NUREG/CP–0123
Supplement 1

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

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

Sponsored by
U.S. Nuclear Regulatory Commission

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

Proceedings prepared by
U.S. Nuclear Regulatory Commission

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ABSTRACT

This supplement provides revisions to papers included in NUREG/CP-0123, and the papers which were not included in the initial publication of NUREG/CP-0123. The 1992 Symposium on Pump and Valve Testing, jointly sponsored by the Board on Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the Nuclear Regulatory Commission, provided 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 provided an opportunity to discuss the need to improve 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 or ideas and perspectives regarding the improvement of inservice testing of pumps and valves at nuclear power plants.

DISCLAIMER AND EDITORIAL COMMENT

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

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

Errata:
Corrections to NUREG/CP-0123

Using Torque Switch Settings and Spring Pack Characteristics to Determine Actuator Output Torques

Bill R. Black, TU Electric

Paper begins at page 321

Correct equations on page 330, left column as follows:

$$MARM = (-0.46\ ft/in^2)(SPD)^2 + (0.226\ ft/in)(SPD) + 0.148\ ft$$

Errata:
Corrections to NUREG/CP-0123

Comanche Peak Check Valve Reliability Program

Diane Stewart, TU Electric

Paper begins at page 437

1. *Page 437 - Right-hand column, add "Extraction Steam" to list of systems.*

2. *Page 439 - Left-hand column, change "(Figure 1)" to "(Figure 1-A)", and see new Figure 1-A (next page).*

3. *Page 441 - Right-hand column, change "(see Table 3)" to "Test Frequency (Table 3)." Table 3 is revised as follows:*

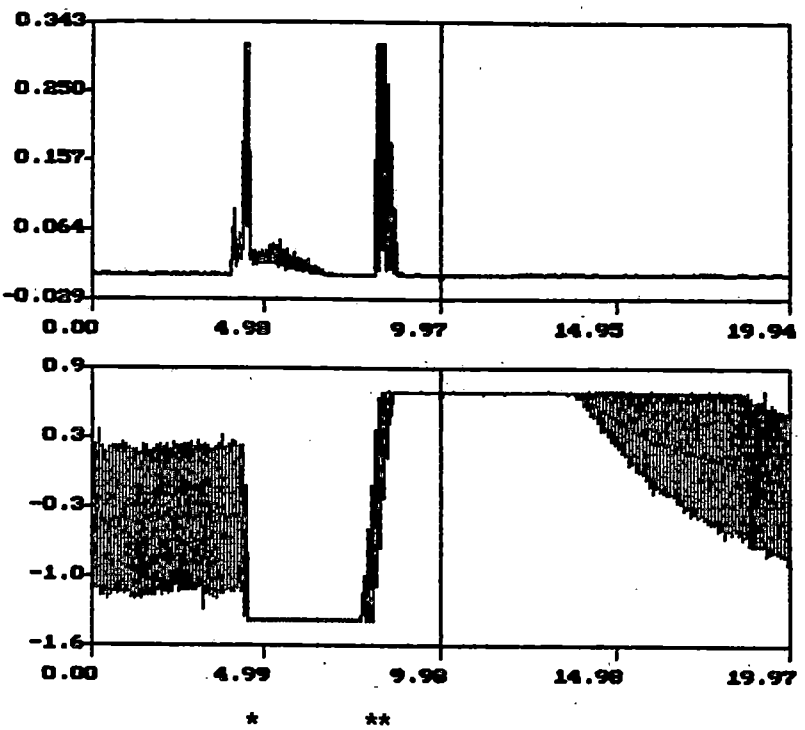
Table 3. *AE test frequency per program priority categories*

Program Priority	AE test frequency Minimum Number of refueling cycles
A	1
B	2
C	4

4. *Page 452 - Right-hand column* ***1RF01 RESULTS****, change to read as follows:*

   Outage activities included one main feedwater pump discharge check valve which is a 20-inch Crane tilt disc, Model 20-7109. There is one associated with each main feedwater pump.

Errata: Corrections to NUREG/CP-0123
FIGURE 1-A



VIEW: [redacted]

PLANT: OPEES
UNIT: 1
TAG: 1DO-0065
DATE: 09/05/91
TIME: 14:56:47
TEST: 10

	PLOT 1	PLOT 2
NAME	EA12A	UN13A
RESOL	1: 50	1: 998
X-axis	seconds	seconds
Y-axis	g's	volts
X	9.9690	9.9840
Y	0.002	0.688
REF X		
REF Y		
Δ X		
Δ Y		

EVENT	TIME
1	
2	
3	
4	
5	
6	
7	
8	
9	
10	

* Valve Closure Impact

valve closure coincides with repressurizer valve opening in out-of-service train (and corresponding pressure drop in in-service train)

** Valve Open Impact

Plot 1: AE, envelope (10g, ± 2V) 1DO-0065

Plot 2: pressure (from tap off Air Dryer) (UN download, ± 2V)

Steady State Impacting in a DG Air Start Check Valve (1-1/2" Rockwell Piston): Simultaneous Use of Two Non-Intrusive Test Methods

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The ASME Codes and Standards Development Process

Mark E. Sheehan, ASME
Director, Nuclear Codes and Standards

INTRODUCTION

The ASME Codes and Standards Department provides a vital service to various industries, only one of which is the nuclear power industry. How do the ASME standards developing committees operate, and what are some of the forces that steer us in the standards writing business?

Make no mistake, standards writing is a business, with pitfalls and obstacles like any other. One of the obstacles is basic, but subtle. Given that standards evolve from successful experience over time, how can ASME make accurate, timely responses to advances in an industry as dynamic as nuclear power?

Standards cannot be allowed to become so definite and inflexible that their revision becomes a difficult and time consuming process. If the standards writers are not responsive to changes in the industry, then the standards could actually obstruct progress.

To ensure quality standards, there is an involved process within ASME that must be followed for every new standard and each revision developed. Hundreds of man-hours of volunteers in industry, academia, government and the general public are put into the writing of a standard. These volunteers come from all around the country, as well as Canada and in some cases other foreign countries.

ORGANIZATION OF COMMITTEES

A 20-member Council on Codes and Standards supervises the Codes, Standards, accreditation and certification activities of the Society. This council is one of five reporting to the Board of Governors. Five supervisory boards and five advisory boards support and advise this Council. Together they organize the administrative tasks and advise the Codes and Standards staff at the New York ASME Office. This staff includes more than 40 engineers who are technically qualified to support committee activities and provide information on ASME Codes and Standards.

More than 100 Main Committees report to the supervisory Boards. These committees are balanced with all categories of interest that relate to the subject represented. These categories of interest can include manufacturers, inspectors, regulators, material suppliers, consultants, users and insurance interests.

Opening Remarks

Each Main Committee has a number of technical committees, working groups and task forces, charged with developing, maintaining or updating the Codes and Standards within their areas of technical expertise. Nearly 4000 volunteers currently participate in the ASME Codes and Standards activity.

CODES AND STANDARDS DEVELOPMENT

The path a Code or Standard must follow from identification of need to publication is a process of various steps as follows:

- **Determination of Need for a Code/Standard**

  The Council on C&S decides whether ASME should undertake work on a new standard. It then assigns the project to the appropriate supervisory board. This board assigns it to an existing Main Committee or recommends establishment of a new Main Committee.

- **Initial Development Process**

  The Main Committee examines the project in detail, determines the limitations of its scope, the concepts to be included and whether more than one standard is needed. It then establishes technical subcommittees and working groups each taking a part of the proposed standard to develop.

- **Draft Code/Standard**

  The proposed sections go through many drafts with the subcommittees or working groups voting on each draft. A record is kept of the votes and the Main Committee is kept up to date through progress reports.

- **Review**

  A final draft, called a proposed standard is then sent to the Main Committee, *which represents all interested parties*, for a vote by letter ballot.

  If approved, it is then sent to the appropriate supervisory board to assure that the ASME procedures for Codes and Standards Writing committees have been followed.

- **Comments**

  At the same time, it is announced via *Mechanical Engineering* magazine and the "Call for Comments" in the ANSI reporter. This gives each proposed standard the widest possible exposure for comments. Anyone may comment on the proposed standard and ASME responds to every comment.

- **Comments (continued)**

  If there are comments, the draft is returned to the working group for consideration and possible revisions. The same procedures of voting and resubmission to the Main Committee are followed for any revisions resulting from public comments.

- **Issuance**

  When there are no outstanding comments remaining, and the draft standard has been approved by the ANSI Board of Standards. ASME then publishes it as an ASME code or standard and offers it for sale.

- **Subsequent Activities**

  ASME provides updates and addenda for codes and standards until they are reissued. The Society also provides interpretations of some of its standards throughout the year by issuing written answers to inquiries. In all cases, questions and answers are published for the information of all the users of that standard.

CODE VERSUS STANDARD

The difference is between a code and a standard is not always apparent. A standard is a set of technical definitions and guidelines so that items can be manufactured uniformly and provide for interchangeability. When a standard is intended to be adopted by governmental bodies - local state or federal - or to be cited in a contractual agreement, it is called a code. The O&M series of documents is unique among ASME documents because it consists of both a Code section and a section dealing with standards and guides.

ASME FUNCTIONS

When ASME publishes codes and standards, these works are copyrighted. The copyright protects the form of expression . . . that is, the published codes or standard, not the underlying test or products covered by the code or standard.

Only ASME can print and publish these codes and standards or grant rights to others to print and publish them. Anyone not authorized to copy this material is in violation of U.S. copyright laws. An example of a violation is when a company purchases one set, then reproduces it to make other sets for use within the company.

DEALING WITH POTENTIAL CONFLICTS OF INTEREST

This involved process for developing standards has some built-in pitfalls. Whenever you gather competent and knowledgeable individuals of recognized

abilities and qualifications, it is inevitable that volunteers on the committees may have the potential for conflicts of interest. The potential for such a conflict exists whenever a person owes loyalty to competing interests.

Differences between representatives with various interests or loyalties do not constitute a conflict of interest as long as the action taken by each would be in the public interest. Everyone should be motivated for the same reason: *to enhance the public good*. Such a situation may be more appropriately described as a "Community of Interests." A real conflict of interest occurs only when loyalty to one interest would drive a course of action different from that motivated by another interest.

> *For example:* While acting for or on behalf of the ASME, a volunteer may, on occasion, be faced with consideration of a matter which directly affects the specific rather than the collective interests of the individual's employer or a competitor.

In such instances, there could be a conflict of interest between exercise of the volunteer's independent professional judgment on behalf of the ASME and the public, and the individual's loyalties and responsibilities to his or her employer or another entity.

How do we guard against such conflicts on ASME committees? The procedures for appointments to standards writing committees provide for a balance or diversified representation among the various categories of interest within the scope of that committee's concern. The Balance of Interest minimizes the instances of potential conflict of interest by preventing situations in which a single interest group could control the action on an issue.

Each of the volunteer committee members has a duty to be aware of the possibility of a conflict of interest between their responsibilities to the public on one hand and to their employer on the other. All volunteers have a fundamental responsibility to refrain from participating in decision making when a competing interest precludes or inhibits the exercise of the volunteer's independent professional judgment on behalf of the public or when the nature of the competing interest is such that the volunteer's continued participation would unreasonably jeopardize the integrity of the decision making process. The ASME staff also has the duty to be aware of potential conflicts of interest.

ANTITRUST CONSIDERATIONS

ASME finds itself in a unique business dilemma: *the better our standards are, the more they are used*. Consequently, the more they are used, the more apparent control they have in a given market. The more control over a market, the more they subject ASME to antitrust liability.

The Federal antitrust laws are intended to encourage and preserve vigorous business competition. These laws affect all parts of the business community, and professional organizations like ASME are no exception. The basic federal

antitrust law - the Sherman Act - was enacted in 1890 to regulate certain business activities in interstate commerce. Most of the substantive antitrust laws that govern businesses today come from the analysis and application of Sections 1 and 2 of the Sherman Act by Federal Courts.

Associations that bring together competitors in a spirit of cooperation have always been subject to close scrutiny under the antitrust laws. Scientific and technical associations like ASME, however, whose members come from a broad spectrum of industry, academic life, government and the general public, do not run as great a risk as do members of a trade association. Trade associations are typically narrow in their constituency and cooperative efforts can be more easily viewed as a collusive effort to enhance their employers' position in a given market.

Although ASME's principal antitrust vulnerability lies in connection with its standards activity, standard writing is perfectly legal - courts and government regulators have long recognized the need for such programs and their beneficial effect. To expand on my previous statement on ASME's unique dilemma, by setting standards and establishing industry codes, ASME is engaged in an activity that may have the effect of limiting competition to those products that meet its specifications. That the ASME's standards have served as a basis for governmental and industry regulations merely increases its power as a standards setting body.

Compliance with ASME standards is a virtual requirement for doing business for a large number of enterprises. Consequently, these standards programs have enormous economic importance. The ASME, the individual volunteer, and staff members bear a heavy responsibility to ensure that ASME's unavoidable impact on competition is principled and controlled.

Generally speaking, antitrust problems can be avoided by maintaining standards programs that are reasonable, impartial and open and which provide appeal procedures that safeguard the rights of individual manufacturers or users. Individual volunteer and staff members have a fundamental responsibility to ensure that their own participation in these programs is consistent with the requirements of the antitrust laws. Adherence to some simple guidelines will help volunteers and staff members avoid potential liability under the antitrust laws. The key word here is ***fairness***.

Fairness means: • that the public has been given adequate notice of the proposed adoption of a standard, • that an accurate record of the consideration of the group is available to interested people, • that there are formal and publicized ways in which those who disagree with any part of the code or standard can appeal to other bodies with the organization, and • that standards are periodically reviewed and revised, as necessary, to ensure that the standards reflect current technology and that all industry members - including those with the most innovative products - have an opportunity to meet them.

CONCLUSION

So now you see that we have come full circle. I started off asking how ASME can respond to advances in technology when the pitfalls and obstacles of the standards writing business require an involved approval process, and I have finished with the revelation that part of the involved approval process is driven by antitrust concerns that require timely response to innovations. *The process has become self-sustaining.* It is for this reason that the ASME standards developing process - which has been so successful up to the present time - will continue to serve industry in the same manner in the future.

Activities of the O&M Committee

Steven D. Weinman, ASME Secretary
Committee on Operation and Maintenance of Nuclear Power Plants

The charter of the O&M Committee is to develop, revise and maintain Codes, Standards and Guides applicable to the safe and reliable operation and maintenance of nuclear power plants. The codes, standards and guides developed by the O&M Committee are intended to serve the operating plants in the best interest of the public and to be submitted to the American National Standards Institute (ANSI) to become American National Standards.

In accordance with ANSI and ASME procedures, in order for the O&M Main Committee to be considered a consensus committee it has to be balanced. That is, all directly and materially affected interests shall have the opportunity for fair and equitable participation without dominance by a single interest. To met this requirement ASME procedures require no more than one third of the Main Committee membership to come from any one category. The makeup of the O&M Committee is shown on Figure 1.

Consensus is established when substantial agreement has been reached by directly and materially affected interests. Consensus requires that all views and objections be considered, and that a concerted effort be made toward resolution. The process is time consuming and laborious, but it does work to produce good documents that are more likely to be accepted by the industry and are often preferred to regulations or regulatory guides.

To fulfill our charter the O&M Committee, including Subcommittees, Subgroups and Working Groups, has 200 volunteers from various segments of the nuclear industry with virtually all of the members operations oriented. There are 35 utilities with members on the O&M Committee with five utilities having five or more members on the Committee. This adds up to a total of 80 O&M Committee members from utilities. Approximately one-third of the O&M Main Committee members are employed by utilities.

There are currently over fifteen NRC personnel who are members of the O&M Committee. They participate at the Main Committee level all the way to Working Group and Task Group level. This substantial NRC participation has many benefits, from the commitment of highly qualified people, to an increased probability that the code, standard or guide will be used.

The O&M Main Committee reports to the ASME Board on Nuclear Code and Standards. The Board supervises the nuclear codes, standards, and related accreditation and certification activities of the Society. The Main Committee is considered the consensus committee for proposals developed by the O&M Committee. Figure 2 presents an organization chart of the O&M Committee.

Figure 1

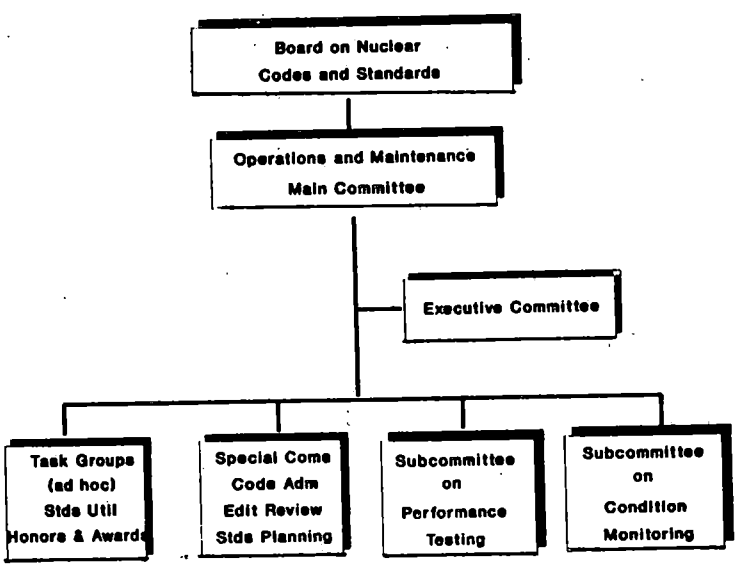
Date June 1992

COMMITTEE ON OPERATION AND MAINTENANCE
BALANCE OF INTEREST

Total 25

AK - Manufacturer	AO - Owner
1. Khuzaie-GE Nuclear Energy 2. McDonough-Westinghouse Elec. 3. Phinney - Babcock & Wilcox 4. Warner-General Atomics 5. Wolf - ABB Combustion Engrg	1. Allen - Tennessee Valley Auth 2. Basile - Consol. Edison of NY 3. Ferguson - Con. Yankee Atomic 4. D. Hansen - Yankee Atomic Co. 5. Hoyle - WPPSS 6. McGaughy - Iowa Elec. L & P 7. Rhodes- Wolf Creek Nucl. Op. 8. Thailer - Pacific Gas & Elect
AT - Regulatory	**AH - Inspection**
1. Baer - US NRC	1. Ferrante - ANI
AA - Constructor	**AI - Laboratory**
1. Mulligan - United Engrs. & C 2. Shortt - Stone & Webster *Finger (Stone & Webster) Alternate to Shortt	1. Hyten-Wyle Laboratories 2. Williams-ORNL
AB - Designer	**AF - General**
1. Dermenjian - Sargent & Lundy 2. R. Hansen - CEGA	1. Johnson-United Energy Srvcs 2. Kufel -NPS Energy Services 3. Rowley-Rowley Consultants 4. Zigler-Science & Engrg Assoc. *Sideris (United Energy Srvcs) Alternate to Johnson *Newhart (NPS Energy Services) Alternate to Kufel

Figure 2



The structure of the Committee is as follows:

> There is the Main Committee which has an Executive Committee, three Special Committees, and two Subcommittees reporting to it. The three Special Committees are on Code Administration, Editorial Review, and Standards Planning. The Subcommittees are on Performance Testing and Condition Monitoring (formerly called Vibration Monitoring).

Figure 3 presents the organization of the Subcommittee on Performance Testing. The structure of this subcommittee is as follows:

> Under the Subcommittee on Performance Testing there are three Subgroups (the Subgroup on Mechanical Equipment, the Subgroup on Pumps and Valves, and the Subgroup on Systems). Under the three Subgroups there are thirteen Working Groups.

Figure 4 presents the organization of the Subcommittee on Condition (Vibration) Monitoring. The structure of this committee is as follows:

> Under the Subcommittee on Condition Monitoring there are four Subgroups (the Subgroup on Piping Systems, the Subgroup on Reactor Internals and Heat Exchangers, the Subgroup on Light Water Reactor Loose Parts Monitoring, and the Subgroup on Rotating Equipment). There are no Working Groups under this Subcommittee.

There are currently two Task Groups under the Main Committee, the Task Group on Standards Utilization and the Task Group on Honors and Awards. Task Groups are formed on an ad hoc basis.

Committee Meetings

All ASME Codes and Standards meetings are open to the public. There is never a charge or registration fee for attending. The O&M Committee meets four times a year. Generally, Task Groups and Working Groups meet on Monday; Working Groups and Subgroups meet on Tuesday; Subcommittees, Special Committees and the Executive Committee meet on Wednesday and the Main Committee meets on Thursday. Meeting locations are rotated around the country. For those interested, a listing of all ASME Codes and Standards meetings are shown in ASME *Mechanical Engineering* magazine.

One of the responsibilities of a consensus committee is to determine what codes, standards or guides are necessary or desirable. This is particularly important as the priorities and emphasis of the nuclear power industry are refocused from design and construction to operations and maintenance.

One of the significant efforts the O&M Committee has been working on for the last two years is to try and improve its planning process. That is, more guidance

Figure 3

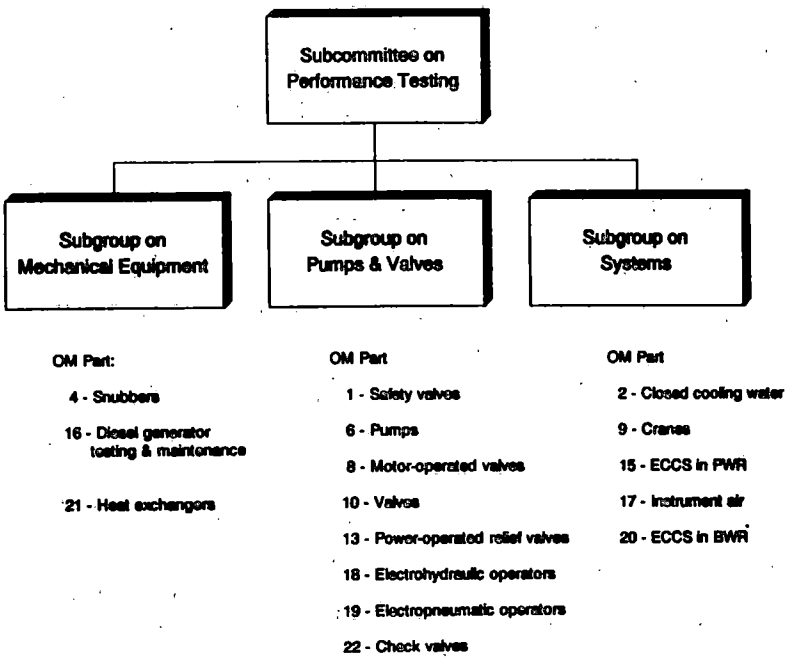
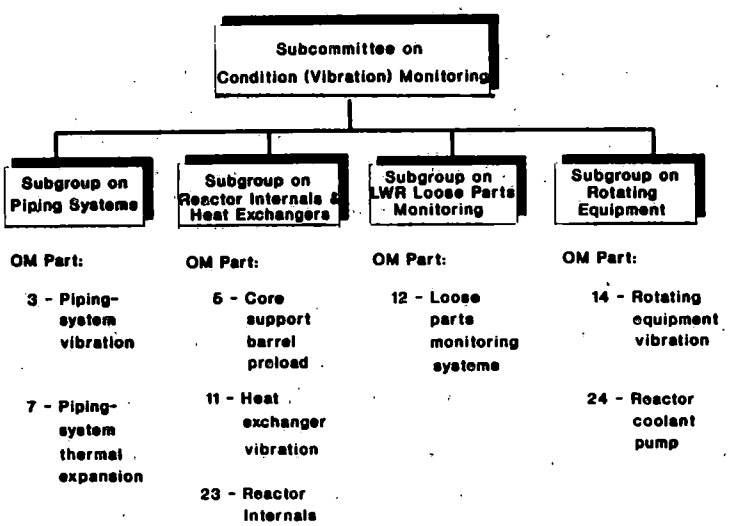


Figure 4



from the Main Committee to the Subgroup and Working Group levels. To assist in the planning process, a new special committee was formed, the Special Committee on Standards Planning. As part of this effort it was agreed that the new Special Committee should review the existing and planned scope statements for consistency and adequacy. There are currently fourteen Parts published and six others Parts agreed to by the Board. Additionally the O&M Main Committee has decided to develop four additional Parts. These 24 scope statements have been reviewed and revised as needed in accordance with this action. They have been Main Committee balloted and in many cases revised and reballoted until all 24 are now approved by the O&M Main Committee. These proposals have now been submitted to the Board for review and approval. If the proposed scope statements are Board approved, some of the existing Standards and Guides will have to be revised to correspond with their revised scope statements.

Additionally to improve our planning process we have revised our procedures to require all new scope statements to be accompanied by a statement of need and to be Main Committee approved prior to a Working Group or Subgroup expending too much effort. This allows the Main Committee to be more selective on what projects the Committee is working on and should make the Committee more responsive to the industry's needs.

To try and streamline the review and balloting process the O&M Committee formed a Special Committee on Editorial Review. Their assignment is to review draft proposals for consistency and accuracy prior to Main Committee balloting and to provide comments as needed. The intent is to have as accurate a draft as possible being Main Committee and Board balloted and eventually published.

Communications

To try and improve communications with the nuclear power industry the Committee formed a Task Group on Standards Utilization. The primary product of this Task Group is a news letter called the O&M News. The O&M News provides current information on the development, publication and utilization of the O&M Code, Standards and Guides. It is published one or twice a year and is mailed to the Vice President Nuclear and the Plant Manager of each utility. Copies are also given out during O&M Committee meetings. For anyone interested in the latest edition there are limited copies available in the back of the room.

Inquiries

Inquiries are questions posed by interested individuals or companies. A Committee response to an inquiry is intended to provide clarification of existing requirements. Inquiries also provide feed back to the Committee from the users. Quite often an inquiry points out areas of a document that need to be clarified. In fact, many of the revisions being prepared to the OM Code, Appendix I are a result of inquiries.

The Current O&M Code, Standards and Guides

The OM-1987 Document (with the three published addenda) has been split into two new documents. One is the OM Code, and the other is the OM Standards and Guides. The OM Code covers the inservice testing (IST) requirements that are being removed from Section XI of the Boiler and Pressure Vessel Code. This distribution is presented in Figure 5.

The OM Code, entitled Code for Operation and Maintenance of Nuclear Power Plants was first published on October 15, 1990. It is divided into four Subsections, as shown on Figure 6:

- ISTA on General Requirements
- ISTB on Inservice Testing of Pumps
- ISTC on Inservice Testing of Valves
- ISTD on Inservice Testing of Snubbers

Subsection ISTA establishes the rules and requirements for inservice testing of components in light-water cooled nuclear power plants. This includes the visual examination of dynamic restraints. The ISTA general requirements have been culled from the applicable portions of Section XI, IWA.

Subsection ISTB establishes the requirements for preservice and inservice testing to assess the operational readiness of certain centrifugal and positive displacement pumps used in nuclear power plants. The pumps covered are those provided with an emergency power source, that are required in shutting down a reactor to the cold shutdown condition, in maintaining the cold shutdown condition, or in mitigating the consequences of an accident.

Subsection ISTC establishes the requirements for preservice and inservice testing to assess the operational readiness of certain valves and pressure relief devices used in nuclear power plants. The active or passive valves covered are those that are required to perform a specific function in shutting down a reactor to the cold shutdown condition, in maintaining the cold shutdown condition, or in mitigating the consequences of an accident.

Subsection ISTD establishes the periodic examination and testing requirements for nuclear power plant equipment dynamic restraints (snubbers) throughout the life of the plant. The Subsection covers the preservice and inservice testing intervals, criteria, and data requirements. The requirements are applicable to the examination and testing of Classes 1, 2, 3 and MC Component snubbers.

The Code currently contains one mandatory appendix, Appendix I on Inservice Testing of Pressure Relief Devices. Appendix I provides the general requirements for periodic performance testing and monitoring of pressure relief devices utilized in nuclear power plant systems. The pressure relief devices covered are those required for protection of systems or portions of systems that perform a

Figure 5

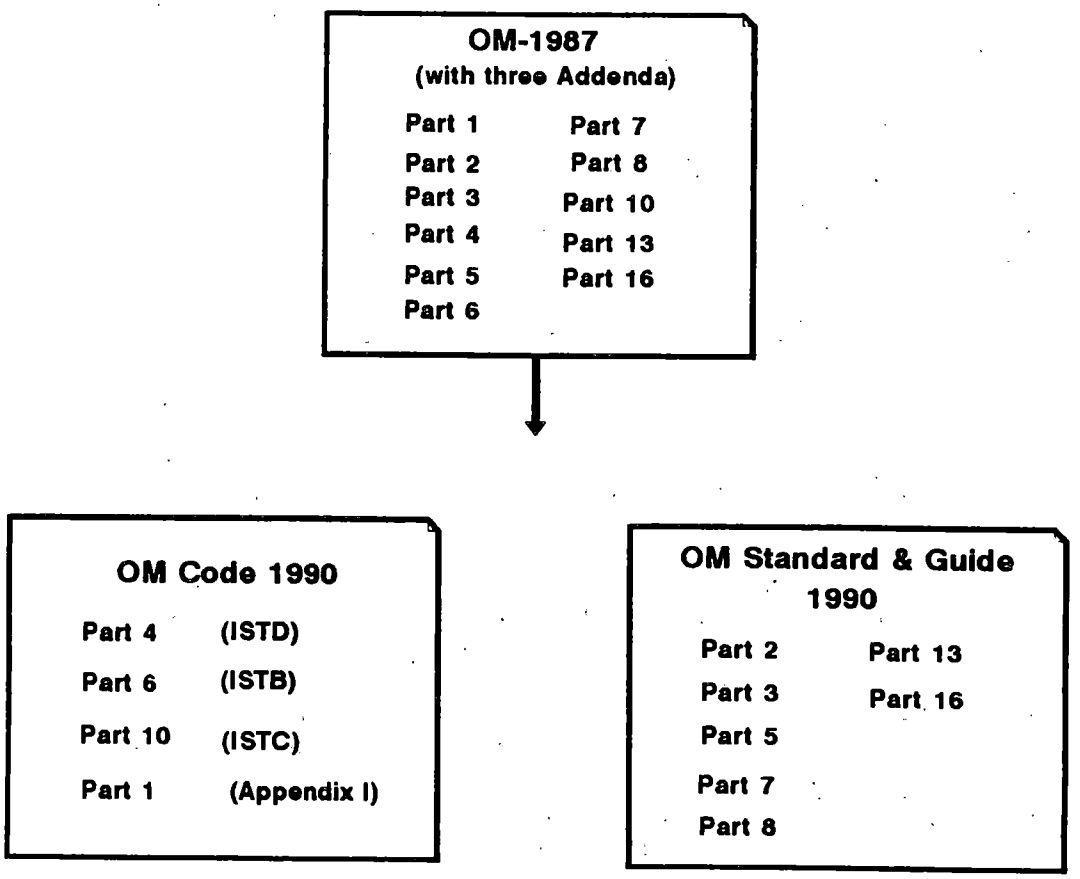

OM-1987
(with three Addenda)
Part 1
Part 2
Part 3
Part 4
Part 5
Part 6
Part 7
Part 8
Part 10
Part 13
Part 16
OM Code 1990
Part 4 (ISTD)
Part 6 (ISTB)
Part 10 (ISTC)
Part 1 (Appendix I)
OM Standard & Guide 1990
Part 2
Part 3
Part 5
Part 7
Part 8
Part 13
Part 16

Figure 6

Operation and Maintenance Code 1990

ISTA General Requirements

ISTB Inservice Testing of Pumps (Formerly Part 6)

ISTC Inservice Testing of Valves (Formerly Part 10)

ISTD Inservice Testing of Snubbers (Formerly Part 4)

Mandatory Appendix I Inservice Testing of Pressure Relief Devices (Formerly Part 1)

Nonmandatory Appendices

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

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. It includes detailed requirements for in-service test and acceptance criteria, temperature stabilization, and correlations on temperature and test media.

The Code also contains four nonmandatory Appendices as follows:

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

To try and improve our codes, standards and guides ASME uses an addenda service. Approved revisions, additions and deletions are incorporated directly into the affected pages. For example if paragraph ISTA 2.1 on page 4 was revised you would receive a replacement page 4. Additionally, there would be a margin indicator next to the affected paragraph to indicate a change has been made. The OM Code is on a yearly addenda service and a three year new edition cycle. That is, after we publish the three yearly addenda, all the published changes are incorporated into a new edition. The addenda are issued in the form of replacement pages. Each of the three addenda are issued on a different colored paper.

The first addenda to the OM Code, OMa-1991 was published on November 15, 1991. It contained no changes to the Code. Essentially that meant the O&M Committee was unable to complete the processing of technical changes to the Code for about one year from the time it was first published. It did contain two interpretations that were approved by the Committee and they are provided as part of the addenda service.

The second addenda to the OM Code, OMb-1992 is being prepared for publication and should be available by the middle of December 1992. It will contain revisions to the OM Code, Subsection ISTD on snubbers. The changes pertain to snubber visual examination requirements and a new visual examination frequency Table. The revised table on snubber examination frequency depends upon the number of previous failures and the size of the snubber population. The revised interval is designed to coincide with planned refueling outages, rather than being strictly associated with a maximum 18 month interval. In fact, if the failure rate is sufficiently low, examination during the subsequent refueling outage may be skipped. In general, the table provides considerably more flexibility to utilities, while at the same time ensuring an acceptable level of plant safety.

In conjunction with the revision to ISTD there will be a nonmandatory Appendix G added to provide guidance to determine the subsequent snubber examination interval. The addenda will also contain ten interpretations that were approved by the Committee and are provide for the users information.

Opening Remarks

The OM Standards and Guides was first published on December 15, 1990. It is arranged in two sections, Standard and Guides which are subdivided into Parts. The first Edition contain four Standards (Parts 2, 3, 13, and 16) and three Guides (Parts 5, 7 and 8). See Figure 7.

The Standards in this document are as follows:

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

The Guides in this document are as follows:

Part 5 *Inservice Monitoring of Core Support Barrel Axial Preload in Pressurized Water Reactors*
Part 7 *Requirements for Thermal Expansion Testing of Nuclear Power Plant Piping Systems, and*
Part 8 *Start-up and Periodic and Performance Testing of Motor-Operators on Valve Assemblies in Nuclear Power Plants*

The OM Standards and Guides is published on a yearly addenda service and a three year new edition cycle similar to the OM Code.

The first addenda, OMa-1991 was published on November 30, 1991. It contained technical changes to Part 3. The primary purpose of the revision to the Part was to make the corrective action portion more comprehensive and to modify the criteria for high frequency effects in piping systems. Also added was a nonmandatory Appendix E to provide guidance on piping vibration excitation mechanisms, piping responses and corrective actions. There were two new parts, Part 12 and 11, added in this addenda:

> Part 12 is on Loose Part Monitoring in Light-Water Reactor Power Plants and was added as a new Standard. Loose part monitoring is required for the reactor vessel and primary coolant system in a PWR and the reactor recirculation system in a BWR. Loose part monitors provide a means for detecting and evaluating metallic loose parts through analysis of transient acoustic signals produced by loose part impacts. OM Part 12 presents the elements necessary to establish an effective program that incorporates current research and operating knowledge.

- NRC personnel are meeting with the OM-12 writing group to discuss possible changes to Regulatory Guide 1.133 to incorporating concepts from Part 12.

Figure 7

Operation and Maintenance Standards and Guides 1990

Standards	Guides
Part 2 Closed cooling waters	Part 5 Core support barrel preload
Part 3 Piping-system vibration	Part 7 Piping-syetem thermal expansion
Part 13 Power-operated relief valves	Part 8 Motor-operated valves
Part 16 Diesel generator testing & maintenance	

- o Part 12 is considered more conservative than the European Standards
- o The Germans and the French are interested in revising their standards to make them compatible to Part 12.

OM Part 12 is an example of predictive maintenance based standards being developed by the O&M Committee.

Part 11 is on Vibration Testing and Assessment of Heat Exchangers and was added as a Guide. This Part establishes general procedures for the assessment of heat exchanger vibration for nuclear power plants. The Part includes guidance for placement and installation of instrumentation, steady-state and transient testing, data reduction and interpretation, and acceptance criteria.

The second addenda to the OM Standards and Guides, OMb-1992, is being prepared for publication and should be available by August or September of this year. It will contain a new part, Part 14 on Vibration Monitoring of Rotating Equipment in Nuclear Power Plants as a guide. The guide provides direction for in place vibration monitoring of rotating equipment for the purpose of scheduling or extending maintenance periods. It provides acceptance limits for vibration, monitoring time intervals, and guidance on trending and instrument selection. See Figure 8.

Future Projects

The O&M Committee is preparing several proposed revisions to the OM Code which may be published in the 1993 addenda. For example:

To ISTB there is a substantive proposal that would require a less frequent, more comprehensive pump test. This test would be performed at or near the pump design flow or pressure. The revision divides pumps into two groups based on operation. Group A would be pumps that are operating continuously or routinely during normal operation, cold shutdown, or refueling operations. Group B would be pumps in standby systems that are not operated routinely except for testing. The change specifies four tests and acceptance criteria. The preservice and comprehensive test employ higher instrument accuracies. The Group B test is basically a standard run test performed quarterly. The Group A test is similar to the current test and is also performed quarterly.

To ISTC there is a proposed revision to permit the use of non-intrusive techniques to verify a full-stroke exercise of check valves to meet the Codes exercising requirements. This revision will require the Owner to ensure the capability of each non-intrusive technique. A definition of non-intrusive testing has also been proposed.

Figure 8

Operation and Maintenance Standards and Guides 1990 (with OMa-1991 Addenda) (and OMb-1992 Addenda)

Standards	Guides
Part 2 Closed cooling waters Part 3 Piping-system vibration Part 13 Power-operated relief valves Part 16 Diesel generator testing & maintenance	Part 5 Core support barrel preload Part 7 Piping-syetem thermal expansion Part 8 Motor-operated valves
(added in OMa-1991) Part 12 Loose parts monitoring systems	(added in OMa-1991) Part 11 Heat Exchanger (vibration)
	(added in OMb-1992) Part 14 Rotating equipment vibration

Additionally to ISTC, there is a proposed revision to permit a sampling technique when using disassembly examination in lieu of check valve exercising. The sample disassemble examination program will group check valves of similar design, application and service condition, and requires a periodic examination of one valve from each group. The details and bases of the sampling program shall be documented in the test plan. The revision will also require a manual exercise, post maintenance testing, and corrective action.

To ISTD there are proposed revisions to add special treatment during operation for the effects of water hammer or steam hammer referred to as a transient dynamic event. Separate evaluation or abbreviated testing for those areas where transient dynamic events may occur is allowed. Transient dynamic event has been added as a failure mode group. In conjunction with this action a nonmandatory Appendix has been prepared that presents the testing plan in a flow chart form, showing the options available for corrective actions when unacceptable snubbers are found. Editorial revisions have also been prepared to improve the Subsection.

Appendix I is presently undergoing an extensive revision. Some of the proposed changes now in the review and approve process are:

1. an expanded corrective action option to include adjustment of replacement, in addition to refurbishment
2. system-level evaluations will be required to assess the impact of failures to meet the performance test acceptance criteria
3. the minimum 10 minute time between successive valve openings has been changed to an Owner specified value
4. post test/verification examinations or exercising has been added to verify operational readiness
5. the initial 5 year test interval requirements are being clarified

The O&M Committee is also preparing several proposed revisions and an addition to the OM Standards and Guides which may be published in the 1993 addenda. For example:

A proposed Part 17 on:

Performance Testing of Instrument Air Systems in LWR Power Plants

The proposed Part will establishes the requirements for preservice and inservice testing to assess the operational readiness of certain Instrument Air Systems used in LWR power plants.

The Instrument Air Systems covered are those which supply air to systems 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 Part does not apply to breathing air systems. The testing requirements in this Part include performance tests to ensure a system

is capable of producing quality air at full design capacity. Other tests look for degradation of all parts of the system. This proposed Part is being written as a standard because of the importance of instrument air and its contribution as a common mode failure mechanism in multi train safety systems.

Under considered is a proposed rewrite of Part 8 on:
Performance Testing of Electric Motor Operated Valve Assemblies in LWR Power Plants. The Part establishes the test methods, intervals, criteria for evaluation of test results, and documentation of tests and corrective action taken as a result of these tests on certain Electric Motor Operated Valve Assemblies found in nuclear power plants.

Some of the more significant changes to the Part are as follows:

- The proposed revision will change the published guide into a standard.
- It will factor in MOV information and streamline the types of tests to be performed.
- There is a greater emphasis on analysis of all testing results to better understand MOV performance.
- The parameters for component testing have been simplified.
- The grouping of valves is recognized as part of the MOV testing program with proper engineering analysis.

The appropriate implementation of this new Part will reduce the number of valves to be periodically tested.

Also being considered is a proposed revision to Part 7 to include the influence of thermal stratification.

Please note the above proposals for 1993 addenda to the OM Code and the OM Standards and Guides are what could be in the addenda. The proposals are not fully approved yet. Additional proposals may be added, or several of the proposals may not receive full approval and therefore will not be published.

Several Long Range Proposals - 1994 Plus

The O&M Committee is presently preparing several new standards and a guide for inclusion in the OM Standards and Guides document. These proposed Parts are in the drafting stage by their respective Working Group or Subgroup and have not been Main Committee balloted. They are as follows:

Part 9 *Standard For Inservice Performance Testing and Inspection of Nuclear Plant Cranes*

Part 15 *Standard For Performance Testing of Emergency Core Cooling Systems in PWR Plants*

Part 18	*Standard For Performance Testing of Electrohydraulic Valve Assemblies in LWR Power Plants*
Part 19	*Standard for Performance Testing of Electropneumatic Valve Assemblies in LWR Power Plants*
Part 20	*Standard for Performance Testing of Emergency Core Cooling Systems in BWR Power Plants*
Part 21	*Standard for Performance Testing of Heat Exchangers in LWR Power Plants*
Part 22	*Standard For Performance Testing of Check Valves in LWR Power Plants*
Part 23	*Guide For Inservice Monitoring of Reactor Internals Vibration in PWR Power Plants*
Part 24	*Standard for Reactor Coolant and Recirculation Pump Vibration Monitoring*

NRC Endorsement

As you may know the NRC is in the final stages of revising 10 CFR 50.55a which will update the Section III and Section XI references to their 1989 Editions. Additionally, a proposed rule has been initiated to incorporate by reference into the NRC regulations the 1992 Editions of Section III and Section XI, and the ASME OM Code.

Conclusion

The O&M Committee has in the past and will continue to try and respond to the problems of the nuclear power industry that are within the purview of the ASME. Your technical contribution to the O&M Committee is certainly welcome, since the broader the input, the better the quality of our documents. Our challenge is to continue to address key issues in a timely manner.

Application Of Hydraulic Network Analysis To Motor Operated Butterfly Valves In Nuclear Power Plants Revision 1

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ABSTRACT

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

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

INTRODUCTION

In 1989, the United States Nuclear Regulatory Committee issued Generic Letter 89-10 (GL 89-10), which recommends that all nuclear power plant licensees in the United States review the operability of motor operated valves (MOVs) in their plants. The letter specifically recommends that both normal operating conditions as well as design basis conditions be addressed. Although in-situ testing may be used to demonstrate MOV operability under normal operating conditions, some design basis conditions, such as those involving downstream pipe rupture, do not lend themselves to such testing. Additionally, large scale flow testing may not be feasible for several reasons, including plant operating constraints, cost, and inaccessibility of some MOVs. Analytical procedures in conjunction with judicious testing of some MOVs may be used to address the concerns of GL 89-10. One of the objectives of these procedures is to determine the actual valve flow rate and pressure drop against which

the MOV is required to operate. In the absence of a well-established and structured methodology to calculate pressure drops and flow rates across the MOV at various disc positions, utility engineers often utilize overly conservative assumptions and analytical procedures. In many cases, torque and thrust predictions based on such assumptions may show that adequately sized MOVs are incapable of operating under the specified conditions. In other situations, the calculated actuation torque or thrust, based on such assumptions, may exceed the structural strength capability of the valve or the actuator. In these situations, the apparent problem is not in the MOV itself, but rather in the underlying assumptions or the use of simplified procedures.

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

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

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

REQUIRED ACTUATION TORQUE FOR BUTTERFLY VALVES

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

Total Seating and Unseating Torque, T_{seat}

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

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

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

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

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

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

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

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

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

The hydrostatic torque may be neglected except for very large valves (typically 30 inches and larger). The hydrostatic torque becomes zero (or negligibly small) when (1) the valve stem is vertical; (2) liquid levels in both the upstream and downstream pipes are the same (either full, empty, or partially full); or (3) the process fluid is air, gas, or steam with insignificant head variation due to density.

In summary, the total torque required to seat or unseat the valve disc, T_{seat}, is the sum of four components: (1) seat torque, T_s; (2) bearing torque, T_b; (3) packing torque, T_p; and (4) hydrostatic torque, T_{hyds}.

Dynamic Torque, T_{dyn}

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

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

The positive sign is used when calculating opening torque and the negative sign is used when calculating closing torque. The peak

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

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

The hydrodynamic torque component, T_{hyd}, is the product of the resultant hydrodynamic force induced by the flowing fluid and its moment arm to the center of disc rotation, Fig. 1. T_{hyd} is a self-closing torque which tends to close the valve. Some exceptions do exist, however. For instance, at high disc opening angles, torque reversals (from self-closing to self-opening) are encountered with single and double offset disc designs when the valve is oriented such that the flow is towards the flat face of the disc (valve shaft is downstream). Continuous operation with the disc at or near the torque reversal position can cause damage to both the valve and the actuator. In many cases the disc travel is limited to a smaller disc angle (60 to 70 degrees open) to prevent such damage. Another exception to the self-closing characteristics of T_{hyd} was reported for compressible flow by Watkins, et al (1986). In their blowdown testing of containment purge valves, it was found that the dynamic torque is self-opening throughout the valve stroke when the shaft is downstream. Butterfly valve *aerodynamic* torque in compressible flow has been recognized to be a complex phenomenon and continues to be a subject for further investigation (Morris and Dutton, 1989).

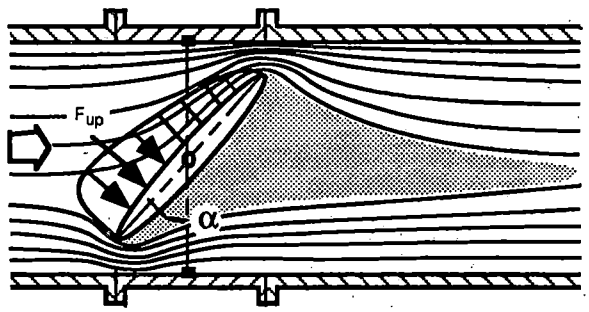


Figure 1. Flow through a symmetric disc butterfly valve

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

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

$$T_{hyd} = \frac{C_t\, d^3\, \Delta P_v}{12} \qquad (5)$$

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

The hydrodynamic torque of butterfly valves with non-symmetric disc designs (such as single and double offset discs) depends on the flow direction. With the shaft upstream (often referred to as curved face forward or CFF), the hydrodynamic torque

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

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

> *As the ratio of valve pressure drop at zero flow ($Q = 0$) to valve pressure drop at maximum flow ($Q = Q_{max}$) increases, the disc opening angle for the peak dynamic torque decreases towards the fully closed position.*

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

Second, as the flow rate decreases, the effect of upstream disturbances on the inlet velocity profile and T_{hyd} diminishes. Thus, at low disc opening angles, where the flow rate is small, the effect of upstream disturbances is small, and in many cases negligible. In centrifugal pump applications with high piping resistance, peak dynamic torque occurs at small disc opening angles. For these applications, the effect of upstream disturbance on the required torque is small and has been found to be negligible in several cases. The presence of upstream disturbances also affects the valve flow coefficient. This in turn affects the

valve pressure drop as a function of disc angle. For instance, the effect of inlet/outlet concentric reducers on the valve flow coefficient at various disc angles is accounted for by introducing a piping factor, F_P, which is a reduction factor (ISA-S75.01, 1985).

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

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

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

HYDRAULIC NETWORK ANALYSIS

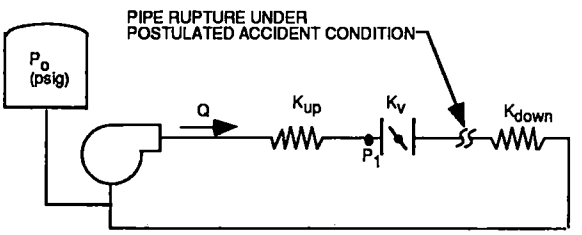
Hydraulic network analysis can be used to determine pressure drop across MOVs and the corresponding flow rate as functions of disc angle for quarter-turn valves, or valve travel for rising stem valves. Depending on the layout of the particular installation, the network may be analyzed using a single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

Hydraulic network analysis assumes that the flow and torque coefficients, as well as choking characteristics of the valve

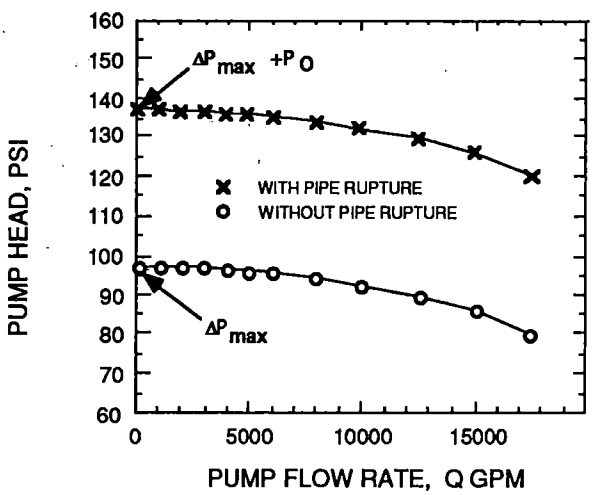
under consideration are known. These valve characteristics should be determined experimentally or obtained from the valve manufacturer for the specific valve design, disc orientation, and fluid media (incompressible or compressible).

Single line Model

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

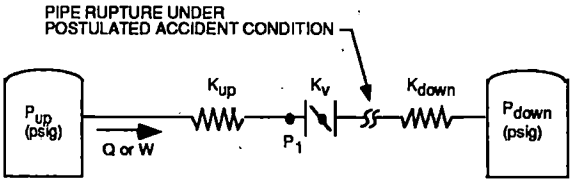


(a) Simplified model of a hydraulic system with a butterfly valve

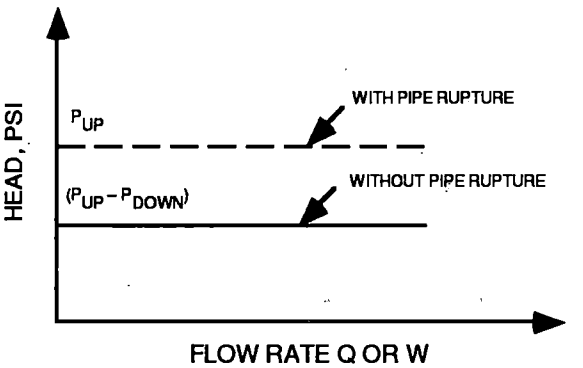


(b) Pumping characteristics with and without pipe rupture (NCWS pump in examples)

Figure 2. Single line model of hydraulic systems with centrifugal pumps



(a) Simplified model of a hydraulic system with a butterfly valve



(b) Effective head with and without pipe rupture

Figure 3. Single line model of hydraulic systems with constant head

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

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

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

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

$$K_{sys} = K_{up} + K_v + K_{down} \quad (7)$$

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

$$Q_{sys} = 236\, d^2 \sqrt{\frac{\Delta P_{sys}}{K_{sys}\, \rho}}$$

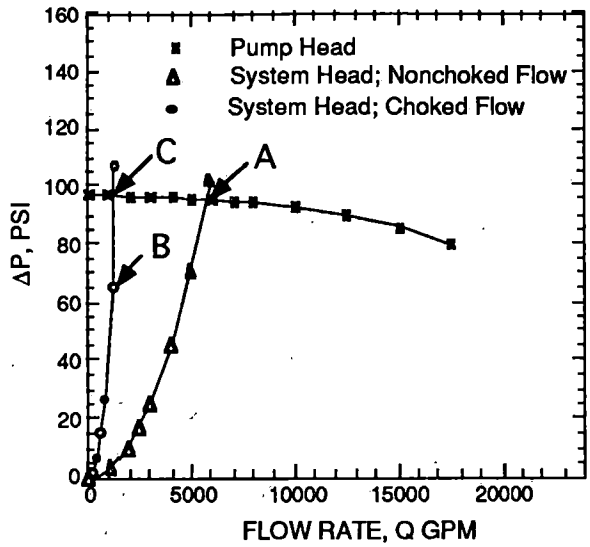
(incompressible flow) (8a)

$$W_{sys} = 1{,}891\, Y\, d^2 \sqrt{\frac{\Delta P_{sys}}{K_{sys}\, \overline{V}_1}}$$

(compressible flow) (8b)

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

$$\Delta P_v = \Delta P_{sys} - \left(\Delta P_{Kup} + \Delta P_{Kdown}\right) \quad (9)$$



$$\Delta P_A = \Delta P_{Kup} + \Delta P_v + \Delta P_{Kdown}$$

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

$$\Delta P_C = \Delta P_{Kup} + \Delta P_v + \Delta P_{Kdown}$$

Figure 4. Graphical solution of a single line model at two disc angles with unchoked and choked flow conditions

As the flow rate increases with disc opening angle, the pressure drop across the valve decreases because of the increase in the pressure drop across the system (other than the valve) as given by $\Delta P_{Kup} + \Delta P_{Kdown}$ (see Equation 9). This procedure should be repeated at various disc positions. The objective is to determine the valve flow rate and pressure drop as functions of disc position throughout the valve stroke.

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

The accuracy of the valve pressure drop predictions depends on how closely the single line model represents the actual network. If the valve is located in the main pipeline with small or no parallel flow

through other branches, these predictions are quite accurate. In the presence of parallel branches with relatively small flow resistance, the pressure drop predictions based on single line assumptions may be too conservative, and more accurate predictions can be obtained by utilizing the parallel line model described next.

Parallel Line Model

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

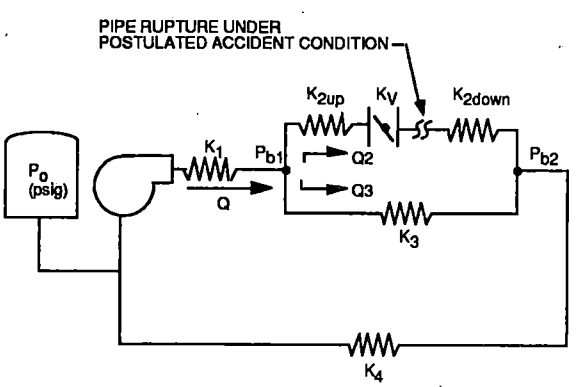


Figure 5. Parallel line model of a hydraulic network with a parallel branch

Resistance Coefficients. Conservative valve pressure drop predictions are obtained by underestimating K_1, K_{2up}, K_{2down}, and K_4, for the same reason discussed in the single line model. Overestimating K_3 also gives conservative predictions because more flow would be forced through the valve branch, resulting in higher pressure drop across the valve.

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

1. Calculate the resistance coefficients (K factors) of the piping components.
2. Normalize the K factors with respect to the nominal size of the MOV.
3. Combine K factors for the different branches (K_{branch}) as follows:

For n resistances in series:

$$K_{branch} = \sum_{i=1}^{n} K_i \qquad (10a)$$

For n resistances in parallel:

$$\frac{1}{\sqrt{K_{branch}}} = \sum_{i=1}^{n} \frac{1}{\sqrt{K_i}} \qquad (10.b)$$

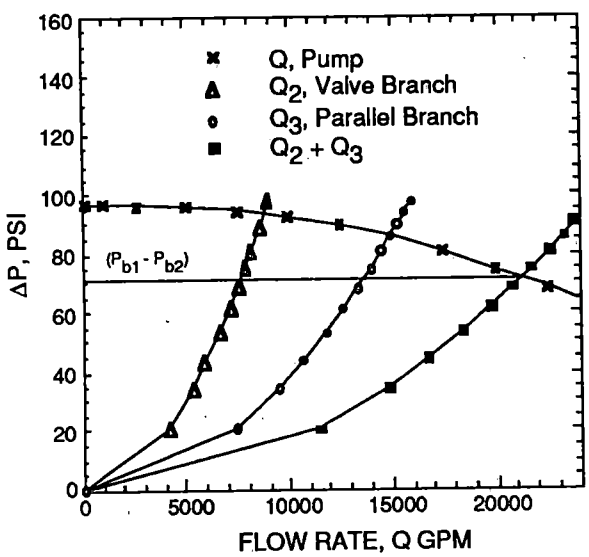
Alternatively, if the flow rate (Q_{branch}) and pressure drop (ΔP_{branch}) across a pipe branch are known from plant performance/test data, Darcy's formula may be used to calculate the branch resistance coefficient (K_{branch}) as follows:

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

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

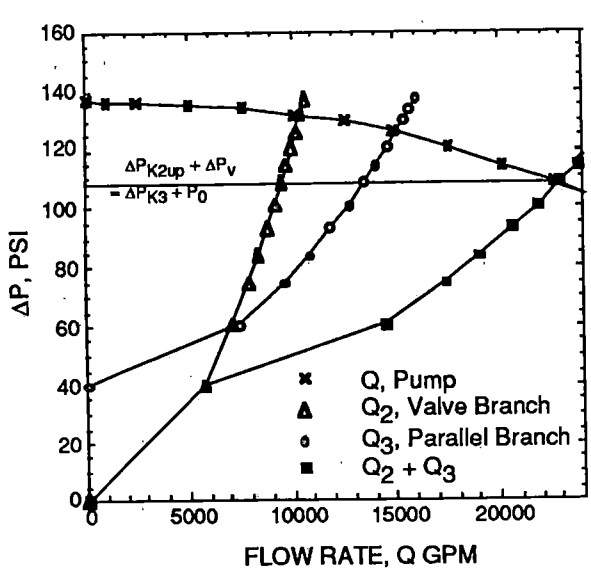
Flow Rate and Valve Pressure Drop. At each disc angle, α, the valve flow rate, Q_2, and pressure drop, ΔP_V, can be determined using either analytical or graphical solution procedures. One such procedure is shown in Figure 6a for normal flow (no pipe rupture) and in Figure 6b for blowdown condition (pipe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

$$Q_{pump} = Q_2 + Q_3$$
$$= 236\, d^2 \sqrt{\frac{(P_{b1} - P_{b2})}{(K_{2up} + K_v + K_{2down})\rho}}$$
$$+ 236\, d^2 \sqrt{\frac{(P_{b1} - P_{b2})}{K_3\, \rho}} \qquad (12)$$



$$(P_{b1} - P_{b2}) = \Delta P_{K2up} + \Delta P_v + \Delta P_{K2down}$$
$$= \Delta P_{K3}$$

(a) Without pipe rupture



(b) With pipe rupture

Figure 6. Graphical solution of parallel line model at disc angle, α, with $K_1 = K_4 = 0$

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

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

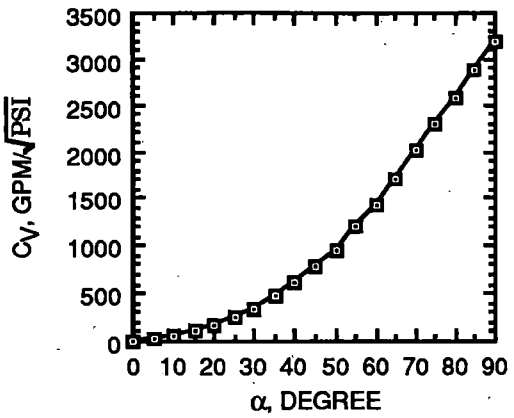
EXAMPLE

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

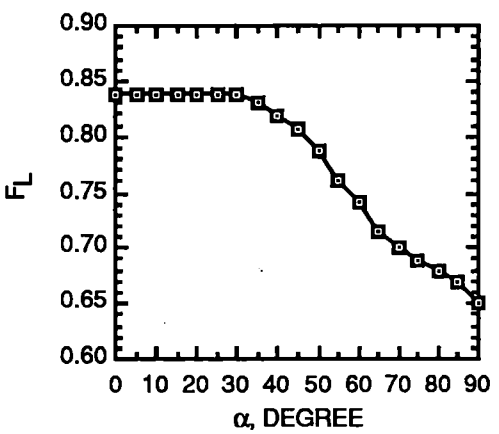
Problem definition

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

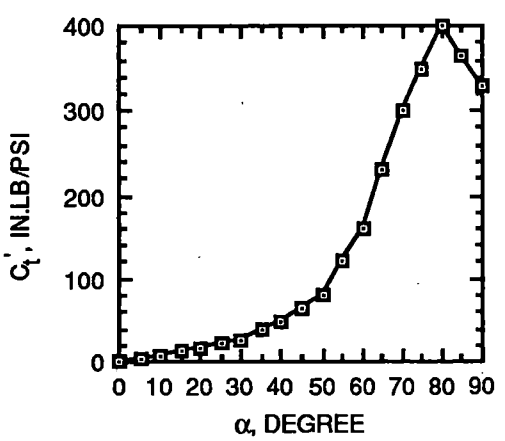
T_s = 116 ft.lb
T_p = 57 ft.lb
μ_b = 0.15
d_s = 1.125 inch



(a) Valve flow coefficient, c_V, vs disc opening angle, α



(b) Liquid pressure recovery factor, F_L, vs disc opening angle, α



(c) Hydrodynamic torque coefficient, C_t', vs disc opening angle, α (shaft upstream)

Figure 7. Flow and torque characteristics for a 10", 150# butterfly valve (shaft upstream)

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

$$K_1 = 10$$
$$K_{2up} = 10$$
$$K_{2down} = 30$$
$$K_3 = 20$$
$$K_4 = 10$$

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

Problem Requirement

1. Find the required actuation torque under (a) normal operating conditions, (b) postulated pipe rupture conditions with surge tank pressure, and (c) postulated pipe rupture conditions with no surge tank pressure. Find these torque requirements using single line and parallel line models.

   These conditions and modeling approaches result in six cases, i.e.:

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

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

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

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

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

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

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

Solution

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

Single Line Model

A single line model of this system is obtained by conservatively neglecting the flow through the parallel branch with line resistance K_3. Analytically, this is equivalent to setting the flow coefficient of the parallel branch to zero, or K_3 to infinity. The hydraulic resistances of the single line model as shown in Figure 2a are as follows:

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

For normal flow conditions (Case 1):

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

For pipe rupture conditions (Cases 2 and 3):

$$K_{down} = 1$$

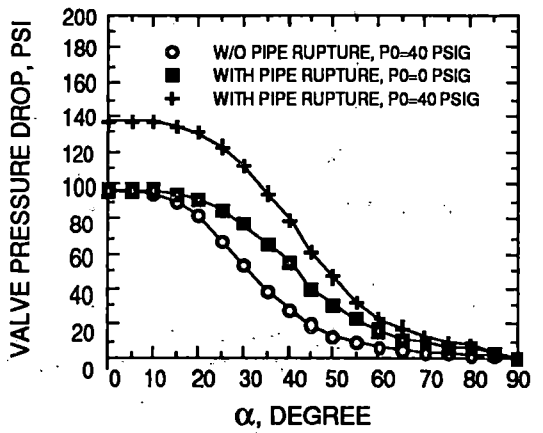
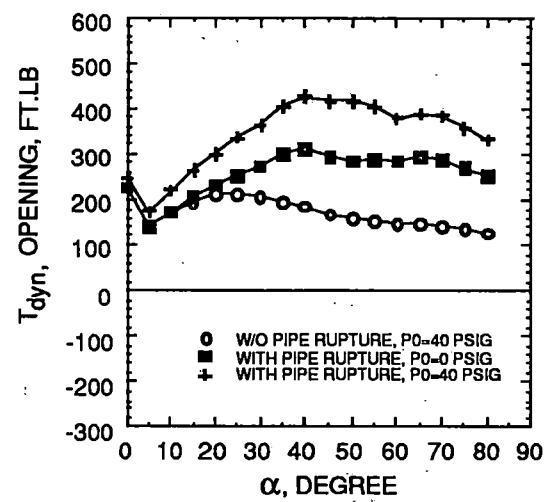
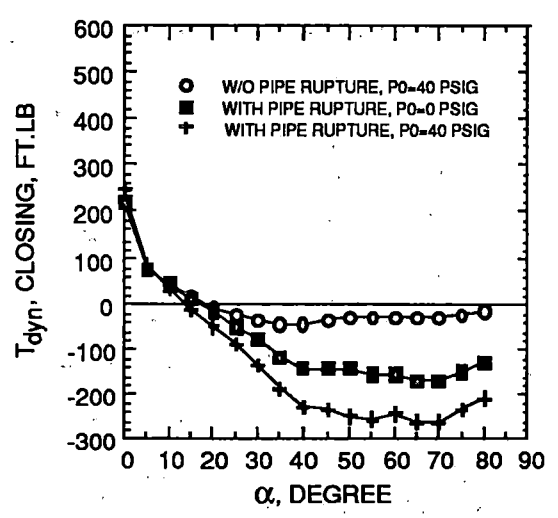


Figure 8. Valve pressure drop versus disc opening angle, using single line model



(a) Opening stroke



(b) Closing stroke

Figure 9. Required actuator torque vs disc opening angle using single line analysis

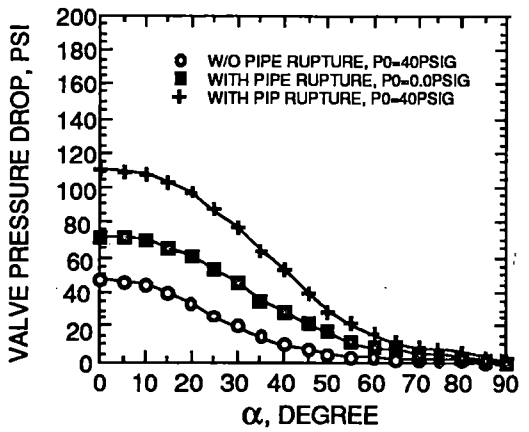
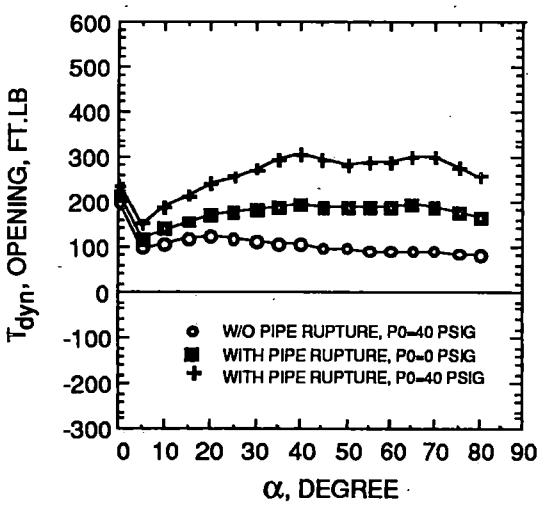
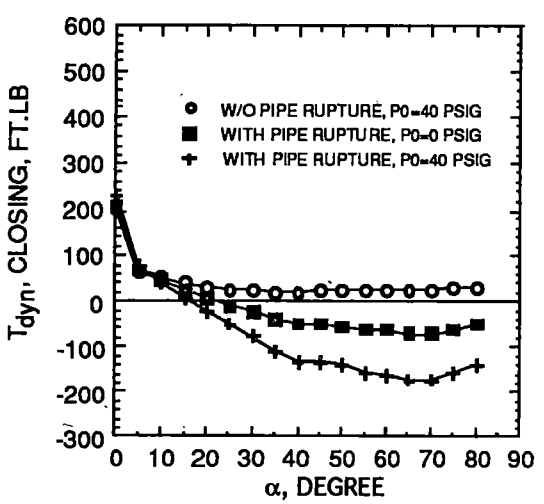


Figure 10. Valve pressure drop vs disc opening angle using parallel line model



(a) Opening stroke



(b) Closing Stroke

Figure 11. Required actuator torque vs disc opening angle using parallel line analysis

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

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

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

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

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

Table 1. Summary of torque requirements for example problem

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

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

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

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

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

1. Under normal operating conditions (Case 1), the required actuation torque is 225 ft-lbs, which is dictated by the total seating and unseating torque.
2. The required opening dynamic torque of 426 ft-lb for the case of pipe rupture with full tank head (Case 3) is about double its value under normal operating conditions (Case 1).
3. The required opening dynamic torques of 310 ft-lb and 426 ft-lb respectively in both pipe rupture cases (Case 2 and Case 3) exceed the total seating and unseating torque.
4. The actuator pullout torque of 296 ft-lb at reduced voltage is less than the required opening dynamic torque for Case 2 and Case 3 .

5. The margin between the actuator pull-out torque of 462 ft-lb at full voltage and the required opening dynamic torque of 426 ft-lbs for Case 3 is less than 10 percent.

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

Parallel Line Model

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

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

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

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

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

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

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

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

CONCLUSIONS

The analytical procedures for performing hydraulic network analysis to evaluate the performance of butterfly MOVs are presented. Both single line and parallel line models described herein can be used to determine valve pressure drops, flow rates, and torque requirements. An example problem from a typical installation in a nuclear cooling water system is used to

demonstrate the application of the analysis procedures. A matrix of three different analyses performed using single line and parallel line models shows a significant reduction in torque requirements when the more realistic parallel line model is utilized. Detailed results are presented for each case. For the example problem, torque requirements for the parallel line model under the limiting operating conditions were found to be 40 percent less than that obtained for the single line model.

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

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

NOMENCLATURE:

Symbol	Description
C_b	Effective bearing torque coefficient, in.lb/psi
C_t	Hydrodynamic / aerodynamic torque coefficient, dimensionless
C_t'	Hydrodynamic / aerodynamic torque coefficient for a specific valve size, pressure class, and flow direction, ($=C_t\, d^3$), in.lb/psi
C_v	Valve flow coefficient at any disc angle, α, $\frac{gpm}{\sqrt{psi}}$
F_L	Liquid pressure recovery factor of a valve without attached fittings, dimensionless
F_p	Piping geometry factor, dimensionless
K	Head loss coefficient of a device, dimensionless
P_1	Upstream absolute static pressure, measured two nominal pipe diameters upstream of valve or valve-fitting assembly, psia
Q	Flow rate, gpm
Q_{choked}	Valve flow rate under choked flow conditions, gpm
Q_{max}	Maximum flow rate when the valve is fully open, gpm
T_b	Bearing torque, ft.lb
T_{dyn}	Total dynamic torque at disc angle α, ft.lb
T_{hyd}	Hydrodynamic / aerodynamic torque, ft.lb
T_{hyds}	Hydrostatic torque, ft.lb
T_p	Packing torque, ft.lb
T_s	Seat torque, ft.lb
T_{seat}	Total seating/unseating torque, ft.lb
$\overline{V}_1$	Specific volume of fluid at inlet conditions, ft^3/lb
W	Flow rate, lb/hr
W_{choked}	Valve flow rate under choked flow conditions, lb/hr
W_{max}	Maximum flow rate when the valve is fully open, lb/hr

Y Expansion factor, ratio of flow coefficient for a gas to that for a liquid at the same Reynolds number, dimensionless

d Nominal valve size, inch

d_s Stem diameter, inch

α Disc opening angle, deg.

μ_b Bearing coefficient of friction, dimensionless

ρ Density, lb/ft^3

ΔP_{choked} Pressure differential at the onset of valve choking, psi

ΔP_{max} Maximum pressure differential at shut-off, psi

ΔP_v Pressure differential across the valve or valve-fitting assembly, psi

ACKNOWLEDGMENTS

The authors are deeply grateful to the many utility engineers who supported the technical development of this work and provided the required system data. In particular, we wish to acknowledge Mr. Robert Elfstrom of MOV Service Group (previously with Davis-Besse Nuclear Power Station), Mr. Edward Smith of Arizona Public Service, and Mr. John Hayes of Toledo Edison. Special thanks to Bobbie Benson for preparing this manuscript.

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Toeing the Line
Meeting Minimum Compliance Regulation

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ABSTRACT

This paper discusses the history of pump and valve testing regulation in the United States, and identifies the result of decades of testing under those rules. While the current regulations for pump and valve testing under OM rules are an improvement over the previous Section XI requirements, the author identifies the essential need for a regulatory distinction between licensees meeting minimum compliance with current rules and licensees that exceed the requirements of existing regulation. In addition, the author discusses some case histories of problems in pump and valve testing under the current set of rules.

Introduction

Over the last 20 years of the nuclear power industry, the evolution of equipment testing to meet established regulatory requirements has been justified on the basis of an overall improvement in equipment reliability and state-of-readiness. In order to meet these goals, pumps and valves have been required to meet some rather elementary surveillance-based tests -- typically measurement of valve stroke times and basic pump parameters. These tests have provided basic information regarding the current condition of the components under test, but have not generally resulted in achieving the goals of the industry, i.e., maintaining safety-related equipment in a higher level of reliability using an appropriate level of effort.

History of Component Testing

The essential regulation governing this process of testing has been 10CFR50.55a and related sections. This regulation has primarily been implemented using ASME Section XI, both for passive component examination (welds, studs, nuts, etc.) and for active component testing (pumps and valves). This testing and inspection has, in general, been implemented via separate groups of people at the plant level, and has been regulated by different individuals at the NRC. It is important to remember that as an industry we have been testing these pumps and valves for at least 15 years (and in many cases, 20

years), and have accumulated an impressive storehouse of knowledge of the results of this testing. What can we say we have learned from our hundreds of reactor-years of testing experience?

For the sake of review, I would like to summarize some of the points that have been made by various individuals over the last few years in response to the effectiveness of pump and valve testing performed thus far. These points may not hold true in every case at every plant, but I believe that the situations identified will be very familiar to many plant individuals.

Point 1 - Trending of AC motor-operated valve stroke times performs no useful function. While it is true that meeting a specific stroke time limit for a given valve provides complete assurance of the valve's current operable condition, it does not really assess the long-term operable condition of the valve. The true concern here is that a measurement of valve stroke time using a stopwatch will not be able to detect the extremely subtle changes in AC motor-operated valve stroke time that indicate long-term degradation is taking place. Typically, the failures found by this testing method are catastrophic in nature (i.e., complete failure of the valve to change position) and therefore maintaining a history of stopwatch stroke times provides little viable data for long-term evaluation of the valve/actuator's condition.

Please note that this is not a call for removing the stroke time requirement completely, in that some regular operability test frequency does provide valuable information about the current state of the valve. Over time, by careful review of historical valve failure data, a reasonable confidence in valve operability is gained by the lack of failures between tests. Therefore, a simple stroke time test, with a comparison of stroke time to maximum allowable stroke time limits, provides us with some assurance of current valve operability. The issue of prediction of valve failures at some time in the future has not been verified. Along with testing in accordance with Generic Letter 89-10, most plants can maintain reasonable confidence in the operability of their AC MOVs via a simple quarterly (or as appropriate) stroke time test.

Point 2 - Trending of Air-Operated Valves (AOVs) provides little usable data for predicting future valve failures. Solely based on AOV stroke time, without taking into account system pressure, air line condition, actuator condition, time since last test, etc., decisions made concerning AOV condition may be based on data which is essentially variable in nature, and may show no real correlation with valve failure potential. Most testing personnel will be familiar with the "sawtooth" data spread many AOVs stroke times evidence over successive tests. In general, this sawtooth behavior is evidence or a lack of control over the essential parameters of AOV testing (system pressure, air line pressure/condition, stem condition, and valve installed condition). In most cases, once the variables in valve stroke time are understood, and hopefully controlled during the test, the AOV stroke time test data provides valuable information about the condition of the valve actuator.

This AOV stroke time effect is similar, although not as extreme in most cases, to the results that have been seen when trending containment isolation valve leakage rates over time. In the case of containment isolation leak rate testing, there seems no simple correlation from one set of data to the next, which precludes feasible prediction of failure. It is important to note that the requirement to trend containment isolation valve leakage rates was one of the significant reductions in testing performed on valves under OM rules, vice ASME Section XI Code. In addition, the specific requirements to analyze the 6-inch and larger valves under Section XI was removed from most plant's testing programs under Generic Letter 89-04.

Point 3 - Use of the OM valve stroke time acceptance criteria allows some very high variability from test-to-test, which would not meet present Code rules. The concern lies in the previous Code rules that allowed what has been termed the "creeping death" syndrome, i.e., valve stroke times which slowly increased without exceeding the 25% level from test-to-test, until the valve fails or reaches its maximum allowable limit. Without a graphical display of the valve's stroke time history, this effect can be difficult to recognize. The OM testing acceptance criteria protect against this failure mode, by setting a fixed reference value against which the current test is always compared. This fixed reference value is taken when the valve is in good operating condition, and free of obvious defects. However, the improvement in using this stroke time acceptance criteria does allow a valve's stroke time to become essentially variable between the ±25% (or ±15%) band of acceptable performance. This means that a valve can vary from test-to-test by as much as 66% without any action being taken to address the condition of the valve. Based on this issue, general recommendations have included analyzing valve data using both criteria (reference value limits and test-to-test limits) to address the operable condition of the valve.

One of the problems in this area lies in the "mixing" of the concepts of surveillance testing with the concepts of performance testing. Surveillance testing provides valuable data on the current operable status of components (whether valves, pumps, sensors, reactor protection system instruments, etc.), but is not really designed to provide long-term data related to subtle degradation modes.

Why then do we perform surveillance testing at all? The surveillance test results do show that the plant is currently able to perform it's intended safety-related response in the event of an accident situation. The results do not show that the plant will be able to perform it's safety-related function at any specific time in the future, since there is no predictive capacity in the results of the surveillance tests. Alternatively, performance testing of components does not generally provide any verification of absolute qualification of the component to meet any specific safety-related value. Performance testing does provide data to assess the condition of the equipment, and will provide data to provide long-term predictions of component condition, as well as measurement of

subtle (and sometimes not so subtle) failure modes.

Therefore we must draw a distinction between the goals and methods of implementing surveillance testing and the goals and methods of implementing performance testing. While these two sets of methods may occasionally utilize the same specific testing techniques, there is no reason that they should have to be the same set of tests.

The Concept of Minimum Compliance

The effect of setting a minimum compliance standard on the nuclear industry has been to focus attention to meeting, at minimum cost, the regulatory standard. More importantly, it has meant that any additional attention paid to performance testing at nuclear plants has often been seen as "unnecessary" and "extravagant" uses of plant manpower and material, without any direct benefit. It is impossible to assume that a company faced by budgetary shortfalls would pass over "discretionary" testing of pumps and valves when looking for costs to recover.

In order to increase component reliability and availability, we are then faced with a choice of two options; (1) allow plant personnel some flexibility to develop their own performance testing program, taking into account their unique and individual problems and concerns, while still meeting the goal and intent of maintaining component reliability at appropriate levels, or (2) raising the standard to encompass all possible situations and circumstances with a very high level of detail. The very basis of nuclear regulation in the US has been away from the prescriptive, detailed listing of requirements and towards the results-oriented position. Taking position 2 would require the NRC (nee ASME) to develop a testing code that is all things to all people, which would be challenging to say the least. The recent issuance of the Maintenance Rule for the nuclear industry provides what is in effect redundant regulation for the group of pumps and valves tested under 10CFR50.55a. I would call for a review of the details of 10CFR50.55a in light of the positions held by the Maintenance Rule, and deal with the issues of pump and valve testing in the same fashion. By providing broad recommendations for implementation by licensees, the NRC could provide incentive for the creation of effective and efficient pump and valve testing programs that maintain the separate goals of verification of operable capability via surveillance testing and assessment of long-term failure potential via component performance testing.

At the conference two years ago, one of the best presentations that I saw, based on commitment to equipment reliability and overall impact, was without a doubt the discussion of vibration monitoring at a fossil-based utility. I was both impressed and saddened by this presentation; impressed that this utility had developed such a detailed and comprehensive vibration monitoring program for all their plant rotating equipment, but saddened that at many nuclear plants around the country, the safety-related pumps actually get less attention during vibration testing than do their balance of plant counterparts. It is depressing to see that the RHR pump's vibration is

being monitored with a hand-held stick probe to meet existing ASME Code rules (mils displacement peak-to-peak) when the circulating water pumps get monthly vibration frequency spectrum analysis. The justification for this situation at many plants is that the plant needs a specific acceptance criteria to meet NRC regulations, and that acceptable value is difficult at best to develop to cover all vibration-based failure situations and failure modes. Therefore, the plant falls back on the minimum compliance rule to perform testing of safety-related equipment.

Alternatives to Minimum Compliance

In lieu of requiring plants to perform a strict set of performance tests designed to fit a general set of circumstances, we should provide a program of alternatives and options, designed to provide flexibility in meeting the defined program goals of assessing component present and future operability. One possible set of alternatives could include:

- Stroke time testing of AC motor-operated valves on a quarterly basis, along with refueling frequency testing to meet Generic Letter 89-10 requirements. Stroke time data would be taken on all tested valves, but only a comparison to maximum allowable stroke times would be performed. No trending of valve stroke times would be required.

- No regular quarterly vibration testing of pumps. Instead, perform extended vibration frequency spectrum analysis of pumps during refueling testing/after maintenance, etc. Base the results of the vibration testing on the engineering judgement of the individuals involved. Failures identified via vibration analysis would have to be dealt with prior to startup, and a successful pump run with supporting documentation be developed at that time.
- During regular quarterly pump testing, perform a full vibration frequency spectrum analysis. Base the acceptance of the analysis on the judgement and experience of the plant personnel. Based on the results of several successive pump tests, allow a relaxation in the pump testing frequency (quarterly to bi-annually, etc.). Pumps which fail via vibration-related causes should be dealt with decisively.
- Use of advanced diagnostic systems on air-operated valves could allow relaxation of the trending/testing requirement for stroke time data.
- Flexibility of scheduling of surveillance testing for any components with a successful long-term testing history. If a given component has shown little or no degradation over time, then the test schedule for that component should be adapted to reflect the low-duty cycle of its use.

Maintaining a Fresh Perspective

Over the past year, I have had the opportunity to support some of the needs of the Krško Nuclear Plant, located in the recently renewed country of Slovenia, in meeting several of the requirements we have

taken for granted in the United States. During development of the Pump and Valve Testing Program for that unit, I found myself providing a significant amount of "justification" for some of the issues in our testing standards. The personnel at Krško were very unhappy that they would soon have to lower their test quality in order to meet our guidelines.

This concept of reducing test quality in order to meet testing standards is not unfamiliar to many of our plants in the United States.

A detailed review of the information gained from several years of testing will provide the entire industry with essential data to support relaxation of the testing frequency or methodology for performance testing of pumps and valves. The use of a fixed testing frequency for plant safety-related equipment should not be considered as a "hands-off" issue with little hope of change, regardless of the historical preponderance of successful test results.

While it is considered possible to make any change necessary via a Relief Request to an existing Pump and Valve Test Program, this can be considered only a partial solution, unless there exists a proactive regulatory condition that supports this level of change.

I would call on our colleagues at the NRC to support proactive change at the strongest level, and to ensure that the desires of the personnel who have a commitment to equipment performance can find some benefit in their extra efforts.

Inservice Testing Bases Program

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ABSTRACT

Inservice Testing (IST) Programs generally address those components necessary for safe operation as specified in Subsections IWP-1100 and IWV-1100 of Section XI, and Position 11 of NRC Generic Letter 89-04. Program tables, generated manually or from computer databases, are frequently used to describe those components to be tested, and testing requirements. Table entries are usually derived from review of system drawings, accident analysis, and other technical documents, but the actual bases for entries are not always documented. Due to the dynamic attribute of IST programs, the continual evaluation of program entries is inhibited if bases for program entries are unclear. When IST Programs have been developed by Architect Engineers, NSSS Suppliers, or other contractors, the bases for the program contents becomes even more nebulous to the program users.

This paper presents a discussion of an inservice testing bases program.

INTRODUCTION

What is an Inservice Testing Bases Program?

An Inservice Test (IST) Bases Program is any systematic approach of developing, and documenting at the component level, the justifications for both the inclusion (or exclusion) of components in the Inservice Test Program, and for the specific testing required for those components. A bases program is a dynamic program which requires administrative controls to maintain it as a current reflection of plant configuration, procedures, and practices. As such, the program should evolve and change as requirements change, plant configuration and procedures change, and as new bases data is developed. Useful bases program output should be provided in easily retrievable component bases criteria.

Every United States nuclear plant operating under Title 10, Part 50 of the Code of Federal Regulation is required to have an Inservice Testing Program. These testing programs are required to conform to both ASME Code and NRC regulatory requirements. However, an Inservice Testing bases program is not required by any regulatory or industry code. Thus, an Inservice Testing bases program is a voluntary non-regulated program which can take on a wide variety of formats.

This paper is an attempt to present our concept of what constitutes the necessary components of an inservice testing bases program. Our paper addresses areas such as:

- Reasons for having an IST bases program.
- Determining what bases are most needed and recommendations on how, and to what extent, bases need to be documented.

- Discussion of some of the possible sources that can be tapped for bases data.

IST BASIS PROGRAM PURPOSE

The single most important impetus to establishing and maintaining an IST bases program is change. Plant configuration, procedures, systems line-up, and safety analysis continuously change. Additionally, pump and valve testing rules and requirements change with new technology, research, and industry experience. Having a clear bases for your present testing activities is essential for efficient and timely reaction to these changes, and to insure that the appropriate level of testing is maintained.

Actions required to comply with regulatory requirements are often recorded in a myriad of documents: FSAR, Technical Specifications, commitment tracking systems, design basis analysis and calculations, various letters between regulators and plant licensees, and memoranda to file. These documents are not always easily retrievable, and because of the large body of documents, it is not always apparent which particular references impact each component of the IST Program. An IST bases program provides a means for cross-referencing, and tracking the various references which impact the IST Program to insure that the appropriate level of testing is maintained.

An IST bases program also becomes a tool for providing consistency in those areas where specific regulatory guidance does not exist. When regulatory guidance does not provide the necessary level of detail, then engineering judgement is often applied and a decision is made as to the type and frequency of testing necessary. When this determination is made by different people or at different times for different components, there exists the risk of inconsistency in the IST Program. By documenting the bases for the level of testing selected, consistency in the testing approach is more likely to be maintained. Documenting test bases provides a full understanding of why the present testing approach is appropriate, and provides the necessary baseline from which to evaluate change.

The IST program typically receives the considerable attention of various reviewers from both within the company, and from external regulators. Frequent questions and concerns arise surrounding the selection of components for inclusion in the IST Program, and in the specifics of component testing. An IST basis program will not completely avoid all inquiries, but by documenting the bases for decisions made in the IST Program, and by establishing an easy method for identifying and retrieving applicable references, the effort required to respond to those questions can be reduced. Additionally, presenting the background and rationale for decisions, and consistently applying those decisions, strengthens management's position.

BASES PROGRAM CONTENT

Our concept of a bases program includes the following documents for the grouping of bases information:

- Guidelines for Development and Implementation of the IST Program
- Component IST Requirements Bases
- Inservice Test Plan

Guidelines for Development and Implementation of the IST Program

Inservice Testing requirements are described in several sources, yet the specifics still remain ambiguous in many areas. The goal of this particular document in the IST Basis program is two-fold: to provide a compilation of all applicable requirements surrounding Inservice Testing, and to provide specific guidance and criteria where the mandated requirements lack the necessary detail. Compiling all the related requirements for Inservice Testing into one document provides an ongoing means for insuring that all commitments are being met. Developing and documenting specific guidance to support and supplement the mandated requirements insures that decisions surrounding the

level of testing for components are applied consistently, rather than on a case by case basis. Ultimately, the level of detail presented should be such that the Component IST Requirements Bases can be developed by applying the requirements contained in this bases document to the individual components.

To provide a complete compilation of all the IST requirements, this bases document must cover all the specific testing delineated in Section XI, subsections IWP and IWV. Therefore, the majority of this document parallels the paragraphs in IWP and IWV. Many of the sections may only be a repetition of the requirements in IWP and IWV. However, some sections may be significantly modified by other mandatory requirements, or may require extensive additional guidance. Therefore, each section should present the following:

- The specific requirements of the applicable section of IWP or IWV
- As modified or extended by other mandatory documents, e.g., Generic Letter 89-04 or IST Plan Safety Evaluation Report
- Detailed criteria and definitions for vague requirements and terms in the referenced mandatory documents. These may be derived from other non-mandatory documents (e.g., Minutes to the Public Meetings on Generic Letter 89-04, or other Codes and Standards) or may simply be the official position of the owner; and
- Reference to all applicable documents used in developing the requirements of each section in order to allow for verification, and to support the positions taken. As an alternative, a Cross-Reference List can be included as an appendix to this bases document.

Appendix A provides an example of the Table of Contents for the Guidelines for the Development and Implementation of an Inservice Test Program. Appendix B provides an example of a Cross-Reference List.

The second goal of this bases document, to provide additional guidance and criteria where the mandated requirements lack the necessary detail, is perhaps the most difficult and potentially, the most controversial. ASME designed Section XI requirements to be deliberately general in many areas so as be relevant to the wide variety of Nuclear Power Plant designs throughout the world. Therefore each owner must interpret the requirements in the context of the design and operation of his particular plant. Naturally, this leads to differing, and sometimes conflicting interpretations of the requirements. These differences may become acute when involving the owner and the regulator. By documenting the rationale and references (if any) to support his position, and by providing evidence of consistency in logic and application, the owner may successfully defend his position. Additionally, the exchange of information, and the dialogue between the owner, the NRC, and ASME could lead to future refinements and improvements in the Inservice Test Code.

Appendix C presents numerous areas where the owner may need to develop additional guidance and criteria. In some cases, additional guidelines and criteria are presented for illustration purposes only. These positions are not to be construed to have the official approval of the Symposium sponsors.

Component IST Requirements Bases

The purpose of the Component IST Requirements Bases is to justify and document the bases for inclusion (or exclusion) of components in the IST Program, and the specific testing required. Each component should be addressed separately (or in small groups where the components perform identical functions), and should address the following:

(1) Component Safety Function:

- Address each Class 1,2,3 pump, power

operated valve, and check valve.
Address manual Class 1,2,3 valves which have a safety function.
Address non-Class 1,2,3 pumps and valves which have a safety function.
Address drivers attached to the Class 1,2,3 pumps which have a safety function.

- For each component capability (open, close, leak tight seal, relieve pressure, provide flow, etc.) detail each condition where that capability would perform a function (there may be multiple conditions/functions for each component capability).
  e.g. Safety Injection check valve

CAPABILITY open

CONDITION LOCA

FUNCTION Provides flow path from the Safety Injection Tanks (SIT) to the Reactor Coolant System (RCS)

FUNCTION Provides flow path from Safety Injection (SI) pumps to the RCS

CONDITION Mode 4-6

FUNCTION Provides flow path for Shutdown Cooling flow to the RCS

FUNCTION Provide alternate boration flow path

CAPABILITY leak tight

CONDITION Modes 1-4

FUNCTION RCS Pressure Boundary (Pressure Isolation Valve (PIV))

- Establish whether each function is a safety function and why.
- Establish whether each safety function is a passive function and why.

(2) Component Test Requirements

For each capability which has a safety function:

- justify acceptance criteria: stroke time, full flow open, (check valves), stroke closed (check valves), partial stroke, leak rate test, torque limits, Safety Analysis vice Sect XI limits, disassembly/inspection, position indication (e.g. solenoid valves).
- Document where tests are performed (matrix).
- Provide a detailed justification for relief requests.

(3) Test Frequency:

- Provide detailed cold shutdown, or refueling frequency justifications.
- Provide detailed justifications where a partial stroke test cannot be performed in lieu of a full stroke.

Inservice Test Plan

The Inservice Test Plan consists of a listing of pumps and valves, and the testing performed on each. In addition, relief from Section XI requirements granted by the NRC are also documented in the IST Plan. The contents of the IST Plan meet, as a minimum, the Section XI requirements for a test plan. It is this document which was previously submitted to the NRC for approval, and is now submitted to the NRC for reference. Essentially, the IST Plan is an abridged and condensed version of the information contained in the Component IST Requirements Bases document.

Although the IST Plan typically contains certain information which in itself constitutes basis information (e.g. the Edition and Addenda of the Code used), it is the general lack of bases information in the IST Plan which necessitates a bases program. The preceding bases documents support the IST Plan by providing the detail and bases for the information contained in the IST Plan. The product of the IST Bases

Program is the IST Plan and the implementing test procedures.

SOURCES OF BASES DATA

The sources of bases data are as varied as the number of document types that can be found at a nuclear power plant. However, certain design, operations, and licensing documents contain information which should be considered when developing and maintaining an IST Bases Program. Such documents include:

- Design documents such as system design criteria, component and system design specifications, piping and instrumentation drawings (P&IDs), seismic qualification packages, safety functional diagrams, instrumentation classification bases, and engineering reports.
- Operations procedures used for system operation and testing.
- Licensing documents such as the Technical Specifications and safety analysis report.

CONCLUSIONS

It is difficult to establish what are the necessary contents and the optimal level of detail in an IST Bases Program. In general, there cannot be too much informantion, if the information is presented in a well organized format. However, there is always a balance that must be achieved between the significant cost of developing an extensive IST Bases Program and the usefulness of that information. This paper has attempted to present only one concept of an IST Bases Program, it is important to establish the beginnings of an IST Bases Program upon which to build. By continuously updating and expanding the bases whenever various aspects of the IST Program are revisited and reconsidered, the Bases Program will eventually achieve the desired level of detail without extensive effort by the owner. This effort will not only produced improved bases documents, but may also lead to refinements in the Inservice Test Plan. The final product is an improved Inservice Test Program with the flexibility to promptly respond to change.

List of Attachments:

Appendix A	Table of Contents (of Basis Document)
Appendix B	Cross-Reference List
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CROSS-REFERENCE LIST

GGNS M189.2

Paragraph No.	Source Document
1.0	10 CFR 50.55a(g)
1.2	IWP-1100 S1, IWV-1100 S1, 10 CFR 50.55a(g), 10 CFR 50 Appendix A GDC-1, Appendix B Crit XI and Generic Letter 89-04 Attachment 1 11 P3
1.4.1	IWP-1200a
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1.4.4	IWV-1200b
2.1	IWV-2100a
2.2.1	IWV-2200a
2.2.2	IWV-2200b
2.2.3	IWV-2200c
2.2.4	IWV-2200d
2.3	10 CFR 50 Appendix J II B
2.4	IWV-2300a
2.5	IWV-2300c
2.6	ANSI N56.8 Section 2
2.7	10 CFR 50 Appendix J II H
2.8	IWV-2300d, IWV-3200 Note 1 and IWP-3111 S1
2.9	Generic Letter 89-04 Meeting Minutes Position 1 Q4 Response
2.14	IWV-2100b
2.15	Generic Letter 89-04 Attachment 1 4a P1 S1
2.16	Generic Letter 89-04 Attachment 1 6 P5 S1
2.19	IWP-3110 S1
2.21	IWV-3413a
4.1	Tech Spec 4.0.5, IWV-3414 S1 and IWP-3400b S1
4.2	IWV-3200 S1 and IWP-3111 S1
4.3	Reg Guide 1.58 (UFSAR Appendix 3A Page 3A/1.58-1)
4.4	10 CFR 50.55a(g), 10 CFR 50 Appendix J and Generic Letter 89-04
4.5	Tech Spec 4.0.2 and 10 CFR 50 Appendix J
4.6	IWV-3426 S1 and IWP-1500 S1&2
4.7	IWA-2120c
5.1	Tech Spec 4.0.5
5.2.2	Tech Spec 4.0.5 and 10 CFR 50.55a(g)
5.2.3	Generic Letter 89-04 Page 3 B P2, P4 S2
5.2.4	Generic Letter 89-04 Attachment 1 P2
6.1	IWP-3400a S1, S2 and S3
6.2	Generic Letter 89-04 Attachment 1 9 P3
6.3	IWP-6230a and b
6.4	Table IWP-3100-1, IWP-3100 S1 and S2
6.5.1	IWP-3110 S1 and S4
6.5.3	IWP-3111 S1

CROSS-REFERENCE LIST

GGNS M189.2 Paragraph No.	Source Document
6.5.4	IWP-3111 S2 and S3
6.5.5	IWP-3112 S1-S4
6.6.1	IWP-3110 S4, IWP-6240c, and IWP-4160
6.6.2	IWP-4150 S1, S2 and S3
6.6.3	IWP-4210
6.6.4	IWP-4240
6.6.5	IWP-4240
6.6.6	IWP-4310 S1 and S2
6.6.7	IWP-4510 S1-S4
6.6.8	IWP-4510 S5
6.6.9	IWP-3100 S1 and S2
6.6.9	IWP-3500a and b
6.6.10	IWP-4600 S1, S2 and S3
6.7.1	IWP-4110
6.7.2	IWP-4140 S1, S2 and S3
6.7.3	IWP-4520a and b
6.7.4	IWP-4130 S1 and S2
6.7.5	IWP-4230 S1 and S2
6.8.1	IWP-3100 S4
6.8.2	Generic Letter 89-04 Attachment 1 8 P6 and IWP-3230b, c and d and IWP-3100 S4
6.8.3	IWP-3230a
6.8.4	IWP-3100 S3 and IWP-3230c
7.1.1	IWV-3200
7.1.2	IWV-3300 and IWV-3412b
7.1.3	IWV-3415 S1
7.1.4	ANSI N45.4-72 4.2
7.1.6	Generic Letter 89-04 Attachment 1 10 P1 and P2
7.2.1	IWV-3411 and IWV-3412a S1 and S4
7.2.2	IWV-3412a S2
7.2.3	IWV-3412a S3
7.2.4	IWV-3416 S1 and S2
7.2.5a	Generic Relief Request #5
7.2.5b	IWV-3412a
7.2.5c	IWV-3412a
7.2.6	IWV-3422, 10 CFR 50 Appendix J III D3, Tech. Spec. 4.4.3.2.2 and Tech. Spec. 4.6.1.2 f/j - 3
7.2.7	IWV-3300
7.2.8a	Table IWV-3510-1
7.2.8b	UFSAR 5.2.2.10
7.2.9	IWV-3610 S1
7.2.10	IWV-3620
7.3.1	IWV-3522 S1, b S1, S3, S5, S6 and S7, Generic Letter 89-04 Attachment 1 2 P1 and P2, and Generic Letter 89-04 Meeting Minutes Q8 Response

CROSS-REFERENCE LIST

GGNS M189.2 Paragraph No.	Source Document
7.3.2a	IWV-3522b S2
7.3.2b	IWV-3522b S2
7.3.2c	IWV-3522b S2
7.3.2d	IWV-3522b S2
7.3.2e	Generic Letter 89-04 Meeting Minutes Q1 Response
7.3.2f	Generic Letter 89-04 Meeting Minutes Q1 Response
7.3.3	IWV-3522b S4 and S5
7.3.4	IWV-3200
7.3.5a	Generic Letter 89-04 Attachment 1 2 P3 a
7.3.5b	Generic Letter 89-04 Attachment 1 2 P3 b
7.3.5c	Generic Letter 89-04 Attachment 1 2 P3 c
7.3.5d	Generic Letter 89-04 Attachment 1 2 P3 7.3.5e Generic Letter 89-04 Attachment 1 2 P3
7.3.5f	Generic Letter 89-04 Attachment 1 2 P5
7.3.5g	Generic Letter 89-04 Attachment 1 2 P6
7.3.5h	IWV-3200
7.3.6	IWV-3417b S1 & S2 and Generic Letter 89-04 Attachment 1 1 P1
7.4.1	IWV-3522a S1 and S2
7.4.2	IWV-3522a S2 and Generic Letter 89-04 Attachment 1 2 P2
7.4.3	IWV-3417b S1
7.5.1	IWV-3421 S1 and S2
7.5.2	IWV-3423 S1 and 10 CFR 50 Appendix J III c1
7.5.3a	IWV-3424a
7.5.3b	IWV-3424b
7.5.3c	IWV-3421 S2 and 10 CFR 50 Appendix J III c1
7.5.4	IWV-3423a - d
7.5.5a	IWV-3423e & f
7.5.5b	Tech. Spec. 4.6.1.2d & h
7.5.6b	ANSI N56.8 3.3.6
7.5.7	IWV-3426 S1
7.6.1	IWV-3610 S1 and S3 and Tech. Spec. 3.1.5d
7.6.2	IWV-3610 S2 and S4 and Tech. Spec. 3.1.5d
7.7.1a	IWV-3512 S1
7.7.1b	ANSI/ASME-PTC 25.3-76 1.01f
7.7.1c	IWV-3512 S2
7.7.1d	ANSI/ASME-PTC 25.3-76 0.01
7.7.1e	ANSI/ASME-PTC 25.3-76 0.03
7.7.2a	IWV-3513 S1 & S2 and Table IWV-3510-1
7.7.2b	IWV-3514 and IWV-3200
7.8.1a	IWV-3413a
7.8.1b	Generic Letter 89-04 Attachment 1 5 P7
7.8.1c	IWV-3413a

CROSS-REFERENCE LIST

GGNS M189.2 Paragraph No.	Source Document
7.8.2	AECM 85/149
7.8.3	AECM 85/149
7.8.4	AECM 85/149
7.8.5	IWV-3417
7.8.6	IWV-3417
7.9	IWV-3620 S1 and S2
8.1	IWV-6210 and IWV-1400 and IWP-6210 S1
8.2	IWV-6220
8.3	IWP-6220a and b
8.4	IWP-6240a through f, IWP-6250 and IWV-6230
Table A	Table IWP-3100-2 and Generic Relief Request #1
Table B	Table IWP-4110-1
Table C	Table IWV-3700-1

IWP-3000/IWV-1100 SCOPE

What is an accident?

- Is this limited to accidents listed in FSAR Chapter 15?
- Is a fire an accident?
  - external toxic gas?
  - evacuation of the Control Room?
  - seismic events?
  - severe weather?
  - flooding?

What systems are required to mitigate the consequences of an accident?

- Is this limited to those systems identified in the FSAR as a safety related system? (chapter 7)

- Does the Post Accident Sampling System mitigate the consequences of an accident?
  - Remote Shutdown Panel?
  - Containment Hydrogen Analyzer?
  - Containment Vacuum Relief?
  - Fire Protection systems?
  - Fuel Pool Cooling?

For a required system, what component functions are required to mitigate the consequences of an accident?

- Components in a required process flow path

- Components that prevent diversion of a required process flow

- Class break boundaries?

- Components that maintain the safety system in a standby condition (e.g. maintain a pressure or volume source)?

- Components that protect a safety system from damage (e.g. system relief valves, vacuum breakers)?

- System control valves?

- Components that are directed to be operated in the Emergency Operating Procedures?

What failures should be considered when determining safety function.

- single active failure?
- operator error?
- any number of non-tested component failures?
- passive component failures?

IWP-4100 INSTRUMENTS

How is instruments accuracy calculated?

IWP-2100 ACTIVE AND PASSIVE VALVES

Are all valves which receive a safety signal active?

Are valves which are 'normally' in their safety position passive, even though they may occasionally be removed from their safety position

- for testing?
- for maintenance?
- during normal evolutions? infrequent evolutions? off normal evolutions?

Can check valves ever be passive?

Does the FSAR Active Valve List fully define active valves?

IWV-2200 CATEGORIES OF VALVES

What is a Category A valve? All valves which have a closed safety function must interrupt (limit) flow. At what point does this become a Cat A function?

IWP-2300 OTHER DEFINITIONS

Exercising - Must check valves always be testing in both directions?

IWP-3300 VALVE POSITION INDICATOR VERIFICATION

Which valve position indications must be verified?

- Only those used to time valves?
- All indications on tested valves?
- All indications which perform a safety function, or are used to verify a safety function?

IWV-3412/IWV-3521 EXERCISING PROCEDURE

What criteria should be used to determine what is impractical?

- Human safety?
- Equipment damage?
- Radiation Exposure?
- Technical Specification requirements?
- Component failure would cause a loss of system function?
- Costs?

IWV-3413 POWER OPERATED VALVES

How is the limiting value of stroke time determined?

An Update To Inplace Testing Of Safety/Relief Valves Utilizing Lift Assist Technology

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ABSTRACT

Inplace testing of safety and relief valves with lift assist devices has received mixed reviews from nuclear power plant testing personnel. While many plants use the technology, most limit it's use to testing main steam safety valves (even though both OM-1-1981 and PTC 25.3-1976 allow it's use for several different service applications). Test coordinator concerns regarding the technology range from lift setpoint accuracy and repeatability, to the quality of the test result output. In addition, OM-1-1981 and PTC 25.3-1976 differ in their approach to the technology.

PTC 25.3-1976 allows the use of lift assist devices to test any compatible valve, regardless of the system service. OM-1-1981 allows the use of lift assist devices for compressible fluid systems. However, it recommends not using such devices for liquid service systems. The reasons for the differences between PTC 25.3-1976 and OM-1-1981 will be discussed along with additional considerations applicable to the utilization of the technology in testing liquid service valves.

Advances in assist devices over the last few years have improved the accuracy, repeatability, and quality of test result output. This paper will show that lift assist technology is quite capable of determining lift setpoints within the accuracy requirements of OM-1 and PTC 25.3. It will also demonstrate that the technology should not be limited to compressible service systems. Also, improvements in test repeatability and output quality will be discussed as a function of the assist device design used and valve characteristics.

Lift assist testing is often preferred over performing inplace testing using direct system pressure. It is often more cost efficient than bench testing because it does not require removal of critical systems from service and the transportation of components. Also, duplicating system temperatures and other environmental factors is not an issue during inplace testing. Valve testing that once required an outage and maintenance period can now be conducted prior to such periods. This approach minimizes the possibility of failures becoming critical path limiting items.

In summary, the full potential of lift assist technology is not widely understood. As a result, the technology is under utilized in most plants, despite its many advantages.

BACKGROUND

In September 1991, I provided engineering support for a vendor at a nuclear power station for main steam safety valve testing utilizing a lift assist device. I was impressed with the ease and expediency of the tests, and was curious to know to what extent the practice was being used in the nuclear industry. During the following months I conducted a phone survey of several power stations to determine: (1) If lift assist testing was being performed at the station, and if so, on which valves was the device used, (2) How familiar were they with the technology, and (3) What opinion did they have of the technology. The result was an interesting mix with most of the responses falling into three major categories: (1) Yes we use the technology and are satisfied with the results, (2) We have used the technology in the past and were not satisfied with the results OR We are not convinced of the technology and thus do not use it, and (3) We are not very familiar with the technology.

Of the stations that have either tried or presently use lift assist for testing, the overwhelming majority only use the technology for steam service valves (main steam line safety valves and some pressurizer safeties for PWRs). The technology is viable for many plant systems and should receive consideration for different uses in the industry.

ADVANTAGES OF LIFT ASSIST TECHNOLOGY

Let us assume that lift assist device brand X is 100% accurate and fully compatible with any system valve. With such a fictional device we can analyze the full potential of this technology in the nuclear industry.

1. Since normal system operating conditions can be utilized in conjunction with a lift assist device to measure a valves set pressure, it is therefore possible to test most valves during power operation without having to remove the associated system from service. Thus, by conducting a test prior to cold shutdown or refueling, a satisfactory valve will no longer be an outage concern. Also, should a valve failure occur that would require a Limiting Condition for Operation (LCO) action statement, the planned reactor shutdown will normally occur before the LCO time period expires.

2. Conducting valve testing inplace under normal system operating conditions eliminates the concern with duplicating or addressing the valve's environmental affects (i.e. test fluid used, back pressure, normal ambient temperature, etc.) during shop testing.

3. For valves in radiological or contaminated systems, testing the valve inplace will virtually eliminating rad-waste, decontamination and transportation associated with shop testing of these valves.

4. Once an established test procedure is in place, the manhours spent preparing for, and performing a valve test with a lift assist device will usually be significantly less than the manhours required to remove, transport, test and reinstall the valve.

5. In most cases, testing an entire population of valves (e.g. main steam safeties) would require less time than it would take to remove the sample group. Testing with a lift assist device would prevent a sample group valve failure from creating a last minute crisis during an outage.

DISADVANTAGES OF LIFT ASSIST TECHNOLOGY

1. Certain relief valve designs preclude utilizing this technology. A valve must have the ability to have it's lift stem exercised by an external mechanical force.

2. Not all systems are accessible during normal power operations and thus would required additional planning and coordination to test these system's valves.

3. Lift assist devices are not created equal. Each device must be evaluated on it's own design and capabilities. Some key attributes a device should have are:

   (a) It will not interfere with the valve's ability to self actuate when the device is attached during a test.

   (b) It conducts a straight lift on the stem to prevent damaging the stem or packing, and also eliminates the need to address the reduced force vector created in the set pressure calculation.

   (c) Will provide immediate assistance with valve closure in the event that a valve sticks after opening. The device should be capable of performing this function without damaging the device or the valve.

   (d) Proven repeatability.

   (e) Has the ability to provide an undisputable proof of valve lift and readily identifies all measured quantities at the time of the lift.

   (f) Proper documentation of test equipment accuracies to ensure that applicable code accuracies are met.

4. If a valve fails during power operation, the station must address the affect the failure has on the safe operation of the reactor. Valve failure concerns are very dependent on system safety function, limiting conditions for operation, and the type of failure the valve experienced.

CODE CONSIDERATIONS

ASME Section XI Code editions prior to the 1986 Code reference ANSI/ASME-PTC 25.3-1976 for conduct of safety and relief valve testing. Section 4.091(a) PTC 25.3-1976 discusses valve set pressure and operational test methods for a valve installed on a fluid system. Subsection (2) specifically states "On valve installations where the system pressure may not be conveniently increased to the set pressure of the valve, the set pressure of the valve may be determined with calibrated hydraulic or pneumatic-assist equipment." This method of set pressure testing is allowed for any fluid system (liquid or compressible) covered in the Power Test Code. Also, section IWV-3512 of the 1980 and 1983 Section XI Codes waives the Category A leak test requirements for applicable valves if a lift assist device is used for set pressure testing.

The 1986 edition of ASME Section XI references ANSI/ASME OM-1-1981 for conduct of safety and relief valve testing. Sections 4.1.1.3, 4.1.2.3, 8.1.1.3 and 8.1.2.3 state "Assist devices may be used for set pressure testing, provided the calibrated accuracy meets the requirements of 1.4." for steam service and compressible fluid services other than steam. Sections 4.1.3.3 and 8.1.3.3 state "Assist devices to determine set pressure are not recommended for liquid service relief valves". Additionally, the station is required to perform seat leakage testing in most cases.

LIQUID SERVICE CONCERNS

Why the difference in handling liquid service valves? Several factors must be considered. When relief valve lifts on an overpressure condition, the valve remains open until the system pressure is reduced by a set amount (blowdown), then the valve shuts to allow proper system pressure restoration. When an assist device lifts a valve, the valve is open for a set amount of time based on the control characteristics of the device. For compressible fluids, the amount of system energy lost during the valve lift is negligible and thus the affect on system pressure is minimal. However, for a non-compressible fluid (liquid service), the amount of system energy lost during a lift of the same duration is much greater and has a significant affect on system pressure. Thus pressure control of an operating liquid service system could be compromised during a relief valve test with a lift assist device.

This dramatic affect on a liquid service system's pressure creates another problem for lift assist usage. Most assist systems are designed to measure the force required to overcome the valve spring force, and the primary method of determining valve lift is by observing the change in this force when the valve lifts (Figure 1). The observed change is due to the increase in the acting area of the disc once the valve lifts. For a liquid service system, the decrease in system pressure during lift is much greater and occurs quicker than the decrease observed in a compressible fluid system. Thus, since the liquid pressure drops quickly after lift, there is no significant increase in force due to the increase in the acting area of the disc. This results in a severe degradation in the ability to determine the point of valve lift by force trace observation.

So why not test liquid service valves when

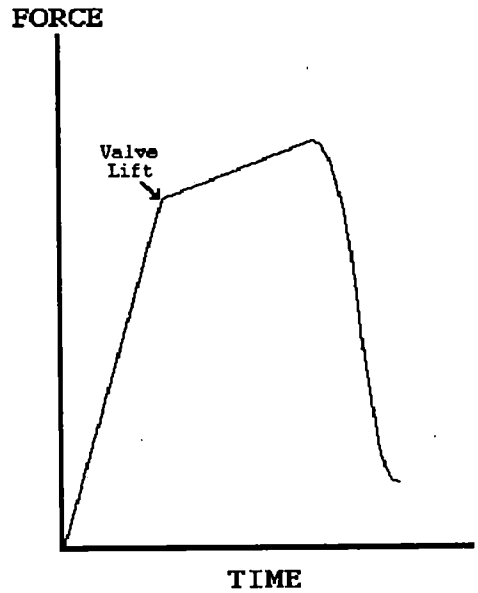

Figure 1. Force Change during valve lift.

system pressure transients are not a concern? Most relief valves inlet piping is a small line attached to a larger main process line. The liquid in the inlet piping is usually in a static condition even when flow is established in the main process line. A corrosion layer will exist in the inlet pipe which is loosened by the combination of mechanical shock and the sudden flow through the line when the relief valve lifts. The amount of flow through the relief valve during a typical lift is not sufficient to flush these corrosion products clear of the line and valve seat area, thus when the valve reseats, some of the corrosion products are trapped between the disc and the seat. The potential for seat leakage under these conditions is greatly increased (and OM-1 requires seat leakage testing). To affectively flush the line to prevent possible seat damage, the valve disc will need to remain open longer than is required to verify set pressure. More system energy is lost thus creating a larger pressure transient.

Catch 22? Not really. It should be noted that it is difficult to create a generic test device that

is capable of accomplishing satisfactorily results for all testing situations. Liquid service testing requires more forethought and planning to utilize this technology effectively. The lift assist devices utilized today are capable of testing liquid service valves with some modification. The ability to determine valve lift is not an insurmountable obstacle. Multichannel recorders can be used effectively to measure more than one test parameter on the same time scale, and thus would allow monitoring for a system pressure drop or an acoustic signature to identify the moment the valve lifts. Multichannel recorders are in use today. The modifications required to test equipment and testing methodology are not significant to improve a lift assist device's ability to determine liquid service valve set pressures. Although OM-1 recommends that assist devices not be used for liquid service valves, it does not forbid the use of such devices.

CAPABILITIES

Accuracy, repeatability, and test result output quality were the primary concerns expressed by power stations during my initial survey. These issues are valid reasons to be critical of an individual lift assist device. If a specific device design can not meet Code criteria and power station standards, then it should not be considered for Code testing.

Accuracy

Accuracy is of paramount importance for any type of safety/relief valve testing. Total accuracy for set pressure testing utilizing a lift assist device is dependent on the following factors:

1. System pressure during the test and the accuracy of the measurement of that pressure. System pressure will determine load cell requirements for the test.

2. The loop accuracy and repeatability of the device test equipment (i.e. load cell, transducers, signal amplifiers, recorders, etc.).

3. Calibration standard accuracy (i.e. ring gage, high accuracy pressure gage, etc.).

The following example illustrates how these factors affect the overall test accuracy for a standard assist device.

Equipment		Accuracy/Repeatability (full scale)
System pressure high accuracy pressure gage (2000 psig)	(A_G)	+ 0.5% - 0.0%
Ring gage accuracy (calibration standard)	(A_{RG})	± 0.5%
Load cell repeatability	(A_{LC})	± 1.0%
Recorder accuracy	(A_R)	± 1.0%

The loop accuracy (A_{Loop}) for the load cell, signal amp and recorder can be calculated by the square root of the sum of the squares of the accuracies (ISA-S67.04, 1982).

$$A_{Loop} = (A_{LC}^2 + A_R^2)^{1/2} \qquad (1)$$

From equation (1), A_{Loop} is found to be 1.41%.

For a safety valve with the following characteristics;

Set pressure	(P_{SET})	1050 psig
Affective diameter	(D)	5.25 inches
System Pressure	(P_{SYS})	800 psig
Discharge to atmosphere		

the Acting Area ($Area_{ACT}$) of the disc is given by;

$$Area_{ACT} = \pi * (D / 2)^2 \qquad (2)$$

and the force required to overcome the spring force (F_{SPR}) is;

$$F_{SPR} = P_{SET} * Area_{ACT} \quad (3)$$

$$F_{SPR} = F_{SYS} + (F_{LAD} - F_W) \quad (4)$$

where:

F_{SYS} = the force due to system pressure (P_{SYS} * $Area_{ACT}$).

F_{LAD} = the force supplied by the lift assist device.

F_W = the force due to the weight of the lift assist components attached to the valve stem. (10 lbf will be used for this example)

Solving equations (2), (3) and (4) for F_{LAD} yields 5422 lbf required by the lift assist device. This would require a 10,000 lbf load cell and ring gage. Since A_{Loop} is ± 1.41%, and A_{RG} is ± 0.5% , then total accuracy of the force recorded will be ± 1.91% or ± 191 lbf for a 10,000 load cell arrangement. The accuracy contribution by the lift assist device (A_{LAD}) is;

$$A_{LAD} = \pm 191 \text{ lbf} / Area_{ACT} = \pm 9 \text{ psig} \quad (5)$$

and the contribution due to A_G is +10 psig to -0 psig. Thus the overall accuracy is +19 psig to -9 psig which is +1.8% to -0.86% of P_{SET}.

If this above system pressure was high enough so that a 5000 lbf load cell could have been used, the accuracy of the device would have been +14.4 psig to -4.4 psig, corresponding to +1.4% to -0.4% of P_{SET}.

The above example shows that existing lift assist devices are capable meeting the accuracy of +2% to -1% required by Section 1.4.1.2 of OM-1-1981. The accuracies used for this example are reasonable values for standard test equipment in the market. Equipment with a higher degree of accuracy is available and should be preferred.

As shown above, the accuracy of a particular device is primarily dependent upon the method of load measurement and means system pressure measurement.

Repeatability

A particular difficulty with the devices used today is the simultaneous measurement of all the required test parameters at the moment of lift. The potential for error increases when separate means of measurement are utilized. The judgement calls and timing necessary for separate measurement makes repeatability more of a human factor than an actual device issue. If a multichannel recorder is used to measure all parameters, human error is removed from the problem and repeatability rests solely on the lift assist device equipment.

Determination of valve lift is the next major repeatability issue. If the moment of lift is not well defined, judgement again becomes the critical factor in repeating test results.

For test devices utilizing a load cell arrangement, the repeatability of the individual load cell used is the major test equipment factor. Load cell variance can be included in the loop accuracy (as done in the accuracy example above) to determine the significance of it's effect on test parameter measurement. If the total accuracy is within the requirements of the Code, then the repeatability of the load cell should not be a concern.

Lastly, we have the valve itself. The first as found lift performed must overcome the breakaway and initial frictional forces inherent of a valve that is not periodically exercised. After the first lift, the valve may experience thermal

effects depending on the valve service. If the valve temperature varies significantly between successive lifts, the spring constant will vary accordingly. The ability to accurately predict the cumulative effect that valve characteristics have on set pressure is difficult at best, thus it is very important to let a valve reach it's equilibrium condition prior to commencing another set pressure test. The repeatability of a valve should be a minor issue when testing is conducted properly.

Test Result Documentation

Recorder output must have sufficient quality, and should be of a nature such that the validity of recorder output can not be disputed. Thermal trace recorders in use today provide excellent quality and permanence. Additionally, a multichannel recorder, when utilized for multiple test parameter input, will provide the actual test parameters at the time of lift in a format that will prevent transcription errors and additional documentation tracking.

Digital and computerized test equipment is also available for incorporation into existing lift assist device designs. By using digital format the need to visually interpret or interpolate is virtually eliminated, thereby improving the overall confidence of the test results.

SUMMARY

Lift assist technology is definitely a viable means of set pressure testing. The accuracy of the technology is consistent with Code requirements and repeatability can be achieved with proper use. As with any unique method of testing, the limitations as well as the capabilities should be fully understood. If the limitations of one device and/or testing methodology restricts the use for a desired application, and the potential benefit gain justifies using the technology (such as radiological systems), then strong consideration should be given to alternative device designs and/or test methods. Liquid service valves should be reviewed for such potential benefits. The capability to adequately test such valves is achievable and should not be overlooked because it does not fit the standard test methods in use today.

REFERENCES

ANSI/ASME OM-1-1981, "Requirements for Inservice Performance Testing of Nuclear Power Plant Pressure Relief Devices".

ANSI/ASME PTC 25.3-1976, "Safety and Relief Valves, Performance Test Codes".

ISA-S67.04, 1982, "Setpoints for Nuclear Safety-Related Instrumentation Used in Nuclear Power Plants".

A Review of Historical Check Valve Failures

by

M.D. Todd
D.A. Casada

ornl

Prepared for: July 21-23, The Second NRC/ASME Symposium on Pump and Valve Testing

OM-22, *Performance Testing of Check Valves* , has been Pursuing Dual Paths in Our Standard Development Process

- Development of a stand-alone OM-22 standard

- Piecemeal changes to the existing ISTC code
  - Non-intrusive testing
  - Disassembly and examination
  - Changes to both areas have been approved

The Working Group Searched for Existing Literature to Support Disassembly and Examination Changes

- Among the issues for which we sought supporting technical bases was definition of appropriate frequency of disassembly & examination
- Only one published document was found: a paper published by Scott (INPO).
- After consideration, the Working Group adopted the paper as our working basis

M.D. Todd
D.A. Casada

Prepared for: July 21-23, The Second NRC/ASME Symposium on Pump and Valve Testing

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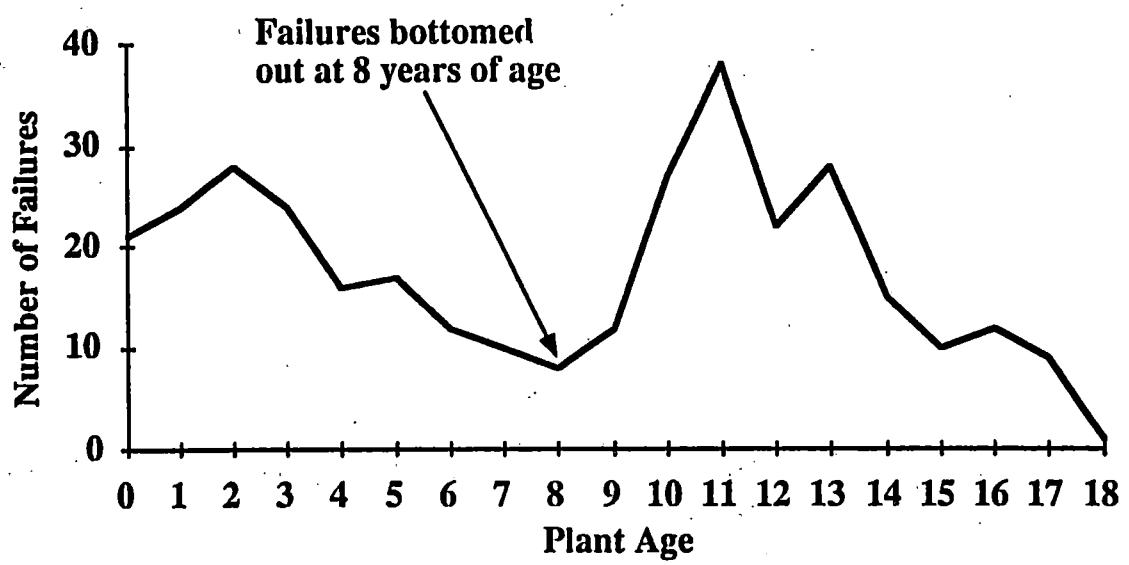
The Working Basis Fundamentally Assumed that an Eight Year Disassembly & Examination Frequency (all Valves in a Group) Would be Adequate

The Working Group's Basis for Disassembly & Examination Frequency



Based on data from: *Check Valve Failure Trends in the Nuclear Industry*, by M. L. Scott, INPO. Proceedings: EPRI Power Plant Valves Symposium II, EPRI NP/GS-6327, July 1989.

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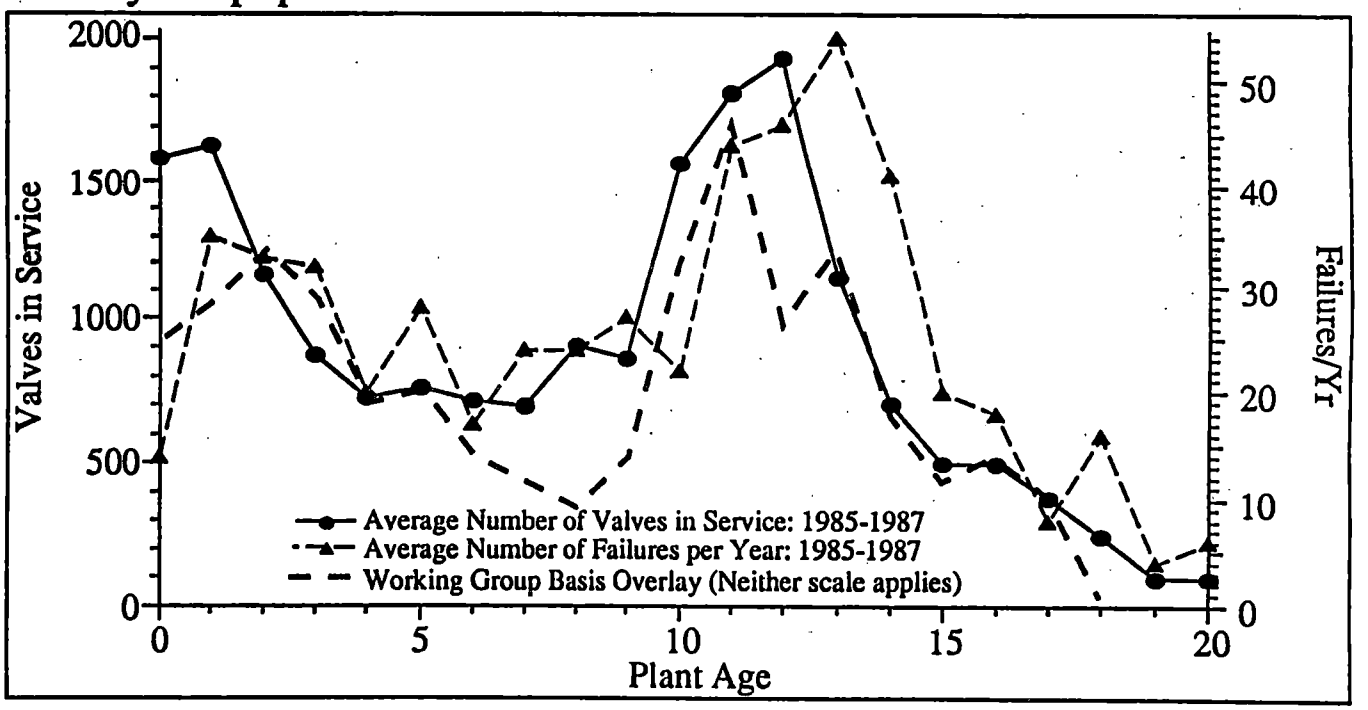
Based on a Request from the NRC, ORNL Performed a Review of Failure Trends and Population Distribution

- The Working Group basis appears to be strongly influenced by the population of valves in service



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As a Result of the Initial Data Review, ORNL is Conducting a More Comprehensive Study of Failure Data

- Evaluating failures that involve significant internals degradation
- Further characterizing the data to consider not just plant age, but a variety of other parameters, such as valve size, method of detection, system of service, failure mode, and others
- The review is currently underway. A draft NUREG/CR report is expected to completed this fall.
- We solicit your critical examination of our study

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The Failure Review Process Involved Five Steps:

- Downloaded all failures occurring through 1990 from NPRDS and reviewed the narrative of each failure
- After reviewing each failure narrative, several failures were eliminated
  - External leakage and minor internal leakage
  - Non-check (e.g., relief) valve failures
  - Failures occurring before 1984
- Characterized each failure in a variety of ways
- Normalized failures to account for population sizes and population service life
- Subjected process to two independent reviews

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The Following Example Using One-inch Valves Shows The Normalizing Process

Year	≤1984	1985	1986	1987	1988	1989	1990
Valves added in service	1365	214	264	111	23	206	7
Years of service (1984-1990)	7	6	5	4	3	2	1
Valve-years	9555	1284	1320	444	69	412	7

Total Failures for One-inch Valves = 50

Total Valve-years For One-inch Valves from Chart Above = 13091

$$\text{Failure Rate} = \frac{\text{Number of Failures For One-inch Valves}}{\text{Total Valve-years For One-inch Valves}} = \frac{50}{13091} = 0.0038$$

$$\text{Population Failure Rate} = \frac{\text{Total Failures For All Sizes}}{\text{Total Valve-years For All Sizes}} = \frac{1235}{134846} = 0.0092$$

$$\text{Relative Failure Rate} = \frac{\text{Failure Rate}}{\text{Population Failure Rate}} = \frac{0.0038}{0.0092} = \boxed{0.417}$$

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The Failures Were Characterized By Several Means:

- NSSS
- Valve size
- Plant age
- Manufacturer
- Plant system and system status
- Failure mode
- Method of failure detection (two approaches)

Several Terms Used in This Presentation Need To Be Defined:

- Programmatic monitoring includes scheduled testing and maintenance, e.g., leak tests, inservice tests, and disassembly and examination
- Examples of non-programmatic detections include pump rotation, elevated temperatures/pressures, or abnormal noise
- Failure seriousness classifications
  - Serious: stuck shut or open, valve part unattached or broken
  - Moderate: excessive leakage, loose parts, reduced flow, restricted motion, corrosion/wear
  - Light: externally-induced (stem or arm) failures, leakage exceeding administrative limit

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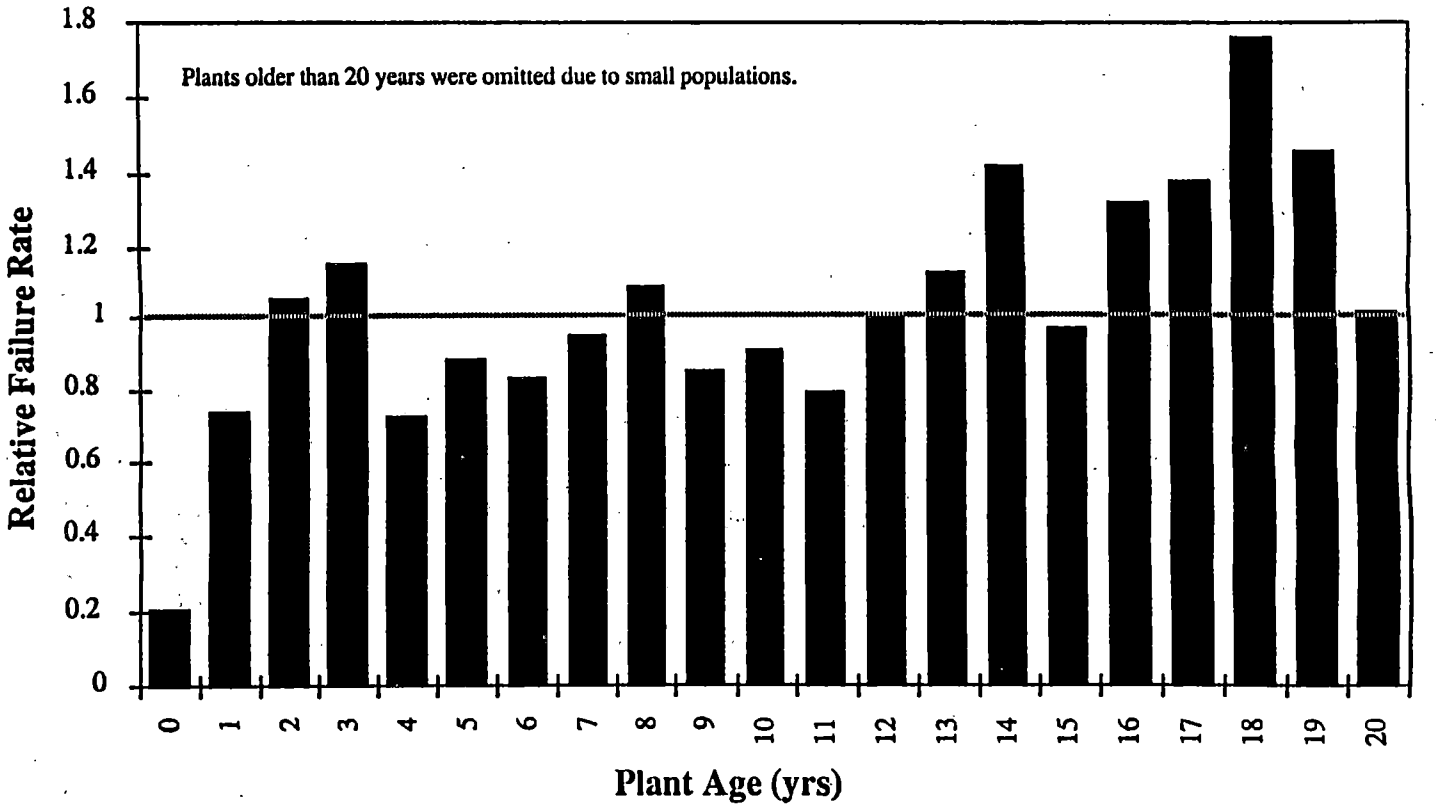
When Considering Individual Plant Ages, Failure Rate Only Roughly Increases with Plant Age



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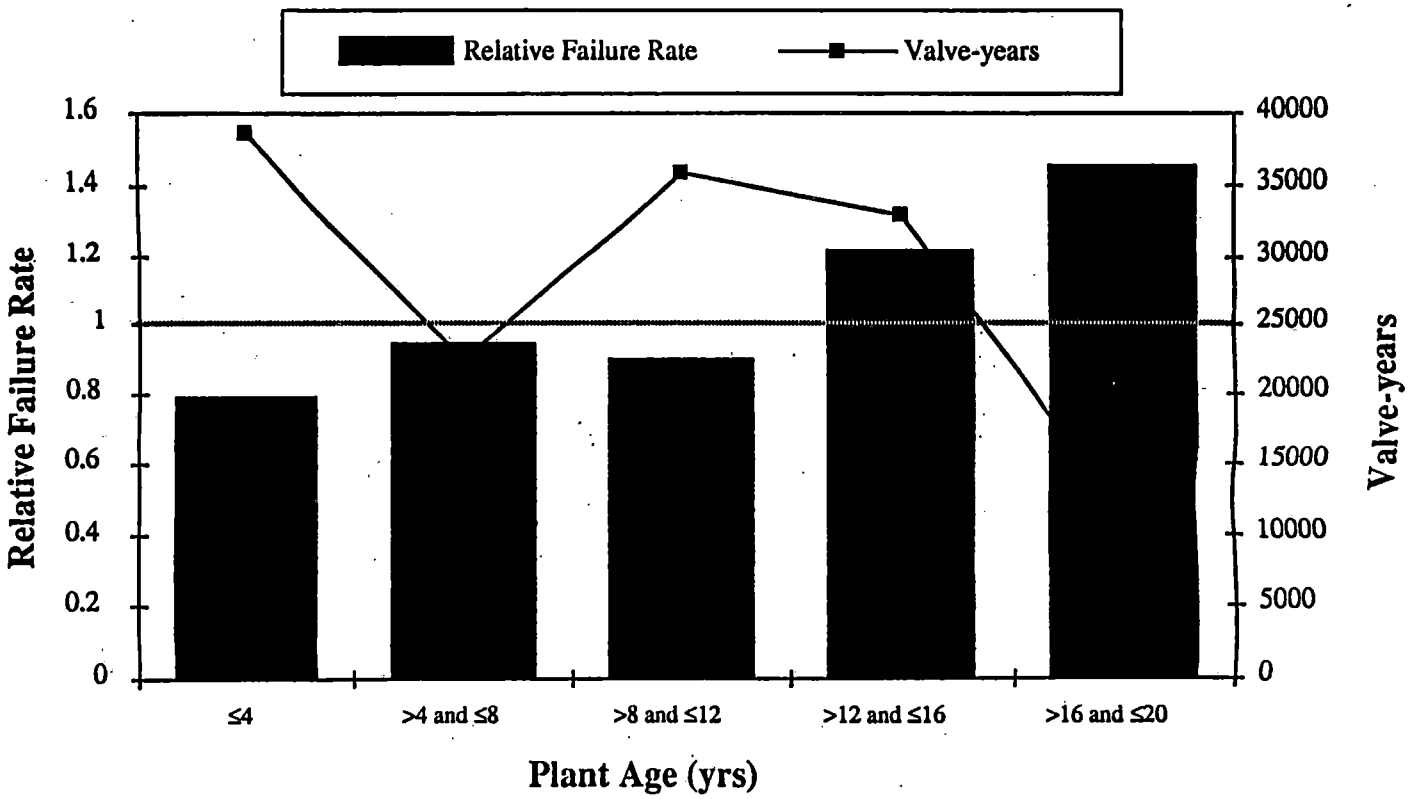
When Plant Ages are Grouped, it is Clear that Failure Rate Is Roughly Proportional to Age



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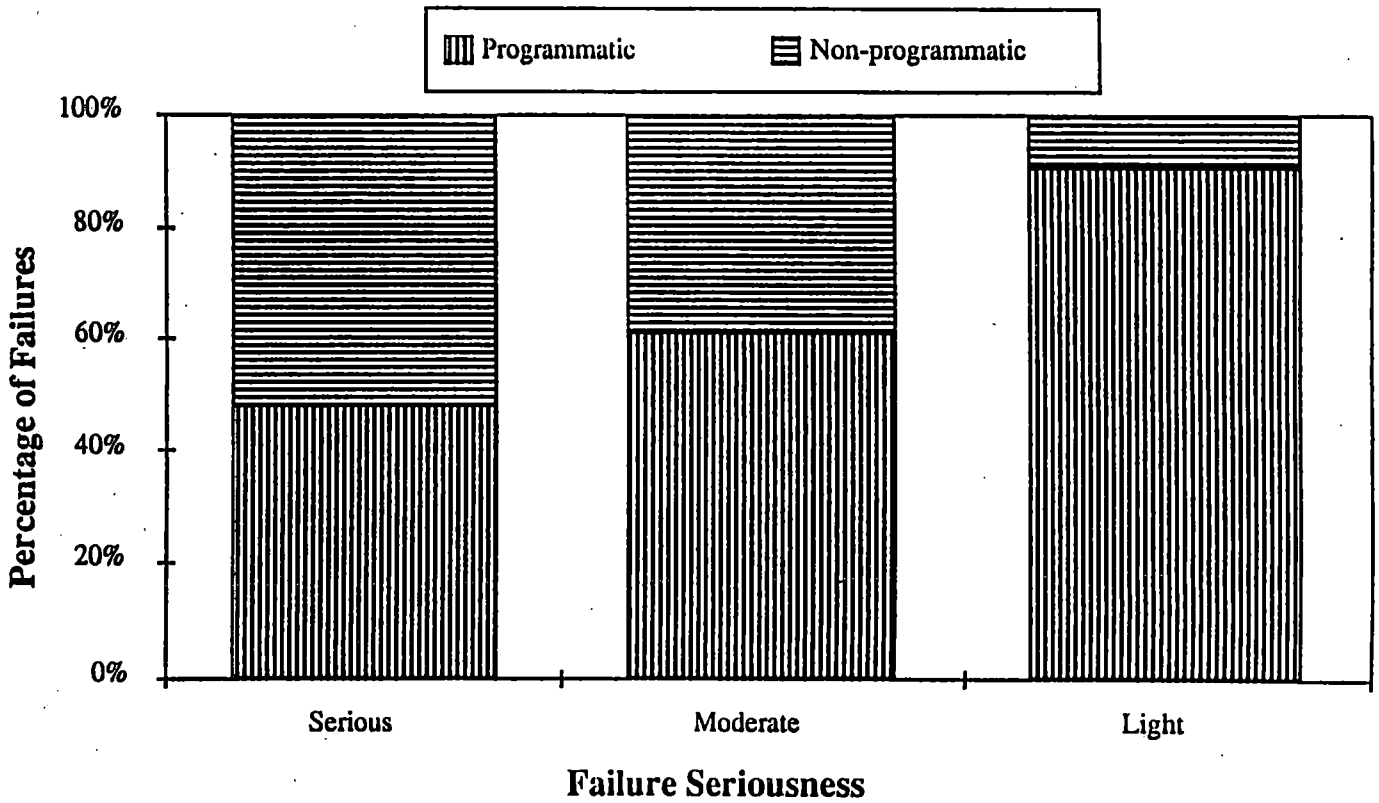
Programmatic Monitoring Is Better at Detecting the Less Serious Failures than the Most Serious Failures



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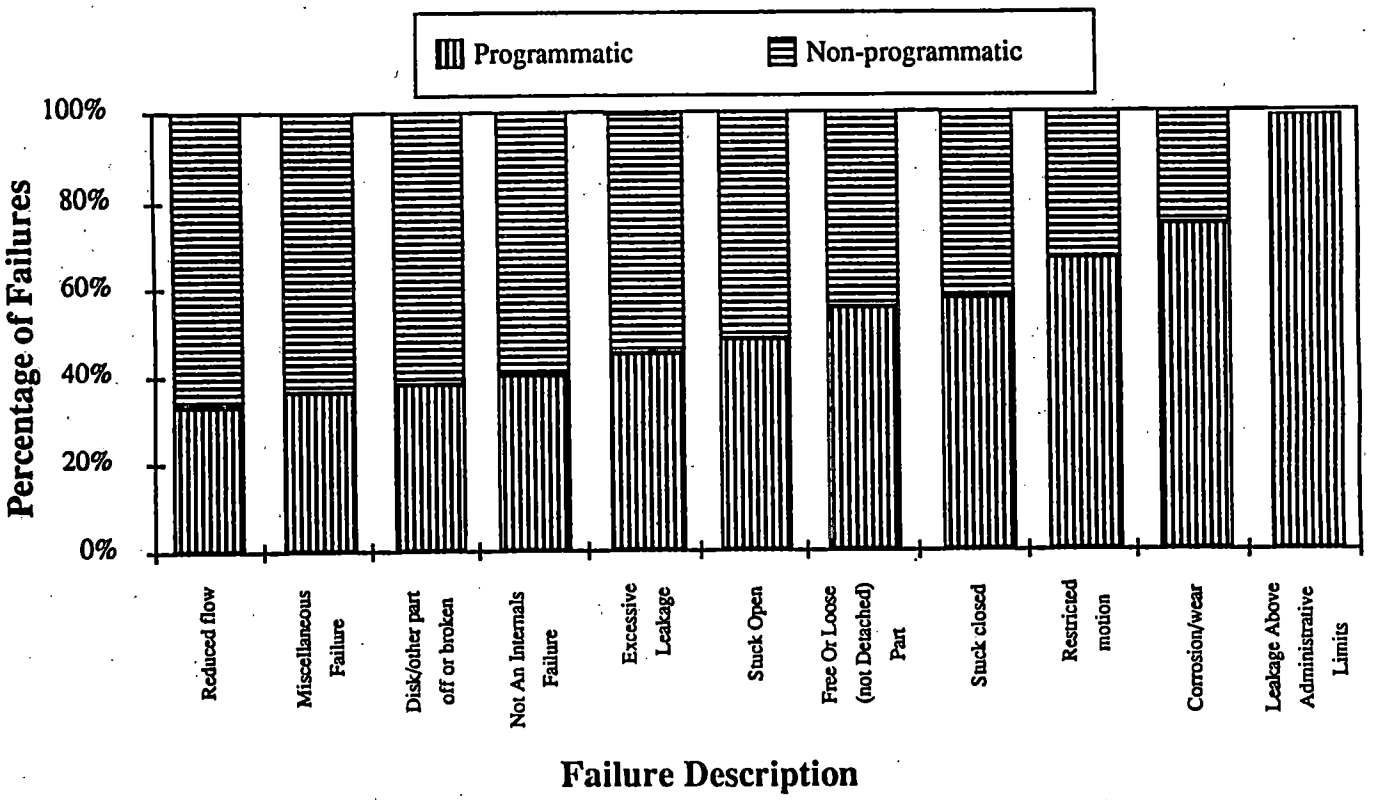
A Significant Number of Failure Modes Are Not Being Detected By Programmatic Monitoring



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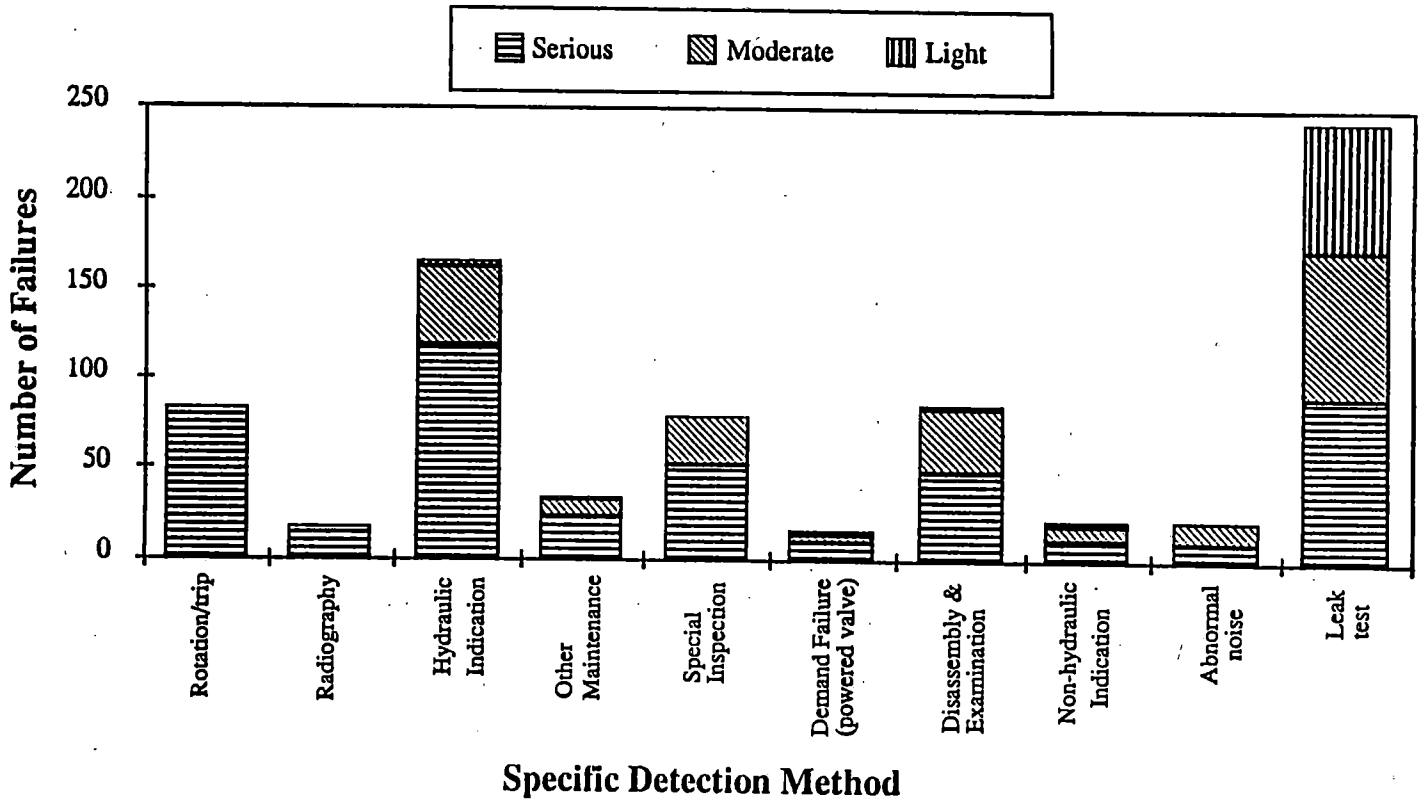
Some Detection Methods Had Greater Success in Detecting The Most Serious Failures



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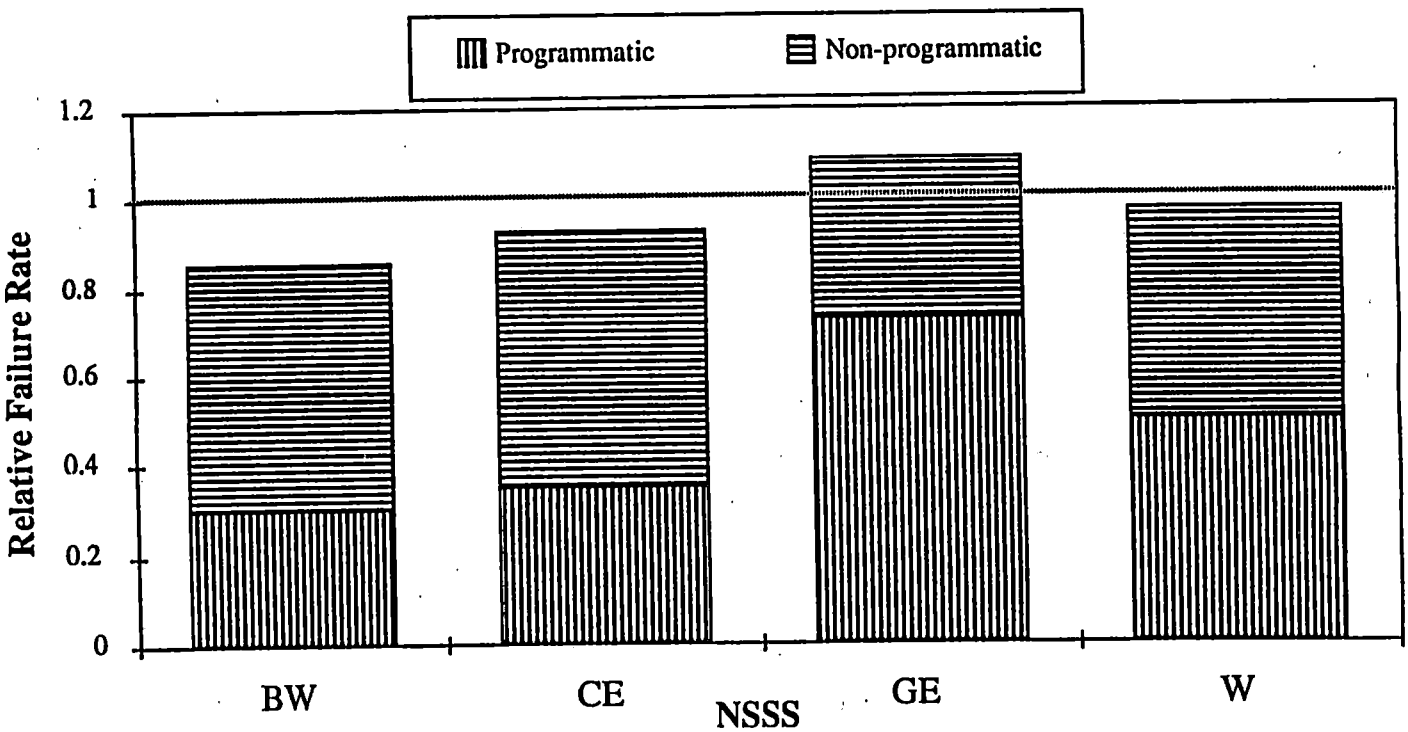
Although the Total Failure Rate of BWR's is Higher, the Failure Rate of Valves Not Detected by Programmatic Monitoring is Lower

- An average 55.5% of BWR and 32.1% of PWR valves are leak tested (6 plant sample)



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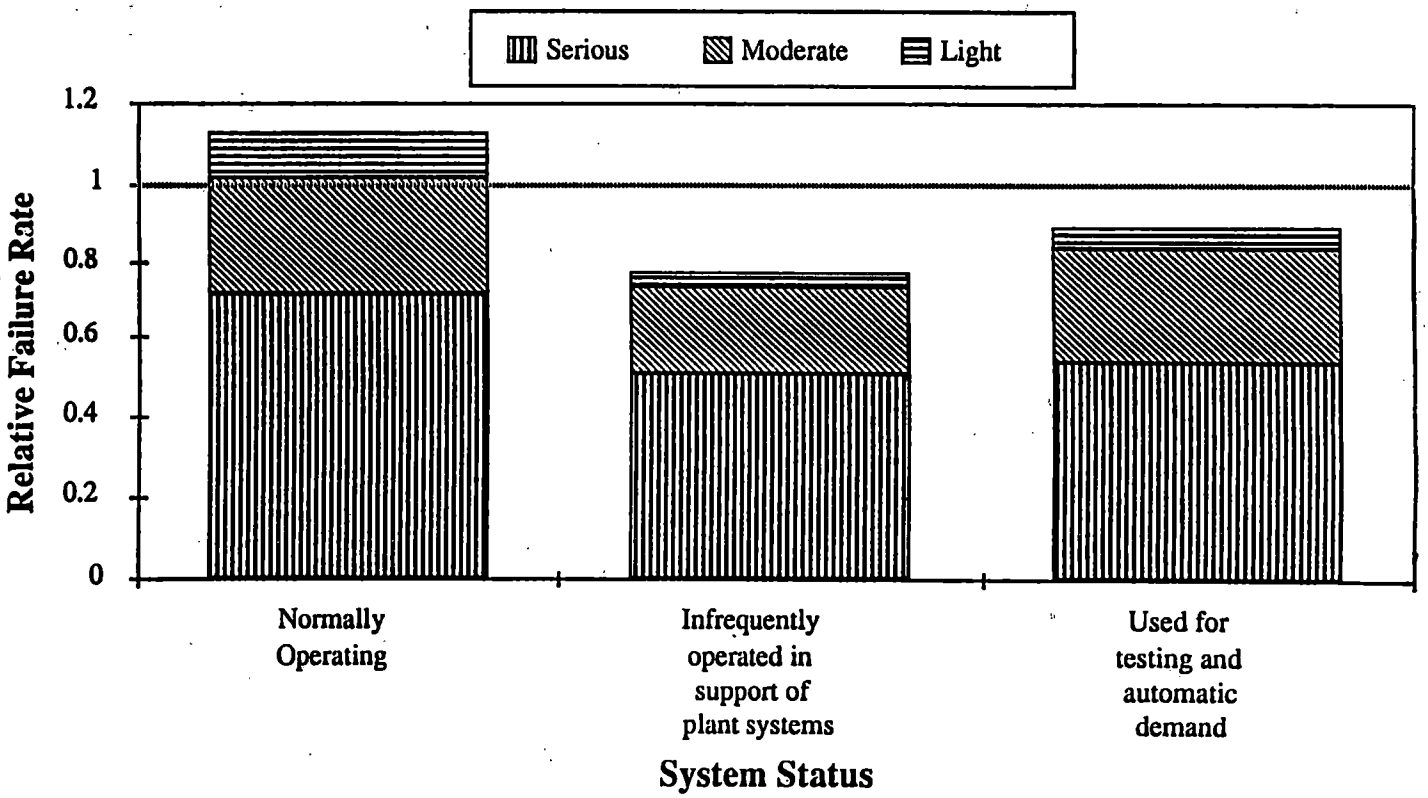
Surprisingly, Normally-operating Systems Do Not Have a Significantly Higher Failure Rate Than Other Classes of Systems



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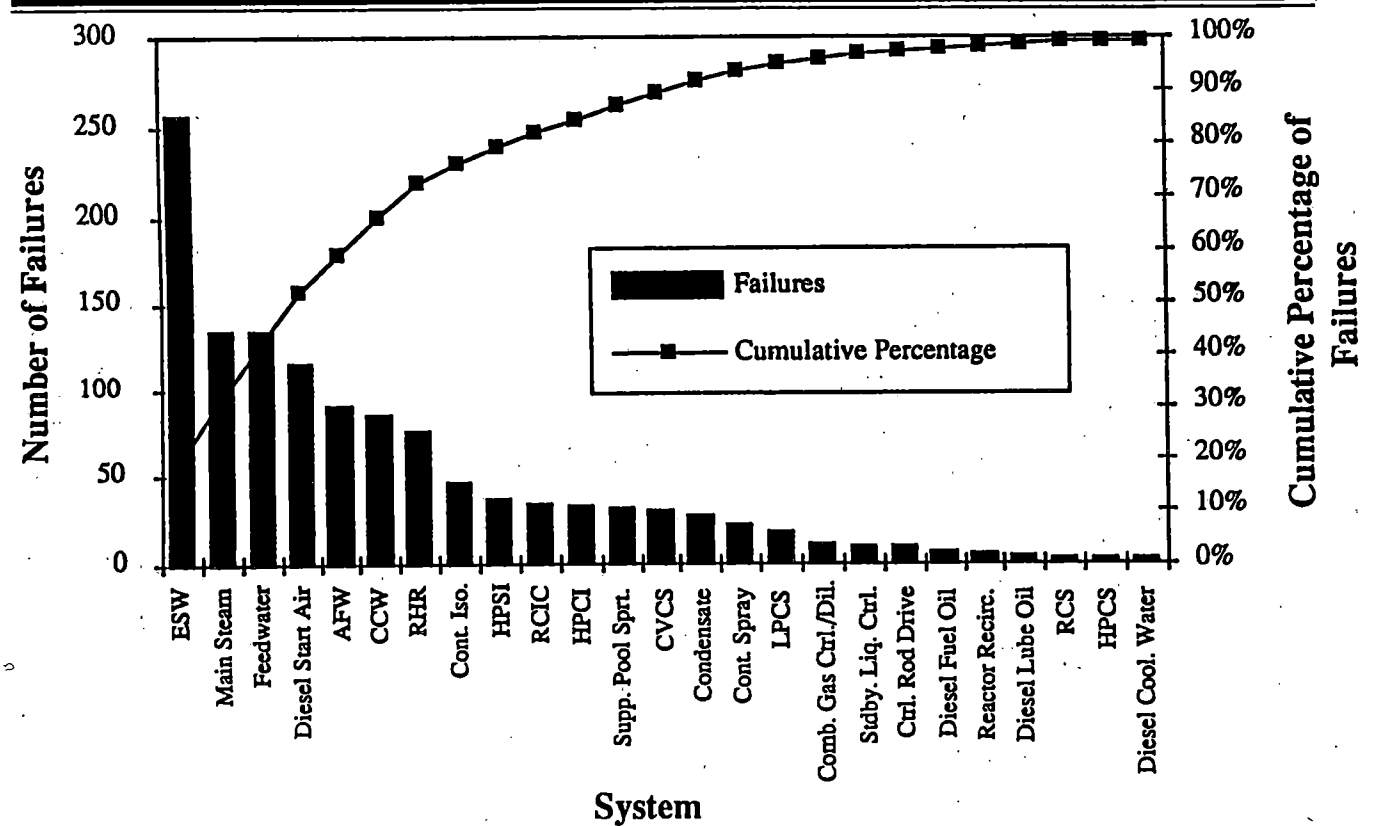
Four Systems Contribute About 50% of the Failures



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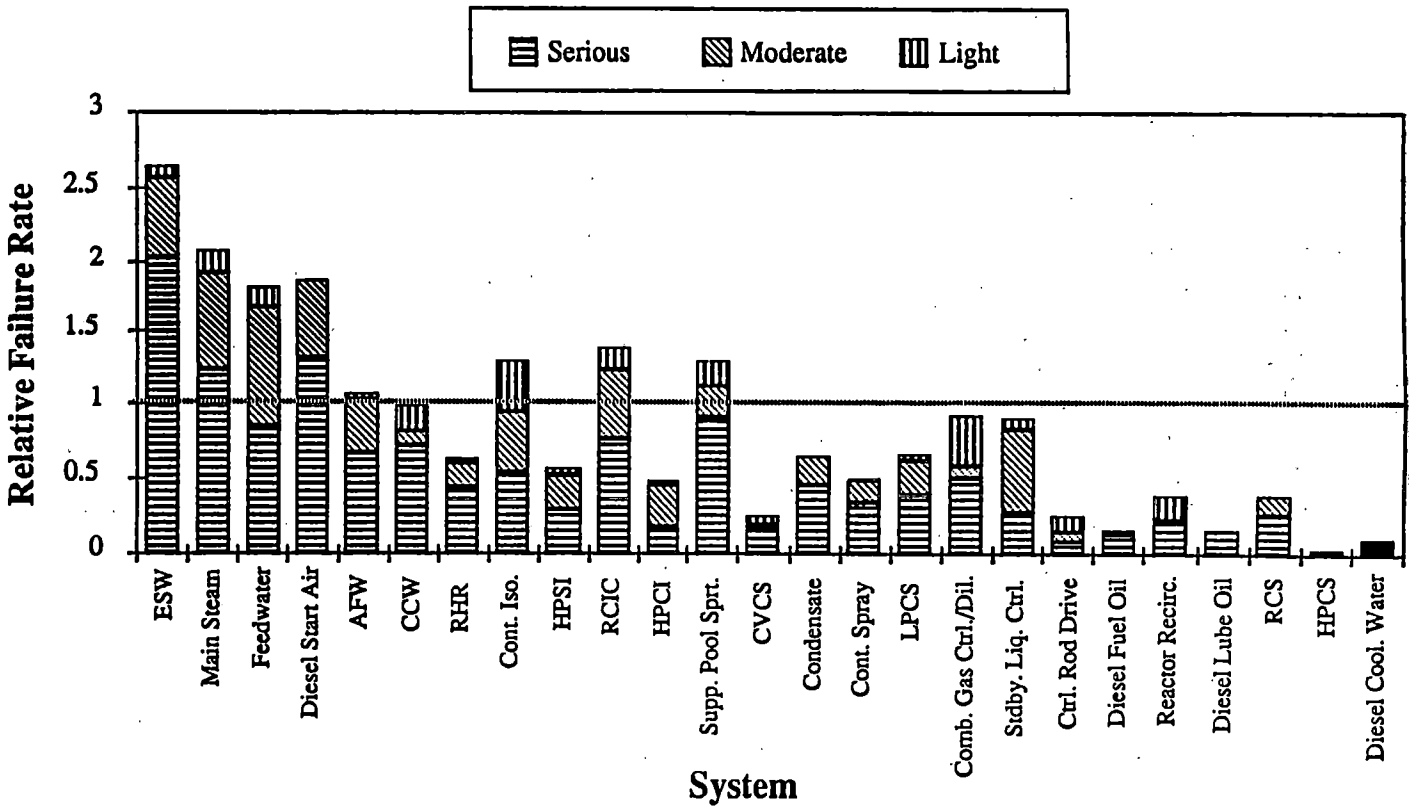
The Four Systems With the Largest Number of Failures Also Have the Highest Failure Rates



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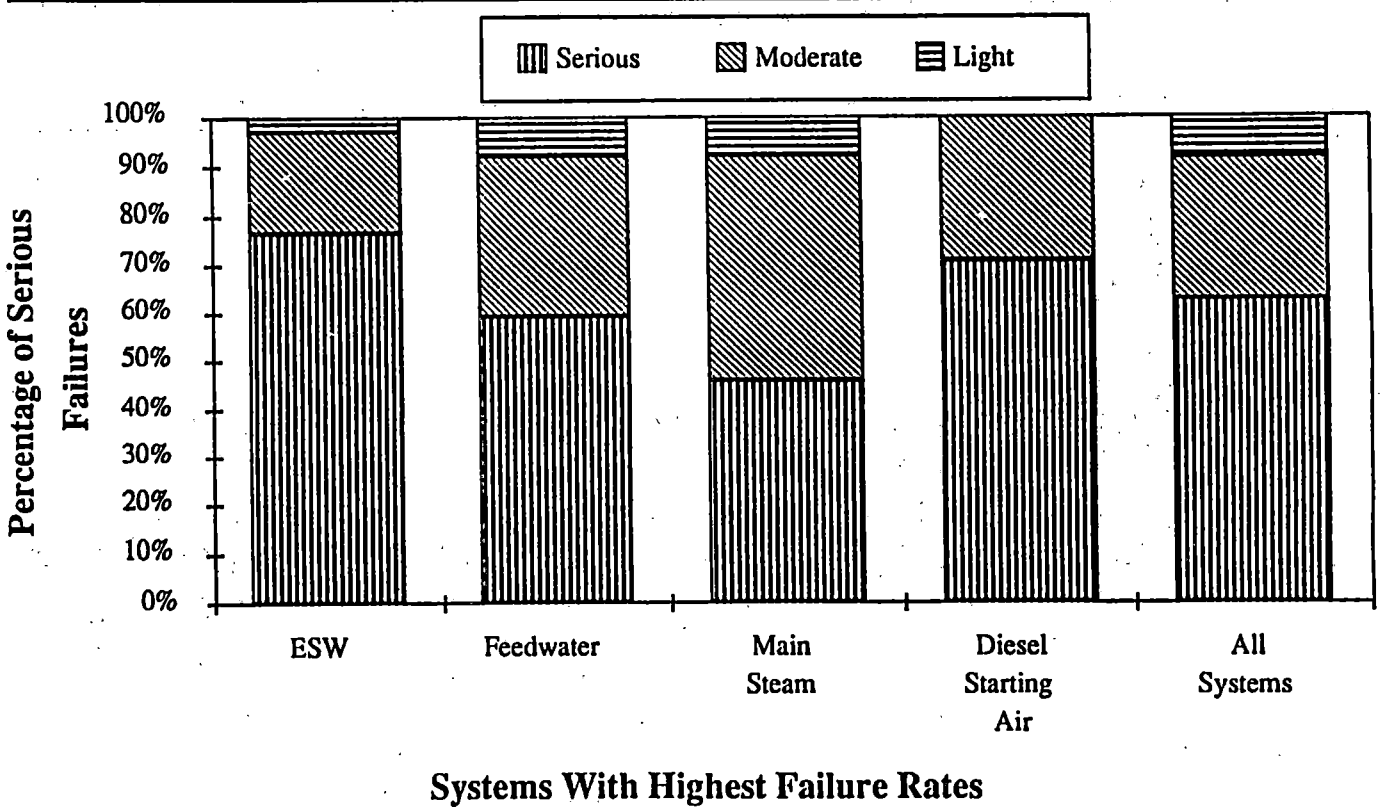
The Most Serious Failures Dominate the High Failure Rate Systems



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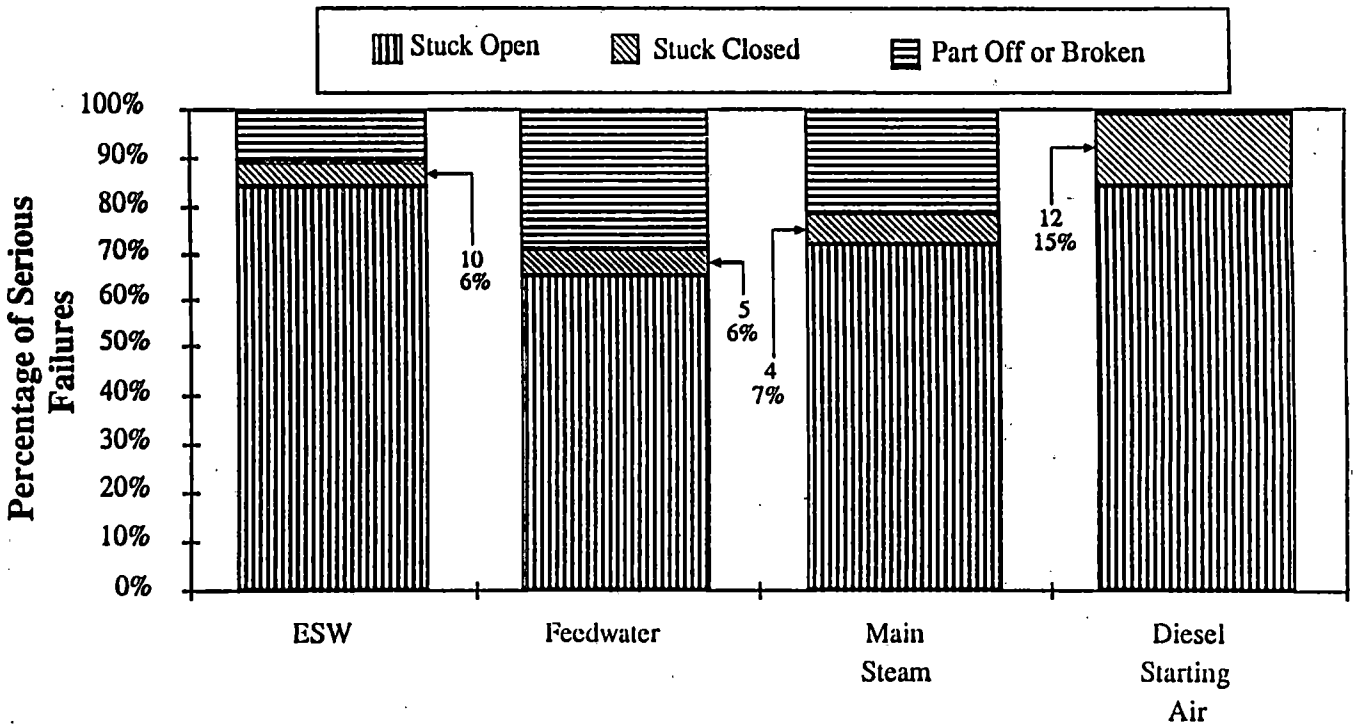
A Noticeable Number of Stuck Closed Cases Have Occurred In All Four of the High Failure Rate Systems



Systems with Highest Failure Rate

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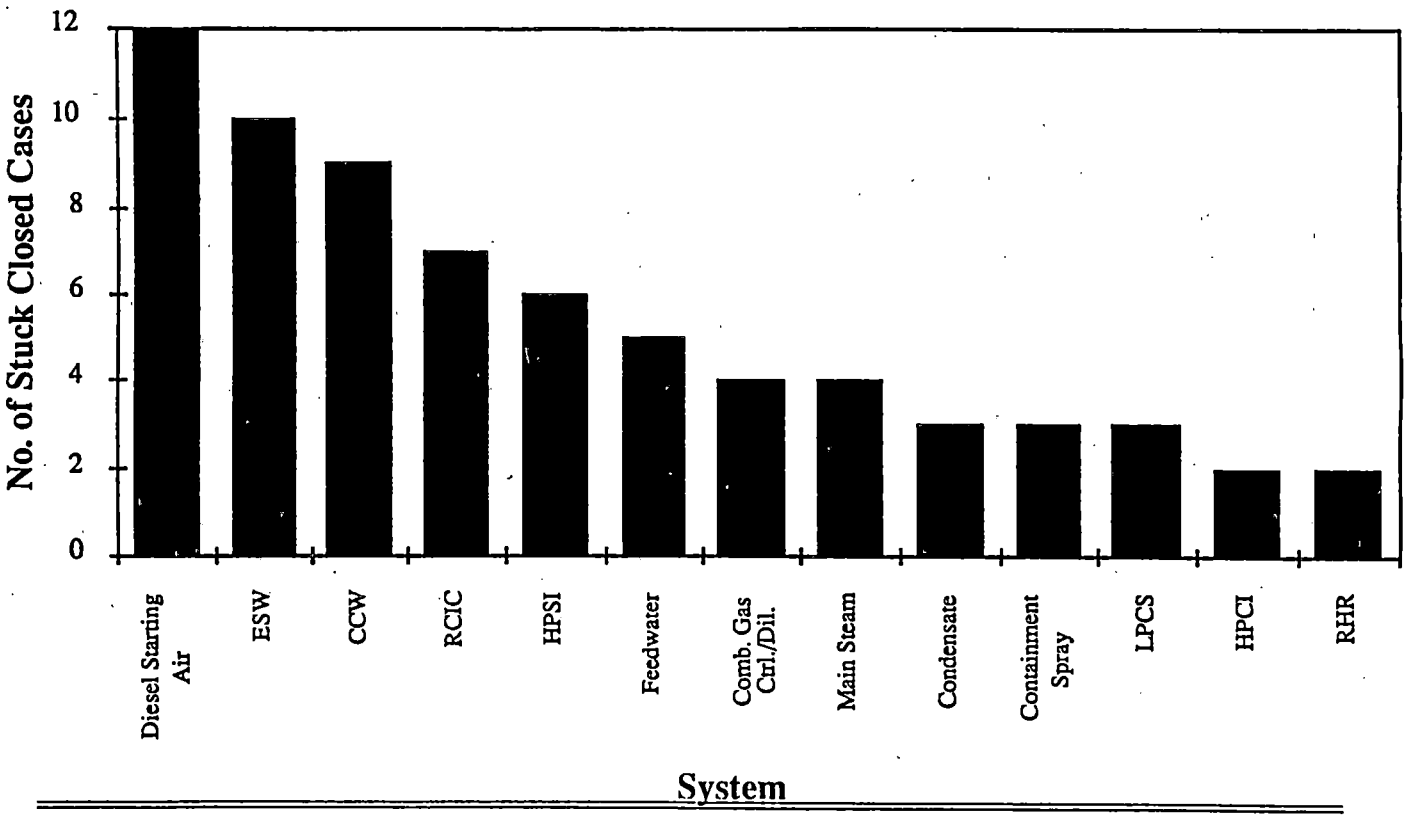
Valves Have Stuck Closed in a Variety of Different Systems



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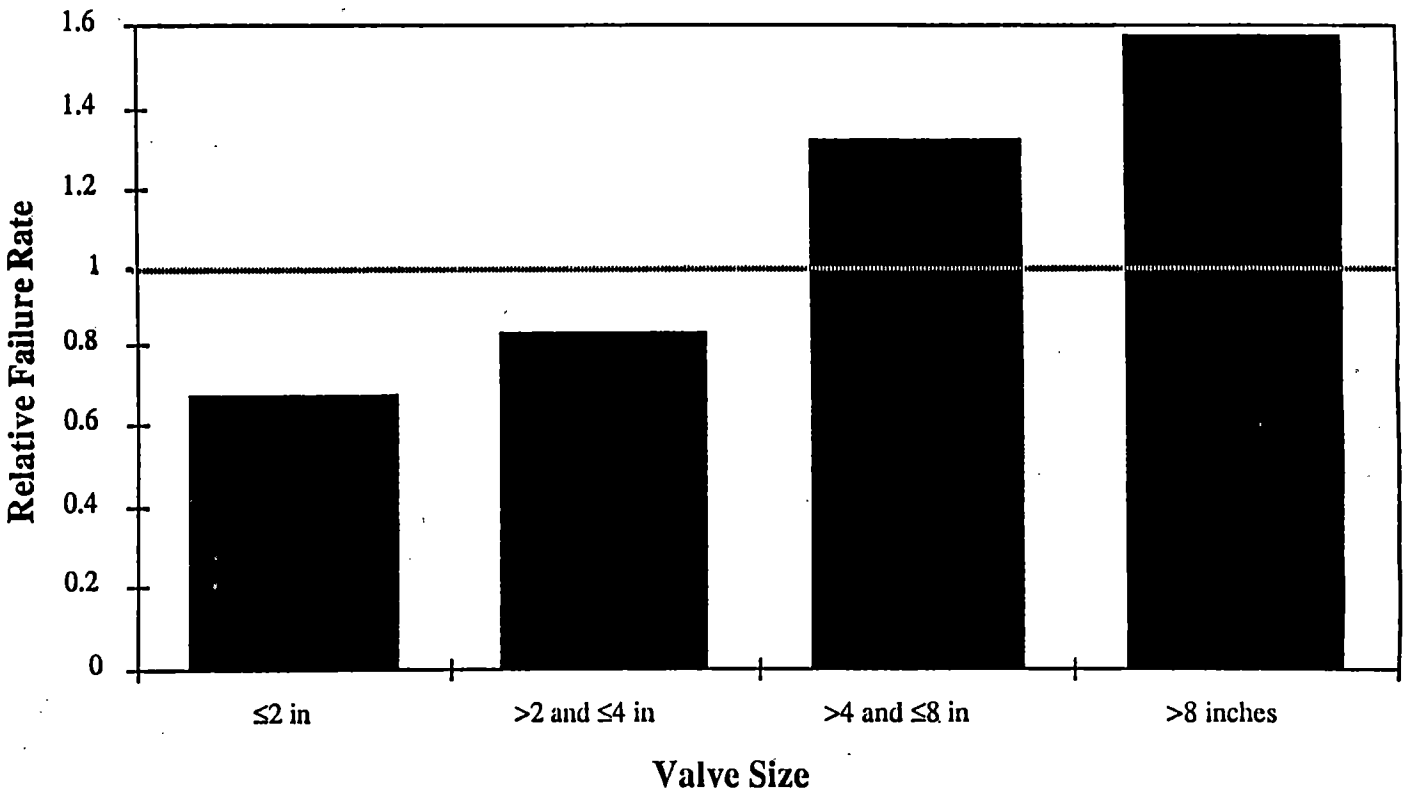
The Failure Rate Clearly Increases with Valve Size



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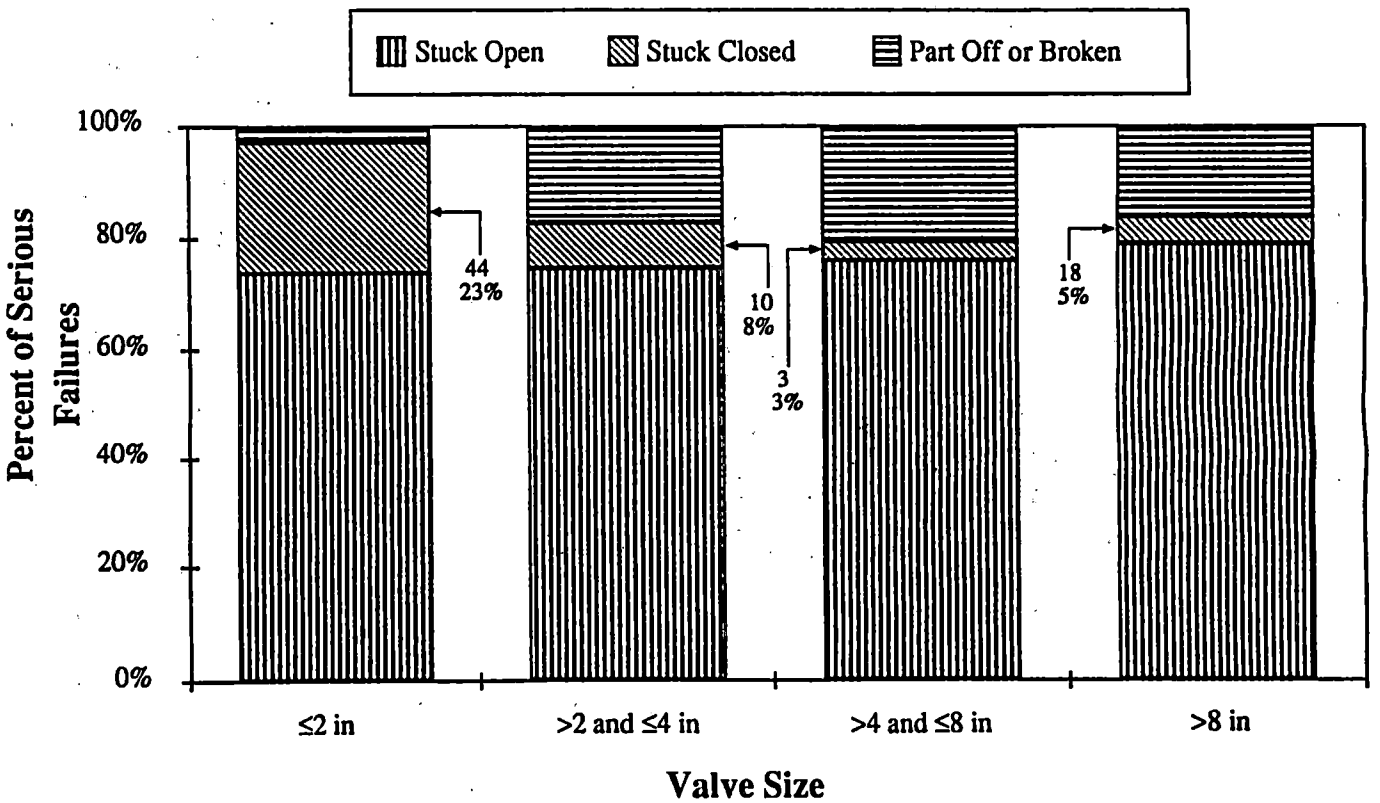
Although Stuck Open Valves are Expectedly the Most Prevalent Of the Serious Failures, Stuck Closed Valves Occur in All Valve Sizes



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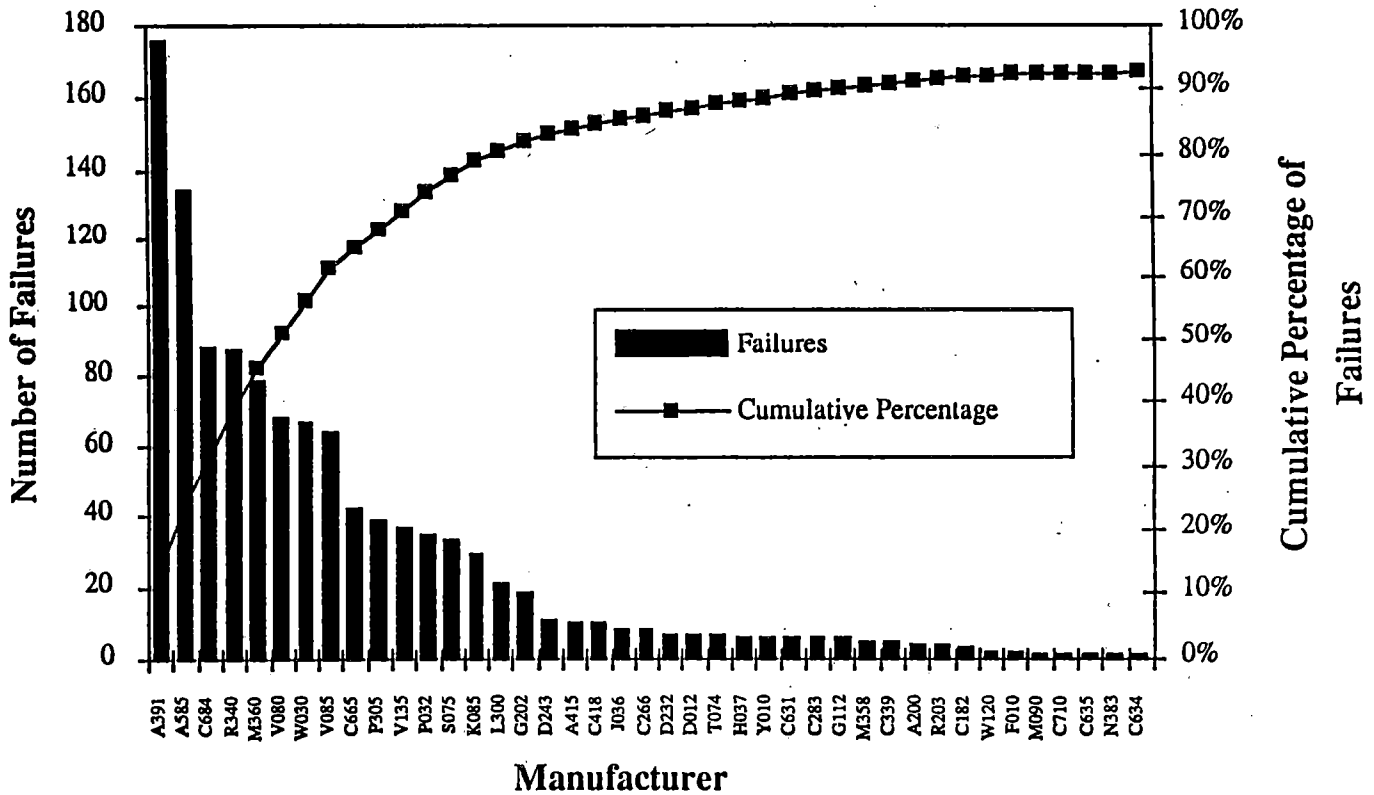
7% of the Manufacturers, Supplying 31% of the Valves in Service, Are Responsible for Almost Half of the Failures



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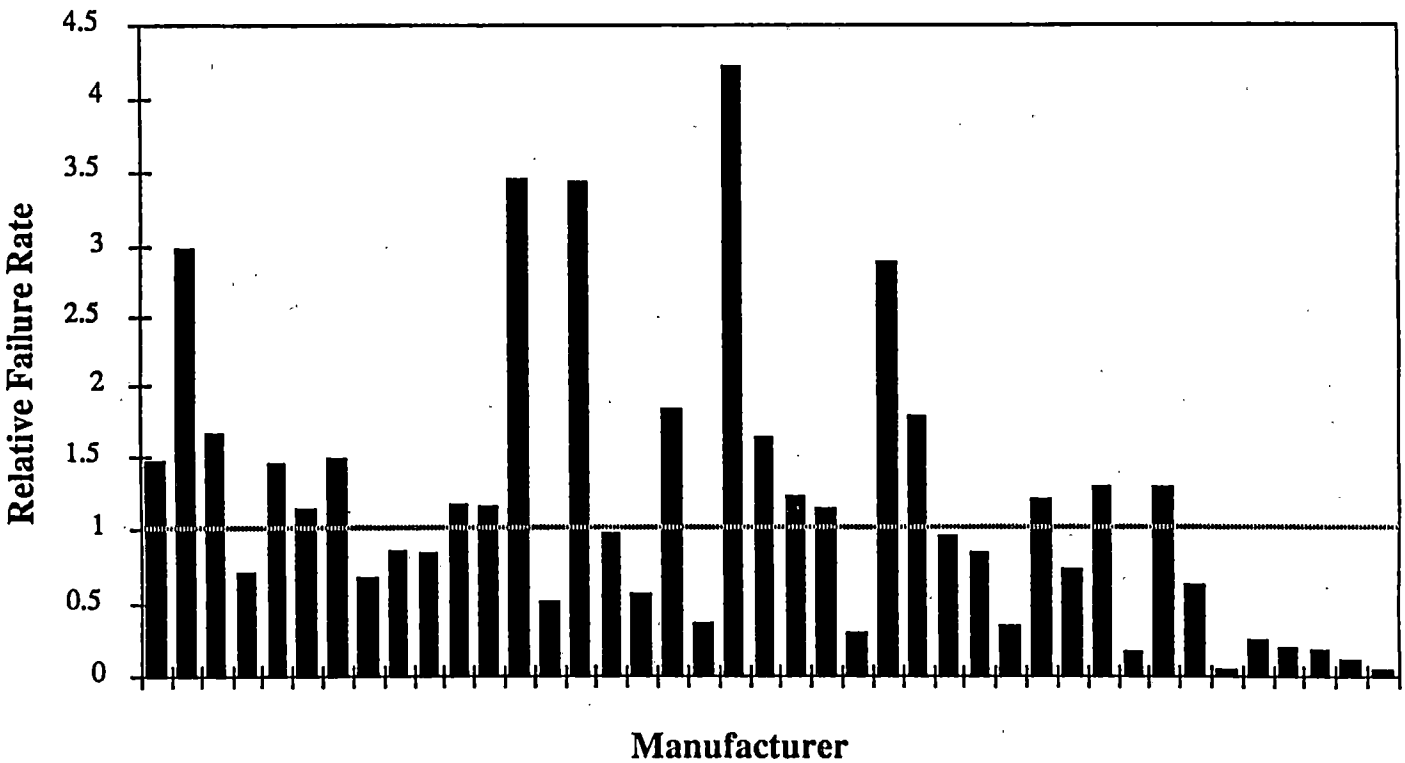
After Normalizing, Five Different Manufacturers Have Considerably Higher Failure Rates



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The Conclusions of This Study Present Several Challenges to OM-22 To Reconsider The Occurrence and Detection of Check Valve Failures

- Programmatic monitoring, as historically applied, is considerably better at detecting the less serious failures than detecting the most serious failures
- Normally operating systems do not have significantly higher failure rates than other systems
- Failure rate is generally proportional to age; does not follow "bathtub curve"
- BWR's have higher overall reported failure rates than PWR's,but this is because they are better at detecting failures

Results of AEOD Survey on Pressure Locking of Double-Disk and Flexible-Wedge Gate Valves

Stuart D. Rubin, U. S. Nuclear Regulatory Commission

Presentation Outline

- Double-Disk and Flexible-Wedge Gate Valve Characteristics
- Review of Pressure Locking Phenomenon and Causes
- Safety Significance of Pressure Locking
- Operational Events and Feedback
- Results of AEOD Survey of Licensee Corrective Actions
- Technical Lessons Learned from Survey
- Conclusions

FLEXIBLE WEDGE AND DOUBLE DISK GATE VALVE CHARACTERISTICS

- Provides for a high level of leak tightness
- Used extensively in nuclear plants
  - BWRs and PWRs
  - Many safety and nonsafety system applications
- High pressure fluid will leak into valve bonnet area
  - During normal open/close cycling of valve
  - Due to flexure of wedge/disk when valve is closed

PRESSURE LOCKING PHENOMENON AND CAUSES

Phenomenon

- Occurs when bonnet pressure is significantly greater than either the upstream and downstream line pressures
- Very large disc-to-seat frictional forces and large downward pressure force on disks are developed
- Motor-operator stalls trying to unseat the disk

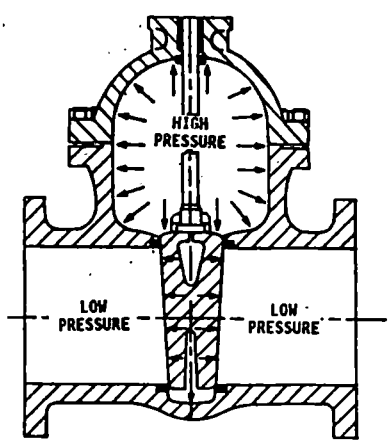


Figure 1

Safety Significant Cause

- Normal leakage past downstream in-line check valve allows high pressure fluid on one of the valve disks
- High pressure causes disk to flex slightly, allowing fluid to leak past disk, pressurizing valve bonnet volume
- Downstream line subsequently depressurizes rapidly (LOCA)
- High pressure fluid gets trapped in bonnet (between disks) while upstream and downstream piping are at lower pressures

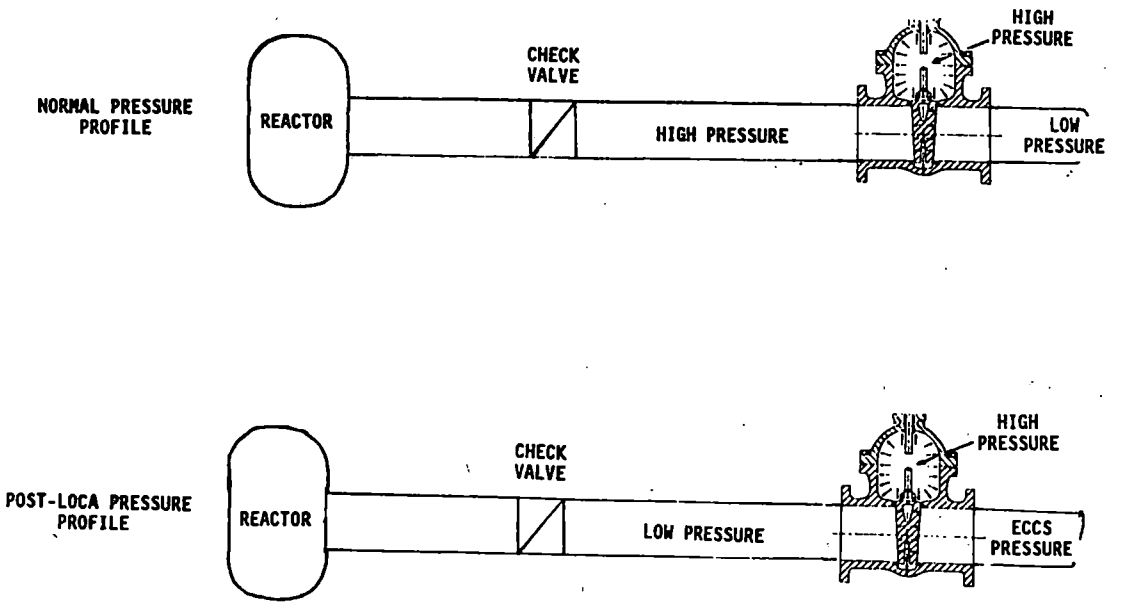


Figure 2

SAFETY SIGNIFICANCE OF PRESSURE LOCKING

- Flexible-wedge and double-disk gate valves are widely used in BWR and PWR front line emergency core cooling systems (ECCS)
- Flexible-wedge and double-disk gate valves are required to open for design basis events in some ECCS applications
- Design basis (depressurization) events can potentially cause consequential pressure locking conditions
- Pressure locking is a potential common cause failure mechanism for redundant and diverse ECCS

OPERATIONAL EVENTS AND FEEDBACK ON GATE VALVE PRESSURE LOCKING

June 1964	Buships 9480.72, "Surface Ship Steam System Valves Operation Prior to Warmup"
December 1969	Residual heat removal suction valve fails at Ginna due to Pressure-Locking
Circa 1977	Westinghouse design change issued to require vent lines on Westinghouse supplied gate valves
March 1977	IEC 77-05, "Fluid Entrapment in Valve Bonnets"
September 1981	Two safety injection valves fail to open at San Onofre 1 during a depressurization event
October 1981	IN 81-31, "Failure of Safety Injection Valves to Operate Against Differential Pressure"
Circa 1981	Two recirculation discharge valves fail to open at a BWR
December 1981	GE Service Information Letter 368, "Recirculation Discharge Isolation Valve Locking"
August 1982	Two residual heat removal suction valves fail to open at a foreigh PWR (also six previous pressure-locking events)
Circa 1982	Foreign authority requires all plants to modify gate valves with potential for pressure locking

November 1983	Industry correspondence on pressure locking issued
September 1983	Residual heat removal (RHR) heat exchanger outlet valve fails to open at LaSalle 1
November 1983	Residual heat removal heat (RHR) exchanger outlet valve fails to open at LaSalle 1
July 1984	AEOD Study on pressure locking issued
December 1984	Industry correspondence on pressure locking and thermal binding issued
February 1988	RHR cross-connect valves fails to open at Vogtle 1
July 1991	Low pressure cooling injection (LPCI) injection valve fails to open at Fitzpatrick
August 1991	PASNY review concludes all Fitzpatrick low pressure ECCS may be lost due to pressure locking: injection and spray valves modified
April 1992	IN 92-26, "Pressure Locking of Motor-Operated Flexible Wedge Gate Valves"
May 1992	AEOD limited field survey of licensee actions to address pressure locking

AEOD SURVEY ON LICENSEE CORRECTIVE ACTIONS

Plants Visited

- Based on experience: Fitzpatrick and Susquehanna
- Two BWRs: Nine Mile Point and Hope Creek
- Two PWRs: Salem and Ginna

AEOD Survey Results

Fitzpatrick

- Pressure locking was not considered credible prior to July 1991 LPCI injection valve failure event

Regulatory/Operability Issues

- After July 1991 event, licensee reevaluated potential for all gate valves
- Licensee concluded pressure locking was a credible potential common cause failure mechanism
- Core spray and LPCI injection valve bonnets were modified with vent lines
- HPCI, RCIC, containment cooling and sump isolation valves evaluations were continuing

Susquehanna

- Pressure locking was not considered credible prior to July 1991 LPCI injection valve failure at Fitzpatrick
- After Fitzpatrick LPCI valve failure, licensee reevaluated potential for pressure locking for all gate valves
- Licensee concluded pressure locking was a credible common cause failure mechanism for LOCA
- Weep hole drilled in core spray and LPCI valves disks for Unit 1 (Unit 2 will be fixed at next outage)
- Review of other gate valves in process

Nine Mile Point

- Pressure locking was not considered a credible problem
- Licensee has no plans to modify gate valves

Hope Creek

- Pressure locking was not considered a credible problem
- Licensee has not modified gate valves
- Procedures in place to remove insulation and cool bonnet if pressure locking occurs
- Currently reevaluating potential for pressure locking

Salem

- Westinghouse design change notice during construction (1977) for Westinghouse supplied gate valves prompted earlier review
- Salem had expanded review for all gate valves
- All potentially susceptible valves modified with vent lines (BIT isolation valves later changed to disk weep holes)

Ginna

- Containment sump isolation valve failed due to pressure locking in 1969
- Sump isolation valves modified in 1970
- Licensee has not modified additional valves following industry or NRC feedback
- Currently reevaluating potential for pressure locking

LESSONS LEARNED FROM SURVEY

- Several valve suppliers are involved: Anchor Darling, Alloyco, Crane, Lunkenheimer, Pacific Valves, Powell, Rockwell, Velan, and Westinghouse
- PWR and BWR front-line safety system valves are among those potentially susceptible: LPCI, CS, LHSI
- Safety concerns involve motor-operator failure and/or unacceptable time delays for valve opening (ECCS injection)
- Relatively inexpensive and proven fixed (e.g., disk weep hole or bonnet vent line) are involved
- Nonconservative aspects of pressure locking analyses were observed:
  - Leakage past in-line check valve assumed to be zero
  - Bonnet out-leakage rate assumed worn disks rather than new/rebuilt disks
  - Bonnet out-leakage rate (for bonnet depressurization) was based on low (LLRT data) differential pressure rather than high differential pressure

Regulatory/Operability Issues

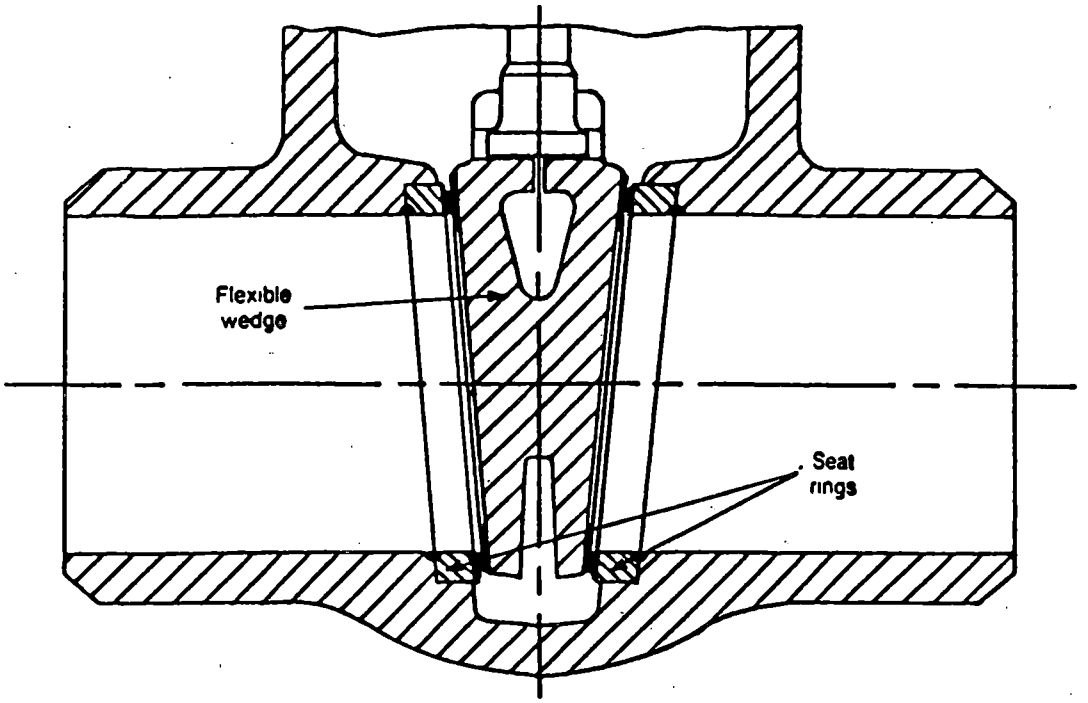
- All valve internal forces were not included or not assumed to act simultaneously
- Motor-operator thrust capability was not conservative
- Bonnet heat-up (pressurization) from nearby breaks or equipment was not considered

CONCLUSIONS

- Pressure locking of double disk and flexible wedge gate valves is a potential consequential and common-cause failure mechanism for BWR and PWR fron-line ECCSs
- Past industry and NRC feedback on pressure locking has not been fully effective in achieving adequate licensee assessments and/or needed corrective actions

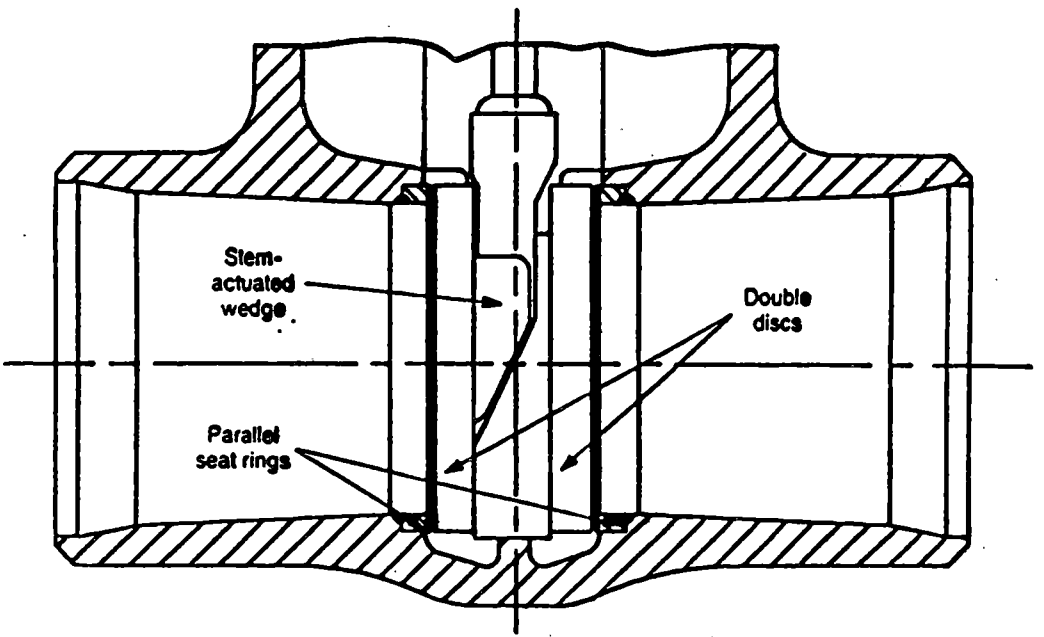
(See Figure 3 and Figure 4 for diagram of flexible-wedge and double-disc parallel-seat gate valves.)

Figure 3



Flexible-Wedge Gate Valve

Figure 4



Double-Disc Parallel-Seat Gate Valve

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The 1992 Symposium on Pump and Valve Testing, jointly sponsored by the Board of Nuclear Codes and Standards of the American Society of Mechanical Engineers and by the Nuclear Regulatory Commission, provided 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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