on the high side. These formulas tend to predict a higher packing load then would normally be seen, not the expected loads.

Once a formula was available it became a standard practice to compare acquired data to the formula. The reviewing of this data forced one to look harder at the specifics under each test. This review pointed out the importance of the need to pack valves consistently and the impact of proper consolidation. An obvious critical point which quickly became apparent was that if diagnostic data was to capture the highest running loads, it must be taken in conjunction with a retorque of the packing.

MOV User's Group Guidelines

Based on the supporting data acquired and the formulas developed, the MOV User's Group in the summer of 1995 established guidelines for Post Maintenance Testing of MOV's due to valve packing work. The guidelines accepted the assertion that "by controlling the method in which valves are packed, it becomes possible to predict packing friction within certain limits." The guidelines were developed to:

> "provide an improved method for predicting packing friction that takes into consideration all the variables that effect

the amount of packing friction generated against the valve stem"

"provide an approach that outlines the information and procedures needed to address post maintenance testing requirements following a repack or packing adjustment."

"issue guidance on the necessary controls when replacing packings to improve packing performance and predictability of packing friction."

PREDICTION OF VALVE PACKING PERFORMANCE

The establishment and acceptance of an upper limit for valve packing loads does little if anything to improve valve packing performance, since packing leaks are caused by inadequate radial sealing loads. Typical problems which could occur are:

galled packing stud
packing gland stuck in the stuffing box
incorrect packing gland torque used
improper packing consolidation
inadequate packing load

To identify potential packing problems requires a minimum acceptance goal. To evaluate if a minimum packing load could be determined two unique reviews of our data of over 200 MOV diagnostic tests was performed. The first review identified any valve which over the past several outages had develop any type of leak no matter how minor (i.e. 1 dpm during hydro) and reviewed the measured running loads in comparison to the upper limit and to other similar valves. The second review imposed a draft lower limit on all data to determine if a good estimate could be developed which would identify all indicated problem valves and would impact few other valves.

An appropriate value which fit very well in the above process was to utilize 40% of Argo's calculated predicted value. On first review this value may seem low, but as noted above, these upper limit values were developed by Argo, not to accurately predict normal packing loads, but to establish a maximum value for packing loads which would rarely if ever be exceed to support MOV sizing and operability calculations. In Appendix B, an case history of how this criteria was utilized to identify a potential valve packing problem is presented.

Once the maximum and minimum limits for packing load have been established it is necessary to implement a packing load monitoring effort in a way which best fits with the sites established methods. The personnel typically handling both MOV and AOV diagnostic equipment which measure the valve running loads have little or no experience or knowledge of valve packing. In the future we plan to provide training to these personnel, so they can provide the first actions to resolve potential problems when they arise. For now, we simply established our limits and requested notification if any readings which were outside these bounds were found. In the last 2 outages, few problems were found but those which were found were legitimate and the work delays incurred in resolution were worth the effort for the future problems they resolved.

FINAL EFFECTIVE PACKING PROGRAM

To establish an effective valve packing program requires the same effort as would be instituted in any area of the power plant where the opportunity for plant improvement and cost savings exists. To be successful requires a long term

commitment to provide adequate resources and funding. Accountability must be clearly assigned and accepted. An implementation plan needs to be develop which clearly identifies the objectives, the path to reaching the objectives and the schedule. As with most maintenance components the 80/20 rule applies to valve packing. 20% of the valves will require 80% of the resources. The program needs to aggressively attack first that 20% of the valves that is incurring the most cost. Typically the troublesome packing areas are associated with AOV and valves on high temperature systems. Numerous papers and books have been published on the general issues associated with establishing a successful program.

The basic intent of this presentation is to point out two specific areas which may not typically be identified when a committed, programmatic approach to valve packing is taken. The misconceptions presented though obvious, can subtly impact a program and need to be aggressively pursued through training, procedures and technical field support. The factoring in of diagnostic MOV and AOV data which is typically acquired and utilized by groups unrelated to valve packing, can be one of the stronger predictive tools available to a utility.

Efforts which can be directed at each of the misconceptions discussed in this paper are summarized below:

VALVES DO NOT REQUIRE PERIODIC REPACKING

The benefit of repacking valves under a controlled, programmatic approach are long term. The man-hours and radiation dose for valve packing do not have to be a continuing expense.

Insure the valve is repacked right the first time.

A little extra cost incurred in technical field support, personal training, or material expense will be off set within a few years.

To eliminate periodic valve repacking, a program must be implemented to insure that radial sealing is maintained via periodic retorque and/or predictive measurements of packing loads.

VALVE STEM SEAL OCCURS WITH ADEQUATE RADIAL LOAD

The least number of packing rings provides the best axial to radial load transfer.

Packing consolidation via retorquing and stroking are key to insuring uniform radial loading of the entire valve packing set.

Attention to detail is necessary to insure axial load is transmitted to the packing (proper fitting glands, clean and lubed packing studs) Trained, committed personnel are required.

Controls must be placed over gland loads, specific values must be given to maintain consistent performance, this area cannot be left to experience or skill level of the worker (unless trained to calculate packing gland loads).

PROPERLY REPACKED VALVES WILL NOT LEAK

Mechanic's expectations need to be developed, that they do not expect or accept packing leaks.

Any packing leak of a repacked valve should be investigated to determine cause.

Maximum gland loads should be used, do not reduce loads to allow for margin when the valve starts leaking.

Eliminate stem leakoff systems, they serve no useful function and hinder the proper radial loading of the valve packing.

VALVE PACKING LOADS ARE PREDICTABLE

Packing loads are predictable when packing is installed under a controlled program with skilled personnel.

Techniques exist to adequately measure packing loads on MOV's and AOV's.

Measured packing loads can be utilized:

- to predict future packing performance
    - determine future frequency for retorquing valve packing
    - trend any packing degradation
    - identify any potential valve packing failures

- confirm proper installation and obtainment of proper radial load
- confirm personnel training and technique
- confirm proper packing consolidation

Confirm valve packing program controls to allow for reduced expensive Post Maintenance Testing.

To be effective, the valve packing program must stress these conceptions of valve packing throughout the organization and establish a commitment to maintaining and improve valve packing performance.

Beyond commitment, an effective valve packing program must have:

- trained personnel
- established program of periodic valve packing retorques
- field feedback from results of valve repacks and retorques
- feedback from MOV and AOV diagnostic testing.

COST BENEFITS OF EFFECTIVE VALVE PACKING PROGRAM

Recent estimates have found that the impact of an effective valve packing program can save a utility $300,000 to $500,000 annually. Though not discussed in this presentation, experience has indicated that with the training and dedication which is inherent from a committed valve packing program, the man-hours per repack can be expected to decrease by at least 25-50 %. This generic estimate is based on the following assumed annual valve packing activities which have typically been seen at nuclear power plants which do not have a formal valve packing program.

- 150-300 valves are repacked each outage based on a combination of corrective repacks of valves with known packing leaks, repetitive repacks based on past practice, and typically an additional scope of valves selected by a planner or maintenance engineer to attempt to "get control" of valve packing.

- Any work involving valve packing (retorque or repacking) of safety related AOV's or MOV's require a full diagnostic test.

- 2-6 "leak repairs" are paid for each cycle to stop valve packing leaks.

ESTIMATED ANNUAL COST SAVINGS

elimination periodic valve repacking (25-50 repacks)	$50,000
reduction in corrective valve repacking	$75,000
elimination of 3-5 "leak repairs"	$10,000
improved valve repacking efficiency from 20 man-hours to 10	$85,000
elimination of 20 MOV diagnostic PMT due to valve packing	$40,000
elimination of 15 AOV diagnostic PMT due to valve packing	$30,000
eliminate 50% of AOV stroking problems	$50,000

CONCLUSION

Though the ASME code views valve packing as outside the scope of the pressure retaining components, it can have significant impact on plant operation and valve operability. The misconceptions of valve packing performance have hindered the resolution of many valve packing problems. By instituting an aggressive, effective valve packing program whose goal is to maintain "leak free" service, substantial annual O&M savings can be obtained. Valve packing loads can be appropriately estimated. By tying valve packing with current efforts on AOV and MOV diagnostic, valuable predictions of valve packing performance are provided without additional cost or impact.

APPENDIX A

VALVE PACKING LOAD PREDICTION FORMULA

PACKING LOAD = 3.1415 x Sg x F x Y x Ds x H x Lf

Sg	= Compressive Stress on the Packing (psi)
F	= Coefficient of friction
Y	= Ratio of Axial to Radial Stress in the Packing
Ds	= Outside diameter of the valve stem
H	= Packing height (uncompressed) in inches
Lf	= Live load factor

TRANSFER RATIO (Y)

Stress	1 to 6 Rings Ratio	7 to 9 Rings Ratio
3000 psi	.65	.55
3,500	.75	.65
4,000	.85	.75
5,000	.85	.85
>5,000	.85	.85

LUBRICATION FACTOR

PACKING TYPE	STEM FINISH	
Composite	(0-32 RMS)	.05
Composite	(33-50 RMS)	.07
Yarn/graf	(0-32 RMS)	.10
Yarn/graf	(33-52 RMS)	.15

LIVE-LOADING FACTOR (LF)

Live-loaded	1.0
Conventional	.75

APPENDIX B
USE OF MOV DIAGNOSTICS TO MONITOR VALVE PACKING PERFORMANCE

A specific example which utilized MOV diagnostic data to predict valve packing degradation is presented below. By comparing previous MOV diagnostic tests (tests 14 & 20) for the same valve, with current data being taken (tests 1, 2, & 5) the following indication was found. A significant decrease in packing load was seen in both the open and close direction. As a standard work practice the packing was retorqued prior to any diagnostic test. Experience indicated that packing loads would probably increase slightly but could remain the same or drop 100 pounds at most.

The drop measured below of over 900 pounds, could not be explained. The valve was repacked and the carbon bushing was found to be cracking causing the packing load to be released. No similar problems with carbon bushings had been experience prior to this or since.

The post repack diagnostic testing (test 9 & 10) demonstrated the expected packing loads indicating proper stem sealing loads. Without this data review, this valve would have either leaked during the OPS hydro causing potential refuel outage delay or started to leak while in service which would have caused the unit to shutdown.

HV141F016 is a 3", 900#, class flex-wedge, gate valve, which provides containment isolation of the Main Steamline line.

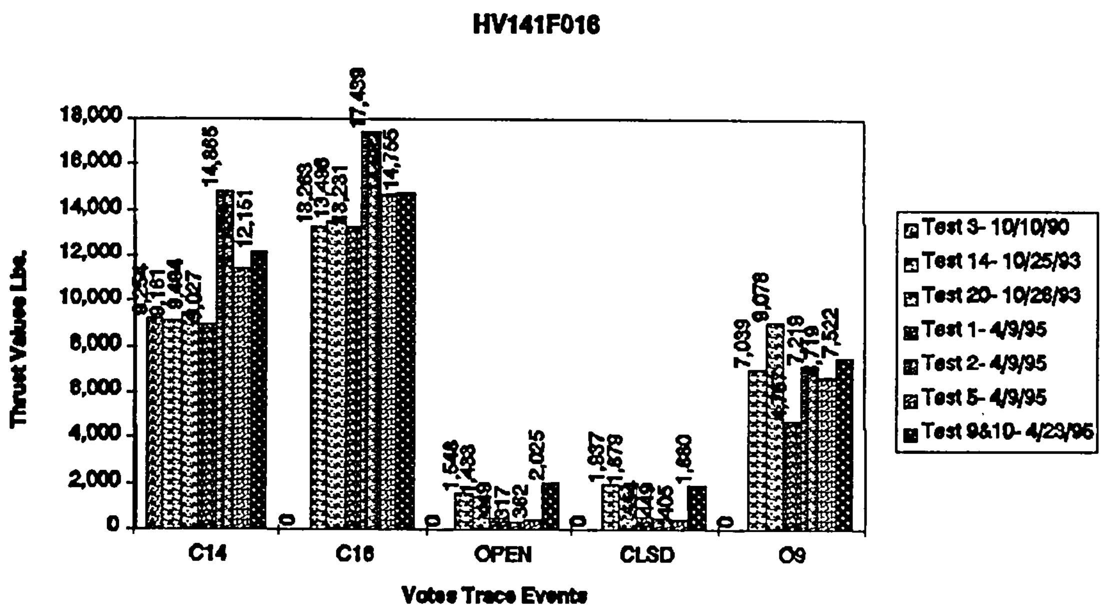

Figure 5. COMPARISON OF MOV DIAGNOSTIC DATA

DIAPHRAGMS IN AIR-OPERATED VALVES

Joseph E. Groeger
Altran Materials Engineering, Inc.

ABSTRACT

The author will present current issues related to diaphgrams in air-operated valves. Altran Materials Engineering, Inc., often performs root-cause analyses for nuclear power plant owners. The author will discuss various analyses that have been performed or are currently underway.

MOTOR DEGRADATION PREDICTION METHODS

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Commonwealth Edison, Quad Cities Station

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ABSTRACT

Motor Operated Valve (MOV) squirrel cage AC motor rotors are susceptible to degradation under certain conditions. Premature failure can result due to high humidity/temperature environments, high running load conditions, extended periods at locked rotor conditions (i.e. > 15 seconds) or exceeding the motor's duty cycle by frequent starts or multiple valve stroking. Exposure to high heat and moisture due to packing leaks, pressure seal ring leakage or other causes can significantly accelerate the degradation. ComEd and Liberty Technologies have worked together to provide and validate a non-intrusive method using motor power diagnostics to evaluate MOV rotor condition and predict failure. These techniques have provided a quick, low radiation dose method to evaluate inaccessible motors, identify degradation and allow scheduled replacement of motors prior to catastrophic failures.

I. INTRODUCTION

Motor failure due to rotor bar/shorting ring separation, cracking or melting has caused MOV failure(s) to operate as designed. This type of failure can occur due to repeated operating cycles, galvanic corrosion, thermal overload sizing inadequacies, repeated resetting of thermal overload relays, high operating loads or moisture intrusion combined with heat.

In the mid 1980s, a nuclear generating station (Reference 1) experienced a number of motor operated valve (MOV) failures due to improperly sized thermal overload relays (TOLs). Typical sizing of thermal overload relay heaters will result in 8 to 15 seconds of sustained locked rotor current before control power is interrupted. IEEE recommends thermal overload sizing to allow 10 seconds or less at locked rotor conditions (Reference 2). There was a possibility that some of the motors had been originally sized and thermal overload relay heaters set to provide 20-40 seconds of locked rotor prior to TOL trip (Reference 1). These settings caused severe MOV rotor damage and in some cases catastrophic failure of magnesium or aluminum rotor MOV motors after a single TOL trip. Both white oxide (magnesium hydroxide) and dark oxide (black or gray/magnesium oxide) was identified during their inspections.

The failure mechanism was found to be rotor bar cracking, sometimes accompanied by evaporation of rotor shorting ring material and separation of rotor core laminations. The rotor material most often found damaged was identified as AM100A, which is approximately 90% magnesium and 10% aluminum. The nuclear generating station initiated a periodic inspection of these motors using a borescope. Due to accessibility, many of these inspections were scheduled for outages, resulting in a significant manpower and dose impact due to having to remove/reinstall the motors or perform the bore scope inspections in place.

Similar problems during Equipment Qualification (EQ) testing in the late 1980s identified that magnesium motors would fail in as few as 4 days under post Loss of Coolant Accident conditions. It has been postulated that motors in normal ambient high temperature and high humidity environments over longer periods of time will have a similar effect.

ComEd's Quad Cities station experienced similar MOV shorting ring rotor failure in 1989. Their investigation concluded that failure due to environmental conditions, i.e. high temperature and humidity resulting in chronic degradation and subsequent failure to operate after a refueling outage (during startup). The motor had extensive corrosion at the iron core stack lamination interface with the magnesium end (shorting) ring.

A white powder covered the magnesium portions of the motor that was found to be magnesium hydroxide. The galvanic potential between magnesium and iron (1.9 volts), combined with the brittle

structure of magnesium, accelerates the corrosion. This effect is exacerbated by the differences in thermal coefficients of expansion and the axial pressure on the shorting ring/rotor bar joint generated from the corrosion by products.

To evaluate motor condition, ComEd attempted to perform a few preventive maintenance techniques, such as borescope inspections (results often inconclusive, some motors not equipped with removable plugs), without much success. Existing motor rotor analysis systems required significantly longer run times to perform evaluations than is typical of a MOV motor (normally 60 seconds or less). These systems were primarily designed for continuous duty motors. Motor removals and inspections were time consuming, required further post maintenance testing and added to radiation dose burdens at the station (the highest heat/humidity areas were inside containment).

ComEd identified that two magnesium alloys were used for the 180 frame size and larger MOV motors: Dow "M" (99% magnesium, 1% manganese) and Dow "G" (90% magnesium, 10% Aluminum). The Dow "M" appeared to be most susceptible to galvanic corrosion, while both alloys would fail if heated above 490°C (914°F). This condition can be reached by motor stall for 15 seconds or greater (Reference 1).

REG Guide 1.106 requires that safety related motor operated valves be thermally protected in such a manner that the motor is given every opportunity to perform it's safety function. In some plants the TOL is bypassed under certain conditions to assure this requirement is met. This sets the stage for further rotor degradation if a locked rotor condition occurs.

ComEd, in pursuit of a non-intrusive method of identification of rotor degradation, worked with Liberty Technologies to validate the use of motor power monitoring equipment analysis techniques from the Motor Control Center (MCC) to assess the condition of the MOV motors.

II. Results

ComEd has tested over 40 motors using this diagnostic technique (including new replacement motors). The motor power evaluation determined three motors to be degraded and subsequent disassembly and inspection substantiated the methodology.

A fourth motor also identified as severely degraded suffered shorting ring separation during disassembly. See figures below. A fifth motor failed during motor diagnostic testing (shorting ring separation). Subsequent review of test results indicated imminent failure.

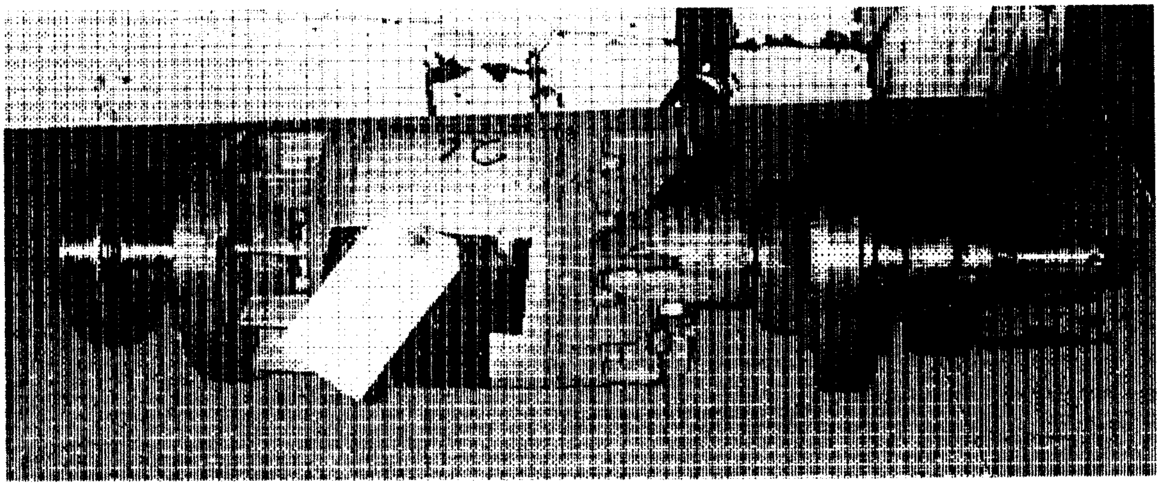

Figure II-1: Full view of the rotor assembly of a degraded motor.

Figure II-2: Enlarged view of the rotor assembly of a degraded motor.

Figure II-3: Close-up view of the broken rotor bars.

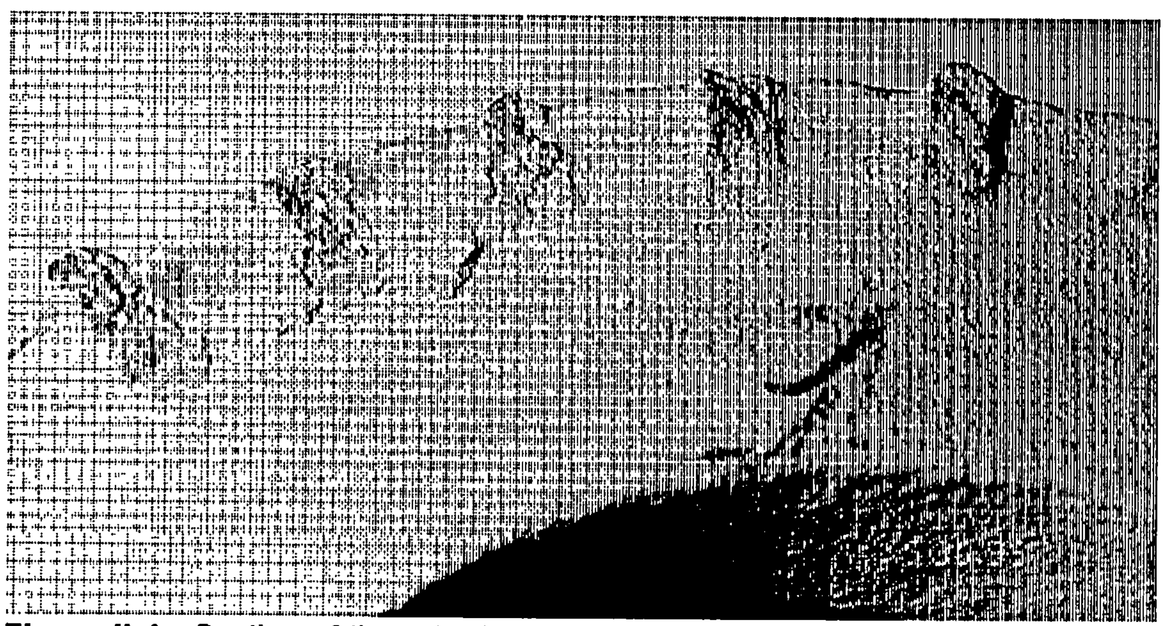

Figure II-4: Section of the rotor illustrating degradation at the end ring connection to rotor bars. This may result in end ring separation.

III. MOTOR POWER MONITOR (MPM)

MPM uses a technique called power signature analysis (PSA) to detect the behavior of electric motors and motor-driven machines, under normal or abnormal loading conditions. By attaching probes to motor feed lines (Figure III-1), electric current and voltage signals are monitored on-line and in real time. There is no need to interrupt a running motor to use MPM.

Sensed signals are conditioned in the Signal Conditioning Unit, and digitized in the portable computer. Test data files are stored in the computer's hard disk and subsequently retrieved and analyzed using MPM applications software tools to plot key variables such as: total real power (TRP), total reactive power (TXP), and total power factor (TPF).

In addition to PSA techniques, MPM can also capture non-power data via spare channels. Strain-gage sensors, accelerometers, thermistors, etc., can be used to detect physical variables such as: temperature, flow, and pressure. With this capability MPM can observe the behavior of machines using established techniques like vibration signature analysis (VSA) and newer techniques such as PSA.

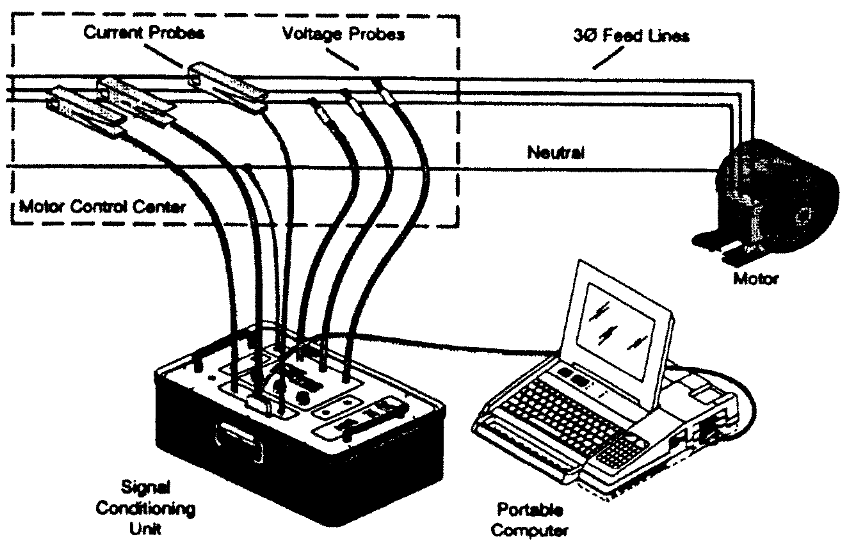

Figure III-1

IV. MOTOR INTEGRITY INSPECTION TECHNIQUES

A. Slip and Pole Pass Frequency

Theory

In the induction motor, an alternating current is supplied to the stator directly and to the rotor by induction or transformer action from the stator. When excited from a balanced polyphase source, it will produce a magnetic field in the air gap rotating at synchronous speed (h) as determined by the number of poles *P* and applied stator frequency *f*.

$$h_1 = \frac{120 \times f}{P}$$

Let us assume that the rotor is turning at a steady speed h in the same direction as the rotating stator field. Let the synchronous speed of the stator field be h_1 (as given by the above equation). The rotor will rotate backwards at a speed (h_1- h) with respect to the stator field, or the *slip* of the rotor is (h_1- h). Slip is usually expressed as a nondimensional quantity, as a fraction of synchronous speed with limits of zero and one. The per unit slip *s* is:

$$s = \frac{h_1 - h}{h_1}$$

Slip is also expressed in percent, where a slip of 100 is equivalent to a per unit slip of unity (1.0).

This relative motion of the stator flux and the rotor conductors induces voltages of frequency *sf*, called the *slip frequency*, in the rotor. The operating speed (h) of an induction motor can never equal the synchronous speed (h_1).

For MOV motors, the rotor "windings" are constructed by forcing molten metal (an injection molding process) into axial cavities that run the length of the rotor, in essence forming the rotor bars and end rings simultaneously. The rotor bars are short-circuited together with the cast aluminum end rings at each end of the rotor. This type of rotor construction results in induction motors which are relatively inexpensive and highly reliable. Because of this type of motor construction, cracked or otherwise electrically disconnected rotor bars are an identified area of serious concern. These disruptions of the rotor circuit give rise to torque and speed pulsations and vibrations often mistakenly assigned to other mechanisms such as mass-unbalance or driven machine deficiencies. Cracked bars or end rings give rise to rotating "hot spots", subjecting the rotor to unwarranted thermal stresses and sometimes actually ejecting molten metal. These high resistance rotating circuits can force new and unintended current paths to be taken through the rotating core laminations.

The pole pass frequency is defined as the number of poles *P* times the slip frequency *sf*.

$$ppf = sf \times P$$

It defines the number of poles passed by the rotating synchronous magnetic field per second. Rotor degradation will result in higher energy levels (decibels) at the pole pass frequency. The pole pass frequency can then be used to assess rotor integrity by evaluating the magnitude of the energy at synchronous speed less the pole pass frequency. The energy magnitude is determined using a standard fast fourier transform (FFT) during the running region.

The effectiveness of this method is a function of motor load. As motor load is increased the slip frequency increases, increasing the pole pass frequency. The resolution of the spectrum increases with increasing load. The motor should be loaded to at least 60% - 70%.

MPM Application (Method 1)

This method is most effective when the motor is heavily loaded, however can detect significant damage at any motor load. At motor loads less than 60-70% of the rated torque, results in the 35dB to 50dB level may not be indicative of rotor degradation. In these cases, Method 2, Motor Current Unbalance Techniques should be applied to verify degradation.

The first step to detecting one or more broken, cracked, or damaged rotor bars using MPM is to find the pole pass frequency. As defined above, the pole pass frequency (*ppf*) is a function of motor speed and by the number of poles. To find the *ppf*, perform a FFT on the reactive power signature. The pole pass frequency is usually the first large peak on the reactive power spectrum as illustrated in Figure IV-1. In this example the motor is a (202-ft-lb, 3600 rpm (2 pole) induction motor).

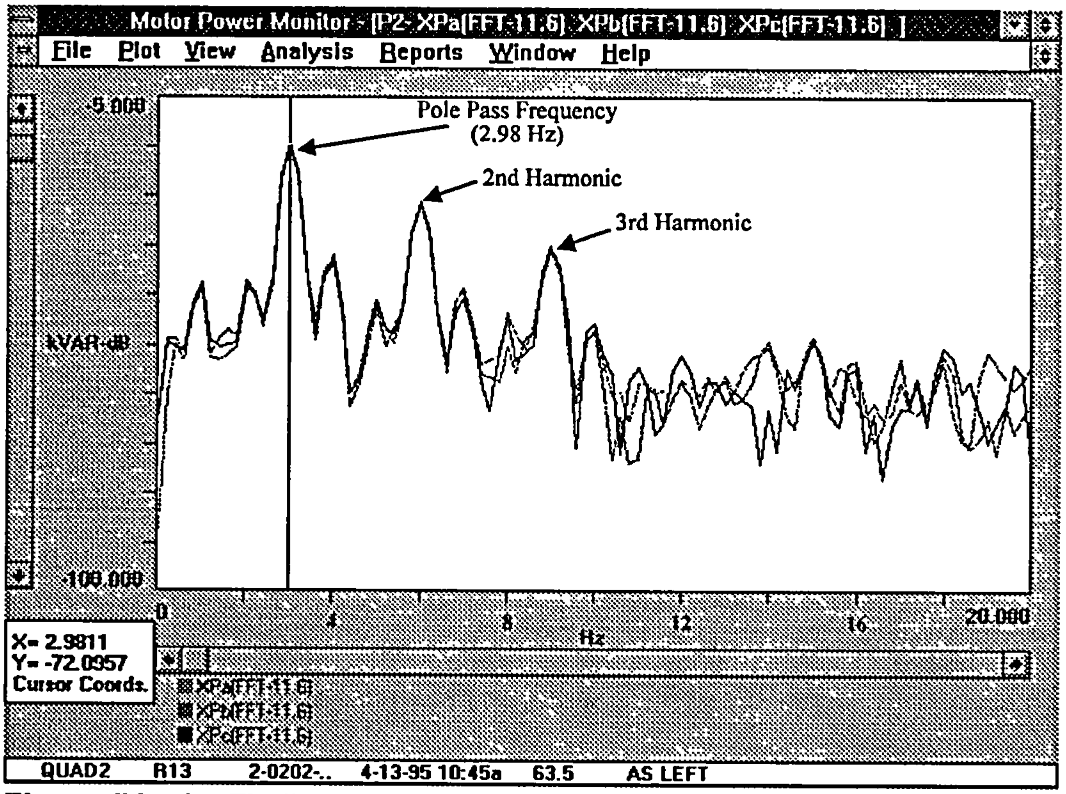

Figure IV - 1: Finding the Pole Pass Frequency. The pole pass frequency is the first large peak (2.98 Hz in this case). Note that the second and third harmonics are visible, which usually indicates a rotor problem.

Normal induction motors operate in ranges of 2-10% slip (*s*), or 0.02-0.10. For a line frequency of 60Hz and knowing the number of poles on the motor, this gives the user a good place to start for evaluating the *ppf.*

For a 3600 RPM motor (2 poles), *ppf* ranges from 2.4 Hz to 12 Hz.
For a 1800 RPM motor (4 poles), *ppf* ranges from 4.8 Hz to 24 Hz.
For a 1200 RPM motor (6 poles), *ppf* ranges from 7.2 Hz to 36 Hz
For a 900 RPM motor (8 poles), *ppf* ranges from 9.6 Hz to 48 Hz

The second step is to perform a standard FFT on the raw motor current in the same time region as the reactive power. Next look for the pole pass modulation frequency which is modulating on the 60 Hz carrier frequency.

Pole Pass Modulation Frequency	=	60 Hz - *ppf*
	=	60 Hz - 2.98 Hz
	=	57.02 Hz

Next, compare the dB level down from the line frequency (60 Hz) to the pole pass modulation frequency (see Figure IV-2).

Once the dB level has been determined, the value can be compared to the following chart (Reference 11) to assess the condition of the rotor bars. In this particular case, the dB level was 27, which is the most severe case with multiple broken rotor bars likely.

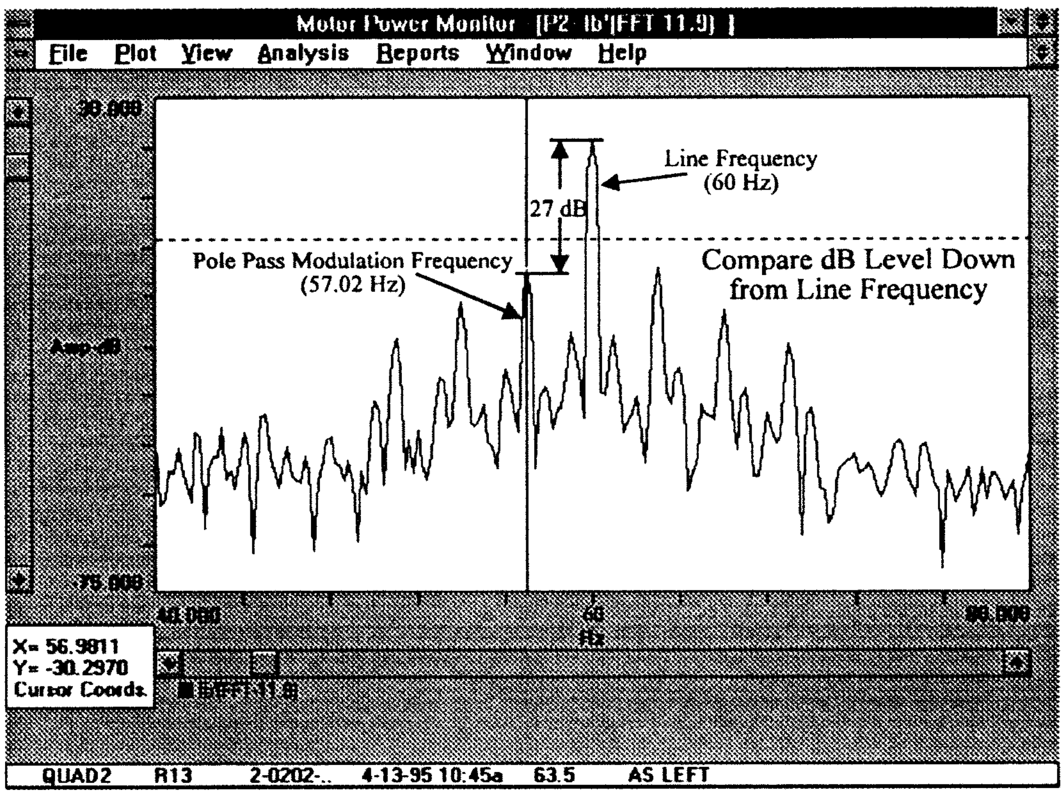

Figure IV - 2: Compare dB level. To find change in dB level, measure the difference from the line frequency to the pole pass modulation frequency. Note the measurement is absolute (i.e. |20| + |-7| = 27 dB).

Chart 1: Condition Assessment of Rotor Bars

60 dB or more	=	Excellent
54-60 dB	=	Good
48-54 dB	=	Moderate
42-48 dB	=	Rotor Bar Crack Developing or High Resistance Joints
36-42 dB	=	2 Bars Likely Cracked/Broken; High Resistance Joints Likely
30-36 dB	=	Multiple Cracked/Broken Bars or End-Rings Indicated
less than 30 dB	=	Multiple Cracked/Broken Bars or End-Rings Very Likely; Severe Problems Throughout

When the ComEd motor was replaced, this procedure was repeated to compare the difference in dB level. As seen in Figure VI-3, the dB level was improved dramatically to 52 dB, which indicates a moderate rotor bar condition.

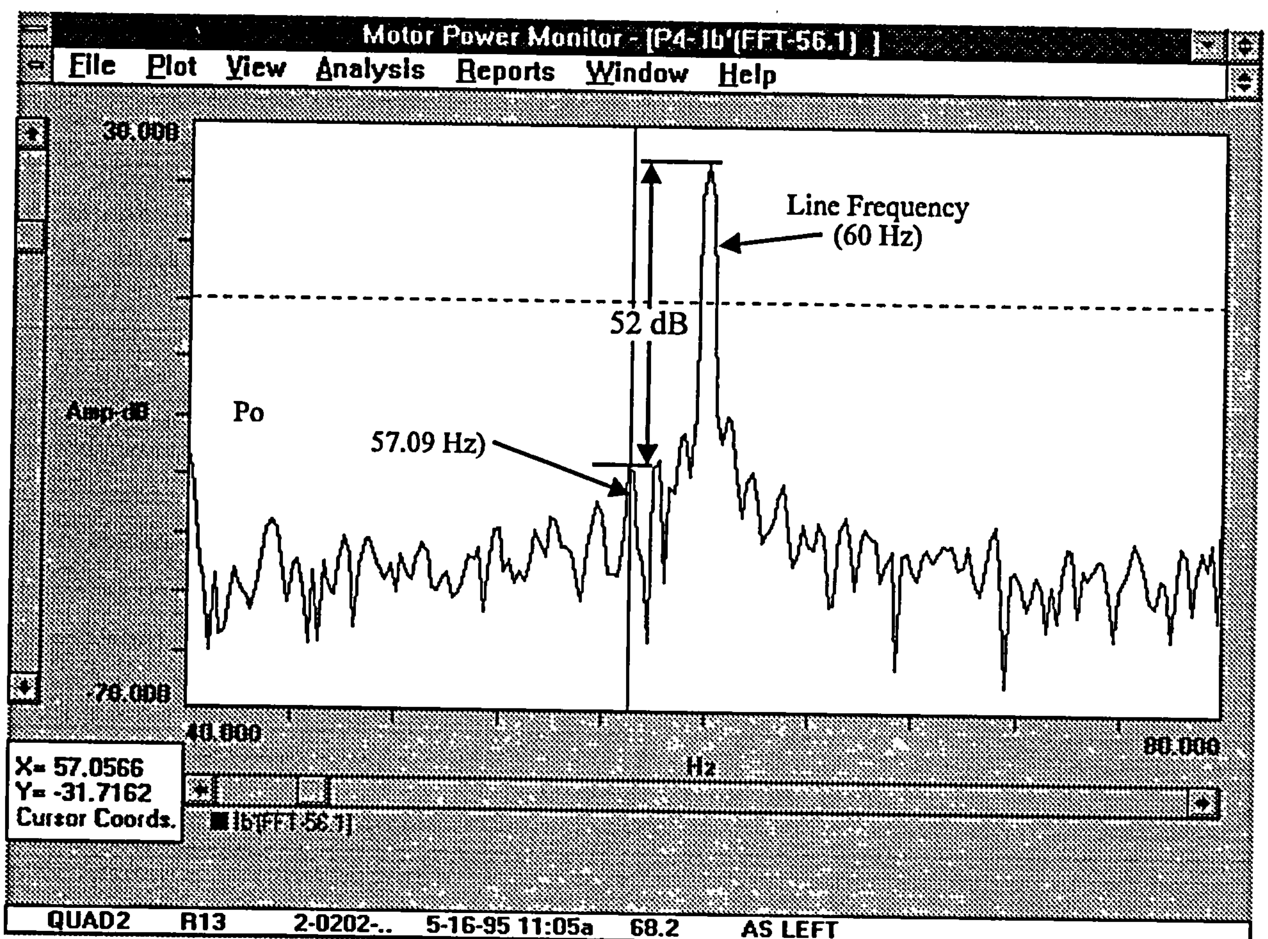

Figure IV - 3: Replacement Motor. Note the large improvement in dB level.

B. Current Unbalance

Theory

For the normal operation of an induction motor, an alternating current is supplied to the stator directly and to the rotor by induction or transformer action from the stator. This current is moved from rotor bar to rotor bar as the rotor "back rotates" relative to the stators rotating field, due to *slip* (caused by the load). This will cause the rotor bars to slip counterclockwise on each rotation of the rotor (with reference to the stator rotation). Figure IV-4 is a cross-section of a typical Stator/Rotor combination. For simplicity sake, the motor has 36 rotor bars and a synchronous speed of 3600 RPM. Also, lets assume the rotor is experiencing 1% slip.

As shown in Figure IV-4, when the broken bar is away from the pole, it's effect on the current flow in the stator is nil. As the rotor continues to back rotate on each revolution (in this example 3.60°/Revolution), eventually the broken bar will lie directly under the pole (see Figure IV-5). In this example, it requires 25 revolutions of the rotor for the bar to lie directly under the North pole. When the broken rotor bar has rotated to this location, the current in the stator modulates because of an interruption in the current flow of the rotor. The motor must still drive the torsional load, therefore the adjacent bars (±3.60°) will carry the required torque producing current. This phenomenon is accentuated when the number of broken bars increases.

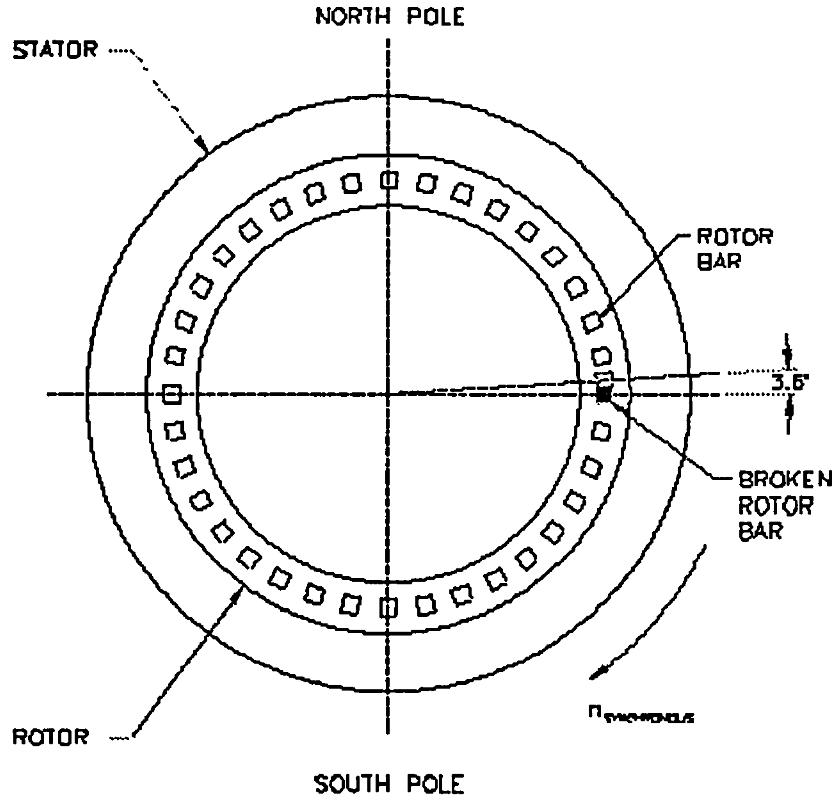

Figure IV-4: Cross-section of a typical Stator/Rotor combination

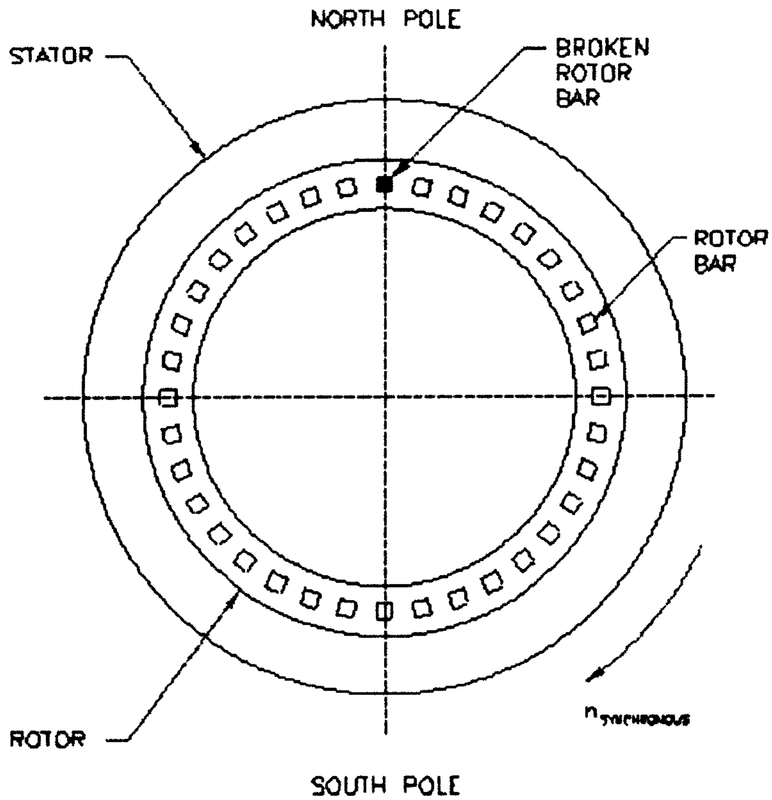

Figure IV-5: Broken rotor bar has moved 90° due to slip.

Utilizing the MPM applications software, one signatures that has been used to confirm the existence of degraded/broken rotor bars is the Current Unbalance Signature (I-Unb). This signature is calculated using the following equation (Reference 9 and 10):

$$\%Unbalance = 100 \times \left[\frac{max(|I_a - I_{mean}|, |I_b - I\text{-}mean|, |I_c - I_{mean}|)}{I_{mean}} \right]$$

In essence, the equation calculates the instantaneous maximum deviation per phase from the instantaneous average. The current values used in the equation are the true RMS currents. IEEE also references other equations (Reference 10) to measure unbalance, but Liberty feels this equation is the most effective. The time domain waveform is displayed in percent unbalance. The engineers from both ComEd and Liberty have used the shape of this signature to confirm the existence of broken rotor bars.

As the rotor degrades, the magnetic field induced by each rotor changes. The changing magnetic field induced by the rotor/pole combination is captured and plotted in the current unbalance signature for each revolution. Lack of symmetry and large fluctuations in the %unbalance signature is a good indicator of broken rotor bars. This assists the engineer in confirming the results of the frequency technique.

MPM Application (Method 2)

Figure IV-6 displays 30 revolutions of an MOV motor known to be bad. Notice the large fluctuations in the % Unbalance and an actual "beating of the signature through the 30 revolutions.

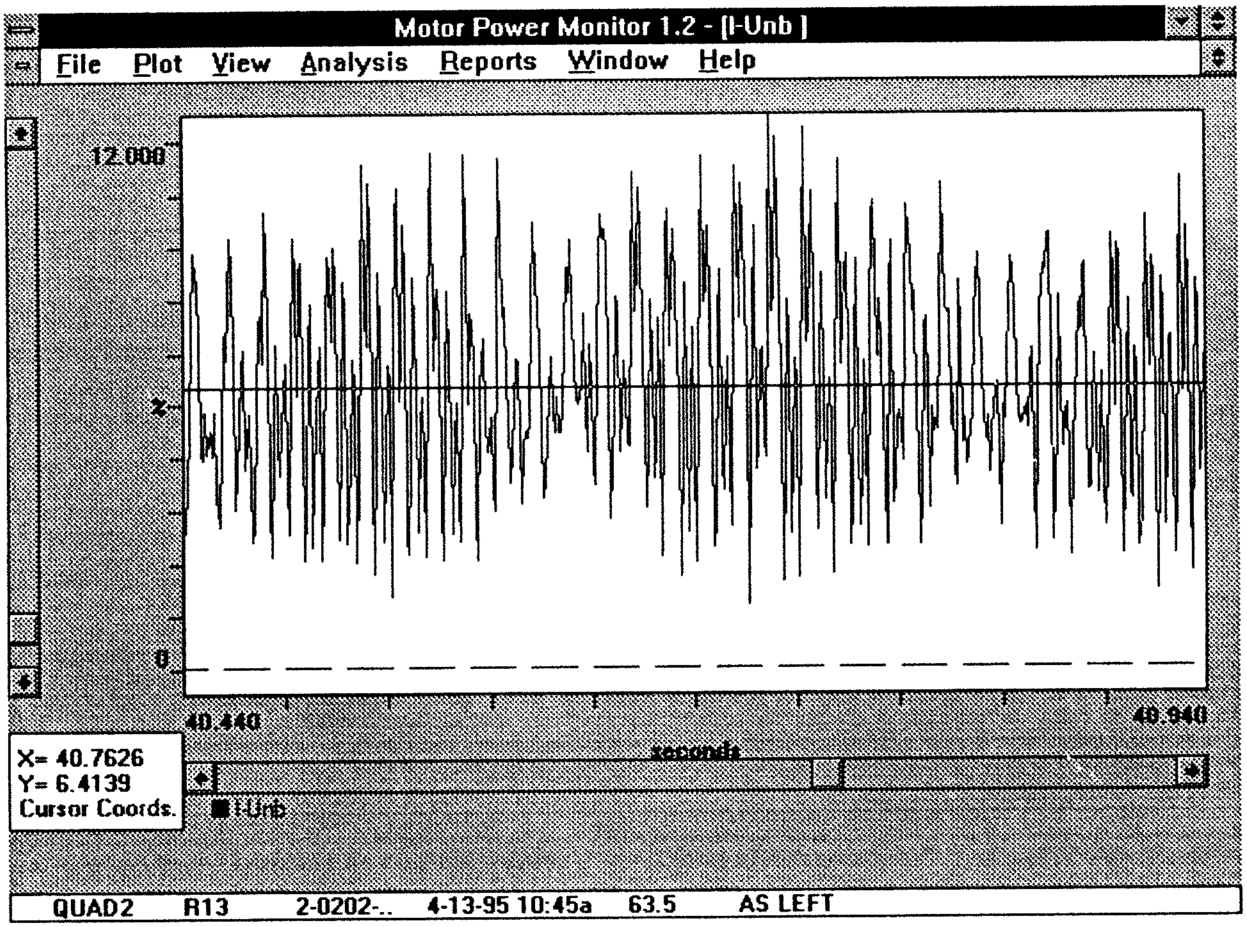

Figure IV-6: Thirty revolutions of an MOV motor. Found to have 13 out of 31 broken Rotor Bars. Note beating of signature and large fluctuations in the %Unbalance.

Figure IV-7 is the same MOV with a replacement motor installed. Again, the signature displays 30 revolutions. Notice this signature does not "beat" like it's counterpart and their are no large fluctuations in the %Unbalance. The modulation seen in this signature is typical of MOV actuator torque/power requirements for overcoming normal running conditions (i.e. packing loads).

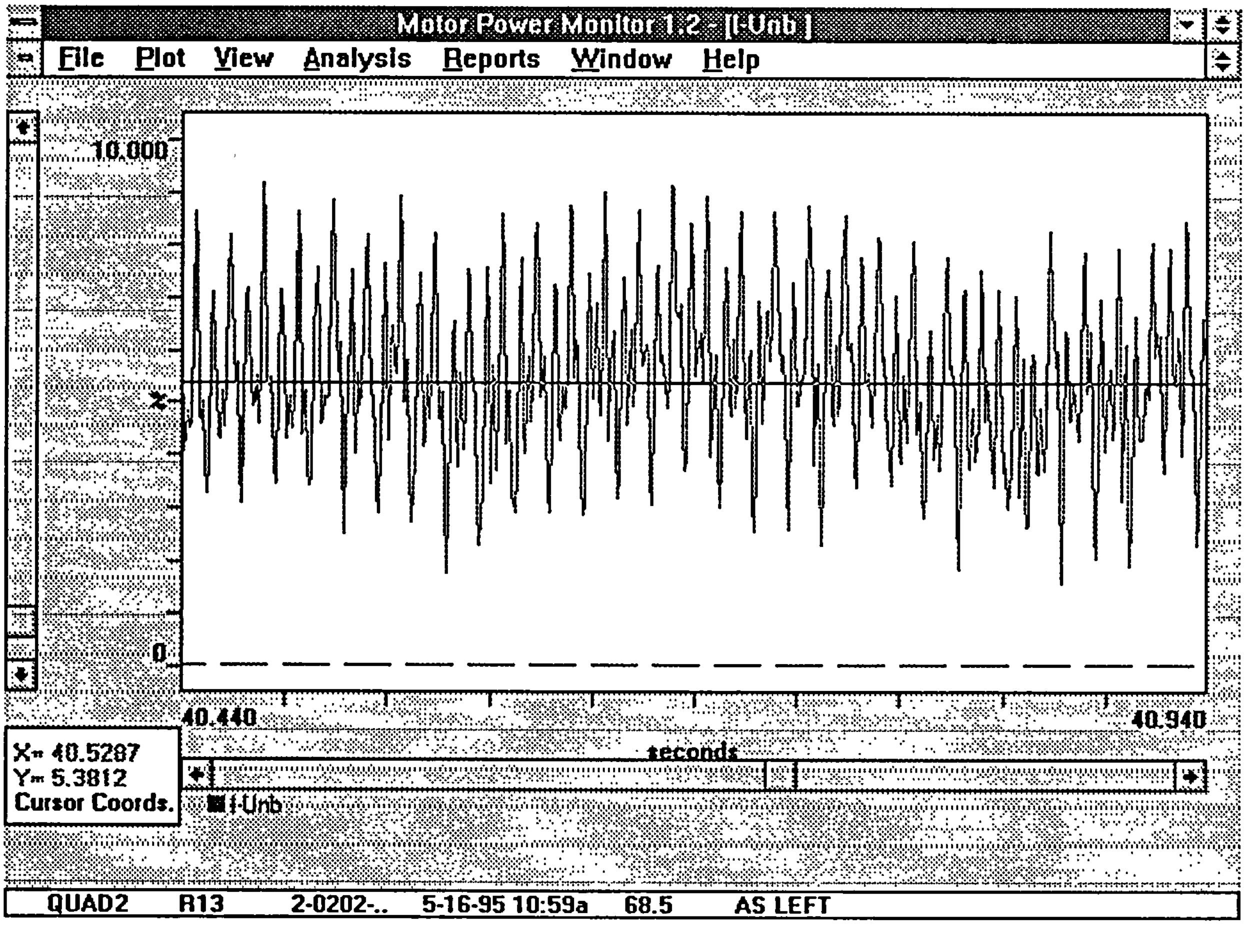

Figure IV-7: Thirty Revolutions of Replacement MOV Motor. Note the signature does not beat, no large fluctuations on the %Unbalance.

Figure IV-8 displays an overlay of both the damaged and replacement motors for 5 complete revolutions. One can observe the symmetry in the Red signature indicating the lack of degradation in the rotor as opposed to the "double hits" in the blue signature. The motor has a nameplate synchronous speed of 3600 RPM (i.e., 2 pole induction motor). The current unbalance trace displays each of the poles as the rotating magnetic field passes each rotor pole.

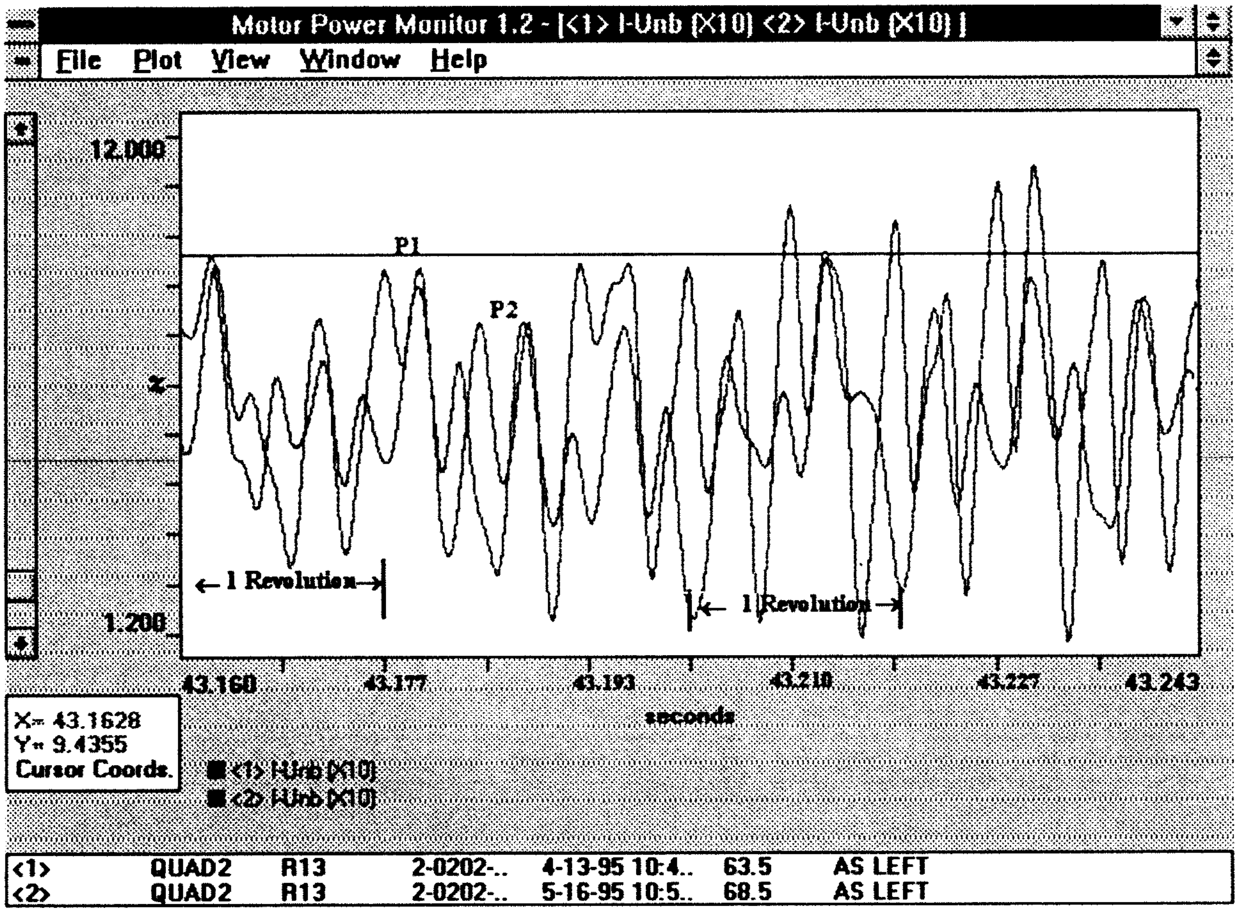

Figure IV-8: Overlay of Current Unbalance signatures for five revolutions of Replacement (Red) and Damaged (Blue) MOV Motor.

The technique of using frequency analysis coupled with the signature analysis of the %Unbalance trace allows the engineer to make a more confident recommendation on the health of the motor.

V. Validation Results via Disassembly and Inspection of Rotors.

The table below summarizes the results of MPM testing and disassembly/visual inspections performed to date. In all of the cases below the running torque at the motor is less than 10% of the rated torque. As a result, the Pole Pass Frequency method is inconclusive in the middle dB ranges (35 to 50 dB). In these cases the motor current imbalance symmetry test is extremely accurate in predicting rotor degradation and identifying degraded rotors requiring replacement.

	Size (ft-lb) speed (rpm) Description	Technique		Post Mortum Disassembly and Inspection Results
		Pole Pass Frequency Noise Level Check Results	Current Imbalance Symmetry Check Results	
1	60-3600 QC1-202-4A	Cracks Developing 46.2dB	Symmetrical	NA
2	60-3600 QC1-202-4B	Cracks Developing 42dB	Symmetrical	No Damage
3	100-3600 QC1-202-5A	Multiple Cracks 40.4dB	Appears to have a slight break from symmetry	Corrosion process starting, visible pitting
4	100-3600 QC1-202-5B	Cracks Developing 52.6dB	Asymmetrical	Cracks and separation of end ring
5	100-3600 QC1-1001-50	Cracks Developing 42.5dB	Symmetrical	No Damage
6	100-3600 QC2-202-5A	Cracks/Severed End Ring 14.1dB	Asymmetrical	Failed during inspection
7	60-3600 QC2-202-4A	Cracks/Severed End Ring 27dB	Asymmetrical	Cracks and separation of end ring
8	60-3600 QC2-202-4B	No Degradation 52dB	Symmetrical	New Motor
9	60-3600 QC1-202-4B	No Degradation 48.4dB	Symmetrical	New Motor
10	60-3600 QC1-202-5A	No Degradation 55dB	Symmetrical	New Motor
11	100-3600 DR2-202-5A	No Degradation 49dB	Symmetrical	New motor
12	100-3600 DR2-202-5B	Cracks Developing 44.2dB	Symmetrical	New Motor

VI. CONCLUSION

Motor Power diagnostics provide an economical method for assessing and trending motor rotor integrity. These methods can predict imminent failures, immediately be utilized to assess effect of locked rotor conditions and evaluate motor capability if motor duty cycle is exceeded during testing or operations. The qualitative evaluation can be used as a go/no-go gauge, the quantitative evaluation can help to assess operability. These methods are simple, effective and can save significant manpower and radiation exposure.

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Replacement of Outboard Main Steam Isolation Valves in a Boiling Water Reactor Plant

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ABSTRACT

Most Boiling Water Reactor plants utilize wye pattern globe valves for main steam isolation valves for both inboard and outboard isolation. These valves have required a high degree of maintenance attention in order to pass the plant local leakage rate testing (LLRT) requirements at each outage. Northern States Power made a decision in 1993 to replace the outboard valves at it's Monticello plant with double disc gate valves. The replacement of the outboard valves was completed during the fall outage in 1994. During the spring outage in April of 1996 the first LLRT testing was performed with excellent results. in is presentation will address the decision process, time requirements and planning necessary to accomplish the task as well as the performance results and cost effectiveness of replacing these components.

INTRODUCTION

Northern States Power Company's (NSP) Monticello Plant, like most domestic boiling water reactors (BWR's), was originally equipped with wye pattern globe valves as main steam isolation valves (MSIV'S). One of the purposes of the MSIV's is to close automatically and tightly in the event of a loss-of-coolant accident (LOCA) to maintain potential radiological releases through the MSIV's to within allowable limits. The MSIV's are leak tested every refueling outage to assure MSIV leakages following a postulated LOCA will be within these limits.

In 1991, Monticello began to experience measured MSIV seat leakage well in excess of allowable limits. The problem was compounded in 1993 when Monticello determined it could no longer perform leak tests with the non safety grade makeup air/nitrogen supply to the MSIV pneumatic actuators valved-in, and began performing the leak tests with the supply isolated. The MSIV's which failed the leak tests were brought into compliance by performing maintenance only. However, a long term permanent fix was needed to assure that future MSIV leakages would be consistently below allowable limits.

A search for a permanent solution to the MSIV leakage problem was initiated in early April 1993 when work began on compiling and evaluating a list of possible alternatives. Some of the alternatives considered were: increasing MSIV allowable leakage limits,

modifying MSIV internals to improve seating, increasing thrust capabilities of existing actuator springs, reconfiguring the steam lines so the MSIV actuators were oriented vertically, replacing MSIV's with gate valves, installing a safety-grade pneumatic supply, installing gate valves as a third MSIV in each steam line to be dedicated for containment isolation only, and replacing the MSIV's with new wye pattern globe valves having improved seating features. The evaluation of these alternatives consisted of preparing and analyzing the following: feasibility and performance studies, preliminary design and specification of equipment and systems, requests for proposals for new equipment, and cost estimates. Following completion of the evaluation, replacement of the existing wye pattern globe valves and pneumatic actuators with double disc gate valves and spring powered actuators, respectively, was selected for the outboard MSIV's. This alternative was determined at the time to be the only one that could provide seat leakages consistently below allowable limits and be relatively cost effective.

Orders for the new outboard MSIV's were placed in October 1993 - approximately 6 months after the evaluation started.

PRODUCTION OF EQUIPMENT

A contract for double disc, gate valves was awarded to Anchor/Darling Valve Company in October 1993 to replace Monticello's existing outboard MSIV's. These gate valves are class 900, 18 x 14 inch venturied type equipped with Hiller actuators. The total weight of valve and actuator is just under nine tons. The actuator is essentially half the total weight. The valve design conditions are 1250 psig at 575°F manufactured from carbon steel with cobalt chrome hard facing (Stellite 6) on the seats and discs. The body, bonnet, and yoke were specifically designed to accommodate the seismic load requirement of 2.50 g's applied in any direction.

The valve bonnet cavity is connected to the upstream side of the valve through a hole in the upstream disc to eliminate the possibility of pressure lockup. External limit switches are mounted on the valve yoke to indicate open and close positions, as well as the 10% close position (which is used as an input in the reactor protection system). A live loaded stuffing box arrangement was used in conjunction with a single stuffing box sized for Garlock EVSP packing.

The actuator was designed and manufactured by the Ralph A. Hiller Company and provides a minimum closing thrust of 55,400 pounds at full valve stroke. This thrust was based on closing the valve at a maximum differential pressure of 1000 psi and flow of 3.4×10^6 lb/hr. (200% of design flow) using a valve factor of 0.4. When as-built spring dimensions are considered and the fact that the valve isolates flow prior to full stroke, the actuator output would be capable of isolating flow with a valve factor of approximately 0.5.

The MSIV is opened with air pressure (270 psig nominal) and closed with mechanical spring force only. The normal closure speed can be controlled from three to ten seconds by adjusting hydraulic flow control valves associated with the actuators' hydraulic dash pot. The MSIV can also be exercised at various slow speeds by controlling the rate of air released from the actuator's air cylinder through an adjustable exhaust nozzle. Each actuator is equipped with Norgren air control valves and Valcor solenoid valve cluster assembly. These valves utilize the same control logic as the existing MSIV's.

Upon completion of manufacturing at Anchor/Darling's factory, the valves were subjected to the standard hydrostatic testing and seat testing specified by ANSI B16.34. The valves were then subjected to low pressure leak rate testing at air pressures of 25 psig and 42 psig. Northern States Power Company personnel performed this testing using the same test equipment and methodology utilized to perform leak tests at its Monticello Plant. All four valves had unmeasurable seat leakage. Actuator thrust measurements were also obtained at Anchor/Darling's factory by NSP personnel using Teledyne and Votes diagnostic equipment. This same equipment is used at Monticello. The test results were considered essential base line data which would be repeated once the valves were installed at the site.

The valves and actuators were manufactured in approximately ten months and delivered to the Monticello site in August 1994 to meet a planned September 1994 outage. The valves were packed and shipped separately from the actuators for ease in handling and installation.

INSTALLATION AND TESTING

The outboard MSIV's are located in the plant's reactor building in a room called the steam chase. One of the walls in the steam chase is adjacent to the plant's turbine building, and forms the secondary containment boundary for the room. This wall is equipped with blowout panels on the turbine deck elevation. These blowout panels were removed and left off for the entire installation period. Transport of equipment and personnel in and out of the steam chase was made through the blowout panel opening. In order to assure secondary containment was intact when required during the outage; all pipe penetrations, floor drains, doors, or temporary openings between the steam chase and reactor building were either sealed at all times or controlled such that the openings were only made when secondary containment was not required.

Installation work began immediately after plant shutdown. The first work activities undertaken were to modify the four existing monorails located over the outboard MSIVs to increase their load carrying capacity to what was needed to lift the heavier new valves and actuators. A new monorail was also installed from the outboard MSIV locations to the turbine deck. No such monorail existed previously and it was needed to provide a means to safely move heavy components in and out of the steam chase.

The existing valves and new valves were removed and installed, respectively, without incident. The yokes, bonnets, stems, and discs of the new valves were removed for convenience of handling and welding the bodies into the piping system. All pipe welding was performed manually and radiographed. No new pipe supports were required. Removal of the existing MSIVs and installation of the new MSIVs was performed in parallel with the installation of a new, non-safety grade, high pressure (270 PSIG nominal) main air supply system for the new MSIV actuators. The system is supplied by two 100% capacity banks of bottled air which were installed in a Menard's type building outside the plant. Stainless steel piping was routed from the building to the actuators in the steam chase. A 550 gallon accumulator was installed in the system on the turbine deck. The accumulator maintains system pressure above a minimum value to assure the actuators will not drift close during valve exercising. Plant ambient operating

temperatures on the turbine deck are cooler than the steam chase so the accumulator also mitigates the effects of thermal expansion of air in the system. Two Haskel air driven air compressors were connected to the main air system as a backup in case replacement air bottles were unavailable.

The same non-safety grade pilot air supply used for the existing outboard MSIVs was reconnected to the new actuators using quick disconnect air fittings. No changes were made to the electrical supplies to the new actuator solenoid valve manifolds and limit switches. However, minor adjustments in cable lengths were required because of slight changes in terminal point locations.

After installation of the valves, permanent platforms were installed around the valves to provide easy access for future operation, testing, and maintenance.

Pre-operational testing of the new valves included a system leakage test, low pressure air leak rate test, stroke timing test, actuator leak test, actuator thrust test, and other functional testing to verify proper operation of valves, actuators, and limit switches. Relief was requested and received from the NRC to perform a system leakage test on the new valves at normal operating pressure in lieu of a system hydrostatic test at 110% of normal operating pressure. The system leakage test was performed at the end of the outage in conjunction with the normal reactor system leakage test.

Operational testing was also performed on the new MSIVs. It involved stroking each valve at incremental reactor power levels during plant startup to verify no unexpected transients were induced. The downstream welds on the valves were also checked for leakage during operation.

Installation and pre-operational testing of the outboard valves including monorail upgrades and main air system installation were completed in approximately 30 days.

PERFORMANCE RESULTS

Since the installation of the new double disc, gate valves in October 1994, the valves have been exercised from full open to ninety percent open on a weekly basis with full steam flow ($\approx 1.68 \times 10^6$ lbs/hr)--seventy two exercise cycles. The valves have also been cycled full open to full close once each quarter at seventy five percent of full flow ($\approx 1.26 \times 10^6$ lb/hr) - six full stroke cycles.

At the first outage following installation of the new MSIVs, in April 1996, leak rate testing was performed with acceptable results. Three valves were tested with the same leak rates as found in October 1994 after installation and one valve had the leak rate increase from 0 to 2.7 SCFH at 25 psig.

The leak rate history to date on the four new MSIVs is shown in Table 1.

The valve actuator thrust output for each valve was also obtained using VOTES and Teledyne diagnostic equipment and the comparison of this data is shown in Table 2.

A reduction in reactor operating pressure of approximately 5 psi was obtained as a result of lower pressure drop losses through the new outboard MSIVs. Reactor operating pressure is a function of turbine control valve set point plus steam line pressure losses between reactor and turbine control valves. The lower reactor operating pressure resulted in two

benefits. The first was an increase in reactor safety/relief valve (SRV) simmer margin - the difference between reactor operating pressure and SRV set point. A high simmer margin is needed to minimize SRV seat leakage. Excessive SRV leakage during operation could cause a plant shutdown which has occurred on numerous occasions at Monticello in the past.

The second benefit was a reduction in moisture content at the turbine throttle resulting in a small turbine efficiency improvement and heat rate reduction. The savings was estimated at approximately $50,000/year, and will be realized for the remaining licensed life of the plant, which at the time of installation, was 17 years.

The benefits of the reduced outboard MSIV pressure drop will increase slightly if Monticello's request to the NRC to operate its reactor at a higher thermal output rating is approved.

CONCLUSION

It is obvious that the replacement of major equipment, such as MSIVs, is very challenging and it may be difficult to develop convincing arguments to justify the cost outlay. It must be recognized at the start that total replacement costs for major equipment are significantly more than the cost of the new equipment; however, the alternative of not replacing poor performing equipment can readily exceed total replacement costs.

It is extremely important to accurately define performance requirements and service conditions early in the planning stage for replacement equipment. With diligent evaluation and planning on the part of the plant operators and a total commitment of support from the equipment manufacturer, such replacement work can be cost effective and completed in a timely manner.

TABLE 1

MONTICELLO PLANT MSIV
AIR LEAK RATE TEST RESULTS

Valve Serial Number	**Outboard A 768-1-2**	**Outboard B 768-1-3**	**Outboard C 768-1-1**	**Outboard D 768-1-4**
Leakage Test Results:				
July 1994 @ Mfg. Plant	25 psig/0 scfh 42 psig/0 scfh	25 psig/0 scfh 42 psig/0 scfh	25 psig/0 scfh 42 psig/0 scfh	25 psig/0 scfh 42 psig/0 scfh
October 1994 @ Monticello	25 psig/0 scfh	25 psig/0 scfh	25 psig/1.5 scfh	25 psig/0 scfh
April 1996 @ Monticello	25 psig/0 scfh	25 psig/0 scfh	25 psig/1.5 scfh	25 psig/ 2.7 scfh

TABLE 2

MONTICELLO PLANT MSIV
AVERAGE CLOSING FINAL THRUST DATA

Valve ID Number	Outboard A AO-2-86A	Outboard B AO-2-86B	Outboard C AO-2-86C	Outboard D AO-2-86D
Average Closing Final Thrust Data:				
July 1994 @ Mfg. Plant	60,257	61,253	61,190	61,967
October 1994 @ Monticello	60,800	59,960	61,760	61,544
May 1996 @ Monticello	58,465	58,400	59,080	59,400

Session 2C

Inservice Testing, General

Session Chair
Robert I. Parry
Senior Engineer
North Atlantic Energy Service

ADVANTAGES OF CUSTOMER/SUPPLIER INVOLVEMENT IN THE UPGRADE OF RIVER BEND'S IST PROGRAM

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ABSTRACT

At River Bend Station, IST testing had problems. Operations could not perform the test with the required repeatability; engineering could not reliably trend test data to detect degradation; licensing was heavily burdened with regulatory concerns; and maintenance could not do preventative maintenance because of poor prediction of system health status. Using Entergy's Total Quality principles, it was determined that the causes were: lack of ownership, inadequate test equipment usage, lack of adequate procedures, and lack of program maintenance. After identifying the customers and suppliers of the IST program data, Entergy management put together an upgrade team to address these concerns. These customers and suppliers made up the IST upgrade team. The team's mission was to supply River Bend with a reliable, functional, industry correct and user friendly IST program. The IST program in place went through a verification process that identified and corrected over 400 individual program discrepancies. Over 200 components were identified for improved testing methods. An IST basis document was developed. The operations department was trained on ASME Section XI testing. All IST tests have been simplified and shortened, due to heavy involvement by operations in the procedure development process. This significantly reduced testing time, resulting in lower cost, less dose and greater system availability.

Inservice testing in accordance with the ASME Boiler & Pressure Vessel Code, Section XI (the Code) is intended to test the operational readiness of components required to perform a specific function in shutting down a reactor or in mitigating the consequences of an accident. IST test data provides an excellent chronology of component degradation over time. At Entergy's River Bend Station, the inservice testing program was a liability instead of a precursor to preventative maintenance. Operations could not perform the test with the required repeatability; engineering could not reliably trend test data to detect degradation; licensing was heavily burdened with regulatory concerns; and maintenance could not do preventative maintenance because of poor prediction of system health status. Since 1990, there were 9 quality assurance findings against IST; 11 NRC level 4 violations issued against IST; and over 30 condition reports submitted against IST.

NRC reports also confirmed management's concerns.

NRC INSPECTION REPORT 50-458/94-06

- "Weaknesses continue to appear in the performance of IST surveillance's on

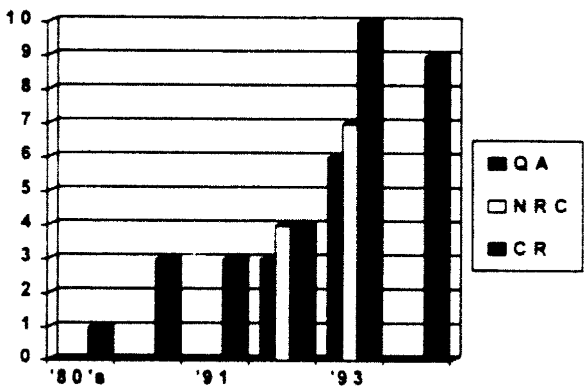
safety-related pumps, further confirming the need for the licensee to implement the IST Program Improvement Strategy in a timely manner. The River Bend Station has had a history of problems with the IST Program. Violations and weaknesses have been documented in NRC inspection reports 50-458/92-26, -92-35, -93-05, -93-25, -93-27, AND -93-31."

Why was River Bend's IST program a liability? An investigation using Entergy's Total Quality principles revealed that the causes could be categorized as: lack of program ownership, inadequate test equipment usage, lack of adequate procedures, and lack of program maintenance.

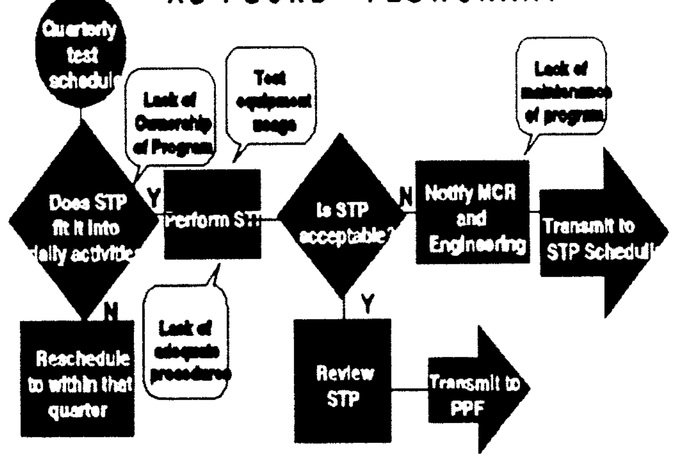
The INVESTIGATION

Reports generated due to the symptoms, were investigated:

- 9 Quality Assurance (QA) findings against IST
- 11 Nuclear Regulatory Commission (NRC) level IV violations issued against IST
- Over 30 condition reports (CR) submitted against IST



'AS FOUND' FLOWCHART



PROBLEM IDENTIFICATION ANALYSIS
AFFINITY DIAGRAM

Lack of Ownership	Inadequate Use of Test Equip	Lack of Adequate Procedures		Lack of Maintenance of Program
• CONDITION	• CONDITION	• CONDITION	—CR 93-0647	Not enough
REPORTS:	REPORTS	REPORTS	—CR 93-0666	reliable data
—CR 93-133A	—CR 90-0539	—CR 90-0548	—CR 93-0741	trends
—CR 93-0584	—CR 91-0151	—CR 92-0109	—CR 94-0059	available to
—CR 91-0037	—CR 90-0292	—CR 86-1475	—CR 94-0075	determine
—CR 94-0252	—CR 94-0237	—CR 92-0110	—CR 94-0056	system
—CR 94-0253	—CR 94-0240	—CR 92-0040	—CR 94-0199	degradation
• QA CONCERN	—CR 93-0667	—CR 91-0444	—CR 93-0667	with certainty.
—92-02-1-STPG-004	QCR P93-11-010	—CR 92-0497	—QCR P93-11-005	
—92-01-1-IIPG-007		—CR 93-0635	—QCR P93-11-006	
—92-01-1-IIPG-008		—CR 93-0133	—QCR P93-11-006	
QCR P93-11-007		—CR 93-0489	QCR P93-11-009	

Lack of Ownership
Since 1986 the IST program has changed hands from Technical staff to Operations to Systems engineering. No one was responsible to ensure the program was properly implemented. Since no one was held accountable for so long, the program suffered.

Inadequate Test Equipment Usage
Test results were not trendable due to improper gauge usage. Permanent plant test equipment was not calibrated to Code standards which required the use of temporary gauges. Due to poor program control in operations, the temporary test equipment used had to be recreated every test which would result in varying test equipment locations and meter ranges. This was not conducive to a repeatable test condition. The operations personnel performing the surveillances did not understand the Code requirements for test reference condition repeatability.

Lack of Adequate Procedures
Tests would not ensure operational readiness of equipment and many times could not be performed without correcting the procedure. It was not uncommon during the performance of a test for the test personnel to stop the test to obtain guidance on test instructions. The instructions were incomplete and often incorrect. This coupled with a cumbersome procedure change process led to poor procedure adequacy and adherence.

Lack of Program Maintenance
Test results often had unreliable trends and some were not trended. Due to poor recreation of test reference conditions, data provided from surveillances could not determine the hydraulic or mechanical condition of safety related equipment. This along with insufficient ownership of the program led to the attitude that the data did not need to be trended.

An in-house assessment of River Bend's IST testing program was performed to verify the causes were correctly identified. The objectives of this assessment were:

1. INSERVICE TESTING PROCEDURES
   Performance objective: IST procedures provide appropriate direction for the support of the IST program including: data gathering, documentation, trending and other activities as required. Newly issued procedures implement the requirements of the PROCEDURE WRITER'S MANUAL.

2. PROGRAM PLAN ADEQUACY
   Performance objective: The River Bend IST program plan adequately and appropriately implements the requirements of the ASME Code, GL 89-04, NUREG-1482 and the ENTERGY standard for inservice testing as outlined in design engineering administrative manual (EP-S-003-00C).

3. PERFORMANCE OF TESTING ACTIVITIES
   Performance objective: Performance of testing activities are completed in accordance with approved procedures, with calibrated test equipment that meets Code specified range and calibration accuracy and in a manner that exhibits good radiological and work practices.

4. TEST RESULTS EVALUATION/ TRENDING/DOCUMENTATION
   Performance objective: The IST program test results evaluations and trending, design basis documentation and administrative controls provide for a complete, systematic, and appropriately documented program. Evaluate test results, trending, and design basis documentation for a sample of components in the selected systems.

5. VALVE TESTING
   Performance objective: Valve testing, selection, methodology, acceptance criteria and identification of corrective actions support the assessment of operational readiness of components and meet both the regulatory and design basis requirements.

6. PUMP TESTING
   Performance objective: Pump testing, selection, methodology, acceptance criteria and identification of corrective actions support the assessment of operational readiness of components and meet both the regulatory and design basis requirements.

7. INSERVICE TESTING GROUP INTERFACES
   Performance objective: IST group interfaces are well-defined and function effectively to accomplish assigned tasks.

8. IST PROGRAM CONFIGURATION MANAGEMENT AND DOCUMENTATION ADEQUACY
   Performance objective: IST program documentation is consistent; working level procedures and plans accurately implement the requirements and commitments of design basis and licensing basis documents.

9. INSERVICE INSPECTION TESTING ENGINEER TRAINING
   Performance objective: IST personnel knowledge, training, qualification, and performance support safe and reliable plant operation.

Entergy's Total Quality principles were employed again to determine the <u>customers</u>

and suppliers of the IST program data. These customers and suppliers were employees who were affected the most by the River Bend IST program. These parties were those that could do the most to aid in the improvement the program.

CUSTOMERS	SUPPLIERS
operations	operations
systems engineering	systems engineering
licensing	design engineering
maintenance	quality control/assurance

Entergy management formed an upgrade team to address the program concerns. The upgrade team itself was staffed with representatives of these customers and suppliers. The empowerment theory enabled the customers and suppliers to develop a reliable, functional, industry correct and user friendly IST program. The team addressed each cause individually:

Lack of Ownership

In February 1994, the program was assigned to the upgrade team. Operations would run the surveillances, Engineering would evaluate the test data to determine if the Code requirements were satisfied; and as a team they would commence the task of Kaizen (improving a small amount every day). An owner was assigned to each test during the upgrade process to track progress and be accountable for test upgrade completion. Administrative controls were established over the program documentation to ensure completion. Roles and responsibilities were defined in the administrative procedure for the operations, systems engineering, licensing, and IST engineering involvement in the upgrade process.

Inadequate Test Equipment Usage

At the start of this project, few installed plant instrumentation met Code calibration and range requirements. A plant modification was planned to install permanent plant instrumentation that satisfied those Code requirements. This would reduce the chance of contamination, shorten test duration, and make test reference conditions easier to recreate. Ultrasonic flow measurement equipment was also made a part of the IST testing program where applicable. This method does not involve breaching the system to obtain a measurement; further reducing the chance of contamination. The operators in the IST team were trained on proper test equipment usage. The operations department was trained at a later time on test equipment principles and ASME Code concerns by the IST upgrade team. As a temporary measure until the permanent instruments are installed, standard test gauge locations and practices were developed. A generic procedure was written to govern the use of temporary instrumentation. This procedure had far reaching applications beyond inservice testing; it has been used in troubleshooting efforts prior to maintenance activities. The operators had all test rigs made and stored for scheduled use. All test connections to systems in question were standardized for consistency.

Lack of Adequate Procedures

Every IST surveillance procedure was rewritten after a field walkdown and a tabletop discussion with operations, systems engineering and inservice testing engineering. The tabletop discussions were important because all parties involved with the test were present to discuss their concerns and make changes as needed. The systems engineer for that system was present to learn of the IST requirements imposed on their systems. The inservice testing engineer was present to ensure that all Code concerns were satisfied.

Operations was there to see that the tests were user friendly to perform. The operations concerns also dealt with ALARA dose practices (As Low As Reasonably Achievable) and test instruction methods. Redundant procedures were deleted. Some tests were divided by safety divisions to fit Entergy's divisional work week philosophy. Surveillances were categorized into quarterly, cold shutdown, and refuel procedures that could be easily integrated into the normal operations department's shift rotation schedule. ALARA concerns facilitated some changes in test methodology that aided in lowering the total dose per test. These rewrites reduced testing time significantly, resulting in less dose and greater system availability. In the examples shown, some notable savings were realized in dose and time. Surveillance STP-201-6311/6312 is the Standby liquid control system pump and valve test. Historically this test took at least 12 hours and involved pumping the contents of a test tank into barrels. It was a very difficult test and had many opportunities for error. The new version of this test takes about an hour to perform.

Some test instruction methodologies that facilitated these savings were:

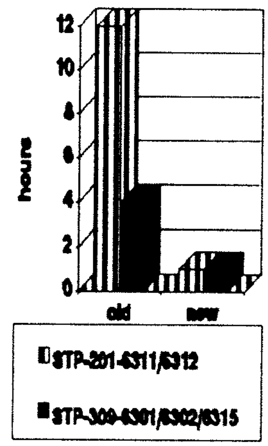
- During the test development bullets were used instead of numbers where the order was unimportant or steps could be done concurrently. This reduced test setup time allowing multiple activities in different locations.

- Test connections were moved to lower dose areas wherever possible. For example, the original pressure isolation valve test involved high dose drywell entries. After the procedure was upgraded, most connections were moved to remote instrument connections outside the drywell where the dose and climate were more favorable.

- Ultrasonic flow measurement equipment does not involve breaching the system in question to obtain a system flow rate, thus reducing the possibility of contamination.

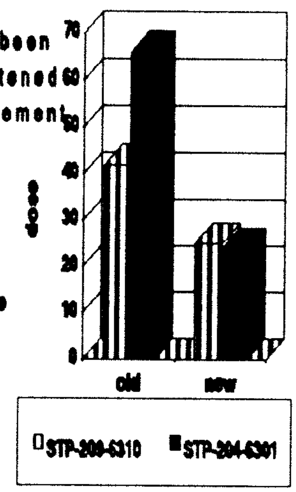
Time Savings

All IST STP's have been simplified and shortened due to heavy involvement by the customer (Operations) in the procedure writing process resulting in less system down time.



Dose Savings

All IST STP's have been simplified and shortened due to heavy involvement by the customer (Operations) in the procedure writing process,resulting in lower radiation dose



Lack of Program Maintenance

The program went through a reverification to ensure Code requirements were satisfied. During this reverification over 200 components were identified for improved testing methods and identified / upgraded over 400 individual discrepancies. An IST component basis document was developed to explain the content and context of the River Bend IST program. Title 10, Part 55.55a of the Code of Federal Regulations and River Bend Station Technical Specification 4.0.5 invoke, by reference, the requirement for inservice testing (IST) of pumps and valves in accordance with the ASME Boiler & Pressure Vessel Code, Section XI (the Code). Consequently the Code requires that the owner of each nuclear power plant prepare a "plan" for testing and inspecting systems and components under the jurisdiction of the Code. With respect to the elements of that plan related to the testing of pumps and valves, Section XI, Subsections IWP and IWV specify, in general terms, the program scope and testing requirements needed to satisfy the Code. Over the period of time since the requirement for Inservice Testing was first incorporated into the Code of Federal Regulations, the NRC, ASME, and the industry have provided additional interpretations to the Code requirements through various mechanisms including, Code cases, Generic Letters, program review meetings, site inspections, etc. It is the intent of this document to:

- establish succinct rules for determining the program scope

- evaluate each of the plant systems and related components in a consistent manner to identify which components should be included in the testing program and to what extent each should be tested.

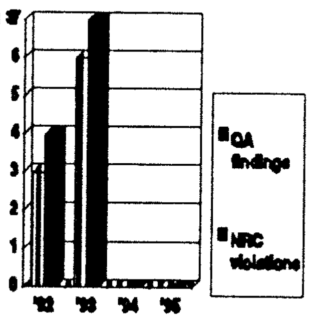
This document establishes the philosophy by which the scope of the ASME Section XI IST Program is determined including which components are to be included and the extent and type of testing required for each. In the course of developing this document, each of the plant systems at River Bend were evaluated with respect to the function of each component and the need for its operability as it relates to the scope of Section XI.

Conclusion

Getting something done is an accomplishment, getting something done right is an achievement. What does River Bend's IST testing program look like today? Is it a liability? The program is owned, controlled and trended by Engineering programs and the surveillances are performed by the normal operations shift crews. At print time no major concerns had surfaced. The procedures are user friendly and can be easily integrated into the normal shift activities. The test data is trended with a new computer program that has the capability to plot component performance over time. The program is considered to be a strength by the ENTERGY management team. The IST upgrade team was recognized in ENTERGY's Peak performer and Team excellence programs for the results that were produced. How did the IST program fare against the initial problem reports? All report categories investigated at the onset of the upgrade went to zero. No price can be set on regulator image of a nuclear facility, but in-house condition reports cost approximately $10,000 each.

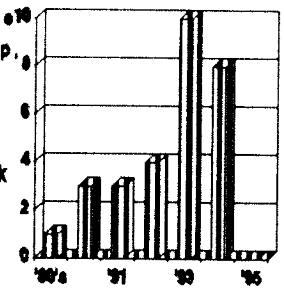
Summary of Results

Reduced from 11 NRC violations and 9 Quality Assurance (QA) findings tozero to date.

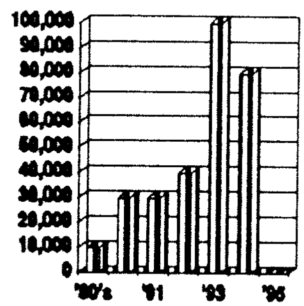


Summary of Results

Reduced from 90 condition reports due to: Lack of ownership, Inadequate test equipment usage, Lack of adequate procedures, and Lack of maintenance of program tozero to date.



What did those condition reports cost !



During the upgrade process, an NRC assessment revealed no problems. The NRC's opinion has changed somewhat since the upgrade.

NRC INSPECTION REPORT November 1994

- "The efforts to improve the inservice test program and IST implementation were considered a strength."

- "The initiation of an assessment of the inservice test program was commendable. The scope of the assessment was excellent and the expertise of the team members was considered a strength."

- "The inspectors considered the establishment of the component basis document to be an excellent effort at clearly defining and documenting the scope and requirements necessary to have an effective IST program."

- "The inspectors considered the training effort established by the licensee to be a strength. The methodology was appropriate and well thought out, and the training materials were considered excellent."

NRC INSPECTION REPORT January, 1995

- "On January 12, 1995, the inspectors observed inservice testing of the high pressure core spray pump pursuant to ASME Code Section XI. The procedure was technically correct, and was written in an excellent manner. This procedure was a significant improvement over procedures the inspectors had reviewed prior to the IST Program Improvement Plan."

REFERENCES:

- River Bend Station Inservice Testing Plan
- NRC Inspection Reports (various)
- ASME Section XI
- ENTERGY's Quality Principles & Practices

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Thank you for your support.

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Summary of Inspection Findings of Licensee Inservice Testing Programs at United States Commercial Nuclear Power Plants

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ABSTRACT

Periodic inspections of pump and valve inservice testing (IST) programs in United States commercial nuclear power plants are performed by Nuclear Regulatory Commission (NRC) Regional Inspectors to verify licensee regulatory compliance and licensee commitments. IST inspections are conducted using NRC Inspection Procedure 73756, "Inservice Testing of Pumps and Valves" (IP 73756), which was updated on July 27, 1995. A large number of IST inspections have also been conducted using Temporary Instruction 2515/114, "Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs" (TI-2515/114), which was issued January 15, 1992. A majority of U.S. commercial nuclear power plants have had an IST inspection to either IP 73756 or TI 2515/114. This paper is intended to summarize the significant and recurring findings from a number of these inspections since January of 1990.

INTRODUCTION

Periodic inspections of pump and valve inservice testing (IST) programs in United States (US) commercial nuclear power plants are performed by Nuclear Regulatory Commission (NRC) Regional Inspectors to verify licensee regulatory compliance and licensee commitments. The testing requirements for pump and valve IST programs are referenced through Title 10 of the Code of Federal Regulations, Section 50.55a, "Codes and Standards." These requirements are specified in the American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section XI, Subsections IWP (for pumps) and IWV (for valves). ASME/ANSI (American National Standards Institute) Operations and Maintenance (OM) Standards Part 6, "Inservice Testing of Pumps in Light-Water Reactor Power Plants," (OM-6) and Part 10, "Inservice Testing of Valves in Light-Water Reactor Power Plants," (OM-10) were incorporated into the 1989 edition of Section XI by reference and subsequently included in the regulations through rulemaking effective September 8, 1992. OM-10 and later editions of IWV reference OM-1, "Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices," to establish requirements for safety and relief valves. The Code of record for a specific plant is dependent on which edition of the Code was referenced in the regulations at either the commencement of commercial plant operation or the date twelve months prior to the start of the next IST program ten-year interval.

IST inspections are currently performed under NRC Inspection Procedure 73756, "Inservice Testing of Pumps and Valves" (IP 73756). IST inspections have also been conducted

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

using Temporary Instruction (TI) 2515/114, "Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs," which was issued on January 15, 1992. In addition, TI 2515/110, "Performance of Safety-Related Check Valves," issued on November 19, 1991, includes an evaluation of IST program check valves. IP 73756 was updated on July 27, 1995, to incorporate elements of TI 2515/114 and TI 2515/110 into the inspection procedure. NUREG-1482, "Guidelines for Inservice Testing at Nuclear Power Plants," was issued April of 1995 to provide additional guidance on IST programs.

This paper is intended to summarize the significant and recurring findings from a number of these inspections since January of 1990. A search of the NRC Nuclear Documents Database yielded inspection reports from a number of plant sites, which have had an IST inspection to either IP 73756 or TI 2515/114. A number of IST findings in this paper have been included that were part of resident inspector reports, diagnostic evaluation team reports, and recent events. The findings have been grouped under the headings of IST Program Scope, Pump Testing Methodology, Valve Testing Methodology, Missed Surveillances, Test Result Analysis, and Relief Requests and Deferred Testing. A general discussion of the applicable Code requirements is included in each major section of this paper. References to "the Code," unless otherwise stated, represent the requirements of Part 6 and Part 10 of the ASME *Operations and Maintenance Standards*, as referenced in Section XI of the 1989 Edition of the ASME Code. Earlier editions of Section XI or the later *OM Code* may have different requirements associated with a particular section.

IST PROGRAM SCOPE

The Code requires that certain Class 1, 2, and 3 components that perform a specific function in shutting down the reactor, maintaining the reactor in the cold shutdown condition, mitigating the consequences of an accident or for over pressure protection of systems that perform the above functions, be included in the facilities IST program. IST program inspections include an assessment of the program scope for a select number of systems. The assessment is performed by reviewing the licensee's Safety Analysis Report (SAR), Technical Specifications, Emergency Operating Procedures, and other design documents. A number of licensees have developed or are developing IST basis documents to provide the bases for component inclusion in or exclusion from their IST program. An IST basis document was noted to be a valuable resource to both the licensee and NRC staff to document component functions and design bases.

The following list of components were identified during NRC inspections to have a safety function but were not included in the plant's IST program. This section discusses scope findings for check valves, power-operated valves, manual valves, and relief devices. There were no significant IST program scope deficiencies noted for pumps. The general areas where concerns were noted included valves used to isolate non-safety related systems or non-essential components to prevent the diversion of flow, valves used to cross-connect trains or modes of operation, and control valves required to reposition to perform their safety function. The safety functions addressed here may or may not be applicable at other facilities. There may, however, be similar valves or scenarios that should be reviewed for applicability at your

facility. When applicable, a reference is made as to whether the components or scenario pertained to a pressurized water reactor (PWR) or a boiling water reactor (BWR). When no reference is provided, the finding, although identified at a particular plant or group of plants, is applicable to all facilities.

Check Valves

- Residual heat removal (RHR) pump discharge check valves perform a safety function in the closed direction to ensure the RHR piping does not drain when the pumps are aligned to take a suction from the suppression pool. If the piping was not filled, a possible water hammer event could occur upon starting the pump and prevent the system from performing its intended function. The licensee's IST basis document stated that the check valves did not have a closed safety function because the water leg pumps would keep the RHR system filled with water. The water leg pumps would not, however, be able to keep the system filled if the valves degraded to the point that the pump would be unable to keep up with the leakage rate due to the pump's limited capacity. (BWR)

- RHR pump suction check valves from the refueling water storage tank (RWST) perform a safety function in the closed direction during the accident injection phase. If one RHR pump were to fail, flow from the running RHR pump, which is in a parallel pump configuration, could be diverted from the reactor core through the RHR cross-connect valves, the open mini-flow valve on the idle pump, and challenge the RWST check valve. The operator would assume all flow from the running pump would be entering the reactor since flow instrumentation was upstream of the cross-connect valves. (PWR)

- Service water (SW) pump cubicle cooler outlet check valves (normally open) perform a safety function in the open direction to provide cooling flow to the pump room coolers. These valves were considered passive category C valves by the licensee, which would exclude them from the IST program. These valves may, however, momentarily close during an accident if power to the SW pump was lost and would have to reopen when the pump restarts. Normally open check valves should not be considered passive. As an active valve, it should be included in the IST program. (PWR)

- Volume control tank (VCT) outlet check valves perform a safety function in the closed direction to prevent reverse flow into the VCT. This scenario could occur after a loss-of-coolant-accident (LOCA) during the recirculation phase when the RHR pumps provide the water supply for the charging pumps. The discharge pressure of the RHR pumps may exceed the VCT relief valve setpoint and provide a leak path outside containment. The check valve would be required to close to isolate the relief valve from RHR system pressure. (PWR)

- Component cooling water (CCW) check valves upstream of the reactor coolant pump (RCP) thermal barriers perform a safety function in the closed direction to isolate high pressure reactor coolant from low pressure CCW piping if the RCP thermal barrier were to leak or fail. (PWR)

Power-Operated Valves

- RHR system cross-connect valves perform a safety function in the closed direction during cold leg recirculation and a safety function in the open direction during hot leg recirculation. The valves need to reposition during an accident to switch from cold to hot leg recirculation. (PWR)

- CCW loop isolation valves, which are normally open, perform a safety function in the closed direction to isolate non-essential loads and non-code class piping during a LOCA. (PWR)

- Component cooling service water (also called residual heat removal service water) control valves perform a safety function in the open direction to supply cooling water to the RHR heat exchangers. These valves are normally closed and are required to open to perform their safety function. The exception in the Code for control valves does not apply, as the valves must open (partially) to perform their safety function. Control valves with a fail-safe function are also not excluded from the Code as addressed in NUREG-1482, section 4.2.9. (BWR)

- RHR pump mini-flow valves (normally open) perform a safety function in the closed direction to isolate mini-flow after RHR system flow reaches a defined flow rate. The SAR stated that if the mini-flow valve failed open, flow would be diverted from the reactor coolant system, but the opposite RHR train would satisfy the minimum flow requirements. In order to meet the single failure criteria, however, both trains must be able to meet the design flow requirements in the event of a failure in the opposite train. (PWR)

- High pressure coolant injection (HPCI) steam line drain valves perform a safety function in the closed direction to prevent the diversion of steam from the HPCI turbine to the drain system. These valves are normally open to drain condensate from the line. If the valves do not close when required, steam flow to the turbine would be reduced and possibly affect turbine performance. (BWR)

- HPCI turbine steam supply valves perform a safety function in the open direction in addition to the close function listed in the program. The normally closed valves must open to provide steam to the HPCI turbine. (BWR)

- Position indication tests for power operated valves with a passive safety function were not included in the IST program as required by OM-10.

Manual Valves

- One plant generically excluded manual valves. As stated in the Code and NUREG-1482, Section 4.4.6, all valves with a safety function must be included in the IST program.

- CCW manual valves perform a safety function to align the common CCW heat exchanger to the unit undergoing post-LOCA recovery. Since the common heat exchanger may be lined-up to the unit not undergoing the accident, the valves would need to be repositioned for the heat exchanger to perform its safety function. (PWR)

Relief Devices

- The CCW surge tank relief valve performs a safety function to protect the system from over pressurization in the event of system in-leakage. (PWR)

- The HPCI and reactor core isolation cooling pump suction relief valves perform an over pressure protection function for their respective systems. (BWR)

- The HPCI rupture discs perform a safety function to protect the steam exhaust piping from an over pressurization. (BWR)

PUMP TESTING METHODOLOGY

The Code requires quarterly testing of all pumps included in IST programs. Pump hydraulic and mechanical performance are assessed by determining, at reference conditions, pump flow, differential pressure, and bearing vibration. Acceptance criteria for these measured values are included in the Code. In addition, test methods and instrumentation requirements are also specified in the Code.

This paper addresses inspection findings in the areas of pump hydraulic testing, vibration testing, acceptance criteria, instrumentation, and issues concerning NRC Bulletin 88-04.

Pump Hydraulic Testing

The Code requires that hydraulic reference values be established from preservice or inservice tests when the pump is known to be operating acceptably that can be readily duplicated during subsequent inservice tests. Licensees have asserted that repeatable reference values are not attainable in certain systems due typically to the inability to easily throttle the pump flow. Guidance has been provided in NUREG-1482, Section 5.2, for requesting relief to use pump reference curves to establish Code alert and required action range acceptance criteria. In addition, NUREG-1482, Section 5.3, states that a $\pm 2\%$ tolerance band around a specific reference value is allowed without approval from the NRC.

- A number of licensees have used pump reference curves without prior NRC approval. In one case, the licensee had developed pump curves for every pump in their IST program to verify Code acceptance criteria without approved relief requests. The licensee incorrectly assumed that this test method was in accordance with the Code requirements. After an evaluation of their pump testing, the licensee determined that most of their pumps could be tested using fixed reference values.

- The HPCI pump surveillance procedure did not set the fixed reference value in a repeatable manner as required by the Code. The established reference value for differential pressure (dp) was 1240 pounds per square inch differential (psid). The test procedure set pump discharge pressure at 1200-1280 pounds per square inch gage (psig) and then subtracted pump suction pressure to determine pump dp. Test data indicated that the fixed dp reference value ranged from 1195 to 1240 psid. As such, actual dp varied by as much as 3.6% from the fixed reference value. If pump discharge pressure had been set at the lower end of the allowed pressure band (1200 psig), dp could have varied approximately 5% from the fixed reference value.

- A number of licensees did not have provisions in their quarterly pump test procedures to maintain the established reference condition for the required period of time to allow hydraulic conditions to stabilize before recording pump flow, dp, and vibration.

- Several pump test procedures did not ensure the measured flow rate was equal to the corresponding reference value. The flow rate was assumed to be constant based on no adjustments made to the recirculation line, which provided the flow path for the pump. The licensee indicated the recirculation line was not instrumented based on the flow rate meter not meeting the 2% of full scale Code accuracy requirement. The technical manual for the flow instrument, however, indicated an accuracy of 0.75% of full scale, which was within the Code accuracy requirements.

Pump Vibration

The Code requires the owner to determine the proper location to measure pump bearing vibration. These locations must be accessible to the vibration probe and used for each subsequent quarterly inservice pump test to obtain valid vibration data. Vibration reference values (V_r) are required to be established when the pump is in good operating condition. OM-6 specifies vibration alert and required action range limits as multiples of the reference value ($2.5{\bullet}V_r$ for the alert limit and $6{\bullet}V_r$ for required action limit). In addition, absolute alert and required action range limits are specified at 0.325 inches per second (in/sec) and 0.700 in/sec respectively.

- The cover of a SW vertical line shaft pump had been rotated, shifting the vibration markings for the vibration probe. In other cases, the vibration probe locations on several pumps were not clearly marked. The marks were worn away (rust) or painted over such that they were no longer visible. As a result, the vibration readings were not meeting the repeatability requirement of the Code.

- The pump test procedures did not contain instructions for the placement of vibration probe to measure bearing vibration. The IST engineer and surveillance personnel stated that they use "skill-of-the-craft" to place the probe for vibration testing. The inspectors observed that the pumps had painted marks in some, but not all of the probe locations for vibration testing. In addition, there was one location marked in which surveillance personnel were not physically able to place the vibration probe on the mark and had to position the probe adjacent to the mark.

- Vibration measurements were only taken in the vertical and horizontal directions for the core spray and RHR pump bearings. OM-6 requires that a measurement shall be taken in the axial direction on each accessible pump thrust bearing housing. For these pumps, the thrust bearing is located in the motor housing and is accessible. A similar finding at another plant on a vertical centrifugal RHR pump prompted the licensee to submit a Code inquiry to clarify the requirements for measuring axial vibration on pump thrust bearings located in the driver.

- The vibration acceptance criteria for four of the five pumps were incorrect. The reference values for the pumps had been revised; however, the test procedure was

not updated to reflect the new alert and required action vibration limits.

- The absolute vibration limits established were non-conservative. The relief request established generic absolute velocity vibration alert and required action limits of 0.236 and 0.314 in/sec, respectively, for most pumps in the IST program. These limits were less than the absolute limits established in OM-6. However, OM-6 also provided reference multipliers to account for "smoother" running pumps which would be more restrictive. The absolute alert limit established in the relief request would have exceeded the required action range limit using the OM-6 multiplier.

Acceptance Criteria

Pump hydraulic and mechanical performance data is required to be assessed to determine if the pump is operating in the alert or required action range. ASME Section XI previously allowed the owner to specify the acceptance criteria when the Code limits were not attainable. This provision was not included in OM-6.

- A pump test contained expanded acceptance criteria ranges without adequate justification. The justification for the expanded ranges was the fluctuation exhibited in the flow meters did not make for consistent readability. This justification did not appear to be adequate based on a review of the test data that indicated the measured flow rates were within the Code allowable limits since the expanded ranges were used. In addition, during an operability test of the pump, the flow rate meter was observed not to fluctuate substantially.

- The Code required reference value evaluations were not performed after pump maintenance activities. After pump maintenance or repair, reference values must be reconfirmed or a new set of reference values established.

- After replacing a pump impeller with an new impeller of a different material, the licensee's post maintenance testing consisted of revalidating the 1970 pump performance curve at only the minimum flow recirculation point. Although the testing satisfied the Code requirements, the post-modification test should have included additional measurements at higher flow rates to provided a greater level of confidence in pump performance.

- The acceptance criteria for several pumps were based on approved relief request pump curves that meet the Code limits, however, the design requirements for the pumps may be more limiting. As a result, pump degradation allowed by the pump curve may not meet design requirements. The IST administrative procedures need to ensure when pump curves are developed or revised, they meet both the pump design and Code requirements.

- Incorrect reference values for the pump dp were used in the test procedure for the turbine driven auxiliary feedwater (AFW) pump. The licensee used an incorrect dp of 1650 psid instead of the required dp of 1701 psid. In addition, the values listed for the acceptable, alert, and required action pressure ranges were incorrect since they were based on 1650 psid.

Instrumentation

- Various pump suction pressure gages at a number of plants were not in compliance with the Code full-scale range requirements of three times the reference value. Licensee actions included replacing the gages, revising test procedures, and submitting relief requests to use the existing gages.

- A flow instrument was used to measure pump flow rate that was not in compliance with the Code full-scale range requirements. Although the licensee had identified the improper gage one year earlier and ordered an ultrasonic flow meter that would meet the Code requirements, the condition and operability determination of the pump had not been documented and the licensee continued to use the original gage.

- Flow instruments were referenced in the test procedures for pumps to measure flow rates greater than 200 GPM. The instruments, however, had a range of only 0-200 gpm.

NRC Bulletin 88-04, "Safety Related Pump Loss"

NRC Bulletin 88-04 was issued to request that licensees review and initiate corrective actions, if necessary, to address two potential design concerns in the recirculation (minimum flow) loops of safety-related pumps. These concerns were: 1) the potential for one pump in a parallel pumping system to dead-head the other pump or pumps in the system; and 2) to verify that the system recirculation capacity was adequate to protect the pump. Inspection findings were related to the review and corrective actions that licensees had implemented.

- Documentation responding to Bulletin 88-04 did not include evidence demonstrating that the high pressure safety injection (HPSI), low pressure safety injection and containment spray pumps can operate satisfactorily in the minimum flow mode during all analyzed plant conditions including a small break LOCA. Specifically, the estimated times of pumps operating in the minimum flow mode following a postulated small break LOCA had not been determined.

- Pumps were tested using the minimum flow lines with a flow rate of 300 gpm, which is below the minimum flow rate of 425 gpm stated in the licensee's response to Bulletin 88-04. Further investigation by the licensee revealed that an orifice was installed in the recirculation line to maintain the flow rate above the vendor recommended minimum of 200 gpm but below the line full flow rate of 425 gpm.

- A pump vendor had specified a maximum run time of 15 minutes at a minimum flow rate of 20 gpm. The time limit was not addressed in the monthly surveillance test procedure. It was concluded that the vendor recommended limits were not being strictly implemented. After discussions with the pump vendor, the licensee decided to decrease the pump overhaul interval to every five years. This decision was based on that strict adherence to the time limit was not practical. Procedures were changed to discourage low flow operation. Previous pump overhauls had not indicated excessive wear.

VALVE TESTING METHODOLOGY

A number of issues were identified with the test of valves in licensee's IST programs. These issues are divided into the following categories as discussed below: stroke time testing, leak rate testing, check valve acceptance criteria, disassembly/inspection of check valves, test methodology, and relief valve testing.

Stroke Time Testing

The Code requires that power-operated valves be stroke timed to each safety position at a quarterly frequency. This testing may be deferred to cold shutdowns or refueling outages if the stroke testing is impractical to perform. The Code requires reference values to be established when the valve is known to be operating acceptably.

- Several power operated valves were identified with safety functions in both the open and closed directions, but only stroke timed in one direction. This is inconsistent with Code requirements. There can be significant differences between the open and close stroke for air or hydraulic operated valves. Although there may not be significant differences for motor-operated valves, the intent of the Code is to perform stroke time testing in each safety direction. This was based on the Code clarification in OM-10 and the NRC's response to question group 41 in NUREG-1482.

- At several plants it was identified that the maximum stroke time allowed by the IST program exceeded the limits identified in the SAR or other design documents. Licensees must ensure that maximum stroke times developed by use of Code multipliers are within the limits established by the plant's design analysis.

- A design change was completed that changed the gear ratio of a motor-operated valve, reducing the valve's stroke time. However, the valve was not re-baselined and the closing time acceptance criteria was not changed. There were no controls to ensure that the IST coordinator was notified of design changes that could impact valve performance.

- Though not required, remote position indication testing should include verifying both the open and closed lights even if the valves only have a safety function in one direction.

Leak Rate Testing

The Code requires that Category A valves be leak rate tested once every two years. Some form of leakage testing may be used for Category C valves to verify closure where the licensee establishes the acceptance criteria.

- Pressure isolation valves were not individually leak tested as discussed in Position 4 of Generic Letter (GL) 89-04 and NUREG-1482, Section 4.4.7.

- Leakage tests of the safety injection tank outlet check valves and the HPSI check valves were not being corrected for maximum service pressure versus test differential pressure in accordance with the requirements of the Code.

- Category C check valves on the instrument air supply line to turbine-driven AFW pump discharge valves were tested without using quantitative leak rate acceptance criteria. The acceptance criteria used was

qualitative and relied on the judgement of the person performing the test. Although these valves are Category C valves, the check valves on the air supply line must close and maintain a certain amount of leak-tightness so the nitrogen gas supply does not leak back through the air system, which would render the discharge valves inoperable. This is discussed in Position 3 of GL 89-04.

Test Acceptance Criteria

The Code requires that check valves be exercised to the position required to fulfill their safety function. To verify the open safety function, GL 89-04, Position 1, states that a check valve's full-stroke capability may be verified by passing the maximum accident flow through the valve. The closure safety function may be verified by a leakage test or other qualified method that demonstrates check valve closure.

- Many licensee's did not have adequate acceptance criteria for verifying the open function of check valves to meet the guidance of Position 1 in GL 89-04. This included not using the maximum accident flow rate, measuring total flow in systems with multiple parallel lines to verify full-stroke exercise of individual check valves in those systems, using pump test flow rates that were less than the check valve maximum accident flow rates, and using pump flow curve acceptance criteria that are less than the check valve maximum accident flow rate. Licensees should ensure that all accident scenarios are reviewed to identify the maximum accident flow rate.

- The acceptance criteria for the HPCI turbine steam exhaust check valves was to monitor the steam exhaust pressure high alarm during the HPCI pump test and ensure that an alarm was not initiated. Since the alarm was set at 150 psig and normal turbine discharge pressure was 20 psig, the acceptance criteria was not adequate to address valve degradation.

- Check valves were not specifically identified in the surveillance procedures or did not have quantifiable acceptance criteria. When a procedure is used to take credit for verifying the open or closed function of a check valve, it must be documented in the procedure with adequate acceptance criteria.

- The use of audible indication to verify check valve closure by itself does not meet the intent of the Code as a positive means and should not be considered acceptance criteria. This issue is discussed in NUREG-1482, Section 4.1.3, Item (3), Page 4-7.

Disassembly and Inspection of Check Valves

Check valves that cannot be full-stroke exercised during any mode of plant operation may be disassembled and inspected (DI) on a sampling basis during refueling outages in accordance with GL 89-04, Position 2. OM-10 allows the disassembly of individual check valves when there are no other means to verify a full-stroke exercise.

- A number of licensee's were not performing and/or documenting a partial stroke of valves after DI or on a periodic basis, even though testing was possible. When a valve is disassembled and inspected per Position 2 of GL 89-04, a partial stroke should be performed if possible.

- Several licensee's were not manually stroking the valve disc and/or documenting the results as required when a valve is DI per Position 2 of GL 89-04.

- IST program was not properly evaluated when swing check valves were replaced with nozzle check valves. The swing checks were verified closed by DI on a sampling basis. The nozzle checks were to be tested with a test probe that can be inserted on the upstream side of the valve to exercise the valve disc. The licensee stated this was a partial DI (removing a plug) and would be done on a the same DI sampling basis. The use of the test probe does not meet the intent of Position 2 for DI and the valves need to be tested with the probe at the Code frequency.

Test Methodology

- The licensee was performing reverse flow testing of stop check valves by using the manual operator. Since the valves were required to close on reverse flow, use of the manual operator was not the proper test method as discussed in the NRC's modified response to question group 25 in NUREG-1482.

- The licensee implemented a non-intrusive techniques (NITs) sampling program for several check valves that did not meet the guidance of NUREG-1482, Section 4.1.2. The licensee intended to use NITs one valve every refueling outage to verify a full open stroke. Since the valves were testable on a quarterly basis, the sampling program must be established on the same frequency (ie. one valve every quarter).

Relief Valve Testing

OM-10, Section 4.3.1, states that safety and relief valves shall meet the inservice test requirements of OM-1. IST programs that were developed to the 1983 or 1981 edition of ASME Section XI use the requirements of PTC 25.3-1976 for relief valve testing.

- Incorrect set points were used in the relief valve maintenance procedure as back pressure was not accounted for in determining the correct setpoint.

- Relief valve testing as required by OM-1 states that valves should be tested under the similar conditions that they would be expected to see during operation or accident conditions. If testing was performed under other conditions, such as at ambient temperature or with a different test media, then certified correlations for setpoint testing need to be developed. The question of what constitutes a certified correlation is unclear at this time and is being addressed through a Code inquiry.

- Licensee's were not always following OM-1 guidance on the sequence of relief valve testing. The purpose of the test sequence is to ensure as-found test results are obtained.

- The test procedure for the main steam safety valves stated that setpoint testing should be stopped and the valve repaired when valve seat leakage is excessive. The Code of record was the 1983 edition of ASME Section XI. The test procedure did not define excessive seat leakage or require additional testing when a test failure was observed.

MISSED SURVEILLANCES

The Code requires that pumps and valves be subject to periodic testing every three months. Certain valves also have additional leak rate and position indication test requirements that are required to be performed once every two years. A common theme with many of the findings in this section is that missed surveillances are frequently caused by inadequate surveillance procedures and administrative controls. This could be due to specific program functions distributed between a number of individuals or departments without an overall program coordinator.

- Two solenoid valves on a charging pump suction vent line were installed in a modification to the charging system in 1990. Reviews by the IST engineer and quality assurance did not identify that these valves should be included in the IST program and stroke timed. Inspectors discovered that the valves were not being stroke tested in a 1993 inspection.

- A licensee quality assurance audit identified reactor makeup water pumps as being within the IST program scope and were added to the licensee's IST program. However, four years later, the NRC identified that surveillance test procedures had not been developed and the pumps had not been tested.

- Vibration measurements were not being taken on the boric acid transfer pumps during quarterly testing. The pumps were added to the IST program during a program update. A relief request was submitted to not require vibration testing due to the pumps being encapsulated in heat tracing. The relief was denied and modifications to the pumps were initiated to provide access for performing vibration testing. Two and a half years after the relief was denied, modifications to provide a safe environment for personnel to conduct vibration measurements were not completed.

- Two check valves were identified in the IST program as being partially-stroke tested at cold shutdowns and DI during refueling outages in accordance with GL 89-04, Position 2. However, the licensee was only performing the partial-stroke tests and no inspections of the check valves had been performed.

- Several test procedures were not revised or written to perform testing of components added to the IST program. The IST program ten-year update added components and/or additional safety functions to numerous components. These safety functions were not verified by test procedures a year after the program was implemented.

- Several valves were installed during a modification in 1985 and included in the IST program since 1989. A 1993 inspection revealed that the required quarterly IST had not been performed since the valves were installed.

- Several service water valves included in the unit 1 surveillance procedure were common to both units. With unit 1 in a outage, the plant scheduler decided the surveillance was not required (assumed system out of service) until unit 1 was ready to resume power operations. As a result, the quarterly testing frequency of the service water valves was exceeded.

TEST RESULT ANALYSIS

The Code requires that test results be compared to the established acceptance criteria to determine if the component has degraded. Acceptance criteria includes limits for requiring increased testing and declaring the component inoperable. Although not required by the Code, a good engineering practice would consist of trending test results to predict potential degradation problems. The following is a list of test result analysis issues identified during NRC inspections.

- Several pumps were discovered to fall within the required action range, but were not declared inoperable because the operations department suspected that the discharge pressure instruments were out of calibration. The instruments were checked and determined to be within their calibration tolerances. A flow instrument, however, was later determined to be out-of-tolerance on the low side. An evaluation performed using the corrected flow data, confirmed that the pumps were operating in the acceptable range.

  Although the pumps were later determined to be operable based on the out-of-tolerance flow instrument, the Code requires that when the tests results are determined to be in the required action range, the pumps shall be declared inoperable. Subsequent to the declaring a pump inoperable, instrument inaccuracies can be taken into account to verify that the pump is in fact operable and capable of performing its intended function.

- A pump was placed on an increased testing frequency after testing revealed a pump dp measurement in the alert range. The increased test frequency continued for one year when it was discovered that the high dp was due to steam turbine operation at a greater speed than the reference value. Numerous other pumps and valves were on increased frequency at this particular plant with the licensee taking little or no action to correct the conditions that caused the equipment to be on increased frequency testing. Although the Code does not limit this practice, a proactive IST program would investigate components operating in the alert range and implement corrective action to return component performance parameters to the acceptable range.

- Valves were not put on an increased testing frequency when their measured stroke time increased 25% from the previous test. IWV-3417 established limits to compare the stroke time test results of power-operated valves to results from the previous test. When these limits are exceeded, the test frequency is required to be increased.

- Additional relief valves in the same group were not tested after a relief valve set pressure test failure. The purpose of testing additional valves is to ensure similar valves are also not exhibiting set pressure problems. This is a requirement under the section of IWV that references PTC 25.3-1976. However, OM-1 does not specifically require additional testing for valve failures except as discussed under replacement with pretested valves. This was not the intent of OM-1 as the ASME OMc Code-1994 Addenda to the ASME OM Code-1990 addressed this discrepancy. This is discussed in NUREG-1482, Section 4.3.9.

- No corrective actions were taken when several valves exceeded the leak rate limits

established in the local leak rate test (LLRT) procedures. The licensee was performing testing per Appendix J as allowed by 10 CFR 50.55a, however, 10 CFR 50.55a(b)(vii) takes exception to Appendix J with respect to corrective action. This exception requires licensees to follow the Code requirements to correct individual valve leakage versus basing corrective actions on the overall $0.6L_a$ leakage rate. This requirement was not specified in the LLRT procedures to ensure it was adequately addressed by the IST group.

- The licensee was not evaluating IST vibration measurement results that met the acceptance criteria, but had significantly changed from previous test results to determine if degradation was occurring. Although not required by the Code, trending of data could be a useful tool to identify component degradation to initiate root cause evaluations and corrective actions before the component becomes inoperable.

RELIEF REQUESTS AND DEFERRED TESTING

When testing of an IST component in accordance with the Code requirements is impractical, results in a hardship, or an alternative method of testing provides an acceptable level of safety, the regulations allow relief to be requested from the Code requirements. Inspectors review granted reliefs and approved alternatives to verify the basis for requesting relief and the implementation of provisions imposed by the NRC. Guidance on preparing relief requests is included in NUREG/CR-6396, "Examples, Clarifications and Guidance on Preparing Requests for Relief from Pump and Valve Testing Requirements." Inspectors also review and evaluate the deferred test justifications contained in IST programs.

Provisional Relief Request not Implemented

- A relief request was approved to perform leak rate testing of pressure isolation valves using the leak testing requirements of 10 CFR 50, Appendix J, with the provision that the licensee meet the requirements of ASME Section XI, Paragraphs IWV-3426 and IWV-3427(a) as stated in the guidance provided in GL 89-04, Position 10. An updated relief request in the licensee's IST program did not contain the additional requirements imposed in the provisional relief request.

- A relief request was granted to perform vibration testing to measure pump vibration amplitude in units of in/sec provided that alert and action levels were at least as stringent as required by the Code. The licensee's implemented alert and required action ranges, however, did not met the Code requirements.

- A relief request was granted to allow only a 1-minute waiting period after flow rate for the standby liquid control pump was established prior to recording data. However, the test procedure did not specify the waiting period required by the relief request.

Reliefs Implemented Without Approval

- Recent IST inspections at three sites revealed that relief requests submitted to the NRC between 1985 and 1993 had been implemented without prior NRC approval. There were questions whether some of the relief requests were impractical. One

relief request was withdrawn after it was determined that testing previously asserted as impractical, was in fact assessed by the licensee to be practical.

- The licensee implemented a relief request that revised vibration limits for several pumps without prior NRC approval. The relief request was denied twice by the NRC due to lack of information.

- A licensee was using OM-10 without an approved relief request. At the time of this inspection, OM-10 had not been endorsed by the NRC through the regulations.

Other Relief Request Issues

- The acceptance criteria used for stroke time testing of power operated valves was not in compliance with the methodology stated in OM-10. The relief request was submitted to perform stroke time testing using reference values versus the methodology committed to in a previous Code edition. The regulations state that subsequent editions of the Code incorporated by reference into 10CFR50.55a, may be used provided that all related requirements of the respective editions are met. The stroke time limits referenced in the relief request were greater than those allowed by the Code. In addition, the maximum stroke time was determined by multiplying the reference value by two, which for motor-operated valves was excessive.

Deferred Test Justifications

The Code allows deferral of testing for valves that are impractical to test at power to cold shutdowns or refueling outages. Deferred test justifications which are included in the IST programs are required to describe the impracticality for testing at the Code frequency. Inspectors review these justifications during IST inspections. The basic deficiency in these justifications noted at a number of plants is an inadequate description of the impracticality for performing inservice testing quarterly or during cold shutdowns. In a few instances, the licensee could not justify the deferred test frequency and committed to change their IST program to reflect the new test frequency and delete or revise the deferred test justification. In most cases, only a justification revision was needed to provide a clearer description of the impracticality.

CONCLUSION

This paper has attempted to summarize the inspection findings contained in NRC inspection reports and other sources dedicated to the area of IST. While the authors make no attempt to assess the current state of IST programs at US commercial nuclear power facilities, a majority of the findings discussed in this paper were the result of problems with administrative and surveillance procedures, detail of systems reviews, and implementation of regulatory requirements. It is hoped that a review of the findings in this paper by individuals responsible for the implementation of their IST program may aid in alerting them to potential weaknesses at their plant and implement any necessary changes to their IST program and plant procedures.

REFERENCES

NRC Generic Letter 89-04, "Acceptable Inservice Testing Programs" and associated "Minutes of the Public Meeting on Generic Letter 89-04," issued April 3, 1989.

NRC Inspection Procedure 73756, "Inservice Testing of Pumps and Valves," revised July 27, 1995.

NRC Temporary Instruction 2515/114, "Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs," issued January 15, 1992.

NRC Temporary Instruction 2515/110, "Performance of Safety-Related Check Valves," issued November 19, 1991.

NRC Bulletin 88-04, "Safety Related Pump Loss," issued May 5, 1988.

NUREG-1482, "Guidelines for Inservice Testing at Nuclear Power Plants," issued April 1995.

NUREG/CR-6396, "Examples, Clarifications and Guidance on Preparing Requests for Relief from Pump and Valve Testing Requirements," issued February 1996.

American Society of Mechanical Engineers (ASME) Boiler and Pressure Vessel Code, Section XI, Subsections IWP (for pumps) and IWV (for valves), 1981, 1983, and 1986 Editions.

ASME/ANSI Operations and Maintenance Standard Part 1 (OM-1), "Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices," 1981 Edition.

ASME/ANSI Operations and Maintenance Standard Part 6 (OM-6), "Inservice Testing of Pumps in Light-Water Reactor Power Plants," 1988 Edition.

ASME/ANSI Operations and Maintenance Standard Part 10 (OM-10), "Inservice Testing of Valves in Light-Water Reactor Power Plants," 1988 Edition.

Performance Test Codes, PTC 25.3-1976, "Safety and Relief Valves," 1976 Edition.

Title 10 of the Code of Federal Regulations, Section 50.55a, "Codes and Standards," 1995 revision.

MOV Reliability Evaluation and Periodic Verification Scheduling

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ABSTRACT

The purpose of this paper is to establish a periodic verification testing schedule based on the expected long term reliability of gate or globe motor operated valves (MOVs). The methodology in this position paper determines the nominal (best estimate) design margin for any MOV based on the best available information pertaining to the MOVs design requirements, design parameters, existing hardware design, and present setup. The uncertainty in this margin is then determined using statistical means. By comparing the nominal margin to the uncertainty, the reliability of the MOV is estimated. The methodology is appropriate for evaluating the reliability of MOVs in the GL 89-10 program. It may be used following periodic testing to evaluate and trend MOV performance and reliability. It may also be used to evaluate the impact of proposed modifications and maintenance activities such as packing adjustments. In addition, it may be used to assess the impact of new information of a generic nature which impacts safety related MOVs.

POSITION

Overview

The amount of margin required to adequately setup an MOV varies significantly from valve to valve. The required margin depends on the type of testing performed, the accuracy to which test data is measured, the predictability of the valve design's performance, and other factors. All of the uncertainties associated with the MOV's performance are combined statistically to determine the total effect of these uncertainties. This total uncertainty is compared to the available margin to determine an MOV's expected reliability. This reliability value is combined with the safety significance of the MOV, as determined in the Probabilistic Risk Assessment (PRA) for the station, to determine the adequacy of the MOV's setup and to determine what type and periodicity of periodic verification testing is appropriate. Major attributes of this methodology are outlined below:

- All Sources of uncertainty should be considered without giving precedence to equipment inaccuracy and torque switch repeatability.
- Each Uncertainty should be quantified using best available data. When actual field test data is available, it is preferred over laboratory test data and arbitrary vendor guidance.
- The methodology used to sum uncertainties should be based on sound statistical methods.
- MOV reliability is determined by comparing existing (nominal) margin to the margin which would provide a 2 sigma (97.6%) confidence level.

- The methodology supports trending of reliability.
- The methodology identifies the most significant sources of uncertainty which impact an MOVs reliability.

Uncertainties

The following is a list of the parameters and effects which provide the most significant contributions to the required margin for an MOV: (This list identifies whether the uncertainty associated with the parameter is of a bias nature, random nature, or both.)

	Nominal Value	Bias Value	Random Uncertainty
Valve Factor	X		X
Torque and Thrust	X		X
Torque Switch Repeatability			X
Spring Pack Relaxation		X	
Load Sensitive Behavior		X	X
Stem Factor	X		X
Packing Load	X		X
Inertia Factor	X		X

An example of an uncertainty for which both a bias effect and a random effect exist is Load Sensitive Behavior. A normal distribution is assumed for this uncertainty as shown below. The bias value for load sensitive behavior is set equal to the greater of zero or the measured value (when available), or the average value observed for the stem lubricant used on the MOV. For DP tested MOVs, the two sigma value is set equal to the bias value plus 5%. For non-DP tested MOVs, the two sigma value is set equal to the average value plus two times the standard deviation for the lubricant. The example below corresponds to the assumed distribution for a non-DP tested MOV using a lubricant which has an average load sensitive behavior of 5% with a standard deviation of 8%.

An example of a parameter for which random uncertainty exists about a nominal value is Valve Factor. For DP tested MOVs, the uncertainty is due to measurement inaccuracy and potential valve factor variation. For non-DP tested MOVs, the nominal valve factor is based on the performance of similar valves and the uncertainty is based on the observed valve to valve variation in valve factor for

similar valves. The example in Figure 1 corresponds to a non-DP tested MOV for which the average valve factor for similar valves is 0.45 with a standard deviation of 0.15.

MOV Safety Significance Rankings
The MOVs at ComEd stations are separated into four categories. These categories are used by this methodology to ensure that MOV margin enhancement actions are properly identified and prioritized. A description of these safety significance categories is provided below:

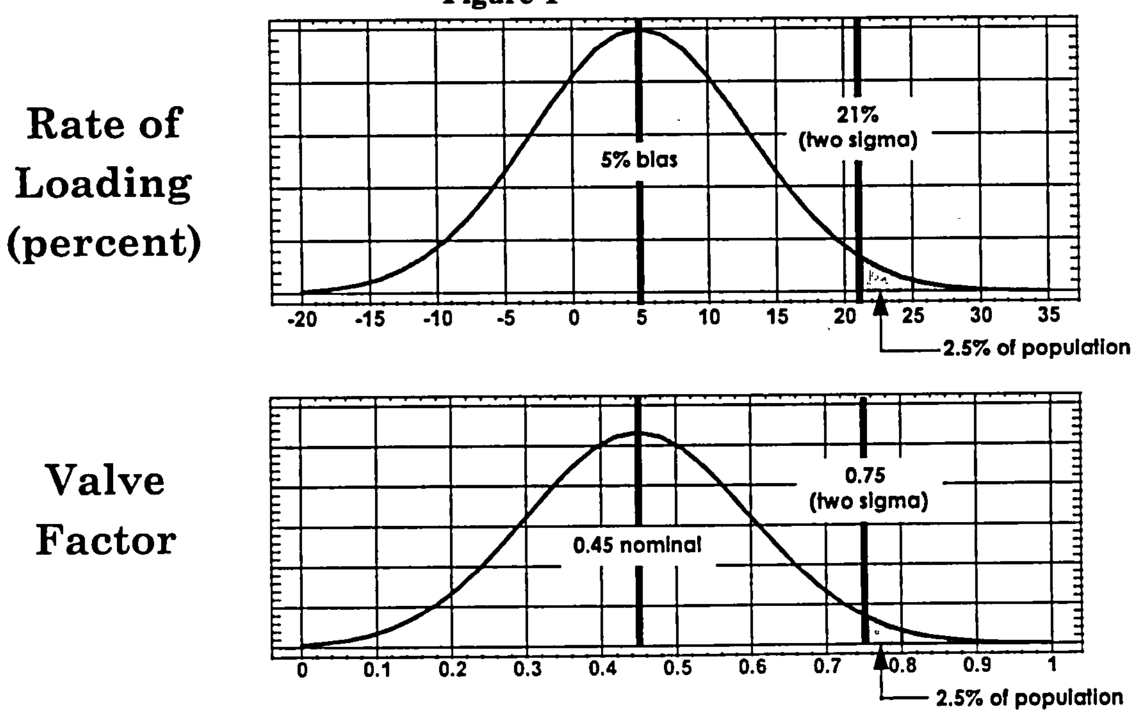
High Safety Significance:
MOVs which are modeled in the PRA and which contribute significantly to the safety of the plant.

Medium Safety Significance:
MOVs which are modeled in the PRA, but which are not in the High Safety Significance category. In addition, some MOVs which are not modeled in the PRA have been placed in this category due to the importance placed on the MOVs during expert panel reviews of the MOV population.

Low Safety Significance:
MOVs which are not modeled in the PRA analysis, but which may be required to change position in response to a design basis event.

Low-Low Safety Significance:
MOVs which are not modeled in the PRA analysis and which are expected to be in their safety position at the time of an design basis event. Examples of MOVs in this category include normally closed containment isolation valves.

Figure 1



Calculating MOV Reliability

The methodology used to calculate MOV Reliability can be summarized in the following three equations.

Actual Margin Equation:

(1)

$$Margin_{nominal} = \frac{AvailableCapability - [NominalRequirement + BiasAdjustments]}{NominalRequirement + BiasAdjustments}$$

2 Sigma Margin Equation:

A reduced margin value is determined for each random uncertainty by calculating the margin when the parameter is set equal to its 2 sigma (conservative) value. The Δ Margin values are then determined for each parameter by calculating the difference between the nominal margin and the margin with the parameter at its two sigma value. The Δ Margin values are then summed as follows to determine the overall margin required to achieve a 2 sigma confidence value.

(2)

$$Margin_{2\Sigma} = \sqrt{\sum_{AllUncertainties} (\Delta MarginWhenUncertaintyIsSetTo2SigmaValue)^2}$$

Reliability Equation:

(3)

$$Reliability = InvNorm\left[\frac{Margin_{nominal} x2}{Margin_{2\Sigma}}\right]$$

Example:

Nominal Margin between CST thrust and MRT	**37%**	
Margin at 2σ valve factor	**11%**	**(26% reduction)**
Margin at 2σ_packing load	**31%**	**(6% reduction)**
Margin at high stem mu degradation (2σ)	**27%**	**(10% reduction)**
Margin at high rate of loading (2σ)	**26%**	**(11% reduction)**
Margin at bounding torque switch repeatability	**31%**	**(6% reduction)**
Margin at 2σ equipment inaccuracy	**24%**	**(13% reduction)**

Sum of uncertainties (at 2σ) = $\sqrt{26^2+6^2+10^2+11^2+6^2+13^2}$ **= 34%**

Therefore, actual sigma level = 2 x (37/34) = 2.18

Reliability Level: 2.18 sigma corresponds to a 98.5% reliability level for a one-sided normal distribution.

Margin Descriptions

Several margins must be evaluated to determine the adequacy of the MOV setup. Margins which are a measure of the reliability of the MOV to perform its design function are categorized as Design Function Margins (DM). Margins associated with actuator torque output capability are categorized as Motor/Gearing Capability Margins (MM). Finally, margins associated with valve or actuator structural limits are categorized as Structural Margins (SM).

The acceptance criteria used for Design Function Margin should be a function of MOV safety significance. For Motor/Gearing Capability Margin and Structural Margin, the standard industry practice of requiring a 2 sigma (97.7% reliability) value for these margins is used as a long-term design requirement.

Margin	Description/Notes
TST/MRT_s	Margin between Torque Switch Trip Thrust and Thrust Required to Seat under Design Conditions (DM)
MGC_C/TST	Margin between Motor/Gearing Capability and Torque Switch Trip Torque (MM)
MGC_C/MRT_s	Margin between Motor/Gearing Capability and Thrust Required to Seat under Design Conditions (DM) See Note 1.
MGC_C/MaxOpen	Margin between Motor/Gearing Capability and Thrust Required to Open the MOV (DM & MM) See Note 2.
Structural/MaxClose	Margin between the most limiting structural limit and the maximum closing thrust (SM)
Structural/MaxOpen	Margin between the most limiting structural limit and the maximum opening thrust (SM) See Note 2.

Note 1: This margin is only applicable to the setup of limit closed MOVs.

Note 2: The thrust required to open is the greater of the static unwedging thrust or the thrust required to overcome open packing load plus open DP load.

Long Term Margin Requirements (for Design Function Margin)

The design criteria for MOVs is based on ensuring that the MOV reliability is consistent with assumptions in the Probabilistic Risk Assessment (PRA) for the station. Typically, reliability values of approximately 99.7% are assumed for MOVs in a PRA.

- This failure rate includes failures other than those attributable to inadequate setup such as MCC problems.
- Therefore, the failure rate due to setup must be less than 0.3%.

High and medium safety significant MOVs are modeled in the ComEd PRAs. Therefore, for these MOVs, ComEd will use 99.87% reliability for the High Margin cutoff which corresponds to the desired long-term design margin. This is equivalent to 3.00 sigma.

Low and low-low safety significant MOVs are not modeled in the ComEd PRAs and do not contribute significantly to plant safety/reliability. A 97.7% reliability is

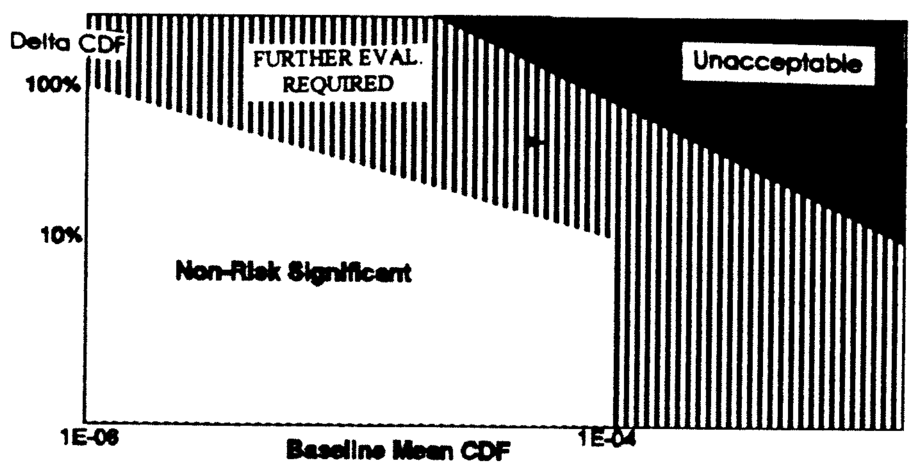
considered adequate as a long-term design requirement for these MOVs. This is consistent with error analysis assumptions for typical diagnostic equipment used to monitor these MOVs performance/setup.

Acceptance Criteria for MOVs at Reduced Design Function Reliability Levels

The section above discussed the long-term design margin criteria for MOVs. A criteria for the minimum margin required to for short term operability must also be defined. The EPRI/NEI criteria provided in Figure 4-1 of Reference 1 is used as the basis for determining whether degraded reliability (below the 99.7% reliability assumed in the PRA) for a specific MOV significantly impacts plant safety.

The MOV's design function reliability is calculated using the methodology discussed in section 4 above. To apply the EPRI/NEI criteria, this reliability value must be converted into a Delta CDF (percent change in core damage frequency). At a 99.7% reliability for the MOV, the Delta CDF is zero since this is the MOV reliability used in the PRA. Delta CDF values corresponding to 0% reliability for each MOV are tabulated for the PRA analysis (Delta CDF 0 values). By interpolating between these values, the Delta CDF can be determined for reliability values between 0% and 99.7%. The delta CDF is compared to Figure 4-1 (below) of Reference 1 to determine whether the reduced reliability is risk significant.

FIGURE 2



Selecting General Reliability Limits for MOV Safety Significance Groups

The acceptance criteria in the section above is valve specific in nature since each MOV has an unique Delta CDF 0 value. Furthermore, many safety-related MOVs are not modeled in the PRA and do not, therefore, have Delta CDF 0 values tabulated. To simplify the operability criteria, ComEd has used the following process to create short-term acceptance criteria for design function reliability:

- For ComEd, the highest baseline CDF value is 3.1 E-05 among the six nuclear stations.

- For this baseline CDF of 3.1 E-05, Figure 4-1 of Reference 1 shows that changes in CDF below 20% are considered non-risk significant.

- The MOVs are grouped in safety significance categories: High, Medium, Low, and Low-Low. Based on the "Delta CDF 0" values, the lowest reliability values which are in the non-risk significant region are determined for each safety significance category.

    - ♦ For MOVs in the High Safety Significance category, a Delta CDF 0 of 6.0 is bounding. For MOVs in the Medium Safety Significance category, a delta CDF 0 of 2.0 is bounding.

    - ♦ Low and Low-Low Safety Significant MOVs are not modeled in the PRA because their contribution to plant safety is not significant. It is considered reasonable to assume that an MOV reliability rate of 0% for any of these MOVs would not double the overall core damage frequency for a plant. For this reason, a delta CDF 0 of 1.0 is considered conservative for MOVs in these categories.

- The equation below uses interpolation to determine the reliability cutoff limit for each safety significance group.

$$\frac{Baseline\ Reliability - Reliability\ Cutoff}{Baseline\ Reliability - 0\%} = \frac{20\%\ (delta\ CDF)}{CDF0_{\max}^{safety\ sig\ category}} \qquad (4)$$

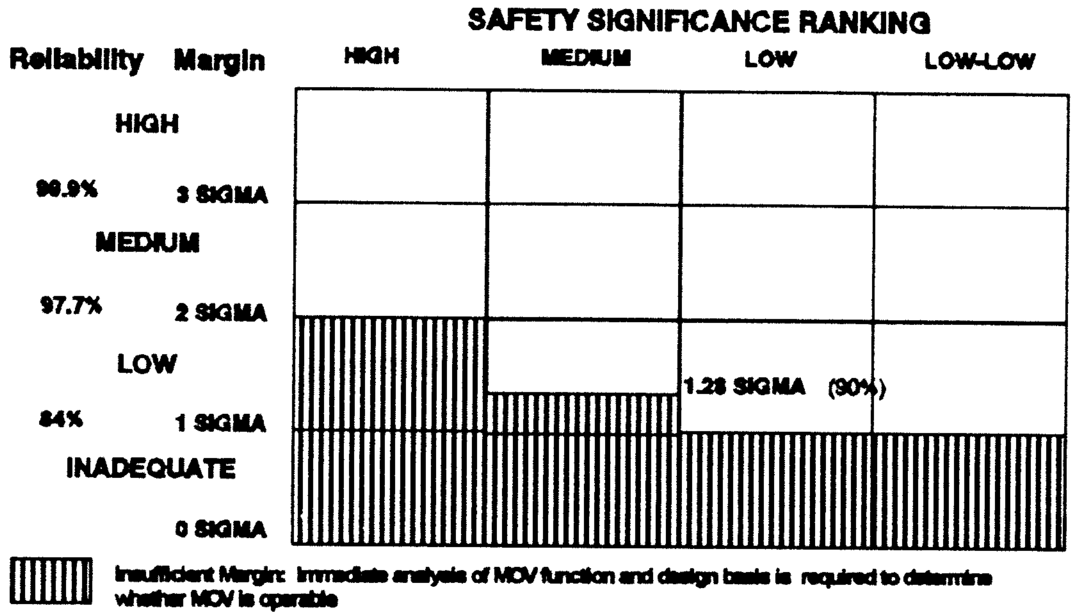
$$\frac{99.7\% - Reliability\ Cutoff}{99.7\% - 0\%} = \frac{20\%\ (delta\ CDF)}{600\%^{high\ s\ s\ category}}, \quad Reliability\ Cutoff = 96.4\% \qquad (5)$$

$$\frac{99.7\% - Reliability\ Cutoff}{99.7\% - 0\%} = \frac{20\%\ (delta\ CDF)}{200\%^{medium\ s\ s\ category}}, \quad Reliability\ Cutoff = 89.8\% \qquad (6)$$

$$\frac{99.7\% - Reliability\ Cutoff}{99.7\% - 0\%} = \frac{20\%\ (delta\ CDF)}{100\%^{low/low\ low\ s\ s\ category}}, \quad Reliability\ Cutoff = 79.8\% \qquad (7)$$

The cutoff limits have been simplified to 97.7% (2 sigma), 90% (1.28 sigma), and 84% (1 sigma). The resulting matrix below shows the area of concern for which an operability review should be performed.

FIGURE 4



Periodic Performance Verification:

- For all safety related MOVs, periodic verification testing must be scheduled. Both the type of testing and the frequency of testing should be based on the amount of margin in the MOV's setup and on the safety significance of the MOV.

- The following types of periodic testing are to be used at ComEd Stations:

(1) **MPM Testing** is thrust verification testing using the VOTES Motor Power Monitor diagnostic testing system. For acceptance criteria, the MOV reliability should be evaluated using the measured value for C14 thrust, an estimated value for C16, and a closing equipment inaccuracy of approximately +/- 16%.

(2) **Static VOTES Testing** is testing with the VOTES diagnostic testing system under static (zero flow, zero differential pressure across MOV) conditions.

(3) **DP VOTES Testing** is testing with the VOTES diagnostic testing system under dynamic (flow and differential pressure across MOV) conditions. In addition, static testing should be performed on the MOV at a time close to the DP test (system conditions permitting).

(4) **VTC VOTES Testing** is testing with the VOTES diagnostic testing system including a VOTES Torque Cartridge under static (zero flow, zero differential pressure across MOV) conditions. Other direct torque measurement devices such as stem mounted strain gauges may be used in lieu of the VTC.

- Tables 1 through 4 can be used to determine the frequency and type of diagnostic testing to be performed. These testing matrices are designed to ensure that more comprehensive and accurate testing methods are used on MOVs which do not meet the long-term design function margin or motor/gearing capability margin requirements. Performing more comprehensive and accurate testing yields to valuable results:

(1) Higher accuracy / more comprehensive testing should identify MOV degradation before MOV operability limits are violated.

(2) Higher accuracy / more comprehensive testing will reduce the uncertainty in the margin calculation. For MOVs which are not degrading, this should improve the predicted reliability of the MOV and may place it in a position on the matrix for which less comprehensive testing is allowed in the future. For example, performing a VTC test once may be all that is needed to push an MOV into the high margin portion of the testing matrix.

TABLE 1

HIGH PRA MOVs		**MGC MARGIN RELIABILITY**		
		≥97.7%	<97.7% & ≥90%	<90%
DESIGN FUNCTION RELIABILITY	≥99.9%	Static VOTES test* every 3 outages	Static VTC test every 3 outages	Static VTC test every outage
	<99.9% ≥97.7%	DP Test** (if practicable) every 3 outages	Static VTC test and DP test** (if prac.) every 3 outages	Static VTC test every outage, DP test** (if prac.) every 3 outages
	<97.7%	DP test** (if practicable) every outage	Static VTC test every 3 outages, DP test** (if prac.) every outage	Static VTC test and DP test** (if prac.) every outage

* A Static MPM test may be substituted for the Static VOTES test provided structural margin reliability is greater than or equal to 99.9%.

** If Design Function Reliability is limited solely by motor/gearing capability to unseat the MOV under static conditions, then VTC testing in lieu of DP testing is appropriate.

TABLE 2

MEDIUM PRA MOVs		MGC MARGIN RELIABILITY		
		≥97.7%	<97.7% & ≥90%	<90%
DESIGN FUNCTION RELIABILITY	≥99.9%	Static VOTES test* every 3 outages	Static VTC test every 3 outages	Static VTC test every outage
	<99.9% ≥90%	DP test** (if practicable) every 3 outages	Static VTC test and DP test** (if prac.) every 3 outages	Static VTC test every outage, DP test** (if prac.) every 3 outages
	<90%	DP test** (if practicable) every outage	Static VTC every 3 outages, DP test** (if prac.) every outage	Static VTC and DP test** (if prac.) every outage

* A Static MPM test may be substituted for the Static VOTES test provided structural margin reliability is greater than or equal to 99.9%.

** If Design Function Reliability is limited solely by motor/gearing capability to unseat the MOV under static conditions, then VTC testing in lieu of DP testing is appropriate.

TABLE 3

LOW PRA MOVS		MGC MARGIN RELIABILITY		
		≥97.7%	<97.7% & ≥84%	<84%
DESIGN FUNCTION RELIABILITY	³99.9%	Static VOTES test* every 6 outages	Static VTC test every 3 outages	Static VTC every outage
	<99.9% ≥97.7%	Static VOTES test* every 3 outages	Static VTC test every 3 outages	Static VTC test every outage
	<97.7% ≥84%	DP Test** (if practicable) every 3 outages	Static VTC and DP Test** (if prac.) every 3 outages	Static VTC test every outage, DP test** (if prac.) every 3 outages
	<84%	DP test** (if practicable) every outage	Static VTC every 3 outages, DP test** (if prac.) every outage	Static VTC and DP test** (if prac.) every outage

* A Static MPM test may be substituted for the Static VOTES test provided structural margin reliability is greater than or equal to 99.9%.

** If Design Function Reliability is limited solely by motor/gearing capability to unseat the MOV under static conditions, then VTC testing in lieu of DP testing is appropriate.

TABLE 4

LOW-LOW PRA MOVS		MGC MARGIN RELIABILITY ≥97.7%	MGC MARGIN RELIABILITY <97.7% & ≥84%	MGC MARGIN RELIABILITY <84%
DESIGN FUNCTION RELIABILITY	≥97.7%	Static VOTES test* every 6 outages	Static VTC every 6 outages	Static VTC every 3 outages
	<97.7% ≥84%	Static VOTES test every 3 outages	Static VTC test every 3 outages	Static VTC test and DP test** (if prac.) every 3 outages
	<84%	DP test** (if practicable) every 3 outages	Static VTC test and DP test** (if prac.) every 3 outages	Static VTC and DP test** (if prac.) every outage

* A Static MPM test may be substituted for the Static VOTES test provided structural margin reliability is greater than or equal to 99.9%.

** If Design Function Reliability is limited solely by motor/gearing capability to unseat the MOV under static conditions, then VTC testing in lieu of DP testing is appropriate.

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A DISCUSSION OF SYSTEM RELIABILITY AND THE RELATIVE IMPORTANCE OF PUMPS AND VALVES TO OVERALL SYSTEM AVAILABILITY

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ABSTRACT

An analysis was undertaken to establish preliminary trends for how component aging can effect failure rates for swing check valves, centrifugal pumps and motor operated valves. These failure rate trends were evaluated over time and linear aging rate models established. The failure rate models were then used with classic reliability theories to estimate reliability as a function of operating time. Reliability theory was also used to establish a simple system reliability model. Using the system model, the relative importance of pumps and valves to the overall system reliability were studied.

Conclusions were established relative to overall system availability over time and the relative unavailabilities of the various components studied.

NOMENCLATURE

λ	=	Failure rate (failures/year)
T	=	Mission time (years)
R_{cv}	=	Reliability of check valve
R_{pump}	=	Reliability of pump
R_{MOV}	=	Reliability of Motor Operated Valve
R_A	=	Reliability of train A
R_B	=	Reliability of train B
$R_{SI,System}$	=	Reliability of overall system

INTRODUCTION

Definition of Reliability. Reliability has been defined as "the probability of a device performing its purpose adequately for the period of time intended under the stated operating conditions"[1]. Probability refers to the chance, or likelihood, that the device will work properly. In fact, the terms chance or likelihood can be used as synonyms to probability. Probability being measured as a decimal ratio, between 0 and 1, and is usually expressed as a percent. Since probability is the chance that something will occur (chance of a success or a failure, depending on what is desired), it is calculated as the ratio of the successes to the total number of possible occurrences or trials.

Failure Rate Calculations. Failure rate is defined as the number of failures per unit of time. When the failure rate of a given type of component is calculated based on failure population data, the number of units in the population (n) and the time of the study (t) are needed along with the quantity of failures (f). The average failure rate is given by λ = f/nt.

Thus if you have seven (7) failures in a total of 231 check valves during an 8 year period where the actual valve-years was 1894 then the average failure rate =7 / 1894 = 0.0037/year. If only one of these failures was for the check valve to stick open then the average failure rate for stuck open check valves is given as 1 / 1894 = 0.0005/year.

Exponential Failure Law. The exponential has been the most used distribution in all aspects of reliability. Because it is the easiest to use and because most complex systems exhibit exponential qualities with an average failure rate to approximate a constant failure rate, it will be used in the following analysis calculations. The general equation for reliability [2] is as follows:

$$R(t) = \exp\left[-\left(\int_0^t \lambda(x)\,dx\right)\right] \tag{1}$$

when $\lambda(x)$ is a constant, $\lambda(x) = \lambda$, the reliability equation becomes

$$R = e^{-\lambda T} \tag{2}$$

The T in the preceding formula is a probability time sample and is usually determined to be the mission time.

If the failure rate can be approximated using a linear aging rate over the mission time ($\lambda = a\ t + \lambda_o$) then the following can be established [3] and used to estimate Reliability.

$$\text{for } \lambda = at + \lambda_o\text{, then } R = e^{-\left(\frac{a}{2}T^2 + \lambda_o T\right)} \tag{3}$$

(Reliability at end of mission time, T)

Failure Rate Estimation. In order to properly and meaningfully predict reliability, the failure rate of the individual component parts must be estimated. During the initial design phase of a system, the failure rates are not known and must be estimated by "educated guess," "extrapolation," and/or "expert opinion." However, as systems go into operation and "actual measurements" of failure rates become available then reliability calculations provide a greater degree of confidence in the results.

DISCUSSION

Over the past several years, Oak Ridge National Laboratory (ORNL) has been reviewing historical failure data for pumps and valves used in commercial nuclear power plants. This work was sponsored by the Nuclear Regulatory Commission's (NRC's) Nuclear Plant Aging Research Program and involved review and characterization of failure records obtained from the Institute of Nuclear Power Operation's (INPO's) Nuclear Plant Reliability Data System (NPRDS) database. Parameters considered included failure area, failure cause, system of service, corrective action, etc. In general these records have presented a review of failures occurring in the 1984-1994 time frame. The actual record sets used in specific evaluations may represent a shorter duration time frame but will be within this 10 year period. These evaluations are discussed in several documents [4], [5], [6], [7], [8].

It should be understood that the NPRDs data base provides probably the best overall data currently available on nuclear plant operations; however, this data has numerous limitations which allow for only general and limited conclusions to be inferred from this study. Data can not be established to provide exact on demand failure rates for specific components. Only basic trends for general types of components at numerous plants of like design can be inferred. Also, the data

has "built-in" the results of both past and current testing programs and the overall impact of many utility maintenance programs. Therefore, neither specific absolute component failure rates and/or specific component repair rates can be segregated from the data only general industry trends established.

METHOD OF APPROACH

In the following discussions, a value for reliability will be estimated for several types of components using the NPRDs data. Because of the nature of the NPRDs data, the term "reliability" should be more exactly described in context of this work as <u>average availability</u>.

<u>Swing Check Valve Failures</u>. For the purposes of this study, the system of service to be evaluated is the Emergency Core Cooling System (ECCS) in a nuclear power plant. Therefore, the swing check valves that are of interest are those studied in the accumulator discharge study [6]. This study covered a total of 231 accumulator check valves manufactured by five different vendors and installed in a total of 39 nuclear plants. Of the 231 valves identified, 142 were at Westinghouse plants, 61 were at CE plants and 28 were at B&W plants. The study covered a time period from 1984-1992 and the ORNL-characterized failures represented 1,894 valve-years (nt) of operational service. When the total of 18 failures are distributed by age brackets (0 - 5 yr., 5 - 10 yr., etc.) and plotted as absolute failure rate at the middle of the bracket then you can obtain the Fig. 1 distribution. For this ECCS study, the critical failure mode for the swing check valves is stuck closed. In the time period studied, no accumulator check valves failed stuck closed. Therefore, in order to estimate a stuck closed failure rate it was assumed that over 20 years this population of check valves would have 3 to fail stuck closed. That estimate was used to develop the Fig. 1 stuck closed failure rate distribution. This Fig. 1 distribution can be conservatively approximated by a constant failure rate of 0.00085 failures/year when used in an exponential failure law to calculate reliability for the swing check valves. The stuck closed failure mode was the only failure rate estimated since this failure mode is the only method by which the swing check valve can disrupt the operation of the ECCS.

The reliability of the swing check valve can then be <u>estimated</u> using the following:

$$R_{cv} = e^{-0.00085T}$$

where: T is the operating time in years

(4)

<u>Pump Failures To Run</u>. The evaluation of pump failure rates used, in as much as possible, the same methodology applied to the check valve studies. Reference [7] provided details of the pump failure study.

Considerable variation in failure rates was found among the examined categories of pumps. Emergency Service Water pumps at PWR plants had a failure rate that was more than twice that of the overall PWR pump population (including ESW pumps), and about 2.7 times that of the studied PWR pumps. At BWR plants, over three-quarters of all reported pump failures, and over 90% of the significant failures occurred in the ESW system. Excluding ESW pumps, the failure rate of significant failures for studied pumps at PWR units was almost nine times that of

BWR units and appears to be aging at a rate six times faster than that for BWR pumps.

Best estimates of pump failure rates and trends are shown on Fig. 2. These trend lines from Fig. 2 have been used to estimate failure rate models for several pump applications. Since over 90% of all U.S. commercial nuclear power plants have been operating longer than 5 years, the wear-in trends have been neglected and either a constant failure rate or a straight line function through the origin were used as mathematical models.

For the ECCS system the appropriate pump model would be for CVCS/HPSI pumps. The reliability of the CVCS/HPSI pumps can be estimated using these trends and exponential failure law as follows:

$$R_{HPSI\,Pump} = e^{-(0.0023T^2)}$$

(5)

Motor Operated Valve Failures to Open. The evaluation of motor operated valve (MOV) failure rates uses the same methodology applied to the pump and check valve studies. A very specific group of MOVs was selected for this study. This group all consisted of the safety injection loop isolation valve in PWRs. A total of 77 valves were identified from 22 of the same plants that were used in the accumulator swing check valve study. This study was for the same time period and provided a total of 646 valve-years (nt) of data. Over this time period, a total of 44 failures occurred in this population of MOVs. Reference [8] provides a discussion of MOV failure data trends by type and system.

For a safety injection system, the loop isolation MOV is normally closed and must open for delivery of safety injection water to the PWR core. Therefore, the failure mode of interest was failure to open. Of the 44 failures, 9 failures were failure to open. When these 9 failures were identified by component age bracket, the failure rate distribution shown on Fig. 3 was established. For this data no specific aging effect is established, this is probably due to an increased repair rate for MOVs at and beyond 10 years of age. However, this data is not available from NPRDs. The MOV failure rate for opening on demand was then averaged and used as a constant value of 0.0139/year. The reliability of the MOV to open is then estimated using the following:

$$R_{MOV} = e^{-0.0139T}$$

(6)

System Reliability. The overall plant ECCS is generally made up of parallel trains each composed of a series of components. The reliability estimates over time for the given components are shown on Fig. 4. The general reliability equations for series and parallel systems are given in Reference [2]. The general schematic for a two train safety injection system for a PWR [10] is shown on Fig. 5. When you use the appropriate equations as shown in Reference [9] you can develop an equation for reliability of each train as follows:

(Train A or B)

$$R_{SI} = \left[1 - (1 - R_{cv})^2\right]^3 R_{cv} \times R_{Pump}\left[1 - (1 - R_{MOV})^2\right]$$

(7)

The two train system reliability is as follows:

$$R_{SI\ System} = 1 - (1 - R_A)^2 = 1 - (1 - R_B)^2 \tag{8}$$

This overall system reliability expression can be used to estimate the total SI system reliability with pump operating hours. This type of reliability estimate is shown on Fig. 6. The Fig. 6 estimate is based on $R_{cv} = 0.9865$, $R_{MOV} = 0.8006$, and the pumps in both trains accumulating operation hours at approximately the same rate (2500 hrs/year). The Fig. 6 estimate indicates that total safety injection system reliability would degrade below 99% at approximately 45,000 hours of pump operation time and below 95% reliability at approximately 66,000 hours. Using this model and NPRDs failure data, if CVCS/HPSI pumps were to develop operating hours at 2,500 hours/year then the SI system reliability could fall below 99% at about 20 years of operation.

CONCLUSIONS

The reader is cautioned to note that the above estimates and calculations are by no means exact and can only be used to develop some general trends. Clearly specific plant failure data and maintainability programs can and do play a major role in the general availability (reliability) of a specific system. Also, mission time is very dependent on plant programs for testing, inspection and repairs. The NPRDs data does not allow these items to be independently reviewed and assessed.

However, general trends can provide some useful information as follows:

- For mission times between 1 and 4 years, pumps and motor operated valves should have about equal values of unavailability, ranging from 0.003 to 0.03 with an average of 0.015 and swing check valves would be an order of magnitude less ranging from 0.0008 to 0.003 with an average of 0.002.
- As a specific type of PWR pump, CVCS/HPSI pumps appear to be aging at a rate 1.5 times that of the general PWR population.
- As plants begin to approach half of their design lifetimes (e.g., 20 years), ever decreasing pump reliability due to the increasing operating hour history could make the pump the dominant component impacting overall Safety Injection System unavailability.

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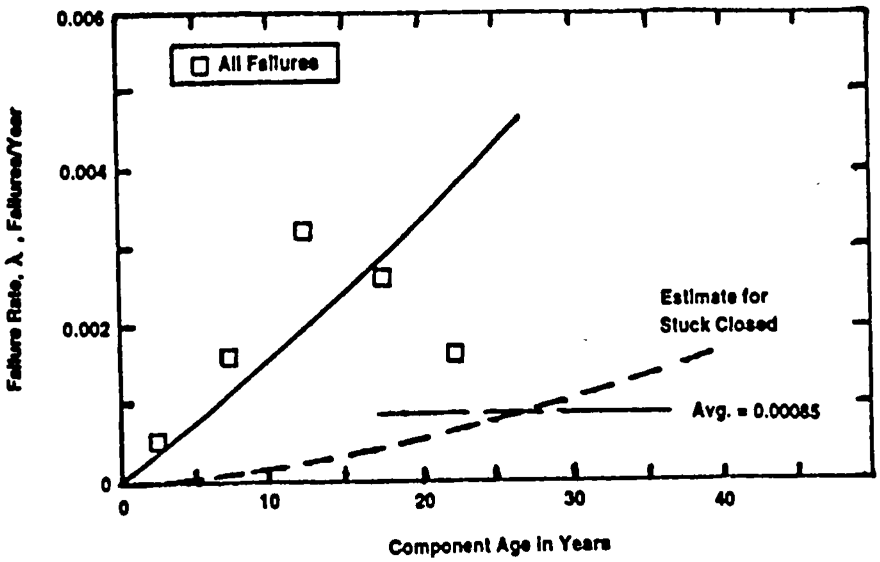

Fig. 1 - Swing Check Valve Failure Rates

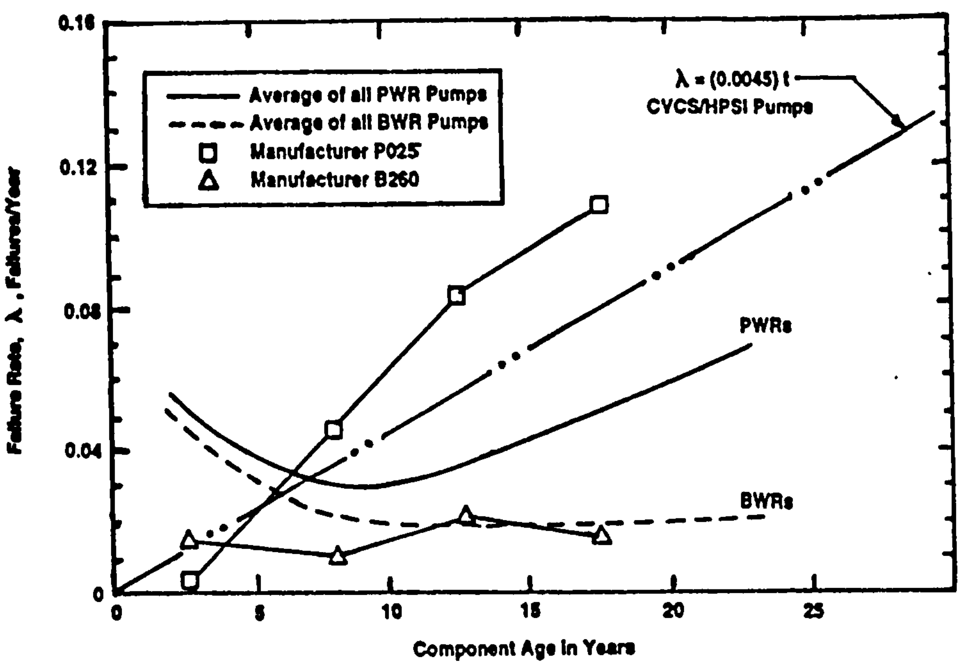

Fig. 2 - Failure Rate as a Function of Time for Safety Related Pumps
(Significant Failures Only)

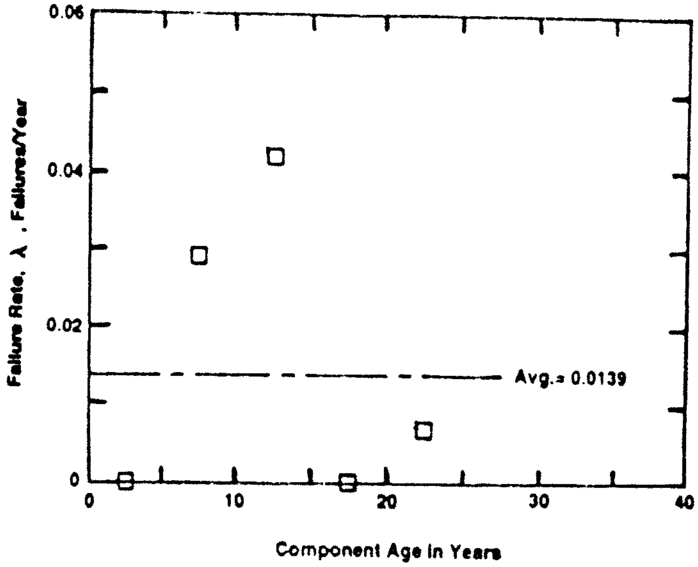

Fig. 3 - Failure Rates as a Function of Time for Motor Operated Valves (MOV - Failure to Open on Demand)

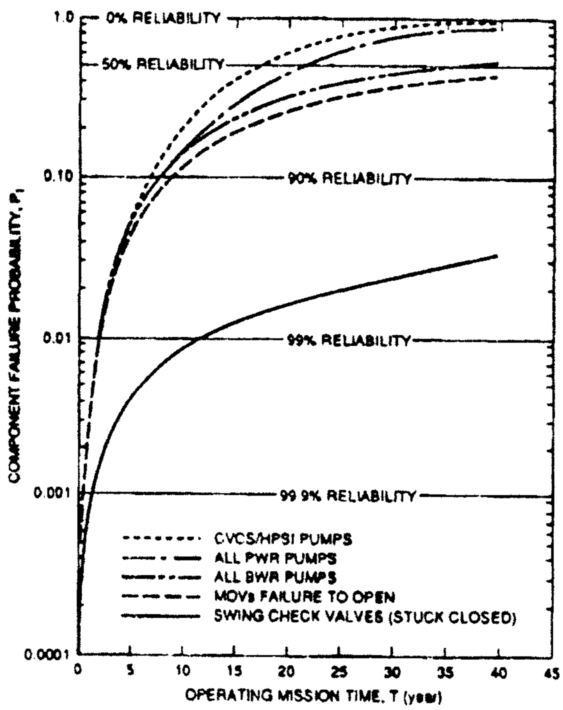

Fig. 4 - Nuclear Plant Component Reliability Curves

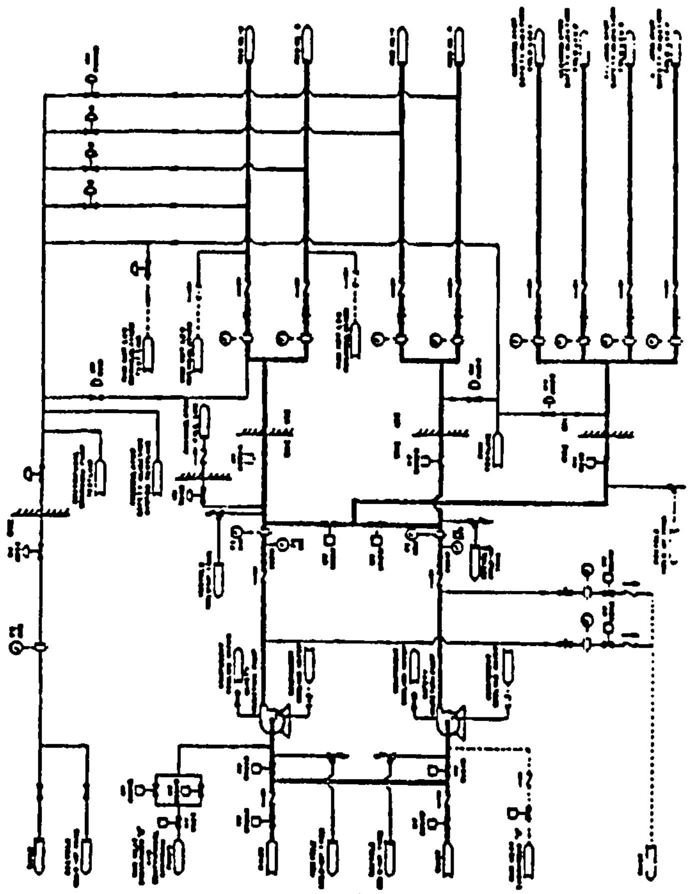

Fig. 5 - Standard Nuclear Plant Emergency Core Cooling System Design

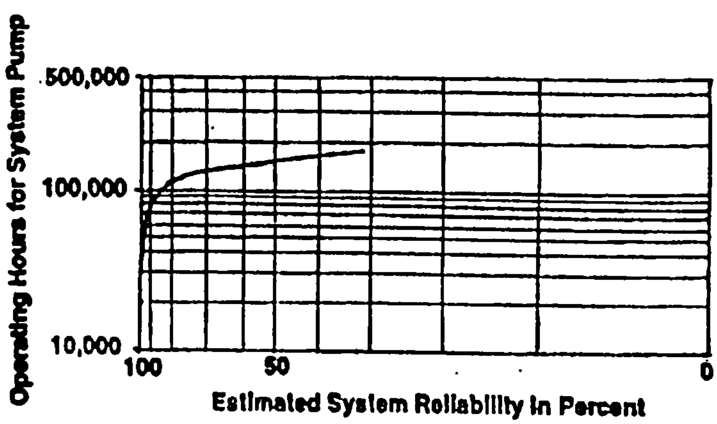

Fig. 6 - Estimated System Reliability for the Safety Injection System as a Function of the CVCS/HPSI Pump Operating Hours.

Performance-Based Appendix J Program for Containment Isolation Valve Leakage Rate Testing

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ABSTRACT

The NRC published a final rule change to 10 CFR 50, Appendix J, "Primary Reactor Containment Leakage Testing for Water-Cooled Power Reactors," in the *Federal Register* on September 26, 1995 (60 *FR* 186, p. 49495). The final rule became effective October 26, 1995. The revised regulations provide a performance-based option for leakage-rate testing of containments ("Option B"). Licensees may voluntarily adopt the option in lieu of compliance with the prescriptive requirements in the regulation ("Option A"). The NRC issued the change as part of an effort to improve the focus of regulations by eliminating prescriptive requirements that are marginal to safety. The final rule allows leakage test intervals to be based on system component performance. Thus, licensees have greater flexibility for cost-effective implementation methods in satisfying regulatory safety objectives.

This paper will explore some of the considerations licensses should address if they choose to implement Option B, "Performance-Based Requirements" for certain of the containment isolation valves. These considerations will highlight some of the necessary changes to the ASME Inservice Testing programs. Applicability to both the IWV and OM Part 10 programs will be discussed.

Optimized Periodic Verification Testing Blended Risk and Performance-Based MOV Inservice Test Program An Application of ASME Code Case OMN-1

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Abstract

This paper presents an application of ASME Code Case OMN-1 to the GL 89-10 Program at the South Texas Project Electric Generating Station (STPEGS). Code Case OMN-1 provides guidance for a performance-based MOV inservice test program that can be used for periodic verification testing and allows consideration of risk insights. Blended probabilistic and deterministic evaluation techniques were used to establish inservice test strategies including both test methods and test frequency. Described in the paper are the methods and criteria for establishing MOV safety significance based on the STPEGS probabilistic safety assessment, deterministic considerations of MOV performance characteristics and performance margins, the expert panel evaluation process, and the development of inservice test strategies. Test strategies include a mix of dynamic and static testing as well as MOV exercising.

1.0 INTRODUCTION

This paper describes a project (Reference [1]) performed by ERIN Engineering and Research, Inc. (ERIN) for the South Texas Project Electric Generating Station (STPEGS). The purpose of the project was to apply ASME Code Case OMN-1 (Reference [2]) to the Generic Letter (GL) 89-10 (Reference [3]) Program at STPEGS. This project was performed as part of the comprehensive risk management program at STPEGS, and as such, is an application of the STPEGS Probabilistic Safety Analysis (PSA) (Reference [4]).

The primary result of the project is a set of recommendations to optimize the MOV inservice test program in a manner that will assure high MOV reliability and reduce the costs of MOV periodic verification testing while maintaining plant safety. A combination of deterministic and probabilistic methods are used which includes a strategy of testing methodologies and frequencies. The methodology and results of this project reflect key input from an Expert Panel working group comprised of cognizant personnel from operations, maintenance, engineering, licensing and risk assessment organizations.

1.1 Background

This project was started after the STPEGS GL 89-10 Program had received successful

closure by the NRC. The initial design basis testing was completed and periodic verification testing was underway when the decision was made by STPEGS management to incorporate risk and performance-based considerations into the program. At stake in this project are the substantial resources being invested on periodic verification testing to comply with GL 89-10 commitments beyond that needed to meet other testing requirements such as those of the In-service Testing Program. This project is motivated by the desire to focus testing resources on those components most important to safety and to avoid costly testing when justified by a combination of deterministic and risk impact considerations. The most important such consideration is the need to eliminate costly testing where the payoff in terms of risk management effectiveness is marginal.

This project can be compared with other similar projects in which the risk and performance-based considerations were incorporated at an earlier stage of the program. The timing of this project was influenced by the priorities of the comprehensive risk management program at STPEGS which had heretofore focused its resources on other applications such as Graded Quality Assurance, Risk-Based Technical Specifications, implementation of the Maintenance Rule (Reference [5]), outage risk management and risk management of on-line maintenance. Consideration of the project timing relative to the advanced stage of implementing GL 89-10 testing had a significant impact on how the PSA models were updated and applied in this project as well as on the results.

1.2 Project Objectives

The objectives of this project were to:

- Develop a methodology to prioritize the periodic verification testing for MOVs in the STPEGS GL 89-10 program.
- Develop a blended Risk and Performance-Based Inservice Testing program for MOVs consistent with ASME Code Case OMN-1 and the PSA Applications Guide (Reference [6]).
- Document the results in a manner that will facilitate future use by STPEGS and review by the US NRC.
- Optimize long-term resource allocation relative to MOV testing.

2.0 SUMMARY OF APPROACH

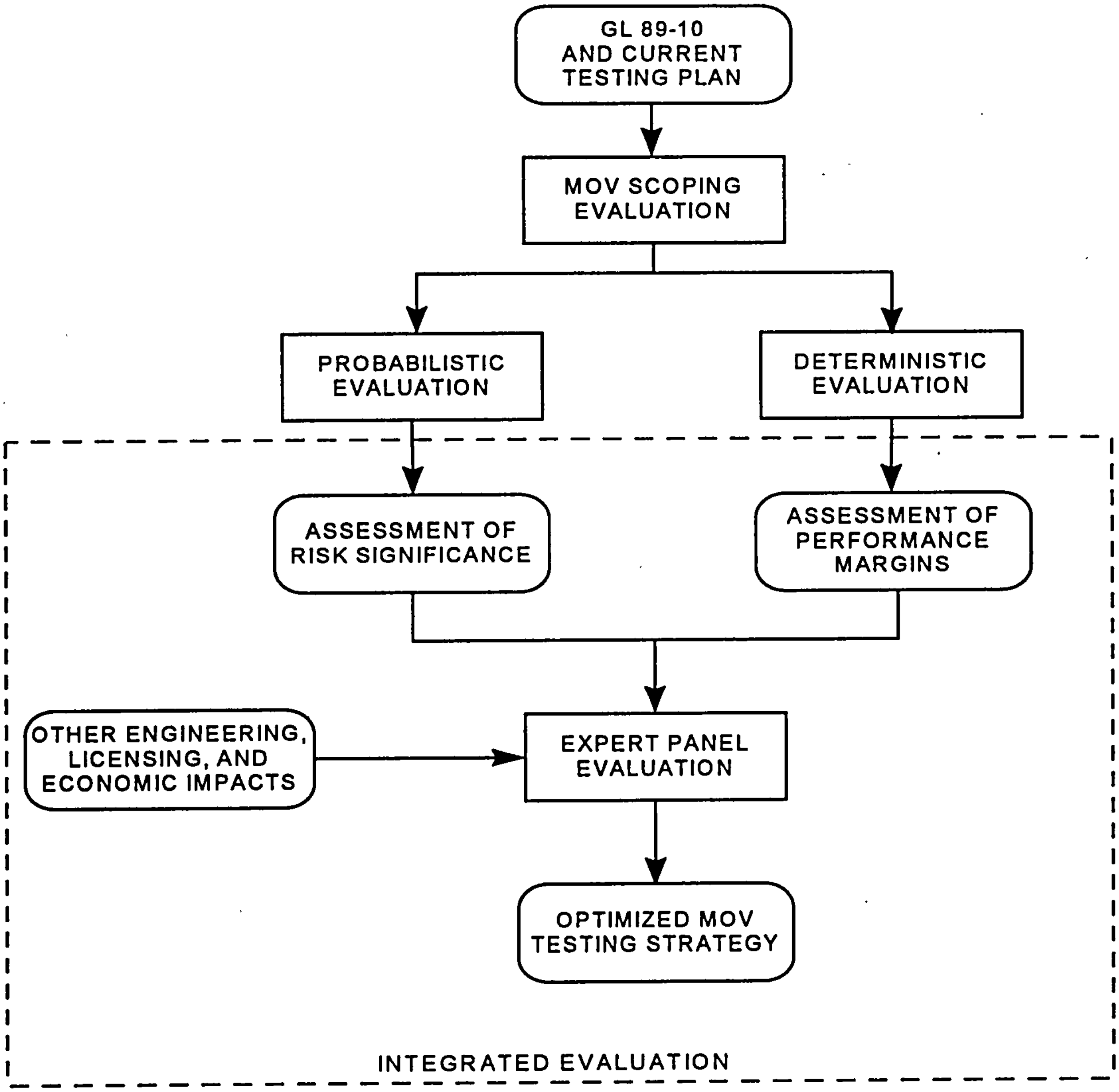
The approach that was employed in this project is illustrated in Figure 1. This approach consists of the following major elements:

2.1 MOV Scoping Evaluation

In this part of the evaluation, the information needed to perform this project is collected and evaluated qualitatively. This information includes the STPEGS GL 89-10 program documentation including list of GL 89-10 MOVs, engineering data for the MOVs, results of MOV tests, and other information needed to establish performance margins and testing requirements.

Figure 1

OVERALL METHODOLOGY FOR MOV TESTING PRIORITIZATION



2.2 Deterministic Evaluation

Key products the deterministic evaluation include the identification of functional failure modes (FFMs) of the MOVs and the definition of functional groups of MOVs that have similar functions, valve types, testability, expected performance characteristics and risk impacts. The primary purpose of this task is to perform a deterministic engineering evaluation of each MOV group to establish the margin between the performance capability of the MOV and the performance needed to successfully accomplish the MOV safety function.

2.3 Probabilistic Evaluation

The purposes the probabilistic evaluation are to determine the risk contribution of individual MOVs for the purpose of ranking the components with respect to safety significance and to evaluate the cumulative effects of the proposed testing strategy on the risk of severe accidents at STPEGS. In order to serve these purposes, it is necessary to first characterize the risk impacts of the technical issues raised by GL 89-10 in terms of the failure rates and common cause failure parameters of the MOVs. Once this is accomplished, risk importance measures are used to affect a preliminary ranking of MOVs in terms of their contribution to the frequency of a core damage event and that of a large early release. The results of this evaluation are combined with the results of the deterministic evaluation and initial input from the Expert Panel to define a proposed testing strategy for all MOVs. The PSA evaluation is then completed by performing a PSA update that measures the cumulative effects of the combined MOV testing strategies.

2.4 Integrated Evaluation

The purpose of this task is to combine the results of the deterministic and probabilistic evaluations to determine the inservice testing strategy for the MOVs. This evaluation is performed in two steps. In the first step, a ranking matrix is used to place each group of MOVs into a unique testing category based on a combination of risk rankings and performance margins. In the second step, the Expert Panel makes adjustments to the testing strategy to account for factors not explicitly accounted for in the previous evaluations. Such factors include consideration of shutdown modes not accounted for in the PSA, release states other than large early releases, and consistency with other licensing positions, etc. The Expert Panel has an influence not only on the incorporation of deterministic factors but also on how the probabilistic aspects are reflected in the final evaluation. This approach reflects the reality that the Expert Panel is representative of the management team that makes essentially all the technical decisions about plant design, operations and maintenance issues.

3.0 EVALUATION METHODOLOGY AND RESULTS

3.1 Deterministic Evaluation

The deterministic evaluation for STPEGS GL 89-10 MOVs involved the following four tasks:

1. Functional Failure Mode Definition
2. Deterministic Criteria Development
3. Performance Margin Quantification
4. Other Deterministic Considerations

The methods used to accomplish these tasks and the corresponding results are described below.

3.1.1 Identification Of Functional Failure Modes. The first step in the deterministic evaluation process involved the identification of the MOV functional failure modes (FFMs) applicable to the STPEGS GL 89-10 program. The identification of GL 89-10 FFMs is a key fundamental step necessary for the appropriate prioritization and safety significance assessment of the MOVs using the PSA. The FFM of a valve is defined as the failure of the MOV to perform a specific design basis function. Failure modes modeled in the PSA such as inadvertent operation (transfer open or closed) and physical valve failures such as pressure locking and thermal binding (plugging) are not appropriate for this evaluation as testing efforts associated with GL 89-10 will neither preclude nor identify these types of failures.

The STPEGS GL 89-10 MOV FFMs were developed through review of the MOV notebooks maintained by the STPEGS MOV program personnel. These notebooks document the design basis requirements for each MOV in the STPEGS GL 89-10 MOV program. To aid in the analysis and discussion of results, the MOVs were classified into functional groups. The functional groups were defined as MOVs having the same basic design basis function. Based on the three independent train configuration of most systems at STPEGS, identification of functional groups was a relatively simple process. The FFMs identified for the MOV functional groups were reviewed in detail by personnel expert in STPEGS plant and system function and operation to assure completeness and accuracy as discussed later in this paper.

3.1.2 Deterministic Evaluation Criteria. The next step in the deterministic evaluation was the establishment of deterministic evaluation criteria. The deterministic evaluation uses quantitative criteria based on MOV performance margins and categorizes the MOVs into three priorities based on plant specific criteria. Consideration of MOV performance margin is appropriate for this prioritization effort in that the likelihood of MOV failure can be considered proportional to performance margin. The deterministic evaluation also uses qualitative criteria based on the engineering judgment of personnel expert in plant, system, and MOV behavior considering safety issues, plant and system operating modes, local environmental conditions and MOV accessibility for repair or manual operation (recovery), and key plant support functions that are beyond the scope of the PSA model.

The performance margins are used to rank GL 89-10 MOVs in accordance with the deterministic criteria provided in Table 1.

Table 1
DETERMINISTIC RANKING CRITERIA

MARGIN RANK			
	LOW	**MEDIUM**	**HIGH**
Dynamic	≤ 10%	10% to 25%	> 25%
Static	≤ 25%	25% to 50%	> 50%
Analytic	≤ 50%	50% to 100%	> 100%

As shown in Table 1, less dynamic test margin is required to rank the MOV as having HIGH margin than with static or analytic margin. Likewise, less static margin is required to rank an MOV as having HIGH margin than with analytic margin. This is due to the increased performance certainties associated with in-situ testing. Increased confidence in MOV performance and performance requirements is provided with a more detailed level of testing. This approach accounts for the progressively greater certainties inherent in demonstrating actual performance capability from dynamic and static test margins. Preference is given to dynamic test margin over static test and analytic margins. For conservatism and consistency with the probabilistic ranking process, the overall deterministic rank of the functional group is the lowest deterministic rank of all of the valves in the group. For example, if only one MOV in the group indicates LOW deterministic margin, the functional group deterministic rank is LOW, and all of the valves in the functional group are treated as having LOW deterministic margin. In most instances, however, the valves in a functional group are similar not only in design basis function, but also in design capability. As such, deterministic margins of individual valves in a group were of similar magnitude.

3.1.3 Performance Margin Quantification. The quantitative deterministic criteria are based on MOV performance margins which must be quantified. MOV performance margins were calculated based on the results of dynamic tests, static tests, and analytical setpoint calculations. The preferred source of performance margin data is the dynamic (differential pressure & flow) test. However, dynamic test data does not exist for all STPEGS GL 89-10 MOVs. For MOVs without dynamic test data available, static test data was used as applicable. In cases where neither dynamic nor applicable static test data was available, analytic setpoint calculation data was used.

Dynamic Test Margins

Rising stem valves have a minimum required stem thrust in the close direction that normally occurs near valve seating and is used to select the appropriate setpoint for Control Switch Trip (CST). Therefore, comparing the measured thrust value at

control switch trip to the maximum measured thrust prior to or at hard seat contact is a valid indicator of MOV performance margin. The thrust at CST is compared to the maximum measured thrust prior to closing during the dynamic test to determine the close dynamic margin for rising stem valves using equation 1.

Although rising stem valves have a calculated minimum required thrust in the open direction, it normally occurs near disc pullout and not at CST. Therefore, comparing the measured thrust value at CST to the minimum required is not necessarily indicative of MOV performance margin. The calculated maximum MOV capability is compared to the maximum measured thrust after disc pullout during the open dynamic test to determine the open dynamic margin for rising stem valves using equation 2.

The maximum open thrust capability is determined from the minimum of the motor torque capability at reduced voltage, actuator thrust and torque ratings, and valve structural limits. The motor torque capability and actuator torque rating were converted to thrust limits using a stem factor assuming a conservative stem coefficient of friction consistent with STPEGS GL 89-10 MOV program.

Butterfly valves have a calculated minimum required torque; however, the minimum required torque does not necessarily occur at CST as it does for rising stem valves. Comparing the measured torque at CST to the minimum required is not necessarily indicative of MOV performance margin. Therefore, the calculated maximum MOV torque capability is compared to the maximum measured torque during the dynamic test using equation 3.

Static Test Margins

To determine performance margins based on static testing, it is necessary to validate MOV capability. For gate and globe valves in the close direction, MOV capability at CST is verified to be above the calculated minimum required thrust. However, for rotating stem valves in both directions and for rising stem valves in the open direction, validation of MOV capability cannot be determined unless a load verification test is performed. For rising stem valves, this thrust verification is performed into a load cell. The measured CST Load Cell Thrust must be greater than the calculated minimum required thrust. Likewise, for rotating stem torque verification using the MOVATS® BARTS equipment, the measured CST BARTS Torque must be greater than the calculated minimum required torque in the appropriate direction. If these conditions are satisfied, the MOV capability has been proven to be above the minimum required thrust or torque. Static performance margins are calculated using equations 4 and 5 for rising stem gate and globe valves, and equation 6 for rotating stem butterfly valves.

Analytic Performance Margins

The final MOV margin calculation is the analytic performance margin. For analytic margins, no test data is used. MOV design capability is compared to the calculated minimum required thrust or torque. Equation 7 is used to calculate analytic performance margins.

$$\text{Close Dynamic Margin} = \left(\frac{\text{Measured Thrust at CST}}{\text{Maximum Measured Thrust prior to or at Hard Seat Contact}}\right) - 1 \qquad (1)$$

$$\text{Open Dynamic Margin} = \left(\frac{\text{Maximum Open Capability}}{\text{Maximum Measured Thrust after Disc Pullout}}\right) - 1 \qquad (2)$$

$$\text{Dynamic Margin} = \left(\frac{\text{Maximum Torque Capability}}{\text{Maximum Measured Torque}}\right) - 1 \qquad (3)$$

$$\text{Close Static Margin} = \left(\frac{\text{Measured Thrust at CST}}{\text{Minimum Required Thrust}}\right) - 1 \qquad (4)$$

$$\text{Open Static Margin} = \left(\frac{\text{CST Load Cell Thrust}}{\text{Minimum Required Thrust}}\right) - 1 \qquad (5)$$

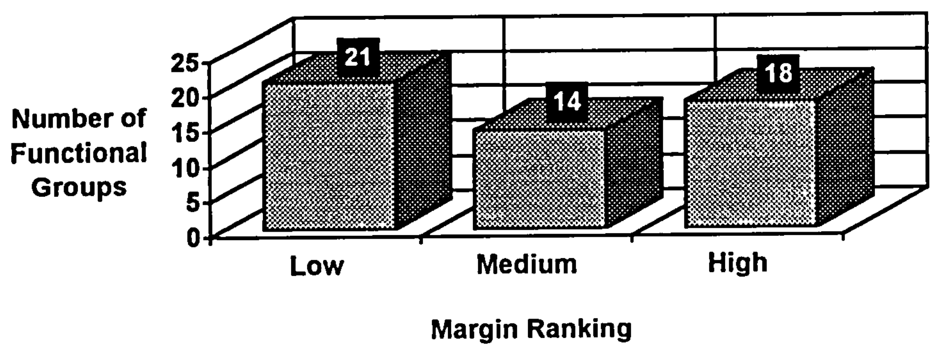
$$\text{Static Margin} = \left(\frac{\text{CST BARTS Torque}}{\text{Minimum Required Torque}}\right) - 1 \qquad (6)$$

$$\text{Analytic Margin} = \left(\frac{\text{Maximum MOV Capability}}{\text{Minimum Required Thrust or Torque}}\right) - 1 \qquad (7)$$

3.1.4 Other Deterministic Considerations. The quantitative criteria discussed above was supplemented by qualitative criteria based on the engineering judgment of personnel expert in plant, system, and MOV behavior. These deterministic considerations include safety issues, MOV design characteristics and plant/system operating modes not addressed by the PSA. For example, a safety-significant issue currently being addressed in the nuclear power industry is the risk of core damage events initiated during shutdown operations. Such considerations are addressed qualitatively in this study. Additional deterministic considerations include local environmental conditions and MOV accessibility for repair or manual operation (recovery), MOV design configurations where service condition loadings may promote MOV operation, and MOVs that frequently demonstrate performance capability through operation at, or near, design basis conditions.

3.1.5 Deterministic Ranking Results. The 302 STPEGS GL 89-10 MOVs were placed in 53 functional groups and ranked in accordance with this deterministic methodology. Of the 53 functional groups, 21 were ranked as having LOW margin, 14 were identified as having MEDIUM margin and 18 were ranked as having HIGH margin. The results of the quantitative performance margin ranking are shown in Figure 2 by number of MOV functional groups.

Figure 2
Deterministic Ranking Results
Number of Functional Groups vs Rank



Based on the functional group ranking, 134 MOVs were ranked with LOW margin, 74 MOVs were ranked with MEDIUM margin, and 94 MOVs were ranked with HIGH margin. The MOV ranking results by deterministic margin are shown in Figure 3.

3.2 Probabilistic Evaluation

The objectives of the probabilistic evaluation are to determine the baseline risk significance of each MOV in the STPEGS GL 89-10 program as well as other MOVs having a significant risk impact and to

evaluate the risk impacts of the proposed testing strategy for these components. The overall methodology for meeting these objectives is illustrated in Figure 4 and consists of three major elements:

3.2.1 Risk Characterization of GL 89-10 Issue. The purpose of this task is to define the cause and effect relationships between the technical issues raised by GL 89-10 and the risk of severe accidents as modeled in the most recent update of the STPEGS Level 2 PSA. As noted in the PSA Applications Guide, determination of the appropriate cause and effect relationships is an essential first step to a successful application of a PSA model. In this application, the PSA is used to establish the risk significance of the GL 89-10 technical issues, to rank the MOVs with respect to their contribution to risk, and to evaluate the risk impacts of various testing strategies under consideration for periodic verification inservice testing. Risk characterization is necessary to determine how to interpret the current results of the PSA and to use the PSA to evaluate MOV testing priorities.

Figure 3
Deterministic Ranking Results
Number of MOVs vs Rank

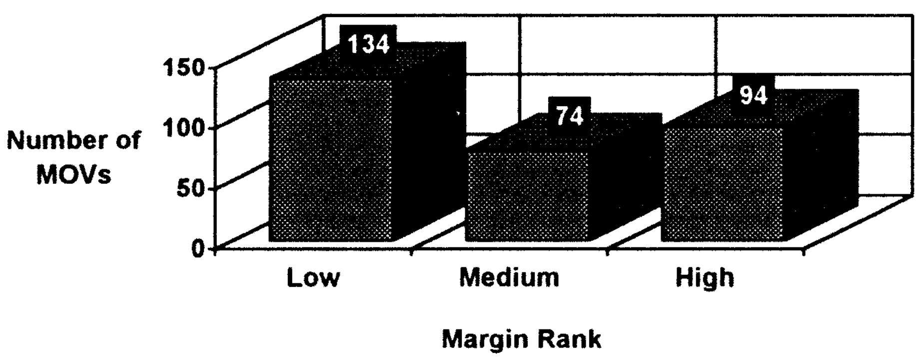
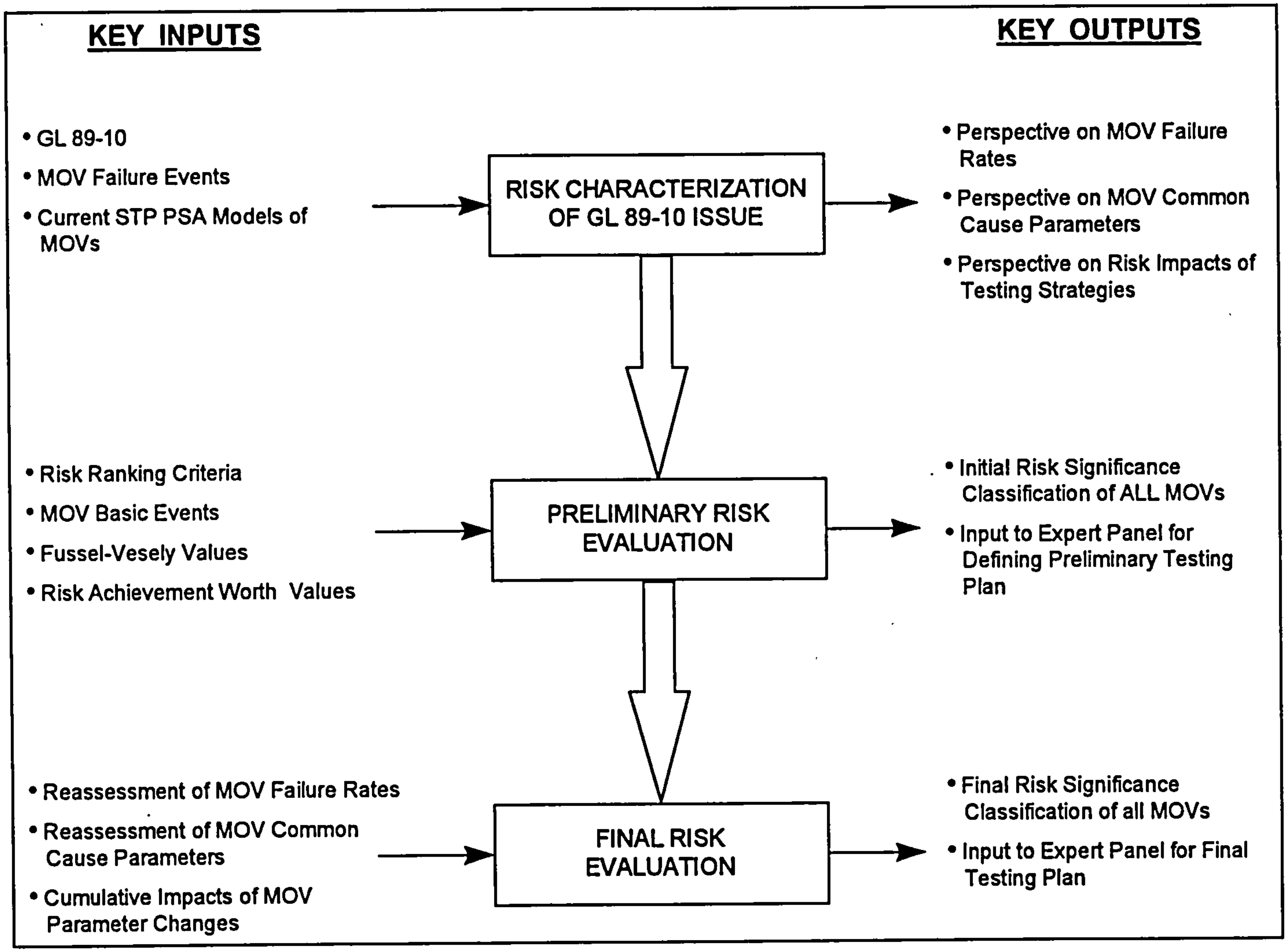


Figure 4

METHODOLOGY FOR PROBABILISTIC EVALUATION



GL 89-10 was issued in response to NRC concerns regarding the reliable performance of MOVs. These concerns were based on insights from several incidents in which the design adequacy of MOVs was questioned. These incidents involved common cause failures of redundant sets of MOVs during plant transient events at several plants.

In these incidents, certain MOVs were found to have such characteristics as undersized valve actuators or actuator motors, improper control switch settings, and other design shortcomings that raised questions about whether these MOVs would function at design basis conditions. Such failure modes are not necessarily identified in periodic inservice or surveillance tests. In several of the operating incidents that led to the concern, it was determined that deficiencies had existed since the initial startup of the plant in question.

The risk characterization of GL 89-10 in the context of a PSA model can be expressed in terms of the answers to the following questions?

- What are the true failure rates of MOVs during design basis or accident sequence conditions for both independent and common cause failures for the baseline PSA evaluation?

- What impact, if any, will the new GL 89-10 performance tests have on these MOV failure rates?

- What is the impact of any revisions to the MOV independent and common cause failure rates on the risks of a severe accident?

The STPEGS PSA models used a distribution for MOV failure rates that were based on a combination of generic industry data and plant specific data. The plant specific data was collected from maintenance work orders and estimates of the number of demands and accounts for about 5 reactor years of operating experience through 1992. The generic and plant specific data was collected primarily from surveillance tests that may have not been able to identify certain failure modes whose conditions are not simulated or detected during surveillance tests. An extensive reexamination of this MOV failure rate data was performed to account for the MOV testing performed as part of the STPEGS GL 89-10 program which resulted in small changes in individual MOV failure rates but significant changes in common cause failure rates. Presentation of this failure rate data reexamination is beyond the scope of this paper.

The Level 1 PSA model was updated to account for the changes in MOV failure rates and common cause beta factors described above. The update was accomplished by updating all the systems and initiating event models that include MOVs, requantifying the event trees and reviewing the results.

The impact of the update of MOV failure rates and common cause parameters on the frequency of core damage was positive but very small indicating a small contribution of MOVs on an cumulative basis. More importantly we know that the change is a reduction in CDF since all the individual model changes were positive. This demonstrates that the testing, analysis, and maintenance activities performed under the

STPEGS GL 89-10 program had a positive impact on plant safety.

3.2.2 Preliminary Risk Evaluation. In this task, the PSA update described above is used to rank the MOVs with respect to their contribution to risk. These rankings are based on component level risk importance measures that are determined for each component in the PSA model. The measures used in this evaluation are the Fussell-Vesely (FV) importance and the risk achievement worth (RAW) importance. The FV importance measures the fraction of the overall risk involving sequences in which the component is postulated to fail, while the RAW importance measures the ratio of the risk assuming the component failure probability is 1.0 to the base case risk level. In this study, risk is measured in terms of the annual average core damage frequency (CDF) and the annual average frequency of a large early release (LERF). This assures that MOVs not important for preventing core damage but instrumental in ensuring the containment integrity during a severe accident are retained in the risk significance categorization.

The probabilistic criteria shown in Table 2 are used to place each MOV into one of three classes of risk significance. The high and medium risk importance criteria are satisfied if either the CDF or the LERF importance measures for a component exceed pre-selected values. Hence, the use of the LERF criterion serves to increase rather than to decrease the population of components ranked with high or medium risk significance.

Table 2
RISK SIGNIFICANCE CATEGORIES FOR PRELIMINARY RISK EVALUATION

RISK SIGNIFICANCE CATEGORY	IMPORTANCE PARAMETER RANGE	
	FUSSELL-VESELY IMPORTANCE RANGE	RISK ACHIEVEMENT IMPORTANCE RANGE
HIGH	FV{CDF} or FV{LERF} > .005	ALL
MODERATE	FV{CDF} and FV{LERF} < .005	RAW{CDF} or RAW{LERF} > 2.0
LOW	FV{CDF} and FV{LERF} < .005 and Truncated or Not Modeled Components Verified as Low Risk	RAW{CDF} and RAW{LERF} < 2.0 and Truncated or Not Modeled Components Verified as Low Risk

The basis for omitting MOVs from the PSA model was reviewed to confirm that there was a justification for classifying the not modeled MOVs as low risk significance. Similarly, MOVs that were fully or partially truncated from the model were

reviewed to confirm that their classification as low risk significance was consistent with a qualitative evaluation and with the PSA results.

MOVs modeled as basic events in the PSA fault trees have a quantifiable impact on the risk of a severe accident. MOVs modeled as part of human recovery actions and MOVs contributing to initiating events also have a quantitative impact on plant risk but this impact is not always solely due to MOV failure. Therefore, the quantitative impact of MOV recovery and initiating events was qualitatively presented to the Expert Panel for consideration in the study.

Several different failure modes for MOVs are modeled in the PSA depending on the accident sequence, the normal valve position, and the function of the valve. These failure modes include:

- Failure to open
- Failure to close
- Initially mis-positioned valve
- Failure to remain in correct position (transfer)
- Internal seated disc leak or rupture
- External leak or rupture

In addition, common cause failures of combinations of MOVs within redundant groups were also considered. For a group of three redundant MOVs in a system, the possibilities for common cause failures of any two valves (3 such combinations) and all three valves (1 combination) are considered as distinct fault tree basic events.

These failure modes are considered in the PSA model as representing MOV specific failures in addition to dependent failures of the valves due to loss of support functions such as electric power and loss of actuation signals. The failure modes addressed by GL 89-10 are failure to stroke (open or close) given a demand under design basis conditions. The PSA considers these modes as well as others such as being mis-positioned, transferring to the wrong position, external leak or rupture and internal leak or rupture of a normally seated valve disc or discs. While all failure modes contribute to the overall risk importance of the MOVs, we consider only the failure to stroke modes in prioritizing valves for GL 89-10 inservice testing. GL 89-10 inservice testing does not impact the frequency of the remaining failure modes and have been excluded from this evaluation.

The STPEGS GL 89-10 testing program includes a total of 151 MOVs at each unit. The STPEGS PSA models 131 of these MOVs, however 55 MOVs in the PSA models are not modeled for the failure to stroke failure modes that are addressed in GL 89-10. Most of these 55 MOVs are normally in the correct position for fulfilling the safety function and are only considered in the PSA due to the potential for being mis-positioned or for transferring to the wrong position. Hence, these 55 MOVs have a finite risk impact but this risk impact is irrelevant to the issues raised by GL 89-10 and monitored by the inservice test program. Therefore, approximately 50% of the STPEGS GL 89-10 MOVs have a quantifiable risk significance associated with GL 89-10 failure modes.

It is necessary to compute component level importance measures from the PSA to rank the MOVs by risk significance. Since each modeled MOV may include many different basic events to account for different failure modes as well as different common cause basic events that involve a particular component, separate risk importance measures are computed for each basic event. Component level risk importance measures, i.e., Fussell-Vesely and Risk Achievement Worth were computed using Equations 8 and 9.

$$FV\{x\} = \sum_{i=1}^{N} fv_i \tag{8}$$

$$RAW\{x\} = 1 + \sum_{i=1}^{N} (raw_i - 1) \tag{9}$$

where:

$FV\{x\}$ = Fussell-Vesely Importance Measure for component x

fv_i = Fussell-Vesely Importance Measure for basic event i

$RAW\{x\}$ = Risk Achievement Worth for component x

raw_i = Risk Achievement Worth for basic event i

N = Number of basic events contributing to failure of component x

A property of Equations 8 and 9 is that for common cause groups of MOVs, there is multiple accounting for the common cause basic events. For example, the basic event for common cause failure of MOVs A, B, and C will be accounted for in all three MOV component level importance measures. This is a reasonable assumption given the fact that each importance measure assumes that all other items in the model are fixed at the base case values. It is conservative in the sense that for any component in a common cause group its risk importance is significantly impacted by the risk importance of the common cause events. This is again not unreasonable in view of the role that common cause failure events have played in defining the technical issues of GL 89-10.

A total of 10 MOVs were classified as high risk significance because their FV importance values for CDF or LERF exceeded .005. Two of these were based on the LERF values and the other eight were based on the CDF values; however, the correction for asymmetry would have added the same two valves into the high risk category based on CDF also. A total of 27 MOVs were classified as medium because their RAW values exceeded 2.0 even though their FV values were below .005. The remaining 114 GL 89-10 MOVs were found to be of low risk significance for one reason or another, but in all cases, it can be safely concluded that their respective risk importances would not meet the medium or high criteria in Table 2.

A key issue in the interpretation of risk importance values is the appearance of numerically different risk importance values for components in groups of similar MOVs whose risk contribution would be expected to be similar. In some cases, this reflects valid differences due the asymmetries of interdependent systems. More often, this phenomenon results from modeling assumptions associated with asymmetries in operation of normally running systems. For example, the

Essential Cooling Water System at STPEGS has three symmetrical trains, where one is normally running, one is in standby and ready to autostart if the normally running train fails, and one requires a manual action to start. Of the six possible configurations that this system can be in normal operation, only one was modeled in the PSA. While the event frequencies were modeled to provide a correct estimate of the annual average core damage frequency, the risk importance of different trains are skewed as a result of this type of modeling assumption. Because of the dependencies of other systems such as Component Cooling Water, Essential Chilled Water and others on Essential Cooling Water, this modeling asymmetry carries through to impact the risk importance of many three and four train systems at STPEGS.

To account for these modeling assumptions in the final evaluation of MOVs, the individual MOVs are put into functional groups and the risk classification of the group is based on the limiting value for the similar valves assigned to the group. For example, two of the three emergency sump recirculation MOVs were classified as high risk significance and the other as medium safety significance due to modeling assumptions. In the final integrated evaluation, all three were put into the same functional MOV group and given a high risk significance ranking.

The risk importance of a given component is primarily established by combining the respective importances of the basic events in the system fault trees that correspond with failure of the component. The PSA model was reviewed to identify other risk contributions that were "buried" in the other aspects of the PSA model. A total of five MOVs were identified in the treatment of operator recovery actions whose actions contribute to the total risk importance of the valve. These MOVs include the two pressurizer PORV block valves that are credited with the operator action to isolate a LOCA through a leaking PORV, two emergency sump recirculation MOVs that are credited in recovering failure of these MOVs to transfer on demand during small LOCA sequences, and the main steam to AFW MOV that is credited for operator recovery of the turbine driven auxiliary feedwater pump. These MOVs were already considered in the system fault trees, however, the risk importance had to be revised to add in the contribution from these operator actions. The addition of this aspect of importance had a significant contribution to all 5 of these MOVs and was responsible for elevating the risk significance classification of 3 of them from medium to high risk significance, the other 2 already having been classified as high.

4.0 INTEGRATED EVALUATION OF PROPOSED TESTING STRATEGY

This project employed a blended approach of deterministic and probabilistic methods to set and evaluate priorities for MOV testing and to help define the most appropriate and cost effective strategies for GL 89-10 periodic verification inservice testing at STPEGS. The probabilistic element of this approach ranked the importance of each MOV based on its contribution to the risk of a severe accident, while the deterministic element ranked importance in terms of performance margins. A unique testing priority is then assigned based on a blending of the

probabilistically and deterministically based rankings.

An approach to integrating the two sets of rankings into a composite test strategy was developed in an iterative process and resulted in the prioritization scheme described in Table 3. This scheme is based on the principle that the highest priority be given to MOVs with a combination of low performance margin and high risk significance and the lowest priority to those MOVs with high performance margins and low risk significance. Three testing priorities denoted as 1, 2 and 3 with 1 as the highest priority were initially established. This scheme was then refined by separating Priorities 1 and 2 into subgroups to further differentiate the MOV test strategies.

Table 3
Test Strategy Categories

Testing Priority Category	Risk Significance Category	Deterministic Margin Category	Initial Testing Strategy
1A	High	Low or Medium	Dynamic
1B	Medium	Low	Static or Dynamic*
2A	High	High	Static
	Medium	Medium or High	
2B	Low	Low	Exercise or Static*
3	Low	Medium or High	Exercise

* Selection based on factors not included in risk and performance margin rankings such as the ability to or feasibility of a dynamic test.

4.1 Blended Deterministic And Probabilistic Evaluations

Once the deterministic and probabilistic rankings have been performed, the two rankings are combined into one final rank to determine the testing priorities and initial inservice test strategies for each functional group. Each MOV group is assigned a risk significant category of HIGH, MEDIUM, or LOW based on the risk importance of each MOV and a deterministic margin of HIGH, MEDIUM, or LOW, with HIGH representing the valves with the largest performance margins. An overall testing priority of 1A, 1B, 2A, 2B, or 3 is assigned with 1A for the highest testing priority as shown in Table 4.

Table 4
STPEGS GL 89-10 MOV
MOV Test Strategy Matrix

Risk Significance Category	**HIGH**	1A	1A	2A
	MED	1B	2A	2A
	LOW	2B	3	3
		LOW	**MED**	**HIGH**
		Deterministic Margin Category		

As indicated above, the assignment of highest testing priority goes with MOVs with a combination of high risk significance and low or medium margins. The lowest priority is only assigned for a combination of low risk significance and medium to high performance margins. Once the final testing priority has been determined, the recommended test strategy can be defined. Table 5 illustrates the type of inservice testing recommended for each of the testing priority categories.

Table 5
Inservice Test Types

Risk Significance Category	**HIGH**	**Performance-based Diagnostic Testing**	DP	DP	Static
	MED		Static or DP*	Static	Static
	LOW	**Exercise Testing**	Exercise or Static*	Exercise	Exercise
			LOW	**MED**	**HIGH**
			Deterministic Margin Category		

* Upgrade of testing methodology to next level based on trends in test performance data or other factors not reflected in risk and margin categories.

Each MOV functional group was placed into a test methodology category. The number of MOVs recommended for each test category are provided in Table 6. This information was presented to an expert panel for evaluation and selection of inservice test strategies.

Table 6
Number of MOVs In Each Test Category

Risk Significance Category	**HIGH**	**Performance-based Diagnostic Testing**	6	10	10
	MED		20	18	16
	LOW	**Exercise Testing**	108	46	68
			LOW	**MED**	**HIGH**
			Deterministic Margin Category		

4.2 Incorporation Of Input From Expert Panel

The results of the deterministic and preliminary probabilistic evaluations and the recommended test strategies were presented to the STPEGS Expert Panel working group for evaluation. The Expert Panel working group included personnel expert in plant, system, and MOV behavior from the operations, maintenance, risk assessment, and design engineering disciplines. The Expert Panel working group was based on the STPEGS Maintenance Rule Expert Panel working group and supplemented to include individuals expert in the MOV program. The working group considered evaluation criteria related to plant, system, and MOV behavior considering safety issues, plant and system operating modes, and key plant support functions beyond the scope of the PSA model.

The plant knowledge, operating experience, and engineering judgment of this panel was used to verify the functional failure modes defined for the MOVs, establish risk-based rankings for MOVs not modeled in the PSA, concur with the application of qualitative deterministic criteria, assure that all significant safety and operational concerns were adequately addressed, concur with the recommended test strategies, and provide an independent check of the results. The draft prioritization results were reviewed on a valve specific and functional group basis. Corrections to MOV specific FFMs and normal positions were identified and

questions relative to PSA model completeness were raised. These comments were resolved and incorporated into the results presented in this paper.

The final testing strategy was determined by the Expert Panel working group by reviewing the initial testing strategy based solely on the combination of risk significance and deterministic margins rankings and by taking into account other factors that are not reflected in these rankings. Such factors include:

- unfeasibility of performing the indicated test
- design features of the valve such a flow assisted operation
- flow characteristics of the system in relation to risk significant failure modes
- capability of less strenuous test in identifying risk significant failure modes
- important design basis functions not reflected in risk ranking
- impact of PSA scope limitations and model simplifications such as exclusion of shutdown states
- importance of release states less severe than large early releases that are not explicitly reflected in the risk ranking scheme

4.3 Integrated Evaluation of MOV Groups

An overview of the impact the Expert Panel working group had in selecting the final testing strategy is provided in Table 7. Key results are discussed below.

Table 7

Impact Of Expert Panel On Inservice Test Strategy

Testing Priority Category	Initial Testing Strategy	Number of Valves (Valve Groups)	Final Testing Strategy
1A	Dynamic	16(4)	6(1) selected for dynamic testing while 10(4) selected for static testing due to unfeasibility of dynamic testing
1B	Static or Dynamic	20(3)	16(2) selected for static; 4(1) for dynamic testing
2A	Static	44(7)	8(1) selected for exercise, 36(6) for static testing
2B	Exercise or Static	108(17)	2(1) selected for dynamic, 50(7) for static and 56(9) for exercise testing
3	Exercise	114(22)	6(1) selected for static, 108(21) for exercise testing
	TOTALS	302(53)	

The initial strategy for priority 1A, the highest priority that was assigned, was diagnostic dynamic testing. This involved 16 MOVs in 4 groups. One of these groups of 6 MOVs was the ECW pump discharge isolation valves which are exercised at design basis differential pressure and flow conditions during the quarterly system surveillance test. In this case, diagnostic testing would be costly and the added benefits over exercising under full differential pressure and flow conditions questionable. While exercise testing would be adequate to verify functional readiness, the selection was made for diagnostic dynamic testing to provide adequate advance warning of possible performance degradation for this high risk significant MOV group. The remaining priority 1A MOVs were not feasible to dynamic test and were therefore recommended for diagnostic static performance testing. These include the pressurizer PORV block valves and the containment sump isolation valves.

For testing priority 1B, the Expert Panel used a default classification of dynamic

testing unless less strenuous testing could be justified. For one group of 4 MOVs, the RCS letdown isolation stop valves, DP testing is not feasible. For another group of 12 MOVs, the RHR pump suction isolation valves, a DP test could be performed but would not be meaningful for the relevant failure modes. Hence, one group of 4 MOVs retained the dynamic testing strategy and two groups totaling 16 MOVs were assigned static testing in Testing priority group 1B.

For testing priority 2A, the initial recommendation of static testing was retained by the Expert Panel for all MOVs except for one group of 8 MOVs. This group was the AFW Pump Discharge Isolation Valves. The risk significant failure mode for this group is failure to open which is flow assisted for these stop-check valves. Neither a static or dynamic test would be meaningful for this configuration, which is adequately supported by an exercise test during AFW pump surveillance testing.

About one third of the 302 MOVs were assigned to Testing Priority Category 2B which has a combination of low risk significance and low performance margins. The Expert Panel used a default classification for these of static testing unless a specific reason could be defined for another classification. One group, the charging line containment isolation valve was actually upgraded to dynamic testing primarily as a surrogate for one group in Category 1B that could not be dynamically tested. The remaining MOVs in Category 2B were distributed roughly equally between diagnostic static testing and exercising for a variety of reasons.

The Expert Panel working group did not change the valve risk ranking or deterministic margin rank; however, test strategies were revised based on test type practicality and other qualitative factors. The inservice test recommendations generated by the Expert Panel working group with comparison to the initial recommendations made to the panel are shown in Table 8.

Table 8
Summary of Final Test Strategies

Stage of Analysis	TEST TYPE		
	Dynamic	Static	Exercise
Prior to Expert Panel	**16**	**64**	**222**
Conclusion of Expert Panel	**12**	**130**	**160**

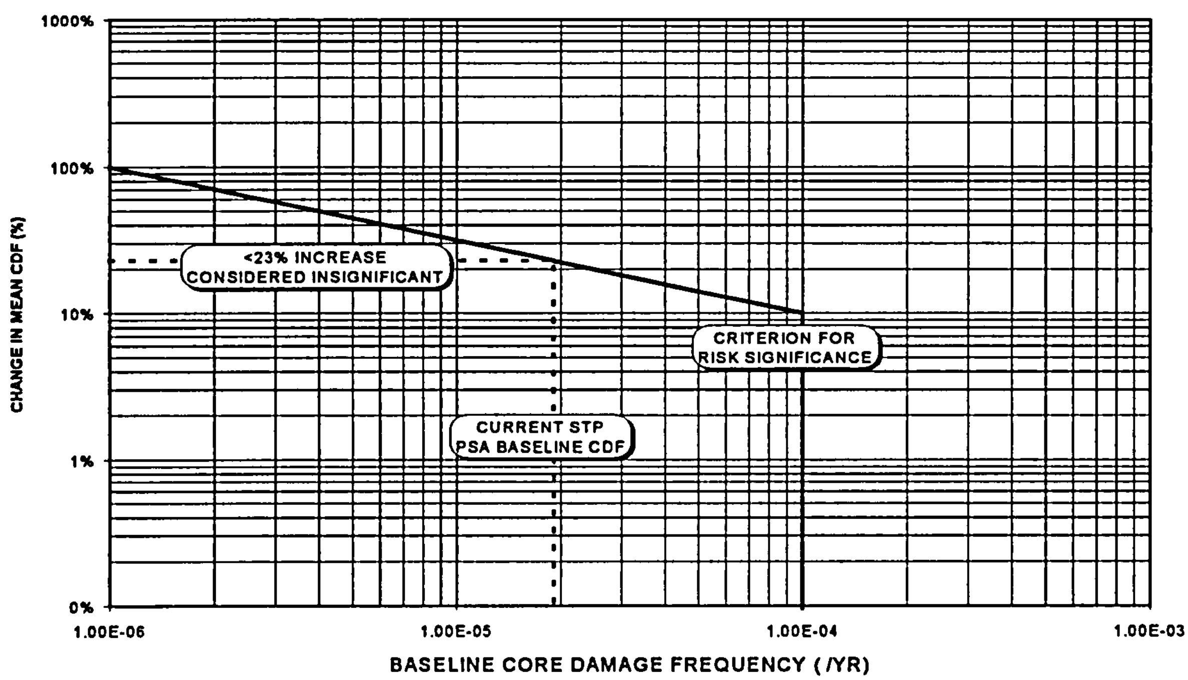
5.0 FINAL RISK EVALUATION OF PROPOSED MOV TESTING STRATEGY

A final risk evaluation is accomplished by updating the PSA to reflect the cumulative risk impacts of the proposed inservice testing strategies. Justification for the proposed MOV test strategies requires that any impact on the risks of a severe accident be within defined acceptance criteria. This risk impact must consider the cumulative risk impacts of the proposed test strategies as applied to the whole set of GL 89-10 MOVs.

5.1 Risk Acceptance Criteria

As described in Section 3.2.1, the STPEGS PSA models were updated to account for issues raised by GL 89-10 as well as the testing performed to date within the STPEGS GL 89-10 program. This PSA update reflects revised MOV failure probabilities based on the GL 89-10 testing and the current inservice test program for MOVs at STPEGS. The recommended MOV inservice test strategies recommended herein will revise the current STPEGS GL 89-10 periodic verification testing commitments as well as the current inservice test program stroke test requirements for these MOVs. As this represents a permanent change to the plant, the permanent change criteria from the PSA Applications Guide is used as acceptance criteria. As illustrated in Figure 5, the permanent change decision criterion is used as basis to verify that any risk impacts are insignificant and allows a 23% increase in CDF.

Figure 5
PSA Applications Guide Permanent Change Criteria
Allowable CDF Increase for STPEGS



5.2 Effect of Test Strategies on MOV Failure Probabilities

The proposed MOV Inservice test strategies have a potential impact on the MOV failure probabilities. As described in Section 6.0, two basic test strategies are recommended; Performance-Based testing and MOV exercising. Performance-based testing differs from exercise testing in that the performance-based test monitors performance trends, quantifies potential performance degradation and predicts potential impending component failure. Exercise testing does not predict impending component failure, rather it verifies the functional readiness of the component by demonstrating that the component has not failed. These two basic test strategies have different effects on MOV failure probabilities as discussed below.

5.2.1 Performance-Based Testing. Performance-based testing is diagnostic testing that monitors and trends MOV performance, quantifies observed performance degradation, and predicts potential MOV failure. The performance-based inservice test is much more effective than current stroke time testing and/or exercising in assessing MOV design basis capability and provides greater confidence in MOV functional readiness. Therefore, it can be reasonably expected that the failure rates for both independent and common cause failures of MOVs where performance-based inservice testing is employed would be improved.

The independent component failure probability is modeled as the sum of a demand failure probability and a time dependent failure probability as shown in Equation 10.

$$P_f = P_d + \lambda T/2 \tag{10}$$

Where:

P_f = Component Failure probability
P_d = Demand Failure rate probability
λ = Standby Failure rate
T = Test intervals for a given component

A similar equation can be expressed for the component common cause failure probability. The difficulty in this formalism is that the two components of the failure probability are unknown. Performance based testing will tend to reduce both the demand failure probability and the time-dependent failure rate for both independent and common cause failures. However, no data exists to either prove or quantify this effect. Therefore, the potential reduction in MOV failure probability due to periodic performance-based testing was not considered in this study which provides a significant level of conservatism.

5.2.2 MOV Exercise Testing. Exercise testing differs significantly from performance-based testing in that it does not predict impending component failure, rather it verifies the functional readiness of the component by demonstrating that the component has not failed. While MOV exercising is not expected to increase the MOV failure rate, it does impact the failure probability. The impact on failure probability comes from the fact that MOV exercising will not identify failure until the failure has occurred. Exercise testing is also less capable then performance-based testing in identifying the existence of common cause failure modes. It is important to note that, should MOVs recommended for exercise testing only begin to experience increase failure rates, corrective actions are taken which may

include the implementation of performance-based testing on the MOVs.

To quantify the risk impacts of revising the inservice test commitment for the MOVs recommended for exercise testing only, it is necessary to compare both the capability and frequency of the exercise test to the existing inservice or surveillance tests. Reduction in either test frequency or capability to identify component failure could lead to an increase in component failure rate and failure probability. These impacts must be considered separately for independent and common cause failure modes.

The approach to quantifying risk impacts of MOV recommended for exercise testing only is to perform sensitivity studies to assess the various levels of increased failure rates and demonstrate that the final risk levels are insensitive to this issue. This is true since there are no data available to quantify possible correlation between component failure rates and changes in test strategy.

5.3 PSA Requantification

The cumulative effect of the recommended MOV inservice test strategies was treated using the guidance and criteria from the PSA Applications Guide. Here the STPEGS PSA model was recalculated by revising the failure probabilities for MOVs recommended for exercise testing only. The revised failure probabilities were inserted into the model and the model resolved. This result was then compared with the CDF and LERF allowable changes from the PSA Application Guide. Two sensitivity studies were performed as described below.

The actual impact on MOV failure probabilities of inservice testing by only MOV exercising is uncertain. Therefore, sensitivity studies were performed to study the impact on plant risk.

The revised MOV failure probability is bounded very conservatively by assuming all MOVs recommended for exercise testing only have a failure probability of 1.0. The first sensitivity study was performed assuming all MOVs recommended for exercise testing only were failed.

A more realistic model for the increased failure probability is to assume the MOV failure probability increases linearly as a function of test interval. MOV exercising is to be performed on a once-per-fuel cycle time-dependent frequency which translates to an 18 month frequency at STPEGS. The MOVs recommended for exercise testing only are currently tested on a 3 month frequency. In this model, the current failure probability is increased by a factor of 6 to account for the recommended change in inservice test interval. The second sensitivity study was performed by increasing the failure probability for MOVs recommended for exercise testing only by a factor of 6.

The final PSA quantification was still in progress at publication of this paper. Final adjustments will be made to the recommended inservice test strategies to insure that possible risk increases indicated by these sensitivity studies satisfy the PSA Applications Guide permanent change criteria.

6.0 IMPLEMENTATION OF TEST STRATEGIES

The primary objective of this study was to optimize the periodic verification testing portion of the STPEGS GL 89-10 program. This optimization included the assessment of MOV safety significance through blended probabilistic and deterministic evaluations and presentation of initial test strategies to an Expert Panel for final selection of the most appropriate strategy. This section of the paper describes the implementation of the optimized test strategies.

Optimization of inservice test strategies was based on MOV safety significance. The American Society of Mechanical Engineers (ASME) has recently published Code Case OMN-1, which provides a performance-based inservice test methodology for MOVs that can be used as an alternative to the prescriptive, time-based inservice test methodologies specified by the ASME OM Code (Reference [7]). The performance-based inservice test methodology of OMN-1 includes a mix of static and dynamic MOV performance testing as well as MOV exercising. OMN-1 also allows the consideration of MOV safety significance in the application of risk-based criteria for MOV testing.

The optimized test strategies recommended for the STPEGS GL 89-10 periodic verification test program are based on the alternative guidance provided in ASME OMN-1 with risk-based criteria included. The risk-based criteria for MOV testing was applied to the development of the specific test strategies in that performance-based testing and exercising is recommended for the more safety significant MOVs and only MOV exercising is recommended for the less safety significant MOVs. Performance-based test frequencies are based on observed performance trends while MOV exercising is implemented on a time directed frequency.

Performance-based testing differs significantly from exercise testing in that the performance-based test monitors performance trends, quantifies potential performance degradation and predicts potential impending component failure. Exercise testing does not predict impending component failure, rather it verifies the functional readiness of the component by demonstrating that the component has not failed. These test strategies are described in the following sections.

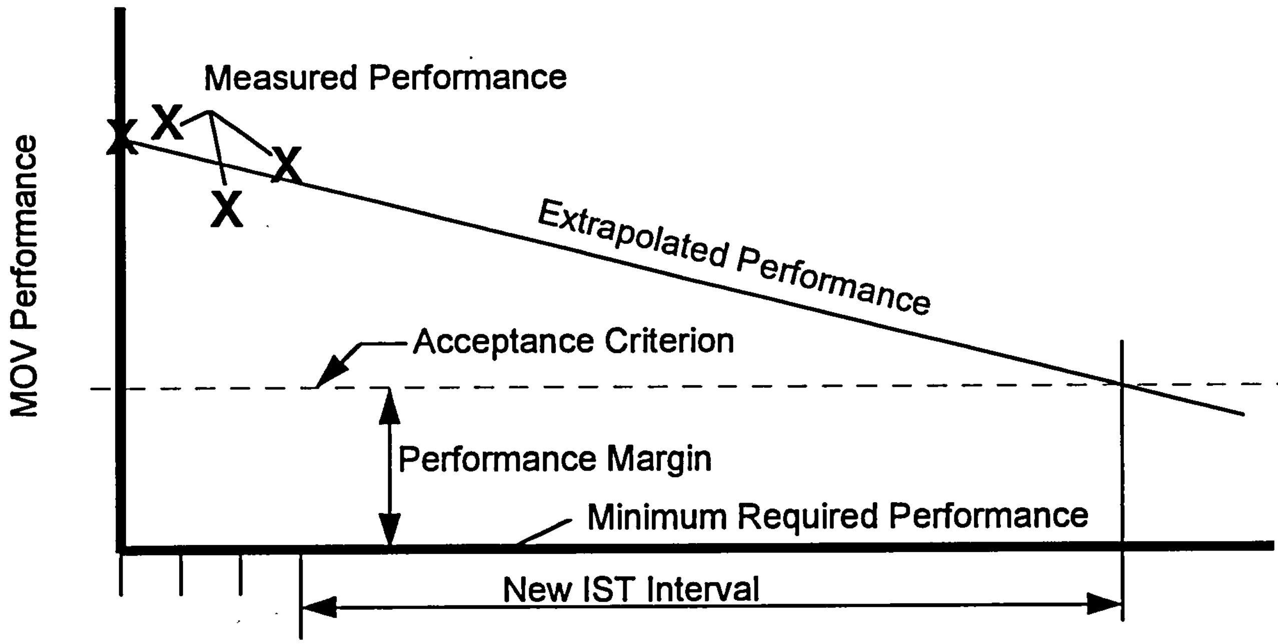
6.1 Performance-based Testing.

The performance-based inservice test model begins with the establishment of performance acceptance criteria and performance of a "preservice" test. The preservice test is to be performed under conditions as near as practicable to those expected during subsequent inservice testing. The preservice test establishes the initial performance data point to which subsequent inservice test performance data will be compared. Test acceptance criteria are established with consideration of uncertainties such as test measurement uncertainty, MOV performance uncertainty, and allowance for potential performance degradation.

Once the acceptance criteria are established and the preservice test performed, the performance-based inservice testing may begin. Acceptance criteria are MOV or MOV group specific and can be based on

thrust, torque, friction, or other measured parameter indicative of MOV performance. The performance-based testing trends measured MOV performance and calculates the next test interval based on observed changes in performance. The performance-based model is shown graphically in Figure 6.

Figure 6
Performance Model For Determining Test Intervals



As shown in Figure 6, the inservice test interval is determined by extrapolation of measured performance. This performance-based model can be applied to single MOVs as well as MOV groups. In applying the methodology to MOV groups, a specific performance-based model is established for each individual MOV and the performance changes from inservice tests within the group are applied to each individual MOV model as percentage changes in performance.

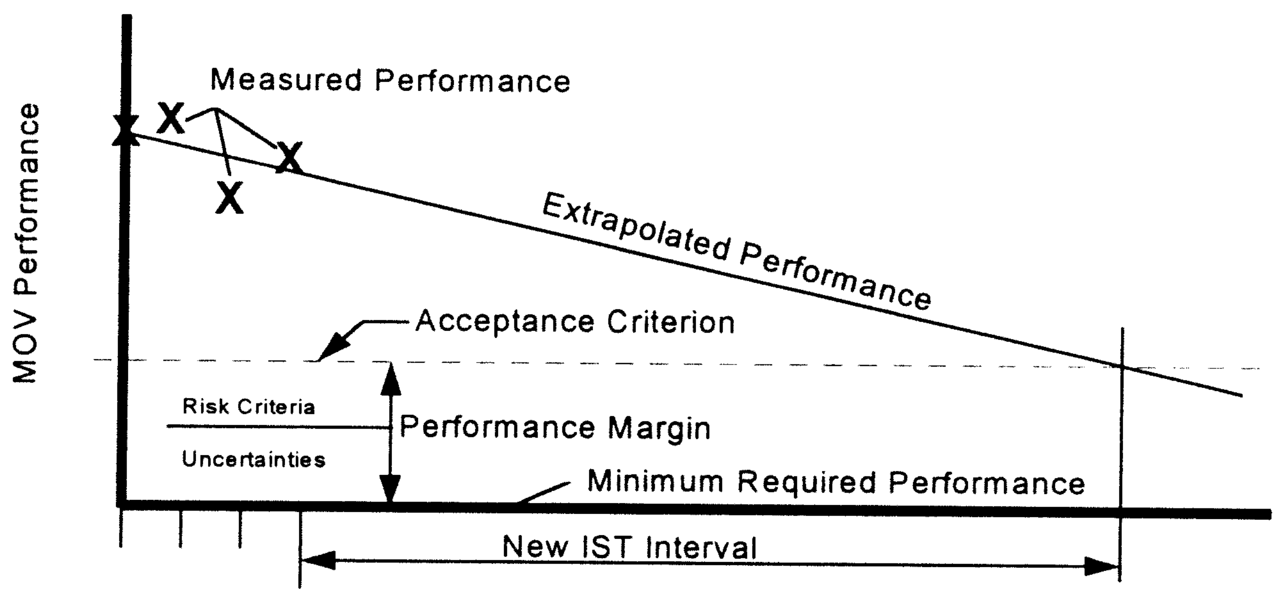
MOV performance uncertainties include such items as torque switch repeatability and measurement uncertainty. Therefore, minor variations in performance are expected. These variations in performance from one test to the next do not necessarily indicate performance degradation nor do they demonstrate lack of degradation. However, three data points are considered adequate for initial extrapolation of performance trends.

Because insufficient performance data currently exists at STPEGS to extrapolate performance trends, the initial inservice test frequency must be time-based. The recommended initial test frequency is two

(2) refueling cycles or three (3) years, whichever is longer, until three (3) data points are available for each MOV. This does not mean that each MOV is recommended for testing on this frequency but that at least one MOV in each MOV group must be tested within this interval. However, each MOV in the group must be tested within the maximum recommended frequency. The recommended maximum inservice test frequency for the performance-based model is ten (10) years.

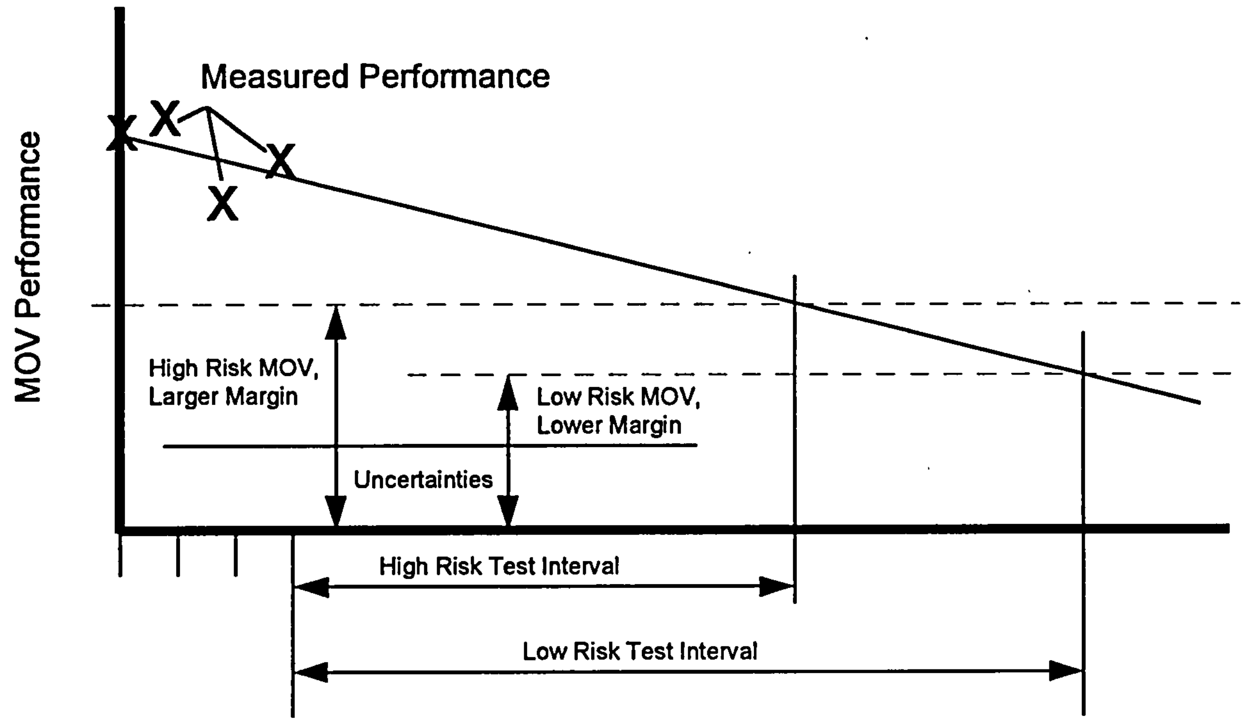
The performance-based model described above does not incorporate risk insights beyond those used to choose the performance-based test strategy. The STPEGS optimized MOV inservice test program also incorporates risk criteria into the performance-based inservice test model. This is accomplished by applying the risk criteria to the performance margin used to establish minimum acceptance criteria as shown in Figure 7.

Figure 7
Incorporation Of Risk Criteria In Performance Margins



The risk-based criteria applied to the performance margin is also graded commensurate with MOV safety significance such that a larger risk-based performance margin is applied to more safety significant MOVs and lower risk-based margins are applied to lower safety significant MOVs as shown in Figure 8.

Figure 8
Performance-based Model
With Graded Risk Criteria Applied



6.2 <u>Exercise Testing.</u>

As discussed earlier, exercise testing differs from performance-based testing in that it verifies the functional readiness of the component by demonstrating that the component has not failed rather than predicting impending failure based on performance trending. MOV exercising is recommended for all GL 89-10 MOVs regardless of safety significance. Inservice testing is limited to MOV exercising for the low safety significant MOVs and was demonstrated to have minimal impact on plant risk.

Exercise testing consists of one full stoke operation of the MOV. Regular MOV operation during normal plant or system operation and maintenance can be credited as an exercise test provided the stroke is documented and successful stroking can be verified. The recommended frequency for exercise testing is one (1) fuel cycle.

Should exercise testing identify a failure, corrective action should be taken consistent with normal plant procedures. However, if exercise testing identifies repeated failures of a single MOV or repeated failures in a group of MOVs, strong consideration should be given to upgrading the test strategy for these MOVs to performance-based testing.

7.0 REFERENCES

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Thermal-Hydraulic Analysis for Changing Feedwater Check Valve Leakage Rate Testing Methodology

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ABSTRACT

The current design and testing requirements for the feedwater check valves (FWCVs) at the Grand Gulf Nuclear Station are established from original licensing requirements that necessitate extremely restrictive air testing with tight allowable leakage limits. As a direct result of these requirements, the original high endurance hard seats in the FWCVs were modified with elastomeric seals to provide a sealing surface capable of meeting the stringent air leakage limits. However, due to the relatively short functional life of the elastomeric seals compared to the hard seats, the overall reliability of the sealing function actually decreased. This degraded performance was exhibited by frequent seal failures and subsequent valve repairs. Thus, adherence to the original licensing requirements has resulted in higher operating costs, longer outages, more difficult testing methods, higher station personnel radiation doses, and an overall degradation in equipment performance.

The original requirements were based on limited analysis and the belief that all of the high energy feedwater vaporized during the LOCA blowdown. These phenomena would have resulted in completely voided feedwater lines and thus a steam environment within the feedwater leak pathway. Given this condition, the appropriate testing criteria would thus be based on air with a relatively tight allowable limit. To challenge these criteria, a comprehensive design basis accident analysis was developed using the RELAP5/MOD3.1 thermal-hydraulic code. Realistic assumptions were used to more accurately model the post-accident fluid conditions within the feedwater system.

The results of this analysis demonstrated that no leak path exists through the feedwater lines during the reactor blowdown phase and that sufficient subcooled water remains in various portions of the feedwater piping to form liquid water loop seals that effectively isolate this leak path. These results provided the bases for changing the leak testing requirements of the FWCVs from air to water. The analysis results also established more accurate allowable leakage limits, determined the real effective margins associated with the FWCV safety functions, and led to design changes that improved the overall functional performance of the valves.

INTRODUCTION

Many Boiling Water Reactor (BWR) feedwater systems in the nuclear industry contain safety related check valves that perform a containment isolation function similar to those at Grand Gulf Nuclear Station (GGNS). In order to ensure proper check valve operation under the guidance of 10CFR Part 50, Appendix J, a leakage test must be performed on these valves during refueling outages. At many plants, this leakage test is performed using air as the test medium based on the belief that the check valves will not remain liquid covered after a large break loss of coolant accident (LOCA) and subsequent reactor vessel depressurization. The air leakage test requires draining the feedwater piping and pressurizing the feedwater piping with air.

It has been a major challenge for the GGNS feedwater check valves (FWCVs) to pass an air leakage test. The air leakage testing criteria is very stringent, particularly for a piping environment that typically consists of subcooled liquid during power operation. As a result of several air test failures, the original high endurance hard seats in the FWCVs, which were designed for the typical liquid operating environment, were modified by the addition of elastomeric seals. The elastomeric seals do not perform well under the typical high temperature (greater than 480 °K), high flow (greater than 950 kg/sec) liquid operating environment in the feedwater piping and consequently do not always pass the air leakage test on the first attempt. In order to test the feedwater check valves with air, the feedwater piping has to first be drained which consumes outage time. Even with the addition of soft seats, the feedwater check valves still require frequent rework to pass the air leakage test. Rework of these check valves is a significant contribution to personnel dose.

Because of problems created by the air leakage test requirement, it is believed that water would be a more desirable testing media for the feedwater check valves. The water test would reduce check valve test setup time, personnel dose and check valve maintenance. Therefore, a water leakage testing methodology for the feedwater check valves was investigated. This investigation involved identifying the accident scenarios that required the air leakage test for the feedwater check valves. The scenarios investigated were those that had the potential to depressurize the feedwater piping, which removes the water from the piping, and allow for radioactive effluents from the reactor vessel to reach the condenser through the feedwater piping. The most conservative scenario identified was a design basis accident (DBA) LOCA with loss of offsite power (LOP), which was a recirculation suction line break (approximately 0.29 m^2 equivalent area). Another obvious scenario investigated was a feedwater line break (approximately 0.034 m^2) in the drywell, which did not result in significant fuel failure; and therefore, did not create the potential for significant dose consequences.

The analysis performed determined if there is enough energy in the feedwater piping to remove the liquid feedwater from the feedwater piping by boiling during depressurization of the piping and reactor vessel after a design basis recirculation suction line break. If enough water is removed from the feedwater piping during depressurization from normal operating conditions to depressurized accident conditions, then a flow path could exist for radioactive effluents from the reactor vessel to the condenser. The flow of radioactive effluents from the reactor vessel through a flow path to the condenser could

produce significant dose consequences in the turbine building which is not part of secondary containment. The goal of this analysis was to determine whether such a flow path existed based on some portion of the feedwater piping remaining water solid throughout the analyzed event.

This paper presents a summary of the thermal hydraulic analysis performed with a RELAP5/MOD3.1 feedwater piping model. The RELAP5/MOD3.1 code was chosen to perform the thermal hydraulic analysis based on the availability of two phase flow and phase change correlations and based on the degree of software qualification already completed for the code. The results of the model provide justification for changing the air leakage test requirement for feedwater check valves to a water leakage test requirement.

FEEDWATER SYSTEM MODELS

A transient thermal hydraulic analysis utilizing RELAP5/MOD3.1 is completed to demonstrate the timing of the feedwater piping depressurization, the heat transferred (including heat transfer associated with the fluid and walls), the magnitude/direction of flow rates in the feedwater piping and the amount/location of water left in the feedwater piping. The accident conditions modeled represent the reactor vessel response for a DBA LOP/LOCA. The calculated amount of liquid inventory left in the feedwater piping is used to assess the reasonableness of water testing for the feedwater check valves based on the potential presence of water seals in the feedwater piping. The RELAP5/MOD3.1 feedwater system model is also used to evaluate the potential for a steam flow path from the reactor vessel to the condenser to assess any potential dose impact. In addition to the liquid inventory information, data such as feedwater piping pressures, reactor vessel pressures and the direction of flow at various points in the feedwater piping (e.g., the inboard and outboard feedwater isolation check valves) are used to support this evaluation.

A model of the feedwater piping from both of the reactor feedwater pumps (including the minimum flow lines to the condenser) to the reactor vessel was built using the RELAP5/MOD3.1 code. The model duplicates the physical layout of the feedwater piping with the model's initial conditions set at 100% power operating conditions. Figure 1 illustrates the physical feedwater piping layout, which is critical to the analysis since lower elevation piping is the most likely collection point for any water left in the lines after depressurization. Figure 2 presents a nodalization diagram for the feedwater system model. Table 1 provides a description of the feedwater system components, other than piping, that are part of the RELAP5/MOD3.1 feedwater system model.

As stated previously, an important aspect of the feedwater model is the accuracy of the feedwater piping elevations. The elevations are crucial to identify piping with potential water loop seals. Once the feedwater model was developed, an input check of the RELAP input deck was made using RELAP5/MOD3.1 to verify continuity of the piping and associated elevations.

Next, a steady-state RELAP5/MOD3.1 model of the GGNS feedwater system was run to assess initial conditions. A subsequent null transient was run with the initial conditions from the steady-state model to verify that the physical inputs, such as flow resistance values (i.e., loss factors), volume sizes and heat

structure input values (for pipe walls and feedwater heaters), were correct. This model was benchmarked against actual plant operating data including flows, pressures and temperatures at various points in the feedwater system. In general, the null transient produced very accurate results when compared with plant operating data. The null transient was also used to provide a restart input file for the transient analysis. Time step and noding sensitivities were assessed to ensure that the most conservative and accurate model was created.

Some assumptions made for the feedwater piping and valve components during the transient analysis have the potential to affect the results of the analysis and should therefore be discussed. The feedwater piping is assumed rigid and its pressure boundary maintained during the reactor vessel depressurization. A plant seismic walk down and evaluation was performed on the feedwater piping to support this assumption. It should be noted that some of the feedwater piping was originally designed as non-seismic, but is still assumed to maintain its pressure boundary in this analysis based on the walk down and evaluation. The closed safety related feedwater inboard (B21-F010A/B) and outboard (B21-F032A/B) check valves are assumed to have a leakage area equal to 1.5% of the total opened check valve area. The 1.5% number is based on an assumed failure of the elastomer seals which are not very reliable under expected normal operating conditions. The maximum elastomer seal area percentage of the inboard and outboard check valve areas is under 1.5%. This number (1.5%) is conservatively used for both styles of check valves. The closed non-safety related reactor feedwater pump discharge (N21-F015A/B) check valves are assumed to have a leakage area equal to 1.5% of the total disk area. The 1.5% number is based on the percentage used for safety related check valves. For the pump discharge valves, a 1.5% leakage area equals 2.02×10^{-3} m^2. The closed non-safety related reactor feedwater pump minimum flow (N21-F503A/B) valves are assumed to have a leakage area equal to 1.5% of the total pipe flow area or 8.76×10^{-4} m^2. These valves are closed throughout the event and are consequently modeled as a regular junction with an area equal to the leakage area. The outboard motor operated containment isolation valves (B21-F065A/B) are assumed opened for the duration of the analysis since they are manually operated valves and no credit is taken for operator action to close the valves.

METHODOLOGY

The feedwater piping model is depressurized by coupling the feedwater sparger connections to reactor vessel pressure data from a previous DBA LOP/LOCA analysis. The reactor feedwater pump minimum flow line valve leakage is exhausted to atmospheric conditions in a condenser volume.

The criterion used for determining a leak path from the reactor vessel to the condenser is whether there are sections of feedwater piping filled with water which could provide a loop seal that would prevent and/or scrub any releases from the vessel to the condenser. The presence of a loop seal is determined using the RELAP5/MOD3.1 code. The loop seals are primarily assessed from the code results by looking at piping modeled at the lower elevations. If the ratio of liquid volume left in the pipe to the total pipe volume is equal to one, then a loop seal is present. The leak path is also evaluated based on the reactor vessel pressures, feedwater piping pressures and the differential pressure effects

on the direction of flows. If steam and liquid flows are being driven towards the reactor vessel and not the condenser based on existing pressures (i.e., feedwater piping pressures are greater than reactor vessel pressures), then release of radioactive effluents is not a concern. Another set of RELAP5/MOD3.1 results used are the volumetric flow rates at the feedwater check valves (B21-F010A/B, B21-F032A/B and N21-F015A/B) and the feedwater system isolation motor operated valves (B21-F065A/B). These flow rates provide a magnitude and direction of the feedwater piping flow. The flow rates' acceptance criteria would be flows going towards the reactor vessel.

Based on the ANSI/ANS-56.8-1981 discussion of containment leakage rate testing, water filled systems are defined as systems that are designed to contain water subsequent to a leakage design basis loss of coolant accident, such that the containment isolation valves seating surface remains water covered (considering the water volume and water leakage of the isolation valve) for at least 30 days. Per the ANSI standard, valves in water filled systems may be tested with water. This definition is established to provide a criteria for preventing the containment atmosphere from escaping to the environment. This analysis determines the requirements for the feedwater check valves' leakage testing based on the blowdown phase of a DBA LOP/LOCA. Subsequent to the blowdown phase (at approximately twenty minutes into the event), the feedwater leakage control system (FWLCS) provides the necessary makeup water to maintain water filled sections of piping and justify water leakage testing of the feedwater check valves.

During the DBA LOP/LOCA and associated feedwater piping depressurization, the check valves may be opening and closing for a number of reasons. As an example, the check valves will be cycling opened and closed initially due to the coast down of the tripped reactor feedwater pumps combined with feedwater vaporization (or flashing to steam) based on the saturation pressure being reached during depressurization. When the safety related feedwater check valves are opened, then fluid flow is towards the vessel in the safety related feedwater piping and neither a pneumatic or hydraulic test is needed.

In summary the feedwater piping depressurization will be evaluated using the RELAP5/MOD3.1 model results based on the following criteria:

1. A volume fraction of one (1) in a volume or subvolume indicates that a pipe or section of a pipe is water (liquid) solid.

2. Feedwater piping pressures greater than reactor vessel pressure indicate that the differential pressure would drive fluid flow towards the reactor vessel.

3. The magnitude and direction of feedwater piping flow rates at the check valves, minimum flow valves and motor operated valves determine if there is steam or water flowing from the reactor vessel to the condenser. The flow rates are assessed at the feedwater spargers, the inboard and outboard check valves, the reactor feedwater pump discharge check valves and the motor operated isolation valves.

4. A closed position of the safety related check valves demonstrates that only the leakage flow area is available for reverse flow, while an open position of the

safety related check valves indicates that no leakage testing is required since flow through the valves is towards the vessel and the valves are not closed.

STEADY-STATE/NULL TRANSIENT MODEL RESULTS

Table 2 documents the RELAP5/MOD3.1 null transient model results for key feedwater flows. The table data, as well as other RELAP5/MOD3.1 flow results, demonstrate accurate flow balances at several merging and branching points in the feedwater system. The null transient model flow results were benchmarked against plant data for 100% operating conditions. The model results matched within 0.5%.

The RELAP5/MOD3.1 model temperature distributions were benchmarked against design feedwater temperature values due to the availability of data. The temperature distribution comparison is made between the RELAP5/MOD3.1 output and temperatures at a design feedwater flow of 2072.8 kg/s. As Table 3 illustrates, the temperatures compare reasonably well. Other operating plant data, such as pressures, were also compared with the null transient model results. These comparisons also indicated that an accurate and steady-state null transient model was achieved.

TRANSIENT DEPRESSURIZATION MODEL RESULTS

When a LOP/LOCA occurs, feedwater flow is assumed to completely stop in five seconds (after the reactor feedwater pumps coast down). The transient RELAP5/MOD3.1 model simulates this coast down using a time-dependent junction. As previously stated, the reactor vessel is modeled as a time dependent volume that uses reactor vessel pressure data from a previous DBA LOP/LOCA analysis. The initial temperature of feedwater entering the reactor vessel is approximately 488 °K with a saturation pressure equal to approximately 2.12×10^6 Pa. It is expected when a subcooled liquid is depressurized or decompressed, that the liquid reaches a point where the local pressure decreases below the saturation pressure as dictated by local temperature. When the liquid's local pressure falls below the saturation pressure, vapor formation occurs. Vaporization removes heat (in the form of the heat of vaporization) from the liquid which reduces the local temperature. Depending upon the existence and magnitude of external heat sources (such as the metal pipe walls), the removal of heat from the liquid due to vaporization eventually returns the local temperature to the saturation value.

The pressure in the feedwater piping near the reactor vessel will closely follow the reactor vessel pressure after the five second reactor feedwater pump coast down. When the reactor vessel pressure initially falls below the feedwater saturation pressure of 2.12×10^6 Pa (for 488 °K), some of the water in the feedwater piping near the reactor vessel will begin flashing to steam. Feedwater continues to flash (or vaporize) in the feedwater piping as reactor vessel pressure decreases below feedwater saturation pressure. The pressure in the feedwater piping follows (e.g., slightly lags behind) reactor vessel pressure and vaporization occurs beginning at feedwater piping nearest the reactor vessel and progressing towards the reactor feedwater pumps. At the same time, vaporization also occurs from the minimum flow feedwater piping going to the condenser (with steam flow in the direction of the condenser). The

condenser pressure is approximately 1.01 x 10^5 Pa. This vaporization progresses towards the reactor feedwater pumps also, but from the opposite direction of the reactor vessel. Therefore, the timing associated with the reactor vessel depressurization is critical to determine the amount of liquid feedwater left in the feedwater piping, the feedwater piping pressures and the direction of flows between the feedwater piping and the reactor vessel and condenser. A brief accident chronology for a design basis recirculation suction line break is given in Table 4 to illustrate the timing of the feedwater piping blowdown.

The following discussion of results is presented for the B loop of the feedwater system only. The results for the A loop are very similar with the exception of effects produced by minor geometry/piping length differences and an additional swing check valve (B21-F803) in the B loop that does not exist in the A loop. The B21-F803 valve is not shown on Figure 1, but is located between the FWLCS connection and B21-F065B.

The B21-F010B check valve results are presented in Figure 3. The steam and liquid volumetric flow rate plots are shown with the check valve position plot. When the check valve is open, flow is in the direction of the reactor vessel. The initial liquid flow in the direction of the reactor vessel ceases as the reactor feedwater pumps trip and coast down. After the feedwater pumps have coasted down and prior to the feedwater saturation pressure being reached, reactor vessel pressure drives liquid flow pass the check valve in the direction of the condenser. The elevation difference between the feedwater spargers and the check valve also produces a water column in the piping which tends to cause liquid flow in the reverse direction (i.e., towards the condenser) for a few seconds. Once the saturation pressure of the highest temperature feedwater is reached in the reactor vessel (at approximately twenty-five seconds), steam flow begins and any steam or liquid volumetric flow is maintained in the direction of the reactor vessel. The inboard check valve position cycles throughout the reactor vessel depressurization. This cycling, combined with the flow direction of the steam and liquid towards the reactor vessel, supports the fact that neither a hydraulic or pneumatic test is representative of the conditions in the feedwater piping during reactor vessel depressurization (prior to initiation of FWLCS). After FWLCS initiation, the hydraulic test would be a representative testing media for the feedwater piping.

The cycling response of one of the outboard feedwater check valves (B21-F032B) is presented in Figure 4. The steam and liquid volumetric flow rate plots are again shown with the check valve position. The general outboard check valve flow response differs from the inboard check valve (B21-F010B) flow response only based on the point in time that cycling stops (approximately ten minutes into the event versus twenty-three minutes into the event). Other than the time frame over which the outboard check valve cycles, the results and conclusions are similar to those for the inboard check valve.

The flow rates at the feedwater spargers are presented in Figure 5. The six feedwater spargers are plotted together on two plots, one indicating liquid flow rates and the other indicating steam flow rates. Like at the inboard and outboard check valves, the liquid flow rates change direction from towards the reactor vessel (during reactor feedwater pump coast down) to towards the condenser, then back towards the reactor vessel after the highest temperature feedwater saturation

pressure is reached in the reactor vessel at approximately twenty-five seconds into the event. Unlike at the check valves, a small steam flow rate exists at the feedwater spargers in the direction of the condenser during the first twenty-five seconds. Once the saturation point is reached, steam flow changes direction to towards the reactor vessel. Throughout the reactor vessel depressurization, the liquid flow rates at the spargers dissipate and any significant steam flow rates at the spargers are in the direction of the reactor vessel. The liquid flow rate ceases at approximately eleven minutes into the event. Some very small oscillations of liquid flow rates towards the condenser do exist prior to this point, but this liquid flow does not advance beyond the safety related check valves as indicated on the check valve flow plots. The steam flow rates are oscillatory during the entire event, but there are no significant flows observed towards the condenser. Small oscillations of the steam flow do proceed in the direction of the condenser, but this flow does not advance beyond the safety related check valves as indicated on the check valve flow plots.

The flow rates at one of the outboard isolation motor operated valves (B21-F065B) are shown on Figure 6. These flows are again similar to the inboard check valve and outboard check valve flows for the first twenty-five seconds. At this point (when the reactor vessel reaches the saturation pressure of the feedwater), the outboard isolation valve flows begin to differ from the feedwater sparger, inboard check valve and outboard check valve flows. The liquid flow rate dissipates like the sparger and check valve flow rates; however, the steam flow rate at the outboard isolation valve oscillates with flows going in the direction of the reactor vessel and the condenser during first ten minutes of the reactor vessel depressurization. The magnitude of the outboard isolation valve flow rates in the condenser direction is very small with respect to the flow rates in the reactor vessel direction. These flow rates in the direction of the condenser and in the direction of the feedwater heater loop seals will not advance to the condenser due to the loop seals at the feedwater heaters. When the flow oscillations settle, the steam flow is in the direction of the reactor vessel and there is no liquid flow rate.

The flow rates at one of the reactor feedwater pump discharge check valves (N21-F015B) are presented in Figure 7. At the onset of the depressurization, the liquid flow rate goes from forward flow to the reactor vessel to reverse flow to the condenser. At approximately five minutes into the depressurization, the liquid flows halt and steam flow appears at the check valve. The steam flow exhausts to the condenser via the minimum flow line. The steam and liquid flowing to the condenser through the reactor feedwater pump discharge check valves are both from existing water in the piping and not from water in the reactor vessel. This determination is based on the initial volume of water in the feedwater piping, the existence of water loop seals near the feedwater heaters and the fact that no steam from the vessel advances beyond the safety related inboard and outboard feedwater check valves. Although a small amount of liquid flow does advance towards the condenser through the safety related inboard and outboard check valves during the first twenty-five seconds of the event, this liquid flow poses no potential increase in dose. This liquid flow is at the same conditions as the normal liquid in the feedwater piping.

The liquid inventory left in the feedwater piping for loop B is documented in Figures 8

and 9 through plots of the liquid volume fraction (i.e., the liquid void fraction). The feedwater piping evaluated is the safety related feedwater piping inside the containment, the feedwater piping between feedwater heaters 5B and 6B and the feedwater piping at the inlet of feedwater heater 5B. The plots of liquid volume fractions for the safety related feedwater piping include pipe sections between the inboard feedwater check valve (B21-F010B) and the reactor vessel, between the inboard feedwater check valve and the B21-F803 check valve, between the B21-F803 check valve and the outboard feedwater check valve (B21-F032B) and beyond the outboard feedwater check valve to the motor operated isolation valve (B21-F065B). Using the liquid volume fraction plots, the largest liquid volume fraction in the safety related piping approaches only twenty percent over thirty minutes.

The liquid volume fraction plots of the feedwater piping around the feedwater heaters include pipe sections between feedwater heaters 5B and 6B and at the inlet of feedwater heater 5B. For each set of the feedwater heater piping plots, the liquid volume fraction results for three sections of pipe are presented. Loop seals are displayed in all pipe sections for a little over the first one and a half minutes of the depressurization. For the piping between feedwater heaters 5B and 6B, the water volume fraction in all three pipe sections next decreases for about nine minutes, then the water volume fraction in two pipe sections (the horizontal and 45 degree [vertical angle] pipe sections) increases again at under eleven minutes into the depressurization. In the pipe section between feedwater heaters 5B and 6B with the largest liquid volume fraction (the horizontal section), the liquid volume fractions vary between six tenths and one from eleven to thirty minutes. In the case of feedwater pipe sections at the inlet of feedwater heater 5B, at least one pipe section remains water solid for over the first four and a half minutes of the depressurization. For the next approximately seven minutes, all three pipe sections at the inlet to feedwater heater 5B contain steam in varying quantities. The liquid volume fraction for the pipe section (the horizontal pipe section) containing the most liquid during this seven minute time period never decreases below eight tenths. At approximately eleven and a half minutes into the depressurization, the liquid volume fraction in at least one of the three pipe sections (the horizontal and/or the vertical pipe section) returns to one and remains equal to one (i.e., water solid) for the rest of the event. It must be noted that when the water volumes at the inlet to feedwater heater 5B are challenged (i.e., when the liquid volume fraction is less than one), the safety related feedwater piping is depressurizing in the direction of the reactor vessel. There are no observable, significant flows from the reactor vessel to the condenser during this time period (from four and half minutes to eleven and a half minutes). Therefore, the water seals established at the inlet of feedwater heater 5B are sufficient to prevent any flow from the reactor vessel to the condenser. During the seven minute time period when the water seals at the inlet of feedwater heater 5B are not water solid, at least one of the pipe sections remains more than eighty percent filled with liquid which would scrub any steam coming from the reactor vessel to reduce any potential dose consequences. Additionally, during the entire thirty minutes of the depressurization, the liquid volume percentage in the horizontal pipe section between feedwater heaters 5B and 6B remains above thirty-five percent (and above sixty percent beyond eleven minutes), which would also help scrub steam coming

from the reactor vessel to reduce any potential dose consequences.

The pressures in various sections of the feedwater piping are documented in Figures 10 and 11 through plots of feedwater pipe section pressures and reactor vessel pressure. The pipe sections are identical to those used for the liquid volume fraction plots. During the first approximately twenty-five seconds of the reactor vessel depressurization, the feedwater piping pressures are less than reactor vessel pressure. The reactor vessel pressure at which the feedwater piping pressures become greater than the reactor vessel pressure ranges from 2.103×10^6 Pa to 2.206×10^6 Pa. These pressures bound the saturation pressure for 488 °K water (the initial feedwater temperature near the reactor vessel), which is approximately 2.12×10^6 Pa. It is concluded when the pressure in the feedwater piping reaches saturation, water flashes to steam which expands towards the vessel and maintains the pressure in the feedwater piping at saturated conditions. When the pressure in the feedwater piping is greater than or equal to the reactor vessel pressure, there is not a steady pressure driving force for steam to leave the reactor vessel and travel towards the condenser. The pressure plots illustrate that the safety related piping has to depressurize for approximately ten to fifteen minutes before the pressure in the safety related piping approaches the reactor vessel pressure. In the case of the piping at the feedwater heaters, the pressure remains slightly (1.38×10^4 Pa to 2.07×10^4 Pa) higher than reactor vessel for all of the event (excluding the first 34 to 36 seconds). The safety-related piping and the feedwater heater piping pressure plots do not demonstrate a steady pressure driving force from the reactor vessel to the condenser.

CONCLUSIONS

In summary, the flow rates at the feedwater check valves and feedwater spargers demonstrate no steam flows to the condenser during the feedwater line depressurization. The flows at the motor operated isolation valves (B21-F065A/B) went in both directions (i.e., towards the condenser and the reactor vessel), but these bi-directional flow rates are due to the saturation pressure of the water in the feedwater piping being reached and steam expanding through the path of least flow resistance causing oscillations between flow towards the condenser and the reactor vessel. The steam flow rates at the feed pump discharge check valves (N21-F015A/B) also went in the direction of the condenser. The flow rates going through the feed pump discharge check valve are not from the reactor vessel because, as illustrated by the safety related check valve flow rates (B21-F010A/B and B21-F032A/B), there is no steam flow leaving the reactor vessel towards the condenser.

From the flow plots, it is concluded that the safety related outboard check valves (B21-F032A/B) are the boundaries where steam flow rates are maintained going towards the reactor vessel. Flow rates outboard of the safety related outboard check valves oscillated between the reactor vessel and condenser direction.

During depressurization, significant liquid inventory is identified at the inlet of feedwater heaters 5A/B. This inventory provides a loop seal for preventing the release of reactor steam. The loop seal is fully established for the first four and a half minutes of the reactor vessel depressurization and after eleven and a half minutes into the depressurization. The liquid inventory between feedwater heaters

5A/B and 6A/B also provides water inventory for scrubbing any reactor vessel steam flows during the reactor depressurization.

Except for the first approximately twenty-five seconds, feedwater piping pressures remain greater than or equal to reactor vessel pressure throughout the reactor vessel depressurization. The higher pressure in the feedwater piping removes any possibility of a steady pressure driving force for flow between the reactor vessel and condenser.

Based on the RELAP5/MOD3.1 analysis performed and the results described above, it is concluded that there is no steam flow from the reactor vessel to the condenser via the feedwater pump minmum flow lines. The analysis results, combined with the assumed operator actions for feedwater leakage control system initiation and non-limiting dose assessments for feedwater line breaks, demonstrate that the appropriate testing medium for the inboard and outboard feedwater check valves (B21-F010A/B and B21-F032A/B) is water. The combination of the flow direction at the check valves and the cyclic nature of their positions through at least the first twenty minutes of the depressurization event indicates that leakage testing should be based on the water conditions present after initiation of the feedwater leakage control system at twenty minutes into the event.

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5. 10 CFR Part 50, Appendix J, 1994.

Table 1: RELAP5/MOD3.1 Major Component Description		
COMPONENT	**TYPE**	**DESCRIPTION**
	VALVES	
113 (213)	Check Valve	Plug Check Valve - B21-F010A(B) Inboard Containment Isolation Check valve
115 (215)	Inertia Check Valve	Swing Check Valve - B21-F032A(B) Outboard Containment Isolation Check valve
226	Inertia Check Valve	Swing Check Valve - B21-F803 RCIC system Check valve
129 (229)	Check Valve	Swing Check Valve - N21-F015A(B) Reactor Feed Pump Discharge check valve
130 (230)	Pipe 130 (230), Junction 7	Air Operated Controller/Valve - N21-F503A(B) Reactor Feed Pump Minimum Flow Control Valve
	PUMPS AND HEATERS	
147 (247)	Time-dependent Junction	Reactor Feedwater Pump A (B)
123 (223)	Pipe 123 (223) 6 vols	Feedwater Heater B005A (B)
121 (221)	Pipe 121 (221) 6 vols	Feedwater Heater B006A (B)
	REACTOR VESSEL	
1, 11, 21, 31, 41, 51 (2, 12, 22, 32, 42, 52)	Time-dependent Junctions	Feedwater sparger connections in the reactor vessel

Table 2: Null Transient Model Flow Rates			
Description	**Feedwater Loop A Flow (kg/sec)**	**Feedwater Loop B Flow (kg/sec)**	**Total Flow (kg/sec)**
Reactor Feedwater Pump Discharge	991.2	1007.7	1998.9
Minimum Flow line	-0.6648	-0.6621	-1.3269
Sum of Feedwater Pump and Min. Flow line flow rates	990.5	1007.0	1997.5
Flow through Startup Piping Header	-11.01	11.01	0.0
Flow past Startup Piping Header	979.5	1018.0	1997.5
Inboard Feedwater Check Valve (B21-F010) - Flow after loops join and split again	999.94	997.3	1997.26

Table 3: Null Transient Model Temperatures versus Design				
Item	**Description**	**Feedwater Loop A Flow (°K)**	**Feedwater Loop B Flow (°K)**	**Design Temperature (°K)**
1	Reactor Feedwater Pump Discharge	413.8	413.8	413.8
2	Entering Feedwater Heater B005	413.8	413.8	413.8
3	Entering Feedwater Heater B006	436.8	436.6	438.5
4	Leaving Feedwater Heater B006A	485.2	484.3	488.5
5	Entering Reactor Vessel	488	488	488.5

Table 4: Accident Chronology - Design Basis Recirculation Suction Line Break Accident	
EVENT	**TIME (Seconds)**
LOP/LOCA initiated Condensate and Condensate booster pumps tripped	0.0
Control Rods complete full core insertion	≅1.5
Reactor Feedwater Pumps flow stopped	5.0
Main Steam Isolation Valves fully close Condenser pressure at atmospheric pressure	11.5
Reactor vessel water level falls below the feedwater sparger elevation	≅20.81
Reactor Vessel pressure reaches feedwater saturation pressure of 2.12×10^6 Pa for 488 °K	≅25.0
Initiation of the High Pressure Core Spray	30
Reactor Vessel pressure reaches feedwater saturation pressure of 7.07×10^5 Pa for 438.5 °K (See Table 3, Item 3)	≅42.14
Reactor Vessel pressure reaches feedwater saturation pressure of 3.7×10^5 Pa for 413.8 °K (See Table 3, Item 1)	≅57.92
Core effectively reflooded (level above top of active fuel)	250
Initiation of FWLCS	1200
Initiation of Residual Heat Removal heat exchanger loops	1800

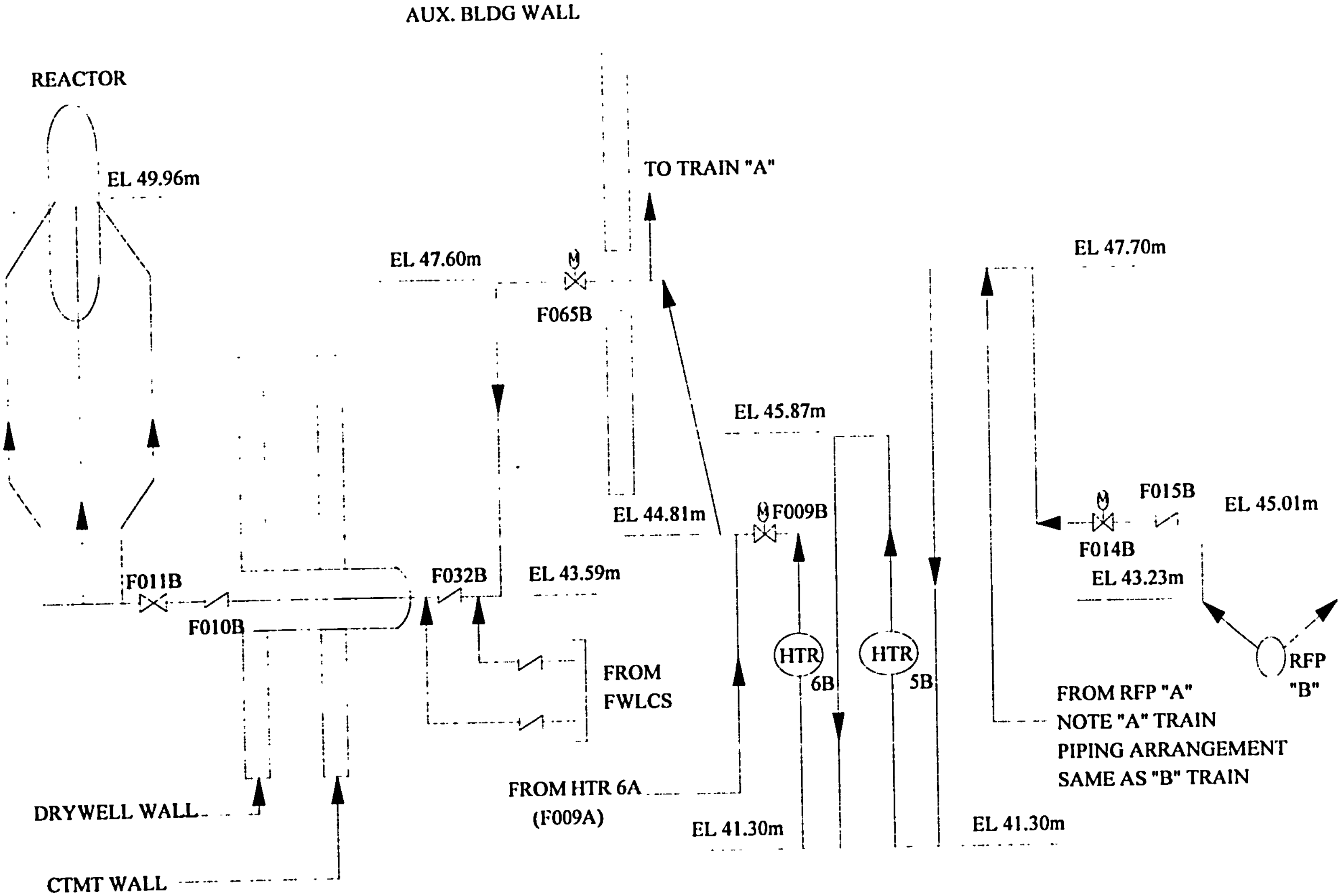


FIGURE 1: Feedwater Piping Layout

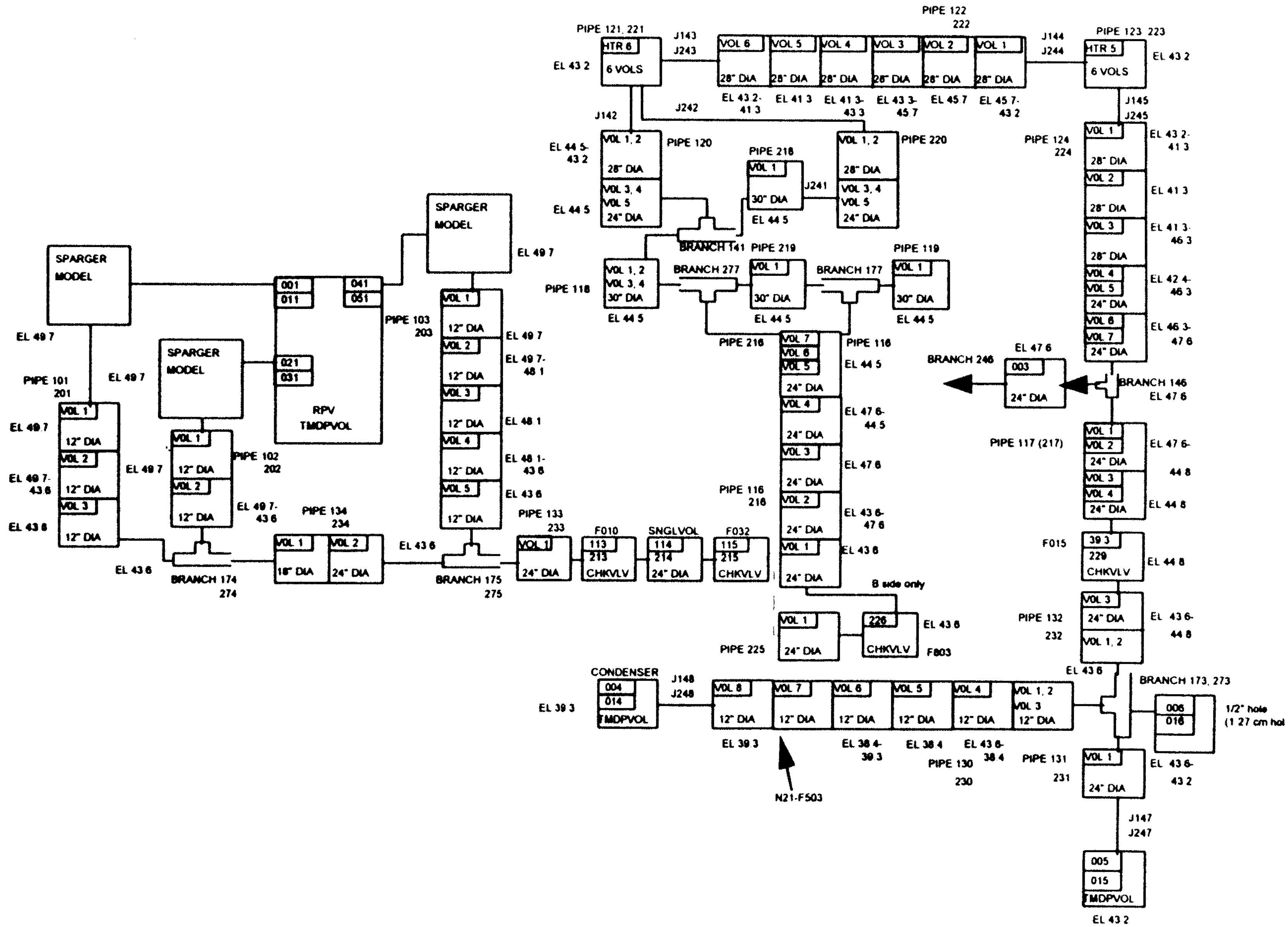
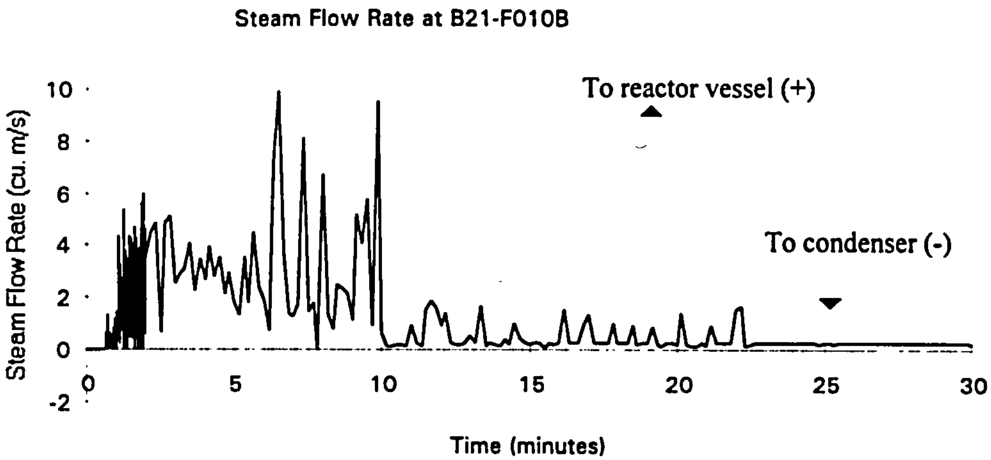
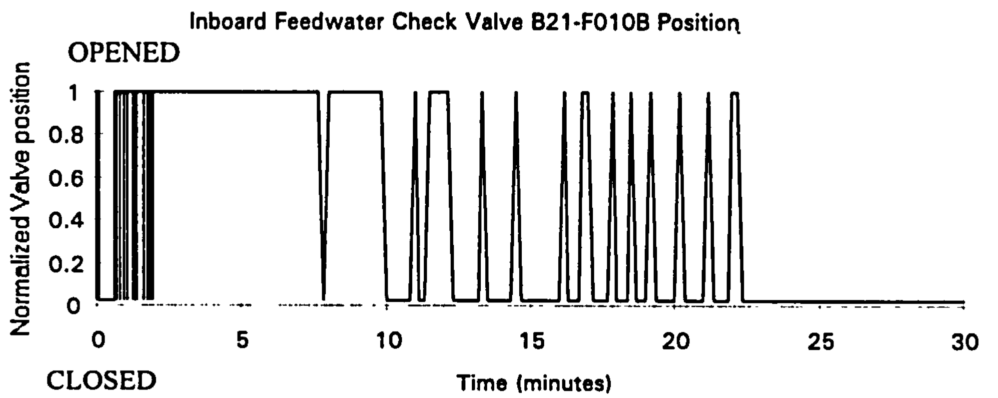


FIGURE 2: Feedwater System Nodalization Diagram

FIGURE 3: B21-F010B Flow Results





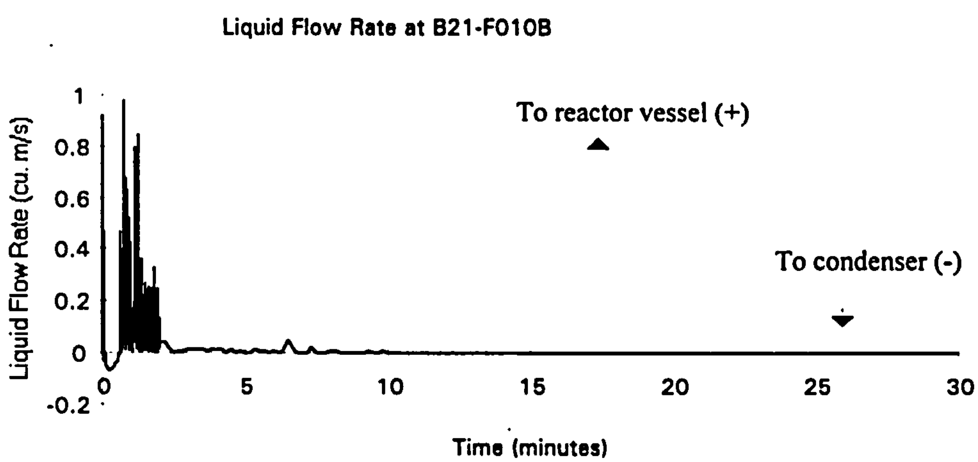


FIGURE 4: B21-F032B Flow Results

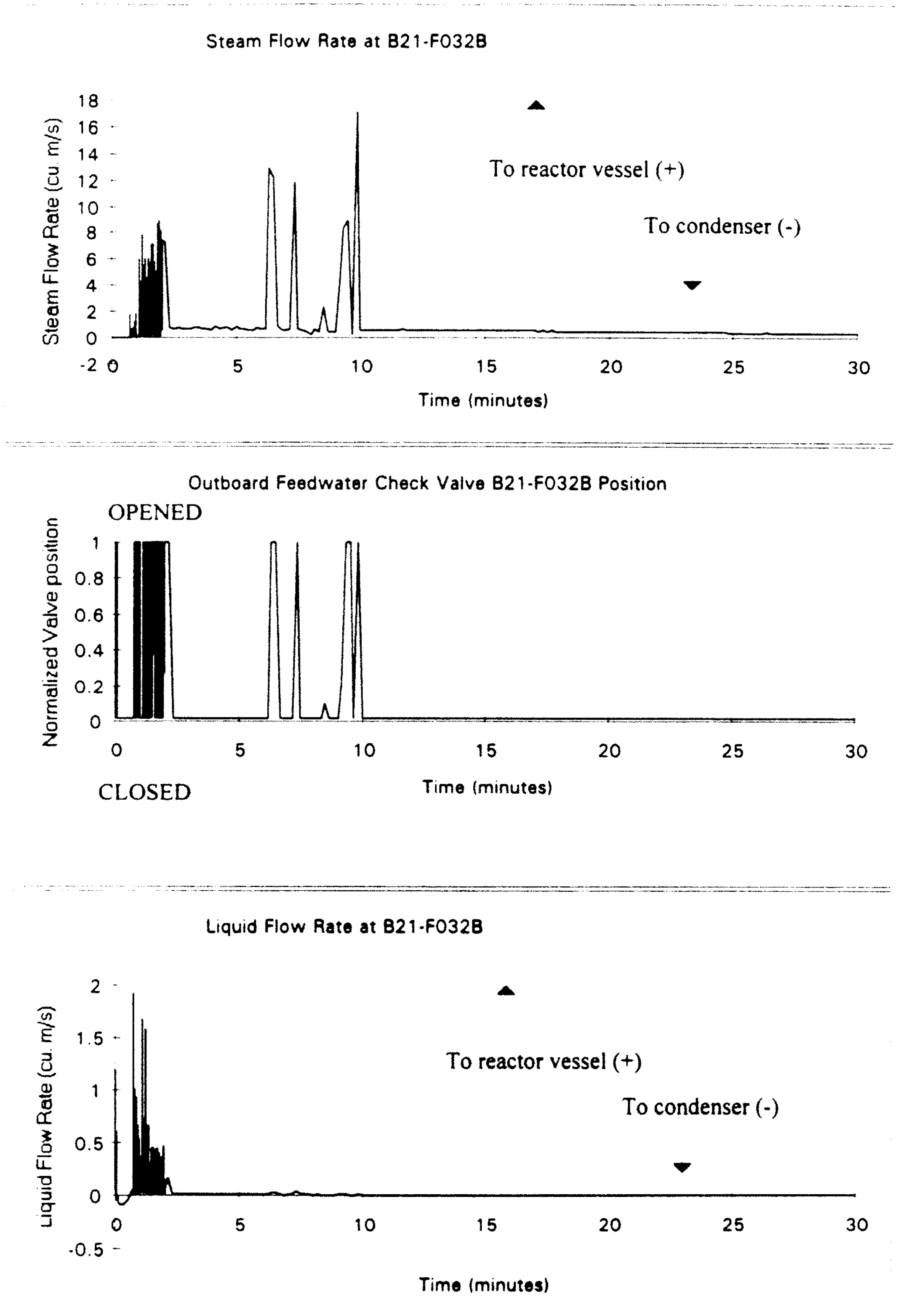
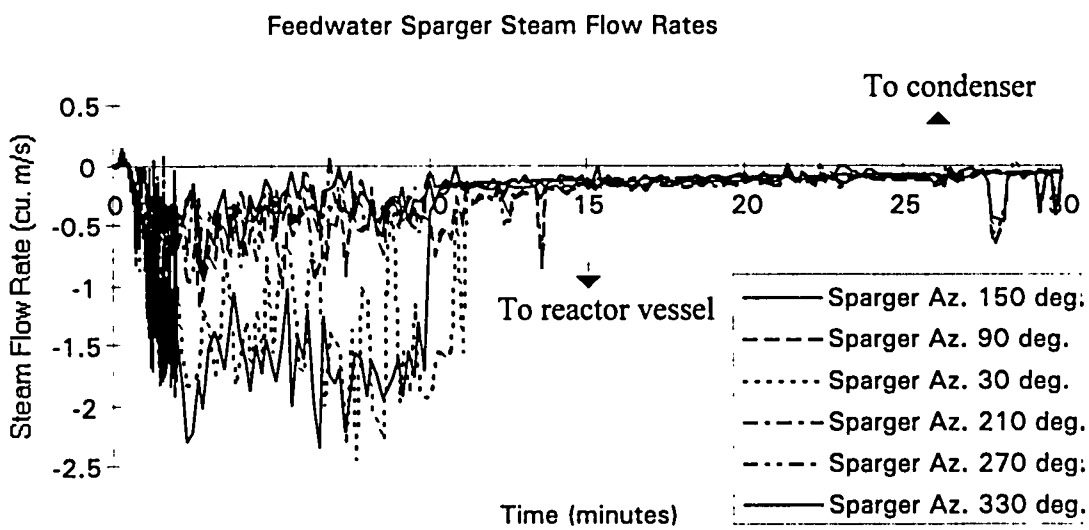


FIGURE 5: Feedwater Sparger Flow Results



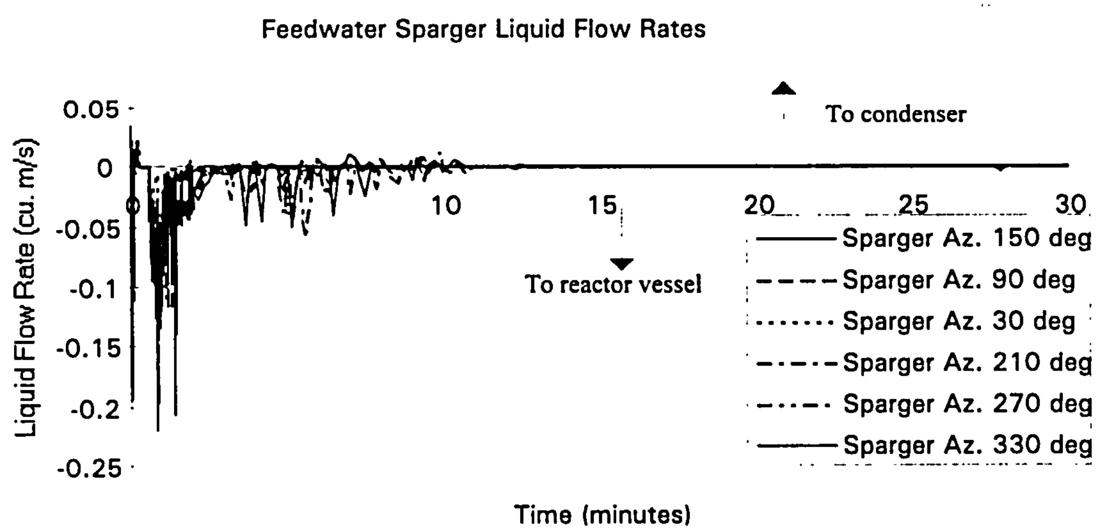
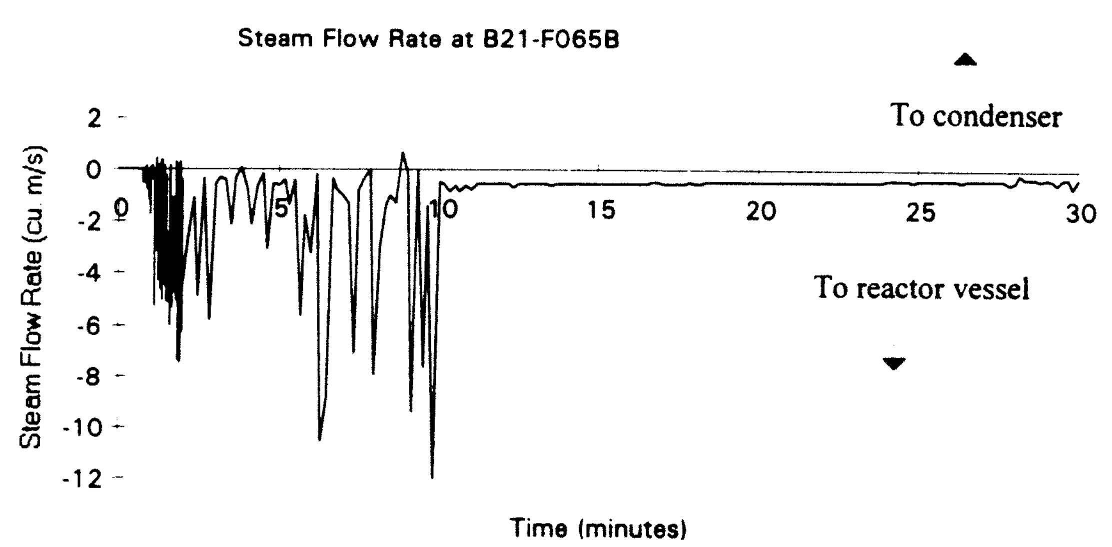


FIGURE 6: B21-F065B Flow Results



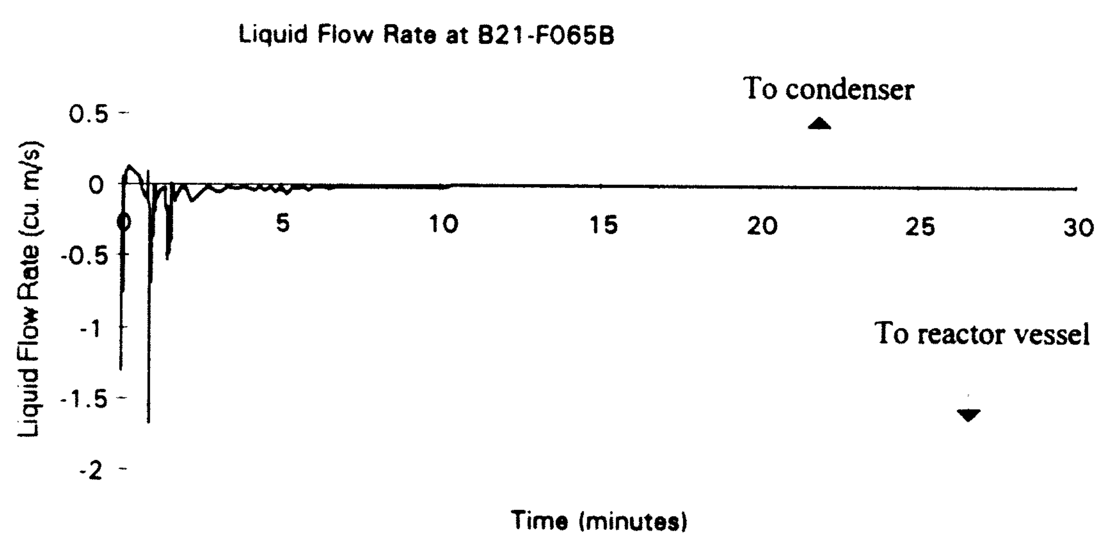
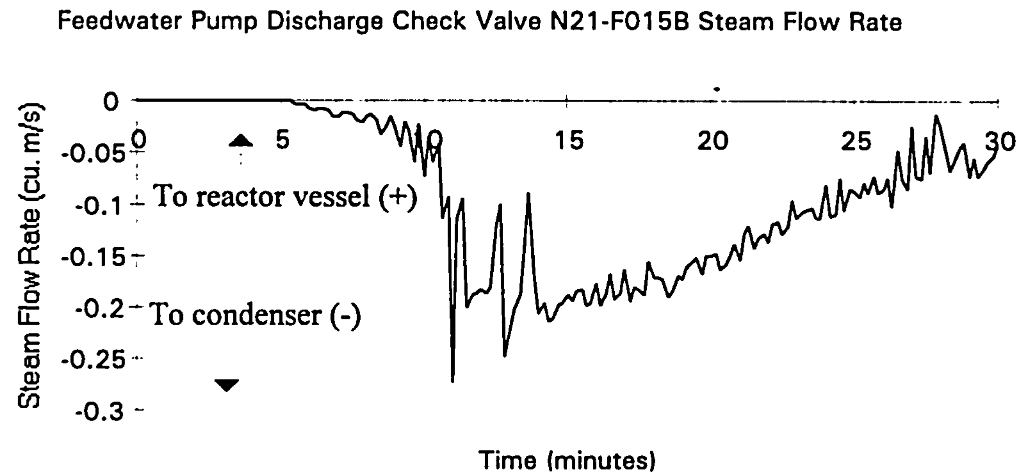


FIGURE 7: N21-F015B Flow Results



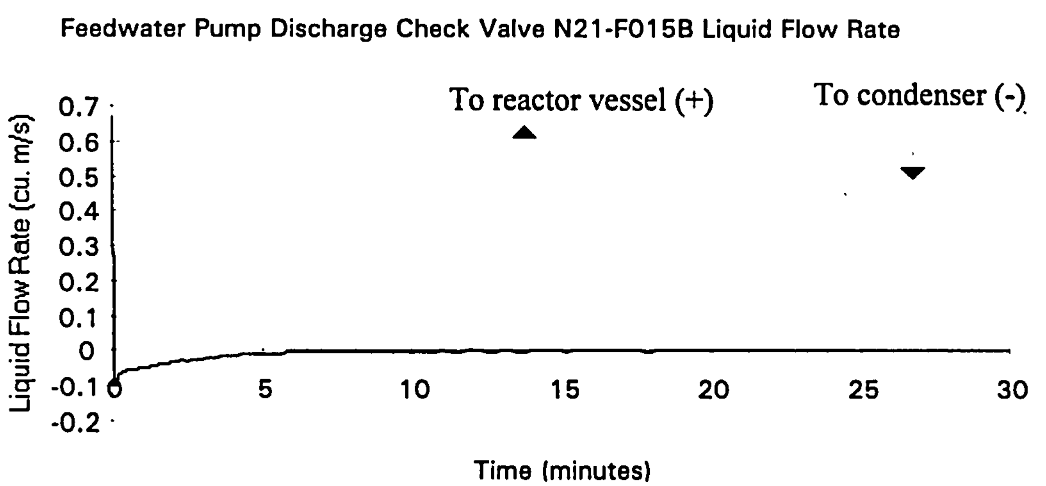
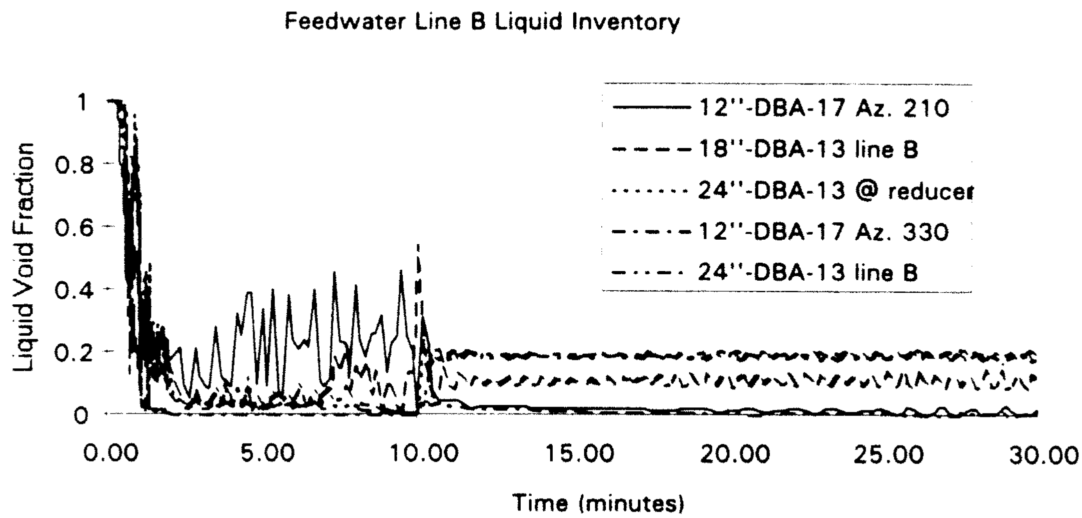


FIGURE 8: Liquid Inventory Results



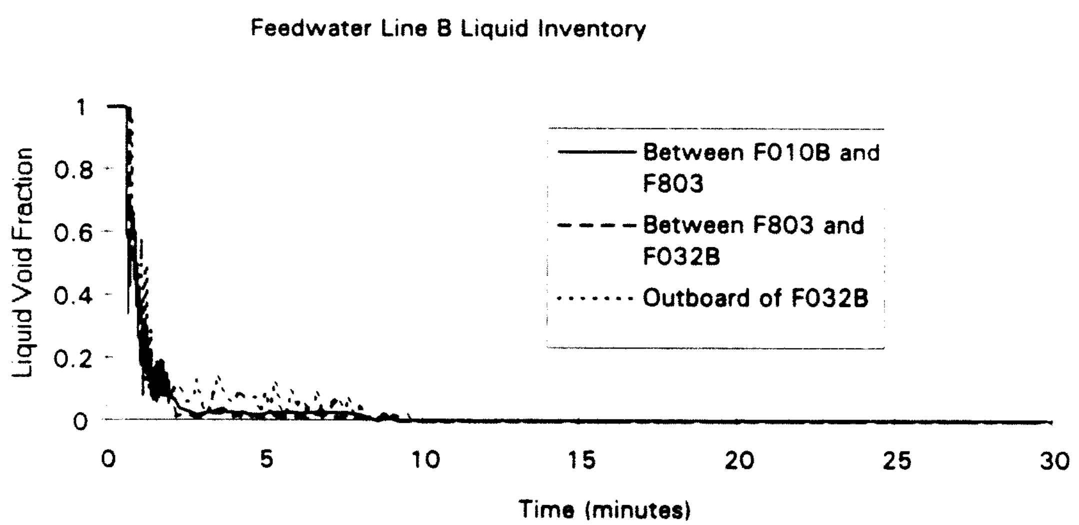
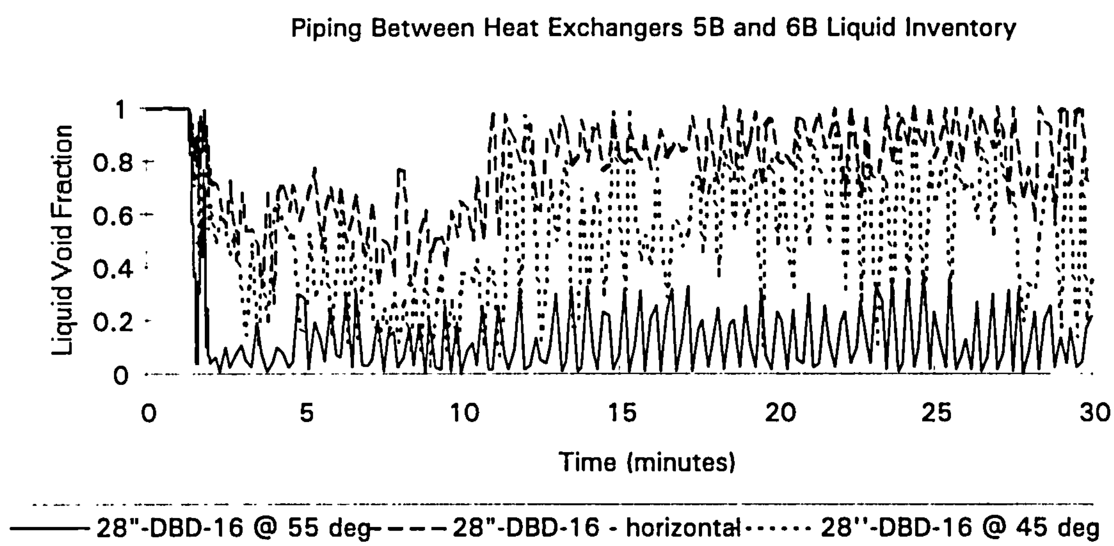


FIGURE 9: Liquid Inventory Results



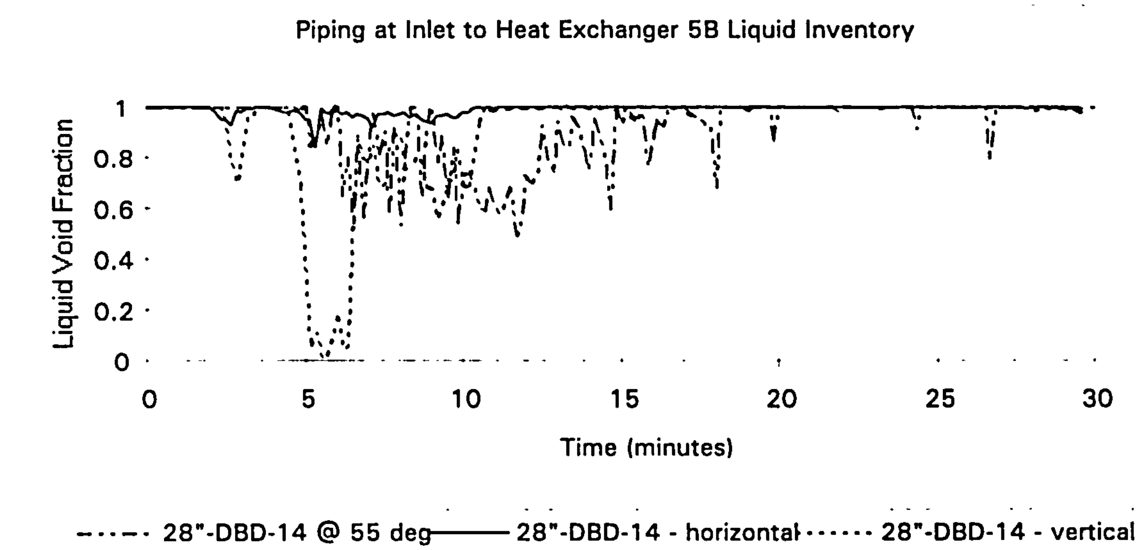
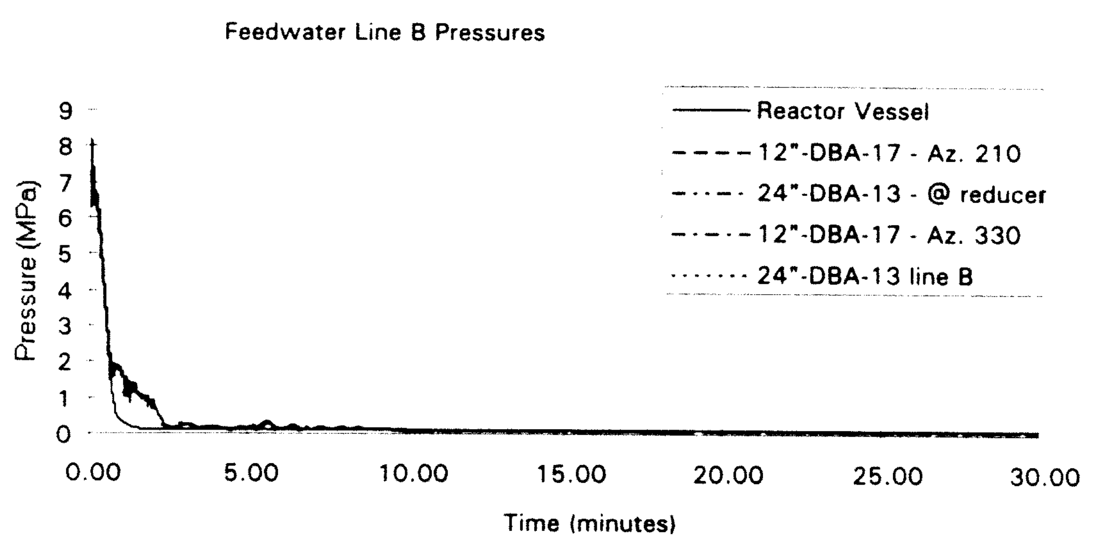


FIGURE 10: Feedwater Piping Pressure Results



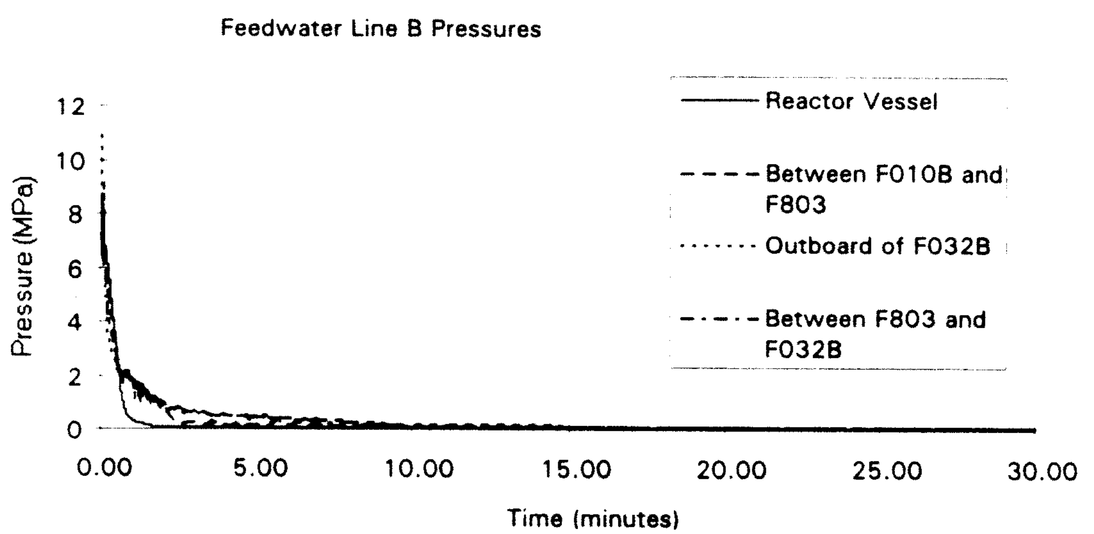
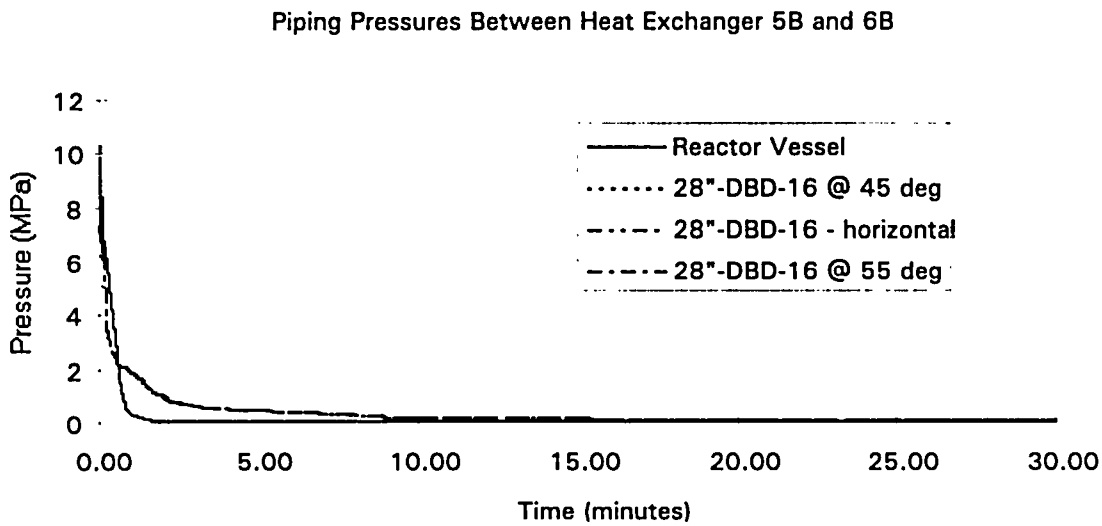
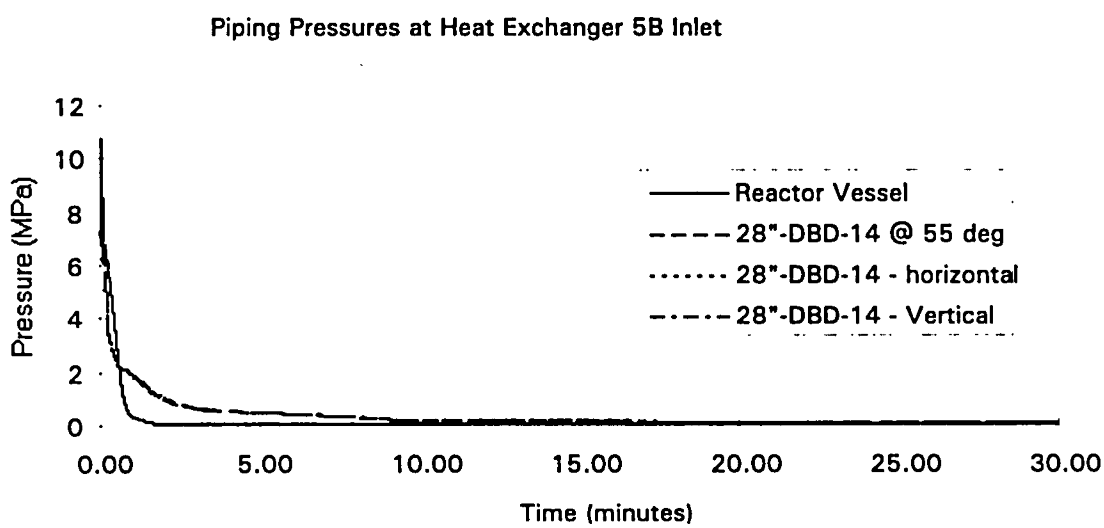


FIGURE 11: Feedwater Piping Pressure Results





Session 3A

Risk-Based Testing of Pumps and Valves

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Risk-Ranking IST Components Into Two Categories

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INTRODUCTION

The ASME has utilized several schemes for identifying the appropriate scope of components for inservice testing (IST). The initial scope was ASME Code Class 1/2/3, with all components treated equally. Later the ASME Operations and Maintenance (O&M) Committee decided to use safe shutdown and accident mitigation as the scoping criteria, but continued to treat all components equal inside that scope. Recently the ASME O&M Committee decided to recognize service condition of the component, hence the comprehensive pump test.

Although probabilistic risk assessments (PRAs) are incredibly complex plant models and computer hardware and software intensive, they are a tool that can be utilized by many plant engineering organizations to analyze plant system and component applications. In 1992 the ASME O&M Committee got interested in using the PRA as a tool to categorize its pumps and valves. In 1994 the ASME O&M Committee commissioned the ASME Center for Research and Technology Development (CRTD) to develop a process that adapted the PRA technology to IST. In late 1995 that process was presented to the ASME O&M Committee. The process had three distinct portions: (#1) risk-rank the IST components; (#2) develop a more effective testing strategy for More Safety Significant Components; and (#3) develop a more economic testing strategy for Less Safety Significant Components.

The risk-ranking process turned out to be a relatively straight forward process because of the availability of the plant probabilistic safety assessment (PSA). First the IST components are risk-ranked using the Fussell-Vesely (F-V) risk indicator to the core damage frequency (CDF) end state. Then a series of plant quantitative studies are performed to provide insight into the identification of the key plant IST components. Then a number of deterministic parameters are identified germane to the component. Finally a plant expert panel is convened to blend the quantitative and deterministic aspects to place the IST components into two categories: More/Less Safety Significant Component. The application of these two component categories to Process (#2) and Process (#3) is the subject of later papers in this Session.

Although some plant PSAs are more sophisticated than others (e.g., Level 1/2/3, internal/external events, shutdown cooling), this risk-ranking process is relatively independent of that degree of sophistication. Thus any nuclear plant can apply this risk-ranking process to their IST program to both improve safety and achieve substantial O&M cost savings. Note that the opinions expressed in this paper are those of my own and not the opinions of the ASME O&M Committee.

BACKGROUND

Currently, the ASME O&M Code provides the requirements for inservice testing for pumps, valves, and snubbers. Historically the

Code has treated all ASME Code Class 1/2/3 components as equal, including approximately 30 pumps, 500 valves, and several hundred snubbers in a typical plant. Some attempt was made about 15 years ago to adjust the scope of inservice testing components to safe shutdown and mitigating the consequences of an accident. In recent years, the service condition and the performance of the components have been partially factored into the requirements.

PRA AS A TOOL FOR RISK RANKING

Although the WASH-1400 Report (NRC 1975) provided great risk insights into the design and operation of nuclear plants, it was the requirements of NRC Generic Letter 88-20 (NRC 1988) that provided the useful tool to estimate the risk of a specific plant. As plant owners started applying the PRA tools that had been developed, a great many insights began to appear. The components historically addressed with inservice testing programs were found to span many orders of magnitude in risk. Clearly all these components were not equal in importance.

Typically 90% of the IST pumps and about 50% of the IST valves are modeled in the PSA. Snubbers are not typically modeled in the PRA so discussion of that IST component will be deferred. An ASME/EPRI Risk-Based IST Pilot Study (EPRI 1995) showed the following general conclusions based on the PSAs of ten plants:

Component	Risk Significant	Less-Risk Significant
IST Pumps	50%	50%
IST Valves	10%	90%

This study evaluated all IST pumps and valves and used a Fussell-Vesely threshold of .001 for determination of risk significance. This indicates that on an average 15 IST pumps and 50 IST valves are risk significant. This initial research work showed distinct value in dividing the IST components into two or more risk categories.

PROCESS FOR COMPONENT IMPORTANCE RANKING

If the ASME were to divide its scope of IST pumps and valves into two or more categories, then a component importance ranking process would have to be devised that could apply to all nuclear plants. This would have to be applicable to all four nuclear steam supply systems (NSSS), all ten architect-engineering designs, all plant vintages, and all PSA modeling techniques. The ASME O&M Committee asked the ASME Center for Research & Technological Development to develop this universal process based on the application of risk technology. The ASME CRTD created a Risk-Based (RB) IST Research Task Force (ten members), a RB-IST Steering Committee (thirteen members), and an independent peer review committee (three members) which, over a three year period, produced the recommended process (ASME 1995).

The first step in the process consists of PSA risk ranking, which divides the IST components into three groups: risk significant

(F-V/CDF > .001); less risk significant (F-V/CDF < .001); and not modeled components. The second step in the process consists of additional PSA sensitivity and mapping studies, such as identification of components with risk achievement worth (RAW)/CDF > 2, F-V/large early release frequency (LERF) > .001, impact of external events, and risk significant pumps and valves that are not in the IST program. The third step in the process consists of identification of a number of deterministic parameters, including effect of component on shutdown cooling, and importance of component in the execution of Emergency Operating Procedures (EOP). Finally the fourth step in the process convenes a plant expert panel to blend the quantitative and deterministic data to categorize the pump or valve as a More Safety Significant Component or Less Safety Significant Component.

PSA ISSUES AFFECTING IST

A number of issues surfaced during the development and refining of this process by the ASME research task force. Some of those issues were scope of the PSA, level of PSA detail, "quality" of the PSA, truncation limits, modeling of initiating events, super-components, common cause failures, multiple component considerations, importance measures and thresholds, and expert panel guidance.

It was recognized that not only is the design configuration of every plant different, but that the PSA modeling techniques were also different. Even for a plant with a late model design and "Cadillac" PSA (including external events, Level 1/2/3, and shutdown) like Seabrook, still only half of the IST components were modeled. Although ideally the plant safety analysis, the EOPs, and the PSA all use identical assumptions, this is not the case in most plants for a variety of good reasons. Thus, a pure risk-based approach cannot be utilized, but a blend of quantitative risk factors and deterministic factors must be used. The plant expert panel becomes the tool to effect this blending.

ASME INITIAL APPROACH TO USING RISK FOR IST - CODE CASE

The ASME CRTD Vol 40-2 Report (ASME 1995) is a comprehensive report that not only describes the component ranking process, but it also describes the process for developing a testing strategy for more safety significant components (MSSCs) and less safety significant components (LSSCs). On the other hand, it is not written in the words of a requirements document. Thus, the next stage is for the ASME O&M Committee to develop the requirements document. Initially that will be an ASME OMN-X Code Case; later that will be a revision to the ASME O&M Code. ASME OMN Code Cases can be developed to provide alternate rules for testing not covered by existing code rules and to gain experience through usage over a period of time.

The ASME O&M Committee has commenced the development of an OMN-X Code Case on Component Importance Ranking. The initial draft of the Code Case, which is subject to change before it is issued in final, is shown inside the table boxes with initial rationale in the text below.

Proposed OMN-X Code Case

Alternative Rules for Scoping Components
for Inservice Testing of LWR Power Plants
ASME OM Code - 1995, Subsection ISTA

Inquiry: What alternative requirements to those of OM Code, Subsection ISTA, may be used for scoping pumps and valves into the inservice testing program requirements in light water reactor power plants?

Reply: It is the opinion of the Committee that, in lieu of the requirements that state that the pumps and valves covered are those that are required to perform a specific function in shutting down a reactor to the safe shutdown condition, in maintaining the safe shutdown condition, or in mitigating the consequences of an accident, the following alternative requirements may be applied.

1. INTRODUCTION

1.1 Scope

This Code Case establishes the requirements that identify two populations of pump and valve components that are applicable to the inservice testing program requirements.

The applicable pumps and valves are those required to perform a specific function in shutting down a reactor to the safe shutdown condition, in maintaining the safe shutdown condition, or in mitigating the consequences of an accident.

This Code Case establishes the methodology and process for dividing the population of pumps and valves into "more safety significant component" and "less safety significant component" categories.

SCOPE OF ASME IST CODE

Note that the applicable pumps and valves remain those defined by the ASME OM Code in both the 1990 and 1995 editions. This is also the scope of the pumps and valves for ASME OM Part 6 (Pumps) and Part 10 (Valves) in ASME/ANSI OM-1987, which is referenced by the ASME B&PV Code, Section XI, Subsection IWP (for pumps) and Subsection IWV (for valves). Since Section XI uses ASME Code Class 1, 2, and 3 as its scoping boundary, any plant that uses Section XI as its IST Code of Record is linked directly to ASME Code Class 1/2/3.

EXCLUSIONS TO ASME IST CODE

Note these are the exclusions currently in the ASME OM Code and in ASME/ANSI OM Part 6 & Part 10. The ASME O&M Committee is currently considering the removal of the pump exclusion on drivers. Since pump failures to start and failures to run may be caused by issues in the pump, the driver, or the control circuit, it would seem appropriate to address all the potential failure causes in an inservice testing program vice just the mechanical issues.

1.2 Exclusions

Pumps in the following applications are excluded from this Code Case, if they are not specifically required to perform a function described in para 1.1 above:

- o drivers, except where the pump and driver form an integral unit and the pump bearings are in the drivers; and
- o pumps that are supplied with emergency power solely for operating convenience.

Valves in the following applications are excluded from this Code Case, if they are not specifically required to perform a function described in para 1.1 above:

- o valves used only for operating convenience such as vent, drain, instrument, and test valves;
- o valves used only for system control, such as pressure regulating valves; and
- o valves used only for system and component maintenance.

DEFINITIONS GERMANE TO RISK-BASED INSERVICE TESTING

There are a number of definitions that are being introduced to the IST world that historically have not been associated with inservice testing. Many of those definitions are associated with the application of the PSA. In addition the following definitions are being specifically addressed in the development of inservice testing strategies:

- functional readiness - the ability of the component to function on demand (i.e., the component has not already failed).
- operational readiness - it is the ability of a component to perform its intended system function when required (i.e., component is operable, electrical power is available to the motor, system valves are appropriately aligned).
- operable - component will operate at designed conditions (e.g., valve will stroke, pump will deliver rated flow) when called upon to do so at a future point in time.

2. SUPPLEMENTAL DEFINITIONS	
Availability	Basic Event
Common Cause (CC) Failure	Core Damage
Core Damage Frequency (CDF)	Cut Set
Decision Criteria	Failure Modes, Cause Analysis
Failure Rate	Functional Readiness
Fussell-Vesely Importance	Importance Measure
Independent Plant Examination	Inservice Test
Inservice Testing Strategy	Initiating Event
Large, Early Release	Large Early Release Frequency (LERF)
Less Safety Significant Comp	Mission Time
More Safety Significant Comp	Operational Readiness
Operable	Performance
Probabilistic Safety Assessment	Probability
Reliability	Risk
Risk Achievement Worth (RAW)	Risk Reduction Worth (RRW)
Truncation Limits	

REQUIREMENTS FOR COMPONENT IMPORTANCE RANKING

Although this Code Case only addresses "component importance ranking", clearly what one intends to do with the ranked component affects to some degree the ranking process. For the MSSC, the testing strategy is intended to look for precursors to component failure and determine functional readiness. For the LSSC, the testing strategy is intended to determine functional readiness (e.g., find components that have already failed within a reasonable period of time).

Clearly, if each plant did not have a PSA available, then this quantitative approach to component importance ranking would be difficult. As mentioned earlier, since the plant PSA is not perfect with respect to the safety analysis, the EOPS, and all plant modes, the plant expert panel is the key organizational function to ensure that appropriate data and information are considered. There are two key words in the previous sentence that greatly affect the functional product of the plant expert panel: "appropriate" and "considered".

The members of the plant expert panel are the decision makers for designation of components as MSSC or LSSC. The current systems configuration must be the basis for the safety analysis report (USAR), the EOPs, and the PSA. Thus, the PSA must be a living PSA, which is maintained up-to-date by some periodic mechanism.

Both the ASME research task force (in their ASME CRTD Vol 40-2 Report) and the O&M Committee (via the Special Working Group on Component Importance Ranking) recognize that there are many specific requirements to the implementation of this Code Case. There is a tradeoff between making the specific requirements of the Code Case prescriptive and numerous to create consistency from plant to plant in this RB-IST application and providing flexibility to the Owner to effectively tap the unique capability of its PSA, its plant design, and the talent of its plant staff.

3. GENERAL REQUIREMENTS

3.1 Plant Specific PSA

The plant specific PSA shall be available to perform component risk ranking to identify risk significant and risk insignificant components.

3.2 Plant Expert Panel

A plant expert panel shall be designated to perform the blended evaluation of quantitative and deterministic engineering data for each component.

3.3 Determination of MSSC and LSSC

The plant expert panel shall evaluate each component, utilizing PSA quantitative information (if component is modeled) and engineering deterministic information, and designate it either MSSC or LSSC.

3.4 Living PSA

The plant specific PSA shall be maintained up to date such that plant modifications affecting the model are reflected in the model.

4. SPECIFIC REQUIREMENTS

4.1 Component Risk Determination

Develop basis for component risk determination by using the plant PSA as follows:

(a) Certify the plant PSA to be technically consistent per Appendix B of the EPRI PSA Applications Guide (TR-105396, August 1995).

(b) Risk rank the pump and valve components defined by para 1.1 above into three CDF figure of merit categories using F-V risk importance measure as follows:

(1) Risk significant: F-V > .001
(2) Less risk insignificant : F-V < .001
(3) Not modeled

Although there are obviously many modeling differences between the PSAs at the various nuclear plants, it is less clear which differences (if any) affect this risk-ranking application for IST components. Some nuclear benchmark standard for PSA "quality" is needed, thus the EPRI PSA Applications Guide was selected. Should a better benchmark standard be developed, then clearly this reference should be reconsidered.

One of the most attractive features of a plant specific PSA is the calculation of risk-ranking parameters based on actual system configuration and best estimate of component reliability. Thus, the F-V importance measure for the CDF end state was considered the best surrogate measure of risk for operability of pumps and valves important for safe shutdown and mitigating the consequences of an accident. The threshold of F-V = .001 was

used to draw the line at a conservative level (0.1%).

4.2 Component Risk Mapping & Sensitivity Studies

Perform several PSA mapping and sensitivity studies for the modeled pump and valve components as follows:

(a) Risk rank the above PSA modeled components into three CDF figure of merit categories using RAW risk importance measure for out of service sensitivity categories as follows:

(1) High out-of-service sensitivity: RAW > 10
(2) Medium out-of-service sensitivity: RAW > 2
(3) Low out-of-service sensitivity: RAW < 2

(b) Risk rank the above PRA modeled components into LERF figure of merit categories using F-V risk importance measure for containment bypass sensitivity categories as follows:

(1) High containment-bypass sensitivity: F-V > .001
(2) Low containment-bypass sensitivity: F-V < .001

(c) Risk rank the above PRA modeled components into two CDF figure of merit categories using F-V risk importance measure for external versus internal events sensitivity categories as follows:

(1) High external-event sensitivity: F-V > .001
(2) Low external-event sensitivity: F-V < .001

(d) Risk rank all the PRA modeled pumps and valves in plant into two CDF figure of merit categories using F-V risk importance measure for non-IST pumps and valves as follows:

(1) High non-IST sensitivity: F-V > .001
(2) Low non-IST sensitivity: F-V < .001

During the deliberations of the ASME research task force during 1993 to 1995, a question continually reemerged: "should other importance indicators be utilized, such as RAW?". Assuming that a rather reliable component always fails certainly skews the risk ranking results. Assuming that operator actions always fail (the human factors aspect), also skews the risk ranking results. After many discussions over a period of two years, the research task force decided that the RAW was not the primary risk ranking importance measure, but that it certainly could provide additional quantitative insights to the plant expert panel for the component importance ranking process. In fact, the "QUAD Chart" was created as an analysis tool for the expert panel. The "QUAD Chart" has RAW on the ordinate and F-V on the abscissa, with thresholds of F-V = .001 and RAW = 2 to provide the four quadrants.

From a theoretical perspective there are dozens of sensitivity and mapping studies that can be done for pumps and valves on a plant PSA. These studies can be performed far easier on some PSAs than others. One of the objectives of the RB-IST Pilot Project was to identify the sensitivity and mapping studies that provide meaningful information for the typical plant. Based on extensive studies performed on the Comanche Peak Station and the impact of the more significant studies on the other nine pilot plants, the above four PSA studies were deemed to be useful for the plant expert panel to consider. Especially of interest was the PSA identification of important pumps and valves that were not in the ASME IST program. Another interesting aspect is the defense-in-depth issue. The original designers of the plant applied many deterministic defense-in-depth criteria, which led to much redundancy in many areas. The RAW is perhaps a very good risk indicator of the level of defense-in-depth.

4.3 Membership on the Plant Expert Panel

(a) Minimum size requirements:

(1) There shall be at least five engineering experts designated as members of the plant expert panel.

(b) Minimum expertise requirements:

(1) The following plant expertise disciplines shall be represented on the plant expert panel:

- o probability risk assessment engineering
- o plant operations (licensed operator)
- o safety analysis engineering

(2) Additional members of the plant expert panel should be selected from the following plant expertise disciplines:

- o system engineering
- o maintenance engineering
- o inservice test engineering
- o component engineering
- o licensing engineering
- o operations engineering

(3) Other plant or nuclear industry experts may be invited to attend some or all of the sessions of the plant expert panel to provide observations, opinions, or recommendations.

(c) Minimum experience requirements:

(1) A minimum of 50 collective man-years of nuclear industry experience must be represented by members of the plant expert panel.

(2) A minimum of 30 collective man-years of specific plant experience must be represented by the probability risk assessment engineering, plant operations (licensed operator), and safety analysis engineering members of the plant expert panel.

Since the plant expert panel has the final decision of categorizing a component as MSSC or LSSC, the panel clearly needs to meet some minimum standard. Based on plant expert panels convened by the RB-IST Pilot Project, certain dynamics were noted. The vast majority of indepth discussions took place between the PSA expert, EOP expert,

and SAR expert. Thus these "experts" are the most important personnel to have on the plant expert panel. Since these "experts" are not necessarily currently assigned in these discipline areas, some mechanism needs to be identified to provide incentives for plant management to allocate their scarce and valuable time to the plant expert panel.

4.4 Management of the Plant Expert Panel

(1) A procedure approved by the Owner's Plant Operations Review Committee must describe the process, including designated members, designated chairman, quorum, attendance records, agenda, motions for approval, percentage of approval required for passage, written accommodation for dissenting opinions, and minutes of meetings.

(2) The plant expert panel elicitation process may be conducted using an interactive group process, the delphi process, or a combination of these processes.

The basis for the decisions of the plant expert panel must be available for audit and for after-the-fact analysis by others. In addition the decision process of the plant expert panel must be repeatable.

There are many ways for the plant expert panel to blend the appropriate quantitative and deterministic data. This Code Case does not desire to tell the plant expert panel how to perform the blending, only what factors are likely important in the blending process. Some plant expert panels will use multi-page work sheets for each component, while others will use multiple data base information sources. The important result is to have a repeatable basis for the decision.

4.5 Component Evaluation by the Plant Expert Panel

(a) In addition to the PSA quantitative data, the following deterministic data shall be considered by the Plant Expert Panel:

(1) design basis information
(2) common cause failures
(3) regulatory commitments
(4) utilization in EOPs
(5) performance reliability
(6) shutdown contribution
(7) containment integrity
(8) effects of external events
(9) effect of component failure on system operability

(b) See Appendix A for example list of component deterministic questions that could be considered by the plant expert panel to determine the component's safety classification.

(c) The risk categorization for each component shall be determined by a consensus of the plant expert panel. If a consensus cannot be obtained, then the component shall be categorized as MSSC.

Some records are necessary to document the component ranking process for future reevaluations based on new information. In addition since the deliberations of the plant expert panel will likely be of interest to the plant regulator, an auditable trail of the decision making process must be available.

5. RESERVED
6. RESERVED
7. RESERVED
8. RESERVED

9. RECORDS AND REPORTS

The Owner shall maintain the following records:

9.1 Plant Expert Panel Records

(a) membership and attendance
(b) member expertise representation
(c) member experience (years of experience in each of the expertise categories)
(d) meeting agendas
(e) meeting minutes
(f) chartering plant procedure

9.2 Component Records

(a) risk significance based on F-V
(b) additional PSA quantitative information
(c) deterministic information
(d) plant expert panel decision on MSSC or LSSC
(e) basis for the MSSC/LSSC decision

Undoubtedly improvements in this component importance ranking process will occur in the future, possible through improvements in the PSA models and very likely through the iterative interaction of PSA experts, SAR experts, and EOP experts.

This ASME OMN-X Code Case to categorize IST pumps and valves into MSSC and LSSC will likely be approved in late 1996. The regulatory endorsement via a regulatory guide or a generic letter will likely occur in 1997. Note that as soon as the ASME approves and publishes the Code Case, it could be referenced in an Owner's request for an alternative to the ASME Code requirements.

CONCLUSION

The use of risk-ranking methods should prove useful in the near future. Once the NRC has approved one or more pilot programs and issued guidance on the approach, plants may want to develop and implement risk-based IST programs.

REFERENCES

ASME 1995: Draft ASME Research Report, "Risk-Based Inservice Testing - Development of Guidelines," dated November 1995.

EPRI 1995: "Risk Based In-Service Testing Pilot Project," Electric Power Research Institute, Document Number EPRI-TR-105869, December 1995.

NRC 1988: Generic Letter 88-20, "Individual Plant Examination for Severe Accident Vulnerabiilties - 10 CFR §50.54(f)," issued November 23, 1988.

Risk-Based Inservice Testing Program Modifications at Palo Verde Nuclear Generating Station

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ABSTRACT

Arizona Public Service Company (APS) is investigating changes to the Palo Verde Inservice Testing (IST) Program that are intended to result in the reduction of the required test frequency for various valves in the American Society of Mechanical Engineers (ASME) Section XI IST program. The analytical techniques employed to select candidate valves and to demonstrate that these frequency reductions are acceptable are risk based. The results of the Palo Verde probabilistic risk assessment (PRA), updated in June 1994, and the risk significant determination performed as part of the implementation efforts for 10 CFR 50.65 (the maintenance rule) were used to select candidate valves for extended test intervals. Additional component level evaluations were conducted by an "expert panel."

The decision to pursue these changes was facilitated by the ASME Risk-Based Inservice Testing Research Task Force for which Palo Verde is participating as a pilot plant. The NRC's increasing acceptance of cost beneficial licensing actions and risk-based submittals also provided incentive to seek these changes.

Arizona Public Service is pursuing the risk-based IST program modification in order to reduce the unnecessary regulatory burden of the IST program through qualitative and quantitative analysis consistent with maintaining a high level of plant safety. The objectives of this project at Palo Verde are as follows:

1. Apply risk-based technologies to IST components to determine their risk significance (i.e., high or low).

2. Apply a combination of deterministic and risk-based methods to determine appropriate testing requirements for IST components including improvement of testing methods and frequency intervals for high-risk significant components.

3. Apply risk-based technologies to high-risk significant components identified by the "expert panel" and outside of the IST program to determine whether additional testing requirements are appropriate.

4. Submit code relief request(s) to the NRC using the Palo Verde pilot project results.

5. Ensure the results and insights from our project are available for use by the industry in a timely manner.

As a result of this pilot program, Palo Verde will be submitting a code relief request to the NRC to change the test frequency of low-risk significant valves that have exhibited good performance to a test interval of every other fuel cycle. This paper will explain the processes used to (1) determine relative risk significance (including the role of the "expert panel"), (2) select candidate valves for test interval extension, and (3) evaluate the impact of the aggregate changes on public risk. It will also provide the results of the pilot effort including as estimate of the cost to implement a risk-based IST program and the cost savings that will be anticipated.

APPLICATION OF RISK-BASED METHODS TO INSERVICE TESTING OF CHECK VALVES

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ABSTRACT

Research efforts have been underway in the American Society of Mechanical Engineers (ASME) and industry to define appropriate methods for the application of risk-based technology in the development of inservice testing (IST) programs for pumps and valves in nuclear steam supply systems. This paper discusses a pilot application of these methods to the inservice testing of check valves in the emergency core cooling system of Georgia Power's Vogtle nuclear power station.

This demonstration study, which has been sponsored by the Westinghouse Owners Group, applies probabilistic safety assessment (PSA) models that have already been developed to meet regulatory requirements for an individual plant examination. The results of the PSA are used to divide the check valves into risk-significant and less-risk-significant groups. This information is reviewed by a plant expert panel along with the consideration of appropriate deterministic insights to finally categorize the check valves into more safety-significant and less safety-significant component groups. The relevant failure modes for the components are also identified.

All of the more safety-significant check valves are further evaluated in detail using a failure modes and causes analysis (FMCA) to assist in defining effective IST strategies. A template has been designed to evaluate how effective current and emerging tests for check valves are in detecting failures or in finding significant conditions that are precursors to failure for the likely failure causes. This information is then used to design and evaluate appropriate IST strategies that consider both the test method and frequency. A few of the less safety-significant check valves are also evaluated using this process since differences exist in check valve design, function, and operating conditions. Appropriate test strategies are selected for each check valve that has been evaluated based on safety and cost considerations. Test strategies are inferred from this information for the other check valves based on similar check valve conditions. Sensitivity studies are performed using the PSA model to arrive at an overall IST program that maintains or enhances safety at the lowest achievable cost.

This paper will discuss the above process and results to show that application of risk-based methods in the development of IST programs can potentially result in significant savings while maintaining a high level of safety.

INTRODUCTION

Inservice testing programs are intended to identify potential malfunctions of equipment before they could lead to unanticipated incidents or accidents. All aspects of inservice testing, including where, when and how to test, affect the benefits of the test in enhancing equipment and plant reliability. Inservice tests are currently based on mandated requirements, such as those for nuclear power plant components in the ASME Operations & Maintenance Code (ASME 1995), technical specification requirements, and U.S. Nuclear Regulatory Commission (NRC) regulations. Most inservice test requirements are based on past experience and engineering judgment and have only an implicit consideration of risk-based information, such as component failure rates and consequence impacts for the specific operating conditions, equipment functions, and environment.

While similar processes have been used in previous programs such as reliability-centered maintenance, this check valve study is a first application to testing required by the ASME Code. The significant feature of the work reported here is the use of risk-based information with performance evaluations such that the effect on safety from changes to IST programs can be determined.

Risk-based processes are used to evaluate the use of technology and resources to improve the effectiveness of testing components, to enhance testing strategies in some areas and reduce testing in many others, to evaluate improvements to plant availability and enhanced safety measures, and to reduce overall operation and maintenance (O&M) costs while maintaining a high level of safety.

RISK-BASED IST PROCESS

The actual process of evaluating testing strategies for components is identified in the following four steps that were developed by the ASME Research Task Force on Risk-Based IST (ASME 1996).

1. **Risk Ranking** - determination of risk significance based on PSA[1] results (core damage frequency) using risk importance measures. Two risk groups are defined (more risk-significant/less risk-significant). An evaluation of the results of PSA and an evaluation of other considerations (shutdown risk, operating history, maintenance history, and other deterministic insights) through the use of an expert panel is performed. Two groups are defined (more safety-significant/less safety-significant).
2. **Component Review & Failure Modes and Causes Analysis** - identification of key characteristics which could influence the determination of effective testing methods (component type, design features, configuration, application, service duty, component age, industry experience, and plant specific experience) and identification of the predominant failure modes and causes for the more safety-significant and selected less safety-significant components.
3. **Test Effectiveness Assessment** - qualitative assessment of the effectiveness of each test based on their ability 1) to detect a failure and 2) to detect significant degradation that is a precursor to failure. An assessment of a level of confidence in the testing methods is made to determine that, if a real demand occurs anytime over the operating interval before the next test, the component will function correctly is also performed.
4. **Strategy Formulation & Evaluation** - for each component, definition of some schedule of tests (full flow test every refueling outage, ultrasonic every 3 years). An assessment of a level of confidence for each strategy, and an evaluation of the value-impact ratio for the various IST strategies, in terms of impact on core damage frequency and testing costs (man-rem exposure and testing costs), is also performed. Finally, the most effective strategy for

[1] PSA models are now in the process of final approval for all United States operating light water reactors as a result of meeting the regulatory requirements of NRC Generic Letter 88-20, "Individual Plant Examination (IPE) for Severe Accident Vulnerabilities," (NRC, 1988). Working with the Nuclear Energy Institute, the Electric Power Research Institute has published guidelines for the application of probabilistic safety assessment models (EPRI, 1995).

the less safety-significant components and best overall IST program for all components is determined.

Once an IST program has been established, a feedback loop is included based on the actual results of testing for the components.

APPLICATION TO CHECK VALVES

The Westinghouse Owners Group (WOG), one of the sponsors of the ASME research project on Risk-Based IST, performed a demonstration project, in parallel with the ASME effort, to apply the methods to the inservice testing of check valves. The check valves that are part of the emergency core cooling system (ECCS) at Georgia Power's Vogtle Units 1 and 2 nuclear generating station were selected for the demonstration effort. All pressurized water reactors have similar ECCS injection valves. This system was selected for study because:

- The system is a dominant contributor to risk based on the IPE/PSAs performed for Westinghouse plants
- The residual heat removal system, which is part of the ECCS, is an important system during shutdown operations
- The ECCS, in general, has a similar design across most plants with a Westinghouse-designed NSSS
- The ECCS has a wide range of valve importance rankings
- The ECCS accumulator check valve testing is costly for many plants
- Application of the methods to the ECCS offers utilities potential cost savings while maintaining a high level of safety

Risk-Ranking IST Results

The initial risk categorization for the WOG pilot-plant study was completed using the PSA results to calculate importance measures for each ECCS check valve group. Based upon industry established threshold values, two check valve groups were initially identified as being risk significant. However, after the expert panel review of shutdown risk importance and the elicitation of other deterministic information, two additional check valve groups were classified as more safety-significant. Thus, four out of 35 valve groups (encompassing a total of 7 check valves) in the plant IST program were determined to be more safety-significant. This risk-ranking process is also consistent with the draft NEI Industry Guideline for Risk-Based IST (NEI, 1996).

Table 1 provides the justification as to why the four valve groups are determined to be more safety-significant. Table 2 identifies some of the risk significant check valve groups by failure mode. For entries under the "Failure Mode" section of Table 2:

- blocks outlined in **bold** identify a failure mode and check valve group that is more safety-significant
- blocks that are clear (or white) identify less safety-significant valve groups and failure modes that were explicitly modeled in the PSA
- blocks with a grey-scaled background indicates all other less safety-significant valve groups and failure modes

This table provides an example of a simplistic presentation of the risk categorization results that can be easily interpreted by other phases of the demonstration study.

A limited number of valves are in the more safety-significant category offering a significant potential savings in the way IST programs are currently developed, particularly for the large population of valves in the less safety-significant category.

Component Review & Failure Modes and Causes Analysis

Oak Ridge National Laboratory (ORNL) conducted a review of historical check valve failure data under a Nuclear Regulatory Commission Nuclear Plant Aging Research Program, reported in NUREG/CR-5944, "A Characterization of Check Valve Degradation

TABLE 1 MORE-SAFETY-SIGNIFICANT ECCS CHECK VALVES SUMMARY		
Check Valve Group		**Justification for Risk Significance**
1204U6090	Safety Injection System (SIS) pump suction from RWST check valve	Failure of valve to open results in failing both SI pumps (single point failure for the SI subsystem)
1204U6083 1204U6084 1204U6085 1204U6086	Safety Injection System (SIS), Accumulator, and Residual Heat Removal (RHR) cold leg loop check valves	Valve rupture is a potential cause of an interfacing system loss-of-coolant accident. Valves have experienced body/bonnet leaks causing a forced plant shutdown and were subsequently found to be in a degraded condition. These valves see loop conditions and have a tendency to unseat and cause pressurization of the low pressure systems (RHR). Valve failing to re-seat is a significant concern.
1208U6189	RWST to charging pumps suction check valve	Valve is a single point failure which causes the loss of ECCS injection via high pressure charging system.
1204U6013	Boron injection inboard isolation check valve (in injection path between BIT and loop branch lines)	Valve is a single point failure which causes the loss of ECCS injection via high pressure charging system.

and Failure Experience in the Nuclear Power Industry," (NRC 1993 and 1995a). The study, published in 1993, involved the review and characterization of failure records from the NPRDS database from 1984-1990. Approximately 5000 failure records were reviewed and after filtering, a data base containing 1227 failure records remained. Of the 1227 valve failures, 1081 (or 88%) involved safety-related valves.

Following completion of this study, data for 1991 was reviewed and analyzed similarly to the 1984-1990 data review. This work was published in 1995 as Volume 2 of NUREG/CR-5944. However, the 1991 data review included identification of specific valve type, such as swing check, tilting disk check, piston/lift check, etc.

All the above data, which is presently being incorporated into a data base being developed by the Nuclear Industry Check Valve Group (NIC), provides useful information on experience with check valves failures and what percentage of failures are attributed to the different types of check valves, failure modes and causes (Hart, 1996). In addition, to develop a risk-based IST program, one needs to know:

- Of all check valves in operation, how many failures per demand have occurred?

- How many times has a test engineer discovered a failure during a test?

Expert opinion provides a structured process to obtain this information and to utilize it in developing and evaluating potential IST strategies (McAllister, et al. 1986).

The objective of this step is to 1) perform a detailed valve review and 2) identify the dominant failure causes for the failure modes for the more safety-significant components. Each of these steps was determined to rely on expert judgment based on background information obtained from the detailed valve review.

The valve review identifies the key characteristics influencing the determination of effective testing methods including:

- valve type (e.g., swing, stop check, piston, tilting disc, lift)
- design features (e.g, actuator, packing, seals)
- configuration (i.e., horizontal, vertical)
- application (i.e., clean or dirty water)

TABLE 2
EXAMPLE SUMMARY OF SOME ECCS CHECK VALVES IN IST PROGRAM AND PRA MODEL VALVE GROUPINGS AND FAILURE MODES (ALVIN W. VOGTLE NUCLEAR PLANT)

Valve ID	System	IST	IPE	Description	Failure Modes					
					Failure to Open	Failure to Close	External Leakage	Internal Leakage	Rupture	Plugged
1204U6147 1204U6148 1204U6149 1204U6150	SI	Y Y Y Y	Y Y Y Y	RHR to cold leg loop 1, 2 ,3, and 4						
1208U4021	CVCS	Y	N	CVCS seal backflush - Pene. #49						
1208U6032	CVCS	Y	Y	CVCS to regen. HTX - Pene. #50						
1208U6124	CVCS	Y	Y	VCT outlet check valve to charging suction header						
1208U4185 1208U4483	CVCS	Y N	Y Y	Boric Acid (emergency boration)						
1208U4499	CVCS	Y	N	Boric Acid to charging pumps						
1208U6189	CVCS	Y	Y	RWST to CVCS check valve						
1208U4284 1208U4299	CVCS	Y Y	YY	Boric acid transfer pump discharge						
1204U6013	SI	Y	Y	Boron injection inboard isolation						
2402U4017	SI	Y	N	N_2 supply to accumulator penet. #42						

- service duty (e.g., standby, continuous)
- current inspection and test programs (e.g., IST, INPO SOER 86-03, preventive maintenance)
- valve age
- industry experience
- plant specific experience

A detailed form to capture this information is provided in the ASME Volume 2 Document (ASME, 1996c) and the WOG check valve report (Westinghouse, 1995). The form is divided into two sections. Having this information available reduces the biases of the experts performing the assessment by drawing out unstated assumptions, considering all relevant issues, and reviewing both industry and plant-specific experience.

Section 1, Valve Description, identifies relevant features of the valve, its design, its installation, its application, etc. that influence the overall level of confidence that the valve will function correctly if a real demand occurs during the next operating interval. This information helps answer the following types of questions:

- How do these factors interact? For example, is this a spring loaded lift valve operating in a dirty water environment? Or, is this a bonnet hung swing valve that could be damaged during reassembly?

- In general, are there any factors specific to this valve that raise special performance or testing concerns?

Section 2, Valve History, records all relevant experience with the valve or similar valves (in design & application) that influence the overall level of confidence that the valve will function correctly if a real demand occurs during the next operating interval. Both plant specific experience and industry experience (e.g., NIC data base, NPRDS) are included.

Following the valve review, the predominant failure modes and causes for the more safety-significant components are identified. The key failure modes that have been determined in the Vogtle ECCS check valve study are:

- fail to open
- fail to close
- disc rupture

Because very little data exists indicating the relative frequency for different failure causes, any proposed test should be able to detect the likely failure causes for failure modes of interest. Working with representatives cognizant of the NIC data base, the general failure causes considered in the evaluation of testing methods are identified in Section 3 of Table 3.

Test Effectiveness Assessment

This step of the process qualitatively assesses the effectiveness of each test for all the more safety-significant check valves, for the significant failure modes and for the identified causes. The more safety-significant failure modes are identified from PSA and expert panel results. The tests are rated based on their ability 1) to detect a failure and 2) to detect significant conditions that could be precursors to failure. The current and emerging testing techniques that are evaluated are described as follows:

- Full and partial forward flow tests verify that the valve opens. However, this test is may not identify internal degradation. Reverse flow tests, which ensure that a valve disk moves to seat or that leakage past a valve is within a predetermined limit, may also not identify internal degradation. Testing of valves in both directions is generally only performed for valves that have safety functions in both the open and close directions. Most utilities perform both of these tests at the same time (i.e., they do not stagger the testing). Exercising of check valves does not always detect degradation of their performance or their ability to perform the intended function over the next interval.

- Leakage rate testing generally requires that certain systems necessary for plant operation are taken out of service for extended periods. Additionally, containment access may be required, resulting in high person-rem exposure. This testing also may not be cost-effective to perform during cold shutdowns because the installation and removal of test equipment could delay plant startup.

- Disassembly and inspection of valves are generally recommended to identify and correct internal valve degradation. No other testing method is as effective in identifying wear or fatigue-cracking. However, this approach can be costly in terms of time, approach can be costly in terms of time, manpower and radiation exposure; and it increases the potential for errors during reassembly of the valve following examination.

- Non-intrusive examinations are under development and initial implementation is underway in the nuclear industry to primarily replace the need for frequent disassembly and inspection, and to better detect conditions that may lead to check valve failures.

  Several utilities have begun to use non-intrusive testing such as acoustic, ultrasonic or vibration techniques. These methods involve data collection with the system in its normal or test configuration. Hand-held devices or mounted sensors are used to collect noise or vibration data which is compared to baseline information previously obtained under similar conditions. These techniques have been used primarily to identify check valve seat leakage or to identify issues in large, normally open check valves.

  For non-intrusive testing techniques, current industry efforts are showing the need for use of more than one technique to obtain meaningful results for check valve performance.

- Condition monitoring has been developed by the ASME OM-22 Code working group. Condition monitoring focuses on optimizing testing, examination and preventive maintenance activities to improve the performance of components. Condition monitoring involves time-history/trending analysis of the test and maintenance history of a valve or group of similar valves in a similar application in order to establish the basis for specifying testing and maintenance activities. This process can be integrated with leakage rate tests, disassembly and inspection, and non-intrusive techniques to verify and trend performance. Based on the analysis of the valve's performance, the right combination of tests is determined considering the feasibility and effectiveness of the tests in monitoring the failure mechanisms identified in the analysis.

For check valves, IST programs primarily test for "demand failures", i.e., an initiating event would create a "demand" for the valve to function correctly. However, some IST programs test for failure during operation, e.g., leakage in normally closed valves. Testing methods either create an "artificial demand" to establish that the valve works, or examine the valve to discover if any significant condition exists, using either intrusive methods (disassembly) or non-intrusive methods.

In either case, the test creates a "level of confidence" that the valve will function correctly if a real demand occurs before the next test. This level of confidence can be directly assessed as the probability that the valve won't fail if there is a real demand.

This evaluation is not the same as assessing a valve specific failure rate. Instead, it is attempting to define some correlation between the effectiveness of various tests and the confidence in the functional readiness of the valve. For example, if some non-intrusive method allows trending of the conditions of a valve, there would be more confidence that a potential failure would be identified at an earlier state. This would lead to the belief that the valve will have a higher probability of being functional.

Because check valves in the ECCS are, in general, very reliable, this evaluation must rely on the judgement of a component IST team of experts. Two general principles guide the elicitation of expert judgement. The expert should consider: (1) all issues relevant to the judgement, and (2) both generic industry experience and specific plant experience. Together, these principles work to minimize the effect of natural bias to focus on recent issues and experience.

A conditioning step that focuses on relevant failure causes is performed prior to asking the component IST team to assess the effectiveness

of the various test methods. Knowing the likely failure causes for a particular valve helps the experts in making a judgement on the effectiveness of a test for a specific valve. In this assessment, an evaluation is done to determine how likely the potential failure causes may exist for each particular check valve considering its background information. The question that is asked of the experts is, "If the valve in question was found to be in a failed state at any given time, how likely would each of the potential failure causes be expected to contribute to that failure?" The valve background information plays an important role in this assessment. For example, a check valve that has been disassembled and inspected some time after initial plant operation and has been found to be in an acceptable state would help to reduce the likelihood of some of the failure causes (abnormal wear, misapplication or improper installation, and initial design, manufacturing, or assembly errors would be considered unlikely). On the other hand, if the valve has never been disassembled and inspected, then these causes may be more likely to exist. Other background factors may suggest that these potential causes are unlikely.

Table 3 shows the form to assess the likelihood of the potential failure causes (Section 3). The failure causes can be assessed by relative percentages or by the scale of "likely", "unlikely", or "possible". Results are shown for check valve 1(2)1204U6013 to exemplify the process.

The effectiveness of the various test methods in detecting those causes is also shown in Table 3 (Section 4). The tests currently available for check valve testing are shown in Table 3.

For the test effectiveness assessment, if the component IST team of experts judge that there is less than 1 chance in 4 (25%) that a given test will detect failure or significant conditions that could lead to failure, the test's effectiveness is rated as low (L). If the probability of detection is greater than 75% (i.e., less than 1 chance in 4 of failing to detect), the test's effectiveness is rated as high (H). Otherwise, the test's effectiveness is rated as medium (M).

In order to perform the above assessment, the component IST team should have the following expertise:

- IST experience, including knowledge of ASME OM Code requirements and associated procedures for performing the test

- maintenance experience associated with check valves

If emerging test methods are being considered in the evaluation, then an outside expert who is knowledgeable in these methods may be needed if this expertise does not exist in-house. This team will also be needed to perform the evaluation in the next step of the process.

Strategy Formulation and Evaluation

From the evaluation of failure causes and the individual assessment of various test methods at detecting the dominant failure causes, a strategy is defined as the complete set of tests and testing intervals that collectively constitute a check valve IST program. An evaluation of the value-impact ratio for the various IST strategies (in terms of impact on core damage frequency and testing costs) is then performed. This assists in the determination of the most effective strategy and best overall IST program.

For each check valve group, an IST strategy alternative is defined, for example:

- perform reverse flow test every refueling outage

- examine the valve using acoustic emission and magnetic flux every three years

- disassemble and inspect the valve every 5 years

To review current strategies and to define new strategies, a tool called a strategy table is employed. Each column in a strategy table represents a decision area relevant to the objectives. Under each decision area, options are listed. Each option represents one way the decision area may be addressed. A strategy is developed by "stringing together" one or more options from each area into a consistent plan of action. Not all strategies need be evaluated, some will be rejected out of hand. The strategy table clearly defines and communicates what options are being considered and what options

are not being considered. Strategies chosen for evaluation should bound the set of alternatives.

For Vogtle, a team of Vogtle IST and maintenance personnel, along with a member of the NIC group and a non-intrusive test method expert, formulated test strategies for a number of check valves by synthesizing the information from the valve review and test effectiveness assessments.

Table 4 is a variation of the strategy table and is used for the formulation and evaluation of testing strategies. Section 5 of Table 4, "IST Strategy Definition" identifies the various test methods and test frequencies in order to formulate strategies. Letters are used in the various blocks to define test strategies. The current test method at its defined frequency is identified and variations of the current test strategy are evaluated (e.g., extended intervals, in combination with another type of test, etc.). Non-intrusive strategies along with a disassembly and inspection at various test intervals are also formulated by the component IST team.

The next step in the process is to evaluate the effectiveness of the strategies. To assist in evaluating the strategies, Section 6 of Table 5, "IST Strategy Evaluation," an ordinal scale from 1 through 9 is developed related to the level of confidence that, if a real demand occurred anytime before the next test, the valve would function correctly. The current IST testing strategy is assigned a score of "5" and other strategies are assessed relative to the current testing strategy. The component IST team comes to a consensus on each score. A low score indicates a test that is least effective at discovering the failure mode/cause, whereas a high score indicates a test that has a high confidence in detecting the failure mode/cause. The intent of this ranking is to evaluate the effectiveness of various testing strategies relative to each other using the valve background information (including industry and plant-specific failure data), the effectiveness rating and any available performance information associated with the various tests, and the engineering judgement of the component IST team members. Usually, the score of the test strategy for check valves is more influenced by the type of test that is performed rather than the test frequency unless failure data suggests otherwise. Therefore, cases exist where the test is kept the same and the frequency is extended, but the score remains constant. For this situation, the reliability of the valve to perform its intended function is shown to be very reliable based on industry and plant-specific failure data. In other cases, however, an extension in frequency results in a lower score because of valve performance issues.

For Vogtle, the strategies that have been formulated and assessed for a more safety-significant check valve, 1(2)1204U6013, are provided.

A comprehensive IST program for check valves must address several objectives. Any attempt to rank alternative strategies depends on a clear understanding of the objectives that the strategy seeks to achieve. For the IST program, the objectives are:

- Safety - insure that valves in the IST program will perform their required safety functions to:
  - shut down the reactor to safe shutdown
  - maintain the reactor in safe shutdown
  - mitigate the consequences of an accident

- Economic - minimize testing costs, including man-rem exposure costs and minimize the impact of valve failure on plant availability and potential damage to other plant systems and equipment.

Appropriate decision criteria need to be defined in order to measure the degree to which any proposed strategy achieves that objective.

TABLE 4
SECTION 5: IST STRATEGY DEFINITION

Unit Name : *Vogtle Units 1 and 2*
Component ID : *1(2)1204U6013*
Failure Mode : *Fails to Open (Fails Closed)*

	routine operator inspection	forward flow		reverse flow	leakage rate		disassemble & inspect				acoustic[1]	magnetic flux	ultrasonic[1]	temper-ature	radio-graphy
test frequency		full pressure	partial pressure		w/o THT	w/- THT	w/o PAT w/o THT	w/- PAT w/o THT	w/o PAT w/- THT	w/- PAT w/- THT	w/- THT	w/- THT	w/- THT	w/- THT	w/- THT
Strategy: Current															
quarterly															
ea refueling outage		A													
every 5 years															
every 10 years		B													
other															
Strategy: Nonintrusive with Condition Monitoring															
quarterly															
ea refueling outage		D									C/D	C/D			
every 5 years															
every 10 years		F									E/F	E/F			
other															
Strategy: Disassemble															
quarterly															
ea refueling outage															
every 5 years															
every 10 years									G	H					
other															

For each strategy, define the strategy by identifying which test will be performed at each frequency, then indicate your *relative level of confidence* in the strategy on the next page.
[1]Acoustic and magnetic flux methods are performed together.

TABLE 5
SECTION 6: IST STRATEGY EVALUATION

Unit Name : *Vogtle Units 1 and 2*
Component ID : *1(2)1204U6013*

Strategy	Score									NPV Cost ($)	Notes
	1	2	3	4	5	6	7	8	9		
Current - A					X					3,092	
Current - B					X					334	
Nonintrusive - C*							X			24,351	
Nonintrusive - D**									X	27,443	
Nonintrusive - E*						X				2,630	If you increase the interval for nonintrusive, then you won't have the data to do condition monitoring
Nonintrusive - F**								X		2,964	
Disassemble G					X					7,284	
Disassemble H							X			7,409	PAT - flow test

Scoring: 1 = worst possible (highest failure rate), 5 = current practice (nominal failure rate), 9 = best achievable (lowest failure rate).
* with flow - no measurement
** with flow and flow measurement

NPV Cost = Net present value cost and is based on a discount rate of 7.5% and assumes a remaining plant life of 30 years. The capital cost for non-intrusive equipment is not included because it could be spread across many values and many other applications. Radiation costs are included.

In order to address the effectiveness of current and alternative testing strategies in meeting the above IST program objectives, the following decision criteria are used:

- Safety - score of various test strategies as related to component failure rate and thus the incremental change in core damage frequency and containment release

- Economic - net present value of revenue requirements to measure testing costs and man-rem exposure

Any IST program represents a cost to the utility. It has value only if it effectively lowers the probability that the utility will avoid future consequences that have a higher cost, where this cost factors into the value that the utility puts on avoiding risk. Risk results from the chance that the value of an alternative (as measured by the decision criteria) will be unacceptable. To structure the factors affecting risk, an influence diagram is used for modeling the risks and values of current and alternative check valve IST strategies.

The decision criterion for the evaluation is the "value-impact ratio."[(2)] The value is measured as the "incremental risk of core damage" and the cost is the "total cost." The incremental risk of core damage is influenced by the "level of confidence in a test strategy" while the total cost is influenced by the "man-rem exposure" and the actual "test cost." All of these factors are influenced by the test strategy.

To assess the safety risk (incremental risk of core damage), the categorization of components into the more safety-significant and less safety-significant groups implies the impact that a strategy will have on core damage frequency. Because check valves are very reliable components, even if the failure rate decreases, no appreciable decrease in core damage frequency can be gained. In addition, a check valve failure rate would have to increase significantly to increase core damage frequency. Therefore, the level of confidence in a test strategy becomes the best measure of value of a test strategy. (Note: this may not apply to other equipment types that show significant changes in core damage frequency as a function of failure rate.)

To determine the costs associated with various test strategies, cost data can be compiled from industry and plant-specific sources on a per valve basis. Capital costs should not be applied to a single valve because they usually can be spread across many valves and many other plant applications and may mislead any decisions if capital costs are included. Net present value calculations should be based on a discount rate to be calculated over a remaining plant life. The net present value (NPV) per valve for valves located outside containment (implies no man-rem costs) and for valves located inside containment (or high radiation areas) can be determined based on test type and test frequency.

Based on this information, each valve (or valve group) can be assessed. The total costs for each different strategy can be shown along with the level of confidence for each valve. A determination as to which test strategy could be employed is determined based on the utility's risk aversion and cost considerations.

For example, as shown in Table 5, for the boron injection isolation check valve (1204U6013), the current strategy (Strategy A, full flow test every refueling) costs \$3092 and has a confidence level of 5, while Strategy B (full flow test every 10 years) has the same confidence level of 5 with a reduced cost of \$334. For this check valve, the level of confidence could be improved to an 8 (Strategy F, acoustic and magnetic flux testing along with a full flow test with Δp measurement every 10 years) at a slightly lower cost than the current strategy's cost. The information also shows that performing acoustic and magnetic flux testing on a refueling outage basis for this valve is more costly by an order of magnitude.

Based on the assessment of each valve (or valve group), the test strategies for the remaining valve groups can be determined. The optimal IST strategy for the check valves can be "built up" from the optimal strategies for each valve or valve group.

(2) See CRTD-Vol. 23 (ASME, 1993) and SECY-95-028 (NRC, 1995b) for further discussion on use of value-impact ratio for ASME and regulatory applications, respectively.

For each of the ECCS check valves evaluated at Vogtle Units 1 and 2, the component IST team was always able to define an equivalent or more effective test strategy in terms of safety at an equivalent or lower cost than the current test strategy. Therefore, the decision as to which of these strategies should be implemented is fairly straightforward. The more difficult decision would involve the case where a more effective test strategy can only be achieved at an increased cost. Some of the factors that would impact this decision include:

- how much impact the more effective test strategy would have on core damage frequency
- how much impact the more effective test strategy would have on plant operation
- how much longer the owner plans to operate the facility

The test effectiveness assessment was completed for the four more safety-significant check valve groups and four less safety-significant check valve groups. The less safety-significant check valve groups were selected to cover different valve designs not analyzed as part of more safety-significant check valve groups and special issues associated with valves (e.g., accumulator check valves). For the eight check valve groups, the best test strategy was identified by selecting the test strategy that provided an equivalent or better level of confidence (a score of 5 or better) in valve operation at the least cost.

For the remaining low safety-significant check valve groups, a match between the analyzed valve groups and the unanalyzed valve groups was attempted to define a suggested test and interval based on:

- check valve type and size
- check valve function
- current test method and interval

Using suggested test intervals, component failure probabilities were modified in the PSA and the core damage frequency was recalculated.

Core Damage Frequency Assessment

For the more safety-significant components, the test effectiveness assessment "level of confidence" is equivalent to or greater for the suggested test method and interval and therefore, the failure probability for the more safety-significant components was assumed to remain the same (no credit for improvement).

For the less safety-significant components, increases in test interval were assumed to impact failure probability. To obtain new probability of failure for a new test period, (from page 5-12 of NUREG/CR-2300, *PRA Procedures Guide*, 1983) the new probability is expressed as:

$$P_{new} = 1 - (1-P_{old})^{t1/t} \quad (1)$$

where

P_{new}	=	new probability
P_{old}	=	old probability
t	=	old test period
t1	=	new test period

The above equation is essentially equivalent to:

$$P_{new} = P_{old} * (t1/t) \quad (2)$$

The basic event probabilities associated with each check valve in the less safety-significant groups were modified (including the common cause probabilities).

The core damage frequency increased by approximately 6% based on an initial set of suggested test intervals. The check valve groups that were contributing the most to the 6% increase were reviewed and more stringent test intervals were proposed for these check valve groups and the core damage frequency was recalculated. This resulted in an increase of approximately 2%. The final selection of test intervals for these check valve groups is dependent on additional qualitative arguments (including any compensatory measures) and utility's risk aversion.

Final Strategy and Estimated Savings

Net present values were calculated for the current strategies and the recommended strategies (method and interval) for all the check valve groups. A savings of $335,000 can be achieved in direct costs with the implementation of a risk-based IST program over the 30 remaining years for each unit. Other indirect

cost savings should also be reduced. These indirect cost savings, on a per refueling outage basis, include:

- **Outage critical path reduction** - shorter defuel window and flow balancing are estimated to result in a 24-48 hour reduction in outage time each outage

- **Program administration** - reduced maintenance work orders, surveillances, and clearances

- **Contractor support** - reduction of contractor personnel

- **Insulation removal and scaffolding requirements** - reduction in requirements

Therefore, the total savings expected from a risk-based program for ECCS check valves is estimated to be a net present value of $3.7 million over the 30 remaining years of each unit. In addition, this program will help to eliminate testing strategies that may damage the component (e.g., from errors made from disassembling and reassembling valves) or that may result in an incident/accident because the potential malfunction was not discovered. This estimate is based on the cost of unplanned shutdown ($340,000 per day) and the average length of unplanned shutdown (2.5 days) or $850,000.

SUMMARY

Once an overall IST program is defined, inservice tests are performed and the results of the tests are then evaluated against the data used in the initial risk prioritization. If failure probabilities need to be changed as a result of these findings, the risk-ranking process is repeated, as appropriate, in order to move components within the IST risk-ranking categories. Changes to the test strategies may also need to be made. The risk-based IST process will benefit utilities, industry, ASME and NRC by better focusing and allocating limited resources to the more safety-significant components. The utilities should experience a reduction in plant operating and maintenance costs associated with risk-based IST, while maintaining a high level of plant safety. The NRC has acknowledged the intent to implement these applications as described in a recent policy statement (1995) and implementation plan (1994).

Discussions are already underway with cognizant industry representatives, such as the Nuclear Industry Check Valve Group and the ASME OM-22 Working Group, to facilitate the development and use of templates for check valves. These templates would be used by nuclear utility representatives in defining risk-based IST programs for check valves.

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A Practical Approach for Implementing Risk-Based Inservice Testing of Pumps at Nuclear Power Plants

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ABSTRACT

The American Society of Mechanical Engineers (ASME) Center for Research and Technology Development's (CRTD) Research Task Force on Risk-Based Inservice Testing has developed guidelines for risk-based inservice testing (IST) of pumps and valves. These guidelines are intended to help the ASME Operation and Maintenance (OM) Committee to enhance plant safety while focussing appropriate testing resources on critical components. This paper describes a practical approach for implementing those guidelines for pumps at nuclear power plants. The approach, as described in this paper, relies on input, direction, and assistance from several entities such as the ASME Code Committees, United States Nuclear Regulatory Commission (NRC), and the National Laboratories, as well as industry groups and personnel with applicable expertise.

Key parts of the risk-based IST process that are addressed here include: identification of important failure modes, identification of significant failure causes, assessing the effectiveness of testing and maintenance activities, development of alternative testing and maintenance strategies, and assessing the effectiveness of alternative testing strategies with present ASME Code requirements. Finally, the paper suggests a method of implementing this process into the ASME OM Code for pump testing.

Introduction

The ASME Research Task Force on Risk-Based IST has developed guidelines for testing pumps and valves. This paper proposes a methodology for the implementation of risk-based IST for pumps. This paper discusses only the process for the IST Component Group I (high importance) pumps in detail. The proposed methodology consists of the following steps:

- o the performance of a component review,
- o the identification of the important failure modes,
- o the identification of significant failure causes that are likely to result in those failure modes,
- o the estimation of failure-cause occurrence rates,
- o the estimation of the effectiveness of testing, maintenance, and inspection

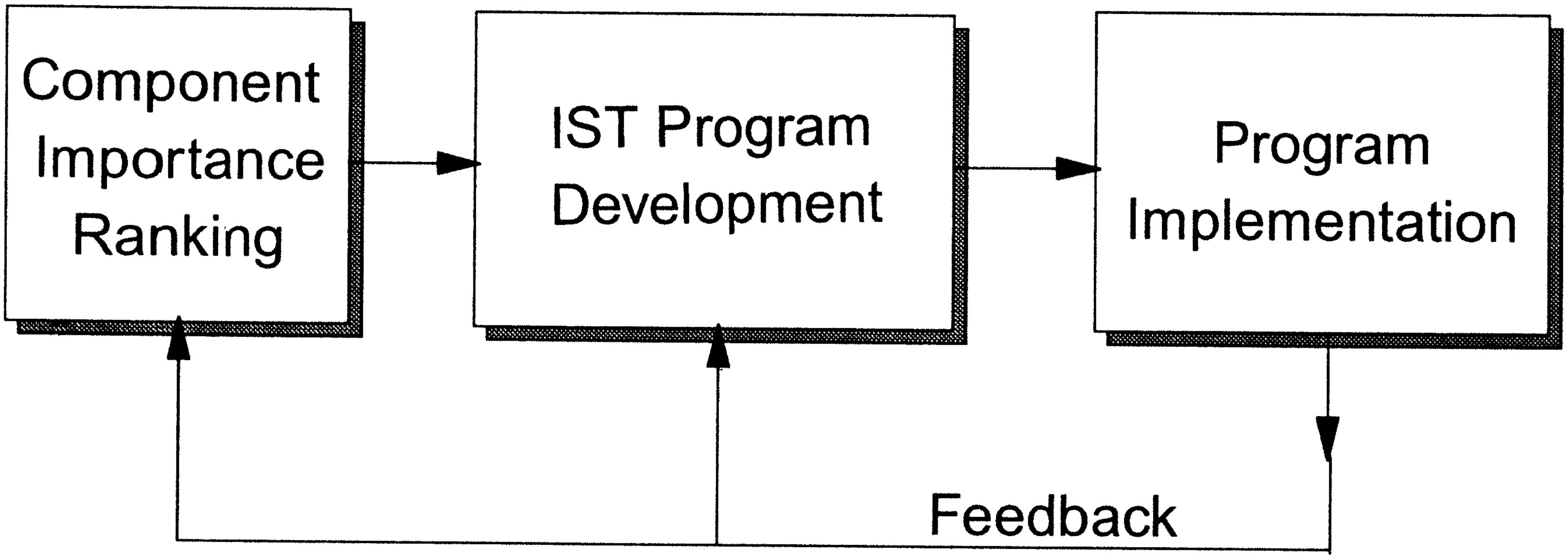


Figure 1 Risk-Based IST Methodology

practices to detect the significant failure causes,

o the development and assessment of potential testing strategies, and

o implementation and feedback.

Risk-Based IST Methodology

Figure 1, "Risk-Based IST Methodology" (ASME 1996), provides a general overview of the risk-based IST process. As shown in the figure, components (pumps and valves) are ranked based on their importance, an IST program is developed, the program is implemented, and results and insights are fed back into the process. The first box, component importance ranking, identifies the safety significant, or IST Component Group I, components and their important failure modes using both probabilistic risk assessment (PRA) and deterministic insights. The process is described in Section 2.2 of the Volume 2 guidelines This step also identifies the less safety significant, or IST Component Group II components. IST programs are then developed for both component groups[1], as shown in the second box. Once appropriate programs are developed, they are implemented through changes to test procedures and new or different tests are performed. Then the results of the program changes are reviewed and the insights gained from this review are fed back into the process. This should be considered part of a living process rather than a one-time program change.

IST Program Development

Figure 2, "IST Program Development Process," provides a more detailed look at the steps that would be followed for IST Component Groups I and II. For pumps, the ASME guidelines address only motor-driven, multi-stage horizontal centrifugal pumps contained in the Auxiliary/Emergency Feedwater (AFW) system to limit the scope of the initial study. There are other pumps and driver types that are significant both from the standpoint of their risk significance and the fact they have IST requirements according to the ASME Code.

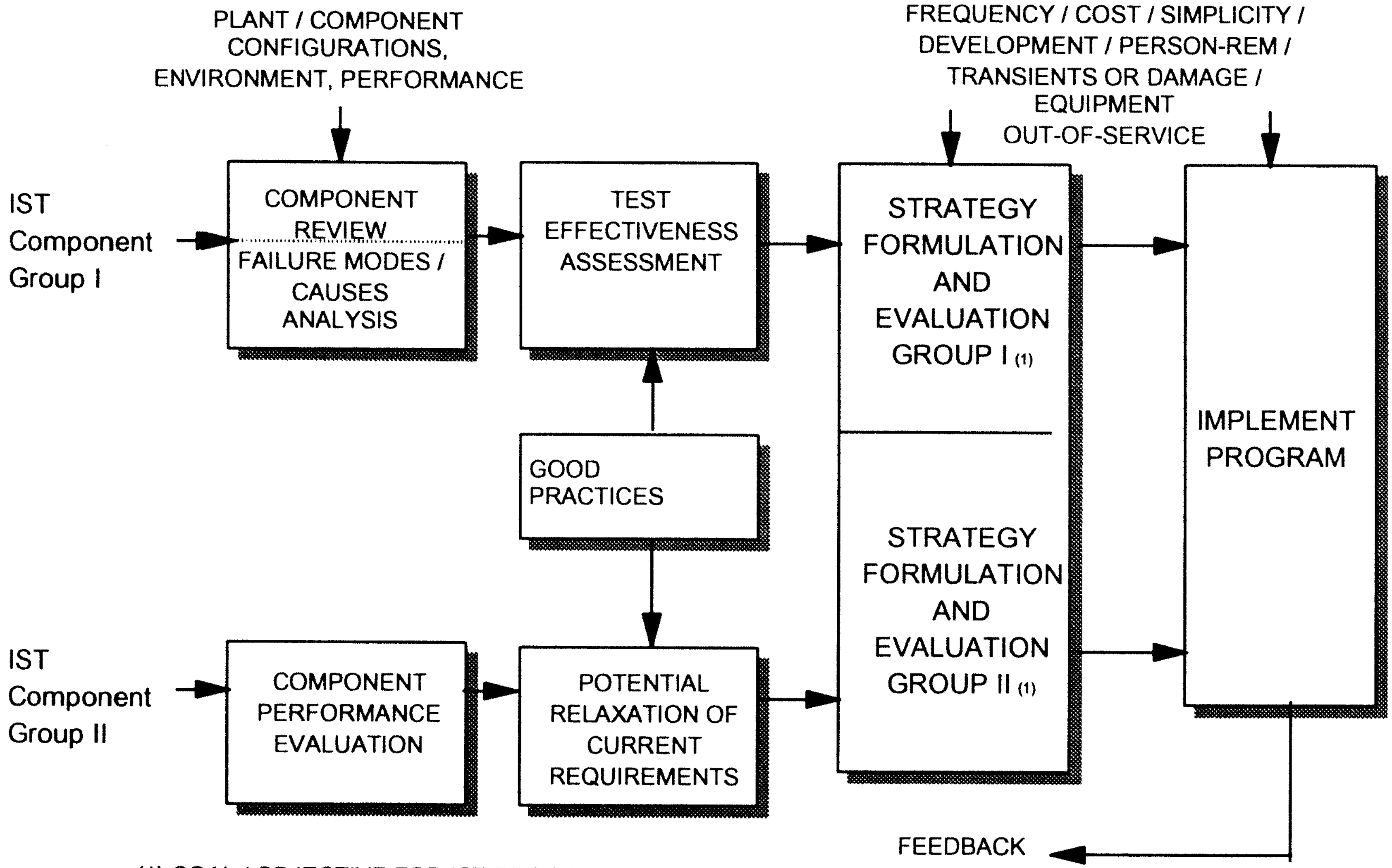
As shown in Figure 2, the first step in the process is a component review to develop background information for the pump and its driver[2]. This is done by performing a pump review. The review process assembles important general and specific information pertinent to the pump and its driver.

The pump review also assembles information regarding the pump's performance history and existing maintenance and testing programs.

A Westinghouse Owners Group (WOG) demonstration project with Shearon Harris Nuclear Plant further explored this process for pumps and (WOG 1996) provides additional information on this topic.

Following the component review, is the failure modes and causes analysis (FMCA). The value of the FMCA process is based on the notion that detailed analysis of the failures of a representative population of components facilitates the identification and ranking of likely failure-causes for the similar subject component. The subsequent implementation of more appropriate testing and maintenance activities will identify and correct the degradations that are the most likely to occur before they result in pump failure.

The FMCA process is used to develop a generic failure-cause list and then to refine it to a plant-specific failure-cause list. The



(1) GOAL / OBJECTIVE FOR IST PROGRAM
COMPONENT GROUP I - PERFORM TEST TO DETECT SIGNIFICANT CONDITIONS THAT ARE PRECURSORS TO FAILURE AND TO VERIFY FUNCTIONA READINESS
COMPONENT GROUP II - PERFORM TEST TO VERIFY FUNCTIONAL READINESS

Figure 2 IST Program Development Process

generic failure-cause list identifies likely failure-causes and their prevalence (cause occurrence percentages). The generic list is developed based on industry failure data from a population representing the subject pump. For use at a particular plant, a plant-specific failure-cause list should be developed. This list should be based on a review of the generic failure-cause list applicable to the pump type under examination and the consideration of plant-specific insights developed during the pump review.

The next part of the risk-based IST process, as shown in Figure 2, is to estimate the effectiveness of test, maintenance, and inspection activities to identify degradations or failures due to the identified failure-causes. These test effectiveness estimates will be used to compare tests and combinations of tests later in the process. Experts familiar with pump maintenance and testing can estimate the effectiveness of various testing, inspection, and maintenance activities at identifying existing pump failures and degradations that will lead to pump failure, if left unchecked.

The test effectiveness estimations are then used with the plant-specific failure-cause list to formulate and assess various strategies for testing pumps. Methods are proposed (ASME 1996) to semi-quantitatively assess the effectiveness of different tests or combinations of tests. These assessments can be relatively simple or quite complex, requiring the assistance of powerful computerized analytical tools. The simple analysis is provided in this paper. A computerized decision analysis model is being developed in cooperation with the WOG (Perdue 1996).

Using these analytical approaches, the effectiveness of regulatory-driven testing activities, such as prescriptive ASME Code tests can be analyzed in detail and assessed. The effectiveness of various potential alternative testing and maintenance strategies can also be assessed in detail using this process.

Detailed Description of Process for More Safety Significant Components

The following discussions describe each step of the proposed risk-based methodology for the more safety significant pumps in greater detail and provides some insights on how to perform the steps effectively. As shown in Figure 2 and described in the ASME Guidelines, the goal/objective for Component Group I is to perform tests to detect significant conditions that are precursors to failure and to verify component functional readiness. Achieving this demands a reasonable level of rigor. It does not necessarily mean that more expensive or difficult tests are appropriate.

Pump Review

The first step of the process is the development of relevant information regarding the pump and its driver for use in the risk-based IST process. We recommend that a pump review form be developed to logically gather and record both general and specific information on the pump, its driver, plant testing practices, requirements, and maintenance history.

The types of information that are available and might be considered during a pump review will need to be modified for use at any specific facility (to reflect the relevancy and availability of information for a particular pump). In general, a pump review form should consider the following information:

- o General Information, to identify the relevant features of the pump and driver, including the operational requirements of the pump,
- o Driver Description, to provide a detailed description of the characteristics of the driver such as the type of motor, bearings, bearing materials, lubricants, etc.,
- o Pump Description, to provide a detailed description of the pump that identifies characteristics, such as number of stages, physical orientation, bearings, type of seals, etc.,
- o Testing and Maintenance History, to record testing, maintenance and all other plant-specific experience with the pumps and review information from similar pumps at other plants in the industry.

The review form should capture the information efficiently and in a timely fashion. In most cases, significant efforts should not be expended simply to "fill in all the blanks," but a reasonable effort should be made to gather all relevant information.

FMCA

The next step of the process is the FMCA, which is used to determine the significant failure-causes for each applicable failure mode. First, the significant failure modes for a component must be identified. This can be done by reviewing applicable safety documents such as the results of a PRA or individual plant examination (IPE) or by using expert opinion.

The distinction between the primary failure modes (ASME 1996) applicable to motor-driven AFW pumps, "failure-to-start" and "failure-to-run," was vague. Additionally, the failure-cause data was not considered supportive of the design of different testing strategies based on either one or the other of these modes. Therefore, the mode "failure-to-start" was treated the same as "failure-to-run" in the development of guidelines.

Once the applicable failure modes are known (the modes are identified during component risk ranking), the significant failure causes must be identified. This was done by doing a detailed review of Licensee Event Reports (LER) and the Nuclear Plant Reliability Data System (NPRDS) data bases. By "binning" each failure into an appropriate failure cause bin and tracking the results, dominant failure causes are identified and an occurrence rate can be established for each significant failure cause.

The FMCA results for pumps should be reviewed by experts, such as members of the ASME OM Working Group on Pumps (OM-6), prior to use in a risk-based IST program. This should help to ensure that significant failure causes are not overlooked or the failure reports misinterpreted and should help to ensure that the information is suitable for its intended purpose.

Additionally, current work by the national laboratories, such as the Idaho National Engineering Laboratory (INEL) or Brookhaven National Laboratory (BNL) and industry groups such as the Electric Power Research Institute (EPRI) or the Instutute of Nuclear Power Operation (INPO) regarding pump (and valve) degradations and failures should be consulted for insights. One such study of pump degradations was recently done by Oak Ridge (Greene 1995).

Test Effectiveness Assessment

Once the significant causes of potential failures are identified, various test, maintenance, and inspection activities can be assessed to determine their effectiveness at identifying the presence of the likely failure causes. This information will be used as part of the test strategy development process. It allows an assessment of the relative merits of potential changes in the testing program, such as comparison of alternatives or the value of performing new or additional tests.

Members of ASME OM committees have followed this process for estimating test effectiveness for a motor-driven multistage boiler feed-type pump (auxiliary feedwater pump) and several types of check valves. Generally speaking, the level of difficulty in assessing test effectiveness is related to the extent and complexity of the component or system under review. Test effectiveness assessments were also done as part of the WOG demonstration project with engineers from Shearon Harris, WOG, and members of the ASME OM Working Group on Pumps (WOG 1996). The results of that work are described in the WOG report.

Various methods might be used to develop the effectiveness estimates. A good method for obtaining effectiveness estimates is to organize a facilitated group session. The group should have experience with testing and maintenance of the component under scrutiny. Other desirable qualities for the group include: experience with design, manufacturing, or operation of the component. When issues and concerns are aired and discussed, an effort should be made to obtain consensus estimates, thus greatly reducing the uncertainty of the individual estimation results. The experts should be asked to evaluate the ability of testing methods to detect an existing failure or predict an impending failure resulting from each of the identified significant failure causes.

The test effectiveness estimates should be based on the assumption that a test can be implemented under idealized conditions. Certain factors specific to any specific installation can affect the realized effectiveness of an actual test or predictive maintenance activity. Individual effectiveness estimates described previously should be modified (increased or decreased) as necessary based on specific considerations at a particular plant. Many of these potential limitations are evident to the operator conducting the testing or to the analyst evaluating the results.

Biasing Factors for Testing and Maintenance Activities (Hartley 92), identifies some effectiveness and test quality considerations for several test and maintenance activities. The effect of these biasing factors may act to improve or reduce the effectiveness achieved using specific test, maintenance, or inspection methods. For example, testing of centrifugal pumps at very low flow rates may mask existing degradation that would be evident at higher flow rates, resulting in a less effective test. There are also some general matters that apply to a wide variety of test effectiveness issues, such as reviewing human factors concerns associated with the testing and maintenance activities. This sort of review may help to identify and resolve problems and increase effectiveness. Well written inspection and testing procedures and personnel training/qualification can also help to achieve consistent high quality results. Therefore, biasing factors should be considered in relation to the estimates of effectiveness in implementing this process at any particular plant.

Once a test's effectiveness to identify or predict each significant failure cause has been assessed, overall test effectiveness can be determined. This can be accomplished by multiplying the test effectiveness for each significant failure cause by the percentage each failure cause contributes to the total failure probability, then summing all the resulting products. This will result in an overall test effectiveness to detect failure due to all the significant failure causes.

Test Strategy Development and Evaluation

The development and evaluation of various alternative testing strategies is an important step in the development of a risk-based test program. The process of developing the background material should have been methodical and reasonably detailed to help limit uncertainties in the results. However, the AFW pump/motor driver combination and its interactions with the plant are complex. There are many complicating factors and interactions, e.g., test induced damage, unavailability, and test frequency, that should be considered when developing and assessing any strategy involving combinations of tests and test frequencies.

A relatively simple process to assess generic strategies using a "strategy rating" is described in the guidelines. There are obvious limitations of this technique. For example, it cannot discriminate between the value of a test performed daily and one performed once every ten years. A related assumption used in PRAs regarding test frequency is that the probability that a component is failed increases linearly following a successful test, until it is reset by another successful test. Frequent testing is a seemingly obvious method for keeping the probability of failure low. However, an ineffective test that cannot detect a failure cause that would render a component inoperative, cannot rule out the presence of a failure and therefore, reset the probability (presumably the probability would continue to increase).

By using risk-based IST techniques and focusing our efforts on the most likely failure causes for important components we will increase our knowledge and awareness of component performance issues. As we gain more knowledge in this manner, we may be able to better understand the relationship between the effectiveness of a test and it's implication (if any) with regard to changes in test frequency.

The simple analysis technique described in the guideline uses the test effectiveness ratings to calculate "strategy ratings." A strategy rating for one cause and one test is simply the test's test effectiveness rating for that cause. Specific tests can be evaluated by comparing their test effectiveness ratings for various failure causes. Multiple tests can be combined and compared using the strategy ratings.

As stated before, the AFW pump and motor combination is complex. A testing strategy for a complex component will likely be composed of several complimentary testing and inspection methods. A strategy rating can be calculated for these more complex situations as illustrated in simple terms in Table 1. The last column of the table shows the test effectiveness rating of the most effective test for each cause. The values in the last column are then summed to yield the strategy rating. This technique does not account for the potential synergies of employing multiple dissimilar tests. Probabilistic calculational techniques might be used to calculate the likely synergies (such as A + B - [A X B]). The calculated strategy rating may be used to

analyze current ASME Code specified tests, simple visual inspection and lube oil analysis, or some other combination of methods. The process of arriving at strategy ratings is simple and intuitive to persons familiar with testing and maintenance processes. As would be expected, strategy ratings, tend to show an improvement when employing multiple tests over those resulting from fewer or single tests. The cost and ease of implementing each strategy depends on the nature of the failure cause and various other issues, such as accessibility for testing or inspection.

Table 1 Example Determination of Strategy Rating for Two Tests

Cause	Test 1 Effectiveness Rating	Test 2 Effectiveness Rating	Largest Effectiveness Rating
Packing leakage, overheating	0.07	0.04	0.07
Bearing wear, corrosion, breakage	0.01	0.04	0.04
Sum of Largest Effectiveness Ratings or "Strategy Rating"			0.11 (0.07 + 0.04)

Analysis was performed for three strategy scenarios in the guideline (ASME 1996). The first, Strategy 1, was essentially equivalent to the currently required ASME Code testing of ASME Section XI, Subsection IWP (ASME 1989). The ratings in the last column of the Table for the two tests of Strategy 1 were summed and yielded a Strategy Rating for the combination of tests of 41%.

The Strategy 1 rating could be a measure of the regulatory effectiveness for an AFW pump at a particular plant whose only regulatory driven requirement was compliance with the ASME 1990 OM Code (ASME 1990) or 1989 Edition of Section XI (ASME 1989). A similar analysis could be performed for the subject component on the tests and maintenances that must be performed to satisfy other regulatory commitments. Strategy 2, was a combination of three tests and inspections. This strategy is based on a recent WOG study that found what many plants are currently doing voluntarily for their critical equipment; that is hydraulic testing, vibration spectral analysis, and visual inspection. The strategy rating for this combination yielded 62%.

The third strategy, incorporated four complimentary tests at two frequencies; quarterly and every two years. The tests are a bump start quarterly, with a concurrent visual inspection. This is followed every two years by a hydraulic parameter test at a high flow rate (near the pump's design flow rate) and a lube oil analysis. The strategy rating for the combination was 62%.

These were somewhat simple methods of assessing various alternative testing strategies. They are based primarily on estimated effectiveness and cause occurrence insights. There are many other important issues that

should be considered in the selection process for a particular strategy. These issues can be considered manually using a framework such as the NRC's Value-Impact Assessment manual, or by using computer programs commercially available or specifically programmed to suit the task. The work by Dr. Perdue and the WOG (Perdue 1996) allows a very detailed and thorough assessment of testing strategies.

Process for Less Safety Significant Components

As discussed in the guidelines, the goal/objective for the IST program Component Group II is to perform testing to verify component functional readiness and not to predict failures or pinpoint levels of degradation. Degradation or failures of these components will have little impact on the plant risk. As shown in Figure 2, the main issues that are considered are component performance related and the components are candidates for relaxation of current testing requirements (e.g, fewer tests and/or extended intervals between tests).

Implementation and Feedback

Once appropriate testing strategies are developed and assessed, the next step is to implement the strategies into the testing program. This is done by appropriate updating of the IST program and modifications to or preparation of new testing procedures. Following that, the effects of changes in the test program should be predicted and monitored as appropriate, the insights should be fed back into the program as part of a living process.

Adoption of Risk-Based Inservice Testing Into the OM Code

The Research Task Force on Risk-Based Inservice Testing completed its work on the guidelines (ASME 1996) in late fall of 1995. A detailed presentation on the results of the research was given to the ASME Operation and Maintenance Committee at the December 1995 meeting in St. Petersburg Beach, Florida. The process promises significant savings in O&M costs and increased operational safety of nuclear power plants by focussing resources on the components with the highest contribution to risk. Various OM Committees, including the Special Committee on Standards Planning, Subcommittee on General Requirements, Subcommittee on Valves, and Subcommittee on Mechanical Equipment and Systems, and various OM Working Groups and other committees are now working to adopt risk-based inservice testing methods.

There are several options for implementing risk-based inservice testing including; 1) the ASME OM Code can be changed to reflect the technology, 2) plants can submit requests for relief from the ASME testing requirements according to the Code of Federal Regulations, 10 CFR 50.55a, or request an authorization to use the method as an alternative to the Code, or 3) Code Cases could be written as alternatives to the Code requirements.

Following, is a brief analysis of each of these options. The first option is to change the ASME Code. This would entail changes to the Code scope (components covered) and significant revisions to testing requirements for affected components. Changes to the Code requirements are difficult and can be very time consuming. It may take years to

incorporate the risk-based methods via this option.

The second option, is for plants to develop alternative programs or to prepare requests for relief from the Code requirements. These requests would have to be reviewed by the US NRC for approval or authorization. The NRC may give conditional approval or deny the requests. The requests would have to be sufficiently detailed to allow adequate review and large numbers of submittals would be very taxing on limited NRC review resources. The NRC is not encouraging this approach.

The third option, that is favored at the time of this writing by the ASME OM Committee is to prepare Code Cases as alternatives to the Code requirements. These Code Cases can be prepared in the form of Appendices to the ASME Code (this would ease incorporation into the Code following approval of the Code Case). Also, the review and approval process is often more efficient and rapid for a Code Case than it is for a Code change. Approval might be gained in a year or less with aggressive committee schedules. The Code Cases can then be followed by prompt incorporation into the Code.

Summary and Conclusions

The ASME Research Task Force on Risk-Based IST has developed guidelines for risk-based testing. The guidelines describe a process of ranking components into two groups and developing, analyzing, implementing, and monitoring testing programs for both groups. We have reviewed the process that was developed for risk-based testing of pumps. The guidelines for Component IST Group I pumps suggest that an implementing plant perform a pump review, FMCA, and test effectiveness assessment. The information from that process is then used to develop, assess and compare different testing strategies (combinations of tests and maintenance activities) and select an appropriate strategy for implementation. A less rigorous process is used for the Component IST Group II pumps. Once programs are developed for both groups, the programs are implemented and the effects of the changes are monitored as part of a living process.

Members of the ASME OM Committees have assisted with the research for both pumps and check valves and been given presentations of the results. The committees are now working to incorporate methods similar to those developed during the research into the ASME Code and are developing Code Cases for various components.

Acknowledgments

The authors acknowledge the invaluable assistance of the other Research Task Force, Research Steering Committee, and Independent Peer Review Committee members, the Westinghouse Owners Group, ASME OM Working Group on Pumps, and others in developing the ASME guidelines. Thanks also to Mr. Richard Gregg of Testing Technologies Company for his input and review of this paper.

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1. The IST program is likely to be much less rigorous for IST Group II components than for Group I components.

2. The boundary of this study includes the pump and its motor driver. During the data analysis for the FMCA, failure data were found outside of the pump and driver boundary. The ASME OM Code 1995, Subsection ISTB, which states the current pump testing requirements, identifies a boundary for testing that generally excludes the driver from the detailed testing requirements.

USE OF EXPERT JUDGMENT IN THE DEVELOPMENT AND EVALUATION OF RISK-BASED INSERVICE TESTING STRATEGIES FOR PUMPS AND VALVES

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ABSTRACT

This paper describes a rigorous approach for quantitatively evaluating inservice testing effectiveness that evolved from two pilot plant studies. These studies prototyped methodologies for designing and selecting inservice testing (IST) strategies in a manner structured to insure that the targeted components will perform their required safety functions while minimizing life cycle inservice testing costs. The paper concentrates on the use of expert judgment in developing test effectiveness measures that move risk-based methods beyond ranking to optimization of plant IST programs. Selected results for check valves and pumps are shown to illustrate the practical significance of the approach.

INTRODUCTION

Nuclear power plant inservice testing programs are intended to uncover incipient degradation or malfunction of equipment so as to prevent an unanticipated failure. The "where, when and how" of such testing is based on mandated requirements emanating from such sources as the Section XI Code for Nuclear Components of the American Society of Mechanical Engineers (ASME), U.S. Nuclear Regulatory Commission (NRC) regulations, and the plant's own technical specifications. While these requirements incorporate considerable experience and engineering judgment, they do not explicitly incorporate "risk-based" information, such as component failure rates, nor do they explicitly factor in the consequences of failure on systems; particularly, safety-related systems. This suggests that some portion of the effort now spent on IST for less risk-significant components is unnecessary, and that more risk-significant components receive less than optimal attention. Consequently, replacing current IST programs with testing strategies that explicitly link the level of attention to failure likelihood and consequence should achieve more safety at lower cost.

The ASME is supporting the investigation and development of risk-based testing methods and tools through a collaborative industry, academic, and regulatory effort. The Westinghouse Owners Group (WOG), a sponsor of this collaborative research, initiated two demonstration projects, one for pumps and another for check valves, with the intent of refining these tools and demonstrating their effectiveness. A full description of this work is beyond the scope of this paper. Instead, this paper focuses on the methods used to incorporate expert judgment into measures of test effectiveness and component reliability in

the following manner: (1) the risk-based IST process is briefly described so as to establish the crucial role played by the test effectiveness and reliability measures, (2) the expert judgment elicitation process is discussed in the context of risk-based methods, (3) the use of expert judgment in the check valve study is described from the perspective of illustrating the challenges posed by certain known cognitive biases and by the requirement to make the most effective use of all available information, and (4) efforts to deal with these challenges in the subsequent pump study are reviewed and illustrative results are presented. These results are in the form of probabilities of (non-) detection and implied changes in failure rates for alternative test strategies. Finally, some conclusions are presented.

RISK-BASED IST: AN OVERVIEW

Figure 1 illustrates the prototypical risk-based IST process developed by the ASME Research Task Force on Inservice Testing and adopted for use in WOG applications. The process starts with a risk ranking of all components in the targeted system. The development of this ranking begins with the use of probabilistic safety assessment (PSA) models to calculate risk importance measures (i.e., "Risk Achievement Worth" and "Fussell-Vesely Importance") for each component. These metrics measure the "at-power" impact of the component function on the plant's core damage frequency (CDF). The next step supplements the PSA model with information on non-modeled components, shutdown risk, etc. An expert panel reviews information from all sources and finalizes the ranking of all relevant components with regard to their safety significance, sorting components into two groups: "More Safety-Significant" and "Less Safety-Significant."

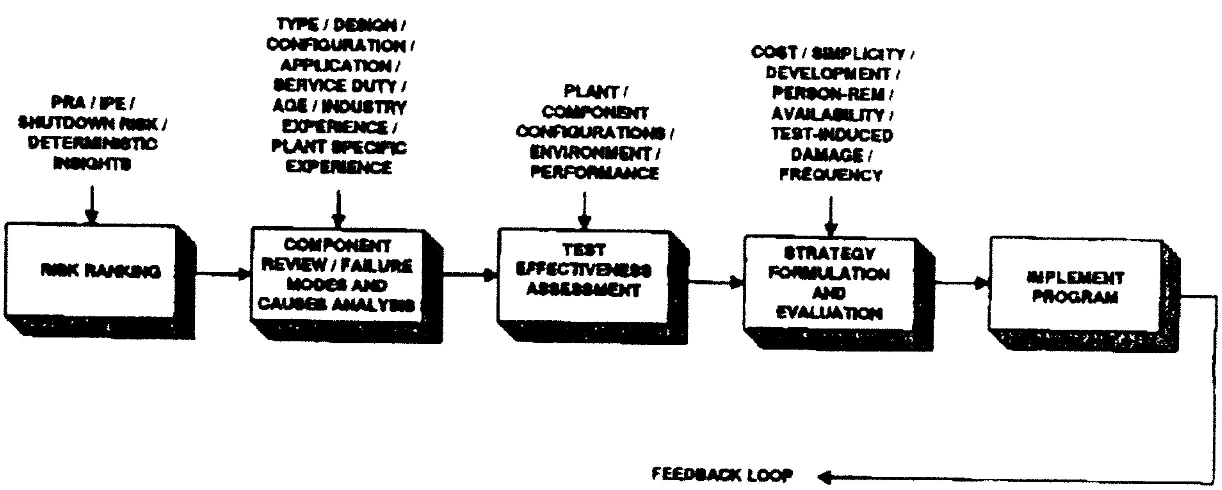

Figure 1. Overall Risk-Based IST Process
(Adapted from ASME Research Task Force – ASME, 1996)

The second major stage analyzes component failure modes and causes. This starts with a review of the component types and specific design features, together with their installation, application, and maintenance history (i.e., all of the factors that might effect component reliability), and then proceeds to study the predominate failure modes and causes for the More Safety-Significant and selected Less Safety-Significant components. The typical output of this stage includes an exhaustive list of potential failure modes and causes and, when available, historical data on the distribution of historical failures across these causes. Such historical data may come from industry data bases such as the Nuclear Plant Reliability Data System (NPRDS), the Nuclear Industry Check Valve (NIC) data base, and from plant-specific data. An additional output of this stage is a comprehensive list of the factors that should be considered when assessing test effectiveness. This list will be used later to condition the expert elicitations.

There are a number of alternative tests available, and it is a matter of common knowledge that they are not all equally effective at detecting each and every potential failure mode and cause. Thus, knowing *how and when* to test can be as important as knowing what to test. The goal of this stage is to provide measures of the relative or absolute effectiveness of the existing and emerging tests that might be employed. Ideally, these measures will combine information from the "failure modes and causes" stage with all available objective and judgmental knowledge on: (1) the frequency of failure causes, (2) the ability of alternative tests to detect such causes, and (3) component characteristics and operating conditions.

As a practical matter, plants will typically apply multiple tests at multiple time intervals, i.e., at different frequencies, such as Test A quarterly and Test C annually. Thus, we define an *IST strategy* as some combination of test types and test frequencies defined for one or more specific components. The goal of the final or "Strategy Formulation and Evaluation" stage is to develop alternative testing strategies and rank those that maintain or improve the level of safety while reducing cost. The Value-Impact Analysis methodology employed in NRC-sponsored studies [NRC (1995)] to evaluate cost versus benefit of proposed regulatory changes can be adapted to the task of quantifying cost and safety impacts of the alternative strategies. The deliverable is a set of recommended inservice testing strategies for the components of interest together with estimates of the safety and economic benefits that will accrue to the plant as a result of implementing the recommendations.

The ultimate goal of a risk-based process for optimizing IST programs is to develop and rank a set of alternative testing strategies for each of the More Safety-Significant and selected Less Safety-Significant components that:

1. insures that components in the IST program will perform their required safety functions
   - shut reactor down to safe shut-down condition
   - maintain reactor in safe shut-down condition
   - mitigate the consequences of an accident
2. minimize life cycle testing cost

3. integrates and balances all available knowledge
   - industry data bases
   - individual IST engineer expertise
4. takes a whole plant perspective
5. provides defensible conclusions
   - quantitative measures of safety risk
   - life cycle whole program testing cost
   - elicits expert judgments that the experts are willing to defend
6. permits individual utilities to incorporate their own level of risk aversion

USING EXPERT JUDGMENT TO ASSESS TEST EFFECTIVENESS

Expert judgment elicitation is recognized as an integral part of the methodology used in the PSAs [NUREG/CR-4550]. In these studies, the use of expert judgment has been extended beyond safety analyses and risk ranking to the formulation and evaluation of testing strategies.

The use of expert judgment has a long history of applications in both nuclear and non-nuclear industries. The branch of Operations Research that originated the formalized elicitation of expert judgment more than thirty years ago is known as Decision Analysis (DA). Combining elements of Probability Theory, Statistical Decision Theory, Game Theory, Systems Engineering and Cognitive Psychology, DA explicitly models the uncertainty and risk that complicate decision making. DA includes tools for specifying and prioritizing goals, generating and screening alternative strategies, defining decision makers' preferences among possible outcomes, valuing tradeoffs across multiple objectives, identifying and quantifying uncertainties affecting risk, and modeling the consequences of these uncertainties. Thus, DA supplies a comprehensive framework for communicating the strategic implications of technical concerns to management.

Why Use Expert Judgment?

The informal use of expert judgment is by no means new. In the guise of "engineering judgment" it has been an essential component of all technical disciplines. There are many situations that we can simply never "know". Take component failure rates for example — for many plant components, there is no way to set up a repeatable, well-designed experiment to gather statistics on component reliability in extreme events. Thus, engineering (expert) judgment is, of necessity, used whenever:

- experimental data is unavailable or unobtainable,
- available data is incomplete, suspect or not representative,
- qualifications or conditions must be considered to properly interpret available data,
- available data does not adequately capture the current "state of knowledge",

among other situations. Indeed, these uses are so commonplace in engineering that they are not explicitly recognized as expert judgment — rather, their use is implicit in most technical decision-making.

To develop and select testing programs that balance risk and consequence, decision makers must integrate technical, regulatory, economic, and financial information. As described in the previous section, engineering analyses of design and performance issues must be combined with the results of PSA's, failure mode and cause analysis (FMCA), and

other safety analyses within the context of the current regulatory climate, while realistically acknowledging economic and financial constraints on the utility.

Lacking formal methods to explicitly model the uncertainty inherent in expert judgment, current IST strategies were developed with a good deal of conservatism, e.g., "stroke every check valve quarterly". Not only does this approach strictly subordinate cost control objectives to safety maximization objectives, it does not guarantee that the most risk-significant components receive optimal attention. That is, we cannot show that current IST strategies maximize plant safety.

Risk-based methods address the issue of identifying and ranking components in order by their contribution to overall plant safety. Decision analytic methods add the necessary additional tools for evaluating test effectiveness and choosing optimal IST strategies. Both risk-based methods and Decision Analysis require the use of expert judgment.

The issue is then, not should we or shouldn't we use expert judgment, but rather, given that we must use expert judgment, how we can use it reliably while avoiding the unquantified conservatism of traditional implicit approaches. These are precisely the topics that Cognitive Psychologists and Decision Science Theorists have been addressing since the 1950's [Kahneman (1974), Spetzler (1975)]. The remainder of this section briefly discusses the principles of expert elicitation.

The Principles of Expert Elicitation

First, we must recognize that uncertainty is inherent in all important decisions. This is true almost by definition, for if there is no uncertainty, the decision is usually so obvious that there is effectively no decision. Qualitative characterizations of uncertainty are ambiguous. One empirical study found that the phrase "a good chance" meant anything from 1-chance-in-4 to a 96% chance to different people [Lichtenstein (1967)]. The very use of numerical probabilities to illustrate this ambiguity shows that the right way to unambiguously characterize uncertainty is to use probability. Thus, the first principle of expert judgment elicitation for decision-making is that uncertainty in expert judgment must be explicitly recognized and assessed quantitatively using probability.

Introducing the use of probability raises the issue of objective versus subjective data. Systematic methods for modeling decision-making are criticized both for being too subjective and not subjective enough. On one hand, we are told that the recommendations of decision analysis models are somehow suspect because they are not based on objective 'hard' data but rely instead on subjective expert judgment. This is a red herring. First, objective data is used whenever available. Most of the time, it isn't. Second, all technical decisions rely on subjective judgment, whether implicit and qualitative, or explicit and quantitative. As one theorist has observed, "there is no more subjective decision than the determination of what is and isn't objective data". We still have decisions to make. The issue is not 'objective versus subjective', because objective is not an option — rather it is 'rational versus arbitrary' and 'comprehensive versus ad hoc'. On the other hand, formal models are criticized for lacking the subjective subtleties that pervade real decision-making. Since probability assessments capture both the real uncertainties in the data and the experts' confidence in their

judgment, 'soft' issues that are normally left out of models can be addressed.

Thus, a second principle of expert elicitation is that probability assessments should be embedded within a comprehensive, systematic process that rationally structures the uncertainties requiring expert judgment, motivates the experts to participate wholeheartedly, and conditions the assessments to minimize the effects of bias. For example, all uncertainties to be assessed should pass the *clarity test*. Imagine a clairvoyant who has perfect knowledge of all events. Could this clairvoyant tells us the outcome for every uncertainty being assessed? Note that clairvoyants can only answer unambiguous questions requiring no interpretation or judgment. Thus, asking whether a component is safe or unsafe would fail the clarity test because it requires the clairvoyant to make assumptions and judgments about what is and isn't safe. Asking instead for the failure rate under a specified testing strategy would pass the clarity test. Applying this simple test helps to insure that experts are not required to synthesize too many different factors in their heads before expressing their judgment.

A third principle of expert elicitation is that uncertainties should be assessed in a way that mitigates and minimizes the effect of bias. Bias is a systematic, often predictable discrepancy between what we say and what we know. Cognitive Psychologists have developed a considerable body of empirical data describing the types of bias we exhibit in making judgment under uncertainty [Kahneman (1982)]. Two basic types of bias have been identified. *Motivational bias* occurs when experts' judgments do not reflect their conscious beliefs, e.g., when their response is motivated by perceived rewards or punishments. *Cognitive bias* occurs when experts' conscious beliefs do not reflect all of their knowledge.

Several approaches can be employed to avoid motivational bias. First, motivate the expert to encourage truthful judgment by explaining the importance of the assessment. Frame assessments so that the expert is communicating knowledge, rather than setting objectives or goals. Decompose the problem into several assessments so that the impact of any one assessment in obscured.

Cognitive bias can be counteracted. Start by drawing out unstated assumptions. Ask the experts to construct scenarios leading to extremes and use the availability of this information to counteract the tendency to anchor on current estimates. Explore the problem from both the specific and general perspective to insure a proper balance of generic industry experience and plant-specific experience. Cognitive bias may be introduced by the way in which information is presented. Apply the clarity test to insure that questions are properly framed. Allow the experts to use the most natural scale for each uncertainty.

Expert judgment can be elicited from individuals or collectively from a team. Individual assessment works well if all experts have common knowledge on the subject of interest, or if one expert dominates the field. However, for risk-based IST, no one person is an expert in all of the relevant factors. For example, synthesizing data on failure rates and consequences with a practical knowledge of testing and familiarity with advanced methods could require three or more experts. Therefore, the team approach is recommended for risk-based IST.

Anytime a group approach is used, the use of a moderator or facilitator is advisable. A facilitator focuses on guiding the elicitation process and should not have a stake in the conclusions. He or she insures that no one expert dominates the discussion, and acts as a "devil's advocate", pushing the experts to consider all of the factors. A facilitator trained in the methods of Decision Analysis monitors the elicitations to insure that the questions have been properly framed and that biases are being addressed.

Expert judgment does not consist solely of probabilistic assessments of uncertainties, but includes the assumptions and reasoning used to reach conclusions. These should be documented to provide an "audit trail". The examples below show forms developed to document the process by which judgments were reached.

Probabilistic methods for quantitatively encoding expert judgment have been successfully used for over 30 years. They have a strong basis in Cognitive Psychology. There is a considerable body of evidence showing that, if you follow the rules, quantitative evaluation of expertise can be accurately assessed. However, they may require a considerable commitment of time and resources to do it right.

TEST EFFECTIVENESS ASSESSMENT FOR CHECK VALVES

The check valve pilot plant application at Southern Nuclear's Vogtle Units 1 and 2 focused on developing check valve inservice test programs for the emergency core cooling system (ECCS). The risk ranking process described above identified seven of 89 check valves as More Safety-Significant (results are fully described in WCAP-14358). The seven More Safety-Significant and selected Less Safety-Significant valves were then subjected to a detailed component review, failure modes and causes analysis, and test effectiveness evaluation.

A panel of experts was convened to identify the most prevalent causes for specific valves and environments for the failure modes of importance from the risk-ranking. As always, the composition of the expert panel is critical. In this case, the panel included plant and utility maintenance engineers as well as recognized industry experts familiar with check valves and testing methods. Using information from the NIC data base, eight generic failure causes were identified (normal wear, improper installation, foreign materials, design, manufacturing or assembly, abnormal wear, human/procedural error, erosion-corrosion/cracking, missing parts during operation). Next, the same panel was asked to assess the likelihood that each failure cause would be the dominant cause should the specified valve be found in a failed state, considering the valve's failure mode, characteristics and history. The experts were required to provide only qualitative assessments on each failure cause in terms of "likely, "unlikely," and "possible," with modifiers, as shown in Table 1.

Table 1
Section 3: Failure Cause Ranking

Unit Name: *Vogtle Units 1 and 2*
Component ID: *1(2)1204U4026, 027, 028 & 029*
Failure Mode: *Fail to Open (Fails Closed)*

FAILURE CAUSE (* TIME/AGE RELATED FAILURE)	RANK
normal wear*	unlikely
abnormal wear*	unlikely
misapplication or improper installation	unlikely
human/maintenance error or procedure problems	likely
foreign materials	unlikely
erosion, corrosion, or cracking*	unlikely
initial design, manufacturing, or assembly	unlikely
missing parts* (during operation)	unlikely

Section 4: Test Effectiveness Rating

routine operator inspection		forward flow				reverse flow		leakage rate				disassemble & inspect								acoustic[2]* *		magnetic flux[2]**		ultrasonic[1]		tempera-ture		radio-graphic[2]	
		full pressure		partial pressure				w/o THT		w/- THT		w/o PAT w/o THT		w/- PAT w/o THT		w/o PAT w/- THT		w/- PAT w/- THT											
SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F	SD	F
		M	H									H	H	H	H	H	H	H	H	M	H	M	H					M	H

Test Effectiveness Rating: **H** (high) ⇒ probability of detection > 75%; **M** (medium) ⇒ 75% > probability of detection > 25%; **L** (low) ⇒ probability of detection > 25%

Abbreviations: THT — time history trending PAT — post assembly testing SD — significant degradation F — failure
1 doesn't work well due to thickness & stainless steel
2 nonintrusive can also determine failure at time of test (first time or one time)
** Acoustic and magnetic flux methods performed together.

This invariably led to the identification of "dominant" causes for each valve, and these dominant causes were then used to condition the subsequent test effectiveness assessments.

Essentially all existing and emerging valve tests were exhaustively reviewed (WCAP-14358 contains this useful review). Ten tests were chosen for evaluation. Individual tests were combined with other tests and specified test frequencies to define a test *strategy*. Table 2 illustrates the "strategy table" used to elicit a set of alternative strategies for a single valve. Thus, Test A is defined as a "forward flow-full pressure" test applied at each refueling outage. Alternatively, we might consider applying the same test only at a ten year interval (i.e., Test C). The table identifies 10 candidate strategies for this one valve (similar tables were constructed for the other valves of interest). Tests either create an artificial demand to see if the valve actually works or look for conditions that could lead to failure, using either intrusive (i.e., disassembly) or non-intrusive methods. The check valve study defined test effectiveness as the "level of confidence" held by the IST engineer that, should a real demand occur anytime over the operating interval before the next test, the valve will perform its intended function.

This level of confidence was originally elicited from the component expert team as the *probability* that the valve will not fail if there is a real demand. However, we were unable to achieve a consensus for the very small probabilities (on the order of 10^{-4}) associated with these highly reliable components. This illustrates the well known problem associated with applying decision analytic probability elicitation methods [Spetzler (1975)] to very low frequency events. Consequently, it was proposed that, as a cognitive aid, these failure rates be assessed using an arbitrary scale of 1 to 9, with 5 being equivalent to the current strategy's failure rate and 1 and 9 representing a range of failure rates from the "worst it can get" to the "best it can get." Table 3 illustrates the check valve study's evaluation of the test strategies defined in Table 2.

TEST EFFECTIVENESS ASSESSMENT FOR PUMPS

For the pump demonstration project, using Carolina Power and Light's Shearon Harris plant, the risk ranking process identified 12 of the plant's 27 pumps currently in the IST program as being more-safety-significant. The detailed component review, failure modes and causes analysis and test effectiveness evaluation then focused on two pump types, vertical single stage (VSSP) and horizontal multistage pumps (HMSP). At Shearon Harris, these pump types are found in the Residual Heat Removal (RHR) and Containment Spray (CS) systems for the vertical pumps and the Motor-Driven Auxiliary Feedwater (MDAFW) and charging/safety injection (C/SI) systems for the horizontal pumps. The boundary considered for these pumps included the pump itself, the motor driver, and electrical control devices (results are fully described in WCAP-14571).

Table 2
Section 5: IST Strategy Definition

Unit Name : _Vogtle Units 1 and 2_
Component ID : _1(2)1204U4026, 027, 028 & 029_
Failure Mode : _Fail to Open (Fails Closed)_

	routine operator inspection	forward flow		reverse flow	leakage rate		disassemble & inspect *				acoustic[1]	magnetic flux[1]	ultrasonic	temperature	radio-graphic
test frequency		full pressure	partial pressure		w/o THT	w/- THT	w/o PAT w/o THT	w/- PAT w/o THT	w/o PAT w/- THT	w/- PAT w/- THT	w/- THT	w/- THT	w/- THT	w/- THT	w/- THT
Strategy: Current															
quarterly															
ea refueling outage		A													
every 5 years															
every 10 years		C													
other (baseline)		B													
Strategy: Nonintrusive															
quarterly															
ea refueling outage		F									D/F	D/F			
every 5 years															
every 10 years		E									E	E			
other															
Strategy: Disassemble															
quarterly															
ea refueling outage							G	H							
every 5 years															
every 10 years															
other															

For each strategy, define the strategy by identifying which test will be performed at each frequency, then indicate your *relative level of confidence* in the strategy on the next page.

* PAT - flow test

[1] Acoustic and magnetic flux methods performed together.

Table 3
Section 6: IST Strategy Evaluation

Unit Name : *Vogtle Units 1 and 2*
Component ID : *1(2)1204U4026, 027, 028 & 029*

Strategy	Score 1	2	3	4	5	6	7	8	9	Notes
Current - A					X					
Current - B	X									Concern that spring may degrade
Current - C					X					
Nonintrusive - D							X			Less effective at longer intervals because there is less data to trend
Nonintrusive - E								X		
Nonintrusive - F									X	
Disassemble G	X									Issue because parts fall out during disassemble, more opportunities for error, internals hung from bonnet and welded bonnet
Disassemble H						X				

Scoring: 1 = worst possible (highest failure rate), 5 = current practice (nominal failure rate), 9 = best achievable (lowest failure rate).

The pump test effectiveness analysis embodies the "competing cause" framework suggested by Siu and Hartley (1995). Specifically, the analysis uses two arrays: (1) a "Probability of Occurrence by Cause" column (v), and (2) a "Test Effectiveness by Cause" matrix (m). The probability of detection for a test is determined as follows. Let $m(ji)$ = the jth test's effectiveness in detecting the ith cause, and $v(i)$ = conditional probability that if significant degradation has occurred the ith cause code is the reason for the failure, then the probability that the jth test will detect significant degradation (given that it exists) is:

$$p(j) = \sum_i m(ji)v(i) \qquad (1)$$

In words, multiply each probability of occurrence by its corresponding conditional probability of detection and then sum across all causes to get the probability that a test will detect significant degradation from whatever source. The following paragraphs described the practical implementation of this model.

Historical data exists on pump failures just as for check valves with the exception that the check valve data from NPRDS has been filtered for use in the NIC data base. However, in the check valve study, we were able to identify only one relevant failure for some very reliable – More Safety-Significant check valves at Vogtle. Moreover, no incidents had been reported for most of the candidate failure causes for these valves. The pump study team reviewed the NPRDS failure information for all WOG plants, 1974 – 1995, and Licensee Event Reports (LERs) for Westinghouse plants from 1984 to 1995. After filtering to insure consistency with the configurations and boundaries described above, a data base of 907 records on failures remained. Only eight of these failures were from Shearon Harris. We took a "Bayesian" perspective that this overwhelming background evidence (i.e., "prior") of industry experience should dominate the relatively slim plant-specific experience; that is, the failure source distributions derived from this industry data base were assumed to apply to the plant. Failures were classified into one of 24 "cause codes" (19 mechanical and 5 electrical) so that the percentage of all failures attributable to each cause code could then be calculated for each of the four pump systems (i.e., RHR, CS, MDAFW, and C/SI). Denote this result as the Probability of Occurrence by Cause column (v). In other words, each element of this column represents the conditional probability of occurrence for each cause code listed for each pump system. These probabilities are based upon "objective" data (leavened, as usual, with judgment about what to leave in and what to leave out).

The study identified 14 current and emerging individual pump tests, ranging from Test A = 1983 ASME Code Version to Test P = Proximity probe. No comparable data exists on the ability of each of these tests to identify above causes and, consequently, expert judgment was required. A panel of experts consisting of members of the ASME OM-6 Working Group on Pumps and one CP&L IST engineer was formed for the purpose of evaluating the effectiveness of the various tests on the two pump types of interest (i.e., VSSP and HMSP). For each cause code (e.g., MTB = Breaker/Control System Failed), the panel was told to assume that the cause in question was in fact the dominant cause of a postulated "significant degradation," and then asked to

assign a test to one of four effectiveness categories:

- High = Greater than 75% probability of detecting the indicated cause.
- Medium = Between 10% and 75% probability of detecting the indicated cause.
- Low = Between 0 and 10% probability.
- Null = No credible probability of detecting the cause (usually, because the relevant quantity is not measured).

Once this sorting task was completed, more precise probabilities were elicited from the same experts for the Medium category using probability elicitation techniques from decision analysis. Mid-category probabilities were assigned to the High and Low categories and a zero probability was attached to the null category. This effort produced a matrix for each pump type with the 14 tests as column headings, 24 cause codes as rows and each element of the matrix showing the conditional probability that the test at the top would detect the cause at the left (given that the latter is active). In short, the result is a Test Effectiveness by Cause matrix (*m*) based entirely on encoded engineering judgment. An example is provided in Table 4.

Substituting the vector *v* and the matrix *m* into equation (1) produces a column of estimated detection probabilities that incorporate the best available objective and subjective information; or, in other words, they embody essentially "all we know." The effectiveness of a test is also a function of how often it is applied. Since a test strategy may call for a combination of different tests, each applied at a different frequency, we need to first normalize the tests with respect to frequency. The problem may be stated as follows. Suppose we are to apply a test with probability of detection = p and probability of nondetection = 1-p on a quarterly basis. What then is the equivalent <u>annual</u> probability of nondetection? The answer is obtained by taking 1-p to the fourth power. More generally, APND = the annualized probability of nondetection is,

$$APND(jf) = (1 - p(j))^{f} \qquad (2)$$

where p = a test probability of detection from equation (1) and f = annual frequency of application (thus, for example, a biennial application would have an annual frequency of 0.5 and f would equal 1, 2, 4 and 12 for annual, biannual, quarterly, and monthly, respectively). The annualized probability of nondetection was calculated for each pump system. Table 5 contains illustrative results for the RHR pump system. This table illustrates the impact that the frequency has on test effectiveness. For example, Test A applied monthly is about 17 times more effective than if applied biennially. Note also that, controlling for frequency, tests differ significantly in their effectiveness. Test G, the Visual exam, is the most effective for RHR and is about 34 times more likely to detect significant degradation than is the least effective test (i.e., J) when both are applied on a quarterly basis. The difference between tests becomes even more striking for lower frequencies (for example, G is 91 times more effective than J when the two tests are applied biennially). These differences reflect the assessed relative abilities of the tests to detect the various failure causes and the relative likelihoods associated with those causes.

TABLE 4 TEST EFFECTIVENESS RATING FOR VERTICAL SINGLE STAGE PUMPS (SIGNIFICANT DEGRADATION)					
Test Identifier	**Test A**	**Test B**	**Test C**	**Test D**	**Test E**
	83	**89**	**94A**	**94B**	**94C**
FAILURE CAUSES					
ELECTRICAL					
MTB - Vibration, Norm Wear, Aging	N	N	N	N	N
MTP - Breaker/Control System Failed	N	N	N	N	N
MTS - Stator Insulation Ground/Degrad	N	N	N	N	N
MMX - Motor Maint/Installation	N	N	N	N	N
MTU - Unknown	N	N	N	N	N
MECHANICAL					
PCS - Mech Seal Leakage/Failure	N	N	N	N	N
PAA - Pump Aux Systems Failure	N	N	N	N	N
PCG - Pump Casing Gasket Leakage	N	N	N	N	N
PAO - Oil Leakage: Fittings, Seals, etc.	H	N	N	N	N
PRB - Pump Bearing Wear or High Tem	L	L	L	N	L
PCP - Pump Packing Leakage	N	N	N	N	N
PRS - Shaft Wear, Out of Balance, Exc	L	M(40)	M(40)	N	M(40)
PUN - Unknown	N	N	N	N	N
PRR - Mech Binding	N	N	N	N	N
PAC - Misaligned Shaft Coupling	N/A	N/A	N/A	N/A	N/A
PRI - Worn/Broken Impeller, Low Clear	H	H	H	M(30)	H
PHV - Vapor Binding	N	N	N	N	N
PCL - Pump Casing Leakage	N	N	N	N	N
PCW - Casing Non-rotating Parts Wear	L	L	L	L	M(30)
PHF - Low Flow or Discharge Pressure	H	H	H	H	H
PRV - High Pump Vibration	H	H	H	N	H
PCH - Overheated Pump Packing	N	N	N	N	N
PAB - Pump Frame Breakage, Cracks, etc.	N	N	N	N	N

TABLE 4 (cont) TEST EFFECTIVENESS RATING FOR VERTICAL SINGLE STAGE PUMPS (SIGNIFICANT DEGRADATION)					
Test Identifier	Test F	Test G	Test H	Test J	Test K
	Vib Spectral	VISUAL	MANUAL	MOTOR PWR	MEGGER MOTOR
FAILURE CAUSES					
ELECTRICAL					
MTB - Vibration, Norm Wear, Aging	H	L	N	N	N
MTP - Breaker/Control System Failed	N	N	N	N	N
MTS - Stator Insulation Ground/Degrad	N	L	N	N	H
MMX - Motor Maint/Installation	N	N	N	N	N
OMTU - Unknown	N	N	N	N	N
MECHANICAL					
PCS - Mech Seal Leakage/Failure	N	H	M(35)	N	N
PAA - Pump Aux Systems Failure	N	L	L	N	N
PCG - Pump Casing Gasket Leakage	N	H	M(35)	N	N
PAO - Oil Leakage: Fittings, Seals, etc.	N	H	M(60)	N	N
PRB - Pump Bearing Wear or High Tem	H	N	N	N	N
PCP - Pump Packing Leakage	N/A	N/A	N/A	N/A	N
PRS - Shaft Wear, Out of Balance, Exc	H	N	N	L	N
PUN - Unknown	N	N	N	N	N
PRR - Mech Binding	M(50)	L	M(50)	L	N
PAC - Misaligned Shaft Coupling	N/A	N/A	N/A	N/A	N
PRI - Worn/Broken Impeller, Low Clear	H	N	N	M(20)	N
PHV - Vapor Binding	L	N	N	N	N
PCL - Pump Casing Leakage	N	H	M(30)	N	N
PCW - Casing Non-rotating Parts Wear	L	N	N	M(20)	N
MTB - Vibration, Norm Wear, Aging	-	-	-	-	
PHF - Low Flow or Discharge Pressure	N	N	N	M(40)	N
PRV - High Pump Vibration	H	L	N	N	N
PCH - Overheated Pump Packing	N	N/A	N/A	N/A	N
PAB - Pump Frame Breakage, Cracks, etc.	L	L	L	N	N

TABLE 4 (cont) TEST EFFECTIVENESS RATING FOR VERTICAL SINGLE STAGE PUMPS (SIGNIFICANT DEGRADATION)				
Test Identifier	**Test L**	**Test M**	**Test N**	**Test P**
	BUMP	**LUBE OIL**	**MCSA**	**PROXIMITY PROBE**
FAILURE CAUSES				
ELECTRICAL				
MTB - Vibration, Norm Wear, Aging	N	M(20)	H	N
MTP - Breaker/Control System Failed	N	N	N	N
MTS - Stator Insulation Ground/Degrad	N	N	L	N
MMX - Motor Maint/Installation	N	N	L	N
OMTU - Unknown	N	N	N	N
MECHANICAL				
PCS - Mech Seal Leakage/Failure	M(35)	M(35)	N	N
PAA - Pump Aux Systems Failure	L	L	N	N
PCG - Pump Casing Gasket Leakage	M(35)	M(35)	N	N
PAO - Oil Leakage: Fittings, Seals, etc.	M(60)	H	N	N
PRB - Pump Bearing Wear or High Tem	N	H	N	N
PCP - Pump Packing Leakage	N/A	N/A	N/A	N/A
PRS - Shaft Wear, Out of Balance, Exc	N	N	N	N
PUN - Unknown	N	N	N	N
PRR - Mech Binding	N	N	H	N
PAC - Misaligned Shaft Coupling	N/A	N/A	N/A	N
PRI - Worn/Broken Impeller, Low Clear	N	N	M(20)	N
PHV - Vapor Binding	N	N	N	N
PCL - Pump Casing Leakage	M(30)	M(30)	N	N
PCW - Casing Non-rotating Parts Wear	N	N	M(20)	N
MTB - Vibration, Norm, Wear, Aging	N/A	-	-	N/A
PHF - Low Flow or Discharge Pressure	N	N	N	N
PRV - High Pump Vibration	N	N	N	H
PCH - Overheated Pump Packing	N/A	N/A	N	N
PAB - Pump Frame Breakage, Cracks, etc.	L	L	N	N

TABLE 5
MID-VALUE OF ANNUALIZED PROB OF NONDETECT BY TEST FREQ-RHR

Test Identifier	Test Freq Identifier					
	Biennial	Annual	Biannual	Triannual	Quarterly	Monthly
TEST A	0.9618	0.925	0.8556	0.7915	0.7321	0.3924
TEST B	0.9831	0.9665	0.9341	0.9028	0.8726	0.6644
TEST C	0.9844	0.969	0.939	0.9099	0.8817	0.6853
TEST D	0.9955	0.991	0.9821	0.9732	0.9645	0.8972
TEST E	0.9844	0.969	0.939	0.9099	0.8817	0.6853
TEST F	0.9124	0.8325	0.6931	0.577	0.4803	0.1108
TEST G	0.6823	0.4655	0.2167	0.1009	0.04696	0.0001035
TEST H	0.8812	0.7765	0.605	0.4682	0.3636	0.04805
TEST J	0.9965	0.993	0.9861	0.9792	0.9723	0.9192
TEST K	0.9726	0.946	0.8949	0.8466	0.8009	0.5137
TEST L	0.884	0.7815	0.6107	0.4773	0.373	0.0519
TEST M	0.8388	0.7035	0.4969	0.3482	0.2449	0.0147
TEST N	0.9492	0.901	0.8118	0.7314	0.659	0.2862
TEST P	1	1	1	1	1	1

Test Names and Descriptions

- Test A – 1983 ASME Code Version
- Test B – 1989 ASME Code Version
- Test C – 1994 ASME Code Version for Group A pumps
- Test D – 1994 ASME Code Version for Group B pumps
- Test E – 1994 ASME Code Version, Comprehensive Test
- Test F – Vibrational Spectral Analysis (full sweep)
- Test G – Visual Inspection of an operating pump
- Test H – Manual Rotation of the pump shaft
- Test J – Motor power test (determine gross power, amperage, and voltage)
- Test K – Megger Motor Test
- Test L – Burnup test
- Test M – Lube oil analysis with wear particle analysis
- Test N – Motor current signature analysis
- Test P – Proximity Probe

The visual test of an operating pump does well only on detection of various leaks – but these leaks account for more than 50% of all failures across all systems. Test J, the Motor Power test, does moderately well at finding broken impellers and low flow or discharge pressure; causes that account for less than 5 percent of the failures.

The following alternative *test strategies* were defined by selecting combinations of pump tests and frequencies (test frequency in parenthesis):

1. *Current Code Strategy* = Test C (quarterly)
2. *Comprehensive Strategy* = Tests E (biennial), M (biennial), D and G (the latter two are quarterly except when E and M are used. Hence, D and G are applied quarterly the first year and triannually the second year)
3. *Visualonly Strategy* = Test G (quarterly)
4. *Base (Harris Strategy)* = Test A (quarterly) and Test M (quarterly)
5. *Vibraspec Strategy* = Test F (quarterly) and Test M (quarterly)
6. *MCSA Strategy* = Test N (quarterly) and Test M (quarterly)
7. *Techspec Strategy* = Tests C (monthly), F (monthly) and G (monthly)
8. *No Testing Strategy* = No inservice tests

The first strategy represents compliance with a portion of the ASME 1994 Code, while (2) incorporates the newer "comprehensive" pump test changes to ASME OM Code (described in Hartley (1994)). Strategy 7 is inspired by an industry-wide survey performed at the beginning of the project which found a number of plants applying the indicated tests as per technical specifications on a monthly basis. The Motor Current Signature Analysis (MCSA) and Vibration Spectral (Vibraspec) strategies were suggested during the test effectiveness elicitation session, while Test G's high assessed effectiveness suggests the Visualonly Strategy. The Base Strategy is the testing assumed to be actual current practice by the Shearon Harris PSA and here serves as the base against which other strategies are compared. Finally, a No Test Strategy serves as a bounding case and a potentially viable option for less safety-significant pumps.

The annualized probability of nondetection for a strategy is determined by multiplying the appropriate individual test nondetection probabilities from Table 5 and analogous tables for the other pump systems. For example, the Techspec Strategy is defined as Tests C, F, and G, all applied on a monthly basis. Thus, for the strategy's annualized probability of nondetection for an RHR pump, go to Table 3 and select the appropriate values from the Monthly column and the indicated test rows (i.e., .6853, .1108, and .001). The product of these three individual test probabilities (= 7.9E-06) is the Techspec Strategy's annualized probability of nondetection for an RHR pump. Table 6 holds the results for all of the strategies as applied to each of the four pump systems. Note that a single strategy's effectiveness can vary across pump system. For example, Current Code's probability of nondetection varies from a high of 88% to a low of 49%. Further, this variation can be significant even within a pump type. For example, RHR and CS are vertical single-stage pumps but the Comprehensive Strategy is about 23 times less effective on CS than on RHR pumps. In short, there is a case for differentiating test strategies by pump system. For a specified pump, the frequency of test application can overwhelm other attributes. Thus, the Techspec Strategy is the most effective

TABLE 6
ANNUALIZED PROBABILITY OF NON-DETECTION BY STRATEGY

Strategy	RHR	CS	MDAFW	C/SI
Current Code	8.8E-1	8.2E-1	4.9E-1	7.2E-1
Comprehensive	3.7E-3	7.7E-2	3.3E-2	2.0E-2
Visual only	4.7E-2	2.5E-1	1.73E-1	1.1E-1
Base(Harris)	1.8E-1	3.63-1	6.3E-1	4.7E-1
Vibraspec	1.2E-1	1.4E-1	1.9E-1	2.6E-1
MCSA	1.6E-1	1.8E-1	3.4E-1	4.3E-1
Techspec	7.9E-6	2.4E-4	6.5E-6	1.2E-5
No Testing	1.00	1.00	1.00	1.00

primarily because it is applied monthly while none of the other strategies are applied any more frequently than on a quarterly basis.

The most surprising performance is that of the Current Code. For all pumps under review, except MDAFW, the Current Code is the least effective strategy except for No Testing. For MDAFW, the Current Code is marginally better than Base (Harris) but substantially less effective than its Comprehensive counterpart. The latter is a combination of four tests applied at frequencies ranging from biennial to quarterly. One of the tests in the Comprehensive test strategy is the Visual, applied quarterly. The latter test is the sole ingredient in the Visualonly Strategy which does quite well; ranking third in all but the Less Safety-Significant CS system. Obviously, visual testing is an important contributor to the Comprehensive Strategy, which is the second most effective strategy in every system.

LINKING TEST EFFECTIVENESS TO PLANT SAFETY

Whether the measured differences in test strategy effectiveness have any practical importance depends on the corresponding impact on plant safety variables. The latter, in turn, are a function of the impact on the affected component's failure rate. An observed component failure rate per year, r, can be expressed as (adapted from Siu and Hartley):

$$r = r(iat) \times APND(bs) \quad (3)$$

That is, abstracting from test-induced failures, extraneous shocks and other "nuisance" influences, the failure rate can be expressed as a multiplicative function of the failure rate in the absence of testing, *r(iat)*, and the annualized probability of nondetection for the "base strategy," *APND(bs)* that has been in use up to that point (for example, the base strategy for an RHR pump at Shearon Harris

has, according to Table 6, a probability of nondetection of 1.8E-1). There is an unobservable variable - *r(iat)* - in (3). However, the aforementioned Value-Impact methodology that serves as a framework for the subsequent optimization calculates risk averted (avoided) relative to the current or base strategy. Consequently, only the fractional change in r is needed and, given that r(iat) is unaffected by the choice of strategy, it is apparent from inspection of (3) that the component's fractional change in failure rate, *dr/r* ,will equal the fractional change in the annualized probability of detection, *dAPND(s)/APND(bs),* associated with moving from the base test strategy (bs) to the strategy in question (s). Formally,

$$dr/r = dAPND(s)/APND(bs) \qquad (4)$$

Referring to Table 6, for example, the substitution of the Comprehensive for the Base strategy would yield a (.0037 − .018)/.018 = minus 0.79 or 79% reduction in nondetection and an estimated equivalent reduction in the RHR failure rate. Finally, denote the specific pump system's contribution to plant core damage frequency as "*pump cdf,*" then the reduction, *dpr*, in potential radiation dose to the public that is associated with the reduction in likelihood of pump failure cumulated over the plant's remaining "life" is:

$$\begin{aligned} dpr &= -(dr/r \times pump\ cdf \times total\ dose \times life) \qquad (5) \\ &= -(dAPND(s)/APND(bs) \times pump\ cdf \times total\ dose \times life) \end{aligned}$$

The value for pump cdf is that used in the PSA model and the value for total dose (per release) can be plant-specific (if available) or generic (as in the pump study). The pump study used similar equations to calculate avoided "accidental occupational" and "routine occupational" radiation exposure measures to go with (5). Test effectiveness evaluations are thus formally linked to all measures of plant safety through (5) and its counterparts.

CONCLUDING REMARKS

The check valve and pump projects led to the development of an approach that can serve as a template for future test effectiveness evaluations. The approach features:

- a practical, rigorous approach for combining expert judgment with objective data to quantify test effectiveness in terms of probabilities of detection,
- an appropriate method for combining individual test effectiveness estimates across multiple test frequencies to obtain an estimate of test strategy effectiveness, and
- linkage of test strategy effectiveness to changes in plant safety factors in a manner that effectively integrates PSA results into the analysis.

Current practice implicitly limits the use of expert panels to the risk-ranking steps. In these studies, we have shown that expert panels can also be used to assess the relative likelihood of various failure causes, the effectiveness of alternative testing methods (both current and proposed), and the relative attractiveness of alternative IST strategies. The decision analytic imperative to focus on the "decision to be made" supplies a key ingredient required in optimization.

The approach moves the Risk-Based IST process beyond risk ranking toward optimization of a testing program by differentiating between candidate test

strategies with respect to their effectiveness and then quantifying the safety impact of these differentials. The judicious and rigorous use of expert judgment plays a pivotal role in this transition.

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A Historical Perspective of Risk-Informed Regulation

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PURPOSE

This paper discusses the application of risk management, from a historical perspective to the present, in the regulation of nuclear power generating facilities licensed pursuant to Title 10 of the *Code of Federal Regulations* (CFR), Part 50, "Domestic Licensing of Production and Utilization Facilities," as it relates to the historical underpinnings and the current actions associated with risk-informed inservice testing.

INTRODUCTION

In Federal studies[1], the process of using risk information is described as having two general components: (1) risk assessment - the application of credible scientific principles and statistical methods to develop estimates of the likely effects of natural phenomena and human factors and the characterization of these estimates in a form appropriate for the intended audience (e.g., agency decisionmakers, public); and (2) risk management - the process of weighing policy alternatives and selecting the most appropriate regulatory action, integrating the results of risk assessment with engineering data with social, economic, and political concerns to reach a decision. This paper discusses largely the second component.

BODY

Early Attempts in Risk Management

Throughout the 1960s, the commercial nuclear power industry grew in numbers of reactors and in reactor size. The growth in the size of reactors and the practice of design by extrapolation raised many complex safety issues. A traditional means of protecting the public from the consequences of a nuclear accident was remote siting (i.e., siting the plants well outside populated areas); however, at the same time that reactors were becoming more complex, the Atomic Energy Commission (AEC), predecessor to the NRC, became more receptive to allowing facilities to be sited closer to urban populations. Without the remote siting, the engineered safeguards built into the plants became of greater importance. The functions of the engineered safeguards were (1) to protect the reactor core by removing heat and reducing pressure and (2) to collect and retain radioactive gases and particles released by any accident that might occur. The final line of defense if the engineered safeguards failed was the containment building which enclosed the reactor.[2]

To decrease the likelihood of a major nuclear power plant accident that could threaten public

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

health and safety, the AEC required multiple backup equipment and redundancies in the design of safety systems. It also employed conservative assumptions about the ways in which an accident might damage or incapacitate safety systems when evaluating proposals for licensing reactors. The greatest concern was the potential for a core meltdown and potential release of fission products caused by a loss-of-coolant accident.

Perhaps the first attempt to assess the probability of a core meltdown was made by a special AEC task force established in 1966. The findings of the task force were reported in October 1967 and offered assurances about the improbability of a core meltdown and the reliability of the emergency core cooling systems design, but acknowledging that a loss-of-coolant accident could cause a breach of containment.[3] The report changed the focus of regulating nuclear power plants *from* the reliance on the containment building to restrict any release of radiation from the plant *to* preventing accidents severe enough to threaten the containment such that the public would be protected from a large release of radiation. Such an approach depended heavily on the proper design and functioning of the emergency core cooling systems.[4]

The AEC continued to fund research efforts and tests to assess the potential for accidents and the adequacy of mitigation systems, particularly the emergency core cooling systems (ECCS). Tests conducted at the Idaho Loss-of-Fluid-Tests facility indicated that, in some situation, the ECCS might not work as designed to flood the core with cooling water.[5]

The AEC, and later the NRC, sponsored a Reactor Safety Study to estimate the public risks that could be involved in potential accidents in commercial nuclear power plants of the type then in use (early 1970s). The study was to make a realistic estimate of these risks and, to provide perspective, to compare the nuclear risks with non-nuclear risks which already existed in society.[6] The study presented the estimated risks from nuclear power plant accidents and made the requisite comparison, but it made no judgment as to acceptability of nuclear risks; however, it drew criticism from both inside and outside the NRC that (1) it failed to account for the many paths that could lead to major accidents and (2) the data in the report did not support its executive summary's conclusions that the relative risks of nuclear power were very small. In response to the criticism, in January 1979, the NRC issued a policy statement that withdrew its full endorsement of the study's executive summary.[7] Shortly thereafter, Unit 2 of the Three Mile Island nuclear generating plant[8] experienced a loss-of-coolant accident.

The Three Mile Island 2 Accident

In the early morning of March 28, 1979, a plant trip occurred due to feedwater system problems. A pressure rise in the reactor coolant system caused the pressurizer power-operated relief valve (PORV)[9] to open. After the initial pressure rise, the PORV should have closed; however, unknown to the operators, it stuck open and effectively created a small-break loss of coolant accident (i.e., the initiating event of the accident). In the 142 minutes that followed, the plant systems functioned as designed, except that two critical valves at the discharge of the emergency feedwater pumps were closed when they should have been open. However, there were a series of operator errors that ultimately resulted in boiling in the core and melting of the fuel before cooling water was finally injected and the accident ceased. The accident

indicated that the human element, which had previously been largely ignored in assessing the risk of operating nuclear power plants, was as critical as the design of the engineered safeguards systems. A special President's Commission studied the accident and made a number of recommendations, including initiation of in-depth studies on the probabilities and consequences (on-site and off-site) of nuclear power plant accidents and the consequences of meltdown. The studies were to include a variety of small-break loss-of-coolant accidents and multiple-failure accidents, with particular attention to human failures.[10] The followup-actions required by the NRC were consolidated in NUREG-0737, "Clarification of TMI Action Plan Requirements," November 1980.

Safety Goals Defined

The NRC issued a policy statement entitled "Safety Goals for the Operations of Nuclear Power Plants," on August 21, 1986[11], based on efforts that began as a result of recommendations of the President's Commission on the accident at TMI. The objective of the policy was to establish goals that broadly define an acceptable level of radiological risk. The Commission stated that it believed that the basic statutory requirement of adequate protection of the public was met, but that its regulatory practices could be improved to provide a better means for testing the adequacy of proposed regulatory requirements. The purpose of the policy statement was "to lead to a more coherent and consistent regulation of nuclear power plants, a more predictable regulatory process, a public understanding of the regulatory criteria that the NRC applies, and public confidence in the safety of operating plants." The Commission noted that the risks from the release of radioactive materials from the reactor to the environment may come from normal operations as well as accidents. The Commission also noted that, through its review and preparation of environmental impact assessments, for all plants licensed to operate, it had found that there would be no measurable radiological impact on any member of the public from routine operation of the plant. The "acceptable risks" were given as follows:

- The *qualitative safety goals* are as follows:

  — Individual members of the public should be provided a level of protection from the consequences of nuclear power plant operation such that individuals bear no significant additional risk to life and health.

  — Societal risks to life and health from nuclear power plant operation should be comparable to or less than the risks of generating electricity by viable competing technologies and should not be a significant addition to other societal risks.

- The following *quantitative objectives* are to be used in determining achievement of the above safety goals:

  — The risk to an average individual in the vicinity of a nuclear power plant of prompt fatalities that might result from reactor accidents should not exceed one-tenth of one percent (0.1 percent) of the sum of prompt fatality risks resulting from other accidents to which members of the U.S. population are generally exposed.

  — The risk to the population in the area near a nuclear power plant of

> cancer fatalities that might result from nuclear power plant operation should not exceed one-tenth of one percent (0.1 percent) of the sum of cancer fatality risks resulting from all other causes.[12]

The safety goals do not define the criteria for an adequate level of protection, but rather are meant to provide a useful tool by which to judge regulatory decisions. The safety goals are to be used in conducting cost-benefit analyses for proposed actions and are to be considered as one of several factors in licensing decisions (i.e., they are not meant to be used as a sole basis for a licensing decision).[13]

The Secretary of the Commission issued a Staff Requirements Memorandum on the implementation of the safety goals on June 15, 1990. The memorandum discusses the safety goals in terms of the basic statutory standard of adequate protection, indicating that the Commission, in formulating the safety goals policy, believed that the current regulatory practice ensured compliance with the standard. The memorandum further stated that the Commission believes that "adequate protection" is a case-by-case finding based on evaluating a plant and site combination and considering the body of the NRC's regulations while the safety goals are to be used in a more generic sense and not to make specific licensing decisions. Therefore, the Commission concluded that it was not necessary to create a generic definition of adequate protection, nor was it necessary to amend the safety goals policy to provide a direct relationship between the safety goals and the concept of adequate protection.

Staff Risk-Informed Guidance for Imposing New Requirements

NUREG/BR-0058, Revision 2, "Regulatory Analysis Guidelines of the U.S. Nuclear Regulatory Commission," final report (November 1995), was issued to, in part, reflect the changes in NRC regulations since the previous revision (1984), especially the backfit rule and the policy statement on safety goals, noting that the policy statement presents a risk-informed philosophy to be used by the NRC staff as part of their regulatory analysis process for proposed actions that may have an impact on commercial nuclear power reactors.

The guidance in NUREG/BR-0058 is intended to apply to all mechanisms used by the NRC staff to establish or communicate generic requirements, guidance, requests, or staff positions (e.g., rules, bulletins, generic letters, regulatory guides, orders, standard review plans, branch technical positions, and standard technical specifications). The use of the safety goals is to be early in the process for determining whether or not to proceed with an action. The safety goal evaluation is designed to answer the question as to when a regulatory requirement should not be imposed on a generic basis because the residual risk is already acceptably low, with the intent that such an evaluation could eliminate some proposed requirements from further consideration independently of whether they could be justified on a value/impact basis. The safety goal evaluation is also to serve the purpose of determining whether the substantial added protection standard of the backfit rule is met (i.e., if the proposed safety goal decision criteria are satisfied, it is to be presumed that the substantial additional protection standard

of the backfit rule is met for the proposed action). The safety goal evaluations are to be based upon three broad guidelines: (1) applicable only to safety enhancement for the affected class of plants as guidance (not requirements); (2) implemented in conjunction with the backfit rule criterion on "substantial additional protection;" and (3) should be integrated with related issues under study to avoid piecemeal evaluation of issues. Additionally, the safety goals are intended to balance accident prevention and accident mitigation by recognizing the relatively poor performance of containment and thereby focusing on greater consideration of accident prevention (i.e., issues intended to reduce core damage frequency). The criteria were selected to provide some assurance that the PRA and data limitation and uncertainties, as well as the variabilities among plants, will not eliminate issues warranting regulatory attention.

Individual Plant Examinations

Generic Letter 88-20, "Individual Plant Examination for Severe Accident Vulnerabilities - 10 CFR §50.54(f)," issued November 23, 1988, requested licensees to perform systematic examinations to identify any plant-specific vulnerabilities to severe accidents and report the results to the Commission. The request stemmed from recognition that the use of PRAs could identify plant-specific vulnerabilities that could be fixed with low-cost improvements. This followed a Commission policy statement on severe accidents that concluded that then existing plants posed no undue risk to the public health and safety and that there was no basis for immediate action on generic rulemaking or other regulatory requirements for the plants (50 *Federal Register* 32138, August 8, 1985).

As discussed in a Staff Requirements Memorandum dated April 28, 1995, from the Secretary of the Commission to the Executive Director for Operations, the Commission was briefed on the Individual Plant Examination (IPE) program conducted in response to GL 88-20. The memorandum stated that the current industry IPE results do not, by themselves without further staff review, provide a complete basis for supporting risk-based regulatory decision making. The NRC's reviews of the IPEs were not of sufficient depth to allow approval of, or concurrence with, the absolute values and conclusions stated in the IPE reports, and do not validate the results. The memorandum suggested that industry coordinate with the NRC staff and initiate actions necessary to develop PRAs that would be acceptable for risk-based regulatory use. Such PRAs would use standardized methods, assumptions, and level of detail. Currently, the variability in the IPE methodology and the underlying data and assumptions made by each plant on a case-by-case basis bring a large measure of uncertainty to the final results of the IPE PRAs. Human performance issues are not easily quantified, adding to the level of uncertainties in the use of IPEs. The uncertainties and the lack of detailed review and validation of the IPE results by the NRC staff limit the use of the IPEs to narrow issues of plant-specific vulnerabilities and not to assessing the overall safety of a specific plant or to determining the relative safety significance of specific components in the plant. The IPEs are best used in the context of the GL 88-20 request for the examination, which was to (1) develop an appreciation of severe accident behavior, (2) understand the most likely severe accident sequences that could occur at a plant, (3) gain a more quantitative understanding of the overall probabilities of core damage and fission

product releases, and (4) reduce the overall probabilities of core damage and product releases through modifications of hardware or procedures, where appropriate (*see* GL 88-20).

Other Recent Activities

Other activities that involve risk-informed regulation are (1) Section 50.65, the "Maintenance Rule" which is a performance-based rule that employees elements of risk-informed decisions in implementation, (2) graded quality assurance, (3) inservice inspection, and (4) inservice testing. The maintenance rule has successfully been issued and is effective as of July 10, 1996. The staff and industry continue to work on the other three programs, largely through pilot projects that are in various stages. The risk-informed inservice testing pilot programs are more fully developed than those of graded quality assurance and inservice inspection.

CONCLUSION

Various elements and considerations make up the underpinnings for the application of risk assessment to the regulatory decision-making process. The Safety Goals have established, in broad terms, the level of acceptable risk. The Staff has established guidance regarding imposition of new requirements. The regulated industry has, via the IPE methodology, assessed the individual plant's risk profile. The next presentation will discuss the integration of these various elements and the NRC policy on the use of PRA and will give the up-to-date information on the inservice testing pilot program reviews and regulatory actions.

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2. NRC NUREG/BR-0175, "A Short History of Nuclear Regulation, 1946 - 1990," J. Samuel Walker, January 1993, at 23 - 27.

3. *Id* at 29.

4. *Id* at 29, 30.

5. *Id* at 31.

6. NRC WASH-1400 (NUREG-75/014), "Reactor Safety Study, An Assessment of Accident Risks in U.S. Commercial Nuclear Power Plants," October 1975.

7. NRC NUREG/BR-0175, "A Short History of Nuclear Regulation, 1946 - 1990," J. Samuel Walker, January 1993, at 46.

8. The Three Mile Island nuclear generating plant is owned and operated by GPU Nuclear Corp. and is located approximately 10 miles southeast of Harrisburg, Pennsylvania, on an island in the Susquehanna River.

9. The PORV functions to relieve pressure during an expected transient event which can occur from normal pressure spikes during the transient. It does not provide overpressure protection for the reactor coolant system.

10. Final Report, "The President's Commission on The Accident at TMI," John G. Kemeny, Chairman, October 30, 1979.

11. See 51 *Federal Register* 30028, August 21, 1986.

12. *Id* at 30028, 30029.

13. *Id* at 30032.

Risk-Informed Inservice Test Activities At The NRC

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ABSTRACT

The operational readiness of certain safety-related components is vital to the safe operation of nuclear power plants. Inservice testing (IST) is one of the mechanisms used by licensees to ensure this readiness. In the past, the type and frequency of IST have been based on the collective best judgment of the NRC and industry in an ASME Code consensus process and NRC rulemaking process. Furthermore, IST requirements have not explicitly considered unique component and system designs and contribution to overall plant risk. Because of the general nature of ASME Code test requirements and non-reliance on risk estimates, current IST requirements may not adequately emphasize testing those components that are most important to safety and may overly emphasize testing of less safety significant components.

Nuclear power plant licensees are currently interested in optimizing testing by applying resources in more safety significant areas and, where appropriate, reducing measures in less safety-significant areas. They are interested in maintaining system availability and reducing overall maintenance costs in ways that do not adversely affect safety.

The NRC has been interested in using probabilistic, as an adjunct to deterministic, techniques to help define the scope, type, and frequency of IST. The development of risk-informed IST programs has the potential to optimize the use of NRC and industry resources without adversely affect on safety.

Background:

Since late 1992, the NRC has been working with ASME Research (i.e., funding to ASME Research via grant and personnel support) to develop guidelines for using probabilistic techniques to help define improved inservice testing requirements. In late 1994, the staff began to encourage pilot applications of risk-informed methods to help define and focus IST requirements. Also in late 1994, the Nuclear Energy Institute (NEI) began to take an interest in the development of risk-informed IST programs. On November 27, 1995, the staff received two requests (i.e., from TU Electric for Comanche Peak Units 1 & 2 and from Arizona Public Service for Palo Verde Nuclear Generating Station Units 1, 2, & 3) to implement risk-informed IST programs in lieu of IST programs constructed pursuant to Section XI of the ASME Code. In September 1995 and November 1995, the NRC also received two separate draft guidance documents, one from NEI and another from ASME Research, on developing risk-informed IST programs. The NEI Risk-Based IST Guidelines was revised and reissued on March 19, 1996. Additional

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revisions are expected as the pilot plant reviews continue and as the NRC develops risk-informed IST Regulatory Guides and Standard Review Plans (SRP).

Issues Associated With Risk-Informed Regulation:

The Nuclear Regulatory Commission (NRC) has taken several recent actions to improve the regulatory process, including many directed toward achieving the goal of risk-informed regulation. These staff actions included the recent update of the agency-wide probabilistic risk assessment (PRA) implementation plan (SECY-95-079), the publication of NRC's final PRA policy statement (August 16, 1995), and the framework for applying PRA analysis in reactor regulation (SECY-95-280). However, more thought and deliberation are needed on how to change existing regulatory structure to reflect the desire to regulate "commensurate with their importance to public health and safety." The staff will continue to work with the industry pilot programs to incorporate PRA methodologies and applications into the regulatory process.

Periodically, the Commission is briefed on the status of the NRC staff's risk-informed initiatives, including risk-informed IST, risk-informed ISI, graded QA and technical specifications. In a March 26, 1996, memorandum from the EDO to the Commission, the NRC staff identified four emerging policy issues associated with these risk-informed initiatives. The four emerging policy issues are:

- **Establishing the role of performance-based regulation in the PRA Implementation Plan**

The staff is currently considering the extent to which performance-based initiatives should be addressed in the PRA Implementation Plan. The staff supports the concept of performance-based regulation provided it will result in improved decision making, more efficient use of agency resources, and a reduction in unnecessary burden on licensees.

The staff has further stated that performance-based initiatives should be selected where objective performance criteria can be established for performance monitoring and where failure to meet the performance criteria results in tolerable conditions for which appropriate corrective actions will be taken.

However, one aspect of the performance-based approach should be incorporated into almost all risk-informed activities (i.e., the feedback of actual experience into the risk-informed activity). As data from performance monitoring of structures, systems, and components (SSCs) are accumulated, the staff expects licensees to evaluate the impact of that performance data on the risk-informed activity.

- **Plant-specific application of the safety goals**

In its Staff Requirements Memorandum (SRM) dated June 15, 1990, the Commission instructed the staff not to apply the safety goals on a plant-specific basis without first requesting Commission guidance. As part of its effort to develop guidance on risk-informed decision-making, the staff will be developing criteria to judge the risk contributions of the proposed regulatory changes and

licensees' proposals. It may be appropriate for these criteria to reference various elements of the safety goals or their subsidiary numerical objectives.

- **Risk neutral versus small increases in risk**

  Related to the safety goal issue is the issue of whether only risk neutral plant changes should be allowed. The industry has requested relaxation of regulatory burdens in some areas, although the calculated plant risk would increase slightly. The question is whether the staff should allow increases in risk, or require compensatory measures in the same or other areas to "neutralize" the risk increases. Regardless of whether the NRC attempts to maintain risk neutrality, all risk informed applications will require adequate maintenance of defense-in-depth.

- **Implementation of changes to risk-informed IST requirements**

  The staff plans to use the acceptable alternative approach allowed by 10 CFR 50.55a (a)(3)(i) for approval of the pilot plants' applications after satisfactory staff review of the pilot plant submittals. The staff expects that the interactions with pilot plant licensees will directly benefit the work on the RGs and SRPs and should lead to refinements in industry guidance documents.

  To provide the permanent approach to risk-informed IST, the staff intends to utilize the experience gained through the pilot applications to modify 10 CFR 50.55a through rulemaking. In the interim, the staff may be willing to authorize alternatives for other plants after approval of alternatives for the pilot plants, but such reviews and approvals would be processed consistent with staff review resources and priorities.

The staff plans to brief the Commission on key policy issues associated with this, and other, risk-informed initiatives before the staff authorizes implementation of risk-informed IST programs at the pilot plant sites.

<u>Ongoing NRC Staff Activities</u>:

The staff is developing a general regulatory guide and SRP section, sufficiently broad in scope, that can be applied in transitioning to a more risk-informed, regulatory decision-making process. These regulatory guidance documents will address general issues such as PRA scope and quality to support screening, risk ranking, and detailed analysis, as well as the definition of the potential role and activities of an expert panel.

In parallel with the development of broad regulatory guidance, the staff is developing a series of application-specific regulatory guides that are tied to specific regulations or program areas such as graded quality assurance, inservice testing of pumps and valves, inservice inspection, and technical specifications. As resources become available, application-specific SRP sections and Inspection Procedures corresponding to these application-specific regulatory guides will be developed.

A critical element or activity associated with the development of regulatory guidance in the area of risk-informed IST is the staff's interactions with the two pilot plant licensees. Pilot programs are underway at Comanche Peak and Palo Verde. The staff is working with these two licensees to define and

establish an acceptable risk-informed IST program. On March 15, 1996, the NRC issued a Request for Additional Information (RAI) to the risk-informed IST pilot plant licensees. There are a number of issues in the RAI that need to be addressed and resolved before the staff will be in a position to approve the licensees' request to implement risk-informed IST programs. Areas that the NRC staff are investigating with the pilot plant licensees include:

- Assuring that the level of detail and scope, and other quality related aspects, of the PRA are sufficient for the IST risk-ranking application and that adequate review of the plant's PRA has been performed.

- Assuring that the risk ranking is robust and that components ranked as low safety significant will remain as low, independent of PRA uncertainties, conservatisms, and modeling assumptions. This can be partially achieved by performing a series of sensitivity and uncertainty studies and also by relying on more than one risk-ranking importance measure.

- More clearly defining the processes and decision criteria used to risk rank components to help the staff develop risk-informed regulatory guides and SRP sections.

- Defining an appropriate risk ranking result validation process, and defining an acceptable change in risk from the change in current IST requirements.

- Assuring that the overall risk-informed IST program does not have an adverse effect on defense-in-depth or the plant's licensing basis.

- Measures to identify important components not in the current IST program as well as to identify the important failure modes of the more risk-significant components so that the effectiveness of the test strategy (i.e., methods and frequency) can be addressed.

- Establishing the technical basis for test interval extensions. It should be based on component design, service condition, and performance as well as on the component's safety significance (i.e., not just on risk insights alone).

- Implementing a performance-based feedback mechanism to ensure that if a particular component's test interval is extended too far it gets expeditiously identified and corrected. There also needs to be performance-based feedback to the PRA models and risk-ranking process.

- Defining the specific processes for implementing the risk-informed IST program at the plants (e.g., system versus component-type implementation schedule) as well as the new starting point (test frequency and method) for each component or group of components.

Pilot plant licensees are expected to supplement their submittal and respond to the staff's RAI in late spring to early summer 1996. The NRC anticipates authorizing an alternative to the existing Code testing requirements, pursuant to 10 CFR 50.55a ¶ (a)(3)(i), pending completion of the staff's regulatory guidance related to risk-informed IST programs. The staff expects to brief the Commission regarding the pilot plant program and related policy issues prior to authorizing implementation. Assuming successful completion of the staff's interaction with the

pilot licensees (for Comanche Peak and Palo Verde) and resolution of risk-informed issues with the Commission, the staff expects to document acceptability of the risk-informed IST pilot programs in safety evaluations.

In addition to the review of the pilot plant submittals, the NRC will continue its evaluation of industry guideline documents on risk-informed IST applications and the development of inservice testing strategies based on the resulting categorization of components. The staff acknowledges the important role of industry-wide experience compiled and made available to all parties through codes, standards, and guidance documents. The NRC believes that use of this process is an important element of a transition by industry and regulatory authorities. The NRC will continue to work with ASME Code Committees to develop appropriate test strategies for each component category, based on generic failure data and failure cause information, and to develop a protocol for adjusting the revised Code test strategies based on plant specific information.

Anticipated Future Staff Activities:

About a year after authorizing implementation of the pilot plant risk-informed IST programs and before issuing a final risk-informed IST RG and SRP section, the staff will assess each pilot plant risk-informed IST program to confirm program effectiveness (e.g., to assess the test strategies for the more safety-significant components to verify that the strategies are effective at detecting component degradation/failure; to ensure that plant-specific component performance data and operational experience are being used effectively to make risk-informed IST program adjustments; and to evaluate program implementation issues). Lessons learned from this evaluation will be incorporated into the final RGs and SRP sections.

The staff will initiate rulemaking (i.e., to revise 10 CFR 50.55a) so that other licensees can voluntarily adopt risk-informed IST programs, without the need for specific NRC review and approval. In the interim, the staff may be willing to authorize alternatives for other plants (e.g., the other 7 pilot plants identified by EPRI: Point Beach, Wolf Creek, South Texas, Seabrook, St. Lucie, Three Mile Island, and Peach Bottom) after approval of alternatives for the pilot plants, but such reviews and approvals would be processed consistent with staff review resources and priorities.

In addition, the NRC staff will also continue to work with NEI as they refine their draft (March 19, 1996) Risk-Based IST Guidelines. Comments on Revision B of the draft NEI Guidelines will be provided later this summer when the staff position on implementation process and approaches, and decision criteria have been further reviewed and evaluated.

Conclusion:

Assuming successful completion of the staff's interaction with the pilot licensees (for Comanche Peak and Palo Verde) and resolution of risk-informed issues with the Commission, the staff expects to authorize these licensees to implement risk-informed IST programs that comport with the proposed RG and SRP. The staff's authorization to the pilot plant licensees and the issuance of the RG and SRP for public comment are expected to occur in late 1996. We expect that during the following year the proposed RG and SRP will undergo minor revision in response to public comments. Also, the staff expects to

assess the pilot plant licensees' implementation of the risk-informed IST programs. Depending on the changes to the RG and SRP and the staff's experience over the period between the initial and final publication of these documents, it may be necessary for the pilot plant licensees to bring their programs into conformance with the final version of the RG and SRP. In addition, we understand that the ASME is currently developing Code cases dealing with various aspects of risk-informed IST strategies. The staff has been following this activity and believes that improved test strategies could be of substantial benefit to risk-informed IST programs. Depending upon when the ASME approves these Code cases, and whether the staff finds them acceptable, we may propose to revise the RG and SRP to include these Code cases.

References:

1. NRR Office Letter No. 504, "Procedure for Development, Implementation and Management of Action Plans."

2. NRC Final Policy Statement on "Use of Probabilistic Risk Assessment Methods in Nuclear Regulatory Activities," FRN 42622 dated August 16, 1995.

3. Proposed Agency-Wide Implementation Plan for Probabilistic Risk Assessment (PRA), SECY-94-219 dated August 19, 1994.

4. Electric Power Research Institute (EPRI) PSA Applications Guide, EPRI TR-105396 dated August 1995.

5. Nuclear Energy Institute Draft (Revision B) "Industry Guidelines for Risk-Based Inservice Testing" dated March 19, 1996.

6. Draft ASME Research Report "Risk-Based Inservice Testing - Development of Guidelines" dated November 1995.

7. TU Electric (Comanche Peak Units 1 & 2) Request for Exemption from 10 CFR 50.55a(f)(4)(i) and (ii) for Inservice Testing (IST) Program (Testing Frequency) dated November 27, 1995.

8. Arizona Public Service (Palo Verde Generating Station) Request for Exemption from 10 CFR 50.55a(f)(4)(i) and (ii) for Inservice Testing (IST) Frequency dated November 27, 1995.

9. Arizona Public Service (Palo Verde Generating Station) Request for Exemption from 10 CFR 50.55a(f)(4)(i) and (ii) for Inservice Testing (IST) Frequency Supplement 1 dated December 20, 1995.

10. Memorandum from Shirley Ann Jackson, Chairman, to James M. Taylor, Executive Director for Operations, dated November 30, 1995, "Follow-up Request in Probabilistic Risk Assessment and Digital Instrumentation and Control."

11. Memorandum from James M. Taylor, Executive Director for Operations, to Shirley Ann Jackson, Chairman, dated January 3, 1996, "Improvements Associated With Managing the Utilization of Probabilistic Risk Assessment (PRA) and Digital Instrumentation and Control Technology."

12. Memorandum from James M. Taylor, Executive Director for Operations, to the Commission, dated March 26, 1996, "Status Update Of The Agency-Wide Implementation Plan For Probabilistic Risk Assessment (PRA) (From March 30, 1995 To February 29, 1996)."

Session 3B

O&M Code Issues on Testing Pumps and Valves

Session Chair
Henry J. Thailer
Supervising Engineer
Pacific Gas & Electric Co.

Activities of the O&M Committee History & Future Perspectives

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ASME International

Abstract

This paper gives an overview of the Committee on Operation and Maintenance of Nuclear Power Plants, hereafter referred to as the O&M Committee, formed in June 1975 when the American National Standard Institute's Committee on Reactor Plants and their Maintenance was disbanded. The O&M Committee's history, structure, current focus and future perspectives will be presented. The purpose of this paper is to give information to industry and the public of the Committee's on-going effort to make accurate and timely responses to the needs of the nuclear industry.

I. History

The O&M Committee was formed in June 1975 when the American National Standards Institute's (ANSI) Committee on Reactor Plants and their Maintenance (Committee N45) was disbanded. The N45 Committee was chartered to promote the development of standards for the location, design, construction, and maintenance of nuclear reactors and plants embodying nuclear reactors, including equipment, methods, and components. The American Society of Mechanical Engineers (ASME) assumed the secretariat of several of the N45 committees that related to the requirements contained in Sections III and XI of the ASME Boiler and Pressure Vessel Code.

After ASME assumed this responsibility, the committee reviewed Section XI documents and determined where O&M Standards could replace current Section XI requirements. The major areas identified in Section XI were Articles IWP (Inservice Testing of Pumps) and IWV (Inservice Testing of Valves). To facilitate development of standards in these areas, the Section XI Subgroup on Pumps and Valves was transferred to the O&M Committee in 1979 and was designated O&M Working Group on Pumps and Valves. In time, a new Section XI Working Group on Pumps and Valves was established in 1984 to review the O&M Standard on Pumps and Valves to assure acceptability to Section XI. Ultimately, the charter of the O&M Committee was to develop, revise and maintain Codes, Standards, and Guides applicable to the safe and reliable operation and maintenance of nuclear power plants.

Originally the O&M Committee operated with two Subcommittees: (1) Subcommittee on Vibration Monitoring, and (2) Subcommittee on Performance Testing. There were five separate standards published in 1981 and 1982 that were consolidated into a single publication designated ASME/ANSI OM-1987. As the Committee's work progressed, the ASME Board on Nuclear Codes and Standards

(BNCS) recognized that the O&M Committee is the appropriate committee to establish inservice testing (IST) requirements and voted to proceed with making the O&M Standard stand on its own, with the objective of eventual deletion of IST documents from Section XI of the Boiler and Pressure Vessel Code where appropriate. A transition was implemented in which certain Parts of OM-1987 addressing pumps, valves, safety valves, and snubbers were incorporated into ASME OM Code-1990 (Code for Operation and Maintenance of Nuclear Power Plants). The remaining Parts were incorporated into ASME OM-S/G-1990, Standard and Guides (for Operation and Maintenance of Nuclear Power Plants). This transition did not result in technical changes to the existing IST requirements.

II. Structure

The O&M Committee's standard development process exhibits a balance of interests such that a single category of interest cannot dominate the actions of the Committee. This is in accordance with ANSI and ASME procedures. To ensure a balance of all interests, ASME procedures require not more than one third of the O&M Main Committee membership to come from any one category of interest. This is shown in Table 1. Further, participation is open to all persons who are directly and materially affected by the activity in question.

Once a balance of interest is established, consensus is of paramount importance in developing a Standard or Guide. Consensus is established when substantial agreement has been reached by directly and materially affected interests. Here, substantial agreement means much more than a simple majority, but not necessarily unanimity. Most important, consensus requires that all views and objections be considered and that an effort be made toward their resolution. Although the process is time consuming, it produces documents which serve the needs of the affected parties and the public. The O&M Main Committee is the consensus committee which considers proposals developed by the subtier committees. In addition, it also considers approval of personnel and administrative items or actions relating to ASME policy or position. The Main Committee reports directly to the ASME Board on Nuclear Codes and Standards which is the supervisory body for all ASME activities related to codes and standards directly applicable to nuclear facilities and technology. An O&M Executive Committee exists as advisory to the Main Committee on administrative and current operational issues as well as future planning matters.

The O&M subtier structure consists of two Special Committees (SpC's), four Subcommittees (SC's), four Subgroups (SG's) and fifteen Working Groups (WG's). These are standing committees established by vote of the Main Committee with a continuing assignment of duties and responsibilities. The Main Committee also considers the size of the subtier committees to numbers which will best serve operational needs and still be representative of all interests. The Main Committee has appointed two Special Committees: (1) SpC on Editorial Review and (2) SpC on Standards Planning. They are part of the official membership of the Committee and in general are given long term assignments. Task Groups are not part of the official membership of the Committee and are discharged upon completion of their specific task assigned. For instance, two current Task Groups are addressing Code administration and honors & awards.

Briefly, at the lower tier of the structure is where all proposals for a new document are first considered. Eventually, a proposal will surface to the concerned Subcommittee for its approval and then proceed to the Main Committee. The point being, the consensus of the committee is considering a proposal which has been thoroughly examined and is most representative of the interested parties. The consensus committee is the highest technical level of approval within ASME Codes and Standards.

III. Code Cases

When the need is urgent, the Committee will consider the use of a Code Case to provide a mechanism for early implementation of a new process or alternatives to existing Code requirements. A Code Case is nonmandatory, and is issued for a three year (renewable) term.

IV. Current Focus

For the past few years, the O&M committee has focused on applying risk based inservice testing (RB-IST) technology to the ASME O&M Code. An ASME risk based inservice testing research project conducted through a period of several years is coming to a close and so now, the O&M committee is starting the inservice testing code implementation process. The implementation process is twofold. First, individuals who have been deeply involved in formulating risk based inservice testing must implement the concepts in codes and standards language as well as format and second, a considerable education process remains which must reach the entire community of prospective users. Briefly, the three key attributes recommended by the research project are (1) a process that utilizes quantitative results and deterministic information to help an expert panel risk rank components in two categories, More Safety Significant Components (MSSC's) and Less Safety Significant Components (LSSC's), (2) a performance based testing program development approach for MSSC's using specific component failure modes, failure causes, and effectiveness rating of alternative testing strategies, and (3) a performance based testing program to assess the functional readiness of the LSSC's. By focusing resources on MSSC's, plant economics will improve substantially. Ultimately, the goal is to revise the ASME O&M Code, but in the near term, the committee hopes to have some code cases available for plants to use in late 1996 for specific application to components.

V. Future Perspectives

A. Internationalization of the O&M Code

The O&M Committee is participating in an overall ASME effort to address the needs of the international community with regard to the proper units used in its codes and standards, and removing impediments to adoption by international bodies, commonly referred to as "denationalization." The Codes and Standards committees will consider all customers, including the U.S., and the product forms being manufactured and used. While being encouraged to proceed in accordance with the Society's direction towards globalization, the format and approach implemented by the various committees can be flexible based upon a determination of the best format for what can be used internationally.

B. Proposed New O&M Part

O&M Part 15 (Performance Testing of Emergency Core Cooling Systems in

Pressurized Water Reactor Plants) is being proposed as a new Part to the O&M Standards and Guides. Part 15 provides added value as a single, comprehensive standard with methodology for developing a test program to assess the operational readiness of pressurized water reactor Emergency Core Cooling Systems (ECCS). Current guidance to plant Owners is not focused and may not ensure the ECCS design basis performance requirements. In addition, guidance provided by this standard is intended to convey that only necessary maintenance or modification testing is performed on ECCS.

C. Proposed Revision to ISTC

Subsection ISTC covers inservice testing of valves in light water reactor power plants. One of the current issues in this Subsection concerns check valves. Check valves mechanically exercised to meet code requirements generally have weighted arms. Concerns with the ability of Subsection ISTC to properly assess operational readiness prompted a survey to acquire specific information. Twenty - nine plants representing eighteen utilities responded. Thirty - four percent of these plants mechanically exercised as many as sixty - six check valves. Recurring problems with acceptance criteria and adequacy of breakaway torque alone were identified. A revision to Subsection ISTC to address these problems is currently under consideration.

VI. Closing Remarks

The O&M committee will continue its efforts to respond to the needs of the nuclear power industry that are within the purview of the American Society of Mechanical Engineers. Input from industry in the form of technical expertise is always welcomed and encouraged. A broader spectrum of input leads to documents which are worthy of the industry they serve.

Table 1: Balance of Interest for Committee on Maintenance and Operation of Nuclear Power Plants Status as of January 1996

AA: 1 Constructor	AB: 2 Designer	AC: 0 Designer/Constructor
• Bernier, Stone & Webster	• Dermenjian, Sargent & Lundy • Vogan, Sargent & Lundy	None
AF: 5 General Interest	**AH: 0 Insurance/Inspection**	**AI: 5 Laboratories**
• Allen, *Barrington Consulting Group* • Richardson, *Richardson & Associates* • Rowley, *Rowley Consultants* • Zigler, *Science & Engineering Associates* •Ferrante, *Consultant*	None	• DiBiasio, *Brookhaven National Laboratories* • Hartley, *INEL (Lockheed)* • Hyten, *Wyle Laboratories* • Williams, *Oak Ridge National Laboratory (Lockheed Martin Energy Systems)* • Johnson, *Lockheed Martin Uranium Enrichment*
AK: 4 Manufacturer	**AM: 0 Material Manufacturer**	**AO: 5 Owner**
• Khuzaie, *General Electric Nuclear Energy* • McDonough, *Westinghouse Electric* • Palmer, *Siemens Nuclear* • Au-Yang, *Babcock & Wilcox Nuclear Technology*	None	• Favreau, *Tennessee Valley Authority* • Martel, *NAESCO* • Pelletier, *Yankee Atomic* • Thailer, *Pacific Gas & Electric* • Zudans, *Florida Power & Light*
AT: 1 Regulatory		
• Baer, *U.S. Nuclear Regulatory Commission*		

Total Number of Members: 23

ASSESSMENT OF CODE IMPROVEMENTS IN THE 1992, 1994, AND 1996 ADDENDA OF THE OM CODE

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ABSTRACT

The NRC has initiated a rulemaking to Section 50.55a of Title 10 to the *Code of Federal Regulations* (CFR) that would allow Owners to voluntarily update their pump and valve inservice testing programs to the 1995 Edition of the American Society of Mechanical Engineers (ASME) *Operations and Maintenance* (OM) Code. The 1992 and 1994 Addenda, and 1995 Edition of the OM Code offers many improvements, e.g., clarifications and relaxations, to the 1989 Edition of Section XI, of the ASME *Boiler and Pressure Vessel Code*, or the 1990 Edition of the OM Code. This paper reviews the code changes that may be advantageous for Owners to use, and discusses their related requirements. Additionally, code improvements in the newly issued 1996 Addenda of the OM Code are discussed, as they may be proposed under 10CFR50.55a(a)(3)(i).

INTRODUCTION

In February 1995, the NRC had initiated a rulemaking to 10CFR50.55a that would require all licensees to update their pump and valve inservice testing (IST) programs one final time to the 1990 Edition of the ASME OM Code and eliminate the 120 month periodic IST program update requirement (Reference 1). Future revisions would not have been required, unless justified under the backfit rule, i.e., 10CFR50.109. The rulemaking allowed voluntary updating to the later addenda and edition of the OM Code. Since then, the NRC has revised their approach and has retained the 120 month update provisions. This change in direction was reported in the NRC Liaison Report at the March 1996 OM Main Committee Meeting. The staff has proposed to revise 10CFR50.55a(b) to include the 1990 Edition of the OM Code, which is identical in technical requirements to OMa-1988, Parts 6 and 10 for pump and valve testing. OMa-1988, Parts 6 and 10 are referenced by the 1989 Edition of Section XI, which is currently referenced in 10CFR50.55a(b). Voluntary use of the 1992 and 1994 Addenda, and the 1995 Edition of the OM Code has also been proposed in the recently revised proposed rulemaking.

The 1992 Addenda, 1994 Addenda and 1995 Edition of the OM Code offer many improvements, e.g., clarifications and relaxations, to the 1990 Edition. There were 14 actions incorporated in ISTB and ISTC (no changes to ISTA were included), which effected 135 paragraph changes. The only change in Subsections ISTA, ISTB or ISTC of the 1992 Addenda corrected the inequality sign in Table ISTB 5.2.2a, as errata. The 1994 Addenda, however, contained numerous technical changes to ISTB, ISTC, and Appendix I. The most noteworthy of these changes is addition of the comprehensive pump test and permission to use analysis to revise pump reference values in ISTB;

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

addition of a sample disassembly and inspection program and non-intrusive testing for check valves in ISTC; and substantial changes to Appendix I on safety and relief valve testing. The 1995 Edition consists of the 1990 Edition incorporating changes in the 1992 and 1994 Addenda, and a number of errata in ISTB and Appendix I. Errata should not be considered to be a part of the addenda or Edition they appear in, as they simply correct publishing errors (e.g., typographical errors or omissions to what the Code Committees had approved). Therefore, they should, as appropriate, be implemented immediately.

A summary of the most advantageous enhancements added in the 1994 Addenda are included in Table 1. When using portions of editions and addenda, all the related requirements must also be used as required by the Code, ISTA 2.2.1(c) and the regulations. When editions and addenda are issued, it is often difficult to determine which are the related requirements, without reviewing the code changes as they are approved by the Code committees. Table 1 includes the related requirements as specified in the actions approved by the Subcommittees and Main Committee of the OM Code. This information was gathered from committee correspondence and meetings. The following paragraphs describe the code changes in more detail.

Table 1 Advantageous Alternate Requirements in 1994 Addenda

Advantageous Alternate Requirements in 1994 Addenda	**Related Requirements**
Use of Comprehensive pump test (ISTB 4 and 5)	ISTB 3.1, 3.2, 6.2, 7
Use of new pump reference values based on analyses and evaluation of trends (ISTB 4.6 and 6.2)	ISTB 6.1, 7.3
Use of check valve sample disassembly and inspection (ISTC 4.5.4(c))	ISTC 4.5.6, 6.2(e)
Requirements for Testing Additional Valves, Appendix I, ¶ I 1.3.3(c), (e) and 1.3.5(c), (e)	¶ I 1.2, 1.3.1(e), 5.1, 9.1
Test Frequency for Class 2 and 3 primary containment vacuum relief valves, ¶ I 1.3.7	¶ I 5.2, 9.2
Test sequence requirements, ¶ I 3.1, 3.3, 7.1, 7.3.	None.
Allowance to defer corrective actions, ¶ I 3.4.1(e), 3.4.2(d), 3.4.3(d), 3.4.5(d), 3.4.7(d), 7.4.1(d), 7.4.2(d), 7.4.3(d), 7.4.5(d), 7.4.6(d)	¶ I 5.1, 9.1
Clarifications:	
Clarifications to ISTB 4.7.1(a), ISTB 5.2.1(e) and 5.2.3(e), Footnote to Table 5.2.2-1, ISTB 5.4	None.
Use of non-intrusive testing for check valve exercising (ISTC 4.5.4(a))	ISTC 2
Instrumentation requirements, ¶ I 1.4.1	None
Scope of Appendix I, ¶ I 1.1	None

1994 Addenda

OMa-1988, Part 6 requires quarterly testing of all pumps included in the scope of the standard. There is no distinction for normally operating or standby pumps. The OM Code in the 1994 Addenda has been revised to address standby and normally operating pumps separately. The quarterly requirements in the 1994 Addenda are essentially the same as OMa-1988, Part 6 for normally operating pumps. For standby pumps, however, the quarterly test only involves the measurement of speed, if the pump is variable speed, and flowrate or differential pressure. Owners may wish to implement the reduced quarterly test requirements on pumps that can only be tested on a minimum flowpath, to minimize pump degradation due to low flow. The related requirements that would be required include: performing a comprehensive pump test biennially, performing a preservice test in accordance with ISTB 4.1 prior to implementing this alternate, establishing reference values in accordance with ISTB 4.3, 4.4 and 4.5; complying with the instrument accuracy requirements of ISTB 4.7.1(a), complying with the test method requirements of ISTB 5, complying with the acceptance criteria of ISTB 6.2, and complying with the record requirements of ISTB 7.

OMa-1988, Part 6 requires pumps in the alert range to have their test frequency doubled until the cause of the deviation is determined and the condition corrected. Pumps in the required action range are required to be declared inoperable until the cause of the deviation is determined and the condition corrected. The OM standard does not allow the use of analyses to remove pumps from the alert or required action range. The 1994 Addenda of the OM Code, however, has added a new provision that allows licensees to establish a new set of reference values when the pump is in the alert or required action range, but whose continued operation at the new values is supported by an analysis (ISTB 4.6 and 6.2). The analyses shall include verification of the pump's operational readiness at both the pump level and system level, the cause of the change in pump performance, and an evaluation of trends indicated by available data. When using this alternate, the Owner must document this analysis in the record of tests (ISTB 7.3) and must trend the test parameters (ISTB 6.1).

OMa-1988, Part 10, paragraph 4.3.2.4(c), allows disassembly every refueling outage, as an alternate to exercising check valves. The NRC, in Generic Letter 89-04, Position 2, also allowed disassembly and inspection, however, the NRC recognized that disassembling all applicable valves every refueling outage may be burdensome to licensees and allowed the use of a sample disassembly and inspection plan (i.e., one valve is inspected every refueling outage and every valve in the group is inspected every 6 years). The NRC guidelines for this plan include extension of the valve disassembly/inspection interval to one valve every other refueling outage or expansion of the group size above 4 valves only in the cases of "extreme hardship" supported by actual in-plant data from previous testing. The Position provides 3 criteria that need to be developed to support extension of the interval to longer than once every 6 years. The 1994 Addenda of the OM Code has added a provision for a sample disassembly and inspection plan. This plan allows a 8 year disassembly/inspection interval, however, it does not include provisions to extend the interval past 8 years or to forego disassembling valves at each refueling outage. The 8-year interval was based on an INPO

Technical Paper, "Check Valve Failure Trends in the Nuclear Industry," by Michael Scott and is consistent with the industry trend to 24 month refueling outages. The use of the OM Code sample disassembly and inspection plan requires the related requirements contained in ISTC 4.5.6, Corrective Action, and ISTC 6.2, Test Plans, be implemented.

The requirements for testing additional valves, when valves fail the set pressure test acceptance criteria, in OM-1-1981, OM-1-1987, and the 1990 Edition of the OM Code are very confusing. The 1994 Addenda clarifies the requirements and additionally allows the use of Owner-established set-pressure criteria or $\pm 3\%$ of valve nameplate pressure as a screening criterion for determining the need to expand the test sample as stated in paragraphs I 1.3.3(c)(i) and 1.3.5(c)(i). Related to Appendix I, paragraphs I 1.3.3(c) and 1.3.5(c) are the requirements in paragraphs I 1.3.1(e), 5.1, and 9.1 for establishing and documenting the acceptance criteria. Also, the 1994 Addenda clarifies that valves subject to additional testing are those of the same valve group, which includes the same system application and service media, as well as manufacturer and type. This change could reduce the number of valves to be additionally tested.

OM-1-1981, OM-1-1987, and the 1990 Edition of the OM Code specify that Class 2 and 3 primary containment vacuum relief valves be tested every six months unless historical data indicates a requirement for more frequent testing. The 1994 Addenda of the OM Code has revised this test frequency to every refueling outage or every two years, whichever is sooner. Use of Appendix I, paragraph I 1.37 would require the licensee to establish and implement the test schedule, as required by the related requirements found in paragraphs 5.2 and 9.2. However, licensees must review their Technical Specifications, before proposing this alternative. As discussed in NUREG-1366, the staff recommends retaining the monthly surveillance testing of suppression chamber to drywell vacuum breakers in boiling water reactors.

OM-1-1981, OM-1-1987, and the 1990 Edition of the OM Code specify a sequence of tests, all of which must be performed prior to maintenance or set pressure adjustment. The 1994 Addenda requires only the visual examination, seat tightness determination and set-pressure determination to be performed prior to maintenance or set pressure adjustment. It allows the other tests to be performed after maintenance or adjustments. Additionally, the 1994 Addenda requires the determination of compliance with the Owner's seat tightness criteria to be performed last, after all the other tests are performed. The test sequence in the 1994 Addenda may be proposed in lieu of the requirements in previous editions. There are no related requirements. It should be noted that this paragraph was also revised in the 1996 Addenda, and is discussed later.

The 1994 Addenda of the OM Code, Appendix I now allows an evaluation in lieu of performing corrective actions immediately to meet the valve's acceptance criteria. The basis for this Code change was to allow valves with minor set-pressure deviations to be evaluated and accepted until the next test, provided that the valve is capable of performing its intended function until the next testing interval or maintenance opportunity, and corrective actions are taken to ensure valve operability. Owners may propose deferring corrective actions provided that they perform an evaluation and comply with the

requirements in paragraphs 3.4.1(e), 3.4.2(d), 3.4.3(d), 3.4.5(d), 3.4.7(d), 7.4.1(d), 7.4.2(d), 7.4.3(d), 7.4.5(d), or 7.4.6(d); and document the analysis of tests which do not satisfy acceptance criteria and the corrective actions, as required by paragraphs 5.1 and 9.1.

The scope of Appendix I of the OM Code was revised in the 1994 Addenda to clarify that this appendix applies to pressure relief devices required to protect systems or portions of systems that perform a specific safety related function. Previously, the scope of Appendix I stated that the appendix only applied to pressure relief devices which are, themselves, required to perform a specific safety related function. However, ISTC 1.1 of these earlier editions stated that the pressure relief devices covered are those for protecting safety related components. Therefore, there is no change in requirements, only a clarification. Additional clarifications to Appendix I are discussed by the NRC in NUREG-1482, Section 4.3.9.

1994 Addenda Clarifications

As discussed in Section 4.3.9 of NUREG-1482, "Guidelines for Inservice Testing at Nuclear Power Plants," the NRC has allowed the use of clarifications provided in later editions and addenda of the Code without approval, provided that they are determined to be clarifications only and are documented in the IST program. The NUREG discusses clarifications made in Appendix I. The 1994 Addenda also includes clarifications concerning pump and check valve testing. These clarifications may be used without NRC approval, prior to the amended rulemaking.
ISTB 4.7.1(a) has been revised to clarify that parameters determined by analytical method shall meet the accuracy requirements of Table ISTB 4.7.1-1. Paragraph ISTB 5.2.1(e) and 5.2.3(e) were revised and now include clarification that vibration measurements are required to be compared to both the relative and absolute acceptance criteria in Table ISTB 5.2.1-1. Additionally, a footnote has been added to Table 5.2.1-1 to clarify that positive displacement pumps, excluding reciprocating pumps, should use the vibration acceptance criteria of centrifugal and vertical line shaft pumps. Previously, no vibration acceptance criteria were supplied for these pumps. ISTB 5.4 was clarified to read "within 3 months before the system is placed in an operable status." The previous wording, i.e., "within 3 months of placing the system," was confusing.

OMa-1988, Part 10, paragraph 4.3.2.4(a) states that check valve obturator movement may be observed by "a direct indicator such as a position indicating device, or by other indicator(s) such as changes in system pressure, flow rate, level, temperature, seat leakage testing or other positive means." The 1994 Addenda of the OM Code, provides additional clarification that "other positive means" includes nonintrusive testing results. Related requirements concerning nonintrusive technique qualification have been added to ISTC 2(b), Owner's Responsibility.

The instrumentation requirements contained in OM-1-1981, OM-1-1987, and the 1990 Edition of the OM Code are very confusing. Although, the 1994 Addenda's requirements for accuracy are somewhat more restrictive, it clarifies the requirements and Owners may use this clarification. There are no other related requirements.

1996 Addenda

The 1996 Addenda of the OM Code was published in April 1996. This Code addenda contains 13 actions approved by the O&M Main Committee affecting ISTA, ISTB, and ISTC. Although this Code Addenda has not been incorporated by the NRC in the proposed rulemaking, alternates that provide "an acceptable level of quality and safety" may be proposed in accordance with 10CFR50.55a(a)(3)(i). Table 2 summarizes code changes and clarifications that may be candidates for proposal to the staff. It should be noted that no addenda was published in 1995.

ISTC has been revised to allow Owners flexibility in establishing maintenance activities (which includes maintenance, testing, and examination) for check valves. As an alternate to the testing or examination requirements of ISTC, Owners may group check valves and establish a Condition Monitoring Program. Considering that the Maintenance Rule, 10CFR50.65, is required to be implemented by July 10th, the activities required by the Code may already be performed under the maintenance rule and the burden of implementing this change may be minimal and the benefit worthwhile.

Table 2 Advantageous Alternate Requirements in the 1996 Addenda

Advantageous Alternate Requirements in 1996 Addenda	Related Requirements
Check valve condition monitoring program in lieu of exercising.	ISTC 4.5.5, and Appendix II.
Check valve exercising in both directions.	ISTC 4.5.2, 4.5.4, and non-mandatory Appendix E.
Valve position verification and exercising requirement exclusion for safety and relief valves in ISTC 1.2.	None.
Deletion of the requirement for performing certain tests on safety and relief valves in sequence I 3.3.1, 3.3.2, 3.3.3, 3.3.5, 7.3.1, 7.3.2(?), 7.3.3, 7.3.5, 7.3.6.	None.
Testing of check valves in series, ISTC 4.5.6.	ISTC 4.5.7 and 6.2(f).
Testing of MOVs, Code Case OM-N-1	See proposed NRC Generic Letter .
Clarifications: Clarification in ISTB 6.2.1 that analysis in accordance with ISTB 4.6 may be used when pumps are in the alert range.	ISTB 4.6, 6.1, and 7.3.
ISTB 4.7.4, pump vibration measurement in the orthogonal direction.	None.
ISTC 4.3.2 and 4.3.3, containment isolation valve testing.	None.

The Code has been revised to require both a forward and reverse flow test, regardless of the safety function of the check valves, to ensure that obturator degradation is detected. The Code change also included some guidance in the Non-mandatory Appendix E concerning post-disassembly and inspection testing. Generic Letter 89-04, Position 2 requires, if possible, that partial valve stroking be performed after reassembly. Appendix E also requires a partial flow test if practicable. However, for valves that can be reverse closure tested, this test should also be performed following reassembly. Albeit this

code change increases the burden on Owners, they may wish to implement this change to ensure a higher level of reliability of their check valves.

ISTC 1.2, Exclusions, has been revised to include an exclusion from the valve position verification and exercising requirements for safety and relief valves. This Code change was written to address the testing of the main steam pressure relief valves with auxiliary actuating device (i.e., automatic depressurization system or ADS valves) in BWRs and would reduce the testing required for these plants.

Appendix I has been revised to delete the requirement to perform tests; other than the visual examination, seat tightness determination, and set-pressure determination; in sequence and before any maintenance or set-pressure adjustments. This code change should eliminate an unnecessary burden for Owners.

ISTC 4.5.6 has been revised to address testing a pair of check valves in series. Related requirements include ISTC 4.5.7, concerning corrective action, and 6.2(f), concerning documentation. This change reflects the guidance provided in NUREG-1482, Section 4.1.1 for preparation of a relief request.

The OM Committee has issued their first Code Case. This Code Case provides alternate rules for testing motor-operated valves, including alternate test frequencies. The NRC, in a Federal Register notice dated February 20, 1996, issued a proposed Generic Letter concerning the periodic verification of design basis capability of safety-related MOVs. In Attachment 1 of the proposed Generic Letter, the NRC states that it "would consider a periodic verification program that provides an acceptable level of quality and safety as an alternate to the current IST requirements for stroke-time testing and could authorize such an alternate, upon application by a licensee, pursuant to the provisions of 10CFR50.55a(a)(3)(i)." The NRC has considered the Code Case as an acceptable periodic verification program given three provisions. These include: (1) The benefits and potential adverse effects are considered when determining appropriate testing, (2) When the test interval is greater than 5 years, an evaluation of information from tests performed during the first 5 years should be performed to validate assumptions made in justifying the longer test interval, and (3) Licensees involved in risk-informed IST programs should specifically address the relationship of the Code Case to their risk-informed initiative.

1996 Addenda Clarifications

In addition to new or revised code requirements, the 1996 Addenda also contains a number of clarifications. The 1994 Addenda included a new provision for establishing new reference values when a pump is in the alert or required action range (ISTB 4.6). ISTB 6.2.2 concerning the required action range provided a direct reference to ISTB 4.6. ISTB 6.2.1 concerning the alert range was not revised, however. The 1996 Addenda, has provided this direct reference in ISTB 6.2.1 to ISTB 4.6 for clarification.

ISTB 4.7.4 has been revised to clarify that when measuring pump vibration, the measurements do not have to be absolutely in orthogonal directions, only approximately. This issue was subject of code interpretation, ASME File #OMI 93-7.

ISTC 4.3.2 and 4.3.3 have been clarified to require that containment isolation valves that also have a leak rate requirement based on other functions, such as reactor coolant pressure isolation, are tested in accordance with ISTC 4.3.3.

Additionally, included with the 1996 Addenda are numerous code interpretations related to pump and valve testing, e.g., concerning testing of thermal relief valves, using rms pump vibration readings, and testing of dual function check valves. The Code Addenda should be consulted for these clarifications.

CONCLUSION

This paper discusses some beneficial alternatives included in the 1992 to 1996 Addenda of the OM Code to the requirements currently contained in the 1989 Edition of Section XI referenced in 10CFR50.55a(b). Until the NRC approves and issues the amended rulemaking that would allow voluntary use of these later Code requirements, these alternatives can only be used if approved by the NRC through the use of 10CFR50.55a(a)(3)(i). The NRC will provide instructions in the revised rulemaking for the use and documentation of alternatives, such that approval pursuant to 10CFR50.55a(a)(3)(i) is no longer required.

REFERENCES

1. *Federal Register,* Volume 60, Number 88, Monday, May 8, 1995, "Regulatory Agenda; 4957, Codes and Standards for Nuclear Power Plants."

2. *Code for Operation and Maintenance of Nuclear Power Plants,* American Society of Mechanical Engineers, 1990 Edition through 1995 Addenda.

3. Generic Letter 89-04, "Guidance on Developing Acceptable Inservice Testing Programs," April 3, 1989.

4. NUREG-1366, "Improvements to Technical Specification Surveillance Requirements," R. Lobel and T.R. Tjader, December 1992.

5. NUREG-1482, "Guidelines for Inservice Testing at Nuclear Power Plants," P. Campbell, April 1995.

6. *Federal Register,* Volume 61, Number 34, Tuesday, February 20, 1996, "Proposed Generic Letter: Periodic Verification of Design-Basis Capability of Safety-Related Motor-Operated Valves (M93706); Opportunity for Public Comment."

Significant Issues and Changes for ANSI/ASME OM-1 1981, Part 1, ASME OMc Code-1994, and ASME OM Code-1995, Appendix I, Inservice Testing of Pressure Relief Devices in Light Water Reactor Power Plants

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ABSTRACT

This paper identifies significant changes to the ANSI/ASME OM-1 1981, Part 1, and ASME Omc Code-1994 and ASME OM Code-1995, Appendix I, "Inservice Testing of Pressure Relief Devices in Light-Water Reactor Power Plants". The paper describes changes to different Code editions and presents insights into the direction of the code committee and selected topics to be considered by the ASME O&M Working Group on pressure relief devices. These topics include scope issues, thermal relief valve issues, as-found and as-left set-pressure determinations, exclusions from testing, and cold setpoint bench testing. The purpose of this paper is to describe some significant issues being addressed by the O&M Working Group on Pressure Relief Devices (OM-1). The writer is currently the chair of OM-1 and the statements expressed herein represents his personal opinion.

Introduction

This paper describes changes to American Society of Mechanical Engineers (ASME), Operation and Maintenance (OM) Code, Appendix I, "Inservice Testing of Pressure Relief Devices in Light-Water Reactor Power Plants," hereafter referred to as the Code. It contains a summary of the chair's personal notes, comments and thoughts. Clarifications for formal Code interpretations, suggested Code changes, or Code inquiries should be submitted to the Working Group. In this way formal evaluation and documentation will be provided to the entire nuclear industry.

In general, ASME/ANSI OM (Part 1) 1987 and ASME OM Code-1990 are a reprinting of ANSI/ASME OM-1 1981, Part 1. Only renumbering changes occurred. ASME Omc Code-1994 and ASME OM Code-1995 contain significant changes. Proposed ANSI/ASME OMa Code-1996 will contain additional changes. Presently the OM Code is being reformatted for better readability by combining similar requirements from various sections. This paper will discuss these changes and reflect on current proposed changes and direction.

Scope Statement

Presently ASME Code-1995, Paragraph ISTA 1.1 Scope establishes the requirements for preservice and inservice testing and examination of certain components in light water cooled nuclear power plants. It identifies the components subject to testing or

examination, personnel responsibilities, test methods, test intervals, parameters to be measured and evaluated, acceptance criteria, corrective action, and record keeping requirements. Components covered by the Code include: a) pumps and valves that are required to perform a specific function in shutting down a reactor to the safe shutdown condition, in maintaining the safe shutdown condition, or in mitigating the consequences of an accident; b) pressure relief devices that protect systems or portions of systems that perform one or more of these three functions; c) dynamic restraints (snubbers) used in systems that perform one or more of these three functions.

The ASME OM-1995 Code scope statement includes testing of all safety related relief valves. Prior to ASME OMc Code-1994, however, the wording "to protect systems or portions of systems" was not included in the Appendix I scope statement. This intent is reflected in previous Code editions. Prior to ASME OMc Code-1994, the requirement for relief valves to protect systems within the owner's inservice test program needed clarification. Part 1 of ASME OM-1987 was invoked by referencing Part 10 of ASME OM-1987; Part 10 was endorsed in the 1988 Addenda of Section XI of the ASME Boiler and Pressure Vessel Code.

The scope of Part 10, ASME OM-1987 contains the words "pressure relief device covered are those for protecting systems or portions of systems that perform a required function in shutting down the reactor to cold shutdown condition, in maintaining the cold shutdown condition, or in mitigating the consequences of an accident". ASME OMc Code-1994 scope statement was revised to be more consistent with the previous scope referencing documents.

ASME OMc Code-1994 also changed the scope statement from "cold shutdown" to "safe shutdown", since a majority of nuclear plants have a licensing commitment for safe shutdown as Mode 3. The owner should identify and document the required shutdown mode. Overpressure protection was changed from the ASME Boiler and Pressure Vessel Code Section III definition to become broader and more generic. Some Working Group members imply ASME OM Code-1995 scope statement references "cold shutdown" condition, while Appendix I reference "safe shutdown" condition. Hence because Appendix I is a subsection of ISTC, then owners should test to the "cold shutdown". Current OM code format changes should correct this.

Requirement for Testing Additional Valves

Additional testing costs are incurred when the as-found set-pressure determination test fails to meet acceptance criteria during the first valve actuation. Upon failure of the first actuation, the owner is required to test two additional valves of the same valve group. Because of this sample expansion, it is important to evaluate, establish, and document the owner's acceptance criteria for as-found pressure determination prior to any Code testing. This would provide specific as-found lift limits based upon system and valve design basis or technical specifications in accordance with ASME OM Code-1995, paragraph I 3.1.1(e), Acceptance Criteria.

Presently ASME OM Code-1995 allows the owner to develop the owner-established set-pressure acceptance criteria for the as-found pressure determination test. The owner may choose between either the $\pm$ three percent of the valve nameplate set-pressure or $\pm$ owner-established set-pressure acceptance

criteria. The owner's choice is "either" of these two conditions and not the most conservative. For either condition a plus and minus set-pressure acceptance criteria is required. The owner should verify and document that the system design or hydrostatic test pressures are greater than the as-found acceptance criteria.

For high energy relief valves which are Class 1 and Main Steam relief devices, ± three percent of valve nameplate set-pressure is recommended as acceptance criteria. Many plants are changing their Technical Specification to reflect this ± three percent as-found acceptance criteria. The negative as-found acceptance criteria tolerance for these high energy relief devices reflects the manufacturer's tolerance and early relief valve lifts that may cause over cooling events. I recommend the owner follow this ± three percent of valve nameplate set-pressure, however the owner can follow the second option and develop the ± tolerance limit of the owner established set-pressure acceptance criteria.

For low energy relief valves which are Class 2 and 3, excluding main steam relief devices, the ± tolerance limit of the owner-established set-pressure acceptance criteria is recommended. All as-found acceptance criteria should be documented. If applicable, calculations should be shown or justification provided regarding how the as-found acceptance criteria values are established.

When attempting to define an as-found acceptance value criteria for low pressure relief valves, ± three percent of manufacturer's tolerance could be used as the ± tolerance limit of the owner-established set-pressure acceptance criteria. Manufacturer's tolerances can be obtained from original design specifications.

In ASME OM Code-1990, prior editions and addenda, the requirement to test additional valves was based on exceeding the "*stamped set-pressure acceptance criteria by 3% percent or greater*". This original statement was developed for testing high energy relief valves. Long-standing Working Group members of the Code Committee indicate that low energy relief valves acceptance criteria was not originally considered in the ASME OM Code-1990 and prior edition and addenda. If followed verbatim, the owner would incur additional and unnecessary valve testing for low energy relief valves.

For example, acceptance criteria prior to ASME OM Code-1990, as "stamped set-pressure criteria by three percent or greater", may be inadequate for low energy relief valves. The industry does not have the technology to test within these limits. In addition, valve manufacturer's tolerances are greater than these limits. Three percent of stamped set-pressure is difficult for the owner to measure for low energy relief valves. A 25 psig stamped setpoint with the as-found setpoint determination test at 26 psig would be considered a failure because 103 percent of 25 psig is 25.75 psig. This is unacceptable because manufacturer tolerance exceeds this acceptance criteria. The author recommends following the requirement of ASME OM Code-1995 for low energy relief devices and documenting the as-found acceptance criteria.

Paragraph I 1.3.3(e)(2) in the ASME OM Code-1990 and previous editions and addenda states "...any valve exceeding its stamped set-pressure by three percent or greater shall be repaired or replaced..." This states that three percent of stamped set-pressure is the acceptance criteria, hence would cause the

owner to test additional valves. ASME Code Inquiry OMI-92-26 requested clarification, and an inquiry was submitted. It read:

> ***Question:*** *In the OM-1990 Code, Appendix I, para. I 1.3.3(d), I 1.3.3(e), I 1.3.5(d), I 1.3.5(e) refer to "stamped pressure criteria". In OM-1990 Code, Appendix I, para. I 1.3.3(e)(2) and I 1.3.5(e)(2) refer to "stamped set-pressure". Do these two phrases mean the same thing?* ***Reply:*** *Yes.*

This response, although correct, may be inadequate because the owner may not be able to measure the three percent of "stamped set-pressure", but only three percent of "stamped set-pressure criteria". For this case, criteria is defined by the owner. In addition, the set-pressure acceptance criteria, stamped set-pressure criteria, and stamped pressure criteria of ASME OM Code-1990 and prior editions, for Valve Not Meeting Acceptance Criteria paragraphs, (including similar previous edition) referenced steps are incorrect. The word "criteria" is subjective and could therefore be defined by the owner. The Working Group will develop a Code Case to correct this.

As-left Set-Pressure Acceptance Criteria

As-left set-pressure acceptance criteria as defined in ASME OM Code-1995, paragraphs I 4.1.1(i), I 4.1.2.(i), I 8.1.1(i), and I 8.1.2(i), Number of Tests, requires "... a minimum of two consecutive openings within acceptance criteria." This requirement is not the same as the as-found set-pressure acceptance criteria. The owner's basis document for relief valve lift pressures should contain both as-found set-pressure determination acceptance criteria and as-left acceptance criteria. The manufacturer's tolerance may be acceptable for as-left set-pressure acceptance criteria. If the as-found is within this as-left set-pressure acceptance criteria, then only one more additional lift within the as-left acceptance criteria is required. The author recommends that the owner-established as-left acceptance criteria be set equal to manufacturer's tolerances, provided it is within the design basis of the system.

Set-Pressure Determination Methodology

Set-pressure determination methodology depends on valve manufacturer. The author recommends that methodology for measuring set-pressure be obtained from the valve manufacturer. Normally, for gas relief valves, there is a defined "pop" sound. For liquid reliefs, the set-pressure can be a pencil stream, excessive flow, or even a two step flow change. Reading the vendor manual or asking the manufacturer will avoid confusion. If the set-pressure determination methodology is atypical, then it should be evaluated and documented in the test procedure or valve basis.

Valve Groups

The owner is required to define the valve group based on manufacturer, type, system application, and service media. Documenting each valve group is required because of the group's subjective basis. If a group is too small, then the owner will be penalized because at least 20 percent of the group needs to be tested each period. The owner must round up when obtaining the 20% of valve groups, hence causing more testing for smaller groups. If the group is too large and two or more failures occur, then the owner will need to test the remaining valves. An optimum group size should be carefully determined.

These groups should be established along with the test schedules. If required, two additional valves for testing can easily be identified.

When to Test Additional Valves

If the as-found set-pressure determination test fails, then the Code requires two additional valves be tested. The Working Group received inquiries about when these two additional valves are expected to be tested. The Code is intentionally silent on this issue. The owner is expected to test two additional valves within a reasonable time period. This period is to be established by the owner. If the cause of failure is believed to be generic with high safety significance, then the owner should evaluate or correct the problem immediately. However if the failure is believed not to be generic or of low safety significance, then the owner is not required by the Code to extend a forced outage, shutdown the plant, or procure replacement valves and parts. The two additional valve tests should occur per planned schedules. The committee felt such a time limit to test these two additional valves might cause the owner to be reluctant to perform on-line testing, or even delay refueling testing. With the Code being silent on testing of additional valves the owner should evaluate all valve failures, taking the appropriate actions and documenting their responses.

No ISTC Stroke Timing or Stem Verification

In 1993, owners questioned the technical basis to stroke time and perform position indication verification of safety relief valves in accordance with ISTC. The Working Group agreed with part of this and since ASME OM Code-1995, Appendix I, requires these tests, the Working Group decided to exclude safety relief valves from the requirements of ISTC. There was no need to follow OM ISTC section when Appendix I required these tests. Appendix I requires stem position indication verification. Stroke timing relief valves was considered ineffective since relief valves are fast-acting. Fast-acting is any valve that strokes under two seconds. These changes have been approved by the Working Group and ASME OMa Code-1996 will reflect these ISTC exclusions. The Proposed ASME OMa Code-1996, exclusions paragraph should read *"Pressure relief valves are excluded from the requirements of ISTC, paragraph 4.1, Valve Position Verification and ISTC, paragraph 4.2, Inservice Exercising Test"*. In addition, Table ISTC 3.6-1 for Class C, Safety and Relief Valves will reference Appendix I.

Thermal Relief Valve Issue

Some owners have interpreted that thermal relief valve tests are not required while other owners feel they are required. Within the last three years, both owners and inspectors became more concerned about thermal relief test requirements and required testing of these valves per Appendix I. Some Working Group members reported it was never the intent of the Code to require testing of thermal relief valves. However it is the opinion of the author that they are covered by scope statement, and testing should be in accordance with the requirements of Appendix I.

The scoping statement wording, "The pressure relief devices covered are those for protecting systems or portions of systems that perform a required function in shutting down a reactor..." was added to ASME OMc Code-1994 to clarify the intent of the Code. Some owners justified the exclusion of thermal reliefs because this wording was missing from their committed edition. Others would not test these thermal relief valves because the

component being served does not need to be "protected" when in service. Because the component was subjective to normal system pressure by inservice conditions, the thermal relief valves had no safety significance. Isolation occurs only during a maintenance function, when it is out of service. However, during maintenance, over-pressurization could occur and damage due to a faulty thermal relief valve could go unnoticed.

The thermal relief valve test issue was addressed by a Code inquiry. The inquiry is to be published in ASME OMa Code-1996. In March of 1995 the Subcommittee on Valves approved the following:

> ***Inquiry:*** *Is it the requirement of the ASME OM Code-1995, Appendix I, OMc Code-1994, Appendix I; OM-1987, Part 1 (as referenced per Section XI); or ANSI/ASME OM-1 1981 to require testing of Class 2 and Class 3 relief valves installed in systems that 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; for pressure relief valves whose only overpressure protection function is to protect isolated components from fluid expansion caused by changes in fluid temperature?* ***Response:*** *Yes, provided that they fall within the scope of ISTC 1.1. as determined by the Owner.*

Since the Subcommittee on Valves agreed that thermal relief valves need to be tested, an appropriate test frequency for this subset needed to be developed. A proposed Code change and Code Case would eliminate the two additional valve tests following as-found set-pressure determination (first actuation) failures, currently required in Appendix I for relief valves. Each thermal relief valve would be tested at least once every 10 years unless performance data indicates that more frequent testing is necessary. The thermal relief valve Code Case would resolve the testing frequency issue in a timely fashion. The Code Case will allow owners to test these valves with no penalty of testing additional valves. Failure databases for these thermal relief valves support this extended frequency. Testing frequency has not changed for Class 1 thermal relief valves.

The approved thermal relief valve Code Case for ASME OM Code-1995 is:

> ***Inquiry:*** *What alternative to ASME OM Code-1995 Appendix I paragraph 1.3.5 (a),(b), and (c) may be used for Class 2 and Class 3 pressure relief valves, which are required to be tested per ASME OM Code-1995, Appendix I, paragraph I 1.1, whose only overpressure protection function is to protect isolated components from fluid expansion caused by changes in fluid temperature?*
>
> ***Reply:*** *It is the opinion of the Committee that in lieu of the requirements specified in ASME OM Code-1995, paragraph I 1.3.5, (a),(b), and (c) testing for Class 2 and Class 3 pressure relief valves whose only overpressure protection function is to protect isolated components from fluid expansion caused by changes in fluid temperature shall be performed once every ten years on each valve unless performance data indicates that more frequent testing is needed to assure valve function. In lieu of tests, the Owner may replace these valves every ten years unless performance data indicates that more frequent replacement is needed to assure valve function.*

This thermal relief valve Code Case and Code is expected to be published in the ASME OMb Code-1997.

Change Wait Time Between Lifts from 10 to Five Minutes

The Working Group members have evaluated the advisability of decreasing the minimum time between successive lifts when testing safety or relief valves. After evaluating similar tests at 10 minute intervals and then at five minute intervals for high energy relief valves, there was no significant effect on lift set point. Based on this research from Westinghouse and Wyle Labs, relief valve minimum hold times have been reduced to five minutes. This change and a similar Code Case is expected to be published in ASME OMb Code-1997.

BWR Control Rod Drive Exclusion

In early 1994, a BWR owner brought to the O&M committee's attention a concern regarding the five year replacement frequency for Boiling Water Reactor (BWR) Control Rod Drive (CRD) rupture disks. There are approximately 145 BWR CRD rupture disks for each General Electric unit. A Working Group member developed and submitted an engineering evaluation of the safety significance of these CRD BWR rupture disks. The evaluation concluded that these rupture disks could be excluded from Appendix I requirements. The evaluation was reviewed and approved by the BWR Owner's Group. This exclusion should save owner replacement costs, reduce radiation exposure and standardize inspections.
This exclusion is now being balloted.

Check Valve Versus Relief Valve

A simple check valve is sometimes used as a vacuum breaker. O&M working groups are attempting to distinguish between a simple check valve or a check valve used as a relief valve.

The concern is whether these valves should be tested in accordance with Appendix I or ISTC. The Code will be modified to clarify this situation. A summary of what has been presented to the Check Valve and Relief Valve Working Groups, but not yet approved is:
If a "check" valve has provisions for adjusting the relief set-pressure and is capacity certified, then it shall be classified as a pressure relief device and tested in accordance with Appendix I. However, if a check valve does not have provisions for adjusting the relief set-pressure or is not capacity certified, then it shall be classified as a check valve and tested in accordance with ISTC. The Appendix I test frequency being considered for these vacuum relief valve applications is once every five years, unless performance data requires more frequent testing.

Cold Setpoint Bench Test

Many manufacturers have stamped pressure relief devices with both a cold setpoint bench test (CSBT) pressure and set-pressure. An inquiry was submitted to determine whether a temperature correlation was required for the cold setpoint bench test, when the relief valve in question was being tested at ambient condition. The inquiry was written in response to a Appendix I question on Paragraph 4.3 or 8.3, Alternative Test Media. These paragraphs state "Pressure relief device may be subjected to set-pressure tests and seat tightness tests using a test medium (fluid and temperature) other than that for which they are designed, provided the test complies with paragraphs...."

Upon investigation of manufacturers practices, the Working Group determined
that a correlation is required. To date only one manufacturer is developing these correlations. It appears that manufacturers have been stamping CSBT pressures using engineering judgment. These relief valves are normally installed on Class 2 and 3 on systems where elevated temperatures or liquid may be found. Owners should be aware of this when pursuing design changes and ordering replacement valves. The owner should specify that a temperature correlation and its basis accompany each relief valve purchased that is expected to be tested at a cold bench setpoint. Presently the Working Group concurrent with ASME, Section III, is investigating and developing an action plan to resolve this issue. It is the opinion of the author that owners will need to obtain correlations. Presently this is considered an issue of code compliance, however the owner should evaluate cold bench setpoint testing to ensure that no unresolved safety question exists.

Some OM-1 members argue that these paragraphs require both fluid and temperature differences for a correlation, but that since the fluid is not changed, no correlation is required. Others members argue the Code's intent is perfectly clear and this is a manufacturing issue, not an IST issue. Section III and other construction codes are pursuing resolution of this issue.

Acknowledgments

The topics presented here are highlights. Time constraints and paper limit me from discussing other issues. For specific inquiries about Code interpretations, written requests should be submitted to the Secretary, Operation and Maintenance, Main Committee, ASME, 345 East 47th Street, New York, N.Y. 10017-2392. Written requests are preferred over verbal requests. Written requests are assigned a tracking number, forcing Working Groups to evaluate and produce a timely written response. Verbal requests for information are not official and may go unnoticed or not addressed.

I would like to thank all code members for their time and effort in code development. All members, except ASME staff, are unpaid volunteers. Many individuals are involved in inquiries, Code Cases and the Codes revisions and perform these tasks on their own time. I would also like to thank the individual Working Group members' organizations for providing the time and funding to attend these Working Group meetings.

OM meetings are scheduled four times each year and all visitors are welcomed.
The OM committee process is a consensus process which establishes requirements based on our personal knowledge and integrity. This consensus process takes time but provides superior results. Hope to see you at the next OM-1 Working Group meeting and let us know what we can do to serve you better.

Code Cases for Implementing Risk-Based Inservice Testing in the ASME OM Code

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INTRODUCTION

Historically inservice testing has been reasonably effective, but quite costly. Recent applications of plant PRAs to the scope of the IST program have demonstrated that of the 30 pumps and 500 valves in the typical plant IST program, less than half of the pumps and ten percent of the valves are risk significant.

The way the ASME plans to tackle this overly-conservative scope for IST components is to use the PRA and plant expert panels to create a two tier IST component categorization scheme. The PRA provides the quantitative risk information and the plant expert panel blends the quantitative and deterministic information to place the IST component into one of two categories: More Safety Significant Component (MSSC) or Less Safety Significant Component (LSSC).

With all the pumps and valves in the IST program placed in MSSC or LSSC categories, two different testing strategies will be applied. The testing strategies will be unique for the type of component, such as centrifugal pump, positive displacement pump, MOV, AOV, SOV, SRV, PORV, HOV, CV, and MV.

A series of OM Code Cases are being developed to capture this process for a plant to use. One Code Case will be for Component Importance Ranking. The remaining Code Cases will develop the MSSC and LSSC testing strategy for type of component. These Code Cases are planned for publication in early 1997. Later, after some industry application of the Code Cases, the alternative Code Case requirements will gravitate to the ASME OM Code as appendices.

BACKGROUND

Historically inservice testing has been reasonably effective, but quite costly. A 1993 EPRI report surveyed eight operating nuclear power plants and found that they spent on an average 812 man-hours per year performing IST pump testing and 9072 man-hours per year performing IST valve testing. That is almost five equivalent people working full-time at each plant performing only IST!

Recent applications of plant PRAs to the scope of the IST program (1995 EPRI report) have demonstrated that of the 30 pumps and 500 valves in the typical plant IST program, less than half of the pumps and ten percent of the valves are risk significant. The risk insignificant valves include some MOVs, some AOVs, lots of CVs, and lots of SRVs.

Since every plant already has a PRA, because of the requirements of NRC Generic Letter 88-20, every plant can benefit from this focusing of plant resources on the true risk significant pumps and valves in the IST program and decreasing the plant resources spent on those components that are risk insignificant (sometimes by many orders of magnitude).

ASME APPROACH FOR CHANGE

The way the ASME plans to tackle this overly-conservative scope for IST components is to use a combination of the PRA and plant expert panels to create a two tier IST component categorization scheme. The PRA provides the quantitative risk information and the plant expert panel blends the quantitative and deterministic information to place the IST component into one of two categories: More Safety Significant Component (MSSC) or Less Safety Significant Component (LSSC).

SOURCE OF THE RISK-BASED IST PROCESS

Based on substantial research performed by the ASME on applying risk technology to ISI over the past decade, the ASME O&M Committee commissioned the ASME Center for Research & Technological Development (CR&TD) to develop a process for applying the risk technology available to the nuclear industry to IST. The CR&TD created a Research Task Force who developed a process over the period 1993 through 1995 that has three basic parts: (1) component importance ranking; (2) focused testing strategies for MSSCs; and (3) generic testing strategies for LSSCs. The results of this research project was published in December 1995 as ASME CRTD Vol 40-2 Report.

Concurrent with this ASME research effort were two key complementary industry efforts by the ten plant Risk-Based IST Pilot Project (results published in December as EPRI TR-105869 Report) and the Vogtle ECCS CV Demonstration Project (results published in October 1995 as W-OG WCAP-14358 Report). Thus the ASME process was verified and validated to some degree before the research report was published.

SCOPE OF THE CURRENT IST PROGRAM

As each Owner knows, the IST Code of Record for each nuclear power plant is currently the ASME Boiler & Pressure Vessel (B&PV) Code, Section XI, Subsections IWP and IWV. Because the B&PV Code uses ASME Code Class 1/2/3 for scoping, that is the scope of the IST program and all IST pumps and valves within that boundary have been treated equally. Note that the O&M Committee has rescoped the IST program in the ASME OM Code (and OM Part 6 & 10) to use safe shutdown and accident mitigation as the criteria, but that scope is not yet effective. One noteable exception to the equal treatment, is the comprehensive pump test in the ASME OM Code, which attempted to differentiate between standby and operating pumps. In fact what has been going on for the last decade within the O&M Committee is to factor in other independent variables, such as service condition and performance.

SCOPE OF THE FUTURE IST PROGRAM

In the future an alternative (risk-based IST) to the scope of the IST program will have one major independent variable (based on risk) and several minor independent variables (based on service condition and performance). The scope will be divided into two categories: MSSC and LSSC using the component importance ranking process. Then MSSC and LSSC testing strategies will be developed for each IST component type. Factors such as service condition, performance, and test effectiveness will almost certainly be utilized in the development of testing strategies for each component type.

Note that when the NRC decides to reference the ASME OM Code vice the ASME B&PV Code in 10 CFR 50.55a for IST programmatic require-ments, the ASME Code Class 1/2/3 scoping boundary will likely go away. In its place will be the safe shutdown and accident mitigation boundary. The PRA, the EOPs, and the SAR will all be important in the definition of that boundary.

TESTING STRATEGY FOR MSSCS

Note that only a few IST components are MSSCs - typically 20 pumps and 100 valves. These are the components that are so important for a plant that merely testing them to see if they have failed is not good enough. We need to devise a testing program to track precur-sors to those failures. Thus an OM Code Case is being developed for each type of component - checkvalves, pumps, MOVs, PORVs, AOVs, SOVs, SRVs, etc.

The "technical heart" of these component OM Code Cases is technical data on the configuration varieties of these components, their failure modes and causes, and the development of effective testing strategies. NIC has been very useful in producing the needed data for this checkvalve "technical heart". The O&M Committee intends to work closely with the various nuclear industry component user's groups to develop this "technical heart" quickly for all the various component types.

TESTING STRATEGY FOR LSSCS

This series of OM Code Cases for each component will also specify a "relaxed" testing scheme for LSSCs - typically 10 pumps and 400 valves at the average plant, provided that they demonstrate satidfactory performance as measured by pre-established performance indicators. These are the components that need some sort of periodic exercise to ensure that they have not already failed. Many of these components are routinely run or exercised during the plant refueling cycle, so the plant already knows their status (without an extensive performance oriented IST surveillance test). The concept is to apply an economical testing strategy. Perhaps that might be periodic exercising, staggered testing, or even test sampling in some sort of grouping scheme.

SCHEDULE FOR OM CODE CASES

The goal of the ASME O&M Committee is to create this series of OM Code Cases in 1996 and perhaps 1997. The first Code Case will be the one for "component importance ranking". It will followed by companion Code Cases for MOVs, CVs, AOVs, SOVs, SRVs, and pumps. Ultimately by the end of 1997 or early 1998, there will be a Code Case that will address every type of component in the IST program.

COST/BENEFIT FOR RISK-BASED IST

Clearly better IST testing for the MSSCs will improve reactor safety and more economic IST testing for the LSSCs will lower plant O&M costs, thereby creating a win-win partnership between industry and the NRC — with the ASME as the bridge between industry and the regulator.

ULTIMATE REVISION TO THE ASME OM CODE

After some trial period of using these OM Code Cases, the O&M Committee currently plans to convert them into appendices to the ASME OM Code. Conceiveably, applying risk technology to the plant IST program

could become such a positive cost/benefit from the point of view of safety and economics that these planned appendices might become the sole mandatory requirements, however that would likely be the year 2005 or 2010.

Enhancing the Effectiveness of IST Through Risk-Based Techniques

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ABSTRACT

Current IST requirements were developed mainly through deterministic-based methods. While this approach has resulted in an adequate level of safety and reliability for pumps and valve, insights from probabilistic safety assessments suggest a better safety focus can be achieved at lower costs. That is, some high safety impact pumps and valves are currently not tested under the IST program and should be added, while low safety impact valves could be tested at significantly greater intervals than allowed by the current IST program.

The nuclear utility industry, through the Nuclear Energy Institute (NEI), has developed a draft guideline for applying risk-based techniques to focus testing on those pumps and valves with a high safety impact while reducing test frequencies on low safety impact pumps and valves. The guideline is being validated through an industry pilot application program that is being reviewed by the U.S. Nuclear Regulatory Commission. NEI and the ASME maintain a dialogue on the two groups' activities related to risk-based IST.

The presenter will provide an overview of the NEI guideline, discuss the methodological approach for applying risk-based technology to IST and provide the status of the industry pilot plant effort.

A REGULATOR'S PERSPECTIVE ON NRC'S PARTICIPATION IN THE OPERATIONS & MAINTENANCE COMMITTEES

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PURPOSE

The purpose of this paper is to give a regulator's perspective on the U.S. Nuclear Regulatory Commission's (NRC) participation in the Operations and Maintenance committees. It discusses the consensus process and the regulatory process associated with the development and use of codes and standards.

INTRODUCTION

As a regulator fairly new to the American Society of Mechanical Engineers (ASME) Operations and Maintenance (O&M) Committee process, I do not have a personal historical perspective as do many of the longer-term, and highly respected, members of the O&M Committee. However, as Branch Chief of the Mechanical Engineering Branch, Division of Engineering, in the Office of Nuclear Reactor Regulation at the NRC for just over two years, I have responsibility for the regulatory agency's review of licensee actions involving the products that come from the efforts of the O&M Committee, as well as responsibility for portions of our activities of interest to other ASME Code groups such as Section III, Section XI, and Qualification of Mechanical Equipment. As a result, I have learned a great deal about the code process in a short time. I would like to give you my perspectives on the process and provide a few thoughts on our direction for the future.

BODY

Historical

ASME has been involved in the development of codes and standards for over 100 years. An essential facet of that development has been the public consensus based process that is founded on strong technical principles and that follows established procedures.

In the 1960's, the Atomic Energy Commission (AEC), NRC's predecessor, encouraged the development of nuclear codes and standards for the materials, design, fabrication, and construction of critical reactor systems and components, and to provide standards to ensure their structural integrity. Out of this developed Section III the ASME *Boiler and Pressure Vessel Code*. The reactor coolant pressure boundary was designated Code Class 1 by the AEC. Inspection throughout the service life (including preservice) was added, as were similar requirements for assuring the quality and integrity of the emergency core cooling systems and cooling water systems designated by the AEC as Code Class 2 and Code Class 3, respectively. In 1973, the requirements for inservice testing (IST) of pumps and valves were added to Section XI of the ASME *Boiler and Pressure Vessel Code*. Later, the responsibility for the IST requirements was transferred to the Operations and Maintenance Committee, and are now in the *Operations and Maintenance Code*.

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

Regulatory Agency's Use of Industry Codes and Standards

The Office of Management and Budget (OMB), under the Executive Branch of the government, in conjunction with the Department of Commerce, provides direction to regulatory agencies for using industry codes and standards in OMB Circular A-119, "Federal Participation in the Development and Use of Voluntary Standards." The OMB Circular A-119 encourages regulatory agency participation on the various industry codes and standards committees and the use of the consensus documents where appropriate and not specifically prohibited by law. The use of industry codes and standards eliminates the cost to the Government of developing its own standards and is considered to be of greater economy and efficiency.

OMB Circular A-119 is consistent with recent Government reform objectives of the National Performance Review. NRC is committed to involvement in the ASME process and to endorsing appropriate ASME Codes (with limitations, if necessary) as a preferred alternative to developing agency regulations. Many of the ASME working groups and committees dealing with Code matters affecting the nuclear industry include an NRC member. NRC involvement with ASME includes not only participation in working groups and committees, but can include direct financial support such as the research grants that were given to ASME - Research for the efforts to develop code criteria for risk-based inspection and testing.

OMB Circular A-119 establishes a policy for agency representatives serving on codes and standards working groups or committees. Individuals who, at Government expense, participate in standards activities do so in their governmental capacities as specifically-authorized agency representatives. Each agency is responsible for establishing procedures to ensure that the participants will, to the extent possible, ascertain the views of the agency on matters of paramount interest and will, as a minimum, express views that are not inconsistent or in conflict with established agency views. The guidance differs from the ASME Committee membership guidance, which indicates, for example, that Main Committee members participate, including voting, as individuals and not as representatives of their employer or of any other organization. Therefore, a regulator serving with the ASME has additional constraints in the interest of the government.

ASME Consensus Process

The consensus process used by the ASME Committees is set in motion by the identification of a need for a technical change or for a new code, standard, or guide. The action is given to a technical subcommittee, subgroup, or working group for development of the technical changes or the new requirements. The revised or new code, standard, or guide moves forward to the appropriate Main Committee, through a letter ballot. If the ballot does not pass, the item is returned to the lower-tier group for further changes before moving forward again. If the ballot passes the Main Committee, it moves forward to the Board on Nuclear Codes and Standards (BNCS). If the item does not pass the BNCS, it is returned to the lower-tier group and the process repeats until the item passes or is dropped from further consideration. Subsequent to the BNCS letter ballot, it is published for public comment in the *ANSI Reporter* and in the ASME *Mechanical Engineering* magazine. If there

are adverse comments, the item may be returned to the lower-tier group (e.g., Working Group). If there are no adverse comments, or if all of the comments are satisfied, the item is routed to the Board of Standards for review. Finally, the item is published.

NRC Rulemaking Process

Ideally, the NRC has participated in the consensus process at each level so that potential staff concerns with a code, standard, or guide could be addressed before final publication by the ASME. In most situations, the ASME product is now in an official form that may be incorporated by reference into the NRC regulations, with limitations if appropriate. The rulemaking process for the ASME Code is managed by the Office of Nuclear Regulatory Research (RES) with support from the Office of Nuclear Reactor Regulation (NRR). Generally, the RES Project Manager responsible for the rulemaking will be a participant in one or more working groups or committee and is very familiar with the code activities associated with the most recently published edition and addenda. A proposed rule change package is prepared in accordance with NRC procedures for rulemaking, regulatory analysis, backfit considerations, and paperwork reduction requirements. The preparation effort is extensive and can take a year or more. It is desirable to prepare a rule change at least once every three years to comport with the issuance of a revised code edition (i.e., the ASME Codes editions are issued every three years, incorporating the annual addenda issued since the last edition).

The proposed rule is reviewed by several offices within the NRC (e.g., Nuclear Reactor Regulation, Analysis of Events and Operational Data, General Counsel, Administration, Information Resources Management) before it is presented to the NRC's Committee for Review of Generic Requirements (CRGR). The Advisory Committee on Reactor Safeguards (ACRS) is provided an opportunity to review the proposed rule. The ACRS may defer review on a proposed rule until the final package has been prepared. The proposed rule must then be submitted to the Executive Director for Operations prior to its publication in the *Federal Register* for public comment. In the new information age, the proposed rules are also entered into the FEDWORLD bulletin board and commentors may submit comments electronically. During the comment period, a public meeting may be held if the proposed rule represents a major change.

Once the comment period has ended, the Project Manager begins development of a final rule change package. The comments are addressed and changes are made as deemed appropriate. The Agency's review and approval process for the final rule is similar to that process followed for the proposed rule. Subsequent to a final rule, the NRC may hold a public workshop to answer questions related to implementation of the revised rule.

Rulemaking is a very important function of the NRC as a regulatory agency. The NRC has the statutory authority under the Atomic Energy Act (AEA) to ensure that the operation of nuclear power plants is consistent with the agency's mandate to protect the health and safety of the public. The AEA gives the NRC the authority to promulgate rules and regulations to accomplish its function. The rulemaking process itself is highly regulated through the Administrative Procedures Act and the actions of the Office of Management and Budget. The "incorporation by reference"

of an industry code has statutory authority in 5 *United States Code* 522(a). The process for incorporation by reference must be in accord with 1 CFR 51. It is a way to shorten otherwise long regulations and essentially means that the code itself becomes the regulation (i.e., it is as if the code was printed directly into
10 CFR 50.55a).

Generic Letters and Regulatory Guides

The NRC has mechanisms other than rulemaking to establish or communicate generic requirements, guidance, requests, or staff positions. In the area of codes and standards, generic letters and regulatory guides are used to some extent. For example, Generic Letter (GL) 89-04, Supplement 1, endorsed NUREG-1482, "Guidelines for Inservice Testing at Nuclear Power Plants," which gave guidance in the use of the O&M Standards. GL 89-10 gave guidance on testing motor-operated valves (MOVs). GL 95-05 was issued to request information from licensees related to possible pressure locking of power-operated valves. The recent generic letter on periodic verification of MOVs endorses (with qualification) the first O&M Code Case on periodic testing of MOVs (OMN-1). This Code Case moved through the ASME consensus process rapidly due to: (1) the intensity of interest, (2) the strong leadership of the OM-8 Working Group chair, and (3) the active participation of NRC and industry throughout the process.

Regulatory guides are also used to indicate the acceptability of ASME Code Cases for use by licensees as alternatives to the code. The applicable regulatory guides are listed in Footnote 6 to 10 CFR 50.55a; however, no regulatory guides have yet been issued on O&M Code Cases. The NRC has recently determined that the process for issuing or reviewing regulatory guides will follow a process similar to rulemaking and allow for a public comment period on a proposed regulatory guide or revision before it is finalized.

Both generic letters and regulatory guides are subject to the guidance in NRC NUREG/BR-0058, "Regulatory Analysis Guidelines of the U.S. Nuclear Regulatory Commission," which was revised in November 1995 to address implementation of the NRC's safety goal policy. The guidance delineates a process for assessing potential generic actions. The NUREG includes a discussion of the background of the NRC's regulatory analysis policy that stems in large measure, not from statutory requirements, but from a number of Executive Orders with which NRC, as an independent agency, is not required to comply but basically does comply. The revised guidance increases the likelihood that the NRC will only impose requirements that substantially affect safety. Many of the ASME Code changes represent improvements but not substantial safety benefits and, hence, would not be subject to imposition.

What if the NRC disagrees with the ASME Committee's view? Conversely what if the ASME Committee disagrees with the NRC participant's view? One example of such a disagreement concerned the alternate piping design criteria. ASME published new rules for piping design in the 1994 Addenda to Section III that significantly relaxed the design allowable stress limits for dynamic loads such as earthquake loads. Staff representatives at various levels opposed the changes because of the agency's concerns that there were unresolved technical issues. The NRC took exception to the new rules in letters to ASME issued in December 1994 and May 1995 and

is not prepared to initiate rulemaking that would ensorse this portion of the ASME Code. The ASME and NRC met in May 1995 to discuss the issues. The NRC initiated a technical review of the criteria by an independent contractor and ASME established a special working group to reexamine the rules and consider the technical review results, when available. Eventually, a resolution may be reached that would result in changes that are satisfactory to all involved parties.

The overall Code change and rulemaking process is lengthy and takes time, both for ASME Committees and NRC. However, the entire process through ASME and the NRC before imposition of requirements on licensees assures wide participation, consensus, and opportunity for public comment. NRC has a responsibility to the public and to Congress to promulgate rules that it believes are necessary to accomplish the agency's statutory mission of protecting the health and safety of the public.

Future Directions

As you are aware, the current regulations endorse the 1989 Edition of Section XI to the Code which also references portions of the 1988 addenda to the 1987 OM Standards with limitations for inservice testing of pumps and valves and inservice inspection of snubbers. The NRC has been working on a rule change that may result in imposition of only portions of later editions of the code, with provisions to allow licensees to voluntarily update to later editions that the NRC has determined are acceptable. Imposition could only be mandated if those portions of the Code meet the agency's backfit criteria. If only portions of the codes are imposed by the NRC in the regulations, as determined according to the guidance in NUREG/BR-0058 and the backfit provisions of 10 CFR 50.109, based on the safety significance of the code changes, the code committees may change their focus away from issues that are of little or no safety significance to be consistent with the regulatory focus. Therefore, it is unknown what actual affect the adjustment in the regulatory approach would have on the ASME process.

The NRC has been moving toward a performance-based regulatory framework as opposed to prescriptive rules (e.g., 10 CFR 50.65, the "Maintenance Rule" and Option B to 10 CFR 50, Appendix J). However, a performance-based rule could endorse a prescriptive code like the O&M Code when it is based on performance criteria such as the new condition monitoring provisions for check valve IST. There is also a trend toward risk-informed regulation which uses probabilistic risk assessment techniques to supplement deterministic approaches in regulatory decision making. Implementation of IST programs will be candidates for employing both performance-based and risk-based regulation. Two pilot programs were recently submitted for NRC review with a goal of achieving NRC approval by September 1996. A regulatory guide on the methodology should be available in the near future.

CONCLUSION

As the electric utility industry moves toward deregulation, the future of nuclear power plants lies in the balance of cost-effective operation and maintenance while maintaining safety. The industry is building on the sound technical and procedural concepts of the past to address the pending concerns of today with an outlook for tomorrow's challenges on the horizon. NRC has maintained, and will continue to maintain, a strong commitment to

involvement in the ASME consensus process. We are faced with a challenge as the previous technical and prescriptive process moves toward incorporating the risk-informed and performance-based concepts. We are challenged to be timely in our actions, both within the ASME and within the regulatory arena, and yet not compromise the inherent quality and conservatism that the public expects in the application and regulation of complex technology.

REFERENCES

Available from American Society of Mechanical Engineers:

Boiler and Pressure Vessel Code
Operations and Maintenance Standards and Codes
Board on Nuclear Codes and Standards Management and Operations Manual

Available from U.S. Government Printing Office:

OMB Circular A-119
United States Code
Code of Federal Regulations
NUREG/BR-0058

Session 3C

General Valve Issues 3

Session Chair
C. Richard Favreau
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A METHOD FOR EVALUATING PRESSURE LOCKING AND THERMAL BINDING OF GATE VALVES

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ABSTRACT

A method is described to evaluate the susceptibility of gate valves to pressure locking and thermal binding. Binding of the valve disc in the closed position due to high pressure water trapped in the bonnet cavity (pressure locking) or differential thermal expansion of the disk in the seat (thermal binding) represents a potential mechanism that can prevent safety-related systems from functioning when called upon. The method described here provides a general equation that can be applied to a given gate valve design and set of operating conditions to determine the susceptibility of the valve to fail due to disc binding. The paper is organized into three parts. The first part discusses the physical mechanisms that cause disc binding. The second part describes the mathematical equations. The third part discusses the conclusions.

INTRODUCTION

Gate valves are used extensively in the nuclear and fossil power plants due to their rugged design, their ability to seal positively, and relative ease of maintenance. The closure member consists of a wedge shaped single disk, or twin disks with a spreading mechanism in-between. Flow isolation and sealing is achieved by wedging the disk into the seat or by spreading the disk halves against the seats. If the design and flow conditions are favorable, the sealing action can be accomplished by the fluid pressure alone. Figure 1 illustrates a solid wedge gate valve with a motor operator. The motor operator is essentially a gear box with an electric motor drive and associated controls to close or open the valve.

The motive force to close or open the valve is provided by an electric, air, or a hydraulic operator. The operator is provided with controls to ensure that sufficient force is delivered to the valve stem to move the valve disk to the open or the closed position without exceeding the design limits of the valve components. The closing force must be of sufficient magnitude to overcome the packing force, the pressure force, and the seat friction force due to the differential pressure across the valve disk. An additional seating force may also be required to ensure proper sealing and leak tightness of the valve. The opening force must be able to overcome the wedging force, the packing force, and the pressure forces. A change in the pressure or the wedging conditions subsequent to the closing may adversely affect the magnitude of the force required to open the valve and may render it inoperable. Pressure locking and thermal binding are two such phenomenon that

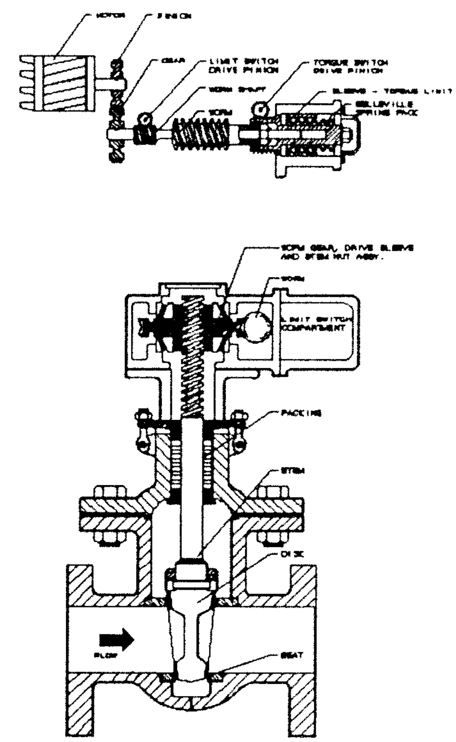

Figure 1 -Gate Valve with a Motor Operator

result from changing pressure andtemperature conditions of the valve.

The design of gate valves make them susceptible to pressure locking and thermal binding [1, 2, 3]. The two-seat design allows the disk to seal simultaneously, both upstream and downstream, and trap fluid in the bonnet cavity. Pressure locking results from subsequent thermal expansion of the trapped fluid which creates additional forces that must be overcome to open the valve. Thermal binding is caused by the differential thermal contraction of the body with respect to the disk, which adds to the force to unwedge the disk and open the valve.

Whether it is pressure locking or thermal binding, the underlying phenomenon is a thermal transient that creates the conditions for disk binding. Therefore, proper evaluation of susceptibility of a valve to disc binding must start with an evaluation of the operating conditions that may result in temperature swings. If such conditions are present, then a quantitative evaluation can be attempted to determine the potential for disk binding. This evaluation should consider the magnitude of the temperature swing, compressibility of the fluid, and the compliances of the valve components.

PRESSURE LOCKING

The conditions leading to pressure locking in a gate valve are illustrated in Figure 2.

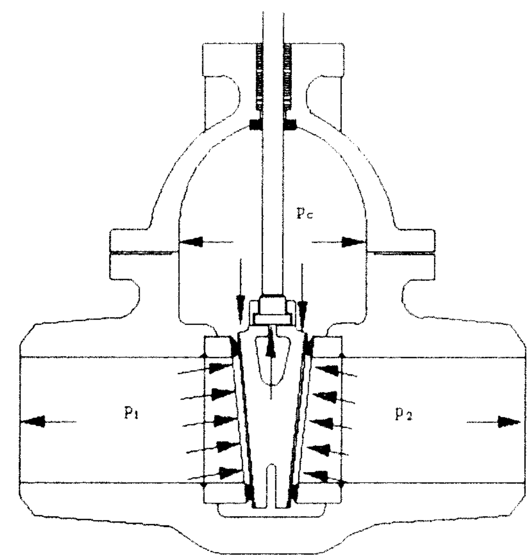

Figure 2 -General Pressure Conditions in a Gate valve

The valve disk is wedged into the seat and has sealed the bonnet cavity. The fluid trapped in the cavity is initially at the line pressure and temperature under which the valve was closed. If the fluid is exposed to heat, such as that produced during system start-up after a cold hydro test, or a rise in the ambient

temperature, it will expand and cause the pressure in the bonnet cavity to increase. Although less likely, the same phenomenon may occur even in steam lines. After the valve is closed, the steam trapped in the bonnet cavity may condense due to heat loss to the ambient. This in turn, lowers the bonnet pressure and allows more steam to be drawn in by leaking past the upstream seat. Collection of the condensate in the bonnet cavity continues until the cavity is completely filled and the pressures are equalized. At this point the conditions are identical to that for a liquid line, i.e., the bonnet cavity is filled with high pressure water and sealed. A subsequent rise in the temperature may lead to pressure locking.

A somewhat similar phenomenon may occur due to line pressure change after the valve is closed. The changing line pressure (upstream, downstream, or both) changes the pressure force acting on the disk, thus directly affecting the opening force. It also changes the unwedging force by changing the interference between the disk and the seat.

Thermal binding occurs due to differential thermal contraction of the valve body (or expansion of the valve disk) after the valve has been closed. The resulting interference between the disk and seat causes the seat to pinch the disk in its place. The pinching creates additional unwedging force which must be overcome to open the valve. Thermal binding is commonly experienced when a valve is closed in hot conditions and is allowed to cool before opened. The valve body cools to a lower temperature than the valve disk and contracts more than the disk. This difference in contraction may be sufficient to cause the disk to bind in the seat. The stiffer the valve body and disk, the higher the potential for thermal binding. Thus, a flexible wedge gate valve is less susceptible to thermal binding than a solid wedge gate valve. Thermal binding may also be caused, although at a less significant scale, by a phenomenon called "stem insertion." Here the valve stem differentially expands more than the valve yoke (or the yoke contacts more than the stem) and drives the disk further into the seat, increasing the wedging. The force required to open the valve increases in proportion to this additional wedging.

FORCE TO CLOSE OR OPEN THE VALVE

The force required to close a gate valve is obtained by adding the component forces acting on the valve stem. These component forces include the packing friction force, stem ejection force, differential pressure (Dp) force, and the wedging force. Figure 3 shows the forces acting on the valve disk and the valve stem during the closing stroke of a flexible wedge gate valve. The expression for the force to close the valve is

$$F_c = F_{pk} + p_1 A_s + (p_1 - p_2) A (S_c - \tan\theta) + k_v s \qquad (1)$$

where

F_c final closing force
F_{pk} packing friction force
p_1 upstream pressure
p_2 downstream pressure
A_s stem area at the packing
A seat area (circular)
S_c seat factor in the closing direction
θ half wedge angle
s stem nut advance after hard seat contact
k_v equivalent stiffness of the valve assembly

The seat area, A, is based on the mean seat diameter. The equation for the seat factor is where μ is the apparent friction coefficient between the disk and the seat. The expression

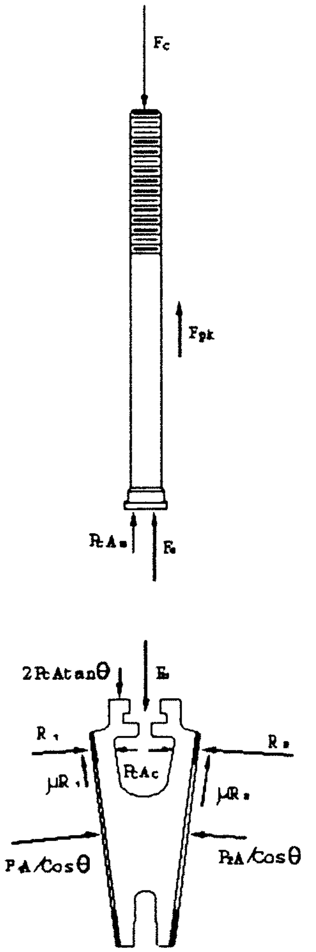

Figure 3 -Force Balance during the Closing Stroke

$$S_c = \frac{Sin\theta + \mu Cos\theta}{Cos\theta - \mu Sin\theta} \qquad (2)$$

(S_c - tan θ) in equation 1 can be recognized as the valve factor

defined by,

$$VF = \frac{\mu}{Cos\theta\,(Cos\theta - \mu\, Sin\theta)} \qquad (3)$$

The equation for the valve stiffness is where k_{ys} and k_{bd} are stiffnesses of the stem-yoke and

$$k_v = \frac{k_{ys}\, k_{bd}}{k_{ys} + k_{bd}} \qquad (4)$$

the body-disk assemblies defined by the following expressions.

$$k_{ys} = \frac{k_y k_s}{k_y + k_s}$$
$$k_{bd} = \frac{k_b k_d}{k_b + k_d} 4 S_c \tan\theta \qquad (5)$$

where k is the stiffness (force per unit deflection) and the subscripts b, d, s, and y stand for the body, disk, stem, and yoke, respectively.

The expression for the opening force is obtained by considering the force balance on the individual components of the valve along with the appropriate compatibility conditions. The expression for the force required to open the valve (which has been subjected to pressure and temperature changes subsequent to its closure) is given below.

$$\begin{aligned} F_o = {} & F_{pk} - p_1 A_s + (p_1 - p_2) A (S_o + \tan\theta) + k_v \gamma s \\ & + k_v \gamma L (\alpha_s \Delta T_s - \alpha_y \Delta T_y) \\ & + 2 k_{bd} \gamma \frac{W}{2\tan\theta} (\alpha_d \Delta T_d - \alpha_b \Delta T_b) \\ & + \Delta p_c A \left[(1 + \frac{k_v}{k_{ys}} \gamma)(2\tan\theta - \frac{A_s}{A}) \right] \\ & + \Delta p_c A \left[\frac{\gamma}{2\tan\theta} (\frac{k_v}{k_d} \frac{A_c}{A} - \frac{k_v}{k_b} \frac{A_b}{A}) \right] \\ & - (\Delta p_1 + \Delta p_2) A \left[(1 - \frac{k_v}{k_{ys}}) S_o + (1 + \frac{k_v}{k_{ys}} \gamma) \tan\theta \right] \end{aligned} \qquad (6)$$

where
Δp increase in pressure
ΔT increase in temperature

A_b bonnet cavity area projected on the plane of the disk
A_c seat area minus the disk hub area
L representative length of the stem
S_o seat factor in the open direction
W width of the disk
α thermal expansion coefficient
γ S_o/S_c

S_o is calculated by substituting ($-\mu$) into equation 2 and taking the absolute value.

In equation 6 above, the first group of terms represent the normal opening force in the absence of thermal binding and pressure locking. Individual components can be recognized as the packing force, stem ejection force, Dp force, and the unwedging force. The second group is the additional unwedging force due to stem insertion. The third group is the additional unwedging force due to thermal binding. The fourth and the fifth groups are the additional forces due to bonnet pressurization. There are four contributing components: stem ejection, disk insertion, disk spreading, and bonnet expansion. The sixth group is the additional force due to line pressure change.

Examination of equation 6 above reveals the following:

- The force required to open a valve is less than the force required to close the valve provided that the conditions (temperature, pressure, seat friction) remain the same.

- Differential thermal expansion of the stem more than the yoke increases the opening force. This additional force, commonly referred to as the stem insertion force, increases in proportion to the overall stiffness of the valve assembly.

- Differential thermal expansion of the disk more than the body increases the opening force. Differential thermal expansion may be brought about by the differences in thermal expansion coefficients, temperatures, or both. The higher the stiffness of the disk-body assembly the higher is the force. Thus, a flexible wedge gate valve is less susceptible to thermal binding than a solid wedge gate valve.

- An increase in bonnet cavity pressure is partially compensated by a reduction of the stem ejection force. It may be further compensated by the body expanding and relaxing the disk in the seat.

- An increase in the upstream or downstream pressure reduces the opening force by relaxing the disk in the seat and by providing a lifting force. A decrease in the line pressure has the opposite effect.

The increase in the bonnet cavity pressure, Δp_c, is generally due to an increase in the temperature of the fluid trapped in the bonnet cavity. Since water is nearly incompressible, even a small increase in temperature may result in a large pressure increase. There are, however, compensating mechanisms to partially offset the pressure increase. These mechanisms are: thermal expansion of the valve body, compliance (ability to expand under pressure) of the bonnet cavity, and presence of noncondensible gases trapped in the bonnet cavity. The effect of the thermal expansion and compliance of the valve body can be accounted for using the equation below.

$$\Delta p_c = \frac{(\beta_w - \beta_c)\Delta T_c}{\kappa_w + \kappa_c} \qquad (7)$$

where

β_c volume expansion coefficient of the bonnet cavity
β_w thermal expansion coefficient of water
κ_c compliance of the bonnet cavity
κ_w bulk modulus of water

The bulk modulus and thermal expansion coefficient of water can be obtained from the thermodynamic properties of water using their formal definitions. Figure 4 illustrates these parameters for saturated water in the temperature range from 100°F to 500°F.

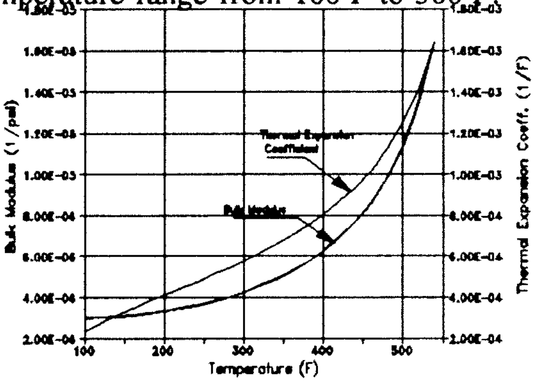

Figure 4 -Bulk Modulus and Thermal Expansion Coefficient for Water

Figure 5 shows the compressibility of water calculated according to Equation 7. The basic property data (pressure, temperature, specific volume) for these figures were obtained from Reference 4 (ASME, 1967). The volume expansion coefficient and the compliance of the bonnet cavity can be estimated using the appropriate stress-strain equations such as those given in Reference 5 (Roark, 1975) or, more accurately determined using finite element analysis. Since these terms are small in comparison to that of water, they can also be ignored for conservatism and simplicity.

Noncondensible gases trapped in the bonnet cavity are very effective in reducing the bonnet pressure rise due to temperature. This

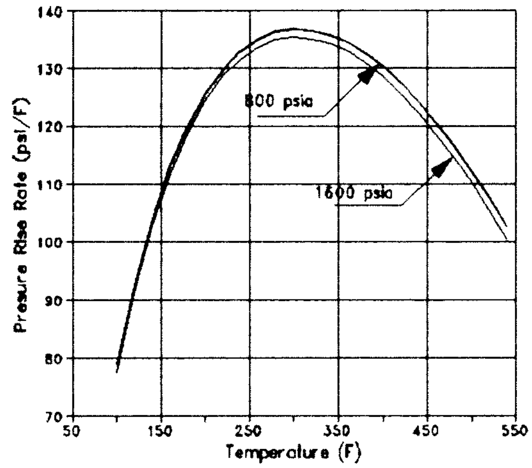

Figure 5 -Compressibility of Water

is primarily due to the compressibility of the trapped gas being several orders of magnitude larger than that of the water. Although this effectiveness decreases with increasing pressures, even a small amount of noncondensible can reduce the pressure rise rate significantly (Figure 6). Provided that the fraction of the trapped gas can be reasonably estimated, its effect on the pressure rise can be calculated by modifying β_w and κ_w in equation 7.

The expression for Δp_c in equation 7 defines its value before the valve is attempted to be opened. As the valve stem is moved in the open direction, the bonnet cavity volume changes due to the stem retracting and the disk advancing into the cavity. Depending on the relative magnitude of these factors, Δp_c may increase or decrease. These effects can be factored into equation 7 in terms of the stem diameter, disk geometry, and the stem-disk gap (if the stem is not self locking at the stem nut or at the worm gear).

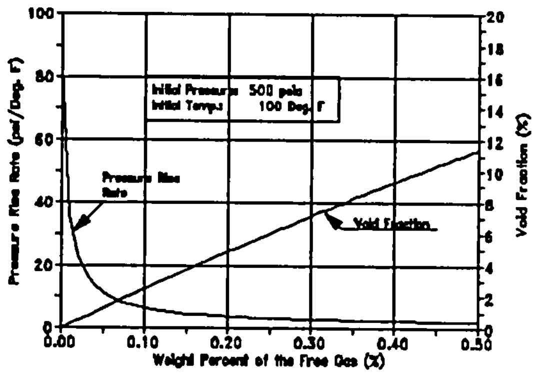

Figure 6 -Effect of Noncondensible Gases on the Compressibility of water

SUMMARY AND CONCLUSIONS

A method is described to assess the operability of gate valves subjected to pressure locking and thermal binding. The force required to open a gate valve that has been exposed to thermal binding or pressure locking conditions can be calculated using equation 6. This equation contains stiffness terms such as k_v, k_{ys}, k_{bd}, k_d, and k_b which may not be readily calculated. However, their ratios, as they are contained in equation 6, are always bounded. These ratios always evaluate to a value between zero and one. Thus, bounding estimates of the opening force can be made by setting these ratios to their limiting value as appropriate. For more accurate estimates, diagnostic test data, if available, can be used to back calculate the stiffness values from the stem force versus stem displacement traces.

The conclusions can be summarized as follows:

1. Bonnet pressurization can result in extremely high pressures if the compliance of the bonnet cavity is small and there are no leakage paths (seat leak, leakage through the packing, or deliberately introduced leak paths such as weep holes, and relief valves). Presence of noncondensible gases trapped in the bonnet cavity significantly reduces the pressure rise.

2. Transient events which change the line pressure or the bonnet pressure can effect the unwedging force by permanently changing the wedging of the disk in the seat. Restoring the initial pressure condiitons does not restore the unwedging force to its pre-transient value.

3. As long as the valve remains wedged, an increase in the upstream or downstream pressure reduces the force required to open the valve.

4. The clearance between the stem and the disk in a self locking valve may offset pressure locking by relieving some of the pressure as the stem is retracted during the opening stroke.

5. Thermal binding can occur only if the valve is subjected to a thermal transient at any time between the closing and the opening strokes. Once this transient has occurred its effect cannot be reversed by restoring the conditions during which the valve was closed.

6. Thermal binding force increases with increasing stiffness of the valve. Thus, a flexible wedge gate valve is less susceptible to thermal binding than a solid wedge gate valve.

7. Thermal binding due to stem insertion is possible but less likely than thermal binding in the disk.

SI CONVERSION

kPa = 0.14504 psi
m = 39.37 in
°C = (°F-32)/1.8
N = 0.22481 lbf (pound force)

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1. "Pressure Locking and thermal Binding of Gate Valves," Institute of Nuclear Plant Operations, SOER 84-7.

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COMMONWEALTH EDISON COMPANY PRESSURE LOCKING TEST REPORT

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ABSTRACT

Pressure Locking is a phenomena which can cause the unseating thrust for a gate valve to increase dramatically from its typical static unseating thrust. This can result in the valve actuator having insufficient capability to open the valve. In addition, this can result in valve damage in cases where the actuator capability exceeds the valve structural limits. For these reasons, a proper understanding of the conditions which may cause pressure locking and thermal binding, as well as a methodology for predicting the unseating thrust for a pressure locked or thermally bound valve, are necessary.

This report discusses the primary mechanisms which cause pressure locking. These include sudden depressurization of piping adjacent to the valve and pressurization of fluid trapped in the valve bonnet due to heat transfer. This report provides a methodology for calculating the unseating thrust for a valve which is pressure locked. This report provides test data which demonstrates the accuracy of the calculation methodology.

DESCRIPTION OF PRESSURE LOCKING PHENOMENA

Pressure locking occurs when the bonnet cavity pressure of a gate valve exceeds the pressure on <u>both</u> sides of the valve disk. The two primary mechanisms that exist for pressure locking of gate valves are described below:

<u>SUDDEN DEPRESSURIZATION</u>

This pressure locking mechanism occurs when a valve is pressurized from one side. Leakage past the valve seat will cause the fluid in the gate valve bonnet to pressurize to the pressure of the high pressure side of the valve disk. Depending on the leak-tightness of the valve seats, this pressurization process may take seconds or hours; however, it is extremely unlikely that the valve seat will be sufficiently leak tight to prevent this process from eventually occurring. If the source of pressure is suddenly removed, then pressure in the bonnet valve will remain trapped. If the valve is called upon to open before the bonnet pressure has decayed to the line pressure, then a pressure locking event occurs.

The time needed for the bonnet pressure to decay is dependent on several factors including leak tightness of valve seats and packing. In addition, when the bonnet fluid is at a high temperature or contains large amounts of air, the bonnet pressure decays much more slowly due to the pressurizer effect. Apparent cases of pressure locking occurring up to a day after the pressure source is removed have been recorded. However, test data presented later in this report suggests that the bonnet pressure is likely to decay within one hour of the sudden depressurization event

occurring. This type of pressure locking is likely to occur when pumps adjacent to closed valves are shut off or when an event such as a LOCA causes pressure on one side of a valve to suddenly drop off.

When the initial differential pressure across the valve disk is sufficient to unseat the high pressure side disk from its seat, then the bonnet pressure following a sudden depressurization event is less than the bonnet pressure at the start of the event. The maximum pressure which can be trapped in the valve bonnet can be calculated by determining the differential pressure at which the valve disk will come back into contact with the valve seat. Until the disk to seat contact is re-established, the bonnet pressure will follow the upstream side pressure. This calculation has been developed by ComEd, but is not provided in this report due to constraints on length.

THERMALLY INDUCED PRESSURE RISE IN BONNET

This pressure locking mechanism occurs when the valve bonnet cavity of a gate valve is filled with liquid that contains little or no air. If a heat source is applied to fluid in the valve bonnet cavity, then expansion of the fluid can cause pressure in the valve bonnet to dramatically increase. The heat source can be fluid in piping adjacent to the valve or external environmental conditions as might be encountered following a high energy line break. Pressurization rates of 20 psi/°F to 60 psi/°F have been recorded during special testing. However, pressurization rates of this nature require the following conditions to exist:

- the valve seats and packing must be very leak tight
- the heat source must provide a high heat transfer rate to the bonnet cavity fluid
- no air can exist in the valve bonnet cavity, or the temperature rise in the valve bonnet cavity must be sufficient to cause the expanding fluid to collapse the air bubbles before the high pressurization rate can be achieved.

PRESSURE LOCKING CALCULATION METHODOLOGY

ASSUMPTIONS

1. The valve disk is assumed to act as two ideal disks connected by a hub. The equations in reference 1 are assumed to conservatively model the actual load due to pressure forces.
2. The coefficient of friction between the valve seat and disk is assumed to be the same under pressure locking conditions as it is under DP conditions.

DESIGN INPUTS

The following design inputs are used in calculating the force required to unseat a pressure locked MOV:

- Design Basis Pressure Conditions at the time of the pressure locking event. This includes the upstream (P_{up}), downstream (P_{down}), and bonnet pressure (P_{bonnet}).

- Valve Disk Geometry. This includes the hub radius (b), hub length (L), mean seat radius (a), seat angle (θ), and average disk thickness (t). Figure 1 below is provided for further clarification. When the hub cross-section is not circular (e.g. many Westinghouse gate valve designs), then an effective hub radius which corresponds to a circle of equal area to the hub cross-sectional area should be used.

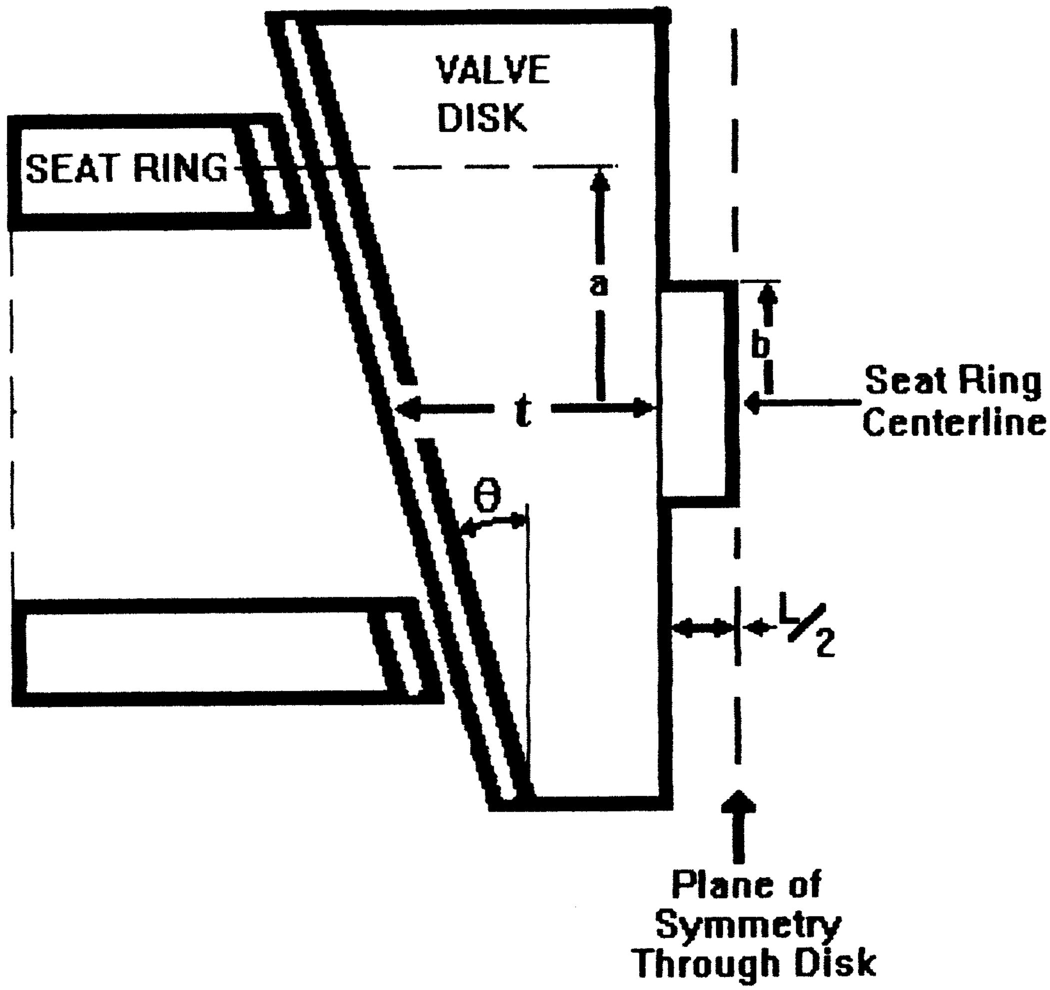
- Valve Disk Material Properties. This includes the modulus of elasticity (E) and the Poisson's ratio (ν) for the disk base material.

- Valve Stem Diameter (D_{stem})

- Static Unseating Thrust (F_{po})

- Coefficient of Friction between Disk and Seat (μ)

FIGURE 1



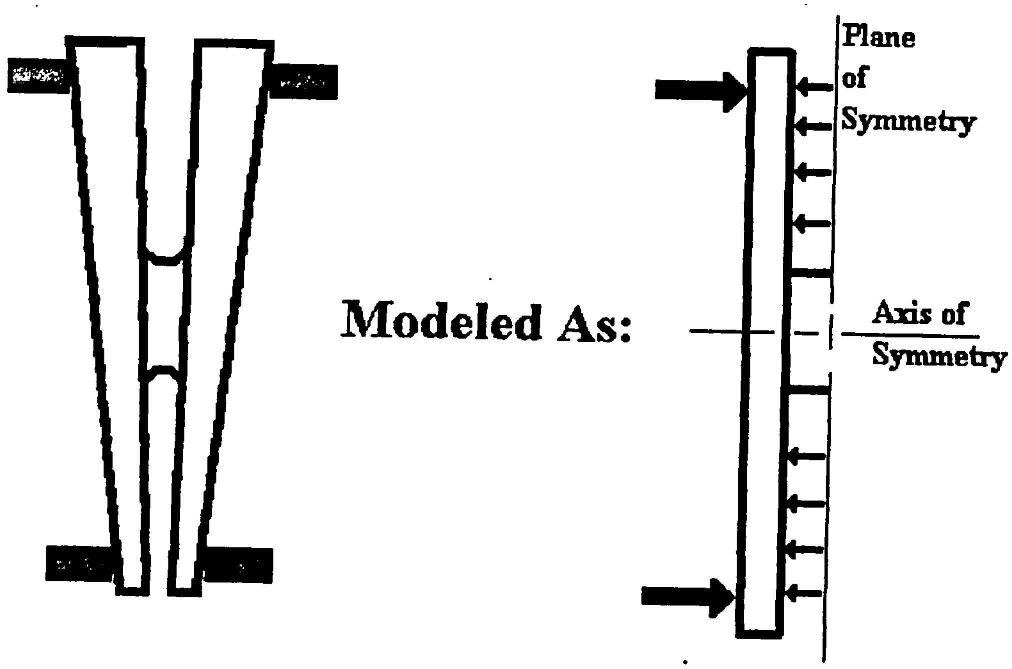
CALCULATIONS

The methodology for calculating the thrust required to open the MOVs under the pressure locking scenario is based on the Reference 1 (Roark's) engineering handbook. This methodology is based in part on calculations developed by MPR Associates (Reference 2). The methodology determines the total force required to open the valve under a pressure locking scenario by calculating the four components to this required force. The four components of the force are the pressure locking component, the static unseating component, the piston effect component, and the "reverse piston effect" component. These components are determined using the following steps.

Pressure Locking Component of Force Required to Open the Valve

The valve disk is modeled as two plates attached at the center by a hub which is concentric with the valve disk. A plane of symmetry is assumed between the valve disks. This plane of symmetry is considered fixed in the analysis.

FIGURE 2



Based on this geometry, the following constants are calculated using the Reference 1 equations:

Average DP Across Disk

$$DP_{avg} = P_{bonnet} - \frac{P_{up} + P_{down}}{2} \tag{1}$$

Disk Stiffness Constants

(*Reference* 1, *Table* 24) $$D = \frac{E \times t^3}{12 \times (1 - \nu^2)} \tag{2}$$

$$G = \frac{E}{2 \times (1 + \nu)} \tag{3}$$

Geometry Factors

(*Reference* 1, *Table* 24) $$C_2 = \frac{1}{4}\left[1 - \left(\frac{a}{b}\right)^2 \left(1 + 2\ln\left(\frac{a}{b}\right)\right)\right] \tag{4}$$

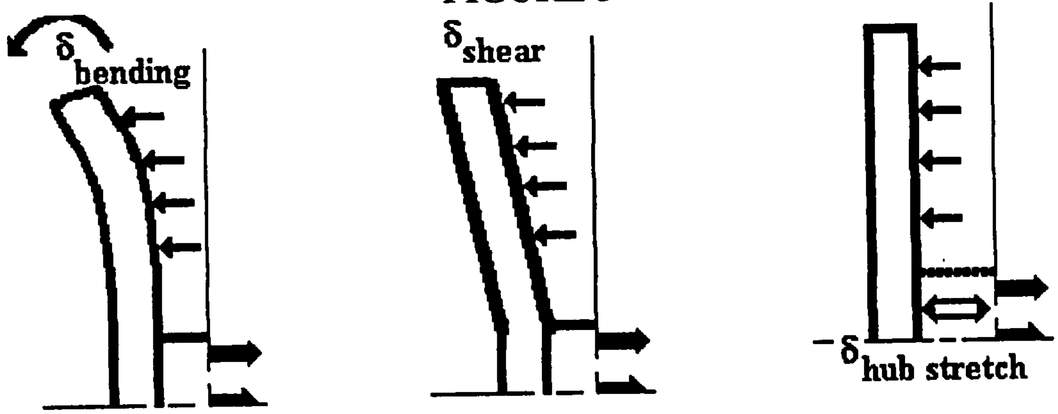
$$C_3 = \frac{b}{4a}\left\{\left[\left(\frac{b}{a}\right)^2 + 1\right]\ln\left(\frac{a}{b}\right) + \left(\frac{b}{a}\right)^2 - 1\right\} \tag{5}$$

$$C_8 = \frac{1}{2}\left[1 + \nu + (1 - \nu)\left(\frac{b}{a}\right)^2\right] \tag{6}$$

$$C_9 = \frac{b}{a}\left\{\frac{1 + \nu}{2}\ln\left(\frac{a}{b}\right) + \frac{1 - \nu}{4}\left[1 - \left(\frac{b}{a}\right)^2\right]\right\} \tag{7}$$

The pressure force is assumed to act uniformly upon the inner surface of the disk between the hub diameter and the outer disk diameter. The outer edge of the disk is assumed to be unimpeded and allowed to deflect away from the pressure force. In addition, the disk hub is allowed to stretch. The total displacement at the outer edge of the valve disk due to shear and bending and due to hub stretch are calculated using the Reference 1 equations.

FIGURE 3



Additional Geometry Factors

(*Reference* 1, *Table* 24) $$L_{11} = \frac{1}{64}\left\{1+4\left(\frac{r_0}{a}\right)^2 - \left(\frac{r_0}{a}\right)^4 - \left(\frac{r_0}{a}\right)^2\left[2+\left(\frac{r_0}{a}\right)^2\right]\ln\left(\frac{a}{r_0}\right)\right\} \quad (8)$$

($r_0 = b$ *for Case* 2L) $$L_{17} = \frac{1}{4}\left\{1-\frac{1-\nu}{4}\left[1-\left(\frac{r_0}{a}\right)^4\right]-\left(\frac{r_0}{a}\right)^2\left[1+(1+\nu)\ln\left(\frac{a}{r_0}\right)\right]\right\} \quad (9)$$

Moment Factors

(*Reference* 1, *Table* 24, *Case* 2L) $$M_{rb} = \frac{-DPavg \times s^2}{C_8}\left[\frac{C_9}{2\times a\times b}\left(a^2 - r_0^2\right) - L_{17}\right] \quad (10)$$

($r_0 = b$ *for Case* 2L) $$Q_b = \frac{DPavg}{2\times b}\left(a^2 - r_0^2\right) \quad (11)$$

Deflection from pressure / bending

(*Reference* 1, *Table* 24, *Case* 2L) $$y_{bq} = M_{rb}\frac{a^2}{D}C_2 + Qb\frac{a^3}{D}C_3 - \frac{DPavg \times a^4}{D}L_{11} \quad (12)$$

Deflection from pressure / shear

$(Reference\,1, Table\,25, Case\,2L)$

$$Ksa = -0.3\left[2\ln\left(\frac{a}{b}\right) - 1\left(\frac{r_0}{a}\right)^2\left(1 - 2\ln\left(\frac{r_0}{b}\right)\right)\right] \quad \textbf{(13)}$$

$(r_0 = b\ for\ Case\,2L)$

$$y_{sq} = \frac{K_{sa} \times DPavg \times a^2}{t \times G} \quad \textbf{(14)}$$

Deflection from pressure / hub stretch

$$P_{force} = \pi\left(a^2 - b^2\right)DPavg \quad \textbf{(15)}$$

$$y_{stretch} = \frac{-P_{force}}{\pi \times b^2}\frac{L}{2 \times E} \quad \textbf{(16)}$$

Total Deflection due to pressure

$$y_q = y_{bq} + y_{sq} + y_{stretch} \quad \textbf{(17)}$$

An evenly distributed force is assumed to act between the valve seat and the outer edge of the valve disk. This force acts to deflect the outer diameter of the valve disk inward and to compress the disk hub. The pressure force is reacted to by an increase in this contact force between the valve disk and seats. The valve body seats are conservatively assumed to be fixed. Therefore, the deflection due to the known pressure load must be balanced by the deflection due to the unknown seat load. The deflection due to the pressure force was previously calculated. The Reference 1 equations are now used to determine the contact force between the seat and disk which results in a deflection which is equal and opposite to the deflection due to the pressure force. This is done by first calculating the amount deflection created by a unit load of seat contact force (w = 1 lbf/in). The equilibrium contact load is then determined by dividing the deflection caused by the unit contact load into the previously calculated deflection due to the pressure force. The equations are provided below.

Additional Geometry Factors

(*Reference* 1, *Table* 24, *Case* 1*L*)

$$L_3 = \frac{r_0}{4 \times a}\left\{\left[\left(\frac{r_0}{a}\right)^2 + 1\right]\ln\left(\frac{a}{r_0}\right) + \left(\frac{r_0}{a}\right)^2 - 1\right\} \tag{18}$$

(*for Case* 1*L*, $r_0 = a$, $\therefore L_3 = L_9 = 0$)

$$L_9 = \frac{r_0}{a}\left\{\frac{1+\nu}{2}\ln\left(\frac{a}{r_0}\right) + \frac{1-\nu}{4}\left[1 - \left(\frac{r_0}{a}\right)^2\right]\right\} \tag{19}$$

Deflection from seat load / bending $(r_0 = a)$

(*Reference* 1, *Table* 24, *Case* 1*L*, $w = 1$)

$$y_{bw} = -\frac{a^3}{D}\left[\frac{C_2}{C_8}\left(\frac{r_0 \times C_9}{b} - L_9\right) - \frac{r_0 \times C_9}{b} + L_3\right] \tag{20}$$

Deflection from seat load / shear $(r_0 = a)$

(*Reference* 1, *Table* 25, *Case* 1*L*, $w = 1$)

$$K_{sa} = -1.2\,\frac{r_0}{a}\,\ln\left(\frac{r_0}{b}\right) \tag{21}$$

$$y_{sw} = K_{sa}\frac{a}{t \times G} \tag{22}$$

Deflection from seat load / hub compr.

$w = 1$, $\therefore$ *Compressive force* $= 2 \times \pi \times a$

$$y_{compr} = -\frac{2 \times \pi \times a}{\pi \times b^2}\left(\frac{L/2}{E}\right) \tag{23}$$

Total Deflection from unit seat load

$(w = 1)$

$$y_w = y_{bw} + y_{sw} + y_{compr} \tag{24}$$

Therefore, the equilibrium contact load distribution (lbf/in) and the corresponding load applied to each seat is calculated using the relationship below:

$$w_{equalibrium} = {}^{y_q}\!/_{y_w}\text{, where } y_w \text{ is calculated for } w = 1 \tag{25}$$

$$Load\ per\ seat = 2 \times \pi \times a \times \frac{y_q}{y_w} \tag{26}$$

Several methods may be used to determine an appropriate seat to disk friction coefficient. Using this friction coefficient and a force balance on the disk to seat interface, the following equation is derived for calculating the stem force required to overcome the increased contact load between the seat and disk:

$$F_{pres\,lock} = \left(2 \times \pi \times a \times \frac{y_a}{y_w}\right) \times \left[\mu \times \cos(\theta) - \sin(\theta)\right] \times 2 \tag{27}$$

where the last 2 corresponds to the number of seats

Static Unseating Force (F_{static})

The static unseating force results from the open packing load and pullout force due to wedging of the valve disk during closure. These loads are superimposed on the loads due to the pressure forces which occur during pressure locking. The value for this load is based on static test data for the MOVs.

Piston Effect (F_{piston})

The piston effect due to valve internal pressure exceeding outside pressure is calculated using the standard industry equation. This force assists movement of the valve stem in the open direction.

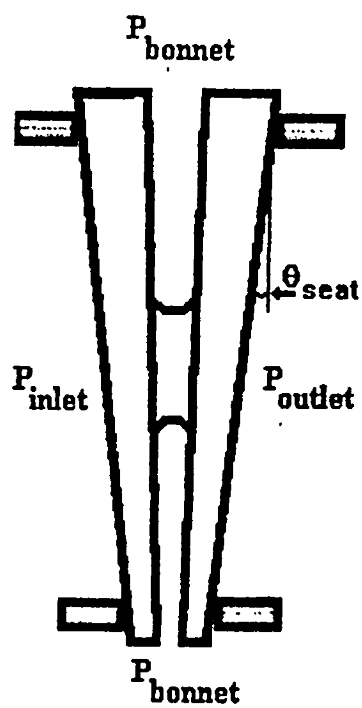
$$F_{piston\,effect} = \frac{\pi}{4} \times D_{stem}^2 \times \left(P_{bonnet} - P_{atm}\right) \tag{28}$$

"Reverse Piston Effect"(F_{vert})

The reverse piston effect is the term used in this calculation to refer to the pressure force acting downward against the valve disk. This force is calculated as follows:

$$F_{vert} = \left[\pi \times a^2 \times \left(2 \times P_{bonnet} - P_{inlet} - P_{outlet}\right)\right] \times \sin(\theta) \qquad (29)$$

FIGURE 4



Total Force Required to Overcome Pressure Locking

As mentioned previously, the total stem force (tension) required to overcome pressure locking is the sum of the four components discussed above. All of the terms are positive with the exception of the piston effect component.

$$F_{total} = F_{pres\,lock} + F_{static} + F_{vert} - F_{piston} \quad (30)$$

DESCRIPTION OF TEST VALVES

ORIGIN

The three test valves were obtained from different sources. The Crane valve is a test valve located at Quad Cities Station. The Westinghouse valve was obtained through the Westinghouse Owners Group. The Borg-Warner valve was obtained from Arizona Public Service.

PAST SERVICE AND TEST HISTORY

The Crane valve is a spare valve which was subjected to blowdown testing at Wyle Laboratories in Huntsville, Alabama. The Westinghouse valve is a test valve which was subjected to limited testing at South Texas Project. The Borg-Warner valve was a spare valve which had not been subjected to previous testing other than that performed at the vendor prior to delivery.

MATERIALS

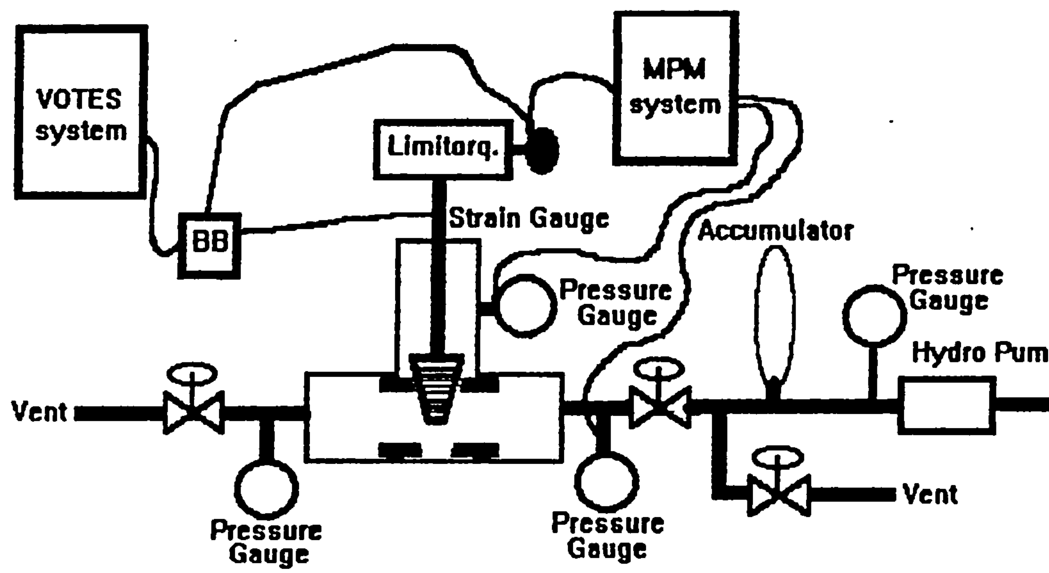
The Crane valve is a carbon steel valve (Model 783-U) which was modified during blowdown testing to contain a stainless steel valve disk and malcolmized guide rail (similar to the Model 783-UL valve design). The Westinghouse valve and Borg-Warner valve were stainless steel valve designs.

DESCRIPTION OF TEST APPARATUS

INSTRUMENTATION AND DATA ACQUISITION SYSTEMS

The figure below shows the basic test setup used for the pressure locking tests. A VOTES data acquisition system and a Motor Power Monitor (MPM) data acquisition system were used to collect stem thrust, actuator torque, and motor power data. In addition, on-line pressure data was collected during the Westinghouse and Borg-Warner valve tests. A hydrostatic test pump and accumulator were used as the pressure source during pressure locking tests and hydropump DP tests:

FIGURE 5



VALVE ORIENTATIONS

For the Crane test, the valve was laid on its side with the stem slightly below horizontal. This configuration was used to ensure that no air pockets would be trapped within the valve body when it was filled with water.

The Westinghouse valve was installed in a test stand with the stem upright. The valve bonnet was vented by bleeding air out of the packing leak-off line.

The Borg-Warner valve was installed in a special test stand which allowed pivoting the valve about its centerline. The valve stem could be put at any angle between upright and sloped downward at a 15 degree angle in either direction. To remove air from the valve bonnet, the valve was rotated on its side and rocked up and down as it filled with water.

DESCRIPTION OF TEST METHODS

STATIC BASELINE TESTS

The test process started with static test strokes to verify the proper installation of the data acquisition systems and to measure static unseating load magnitude and repeatability.

LOCAL LEAK RATE TESTS

Local leak rate tests of the valves were performed to measure seat tightness. These tests were performed at multiple torque switch settings in some cases.

DP TESTS

DP Tests in the open direction were performed by pressurizing the valve from one side with the hydropump and then stroking the valve open. Test data indicates that the differential pressure was maintained across the valve disk while the disk slid across the valve seat. The purpose of the DP tests was to precondition the valve seats and disks and to monitor the seat-to-disk friction coefficient. The DP tests were performed until a stable friction coefficient was achieved.

PAIRED STATIC / PRESSURE LOCKING TESTS

A series of pressure locking tests was performed for each valve. Inlet pressure, outlet pressure, bonnet pressure, and static seating force were varied during these tests. Static baseline tests to measure the static unseating load were performed between the pressure locking tests. The closure strokes for the static tests were performed at the same initial conditions (pressure and seating force) as the closure strokes prior to the pressure locking tests so that the change in unseating load due to pressure locking could be accurately determined.

BONNET DEPRESSURIZATION TESTS

To measure the seat tightness, bonnet depressurization rate tests were performed. The entire valve assembly (including the valve bonnet) was pressurized while in the closed position. Then the upstream and downstream pressure were vented. The bonnet pressure as a function of time was measured.

THERMALLY INDUCED BONNET PRESSURIZATION TESTS

To measure the potential for pressure locking due to bonnet fluid heat-up, thermally induced bonnet pressurization rate tests were performed on the Westinghouse and Borg-Warner valves. After venting air from the valve bonnet cavity, each valve was closed while filled with water at approximately 100 psig. The valve bonnet was then heated using an outside heat source. The pressure of the fluid in the valve bonnet was measured directly. The temperature of fluid in the valve bonnet for the Borg-Warner valve and the temperature of the outside of the valve bonnet for the Westinghouse valve were measured. Initial pressurization rates between 0.5 and 2.0 psi/degree F were measured. Much higher ultimate

pressurization rates were witnessed during the Borg-Warner tests. The data from this testing is not presented in this report, but is available from ComEd upon request.

PRESSURE LOCKING TEST DATA

The following table provides the pressure locking test results comparing the measured pressure locking unseating load to the predicted pressure locking unseating load:

TABLE 1

Valve	Test #	TSS	Static Unseating Thrust	Bonnet Pressure	Predicted Increase	Measured Increase	Percent Conservatism (Non-Cons.)	Notes
Crane 10"	6	1	25000	650	5103	4539	-2%	6
Crane 10"	7	1	25000	850	7213	8191	4%	6
Crane 10"	9	1	26000	1040	9421	11500	8%	6
Crane 10"	10	1	26000	1040	9922	12140	9%	6
Crane 10"	13	1	28000	1195	19462	22140	10%	
Crane 10"	14	1	28000	1375	22974	25480	9%	
Crane 10"	15	1	28000	1375	23126	25480	8%	
Crane 10"	34	2.5	38000	655	6243	5796	-1%	6
Crane 10"	35	2.5	38000	655	5142	5796	2%	6
Crane 10"	38	2.5	37500	1055	13164	13870	2%	6
Crane 10"	39	2.5	37500	1055	13065	13870	2%	6
Crane 10"	42	2.5	40000	1365	30028	29190	-2%	
Crane 10"	43	2.5	40000	1165	30428	24913	-14%	5
Crane 10"	46	2.5	40000	1575	32231	33680	4%	
Crane 10"	47	2.5	40000	1575	31931	33680	4%	
Crane 10"	50	2.5	40000	1775	37749	37950	1%	3,4
West. 4"	30	2	1450	496	1537.6	1555	-1%	
West. 4"	31	2	1450	514	1593.4	1538	2%	
West. 4"	33	2	900	1000	3100	3007	2%	
West. 4"	35	2	900	1000	3100	2990	3%	
West. 4"	37	2	50	1500	4650	4775	-3%	
West. 4"	39	2	50	1500	4650	4672	0%	
West. 4"	42	2	-400	2000	6200	5989	4%	
West. 4"	44	2	-400	2000	6200	6126	1%	
Borg-W. 10"	43	2	16935	205	5691	8532	4%	1
Borg-W. 10"	48	1	7882	209	5802	7386	19%	1
Borg-W. 10"	50	1	7782	402	11160	13004	16%	1
Borg-W. 10"	52	1	7906	630	17489	18799	23%	1
Borg-W. 10"	54	1	7882	694	19265	20514	23%	1
Borg-W. 10"	56	1	5023	919	25511	36849	-164%	1,2
Borg-W. 10"	74	2	17477	208	6225	10167	-2%	1
Borg-W. 10"	75	2	17477	213	6375	10765	-5%	1
Borg-W. 10"	77	2	17751	391	11703	16155	-5%	1
Borg-W. 10"	78	2	17751	402	12032	16853	-7%	1
Borg-W. 10"	80	2	17949	467	13977	22172	-26%	1,2

Valve	Test #	TSS	Static Unseating Thrust	Bonnet Pressure	Predicted Increase	Measure d Increase	Percent Conservatism (Non-Cons.)	Notes
Borg-W. 10"	81	2	17949	219	6555	10591	-2%	1
Borg-W. 10"	83	2	17700	110	3292	7757	-5%	1
Borg-W. 10"	84	2	17700	55	1646	5171	0%	1
Borg-W. 10"	86	2	17352	0	0	3628	0%	3
Borg-W. 10"	95	1	8000	0	0	3132	0%	3
Borg-W. 10"	96	1	8000	557	16671	19035	9%	1
Borg-W. 10"	97	1	8000	504	15085	18189	0%	1

NOTES:

1. The percent conservatism values are calculated after a "memory effect" of 3100 lbf (at TSS=1) or 3500 lbf (at TSS=2) is added to the predicted pressure locking load. Testing indicated that the process of applying and then relieving pressure against one side of the closed valve was sufficient to cause the unseating force to increase by these amounts, even when no pressure was captured in the valve bonnet. This effect was only noted for the Borg-Warner test valve.

2. When bonnet pressure significantly exceeds the pressure class rating of the test valve, the pressure locking calculation methodology appears to become non-conservative.

3. Tests 86 and 95 were performed to quantify the "memory effect" for the Borg-Warner valve. These tests were performed like a pressure locking test in that high pressure (~ 600 psig) was put against one side of the valve disk and then bled off. However, any pressure that entered the valve bonnet was relieved prior to the opening stroke.

4. The AC motor for the test valve stalled during this test and the valve did not fully unseat. Test data suggests that open valve motion was initiated prior to the stall. Consequently, the measured increase due to pressure locking is believed to be correct.

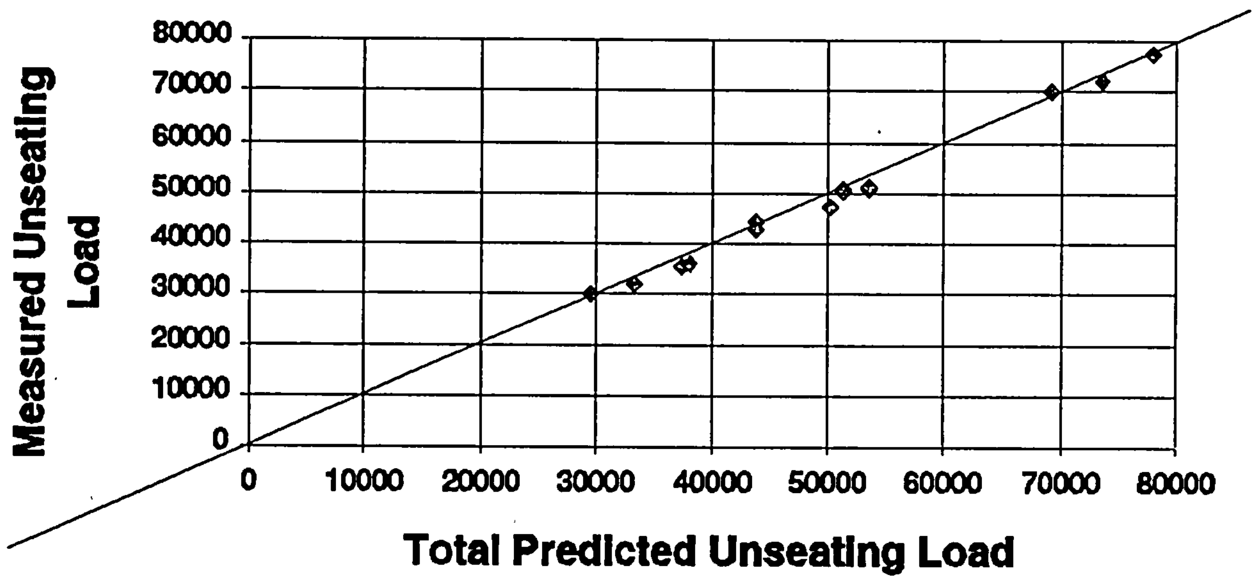
5. The pressure data for this test is questionable and is being evaluated at this time.

6. The upstream and downstream pressure during these tests was approximately 350 psig. This was done to approximate the LPCI and LPCS injection valve pressure conditions which could exist in the event of a LOCA.

Graphs 1 through 6 provide the data in Table 1 for the three test valves. The total measured unseating load versus the total predicted unseating load and the pressure related portion of the measured load versus the predicted pressure related portion of the unseating load are plotted for each valve.

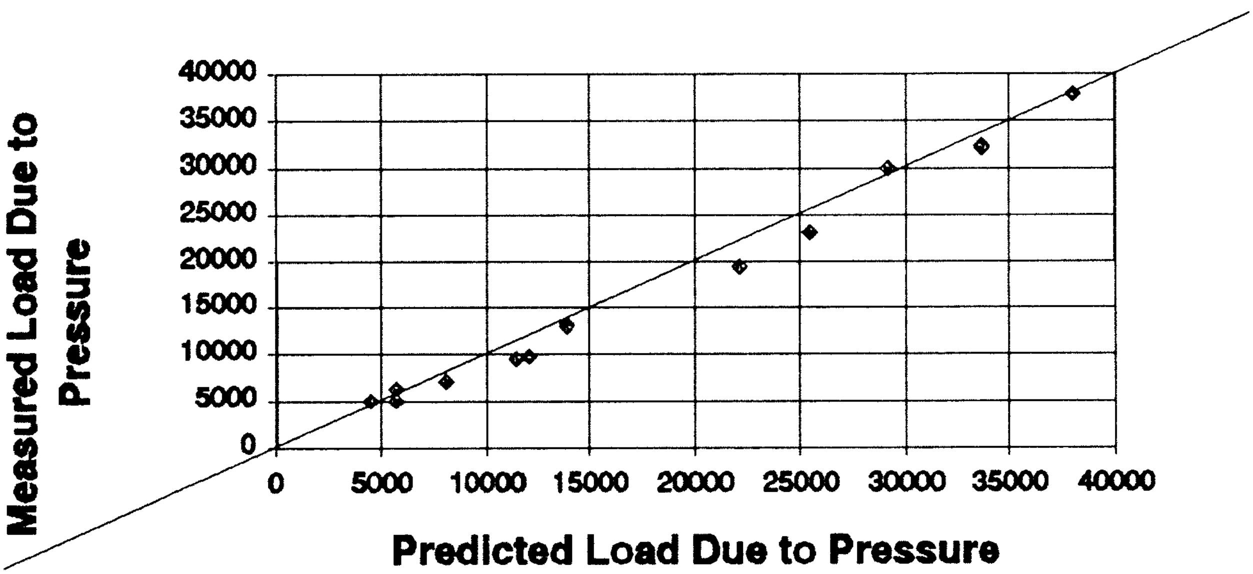
GRAPH 1

Predicted Unseating Thrust Versus Measured Pressure Locking Unseating Force for Crane Valve



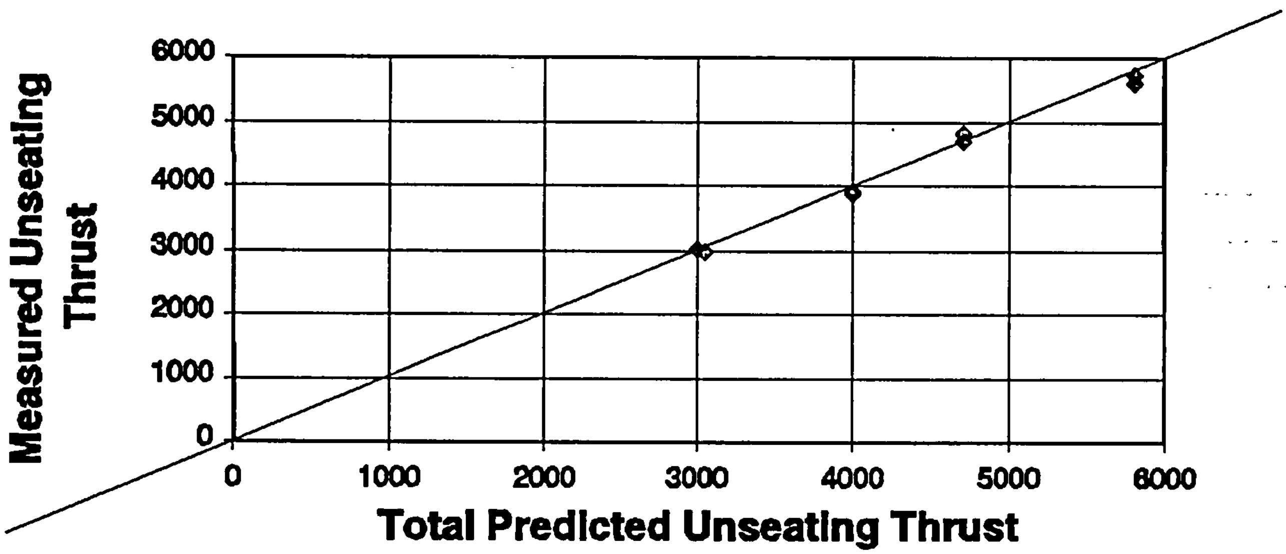
GRAPH 2

Predicted Versus Measured Portion of Pressure Thrust Due to Pressure Forces for Crane Valve

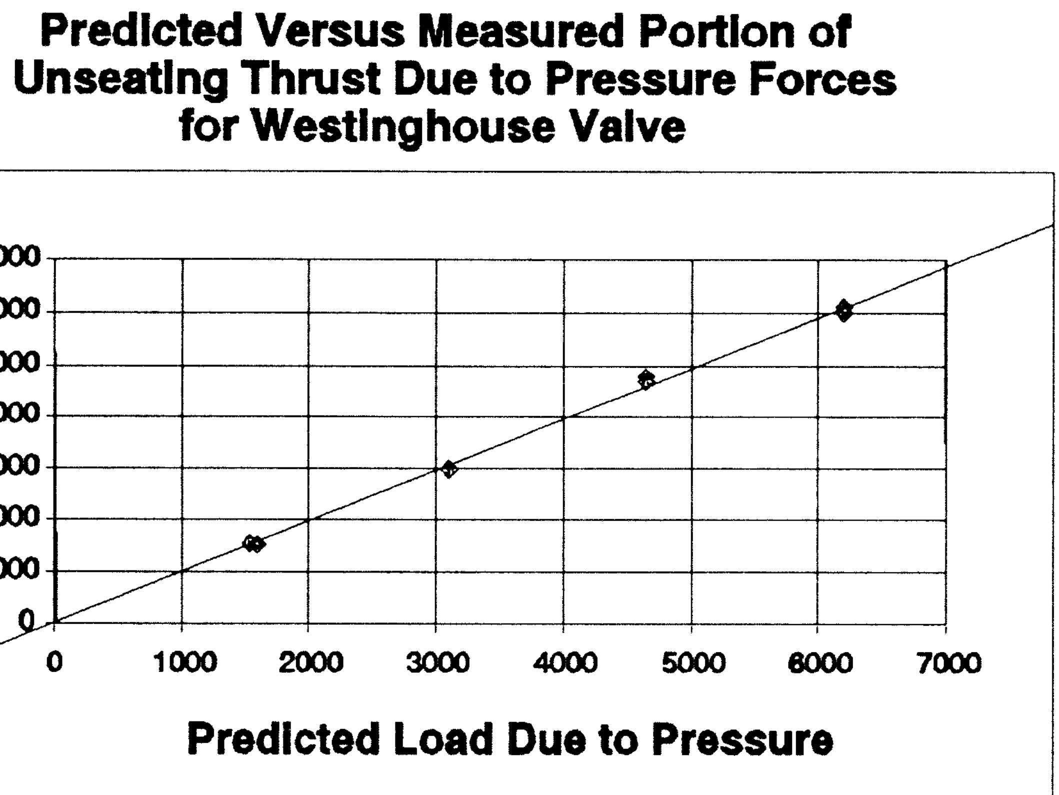


GRAPH 3

Predicted Unseating Thrust Versus Measured Pressure Locking Unseating Thrust for Westinghouse Valve



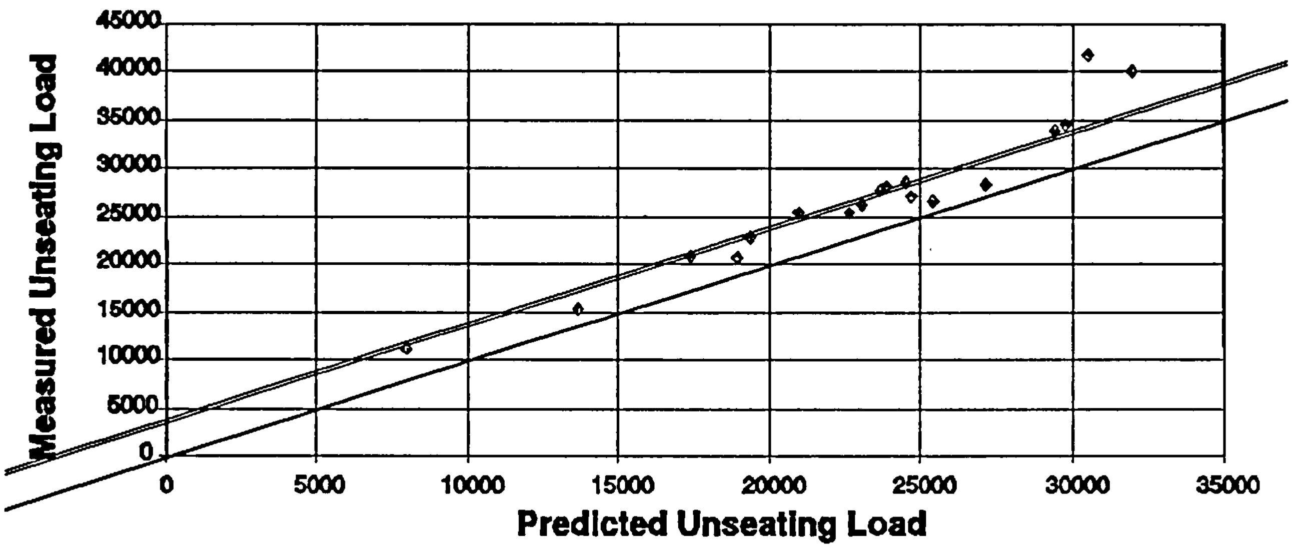
GRAPH 4



Predicted Versus Measured Portion of Unseating Thrust Due to Pressure Forces for Westinghouse Valve
Measured Load Due to Pressure
7000
6000
5000
4000
3000
2000
1000
0
0
1000
2000
3000
4000
5000
6000
7000
Predicted Load Due to Pressure

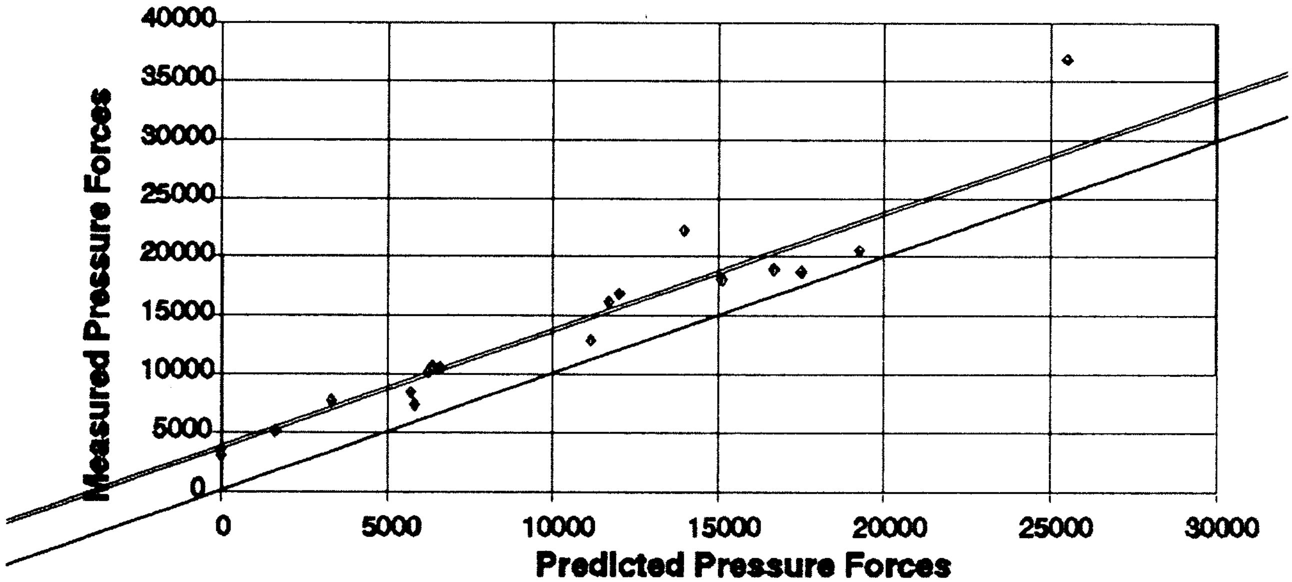
GRAPH 5

Predicted Unseating Thrust Versus Measured Pressure Locking Unseating Thrust for Borg-Warner Valve



GRAPH 6

Predicted Versus Measured Portion of Unseating Thrust Due to Pressure Forces for Borg-Warner Valve



PRIMARY DIFFERENCES BETWEEN THE COMMONWEALTH EDISON PRESSURE LOCKING CALCULATION AND THE PRESSURE LOCKING CALCULATION METHOD PUBLISHED IN NUREG/CP-0146

The ComEd methodology is based on calculating the contact load at the edge of the disk which results in an equal and opposite disk deflection to that caused by pressure trapped between the disks. The ComEd methodology differs in several ways from the methodology described in the Reference 4 NUREG.

- The NUREG Methodology ignores disk deflection due to hub elongation. This is non-conservative. For typical disk geometries, the expected impact of ignoring this effect is less than 5%.

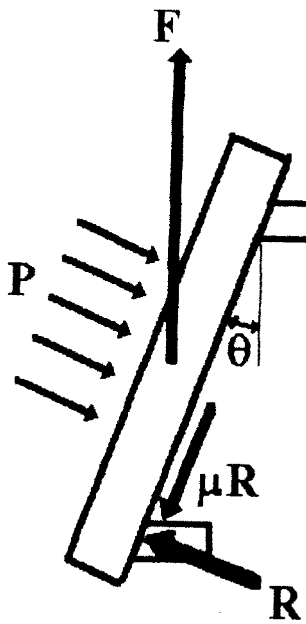
- The NUREG Methodology is based on using Table 24 of Roark's equations for calculating forces in the disk. This table ignores disk deflection due to transverse shear stresses. Section 10.3 of Roark's Equations discusses the conditions under which deflection due to shear is negligible. For typical disk geometries the deflection due to shear is often not negligible. Table 25 of Roark's Equations provides the equations for calculating disk deflection due to shear. Ignoring deflection due to shear is non-conservative. For small valve sizes where the disk thickness to disk diameter aspect ratio is large (>0.3), ignoring shear may result in under predicting the disk to seat contact load by 10% or more.

The ComEd methodology treats the vertical pressure force on the disk separately from the pressure locking load caused by the increased contact load between the seat and disk. The NUREG methodology relies on use of the open disk factor for translating the increased seating contact force into an increased unseating load. The open disk factor is based on a free body diagram in which the disk hub is unloaded. This is not the case for pressure locking. The NUREG treatment of these two components to the pressure locking unseating load is non-conservative. This source of non-conservatism is generally much more significant than the other concerns mentioned above for the NUREG method and is the primary ComEd concern with the NUREG method.

The derivations on the following pages are provided to support the discussion above.

OPEN SEAT FACTOR DERIVATION (Opening a valve against a differential pressure)

FIGURE 6



F = Stem Force (tension)

P = Pressure Force

= DP x Seat Area

R = Seat Reaction Force

μR = Seat Friction Force

θ = Seat Angle

Disk Factor (VF) = F / P (by definition)

Sum of forces in x-direction:

$$\sum F_x = P\cos\theta - R\cos\theta - \mu R\sin\theta \tag{31}$$

$$R = P\frac{\cos\theta}{\cos\theta + \mu\sin\theta} \tag{32}$$

Sum of forces in y-direction:

$$\sum F_y = F - P\sin\theta + R\sin\theta - \mu R\cos\theta \tag{33}$$

$$F = P\sin\theta - \left(P\frac{\cos\theta}{\cos\theta + \mu\sin\theta}\right)(\sin\theta - \mu\cos\theta)$$

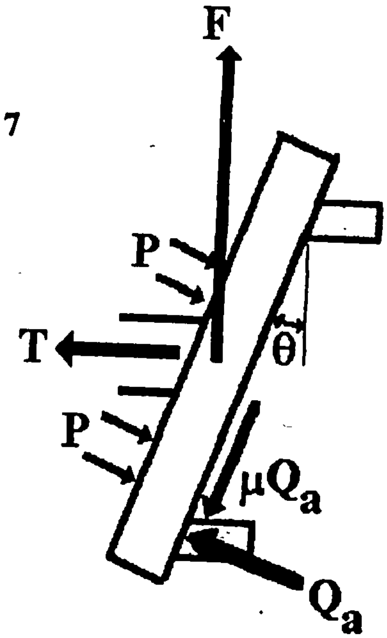
$$F = P\left[\frac{\sin\theta(\cos\theta + \mu\sin\theta)}{\cos\theta + \mu\sin\theta} - \frac{\cos\theta(\sin\theta - \mu\cos\theta)}{\cos\theta + \mu\sin\theta}\right]$$

$$\frac{F}{P} = \frac{\sin\theta\cos\theta + \mu\sin^2\theta - \cos\theta\sin\theta + \mu\cos^2\theta}{\cos\theta + \mu\sin\theta}$$

$$\frac{F}{P} = \frac{\mu}{\cos\theta + \mu\sin\theta} \tag{34}$$

PRESSURE LOCKING SUM OF FORCES

FIGURE 7



F = Stem Force (tension)

P = Pressure Force
= DP x Seat Area

Q_a = Seat Reaction Force
(calculated using Roark's)

μQ_a = Seat Friction Force
θ = Seat Angle

T = Disk Hub Tension

Note that the sum of the forces in the x-direction is different than for the seat factor case due to the hub tension force T. Consequently, the Q_a value is a typically a much lower portion of the P value under pressure locking than it is for the seat factor calculation. (This is the benefit of using Roark's equations for calculating the seat load increase.) Therefore, the sum of the forces in the y-direction should be solved for directly from the free body diagram above, as follows:

$$\sum F_Y = F - \mu Q_a \cos\theta - P\sin\theta + Q_a \sin\theta \tag{35}$$

$$\therefore F = Q_a(\mu\cos\theta - \sin\theta) + P\sin\theta \tag{36}$$

The first term in the equation above is the pressure locking load term in the ComEd methodology. The second term in the equation above is the F_{vert} or reverse piston effect term in the ComEd methodology. The ComEd method adds these two terms to the static unseating load and then subtracts the stem rejection load to get the predicted unseating load under pressure locking conditions.

Rather than use these equations, the NUREG method applies the open seat factor to the Q_a value. Because of the relationship in equation 37 below, the NUREG method substantially under predicts the vertical pressure force portion of the required thrust.

$$Qa < P\cos\theta \,/\, (\cos\theta + \mu \sin\theta) \tag{37}$$

XVI. REFERENCES

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2. MPR Calculations 101-013-1, "Effect of Bonnet Pressure on Disc to Seat Contact Load", dated 3/23/95; and 101-013-4, "Estimate of Valve Unseating Force as Function of Bonnet Pressure", dated 3/23/95.

3. Electric Power Research Institute, Nuclear Maintenance Applications Center, 1990, *Application Guide For Motor-Operated Valves in Nuclear Power Plants*, EPRI/NMAC Report NP-6660-D, March.

4. Smith, D.E., 1994, "Calculation to Predict the Required Thrust to Open a Flexible Wedge Gate Valve Subjected to Pressure Locking", *Proceedings of the Workshop on Gate Valve Pressure Locking and Thermal Binding*, NUREG/CP-0146, July 1995.

PRESSURE LOCKING TEST RESULTS

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ABSTRACT

The U.S. Nuclear Regulatory Commission (NRC), Office of Nuclear Regulatory Research, is funding the Idaho National Engineering Laboratory (INEL) in performing research to provide technical input for their use in evaluating responses to Generic Letter 95-07, "Pressure Locking and Thermal Binding of Safety-Related Power-Operated Gate Valves." Pressure locking and thermal binding are phenomena that make a closed gate valve difficult to open. This paper discusses only the pressure locking phenomenon in a flexible-wedge gate valve; we will publish the results of our thermal binding research at a later date. Pressure locking can occur when operating sequences or temperature changes cause the pressure of the fluid in the bonnet (and, in most valves, between the discs) to be higher than the pressure on the upstream and downstream sides of the disc assembly. This high fluid pressure presses the discs against both seats, making the disc assembly harder to unseat than anticipated by the typical design calculations, which generally consider friction at only one of the two disc/seat interfaces. The high pressure of the bonnet fluid also changes the pressure distribution around the disc in a way that can further contribute to the unseating load. If the combined loads associated with pressure locking are very high, the actuator might not have the capacity to open the valve. The results of the NRC/INEL research discussed in this paper show that the relationship between bonnet pressure and pressure locking stem loads appears linear. The results also show that for this valve, seat leakage affects the bonnet pressurization rate when the valve is subjected to thermally induced pressure locking conditions.

INTRODUCTION

Background

When a wedge gate valve opens against an ordinary differential pressure load, the actuator must provide enough force to unwedge the disc from the seats and to overcome the resistance created by friction at the downstream disc/seat interface. (Other loads, such as the packing load, the stem rejection load, and the vertical pressure load on the disc also contribute to the total stem load by either assisting or resisting stem movement.) Under differential pressure conditions, the upstream pressure tends to decrease the disc load at the upstream disc/seat interface and increase the load at the downstream disc/seat interface. (This scenario can apply in varying degrees to wedging parallel and split wedge gate valves as well). Typical formulas for estimating valve operating requirements are based on differential pressure times disc area times friction at only one sealing surface.

Pressure locking occurs when the valve bonnet pressure is higher than both the upstream and downstream pressures. In most gate valves (including most flex-wedge gate valves, split wedge gate valves, parallel disc gate valves, and double disc gate valves), the bonnet cavity communicates with the area between the disc faces. The effect is that the pressure of the fluid between the discs presses both the upstream and downstream disc surfaces against the seats, introducing resistance to motion at both disc/seat interfaces rather than just one. As a result, the total force necessary to unwedge/unseat the valve disc can be higher than in the normal differential pressure situation. The various forces involved are indicated in Figure 1. At its worst, pressure locking can cause the valve to be locked in the closed position, such that the actuator is unable to open it.

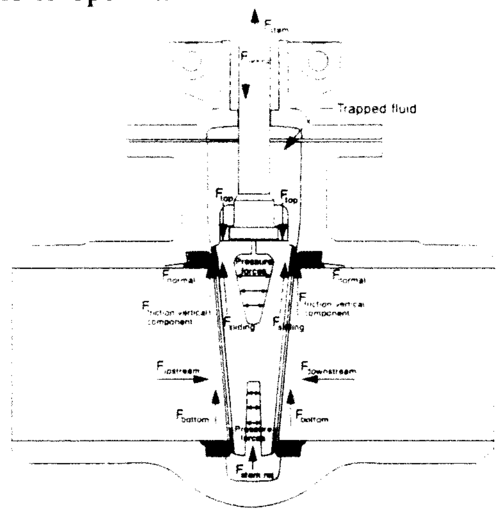

Figure 1: *Diagram of a gate valve, showing the various forces involved when pressure locking occurs.*

Pressure locking loads are much more difficult to predict than the design basis differential pressure load described above, especially with flex wedge gate designs. (The flex-wedge design is the most widely used of all gate valve disc designs.) With parallel disc, double disc, and split wedge gate valves, both discs respond equally and independently to the pressure of the fluid between the discs. However, the disc assembly in a flex-wedge valve is made from a single piece of metal, with the upstream and downstream halves of the disc connected in the center by a hub. As with the parallel disc design, the two discs respond to the pressure of the fluid between them, but the area exposed to the pressure is less, because of presence of the hub.

With the flex-wedge design, not all of the disc area exposed to the bonnet pressure will result in additional force at the disc/seat interface. For this reason, we distinguish between the disc area exposed to the bonnet pressure, and the effective disc area responding to the

bonnet pressure. When exposed to pressure locking loads, the disc assembly acts as a split load path component, with part of the pressure load deforming the disc and pressing it against the valve body seat, and part of the pressure load being reacted in the hub. The stiffer the disc, the more the pressure load is transferred to the hub. In simple terms, then, for each disc stiffness there is an effective area responding to the pressure load. Within a given pressure class of valve, the more flexible disc design will respond with a higher effective area.

One other feature of the flex-wedge gate design contributes to the effects of pressure locking. The angle of the disc, usually about 5 degrees from vertical in an upright valve, creates an effective area on the disc that is acted on by the bonnet pressure and the downstream pressure, and another acted on by the bonnet pressure and the upstream pressure. These areas are typically modeled as elliptical areas corresponding to the downstream and upstream orifices in the valve when viewed from above. The corresponding forces are indicated as F_{top} and F_{bottom} in Figure 1. In a six-inch valve opening against normal differential pressure (and assuming that the bonnet pressure is equal to the upstream pressure), there is no upstream vertical pressure load, and the downstream vertical pressure load (F_{top} - F_{bottom}), which resists opening, is offset by the stem rejection load ($F_{stem\ rej}$), which assists opening. However, in the pressure locked case, both the upstream and downstream vertical loads act to resist opening, resulting in yet another increase in the opening load, as compared to the normal differential pressure opening situation. The extent of the increase is dependent on the bonnet, upstream, and downstream pressures.

Taken together, the load increases described in the preceding paragraphs can cause the thrust needed to open a pressure locked valve to be higher than the value typically calculated by industry formulas for the design basis differential pressure conditions. If the higher thrust demands exceed the capability of the actuator, the valve will fail to open.

The bonnet pressure that causes pressure locking can be either hydraulically or thermally induced. Hydraulically induced pressure locking can result from various operational sequences involving low-pressure system interface with high-pressure systems, or from system depressurization during an accident. For example, a valve closed at high pressure might experience pressure locking if an attempt is made to reopen the valve after both the upstream and downstream sides have been depressurized, and with the high pressure remaining in the bonnet. Such a scenario occurred, for example, in 1991 at the Fitzpatrick nuclear power plant. Thermally induced pressure locking can occur by thermal expansion of water trapped in the bonnet. For example, a valve closed under cold conditions might experience pressure locking if the valve were later heated by a slug of hot fluid coming into contact with the closed disc, or by a line break inside the containment.

The instance of pressure locking that occurred at the Fitzpatrick station was hydraulically induced (Information Notice 92-26). The utility hydro-tested the piping between the inboard and outboard low-pressure coolant injection (LPCI) valves. The inboard LPCI valve is a 24-in. flexible wedge motor-operated valve. After the hydro-test, the utility depressurized the piping between the valves and filled and vented the system to return it to service. About 10 hours later the utility commanded the inboard valve to open.

The valve actuator was energized for about 30 seconds before the circuit breaker tripped. (The normal stroke time for this valve is 120 seconds.) The valve had failed to open. The root cause of the failure was pressure locking.

The magnitudes of possible loads due to pressure locking are specific to the valve design and to the plant system. The flexibility of the disc assembly is an important factor. Pressure locking can happen to valves that are required to be leak-tight in both directions, valves that have little or no leakage, or valves that have not been modified to prevent pressure locking. Typical modifications to gate valves to prevent pressure locking include venting the bonnet to the high-pressure side by drilling a hole through the disc, or by installing a vent line between the bonnet and the upstream side, with the line equipped with a check valve or a power-operated valve.

Purpose

The purpose of the INEL testing described in this paper is to provide technical information to the NRC in support of their effort to evaluate the pressure locking issue. (Thermal binding is being addressed separately.) The INEL effort consists of laboratory testing of two gate valves--a flex-wedge gate valve and a parallel disc gate valve. This paper describes the testing and results for the flex-wedge valve; results from testing of the parallel disc valve will be reported later, after testing and analysis are complete. The test program is designed to address the following questions:

1. Assuming low or zero pressure on the upstream and downstream sides of the disc, how much additional force is needed to open the valve as the bonnet pressure increases?
2. As the temperature of the fluid in the bonnet increases, how much does the bonnet pressure increase?
3. How does the presence of air entrapped in the bonnet affect these pressure increases and the resulting thrust requirements? (Does the air bleed off? Does it dissolve in the coolant?)
4. How does valve leakage affect the bonnet pressure?

TEST SETUP

The flexible wedge valve tested in this project is a 6-inch, 600-lb-class Walworth valve. We believe that the more flexible the disc in a flex-wedge gate valve, the greater the response to the pressure locking load. ASME and other valve design codes provide stress rules that establish minimum disc thickness (stiffness) for the various pressure classes. Because the wedge in this particular design is relatively flexible for its pressure class, we assumed that it would respond with a relatively high effective pressure area. This valve had been used in previous testing. Before the pressure locking tests, the valve sealing surfaces were reconditioned, and leakage was well below accepted limits.

To preclude the possibility of stalling the motor, we equipped the valve with an SMB-0 actuator with a 25-ft-lb motor, which is about twice as powerful as would be used in a typical safety-related application for a valve this size in a commercial power plant. Figure 2 shows the test setup and instrumentation for the pressure locking tests.

Tests were conducted at various pressures and temperatures imposed on the upstream side, the bonnet, and the downstream side of the

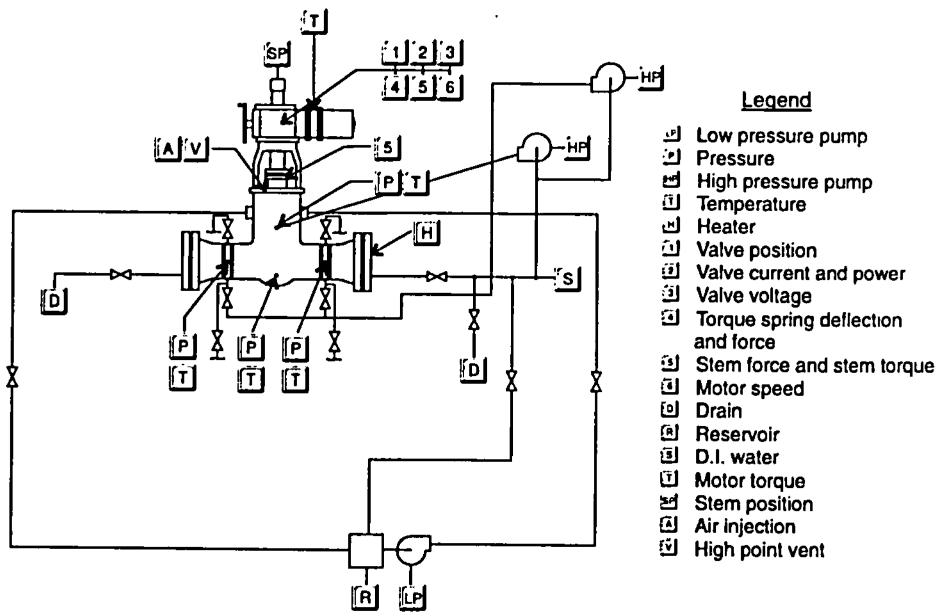

Figure 2: *Test setup and instrumentation for the pressure locking tests.*

valve. Before any testing, the valve sealing surfaces were preconditioned to provide a reasonable disc friction factor. Preconditioning required more time than expected.

Stem force, stem torque, stem position, motor current and voltage, and other valve and actuator parameters were monitored at a frequency of 600 samples per second during valve operation. Upstream, downstream, and bonnet fluid temperatures and pressures were monitored at a frequency of one sample per second during heatup and pressurization.

The valve was subjected to two sets of tests: cold pressure locking tests, and thermally induced pressure locking tests. The cold pressure locking tests focused on the relationship between bonnet pressure and unwedging load. The hot pressure locking tests were intended to examine the relationship between bonnet temperature and bonnet pressure. The hot testing included tests to study the effects of valve leakage on the thrust requirement, as well as the effects of air entrapped in the bonnet. Cold tests were performed first. After the hot tests, another round of cold tests was conducted. The following discussion describes the tests and the results.

COLD PRESSURE LOCKING TESTS

The cold pressure locking tests were designed as a parametric study to evaluate the relationship between the opening thrust and the fluid pressures occurring at various locations in the valve.

Testing

The test matrix, shown in Table 1, consisted of various upstream, downstream, and bonnet pressures distributed across the full range of possible conditions. In addition to the tests shown in the table, we performed baseline valve strokes periodically throughout the testing; the baseline stokes allowed us to evaluate the wedging versus unwedging relationship, obtain upstream and downstream seat friction values, and determine the load due to packing friction. The baseline strokes included a no-pressure (static) valve closing and opening cycle and two differential pressure opening stokes, one with the downstream side and the bonnet pressurized, and one with the upstream side and the bonnet pressurized.

For the cold pressure locking tests, the valve was filled with deionized water under normal environmental conditions, and any trapped air pockets were eliminated. Each step began with the valve open and pressurized to 1200 psig. The valve was then closed and the pressures in the upstream leg, downstream leg, and bonnet were bled down to the test settings shown in Table 1. The valve was then opened and the stem force required to extract the valve disc was measured. A typical thrust trace from such an opening stroke is shown in Figure 3. As mentioned earlier, the valve did not leak significantly; pressure locking occurred easily.

An additional set of tests was conducted with the valve bonnet pressurized by another method. Instead of pressurizing the entire assembly and then closing the valve, we pressurized the bonnet with the valve already closed. The test parameters with the bonnet pressurized by this method are shown in Table 2.

Results

Figure 4 shows stem force traces for five tests where the bonnet was pressurized at 0, 300, 600, 900, and 1200 psig. The traces have been truncated to focus on the unwedging portion of the opening stroke. The plot shows a linear relationship between the bonnet pressure and the unwedging/unseating load.

THERMALLY INDUCED PRESSURE LOCKING TESTS

Tests similar to those described above were performed on the valve at elevated temperatures. The effort in these tests was to evaluate the impact of temperature changes on the rate of bonnet pressurization and on the associated thrust requirements to unseat the valve during opening. The effects of valve seat leakage on bonnet pressurization were also investigated, as were the effects of air entrapped in the bonnet.

We used two different methods to heat the water in the bonnet. One method was to heat the upstream fluid (by means of a coil installed in the upstream leg of the valve), such that the heat was conducted to the bonnet through the disc and the valve body. The other method was to heat the bonnet from the outside using heat tape wrapped around the valve body. These two heating methods were intended to simulate ways that a valve might be heated in a plant scenario, namely, a slug of hot fluid coming into contact with a cold valve, and a cold valve being heated from the outside during a loss-of-coolant accident. As in the cold pressure locking tests, baseline static strokes and differential pressure strokes were performed throughout the test series to determine wedging/unwedging loads, valve seat friction values, and the packing load.

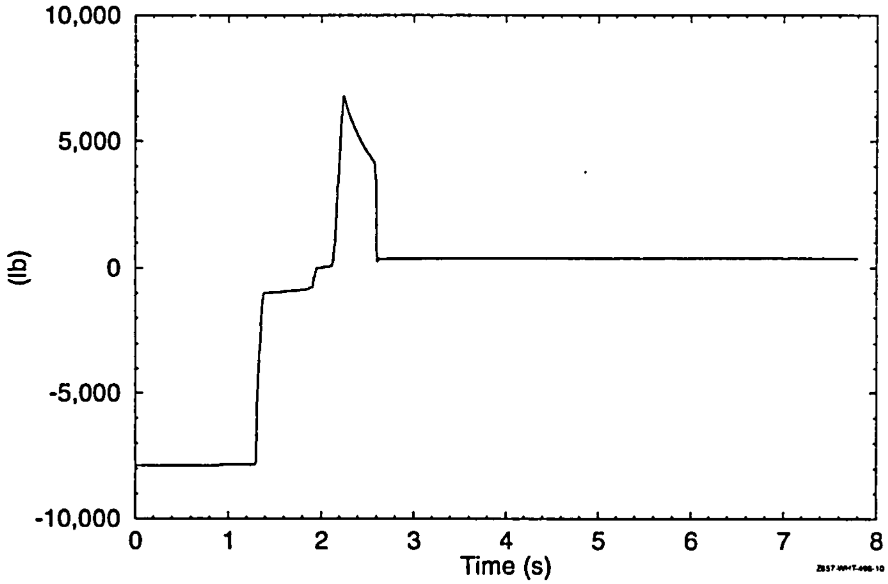

Figure 3: *Thrust trace from a typical opening stroke in a pressure locking test.*

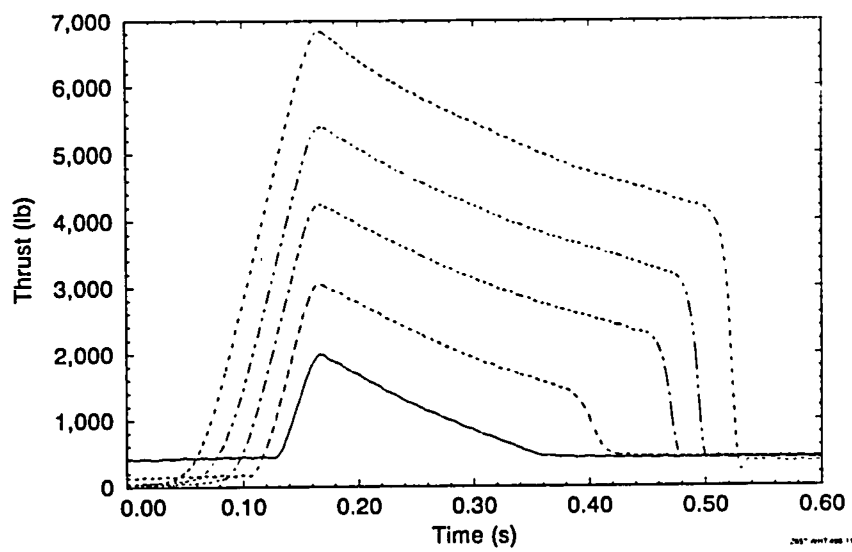

Figure 4: *Opening thrust measurements from pressure locking tests at five different pressures.*

Pressure, Temperature, and Stem Load

For the tests with internal heating of the upstream leg of the valve, the valve was filled with deionized water under normal environmental conditions, and any trapped air pockets were eliminated. The valve was closed and the upstream leg and bonnet were pressurized to 50 psig. The downstream leg was depressurized by opening the downstream high point vent line. The upstream side of the valve was heated at 80°F per hour to an upstream fluid temperature of 290°F while the upstream pressure was bled off to maintain the pressure at 50 psig during heatup. The discharge from the downstream high point vent line was measured as an indication of leakage from the bonnet to the downstream side. The pressures and temperatures were based on an accident scenario that resulted in a containment pressure of 50 psi with the fluid remaining subcooled.

The assumption that the bonnet leaked only to the downstream side was based on the results of earlier check-out tests, during which each chamber (bonnet, upstream side, downstream side) were pressurized. During these check-out tests, the bonnet leaked to the downstream side, but the upstream side maintained its pressure. Therefore the bleed-off to maintain 50 psig in the upstream leg during the heatup tests was due to the expansion of the upstream water only, and the discharge from the downstream side was mainly due to leakage from the bonnet.

The bonnet pressure was also monitored during heatup and compared to the measured leakage to establish the relationship between leakage and thermally induced bonnet pressurization. If and when the bonnet pressure reached 1200 psig, we opened the valve and measured the stem load.

The disc friction factors for both seats tended to be lower in the early tests than in the later tests. The friction factors stabilized during the hot tests and remained at the stable, high value during the subsequent round of cold tests. This is consistent with industry experience for elevated-temperature preconditioning. Because we regularly monitored the disc friction factor by conducting baseline tests, changes in the friction factor did not detract from the usefulness of the stem force measurements taken during the pressure locking tests.

Two of the heatup tests produced bonnet pressures capable of causing pressure locking loads. The results from opening strokes during those two tests showed that the required thrust is linear with pressure. The performance of the valve at elevated temperature, evaluated with consideration for the change in the disc friction factor, was consistent with the results from the earlier cold tests.

Effects of Valve Leakage

Leakage was monitored in tests with both internal and external heating. Table 3 presents leakage data from the various heatup tests. For this valve, it was necessary to pressurize the bonnet initially to establish a tight seal so that thermally induced pressure locking could occur. Table 4 shows the leakage rates for cold pressurization of the bonnet. All the measured leakage rates were very small, well within code requirements for this 6-in. Walworth valve.

When we heated the unpressurized bonnet water, the leakage was sufficient to prevent pressurization as the fluid expanded. In contrast, when we began the heating test with the bonnet already pressurized to 700 psig,

pressurization due to thermal expansion occurred. We assume that the initial pressure (before heating) caused the disc surfaces to seal to the seat surfaces more effectively than in the tests without initial pressure.

Note that evidence of valve leakage during a differential pressure test does not necessarily mean that leakage will prevent pressure locking. The disc elastic response during a pressure locked condition and its match-up to the seat is different in the pressure locking case, as compared to the differential pressure case. The effect of this difference was evident in the results we saw, where the leakage rates in the differential pressure tests showed no relationship to the leakage rates in the bonnet pressure tests. This demonstrates that the disc is matched up to the seat surface differently.

Effects of Entrapped Air

We performed other tests, with heat applied from the outside of the valve, to determine the effect of entrapped air on the bonnet pressure and on the associated opening stem load. We also investigated the extent to which the air will remain entrapped during operation, that is, whether the air pocket will remain intact, bleed off, or dissolve into the water.

Parametric tests were performed with air pockets representing 0.0, 0.5, 1.0, and 2.0 percent of the total valve volume. The valve was filled with deionized water, as in the previous tests, and air pockets were established by draining a known volume from the lower drain line while allowing air to enter through the high bonnet vent. Once the appropriate air volume was established, all external tubing was isolated and the valve was heated to 290°F using external heaters (heat tape). The pressure was monitored during the test and the valve was depressurized to 50 psig any time the pressure reached 1200 psi.

Figure 5 shows results from the heatup test with no entrapped air. The pressure increases rapidly after the bonnet pressure reaches 200 psig. Subsequent pressurization (following scheduled depressurizations) are very repeatable. Figures 6 through 8 show the same kind of data, but from tests with entrapped air volumes of 0.5, 1.0, and 2.0 percent air by volume. Here the behavior is similar, except that the initial pressurization begins at a higher temperature. As in the no-air test, subsequent repressurizations following depressurizations occur immediately. The fact that the presence of an air pocket delays the first pressurization but not the subsequent pressurization may indicate that the air pocket is either collapsed or forced into solution by the first pressurization cycle.

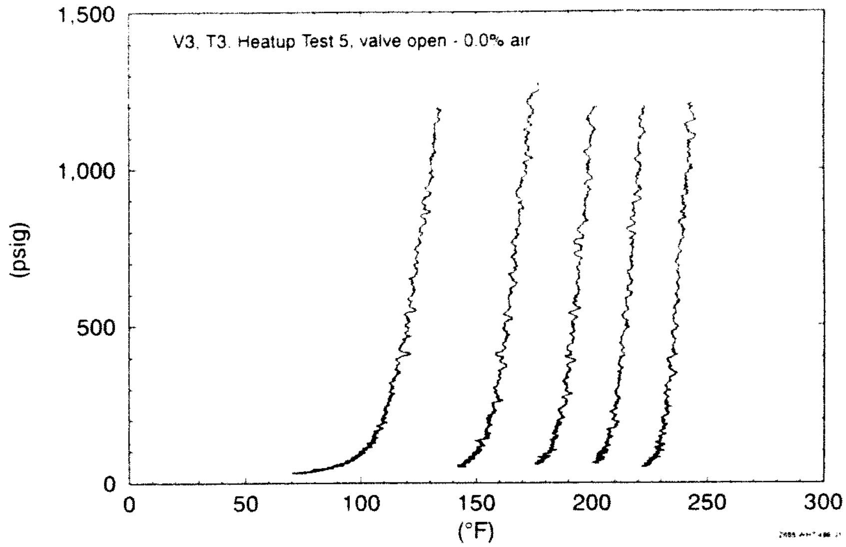

Figure 5: *Pressure versus temperature with no entrapped air.*

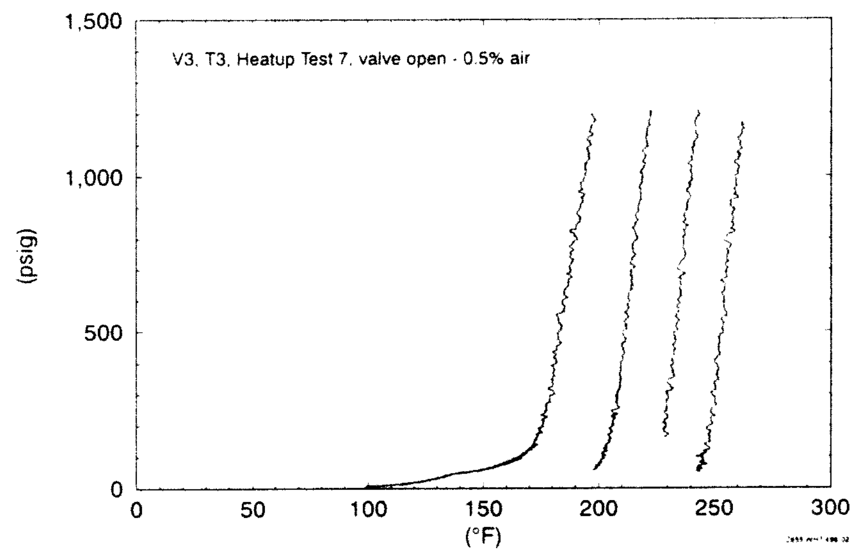

Figure 6: *Pressure versus temperature with 0.5% entrapped air.*

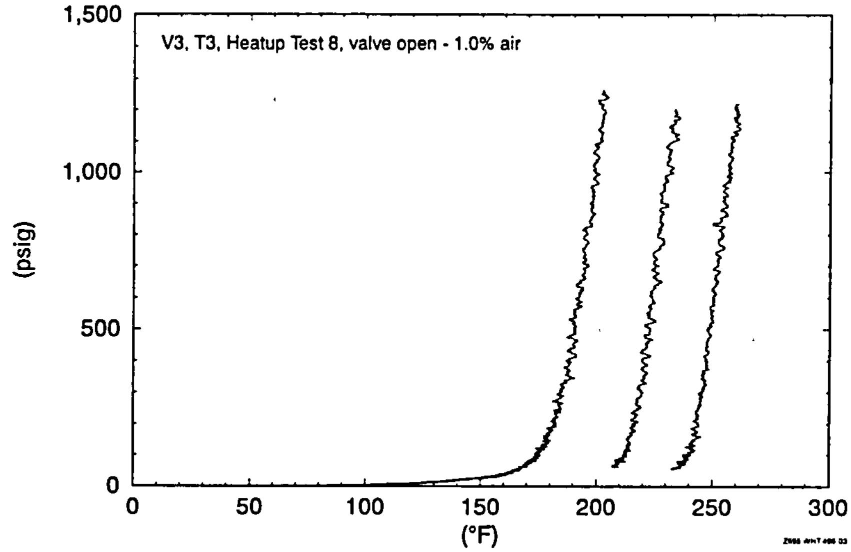

Figure 7: *Pressure versus temperature with 1.0% entrapped air.*

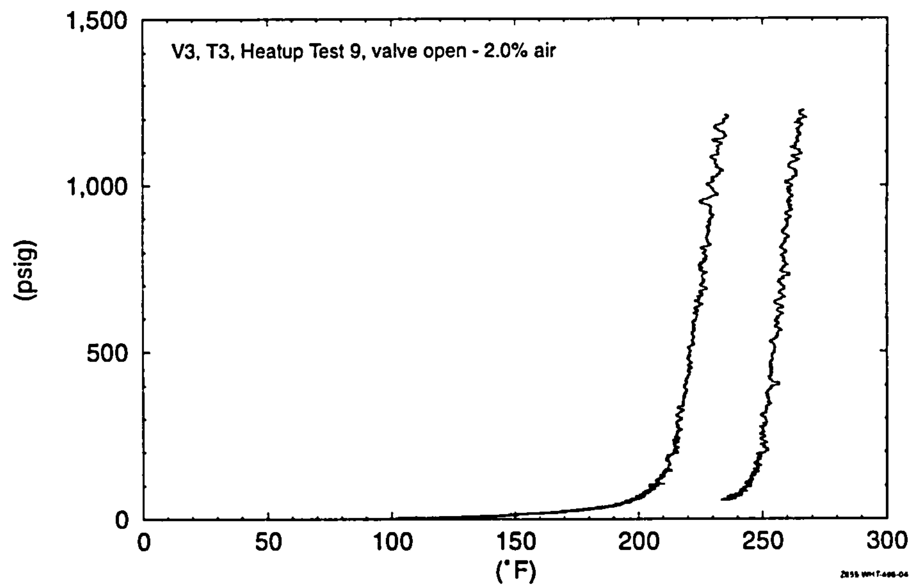

Figure 8: *Pressure versus temperature with 2.0% entrapped air.*

CONCLUSIONS

For this valve, the stem force required to overcome pressure locking loads appears linear with pressure. The flexibility of the disc determines the effective disc area responding to the bonnet pressure, which in turn affects the thrust needed to open the valve.

Small amounts of leakage can affect the pressurization rate in a thermally induced pressure locking scenario. Not enough data are available to quantify the leakage rate that would be necessary to prevent pressure locking from occurring. The leakage rates measured with a differential pressure across the valve were different from the leakage rates measured with only the bonnet pressurized.

We infer from this result that leakage in a differential pressure test is no guarantee that pressure locking will not occur.

In a thermally induced pressure locking situation, the presence of air trapped in the bonnet may delay the pressure buildup in the bonnet, but it will not prevent it.

To date we have completed testing of one valve. More data are needed to determine the extent to which these results apply to other gate valves.

Table 1. Valve Opening Thrust Versus Valve Pressure Parametric Tests.

Downstream pressure (psi)	Bonnet pressure (psi)	Upstream pressure (psi)
0	0	0
0	300	0
0	600	0
0	900	0
0	1200	0
300	1200	0
600	1200	0
900	1200	0
1200	1200	0
0	1200	300
0	1200	600
0	1200	900
0	1200	1200
300	1200	600
300	1200	300

Table 2. Direct Bonnet Pressurization Tests.

Downstream pressure (psi)	Bonnet pressure (psi)	Upstream pressure (psi)
0	300	0
0	600	0
0	900	0
0	1200	0

Table 3. Leak rate data during heat-up tests.

Initial bonnet pressure (psig)	Average heat rate (°F/hr)	Average leak rate from bonnet to downstream (cm^3/min)	Range of bonnet to downstream leak rates (cm^3/min)	Average leak rate from upstream to maintain 50 psig (cm^3/min)	Range of upstream leak rates to maintain 50 psig (cm^3/min)
50	45	16.2	12.5 - 22.2		
600	NA	41.1	NA		
1200	NA	1.5	NA		
50	66.6	1.6	1.45 - 1.83		
50	54.1	1.2	0.91 - 3.12	2.33	1.38 - 3.12
50	85.2	24.2	4.97 - 130	2.15	1.82 -2.49
700	13.8	2.3		9.7	

Table 4. Cold leak rate data.

Bonnet pressure range (psig)	Leak rate from bonnet to downstream (cm^3/min)
1104 - 1326	0.216
504 - 640	1.233
1039 - 1294	0.114
572 - 670	0.078

EFFECTS OF PRESSURE AND TEMPERATURE ON GATE VALVE UNWEDGING

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ABSTRACT

The stem thrust required to unwedge a gate valve is influenced by the pressure and temperature when the valve is closed and by the changes in these conditions between closure and opening. "Pressure locking" and "thermal binding" refer to situations where pressure and temperature effects cause the unwedging load to be much higher than normal. A model of these phenomena has been developed. Wedging (closure) is modeled as developing an "interference" between the disk and its seat rings in the valve. The effects of pressure and temperature are analyzed to determine the change in this disk-to-seat "interference". Flexibilities of the disk, body, stem and yoke strongly influence the unwedging thrust. Calculations and limited comparisons to data have been performed for a range of valve designs and scenarios. Pressure changes can increase the unwedging load when there is either a uniform pressure decrease, or a situation where the bonnet pressure exceeds the pressures in the adjacent piping. Temperature changes can increase the unwedging load when: (1) valve closure at elevated system temperature produces a delayed stem expansion, (2) a temperature increase after closure produces a bonnet pressure increase, or (3) a temperature change after closure produces an increase in the disk-to-seat "interference" or disk-to-seat friction.

INTRODUCTION

Background

Wedge gate valves are commonly used in nuclear power plant fluid systems. When a wedge gate valve is closed, it is seated or "wedged" between two seat rings. The wedge angle is small enough and the friction high enough that the disk is typically self-locking between the seat rings, i.e., a force is required to extract, or unwedge, the disk when the valve is opened. The unwedging force can be affected by pressure and temperature changes in the valve and adjacent piping which occur between the time of closure and subsequent opening. Under some situations, increases in the unwedging load can occur. Pressure locking and thermal binding are terms which refer to situations with a markedly increased unwedging load. Pressure locking typically refers to the situation where the bonnet pressure is greater than that in the surrounding piping,

thereby promoting higher disk-to-seat contact load. Thermal binding refers to a situation where a valve temperature change increases the disk-to-seat load, e.g., as a result of differential thermal expansion.

Previous work investigating pressure locking and thermal binding includes analytical studies (Dogan, 1994, Izekoye, 1994, Wang and Kalsi, 1992). However, there is little data available to verify the analytical studies of these effects. The NRC hosted a workshop (Brown, 1995) which provided additional insights and experience from power plant personnel. In 1995, the NRC issued Generic Letter 95-07, which requested that plants evaluate safety-related gate valves for susceptibility to pressure locking and thermal binding, and take corrective actions where appropriate.

To evaluate pressure locking and thermal binding, the authors have developed analytical models to predict gate valve unwedging thrust, including the effects of pressure and temperature on unwedging. These models have proven to be useful to determine which changes in pressure and temperature create significant thrust increases and to quantify the unwedging thrust. Limited data have been compared to the models, with encouraging results.

Objectives

The objectives of this paper are to:

- describe analytical models for determining the influence of pressure and temperature on unwedging thrust;
- describe approaches for using the models to predict unwedging thrust; and
- present the results of calculations and limited comparisons to test data.

Finally, this paper summarizes the influences of pressure and temperature on unwedging thrust which are most likely to affect gate valve operation.

OVERVIEW

When a wedge gate valve is closed, the disk is squeezed, or compressed slightly, between the seat rings, and the seat rings are expanded slightly. In effect, an "interference" is established; the disk wants to occupy more space than is available between the seat rings, and the disk-to-seat loads force both the disk and the seat rings to deflect so that the disk fits between the seat rings. Pressure changes in the upstream pipe, bonnet, or downstream pipe change the loads applied to the disk and body, and cause deflections in these components. These deflections change the "interference" between the disk and the seat rings. For example, a pressure increase that tends to expand the valve body will move the seat rings further apart and will decrease the "interference", or a bonnet pressure increase that tends to make the faces of a flexible wedge disk move further apart will increase the "interference".

Temperature changes can directly affect the disk-to-seat "interference" as a result of differential thermal expansion, or temperature changes can indirectly affect the "interference" by:

- causing the seating load exerted by the stem to increase above that used to initially close the valve; and
- causing the pressure (particularly in the bonnet) to increase.

Further, disk-to-seat friction coefficient is a function of temperature. Therefore, temperature changes can affect the unwedging thrust even if the disk loads and the disk-to-seat "interference" remain unchanged.

The analytical model described in this paper addresses the effects discussed above. Figure 1 provides a summary of the phenomena and mechanisms by which pressure and temperature influence the unwedging load. As shown in the figure, there are six basic situations which need to be considered. The situations are described individually in the model description below.

MODEL DESCRIPTION

Wedging and Unwedging

During disk wedging with uniform pressure in the valve and adjacent piping (i.e., when there is no flow during closure), the loads applied to the disk are the stem thrust and the contact loads at the disk-to-seat interfaces (Figure 2). The contact load at each interface can be treated as having normal and friction components. These force components can be determined as a function of the maximum thrust exerted by the stem on the disk to close the valve (F_i), the seat half-angle (Θ) and the disk-to-seat friction coefficient (μ). A simple solution for the normal load (R) at each disk-to-seat interface, which assumes equal friction coefficients at the two disk-to-seat interfaces, is provided in Equation (1).

$$R = \frac{F_i}{2\,(\sin\Theta + \mu\cos\Theta)} \qquad (1)$$

The disk-to-seat contact load (R) will cause deflections in the disk and in the body seat rings. These deflections can be characterized as follows:

$$\delta_{DR} = R * G_{DR} \tag{2}$$

$$\delta_{SRR} = R * G_{SRR} \tag{3}$$

Where

δ_{DR}	=	disk deflection in pipe-axis direction (change in disk face-to-face dimension) due to normal load
δ_{SRR}	=	seat ring deflection in pipe-axis direction (change in seat ring face-to-face dimension) due to normal load
G_{DR}	=	compression of the disk in the pipe-axis direction for unit normal loads on the seating surfaces
G_{SRR}	=	expansion of the seat rings (and body) in the pipe axis direction for unit normal loads on the seating surfaces

The "interference", i_R, is the sum of the disk and seat ring deflections.

$$i_R = \delta_{DR} + \delta_{SRR} \tag{4}$$

When the valve is unwedged, the stem thrust which pushes the disk into the seat rings first is removed and then a stem thrust which pulls the disk out of the seat rings is applied. The disk-to-seat friction forces which opposed disk insertion reverse their directions and oppose disk extraction. If it is assumed that the "interference" (i_R) and contact load (R) remain constant until unwedging motion occurs, the extraction, or unwedging, thrust (F_e) is determined by Equation (5).

$$F_e = F_i \left[\frac{\mu \cos \Theta - \sin \Theta}{\mu \cos \Theta + \sin \Theta} \right] \tag{5}$$

When the compressive stem thrust is removed and the tensile stem thrust is applied, internal elastic distortions of the disk can change the disk-to-seat interface load, even though the disk does not move relative to the seat rings. Such deflections will tend to reduce the contact load; hence, actual unwedging loads are typically less than the value given by Equation (5).

Equations (1) through (5) cover the situation with constant, uniform pressure and temperature during wedging and unwedging. A similar, but slightly more complicated set of equations can be developed to address the effect of constant but nonuniform pressure, i.e., a differential pressure (DP) between upstream and downstream piping. If the DP remains constant throughout

the wedging and unwedging process, the effect is that a portion of the closure thrust goes to overcoming the DP and the remaining portion wedges the disk between the seat rings. The disk-to-seat interface loads are different at the upstream and downstream seating surfaces, and the deflections of the disk and seat rings need to be calculated by considering the "halves" of the configuration separately and then adding them. Assuming that the "interference" does not change, the opening thrust also includes components to overcome DP as well as to unwedge the disk.

Influence of Pressure Changes

Pressure changes between the time the valve is closed and when it is opened can occur independently in the bonnet, upstream piping and downstream piping. These pressure changes are referred to as ΔP_b, ΔP_u, and ΔP_d, respectively (Figure 3). These pressure changes affect the disk-to-seat contact loads at the two disk-to-seat interfaces. The changes in disk-to-seat contact load are not statically determinant and can be calculated only by considering the flexibility of the components. A solution can be obtained by determining the changes in the disk face position and seat ring face position at each of the two interfaces, due to the applied pressure loads in the absence of reaction loads at the interfaces. The net result of these position changes is an increase or decrease in the amount of "interference" between the disk and seat rings. This change in "interference" determines the increase or decrease in disk-to-seat contact load. With that force, a free body of the disk can be used to determine the thrust required to unwedge the disk.

The seat rings deflect because the body expands elastically as its internal pressure is increased.

$$\delta_{SRP} = \Delta P_b * G_{SRP} \tag{6}$$

Where

δ_{SRP} = seat ring deflection in the pipe axis direction (change in seat ring face-to-face dimension) due to pressure in the valve body.

G_{SRP} = expansion of the seat rings (and body) in pipe-axis direction due to a unit internal pressure in the valve body.

The seat ring deflections could also be affected by the pressures in the upstream and downstream piping (or by the axial and bending loads in these pipes), but those effects are beyond the scope of the present study.

The disk deflects because the disk halves bend, and the disk hub stretches or compresses in response to pressure changes (Figure 4). The bending deformation is determined by differences between the bonnet pressure and either the upstream or downstream pressure. For the case where $\Delta P_u = \Delta P_d$:

$$\delta_{DP} = 2 * (\Delta P_u - \Delta P_b) * G_{DP} \tag{7}$$

Where

δ_{DP} = disk deflection in pipe axis direction due to pressure

G_{DP} = deflection of the disk half in the pipe axis direction due to a unit differential pressure across the disk

If $\Delta P_u \neq \Delta P_d$, the solution is more complicated and can be obtained by considering the disk halves deflecting separately. The deflections of the seat rings and disk change the "interference" created when the valve was wedged. As a result, the disk-to-seat normal loads change to be consistent with the new value of "interference". The disk-to-seat contact loads are obtained by simultaneously solving the free body equilibrium and deflection equations.

With the sign convention shown, the deflections in Equations (6) and (7) both tend to reduce the "interference". Hence, increases in upstream or downstream pressure will tend to make the valve easier to unwedge; an increase in bonnet pressure can either make the valve easier or harder to unwedge, because of the competing effects of body expansion and disk expansion. For example, in a valve with a solid or very stiff flexible disk, a pressure increase may cause the body to expand enough to loosen the disk in the seat rings.

Bonnet Pressure Increase Due to Temperature Change

An increase in temperature will tend to increase the bonnet pressure of gate valves in water systems due to thermal expansion of the water. The pressurization rate with temperature is dependent on:

- water expansivity and compressibility, as expressed by the coefficient (dP/dT). For water, this coefficient increases with temperature.
- presence of compressible gas (e.g., air) in the bonnet.
- the valve bonnet thermal expansivity and elasticity.
- effective bonnet volume growth due to stem extraction through clearances prior to disk unseating.
- bonnet leakage through seats, gaskets or packing.

An expression which can be used to determine the change in bonnet pressure is developed based on the principle that the water volume plus gas bubble volume at the final condition is equal to the available bonnet volume.

$$(M - M_L) * \nu_s (P_2, T_2) + V_{BUB2} = V_{BON} * F_T * F_P + V_S \quad (8)$$

Where

M	=	initial water mass = $(V_{BON} - V_{BUB1})/\nu_1\ (P_1, T_1)$
V_{BUB2}	=	final bubble volume = $V_{BUB1} * (P_1/P_2) * (T_1/T_2)$
F_T	=	thermal growth factor = $(1 + \alpha_b\ (T_2 - T_1))^3$
F_P	=	elastic growth factor = $(1 + G\ (P_2 - P_1))^3$
M_L	=	mass of water leakage
V_{BON}	=	bonnet internal volume
V_{BUB1}	=	initial bonnet gas bubble volume
V_S	=	volume increase due to stem motion
ν_1, ν_2	=	initial, final water specific volume
P_1, P_2	=	initial, final bonnet pressure
T_1, T_2	=	initial, final bonnet temperature
α_b	=	bonnet thermal expansion coefficient
G	=	fractional growth in body linear dimensions due to a unit pressure increase

If the expressions in the definitions are substituted into Equation (8), a single expression which can be used to solve for P_2 (final pressure) is obtained. Because P_2 appears three places in the equation (in the specific volume property function, in the final bubble volume, and in the elastic growth factor), an iterative solution is required. The inputs to the equation are determined from valve physical dimensions, material properties, water properties, and application-specific attributes (leakage, bubble volume, stem motion). In the absence of application specific data, zero can be conservatively used for all three parameters. The bonnet flexibility (G) can be determined from finite element models or from more approximate methods such as treating the bonnet as a cylinder. Approximate methods have been used in the calculations described in this paper.

Disk-Body Differential Thermal Expansion

Figure 5 shows the disk, seat ring and body interaction. The model addresses uniform temperature changes in these components; the effects of nonuniform changes which might occur are beyond the scope of this paper. The effect of a uniform temperature change is to change the disk-to-seat "interference" as follows:

$$\frac{\Delta i}{2 * (T_2 - T_1)} = (\alpha_o - \overline{\alpha}_{bd}) * g + (\alpha_d - \alpha_b) * w_d/2 + (\alpha_s - \alpha_b) * s \qquad (9)$$

Where

α_o, α_b, α_d, α_s	=	overlay, body, disk, seat thermal expansion coefficient
$\overline{\alpha}_{bd}$	=	½ $(\alpha_b + \alpha_d)$
g, w_d, s	=	dimensions on Figure 5
T_1, T_2	=	initial, final temperature

If the disk, seat rings and body are not identical materials (either $\alpha_b \neq \alpha_d$ or $\alpha_s \neq \alpha_d$), then the second or third (or both) terms dominate the right-hand side of Equation (9). Significant "interference" increases can occur when $(T_2 - T_1)$ and either $(\alpha_d - \alpha_b)$ or $(\alpha_s - \alpha_b)$ are positive, i.e., heating of a valve where the disk or seat rings are more expansive than the body. Similarly, if these terms are negative (i.e., cooling of a valve where the disk or seat rings are less expansive than the body), then "interference" increases can occur. Usually, the disk, seat rings and body are similar materials, and the second and third terms disappear. In rare situations, an austenitic stainless steel disk is used in a carbon steel body ($\alpha_d > \alpha_b$), which results in a sensitivity to heating.

If the disk, body and seat rings are identical materials, only the first term on the right hand side of Equation (9) has an influence. The thermal expansion coefficient of Stellite 6 (the most common overlay material) is approximately halfway between that of carbon steel (lower value) and austenitic stainless steel (higher value). Accordingly, heating of a carbon steel valve and cooling of a stainless valve by similar amounts creates nearly identical increases in "interference". The effect of this "interference" change is evaluated using the flexibility model described previously. Typically, with similar disk, seat ring and body materials, the effect of the overlay is small for flexible wedge gate valves, but can be significant for solid wedge gate valves.

Stem-Body Differential Expansion

This effect occurs when the stem coefficient of thermal expansion is different from that of the body and bonnet. A uniform temperature change in temperature produces a force change in the stem, which is calculated by computing a differential stem length and dividing by the flexibility.

$$\Delta F_s = \frac{\Delta l}{G_s} = \frac{(\alpha_{st} - \alpha_b) * l * (T_2 - T_1)}{G_s} \qquad (10)$$

Where

ΔF_s	=	change in stem thrust
Δl	=	length change
G_s	=	flexibility (discussed below)
α_{st}, α_b	=	stem, body thermal expansion coefficient
l	=	stem-to-body overlap length
T_1, T_2	=	initial, final temperature

In Equation (10), G_s is the flexibility of the stem and its structural restraint which would typically include the disk, seat rings, body, bonnet, yoke and actuator (including the attachments between these components). G_s is the deflection of the stem relative to the disk for a unit load applied at their junction. If the stem can grow freely with little or no load increase (such as may occur with an air actuator), then this flexibility is essentially infinite and Equation (10) yields a thrust change of zero. If, however, the stem is structurally constrained (as occurs in most

MOVs), then G_s can be very low and a stem thrust change due to a temperature change can be expected. Values of G_s are a challenge to determine analytically. A review of load and displacement data taken on 19 motor-operated gate valves sized from 2½ to 18-inches showed that G_s ranges from 2×10^{-6} in/lbf to 10×10^{-6} in/lbf. Valves with G_s values above 4×10^{-6} in/lbf tend to be exclusively low pressure class (150-lb and 300-lb) valves.

The ΔF_s value determined in Equation (10) is a change in the seating thrust, i.e., an increase tends to wedge the disk more tightly into the seat rings, just as if the valve had been closed with increased thrust during its closing stroke. As shown in Equation (10), increases in seating thrust occur either when:

- $(\alpha_{st} - \alpha_b)$ and $(T_2 - T_1)$ are both positive (i.e., heating with the stem more expansive than the body), or

- $(\alpha_{st} - \alpha_b)$ and $(T_2 - T_1)$ are both negative (i.e., cooling with the stem less expansive than the body).

Friction Coefficient Change

As shown in Equation (1), the disk-to-seat contact load developed during wedging is a function of the disk-to-seat coefficient of friction when the valve is closed. In general, a greater disk-to-seat contact load will be developed when the disk-to-seat friction coefficient is lower. If the friction coefficient increases after the disk is seated, the required unwedging load increases because the disk-to-seat sliding load increases proportionally to the friction coefficient. Separate effects and gate valve research indicates that disk-to-seat friction coefficient for self-mated Stellite 6 tends to increase as the temperature decreases.

As discussed in Harrison, et. al. (1994), the friction coefficient decreases from about 0.6 at room temperature to about 0.4 at 650°F. The average sensitivity coefficient is about -(0.2)/(580°F) or -3.4×10^{-4} F^{-1}. Using equations (1) and (5), the change in unwedging load per unit change in temperature is dependent on the initial friction coefficient and on the seat angle. Figure 6 shows the sensitivity for typical values of μ and seat angle; a typical value is -1×10^{-3} F^{-1} (i.e., 0.1% increase in unwedging force for each degree of temperature decrease with the valve in a seated position).

Influence of Elevated System Temperature

If a valve in a system at elevated temperature has been open for a period of time, a portion of the stem exterior to the valve will be at a temperature below that inside the valve. A temperature gradient will exist in the stem. If the valve is subsequently closed, typically requiring less than a minute, the valve will be seated before the stem has any significant change in its temperature profile. Over a period of time, though, the stem will heat up (i.e., the temperature gradient will shift outward away from the valve), and the stem will expand. If the portion of the stem inside of the valve is at T_f (fluid temperature) and the portion of the stem

outside the valve is at T_a (ambient temperature), the net growth of the stem as the gradient shifts is given by:

$$\Delta l_s = \alpha_{st} l_s (T_f - T_a) \tag{11}$$

Where

Δl_s = stem length change
α_{st} = stem thermal expansion coefficient
l_s = stem stroking length

If the stem can grow freely with little or no load increase (such as may occur with an air actuator), then this thermal growth after closure has no effect. If however, the stem is structurally constrained (as occurs in most MOVs), then this thermal growth produces an increase in the stem compression load and an increase in the load pushing the disk into the seat rings.

$$\Delta F_s = \Delta l_s / G_s \tag{12}$$

In Equation (12), G_s is the flexibility of the stem and its structural restraint, discussed previously in conjunction with Equation (10).

Approach for Calculations Using Model

The approach for calculations using the model is as follows:

1. Determine seat contact load and seat "interference" for the closure stroke at maximum developed thrust using Equations (1) and (4). Typically, a stroke with no DP between upstream and downstream during closure will create the highest seat contact load and "interference". It is not apparent whether high system pressure or low system pressure will create the largest unwedging load and both may have to be examined.

2. Determine if there is thrust relaxation after the maximum closure thrust is developed. Many MOVs relax after seating. In the absence of data for a specific valve, zero relaxation typically can be assumed to get a maximum unwedging load.

3. Determine if the valve was closed with the system fluid temperature greater than the ambient temperature. If so, the seating stem thrust increase due to delayed stem expansion should be calculated using Equations (11) and (12). Recommended values for l_s and G_s are the stroke length and 2 x 10^{-6} in/lb. This increase can be reduced by the relaxation determined in Step 2, if any.

4. Determine the values of upstream and downstream piping pressure at the time of valve unwedging. The lowest values of these pressures will yield the maximum unwedging force.

5. Determine the valve temperature change from time of closure to time of opening.

6. Determine the value of bonnet pressure at the time of valve opening. This pressure will be the bonnet pressure at closure adjusted for increases or decreases. Increases can occur due to overall system pressure increases in the time after the valve is closed (which can be "trapped" in the bonnet even if the system pressure subsequently reduces) or due to bonnet heating. Decreases can occur due to leakage to ambient or the adjacent piping. Although the maximum bonnet pressure is typically of concern, in some cases the minimum value may have to be determined. When determining a maximum value, the effect of bonnet heating needs to be determined using Equation (8). If values for air volume, leakage mass and stem withdrawal volume cannot be justified, it is bounding to use zero for these parameters in calculations of maximum bonnet pressure.

7. Determine if there is an increase in stem seating thrust due to stem-body differential expansion and, if so, calculate the value with Equation (10). Use the recommendations in Step 3 for l_s and G_s. An increase can be reduced by the relaxation determined in Step 2, if the relaxation was not already credited in Step 3.

8. Calculate the seat contact load and disk-to-seat "interference" using the maximum seating load as increased by the sum of the values determined in Steps 3 and 7.

9. Calculate the change in disk-to-seat "interference" due to differential expansion of the disk, seat ring and body using Equation (9).

10. Calculate the friction coefficient for unwedging based on the friction coefficient for closing plus $(-3.4 \times 10^{-4})\ \Delta T$, where ΔT is the final temperature minus the initial temperature. Note that temperature decreases (negative ΔT) produce increases in μ, which is the case which produces increases in unwedging thrust.

11. Calculate the change in disk-to-seat "interference" produced by the action of bonnet, upstream and downstream pressure, using Equations (6) and (7).

12. Based on the disk-to-seat "interference" determined in Step 8 and the changes determined in Steps 9 and 11, determine the new disk-to-seat "interference" and contact load.

13. Based on the disk-to-seat contact load and disk-to-seat friction coefficient, determine the unwedging load by free body equilibrium.

CALCULATIONS AND COMPARISON TO DATA

Bonnet Pressure Locking Conditions

From the authors' involvement in a gate valve test program (which did not address pressure locking), eight gate valve tests which inadvertently had bonnet pressure locking conditions were

identified. These eight tests cover:

- six separate valves
- five valve manufacturers
- valve sizes from 3 to 18 inches
- valve pressure classes of 150, 300, 900 and 1500 lbs
- valve bonnet pressures of 50 to 1600 psi

In each test, the bonnet pressure, upstream pressure, downstream pressure and stem thrust were measured, and the valve dimensions were known. Based on this information, an unwedging thrust was calculated using the model. Simple flexibility models (e.g., plates and cylinders) were used in the calculations. Figure 7 shows a comparison of predictions and measurements which indicates that the model is a satisfactory predictor.

Pressure Change Conditions

Because of the body flexibility, a change in unwedging load can occur when the pressure changes even if the bonnet is not pressurized relative to the pipe. The authors did not have data for this situation, but example calculations were performed for two flexible wedge valves to quantify the predicted effect. One of the valves had a disk with a low stiffness and the other had a very stiff disk. In the calculations, each valve was closed at its maximum pressure and then opened at zero pressure. For reference, calculations with both closing and opening at zero pressure were also performed. For the valve with the low stiffness disk, pressure had a negligible effect on the unwedging load. For the valve with the high stiffness disk, the unwedging load increased by 25% when the pressure was reduced with the valve closed, compared to the constant pressure case. Accordingly, for some valves and conditions, the effect of a pressure reduction (without bonnet pressurization) is predicted to cause a significant increase in unwedging thrust.

Thermally-Induced Bonnet Pressure Increase

Limited data on bonnet pressure increase due to heating were identified in Missimer (1984). Calculations were performed to predict the pressure increases, and are compared to the measured values in Table 1. For reference, the predicted pressure increases for a rigid body (no elastic expansion) are also shown on the table. Simple cylinder flexibility models of the valve body were used because detailed body drawings were unavailable. Also, based on the way that the tests were conducted, there was no trapped air, no body leakage and no stem withdrawal volume. These parameters were set equal to zero in the calculations. The predictions for rigid valve body bound all of the data. The body expansion effect reduces the predicted pressure rise and better agreement with the data is obtained.

Thermal Binding

From the gate valve test program mentioned earlier, six tests in which a valve was cooled while in the closed position were identified. Minor bonnet pressure locking also occurred in theses tests. The six tests covered:

- five separate valves
- four valve manufacturers
- valve sizes from 2½ to 6 inches
- carbon and stainless steel valves
- temperature decreases of 60° to 240°F

The six valves all had negligible stem-to-body and disk-to-body differential expansion. Accordingly, the thermal binding mechanism was friction coefficient change. The model described in this report was used to predict the unwedging thrust for the thermal binding tests. The predictions are compared to data in Figure 8. As seen, the model appears to be a satisfactory, yet bounding, predictor.

CONCLUSIONS

1. The major mechanisms by which pressure and temperature can influence gate valve unwedging load have been identified (see Figure 1).

2. Models for predicting the influence of pressure and temperature on unwedging load have been developed, for each of the fundamental mechanisms.

3. Increases in unwedging thrust due to pressure can occur when:

    - the bonnet is pressurized relative to the piping
    - the overall pressure is decreased

4. Increases in unwedging thrust due to uniform temperature changes can occur when one or more of the following occurs:

    - Heating [cooling] of a valve where the stem expansivity exceeds [is less than] that of the body.
    - Heating [cooling] of a valve where the disk expansivity exceeds [is less than] that of the body.
    - Heating [cooling] a carbon steel [stainless steel] solid wedge gate valve, due to overlay-body differential expansion.
    - Cooling a valve with Stellite 6 disk-to-seat interfaces, due to the friction coefficient increase.

5. An increase in unwedging thrust can occur due to stem expansion after closure if the fluid system temperature exceeds ambient.

6. Based on limited data, the models appear to be satisfactory, bounding predictors of unwedging thrust. However, additional data should be obtained, particularly covering:

   - uniform pressure decreases
   - solid wedge gate valves
   - valves with different stem and body thermal expansivities
   - valves in elevated temperature systems

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Table 1

Comparison of Bonnet Pressure Rise Measurements and Predictions

Valve	Initial Temperature (°F)	Measured dP/dT (psi/°F)	Measured Final Pressure (psia)	Predicted Final Pressure (psia)	
				Elastic Body	Rigid Body
2-inch	67	32.1	515	503	551
	104	52.5	445	468	509
	70	20.0	445	633	688
	94	21.0	475	817	883
4-inch	70	17.0	385	552	877
	90	18.6	510	853	1293
	80	22.2	340	399	552
	70	11.4	390	738	1143
	95	19.7	410	680	1037
6-inch	68	20.0	215	208	263
	80	14.6	265	443	659
	79	25.0	265	276	377

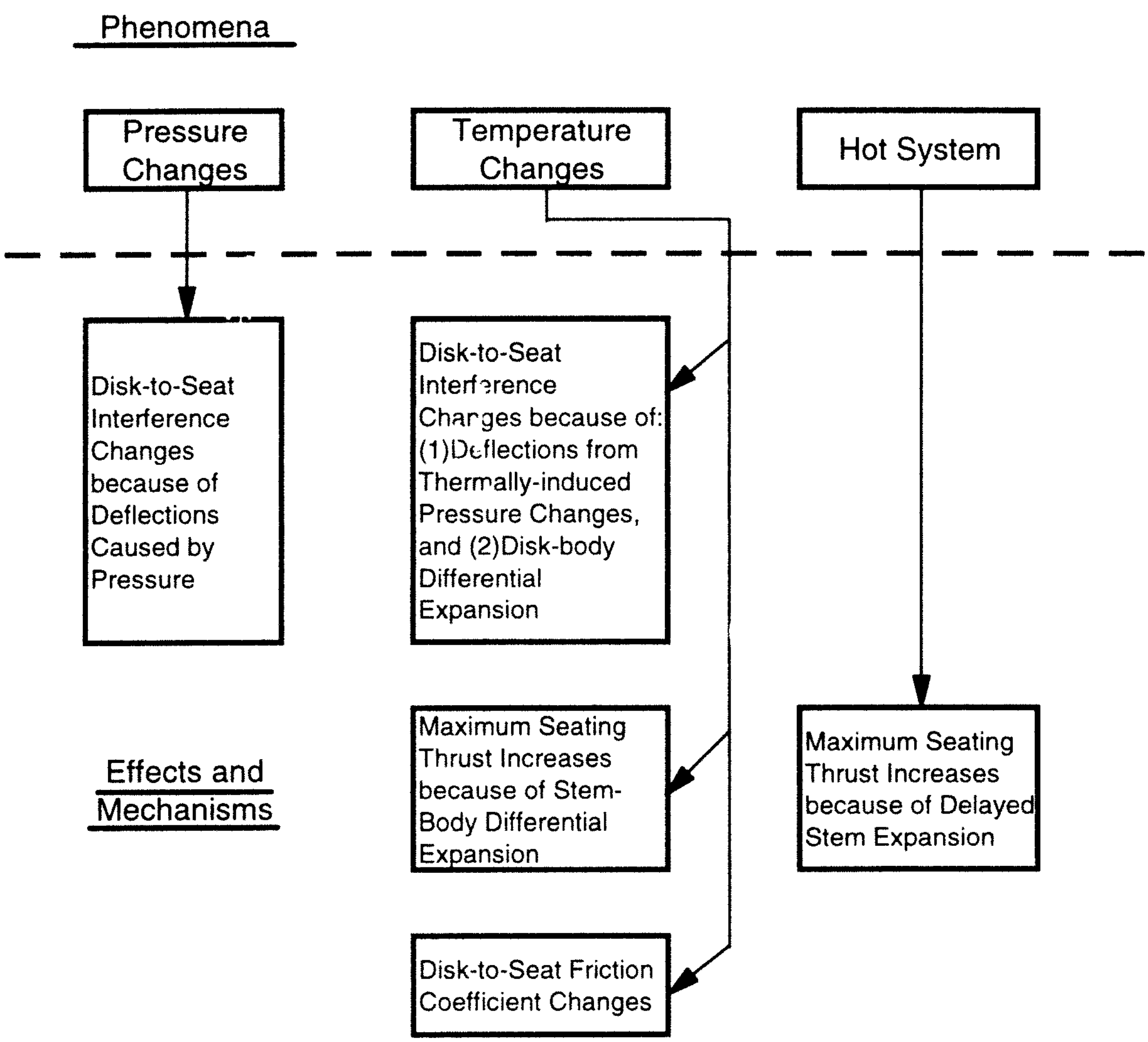

Figure 1. Phenomena and Mechanisms by which Pressure and Temperature Influence Unwedging Load

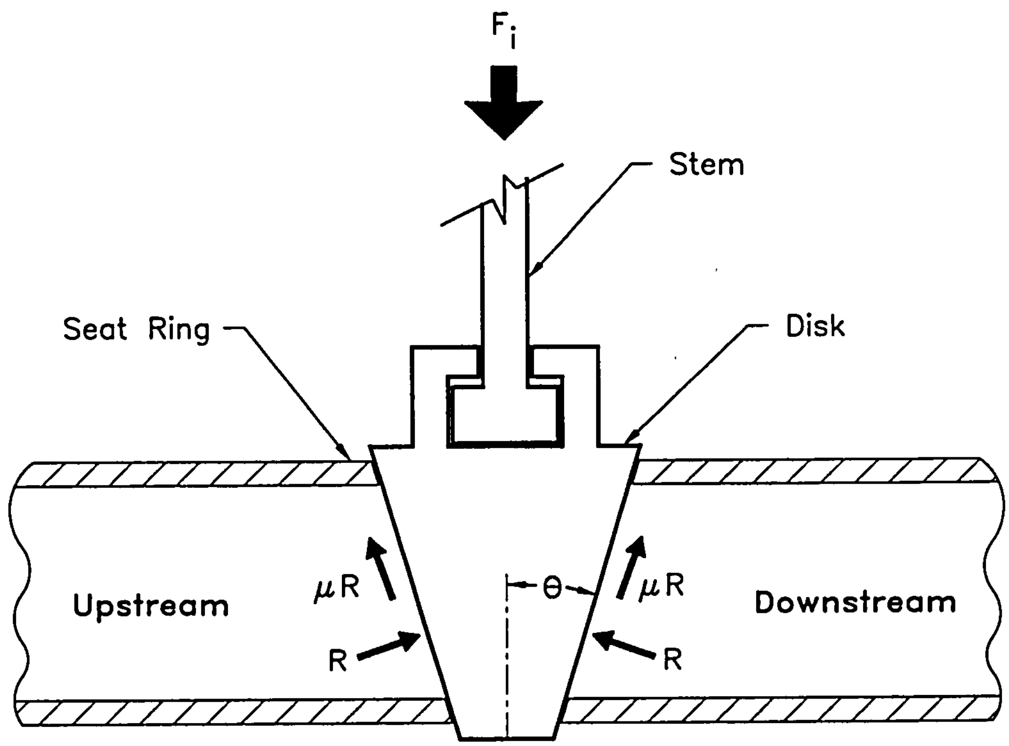

Figure 2. Forces on Gate Valve Disk During Insertion

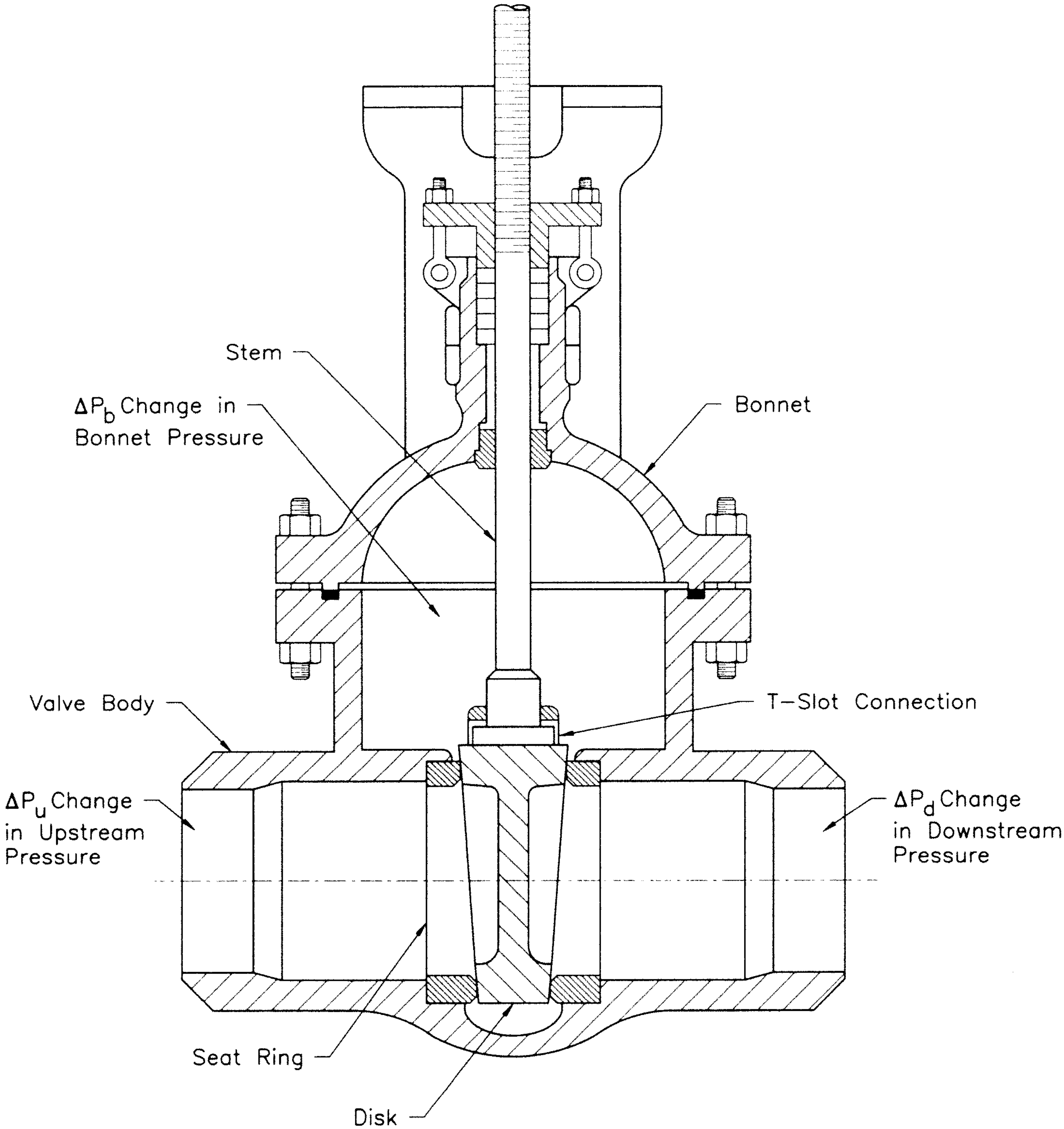

Figure 3. Pressure Changes with Valve Closed

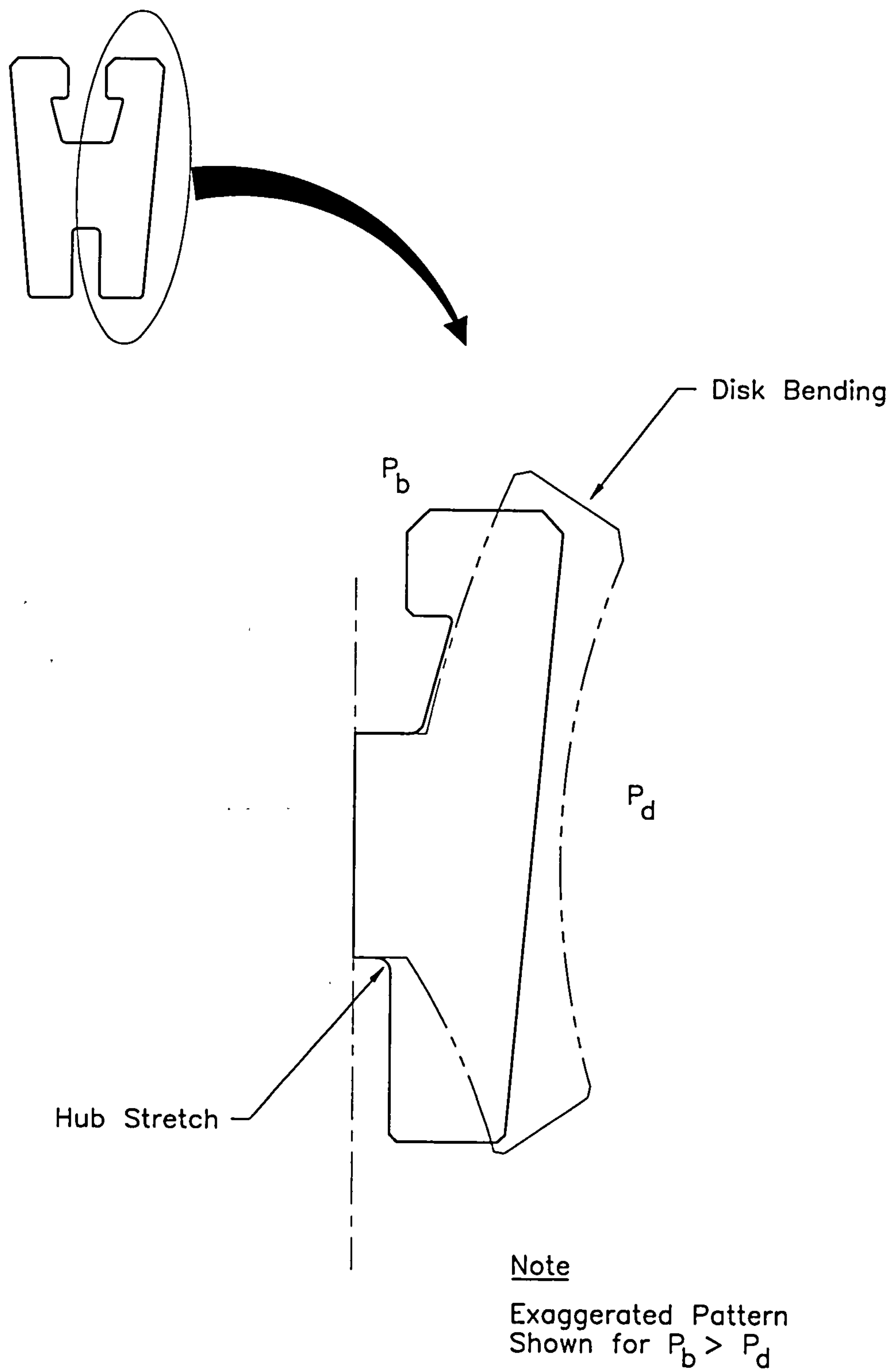

Figure 4. Deflection Pattern of Flexible Wedge Disk (Without Seat Reaction Loads) Due to Pressure Loads

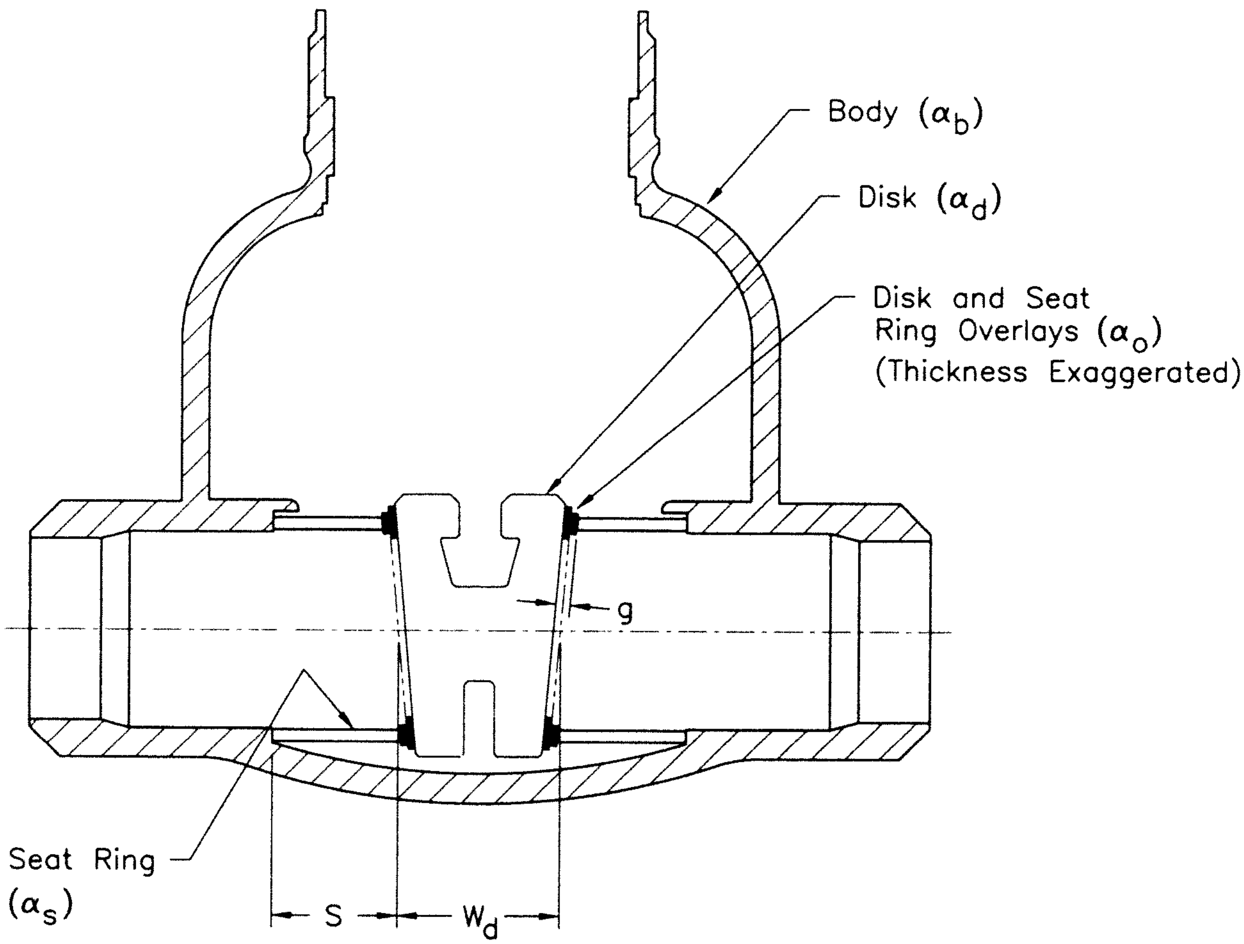

Figure 5. Components Involved in Disk-to-Body Differential Thermal Expansion

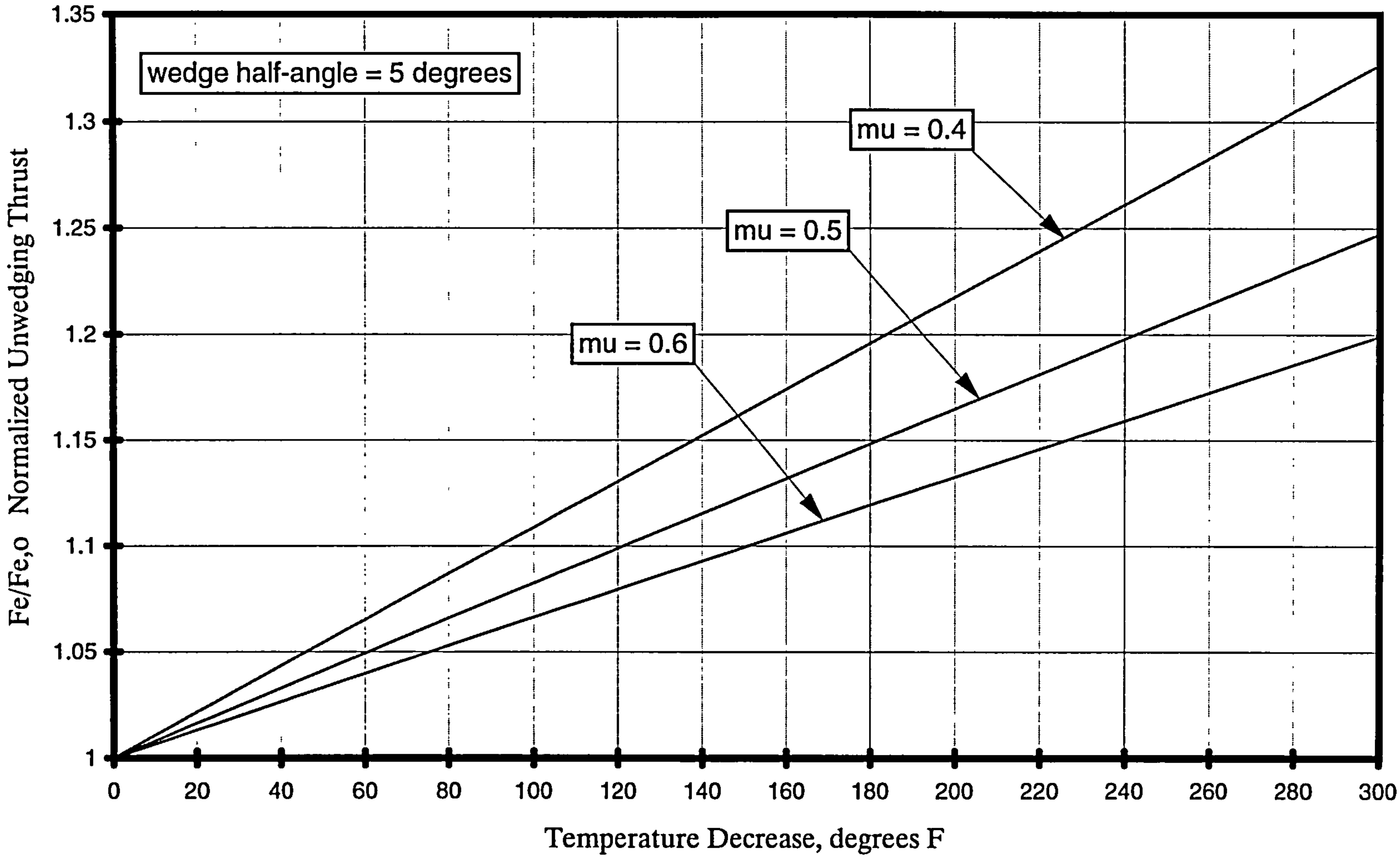

Figure 6. Effect of Temperature Change on Unwedging Load due to Friction Coefficient Change

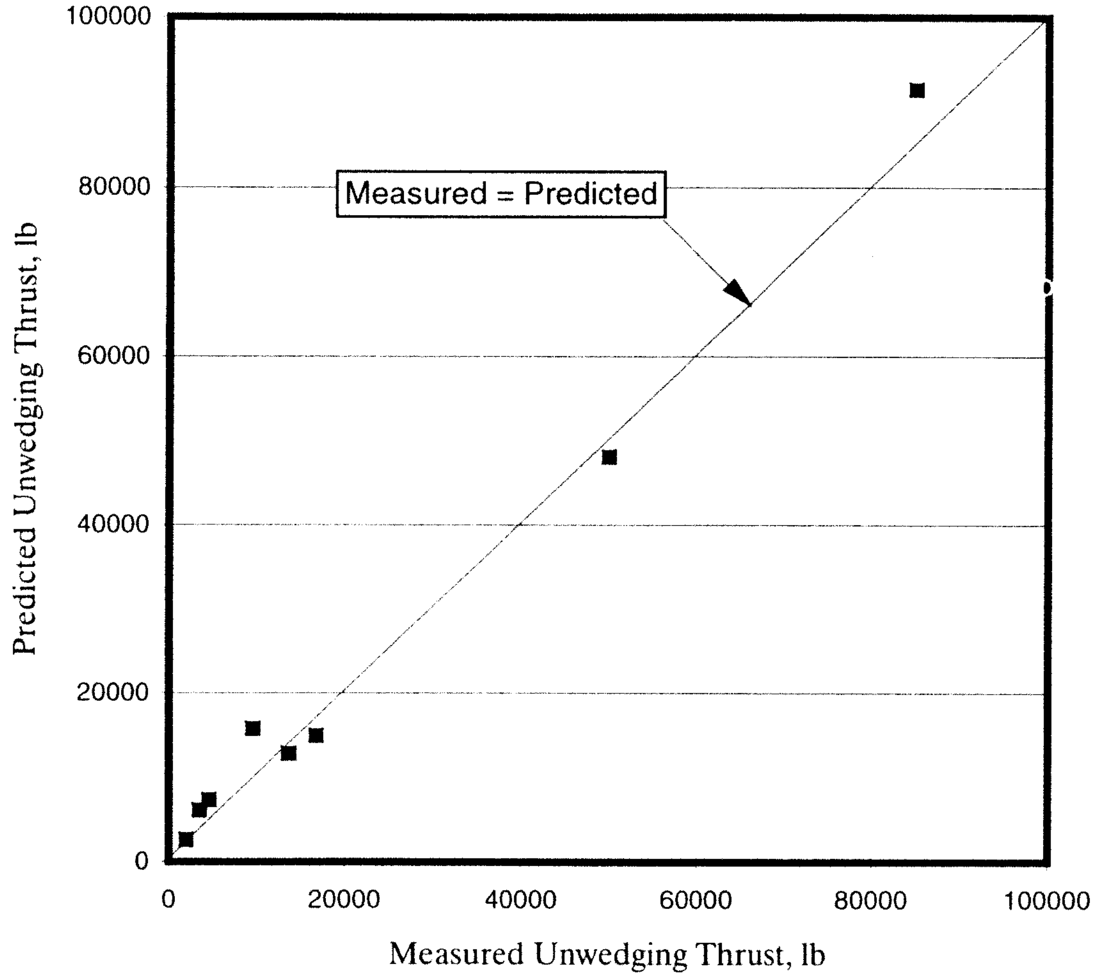

Figure 7. Predicted vs. Measured Unwedging Thrust for Pressure Locking Tests

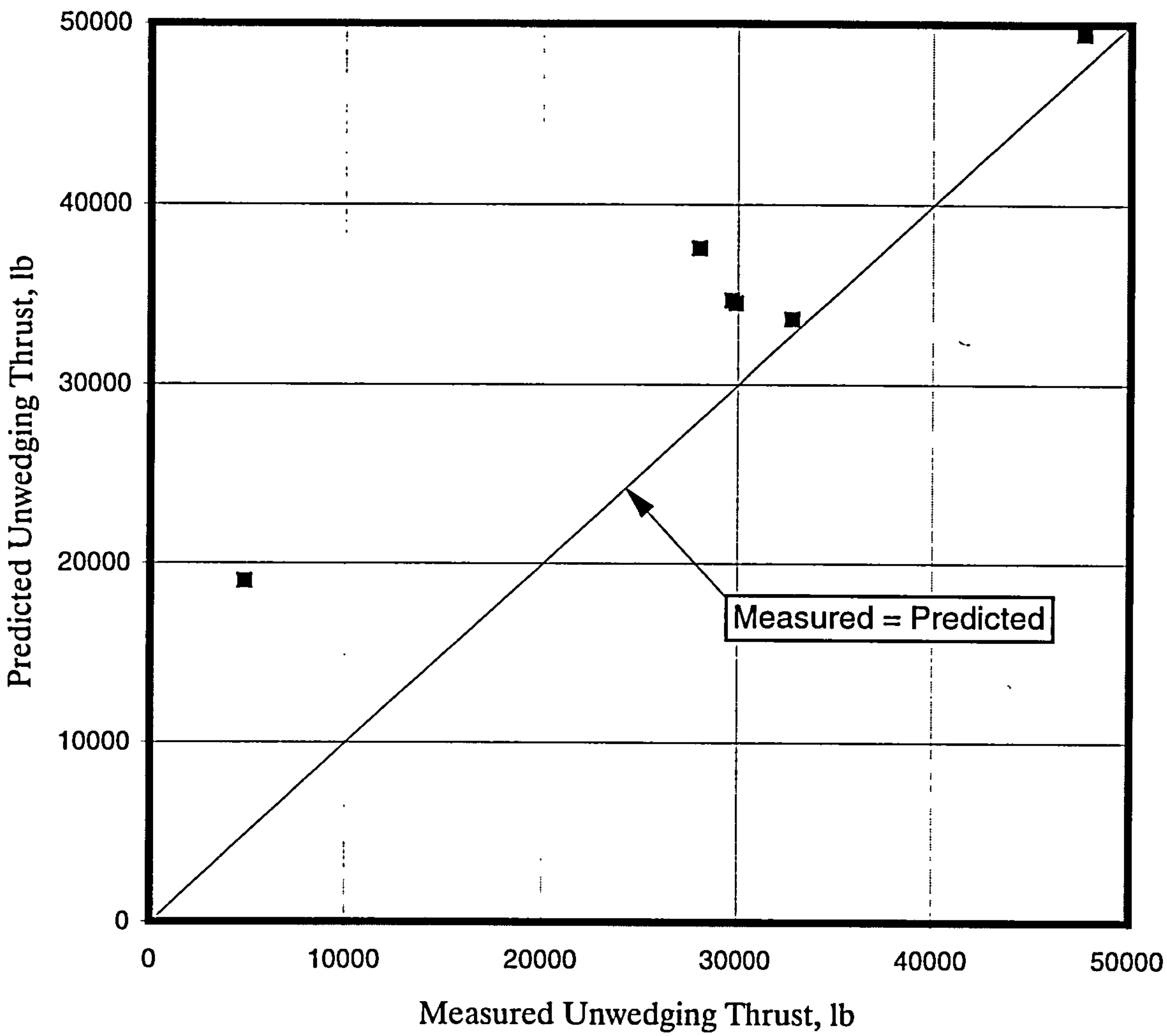

Figure 8. Predicted vs. Measured Unwedging Thrust for Thermal Binding Tests

HARDFACING MATERIALS USED IN VALVES FOR SEATING AND WEAR SURFACES

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Anchor/Darling Valve Co.

ABSTRACT

Most valves and essentially all critical service valves utilize hardfacing materials for seating and wear surfaces to minimize wear and galling. The type of hardfacing materials used, the methods of deposition, and the quality of the final product all contribute to the wear characteristics, required operating force, and life of the final product. Over the last forty years the most prevalent hardfacing materials furnished to the commercial nuclear industry consisted of cobalt base and nickel base materials. In the last several years there has been extensive development and evaluation work performed on iron base hardfacing materials. This presentation will address the wear characteristics of the various materials and the importance of consistent quality of deposited materials necessary to achieve optimum product performance and longevity.

INTRODUCTION

For many years, and prior to the construction of commercial nuclear power plants, the valve industry has been using hardened materials and weld deposited hardfacing materials for wear applications.
By the time commercial nuclear power plants were being planned there were several valve companies manufacturing equipment using deposited hardfacing materials for seating surfaces. The primary hardfacing materials at the time were the various grades of CoCRW alloys. During the 1950's the nuclear industry was less concerned with the long term potential for radiation exposure of maintenance personnel and most valve specifications for nuclear service <u>required</u> valve seats to be hardfaced with Stellite material. During the sixties, seventies and eighties there were <u>many</u> <u>tons</u> of Cobalt based material furnished in valves that were placed in commercial nuclear power plants. In the mid sixties there was a significant amount of development work and testing performed on the nickel based hardfacing materials for the Naval Reactor Program. Since that time there have been many utilities that have replaced cobalt based wear surfaces with nickel based wear surfaces. The nickel based materials exhibit less desirable wear properties than the cobalt based materials and also are more difficult to deposit.

In the late eighties the commercial nuclear power industry became more concerned with the problems associated with the cobalt based materials, the most significant being the radiation exposure to maintenance personnel, and through EPRI became very proactive to pursue suitable replacement materials. As a result of the research and development work initiated by EPRI, the iron based material referred to as NOREM is now being seriously considered as an appropriate replacement for the cobalt based materials. This presentation will describe the efforts and progress that Anchor/Darling Valve Co. has made over the last several years with regard to replacing cobalt based materials in our products.

DESCRIPTION OF TESTS

Many types of wear and galling tests have been conducted on materials by various laboratories and test organizations and after evaluating several test methods Anchor/Darling Valve Co. selected to use the materials laboratory at Rensselaer Polytechnic Institute (RPI) to evaluate the materials of interest. This provided us with an independent evaluation of material couples and a consistent method that we feel is representative of the sliding wear experienced in the operation of gate valves. This testing method, shown in Figure 1, utilizes an

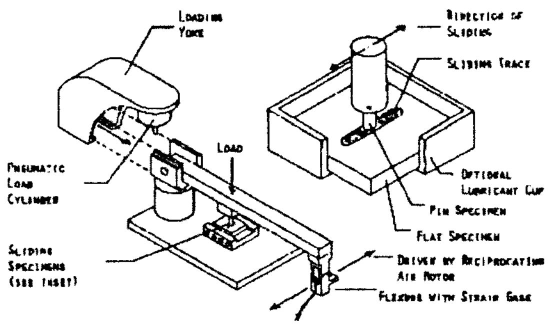

FIG. 1 Schematic of Reciprocating Pin-on-Flat Test Rig

oscillating motion to slide the pin across the plate. The pin is loaded through the oscillating arm to achieve the desired stress between the pin and plate and the friction force is determined by means of a strain gage transducer mounted on the arm. The plate, on which the pin slides, is held in a cup which contains any desired lubricant. For room temperature tests the lubricant is normally common tap water. For elevated temperatures the upper and lower specimen holders are heated and super heated, dry steam impinges directly in front of the pin sliding on the plate.

For all test couples the pins and plates are lapped to produce a surface finish of approximately one to two micro inch finish as measured with a Profilameter or Optical Profiler. The average surface roughness is measured before and after all tests.

Also before and after each test the pins and plates were weighed and weight change was recorded for information.

The friction coefficient for each couple was determined at the start of each test, after 100 cycles and after 1000 cycles.

For each set of data reported there were three sets of wear data collected and averaged.

PRODUCTION OF TEST MATERIAL

It is essential that the production of test material is representative of what can be produced for actual component parts. Especially when using welding techniques to deposit various materials the test samples should be produced using the welding material form and welding process that will be used in the production of component parts. For instance it may not be difficult to produce wear test samples with undiluted weld metal, however it may be very impractical to produce component parts with undiluted weld metal. Experience and test data indicates that dilution in the deposited weld material at the thickness that the material will be used will significantly affect the wear properties. Ideally, the thickness of the hardfacing deposit tested should be equivalent to the minimum thickness qualified in the production procedure. The quality of the welding material used for wear test specimens must also be known and controlled in order to obtain consistent results in production. And finally the personnel performing the welding processes must be consistently qualified using procedures that will ultimately be used in the production of parts.

WEAR TEST RESULTS

The results of laboratory wear tests that have been performed on the materials of interest are shown in Table 1. The cobalt based materials are listed as grades of Stellite, the nickel based material as Deloro 50 and the iron based material most recently developed as NOREM. This data shows that the wear characteristics of iron based NOREM

material compares favorably with the cobalt based Stellite 6 and 156 material and that the cobalt based Stellite 21 and nickel based Deloro 50 have less desirable wear characteristics in sliding wear.

DISCUSSION

The results of the laboratory test data have essentially been confirmed from years of experience reported from users. It must be recognized that the greatest amount of reported experience pertains to the cobalt based materials. These materials have been used predominantly in U.S. plants as well as off shore plants for nearly forty years. Over the last twenty years there has been a significant amount of nickel based materials furnished for valves as well as replacement parts, however there is little user feed back pertaining to these materials. For the iron base materials we have just begun. The first valves that Anchor/Darling Valve Co. furnished with iron based material were manufactured in 1993. Since that time we have furnished many valves and parts with iron based materials for seats and wear surfaces, however there is essentially no user feed back regarding performance at this time.

One of the more interesting and challenging opportunities that we have experienced to date was the replacement of cobalt based material with iron based material in the 24" Y-type feed water check valves in-situ. It was very challenging to rework valves with iron based materials with the limited accessibility and machining capability as well as time restraints, however the completion of this rework demonstrated that such rework could be accomplished.

We believe that the wear characteristics for all of the weld deposited materials is dependent on the consistent quality and the control of base metal dilution at the finished surface of the deposited material. The base material dilution in cobalt based and nickel based hardfacing materials can be readily evaluated on the final wear surface and we have confidence that better wear characteristics are achieved when iron dilution is below five percent. Dilution should be measured on the surface of the final machined part as it will be furnished for installation.

We have found that controlling the iron in the welding material to be deposited and simply relying on the qualified procedure is inadequate to assure consistent hardfacing quality. Presently we are utilizing a metal analyser to determine iron dilution on finished seating surfaces. For the iron based materials the determination of base material dilution is not as simple and we continue to evaluate the effects of base metal dilution on the final wear surface. It appears obvious that more consistency can be obtained in deposition of hardfacing materials in a controlled plant environment than can be achieved in the field, however the success of field changes will most likely improve as more work is performed and more experience gained.

CONCLUSIONS

The cobalt based and nickel based hardfacing materials have been used in the production of valves for many years and there are many years of service experience associated with these materials. The iron based hardfacing material is presently being used in the production of valves and there is little service experience associated with this material.

The test data acquired to date for the iron based material is encouraging and appears to be a suitable substitute for the replacement of cobalt based materials in our nuclear plants. Hardfacing procedures and personnel have been qualified for various welding processes using the iron based material and applied to carbon steel and stainless steel component parts. We have experienced a higher degree of difficulty in application of the iron base material and nickel base material versus the cobalt base material. We would expect this degree of difficulty to diminish as more experience is gained.

We have confidence that, with due diligence, we

can also replace hardfacing material "in situ" on larger size valves. For repairs in place, it is essential that personnel training and extensive full scope mock-up training be given careful consideration.

TABLE 1 - Wear Test Results

Cobalt Base, Nickel Base and Iron Base Materials

TEST LOAD (PSI)	TEST MATERIAL (SELF MATED)	TEST TEMP (°F)	AVERAGE SURFACE ROUGHNESS (x 10[illegible] IN.) BEFORE TEST	AFTER TEST	AVERAGE WEIGHT CHANGE (x 10[illegible] GRAMS) PIN	PLATE	AVERAGE FRICTION COEFFICIENT START	AFTER 100 CYCLES	AFTER 1000 CYCLES
15,000	STELLITE 6	RT[1]	0.50	4.00	(2.30)	(16.30)	0.24	0.26	0.26
15,000		600	2.10	31.80	(14.70)	(5.00)	0.24	0.33	0.37
15,000	AMAX B4[2]	RT	1.27	5.60	(1.70)	(4.00)	0.14	0.26	0.25
15,000		600	1.51	29.50	(2.00)	(13.30)	0.26	0.33	0.35
15,000	STELLITE 21	RT	0.83	4.80	(3.67)	(12.00)	0.24	0.34	0.38
15,000		600	0.87	136.80	(4.33)	(23.00)	0.33	0.42	0.42
15,000	STELLITE 156	RT	1.43	6.00	(3.00)	(5.00)	0.17	0.26	0.31
15,000		600	1.42	124.43	(5.33)	(5.67)	0.30	0.27	0.26
15,000	DELORO 50	RT	0.84	5.00	(3.00)	(15.70)	0.26	0.29	0.30
15,000		600	0.80	17.30	(6.00)	(15.67)	0.44	0.35	0.35
15,000	NOREM 02	RT	0.90	12.00	(7.60)	(5.00)	0.17	0.25	0.27
15,000		600	0.94	91.30	(14.00)	(3.30)	0.32	0.34	0.36
15,000	NOREM B4	RT	0.61	4.30	(2.70)	(16.30)	0.14	0.27	0.27
15,000		600	0.79	31.80	(2.30)	(13.70)	0.28	0.34	0.36

TABLE 1

1. RT = Room Temperature
2. Early developmental version of NOREM

Detecting MOV Stem to Stem-Nut Lubricant Degradation by Reviewing MOV "Seating Time"

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Northeast Utilities

OBJECTIVE

The objective is to describe a methodology, suitable for IST Programs, that obtains sufficient data to assess and trend the overall condition of AC and DC MOVs. Using existing diagnostic test equipment, recording the time from Hard Seat Contact to the Torque Switch Trip point (TST), "Seating Time (msec)", can provide a means to quickly detect for signs of lubricant degradation in the interface between the stem and stem-nut for MOVs. Obtaining and trending the "Seating Time" parameter can be easily accomplished using the sensors which provide motor current, torque or thrust. From the initial baseline test to subsequent tests, it is the change in time between Hard Seat Contact to Torque Switch Trip point that is utilized to monitor changes in the coefficient of friction between the stem and stem-nut.

While many Licensees have begun their MOV Periodic Verification Programs, obtaining the "Seating Time" during Refueling Outages and comparing the results from subsequent outages can be an input to the justification for changing the MOV baseline test frequency.

INTRODUCTION

Numerous plants have completed their NRC GL89-10 MOV program. Actuators and valves have been overhauled, modified and, in many cases, replaced with larger or more reliable components. Since many Licensees have begun their MOV Periodic Verification Program, initial assumptions regarding the frequency of periodic baseline testing for the valves have been made. Testing of these MOVs more frequently than the baseline test frequencies is being proposed by many licensees, as a means to verify the Licensee's actuator preventative maintenance programs and stem to stem-nut lubrication frequencies.

IST Coordinators are now faced with implementation of this MOV Periodic Verification Program and the IST program, which requires licensees to periodically stroke time test the actuators to meet ASME code requirements and to detect for any actuator degradation. Meshing of these two programs is the goal.

Inservice Test Coordinators (IST) Coordinators, the NRC, and Industry groups have long known that stroke time testing of AOVs and MOVs provides little means to detect actuator and valve degradation.[1,2] For AOVs and DC MOVs, a noticeable increase in the actuator's stroke time is indicated when a change in the actuator's output has occurred. However, this does not hold true for AC MOVs since the stroke time test simply indicates that the valve functions and position indication lights imply valve

movement. The testing does not provide a means to detect actuator or stem to stem-nut lubrication degradation. This paper will explore the usefulness and flexibility of obtaining "Seating Time" data obtained at the valve or MCC as a useful parameter in detecting stem to stem-nut lubricant degradation.

BACKGROUND

Stroke time testing for DC and AC MOVs provides minimal useful information to monitor and detect degradation in the lubrication characteristics of the stem and stem-nut interface. As periodic verification programs develop, the frequency of baseline testing will vary from once every Refueling Outage (RFO) to once every six RFOs. If a more frequent testing period is required, a simpler testing technique, at the MCC or at the valve, is necessary to monitor actuator performance and confirm or adjust the baseline test period.

Licensees have performed extensive diagnostic testing of MOVs during the past few years as the means to meet the requirements of GL 89-10. Part of these diagnostic tests included the acquisition of stem torque and thrust during the open and close stroke as a means to verify the actuator's capability to deliver the required thrust. The results of this testing demonstrates, with design margin, that these valves will perform their design basis safety function. To ensure these actuators will consistently produce the necessary thrust output repeatedly over time, many plants have replaced actuator spring packs with slotted spring packs, re-oriented actuators, installed cartridge cap grease reliefs, and installed limiter plates. For most MOVs, especially those in high temperature and harsh environments, degradation of the lubricant at the stem to stem-nut interface is now the major credible short-term problem for ensuring consistent thrust output.

Most MOV test equipment used in the industry have a minimum data sample rate of 1000 samples/sec, which provides sufficient resolution to detect the disk pullout point, hard seat contact point, torque switch trip point, etc., by measuring and recording the motor current, stem torque and/or thrust sensor outputs. Typically, diagnostic test equipment requires calibrated sensors to measure the appropriate thrust and torque parameters to assess the condition of the MOV and to determine the coefficient of friction between the stem and stem-nut interface. Periodic testing of actuators is complicated by recalibration or re-verification of sensor accuracy, requiring, in some cases, removal of the actuator for installation of calibrated test equipment, or replacement and recalibration of sensors.

Diagnostic test systems have one thing in common: a high data sample rate that can record opening and closing events fairly easily and accurately. This is possible since all of the sensors use time as a parameter.

"SEATING TIME"

When diagonstic testing with these systems is performed, closing and opening stroke events are referenced, in milliseconds, from the beginning of the particular stroke. In reviewing these events, a relationship during the closing stroke is evident between the rate of thrust developed, the general condition of the MOV (lubricated, degraded lubrication, etc.) and the time it takes the valve to travel from hard seat contact to the torque switch trip point. Some examples of this are as follows:

At the Northeast Utilities L. F. Sillan Jr. Nuclear Training Center, two gate valve MOVs were instrumented and diagnostically tested to measure torque and thrust at the stem and to determine the relationship of torque and thrust as the coefficient of friction in the stem to stem-nut area is increased. First, the valve stems were lubricated with clean lubricant (Mobil 28) and statically tested. Then, a series of tests were conducted while the lubricant was gradually contaminated with an abrasive to increase the coefficient of friction between the stem and stem nut. The resulting diagnostic test traces, Figures 1 and 2, show several things:

- The rate of thrust buildup (change in thrust/change in time) is constant after the hard seat contact point, between the traces with clean grease and contaminated grease.

- As the stem to stem-nut lubricant is degraded, the rate of torque buildup (change in torque/change in time) increases up to the torque switch trip point.

- The total torque transmitted to the stem by the actuator is greater when the grease is contaminated (high coefficient of friction) than when the grease is clean (lower coefficient of friction). Clearly, the conversion from torque to thrust is not as efficient for the stem with contaminated grease as compared with the clean stem. Since the conversion of stem torque to thrust is degraded, the energy that is not converted is absorbed by the stem and actuator.

Identifying closing events on these thrust and torque traces, Figures 3 & 4, one can also conclude:

- With a constant torque switch trip setting, the "Seating Time" is shorter with the contaminated grease verses the clean grease. One concludes there exists a linear relationship between the "Seating Time" and the subsequent thrust output (and change in coefficient of friction).

- The "Seating Time" between tests can be obtained from the current, torque or thrust trace.

- The "Seating Time" can be obtained from an uncalibrated torque, thrust or motor current sensor, provided the hard seat contact and torque switch trip points are discernible on the traces.

The same conclusions can be drawn when applying these observations to actual test data obtained in the plant during the Millstone Unit 1 RFO15. The stem and stem-nuts for two MOVs (one AC, the other DC) were reworked to improve the stem to stem-nut coefficient of friction. Torque and thrust data were obtained before and after the stem to stem-nut rework. (Refer to Figures 5 & 6 for DC MOV 1-SD-2A). Although the actuator torque generated before and after the rework is constant, an increase in thrust is clearly evident. Corresponding to the change in thrust output is an improvement in the coefficient in friction, resulting in increased seating time.

It can be concluded that data obtained from measuring and trending the "Seating Time" will provide several clues to the health of the MOV:

- Detect degradation of the stem to stem-nut lubricant or torque-to-thrust conversion.

- Detect thrust output changes or changes in the coefficient of friction, assuming the developed torque at torque switch trip point does not vary from test to test.

There are some pitfalls:

- The technique cannot be used to evaluate changes in packing drag (other techniques would need to be employed).

- The technique is unsuitable if subsequent changes to the valve or actuator would invalidate comparison to the original baseline torque or thrust test.

- If the torque switch trip point unexpectedly changes, degradation of stem to stem-nut lubrication could be masked if the torque, delivered at the torque switch trip point, is increased. However, since most actuators have been refurbished and tested to maintain optimum and repeatable torque capability, it can be concluded that the probability of an actuator's torque output to unexpectedly change at torque switch trip would be less than for changes in the stem to stem-nutÕs coefficient of friction.

Obtaining the "Seating Time" is fairly straightforward, and prediction of a worst case seating time can be estimated, as shown in Figure 7. The traces do not need to be "Zeroed", nor do they need to be obtained from calibrated thrust, torque or motor power sensors. However, the technique does require instrumentation with sufficient acquisition speed and resolution to permit the user to consistently observe the change in signal magnitude associated with hard seat contact and with torque switch trip.

Because the thrust buildup from hard seat contact to torque switch trip is constant with changing coefficient of friction in the stem stem-nut interface, we can predict a worst case seating time based upon the highest tolerable coefficient of friction for the MOV. The benefit of predicting the worst case, prior to as-found testing, permits assessment of the stem to stem-nut condition from a quick review of data obtained at the MCC or at the valve.

IST Coordinators may wish to develop acceptance criteria, as shown in Figure 8, that can be quickly used to assess and determine any action necessary, based on the as-found condition of an AC or DC MOV during testing. If the seating time falls outside an acceptable range during a periodic MOV valve test at the MCC or the valve, the valve would be placed in the "Required Action Range" and repaired, or the condition analyzed to confirm the actuator can perform its safety function. This action is similar to the IST Program rules in ASME ANSI OM-6 and OM-10. The frequency of baseline or interim testing could then be adjusted accordingly. The remaining ANSI OM-6 and 10 rules would also apply; i.e., prior to returning the valve to service, a test demonstrating satisfactory operation shall be performed. Similarly, "If it is necessary to establish an additional set of reference values, an inservice test shall be run at the conditions of an existing set of reference values and the results analyzed. If acceptable, a second test run at the new

reference conditions shall follow as soon as practical."[3]

CONCLUSION

From an IST prospective, the technique could be a much better parameter to trend and meet the real intent of the ASME XI Code, rather than the simple stroke time testing methodology currently utilized. This concept reduces some of the complicated data presently obtained for MOV periodic verification program to a form which IST programs can trend. Obtaining and trending the seating time can be a meaningful parameter to judge the overall health of a MOV. Data can be obtained using a thrust, torque, or current sensors even if the sensors are not calibrated.

This presentation concludes that recording and trending seating time, obtained during MOV static testing, is a viable means to assess and trend the condition of MOVs. The data can be used to verify satisfactory performance of the MOV and provide a means to detect degradation in stem-nut lubrication. Using strip chart recorders and similar non-intrusive diagnostic tools, one can justify and adjust, as necessary, a suitable preventative maintenance frequency on a valve-by-valve basis.

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2) W. Fiock, to BWROG VTRG memo, OG93-1076-112, "Final Report: Correlation between Utility Inservice Testing Programs and GL 89-10", December 6, 1993.

3) ASME OM Code-1990, Subsection ISTB 4.5, page 10.

Thrust Trace, MOV1, Open to Close

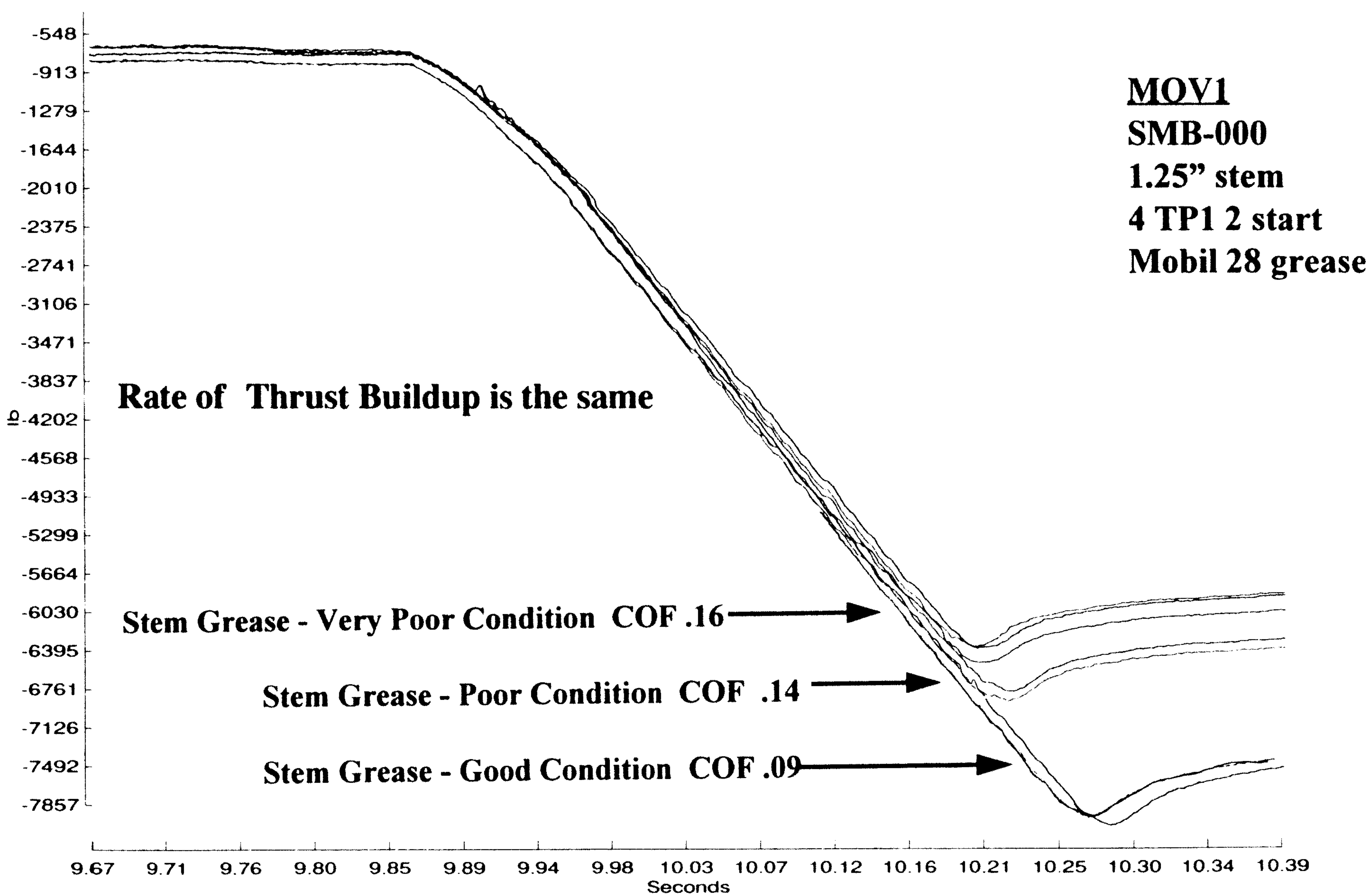


Figure 1

Torque Trace, MOV1, Open to Close

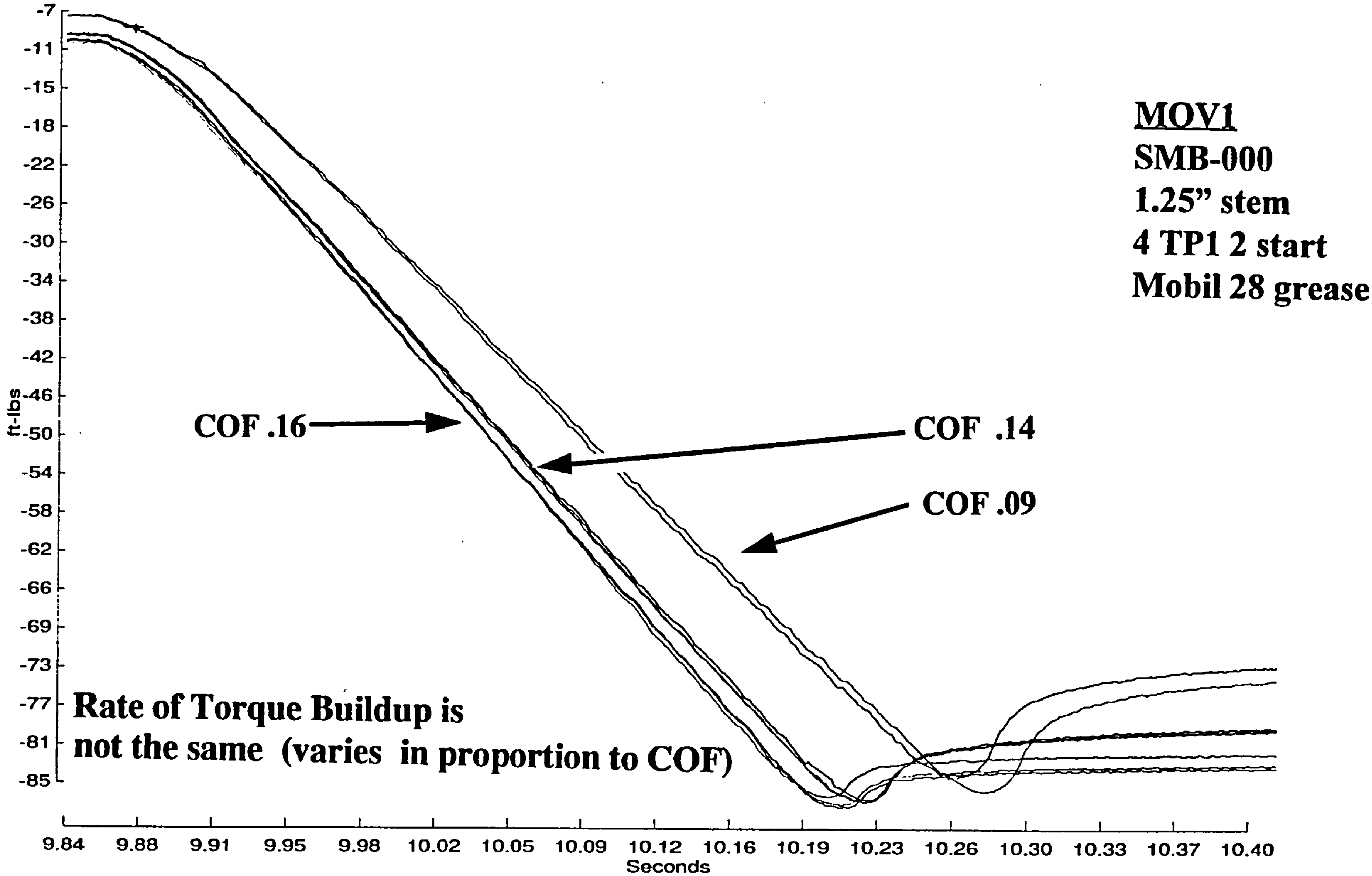


Figure 2

NUREG/CP-0152

3C-84

MOV1

Thrust & Motor Power Trace, MOV1, Open to Close

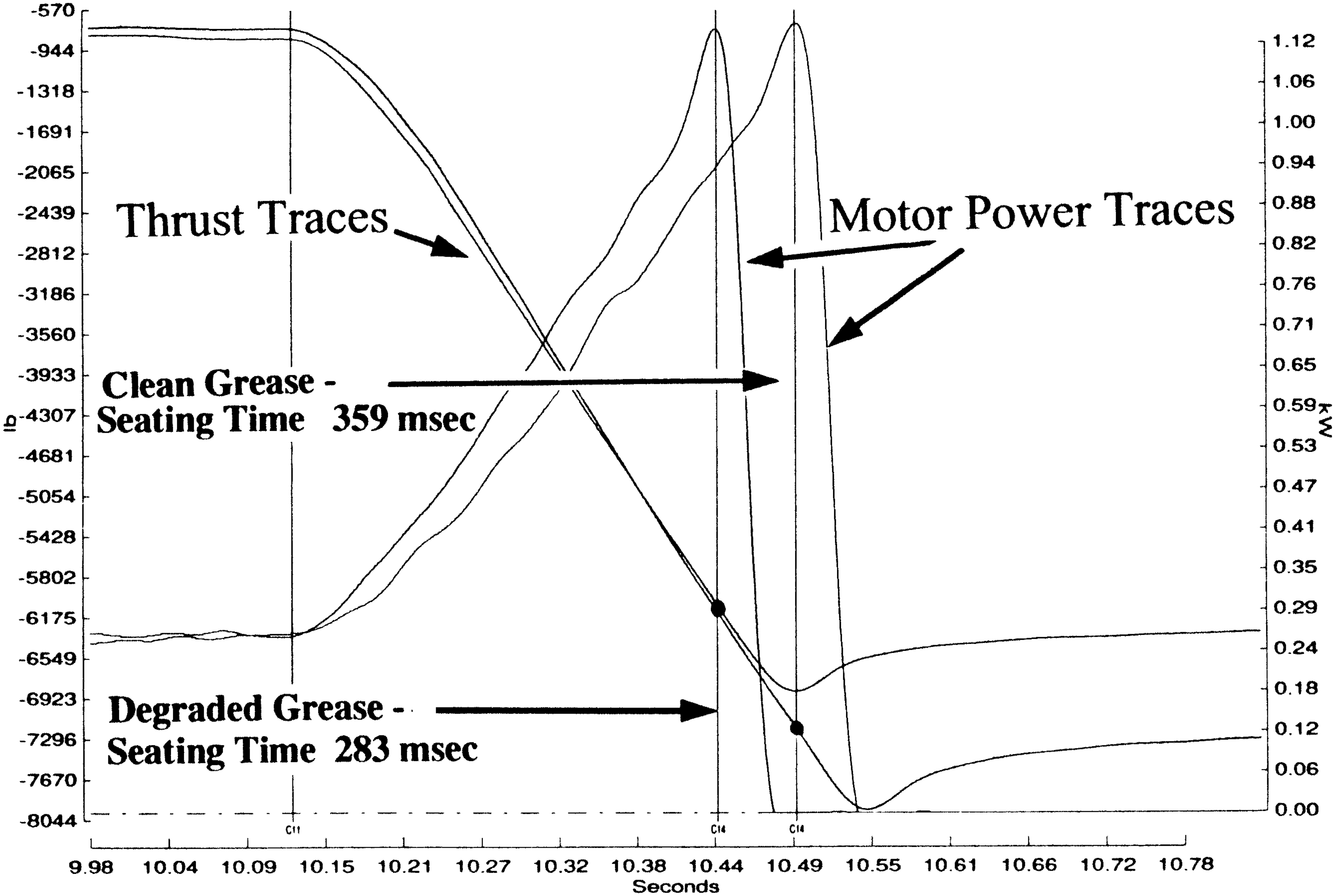


Figure 3

MOV1

Torque & Motor Power Trace, MOV1, Open to Close

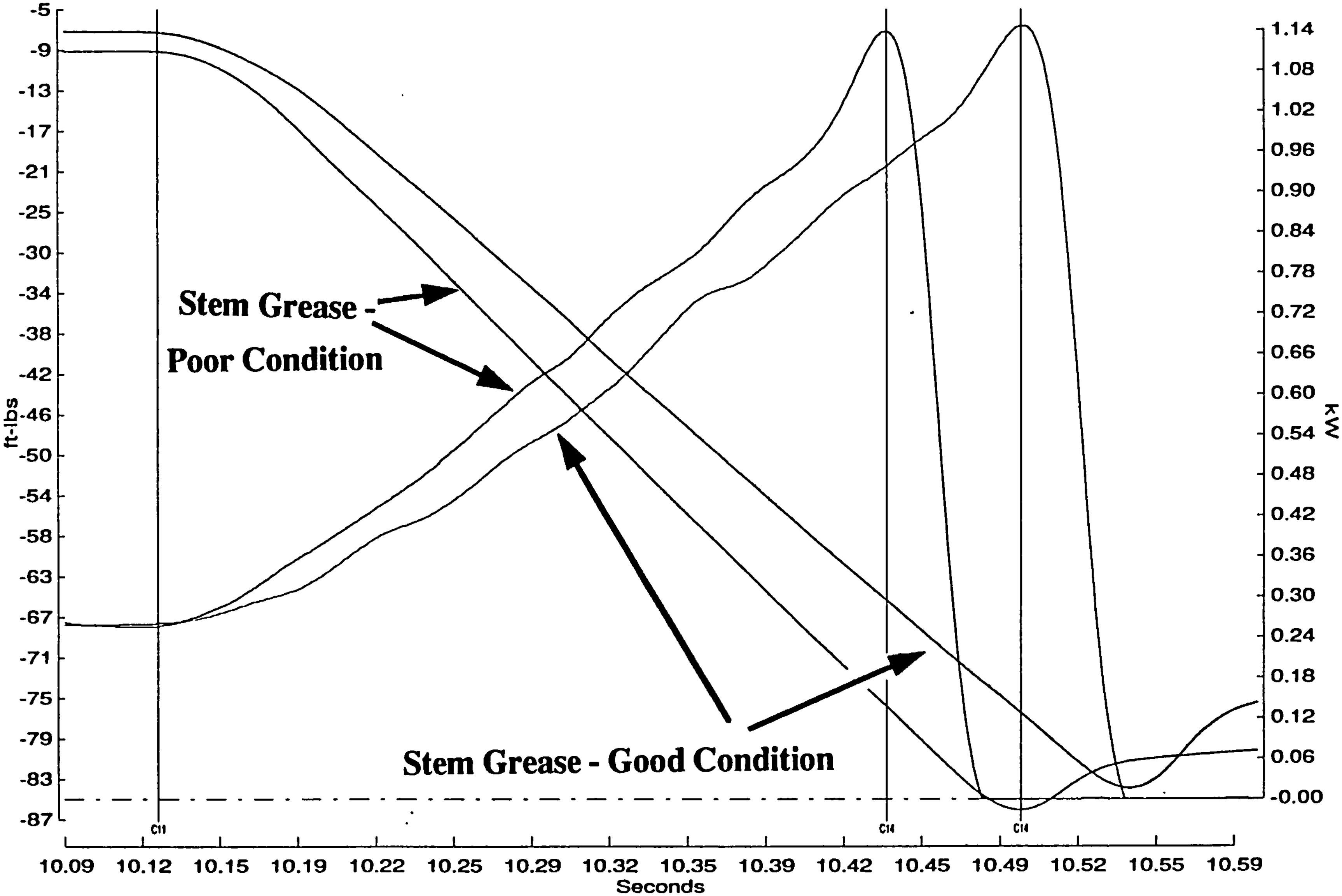


Figure 4

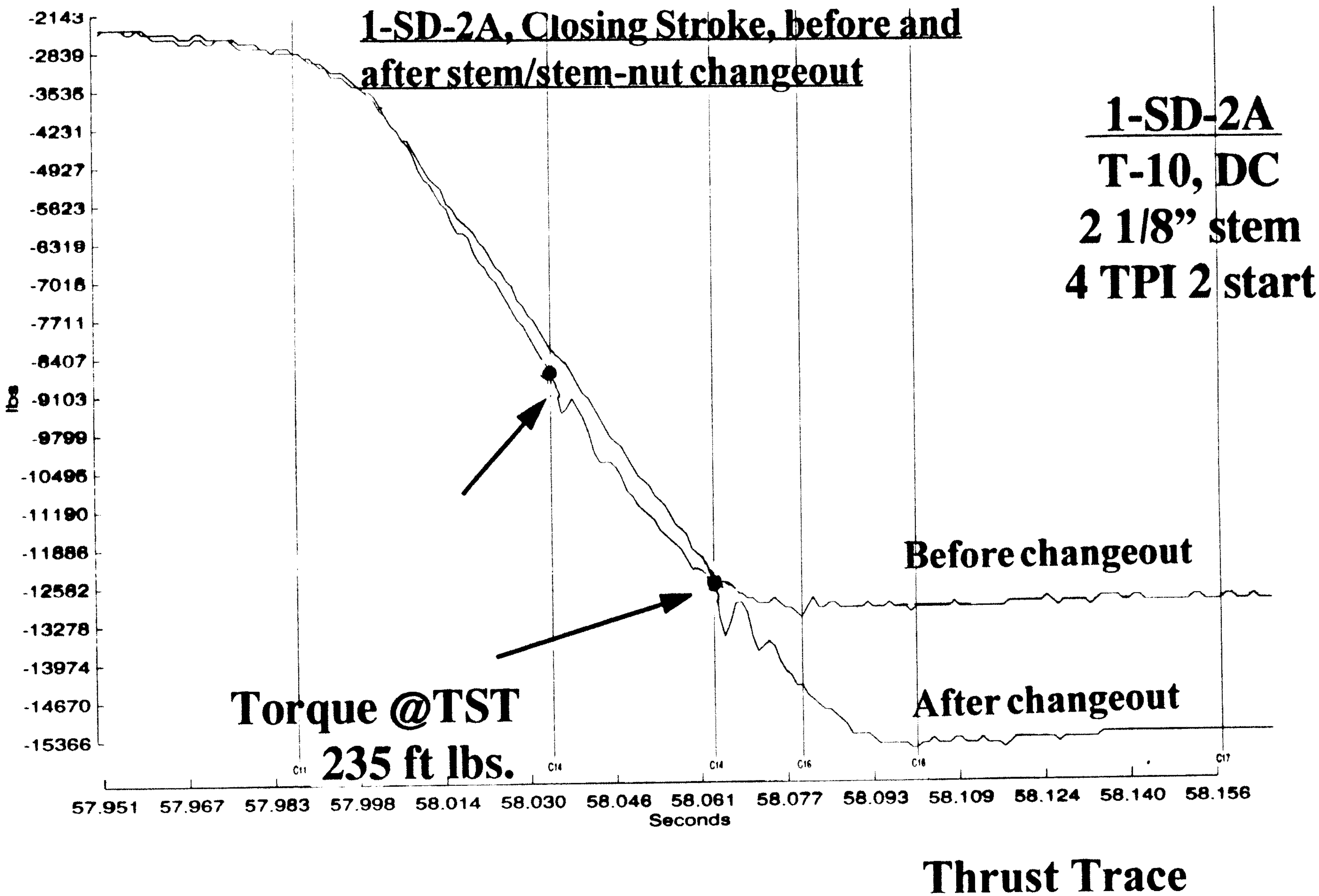
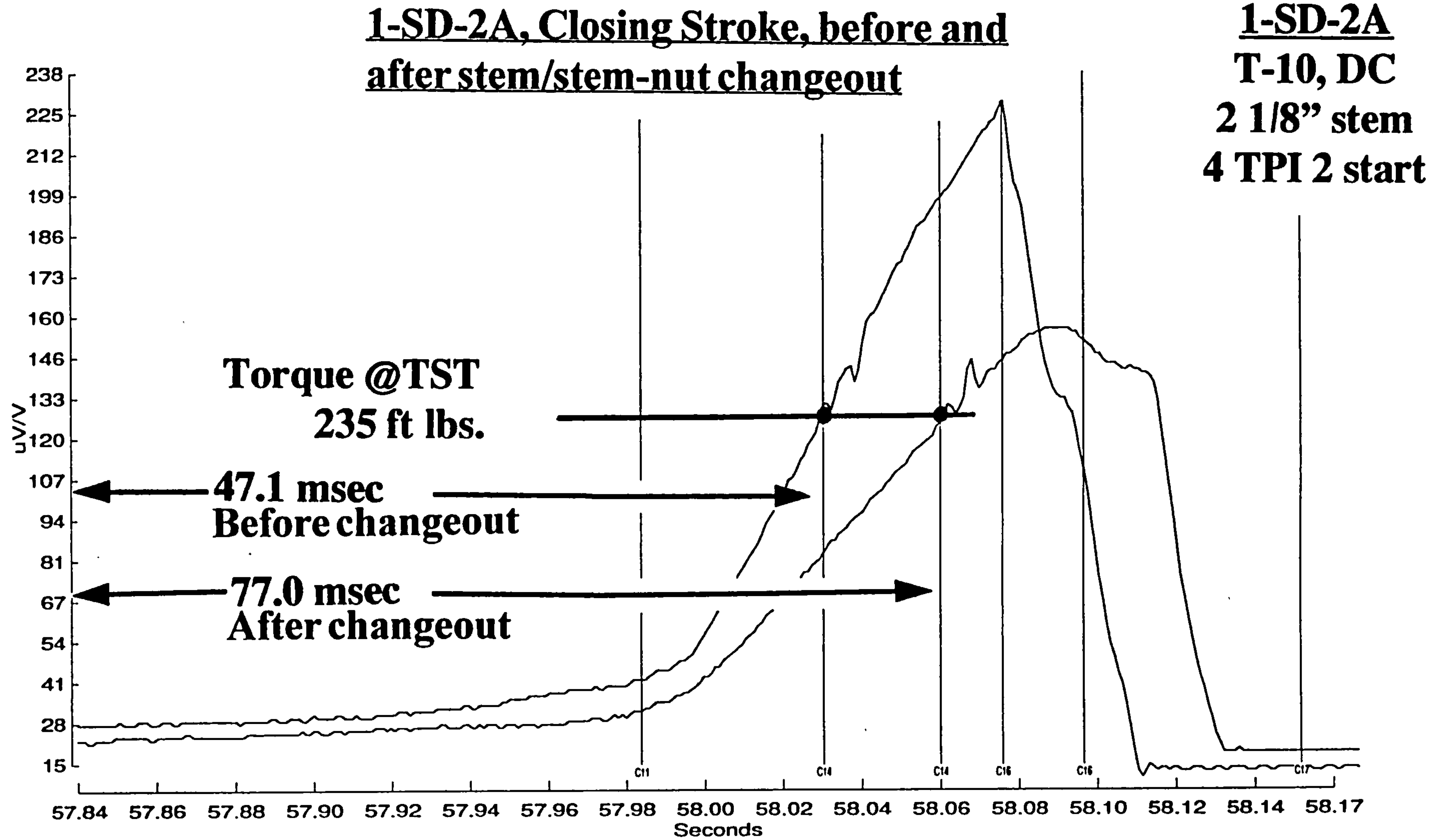

1-SD-2A, Closing Stroke, before and after stem/stem-nut changeout
1-SD-2A
T-10, DC
2 1/8" stem
4 TPI 2 start
Before changeout
After changeout
Torque @TST
235 ft lbs.
lbs
-2143
-2839
-3535
-4231
-4927
-5623
-6319
-7015
-7711
-8407
-9103
-9799
-10495
-11190
-11886
-12582
-13278
-13974
-14670
-15366
57.951
57.967
57.983
57.998
58.014
58.030
58.046
58.061
58.077
58.093
58.109
58.124
58.140
58.156
Seconds
Thrust Trace

Figure 5



Torque Trace

Figure 6

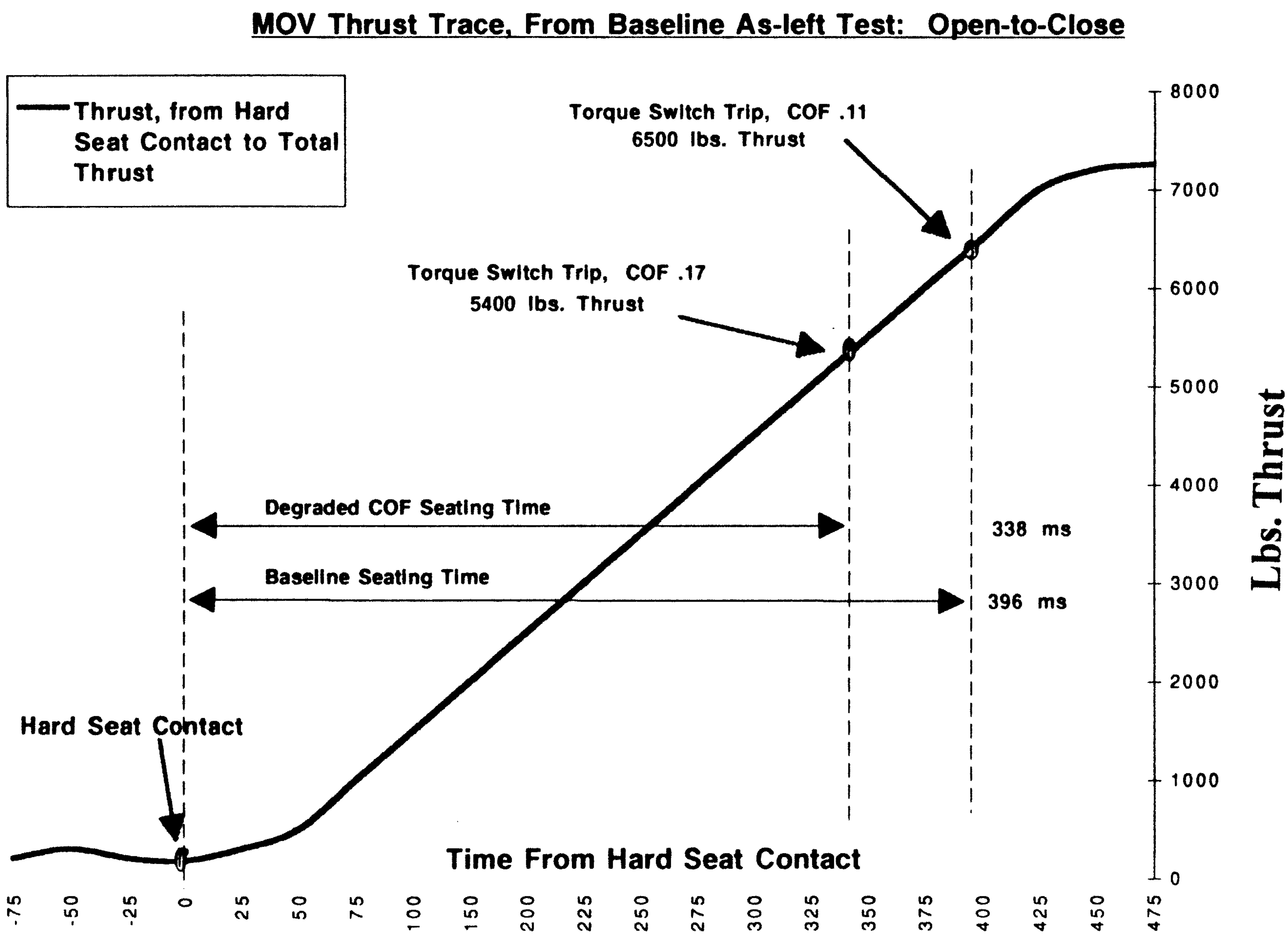

MOV Thrust Trace, From Baseline As-left Test: Open-to-Close
Thrust, from Hard Seat Contact to Total Thrust
Torque Switch Trip, COF .11
6500 lbs. Thrust
Torque Switch Trip, COF .17
5400 lbs. Thrust
Degraded COF Seating Time
338 ms
Baseline Seating Time
396 ms
Hard Seat Contact
Time From Hard Seat Contact
Lbs. Thrust
8000
7000
6000
5000
4000
3000
2000
1000
0
-75
-50
-25
0
25
50
75
100
150
200
225
250
275
300
325
350
375
400
425
450
475

Figure 7

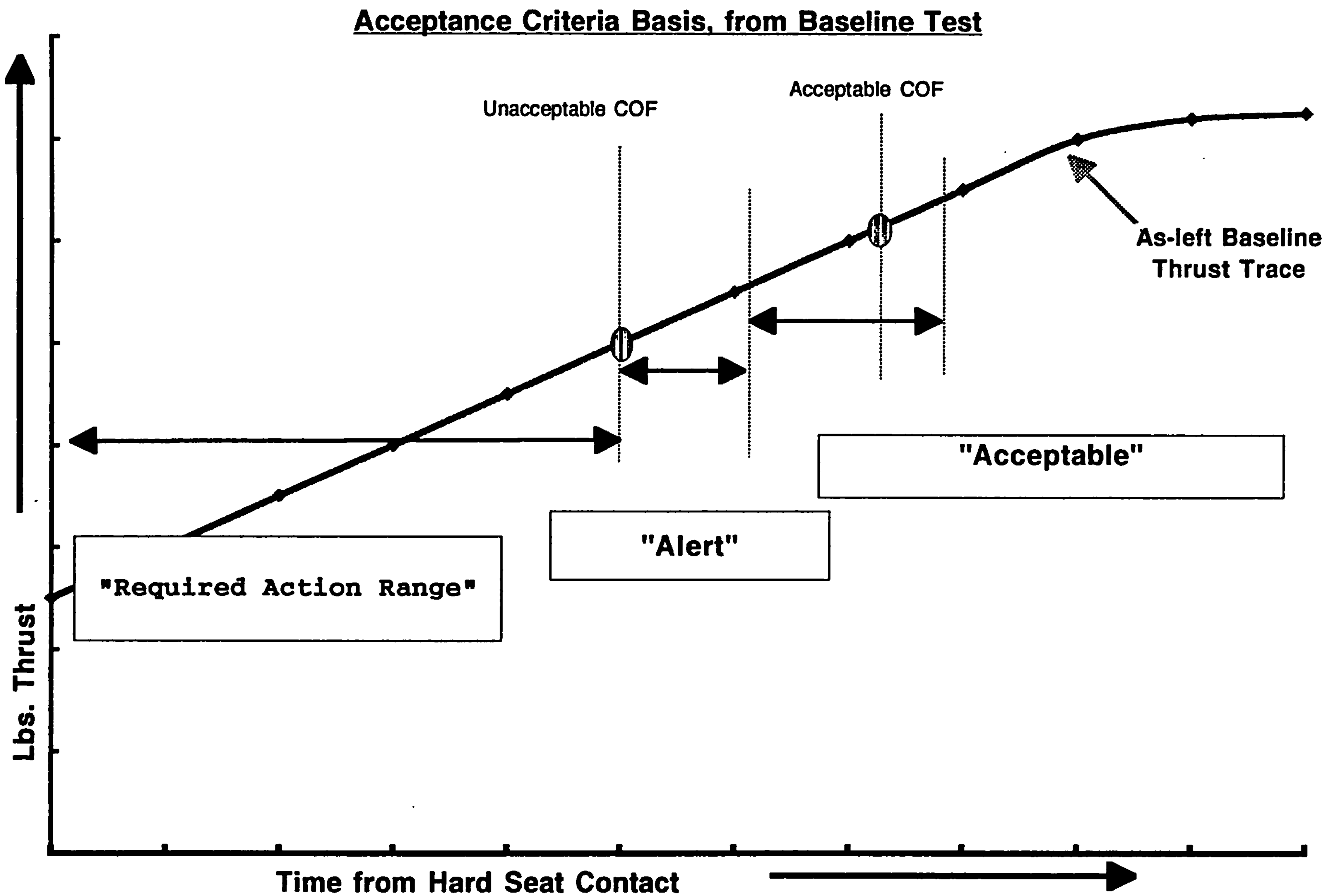

Acceptance Criteria Basis, from Baseline Test
Unacceptable COF
Acceptable COF
As-left Baseline Thrust Trace
"Acceptable"
"Alert"
"Required Action Range"
Lbs. Thrust
Time from Hard Seat Contact

Figure 8

Monitoring Systems for Motor-Operated Valves

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ABSTRACT
INTRODUCTION AND BACKGROUND

On June 28, 1989, the U.S. Nuclear Regulatory Commission (NRC) issued Generic Letter 89-10 to all holders of nuclear power plant operating licenses and construction permits. The generic letter recommends that owners develop and implement a program to ensure that Motor Operated Valve (MOV) switch settings (torque, torque bypass, etc.) be selected, set and maintained for the life of the plant.

GL 89-10 provided limited guidance for long term maintenance of MOV switch settings, therefore, the U.S.N.R.C. proposed Generic Letter 96-XX to reiterate the importance of periodic verification as well as provide further guidance for long term maintenance of MOV switch settings.

To ensure that a MOV continues to meet design basis criteria over time, it is proposed to monitor "the performance" of the MOV on a regular basis. This would not be feasible using existing acquisition equipment used for baseline testing. Instead a digital monitoring system is proposed to provide an alternate method for periodic verification and trending.

The digital monitoring system was designed to utilize the existing base of stem sensors used for measuring thrust and torque during baseline testing. In this manner, dynamic thrust and torque measurements can be obtained during in-service testing and MOV performance can be trended over time.

The digital monitoring system is located in an easy to access area and connected to the MOV via low voltage cables. The system will monitor all connected channels on a continuous basis, however, only retain signal data when a specified event occurs (i.e. energize motor). Acquired data can be removed from the system using the commercially available data cartridge.

This paper will present further detail of the design and capabilities of the digital monitoring system as well as provide actual in-situ test results.

Session 4

Regulatory Issues

Session Chair
Thomas G. Scarbrough
Senior Engineer
U.S. Nuclear Regulatory Commission

LONG-TERM PERFORMANCE OF MOTOR-OPERATED VALVES

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ABSTRACT

The U.S. Nuclear Regulatory Commission (NRC) requires that motor-operated valves (MOVs) important to safety be designed, fabricated, erected, and tested to quality standards commensurate with the importance of the safety functions to be performed. Despite these requirements, operating experience and research revealed problems with the performance of MOVs in operating nuclear power plants. In response to the concerns about MOV performance, the NRC issued Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance," and its supplements. Most licensees have completed the aspects of their GL 89-10 programs associated with the review of MOV design bases, verification of MOV switch settings initially, testing of MOVs under design-basis conditions where practicable, and improvement of evaluations of MOV failures and necessary corrective action. Licensees are establishing processes to ensure that the long-term aspects of their MOV programs, such as periodic verification of MOV capability and the trending of MOV problems, are maintained. The NRC staff is developing a generic letter to address periodic verification of MOV design-basis capability.

<u>Overview</u>

Many fluid systems at nuclear power plants depend on the successful operation of motor-operated valves (MOVs) in performing their safety functions. For example, MOVs may be required to open to allow cooling water to be supplied to the reactor core, steam generators, or containment building. They may be required to open to allow steam flow for turbine-driven pumps in safety systems that supply cooling water to the reactor core, steam generators, or containment building. MOVs may be required to close to prevent loss of coolant from the reactor core or to isolate the reactor containment. To ensure plant safety, they must be capable of performing their functions under design-basis conditions, which may include high differential-pressure and flow, high ambient temperature, and degraded motor voltage.

The complex nature of the MOV and the varied conditions under which it must operate demand that careful attention be paid to all applicable activities, from design to replacement, in order to ensure reliable operation. In the design of the MOV, a suitable analysis must be performed using valid engineering equations and parameters to ensure that the MOV will operate, as intended, under normal plant operations and during design-basis events. Manufacturing, installation, preoperational testing, operation,

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inservice testing, maintenance, and replacement of the MOV must be conducted by trained personnel using proper procedures. Surveillance must be performed and testing criteria must be applied on a soundly based frequency in a manner that suitably detects questionable operability or degradation of the MOV. Moreover, these activities must be conducted in accordance with a strong quality assurance program.

Regulatory Requirements

NRC regulations require that components that are important to the safe operation of a nuclear power plant, including MOVs, be treated in a manner that provides assurance of their performance. Appendix A, "General Design Criteria for Nuclear Power Plants," and Appendix B, "Quality Assurance Criteria for Nuclear Power Plants and Fuel Reprocessing Plants," to Part 50 of Title 10 of the *Code of Federal Regulations* (10 CFR Part 50) contain broad-based requirements in this regard. In 10 CFR 50.55a(f), the NRC requires licensees to comply with Section XI of the American Society of Mechanical Engineers Boiler and Pressure Vessel Code (ASME Code).

MOV Problems

Despite the NRC regulations, operating experience at nuclear power plants revealed weaknesses in many activities associated with MOV performance. For example, some engineering analyses used in the initial design sizing and setting of MOVs were inadequate in predicting the thrust and torque required to open and close valves under design-basis conditions. Both regulatory and industry research programs later confirmed the weakness in the initial design and qualification of MOVs. Shortcomings in maintenance programs, such as inadequate procedures and training, also resulted in poor MOV performance. Typical inservice testing, consisting of measurement of valve stroke times under zero differential pressure and flow conditions, has been shown to be insufficient to detect certain deficiencies that could prevent MOVs from performing their safety functions under design-basis conditions.

NRC Action Plan on MOV Performance

In NUREG-1352 (June 1990), "Action Plans for Motor-Operated Valves and Check Valves," the staff described activities to help resolve the concerns about the performance of MOVs and check valves. These activities include evaluating the adequacy of current regulatory requirements and guidance, developing inspection guidance, coordinating NRC inspections, conducting regular meetings between the headquarters and regional staff, completing NRC research programs, cooperating with industry groups, evaluating the efforts of the NRC staff and the industry, and participating in organizations to prepare national codes and standards. The staff's action plan is periodically updated.

Generic Letter 89-10

Nuclear power plant operating experience, valve performance problems and MOV research revealed that the focus of the ASME Code on stroke time and leak-rate testing for MOVs was not sufficient in light of the design of the valves and the conditions under which they must function. For this reason, on June 28, 1989, the NRC staff issued Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance." In GL 89-10, the staff requested that licensees and construction permit holders ensure the capability of MOVs in safety-related systems

to perform their intended functions by reviewing MOV design bases, verifying MOV switch settings initially and periodically, testing MOVs under design-basis conditions where practicable, improving evaluations of MOV failures and necessary corrective action, and trending MOV problems. The staff requested that licensees complete the GL 89-10 program within approximately three refueling outages or 5 years from the issuance of the generic letter. Construction permit holders were requested to complete the GL 89-10 program before plant startup or in accordance with the licensee schedule (above), whichever was later.

Recommendation "d" of GL 89-10 requested that licensees and construction permit holders prepare procedures to ensure that correct MOV switch settings are maintained throughout the life of the plant. GL 89-10 stated that it may become necessary to adjust MOV switch settings because of wear or aging and that additional measures beyond ASME Code stroke-time testing should be taken to adequately verify that the switch settings ensure MOV operability. GL 89-10 suggested that licensees should periodically verify MOV capability every 5 years or every 3 refueling outages. Recommendation "h" of GL 89-10 requested that licensees evaluate trends in MOV performance every 2 years or at each refueling outage.

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

GL 89-10 and its supplements provide only limited guidance regarding periodic verification and the measures appropriate to ensure continued design-basis capability. The NRC staff is developing a new generic letter to give more complete guidance regarding periodic verification of safety-related MOVs. Although this guidance could have been discussed in a supplement to GL 89-10, the staff considered it appropriate to prepare a new generic letter to allow the staff to close out its review of GL 89-10 programs as promptly as possible.

Most nuclear power plant utilities have completed the verification of the design-basis capability of their GL 89-10 MOVs. The NRC staff has been closing its review of individual GL 89-10 programs on the basis of the completion of the design-basis verification of safety-related MOVs at each nuclear power plant and the utility's establishment of a program for periodic verification of MOV design-basis capability and for the trending of MOV problems. The staff may conduct a more complete review of licensee programs for MOV periodic verification as part of the implementation of a new generic letter.

Substantial licensee resources have been required to implement MOV programs in response to GL 89-10. However, the licensees' GL 89-10 programs have led to the identification and resolution of numerous weaknesses in the design, qualification, and maintenance of MOVs, and in corrective action for and trending of MOV problems. Through its inspection program, the staff has found that licensees have made significant progress in improving the design, qualification, and maintenance of MOVs.

Long-Term Aspects of MOV Programs

Over the last several years, nuclear power plant licensees have tested a large number of MOVs under static and dynamic conditions as

part of the implementation of their GL 89-10 programs. From these tests, licensees identified significant weaknesses in the design and qualification of MOVs used in nuclear power plants. These weaknesses caused MOVs to fail to operate properly during testing. Further, some MOVs operated adequately under test conditions, but analyses of the test results subsequently revealed that the MOVs might not have performed their safety functions under design-basis conditions. Licensees have expended significant resources to ensure that, despite the potential weaknesses in the original design and qualification, MOVs are currently capable of performing their safety functions under design-basis conditions.

In completing their GL 89-10 programs, licensees may have placed their confidence in the current design-basis capability of some safety-related MOVs on the thrust/torque requirements obtained directly from dynamic testing without additional margin for age-related degradation. For some valves, licensees may have employed other methods (such as grouping) to establish design-basis capability. In some cases, the thrust/torque requirements obtained from the dynamic tests were significantly less than the thrust/torque required to operate apparently identical MOVs. Further, NRC and industry testing indicates the potential for the thrust/torque required to operate a valve to increase with service until a plateau is reached. Aging can also decrease the thrust/torque output of motor actuators. Therefore, an effective program for periodic verification of MOV design capability requires an understanding of the performance of safety-related MOVs and the manner in which that performance can change with aging.

Static diagnostic tests produce information on the thrust/torque output of the motor actuator and changes to the motor-actuator output as a result of aging effects. The thrust and torque required to operate a valve are highly dependent on the differential pressure and flow across the valve disk, which are not present during static testing. Therefore, dynamic tests can provide information on the thrust/torque requirements and changes to those requirements as a result of aging effects. Efforts are under way within the nuclear industry to develop methods to obtain information from static tests that would allow prediction of valve dynamic performance. As discussed below, the Electric Power Research Institute (EPRI) has developed an analytical methodology that, when combined with static test data, provides bounding information on the thrust/torque requirements to operate valves under dynamic conditions.

Although there may be benefits to performing dynamic testing to ascertain the thrust/torque requirements and changes to these requirements as a result of aging, there are also potential detriments to dynamic testing (e.g., blowdown testing by EPRI resulted in damage to some valves). The NRC staff has not concluded that dynamic testing is the preferred method for periodic verification testing and believes dynamic testing may not be appropriate for certain situations.

The proposed method for periodic verification testing and demonstration of a particular valve's acceptability and ability to perform consistent with its design basis are the responsibility of the licensee. The proposed method for MOV periodic verification testing may be dependent on the valve and its application as well as the valve's performance

history and its contribution to overall plant risk. Various approaches can be taken to establish a periodic verification program that provides confidence in the long-term capability of MOVs to perform their design-basis safety functions. With each approach, potential degradation that can result in (1) the increase in thrust or torque requirements to operate the valves and (2) the decrease in the output capability of the motor actuator need to be addressed.

Electric Power Research Institute MOV Program

An MOV testing program conducted by EPRI has yielded significant information regarding the long-term design-basis capability of safety-related MOVs. In addition to finding that the thrust required to operate gate valves is typically greater than the thrust originally predicted by valve vendors, EPRI found that the thrust required to operate gate valves can increase with valve strokes until a plateau is reached.

In addition to information applicable to MOV periodic verification, the EPRI program has revealed performance characteristics of MOVs that might adversely affect a licensee's determination of the current capability of certain MOVs. In particular, EPRI found that a high percentage of gate valves were damaged during hot-water and steam-blowdown testing with thrust requirements unable to be predicted. For MOVs that might be damaged under such conditions, EPRI established possible modifications to valve internals for proper clearances and for rounding sharp edges. EPRI found that reliable prediction of globe valve thrust requirements requires an appropriate seat or guide area in thrust calculations. Although EPRI tested only one globe valve under high-temperature and blowdown conditions, the test revealed significantly higher thrust requirements than predicted. EPRI also found that load-sensitive behavior (or rate of loading) can reduce actuator thrust output under dynamic conditions. EPRI has furnished the results of its MOV tests to licensees through industry meetings, and the NRC staff has disseminated the results of the tests to licensees through information notices on the EPRI test program and at public meetings. Some licensees have already incorporated this information into their MOV programs.

The Nuclear Energy Institute (NEI) submitted EPRI Topical Report TR-103237, "EPRI MOV Performance Prediction Program," describing the methodology developed by EPRI to predict dynamic thrust and torque requirements for gate, globe, and butterfly valves without dynamic tests by licensees. On March 15, 1996, the NRC staff issued a safety evaluation (SE) which approves (with certain conditions and limitations) the topical report for use and reference.

Boiling Water Reactor (BWR) Owners' Group MOV Risk Ranking

The BWR Owners' Group Topical Report NEDC 32264, "Application of Probabilistic Safety Assessment to Generic Letter 89-10 Implementation," describes a methodology to rank MOVs in GL 89-10 programs with respect to their relative importance to core-damage frequency, including appropriate considerations regarding other consequences to be added by an expert panel. On February 27, 1996, the staff issued an SE on the topical report. The staff considers the methodology acceptable (in accordance with conditions or limitations contained in the NRC staff's SE) for ranking MOVs in BWRs because the

plant-specific insights are supplemented by generic insights and expert review involving additional considerations, such as external events and shutdown issues. In addition, the MOV rankings are used in combination with deterministic considerations that ensure a minimally acceptable frequency of testing is established even for the least risk-significant valves.

NRC Research Activities

In the 1980s, the NRC Office of Nuclear Regulatory Research (RES) sponsored a test program by the Idaho National Engineering Laboratory (INEL) to determine the thrust required to operate motor-operated gate valves under dynamic flow conditions. The EPRI valve test program confirmed the findings of the NRC's smaller scale test program. More recently, preliminary results from the testing of valve material samples sponsored by RES indicate that valve friction can increase with aging.

With respect to MOV ranking, RES sponsored a study of appropriate frequencies of periodic testing of MOVs based on their risk significance. This work is summarized in an article titled "Risk-Based Approach for Prioritizing Motor-Operated Valves" in NUREG/CP-0137, *Proceedings of the Third NRC/ASME Symposium on Valve and Pump Testing*. Additional work in this area is progressing.

American Society of Mechanical Engineers (ASME) Code Activities

Licensees are currently bound by the requirements in their Code-of-record regarding stroke-time inservice testing (IST), as supplemented by relief requests approved by the staff. Licensees have also verified MOV design-basis capability pursuant to their GL 89-10 commitments. The staff has long recognized the limitations of using stroke-time testing as a means of monitoring the operational readiness of MOVs (see GL 89-04, Supplement 1, "Guidance on Developing Acceptable Inservice Testing Programs") and has supported industry efforts to improve MOV periodic monitoring under the IST program and GL 89-10. As such, the staff would consider a periodic verification program that provides an acceptable level of quality and safety as an alternative to the current IST requirements for stroke-time testing and could authorize such an alternative, upon application by a licensee, pursuant to the provisions of 10 CFR 50.55a(a)(3)(i).

The ASME Operations and Maintenance Code Committee has developed a method to verify MOV design-basis capability by periodic testing. Through non-mandatory ASME Code Case OMN-1, "Alternative Rules for Preservice and Inservice Testing of Certain Electric Motor Operated Valve Assemblies in LWR Power Plants, OM Code 1995 Edition; Subsection ISTC," ASME is allowing the replacement of frequent stroke-time testing with periodic exercising of all safety-related MOVs once per cycle and periodic diagnostic testing under static or dynamic conditions, as appropriate, on a frequency to be determined on the basis of margin and degradation rate.

When implementing the code case, the benefits (such as identification of decreased thrust output and increased thrust requirements) and potential adverse effects (such as accelerated aging or valve damage) need to be considered when determining appropriate testing for each MOV.

The code case states that the maximum IST

frequency shall not exceed 10 years. In addition to this maximum test interval, where a selected test interval extends beyond about 5 years, it is important to evaluate information obtained from valve testing conducted during the initial time period to validate assumptions made in justifying the longer test interval.

Some licensees are developing risk-informed IST programs. As part of an industry pilot effort, two licensees have submitted requests to utilize a risk-informed approach to determine IST frequencies for certain components, in lieu of testing these components according to the frequencies specified in the ASME Code. The relationship of the code case to these pilot initiatives needs to be addressed.

Plant-Specific Programs

The staff has found effective programs for periodic verification of safety-related MOV design-basis capability at nuclear power plants to be characterized by several attributes, as follow:

- A risk-informed approach may be used to prioritize valve test activities, such as frequency of individual valve tests and selection of valves to be tested.
- The valve test program provides adequate confidence that safety-related MOVs will remain operable until the next scheduled test.
- The importance of the valve is considered in determining an appropriate mix of exercising and diagnostic testing. In establishing the mix of testing, the benefits (such as identification of decreased thrust output and increased thrust requirements) and potential adverse effects (such as accelerated aging or valve damage) are considered when determining the appropriate type of periodic verification testing for each safety-related MOV.
- All safety-related MOVs covered by the GL 89-10 program are considered in the development of the periodic verification program. The program includes safety-related MOVs that are assumed to be capable of returning to their safety position when placed in a position that prevents their safety system (or train) from performing its safety function; and the system (or train) is not declared inoperable when the MOVs are in their nonsafety position.
- Valve performance and maintenance are evaluated and monitored, with the periodic verification program periodically adjusted as appropriate.

Licensees of several facilities (for example, Callaway, Monticello, and South Texas) had established MOV periodic verification programs that the staff found acceptable during closure of its review of GL 89-10 programs. One approach to MOV periodic verification that the staff found acceptable is to diagnostically test each safety-related MOV every 5 years (or every 3 refueling outages) to determine thrust and torque motor-actuator output and any changes in the output. A specific margin to account for potential degradation such as that caused by age (in addition to margin for diagnostic error, equipment repeatability, load-sensitive behavior, and lubricant degradation) is established above the minimum thrust and torque requirements determined under the

GL 89-10 program. The selection of MOVs for testing, and their test conditions, takes into account safety significance, available margin, MOV environment, and the benefits and potential adverse effects of static and dynamic periodic verification testing on the selected MOV sample. Measures such as grouping and sharing of valve performance between facilities are appropriate to minimize the need to conduct more rigorous periodic verification tests.

Pressure Locking and Thermal Binding of Gate Valves

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ABSTRACT

Pressure locking and thermal binding represent potential common mode failure mechanisms that can cause safety-related power-operated gate valves to fail in the closed position, thus rendering redundant safety-related systems incapable of performing their safety functions. Supplement 6 to Generic Letter 89-10, "Safety-Related Motor-Operated Gate Valve Testing and Surveillance," provided an acceptable approach to addressing pressure locking and thermal binding of gate valves. More recently, the NRC has issued Generic Letter 95-07, "Pressure Locking and Thermal Binding of Safety-Related Power-Operated Gate Valves," to request that licensees take certain actions to ensure that safety-related power-operated gate valves that are susceptible to pressure locking or thermal binding are capable of performing their safety functions within the current licensing bases. Over the past two years, several plants in Region I determined that valves in certain systems were potentially susceptible to pressure locking and thermal binding, and have taken various corrective actions. The NRC Region I Systems Engineering Branch has been actively involved in the inspection of licensee actions in response to the pressure locking and thermal binding issue. Region I continues to maintain an active involvement in this area, including participation with the Office of Nuclear Reactor Regulation in reviewing licensee responses to Generic Letter 95-07.

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

NRC Staff Review of Licensee Responses to Pressure-Locking and Thermal-Binding Issue

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ABSTRACT

Commercial nuclear power plant operating experience has indicated that pressure locking and thermal binding represent potential common mode failure mechanisms that can cause safety-related power-operated gate valves to fail in the closed position, thus rendering redundant safety-related systems incapable of performing their safety functions. In Generic Letter (GL) 95-07, "Pressure Locking and Thermal Binding of Safety-Related Power-Operated Gate Valves," the U.S. Nuclear Regulatory Commission (NRC) staff requested that nuclear power plant licensees take certain actions to ensure that valves susceptible to pressure locking or thermal binding are capable of performing their safety functions within the current licensing bases of the facility. The NRC staff has received summary information from licensees in response to GL 95-07 describing actions they have taken to prevent the occurrence of pressure locking and thermal binding. The NRC staff has developed a systematic process to help ensure uniform and consistent review of licensee submittals in response to GL 95-07.

1.0 INTRODUCTION

Pressure locking can occur in flexible-wedge and double-disk gate valves when pressurized fluid becomes trapped within the valve bonnet. The valve actuator may not be capable of overcoming the additional thrust required as a result of the differential pressure created across both valve disks by the pressurized fluid in the valve bonnet. For example, the fluid may enter the valve bonnet during normal cycling of open and closed valves, (1) when a fluid differential pressure across a disk causes the disk to move slightly away from the seat, creating a path to either increase the fluid pressure or fill the bonnet with fluid, or (2) for a steamline valve, when differential pressure exists across the disk and the valve orientation permits condensate to collect and enter the bonnet. Surveillance testing can cause a valve to experience pressure locking or thermal binding. For example, an inboard isolation motor-operated valve (MOV) in the reactor core isolation cooling (RCIC) system steamline at a boiling-water reactor (BWR) plant failed in the closed position after routine surveillance testing. Pressure locking and thermal binding can occur to varying degrees and may render a valve incapable of operating.

Various plant operating conditions can introduce pressure locking. Pressure in the valve bonnet might be higher than anticipated when (1) the gate valve is in a line connected to a high-pressure system or (2) the temperature of the fluid in the valve bonnet increases, causing thermal expansion.

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Temperature in the valve bonnet might increase in response to heatup during plant operation, a rise in ambient air temperature caused by leaking components or postulated pipe breaks, or thermal conduction or convection through connected piping. Over time, bonnet pressure could decrease by leakage past the seating surfaces or stem packing. However, during the time to depressurize, the valve may remain pressure locked, and the system may not be able to perform its safety function. Also, valve actuator operation at locked rotor conditions could degrade the motor torque capability of a motor-operated gate valve.

Thermal binding is generally associated with a wedge gate valve that is closed at high temperature and is allowed to cool before attempted reopening. Mechanical interference occurs because of contraction of the valve body on the disk wedge. Thus, reopening the valve might be prevented until the valve and disk are reheated. Solid-wedge gate valves are most susceptible to thermal binding. However, flexible-wedge gate valves experiencing significant temperature changes or operating with significant upstream and downstream temperature differences may thermally bind.

Pressure locking or thermal binding occurs as a result of the valve design characteristics (wedge and valve body configuration, flexibility, and material thermal coefficients) and system operating conditions. These conditions can occur when the valve is subjected to pressures and temperatures that may not have been considered as part of the design basis for valves in many plants.

1.1 History of Events and Generic Communications

The nuclear power industry has issued several event reports describing failure of safety-related gate valves to operate as a result of pressure locking or thermal binding. Several of the industry's generic communications have given guidance for identifying susceptible valves and for taking appropriate preventive and corrective measures. In addition to events at U.S. nuclear power plants, French experience with pressure locking events was documented in NUREG/CP-0137, "Proceedings of the Third NRC/ASME Symposium on Valve and Pump Testing, Volume 2, (July 1994)."

In Generic Letter (GL) 89-10, "Safety-Related Motor-Operated Valve Testing and Surveillance" (June 28, 1989), the NRC staff asked licensees to provide additional assurance of the capability of safety-related MOVs and certain other MOVs in safety-related systems to perform their safety-related functions. The NRC staff asked licensees to review MOV design bases, verify MOV switch settings both initially and periodically, test MOVs under design-basis conditions where practicable, improve evaluations of MOV failures and necessary corrective action, and trend MOV problems. In Enclosure 1 to Supplement 6 of GL 89-10 (March 8, 1994), the NRC staff described one acceptable approach that licensees could use to address pressure locking and thermal binding of motor-operated gate valves.

In March 1993, the NRC issued NUREG-1275, Volume 9, "Operating Experience

Feedback Report - 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. The NRC staff held a public workshop on February 4, 1994, to discuss pressure locking and thermal binding of gate valves, including prioritization of susceptible valves for corrective action. A summary of the public workshop is contained in NUREG/CP-0146, "Workshop on Gate Valve Pressure Locking and Thermal Binding."

Recently, the NRC has issued several information notices (INs) to alert licensees to the potential for gate valves to experience pressure locking:

- NRC IN 96-08, "Thermally Induced Pressure Locking of a High Pressure Coolant Injection Gate Valve," dated February 5, 1996
- NRC IN 95-30, "Susceptibility of Low-Pressure Coolant Injection and Core Spray Injection Valves to Pressure Locking," dated August 3, 1995
- NRC IN 95-18, "Potential Pressure-Locking of Safety-Related Power-Operated Gate Valves," dated March 15, 1995, and Supplement 1, dated March 31, 1995
- NRC IN 95-14, "Susceptibility of Containment Sump Recirculation Gate Valves to Pressure Locking," dated February 28, 1995

Several plants have experienced either pressure locking or thermal binding in safety-related and non-safety-related systems. These cases are discussed in NUREG-1275, Volume 9. Examples of gate valves involved in pressure-locking events are the following:

- low-pressure coolant injection and low-pressure core spray system injection valves
- residual heat removal (RHR) system hot-leg crossover isolation valves
- RHR containment sump and suppression pool suction valves
- high-pressure coolant injection (HPCI) steam admission valves
- RHR heat exchanger outlet valves
- emergency feedwater isolation valves
- RCIC steamline isolation valves

Examples of gate valves involved in thermal-binding events are the following:

- reactor depressurization system isolation valves
- RHR inboard suction isolation valves
- HPCI steam admission valves
- power-operated relief valve block valves
- reactor coolant system letdown isolation valves
- RHR suppression pool suction valves

- containment isolation valves (sample line, letdown heat exchanger inlet header)
- condensate discharge valves
- reactor feedwater pump discharge valves

1.2 Recent Experiences

As discussed in NUREG-1275, Volume 9, the NRC staff became concerned that the operational configurations of safety-related power-operated gate valves may cause them to fail to open during a design basis event as a result of pressure locking or thermal binding. Although NUREG-1275, Volume 9, includes examples of valves in non-safety-related systems and valves which do not have a safety function in the open position, the NRC staff found that valves in safety significant applications may fail to open due to similar conditions.

After NUREG-1275, Volume 9 was issued, the NRC staff made a number of site visits to discuss pressure locking and thermal binding with licensees in order to (1) gather information on the technical issues related to generic and plant-specific valve and system characteristics and (2) determine the implementation status of previous industry guidance for identifying susceptible valves and application of preventive and corrective measures. NRC surveys showed that in response to the number of generic industry communications on the subject, some licensees have performed multiple reviews of pressure locking and thermal binding. However, the staff found only limited instances of corrective actions taken to alleviate the effects of these phenomena.

In Enclosure 1 to Supplement 6 of GL 89-10, the NRC staff stated that licensees are expected under existing regulations to take actions to ensure that safety-related motor-operated gate valves susceptible to pressure locking or thermal binding are capable of performing their required safety functions, and described an acceptable approach that licensees could use to address pressure locking and thermal binding of motor-operated gate valves as part of their GL 89-10 programs. The information on pressure locking and thermal binding of motor-operated gate valves given in Enclosure 1 to Supplement 6 of GL 89-10 was intended as timely notification of operating experience feedback. During inspections of GL 89-10 programs, the staff found that the actions taken by licensees to address pressure locking and thermal binding of motor-operated gate valves were varied. Although many licensees had conducted some level of review of the potential for pressure locking and thermal binding of their motor-operated gate valves, few licensees had either (1) thoroughly evaluated the capability of the motor actuators to overcome the phenomena in light of recent information regarding MOV and system performance or (2) taken corrective action to prevent the phenomena from occurring. In view of these inspection results, the NRC staff determined that issuance of a new generic letter was appropriate.

2.0 GENERIC LETTER 95-07

On August 17, 1995, the NRC issued GL 95-07, "Pressure Locking and Thermal Binding of Safety-Related Power-Operated Gate Valves," to request that licensees perform or confirm that they had previously performed, (1) evaluations of the operational configurations of safety-related, power-operated (including motor-, air-, and

hydraulically operated) gate valves for susceptibility to pressure locking and thermal binding and (2) further analyses, and any needed corrective actions, to ensure that safety-related power-operated gate valves that are susceptible to pressure locking or thermal binding are capable of performing the safety functions within the current licensing basis of the facility.

2.1 Summary of Actions Requested in Generic Letter 95-07

In GL 95-07, the NRC staff requested that, within 90 days, licensees (1) perform a screening evaluation of the operational configurations of all safety-related power-operated gate valves to identify those valves that are potentially susceptible to pressure locking or thermal binding and (2) document a basis for the operability of the potentially susceptible valves or, where operability cannot be supported, take action in accordance with individual plant technical specifications. The staff established the recommended actions for the 90-day screening evaluation in GL 95-07 to provide confidence that no short-term safety concerns existed regarding particular valves as a result of pressure locking or thermal binding. The purpose of this action was for licensees to ensure that no critical deficiencies existed in past evaluations and take action if appropriate. The essence of the 90-day screening evaluation was to conduct an initial assessment, using current knowledge, of all safety-related power-operated gate valves to ensure they were capable of performing their safety functions if they were susceptible to pressure locking or thermal binding.

In GL 95-07, the NRC staff requested that, within 180 days, licensees evaluate the operational configurations of safety-related power-operated gate valves to identify valves that are susceptible to pressure locking or thermal binding, perform further analyses as appropriate, and take needed corrective actions (or justify longer schedules) to ensure that the susceptible valves identified are capable of performing their intended safety functions under all modes of plant operation, including test configurations. The purpose of the 180-day requested actions was for licensees to further analyze or identify, schedule and take corrective action for those susceptible valves, in a timely manner, to ensure that they are capable of performing their intended safety functions under all modes of plant operation.

In GL 95-07, the NRC staff stated that, if a licensee has performed an evaluation of motor-operated gate valves to identify those susceptible to pressure locking or thermal binding and has performed additional analyses and taken needed corrective actions for susceptible valves, in a manner that satisfactorily implements the guidance in Supplement 6 to GL 89-10 (or equivalent industry methods), the licensee need not take any additional action for MOVs.

2.2 Information Requested in Generic Letter 95-07

The NRC staff requested that licensees submit a summary description of actions taken in response to GL 95-07, including the following:

(1) the susceptibility evaluation of operational configurations and further analyses performed in response to the 180-day requested actions, including the bases or criteria for determining that valves are or are not susceptible to pressure locking or thermal binding

(2) the results of the susceptibility evaluation and the further analyses, including a listing of the susceptible valves identified

(3) the corrective actions, or other dispositioning, for the valves identified as susceptible to pressure locking or thermal binding, including (a) equipment or procedural modifications completed and planned (including the completion schedule for such actions) and (b) justification for any determination that particular safety-related power-operated gate valves susceptible to pressure locking or thermal binding are acceptable as is

3.0 REVIEW PROCESS

The NRC staff is performing a thorough technical review of the summary information submitted by licensees in response to GL 95-07. The staff has developed a systematic approach to help ensure a consistent review of each submittal. A team of staff members from NRC Headquarters and each regional office was formed to review the GL 95-07 licensee submittals. Much of the review effort has been completed and, where applicable, requests for additional information have been sent to licensees.

3.1 Scope

In GL 95-07, the NRC staff requested that licensees evaluate the operational configurations of all safety-related power-operated gate valves to determine whether the valves are susceptible to pressure locking or thermal binding. To verify that licensees have included all potentially susceptible safety-related power-operated gate valves in their GL 95-07 review process, the staff performed a screening review of gate valve populations for each plant. Discussions are held with licensees regarding valves that were not addressed in their submittals.

Through review of operating experience feedback, the staff has found that gate valves in certain systems are more likely to experience pressure locking or thermal binding and represent the highest risk significance for this issue. Therefore, to provide an appropriate focus for further evaluations, the staff reviewed in detail the following systems (where applicable) to ensure that they are properly included in licensees' GL 95-07 programs:

General Electric BWRs

- reactor coolant
- low pressure coolant injection
- low pressure core spray
- residual heat removal
- high pressure coolant injection
- high pressure core spray
- reactor core isolation cooling
- containment spray
- drywell spray
- core spray
- isolation condenser

Westinghouse Pressurized Water Reactors (PWRs)

- reactor coolant
- residual heat removal
- safety injection
- high pressure safety injection
- containment spray
- auxiliary feedwater

Combustion Engineering PWRs

- reactor coolant

- low pressure safety injection
- high pressure safety injection
- auxiliary feedwater
- containment spray

Babcock and Wilcox PWRs

- reactor coolant
- decay heat removal
- low pressure injection
- emergency core cooling
- emergency feedwater
- high pressure injection

Other systems may contain potentially susceptible valves and are reviewed on a plant-specific basis.

3.2 Operational Configurations

The staff reviews each licensee's evaluation of the operational configurations of safety-related power-operated gate valves to determine valves susceptible to pressure locking or thermal binding. Focusing primarily on the systems discussed above, the staff independently checks the licensees' identification of susceptible valves. With regard to valves within the scope of GL 95-07:

- Flexible-wedge and double-disk gate valves that could entrap system pressure during plant operation and experience a significant decrease in pressure in the attached piping during a design-basis event are considered susceptible to depressurization-induced pressure locking.

- Flexible-wedge and double-disk gate valves that may become water filled during plant operation and experience an increase in temperature resulting in an uncontrolled rise in bonnet pressure are considered susceptible to thermally induced pressure locking.

- Solid-wedge and flexible-wedge gate valves that are shut at a high temperature and experience a significant cooldown are considered susceptible to thermal binding.

Valves that meet these criteria are considered susceptible to pressure locking or thermal binding for the purposes of the GL 95-07 review. To determine a particular valve's susceptibility to pressure locking or thermal binding, system piping and instrumentation diagrams are used to evaluate heat and pressure sources that could lead to these phenomena.

3.3 Further Analysis and Disposition

In GL 95-07 the NRC staff requested that licensees perform further analyses as appropriate and take needed corrective actions for safety-related power-operated gate valves that are susceptible to pressure locking or thermal binding. Licensee actions and justifications for valves susceptible to these phenomena include equipment modifications, procedure modifications, analysis and testing and operational experience. The staff has developed the following technical review guidelines regarding these actions and justifications:

Equipment Modifications

The staff considers equipment modifications to be the least difficult alternative to justify in addressing pressure locking of susceptible gate valves. Examples of possible modifications to prevent pressure locking are given in NUREG-1275, Volume 9. Modifications to

prevent thermal binding are also possible, such as replacing a wedge gate valve with a parallel-disk gate valve.

Procedure Modifications

The staff considers procedure modifications to be a strong alternative for preventing thermal binding of gate valves. Procedure modifications, however, are less likely to be a justifiable alternative for preventing pressure locking of gate valves.

Analysis

The staff considers the prediction of the thrust required to overcome pressure locking or thermal binding to be very difficult. A licensee may be able to justify adequate actuator capability in response to pressure locking for certain (e.g., small) valves. Because of the uncertainties in valve geometries and material expansion and contraction characteristics, the staff believes that a licensee will need to expend considerable effort to justify this alternative in a manner adequate to resolve concerns regarding thermal binding.

Testing and Operational Experience

A licensee may be able to demonstrate through an in-situ or prototype test that the actuator has adequate capability to overcome pressure locking for a particular valve. The staff considers this alternative difficult to justify for thermal-binding concerns because of the uncertainty in modeling actual plant and valve conditions. A licensee may be able to demonstrate adequate capability of the actuator to overcome pressure locking on the basis of test information on the particular valve or similar valves from other sources, together with an analysis to demonstrate applicability. As with the analysis option, the staff considers this alternative difficult to justify for thermal-binding concerns.

For valves susceptible to pressure locking or thermal binding, the staff reviews the licensee's further analysis and dispositioning to ensure that sufficient assurance that the valve can perform its safety functions has been provided. The level of assurance should be commensurate with the safety significance of the particular valve. If a licensee's submittal does not contain sufficient information, the staff holds discussions with the licensee or requests additional information.

The staff believes that a corrective action schedule (if corrective actions are needed) may be based on risk significance, including consideration of common-cause failure of multiple valves. Plant operation and outage schedules may also be considered in developing corrective action schedules.

The NRC regulations require an analysis under Section 50.59 of Title 10 of the Code of Federal Regulations (10 CFR) for any valve modifications and the establishment of adequate post modification and inservice testing of any valves installed as part of the modification. In cases where a valve has been modified to add a pressure equalizing path, licensees may need to evaluate the effect of unidirectional leak tightness.

In addition, recent operational experience has indicated that, for emergency core cooling system injection valves, leaking downstream check valves may increase the differential pressure under which inservice testing of the injection valves is performed. This increase in differential pressure may be significantly above design values for the valve and result in damage to the actuator. A pressure equalizing

path installed to preclude pressure locking will not eliminate this potential problem.

As required by Appendix B to 10 CFR Part 50, the licensee may need to establish training for plant personnel to perform any necessary actions and to incorporate specific precautions into or revise the existing plant operating procedures. For example, plant personnel might periodically stroke certain valves to reduce the potential for thermal binding.

3.4 Gate Valve Testing

To aid the staff in evaluating the acceptability of licensee actions taken in response to GL 95-07, the NRC Office of Nuclear Regulatory Research has sponsored gate valve testing at the Idaho National Engineering Laboratory (INEL). These tests are intended to study the following issues:

- the relationship between bonnet pressurization and thrust required for a limited sample of flexible-wedge and parallel-disk gate valves
- the impact of a changing temperature environment in the vicinity of a gate valve bonnet on the rate of bonnet pressurization and the associated thrust required to overcome thermally induced pressure locking, including the effects of air entrapped in the valve bonnet, for a limited sample of flexible-wedge and parallel-disk gate valves
- uncertainty in the ability to calculate leakage rate and its effect on depressurization-induced and thermally induced pressure locking
- the extent to which air will remain entrapped in a valve bonnet during plant operation
- the occurrence of thermal binding in sample valves, including the magnitude of temperature difference across the valve and the rate of temperature difference.

INEL has completed the initial phase of testing for a flexible-wedge gate valve. The NRC is analyzing the results of this testing. INEL is conducting a second series of tests on a double-disk gate valve. The staff plans to make the INEL test results available to the industry as soon as practicable.

4.0 CONCLUSION

Operating experience has shown that pressure locking and thermal binding of safety-related power-operated gate valves represents an issue of high safety significance. In GL 95-07, the staff requested that licensees take actions to ensure that valves susceptible to pressure locking or thermal binding are capable of performing their safety functions within the current licensing basis of the facility. The staff has received summary information from licensees describing actions taken in response to GL 95-07 and has implemented a systematic review process to help ensure consistent and timely resolution of this safety issue. The staff anticipates completing the review of licensee submittals and issuance of safety evaluation reports by September 1996. As a followup to the initial review, the staff may develop an NRC Inspection Manual temporary instruction for limited followup inspections of GL 95-07 programs.

Industry Activities to Improve Valve Performance

Clive Callaway
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ABSTRACT

Motor-operated valve issues refuse to go away. For over a decade the industry and the NRC have been focusing extraordinary resources on assuring these special component operate when called upon. Now that we have fixed the design deficiencies, we are focusing on assuring that they perform their safety function within the current licensing basis for the remainder of plant life. NEI supported the efforts by ASME to develop OMN-1 and was encouraged that the industry and the NRC worked together to develop risk and performance based approaches to maintain MOV performance.

Improvements in Inservice Testing Regulatory Guidance

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Purpose

The purpose of this paper is to summarize regulatory actions related to inservice testing (IST) that have occurred since the last symposium held in July 1994. Other papers are focusing on regulatory actions currently under development, such as risk-informed standard review plans and regulatory guides. As such, this paper does not address those subjects.

Introduction

Over the last few years, the NRC has issued guidance to licensees toward improving the quality of requests for relief or alternatives to the code requirements and bring some measure of consistency to the implementation of the code requirements. Based on the majority of the recent submittals, the overall quality has improved. If all of the steps in the internal process for responding to the requests are timely, the review of an updated program can be completed in six to nine months. When a single or only a few requests are submitted, the staff is generally able to complete an evaluation in less than three months. When schedules are related to outages or exigent circumstances, we respond accordingly. The improvements in schedules can be attributed to publication of staff guidelines documents, licensees' efforts in preparing more complete descriptions and justification and an increase in the number of NRC technical staff reviewers assigned to IST.

Body

REPORTS: Two major IST-related guidance documents have been issued in the last two years. These documents contain information on all aspects of IST programs from the regulatory basis, determination of the applicable code requirements, program development, program implementation, and relief requests. The reports were intended to be for information only and not to establish any new requirements. The issues discussed in the reports were selected from previous relief requests or questions from licensees or for the purpose of updating earlier guidance.

NUREG-1482, Guidelines for Inservice Testing at Nuclear Power Plants - The final report was issued in April 1995. Supplement 1 to Generic Letter 89-04, "Guidance on Developing Acceptable Inservice Testing Programs," dated April 4, 1995, endorsed the recommendations in the report and gave approval of certain sections such that licensees could make use of provisions of the American Society of Mechanical (ASME) Operations and Maintenance Standards, Part 6 and Part 10, for IST. A number of licensees have reviewed and revised their IST programs to implement many of the recommendations from the report.

NUREG/CR-6396, INEL-95/0512, Examples, Clarifications, and Guidance on Preparing Requests for Relief from Pump and Valve Inservice Testing Requirements - This report was issued in February 1996. The report consolidates guidance and requirements for a number of testing issues that have been the

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

subject of relief requests. Examples are given of actual requests and evaluations that may be applicable to other similar plants. The purpose for the report was to supplement the information given in NUREG-1482 from a more detailed working level perspective as opposed to a broader regulatory perspective.

Inspection Procedure Revision: Inspection Procedure 73756, "Inservice Testing of Pumps and Valves," was revised July 27, 1995. The revision incorporates Generic Letter 89-04 and NUREG-1482 guidance. A more structured inspection procedure is included. The inspection procedure may be performed as a stand-alone inspection or as part of a larger inspection (e.g., service water system inspection, surveillance inspection, special inspections). A total of 80 hours of direct inspection effort at a plant site is estimated for the staff to complete the procedure. The number of systems and test data selected for review will be the determining factor for the actual length of an inspection.

Generic Communications of Interest in the IST Area: There have been a few generic communications issued that may be somewhat related to, or may impact, IST activities.

NRC Administrative Letter 95-05, "Revisions to Staff Guidance for Implementing NRC Policy on Notices of Enforcement Discretion" IST engineers should be aware that relief requests or authorizations or alternatives are considered the first level to pursue when a code noncompliance is identified that cannot be resolved in the current plant mode (it is assumed that any operability questions are also addressed). An example would be a missed surveillance (either actually missed, inadequate, or not previously identified) for a valve that can be tested only when the plant is in a cold shutdown or refueling outage. If, however, a relief request or an alternative cannot address the particular situation (e.g., time constraints or multiple trains or systems effected), a licensee may need to seek enforcement discretion. Guidance on the NRC's policy on Notices of Enforcement Discretion is given in this administrative letter.

NRC Information Notice 93-83, and Supplement 1, "Potential Loss of Spent Fuel Pool Cooling After a Loss-of-Coolant Accident or a Loss of Offsite Power" This notice discusses a potential loss of cooling to the spent fuel pool. Other recent events have raised the visibility of spent fuel pool cooling issues. As the issues relate to IST, licensees should ensure that any components in the system that function within the scope defined for IST are in the IST program. Similarly, a licensee may have made commitments to include components in the IST program. For most plants, as a minimum, the valves that allow flow from the ultimate heat sink (or service water) are included in the IST program, even if the valves are manual. Even if the components are not within the scope of 10 CFR 50.55a for the IST program, it may be used as an acceptable testing program as discussed in Generic Letter 89-04, Position 11.

NRC Information Notice 95-57, "Risk Impact Study Regarding Maintenance During Low-Power Operation and Shutdown" IST engineers should be aware that taking a single train of a system, in particular the service water system, out of service for testing, even during low-power operation and shutdown, could increase plant risk. If an IST schedule falls within the operational modes of concern, consideration should be given to alternative schedules, and evaluation of the risk impacts

of other systems/components that may be out of service.

NRC Information Notice 96-03, "Main Steam Safety Valve Setpoint Variation as a Result of Thermal Effects" When testing during coastdown at Arkansas Nuclear One, Unit 2, the lift points of several main steam safety valves were found as much as 6 percent above nominal setpoint. Testing at Wyle Laboratories identified that the valve lift varied considerably depending on the ambient temperature during testing. The in-situ tests were performed with the valves in an environment of approximately 35 degrees C (95 degrees F). Testing at Wyle Laboratories was conducted in an insulated box with an ambient temperature of 60 degrees C (140 degrees F). When Wyle Laboratories adjusted the temperature to the in-situ temperature, similar results were observed (i.e., high lift points). OM-1, 1987, requires that the ambient temperature of the operating environment shall be simulated during the set pressure test. Licensees that have not yet adopted OM-1-1987 should be aware that unless a specific temperature is specified, test facilities use a default set of thermal environmental conditions. Wyle Laboratories indicated that specification of test temperatures has not been the past practice of most licensees, but that recently, licensees are beginning to specify the test environmental conditions.

NRC Information Notice 96-15, "Unexpected Plant Performance During Performance of New Surveillance Tests" The NRC has encouraged licensees to test valve position indication at remote panels, though not specifically a requirement in the ASME Code. At the Hatch Plant, a new surveillance was being implemented which included component operation from the remote shutdown panel. A number of problems with the controls from the remote shutdown panel were identified. IST engineers should consider whether it would be advantageous to perform testing from the remote shutdown panel at a selected frequency (e.g., every sixth test) or at a minimum to periodically verify that the position indication for a valve is accurate.

NRC Information Notice 96-17, "Reactor Operation Inconsistent with the Updated Final Safety Analysis Report" Though this notice does not describe problems directly related to IST, IST activities based on incorrect information in a final safety analysis report could be inadequate for monitoring components. Generally, the scope of IST is dependent on the safety analysis and could be incorrect if the safety analysis is not updated properly. A licensee has an advantage if it has created a design basis document for the IST scope.

NRC Information Notice 96-22, "Improper Equipment Settings Due to the Use of Nontemperature-Compensated Test Equipment" The notice included main steam safety valve lift settings as one of the components tested with nontemperature-compensated gauges at the Farley Nuclear Plant. It is possible that test gauges used for pump testing might have the problems discussed in the notice.

Issues: Even with the broad scope of material covered in the IST-related reports, new issues arise periodically as inspections are being performed or as licensees are engaged in a review of their IST programs. Issues that might be of a general interest are discussed below.

Applicability of GL 89-04 Positions to OM Part 6 and Part 10 - NUREG-1482 discusses "current considerations" for each of the positions of the original GL 89-04. Licensees should use the positions for guidance in developing IST programs to the requirements of Part 6 and Part 10 of the *Operations and Maintenance Standards* when no changes to the code have addressed the specific concerns identified in the positions. For example, Position 2 gives guidance on using a sampling program for disassembly and inspection while Part 10 specifies that disassembly and inspection may be used as an alternative to stroking with flow or a mechanical exerciser, but does not address a sampling program. The NRC has allowed a sampling program to continue according to the guidance in Position 2 even when a licensee updates a program to meet the requirements of Part 10.

Similarly, Position 9 discusses pump testing using uninstrumented recirculation lines. Part 6 of the *Operations and Maintenance Standards* does not specifically preclude testing on minimum flow, but does retain requirements to measure flow. Therefore, the guidance in Position 9 would continue to be applicable until the code addresses the concerns (note that the 1994 Addenda and the 1995 Edition of the OM Code includes options for a comprehensive pump testing scheme that would address the concerns identified in Position 9, but the NRC has not yet incorporated the addenda or edition in the regulations).

Examples of GL 89-04 positions which have been addressed by Part 6 and Part 10 of the *Operations and Maintenance Standards* are check valve testing with flow (Position 1), backflow testing of check valves (Position 3), limiting values of power-operated valve stroke times (Position 5), and rapid acting valves (Position 6).

Performing Check Valve Disassembly and Inspection Other Than During Refueling Outages - Possibly due to licensees' efforts to minimize the length of refueling outages, we have received a number of requests to allow disassembly and inspection of check valves during periods other then refueling outages. The requests generally state that a comparable period of time (e.g., 18 months or 24 months) would be used as the test interval. Some of the requests have asked to defer disassembly and inspection of additional valves in the group for a specified period of time (e.g., 30 days) when a problem is identified in the regularly scheduled valve. Deferral of the remaining valves is not acceptable unless the licensee makes a determination that the operability of the remaining valves is not in question. For example, the guidance of Generic Letter 91-18, "Information to Licensees Regarding Two NRC Inspection Manual Sections on Resolution of Degraded and Nonconforming Conditions and on Operability," would be appropriate to consider for this situation.

While the NRC does not encourage licensees to enter limiting conditions for operation to perform an activity such as disassembly and inspection, there may be some cases where the activity poses no increased risk to the plant. The licensee must make the determination that disassembly and inspection of a particular check valve can be safely performed during power operating conditions or during cold shutdowns other than refueling outages. NRC Inspection Manual 9900, "Technical Guidance, Maintenance - Voluntary Entry into Limiting Conditions for Operation Action Statements to Perform Preventative Maintenance," gives guidance to NRC

inspectors regarding inspection of these activities. In addition, 10 CFR 50.65, "Requirements for Monitoring the Effectiveness of Maintenance at Nuclear Power Plants," is now effective (July 10, 1996). Paragraph (a)(3) of Section 50.65 requires that licensees assess the total impact on plant safety before taking plant equipment out of service for monitoring or preventative maintenance.

Testing of Check Valves to Verify Obturator/Disk Movement - Both IWV of Section XI and Part 10 of the *Operations and Maintenance Standards* include requirements to verify disk travel (IWV) or obturator movement (Part 10). When a check valve is tested in both directions or is disassembled and inspected (with disk exercising), there is assurance that disk travel has occurred and all of the requirements for exercising can be considered met. If a valve is tested only in one direction, the testing will not necessarily meet all of the requirements for disk travel. For example, if a design flow test for verifying the opening capability of a valve is the only testing performed, the disk may be dislodged or, for a bonnet-hung disk, may be rotated. We have had actual examples of these conditions occurring. If nonintrusive equipment is used, the flow test may be adequate to show that disk travel occurs. Otherwise, it may be necessary to supplement the flow test with some other technique to verify that there is disk movement (i.e., that the disk is intact and properly oriented). We are considering whether this information needs to be disseminated to licensees through an information notice. This is not a new requirement, but it may be that there are inadequacies in testing.

Leak Testing of Containment Isolation Valves When Using Option B of 10 CFR 50, Appendix J - As published in the *Federal Register* on September 26, 1995 (60 *FR* 49495), a performance-based option was added to Appendix J for local leakage rate testing (LLRT) of containment isolation valves (CIVs) subject to Type C testing. The option allows licensees to extend the testing interval for good performing valves. For plants not yet updated to Part 10, the leakage testing of CIVs is not separated from other valves with a leak-tight function. A few plants have elected to use option B and have addressed the provisions for IST through a request to the NRC for using the requirements of Part 10 of the *Operations and Maintenance Standards*. The NRC mandated only the provisions in Part 10 for "analysis of leakage rates" and "corrective actions" in 10 CFR 50.55a(b)(2)(vii) when incorporating the 1989 Edition of the ASME Code into the regulations. The requirements for the leakage testing interval were not imposed, and, thereby, the schedule of Appendix J provisions would apply. For those valves which are verified closed by leakage testing during LLRT (generally check valves), the extended interval is not acceptable because the closure test is already deferred from quarterly; however, licensees may consider the provisions for condition monitoring of check valves, along with risk assessment reviews of the particular valves, and possibly justify an extension of the testing interval so that the same interval for LLRT and IST can be used for any leakage testing of the CIVs (note that conditioning monitoring could be applied to *all* check valves in the IST program - not just the CIVs).

Implementing the Safety Relief Valve Testing Frequency Per OM-1 - An inquiry was submitted to the OM Committee asking how

to implement the test schedule for safety relief valves. The schedule specified by OM-1 is based on minimum percentages specified in months, with Class 1 valves tested on a five-year schedule and Class 2 valves tested on a ten-year schedule (*Note:* PWR main steam safety valves are tested at the same schedule as Class 1 valves). When transitioning from Section XI to OM-1, there are two groups of valves to consider: (1) valves that were tested in the previous interval, and (2) valves that are being added to the program that were not previously within the scope of the IST program. For valves that were tested in the previous interval, there are two schedules that must be met: (1) the minimum percentages specified in months from the beginning of the interval, and (2) testing within five or ten years from the previous test, as a minimum. For valves that are added to the program, testing should be scheduled from the interval start date. It is not necessary to immediately test all of the valves that were not previously tested, but all of these valves must be tested within either 5 years or 10 years, depending on the class of valves. When a group of valves is small, testing of valves that have already been tested within either 5 or 10 years may occur in order to meet the minimum percentages specified in months.

Conclusion

The IST process could be streamlined to be more efficient and more consistent from one plant to the next, as could be many activities that are managed at a "program" level. Guidance, symposia, and code meetings are useful in that the more information available to licensees, the better informed their decisions can be. For example, where one plant may be conducting testing that is difficult, another may have requested an alternative that accomplishes the intent for monitoring a component, but is less difficult to implement. The licensing bases of each of the plants are often different enough that IST programs are customized for each unit (or for two similar units at a single site) and the scope may be different from one plant to the next based on the time of construction. Moving toward a more uniform system, such as risk-informed IST programs, could allow resources to be better focused and could eliminate much of the inconsistencies in the scope of the IST program and in the number of relief or alternative requests from one plant to another.

NRC guidance for the use of risk-informed techniques will improve consistency for development of new IST programs. The guidance in the recent NUREG reports should improve efficiency in the development of the programs and in the review and inspection process. Certain changes to the regulations that are being considered can improve the efficiency of the programs for the remaining life of the plants in that relief requests and requests for alternatives would remain applicable from one interval to the next unless there are technological or safety changes that are backfit on licensees. If the regulations are changed to provide for a voluntary updating process, licensees will be given more flexibility in the process, yet safety will be maintained at an acceptable level.

Nuclear Power Plant Safety Related Pump Issues

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ABSTRACT

This paper summarizes of a number of pump issues raised since the Third NRC/ASME Symposium on Valve and Pump Testing in 1994. General issues discussed include revision of NRC Inspection Procedure 73756, issuance of NRC Information Notice 95-08 on ultrasonic flow meter uncertainties, relief requests for tests that are determined by the licensee to be impractical, and items in the ASME OM-1995 Code, Subsection ISTB, for pumps. The paper also discusses current pump vibration issues encountered in relief requests and plant inspections which include smooth running pumps, absolute vibration limits, and vertical centrifugal pump vibration measurement requirements. Two pump scope issues involving boiling water reactor waterleg and reactor core isolation cooling pumps are also discussed. Where appropriate, NRC guidance is discussed.

INTRODUCTION

The staff has encountered a number of pump related issues since the Third NRC/ASME Symposium on Valve and Pump Testing in 1994. This paper discusses pump issues involving NRC inspection procedures, information notices, relief request evaluations, ASME Code revisions, vibration issues, and scope issues. Some of the issues discussed include current staff positions and actions in these areas.

GENERAL ISSUES

NRC Inspection Procedure 73756

NRC Inspection Procedure (IP) 73756, "Inservice Testing of Pumps and Valves," was revised on July 27, 1995, to incorporate elements of NRC Temporary Instruction (TI) 2515/114, "Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs," issued on January 15, 1992, and TI 2515/110, "Performance of Safety-Related Check Valves," issued on November 19, 1991. In addition, inspection guidance was added to the IP to evaluate licensee actions taken in response to NRC Bulletin 88-04, "Safety Related Pump Loss." The previous revision of IP 73756 was issued on March 16, 1987. Three inspections in Region I have been performed using IP 73756. Findings from these inspections related to pumps have been included in the paper titled "Summary of Inspection Findings of Licensee Inservice Testing Programs at United States Commercial Nuclear Power Plants" presented in Session 2(c) of this symposium.

Ultrasonic Flow Meter Uncertainties

NRC Information Notice 95-08, "Inaccurate Data Obtained With Clamp-On Ultrasonic Flow Measurement Instruments," described a case where technical specification (TS) system flow rate requirements were not met due to measurement inaccuracies using an ultrasonic flow meter in two successive refueling outage

design basis tests. Subsequent to the TS test failure, the licensee noted flow variations between plant process instrumentation and the ultrasonic flow instrumentation when the system was reconfigured. Subsequent testing by the manufacturer at a national laboratory noted that flow data varied by 5% at different locations around a pipe at a position approximately 15 pipe diameters downstream from a single bend in the test loop.

The licensee concluded that the inaccuracy was most likely caused by flow effects associated with the piping configuration. Other factors can also influence the accuracy of ultrasonic flow instrumentation including pipe dimensions, fluid flow effects, transducer mounting, and instrument setup and operation. These same factors can also affect repeatability of the instrument. The licensee noted that the instrument was not experiencing unusual flow fluctuations that could have been an indication of a problem with the instrument accuracy. In the past, when authorizing the use of ultrasonic flow elements with accuracy requirements outside the Code requirements, the staff has cautioned licensees regarding the correct use of the instruments and on assurance of repeatability.

Relief Requested Under Impracticality

Section 6.3 of NUREG-1482 addresses cases where relief requests submitted on the basis of impracticality may result in a period of noncompliance. Licensees submitting IST relief requests based on impracticality under 10 CFR 50.55a(f)(6)(i) need to ensure that the Code testing is in fact "impractical" to perform rather than being an "alternative" to Code requirements that could be met. Below is an example where pump testing identified by the licensee as impractical was, in fact, practical to perform. This particular relief request was implemented without prior NRC approval.

A relief request had been submitted stating it was impractical to individually test each pump in a parallel two-pump train (two trains in the system, four total pumps) because a control valve in each train of the system was not capable of supporting single-pump operation without experiencing extreme valve cavitation. The licensee's TS required both pumps in both trains to be operable. The licensee had proposed to test both pumps in each train simultaneously using more stringent hydraulic acceptance criteria. However, the reviewer noted that the licensee's TS allowed one pump to be inoperable in each train up to 30 days while the plant remained at power before requiring a plant shutdown. Further discussions with the licensee revealed that two of the pumps had been tested individually in preoperational testing without control valve cavitation problems. A subsequent test by the licensee confirmed the previous single-pump test results. The licensee then withdrew the relief request and stated that these pumps would be tested in accordance with the Code requirements.

Licensees should be sensitive to the issue of implementing requests prior to NRC approval and be cognizant of the difference between "impracticalities" based on design limitations and "alternatives" to the Code requirements based on other reasons such as a different method of testing or a hardship (but not impracticality) in performing the required Code testing.

Acceptance Criteria Analysis

With the approval of the ASME OM-1995 Code, a few licensees have proposed an alternative to use the acceptance criteria action

range requirements of ISTB 6.2.2 in lieu of the requirements specified for pumps operating in the required action range in OM-6, Section 6.1. The 1995 Code states that a pump shall be declared inoperable when it falls within the required action range until either the cause of the deviation is determined and corrective action completed or *an analysis of the pump is performed and new reference values established in accordance with ISTB 4.6* (author's emphasis). The staff would expect that the analysis would include the cause of the degradation, the safety margin of the pump, and the basis for determining that further degradation will not occur before the next test or that the pump will not fail before repairs can be performed.

This section of the OM-1995 Code was added, in part, to include the guidance essentially already provided in NRC Generic Letter (GL) 91-18. Also, similar requirements were in earlier editions of Section XI of the ASME Code. Licensees can perform the analysis allowed in ISTB 6.2.2 under current NRC guidance (i.e., GL 91-18). However, licensees should use this analysis cautiously as it is not intended to be used regularly to evaluate the operability of all pumps that fall into the required action range in order to declare the pump operable and define new reference values while obvious degradation is occurring. The use of this analysis is expected to be a rare occurrence. Repeated application of analysis could lead to stair stepping the Code alert and required action range limits downward to the safety limit of the pump. The licensee should have an understanding of the margin of each safety-related pump above its design-basis hydraulic requirements. The analysis, which should include detailed justification and any change in the pump reference values, must be documented in accordance with the Code requirements. In addition, it is not foreseen that this section in the Code could be used for pump bearing vibration readings, as there are no defined safety margins related to pump vibration.

Alternative requests have been granted to use OM-1995, Paragraph ISTB 6.2.2. Paragraph ISTB 4.6 is a related requirement and must be applied in conjunction with ISTB 6.2.2 (i.e., an analysis is acceptable under 6.2.2 only when new reference values are assigned prior to declaring the pump operable).

Code Design Basis Testing Requirements

The main purpose of pump inservice testing is to detect the onset of either mechanical or hydraulic degradation from an established reference condition. The quarterly frequency test is not a design basis test. The Code does not require pump IST to be performed at the pump design flow rate or pressure. Generic Letter 89-04, Position 9, allows licensees to perform pump testing at full or substantial flow conditions during cold shutdowns or refueling outages if there is no means to measure flow rate through pump recirculation lines during quarterly testing. Unless relief has been granted, the licensee still must measure and record pump differential pressure and vibration quarterly.

The 1995 Edition of the ASME OM Code, Subsection ISTB, requires pump testing at an established reference point within ±20% of the design flow rate at a minimum of once every two years.. While licensees may perceive this testing as more difficult to implement, especially for pumps currently tested using GL 89-04, Position 9, the quarterly testing requirements may be less stringent. At least one licensee has expressed an interest in adopting the OM-1995 ISTB

pump testing methodology. To implement, licensees must propose an alternative to use the OM-1995 pump testing requirements for all, or a group of, pumps in their IST program. All related requirements must be incorporated by the licensee into their IST program.

VIBRATION ISSUES

"Smooth-Running Pumps"

Pumps which have vibration reference values below 0.1 inches/second are referred to informally as "smooth-running pumps." Concerns had been raised that application of the vibration reference value multipliers (required by OM-6 and ISTB) to these pumps could result in the Code corrective action requirements being imposed on pumps that were in fact operating very smoothly. The ASME OM Committee, Working Group on Pumps, evaluated whether a change to the Code to address this issue was warranted. The working group and associated Task Group on Pump Vibration had spent considerable time attempting to develop acceptable code changes. A proposal was sent to the Subcommittee on Mechanical Equipment & Systems in December of 1994, but based on discussions in the subcommittee meetings, was sent back to the working group to develop a more understandable proposal with better consensus.

Subsequent to the Code subcommittee action, a safety-related pump at a plant with an approved smooth-running pump alternative experienced a bearing failure that was not detected by inservice testing but was evidenced through the plant predictive maintenance program. The periodic monitoring had noted an increasing upward trend in vibration that was below the alert range allowed by the alternative request. This finding was brought to the attention of the Code Committee. The task group, with concurrence from the working group, decided to make this issue "inactive" based on a lack of need for this change and the additional resources potentially needed to further investigate this issue to form a consensus.

A few general alternative requests have been granted by the NRC to establish a minimum vibration reference value and set alert and required action range values based on a minimum reference value for pumps classified in a licensee's IST program as smooth running. The plant that experienced a pump failure was implementing an approved alternative. If licensees intend to submit alternative requests to use minimum reference values, the requests should be pump specific and include justification as to how the current inservice testing methodology will detect pump degradation.

Code Absolute Vibration Limits

One major change incorporated in the Code vibration requirements with the adoption of OM-6 was the inclusion of *absolute* alert and required action limits. When some licensees were in the process of updating to the new requirements, they discovered that certain pumps did not meet the Code absolute limits. A number of alternative requests have been received to adjust the vibration absolute alert and required action range acceptance criteria based on the claim that the normal operating level of some pumps exceeded the Code absolute limits even though the pumps were operating acceptably under the previous program requirements.

Alternative requests that did not provide specific information on the pump vibration

history and efforts to improve performance, if appropriate, have generally not been approved. However, a number of alternative requests have been granted to raise the alert limit. In addition, one relief request was granted to raise the required action limit when the pump was tested using the recirculation loop (the Code alert and required action range absolute limits were used when the pump was tested at full-flow conditions). The alternatives specified, either as requested by the licensee or authorized provisionally by the NRC, that the alternative only applied to the specific direction (horizontal, vertical, axial) in which the vibration levels had routinely exceeded the absolute limit. Acceptance criteria for the other vibration directions would be in accordance with the Code requirements. Several licensees were able to demonstrate, through the use of historical pump vibration data, that a specific pump did not exhibit degrading trends and current vibration levels were not damaging the pump.

There are four key components that the staff considers in evaluating these particular alternative requests. The licensee should have sufficient vibration history from inservice testing which verifies that the pump has operated at this vibration level for a significant amount of time. "Spikes" in the test data, if present, should be justified. Second, the licensee should have consulted with the pump manufacturer (or a vibration expert which may be on the licensee's staff) about the level of vibration the pump is experiencing to determine if the operation of the pump is acceptable. Third, the licensee should describe attempts to lower the vibration below the defined Code absolute levels through modifications to the pump. This could include such action as changing impellers, stiffening the pump base plate, or improved balancing. Any pump changes to lower the vibration levels should be discussed in the alternative request. Fourth, the licensee should perform a spectral analysis of the pump-driver system to identify all contributors to the vibration levels. Information on vibration history and spectral plots may be included in the alternative requests. The inclusion of this information in the alternative request, although not assuring NRC authorization, will streamline the review process.

Vertical Centrifugal Pump Vibration Requirements

OM-6, Section 4.6.4(a), requires that vibration measurement for centrifugal pumps be taken in the axial direction of each pump thrust bearing. In some types of large vertical centrifugal pumps (i.e., boiling water reactor residual heat removal pumps), the thrust bearings are located in the driver. OM-6, Section 1.2, allows exclusion of drivers from the Code requirements except where the pump and driver form an integral unit and the bearings are in the driver. Since the pump itself has no thrust bearings, and the thrust load is carried by the driver bearing, the driver is considered an integral part of the pump. During two recent IST inspections, it was discovered that both licensees did not consider this motor bearing axial vibration point to be within the scope of the Code. The inspectors concluded that this vibration point is within the scope of the Code requirements and this position was reinforced by the NRC representative on the ASME OM Committee Working Group on Pumps. The licensees independently stated at the time of the inspections that a Code inquiry would be submitted to address this issue. As of this writing, no inquiry is known to be submitted to the OM Committee.

PUMP SCOPE ISSUES

Waterleg Pumps

NRC Region IV requested the Office of Nuclear Reactor Regulation (NRR) to evaluate whether waterleg pumps at a particular boiling water reactor (BWR) plant should be included within the scope of the plant's IST program. Waterleg pumps (also refered to as keep-fill, line-fill, holding, jockey, stay-fill, or safeguards pipe-fill pumps) are used to pressurize the discharge lines of many BWR emergency core cooling systems (ECCS) to keep them filled with water. Several BWR plants' final safety analysis reports (FSARs) state that the coolant be delivered to the reactor rapidly when a particular ECCS system is called upon to function. In addition, these reports cite the potential physical damage that could occur from large momentum forces generated by water moving through empty ECCS discharge lines. Therefore, many BWR TS require that ECCS discharge lines be filled with water and specify surveillance requirements to verify the filled condition. After initiation of ECCS pump flow, the associated waterleg pump generally does not have any other function. Some BWR designs do not have waterleg pumps but employ other systems to maintain the discharge lines full of water. Based on a review of BWR TS, FSARs, and IST programs, it appears that licensees are not consistent in the determination of the status of waterleg pumps in their IST programs.

The NRR evaluation concluded that maintaining the ECCS lines full of water in order to facilitate a quick injection of water into the reactor vessel is a function that mitigates the consequences of an accident. For a pump to be included in the IST program, it must be connected to an emergency power source. A waterleg pump must be included in the IST program if it is a Code Class (or equivalent) pump, designated as a safety-related component in the FSAR, and supplied with emergency power. When no specific designation exists, then the inclusion in the IST program should be based on whether there are other means to maintain the discharge line full of water. The waterleg pumps should be included in the licensee's IST program if they are the only means available to maintain the ECCS lines full of water during plant operation. If there are other safety-related components that are credited to maintain the discharge lines filled with water, and the waterleg pumps are not credited in the plant's accident analysis, inclusion of the waterleg pumps in the IST program is at the discretion of the licensee.

Reactor Core Isolation Cooling Pumps

The pump section of one recent plant ten-year IST program submittal stated that the reactor core isolation cooling (RCIC) system did not fall within the scope requirements of the ASME OM Standards, Part 6, as implemented by 10 CFR 50.55a (i.e., not ASME Class 1, 2, or 3), was not covered by the regulatory position of Regulatory Guide 1.26, and was not designed to facilitate performance of ASME Code pump testing. The staff reviewed the plant FSAR and TS and concluded that the RCIC system meets the requirements of Regulatory Guide 1.26, and its components are classified as Quality Group Classification A and B which correspond to ASME Safety Class 1 and 2, respectively, and are capable of being tested in all modes of plant operation. For this facility, the staff believes that the RCIC system is within the licensing basis of the plant and that the specific components (i.e., pumps and valves) have a required safety function to bring the

reactor to the cold shutdown condition as specified in the scope requirements of the ASME Code and 10 CFR 50.55a.

A review of other BWR IST programs was conducted to determine the extent of RCIC pump inclusion in their IST programs. Of the 22 BWR sites that have a RCIC system, 16 include the pumps in their IST program. Only two sites exclude the RCIC pumps from their IST program (current IST programs were not available for four sites). In addition, the RCIC pumps included in plant IST programs appeared to be subject to the Code requirements. One licensee had recently gained approval to change the classification of certain functions of their RCIC system from safety-related to quality-related based on, in part, a commitment by the licensee to retain the RCIC system in its IST program.

No general conclusions can be drawn from these findings. The inclusion of RCIC pumps in plant IST programs is based on the plant FSAR, any TS surveillance requirements referring to IST for these pumps, and any commitments made to the NRC during or subsequent to plant licensing. Plants that currently exclude the RCIC pump from their IST program should examine their justification for this position and either revise their program to include these pumps or maintain documentation on site for inspection justifying the exclusion. The staff is continuing to address RCIC system scope issues on a plant-specific basis.

CONCLUSION

The purpose of this paper has been to make licensees aware of a number of pump issues that the staff has encountered since the Third NRC/ASME Symposium on Valve and Pump Testing in 1994. Licensees who believe that some of the items discussed are applicable to their facility may wish to review their current IST program, consider the discussion in this paper, and modify their program as appropriate.

REFERENCES

NRC Generic Letter 89-04, "Acceptable Inservice Testing Programs" and associated "Minutes of the Public Meeting on Generic Letter 89-04," issued April 3, 1989.

NRC Generic Letter 91-18, "Information to Licensees Regarding Two NRC Inspection Manual Sections on Resolution of Degraded and Nonconforming Conditions and on Operability," issued November 7, 1991.

NRC Inspection Procedure 73756, "Inservice Testing of Pumps and Valves," revised July 27, 1995.

NRC Temporary Instruction 2515/114, "Inspection Requirements for Generic Letter 89-04, Acceptable Inservice Testing Programs," issued January 15, 1992.

NRC Temporary Instruction 2515/110, "Performance of Safety-Related Check Valves," issued November 19, 1991.

NRC Information Notice 95-08, "Inaccurate Data Obtained With Clamp-On Ultrasonic Flow Measurement Instruments," issued January 30, 1995.

NRC Bulletin 88-04, "Safety Related Pump Loss," issued May 5, 1988.

NUREG-1482, "Guidelines for Inservice Testing at Nuclear Power Plants," issued April 1995.

ASME/ANSI Operations and Maintenance Standard Part 6 (OM-6), "Inservice Testing of Pumps in Light-Water Reactor Power Plants," 1988 edition.

ASME OM-1995, "Code For Operation and Maintenance of Nuclear Power Plants," Subsection ISTB, issued February 28, 1995.

Title 10 of the Code of Federal Regulations, Section 50.55a, "Codes and Standards," 1995 revision.

RECENT PERFORMANCE EXPERIENCE WITH U.S. LIGHT WATER REACTOR SELF-ACTUATING SAFETY AND RELIEF VALVES

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INTRODUCTION

Over the past several years, there have been a number of operating reactor events involving performance of primary and secondary safety and relief valves in U.S. Light Water Reactors. There are several different types of safety and relief valves installed for overpressure protection of various safety systems throughout a typical nuclear power plant. The following discussion is limited to those valves in the reactor coolant systems (RCS) and main steam systems of pressurized water reactors (PWR) and in the RCS of boiling water reactors (BWR), all of which are self-actuating having a setpoint controlled by a spring-loaded disk acting against system fluid pressure. The following discussion relates some of the significant recent experience involving operating reactor events or various testing data. Some of the more unusual and interesting operating events or test data involving some of these designs are included, in addition to some involving a number of similar events and those which have generic applicability.

DISCUSSION

Thermal Effects

There have been several interesting events involving temperature change phenomena for both PWR pressurizer and main steam safety valves (PSVs and MSSVs). There are thought to be several effects which can change the valve setpoints when changes in valve temperature occur, either on a test stand or as installed on the operating system. One of the identified effects involves differential thermal expansion between individual parts of any given valve. Differential vertical (or axial) expansion between the valve body and nozzle can affect the amount of compressive force between the valve disk and its seat simply by pushing the two together or relaxing some of the compression, depending on whether the nozzle grows more than the body or vice versa. Similarly, differences in thermal growth between the valve setpoint spring and the supporting bonnet can cause changes in the setpoint. Another effect is that as a valve heats up, the nozzle and disk diameters expand which creates a larger disk area over which system fluid pressure acts. This can cause the valve setpoint to drop due to the resulting greater steam pressure force on the valve disk.

At one operating plant event where a PSV opened with a low setpoint it was found that the valves had been misadjusted because thermal effects had not been accounted for during testing. At this plant which has carbon steel PSV bodies with stainless steel nozzles, the differential thermal growth between the body and nozzle caused a maximum of 6% difference in the setpoint. The setpoint increased as the valve heated up, but the increase was found to be somewhat transient, occurring mostly while the valve was heating up, and once thermal equilibrium was achieved, the setpoint came back down. This plant is unique in having carbon steel PSV bodies, and tests performed for other stainless steel models do not indicate as much

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

difference in setpoint with changes in temperature. Further information regarding these circumstances can be found in NRC Information Notice 93-02.

In another set of circumstances, which is discussed in NRC IN 96-03, MSSVs were found to be misadjusted due to not properly simulating the operating ambient thermal environment around the outside of the valves. The test facility where the valves had been set had insulated the valves in such a way as to increase the valve temperature from 95 degrees F (which is the normal operational environment) to 140 degrees F in the bonnet area. The effect of this increased temperature caused the setpoints to lower by up to 6%, which in turn means that they were adjusted higher. When installed in the plant at the lower temperature environment, they were 6% low.

An interesting phenomenon related to the above thermal difference effects is that of the effect of subcooled loop seal liquid upstream of some PSV configurations. The purpose of the loop seal design is to provide greater seat tightness since the denser, more viscous liquid water has more resistance to leaking past the seat than does steam or other noncondensible gases. While, it might be supposed that the setpoint would vary greatest for such a configuration, a significant amount of comparative test data demonstrates that the expected difference between cold loop seal conditions (about 100 degrees F) and hot steam conditions (about 650 degrees F) is not more than about 1%. This conclusion was reached by very carefully and precisely measuring the disk lift at incipient disk opening. If the lift is not precisely measured, a mistaken conclusion may be reached regarding the setpoint value since the relatively viscous water against the seat will not cause the disk to "pop" open as one expects for steam inlet conditions. If the test pressure is increased without making a very precise lift measurement, the increased pressure could be mistakenly interpreted as meaning that the setpoint is too high. If the valve setpoint is then adjusted to a lower value to compensate for this observation, the valve could open early during an overpressure event. For this reason there is general agreement in the industry (including the ASME OM Committees) that valves installed with loop seals should be setpoint tested with steam thus avoiding the difficulty in setting the valves. (It should also be noted that the dynamic nature of the loop seal discharge and the characteristic time delay before the valve can fully open can result in higher peak system pressures which should be evaluated.) Further information on specific events related to PSV setpoints for loop seal installations can be obtained in NRC IN 89-90 and Supplements 1 and 2.

It should be noted that, in Appendix I of the OM Code, the O&M Committee has recognized the variation of valve set pressure with temperture of both the fluid being relieved and the ambient temperature of the valve itself. Paragraphs I 4.1.1(d) and (e) and I 8.1.1(d) and (e) require that the relief valve operating environment be simulated and the valve be at thermal equilibrium before set pressure testing.

Corrosion Bonding and Mechanical Sticking

There have been numerous events involving corrosion bonding and mechanical sticking of safety valve disks to their seats. Most notable are several events in BWRs where pilot valves (which control opening of the main valve disks) have exhibited corrosion bonding. These pilot disks and their seats are composed

of cobalt alloy metals (i.e., Stellite 6 and 6B) which are fairly noble to corrosion in most other reactor water or steam applications. However, in this specific application in BWR safety/relief valves (S/RVs) being situated on top of the main steam lines in a slightly subcooled temperature, there has been significant corrosion and accompanying mechanical bonding or sticking of the pilot disk conical wedge seating area. The specific corrosion mechanism is believed to be fairly well understood and involves a combination of both unique configuration and environment. Inside the valve in the pilot disk area, an initial mixture of steam and small amounts of radiolytically produced oxygen and hydrogen collect. The saturated BWR steam inside the valve is then continually condensed along the inside surfaces of the valve body with the resulting condensate draining back to the steam lines. The replenishing steam is also condensed, but the additional noncondensible gases stay inside the valve. This allows the noncondensible oxygen and hydrogen gases to concentrate until the valve internals eventually contain almost pure oxygen and hydrogen. This is an extremely corrosive environment and occurs within a fairly short time after the reactor is operating. The resulting corrosion and bonding/sticking in the conical pilot seat area has been significant, and in some cases has actually caused the disk to stick well beyond the capability of a testing facility to determine the actual setpoint. There are two different modifications which have been implemented at some BWR plants for resolving the high setpoint concern with these S/RVs. One modification is to place a catalyst material inside the valves in the pilot disk area to recombine the oxygen and hydrogen back into water. Trial operating periods using this method have shown generally favorable results, but this method is still being evaluated for its effectiveness. The other method is to install additional pressure sensing and actuation controls to actuate the S/RVs with external power. The staff recently approved a topical report for this approach and it has already been implemented at a few plants. Of note, corrosion bonding or mechanical sticking of direct-acting spring-actuated BWR S/RVs (i.e., having no pilot valve) has not been significant or extensive so far as the NRC staff is aware. It is believed that there is good turbulent fluid mixing inside these valves such that noncondensible gases do not concentrate, or if they do, their effect is not significant against the pressure actuating forces on these large valve disks.

Generally, with a very few exceptions, there has not been significant internals corrosion or mechanical bonding in PWR PSVs or MSSVs; however, recent testing at one plant has indicated that there may be some type of mechanical bonding between the valve disk and seat. In-place testing of the plant MSSVs with a lift assist device revealed that after a plant cooldown and subsequent heatup, the valve setpoint would drift high by several percent in some cases. It has been postulated that there could be a mechanical cold welding/galling phenomenon caused by differential thermal expansion, during the cooldown and heatup cycle, in the radial direction between the disk and its mating nozzle seating surface. This specific experience is contrary to general experience at other plants and could be unique to circumstances at that plant. However, the NRC staff is interested in any generic implications which these results may have. This licensee is continuing to evaluate the phenomena which may be involved by pursuing laboratory testing of the disk and seat materials, and by augmented testing of the MSSVs in service more frequently than that required by the ASME Code.

Test Equipment Calibration and Accuracy

In general, plant licensees and off-site test facilities have not experienced a large number of occasions where test equipment was not adequately calibrated which caused safety and relief valve setpoints to be inaccurate. The quality assurance procedures required for the associated equipment are usually sufficient to prevent errors due to miscalibration. However, experience has shown that it is important to completely evaluate all parameters which could affect the end result. For example, there are several different devices in use which allow safety valves (mostly MSSVs) to be tested in place at normal operating system pressures without the need to remove them from the system and ship them to a test facility. While performing comparative testing of MSSVs at one plant, it became apparent that the results of setpoint testing using the in-place device was consistently different from results obtained from an off-site facility. It was then discovered that by not accounting for the exact disk area which the system pressure acts against in the setpoint equation used to compute the setpoint from the test data, an error was introduced. The setpoint equation consists of two terms: one which is the actual system pressure during the testing and the other involving the equivalent additional pressure provided by the lifting device itself. This second term requires the seat area as a factor. The testing device vendor had used the mean seat area in the setpoint equation, which was somewhat different from the actual value which was later determined experimentally. In this case, the correct calibration of the testing device depended not only on the ability of laboratory personnel to set gauges and follow procedures correctly, but also on an aspect more of an engineering or analytical nature. This issue is further discussed in NRC IN 94-56.

The use of nontemperature-compensated test equipment has also resulted in errors in safety valve setpoint determinations. As an example, for one type of pressure test gauge, it was discovered that environmental temperature variances can cause up to a 3 psi error for each 5 degrees F change from the reference temperature of 70 degrees F. This is a very significant error which could lead to misadjustment of valve setpoints at operating temperature. This issue is further discussed in NRC IN 96-22.

CONCLUSION

The failure of primary and secondary safety and relief valves to meet plant Technical Specification setpoint requirements or values assumed in plant design-basis analyses can have some safety significance, depending on its severity and the system operating and structural integrity margins. Review of numerous analyses of limiting design-basis events indicates there is some margin to compensate for some out-of-tolerance setpoint conditions; however, there continues to be a significant number of events where the ability of these valves to perform necessary safety functions has been significantly reduced.

Safety and relief valves are relatively simple devices when compared to many other power plant components requiring many more moving parts, external energy supplies, and controls. Their basic design has, for the most part, remained unchanged for perhaps 100 years or more. Their ability to relieve excessive system pressure, when they perform well, has been demonstrated in numerous full scale tests and actual operating events.

However, as outlined by the occurrences discussed above, they have had a fair share of attention as components which have performed poorly in certain circumstances. The NRC staff believes that continued development of improvements to existing valve designs, research into new designs, and the factoring of operating experience into maintenance and testing standards are all necessary in assuring adequate safety and relief valve performance in the future.

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10. SUPPLEMENTARY NOTES

11. ABSTRACT *(200 words or less)*

The 1996 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.

12. KEY WORDS/DESCRIPTORS *(List words or phrases that will assist researchers in locating the report.)*

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